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SKYHOOK CONTROLLER APPLIED IN NONLINEAR VEHICLE SUSPENSION WITH MAGNETORHEOLOGICAL DAMPER

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Abstract: *This work considers the dynamics and control of a vehicle suspension system through a nonlinear quarter-car model. The control is performed through the Skyhook controller, which is commonly used only to reduce the chassis mass displacement amplitudes and the acceleration of the chassis. Numerical simulations were performed to analyze not only the chassis displacement and acceleration reduction levels but also the wheel axis displacement reduction levels in order to verify if the proposed controllable system has advantages over the uncontrolled system. The actuator considered in this work was the magnetorheological damper due to its advantages related to various applications in the engineering area. The numerical results showed that considering a step-type excitation control was efficient to reduce the chassis displacement levels, wheel shaft mass accommodation time and chassis mass acceleration. In this way, it was possible to consider that the controller was able to add greater passenger comfort and dirigibility to the vehicle at the same time.*

Keywords: *Magnetorheological Damper, Nonlinear Vehicle Quarter-car Model, Skyhook Controller.*

1. INTRODUCTION

The technique of modeling a vehicle suspension system using only a quarter of the vehicle, has been adopted by several authors, example of this are the works of Picado (1998), Tusset (2008), Do *et al.* (2010), Hu *et al.* (2012), Oliveira (2015) and Torres (2016). The reason why so many authors deal with this model, is the fact that it represents the displacement of the chassis and wheel and present the existing relations between the two systems, in which it allows a more efficient study of the comfort, vehicle control system.

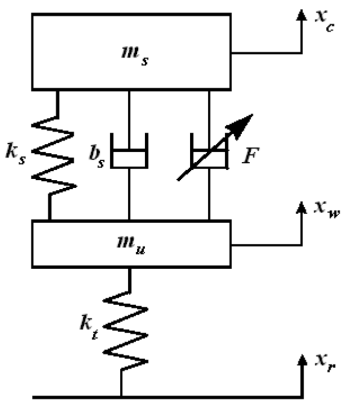

Figure 1. Quarter-car model with two degrees of freedom according to Tusset (2008).

The quarter-car model is based on isolating a quarter of the vehicle and performing the isolated analysis of this section, and assuming that the suspensions of the other wheels will exhibit the same dynamics, in this way, one can significantly decrease the equation that describes the problem. However, there are limitations to this model, according

to Torres (2016), this isolates the suspension of the effects of other parts of the vehicle on it, and it is not possible to analyze the pitch and roll movements. Through Fig. 1 it is possible to analyze the representation of this model.
According to Gaspar *et al.* (2003), the spring k_S and the conventional damper b_S components showed in Fig.1, have linear and nonlinear components, in which they are described by Eq. (1) and Eq. (2).

$$k_s(x_w - x_c) = k_s^l\left(x_w - x_c\right) + k_s^{nl}(x_w - x_c)^3 \tag{1}$$

Where:
k_s^l : Coefficient representing the linear actuation range of the spring;
k_s^{nl} : Coefficient representing the nonlinear actuation range of the spring;

$$b_s(\dot{x}_w - \dot{x}_c) = b_s^l(\dot{x}_w - \dot{x}_c) - b_s^y\left|\dot{x}_w - \dot{x}_c\right| + b_s^{nl}\sqrt{\left|\dot{x}_w - \dot{x}_c\right|}\,\mathrm{sgn}(\dot{x}_w - \dot{x}_c) \tag{2}$$

Where:
b_s^l : Coefficient that affects damping force linearly;
b_s^{nl} : Coefficient that acts nonlinearly on the damper;
b_s^y : Coefficient that represents the characteristics of the asymmetric behavior of the damper.

Applying Newton's second law and considering as initial condition the static equilibrium in Fig. 1, it was derived the equations of motion for the quarter-car model according to Tusset (2008), since Eq. (3) is derived from the sprung mass, and Eq. (4) from the unsprung mass:

$$\begin{aligned} m_S.\ddot{x}_C &= b_s^l(\dot{x}_w - \dot{x}_c) - b_s^y\left|\dot{x}_w - \dot{x}_c\right| + b_s^{nl}\sqrt{\left|\dot{x}_w - \dot{x}_c\right|}\,\mathrm{sgn}(\dot{x}_w - \dot{x}_c) + \ldots \\ & k_s^l\left(x_w - x_c\right) + k_s^{nl}(x_w - x_c)^3 - F \end{aligned} \tag{3}$$

$$\begin{aligned} m_u.\ddot{x}_w &= -b_s^l(\dot{x}_w - \dot{x}_c) + b_s^y\left|\dot{x}_w - \dot{x}_c\right| - b_s^{nl}\sqrt{\left|\dot{x}_w - \dot{x}_c\right|}\,\mathrm{sgn}(\dot{x}_w - \dot{x}_c) - \ldots \\ & k_s^l\left(x_w - x_c\right) - k_s^{nl}(x_w - x_c)^3 - k_t\left(x_w - x_r\right) + F \end{aligned} \tag{4}$$

Where:
m_S : Chassis weight;
m_u : Mass of the wheel axle;
k_t : Represents the tire as a bundle of springs;
x_r : Vertical Tire Movements;
x_w : Vertical movements of the wheel axle;
x_C : Vertical chassis movements;
F : Force applied by magnetorheological damper.

The parameters of the vehicle suspension system treated in this work are presented in Tab. 1.

Table 1. Parameters to suspension system. Adapted from Gaspar *et al.* (2003)

m_s	m_u	b_s^l	b_s^{nl}
290 kg	40kg	700 Ns/m	200 Ns/m
b_s^y	k_s^l	k_s^{nl}	k_t
400 Ns/m	235.102 N/m	235.104 N/m	190.103 N/m

The controllable actuator used is the magnetorheological damper which, according to Carneiro (2009), presents highly reliable operating characteristics, adaptability to various processes and a low implementation cost, in this way, being applied in several areas such as: automotive mechanical engineering, civil engineering, medical engineering and others.

The force represented by F in Fig.1 is given by the action of this actuator, being represented by the LuGre model according to Eq. (5).

$$F = \sigma_a.z + \sigma_0.z.v + \sigma_1.\dot{z} + \sigma_2.\dot{x} + \sigma_b.\dot{x}.v \tag{5}$$

Where:

F : Total force applied by the magnetorheological damper [N];

v : Electrical voltage applied to the magnetic field excitation coil of the magnetorheological damper [V];

$\dot{x}$: Piston speed [m/s];

σ_0 : Rigidity of z influenced by v being equal to 8.10^5 [N/m.V];

σ_1 : Damping coefficient of z equal to $1,6.10^3$ [N.s/m];

σ_2 : Viscous damping coefficient equal to $1,5.10^2$ [N.s/m];

σ_a : Rigidity of z equal to 4.10^5 [N/m];

σ_b : Viscous damping coefficient influenced by v equal to 8.10^2 [N.s/m.V];

a_0 : Constant value equal to 3.10^{-3} [V/N];

z : Variable responsible for the representation of the hysteresis, being determined by Eq. (6).

$$\dot{z} = (\dot{x}_C - \dot{x}_w) - \sigma_0.a_0.|\dot{x}_C - \dot{x}_w|.z \tag{6}$$

where $\dot{x}_C$ and $\dot{x}_w$ are respectively the chassis displacement speed and the displacement speed of the axle of the wheel. The voltage v in Eq. (5) is obtained through the action of the Skyhook controller. This controller was proposed by Karnopp *et al.* (1974) initially with the purpose of reducing the vibration of the chassis, thus allowing a greater comfort to the passengers, however according to Sá (2006) this can also be used as a reference for comparison of more sophisticated controls. This controller is based on considering that sprung mass m_S , or in other words the mass of the chassis has the inertial frame in the sky, through the approach of a fictitious damper as visualized through Fig. 2a. However, this fictitious approach cannot be implemented physically, then it is necessary to use a controllable damper that can cause the system the same effect of the fictitious damper as shown in Fig. 2b. Through this real approach, the skyhook controller intends to propose, according Tusset (2008), greater passenger comfort compared to the passive suspension system.

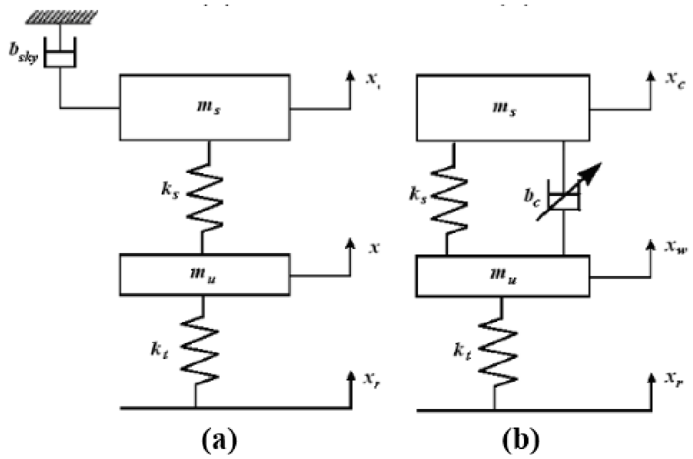

Figure 2. Approach of the damper to skyhook controller. (a) Fictional approach. (b) Real Approach according to Tusset (2008)

According to Paré (1998), the logic that describes the operation of the Skyhook controller can be written according to Eq. (7):

$$u = \begin{cases} b_{sky}.|\dot{x}_c| & se, \quad \dot{x}_c\left(\dot{x}_c - \dot{x}_w\right) \geq 0 \\ 0 & se, \quad \dot{x}_c\left(\dot{x}_c - \dot{x}_w\right) < 0 \end{cases} \tag{7}$$

where u is the control signal applied to the suspension system and b_{sky} is an empirical gain related to the magnitude of the absolute velocity of the chassis mass.

2. COMPUTATIONAL PROCEDURE

The Fig. 3 shows the quarter-car model, considering the action of the controllable actuator in parallel with the conventional damper.

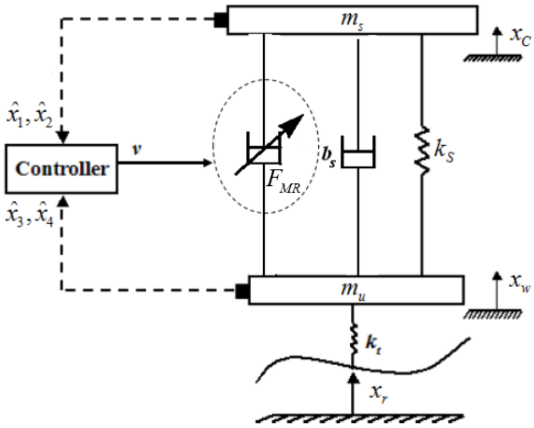

Figure 3. A quarter-car model considering the actuator magnetorheological damper in parallel with the conventional damper.

To analysis and design of the Skyhook controller, the system described by Eq. (8) must be considered.

$$
\begin{aligned}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= -\frac{k_s^l}{m_s}x_1 - \frac{b_s^l}{m_s}x_2 + \frac{k_s^l}{m_s}x_3 + \frac{b_s^l}{m_s}x_4 + \frac{k_s^{nl}}{m_s}(x_3 - x_1)^3 - \frac{b_s^y}{m_s}|x_4 - x_2| + \\
&\quad \frac{b_s^{nl}}{m_s}\sqrt{|x_4 - x_2|}\,\mathrm{sgn}(x_4 - x_2) - \frac{1}{m_s}.F_{MR} \\
\dot{x}_3 &= x_4 \\
\dot{x}_4 &= \frac{k_s^l}{m_u}x_1 + \frac{b_s^l}{m_u}x_2 - \frac{\left(k_s^l + k_t\right)}{m_u}x_3 - \frac{b_s^l}{m_u}x_4 - \frac{k_s^{nl}}{m_u}(x_3 - x_1)^3 + \\
&\quad \frac{b_s^y}{m_u}|x_4 - x_2| - \frac{b_s^{nl}}{m_u}\sqrt{|x_4 - x_2|}\,\mathrm{sgn}(x_4 - x_2) + \frac{k_t}{m_u}w + \frac{1}{m_u}.F_{MR} \\
\dot{x}_5 &= (x_2 - x_4) - \sigma_0.a_0.|x_2 - x_4|.x_5
\end{aligned}
\tag{8}
$$

Where: $x_1 = x_c$, $x_2 = \dot{x}_c$, $\dot{x}_2 = \ddot{x}_c$, $x_3 = x_w$, $x_4 = \dot{x}_w$, $\dot{x}_4 = \ddot{x}_w$, $\dot{x}_5 = \dot{z}$, $w = x_r$ and $F_{MR} = F$.

In Eq. (8), it can be verified that the force of actuation of the magnetorheological damper is given by F_{MR}, being described by Eq. (5). In order to simulate the Skyhook controller, it is necessary that the controller estimate the voltage value *v* that will be applied to the actuator coil in order to obtain a force F_{MR}, which is equivalent to the force *u* calculated by the skyhook controller (Tusset and Balthazar, 1998).

The voltage that is applied to the coil of the MR damper varies from 0 to 5 V, thus it is necessary to find a value in this range that satisfies the condition *F* = *u*. To find these values, the bisection method was used, in which, according to Gonçalves (2017), this is an effective method for measuring the signal bands that will be applied to the coil of the MR damper.

3. RESULT AND DISCUSSION

Figure 4 shows the response of the skyhook controller (red solid line) compared to the passive system response (black dashed line) for the displacement x_C, acceleration $\ddot{x}_C$ and jerk $\dddot{x}_C$ of the mass of the chassis, and the displacement of the mass of the wheel axis x_w.

The excitation considered for these simulations can be seen from Eq. (9).

$$w(t) = \begin{cases} 0, & if \quad t < 2 \\ 0{,}10, & if \quad t \geq 2 \end{cases} \tag{9}$$

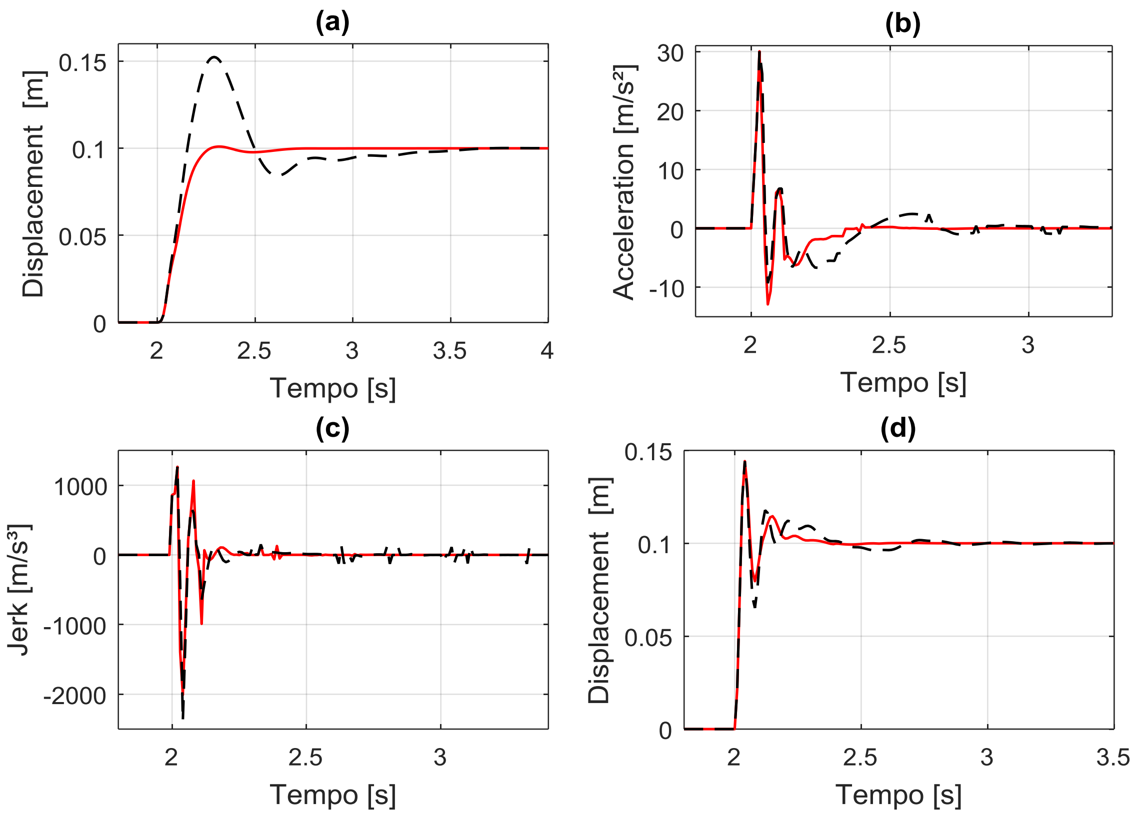

Figure 4. Comparison of the dynamics of the passive system and controllable system through the Skyhook controller. (a) Displacement of the chassis mass. (b) Acceleration of the mass of the chassis. (c) Jerk of the chassis mass. (d) Mass displacement of the wheel axle

In order to make the analysis of the performance of the skyhook controller easier in comparison to the passive system, in Fig. 5 a graph of grouped columns is displayed showing, in percentage, the contribution that the controller offered for each analyzed parameter of the system.

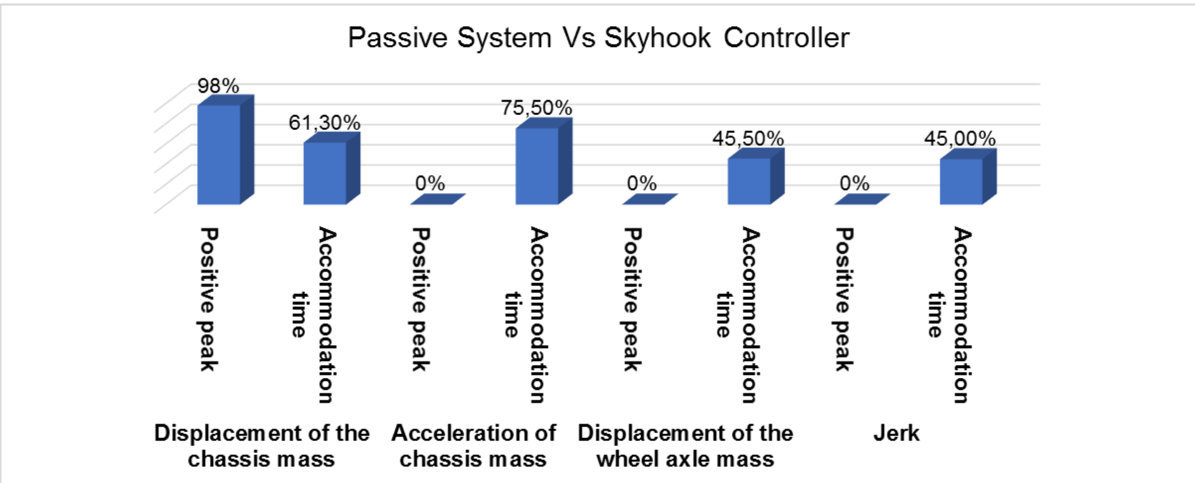

Figure 5. Contribution of the skyhook controller to the reduction of the analyzed parameters, considering the dynamics of the magnetorheological damper

It can be stated from Fig. 4 and Fig. 5 that the skyhook controller was effective in reducing the chassis mass displacement amplitude, as well as reducing the acceleration and jerk accommodation time for the chassis. These data show that the controller was able to generate levels of comfort superior to that of the passive system. It is important to note that although the controller was unable to reduce the acceleration and jerk peaks, it did not compromise passenger comfort, since the oscillations for the controlled system remained near to that of the uncontrolled system. Regarding the mass of the wheel axle, it is possible to affirm that the controller was unable to reduce the first peaks of displacement in relation to the passive system, however this was efficient in reducing the time of accommodation, thus adding a longer contact time between tire and lane, generating greater dirigibility to the vehicle.

4. CONCLUSION

In this work the skyhook controller was used in a nonlinear vehicle suspension system model considering as actuator a magnetorheological damper. The purpose of this control technique was to analyze if the controller was able to reduce the displacement amplitudes of the chassis and the wheel axis, as well as to reduce in the levels acceleration of the chassis mass and the time of accommodation of these parameters. As it can be seen, the controller has been efficient, thus being able to raise the levels of comfort and dirigibility of the vehicle in comparison to the passive system.

5. REFERENCES

Carneiro, R. B., 2009, "Controlo Semi-ativo de Vibrações em Estruturas Utilizando Amortecedore Magnetorreológico", Universidade de Brasília. Brasília. 2009.

Do, A. L., Sename, O. and Dugard, L, 2010, "An lpv control approach for semi-active suspension", IEEE. American Control Conference (ACC), p. 4653-4658.

Gaspar, P., Sazaszi, I. and Bokor, J., 2003, "Active suspension design using linear parameter varying control", 2. ed., IJVAS, v. 1.

Gonçalves, M. A., 2017, "Controle SDRE Aplicado em Suspensão Veicular com Amortecedor Magneto-Reológico". Dissertação de Mestrado - Universisidade Tecnológica Federal do Paraná. Ponta Grossa, p. 138.

Hu, Y., Li, C. and Chen, M. Z., 2012, "Optimal control for semi-active suspension with inerter", IEEE. Control Conference (CCC), China, n. 32, p. 2301-2306.

Karnopp, D., Crosby, M. and Harwood, R. A., 1974, "Vibration control using semi-active force generators", Journal of engineering for industry, Transactions of the ASME, 187, p. 619-626.

Oliveira, K. F., 2015, "Controlo semi-ativo da suspensão de um veículo automóvel", Dissertação de Mestrado em Engenharia Industrial - Escola Superior de Tecnologia e Gestão Insituto Politécnico de Bragança. Bragança, p. 106.

Paré, C. A., 1998, "Experimental Evalution of Semi-ative Magneto-Rheological Suspensions for Passenger Vehicles", Thesis for the degree of Master of Science in Mechanical Engineering - Virginia Polytechnic Institute and State University. Virginia.

Picado, R. M, 1998, "Controle Semi-Ativo de Suspensões Automotivas", Dissertação de Mestrado - Universidade Estadual de Campinas, Campinas.

Torres, T. R., 2016, "Controle preditivo aplicado a um modelo não linear de suspensão automotiva semiativa com amortecedor magneto-reológico", Universidade Federal da Bahia. Salvador, p. 201.

Tusset, A. M. and Balthazar, J. M., 2013, "On the chaotic suppression of both ideal and non-ideal duffing based vibrating systems, using a magnetorheological damper". Differential Equations and Dynamical Systems, p. 1-17.

Tusset, M, 2008, "Controle Ótimo Aplicado em Modelo de Suspensão Veicular Não-Linear Controlada Através de Amortecedor Magneto-Reológico", Escola de Engenharia da Universidade Federal do Rio Grande do Sul, Porto Alegre, p. 181.

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