15th Triennial World Congress, Barcelona, Spain

COMFORT IMPROVEMENT OF AGRICULTURAL VEHICLES BY PASSIVE AND SEMI-ACTIVE SUSPENSIONS

K Deprez * **D Moshou** * **H Ramon** * **J De Baerdemaeker** *

* *K.U.Leuven,Laboratory for Agro-Machinery and Processing,*
Kasteelpark Arenberg 30, B-3001 Leuven, Belgium,
Tel.:+32 16 321478, *Fax:*+32 16 321994,
email: koen.deprez@agr.kuleuven.ac.be

Abstract: Growing awareness that uncomfortable agricultural vehicles endanger the health of the operators and stricter norms concerning this, make the design of an effective cabin suspension inevitable on agricultural vehicles. The comfort problem originates from the vibrations transmitted to the driver caused by the unevenness of the road or soil profile.
This paper investigates the effect a passive and semi-active cabin suspension has on the comfort of the drivers. By optimizing the parameters of the passive suspension and those of the semi-active control laws, satisfactory improvement of the comfort can be achieved.

Keywords: vehicle suspension, optimization, passive, semi-active

1. INTRODUCTION

Today the "market value" of cars does not only depend on performance and price. Safety, comfort and environment friendlyness tend to be of equal importance and are for some manufactures the main promotion topics. Driving force behind it are norms and directives concerning these issues but also the raising awareness of the consumer. Off-road vehicles do not escape this trend in the market. This paper deals with the comfort aspect of agricultural vehicles narrowed down to the whole-body vibration levels the operators are exposed to when handling these machines.

Extensive research in the past pointed out 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) (Griffin 1990, Hulshof and van Zanten 1987). Origin of this discomfort are vibrations transmitted to the driver caused by the unevenness of the road or soil profile, moving elements within the machine or implements. High low-frequency levels between 0.5 and 10 Hz are transmitted to the seat during field operations and the cyclic motions like those caused by vehicle's tires hitting the road are situated in the frequency range of 2 to 20 Hz (Hostens and Ramon 2000, Clijmans *et al.* 1998). Especially the backbone is sensitive in this frequency range for severe physical damage (Pope and Hansson 1992, Boshuizen *et al.* 1992). The damage is caused through "cumulative trauma" and therefore difficult to assess.

Unexpectedly the impact on economy is huge. Not only leads discomfort during work to performance problems (Fairlay 1995) 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 a year for the United States is estimated at $ 90 billion. So to the government, the operators and the manufactures it is of common interest to deal with this problem.

The European Parliament already acted. The general standards like The ISO 2631 (ISO 1985) and the BS 6841 (BS 1987), used for all types of vehicles in which whole-body vibrations exposure occur, were improved in a new Machinery Directive 98/37/EC. Annex I par. 1.5.9 and 3.6.3 which will become 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 the whole-body vibrations.

To meet this directive manufactures will have to fine-tune their suspensions in seats, cabins and axles. This paper reports what results can be achieved with a passive and semi-active hydropneumatic cabin suspension. The parameters of the passive suspension and those in the semi-active control laws are optimized using a global optimization technique.

2. SET-UP AND TOOLS

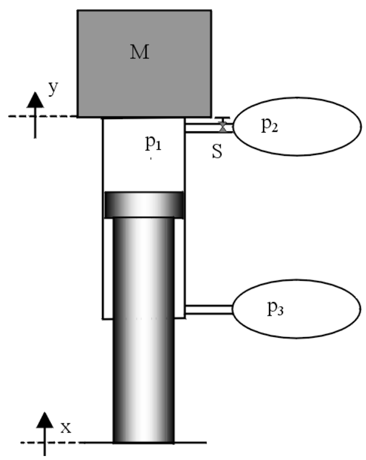

Fig. 1. Scheme of the oil system

Figure 1 shows the hardware used by this paper. It consists of a hydraulic cylinder, two nitrogen bulbs and a current driven hydraulic valve. The choice for this sort of set-up follows out of two considerations. First, to be able to attenuate vibrations it is necessary to develop a system with a natural frequency below the frequency band one wants to attenuate. This feature is provided with the two nitrogen bulbs. Second, to prevent amplification of the natural frequency of the suspension, sufficient means of damping are necessary. To achieve this, oil damping is used which gives significant better results than gas damping (Deprez *et al.* 2001).

This hardware was modeled in SIMULINK (MathWorks 1999*b*) making it possible to link the model to the optimization procedures available in Matlab (MathWorks 1999*a*). Two laws used in this model make it nonlineair. First, the nitrogen bulbs compress and expand according to the adiabatic law:

$$pV^{\kappa} = constant, \tag{1}$$

in which p stands for the pressure, V for the volume and κ for the ratio of specific heats. The second nonlinear equation describes the mass flow of oil through the valve:

$$\dot{V} = S\sqrt{\Delta p}, \tag{2}$$

in which S stands for the surface of the opening of the valve, Δp the difference in pressure over the valve and $\dot{V}$ for the oil mass flow. The modeling leads to a 'base motion' model of a quarter-cabin.

Optimization of this nonlinear model implies that true input signals have to be used. Those were provided during field measurements. Under road and field conditions the vibrations entering the cabin of a combine harvester were measured. The measurements used here were those at 4 km/h on the field, at 11 km/h on an unpaved road and 28 km/h on a paved road.

The most effective way of evaluating a suspension is to measure it's comfort improvement. This is difficult since comfort is strongly subject related. Nevertheless the standards like ISO 2631 and BS 6841 try to grasp this feeling by using objective comfort parameters. This paper uses the effective Root Means Square (effRMS) and the Vibration Dose Value (VDV), because these parameters are used by the standards. These values are calculated after the acceleration data is filtered. This is to eliminate those frequencies that have no influence on the comfort and health of the drivers. The filter of figure 2 is used which clearly emphasis the frequencies between 2 and 10 Hz (Griffin 1990).

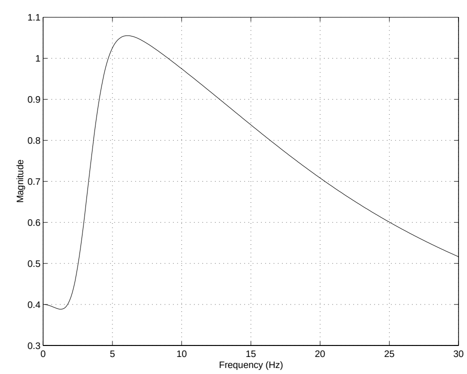

Fig. 2. Magnitude plot of the BS6841 frequency weighing filter used when considering the effect of vertical DOF vibrations on comfort and health

The general formula for VDV is:

$$VDV = \left[\frac{T_s}{N}\sum_{n=1}^{n=N} a^4\right]^{\frac{1}{4}}, \tag{3}$$

with T_s the measured time, 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:

$$effRMS = \left[\frac{1}{N}\sum_{n=1}^{n=N} a^2\right]^{\frac{1}{2}}. \tag{4}$$

The RMS value is time independent and gives an idea of the general level of vibrations.

3. OPTIMIZATION OF THE PASSIVE SUSPENSION

The model developed in SIMULINK has 7 parameters: internal pressure, volume of the two nitrogen bulbs, opening of the valve and dimensions (length, diameter of shaft and cylinder) of the hydraulic cylinder. The optimization of the suspension is based on the Frequency Response Function (FRF) of the system. An optimal FRF was put forward and the squared difference between the optimal and the real FRF, frequency line by frequency line, is minimized. Since the suspension is nonlinear, this step has to be taken carefully. Different signals will result in different FRFs but the reason why this approach was preferred above reducing the total amount of accelerations in time domain, was to tackle the problem of sea sickness. A slow fluctuating movement results in low accelerations and good effRMS and VDV values, but is inadmissible. This can be prevented in frequency domain by emphasizing the fact that the amplification at the natural frequency of the system should be as little as possible. This can be done by weighing the squared error.

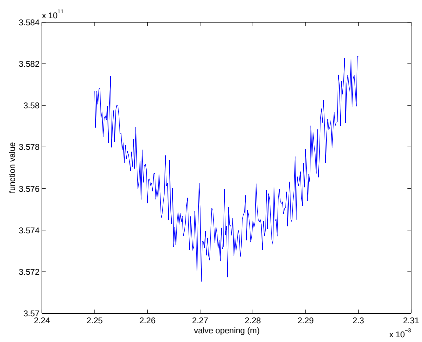

Fig. 3. Objective function near global minimum when changing one parameter

Figure 3 gives and idea how the squared error evolves when one parameter is changed. The objective function is very "noisy". The normal Matlab techniques have problems optimizing such a function and get stuck in the various local minima. The technique used to avoid this problem is called DIRECT (Jones *et al.* 1993). It's a global optimization technique categorized under 'branch and bound'. The method was implemented in the Matlab routine glbSolve (Bjőrkman and Holmstrőm 1999) and is fairly easy to use. Which reduction of effRMS and VDV can be achieved in this way is shown and discussed in section 5.

4. SEMI-ACTIVE SUSPENSION

It is obvious that an active suspension will give the best results for such a suspension but no manufacturer will be inclined to make the initial and operational costs associated with it, especially not if enough benefits can be get at less expense. Semi-active claims to be a good and economically sound alternative (Karnopp 1990). This is tested here by using two semi-active control laws. In the first law the semi-active part of the damping force is proportional to the relative velocity between input and output and proportional to the absolute output velocity. The law, semi-active(1), is given by:

$$F_D = \begin{cases} -\beta(\dot{y}-\dot{x}) - \beta_{SA}|\dot{y}|(\dot{y}-\dot{x}) & \\ & \text{for } \dot{y}(\dot{y}-\dot{x}) > 0 \\ -\beta(\dot{y}-\dot{x}) & \\ & \text{for } \dot{y}(\dot{y}-\dot{x}) < 0 \end{cases} \quad (5)$$

in which F_D is the damping force, $\dot{x}$ the input velocity, $\dot{y}$ the output velocity entering the cabin, β the damping constant of the passive part of the damping force and β_{SA} the damping constant of the semi-active damping force.

The second law, semi-active(2), increases the importance of the absolute velocity by squaring it resulting in:

$$F_D = \begin{cases} -\beta(\dot{y}-\dot{x}) - \beta_{SA}\dot{y}^2(\dot{y}-\dot{x}) & \\ & \text{for } \dot{y}(\dot{y}-\dot{x}) > 0 \\ -\beta(\dot{y}-\dot{x}) & \\ & \text{for } \dot{y}(\dot{y}-\dot{x}) < 0 \end{cases} \quad (6)$$

By switching between a high and a low damping force, those two semi-active laws try to prevent that the damping force enlarges the vibrations (Ferraresi *et al.* 1997).

To choose the β_{SA} parameters in the semi-active laws in an appropriated way, the global optimization technique of section 3 is used again. For the two laws, β_{SA} is optimized with respect to the objective comfort parameters effRMS and VDV. The results obtained are shown and discussed in section 5.

5. COMFORT IMPROVEMENT

To evaluate the comfort improvement of the three suspension systems, passive and semi-active (1) and (2), the comfort parameters effRMS and VDV are calculated when the system is excited with the input signals measured during the field experiments. Those values are compared with each other and with the values if there is no suspension at all. Table 1 shows the results for three road types and for the three suspension systems.

Looking at the results of the passive suspension one sees clearly that a reduction by 60% in effRMS and by almost 80% in VDV is possible on the paved road. The reductions are even larger on the field and unpaved road. This is normal since low frequencies around 2 Hz excite the cabin much more on the road at high speed than in the field and the passive suspension reduces high frequencies more than frequencies near to its resonance frequency. Since the passive suspension

input profile	suspension	VDV $[m/s^{1.75}]$	effRMS $[m/s^2]$
field($4km/h$)	non	4.135	1.009
	passive	0.237	0.058
	semi-active(1)	0.121	0.025
	semi-active(2)	0.128	0.026
unpaved road ($11km/h$)	non	3.051	0.686
	passive	0.387	0.094
	semi-active(1)	0.279	0.054
	semi-active(2)	0.381	0.059
paved road ($28km/h$)	non	6.353	1.502
	passive	2.067	0.352
	semi-active(1)	0.340	0.063
	semi-active(2)	0.519	0.071

Table 1. Calculated comfort parameters for different road profiles using different suspension systems

is the underlying system when using the semi-active control laws, this phenomenon can also be seen when comparing those results.

The first semi-active control law results in a better suspension than the second. So over emphasizing the absolute velocity is not good. It is obvious that other semi-active control laws, using absolute accelerations, can be tried out but it is already clear that semi-active suspensions have an added value in improving the comfort, especially in the reduction of vibrations under road conditions.

Since it is difficult to comprehend the comfort improvement on the basis of the values of table 1, figure 4 visualizes the effect by plotting the input acceleration signal together with the signals entering the cabin when using a passive or semi-active suspension system.

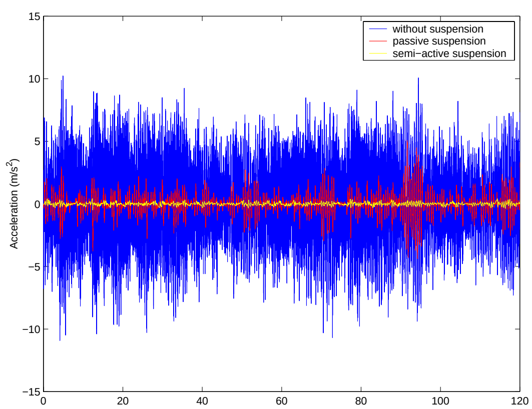

Fig. 4. Acceleration data with and without suspensions

More important to manufactures of agricultural machinery is to see if the new directive can be met. Machinery Directive 98/37/EC. states a maximum value for effRMS of $1.15m/s^2$, but in future this value will be lowered. Without suspension only driving on the paved road gives effRMS values above this level but the field is not to far off. For the VDV the directive states that the maximum dose is $21m/s^{1.75}$ in a period of 8 hours. This is not a severe value since a VDV of $15m/s^{1.75}$ is considered as health threatening (Griffin 1990). Equation 7 gives the relation between the VDV_{T_s} value measured in a time T_s and the dose VDV_T one is exposed to in a time T.

$$\frac{(VDV_{T_s})^4}{(VDV_T)^4} = \frac{T_s}{T} \tag{7}$$

When following the directive and calculating the resulting VDV dose for eight hours, only driving on a paved road would not be allowed without suspension. When using on the other hand a VDV of $15m/s^{1.75}$ a one hour drive on the road or working 6 hours in the field would mean threatening ones health. Whatever VDV norm is used an optimized passive suspension would already provide the necessary vibration reduction.

6. CONCLUSION

To meet new directives concerning comfort on agricultural vehicles and to keep a competitive position, manufactures have to invest more time in the design of good suspension systems. The results shown in this paper prove that an improvement of comfort resulting in a reduction of comfort values by more than 60% are possible using an optimized passive suspension system. With a small additional investment it is possible to reduce the levels a great deal more (up to 90%).

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the Belgian Ministry of Agriculture for the financial support through project S5960. This study has been made possible with the cooperation of New Holland Belgium.

7. REFERENCES

Bjőrkman, M. and K. Holmstrőm (1999). Global optimization using the direct algorithm in matlab. *Journal of Advanced Modeling and Optimization* **1**(2), 17–27.

Boshuizen, H.C., P.M. Bongers and C.T.J. Hulshof (1992). Self-reported back pain in work lift truck and freight container tractor drivers exposed to whole-body vibration. *Spine* **17**, 1048–1059.

BS (1987). *Measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock BS 6841*. British Standards Institution, London.

Clijmans, L., H. Ramon and J. De Baerdemaeker (1998). Structural modification effects on the dynamic behavior of an agircultural tractor. *Transactions of the ASAE* **41**(1), 5–10.

Deprez, K., D. Moshou, I. Hostens, J. Anthonis and H. Ramon (2001). Hardware design and optimization of a 1 dof suspension. *Proceedings of the Fourth International Symposium on Mathematical Modelling and Simulation in Agricultural and Bio-Industries, 12-14 June, Haifa, Israel*.

Fairlay, T.E. (1995). Predicting the discomfort caused by tractor vibration. *Journal of Ergonomics* **38**(10), 2091–2106.

Ferraresi, C., G. Quaglia and M. Sorli (1997). Force control laws for semi-active vehicular suspensions. *European journal of mechanical engineering* **42**(3), 145–151.

Griffin, M.J. (1990). *Handbook of Human Vibrations*. Academic Press. Harcourt Brace & Company, Publishers, London.

Hostens, I. and H. Ramon (2000). Descriptive analysis of agricultural machinery cabin vibrations and their effect on the human body. *Proceedings of ISMA25, International Conference on Noise and Vibration Engineering, 13-15 September, Leuven, Belgium*.

Hulshof, C. and B.V. van Zanten (1987). Whole-body vibration and low back pain. *International Archives of Occupational and Environmental Health* **59**, 205–220.

ISO (1985). *Evaluation of human exposure to whole-body vibrations ISO 2631/1*. International Organization for Standardization, Geneva.

Jones, D.R., C.D. Perttunen and B.E. Stuckman (1993). Lipschitzian optimization without the lipschitz constant. *Journal of Optimization Theory and Applications* **79**(1), 157–181.

Karnopp, D. (1990). Design principles for vibration control systems using semi-active dampers. *Transactions of the ASME* **112**, 448–455.

MathWorks (1999*a*). *Matlab 5.3.1*. Copyright © by The MathWorks Inc.

MathWorks (1999*b*). *Simulink toolbox for Matlab*. Copyright © by The MathWorks Inc.

Pope, M.H. and T.H. Hansson (1992). Vibration of the spine and low back pain. *Clinical Orthopaedics and Related Research* **279**, 49–59.