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Modeling and Optimization of Passive Seat Suspension

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Abstract This paper deals with modeling and optimization of a working machine seat suspension system parameters. Experimental work in the past shown that it is possible to replace even more complicated, for example parallelogram or scissor mechanisms, with simpler dynamic models of 1 or 1.5 degree of freedom (Zener's model). Optimization of the damper and spring settings is performed using a two-objective function optimization technique. This enables to minimize not only the exerted vertical vibration acceleration of the seat squab, but also the seat squab relative vertical displacement (stroke) in regard to the working machine cabin. The first component of the objective function expresses comfort of the seat, the second one expresses the safe handling of the working machine. This optimization technique enables to propose so called "soft", "medium", or "hard" seat suspensions according to the value of the weighting coefficient. The paper also points on the possibility of improving the dynamic characteristics of the seat with the use of a dynamic vibration absorber. The expediency of its application is especially in working machines without significant changes in the seat excitation frequency spectrum.

Keywords: *passive seat suspension, comfort, optimization, frequency spectrum, dynamic vibration absorber*

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1. Introduction

Modeling and optimization of the seat suspension system parameters is an unavoidable part in designing the seat that has to ensure required comfort of the driver and also the safe handling of the working machine. Prolonged exposure to vibrations in working environment leads to early fatigue and in some cases to a deteriorating health state. The vertebra is the most effected. However, other inner organs may also be affected. Epidemiological studies in working environment [1,2] stressed the need for proper dynamic characteristics of seat suspension systems.

Nowadays standard commercial seats are suspended by use of passive suspension systems [3,4,5,6], which are often modeled as systems of one degree of freedom (DOF). It turned out that more accurate modeling of the driver seated on the seat as a multi-body vibrating system does not lead to the significant seat dynamic characteristics modification. Reversely, it significantly complicates its modeling and also optimization. For example paper [3] shows, that in the case of the seat with a low natural frequency its loading with one rigid–body mass (equal to 73% of the driver's weight) leads to similar results compared with the loading by multi- body linear oscillators imitating dynamics of the real driver.

The next problem in modeling the seat is its cushion, which shows certain elastic and damping properties. Its accurate modeling requires individual experimental identification of the stiffness and damping coefficients. Nowadays, in the field of passive suspension systems there is an enlargement in the use of dampers with switchable or continuously adjustable characteristics of damping. In this case the adjusting the coefficient of damping is left on the driver and his subjective perception of comfort. Owing to inexperience of drivers this method is left often unused. In paper [3] is shown, that in modeling a locomotive driver's seat with parallel ordering of the spring and damper it is necessary to consider besides the stiffness of the spring (or equivalent spring stiffness, if the seat mechanism is made of for example a parallelogram or scissor mechanism) and the damping coefficient of the viscous damper also an additional spring serially ordered with the damper. Its stiffness it is necessary to obtain experimentally.

In the paper also the advantage in the application of a vibration absorber (installed under the seat squab) which enables to improve the dynamic characteristics of the seat will be shown. From the design point of view, it is advisable to realize it as a mass placed on the elastic beam fixed to the seat frame. The advantage of this design is the relatively simple change of the bending stiffness of the beam, by setting the beam length, on which is the mass located.

Passive suspension systems however reached their top and therefore active suspension systems [7,8,9] are gradually more and more used in practice. Their dynamic characteristics are significantly better compared to the dynamic characteristics of passive suspension systems, but they are still more expensive, more complex and therefore less reliable than passive suspension systems. Moreover, active suspension systems require an external source of energy and their energetic demand is high.

A compromise between passive and active suspension systems are semi-active suspension systems [10,11,12,13], which can be realized by the use of magnetorheological (nowadays most used), electrorheological or friction dampers. Comparison of the dynamic properties of active and semi-active suspension systems in vehicles can be found e.g. in [14].

The problem of designing seat suspension systems is complex owing to the necessity to meet usually two conflicting demands. The driver comfort should be as high as possible, while the seat squab relative vertical displacement (stroke) in regard to the working machine cabin must be minimized. This leads to utilization of optimization methods [9,11,15,16].

Basic function of a traditional dynamic absorber is to limit the vibration amplitude of the primary mass. For example in [17] effectiveness of the dynamic absorber is analysed.

2. Dynamic and Mathematical Models

In this chapter the mathematical models of three different types of the seat dynamic models will be derived.

2.1. Dynamic and Mathematical Models of the Basic Seats (Cases A and B)

For comparison of the importance of the added spring ordered serially with a viscous damper (Zener's model) we consider as the first one the basic dynamic model of the seat of 1 DOF (case A) with a spring of equivalent stiffness k and a viscous damper with a constant coefficient of damping c, the Figure 1. We will consider that this coefficient of damping is continuously adjustable between the lower and upper bound. The Figure 2 represents 1.5 DOF Zener's model (case B) with an added spring with stiffness coefficient k_t, serially ordered with a damper with the continuously adjustable coefficient of damping c and with a parallelly ordered spring with equivalent stiffness coefficient k_1, which is considered also as continuously adjustable (realized for example by a pneumatic spring). In the Figure 1 and Figure 2 m_s represents the weight of the seat squab and all the masses connected with it, m_{op} represents 73% of the driver's weight.

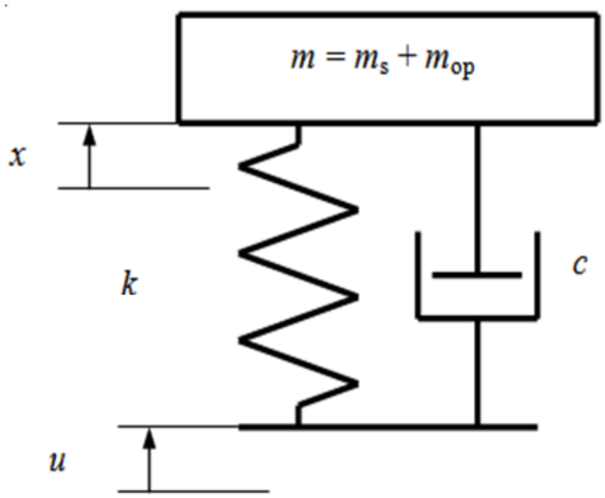

Figure 1. Dynamic model of the seat (case A)

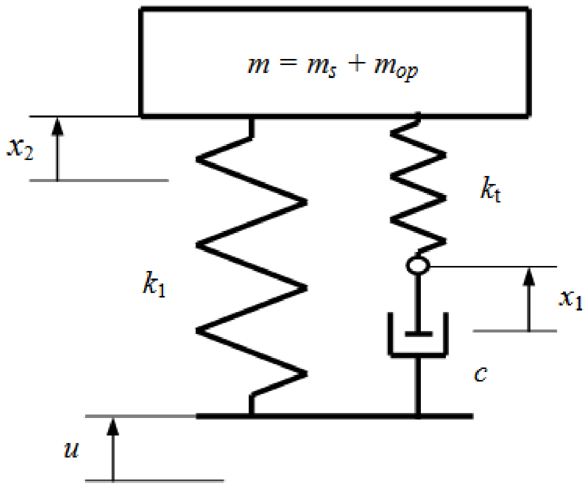

Figure 2. Dynamic model of the seat (case B)

Coordinate u represents the specified kinematic excitation of the seat, excited from the working machine cabin floor, coordinate x (the Figure 1) and x_2 (the Figure 2) define the position of the seat squab, x_1 defines the position of the point of connection between the damper c and the additional spring k_t.

Equation of motion for the case A (the Figure 1) can be written as follows

$$\left(m_s + m_{op}\right)\ddot{x} + c\left(\dot{x} - \dot{u}\right) + k\left(x - u\right) = 0 \tag{1}$$

and for the case B

$$\begin{aligned} &\left(m_s + m_{op}\right)\ddot{x}_2 + k_t\left(x_2 - x_1\right) + k\left(x_2 - u\right) = 0, \\ &k_t\left(x_2 - x_1\right) - c\left(\dot{x}_1 - \dot{u}\right) = 0. \end{aligned} \tag{2}$$

2.2. Dynamic and Mathematical Model of the Seat with the Dynamic Vibration Absorber (Case C)

The dynamic model of the seat with the dynamic vibration absorber is schematically illustrated in the Figure 3. The dynamic vibration absorber is of mass m_2, stiffness of the elastic beam (placed under the seat squab) is k_3. The meaning of the other parameters is obvious.

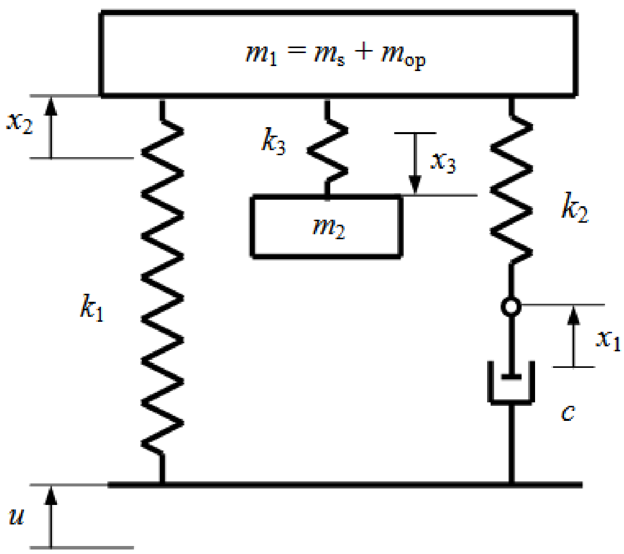

Figure 3. Dynamic model of the seat with the dynamic absorber (case C)

Equations of motion for the case C can be written in the form

$$\begin{aligned} &m_1 \ddot{x}_2 + k_1 (x_2 - u) + k_2 (x_2 - x_1) - k_3 (x_3 - x_2) = 0, \\ &m_2 \ddot{x}_3 + k_3 (x_3 - x_2) = 0, \\ &k_2 (x_2 - x_1) - c(\dot{x}_1 - \dot{u}) = 0. \end{aligned} \tag{3}$$

2.3. Kinematic Excitation of the Seat

Experimentally obtained excitation will be used. It was measured experimentally inside the driver's cabin (on its floor under the driver's seat in vertical direction) in the bucket-wheel excavator Schrs 1320. For the predesign of the parameters of the dynamic vibration absorber it is beneficial to do the frequency analysis of the seat excitation for determination of the interval comprising the dominant frequencies.

The displacement time response of the excitation is shown in the Figure 4. In the Figure 5 its frequency spectrum is shown.

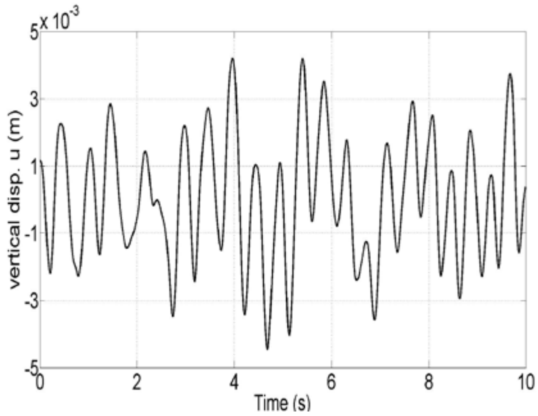

Figure 4. Displacement time response in the Schrs 1320 cabin

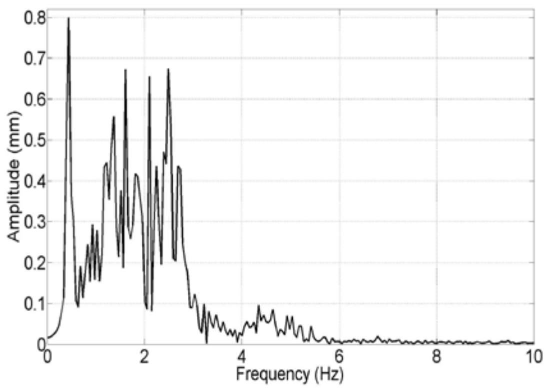

Figure 5. Frequency spectrum of excitation

From the frequency spectrum flows that the most distinguished frequency components lie in the interval from 1 to 3 Hz, but the dominant component, which is assigned by the peak, is located at approximately 0.5 Hz.

3. Optimization of the Seat Suspension Parameters

In the chapter the optimization problem is formulated and optimization results are given for all three suspension systems A, B and C.

3.1. Formulation of the Optimization Problem

For all cases of suspension A, B and C the objective function will be assumed the same

$$f_{op} = w \frac{a_{ef}}{a_{ef\ NOM}} + (1-w) \frac{x_{rel,ef}}{x_{rel,ef\ NOM}}. \tag{4}$$

It is a multiobjective optimization, where w is the weighting coefficient (gaining values from 0 to 1), the first component is characterized by a_{ef} which defines the effective acceleration of the seat squab ($\ddot{x}$ in the case A, or $\ddot{x}_2$ in both cases B and C) and $x_{rel,ef} = x - u$ in the case A, or $x_{rel,ef} = x_2 - u$ in the cases B a C, define the relative displacement effective value. For this quantities the following equations hold

$$a_{ef} = \sqrt{\frac{1}{T} \int_0^T a^2(t)\, dt} \tag{5}$$

and

$$x_{rel,ef} = \sqrt{\frac{1}{T} \int_0^T x_{rel}^2(t)\, dt}. \tag{6}$$

Utilization of optimization methods is needed, because the minimization of a_{ef} usually leads to its decreasing (increase in comfort), but also increasing of $x_{rel,ef}$ (decreasing the ability of clear sightedness of the driver) and vice versa.

Both effective values mentioned above are in equation (4) devided by their nominal values (defined by the mean values of the optimization parameters in their search intervals), because values of a_{ef} and $x_{rel,ef}$ are not commensurable, which could cause problems in the numerical optimization.

In the cases A and B the optimization paramaters (optimization variables) are the spring stiffness k and the coefficient of damping c. Their search intervals are: $k \in \langle 3300, 9300 \rangle$ (N/m), $c \in \langle 500, 4000 \rangle$ (N.s/m). The value of the coeficient of stiffness k_2 in the case B is 76800 N/m [3].

3.2. Optimization Results

For optimization the Matlab Optimization Toolbox was used [15]. After the numerical optimization it could be possible to draw in the coordinate system composed of axes a_{ef} and $x_{rel,ef}$ a curve that would correspond to values of the weighting coefficient w within the interval from 0 to 1. It is clear that if w = 1 we get a so called "soft" suspension of the seat, that prefers comfort of the driver, value w = 0 represents a so called "hard" suspension, which prefers the ability of clear sightedness, and value w = 0.5 a so called "medium" suspension with a balanced emphases on both components of the objective function under consideration.

For the case A with the value of w = 1 were obtained the following optimum values of optimization parameters: k_{op} = 3300 N/m, c_{op} = 500 N.s/m. The corresponding value of the effective acceleration of the seat is a_{ef} = 0.1598 m/s^2 and the value of the effective relative displacement is $x_{rel,ef}$ = 0.0018 m. The relative displacement $x_{rel}(t)$ and acceleration $\ddot{x}(t)$ time responses of the seat squab are shown in the Figure 6 and Figure 7.

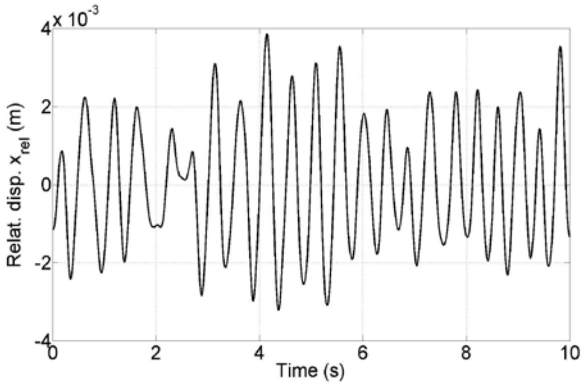

Figure 6. Time response of the seat relative displacement for the case A

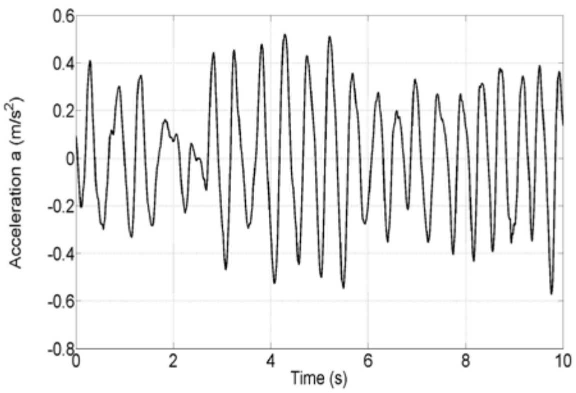

Figure 7. Time response of the seat acceleration $\ddot{x}(t)$ for the case A

For the case B the optimal "soft" suspension (w = 1) is characterized again by the following values: k_{op} = 3300 N/m, c_{op} = 500 N.s/m. The value of the seat effective acceleration is a_{ef} = 0.1710 m/s^2 and the value of the effective relative displacement is $x_{rel,ef}$ = 0.0019 m. The relative displacement $x_{\text{rel}}(t)$ and acceleration $\ddot{x}(t)$ time responses of the seat are shown in the Figure 8 and Figure 9.

From the comparison of the values of the effective acceleration a_{ef} for the cases A and B flows, that there is an increase in the value of the effective acceleration by 7%. The increase in the value of $x_{rel,ef}$ is negligible. It is obvious, that the influence of the additional spring serially ordered with the viscous damper is not significant, see Figure 6 and Figure 8 and also Figure 7 and Figure 9. On the other side the 7% decrease in the value of the effective acceleration is not negligible.

In the case C with the dynamic vibration absorber optimization was realized in such a way, that the parameters taken from the "soft" suspension (the case B) were considered, and for the selected weight of the dynamic vibration absorber m_2 = 6 kg the minimum value of a_{ef} for the case C was obtained by 1– dimensional optimization, where the only optimization variable was the stiffness coefficient k_3. For the optimum value of this coefficient k_{3op} = 1760.52 N/m there was determined the value of a_{ef} = 0.142780 m/s^2 and $x_{rel,ef}$ = 0.0019 m. Comparing values of the effective acceleration in the cases B and C it is possible to allege the decrease in the value of the effective acceleration by 16.5 %. The value of the effective relative displacement was unchanged.

The Figure 10 shows the time responses of the seat relative displacement $x_{\text{rel}}(t)$ and the Figure 11 the time responses of the seat acceleration $\ddot{x}_2(t)$ of the seat squab for the cases B and C.

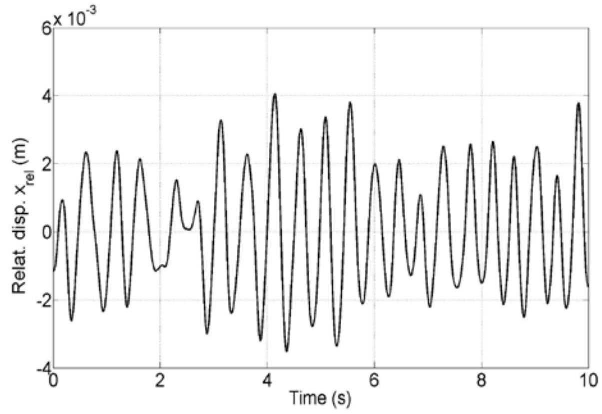

Figure 8. Time response of the seat relative displacement for the case B

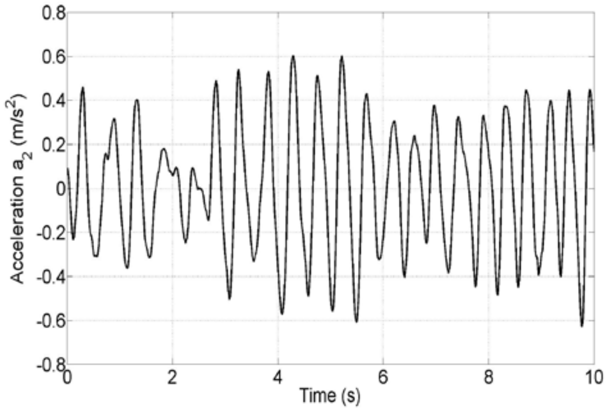

Figure 9. Time response of the seat acceleration $\ddot{x}_2(t)$ for the case B

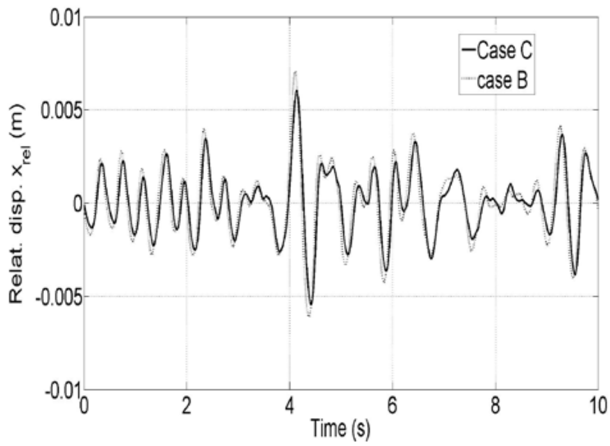

Figure 10. Time responses of the seat relative displacement for the cases B and C

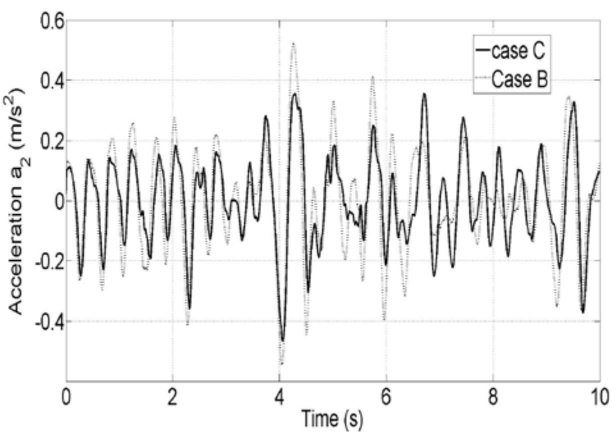

Figure 11. Time responses of the seat acceleration $\ddot{x}_2(t)$ for the cases B and C

4. Conclusion

In the paper three types of a working machine cabin seat were modelled and optimized. The first one is a 1 DOF model, the second one is 1.5 DOF model (so called Zener's model) and the last one is a 2.5 DOF model with the dynamic vibration absorber. It was shown that Zener's model with an additional spring serially ordered with the damper does not significantly influence the dynamic characteristics of the seat.

The multiobjective optimization of the seat suspension system parameters (its stiffness and damping paramaters) is of great importance, because it gives the optimum values of the optimization paramaters from their search intervals. It allows to project the so called "soft", "medium" or "hard" seats, according to what is preferred – comfort or the ability of clear sightedness (the safe handling of the machine) or balance of both criteria.

A relatively significant enhacement is provided by using the dynamic vibration absorber. With its appropriate realization (with the use of an elastic beam with an adjustable length) it is also possible to adjust position of the weight according to the subjective feeling of comfort of the working machine operator.

Acknowledgement

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