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An improved design of air suspension for seats of mobile agricultural machines

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Abstract

Different suspension systems of the type 'full travel' used in agricultural machinery seats over the last 20 years are evaluated. The effects of the type of spring and damper are discussed through theoretical analysis and experimental tests. A new improved passive suspension system is proposed as most promising and includes an air spring with additional air volume and variable air damping. A comparison is made with existing passive systems. The theoretical analysis and laboratory tests show that the new proposed system provides the best vibration attenuation, when considering passive suspension systems, for agricultural machinery or other vehicles with similar vibration inputs.

1. Introduction

Increased vehicle speed and capacity, induced by higher work costs, create a lot of vibration problems, reducing the vehicle lifetime, working precision and drivers comfort [1]. The link between occupations involving driving mobile machinery and the occurrence of back pain has been found in many studies [2–4]. Some of the risk factors, defined as having the greatest influence on back pain, are long-term sitting, (sustained) posture, task handling and whole-body vibration. Many epidemiological studies mention the influence of vibrations on health and comfort. However, only a few had been able to define a dose–effect relationship [5]. The vibration that people are exposed to while sitting and driving is rarely linked to the cause of discomfort or pain since vibrations weaken the spine through a 'cumulative trauma' which is very difficult to assess [6,7]. Although it is hard to define exact causes for each of these problems, vibration attenuating

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seats and correct ergonomic layout of vehicle interior may reduce the risk of recurrence [8]. Suspension systems should be developed in such way that they minimize the transmission of harmful vibrations and shocks as much as possible.

A number of studies show that the musculo-skeletal system possesses a somewhat resonant response, with motion entering at the seat/buttocks interface being magnified at certain frequencies, and at certain locations within the body [9,10]. Specifically, they show that the largest motion occurs, under sinusoidal vibration at least, in the upper lumbar and lower thoracic region of the spinal column. This suggests that this is the part where damage caused by vibration and shock is most likely to be found. The other specific finding is that the frequency of maximum response (in the lumbar/thoracic region) to vertical motion is at around 5 Hz, varying between different subjects, generally in the range 4–6 Hz.

The low-frequency range (2–8 Hz) is crucial for good driver comfort and health, and work efficiency. The suspension system of suspension seats is most likely to hit the end-stop buffers when excited at its resonance frequency, and so performance at this frequency will influence its end-stop impact performance. Burdoff and Swuste [11] measured the isolation of 11 suspension seats in the laboratory, using standardized vehicle vibration spectra, and also in vehicles driving over typical surfaces. The results showed that 19 of the 24 transmissibility measurements in vehicles were greater than the corresponding transmissibilities measured in the laboratory. They concluded that laboratory measurements of the dynamic response of suspension seats did not provide an adequate basis for predicting their performance in the field. A reason for the difference may be the non-linearities caused by friction (with low-magnitude vibration) and end-stop impacts (with high-magnitude vibration). In a field survey of the whole-body vibration experienced by tractor drivers, Stiles et al. [12] found that 45% of seats increased the acceleration levels experienced by the driver. Much of the increase was said to be due to end-stop impacts. It has been suggested that for some drivers the end-stop impacts can be so severe that they would rather weld the suspension system to avoid end-stop impacts. Seidel et al. [13] already mentioned the important influence of shocks on the development of fatigue within the human back region.

But not only the ideal performance of the spring–damper unit of the suspension system is of interest. Friction problems play a larger role for certain excitation magnitudes. Wu and Griffin [14] defined five stages of suspension performance depending on excitation magnitude. A seat exposed to increasing magnitudes of vibration is initially friction locked (stage 1). As the vibration magnitude increases, the seat begins to 'break away' producing jolting on the seat surface and causing the *SEAT* (seat effective amplitude transmissibility) value to increase (stage 2). As friction becomes less dominant, the *SEAT* value decreases and the seat acts to reduce the vibration on the seat surface. Further increases in magnitude leads to stages 4 and 5, where end-stop impacts occur as the available suspension travel is exceeded. The *SEAT* value increases as the end-stop impacts cause severe vibrations at the seat surface.

In this document, an evaluation of old and new seat suspension systems is given. It starts approximately 20 years ago, at the moment that the scissor system as a supporting structure of the suspension system took strong hold. Only the conventional type of seat is taken into consideration ('full travel' type of suspension with free travel of 100 mm). This mechanism proved to be very safe and was low in production cost. The same supporting structure is used in all seats under investigation. As agricultural machinery is within the vehicle group with the highest vibration exposure and this certainly in the low-frequency range, the performance for the vibration inputs of a combine is examined. The frequency range of interest is 0.7–15 Hz. The reasoning behind the improvements is explained through the theoretical principles of the used systems. Illustrations are provided based on measurements on different vehicles and suspension systems, performed by the Laboratory of Agro-Machinery and -Processing. An optimum passive seat suspension system is proposed.

2. Present systems

2.1. Mechanical suspension with constant damping

2.1.1. Theoretical model

The working principle of the suspension system approximates a base excitation on a one-degree-of-freedom spring–damper unit, resulting in damped harmonic vibrations, and given by Eq. (1) (see also Fig. 1)

$$\frac{Y(s)}{X(s)} = \frac{cs + k}{ms^2 + cs + k} \tag{1}$$

with c being the damping, k the spring stiffness and m the mass.

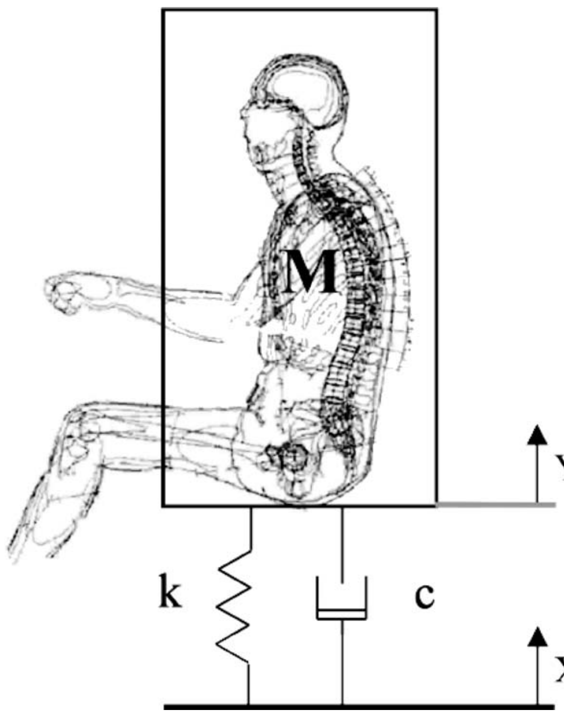

Fig. 1. Graphical representation of one-degree-of-freedom spring–damper unit in a seat suspension with damping c, spring stiffness k and mass M.

The eigenfrequency ω_n of the system is given by

$$\omega_n = \sqrt{\frac{k}{m}}. \tag{2}$$

The use of a hydraulic damper will lower the peak at the eigenfrequency. The more damping c the more this peak will be reduced. However, an increase in damping is correlated with a higher transmissibility λ of higher frequencies to the driver (Fig. 2). For different values of the damping c all curves will go through the same point $\lambda = 1$ when

$$\frac{\omega}{\omega_n} = \sqrt{2}. \tag{3}$$

2.1.2. Seat performance parameters

The seat dynamic behaviour is influenced by a number of parameters. The grade of influence, for a certain set of parameters is determined by the excitation magnitude. For the spring component part, not only the stiffness will play a role but also the linkage friction. For the damper component part, there exists non-frictional damping which provides the required damping under ideal circumstances, and there exists Coulomb damping. The friction through Coulomb damping is generated by the oil seals and the rod against the wall in the cylinder. The non-frictional damping is generated by the oil flow through the damping holes (viscous damping). At high frequencies, the displacements are so small and the direction of the force changes so fast that the oil has no time to pass through the cylinder holes which provide the damping effect. Because the oil is not compressible, the damper will react as a rigid structure through which vibrations can be

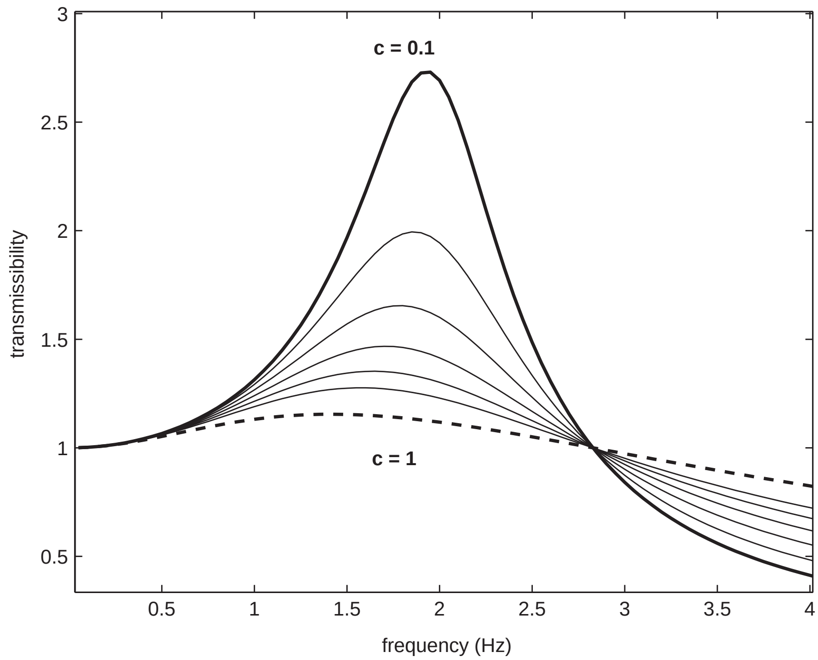

Fig. 2. Transmissibility λ as a function of frequency ω given for different values of the damping c.

transmitted to the driver. Although the damping c of an oil damper is called constant, at higher frequencies it tends to infinity. Still Rakheja et al. [15] used a linearized suspension seat model, and found that Coulomb friction was more influential than viscous damping and that the coefficients of both should be reduced to improve the ride performance. In a model from Gunston et al. [16] all the above elements were introduced. His findings were that reducing the non-frictional suspension damping has negligible effect on seat performance at low magnitudes, a small effect at moderate magnitudes (50% change resulting in a $<5\%$ improvement), and a large detrimental effect at high magnitudes (up to 150% worse) due to increased occurrence and severity of end-stop impacts. Reducing the suspension friction was beneficial at low to moderate magnitudes (50% reduction giving an improvement of up to 40%), but detrimental at high magnitudes as the reduced overall damping resulted in more end-stop impacts. According to Gunston et al., ideally the friction should be reduced as much as practically possible, and the non-frictional damping should be used to control the occurrence of end-stop impacts.

2.1.3. Restrictions of the mechanical system

The mechanical suspension system works with an integrated package of springs. These springs are usually made of steel. By means of adjusting the initial tension (pre-load), the spring package can be adjusted to the current driver's weight to place the seat at mid-position. Oil pressure dampers (hydraulic telescopic type is most common) take the task of damping in this mechanical system.

The ideal solution would be to take the resonance frequency of the spring as low as possible. The lower the resonance frequency, the narrower the base of the resulting peak becomes and the faster the vibrations will be attenuated at higher frequencies. This can be derived from Eq. (1) and seen in Fig. 3.

Unfortunately there is a restriction. From Eq. (4), it can be concluded that there is an inverse quadratic relation between the eigenfrequency and the static deflection Δ_{static} of the suspension:

$$\Delta_{\text{static}} = \frac{g}{\omega_n^2}. \tag{4}$$

This implies that for an eigenfrequency of 2 Hz, the static deflection (because of the weight on top) is 6 cm but for an eigenfrequency of 1 Hz this static deflection is already 25 cm. To construct a seat with an eigenfrequency of 1 Hz is therefore practically not feasible, as a spring length of at least 0.5 m would be needed (static deflection + diameter spring rod multiplied by the number of coils + deflection through vibration input). Furthermore, for the same amount of power input, the seat movement at lower frequencies is much higher. To keep the deflection of the seat within the end stroke range, higher damping will be required. As shown in Fig. 2 and Eq. (1), this higher damping will cancel out the advantages of a system with a low eigenfrequency.

Another reason for retaining higher stiffness is the friction, which causes difficulties in setting the ride height to compensate for the driver's weight. Mechanical suspension systems were built with an eigenfrequency around 1.5 Hz showing a good trade-off between stiffness and damping (low eigenfrequency with high damping, high eigenfrequency with low damping and accurate pre-load for height adjustment).

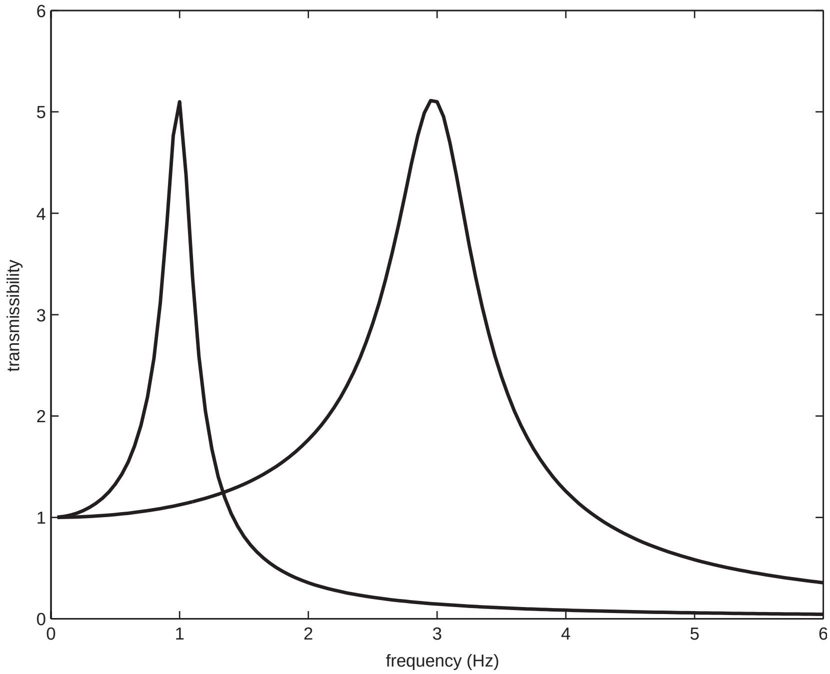

Fig. 3. Transmissibility λ as a function of frequency ω given for different values of the stiffness k and equal damping c.

2.2. Air suspension with constant damping

2.2.1. Theoretical model

The behaviour of this suspension system can be easily described by changing the force of the mechanical spring given by $F = k(x - y)$, with

$$F_{\text{airspring}} = \frac{P_0 V_0^{\kappa} A}{V_{(x,y)}^{\kappa}} - P_A A - mg \tag{5}$$

with P the air pressure, V the corresponding air volume function of x and y, κ the value for the ratio of specific heat, P_0 the initial pressure, V_0 the initial volume, A the effective area of the air spring, P_A the ambient pressure and g the earth gravity.

This equation is based on the laws of adiabatic processes for ideal gases. It is proven to be a suitable approach [17]. The damping is applied in the same configuration as for the mechanical suspension system.

2.2.2. Comparison between mechanical spring and air spring

In time, the mechanical spring became replaced by the air spring. Although the air spring already existed for more than 30 years, it was not for 10 years that the improved performance was recognized and that it was implemented on large scale on agricultural machinery. This suspension system proved to give better comfort to the drivers. A case study from Hostens et al. [18] indicates the better behaviour of air suspension systems through interpretation of power spectra and *SEAT* values.

To clarify the mechanism behind the better performance of air spring systems the models are compared. Through field experiments, the eigenfrequency was found for both seats in the 1.8–2 Hz range with a maximum transmissibility of 2.5. For both models, the appropriate damping and spring stiffness (for the suspension system with mechanical spring) values are calculated to obtain the same eigenfrequency and corresponding amplitude. The spring parameter (for the mechanical spring) and damper parameters (for the hydraulic damper) used to fit the models are: $k = 34\,000$ and $c = 3200$ for the mechanical seat and $c = 12$ for the air spring seat. In Fig. 4, the FRF is given for both systems. The input signal is a multi-sine in the frequency range 0.7–15 Hz. It provides high acceleration levels for the whole frequency band and a higher shock content than a pure sinusoidal signal as a swept-sine, thus representing better real-field signals. The influence of elements of friction as the damper friction and the spring linkage friction and the human dynamics is not taken into account, as it concerns only a comparison study.

These simulations prove that the non-linear characteristics of the air spring provide a certain inherent amount of damping, which only manifests itself in the low frequencies. The need for less added constant damping through a hydraulic damper makes better attenuation of vibrations at higher frequencies (> 3 Hz) possible. The damper friction losses will play a smaller role, therefore providing better attenuation at lower vibration magnitudes.

A second benefit of the air spring system can be illustrated by Fig. 5, where a load/deflection curve for an air spring is given. It shows how close to the recommended design height (middle line) the force (kN) stays the same, which results in a constant stiffness. However, when largely pressed or stretched and thus approaching the end stroke of the seat, the generated force/unit of length

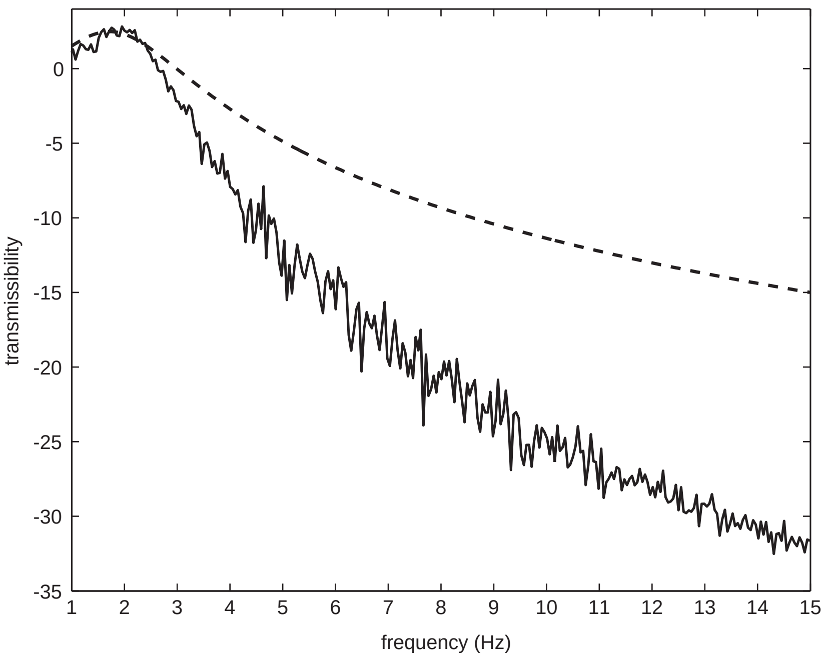

Fig. 4. Frequency response function for mechanical (dashed line) and air spring model (solid line) with a multi-sine (0.7–15 Hz) as input signal.

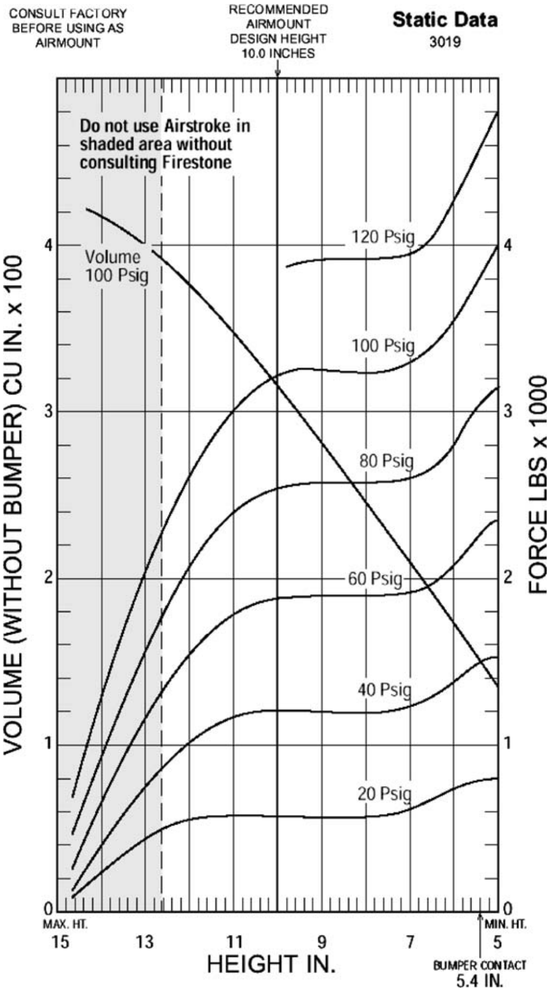

Fig. 5. Static data chart, also referred to as the load/deflection (L/D) curve for an air spring.

increases, resulting in a larger stiffness of the air spring. Due to this property, the seat is better armed against end-stop.

An extra benefit is that not only can pre-load be adjusted for setting the ride height (mid-position) for the driver's weight, also this adjustment can be automatic, eliminating the possibility of operator error or laziness. This is very important as can be seen in the study from Wu and Griffin [14]. They performed tests using a seat with 40 mm stroke and adjusting the mid-position to three different positions (5 mm below, 5 mm above, and at the mid-position). When the magnitude on the vibration simulator was low, the whole seat VDV (vibration dose value) ratios were the same. With increasing input magnitude, both the higher and the lower positions result in end-stop impacts occurring at lower input magnitudes. For example, when the input magnitude was 2 ms^{-2}, and the setting was 5 mm below the mid-position, the whole seat VDV ratio is 28% greater than when set at the mid-position.

2.3. Restrictions of the air suspension with constant damping for agricultural machinery

The vibration energy as shown in the spectrum for a specific vehicle will determine in the end the performance of the seat suspension. For vehicles like buses and trucks, the soft suspension system on the wheel basis or cabin provides one large peak around 1 Hz after which a fast decline is visible in the spectrum (Fig. 6). This indicates that with proper damping good vibration attenuation is provided and end-stop will not occur. The importance of appropriate damping can be demonstrated by driving a bus with high and low damping over the same comfort track in 'Lommel proving grounds' (Fig. 7). The accelerations measured using the seat interface for transducers indicating body accelerations received (SIT-BAR) show values up to 2 g. This indicates the occurrence of end-stop impacts, introducing energy in the entire spectrum including the sensible frequency range of the spine (4–8 Hz) (Fig. 8).

The differences in vibration input between buses and agricultural vehicles are significant in the low-frequency range (0.7–3 Hz). In Fig. 6, one can observe the small 1 Hz peak for the bus and the broader and higher 1.5–2.5 peak for a combination. Many agricultural mobile machines will show similar excitation when driving on specific roads, it can be observed that great differences in excitation exist, when comparing the *SEAT* index for a combine driving on different surfaces with different speed and different tire pressure. The vibrations encountered when driving in field conditions are no problem for the seat suspension. On the road, where the bus shows *SEAT* values always lower than 90 with minimum values around 50, the combine exceeds 100 regularly (Fig. 9). The eigenfrequency for the seat suspension lies in the same range as the excitations at the cabin floor when driving on road surfaces. And when driving on a particularly heavy road, such as

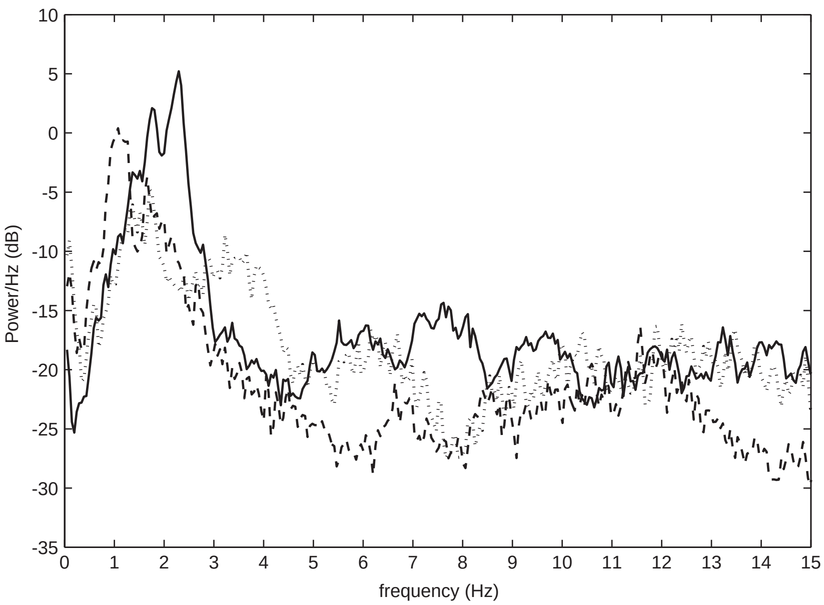

Fig. 6. Power spectra of accelerations measured, using a SIT-BAR, on an air-suspended seat for combine (solid line), bus (dashed line) and small van (dotted line) when driving on a comfort track.

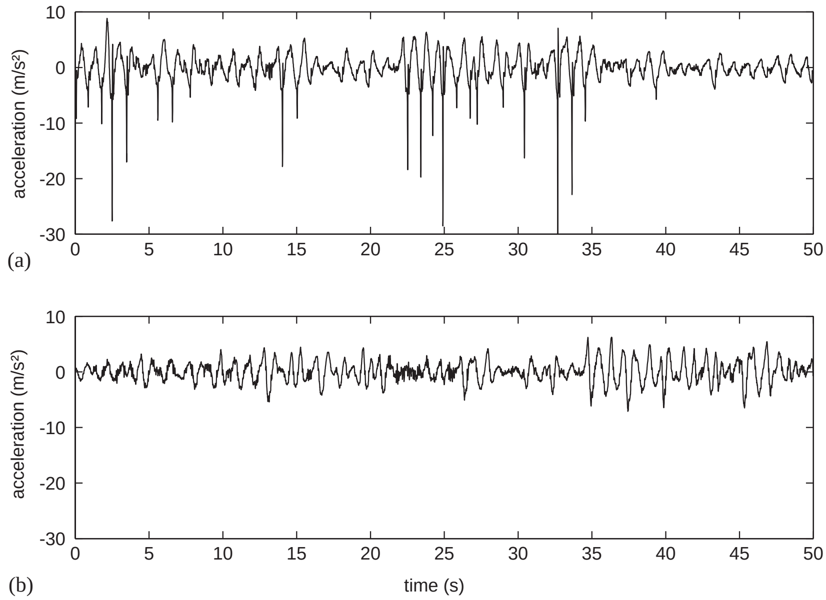

Fig. 7. Accelerations measured, using a SIT-BAR, on a bus seat with low damping (a) and high damping (b) when driving on a comfort track.

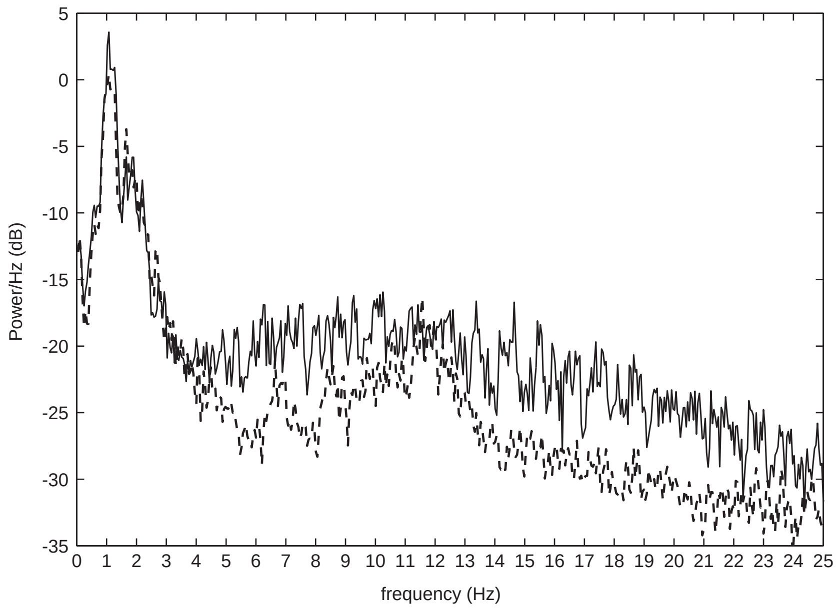

Fig. 8. Power spectra of accelerations measured, using a SIT-BAR, on a bus seat with low (solid line) and high dampings (dashed line) when driving on a comfort track.

the comfort track, the occurrence of end-stop impacts is inevitable (Fig. 10). In most cases, the driver is not warned in advance of such an event and his body will not be prepared. It is not very clear yet what impact these shocks have on the general health of drivers.

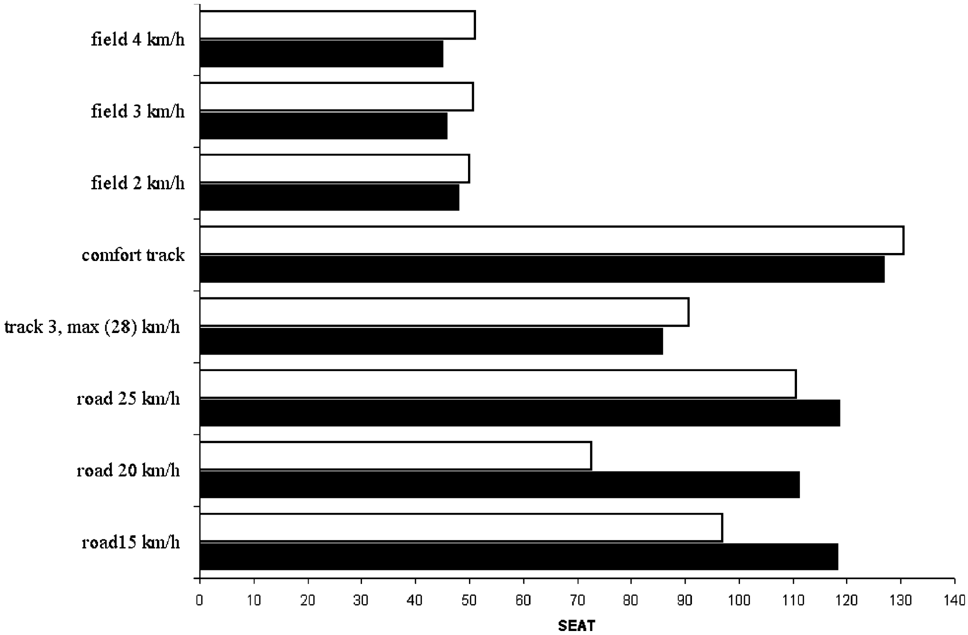

Fig. 9. *SEAT* values given for a combine driving with certain speed over a specific surface with high (black) and low tire pressure (white).

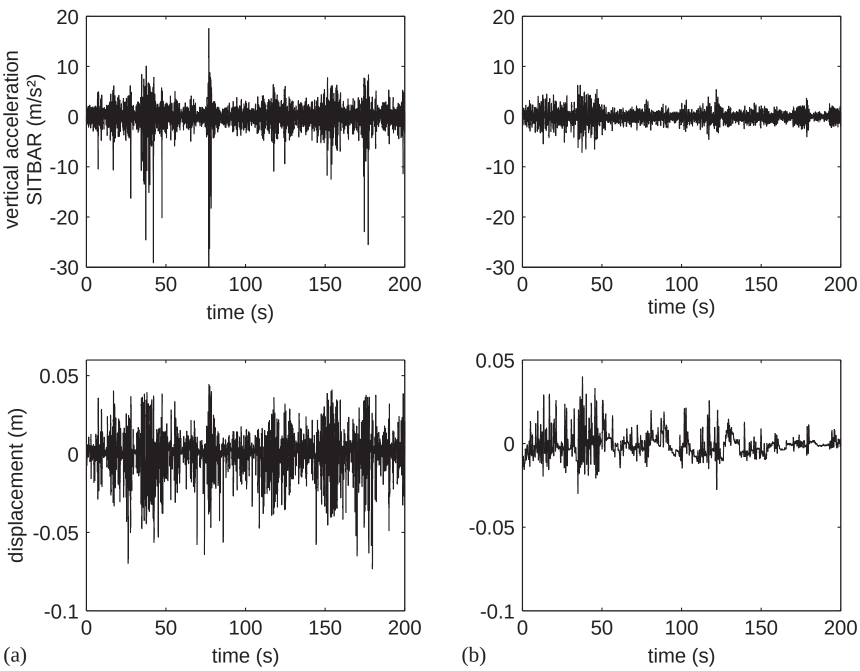

Fig. 10. SIT-BAR accelerations and displacements between seat and floor for combine (a) and bus (b) when driving over a comfort track.

Though under 2 Hz the human body behaves rigidly, when exposed to high magnitudes, great stabilization of the back muscles is needed for sustained posture. The great differences in displacement between seat and cabin floor produce in addition a constant posture change. The constant posture change causes a large labour profile for a driver of an agricultural vehicle. The result is muscle fatigue (from back shoulder, neck and leg muscles), a weakened back and eventually the occurrence of low back pain (LBP). Because of the cumulative trauma which is believed to be the process behind the development of LBP, it is very difficult to assess the influence of the seat travel. In order to provide a solution concerning the large displacements and the possible occurrence of end-stop impacts and at the same time preserve the comfort provision in high frequencies as in the standard air spring suspension system a number of restrictions can be formulated:

- the resonance frequency needs to be lower than the lowest frequency with high acceleration excitation;
- the damping factor has to be high enough to provide limited low-frequency seat travel and at all times a good prevention of end-stop;
- the damping factor should be chosen to guarantee optimal vibration attenuation above 3 Hz.

For the development of an air spring with a lower eigenfrequency, the limitations are determined by Eq. (6) as a linear approximation of the reality [19]:

$$k = \frac{\mathrm{d}F}{\mathrm{d}l} = \frac{mg}{h} = \frac{-p\kappa A^2}{V} \tag{6}$$

with h being the height, A the surface and V the volume of the air spring.

To lower the spring characteristic k, the height of the air spring has to increase. This is the limiting factor in this set-up. The place normally reserved in the seat for the air spring is limited and it must be constructed such that the seat can be folded completely to the ground for transport (the packaging situation). Another limitation is the hydraulic damper. Lowering the stiffness means increasing the damping for optimal performance. This will introduce difficulties in setting the ride height for the driver's weight, as the influence of friction increases.

3. New systems

3.1. Air suspension with variable air damping

By using an air damper, the problems encountered with the hydraulic damping can be avoided. As with the oil damper at high frequencies, the air has no time to pass through the damping holes because of the fast change of displacements and the low level of displacements. However, through the compressibility of gas, the air trapped on both sides of the cylinder will behave as springs. This implies that the non-frictional damping will not add to the friction losses at high frequencies and low magnitudes. The provision of higher damping in the air damper will result in a reduced peak at the eigenfrequency without decreasing the attenuation at higher frequencies. Due to this fact, the air damping is also called variable air damping in comparison to the constant hydraulic damping. This air damping can be built into the air spring resulting in one compact unit (integrated damper).

An extra benefit of the air damping (integrated) includes the non-existence of Coulomb friction in the cylinder seals, as they are replaced by the air spring, and strong reduction of Coulomb friction from the rod against the cylinder.

3.2. Air suspension with lowered eigenfrequency and variable air damping

From Eq. (6) it can be noticed that the spring characteristic k can also be lowered by increasing the volume V but without changing the surface A of the air spring. This can be achieved by connecting to the air spring a second closed volume of air that cannot be changed in size (enclosure in hard material). If the extra air volume would be infinitely large, the displacement needed for the spring to be activated would go to infinity. Consequently, the resonance frequency will become infinitely close to zero. This is of course not achievable. Again a compromise has to be sought between the additional air volume, which determines the resonance frequency and the necessary extra damping. The insertion of a throttle valve between air spring and additional volume produces the necessary damping. The same principle as in Section 3.1 can be used. At low frequencies, a large amount of air passes through the throttle valve at slow rate. Therefore, large effective damping will be generated. At higher frequencies, the same amount of air passes the valve with higher speed generating a higher resistance in the valve. The damping value will become so high that the valve blocks, resulting in a nearly pure spring behaviour of the system with low effective damping. This way of damping completely eliminates the Coulomb friction generated by the suspension system. When considering passive suspension systems, the mechanism explained above will provide the best results.

All the benefits are summed up here:

- *use of an air spring*:
  - no linkage friction exists as in the mechanical spring;
  - contains inherent damping, so that less extra damping is required;
  - when largely stressed or stretched the stiffness increases; therefore, better protecting the driver against end-stop;
- *use of an extra air volume to lower the eigenfrequency*:
  - the original size of the air spring is preserved (changing the height of the air spring would create problems with the packaging situation);
  - as seen before does a lower eigenfrequency, according to Eq. (5) when dealing with a harmonic damped system, imply that the peak will have a more narrow base. This implies that with appropriate damping only in a small frequency range the input signal gets amplified;
  - due to the lowered eigenfrequency, the frequencies mostly excited in agricultural machinery will receive better attenuation and therefore the possibility of end-stop impacts decreases;
- *use of air damping with a throttle valve*:
  - variable damping can be inserted, which means that with proper tuning the pure spring condition can be reached in regions at a safe distance from the resonance frequency and high damping can be provided at the resonance frequency;

◦ no friction losses (viscous and Coulomb) in the suspension system exist as with hydraulic damping, which significantly improves the performance at low excitation magnitudes.

A disadvantage of the system is that the extra air volume is too large to be placed under the seat and causes problems in the packaging situation. However, if the connecting tube between air spring and air volume is sufficiently wide, a tube length up to 1.5 m will not interfere with the optimal performance of the suspension system as concluded by the authors. Therefore, placement of the air volume next to the seat or integrated in the cabin are possible solutions. This disadvantage is inferior to the comfort and health improvement using the new system.

The criteria to optimize the set up is the same as for Section 2.3.

To illustrate the behaviour of the optimal system with extra air volume the comparison is made with the other types of suspension system (Fig. 11). The input signal is a shaped multi-sine with high acceleration amplitudes in low frequencies (1–4 Hz) and low acceleration amplitudes in high frequencies (5–10 Hz). The new system is developed by the Laboratory of Agro-Machinery and -Processing within the EC project SAFEGUARD. The differences are significant. Performance in both low and high frequencies is improved. In future, the evaluation of such systems in real use will determine the impact of the improvements.

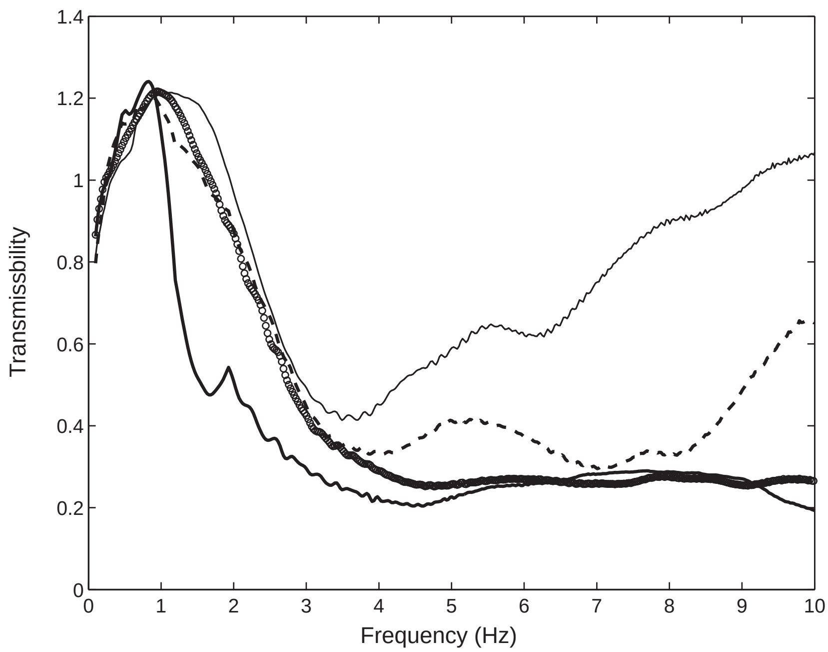

Fig. 11. Frequency response function of four different seat suspension systems (solid line: mechanical suspension with hydraulic damper; dashed line: air suspension with hydraulic damper; circle: air suspension with integrated air damper; thick solid line: air suspension with extra air volume and air damping through a throttle valve) when excited with the same swept-sine in the frequency range 0.7–15 Hz.

4. Conclusions

The occupation of driving agricultural machinery is closely linked to the occurrence of LBP. And though it is not yet clear what role whole-body vibrations play in the occurrence of LBP, prevention is the keyword. In agricultural machinery, drivers often encounter high excitation in low frequencies (1.5–5 Hz) when driving under road conditions. Many current seats have their eigenfrequency is this frequency range. The result is bad attenuation and the occurrence of end-stop impacts. Attenuation performance is highly dependent on the excitation magnitude. The reason lies with the non-friction damping and the friction damping. This is certainly true for the mechanical spring with hydraulic damper unit. By introducing the air spring, inherent damping makes less extra damping necessary, therefore improving the attenuation performance in high frequencies. The use of air damping further eliminates the friction damping. By adding an extra air volume, that can be placed next to the seat or integrated in the cabin, a lower resonance frequency is possible without creating problems in the placement of the air spring and packaging. Combining the three features (air spring, extra air volume, and air damper (throttle valve)) creates a suspension system that provides the best vibration attenuation, when considering passive suspension systems, for agricultural machinery or other vehicles with similar vibration inputs.

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