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Optimization of agricultural tractor cab suspension using the evolution method

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Abstract

The use of cab suspensions on agricultural tractors increases the possibilities to reduce the vibration load on the driver within the limits imposed by the major constraint, i.e. the available travel space.

This paper describes a model for optimization of parameters describing the characteristics of a passive nonlinear cab suspension. The optimization method is based on an evolution algorithm. The objective has been to minimize the total vibration load on the driver.

The studies performed show that the method is a very useful tool for optimization of cab suspension characteristics. The absolute optima were not always found but the differences in the values of the objective function were small.

Keywords: Tractors, agricultural; Tractor cab suspensions; Optimization; Passive suspensions

1. Introduction

The vibration load on the driver of an agricultural tractor influences the driver's health and causes not only decreased quality of the performed job but also a lower degree of utilization of the machine.

A suspended seat, which is a relatively inexpensive and robust construction, is normally used to decrease vibrations. The main problem with seat suspensions is the relative motion between the driver and the cab. It is also very difficult to achieve effective seat suspension in horizontal and rotational directions, which is very important on an agricultural tractor, where this type of vibration is particularly common (Suggs and Huang, 1969).

Better vibration protection can be obtained if the whole cab is attached to the rest of the vehicle by means of a suspension (Hilton and Moran, 1975; Kauss and Weigelt, 1980). Benefits obtained are the possibilities for damping in six degrees of

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freedom (d.o.f.), elimination of relative movement between the driver and the cab and increased possibilities to reduce structurally transmitted noise.

A suspension system placed between the wheels and the rest of the vehicle offers even higher potential to decrease vibration levels. This solution needs, however, advanced and expensive techniques to counteract static loads and is not yet a realistic alternative.

Important variables involved in the measurement of vibration exposures are magnitude, frequency, direction and duration. An international standard for evaluation of human exposure to whole body vibrations was published in 1974 (ISO 2631, 1974). The standard includes exposure time limits for humans in vibrating environments.

The human body shows different sensitivity to vibrations depending on direction and frequency content. In the vertical direction, the body's most sensitive range is 4–8 Hz, while sensitivity in the horizontal directions is highest between 1 and 2 Hz. In general, the body is also more sensitive in horizontal directions compared with the vertical direction. To enable an evaluation of the total load, ISO 2631 defines a vector sum as the weighed sum of the vibration loads in the three linear directions.

A method for measurement and analysis of the vibration load on the driver of an agricultural tractor is also described in a standard (ISO 5008, 1979). ISO 5008 also describes two test tracks, one rougher and one smoother, designed to imitate normal driving conditions in agriculture, and useful for comparative studies of different vibration damping systems.

The purpose of this paper is to optimize parameters in a passive cab suspension with respect to different types of generally defined constraints. The objective function used has been the vibration load on the driver, measured according to ISO 2631 (1974).

A cab suspension with springs and dampers is a complex construction with numerous parameters influencing the vibration damping capacity. An analytic simulation model describing the system has been developed and offers possibilities to investigate the influence of different suspension principles and variables in a scientific way.

A model for studying different types of cab suspensions can describe the whole vehicle including the ground-tyre interplay or just the frame–cab interplay. The possibilities to describe, with reasonable precision, the transmission of ground irregularities through the rubber tyres to the frame are small. Incomplete knowledge of the tyres' dynamic properties and of the ground-tyre interaction are contributory causes (Crolla, 1980; Stayner et al., 1984). More recent work on tyre dynamics has resulted in the errors decreasing, but even better tyre dynamic descriptions are required (Lines et al., 1992).

The optimization of a multi-dimensional nonlinear function, including local minimas, is a difficult task. Normally used algorithms, such as the gradient method, lead to major problems, especially with the local minimas, and cannot be used. Algorithms based on the evolution strategy have shown very good characteristics in this kind of optimization (Muth, 1982) and are, therefore, used for the optimizations described in this paper.

The evolution strategy, as the name indicates, is inspired by biological evolution, with mutations and selections. A range of parameter vectors are formed with statistically decided deviations from an initial estimate vector. Suspensions designed according to each of the parameter vectors are then studied in simulations and the suspension resulting in the most favourable value on the objective function is chosen as base for the next generation of estimates. The procedure is repeated until the function has been optimized with necessary precision.

2. The simulation model

The simulation model is defined only to describe the frame-cab system. A model describing this system is not as complicated as a model also describing the ground-tyre interplay in six d.o.f., which decreases the simulation times and eliminates the influences of incomplete tyre models. The cab's mass is assumed to be so small, in relation to the rest of the vehicle, that cab movement does not affect the movements of the rest of the vehicle. Accelerations measured at the vehicle frame when driving over representative surfaces are used as input to the model.

The simulation model is designed for time domain simulation and describes the movements in all six d.o.f. The algorithm is based on differential equations, describing the cab's movements when influenced by defined forces from the suspension elements (Symon, 1971). The cab and the frame are assumed to be solid bodies without elasticity.

The Simnon software package (Elmqvist et al., 1986) has been used to solve the differential equations describing the cab's movements. Simnon is a commercially available software especially designed for simulation in the time domain, with several integration algorithms implemented. A Runge–Kutta algorithm of the order 4/5 and a maximum time step of 0.005 s has been used. The simulation model is described in greater detail in Hansson (1993).

2.1. The simulation model validation

In order to validate the simulation model, a full-scale model of the cab and the suspension was built. The suspension elements were mounted onto a surrounding foundation analogous to a vehicle frame. The vibrations at the cab and at the "frame" were recorded during a time period when the system was placed on a five d.o.f. hydraulically operated vibrating platform.

The movements of the frame, together with measured suspension parameters, were then used as inputs to the simulation model. The model was programmed to calculate the vibrations transmitted to the cab. Comparing the measured vibrations in the cab with the corresponding results from the simulation model, offered a possibility to examine the correctness of the simulation program.

The validation measurements and the results are described in detail in Hansson (1993). The results from the validation indicate that the simulation model is able to calculate the transmission of vibrations with high accuracy. The disagreement found between the simulated and measured movements is more likely to be dependent on elasticity in the foundation, slip-stick effects in the suspension elements, and measurement noise than on errors in the simulation model algorithm.

3. The cab and suspension characteristics

3.1. The geometry

The geometry for the cab suspension used for the optimizations is relatively symmetric because it offers possibilities to calculate approximate natural frequencies and degrees of damping in the different dimensions. The geometry has been the same in all the performed optimizations.

The cab suspension is described schematically in Fig. 1. The cab is standing on the four vertical elements (elements 3–6 in Table 1) shown in the figure. These elements have an ordinary spring and damping function in the axial direction. The axial functions then mainly influence the vibrations in the z dimension and the rotational movements around the x and y axes.

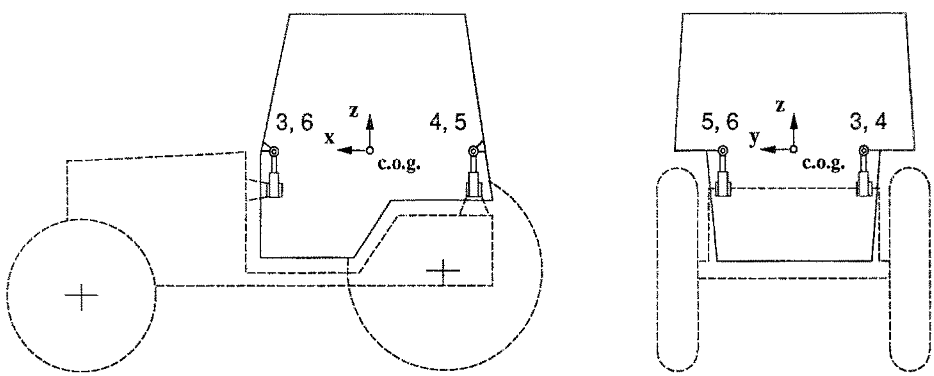

Fig. 1. Schematic view of the cab suspension. Element 3 is located in the cab's front right corner; element 4 in the rear right corner; element 5 in the rear left corner; and element 6 in the front left corner.

Table 1
The coordinates for the cab and frame end points of the elements in the basic suspension measured relative to the cab's c.o.g.

Element	Cab			Frame		
	x (m)	y (m)	z (m)	x (m)	y (m)	z (m)
1a	0	0.60	0	−0.40	0.60	0
1b	0	−0.60	0	−0.40	−0.60	0
2a	−1.00	0	0	−1.00	−0.40	0
2b	1.00	0	0	1.00	−0.40	0
3	1.00	−0.60	0	1.00	−0.60	−0.40
4	−1.00	−0.60	0	−1.00	−0.60	−0.40
5	−1.00	0.60	0	−1.00	0.60	−0.40
6	1.00	0.60	0	1.00	0.60	−0.40

The lower parts of the vertical elements are mounted elastically in the frame. The elastic mounting contributes to a torque being needed to bend the element from the balanced vertical position. The torque affects the cab with a force in the element's radial direction. The connection between an element and the cab is constructed as a frictionless ball joint. The relation between the bending angle and the resulting torque depends on the design of the elastic bushing. A robust rubber bushing, a coil spring, etc., can be used.

It is assumed that the torque produced in the bushing is frequency-independent, and that the bushings work without energy dissipation. If the bushings have a linear relation between the applied torque and the bending angle, the suspension gets linear spring characteristics in the horizontal directions.

Traditional symmetrically mounted damping elements (elements 1a, b, and 2a, b in Table 1) are used to damp the movements in the horizontal directions.

The coordinates for the suspension elements' cab and frame end points in the balanced position are given in Table 1. The coordinates are measured relative to an origin at the cab's centre of gravity (c.o.g.).

The characteristics for the cab used were: $I_{xx} = 444$ kg m^2; $I_{yy} = 506$ kg m^2; $I_{zz} = 213$ kg m^2; $m = 580$ kg.

The cab's c.o.g. position was: x: 0.60 m in front of the rear axle; y: at the tractor's symmetry line; z: 1.50 m above the ground.

3.2. The suspension element characteristics

A passive cab suspension implemented with linear elements must have low natural frequencies in all dimensions to provide good vibration insulation. The low natural frequencies result in large suspension strokes and the need for a large amount of free space to avoid over-travel under rough conditions.

Nonlinear passive damping elements with harder damping at increasing strokes make it possible to use suspensions with low natural frequencies, even with a relatively low amount of free space. Nonlinear spring elements also reduce the strokes, but in contrast to nonlinear damping elements, they also cause substantial increases to the vibration load in the cab (Hansson, 1993).

The undamped natural frequency (f_n) for a suspension is defined by:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

where k = the added spring constants of the springs working in the studied direction and m = the cab's mass.

The degree of damping (R) is defined by the size of the damping constant (c) in relation to the critical damping constant (c_c):

$$R = c/c_c$$

where

$$c_c = 2\sqrt{km}$$

The working principle for the nonlinear dampers used in the simulations can be described by:

$$F_D = -c(l_e)\frac{dl_e}{dt}$$

where the damping constant is dependent on the element's length:

$$c(l_e) = c_0\left(1 + \left|\frac{l_e - l_{e0}}{PK1_c}\right|^{PK2_c}\right)$$

$$R_0 = c_0/c_c$$

where: F_D = damping force; l_e = element length; l_{e0} = element length in the balanced position; c_0 = damping constant in the balanced position; R_0 = degree of damping in the balanced position; $PK1_c$ = progressivity constant 1; and $PK2_c$ = progressivity constant 2.

The constants $PK1_c$ and $PK2_c$ decide the damping elements' nonlinear characteristics. $PK1_c$ can be defined as the deviation from the balanced position where c is twice as high as c_0. $PK2_c$ describes the shape of the curve; $PK2_c$ = 2 defines a quadratic curve, etc.

The parameters f_{nV}, $PK1_{cV}$, $PK2_{cV}$ and R_{0V} have been used to describe the characteristics for vertically working elements and f_{nH}, $PK1_{cH}$, $PK2_{cH}$ and R_{0H} for horizontally working elements. The suspension elements' characteristics are completely defined by these eight parameters.

The connection between the suspension's defined natural frequencies and damping characteristics and the parameter values used in the suspension model are described in Hansson (1993).

4. The optimization method

The optimization method is based on an evolution algorithm. Such algorithms have shown very good characteristics in optimization of multidimensional functions including local minimas and are, therefore, used for the optimizations described in this paper. The evolution strategy, as the name indicates, is inspired by biological evolution, with mutations and selections. A range of parameter vectors are formed with statistically decided deviations from an initial estimate vector. Suspensions designed according to each of the parameter vectors are then studied in simulations and the vector resulting in the lowest vibration load on the driver is chosen as base for the next generation of estimates. The procedure is repeated until the function has been optimized with necessary precision.

4.1. General strategy for the evolution method

Let μ individuals be parents to λ children. Choose the μ best individuals among the children and let them be parents to the next generation of λ children. The number of children must be higher than the number of parents. The best individuals

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objective function calculated. When all the nine vectors were tested, the one with the lowest value for the objective function was chosen as initial estimate for the next generation and s_{i+1} was decided according to the rules described above and with $\alpha = 1.5$.

Normally, the evolution has been simulated for 50 generations. In cases when the algorithm failed to converge after 50 generations, another 10–20 generations were simulated.

The evolution method normally works with the same standard deviations for all parts in the parameter vector. In a cab suspension the parameter magnitude is very variable and each parameter has been given an initial standard deviation of its own. This value has then been changed according to the principles described earlier. The technique is equivalent to a type of parameter scaling, and resulted in no further scaling being needed. The programming and the interpretations of the results have therefore been easier to perform.

When a constraint was exceeded (the travel became too large) the simulation was immediately interrupted. The parameter vector was exchanged for a new one, formed with the same standard deviations, and the optimization could continue. Therefore, when the "best" parameter vector was to be selected there were always nine possibilities and none of them had broken the restriction.

The possibility to penalize the parameter combination breaking the restriction instead of exchanging it was also tried, but problems arose when the optimal solution was close to the restriction. Then, in some cases, 6–7 out of 9 of the parameter guesses were penalized. The best of the remaining 2–3 did not always bring the algorithm closer to the optimum. Therefore, the algorithm became unstable and had problems in reaching the optimum.

The numerous repetitions of the simulation program and the analysis of the result required 4–8 days for the calculation times of the reported optimizations, depending on the amount of constraints defined. The computer used was an IBM compatible PC with 80386 and 80387 processors working with 33 MHz clock frequency.

5. The simulation model input

Two input signals are used in the simulations. The first signal is measured when driving along the smoother test track (track 1), described in ISO 5008 at the prescribed velocity of 12 km/h. This track can be described as some kind of average tractor driving surface. The second input used is measured when driving along the rough test track (track 2), described in the same standard. To get extreme vibration values, the speed was increased from the prescribed 5 to 6 km/h. The vibration load was then so high that the driver almost lost control over the vehicle on the test track.

The tractor used in the measurements was a Volvo-BM T-650, which is a traditionally designed medium size tractor (55 kW) with two-wheel drive, chosen to represent an average agricultural tractor. Premeasurements showed that the angular vibrations around the z axis were very small and could be neglected. The movements were therefore measured in five d.o.f. The RMS values for the input signals used are described in Table 2.

Table 2
The input signals RMS values calculated for a point located in the tractor's symmetry line, 1.50 m above the ground and 0.60 m in front of the tractor's rear axis

Direction	Vibration level RMS (m/s^2 rad/s^2)			
	Track 1		Track 2	
	ISO 2631	Not weighted	ISO 2631	Not weighted
x	0.79	1.11	1.87	2.02
y	1.03	1.68	1.34	1.92
z	1.76	2.16	3.06	4.07
xr	–	1.92	–	2.33
yr	–	1.47	–	1.88
vector sum	2.52	–	4.44	–

6. Optimizations

6.1. Effects of available travel space

The purpose of the first optimizations was to study how the possibilities to work out an effective suspension are affected by the dimensions of the available travel space. The purpose was also to study how the optimum suspension characteristics are affected when the available travel space is changed.

The optimization algorithm was programmed to optimize the vertical and horizontal suspension parameters in a suspension with linear spring and nonlinear damping characteristics. The suspension's vertical and horizontal natural frequencies, together with the nonlinear characteristics of the vertical and horizontal shock absorbers, were optimized (totally eight parameters: f_{nV}, f_{nH}, R_{0V}, R_{0H}, $PK1_{cV}$, $PK1_{cH}$, $PK2_{cV}$, $PK2_{cH}$).

A limit for the cab's deviation from the balanced position, measured at the cab's c.o.g. when driving on track 2 at 6 km/h, was defined as constraint. The vibration load in that test was estimated as the worst possible before the driver risked losing control of the vehicle. The driver would have probably reduced the driving speed before this extreme vibration load appeared. The limit for the maximum deviations varied between 0.075 and 0.125 m in the x, y and z directions. The lower limit for the horizontal natural frequency was defined to 0.5 Hz and to 0.75 Hz for the vertical direction.

The goal for the first study was to find a suspension working well in both normal and extremely rough conditions. The objective function was defined as the mean value of the vibration load when driving on track 1 at 12 km/h and when driving on track 2 at 6 km/h.

The large number of optimized parameters resulted in many local minima being found in the objective function. In several cases it was found that repeated optimizations resulted in other optimal parameter vectors. The solutions thus cannot be considered as representing absolute minima. The differences in the objective function were, however, in no case more than 1.5% for the different

Table 3
Optimum suspension characteristics for the defined constraints

Average space (m)	f_{nV} (Hz)	f_{nH} (Hz)	R_{0V}	R_{0H}	$PK1_{cV}$ (m)	$PK1_{cH}$ (m)	$PK2_{cV}$	$PK2_{cH}$
0.075	0.75	0.50	0.94	1.08	0.052	0.055	1.47	2.50
0.10	0.75	0.50	0.74	0.82	0.089	0.085	2.40	3.40
0.125	0.75	0.50	0.64	0.67	0.141	0.123	2.78	5.85

Table 4
Vibration levels when driving with the optimized suspension on track 1 at 12 km/h

Average space (m)	x (m/s^2)	y (m/s^2)	z (m/s^2)	Vector sum (m/s^2)	xr (rad/s^2)	yr (rad/s^2)
0.075	0.56	0.82	0.81	1.60	0.71	0.83
0.10	0.54	0.80	0.64	1.50	0.67	0.72
0.125	0.54	0.81	0.59	1.49	0.68	0.67

Table 5
Vibration levels when driving with the optimized suspension on track 2 at 6 km/h

Average space (m)	x (m/s^2)	y (m/s^2)	z (m/s^2)	Vector sum (m/s^2)	xr (rad/s^2)	yr (rad/s^2)
0.075	1.43	0.97	2.20	3.27	1.22	1.77
0.10	1.23	0.92	1.95	2.90	1.06	1.65
0.125	1.10	0.88	1.74	2.63	0.87	1.50

solutions. The algorithm showed no tendencies to be instable and seemed to be rather insensitive to varying initial parameter values.

The optimum suspension characteristics with the defined constraints are reported in Table 3. The resulting vibration loads when driving with the optimized suspension on the test tracks are reported in Tables 4 and 5. The vibration levels in the x, y and z directions are weighed values.

The results show, as expected, that the possibilities to work out a suspension with good vibration damping capacity increase when the available travel space is increased.

The optimum natural frequencies in both the vertical and horizontal directions are located at the defined lower limits. The optimum damping in the balanced position (R_{0V} and R_{0H}) decreases when more space is available. At the same time, $PK1_{cV}$ and $PK1_{cH}$ are increasing. The results also show that the optimum suspension becomes more progressive (bigger $PK2_{cV}$ and $PK2_{cH}$) when the available space is increased.

6.2. Suspension optimized for lower average vibration loads

In the optimizations described earlier, the objective function was defined as the average of the vibration load on the smoother and on the rougher track. For normal

Table 6
Optimum suspension characteristics under defined constraints

f_{n}V (Hz)	f_{n}H (Hz)	R_{0V}	R_{0H}	$PK1_{cV}$ (m)	$PK1_{cH}$ (m)	$PK2_{cV}$	$PK2_{cH}$
0.75	0.5	0.59	0.74	0.078	0.081	3.96	1.48

Table 7
Vibration levels when driving with the optimum suspension on the different test tracks

Track	Velocity (km/h)	x (m/s^2)	y (m/s^2)	z (m/s^2)	Vector sum (m/s^2)	xr (rad/s^2)	yr (rad/s^2)
1	12	0.54	0.81	0.59	1.47	0.68	0.68
2	6	1.25	0.92	2.04	2.98	1.27	1.88

driving situations, the smoother test track is probably a better description of the average compared with the very rough test track (no. 2). The vibration loads are only exceptionally as large as when driving on track 2 at 6 km/h.

This study was performed to find out the optimum suspension characteristics when the suspension damping capacity on the smoother track was weighed four times higher than the capacity on the rougher track. Then it is possible to calculate how the optimum characteristics change when more attention is paid to the damping capacity in more normal driving situations.

As previously, the lower limit for the horizontal natural frequency was defined to 0.5 Hz and to 0.75 Hz for the vertical direction. The maximum permitted deviation from the balanced position was constrained to 0.10 m in each direction measured at the cab's c.o.g.

The optimum suspension parameters are shown in Table 6. The vibration levels for the simulated driving over the two test tracks are shown in Table 7.

The suspension optimized in this study has, logically, better damping capacity on the smoother track than the suspension described earlier. The suspension's capacity on the rougher surface is, also logically, poorer than the earlier optimized suspension. The changes are, however, rather small in both cases.

6.3. Optimization of linear suspension

Optimizations have also been performed to study the characteristics of a suspension with linear characteristics meeting the same demands as in the earlier studies. The vibration damping capacity for the two principles can also be compared to get an understanding of the advantages of nonlinear elements.

The optimized parameters in this study were the suspension's vertical and horizontal natural frequency and the vertical and horizontal degree of damping. The lower limit for the horizontal natural frequency was defined to 0.5 Hz and to 0.75 Hz for the vertical direction. The objective function was defined as the mean value

Table 8
The optimum suspension characteristics with the constraints defined in the text

f_{nV} (Hz)	f_{nH} (Hz)	R_V	R_H
0.75	0.5	1.19	1.23

Table 9
Vibration levels when driving with the optimum suspension on the different test tracks

Track	Velocity (km/h)	x (m/s^2)	y (m/s^2)	z (m/s^2)	Vector sum (m/s^2)	xr (rad/s^2)	yr (rad/s^2)
1	12	0.56	0.80	0.81	1.59	0.71	0.82
2	6	1.37	0.96	1.97	3.06	0.96	1.49

of the vibration load when driving on track 1 at 12 km/h and when driving on track 2 at 6 km/h. The available travel space was 0.10 m.

The optimized suspension is described in Table 8. The vibration levels for the simulated driving over the test tracks are shown in Table 9.

The linear suspension optimized with the 0.10 m space constraint is very heavily damped. The natural frequencies in both dimensions still become the same as the values defined as lower limits. The optimized linear suspension gets, as expected, lower vibration damping capacity than the nonlinear suspension optimized for the same amount of available space. The weighted value (vector sum) increases by 6% on the smoother track and by 5.5% on the rougher track.

6.4. Effects of the lower limits for natural frequencies

The results of the optimizations performed earlier showed that the optimal natural frequencies are the same as the ones defined as lower limits. This indicates that the magnitude of the limits influences the possibilities for vibration damping.

An increased horizontal natural frequency gives rise to less static suspension travel when driving on side slopes. An increased vertical natural frequency gives less static travel when the driver's weight varies or when driving with a passenger in the cab.

The purpose of this optimization was to study the changes of optimum suspension characteristics when the lower limits for natural frequencies were increased by 0.25 Hz in each direction.

The vibration levels for the two test tracks were equally weighted and the optimization was performed for 0.10 m available space. The optimized parameters were f_{nV}, f_{nH}, R_{0V}, R_{0H}, $PK1_{cV}$, $PK1_{cH}$, $PK2_{cV}$ and $PK2_{cH}$.

The optimized suspension is described in Table 10. Vibration levels when driving on the two tracks are shown in Table 11.

The lower limits defined for the natural frequencies have major effects on the possibilities to work out an effective suspension. Increasing this value by 0.25 Hz in each direction results in an increasing vibration load in all directions. The increase is especially marked when driving on the very rough surface.

Table 10
The optimum suspension with the constraints defined in the text

f_{nV} (Hz)	f_{nH} (Hz)	R_{0V}	R_{0H}	$PK1_{cV}$ (m)	$PK1_{cH}$ (m)	$PK2_{cV}$	$PK2_{cH}$
1.00	0.75	0.67	0.82	0.110	0.155	1.80	3.94

Table 11
Vibration loads when driving with the optimum suspension on the different test tracks

Track	Velocity (km/h)	x (m/s^2)	y (m/s^2)	z (m/s^2)	Vector sum (m/s^2)	xr (rad/s^2)	yr (rad/s^2)
1	12	0.65	0.95	0.77	1.79	0.83	0.83
2	6	1.59	1.12	2.23	3.52	1.17	1.74

It is interesting that the optimum suspension with the increased limits for natural frequencies, in contrast to the suspensions optimized earlier, did not use the whole available travel space in the x and y directions. When driving with the optimum suspension on track 2 at 6 km/h, the maximum travel was only 0.075 m in the x and 0.080 m in the y direction.

7. Discussion

The geometry for the cab suspension used for the optimizations is relatively symmetric because it offers possibilities to calculate approximate natural frequencies and degrees of damping in the different dimensions. The results obtained are then also useful when designing suspensions based on other types of suspension element principles, which is very important.

The lower limit for the horizontal natural frequency was defined to 0.5 Hz and to 0.75 Hz for the vertical direction. The low natural frequencies make it possible to design suspensions with high vibration damping potential. In order to avoid too large static deviations, it may, however, be necessary to add some kind of slowly reacting self-levelling system.

In the performed studies, the suspension element characteristics have been optimized with respect to various constraints. The method is equally well suited for optimizations of parameters deciding the suspension geometry or combinations of element characteristics and geometry.

The results show that a suspension based on nonlinear damping elements is more effective than a suspension with linear damping characteristics. Another advantage with the nonlinear dampers is the lowered sensitivity for extreme vibration loads. The reason is that the damping at extreme element strokes is very high and, therefore, the protection for overtravel becomes more effective than for a linear suspension.

A measure of the vibration damping potential in general for a passive cab suspension is obtained if the vibration levels in Table 9 using the optimized passive suspension are compared with the levels valid for a rigid cab mounting as presented in Table 2. When driving on track 1 at 12 km/h the weighed vector sum value is decreased from 2.52 m/s^2 to 1.47 m/s^2, which results in the driver being able to stay in the cab 2.9 times longer if the cab suspension is used. When driving on the rougher track at 6 km/h the vector sum decreases from the very high 4.44 m/s^2 to 2.92 m/s^2, which corresponds to a 2.2 times longer stay.

If the vibration levels for particular dimensions are compared, the results show that the cab suspension gives substantial advantages in all dimensions that are examined in the standard. The dimension with highest benefits when driving on the smooth track is the z dimension. Part of this benefit should also be possible to achieve with a conventional seat suspension, and thus benefits in the other dimensions where the seat suspension has insignificant effects, are just as important.

8. Conclusions

The evolution method shows satisfactory characteristics in the optimizations made in the study. The absolute optima were not always found, but the differences in the values of the objective function were small.

The optimization results show, as expected, that the possibilities to work out a suspension with good vibration damping capacity increase when the available travel space is increased. The results also show that the optimum suspension becomes more progressive when the available space is increased.

The suspension optimized with more weight on the performance on smoother surfaces has, logically, better damping capacity on the smoother track than the suspension optimized with equal weight on the two surfaces. The suspension's capacity on the rougher surface is, also logically, poorer than for the other suspension. The changes are, however, rather small in both cases.

The linear suspension optimized with the 0.10 m space constraint is very heavily damped. The optimized linear suspension gets, as expected, lower vibration damping capacity than the nonlinear suspension optimized for the same amount of available space. The weighted value (vector sum) increases by 6% on the smoother track and by 5.5% on the rougher track.

The lower limits defined for the natural frequencies have major effects on the possibilities to work out an effective suspension. Increasing this value by 0.25 Hz in each direction results in an increasing vibration load in all directions.

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