Research Article

Optimisation of the Nonlinear Suspension Characteristics of a Light Commercial Vehicle

Dinçer Özcan, Ümit Sönmez, and Levent Güvenç

Mekar Mechatronics Research Labs, Department of Mechanical Engineering, İstanbul Okan University, Akfırat-Tuzla, TR-34959 İstanbul, Turkey

Correspondence should be addressed to Levent Güvenç; levent.guvenc@okan.edu.tr

Received 9 October 2012; Revised 16 December 2012; Accepted 30 December 2012

Academic Editor: Shinsuke Hara

Copyright © 2013 Dinçer Özcan et al. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

The optimum functional characteristics of suspension components, namely, linear/nonlinear spring and nonlinear damper characteristic functions are determined using simple lumped parameter models. A quarter car model is used to represent the front independent suspension, and a half car model is used to represent the rear solid axle suspension of a light commercial vehicle. The functional shapes of the suspension characteristics used in the optimisation process are based on typical shapes supplied by a car manufacturer. The complexity of a nonlinear function optimisation problem is reduced by scaling it up or down from the aforementioned shape in the optimisation process. The nonlinear optimised suspension characteristics are first obtained using lower complexity lumped parameter models. Then, the performance of the optimised suspension units are verified using the higher fidelity and more realistic Carmaker model. An interactive software module is developed to ease the nonlinear suspension optimisation process using the Matlab Graphical User Interface tool.

1. Introduction

Vehicle suspension design and performance problems have been studied extensively using simple car models such as two degrees-of-freedom (d.o.f.) quarter car, four or six d.o.f. half car, or seven d.o.f. full car models. Usually, the suspension design methodologies are based on analytical methods where a linear vehicle model is investigated by solving linear ordinary differential equations. Laplace and Fourier transforms are valuable tools that are used while investigating suspension units with linear characteristics. The performance functions represented by transfer functions in Laplace and/or Fourier domains are considered to be related to ride comfort, tire forces, and handling criteria versus road roughness input to achieve an optimum design. On the other hand, the investigation of nonlinear suspension characteristics must be based more on numerical methods rather than analytical methods due to the more complicated nature of the problem. In this investigation, both linear and nonlinear spring and damper characteristics of a light commercial vehicle are considered and used in an optimisation study.

Lumped parameter suspension models are used in this paper. Mass, stiffness, and damping are distributed spatially throughout a mechanical system like a suspension. Mass, stiffness, and damping are therefore functions of spatial variables (x , y and z coordinates) in a mechanical system, resulting in what is called a distributed parameter system since the mass, stiffness, and damping parameter values are distributed over the mechanical system. An easier approach is to lump the continuous mass, stiffness, and damping characteristics into ideal mass, stiffness, and damping elements. The result is a lumped parameter system, as the mass of the mechanical system is concentrated at the ideal mass element, the stiffness of the mechanical system is concentrated in the ideal stiffness element (the spring), and the damping of the mechanical system is concentrated in the ideal damping element (the damper). The lumped parameter models used in this paper are the quarter car and half car suspension models.

The optimisation requirements of suspension systems and the state-of-the-art of suspension research in the last decade are reviewed first. It should be noted, however, that the available literature is vast and only a small portion

of it can be presented here. This paper includes the well-known ride, handling trade-off optimisation, and geometrical optimisation of light commercial vehicle suspension systems. Some heavy vehicle suspension optimisation papers are also reviewed due to their conceptual contribution to the subject.

Vehicle suspensions can be regarded as interconnections of rigid bodies with kinematic joints and compliance elements such as springs, bushings, and stabilizers. Design of a suspension system requires detailed specification of the interconnection points (or so called hard points) and the characteristics of compliance elements. Tak and Chung [1] proposed a systematic approach of achieving optimum geometric design of suspension systems where design variables are determined to meet some prescribed performance targets expressed in terms of suspension design factors, such as toe, and camber, compliance steer. Koulocheris et al. [2] proposed to combine deterministic and stochastic optimisation algorithms for determining optimum vehicle suspension parameters. They showed that such combination yields significantly faster and more reliable convergence to the optimum. Their method combines the advantages of both categories of deterministic and stochastic optimisation. They used a half car model of suspension systems, subject to various road profiles considering the improvement of the passenger ride comfort, leading to minimisation of the maximum acceleration of the sprung mass while paying attention to the geometrical constraints of the suspension as well as the necessary traction of the vehicle.

Maximizing tractive effort is essential to competitive performance in the drag racing environment. Antisquat is a transient vehicle suspension phenomenon which can dramatically affect tractive effort available at the motorcycle drive tire. Wiers and Dhingra [3] addressed the design of a four-link rear suspension of a drag racing motorcycle to provide anti-squat. This design increases rear tire traction, thereby improving vehicle acceleration performance. For the drag racing application considered, any increase in normal forces at the tire patch helps improve race competitiveness. Mitchell et al. [4] used a genetic algorithm for the optimisation of automotive suspension geometries considering the description of a suspension model and a scoring method. Their approach is to design with a unit-free measure of fitness for each test and then to combine these with a weighing function. They showed that the genetic algorithm and the scoring mechanism worked effectively and significantly faster than the more common grid optimisation technique. Raghavan [5] presented an algorithm to determine the attachment point locations of the tie rod of an automotive suspension, in order to achieve linear toe change characteristics with jounce and rebound of the wheel. This linear behaviour is advantageous for achieving good ride and handling. Raghavan's procedure can be applied to all suspension mechanism types such as short-long arm, McPherson struts, and five-link front and five-link rear suspensions.

The design of suspension systems generally demands a compromise solution to the conflicting requirements of handling and ride comfort. The following examples demonstrate this compromise.

- (i) For example, for better comfort a soft suspension and for better handling a stiff suspension is needed.
- (ii) A high ground clearance is required on rough terrain, whereas a low centre of gravity height is desired for swift cornering and dynamic stability at high speeds.
- (iii) It is advantageous to have low damping for low force transmission to the vehicle frame. On the contrary, high damping is desired for fast decay of oscillations.

Considering the aforementioned requirements, Deo and Suh [6] proposed a design for a customizable automotive suspension system with independent control of stiffness, damping, and ride height. Their design enabled the achievement of desired performance depending on user preference, road conditions, and maneuvering inputs while avoiding the performance trade-offs.

Goncalves and Ambrosio [7] proposed a methodology in order to investigate flexible multibody models for the ride and stability optimisations of vehicles. Their methodology allows the use of complex shaped deformable bodies, represented by finite elements. The ride optimisation is achieved by finding the optimum of a ride index that is the outcome of a metric that accounts for the acceleration measured at several key points of the vehicle, weighed according to their importance for occupant comfort. Duysinx et al. [8] developed a mechatronic approach to model, simulate and optimize a passenger car (Audi A6) incorporating a controlled semiactive suspension. They paid particular attention to the formulation of the mechatronic model of the car and compared two different modelling and optimisation approaches. The first approach is carried out in the Matlab-Simulink environment and the derivation of the equations is based on a symbolic multi-body model. The optimisation procedure has also been investigated in Matlab. Their second approach relies on a multi-body model based on the finite-element method where the optimisation has been realized with an open-source industrial optimisation tool.

Eskandari et al. [9] optimized the handling behaviour of a midsized passenger car by altering its front suspension parameters using Adams/Car software. They utilized an objective function combination of eight criteria indicating handling characteristics of the car and reduced the amount of optimisation parameters, by implementing the design of experiments method capabilities. The amount of the parameters was reduced from fifteen to ten by using a sensitivity analysis. A similar Adams/Car-based study was conducted by Boyalı et al. [10]. More recently, He and McPhee [11] reviewed the state-of-the-art related to modelling approaches, considering vehicle system models, design variable and performance criteria definitions, optimisation problem formulation methods, optimisation search algorithms, sensitivity analysis, computational efficiency, and other related techniques. They applied these techniques to the design synthesis of ground vehicle suspensions and proposed a methodology for automated design synthesis of ground vehicle suspensions.

Li et al. [12] considered a five-link suspension optimisation for improving the ride safety and comfort using

Adams/Insight. They investigated the relations among multi-link suspension structural parameter, wheel location parameter, and wheel track. Uys et al. [13] reported an investigation to determine the spring and damper settings that will ensure optimal ride comfort of an off-road vehicle, on different road profiles and at different speeds. Spring and damper settings can be set either to the ride mode or the handling mode, and therefore a compromise ride-handling suspension is avoided. They found that optimizing for a combined driver plus rear passenger seat weighed root mean square vertical acceleration rather than using driver or passenger values only returns the best results. Their results indicated that optimization of suspension settings using the same road and constant speed will improve ride comfort on the same road at different speeds and these settings will also improve ride comfort for other roads at the optimisation speed and other speeds, although not as much as when optimisation has been done for the particular road. In the present paper, one vehicle speed and one road profile were used for optimization taking this statement from [13] into account. They also concluded in [13] that for improved ride comfort, damping generally has to be lower than the standard (compromised) setting, the rear spring as soft as possible, and the front spring ranging from as soft as possible to stiffer depending on road and speed conditions. Ride comfort is the most sensitive to a change in rear spring stiffness.

The roll steer of a front McPherson suspension system is studied, and the design characteristics of the mechanism are optimized by Habibi et al. [14] using the genetic algorithm method. The roll steer affects handling, and dynamic stability of the vehicle due to variation of the angles of the wheel and the suspension links (i.e., camber, caster, and toe). However, these changes cause other problems. In their paper, Habibi et al. [14] used a genetic algorithm method to determine the optimum length and orientation of the mechanism's members to minimize the variations of the toe, camber, and caster angles. They defined a performance index which expresses the overall variations of the main parameters in the whole range of rolling of the body. A general formulation for multi-body flexible systems, with linear elastic deformations, is considered by Gonçalves and Ambrosio [15] whose application involved a road vehicle where flexibility plays an important role in ride and handling dynamic behaviour. Using finite elements to describe the flexibility of the body and the modal superposition method has the advantage of greatly reducing the dimensionality of the system. The results presented in [15] showed that the use of the detailed vehicle model within the framework of ride optimisation leads to a measurable improvement of the comfort conditions for different road profiles and driving conditions.

An optimum concept to design "road-friendly" heavy vehicles with the recognition of pavement loads as a primary objective function of vehicle suspension design was investigated by Sun [16]. A walking-beam suspension system is used as an illustrative example of the vehicle model to demonstrate the concept and process of optimisation. Dynamic response of the walking-beam suspension system was obtained by means of stochastic process theory. Using the direct update

method, optimisation is carried out when tire load magnitudes are taken as the objective function of suspension design. Their results showed that tires with high air pressure could lead to more damage in pavement structures, and increasing suspension damping and tire damping can reduce the tire loads and pavement damage.

This paper concentrates on optimisation of the nonlinear shape of the spring and damper of a light commercial vehicle. The work reported in the paper is motivated by the fact that the spring and damping characteristics in an actual road vehicle are designed to be nonlinear on purpose by the automanufacturer. Larger spring forces are used in the rebound motion of the wheel in order to keep forcing an appropriate level of tire-road contact at all times, for example. When the automanufacturer starts working on a new model, a previous, successful suspension design is used as the base characteristic which is modified to fit the characteristics of the new vehicle model. The work presented here follows the same approach as spring, and damper characteristics of an existing base design are used as the starting point in the optimisation. The optimisation procedure is embedded into an interactive Matlab Graphical User Interface to offer ease of use to suspension designers.

The organization of the rest of the paper is as follows. The vehicle models used in this study are introduced in Section 2 along with the scope of this paper. Some of the performance criteria available in the literature are discussed in Section 3. The optimisation procedure used is the topic of Section 4, while the optimisation results obtained using this procedure are treated in Section 5. The paper ends with the usual conclusions and recommendations being presented in Section 6.

2. Vehicle Models Used and Scope of the Investigation

In this section, the scope of the current investigation is summarized by presenting the vehicle models that are used in this study. The use of a complex three-dimensional model of the vehicle, with a detailed description of all suspension systems and road/tire interaction, is necessary to fully investigate the problem. However, such models are computationally expensive especially when used in an iterative optimisation design process. A good alternative which is used here is the optimisation of a subsystem of a complex model. The suspension subsystem is very important in terms of vehicle dynamics. Its spring and damper load deflection characteristics are treated as the basic design variables here.

The ride comfort optimisation is achieved by finding the optimum of a ride comfort index which results from a metric that accounts for the linear and the angular accelerations of the model's suspended mass centre and properly combined in a cost function, considering their importance for the comfort of the occupant. Two lumped parameter models are built in Matlab considering the independent front suspension unit (a quarter car model) and the rear axle suspension unit (a half car model) of a light commercial vehicle. Vertical displacement z_s and acceleration a_s of the suspended mass and the

tire force F_{Tire} of the quarter car model are considered as the key variables in ride comfort and handling, respectively. Similarly, vertical and angular displacement z_s and ϕ_s of the mass centre and the tire forces at both of the rear wheels of the half car model are selected as the key variables for the rear suspension half car model.

The dynamics of the quarter car and the half car rear axle suspension models are governed by nonlinear differential equations of motion. The road profile described in Cebon [17] is used in the optimisation study and simulations to determine suitable linear and/or nonlinear spring and nonlinear damper characteristics. Considering a nonlinear model, a suspension characteristic optimisation routine written as an m-file in Matlab gives more insight than using commercial vehicle simulation/analysis software. In some cases, this option (nonlinear suspension characteristics optimisation) is not readily available in commercial vehicle suspension packages. The main objective and the contribution of this investigation are to determine the optimum nonlinear functions of the damper and the spring characteristics for the improvement of the passengers' ride comfort and vehicle handling leading to minimisation of the chosen objective function.

Basic shapes representing the spring and the damper characteristics (force versus deflection for the spring and force versus velocity for the damper) are used according to automotive manufacturer's specifications. Basic functional shapes in each operating mode (extension or compression regions of the spring and the damper) are predetermined and the functional fits to these shapes are obtained. These functions and their linear combinations are then scaled searching for the optimum characteristics. The emphasis of this investigation is placed on finding nonsymmetric optimum nonlinear functions of the spring and the damper force characteristics. Optimised functional relations are then incorporated into a model built in a high fidelity, realistic commercial vehicle dynamics software to evaluate the performance of the vehicle model with the optimised suspension. Carmaker software [18] is used for this purpose to study the handling behaviour of the car in standard tests (double-lane change, fishhook, etc.). The Carmaker vehicle model [18] is a highly realistic one that incorporates engine dynamics, tire dynamics, steering dynamics, suspension dynamics, vehicle sprung body dynamics, longitudinal and lateral dynamics, a driver model, and road and environment models. The investigation also looks at a scenario where ride comfort and handling are simultaneously required. The cornering behaviour of a road vehicle is an important performance mode often equated with handling. In order to analyze both of ride and handling requirements, a double lane change manoeuvre is performed after travelling over an irregular road profile with a disturbance, and then the vehicle comes to a stop. The simulation results of optimised nonlinear damper and nonlinear spring characteristic functions are compared with those of the optimised linear ones in simulations.

2.1. Suspension Models Used. The mathematical models of the quarter car and the half car representing one of the

FIGURE 1: Quarter car model.

front quarters and the rear axle suspension unit of a light commercial vehicle are presented in this section.

2.1.1. Quarter Car Model. The quarter car model subject to road disturbances is shown in Figure 1. The equation of motion considering the vertical displacement of the vehicle body with linear spring and nonlinear damper characteristics may be written as

$$m_s \ddot{z}_s + B_F \{\dot{z}_s - \dot{z}_u\} + k_s (z_s - z_u) = 0. \quad (1)$$

Similarly the equation of motion of the vehicle wheel may be written as

$$m_u \ddot{z}_u + B_F \{\dot{z}_u - \dot{z}_s\} + k_s (z_u - z_s) - b_u (\dot{z}_u - \dot{z}_r) - k_u (z_r - z_u) = 0, \quad (2)$$

where $B_F \{\dot{z}_u - \dot{z}_s\}$ represents the nonlinear functional relation of suspension damper force versus velocity characteristic.

2.1.2. Half Car Model. The rear solid axle suspension unit of the light commercial vehicle considered here is represented with a half car model (see Figure 2) and subjected to road disturbances coming from both sides of the track (the left and the right wheels). The torsional antiroll bar is also considered in the half car model.

The half car model can represent the bounce (z_s , z_u) and roll motions (ϕ_s , ϕ_u) of the car body and solid rear axle. Therefore, it has four d.o.f. The equations of motions of the sprung (car body) and unsprung (rear axle) masses considering the bounce and roll motions may be written as

$$m_s \ddot{z}_s + F_{\text{sleft}} + F_{\text{sright}} = 0 \quad (3)$$

for the bounce motion of the sprung mass and

$$m_u \ddot{z}_u - F_{\text{sleft}} - F_{\text{sright}} + F_{\text{uleft}} + F_{\text{uright}} = 0 \quad (4)$$

for the bounce motion of the rear axle. The roll motion of the sprung inertia is given by

$$I_{xx} \ddot{\phi}_s + F_{\text{sleft}} \frac{L}{2} - F_{\text{sright}} \frac{L}{2} = 0 \quad (5)$$

FIGURE 2: Half car model.

and the roll motion of the rear axle (considering rotational unsprung inertia I_{uxx}) is represented by

$$I_{uxx}\ddot{\phi}_u + K_{\phi u}\phi_u - (F_{sleft} + F_{uright})\frac{L}{2} + (F_{sright} + F_{uleft})\frac{L}{2} = 0, \quad (6)$$

where the forces F_{sleft} , F_{sright} , F_{uleft} , and F_{uright} acting on the sprung mass are given by

$$\begin{aligned} F_{sleft} &= B_{FL} \left\{ \left(\dot{z}_s + \dot{\phi}_s \frac{L}{2} \right) - \left(\dot{z}_u + \dot{\phi}_u \frac{L}{2} \right) \right\} \\ &\quad + K_{FL} \left\{ \left(z_s + \phi_s \frac{L}{2} \right) - \left(z_u + \phi_u \frac{L}{2} \right) \right\}, \\ F_{sright} &= B_{FR} \left\{ \left(\dot{z}_s - \dot{\phi}_s \frac{L}{2} \right) - \left(\dot{z}_u - \dot{\phi}_u \frac{L}{2} \right) \right\} \\ &\quad + K_{FR} \left\{ \left(z_s - \phi_s \frac{L}{2} \right) - \left(z_u - \phi_u \frac{L}{2} \right) \right\}, \\ F_{uleft} &= k_{uL} \left(z_u + \phi_u \frac{L}{2} - z_{RL} \right), \\ F_{uright} &= k_{uR} \left(z_u - \phi_u \frac{L}{2} - z_{RR} \right), \end{aligned} \quad (7)$$

where L stands for the track width, and I_{xx} and I_{uxx} represent the moments of inertia of the sprung mass and axle, respectively. $K_{\phi u}$ is the torsional spring stiffness of the antiroll bar which is calculated as

$$K_{\phi u} = \frac{GJ}{L_b}, \quad (8)$$

where G is the shear modulus, J is the polar moment of inertia, and L_b is the length of the stabilizer bar which is

subjected to torsional stress. In the simulations and in the optimisation process, spring and damper characteristics of the left and right sides are assumed to be identical.

3. Performance Criteria Available in the Literature

3.1. Some Performance Indices Available in the Literature. The choice of subobjective functions and their weights in the combined (main objective) function plays a very critical role in the optimisation process. In this section, the literature review of the vehicle suspension objective functions and the performance indices are summarized, and our approach to the objective function formulation is presented. The nonlinear stiffness and damping characteristics are optimized by Koulocheris et al. [2] considering a half car model subject to different types of road irregularities. As an objective function, the maximum value of vertical acceleration of the vehicle body at the passenger seat is minimized from the view point of ride comfort. The objective function is formed according to the quadratic penalty given by

$$f(z) = \max(\ddot{z}_s) + M \sum c_i^2(z), \quad (9)$$

where $\ddot{z}_s$ is the vertical acceleration of the vehicle mass, M is the penalty parameter, and c_i are the constraint functions for parameter vector z .

Geometrical parameters of the suspension were considered in Mitchell et al. [4] when determining the fitness of a given suspension design. Since these parameters are not all at the same magnitude or even have the same units, coming up with a single fitness value is difficult. Their basic approach was to carry out the design with a unitless measure of fitness for each test and then to combine these results with a weighing function. Several functions were analyzed and compared while evaluating the speed and the accuracy of the method using the genetic optimisation algorithm. A first-order normal distribution was chosen due to its convergence speed. The score equals to 100 for the ideal score and 28.3 at the bound. Each metric score (score_{*i*}) is combined by way of a weighing function (W_i). Then, the scoring metric and total score are normalised using

$$\text{Total Score} = \frac{\sum_i W_i \text{Score}_i}{\sum_i W_i}. \quad (10)$$

The coordinates of the front and rear suspension hard points, the stiffness and damping properties of the front and rear suspension springs and damper, sprung mass, gear ratio, the inertia of steering wheel and so forth. were selected by Li et al. [19] as the design variables, considering vehicle handling. The objective evaluation index was adapted to evaluate the performance of vehicle handling. The index included course following indices, driver burden indices, indices for the risks of roll over, index for driver's road feeling, and index for lateral slip. The double lane change maneuver was selected for a virtual test, and the objective evaluation index was calculated.

The objective function in Eskandari et al. [9] was selected to represent several aspects of the handling behaviour of a vehicle and is of the form

$$F = \sum_i W_i X_i, \quad (11)$$

where X_i represent yaw velocity overshoot, yaw velocity rise time, lateral acceleration overshoot, lateral acceleration rise time, roll angle steady-state response, RMS of the understeering coefficient, RMS of the steering torque, and RMS of the steering sensitivity. Determination of the weighing factors W_i was made based on the importance of each quantity and is adjustable.

A global performance index is considered in Tak and Chung [1] as the linear combination of each individual performance index. Through kinematic analysis, toe and camber curves were obtained, and target values of the toe and camber curves were set up. The squared value of the reaction force at each tie rod was also included in the performance index. The global performance index was determined as the weighed linear combination of the wheel angles and reaction forces at the joints as

$$I = W_1 I_{\text{toe}} + W_2 I_{\text{camber}} + \dots + W_j I_{\text{roll rate}} + \dots + W_k I_{\text{reaction forces}}. \quad (12)$$

The performance of an active suspension system was evaluated by Jonasson and Roos [20], covering comfort and road holding capabilities as well as the energy consumption of the system. The formulation of three different performance indices was considered: two of them are based on the RMS norm described by

$$|z|_{\text{RMS}} = \sqrt{\frac{1}{\tau} \int_0^\tau z^2 dt}. \quad (13)$$

Comfort is strongly related to the vertical accelerations of the vehicle body. Therefore, the performance index for comfort is formulated considering vertical accelerations. The comfort index for a vehicle with an active suspension system is weighed with respect to the acceleration of the body in a conventional system and is described as a ratio $I_1 = |\ddot{z}_{\text{active}}|_{\text{RMS}} / |\ddot{z}_{\text{passive}}|_{\text{RMS}}$. A value of more than one means that the current design is inferior as compared to the passive suspension. Road holding capability is directly related to the variation in vertical tire force, a constant tire force is ideal, and therefore the second index is described by $I_2 = |F_{T,\text{active}}|_{\text{RMS}} / |F_{T,\text{passive}}|_{\text{RMS}}$. The third index I_3 of the objective function considers the energy used by the active system as represented with $I_3 = |P_{\text{loss,active}}| / |P_{\text{loss,passive}}|$. Then, these indices are combined to form an overall objective function for the optimisation algorithm

$$I = W_1 I_1 + W_2 I_2 + W_3 I_3. \quad (14)$$

3.2. Use of Frequency Weighing Based on the ISO2631 Standard. In order to optimize ride characteristics, human sensitivity to vibrations needs to be considered. For that purpose,

FIGURE 3: Frequency weights as specified in ISO 2631-1 standard.

the vertical motion is weighed according to the ISO 2631 [21] standard. The different characteristics of the excitation, including magnitude, frequency, axis, and duration based on the human tolerance for vibrations should be considered. As suggested by the ISO 2631 standard, the complete acceleration time histories for each of the target points are measured. Then, each Cartesian component of the acceleration history is decomposed into a Fourier series. After that, a frequency weight given in ISO 2631 standards is multiplied by each term of the Fourier series. The single objective function value is determined as the sum of the weighed terms of the Fourier series previously obtained in the decomposition process. Frequency weights of acceleration as specified in ISO 2631-1 standards are shown in Figure 3.

Figure 4 shows the body acceleration response of a quarter car model to a chirp (swept sine) signal, its power spectral density (PSD), and their weighed counterparts obtained after using the ISO 2631 standard. The original and the weighed signals are presented in time domain (top plot) for comparing their magnitudes and in the frequency domain (bottom plot) for comparing their power spectral densities (see Figure 4).

The target accelerations of the vehicle models are weighed according to the ISO 2631 standards in this investigation [21, 22]. The cases belonging to each vehicle model (total of two cases) are investigated. The performance indices of these cases are presented in the simulation section (Section 6).

4. The Optimisation Procedure

In this section, the optimisation procedure is explained in detail. In the current investigation, the complexity of the optimisation problem is reduced by deciding on the basic shape/behaviour of the force versus displacement and force versus velocity characteristics and then scaling them up or down during optimisation. This is motivated by the suspension design procedure used in the automotive industry where a previous, successful design is used as the base design, modified for the vehicle model being developed. The procedure used here has two steps.

FIGURE 4: Body acceleration signal is weighed according to ISO2631-1 standards.

FIGURE 5: Normalized damper characteristic and its functional fit (front).

FIGURE 6: Normalized damper characteristic and its functional fit (rear).

- (1) A function like a polynomial, rational, or an exponential function that can fit the basic initial data of force versus displacement/velocity profile is chosen first.
- (2) Then, a scaling factor C_{opt} is used for the function, such that

$$F_{\text{opt}}(x) = C_{\text{opt}} F(x). \quad (15)$$

The optimisation procedure requires computations of the objective function at each iteration step. The performance index used here includes both time and frequency domain analyses. The performance index evaluation can be summarized with the following steps.

- (1) First, the simulations of the quarter and the half car models subjected to a road excitation are carried out in the time domain. Note that while running the optimisation routines of the vehicle suspension models, two aspects could significantly affect results and might cause errors in the optimisation process as follows:
 - (a) it is preferable to consider the steady state response of the vehicle run. Since a constant vehicle speed is assumed, it takes some time for the vehicle to reach the steady state conditions. Therefore, the beginning part of the time domain simulation containing the transient response is omitted,
 - (b) attention should be paid to the static deflections (due to weights) and the initial conditions considering the static equilibrium points for the springs.
- (2) Then, the target point's accelerations are weighed according to ISO-2631 standards in the frequency domain and used as part of the objective function.
- (3) A suitable global objective function is established according to the needs of automotive manufacturers. This is the most subjective step of the methodology. Since the choice of the objective function and weighing of the particular objectives will result in different

FIGURE 7: Normalized spring characteristic (rear).

FIGURE 8: Road profile PSD versus wavenumber.

FIGURE 9: Generated road profile versus time.

FIGURE 10: Objective function's run history.

FIGURE 11: Change of indices throughout optimisation process.

optimum outcomes. Our choice of objective functions (performance index) for the current paper is explained in the previous section and in the following section on optimisation results.

- (4) As the final step, the optimisation type and algorithm are selected, and the optimisation step is performed in Matlab. The optimisation toolbox SQP algorithm with Quasi-Newton line search is implemented. The SQP algorithm like Simplex, Complex, and Hook-Jeeves belongs to the family of local search algorithms. The local search algorithms converge to the nearest optimum, since they depend upon the starting values of the design variables. Examples in the following section illustrate the simulation result for quarter car and half car vehicle models used here. Numerical simulation results show that the SQP algorithm can efficiently and reliably find the optimum in the neighbourhood of the initial point.

Finally, the optimum spring and damper characteristics obtained should be checked to see if they are manufacturable.

FIGURE 12: Optimum normalised linear spring stiffness and nonlinear damper characteristic scaling coefficients.

FIGURE 13: Performance indices throughout the optimisation process.

FIGURE 14: Default and optimized damper characteristic curves.

FIGURE 15: Default and optimized spring stiffness.

The nonlinear damper characteristic of the front independent suspension and rear solid axle of a light commercial vehicle are presented in normalized form in Figures 5 and 6, respectively. Since the shape of the curve is essential for manufacturing, an appropriate functional representation should be used in the optimisation process. A function with the following form is suitable for the whole range of the damper data

$$F(v) = Ae^{-kv} + Be^{qv}. \quad (16)$$

Equation (16) is used as an empirical curve fit to real damper characteristics. The parameters A , B , k and q in (16) are, therefore, determined empirically. v in (16) represents the

independent variable, which is suspension velocity here. The real experimentally determined suspension damping characteristics are shown as solid lines in Figures 5 and 6. The empirical data fit obtained using (16) is shown as dashed lines in Figures 5 and 6. The close fit between the real and fit (using (16)) damping characteristics shows that (16) provides a highly accurate representation of the real damping characteristics considered in this paper.

For the nonlinear modelling of the spring of the rear axle, a lookup table which represents the nonlinear characteristic of the spring is used (see Figure 7). The spring characteristic of the front independent suspension is linear in all optimisation processes.

FIGURE 16: Values of the objective throughout two-dimensional search domain.

FIGURE 17: Left and right side of the road generated with Robson Method.

FIGURE 18: Total objective function progress.

5. Optimisation Simulation Results

The optimisations results for the front and rear suspension units of the quarter and the half car models are presented in this section. The performance indices used in the optimisation process are presented in the following Sections (5.1 and 5.2). Finally, Carmaker software is used to check and to

FIGURE 19: Change of indices throughout optimisation process.

TABLE 1: Different road classes.

Road class	$C/10^{-8} \text{ m}^{0.5} \text{ cycle}^{1.5}$
Motorway	3–50
Principal road	3–800
Minor road	50–3000

confirm the optimised results using a high fidelity, full vehicle model.

5.1. Quarter Car Optimisation Results. Optimisation runs are performed using the quarter car model in this part of the investigation considering the vehicle with a nonlinear damper and linear spring unit subjected to a generated road. It is required to have road surface input profiles for the realistic response of the vehicle dynamic simulations. These input profiles may come directly from the measurements made by a test vehicle. It is also possible to artificially generate the random road profiles like Robson's Method, presented in Kamash and Robson [23]. In Robson's Method, the road profile spectra can be given by

$$G(n) = Cn^{-w}, \quad (17)$$

where w is a constant which is equal to 2.5, n is the wavenumber in cycle/m as described in ISO-8608, and $G(n)$ is the displacement spectral density in m^3/cycle . For different classes of roads C takes the values listed in Table 1 [24].

In the conventional spectral analysis the process is simply squaring the spectral coefficients which are determined by using discrete Fourier transform. It is required to have uniformly distributed random phases between 0 and 2π . The inverse Fourier transform is given by:

$$z_{r1} = \sum_{k=0}^{N-1} \sqrt{G_k} e^{-i(\theta_k + (2\pi kr/N))}, \quad r = 0, 1, 2, \dots, (N-1), \quad (18)$$

where G_k is described by

$$G_k = \left(\frac{2\pi}{N\Delta} \right) G_{11}(\gamma_k) \quad (19)$$

FIGURE 20: Spring and damper characteristics scaling coefficient progress.

FIGURE 21: Objective function components' progress.

FIGURE 22: Optimized and the default spring characteristics.

and γ_k is given by

$$\gamma_k = \frac{2\pi k}{N\Delta}, \quad (20)$$

where γ_k is the wave-number in rad/m, $G_{11}(\gamma_k)$ is the target spectral density, Δ is the distance interval between two spot heights, and θ_k is a set of independent random phase angles uniformly distributed between 0 and 2π . The road profile was generated via Robson's Method as is given in Kamash and

Robson [23] and also in Cebon [17]. The road profile used in the optimisation has $C = 250$ (see Figures 8 and 9).

The objective function has two components which are the weighed body acceleration RMS value and the penalty function of tire force difference. The objective function is defined as

$$I = w_1 \ddot{Z}_s + w_2 \Delta F_{\text{tire}}, \quad (21)$$

where $\ddot{Z}_s$ is the RMS value of the weighed body acceleration, and ΔF_{tire} is the difference between the maximum and minimum tire force picked from the tire force history. The total objective function changes as shown in Figure 10 during the optimisation run, and an optimum point is reached after 35 iterations. The minimisation of both indices in the objective function as the process goes on is illustrated in Figure 11 via representative steps. Figure 12 shows the normalised values of the design variables: the spring stiffness and damper characteristic scaling coefficients for the quarter car model. The spring stiffness increases, and the damper characteristic scaling coefficient decreases until an optimum point is reached. How the body acceleration RMS value $\ddot{Z}_s$ and the tire force change ΔF_{tire} change during the optimisation run are displayed in Figure 13. The initial and optimised damper and spring characteristics are shown in Figures 14 and 15, respectively. Using a brute force approach, the objective function is evaluated on a grid of spring stiffness and damping curve scaling coefficient values. The resulting plot in Figure 16 demonstrates that the optimisation results shown in Figure 13 are accurate.

FIGURE 23: Optimised and default damper characteristics.

FIGURE 24: Values of the objective throughout two-dimensional search domain.

FIGURE 25: Snapshot of Carmaker animation.

5.2. Half Car Model. The half car model embodying a nonlinear rear suspension unit of the light commercial vehicle incorporating nonlinear dampers and nonlinear springs whose basic characteristic curve shapes are given in the previous sections is considered in this subsection. The performance index used in the previous case (the quarter car) is also considered here with the addition of a function including body roll motion. The objective function (performance index) used

FIGURE 26: Roll angle comparison double lane change.

FIGURE 27: Roll rate comparison double-lane change.

FIGURE 28: Yaw rate change during the manoeuvre.

FIGURE 29: Vertical vehicle body acceleration PSD plots at four corners with high order model.

in optimisation is made up from three functions, as described below:

$$I = w_1 \ddot{Z}_s + w_2 \Delta F_{\text{tire}} + w_3 \Phi_s, \quad (22)$$

where Φ_s represents the RMS value of the roll angle of the vehicle. The difference in the performance index for this case lies on the existence of the roll angle.

The left and the right sides of the road being followed are shown in Figure 17, and they are generated independently using the Robson Method. Since both sides of the road are independent of each other, they have zero correlation between them. Arbitrary body roll motions will be induced as a result of this road profile in the simulation runs. The rear axle nonlinear suspension unit characteristics are optimized considering body roll angle.

The optimisation results for this case are presented in Figures 18–21. The change of the performance index during the course of the optimisation run that is given in Figures 18 and 19 emphasizes the minimisation of all three indices

step by step throughout the process. How the nonlinear spring and nonlinear damper scaling coefficients change during the optimisation is shown in Figure 20. The changes of components of the performance index are shown separately in Figure 21. Figures 22 and 23 show the initial and optimised damper and spring characteristic curves, respectively. Figure 24 demonstrates the search domain of the objective function utilized according to two design parameters changing in a predetermined interval. Based on these figures, it is seen that after the optimisation the spring characteristic tends to be softer, while the damper characteristic tends to be more damped.

5.3. Handling Tests Using Carmaker Software. Finally, the ride handling and comfort performances of the optimized suspension parameters are confirmed by employing the high fidelity Carmaker model of the considered vehicle (see Figure 25). A standard double lane change maneuver is utilized to observe the body roll improvement which mostly indicates better

FIGURE 30: Vertical wheel acceleration PSD plots at four corners with high order model.

handling. The simulation results for roll angle and roll rate in Figures 26 and 27 show that the nonlinear optimisation has improved roll handling performance. Figure 28 shows that the yaw rate handling has not changed significantly. A test track with road irregularities which contains various frequencies is employed for the confirmation of comfort performance of the optimized suspension settings. The simulation results in Figure 29 show the ride comfort improvement in the form of lower vehicle body vertical acceleration PSD values at the four corners of the vehicle. The simulation results in Figure 30 show that the tire accelerations have not changed significantly, illustrating the maintenance of the previous road holding characteristic.

6. Conclusions

Light commercial vehicle front and rear suspension units incorporating both linear or nonlinear springs and nonlinear

dampers were optimized to improve vehicle ride and handling. The nonlinear equations of motions of the quarter car and the half car representing the front and rear suspension models were presented and simulated in the Matlab/Simulink environment. Several aspects of performance criteria were considered for ride comfort and handling such as RMS values of weighed body acceleration, the range of tire forces, and RMS values of body roll angle. For each aspect of performance, time-domain performance measures were evaluated after the optimisation run. A simple optimisation methodology of nonlinear suspension unit was presented, incorporating typical data provided by car manufacturers for the initial characteristics. The methodology was based on keeping the shape of the damper and spring properties and curve fitting a proper function to these data and then scaling it throughout the optimisation process. Finally, the advantage of the nonlinear optimised suspension unit compared to a default suspension unit was demonstrated using a double lane change maneuver with a high fidelity full vehicle model in

Carmaker. In order to generalise the nonlinear suspension unit optimisation problem, an interactive Matlab toolbox was constructed and used in obtaining the results presented here.

The fact that the improvement between original performances and optimized ones is not too big in the results presented in the paper is due to the fact that we started with an already optimized suspension which had been designed earlier. We decided against creating a nonideal starting value for the suspension parameters and showing a large improvement in performance. We were indeed able to show that the suspension design provided to us was very close to its optimum configuration, and only small performance achievements were possible. This is also a very useful outcome as the suspension designer can analytically evaluate his design against the optimal one. The method presented in the paper is fully automated in the form of a Matlab program and its interactive graphical user interface. The results are also obtained much faster as compared to the conventional method of suspension design including a lot of trial and error.

Abbreviations

DOF: Degree of freedom
 I_i : Performance index, $i = 1, 2, 3 \dots$
 I : Total index
 RMS: Root mean square
 PSD: Power spectral density.

List of Symbols

Quarter Car Model

m_s : Sprung mass, kg
 m_u : Unsprung mass, kg
 z_s : Sprung mass vertical displacement, m
 z_u : Unsprung mass vertical displacement, m
 z_r : Road irregularity, m
 b_u : Linear tire damping coefficient
 k_s : Linear spring stiffness
 k_u : Linear tire stiffness
 B_F : Nonlinear damper characteristic function.

Half Car Model

m_s : Sprung mass, kg
 m_u : Unsprung mass, kg
 z_s : Sprung mass vertical displacement, m
 z_u : Unsprung mass vertical displacement, m
 z_{RL}, z_{RR} : Left and right road irregularities, m
 k_{uL}, k_{uR} : Left and right linear tire stiffness, N/m
 B_{FL}, B_{FR} : Left and right nonlinear damper characteristic functions, Ns/m
 K_{FL}, K_{FR} : Left and right nonlinear damper characteristic functions, N/m
 K_{Θ} : Antiroll bar torsional stiffness, Nm/rad
 L : Track width, m
 ϕ_s : Sprung mass roll angle, deg/sec

ϕ_u : Unsprung mass roll angle, deg/sec
 I_{xx} : Sprung mass moment of inertia about x -axis
 I_{uux} : Unsprung mass moment of inertia about x -axis.

Optimisation Process

w_i : Weighing, $i = 1, 2, 3 \dots$
 ΔF_{tire} : Tire force change, N
 $\Delta F_{\text{tire opt}}$: Optimised value of tire force change, N
 $\ddot{Z}_s$: RMS of weighed sprung mass acceleration
 $\ddot{Z}_{\text{spot}}$: Optimised value of the RMS of weighed sprung mass acceleration
 Φ_s : RMS of sprung mass roll angle.

Acknowledgments

The authors thank Yıldırar Koray for his help in the road profile used. The authors thank Server Ersolmaz, Erhan Eyol, Mustafa Sinal. and Selçuk Kervancioğlu of Ford Otosan for helpful discussions on suspension design. The authors also thank Ford Otosan and the EU FP6 project AUTOCOM INCO-I6426 for their support. The authors dedicate this paper to the dear memory of their second coauthor Professor Ü. Sönmez who passed away so suddenly and unexpectedly.

References

- [1] T. Tak and S. Chung, "An optimal design software for vehicle suspension systems," in *SAE Automotive Dynamics & Stability Conference*, Troy, Mich, USA, May 2000, SAE Paper no: 2000-01-1618.
- [2] D. Koulocheris, H. Vrazopoulos, and V. Dertimanis, "Optimisation algorithms for tuning suspension systems used in ground vehicles," in *Proceedings of International Body Engineering Conference & Exhibition and Automotive & Transportation Technology Conference*, Paris, France, July 2002, SAE Paper no: 2002-01-2214.
- [3] P. C. Wiers and A. K. Dhingra, "On the beneficial effects of anti-squat in rear suspension design of a drag racing motorcycle," *Journal of Mechanical Design*, vol. 124, no. 1, pp. 98–105, 2002.
- [4] S. A. Mitchell, S. Smith, A. Damiano, J. Durgavich, and R. MacCracken, "Use of genetic algorithms with multiple metrics aimed at the optimization of automotive suspension systems," in *Proceedings of Motorsports Engineering Conference and Exhibition*, Dearborn, Mich, USA, November 2004, SAE Paper no: 2004-01-3520.
- [5] M. Raghavan, "Suspension design for linear toe curves: a case study in mechanism synthesis," *Journal of Mechanical Design*, vol. 126, no. 2, pp. 278–282, 2004.
- [6] H. V. Deo and N. P. Suh, "Axiomatic design of customizable automotive suspension," in *Proceedings of the 3rd International Conference on Axiomatic Design Seoul (ICAD '04)*, 2004.
- [7] J. P. C. Goncalves and J. A. C. Ambrosio, "Road vehicle modelling requirements for optimization of ride and handling," *Multibody System Dynamics*, vol. 13, pp. 3–23, 2005.
- [8] P. Duysinx, O. Brûls, J. F. Collard, P. Fiset, C. Lauwerys, and J. Swevers, "Optimization of mechatronic systems: application

- to a modern car equipped with a semi-active suspension," in *Proceedings of the 6th World Congresses of Structural and Multidisciplinary Optimization*, Rio de Janeiro, Brazil, 2005.
- [9] A. Eskandari, O. Mirzadeh, and S. Azadi, "Optimization of a McPherson suspension system using the design of experiments method," in *SAE Automotive Dynamics, Stability & Controls Conference and Exhibition*, Novi, Mich, USA, February 2000.
 - [10] A. Boyalı, S. Öztürk, T. Yiğit, B. Aksun Güvenç, and L. Güvenç, "Use of computer aided methods in optimisation in vehicle suspension design and in designing active suspensions," in *Proceedings of the 6th Biennial Conference on Engineering Systems Design and Analysis (ESDA '02)*, Istanbul, Turkey, 2002.
 - [11] Y. He and J. McPhee, A review of automated design synthesis approaches for virtual development of ground vehicle suspensions SAE Paper no: 2007-01-0856, 2007.
 - [12] L. Li, C. Xia, and W. Qin, "Analysis of kinetic characteristic and structural parameter optimization of multi-link suspension," in *Proceedings of the 14th Asia Pacific Automotive Engineering Conference*, Hollywood, Calif, USA, August 2007, SAE Paper no: 2007-01-3558.
 - [13] P. E. Uys, P. S. Els, and M. Thoreson, "Suspension settings for optimal ride comfort of off-road vehicles travelling on roads with different roughness and speeds," *Journal of Terramechanics*, vol. 44, no. 2, pp. 163–175, 2007.
 - [14] H. Habibi, K. H. Shirazi, and M. Shishesaz, "Roll steer minimization of McPherson-strut suspension system using genetic algorithm method," *Mechanism and Machine Theory*, vol. 43, no. 1, pp. 57–67, 2008.
 - [15] J. P. C. Goncalves and J. A. C. Ambrosio, "Optimization of vehicle suspension systems for improved comfort of road vehicles using flexible multibody dynamics," *Nonlinear Dynamics*, vol. 34, no. 1-2, pp. 113–131, 2003.
 - [16] L. Sun, "Optimum design of "road-friendly" vehicle suspension systems subjected to rough pavement surfaces," *Applied Mathematical Modelling*, vol. 26, no. 5, pp. 635–652, 2002.
 - [17] D. Cebon, *Handbook of Vehicle-Road Interaction: Vehicle Dynamics, Suspension Design, and Road Damage*, Taylor & Francis, Boca Raton, Fla, USA, 1999.
 - [18] CarMaker Reference Manual and User's Guide.
 - [19] M. Li, Z. Changfu, P. Zhao, and J. Xiangping, "Parameters sensitivity analysis and optimization for the performance of vehicle handling," in *Proceedings of the 14th Asia Pacific Automotive Engineering Conference*, Hollywood, Calif, USA, August 2007, SAE Paper no: 2007-01-3573.
 - [20] M. Jonasson and F. Roos, "Design and evaluation of an active electromechanical wheel suspension system," *Mechatronics*, vol. 18, no. 4, pp. 218–230, 2008.
 - [21] BS 6841, "British standard guide to measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock," 1987.
 - [22] International Standard ISO-2631-1, *Mechanical Vibration and Shock-Evaluation of Human Exposure To Whole-Body Vibration, Part 1: General Requirements*, 2nd edition, 1997.
 - [23] K. M. A. Kamash and J. D. Robson, "The application of isotropy in road surface modelling," *Journal of Sound and Vibration*, vol. 57, no. 1, pp. 89–100, 1978.
 - [24] International Standard ISO-8608, *Mechanical Vibration—Road Surface Profiles—Reporting of Measured Data*, 1st edition, 1995.