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Comfort improvement of a nonlinear suspension using global optimization and in situ measurements

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Abstract

The health problems encountered by operators of off-road vehicles demonstrate that a lot of effort still has to be put into the design of effective seat and cabin suspensions. Owing to the nonlinear nature of the suspensions and the use of in situ measurements for the optimization, classical local optimization techniques are prone to getting stuck in local minima. Therefore this paper develops a method for optimizing nonlinear suspension systems based on in situ measurements, using the global optimization technique DIRECT to avoid local minima. Evaluation of the comfort improvement of the suspension was carried out using the objective comfort parameters used in standards. As a test case, the optimization of a hydropneumatic element that can serve as part of a cabin suspension for off-road machinery was performed.

1. Introduction

Today the "market value" of cars does not only depend on performance and price. Safety, comfort and environment friendliness tend to be of equal importance and are for some manufacturers the main selling points. Driving this trend are norms and directives concerning these issues and also the rising awareness of the consumer. Off-road vehicles do not escape this

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trend in the market. This paper deals with the comfort aspect narrowed down to the whole-body vibration levels the operators are exposed to when handling these machines.

Extensive research in the past has demonstrated that truck drivers, agricultural machinery operators, subway operators, tractor drivers and construction vehicle operators are common victims of low back problems (truck drivers are four times more likely to have a herniated disk, compared to people not involved in this line of work) [1,2]. The origin of this discomfort is vibrations transmitted to the driver, which are caused by the unevenness of the road or soil profile or by moving elements within the machine or implements. Increased low-frequency levels between 0.5 and 10 Hz are transmitted to the seat during field operations and cyclic motions like those caused by vehicle's tires hitting the road elevate the vibration levels in the frequency range of 2–20 Hz [3,4]. The spine is especially susceptible to severe physical damage in this frequency range [5,6]. The damage is caused through "cumulative trauma" and is therefore difficult to assess.

Unexpectedly, the impact on the economy is huge. Not only does discomfort during work lead to performance problems [7], but low back pain is the leading major cause of industrial disability in those younger than 45 years, and accounts for 20% of all work injuries. The total cost per year for the United States is estimated at $90 billion [8]. So, it is of common interest to the government, the operators and the manufacturers to deal with this problem.

The European Parliament has already acted. General standards like ISO 2631 [9] and BS 6841 [10], used for all types of vehicles in which whole-body vibrations exposure occurs, were improved in a new Machinery Directive (98/37/EC Annex I par. 1.5.9 and 3.6.3) which will come into force in the near future. The new directive imposes limits on the daily use of equipment and machinery in order to restrict the cumulative trauma caused by whole-body vibrations. To meet this directive several aspects need to be taken into account in the design or improvement of mobile machinery one of which is the development of adequate suspension systems in seats, cabins and axles [11].

This paper discusses an approach in which the optimization of nonlinear suspension systems was performed based on in situ measurements in combination with a global optimization technique. As an example the method was tested on a hydropneumatic suspension device that could be suitable for use in a cabin suspension design.

2. Optimization procedure

2.1. Overview

In order to evaluate the behavior of off-road machinery a large number of test drives are performed under different conditions e.g. road profile, motor load, use of implements, etc. During these tests, a large amount of data is collected varying from the vibration levels at different positions on the machine to the noise levels produced in- and outside the cabin. These tests are also used to monitor the behavior of the suspension systems on the machine by measuring the vibration levels before and after the suspension. The outcome of all these tests is not only diagnostic information on the behavior of the machine but also a large amount of additional data.

If vibrational data, displacement, velocity or acceleration measurements, collected during such test drives could be used for preliminary studies to determine if new suspension systems would perform adequately under real conditions, the high costs for design and testing of prototypes could be reduced drastically. It can be seen as an additional test to prevent setting up prototypes which would result in mediocre results.

2.2. Implementation

To make use of these in situ measurements in an optimization procedure, the following four steps have to be followed:

(1) data collection,
(2) modelling,
(3) goal definition,
(4) optimization.

The whole procedure was developed using Matlab [12] and Simulink [13]. The main advantage of using a single platform was that all steps were performed in the same environment and problems like data conversion were avoided.

2.2.1. Data collection

When monitoring a machine during a field test, vibration levels are generally recorded at different positions on the machine. Those of interest here are the motions situated at the base of a suspension device that has to be replaced by a new one or at places where a suspension will be introduced. Examples of this are the vibrations at the support of a cabin and vibrations of the cabin floor at the place where the seat is situated.

2.2.2. Modelling

In order to be able to optimize a suspension all of it, or at least a part of it, has to be modelled using physical principles. Since the method uses in situ input measurements, it is not necessary to limit the modelling to the linear case. The use of real measurements makes it possible to give a correct interpretation to the behavior of a nonlinear model in both time and frequency domain.

2.2.3. Goal definition

Since the aim of the presented work was to design suspension devices that give an added value with respect to the improvement of comfort, the goal functions were selected based on this.

In the time domain it was possible to optimize the suspension with respect to comfort improvement by using the objective comfort parameters stated in the standards ISO 2631 and BS 6841. Objective comfort parameters attempt to estimate the subjective discomfort by calculating a value based on filtered acceleration data. The filtering eliminates those frequencies that have no influence on the comfort and health of the drivers. Commonly used objective comfort parameters are the vibration dose value (VDV) and the effective root means square (effRMS) [1].

The general formula for VDV is

$$\mathrm{VDV} = \left[\frac{T_s}{N} \sum_{n=1}^{n=N} a^4 \right]^{1/4} \tag{1}$$

with T_s the measurement period, N the number of points and a the frequency weighted acceleration data. This parameter is time dependent and gives an objective measure of the amount of vibrations a person had to experience within a certain period. EffRMS is given by

$$\text{effRMS} = \left[\frac{1}{N}\sum_{n=1}^{n=N} a^2\right]^{1/2} \tag{2}$$

with N and a defined in Eq. (1). The RMS value is time independent and gives an idea of the general level of vibrations.

Despite the fact that the time domain approach was straightforward, an optimization using a goal function determined in the frequency domain was also used. This was carried out by postulating a desired magnitude of the frequency response function (FRF) and using the deviation between this desired magnitude and the magnitude of the measured FRF in a certain frequency band as the function to be minimized. The formula for this deviation, DIFF, is given by

$$\text{DIFF} = \left[\frac{1}{K}\sum_{k=1}^{k=K} W_k(A_k - B_k)^2\right]^{1/2}, \tag{3}$$

in which K stands for the number of frequency lines used, W_k for the weighing factor at frequency line k, A_k for the desired magnitude and B_k for the measured magnitude at frequency line k, all expressed in a linear scale.

Fig. 1 gives an example of how optimization using DIFF can be applied. The desired FRF for a one degree of freedom suspension system is labelled as "aim" in Fig. 1. Applying different weighing functions in the formulation of DIFF yields different solutions. Putting emphasis on

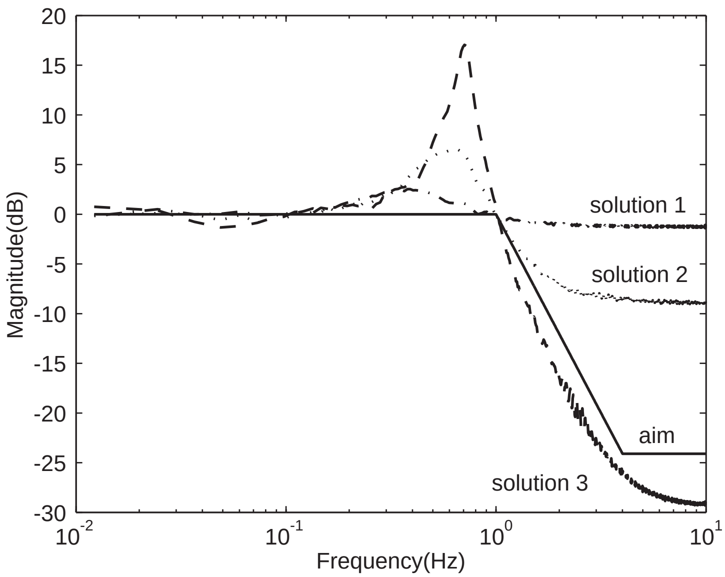

Fig. 1. Magnitude plot of the frequency response function of three solutions obtained by optimization in the frequency domain using different weighing functions together with the desired magnitude plot.

suppressing the resonance peak leads to solution 1 while stressing the behavior above resonance results in solution 3. The weighing function should be chosen in such a way that it reflects the designers purpose for the suspension.

The aim of using two different goal functions was to show that differences between both approaches can be exploited towards obtaining different optimal suspension designs.

2.2.4. Optimization

Once the goal function is established, the problem can be described in the following way:

$$\begin{array}{ll} \min & f(x) \\ \text{s.t.} & x \in [u, v] \end{array} \tag{4}$$

$$\text{with } [u, v] := \{x \in \mathbf{R}^n \mid u_i \leqslant x_i \leqslant v_i, i = 1, \ldots, n\}.$$

In this equation $f(x)$ is the goal function, x represents the parameters and u and v the parameter bounds. Derivative-based optimization techniques do not have a problem with optimizing smooth functions but Fig. 2 gives an example of a goal function obtained during optimization of the suspension of Section 3. This function could not be optimized using the classical optimization techniques programmed in Matlab since those methods got stuck in the various local minima. To overcome this problem the use of a global optimization technique was proposed.

There exists a variety of global optimization techniques that can deal effectively with the local minima problem but the algorithms can be classified into two main groups: heuristic methods, like evolutionary algorithms and simulated annealing, that find the global minimum with a certain probability and deterministic methods that guarantee to find the global minimum with a required

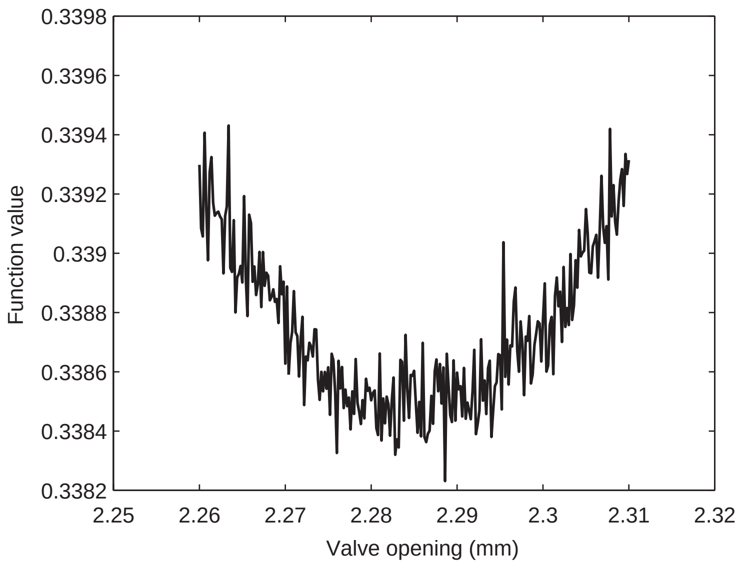

Fig. 2. Objective function near global minimum when changing one parameter of the hydropneumatic passive suspension system.

accuracy [14]. The method used here is called DIRECT. It is a deterministic, "branch and bound" technique implemented in the Matlab routine glbSolve [15].

DIRECT is an algorithm, presented by Jones et al. [16], for finding the global minimum of a multivariate function subject to simple bounds. The algorithm is an extension of the standard Lipschitzian approach that eliminates the need to specify a Lipschitz constant. This is done by carrying out simultaneous searches using all possible constants from zero to infinity. In Ref. [16], Jones et al. introduces a different way of looking at the Lipschitz constant. In particular, the Lipschitz constant is viewed as a weighing parameter that indicates how much emphasis to place on global versus local search. In standard Lipschitzian methods, this constant is usually large because it must equal or exceed the maximum rate of change of the objective function. As a result, these methods place a high emphasis on global search, which leads to slow convergence. In contrast, the DIRECT algorithm carries out simultaneous searches using all possible constants, and therefore operates on both the global and local level. Once the global part of the algorithm finds the basin of convergence of the optimum, the local part exploits it automatically. This accounts for the fast convergence of the DIRECT algorithm.

3. Hydropneumatic suspension

A nonlinear model of the hydropneumatic suspension of Fig. 3 was used to test the optimization procedure presented in Section 2. The aim was to determine if a hydropneumatic suspension could be used to improve the comfort inside the cabin of a combine harvester. The system consisted of a hydraulic cylinder with rod diameter of 18 mm and cylinder diameter of 32 mm, two nitrogen bulbs with an adjustable volume varying between 0.5 and 2 liter at rest and a valve with a maximum opening diameter of 10 mm. The pressure inside the bulbs and their volume

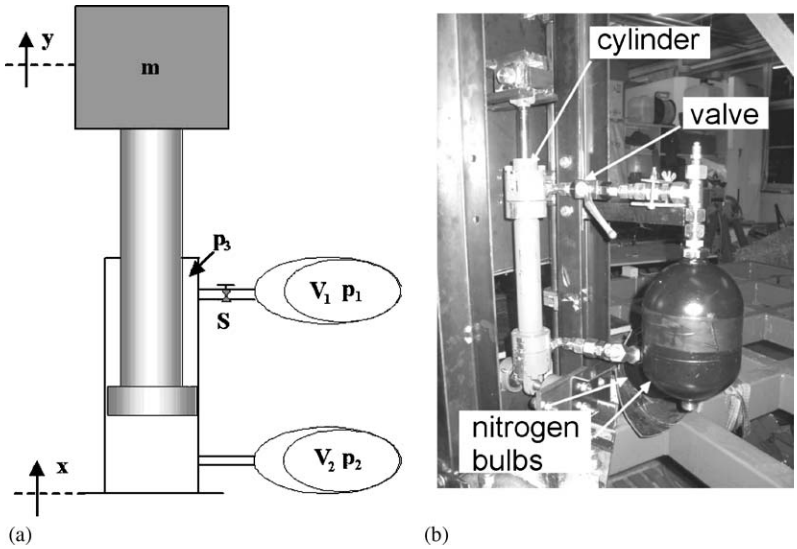

Fig. 3. Scheme and photo of the hydropneumatic suspension system: (a) schematic and (b) photo.

combined with the mass that had to be supported, in this case 200 kg, determined the natural frequency of the suspension while the opening of the valve determined the damping. This type of suspension is used as axle suspension by some sprayer manufacturers since it is able to provide a low spring rate which is necessary to enhance the ride comfort of such machines. Giliomee and Els discussed the use of a semi-active variant for armed fighting vehicles and heavy off-road vehicles [17]. This form of suspension is highly tunable, which makes it suitable as a test-case.

3.1. Data collection

The input signals were provided from road and field measurements. Under road and field conditions, the vibrations at the base of the cabin suspension of a combine harvester were measured using accelerometers. The measurements used here were obtained at a driving speed of 4 km/h on the field, 11 km/h on an unpaved road and 28 km/h on a paved road. The input signals were recorded for a period of 122.88 s at a sampling rate of 200 Hz. The VDV value for the paved road was 5.606 $m/s^{1.75}$, for the unpaved road 3.049 $m/s^{1.75}$ and for the field track 4.108 $m/s^{1.75}$.

3.2. Modelling

In order to model the suspension of Fig. 3, the relation between the movement of mass y and the input x was deduced. This led to a so-called "base motion" model of a quarter cabin. Applying Newton's law to the mass m resulted in the following equation:

$$m\ddot{y} = p_2 S_{\mathrm{cyl}} - mg - p_3(S_{\mathrm{cyl}} - S_{\mathrm{rod}}) - F_W \tag{5}$$

with p_2 and p_3 the pressures in the system (see Fig. 3(a)), S_{cyl} the surface of the piston, S_{rod} the surface of the rod, g the gravitational acceleration and F_W the friction force acting between piston and cylinder.

The nitrogen bulbs compress and expand according to the adiabatic law:

$$pV^{\kappa} = \text{constant}, \tag{6}$$

in which V stands for the volume and κ for the ratio of specific heats. The volume in this equation could be determined using the initial volume of the nitrogen bulbs and the positions x and y. This equation gave the solution for the pressures p_1 and p_2.

Pressure p_1 could be related to pressure p_3 and the velocities $\dot{x}$ and $\dot{y}$ using Eq. (7) which describes the rate of oil flow through the valve:

$$(S_{\mathrm{cyl}} - S_{\mathrm{rod}})(\dot{y} - \dot{x}) \cong S\sqrt{p_1 - p_3} \tag{7}$$

with S the surface of the opening of the valve.

The only unknown in Eq. (5) is the friction force F_W. The friction between the cylinder and the rod make it subject to stick–slip behavior and is therefore described using the following function [18]:

$$F_W = F_C + (F_s - F_C)\mathrm{e}^{-(\dot{y}-\dot{x}/\dot{x}_s)^2} + F_v(\dot{y} - \dot{x}), \tag{8}$$

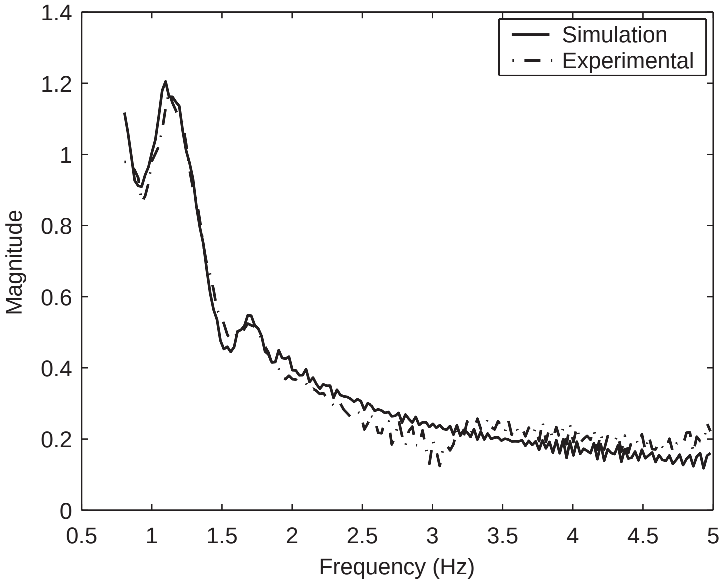

Fig. 4. Comparison between the frequency response function of model (full line) and physical system (dashed line) for a swept sine excitation signal.

in which F_C the minimum Coulomb friction, F_s the level of static friction and F_v the viscous friction term. The parameter $\dot{x}_s$ is empirical and was set to 0.003 m/s. It regulates the transition between the static friction and the Coulomb friction and it is not a very critical parameter in this model, since variations of it up to 100% change the VDV of the signals with less than 1%.

The combination of Eqs. (5)–(8) resulted in a nonlinear model of the hydropneumatic suspension. To verify the correctness of this model Fig. 4 compares the FRFs of the model and the physical system when excited using a swept sine displacement signal with a frequency content between 0.8 and 5 Hz, a constant sweep rate, a duration of 41 s and an amplitude of 1.5 cm. When using the following values for the coefficients: $F_s = 220\,\mathrm{N}$, $F_C = 100\,\mathrm{N}$, $F_v = 2\,\mathrm{N\,s/m}$, $p_3 = 3.5$ MPa, $S = 9.6\,\mathrm{mm}^2$ and nitrogen volumes of 1.5 liter, it can be deduced from Fig. 4 that in most frequencies there is a good general agreement between the model and the experimental results.

Fig. 5 compares the output of the model with the measured output when the system is excited using a multisine having the same frequency content and duration as the swept sine and a maximum amplitude of 1.5 cm.

Since the goal of the presented work was to investigate for improvements concerning comfort behavior, it was of great importance that the VDV of the output of the model was closely related to the VDV of the measured signal. The comparison was performed by using

$$\frac{|\mathrm{VDV_{real}} - \mathrm{VDV_{model}}|}{\mathrm{VDV_{real}}} = \frac{|1.794 - 1.673|}{1.794} = 0.068. \tag{9}$$

The error in the prediction of the VDV was approximately 7% for the multisine shown in Fig. 5 and even less for the swept sine. The model was considered to be a useful simulation of the real suspension system.

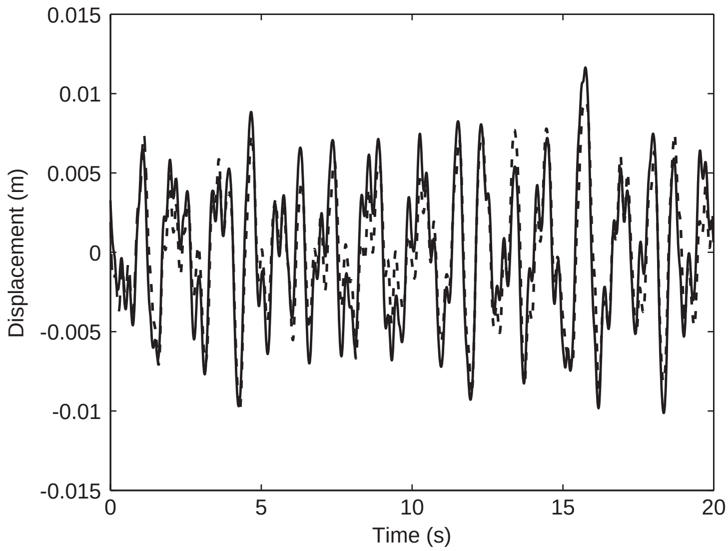

Fig. 5. Comparison in time domain between the output of the model (full line) and that of the physical system (dashed line) for a multisine excitation signal.

3.3. Goal function

In the time domain the VDV of the output acceleration was used as a goal function as given by Eq. (1).

In the frequency domain the desired FRF resembled the aim of Fig. 1. The magnitude up till 1 Hz was equal to 1 and between 1 and 4 Hz it dropped at a ratio of −40 dB/decade to an attenuation level of approximately −17 dB above 4 Hz. The goal function in the frequency domain was given by Eq. (3). A uniform weighing has been applied in the "DIFF" formulation.

3.4. Optimization

During the optimization procedure, four parameters in the system were used as variables: the valve opening S, the static pressure p_2 and the volume of both nitrogen volumes in the bulbs V_1 and V_2 (see Fig. 3(a)). Optimization was performed in time and frequency domain using three different input signals called "field", "paved road" and "unpaved road". This resulted in six different optimal solutions. Tables 1 and 2 give, respectively, for each optimal suspension, the values of the goal function for the time and the frequency domains optimization for each of the three input signals, together with the values for the four optimized parameters. The bold figures in both tables indicate the minimum value of the goal function retrieved by the optimization routine.

3.5. Discussion

The optimization procedure was able to determine the best solution for each of the six optimizations since the bold figures are the lowest in their category. This also means that 6 distinct

Table 1
Calculated goal function value for suspensions developed using time domain optimization

Based on	Input profile	VDV ($m/s^{1.75}$)	DIFF	Parameters
Paved road	Paved road	**1.132**	3.256	$p_2 = 2.51$ MPa
	unpaved road	1.105	2.979	$S = 19.4\,mm^2$
	field	1.056	8.169	$V_1 = 1.21$ l, $V_2 = 1.12$ l
Unpaved road	Paved road	1.165	2.513	$p_2 = 2.51$ MPa
	unpaved road	**1.073**	2.629	$S = 19.1\,mm^2$
	field	1.327	9.178	$V_1 = 1.35$ l, $V_2 = 0.88$ l
Field	Paved road	1.717	1.174	$p_2 = 2.55$ MPa
	unpaved road	1.186	1.426	$S = 14.7\,mm^2$
	field	**1.022**	9.104	$V_1 = 1.73$ l, $V_2 = 1.58$ l

suspension systems are retrieved. The reason for that lies in the use of a nonlinear model and the fact that although both goal functions aimed at the improvement of the comfort behavior, the expression for it is different.

This could raise the question if this method has a purpose since every measurement of the machine under distinct operating conditions will result in a different input signal and could lead to a different optimal suspension system. Therefore it is crucial to select those input signals that give a good representation for the general behavior of the machine under normal working conditions. Table 1 shows also that the optimal suspension for one input signal does not mean bad behavior for the other input signals. In this case one passive suspension system can be used. If on the other hand higher performance is required, the optimal solutions can be used in a semi-active approach of the suspension system.

Table 1 shows that the optimization based on the VDV resulted in three almost equal suspensions with a low natural frequency (due to low pressure and large volumes of nitrogen) and small amount of damping (due to large valve opening). This was not the case when the optimization was performed based on DIFF which resulted in three distinct suspension systems (see Table 2). The explanation for this lies in the fact that VDV focuses on low acceleration levels which can be provided for all input signals from a suspension with low natural frequency. DIFF on the other hand shapes the transmissibility of the suspension and in this way it also restricts high displacements at low frequency which will not show up in the VDV since the acceleration level of low frequency displacements is low. What can be learned from this for the design of suspension systems is that the choice of goal function is a crucial factor. In this specific case the suspension systems of Table 2 will be preferred against those of Table 1 despite the fact that their VDV is higher. Slow movements due to the low natural frequency of the latter can lead to the seasickness phenomenon for the driver, which has to be prevented.

Both tables also show that whichever optimization method was used, a reduction of the VDV in comparison with the input signals between 30% and 75% could be established. Based on this information only, even in light of the 7% error in predicting the VDV, it can be said that using

Table 2
Calculated goal function value for suspensions developed using frequency domain optimization

Based on	Input profile	VDV ($m/s^{1.75}$)	DIFF	$p_2 = 2.51$ MPa
Paved road	Paved road	2.442	**0.424**	$p_2 = 4.12$ MPa
	unpaved road	1.646	1.251	$S = 9.3\,mm^2$
	field	1.357	11.007	$V_1 = 1.31$ l, $V_2 = 0.97$ l
Unpaved road	Paved road	2.867	0.660	$p_2 = 2.58$ MPa
	unpaved road	1.562	**0.598**	$S = 7.1\,mm^2$
	field	1.064	8.972	$V_1 = 1.31$ l, $V_2 = 0.75$ l
Field	Paved road	1.170	3.498	$p_2 = 2.55$ MPa
	unpaved road	1.139	2.846	$S = 17.9\,mm^2$
	field	1.359	**7.274**	$V_1 = 1.36$ l, $V_2 = 1.02$ l

this type of hydropneumatic element in the design of a cabin suspension can give satisfactory results. In this way the presented method provides a tool for deciding if a suspension system can be optimized to perform adequately under real conditions.

During the optimization, the advantages of DIRECT became clear. The combination of local and global search, resulted in fast convergence to the global minimum of the goal function. The fast convergence compensated to a great extend the disadvantage of not having a stopping criterium. In future work the use of different optimization techniques can be considered. Other aspects that can be included in future work are the optimization of MIMO suspension systems or further investigation about parameter optimization of semi-active control laws.

4. Conclusion

The presented paper demonstrated how vibration measurements collected during field tests could be used in the optimization of a nonlinear suspension system. After establishing the model of the suspension, optimization was performed using the global optimization technique DIRECT. The used goal functions were based on the objective comfort parameter vibration dose value and the frequency response function.

By using in situ measurements the method was able to predict the behavior of a nonlinear suspension system under real conditions without actually having to build a prototype. It can be considered as a design approach incorporating a feasibility assessment.

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