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FUZZY CONTROL STRATEGY OF A NONLINEAR FULL-CAR MODEL USING A MEGNETHORHEOLOGICAL (MR) DAMPER

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Abstract. *This paper presents the control strategy of a nonlinear full-car model using a magnetorheological (MR) damper. A two step design approach is used for modeling and control of the mechanical and electrical parts of the suspension system with the MR damper. Firstly, the fuzzy control problem is formulated in order to design the damping force control signal for a nonlinear suspension system. Then the values of the control damping force functions are transformed into electrical control signals. The numerical simulations show the effectiveness of this method for the semi-active control of the full-car suspension.*

Keywords: *Fuzzy Control, Full-Car, MR Damper, Vehicular Suspension*

1. INTRODUCTION

To improve the ride quality, the vibrations of a vehicle body should be reduced. In recent years, much attention has been paid to the design of control techniques of the vehicle suspension systems in order to reduce the vibrations (Liu et al., 2008). Three types of vibration control methods have been proposed and implemented successfully, namely, the passive, the active and the semi-active method for control suspension systems (Sun et al., 2005).

The vehicle suspension is one of the most crucial systems in vehicles, which connects the wheels to the upper body (sprung mass) of the vehicle. It is composed of springs and dampers that can be controlled in order to manage handling, safety, and comfort for the passengers (Laoufi and Eskandarian, 2009).

More recently, magnetorheological (MR) fluids, which can change their viscosity significantly on the application of suitable magnetic fields, respectively, have also been used in vehicle suspensions as a semi-active suspension system by a number of researchers (Ahmadian and Pare, 2000; Yao et al. 2002).

This paper presents the fuzzy control strategy of a nonlinear full-car model using a magnetorheological (MR) damper. The fuzzy control problem is formulated in order to design the damping force control signal for a nonlinear suspension system. Then the values of the control damping force functions are transformed into electrical control signals using fuzzy control.

The MR damper (u_{nm} , n = front or rear, m=right or left) is considered in parallel with the passive suspension (Fig. 1). In order to compare the performance of the proposed control with the performance of the passive suspension is considered the introduction of a MR damper with maximum force of 100 [N/m/s] and a damper with MR maximum force of 600 [N/m/s].

Figure 1 shows the nonlinear full-vehicle model used in this work: the vehicle body is represented by a seven degree-of-freedom rigid cuboid with mass m_s. The heave, pitch and roll motions of the sprung mass are considered. The four unsprung masses (front-left, front-right, rear-left and rear-right) are connected to each corner of the rigid cuboid. It is assumed that the four unsprung masses are free to bounce vertically. The suspensions between the sprung mass and unsprung masses are modeled as nonlinear spring and nonlinear dampers elements, while the tires as modeled as linear springs.

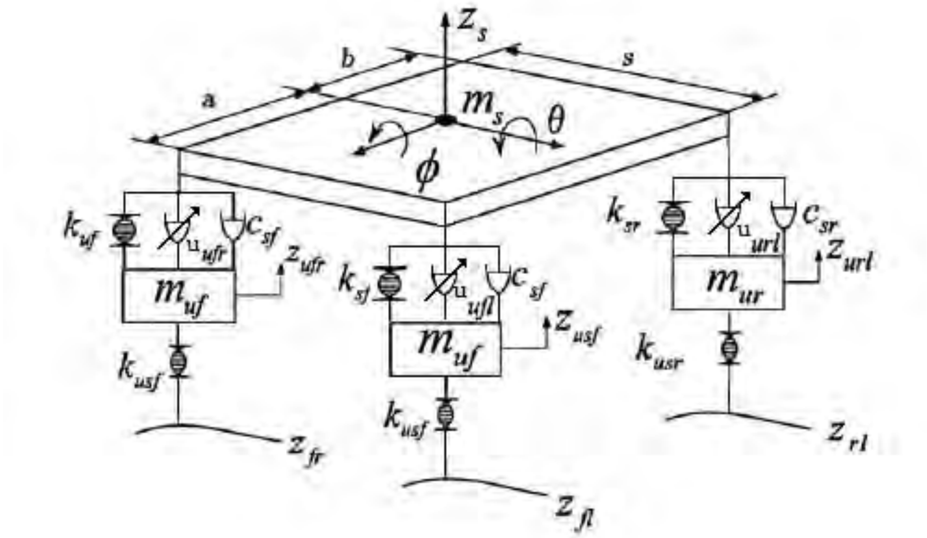

Figure 1. Nonlinear full-car model (Zhu and Ishitobi, 2006).

2. MATEMATICAL MODEL

The nonlinear spring forces of front and rear suspensions are given by (Gaspar et al., 2003):

$$Fs_{nm} = K_{nm}^{l} u_{nm} + K_{nm}^{nl} (u_{nm})^3 \tag{1}$$

where K_{nm}^{l} is the linear coefficient stiffness, K_{nm}^{nl} is the nonlinear coefficient stiffness, u_{nm} is the deformation of the spring, and (n = front or rear, m=right or left).
The nonlinear damping forces of front and rear suspensions are given by (Gaspar et al., 2003):

$$Fc_{nm} = C_{nm}^{l} \dot{u}_{nm} - C_{nm}^{y} |\dot{u}_{nm}| + C_{nm}^{nl} \sqrt{|\dot{u}_{nm}|}\, \mathrm{sgn}(\dot{u}_{nm}) \tag{2}$$

where $\dot{u}_{nm}$ is the relative velocity between the extremes of the damper, C_{nm}^{l} coefficient effects the damping force linearly while C_{nm}^{nl} has nonlinear impact on the damping characteristics, C_{nm}^{y} describes the asymmetric behavior of the characteristic and $\mathrm{sgn}(\dot{u}_{nm})$ is the signal function.
The tire stiffness can be considered as a linear spring so the tire force is:

$$Ft_{nm} = K_{nm}^{t} v_{nm} \tag{3}$$

where v_{nm} is the deformation of the tire and K_{nm}^{t} is the linear coefficient stiffness.

Considering the vertical force, pitching force and roll force acting on sprung mass, moreover the force acting on unsprung masses, and using the nonlinear characteristics of springs and dampers, the force balance equations acting on suspension, and the accelerations are formalized in the following.

Sprung mass:

$$\begin{aligned}
\ddot{Z}_s &= \left(\frac{1}{m_s}\right)\left(-Fs_{fl} - Fc_{fl} - Fs_{fr} - Fc_{fr} - Fs_{rl} - Fc_{rl} - Fs_{rr} - Fc_{rr}\right) \\
\ddot{\phi} &= \left(\frac{1}{I_\phi}\right)\left(\frac{s}{2}\cos(\phi)\right)\left(-Fs_{fl} - Fc_{fl} + Fs_{fr} + Fc_{fr} - Fs_{rl} - Fc_{rl} + Fs_{rr} + Fc_{rr}\right) \\
\ddot{\theta} &= \left(\frac{1}{I_\theta}\right)\left(a\cos(\theta)\left(Fs_{fl} + Fc_{fl} + Fs_{fr} + Fc_{fr}\right) - b\cos(\theta)\left(Fs_{rl} - Fc_{rl} + Fs_{rr} + Fc_{rr}\right)\right)
\end{aligned} \tag{4}$$

Front unsprung masses:

$$\ddot{Z}_{ufl} = \left(\frac{1}{m_{uf}}\right)\left(Fs_{fl} + Fc_{fl} - Ft_{fl}\right)$$
$$\ddot{Z}_{ufr} = \left(\frac{1}{m_{uf}}\right)\left(Fs_{fr} + Fc_{fr} - Ft_{fr}\right) \quad (5)$$

Rear unsprung masses:

$$\ddot{Z}_{url} = \left(\frac{1}{m_{ur}}\right)\left(Fs_{rl} + Fc_{rl} - Ft_{rl}\right)$$
$$\ddot{Z}_{urr} = \left(\frac{1}{m_{ur}}\right)\left(Fs_{rr} + Fc_{rr} - Ft_{rr}\right) \quad (6)$$

3. MAGNETORHEOLOGICAL (MR) DAMPER

The mechanism of the magnetorheological damper is similar to the mechanism of the hydraulic shock absorbers which force is caused due to the passage of fluid through an orifice. This variable resistance to fluid flow allows using the magnetorheological fluid in viscous and other electrically controllable devices. Thus the magnetic properties of the fluid allow its use as a damper controlled by an electric voltage [volts] or an electric current [Amps] (Tusset et al., 2013).

Figure 3 shows the characteristics of change of damping force of a damper as a function of the speed and electrical current applied to the coil of the damper. The operating field indicates the damping force can be changed by changing the velocity of the damper rod or the electric current applied to the coil. Due to the rheological characteristics of the materials there is a saturation point of force in relation to the speed of displacement of the damper rod, without major changes to speeds higher than 0.4 [m /s]. Saturation can also occur in relation to current applied to the coil, increasing the electric current and not increasing the damping force for a given speed, being used between 0 current [Amps] and 1.5 [Amps]. Using a special control system and considering saturation levels may be required to have some characteristic curves of the damping force that can be generated as a function of speed, as the curves of Figure 3 (McManus et al., 2002).

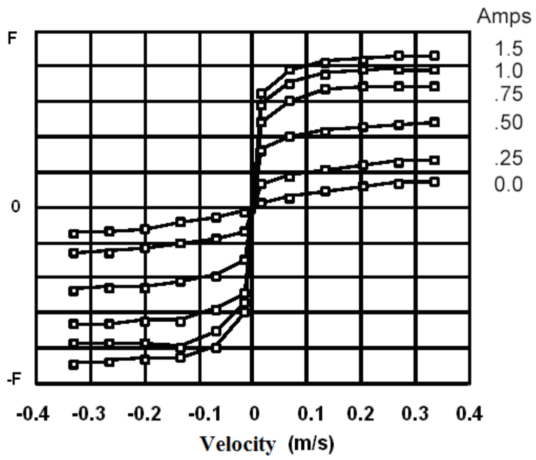

Figure 2. Force–velocity characteristics of a MR fluid damper as a function of the coil current.

In Figure 3 it can be observed variation characteristics of damping force depending on the speed of displacement of the damper piston and electric current applied to the coil. One way to determine the electric current applied to the coil is via Fuzzy Control (Tusset el al., 2009).

In Figure 3a, 3b and 3c it can be observed membership functions for velocity [m/ s], force [N] and current [Amps] as well as the type of membership function and its superposition (Tusset el al., 2009).

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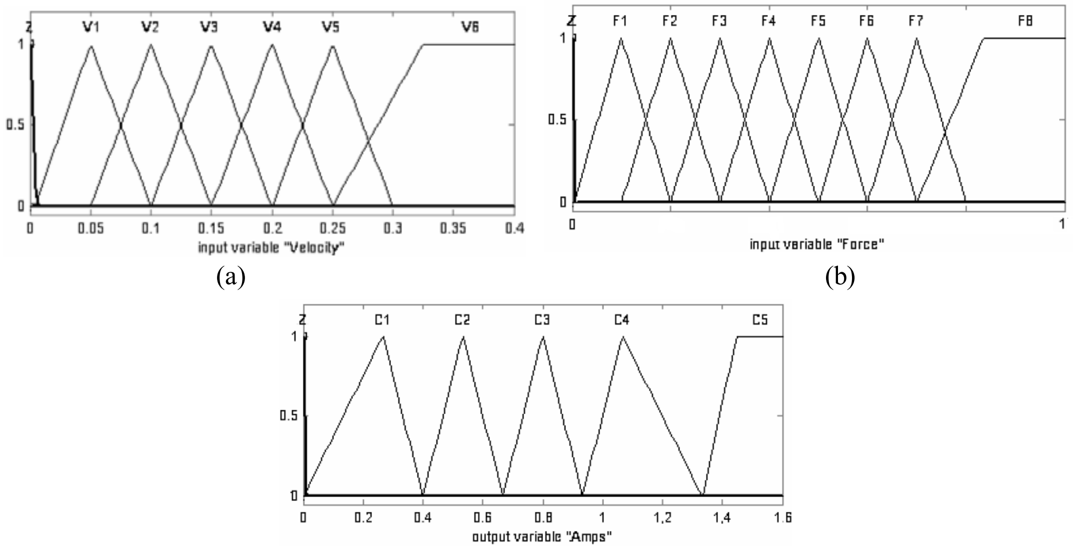


(a) (b)

(c)

Figure 3. (a) Input membership function (Velocity). (b) Input membership function (Force). (c) Output membership function (Amps).

For calculating the defuzzification was defined minimum for the method (and) maximum for the method (or) implication minimum, maximum aggregation, and performing the defuzzification (centroid).

The consequent application in the fuzzy control will be obtained as follows: **If (Velocity is ...) and (Force is ...) then (Amps is ...)**. The rules to be applied can be seen in Table 1.

Table 1. Map of fuzzy rules for the current control (Tusset el al., 2009).

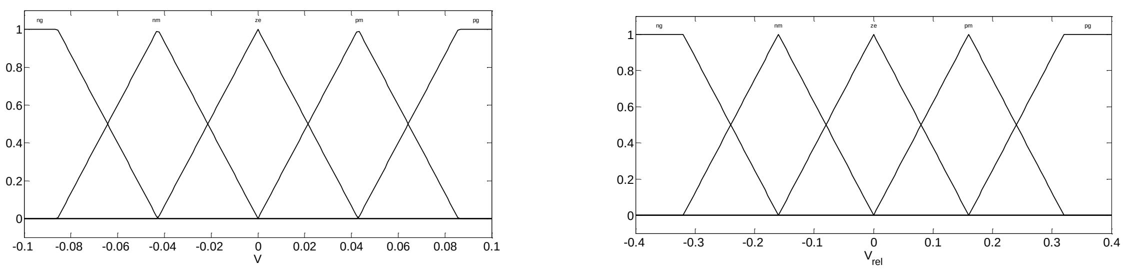
VELOCITY

FORCE

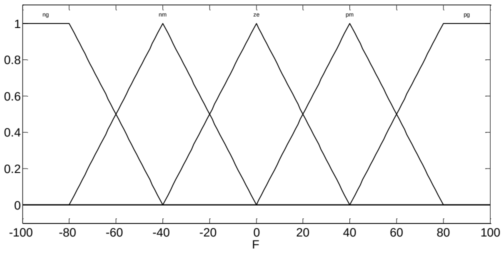
Amps	Z	V1	V2	V3	V4	V5	V6
Z	Z	Z	Z	Z	Z	Z	Z
F1	C1	Z	Z	Z	Z	Z	Z
F2	C1	C1	C1	C1	C1	C1	C1
F3	C2	C2	C2	C1	C1	C1	C1
F4	C3	C2	C2	C2	C2	C2	C2
F5	C4	C3	C2	C2	C2	C2	C2
F6	C5	C4	C3	C3	C3	C3	C3
F7	C5	C5	C4	C4	C3	C3	C3
F8	C5	C5	C5	C5	C5	C5	C5

4. FUZZY CONTROL

For determination of the proposed fuzzy control was used the classical Mamdani, with five membership functions with a symmetrical and evenly spaced in order to smooth out any discontinuities of strength and minimize problems such as bumps and noises. The choice of the membership functions (velocity) and (relative velocity) is due to the fact that the relationship between them to determine the level of damping between m_s and m_u (Pinheiro, 2004; Tusset et al. 2006). Figures 4a, 4b and 4c are defined membership functions relating to velocity, relative velocity and actuation force u_{nm}.



(a) (b)



(c)

Figure 4. (a) Input membership function (Velocity). (b) Input membership function (Relative Velocity). (c) Output membership function (Force).

The rules applied on the control can be seen in Table 2.

Table 1. Map of fuzzy rules for the force control (Pinheiro, 2004).

F		***V***				
		NG	*NM*	*ZE*	*PM*	*PG*
V_{rel}	*NG*	NG	NG	NM	NM	NM
	NM	NG	NM	NM	NM	NM
	ZE	NM	NM	ZE	PM	PM
	PM	PM	PM	PM	PM	PG
	PG	PM	PM	PM	PG	PG

5. SIMULATION RESULTS

For numerical simulations we used the following values of the parameters (Gaspar et al., 2003; Zhu and Ishitobi, 2006):

Table 2. Numerical values of the system parameters.

Sprung mass, m_s	1500 [kg]
Roll axis moment of inertia, I_ϕ	460 [kg m]
Pitch axis moment of inertia, I_θ	2160 [kg m]
Front and rear unsprung mass, m_{unm}	40 [kg]
Front and rear suspension spring stiffness, K^l_{nm} ; K^{nl}_{nm}	10^4; 10^6 [N/m]
Front and rear suspension damping coefficient, C^l_{nm} ; C^{nl}_{nm} ; C^y_{nm}	1000; 400; 400 [N/m/s]
Tire spring stiffness, K^t_{nm}	190000 [N/m]
Length between the front of vehicle and the center of gravity of sprung mass, **a**	1 [m]
Length between the rear of vehicle and the center of gravity of sprung mass, **b**	1.5 [m]
Width of sprung mass, **s**	1.5 [m]

The impulse displacement function ($\delta(t) = 0.2m$) is used to describe the excitation caused by road surface. Thus the forcing function to tires in front-right, front-left, rear-right and rear-left are approximated by:

$$Z_{fl} = \delta(0) = 0.2m \tag{7}$$

$$Z_{fr} = \delta(0.05) = 0.2m \tag{8}$$

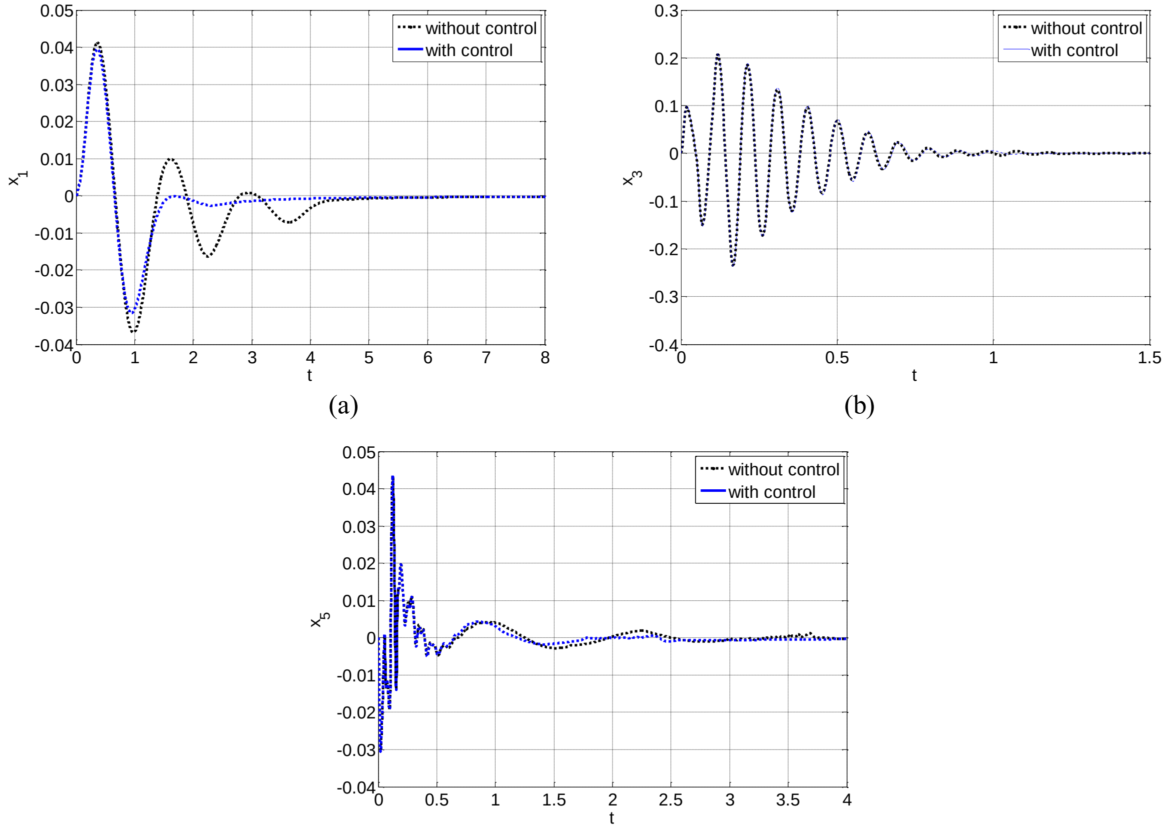
$$Z_{rl} = \delta(0.1) = 0.2m \tag{9}$$

$$Z_{rr} = \delta(0.15) = 0.2m \tag{10}$$

5.1 Control of MR damper with a 100 [N/m/s]

The objective using a reduced damper in parallel with the passive suspension is to analyze the possibility to improve the comfort of passengers without reducing the stability of the vehicle with reduced implementation costs.

In Figure 5 we can observe the displacements of the sprung mass:

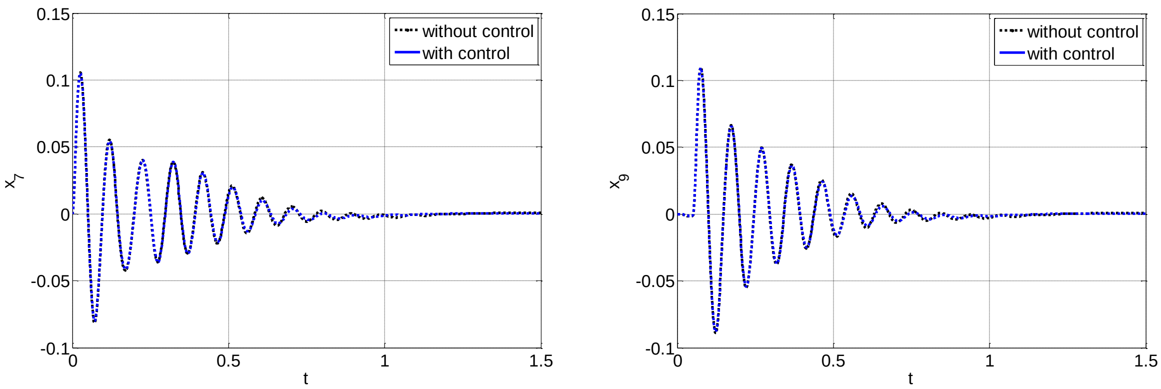


(a) (b)

(c)

Figure 5. Sprung mass: (a) Vertical displacement. (b) Pitching movement. (c) Roll movement.

As can be seen, the introduction of damper resulted in a gain in comfort by reducing the displacement of the vehicle body. Figure 6 shows the displacements of the front unsprung masses.



(a) (b)

Figure 6. Displacement of the front unsprung masses. (a) Left. (b) Right.

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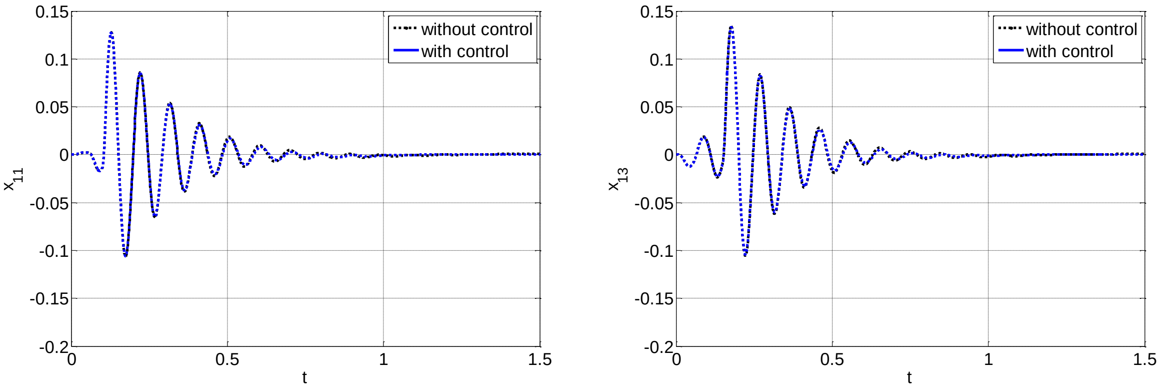
Figure 7 shows the displacements of the rear unsprung masses.

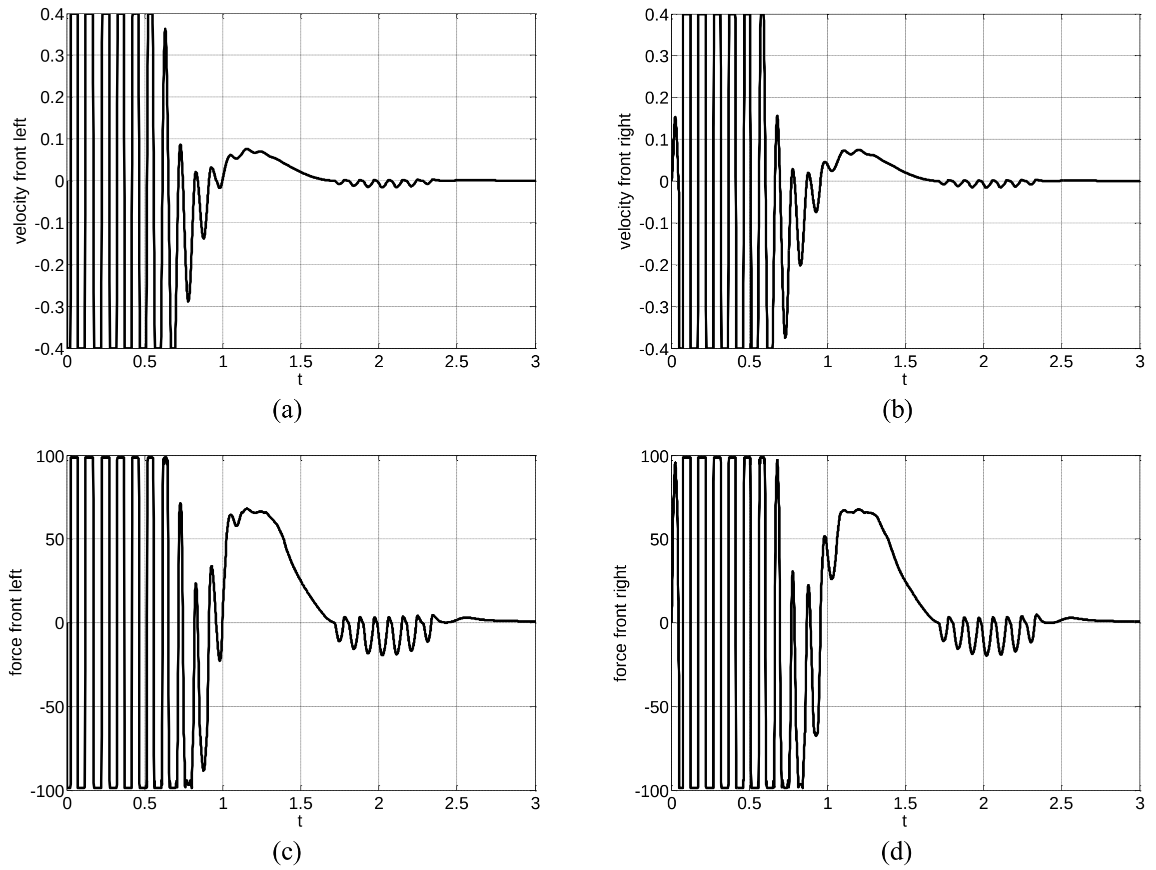


(a) (b)

Figure 7. Displacement of the rear unsprung masses. (a) Left. (b) Right.

As can be seen in Figures 6 and 7 due to the reduced size of the damper, the control does not change the displacement of unsprung masses, being appointed to ensure the comfort of passengers it has not reduced the stability of the vehicle.

In Figure 8 we can observe the values used in the fuzzy control as much force as the electrical current applied to the front suspension of the vehicle.



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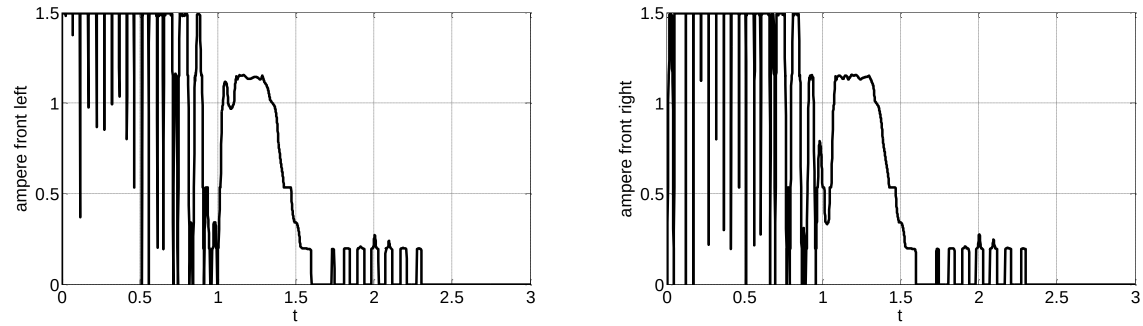
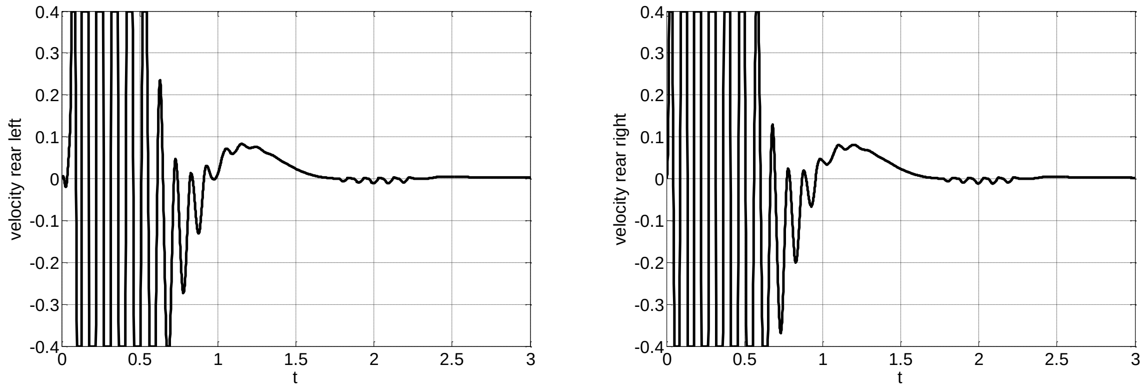


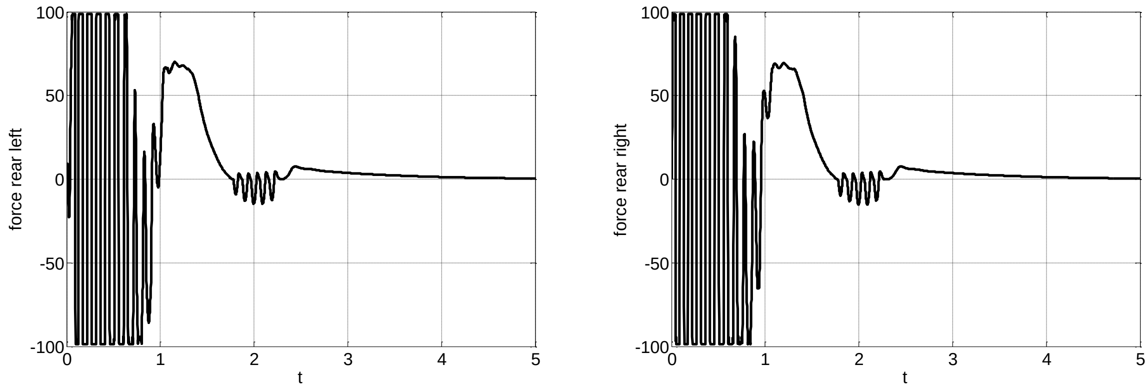
(e) (f)

Figure 8. Signs used in fuzzy control (Front).

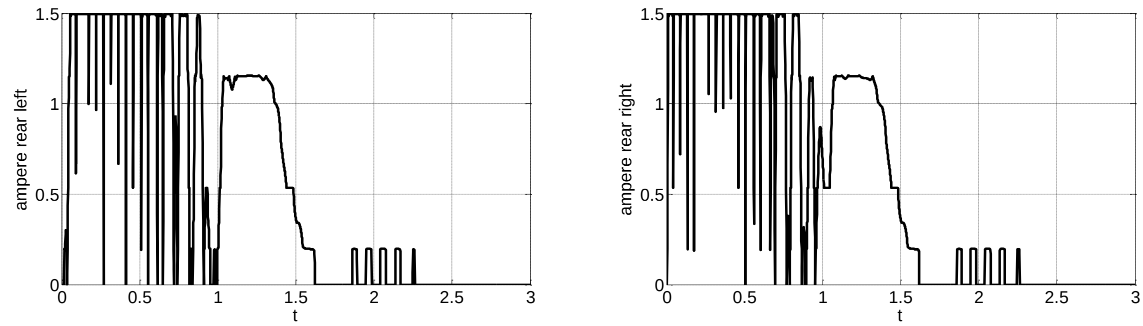
In Figure 8 we can observe the values used in the fuzzy control as much force as the electrical current applied to the rear suspension of the vehicle.



(a) (b)



(c) (d)



(e) (f)

Figure 9. Signs used in fuzzy control (Rear).

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5.2 Control of MR damper with a 600 [N/m/s]

For the cases where comfort and stability are a design requirement, it is necessary to use a larger damper capacity than previously used. Considering then a damper with 600 [N/m/s], and substituting in Figure 4c by Figure 10, we have a more robust control.

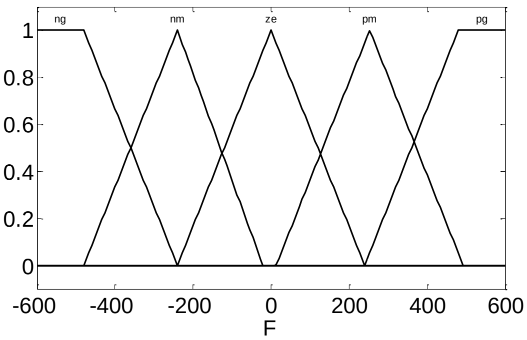


Figure 10. Output membership function (Force).

In Figure 11 we can observe the displacements of the sprung mass:

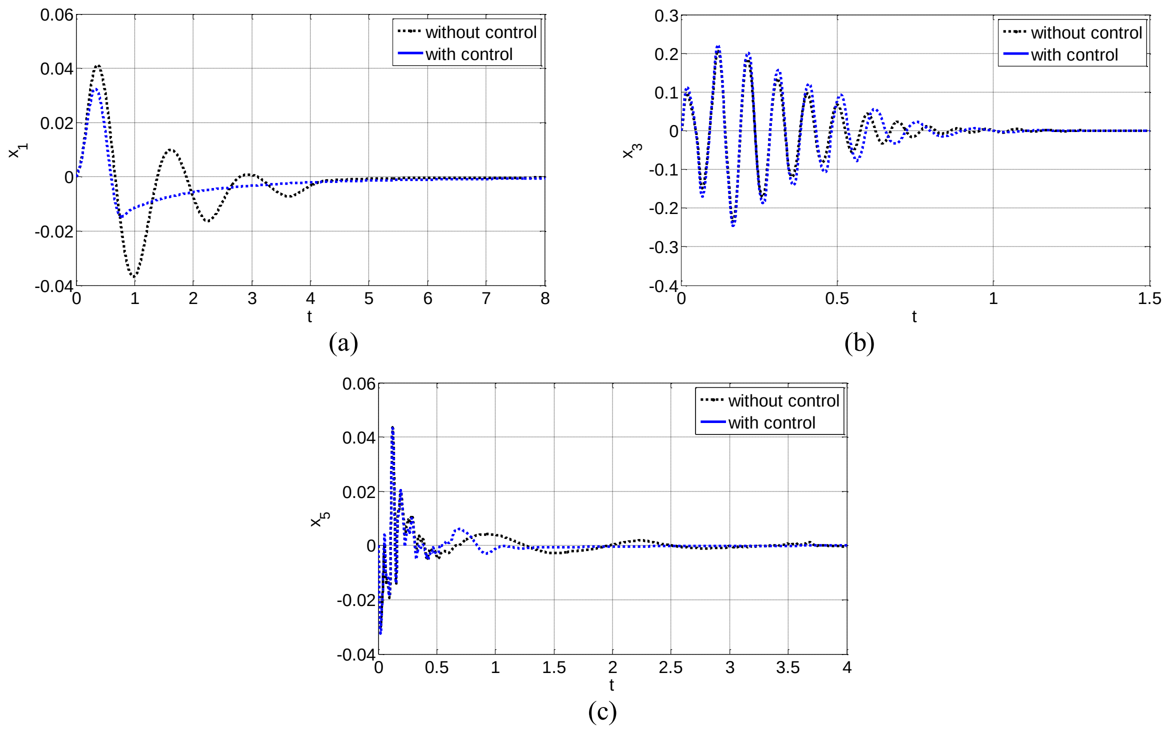
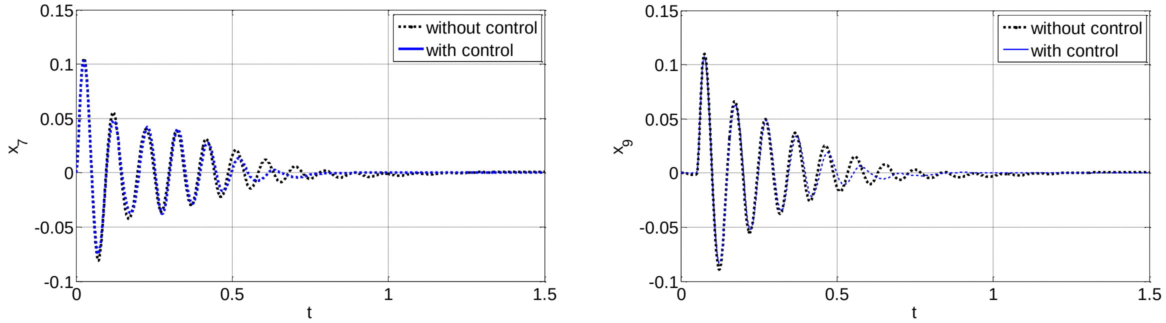


Figure 11. Sprung mass: (a) Vertical displacement. (b) Pitching movement. (c) Roll movement.

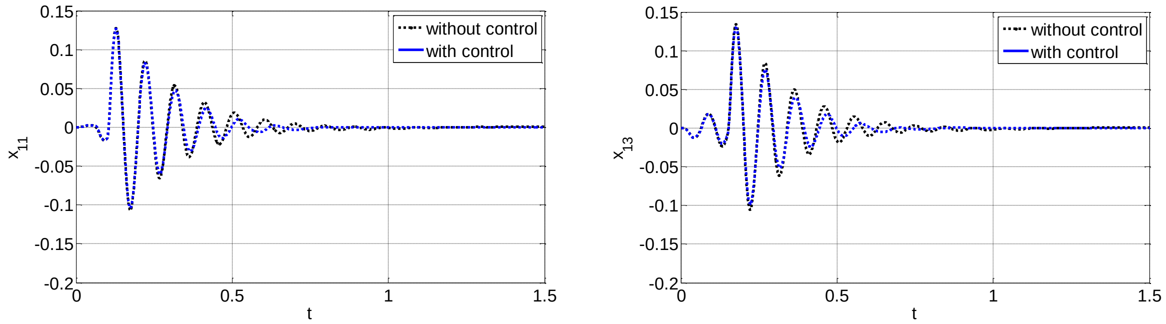
As can be seen, the introduction of damper resulted in a gain in comfort by reducing the displacement of the vehicle body. Figure 12 shows the displacements of the front unsprung masses.



(a) (b)

Figure 12. Displacement of the front unsprung masses. (a) Left. (b) Right.

Figure 13 shows the displacements of the rear unsprung masses.

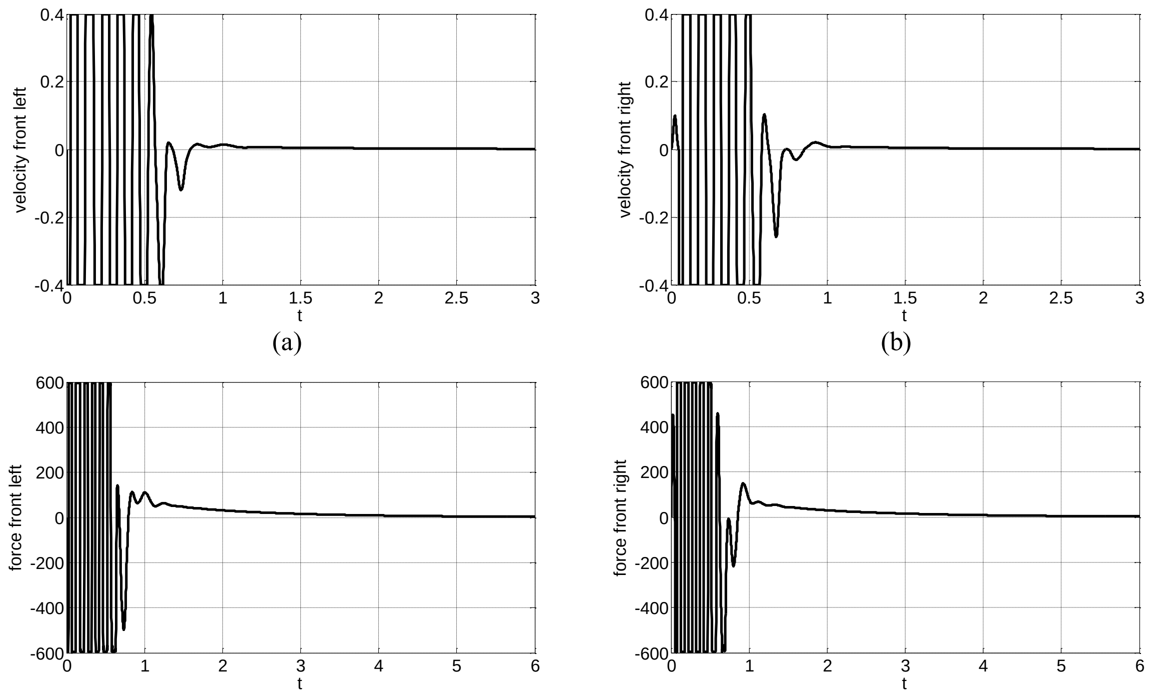


(a) (b)

Figure 13. Displacement of the rear unsprung masses. (a) Left. (b) Right.

As it can be seen in Figures 12 and 13 the damper improved the stability of the vehicle, reducing the displacement of the wheels.

In Figure 14 we can observe the values used in the fuzzy control as much force as the electrical current applied to the front suspension of the vehicle.



(c) (d)

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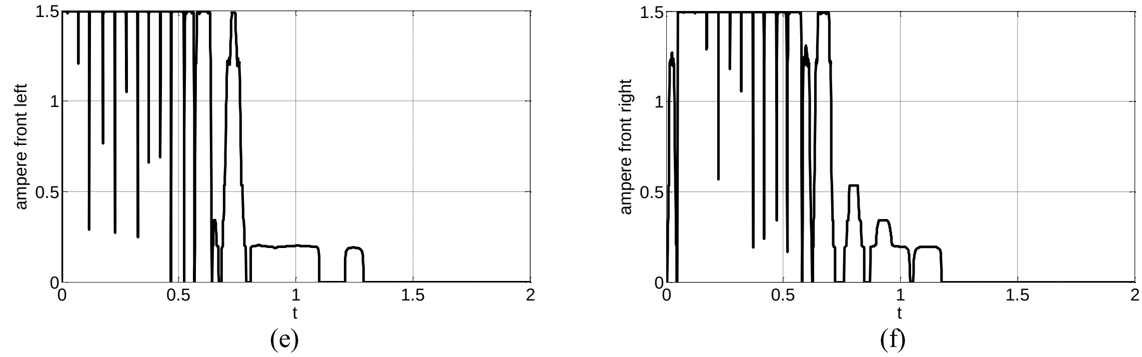

Figure 14. Signs used in fuzzy control (Front).

In Figure 15 we can observe the values used in the fuzzy control as much force as the electrical current applied to the rear suspension of the vehicle.

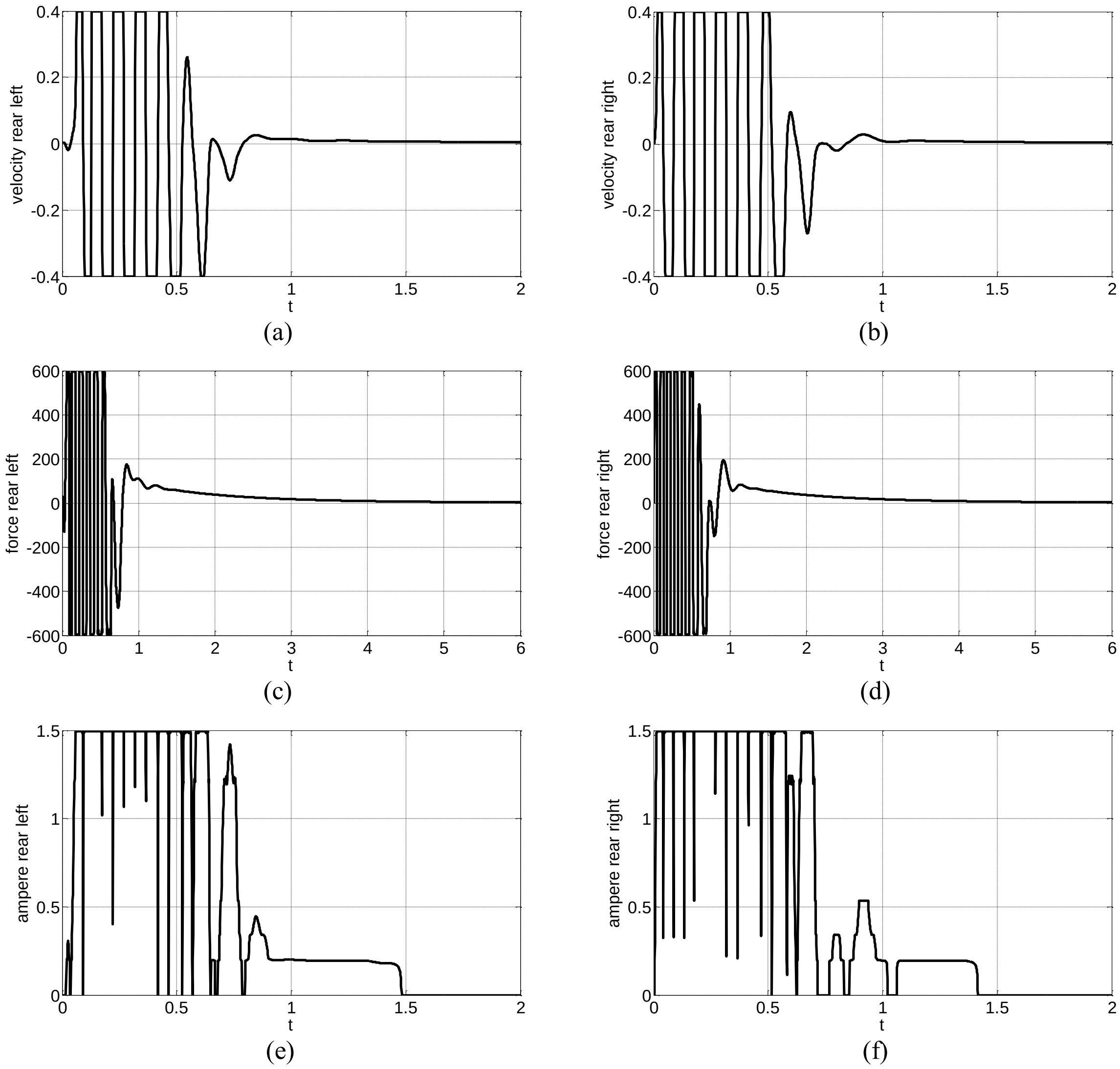

Figure 15. Signs used in fuzzy control (Rear).

6. CONCLUSIONS

The results demonstrate that the fuzzy control can control the active vehicle suspension damper using MR. Using a nonlinear seven degree-of-freedom ground model it is possible to obtain more realistic numerical results. The simulations show two possible control strategies: first saw the introduction of a MR damper with 14.29% of the linear component of the passive damper, and found a good choice for comfort at low cost, the second strategy showed the introduction a MR damper with 85.71% of the component passive linear damper, and found that control improves comfort and stability, but will result in a higher cost compared to the first strategy.

7. ACKNOWLEDGEMENTS

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