

# APPLICATION OF A DISTURBANCE OBSERVER BASED CONTROLLER TO IMPROVE VEHICLE RIDE

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In this paper the application of a disturbance observer based control algorithm is proposed to improve the vehicle ride performance of a vehicle that has magnetorheological damper (MR Damper) suspension. The vehicle is modeled as a quarter car model by employing friction Luge model of the MR damper. The proposed methodology is demonstrated on four different road profiles including step, ramp, bump, sine and rough roads as the disturbance. The controller performance of the proposed algorithm is benchmarked relative to the conventional PID controller on the basis of the rejection of the disturbance. The simulation results show that the DOB based controller outperforms the conventional PID when the control command signal is comparable to each other for all case studies. In addition, DOB can still reject the disturbance, while the conventional PID controller becomes unstable when the proportional gain is increased for better performance. The DOB controller performs quite well for the sine road profile with the excitation frequency close to the resonance frequency of the sprung mass as one of the most challenging case studies. The PID controller cannot suppress the road disturbance under such a condition.

Keywords: Vehicle ride, Quarter car model, Magnetorheological damper, Disturbance observer

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## 1. Introduction

Automotive OEMs spend considerable time and budget in order to optimize ride comfort as it is one of the most important vehicle dynamics attributes. Vehicle ride is closely tied to the vibration of vehicle body due to excitations from road and engine. There are many factors affecting vehicle ride such as road roughness, rotational unbalance of tire/wheel and stiffness variation of tire, driveline and engine as well as vehicle suspension systems.

Vehicle suspension system plays an important role on the vehicle ride and handling characteristics by providing vertical compliance and isolating the chassis from the road surface. There are mainly three types of suspension systems: i) passive, ii) active, and iii) semi-active systems [1]. Passive suspension systems consist of conventional springs and dampers and are extensively used in passenger cars [2]. On the contrary, the active suspension systems use an actuator in order to generate force on the suspension system to improve ride quality [3]. Energy dissipation characteristics of a semiactive suspension system can be made variable using a controllable damper [4]. Active and semiactive suspension systems have been extensively used as part of vehicle control devices. However, the use of semi-active suspension systems is more common than active systems in automotive applications [5].

There has been a vast body of literature on the modeling of MR dampers to predict and improve the dynamics of mechanical systems. During the past decades, various types of mathematical models have been developed. The most commonly used models in the literature are Bouc-Wen model [6], modified Bouc-Wen model [7], Dahl model [8], and Lugre friction model [9]. Using these models, researchers proposed several control methods for MR dampers. One of the most applied control algorithm was  $H_\infty$  control ([10]-[13]). Other common control algorithms including  $L_1$  adaptive control [14], model predictive control [15], fuzzy logic control [16] have been demonstrated on vehicle applications in the literature. Despite their well-demonstrated robustness, the algorithmic complexity associated with these controllers may introduce computational burdens and practical issues. Therefore, we argue that a simplistic, yet robust controller should be considered.

To this end, disturbance observer based controllers appear to provide practical, simple and computationally affordable solutions while ensuring robustness [17,18]. With this goal in mind, the main objective of the paper is to evaluate the performance of the observer based control algorithm for MR damper control system in comparison to conventional PID controller regarding their performance in attenuation of undesired vibration for the driver. Visual comparison of the sprung mass displacement is used to compare the performance of the aforementioned control algorithms.

The paper is organized as follows: first, description of the mathematical models that will be used in representing the plant models is given in Section 2. Next, the proposed methodology of disturbance based controller is explained in Section 3. The methodology is demonstrated on five different road profiles and its performance is compared to conventional PID controller in Section 4. Finally, the paper is concluded in Section 5.

## 2. Description of the Mathematical Model

The use of quarter-car models (one DOF or two DOFs) in automotive product development cycle is common in order to investigate ride metrics, suspension packaging and dynamic load transfer under road excitation mainly due to its simplicity and reasonable representation of the actual system [19, 20].

The schematic model of the MR damper is shown in Fig. 1 as part of the quarter car model and represented as parallel to the suspension stiffness, represented by  $k_s$ . In this model, the sprung mass represents the vehicle body, while the unsprung mass includes the mass of wheel/tire assembly including suspension knuckle and spindle. A linear spring and viscous damping are used in order to model the forces acting between them. Similarly, tire is represented by a linear spring. In order to simplify the modeling, the tire damping is ignored. In the model,  $x_s(t)$ ,  $x_u(t)$  and  $x_d(t)$  are the displacements of the sprung mass, unsprung mass and road input (disturbance), respectively. Parameters of the quarter car model [21] are given in Table 1.

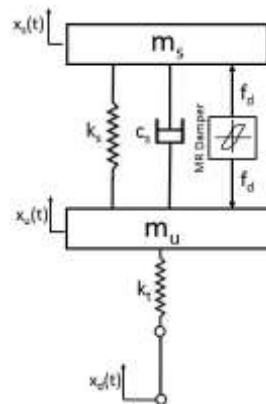


Figure 1: The schematic model of the quarter car and MR damper

Table 1: Parameters of the quarter car model [21]

Parameter	Parameter Value
Sprung Mass, $m_s$	267 kg
Suspension Stiffness, $m_s$	18,742 N/m
Suspension damping, $c_s$	700 N.s/m
Unsprung mass, $m_u$	36.6 kg
Tire Stiffness, $k_t$	193,915 N/m

The governing equations of motion for the linear quarter car model are given by Eq. (1) representing the sprung mass, and Eq. (2) representing the unsprung mass.

$$m_s \ddot{x}_s + k_s(x_s - x_u) + b_s(\dot{x}_s - \dot{x}_u) + f_d = 0 \quad (1)$$

$$m_u \ddot{x}_u - k_s(x_s - x_u) - b_s(\dot{x}_s - \dot{x}_u) - k_t(x_r - x_u) - f_d = 0 \quad (2)$$

The MR damper model is based on modified dynamic Lugre model due to its relatively simplified representation and its extensive use in the literature. The damper force is determined from Eq. (3) and Eq. (4).

$$f_d = \sigma_a z + \sigma_0 z V + \sigma_1 \dot{z} + \sigma_2 \dot{x} + \sigma_b \dot{x} V \quad (3)$$

$$\dot{z} = \dot{x} - a_0 |\dot{x}| \dot{x} + \sigma_b \dot{x} V \quad (4)$$

where  $x$  and  $\dot{x}$  are the displacement and velocity of the sprung mass relative to unsprung mass, respectively. The internal state  $z$  represents the fluid deformation and  $V$  is the input voltage to MR damper. The MR damper model is characterized by 6 parameters ( $\sigma_a$ ,  $\sigma_0$ ,  $\sigma_1$ ,  $\sigma_2$ ,  $\sigma_b$ ,  $a_0$ ) as listed in Table 2.

Table 2: Parameters of the MR damper model [13]

Parameters	Parameter Value
$\sigma_a$ (N/m)	10,000
$\sigma_0$ (N/mV)	320,000
$\sigma_1$ (Ns/m)	3.21
$\sigma_2$ (Ns/m)	1,153.3
$\sigma_b$ (Ns/mV)	34,622
$a_0$ (V/N)	840

### 3. Disturbance Based Observer Controller

First proposed by Ohnishi, a disturbance observer-based controller is a robust control method that effectively attenuate disturbances acting on the system using the inverse of the nominal plant model [17]. Despite the fact that it is a model-based method, it only needs the nominal plant and can reject disturbances due to modeling uncertainties [22]. Its effectiveness was proven for multi-DOF mass-spring-damper systems as well [18].

Fig. 2 displays the proposed DOB-based controller in which two cascaded PID controllers were employed. The inner loop regulates the MR damper force while the outer loop regulates  $x_2$  reference. From Eq. (1), the transfer function between  $f_d$  and  $x_s$  can be obtained via Laplace transformation as follows:

$$\frac{x_s(s)}{f_d(s)} = P_n(s) = \frac{1}{m_s s^2 + b_s s + k_s} \quad (5)$$

Therefore, one can compute  $x_s$ , given  $f_d$  and  $P_n(s)$ . However, due to unknown road profile, an inevitable disturbance  $d$  acts upon the output. If we can measure the output, we may observe  $x_s + d$ , while the model-based computation reveals only  $x_s$ , since  $x_s = P_n(s) f_d$ . The difference between these two

reveals the disturbance. Since the observed disturbance is in displacement level, we need to multiply it with  $P_n^{-1}(s)$  to compute the force variation that corresponds to  $d$ . Therefore, one can construct a DOB block based on the nominal quarter car model to attenuate disturbances caused by the unknown road profile. To do so, we make use of the inverse of the nominal plant  $P_n(s)$ . A 2<sup>nd</sup> order Butterworth filter (low-pass) with a cut-off frequency of 200 Hz is utilized in the DOB controller in a way to handle causality problem. The PID control gains are listed in Table 3. The controller gains were tuned empirically in accordance with the standard rules of controller design; stability and non-excitation of unmodeled non-linear dynamics. For details regarding the DOB design, refer to [23] and [24].

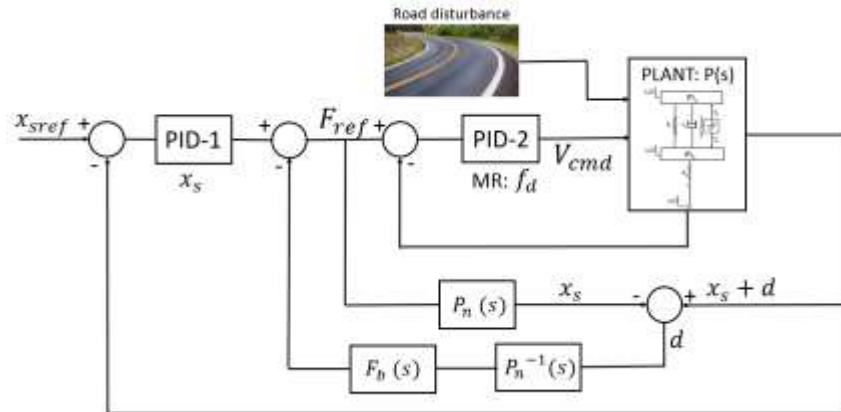


Figure 2: The block diagram for DOB controller.

#### 4. Case Studies

The proposed application of the DOB controller to quarter car model is demonstrated for five cases: step road (Section 4.1), sine road (Section 4.2), bump road and rough road (Section 4.3), and ramp road (Section 4.4). The road profiles are shown in Fig. 3.a to Fig. 3.e.

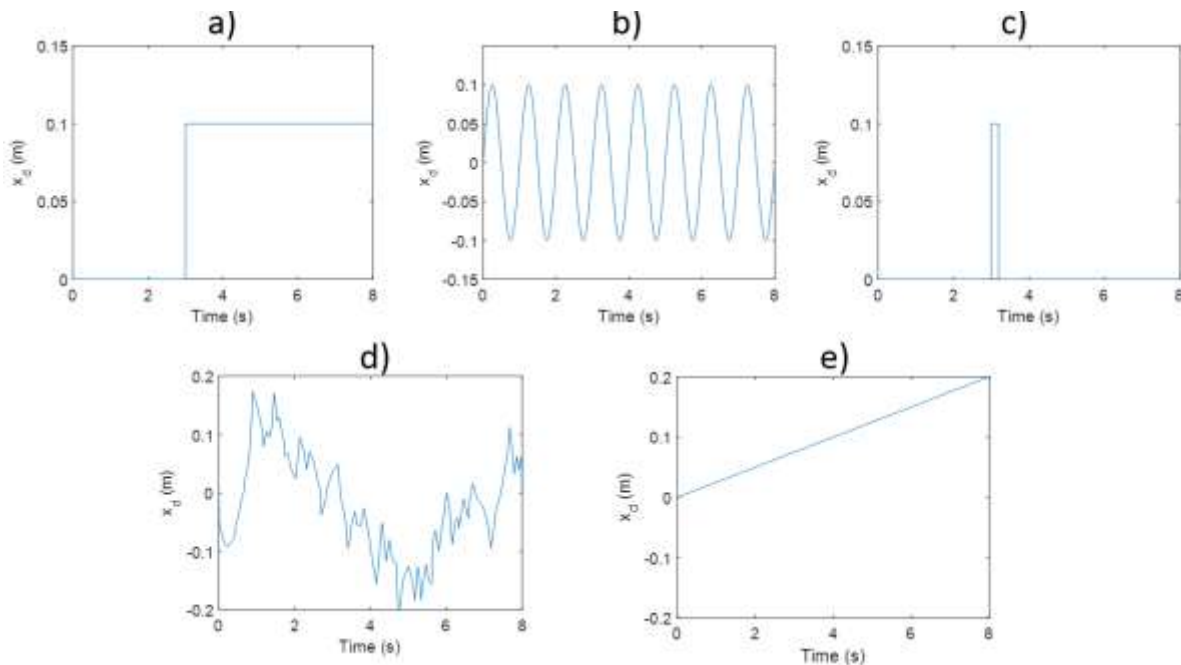


Figure 3: Road profiles a) step, b) sine, c) bump and d) rough road, and e) ramp road

Table 3: Controller gains for PID (MR) and PID ( $x_s$ )

	Parameters	Parameter Value
Controller gains for MR	Proportional gain	1
	Derivative gain	0.5
	Integral gain	10
Controller gains for $x_s$	Proportional gain	100
	Derivative gain	5
	Integral gain	0

### 4.1 Step Road

The quarter car model is simulated on a step road with an amplitude of 100 mm, Fig 3.a. Simulations were conducted using MATLAB 2018b / Simulink via a PC with Intel® Core™ i7-8550 CPU running of 2.00 GHz sampling and 16 GB RAM. The comparison of the performance of both controllers is shown in Fig. 4.a, where the sprung mass displacement is plotted as a function of time. The gains of the controllers are tuned such that the controller command is comparable in terms of best performance. As can be seen from the results, the performance of the DOB controller is more favorable in terms of robustness. The steady state displacement of the sprung mass is 0.09 mm for DOB controller, and 4.5 mm for the conventional PID controller. The results show that the the response is very oscillatory for the conventional PID, even though the DOB cancels out the disturbance quite well. When the proportional, derivative and integral gains of the conventional PID controller are increased to improve the reduce the amplitude of the sprung mass vibrations, the response becomes unstable as shown in Fig. 4.b. In such a case DOB still keeps the system stable.

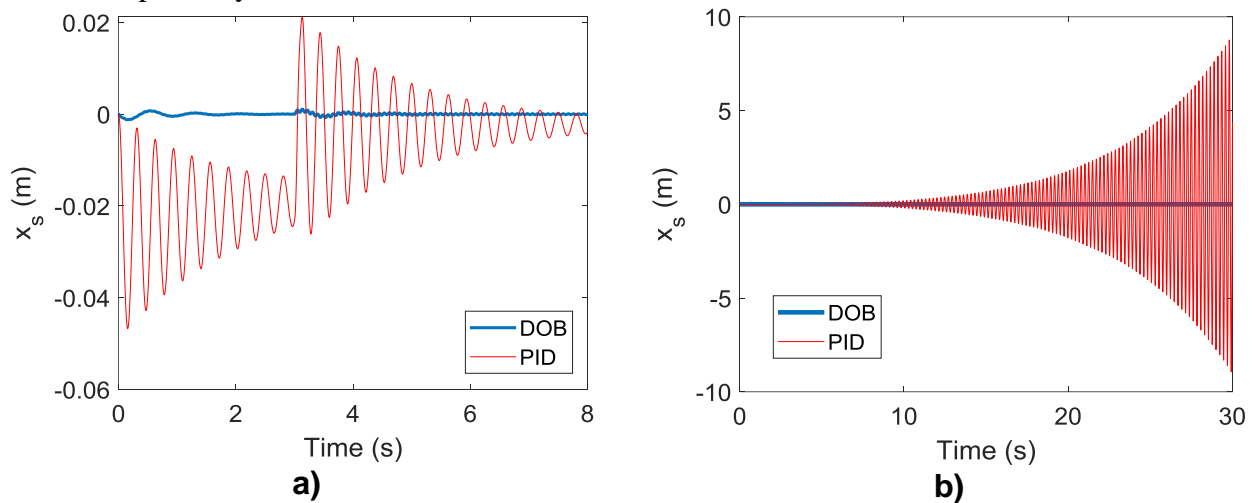


Figure 4: a) response for DOB and PID controller with comparable controller command, b) the response when the PID gains are increased such that the controller becomes unstable

### 4.2 Sine Road

The quarter car model is simulated for two sine roads with frequencies of 1 Hz and 5 Hz to compare the performance of DOB controller and the conventional PID controller. The results for sprung mass displacement are plotted in Fig. 5.a (1 Hz) and Fig. 5.b (5 Hz). The results show that DOB controller outperforms the conventional PID controller. The steady state displacement of the the sprung mass is 1.5 mm for road profile with 1 Hz excitation and 0.5 mm for road with 5 Hz excitation for DOB controller. DOB controller performs better for the latter case since the excitation frequency is more separated than the resonance frequency of the sprung mass (1.28 Hz. from modal analysis). The same conclusion can

be made for the conventional PID controller. Even though, the PID controller can not perform well, its performance is better on a road with an excitation frequency of 5 Hz than that of 1 Hz.

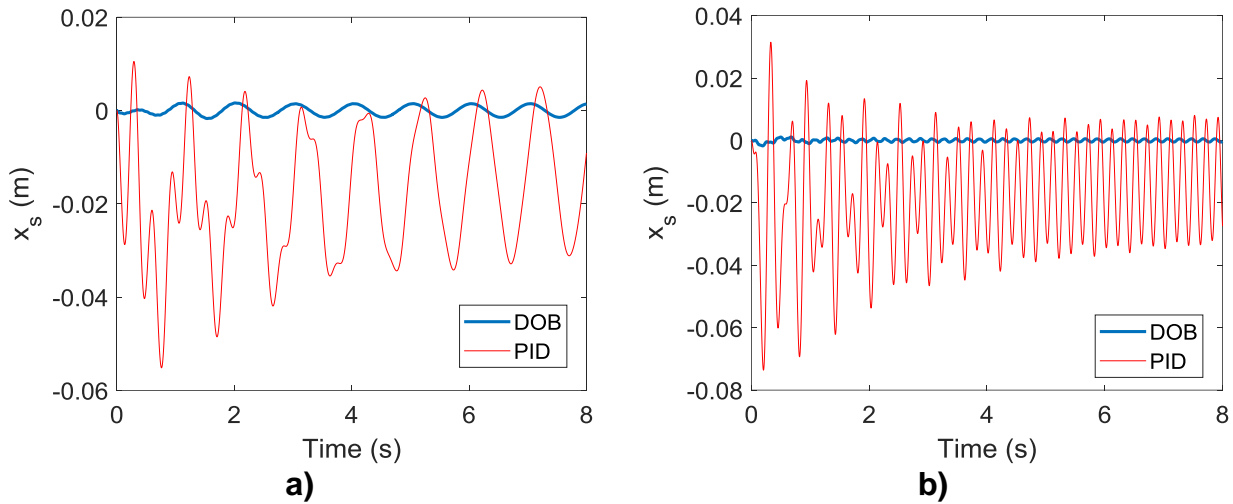


Figure 5: Sprung mass response on sine road a) excitation frequency of 1 Hz, b) excitation frequency of 5 Hz

### 4.3 Bump and Rough Roads

The results for the displacement of the sprung mass on bump-road and rough road are shown in Fig. 6.a and Fig. 6.b, respectively. Even though the performance of the conventional PID controller can be improved by tuning the proportional, derivative and integral gains, such tuning results in the instability of the response. The performance of the DOB controller for the comparable controller command is much better than the conventional PID as in the step road.

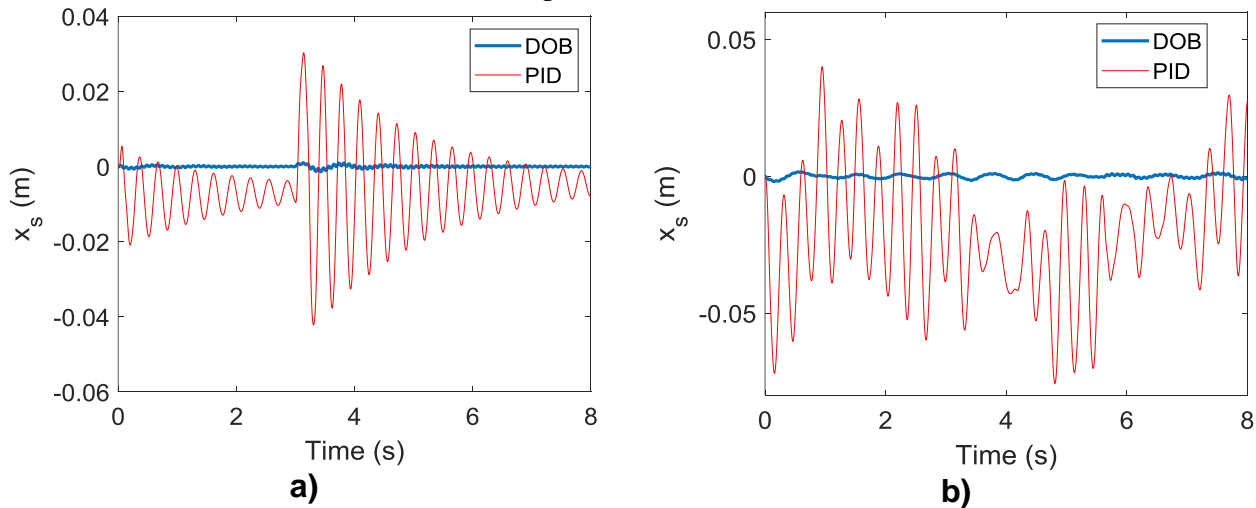


Figure 6: Sprung mass response on a) bump road, b) rough road

### 4.4 Ramp Road

The displacement of the sprung mass on ramp road is shown in Fig. 7. Similar to the other case studies, the DOB controller is able to suppress the disturbance. The amplitude of the displacement of sprung mass is 0.8 mm for DOB controller. The response speed and damping characteristics are also favorable. The amplitude of the oscillations is much greater for the conventional PID controller compared to DOB. In addition, the response is very oscillatory.

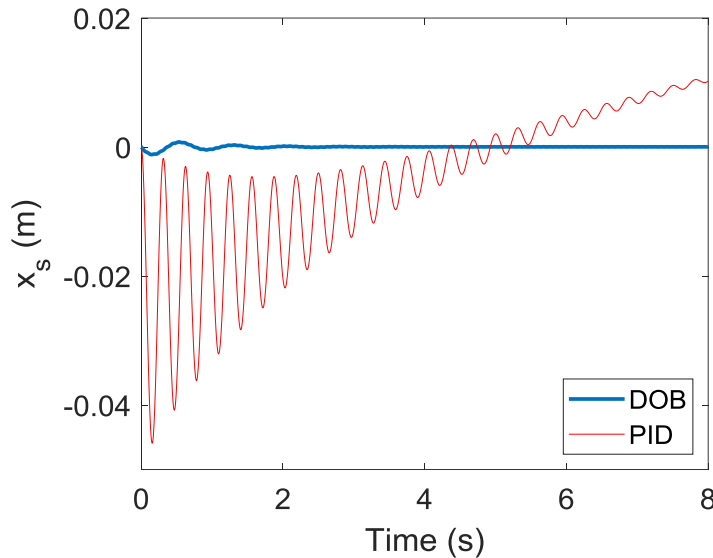


Figure 7: Sprung mass response on ramp road

## 5. Conclusions

In this study, a disturbance observer based control algorithm to a quarter car representation of an automotive with MR damper is proposed to improve the vehicle ride performance. The proposed methodology is demonstrated on five different road profiles and the results are compared to the conventional PID controller. The simulation results show that the DOB based controller outperforms the conventional PID when the control command is comparable to each other for all case studies. The simulation results also show that the conventional PID controller can not attain the performance of the DOB controller even if the controller gains are increased since the system becomes unstable. The application of the DOB controller to full vehicle models is acknowledged as future work. Benchmarking of the performance of the controller to the other commonly used controllers such as  $H_\infty$  control,  $L_1$  adaptive control and sliding mode control are also considered as future work.

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