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On passive displacement-dependent damping for cab suspension

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1. Introduction

Off-road vehicle drivers are often exposed to severe low frequency, whole body vibrations and shock due to irregularities in the terrain. Vibration exposure is an important risk factor for lower back disorders [1, 2]. Cabin and seat suspensions are developed to isolate the driver from vibration sources [3]. Soft suspensions are generally desirable as they provide a good comfort level and high attenuation of high frequency excitations [4]. Suspensions with low stiffness have a large dynamic deflection when exposed to either shocks or low frequency vibration of high magnitude. However, the suspension rattle space is usually very limited due to the mechanical structure of the vehicle. A typical cabin is originally designed to be an integral part of the vehicle structure. When suspension is added to the design, compromises have to be made with the suspension rattle space.

A limited rattle space increases the risk of end-stop impacts [5]. If the vehicle traverses a large obstacle, the limits on the travel might be exceeded causing an impact as the suspension mechanism hits the limit of its compression or extension. To reduce the acceleration during impact, rubber end-stop buffers are mounted at the extremes of the travel. However, the rubber buffer acts mainly as an energy store rather than an energy dissipater. The stored energy is converted back to cabin kinetic energy, which leads to a higher vibration level. Thus it would be beneficial to replace the energy stores with energy dissipaters. In practice the energy dissipation can be realized by replacing the rubber buffers by a displacement-dependent damping factor. Modern electronics and material technology provide the possibility to realize the displacement -dependent damping electronically. Magnetorheological (MR) or electrorheological (ER) dampers with microprocessor control can be utilized [6]. A more robust and straightforward way is to make use of common hydro-pneumatic dampers. They offer a convenient way to implement such nonlinear behavior hydraulically without additional electronics.

This paper focuses on characterization of the displacement-dependent damping curve. A continuous function for the viscous damping factor is presented and its parameters are discussed. The effect and selection of the parameters is studied by means of numerical simulations.

2. Viscous damping coefficient as a function of suspension stroke

It was found convenient to represent the displacement-dependent viscous damping coefficient by a single continuous equation. An important objective is to minimize the number of parameters in the equation. A change in one parameter should not have an influence on other parameters or the influence should be as small as possible. The proposed displacement-dependency of the damping coefficient is

$$C(x) = C_{max} - \frac{tanh\big(\alpha(x+\beta)\big) - tanh\big(\alpha(x-\beta)\big)}{2\big(C_{max} - C_{min}\big)^{-1}}$$

The displacement-dependency function defines a symmetric behavior with respect to the central position of the piston stroke. The curve can be characterized with four parameters: C_{max}, C_{min}, α and β which determine the maximum value of the damping coefficient, the minimum value of the damping coefficient, the slope of the curve when moving over to C_{max} and the width of the constant damping coefficient region, respectively. The shape of the curve is illustrated in (Fig. 1).

Variation of the parameters affects the curve as illustrated in Figs. 2 and 3. In Fig. 2 the value of C_{max} has been varied and in Fig. 3 the slope (β) has been varied. By varying the parameters it is possible to vary the area under the curve. The area under the curve is related to the amount of dissipated energy and the need of rattle space. The smaller the area under the curve, the higher is the need of rattle space. So the area should be kept above a certain value to prevent bottoming.

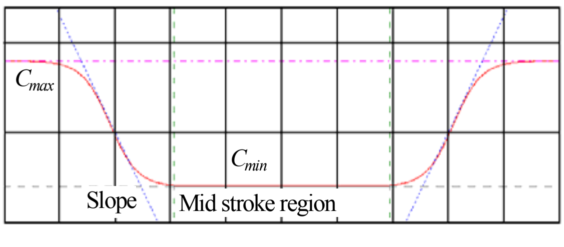

Fig. 1 Parameters for the displacement-dependency

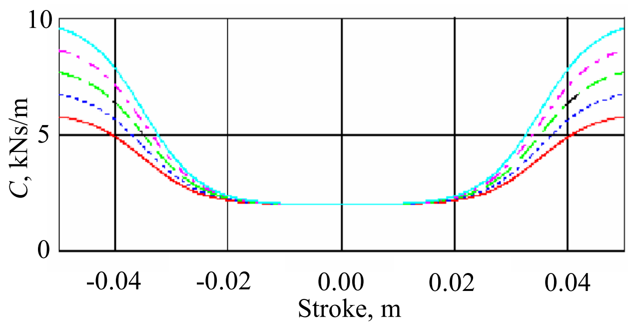

Fig. 2 Damping curve shapes

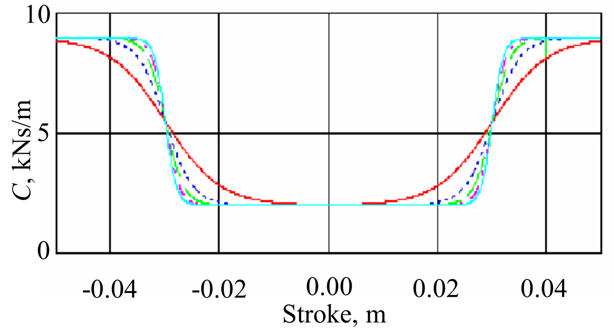

Fig. 3 Damping curve shapes

3. Simulation model

A numerical full-vehicle simulation model (MSC.Adams) was used as an environment to study the effects of displacement-dependent damping (Fig. 4). The model is comprised of the sub-models representing tyres, front-axle kinematics, main frame and the cabin. Also the driving base (soil) was modeled. The simulations used in this paper involved two different bases, namely ISO-5008 (2002) smooth and rough. The vehicle used in simulations represents a general off-road vehicle with a suspended cabin and rigid axles. The suspension is of the 4-point type and the springs are linear. The mass of the vehicle is around 4000 kg. The cabin suspension rattle space is ±50 mm. This paper deals only with vertical motion.

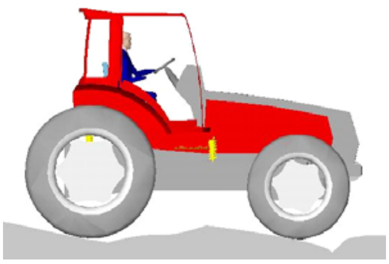

Fig. 4 Off-road vehicle in the simulation model

4. Selection and definition of the damping parameters

A good starting point for the selection of damping curve parameters is the mid-stroke damping value parameter C_{min}. It defines the damping factor when the vibration excitation is insignificant and the suspension displacement amplitude is small. The value of the C_{min} can be selected to give smooth and comfortable behavior in road driving. This selection criteria ignores more severe driving conditions and it will certainly lead to too lightly damped suspension for off-road use.

To tackle the suspension bottoming in off-road use, the shape of the damping curve should be something like that presented in Fig. 1. The rising damping coefficient increases the mean damping in conjunction with increasing suspension travel. The selection of the parameter β affects the width of the constant damping region. The selection approach adapted in this paper is based on the suspension travel on the ISO-smooth test track. The parameter β is selected so that the suspension works linearly over the ISO-smooth track.

Parameter α affects the steepness of the damping curve. A bigger α leads to a steeper rise of the damping factor. The rise in damping factor affects the acceleration change rate. It is known that human comfort is related to acceleration change rate. A jerking motion is felt uncomfortable, so the change in damping curve should be smooth and mild. A sudden change in the damping curve would effectively destroy the benefits of nonlinear damping.

Parameter C_{max} affects the maximum value of the damping factor. C_{max} has to be large enough to prevent suspension bottoming, but it should not overshoot as the acceleration change rate should be kept low.

A set of simulations was conducted to find the usable values for different variables. The first step involved finding the suitable value for parameter C_{min}. The vehicle was driven on the ISO-smooth track and the suspension travel with a low constant damping factor was considered. Next the slope of the displacement-dependency function was determined by making use of the crest factor. The crest factor is a dimensionless quantity defined as the ratio between peak acceleration and the acceleration RMS value. If the acceleration signal contains an instantaneous single shock, the crest factor increases, but the RMS value is not substantially affected.

5. Results

The value of the linear damping coefficient is chosen to be 2 kNs/m based on two facts: with the speed of 8 km/h and the value of 3 kNs/m for the linear damping coefficient on the ISO-rough track, only one end stop impact can be found. By using the same speed and track, the value of 1 kNs/m is too low which means that the maximum value of the displacement-dependent damping is going to be too large. Fig. 5 illustrates the left rear damper rattle space needed for the ISO-smooth track with a speed of 30 km/h and linear damping coefficient value of 2 kNs/m.

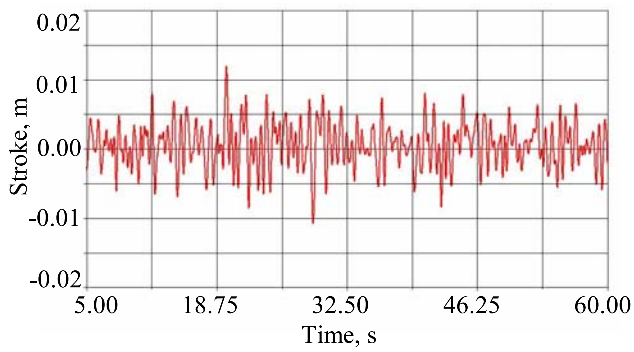

Fig. 5 Rattle space on ISO-smooth track

From the Fig. 5 it can seen that the length of the mid stroke region can be chosen to be about 12 mm – 15 mm for both directions. The value of parameter β in this case is going to 0.035. The width of the region depends a bit on the chosen slope of the curve, but when the parameter α is chosen to be 100 (smooth), the width of the linear region is about ±15 mm.

The maximum value of damping coefficient can be found by simulations. Starting from a low value and examining the rattle space needed, it can found that the value of 8 kNs/m still produces bottoming, and the value of 9 kNs/m does not. The optimal maximum value lies somewhere between these values, more close to 9 kNs/m than 8 kNs/m, so it is reasonable to choose the maximum value of the displacement-dependent damping coefficient to be 9 kNs/m. The resulting damping coefficient curve is illustrated in (Fig. 6)

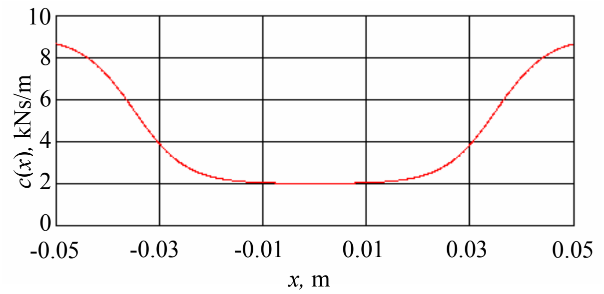

Fig. 6 Displacement-dependent damping coefficient

Fig. 7 illustrates the calculated RMS-accelerations for vertical vibrations measured from the cabin floor below the driver sip-point. The measurements are from the ISO-rough track with a speed of 8 km/h. The columns represent the RMS given by damping coefficient values 3, 6, 9 kNs/m and the displacement-dependent damping, from left to right respectively.

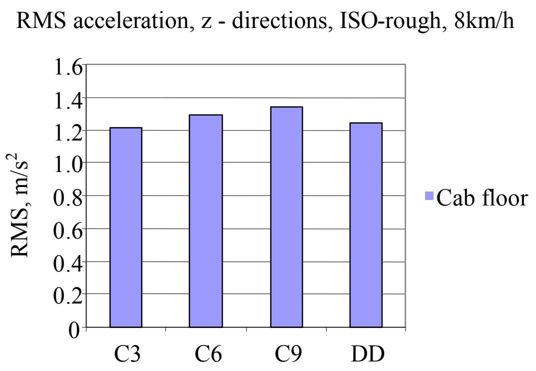

Fig. 7 Vertical RMS-accelerations on ISO-rough

From the figure it can be seen that the displacement-dependent damping coefficient produces almost as low RMS as the 3 kNs/m. It is obvious that the position dependent damping will not affect the RMS compared to the linear damping coefficient of 2 kNs/m on the ISO-smooth track.

6. Conclusions

This paper deals with damping of the vibrations in the vertical direction, concentrating on preventing end stop impacts. It was shown that the vertical RMS can be held below a certain value and the bottoming can be prevented at the same time. During the simulations it was noticed that the RMS values for horizontal directions (especially in the *y*-direction) increased when the linear damping coefficient was reduced. It is clear that the bottoming is partly caused by horizontal vibrations, but the displacement-dependent passive damping will not negate the fact that, by improving the RMS in the vertical direction, the RMS in the horizontal direction will degrade. This is a drawback because the horizontal vibrations are considered to be a greater source of discomfort than vertical vibrations: In standards the accelerations in horizontal directions have a multiplier of 1.4. The major conclusion is that there should be two independent damping systems for cab suspension; for vibrations in the vertical direction and for vibrations in the horizontal direction.

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PASYVUS NUO POSLINKIO PRIKLAUSANTIS KABINOS PAKABOS SLOPINIMAS

R e z i u m ė

Tikslas – žemės ūkio paskirties traktoriaus kabinos pasyvios keturių taškų pakabos tyrimas, atliekant kompiuterinį modeliavimą (MSC ADAMS). Modeliuojant taikytas tiesinis slopinimas ir standumas esant lygiems ir nelygiems ISO standarto keliams. Buvo matuojami kėbulo, kabinos grindų ir vairuotojo sėdynės poziciniame taške (SIP) veikiantys pagreičiai, aptarta būtinybė numatyti kabinos pakabos minimalaus slopinimo diapazoną.

Pateiktame darbe akcentuojami kėbule veikiantys pagreičiai, perduodami kabinos grindims. Pagreičiai skaičiuoti kiekviename matavimo taške. Sėdynės pakabos parametrai buvo pastovūs.

Pastebėta, kad mažėjant standumui, mažėja vairuotojui perduodami pagreičiai abiejų tipų keliuose, esant įvairiems greičiams. Mažinant slopinimo laipsnį, amortizatoriaus stūmoklis dažniau smūgiuoja į ribotuvą. Tokios tendencijos išvengiama naudojant nuo padėties priklausantį amortizavimą, parenkant jį taip, kad amortizavimas būtų nedidelis vidutiniame diapazone ir gerokai padidėtų prieš smūgiuojant į ribotuvą.

Pateikiami kėbulo perduodami kabinos grindims ir vairuotojo sėdynei pagreičiai. Nuo padėties priklausantis amortizavimas sumažina smūgių į ribotuvą skaičių, bet mažai padidina perduodamus pagreičius. Nuo padėties priklausantis pasyvus slopinimas leidžia pagerinti bendras slopinimo charakteristikas. Nuo amortizatoriaus stūmoklio padėties priklausantis slopinimas gali būti nusakomas keturiais parametrais. Pasiūlyta šių parametrų parinkimo procedūra.

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ON PASSIVE DISPLACEMENT-DEPENDENT DAMPING FOR CAB SUSPENSION

S u m m a r y

The aim of this research was to study the behaviour of a passive 4-point agricultural tractor cab suspension through computer simulation analysis (MSC.ADAMS). Simulations were carried out using linear damping and stiffness on ISO-smooth and ISO-rough standard tracks. Accelerations from the vehicle body, the cab floor and the driver sip-point were measured and the need for the cab suspension rattle space was considered. This paper concentrates on the accelerations transferred from the vehicle body to the cab floor. The RMS–accelerations were calculated from each measuring point. The seat suspension parameters were held constant.

It was noticed that lower damping gives lower RMS–acceleration experienced by the vehicle driver on both tracks at different driving speeds. By decreasing the degree of damping an increased incidence of damper piston bottoming was found. This tendency was prevented by means of position-dependent damping. The position dependency was chosen so that the damping remains low in the mid-range and increases considerably when approaching the end-stops.

As a result, the RMS–accelerations transferred from the vehicle body to the cab floor and from the vehicle body to the driver sip-point are represented. With position-dependent damping, the number of end-stop impacts reduced with a small increase of transferred RMS–acceleration. With position-dependent passive damping, good overall damping characteristics can be attained. Damping as a function of piston position can be stated with four parameters. A selection procedure for these parameters is proposed.

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ПАССИВНАЯ ОТ СДВИГА ЗАВИСЯЩАЯ АМОРТИЗАЦИЯ ПОДВЕСКИ КАБИНЫ

Р е з ю м е

Проведено исследование пассивной четырехточечной подвески кабины сельскохозяйственного трактора путем проведенного моделирования (MSC ADAMS). При моделировании использовались линейная амортизация и жесткости для соответствующих ISO стандарту ровных и неровных дорог. Измерялись корпусные ускорения, а также ускорения пола кабины и в характерной точке водительского сидения, обсуждена необходимость учета холостого хода.

В представленной работе акцентируются действующие корпусные ускорения, передаваемые полу кабины. Получены ускорения в каждой измеряемой точке. Параметры подвески сидения не изменялись.

Замечено, что при меньшей жесткости уменьшаются водителю передаваемые ускорения на дорогах обоих типов при разных скоростях. С уменьшением степени гашения, наблюдались более частые соударения поршня об ограничитель. Во избежание таких тенденций используется от положения зависящее гашение, подобранное таким способом, чтобы в среднем диапазоне гашение было бы незначительным, а при соударениях об ограничитель гашение увеличивалось бы.

Получены ускорения, передаваемые в корпус, пол кабины и в характерную точку водительского сидения. От положения зависящая амортизация уменьшает число соударений по ограничителю, но частично увеличивает действующие ускорения. От положения зависящая пассивная амортизация дает возможность получить хорошие общие амортизационные параметры и может быть характеризована четырьмя параметрами. Предложена процедура подбора этих параметров.

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