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Application of a magneto-rheological damper and a dynamic absorber for a suspension of a working machine seat

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Abstract

The article deals with two alternatives of semi active suspensions of a seat of working machine. A magneto-rheological damper is used in the first case and in the second case the suspension of the seat is a combination of magneto-rheological damper and dynamic absorber. The dynamic absorber is composed from passive elements. In both cases, the dampers are controlled by the well known Sky hook algorithm. The dynamic analysis is focused on the influence of the applied passive dynamic absorber on the reduction of the seat displacements. The passive parameters of the seat suspension and the dynamic absorber were evaluated based on the optimization process using genetic algorithms according the defined minimization function.

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Keywords: magneto-rheological; damper; genetic algorithms; dynamic absorber; seat

1. Introduction

Operators of earth-moving machines, bus and truck drivers during their work are exposed to vibration due the unbalanced rotating parts of the machines or by the road surface unevenness. The biggest problem of operators and drivers of earth-moving and mining machines is that they are exposed to vibrations in a frequency range from almost

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0 to 20 Hz [1]. This kind of discomfort causes that operator to lose concentration, get tired early and after several years it can have dramatic effects on their health.

Nomenclature

m_i	suspended mass
x_i	mass displacement
$\dot{x}_i$	mass velocity
y	kinematic excitation
b_i	coefficient of damping
k_i	spring stiffness
F_s	MR damper generated force
c_d	velocity and current dependent coefficient of damping for MR damper
α	shaping parameter of the hysteresis loop
$z, \dot{z}$	evolutionary functions of the hysteresis loop
A	hysteresis parameter
γ	hysteresis parameter
β	hysteresis parameter
k_d	spring stiffness of the MR damper
I	electric current
d_{ef}	effective value of displacement
d_{is}	displacement
T	computation time

It is well known that some of the human viscera have their natural frequencies in the range from 4 to 10 Hz [2,3]. The suspension of an operator's seat in mining machines or agricultural machines is sometimes the only way how to reduce transitionsof vibration on anoperator. In a working condition with low frequencies of vibration passive seat suspension are not suitable, due to fact that they can amplify the amplitudes of vibration with frequencies close to the natural frequencies. Therefore several proposals were made to improve the passive seat suspension by applying dynamic absorbers to the seats. A solo dynamic absorber is good for only reducing vibrations in narrow frequency ranges, therefore Li in work [4] proposed a multiple tuned mass damper. The natural frequencies of the absorbers were distributed non-uniform. This enabled the absorbers to reduce the vibrations in a wider frequency range. Eason in work [5] proposed a dynamic absorber composed from passive and semi active elements aligned in series. This enabled reduction of the vibration of the investigated three degree system of freedom in a wider frequency range. The authors showed that if the mass of the semi active absorber is four orders lower it reduces vibration greatly in a wide frequency range. The application of semi active dampers led Segľa in work [6] to investigate the influence of a dynamic absorber on a semi active suspended seat. The reduction of vibration using an idealized semi active suspension with a dynamic absorber improved the reduction of vibration only for about 7%. A lot of research was made in the field of suspending the seats, and cars by using semi active dampers and their results showed great improvements of reducing vibrations [7-9]. Idealized models of semi active dampershaveno use in real conditions, because the properties of these dampers such as time delays and the damper hysteresis have to be considered. The hysteresis of the damper can be formulated using parametric or non-parametric equations as it can be seen in [10]. The semi active damper has to be controlled. Several well know control algorithms were proposed such as sky hook, balance control, ground hook control and other. The right choice of a control strategy has to be made, because not every control will help to reduce the vibrations is the dangerous 4-10 Hz frequency zone. Park in his study of a smart seat [11] used a magneto-rheological damper controlled by a Lyapunov type robust control considering the dampers time delay. He reduced the transitions of vibration on the operator seat in an extremely wide frequency zone from 20 to 140 Hz. Only a small reduction of vibrations in the area from 4-10 Hz was observed. Gugliemino used in his research [12] a hybrid balance control strategy. This enabled him to reduce the vibration transmitted on the truck drivers cabin and also a high reduction of the forces transmitted to the road. The right choice of and semi active damper and the control strategy does not give us a 100% chance that the suspension is designed right. Therefore it is necessary to optimize the suspension parameters according the chosen objective function or functions, and also the method of the optimization process as it can be found in [13,14].

The aim of this paper is to study the influence a dynamic absorber applied on an operator seat, which is suspended by a magneto-rheological damper. The parameters of the seat suspension and the dynamic absorber were evaluated using optimization based on genetic algorithms.

2. Mathematical model

The operator seat will be considered as a simple suspended mass. The suspension is realized in the first case by a coil spring and a magneto-rheological (MR) damper, the second investigated alternative is a suspended seat with a MR damper with an application of a dynamic absorber (DA) as it can be seen in Fig. 1 a) and b).

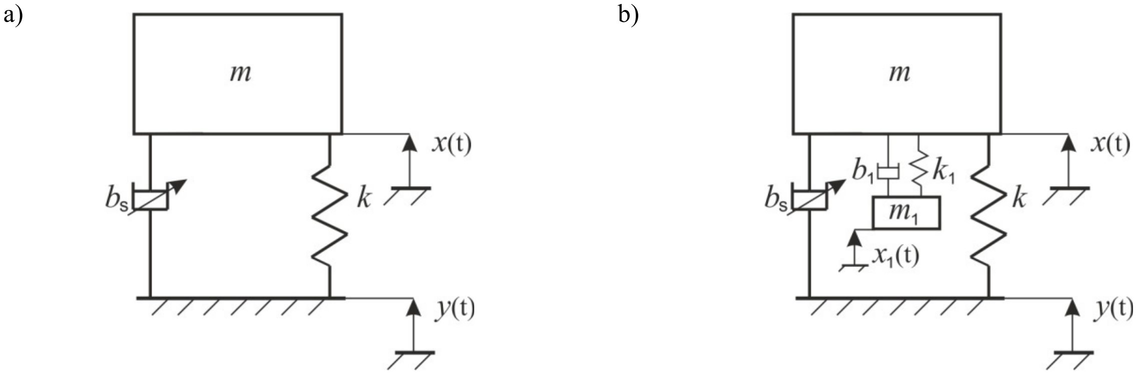


Fig.1The seat suspension a) without a DA; b) with a DA

The equationof motion of the first investigated model is as follows

$$m\ddot{x} + F_s + k(x - y) = 0\,. \tag{1}$$

The equations of motion for the seat with an applied DA are

$$m\ddot{x} + F_s + k(x - y) + b_1(\dot{x} - \dot{x}_1) + k_1(x - x_1) = 0\,, \tag{2}$$

$$m_1\ddot{x}_1 - b_1(\dot{x} - \dot{x}_1) - k_1(x - x_1) = 0\,. \tag{3}$$

The magneto-rheological damper used in the study is RD 1097-01 manufactured by Lord Corporation. This type of damper is appropriate for cases where a high degree of controllability is needed. As is known the MR dampers have their hysteresis. In this study a Bouc-Wen model was adopted, which captures the hysteresis of the damper properly when the coefficients that affect the shape and magnitude of the hysteresis loop are chosen properly. This can be a difficult task and the evaluation has to be done numerically, because the Bouc-Wen equations are non-linear. The generated MR damper force F_s is in this case dependent on two parameters. The first one is a velocity and the second is an applied electric current on the MR coil. Therefore numerical results had to be verified experimentally.

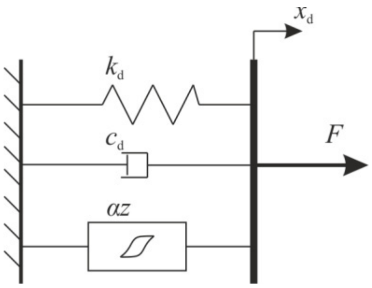


Fig.2Bouc-Wen hysteresis model of the MR damper

The force generated by the damper can be expressed as follows

$$F_s = c_d(\dot{x} - \dot{y}) + \alpha z \, , \tag{4}$$

$$\dot{z} = A(\dot{x} - \dot{y}) - \gamma \left| (\dot{x} - \dot{y}) z^{n-1} \right| z + \beta (\dot{x} - \dot{y}) \left| z^n \right| , \tag{5}$$

where c_d, β, γ, A, n are parameters of hysteresis model.
The parameters used in this study were adopted form experimental data which can be found in detail in [14]. The parameters of the hysteresis model are defined

$$[c_d, \beta, \gamma, A, n] = [0, -7.0773, 10.614, 36.2095, 1], \tag{6}$$

and the current dependent parameters are

$$\alpha(I) = 72.8I^3 - 42.88I^2 + 14.83I + 0.29 \, , \tag{7}$$

$$c_d(I) = -9.37I^4 + 10.22I^3 - 4.33I^2 + 0.89I + 0.02 \, . \tag{8}$$

In both cases the same kinematic excitation y(t) is considered. The kinematic excitation was measuredin the vertical direction of working machine cabin floor. The kinematic excitation is documented in Fig. 3.

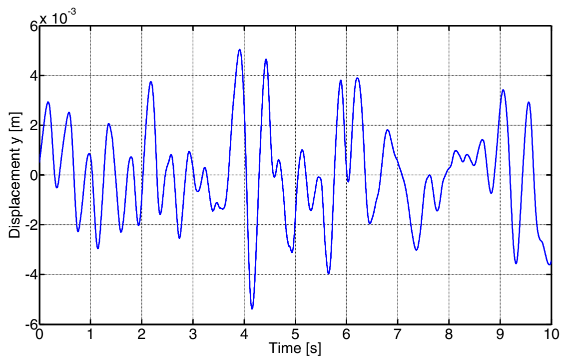


Fig. 3 kinematic excitation y(t)

3. Optimization

The parameters of the MR damper were obtained according experimental results from [14]. The parameters of stiffness and coefficient of damping, the mass of the DA and the stiffness of the DA were evaluated using genetic algorithms in Matlab/Simulink workspace. The objective function was the minimization of the seat absolute displacement

$$f_{\mathrm{ob}} = \min(x)\,. \tag{9}$$

4. Sky hook control

MR damper of the seat was controlled by the well known sky hook control strategy. The strategy was set as an *on-off* control. The current of the damper in the *off* state is 0 A. In the *on* state the current is at a maximal set value 0.4 A. The sky hook control law depends on the absolute and relative velocities of the sprung mass and it can be formulated as follows

$$I = \begin{cases} 0{,}4A & \text{if} \quad (\dot{x}-\dot{y})\dot{x} > 0 \\ 0 & \text{if} \quad (\dot{x}-\dot{y})\dot{x} < 0 \end{cases} \tag{10}$$

5. Numerical results

The input of the optimization process was the kinematic excitation y(t). The task of the optimization was to evaluate the values of the suspension and DA parameters so that the absolute displacement of the seat would be minimal. For faster evaluation lower and upper bounds were added in the genetic algorithms. The lower and upper bounds are shown in Table1 for the seat without DA and for the seat with an applied DA in Table 2.

Table 1.Optimized suspension parameters for the seat without DA

	Bounds	Result
Stiffness k	[3000, 10000]	9363 N/m

Table 2. Optimized suspension parameters for the seat with DA

An example of a column heading	Bounds	Results
Stiffness k	[2500, 10000]	3014 N/m
Stiffness k_1	[500, 2500]	1580 N/m
Damping coefficient b_1	[0, 500]	1 Ns/mm
Absorber mass m_1	[2, 9]	5,872 kg

From the optimized results it is clear that the DA affected the parameters vastly. The spring stiffness of the seat spring was almost 3x smaller compared to the seat without a DA. But this didn't affect the absolute displacements of the seat in a wide range. The results of the absolute displacements of the seats are shown in Fig. 4.

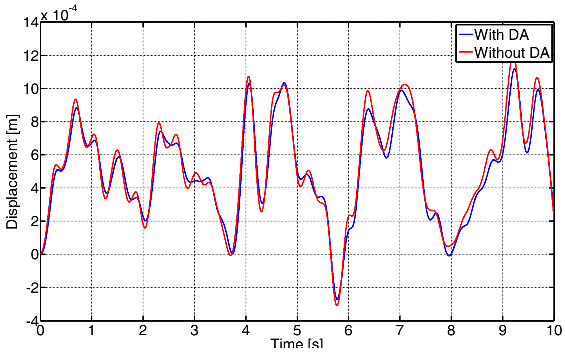


Fig. 4 displacements x(t) over time

From Fig.4 is not possible to evaluate the affectivity of the applied dynamic absorber. In some intervals the applied DA reduced the amplitudes of the displacements, in some interval the amplitudes raise. For better understanding the effective values of displacements had to be evaluated according equation

$$d_{ef} = \sqrt{\frac{1}{T}\int_0^T dis^2(t)\,dt}\,, \tag{9}$$

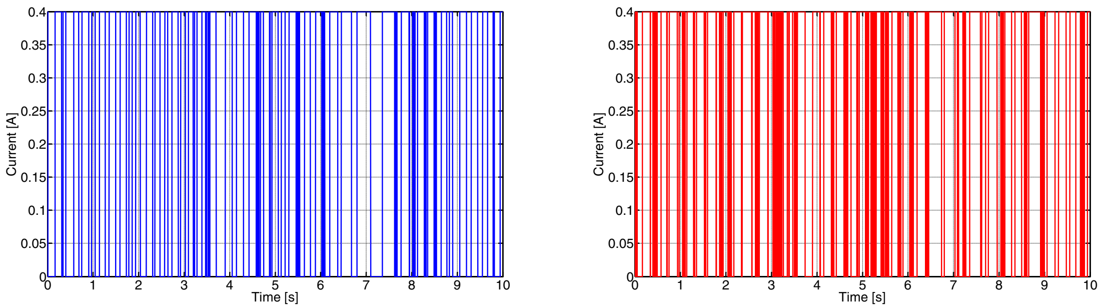
The effective values of displacements for both investigated cases are documented in Table 3.

Table 3. Effective values of displacements

	displacements[m]
Without DA	$6{,}0729.10^{-4}$
With DA	$5{,}8142.10^{-4}$

From the effective values of displacements it is clear that the applied DA reduced the vibration only for about 5%. The better reduction of the applied DA is disputable. From the effective values of displacements it is clear that the DA helped reduce the vibration only by 5% but on the other hand, from Fig. 4 it can be seen that in some intervals the maximal amplitudes of displacements raise. In both cases the behavior of the MR damper was affected by the optimized parameters and by the control law and vice versa. For illustration the change in the electric current in both cases is illustrated in Fig. 5.

a) b)



6. Conclusion

In this paper the affectivity of a DA on seat suspended by a MR damper was studied. From the results of the optimization it is clear that the application of the DA can vastly changes the values of the identified parameters. The change in the seat spring stiffness changed about for 3x, but the effect of the applied DA helped to reduce the effective values of displacements only for 5%. From the time response of the seat displacements it is clear that the responses were affected by the objective function and by the control strategy of the MR damper control. In this case the effect of the applied DA is negligible because the MR dampers reduced the maximal amplitudes to values that are almost unnoticeable for the operator situated on the seats. The maximal amplitude of the displacement was less than 1.5mm in both investigated cases.

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