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Experimental Implementation of Model Predictive Control Scheme for Control of Semi-active Suspension System^{*}

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Abstract: In this paper, a model predictive control (MPC) scheme is developed and experimentally validated for control of a quarter car system equipped with semi-active (SA) suspension system, which is stationed on the INOVE test platform. The work can be described in two folds which are a) parametric modelling of Electro-Rheological (ER) damper based SA suspension system (ER-SA) and b) implementation of MPC with discretized set of inputs, which in this case are the set of duty cycle (DC) dependent pulse width modulation (PWM) signals that operates the ER-SA suspension system. In the former work, a phenomenological parametric damper model is utilized to describe the ER damper's dynamic input/output characteristics by virtue of non-linear least squares (NLS) data fitting method. The latter method utilizes this model into the MPC framework for control of the quarter car system. The MPC controller was practically implemented on the INOVE test platform and results display better performance of the MPC controller in comparison with passive damping and modified Skyhook controller.

Keywords: Semi-active suspension system, Model predictive control, System Identification, Vertical dynamics

1. INTRODUCTION

Vehicle suspension systems plays a pivotal role in guaranteeing safety and comfort for the onboard passengers. There exists plethora of suspension systems depending upon the mode of operation and its technology, however under a bird's eye view, the entire spectrum can be briefly classified as passive, semi-active and active suspension systems. Amongst the three classes, SA suspension systems are quite popular in the automotive industry due to multitude of reasons such as negligible power demand, safety, low cost and weight and significant impact on vehicle performance (see Savaresi et al. (2010)). Some of the prominent SA technologies are a) Electro-hydraulic (EH), b) Electro-rheological (ER) and c) Magneto-rheological (MH) based system. In this paper, the subject of focus is considered around modeling and control of ER-SA suspension system on the INOVE test platform (see Vivas-Lopez et al. (2014)). The INOVE test platform discussed is a 1:5-scaled baja style racing car which consists of 4 controllable ER-SA dampers and 4 DC motors to generate different road profiles for each wheel corner. The INOVE platform is shown in Fig. 1.

Model predictive control (MPC) is indisputably one of the most advanced and efficient control design methodology.

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Ever since its inception and success of its application in petro-chemical industries, MPC has gradually trickled down into other streams of engineering such as automotive, aerospace, biomedical etc. The fundamental reason for its widespread application is due to the ability to provide optimal performance in the presence of system constraints (physical limitations/safety constraints). Despite its enormous advantages in terms of optimal performance and constraint satisfaction, one of the major shortcoming is that the entire MPC controller hinges upon the model utilized in the control design. Due to the predictive nature of the controller, utilizing an erroneous model would ensue poor performance due to the mismatch of models between the plant and the controller. Thus, it is of paramount importance to build high fidelity model such that the mismatch is reduced and tangible performance benefits from the MPC controller are obtained. Given this prelude and in the same spirit, this paper addresses the following problems a) modeling of the ER-SA suspension system and b) inclusion of the developed model into the MPC framework and implementation on the INOVE test platform for a quarter car system.

In general, any SA damper modeling can be broadly classified into parametric and non-parametric based approach. Under the former regime, the structure of the model is dictated by the physics of the system, which is modeled by means of first principles techniques and by contrast, the latter method obscures the underlying physics of the



Fig. 1. INOVE test platform

system and this leads to a flexible model structure, which is modeled by means of empirical techniques. A detailed survey of different types of parametric/non-parametric modeling for SA suspension systems are given in Butz and Von Stryk (2002).

In this paper, the parametric method is of primary interest and some of the popular parametric models (to name a few) include the Bingham model (Stanway et al. (1987)), the phenomenal Bouc-Wen model (Spencer Jr et al. (1997)), the nonlinear viscoelastic-plastic model (Kamath and Wereley (1997)), the nonlinear bi-viscous model (Stanway et al. (1996)), Guo's damper model (Guo et al. (2006)) etc. In this work, Guo's damper model is utilized to describe the ER-SA suspension system for the INOVE test platform due to its simplicity and parsimony to describe the ER-SA damper characteristics.

Pertinent to MPC implementation for SA suspension system, there has been several research conducted in the past. Canale et al. (2006) proposed the fast MPC method, where the optimal control input is computed offline by means of set membership approximation technique. In hybrid MPC approach, proposed in Giorgetti et al. (2006), the system is modeled as hybrid dynamical system and the optimal control input is computed offline by solving a multi-parametric program for a mixed-integer quadratic program (MIQP). In Rathai et al. (2019), reachability based MPC for SA suspension system is proposed and implemented for a quarter car model. A detailed literature review of different control strategies are presented in Tseng and Hrovat (2015) and Poussot-Vassal et al. (2011). In this paper, a parameterized based MPC design is proposed and utilized to control the ER-SA damper system for a quarter car system at the INOVE test platform. The crux of the method is to search (by simulation) rather than to solve (by optimization) for the optimal control sequence. The key advantage of the method is that, it can deal with highly nonlinear dynamics, objective and constraint functions and also, more computationally efficient.

This paper is organized as follows. Section 2 discusses about the system dynamics and the nonlinear ER damper model in detail. Section 3 discusses the list of experiments conducted to estimate the ER-SA damper system's parameters. Section 4 discusses the proposed MPC scheme in detail. Section 5 expounds the different controllers utilized to compare the performance with the proposed MPC controller. Section 6 discusses the results and real-time implementation of the different controllers on INOVE test platform and finally, the paper is concluded with conclusions and future works in Section 7.

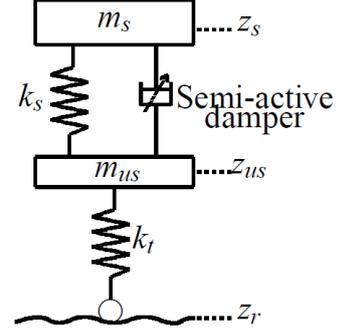


Fig. 2. Quarter car vehicle model

2. MODEL FORMULATION

2.1 Quarter car model

The vertical dynamics model for the quarter car system equipped with ER-SA damper system (around the equilibrium) is defined with

$$\begin{aligned} m_s \ddot{z}_s &= -k_s(z_s - z_{us}) - F_{ER} \\ m_{us} \ddot{z}_{us} &= k_s(z_s - z_{us}) + F_{ER} - k_t(z_{us} - z_r) \end{aligned} \quad (1)$$

where, m_s , m_{us} are the sprung mass and unsprung mass respectively, k_s , k_{us} are the stiffness coefficients of suspension system and the tire respectively, z_s , \dot{z}_s are the sprung mass position and velocity respectively, z_{us} , \dot{z}_{us} are the unsprung mass position and velocity respectively, z_r is the vertical road position and F_{ER} is the force exerted due to the ER-SA damper system. The set of differential equations (1) are compactly expressed in state space form with

$$\dot{x}(t) = A_c x(t) + B_c F_{ER}(t) + B_{cd} d(t) \quad (2)$$

where, $x = [z_s \ z_{us} \ \dot{z}_s \ \dot{z}_{us}]^T$ are the system states, $d(t) = z_r$ is the disturbance input from the road profile. $A_c \in \mathbb{R}^{4 \times 4}$, $B_c \in \mathbb{R}^{4 \times 1}$ and $B_{cd} \in \mathbb{R}^{4 \times 1}$ are the system matrix, input matrix and disturbance matrix respectively. $z_{def} = z_s - z_{us}$ and $\dot{z}_{def} = \dot{z}_s - \dot{z}_{us}$ are the deflection position and velocities between the chassis and the wheel. Fig. 2 illustrates the quarter car model of the system.

2.2 Quasi-static nonlinear ER damper model

The ER-SA damper force F_{ER} defined in (2) is expressed using the quasi-static nonlinear damper model (Guo's damper model) with

$$F_{ER} = f_c u \tanh(a_1 \dot{z}_{def} + a_2 z_{def}) + c_0 \dot{z}_{def} \quad (3)$$

where, f_c , a_1 , a_2 are appropriate model parameters, u is the PWM-DC input for the system such that $u \in \mathbb{U}$, where $\mathbb{U} := [u_{min}, u_{max}]$ such that $0 \leq u_{min} < u_{max} \leq 1$. c_0 is the nominal damping coefficient of the ER-SA damper system.

3. PARAMETER ESTIMATION

3.1 ER-SA damper response time estimation

To study the dynamic characteristics of the ER-SA damper system, it is important to estimate the response time of the system, which is indirectly estimated by finding the peak response time of the system (T_r). The following experiment was conducted to estimate T_r :

- The platform was excited with a pseudo binary random sequence (PRBS) road profile with an amplitude of 5 mm for a duration of 20 s.
- The PWM-DC signal was flipped from u_{min} to u_{max} at time 10 s (a step change in input) and the ER-SA damper force (F_{ER}) was measured.

In order to zero-in the point of transition (high frequency content), which provides the necessary cues for T_r estimation, time-frequency analysis was performed by means of wavelet transform. Fig. 3 illustrates the wavelet analysis performed on F_{ER} signal with Morlet wavelet basis functions.

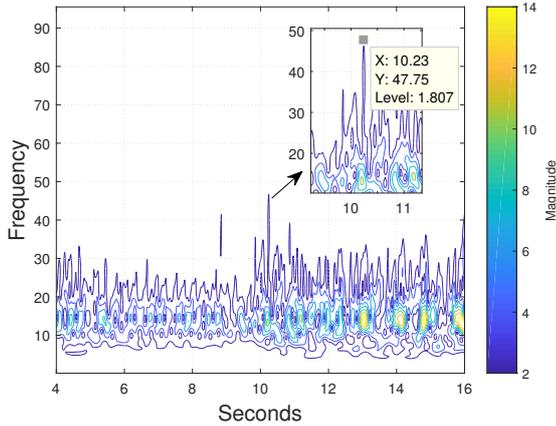


Fig. 3. Wavelet analysis of F_{ER} signal

It is evident from the contour plot that the peak in the frequency and energy content at time 10.23s, provides the necessary cue to approximate the peak response time, i.e. $T_r \approx 230$ ms. This inference serves three purposes which are

- The at-most period for PWM-DC transition to completely capture the dynamical behavior of the ER-SA damper system.
- The look ahead period (horizon) for the MPC controller.
- Estimation of sampling time (T_s) for the entire system, which is computed using the general measure with $T_s \in [\frac{T_r}{10}, \frac{T_r}{5}]$ (Astrom and Wittenmark (1982)).

3.2 Design of experiments

In order to obtain the best model parameters for the ER-SA damper system, it is imperative to conduct informative experiments and collect the input/output data that captures the dynamic behavior of the system. Conditioned upon the previous requirement, the test involved the following scenario:

- PRBS signal based road excitation with an amplitude of 5 mm for a duration of 20 s.
- A PRBS based input PWM signal between the interval $[u_{min}, u_{max}]$ with a holding period of T_r .

The rationale to adopt this scenario is to induce persistent excitation and minimize the crest factor for input design (Ljung (1987)). The ER-SA damper system was operated upon the aforementioned scenario and all the input/output data were collected for parameter estimation stage. Fig. 4 and Fig. 5 illustrates the force vs deflection position and velocity respectively for selected PWM duty cycle values.

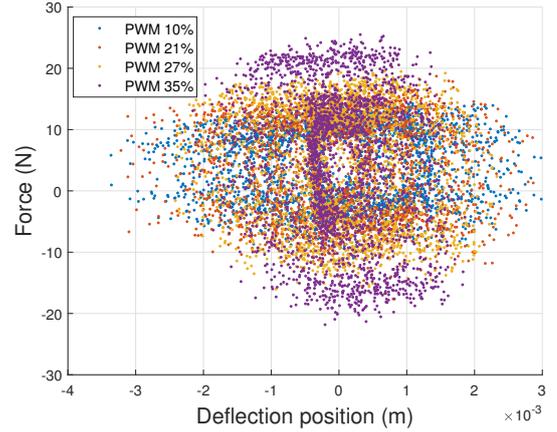


Fig. 4. F_{ER} vs z_{def} plot for different PWM signals

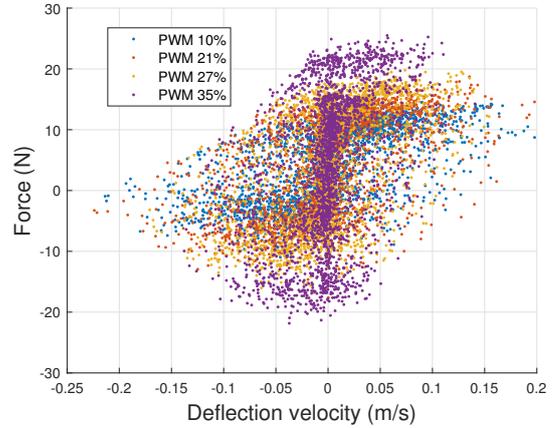


Fig. 5. F_{ER} vs \dot{z}_{def} plot for different PWM signals

3.3 NLS based data fitting

The input/output N -sample dataset are expressed with $\{\mathcal{D}_x(i)\}_{i=1}^N$ and $\{\mathcal{D}_y(i)\}_{i=1}^N$, where $\mathcal{X} = [z_{def} \dot{z}_{def} u]$ and $\mathcal{Y} = F_{ER}$. Let the unknown parameters be represented with $\theta = [f_c \ a_1 \ a_2 \ c_0]$, then the NLS objective function is defined with

$$\chi^2(\theta) = \frac{1}{N} \sum_{i=1}^N \left(\psi(\mathcal{D}_y(i)) - \hat{\psi}(\mathcal{D}_x(i)|\theta) \right)^2 \quad (4)$$

where, ψ and $\hat{\psi}$ are the true and estimated functions for the data fitting problem, which in this case is the ER-SA suspension force, i.e. F_{ER} . The optimal parameters are

computed by solving the following nonlinear optimization problem

$$\theta^* = \underset{\theta \in \Theta}{\operatorname{argmin}} \chi^2(\theta) \quad (5)$$

where, $\Theta \subset \mathbb{R}^4$ is the constraint set for the parameters, which is defined with $\Theta := \{\theta \in \mathbb{R}^4 \mid \{\theta_1, \theta_4\} \in \mathbb{R}_{\geq 0}, \{\theta_2, \theta_3\} \in \mathbb{R}\}$. The estimated model was validated using K -fold cross validation method with $K = 5$ and the accuracy of the model was estimated to 4.65 units. The estimated model parameters are listed in the table 1. Fig. 6 illustrates the predicted vs measured ER-SA damper force (F_{ER}) for a single track of input/output data.

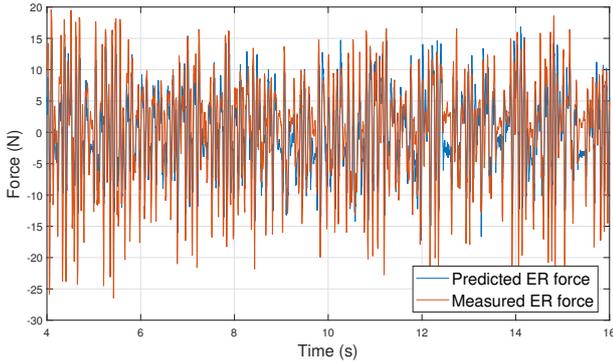


Fig. 6. Predicted F_{ER} and measured F_{ER}

Table 1. Estimated ER-SA damper parameters

Parameter	Symbol	Value (SI unit)
Force parameter	f_c	21.38(N)
Deflection position parameter	a_1	178.93(1/m)
Deflection velocity parameter	a_2	23.21(s/m)
Nominal damping coefficient	c_0	71.03(Ns/m)

4. MODEL PREDICTIVE CONTROL

4.1 Objective requirements

In this paper, the objective requirement considered for experimentation on the INOVE test platform is comfort. Qualitatively, the prime goal of the comfort based objective design is to guarantee the comfort for the on-board passengers. Quantitatively, this tantamount to minimizing the vertical acceleration of the chassis (\ddot{z}_s), which is obtained from (1). The comfort objective for the given look ahead period (T_l) is expressed as

$$J_{T_l}^{com} = \int_0^{T_l} (\ddot{z}_s(t))^2 dt \quad (6)$$

4.2 Constraint requirements

The constraints for the semi-active suspension system primarily arises from the physical limitations of the system. For the MPC design, three constraints are included in the problem formulation which are

C.1 ER-SA damper input constraints:

- (a) Max/Min damper force constraint: The ER-SA damper force, i.e. $F_{ER} \in [\underline{F}, \overline{F}]$, where \underline{F} and \overline{F} are the minimum and maximum saturation forces for the ER-SA damper system.

- (b) PWM input constraint: $u \in \mathbb{U}$.

C.2 State constraints: Max/Min deflection between the chassis and wheel position: This forms a linear state constraint such that $z_{def} \in [z_{def}^{min}, z_{def}^{max}]$, where $z_{def}^{min}, z_{def}^{max}$ are the minimum/maximum deflection position between the chassis and the wheel.

C.3 Road disturbance assumption: The road disturbance is assumed to be constant over the prediction horizon (T_l) i.e. $d^+ = d(t)$, where $d(t)$ is the road profile measured at the current time instant t .

4.3 Proposed MPC approach

The proposed MPC algorithm (Rathai et al. (2018)) is sequentially presented as follows

Algorithm:

- (1) The input u of the non-linear system (2) is finitely parameterized in time with N_δ equidistant points over the look ahead period T_l with $\{\delta_0 \dots \delta_{N_\delta-1}\}$ time stamps with an interval of $\frac{T_l}{N_\delta}$ and $T_l = \delta_{N_\delta-1}$ and in space, the set \mathbb{U} is discretized with N_s points such that $u \in \{\phi_1, \dots, \phi_{N_s}\} \subset \mathbb{U}$, where ϕ_i is a discretization point in \mathbb{U} . The input sequence over the horizon is compactly represented with $\mu(\delta_j | \{u_i(\delta_j)\}_{i=1}^{N_s}, t)$, $\forall j \in \{0, \dots, N_\delta - 1\}$, i.e. at a given time instant δ_j , there exists N_s possible input values and this spans for all given time stamps.
- (2) The explicit/implicit ODE solver for the non-linear system in equation (2) is simulated for all input sequences along space and time under the road profile assumption mentioned in section 4.2 - **C.3**.
- (3) The optimal control sequence is computed with respect to the objective and constraints by plugging the simulated trajectory onto the cost function (6) and the constraint functions mentioned in section 4.2 - **C.1, C.2**. The constraints are handled algorithmically that if a particular input sequence violates the constraints, then the input sequence is discarded and the solver is proceeded with another control sequence until the minimum cost is obtained.
- (4) In case, if no input sequence satisfies the constraints, then the input sequence which least violates the constraints is considered as the optimal input sequence.
- (5) This procedure is repeated in receding horizon policy method at every sampling period (T_s) and the optimal control input is $u^*(0) = u^*(\delta_0)$.

For the considered case of quarter car semi-active suspension system, N_s is assumed as variable (space discretization) and $N_\delta = 1$ (time discretization) and the solver utilized is a simple fourth order explicit Runge-Kutta (RK) method with fixed integration step $h = 1$ ms. The model parameters for INOVE quarter car platform and proposed MPC design are listed in table 2.

5. COMPARISON CONTROLLERS

5.1 Modified Skyhook controller

Skyhook controller is one of the most prominent and well known controller for semi-active suspension system (Karnopp et al. (1974)). The modified skyhook controller

Table 2. Model parameters for INOVE quarter car platform and proposed MPC design

Parameter	Symbol	Value (SI unit)
Chassis quarter car mass	m_s	2.27(kg)
Unsprung mass	m_{us}	0.25(kg)
Suspension stiffness	k_s	1396(N/m)
Tyre stiffness	k_t	12270(N/m)
Max/Min damper force	\bar{F}, \underline{F}	$\pm 21(N)$
Max/Min deflection position	$z_{def}^{max}, z_{def}^{min}$	$\pm 0.005(m)$
Min PWM duty cycle	u_{min}	0.1
Max PWM duty cycle	u_{max}	0.35
Look ahead period	T_l	0.23(s)
Sampling period	T_s	0.023(s)

is an extension to the skyhook controller where the ER-SA damper system's PWM-DC signal swings between minimum and maximum value conditioned upon a switch condition. Mathematically, the controller is expressed with

$$u = \begin{cases} u_{max}, & \text{if } \dot{z}_s \dot{z}_{def} \geq 0 \\ u_{min}, & \text{if } \dot{z}_s \dot{z}_{def} < 0 \end{cases} \quad (7)$$

5.2 Nominal passive suspension

The nominal passive suspension is a typical passive suspension system, however the term nominal indicates that the PWM-DC for the ER-SA damper system is fixed to the mean value of the minimum and maximum values of the PWM-DC, i.e. $u_{nom} = \frac{u_{min} + u_{max}}{2}$. The value is held constant over the entire period of operation.

6. RESULTS AND IMPLEMENTATION

The proposed MPC controller and the comparison controllers were programmed in MATLAB/Simulink environment and was implemented on the INOVE test platform. Two road profile tests were conducted to validate the performance of the proposed MPC controller, which are a) Chirp road profile test and b) Bump road profile test.

6.1 Chirp road profile test

The test involved a chirp road profile with amplitude of 2.5 mm and frequency sweep from 5 Hz to 22 Hz (this corresponds to the comfort frequency range for the INOVE test platform). The road profile is shown in Fig. 7. The PWM-DC control inputs for different controllers is shown in Fig. 8. It is clearly evident that the proposed MPC utilizes the control authority in a judicious manner such that to minimize the vertical chassis acceleration. The RMS values of the chassis acceleration for the test and the percentage gain with respect to nominal passive damping are listed in Table 3. The RMS values clearly evinces the fact that the proposed MPC method fares better the nominal passive damping and modified skyhook controller.

Table 3. RMS values for comfort objective for chirp road profile

Controller	RMS (m/s^2)	% Gain
Nominal passive damping	6.87	0
Modified Skyhook controller	6.66	3.05
Proposed MPC controller	6.42	6.5

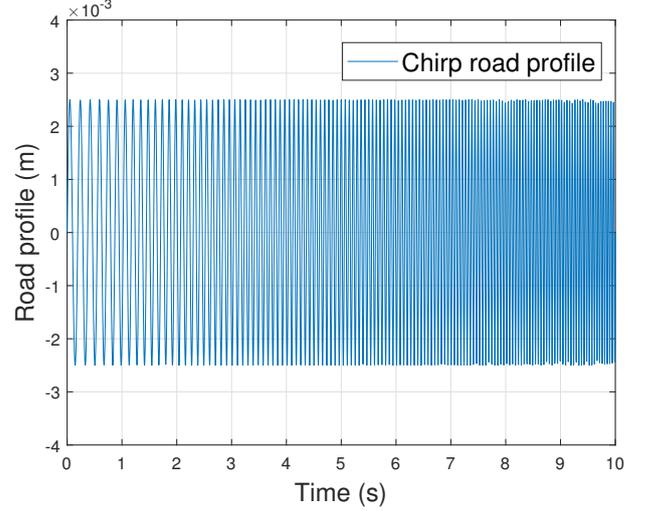


Fig. 7. Chirp road profile

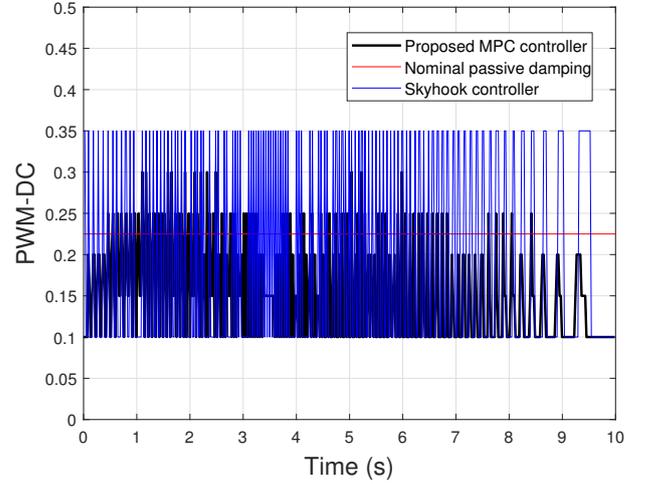


Fig. 8. PWM-DC input for different controllers for chirp road profile

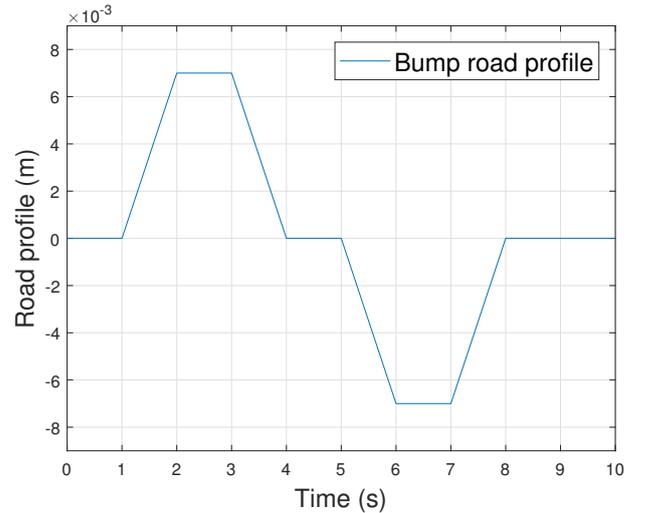


Fig. 9. Bump road profile

6.2 Bump road profile test

The INOVE test platform was excited with bump road profile, shown in Fig. 9 with peak amplitude of 7 mm

and duration of 10 s. The recorded chassis acceleration is shown in Fig. 11. From the chassis acceleration plot, it is evident that the proposed MPC method mitigates the peak chassis acceleration at bump points. The PWM-DC control input is shown in Fig 11.

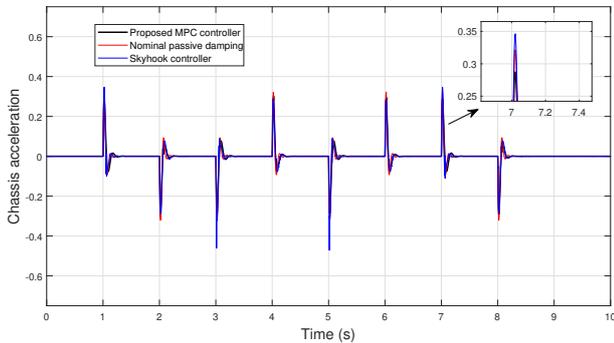


Fig. 10. Chassis acceleration for bump road profile

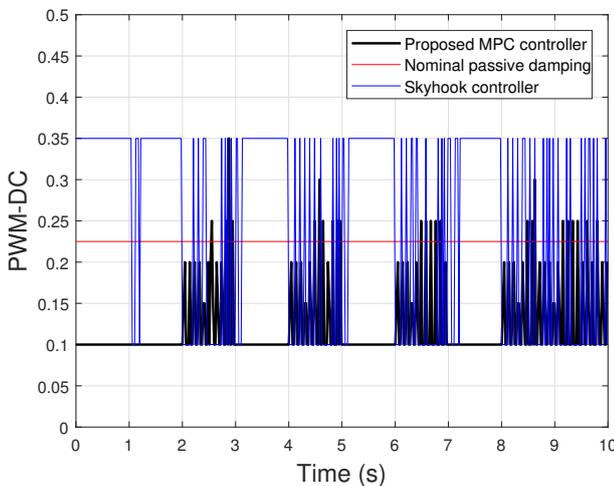


Fig. 11. PWM-DC input for different controllers for bump road profile

7. CONCLUSIONS AND FUTURE WORKS

From the tests conducted on the INOVE test platform, by and large, the proposed MPC method fares well compared to the nominal passive damping and modified skyhook controller. For the future works, the following are in the pipeline

- (1) Extension of the proposed method to full car model of the INOVE platform by means of distributed control methods (Alamir et al. (2017)).
- (2) Inclusion of road models into proposed MPC control design. The parameters such as road roughness coefficient, road type (ISO standards) provide necessary information to develop stochastic road models.

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