Simulation study on ride comfort enhancement for a heavy duty vehicle with fractional order sliding mode controller

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Abstract

The main goal of vehicle active suspension system is to upgrade the ride quality and stability of heavy duty vehicle (HDV) regardless of road surface. The mathematical modelling of HDV with two degrees of freedom is developed and simulated in MATLAB/Simulink environment. The simple and effective control techniques such as proportional integral and derivative (PID) controller, sliding mode controller (SMC) and fractional order sliding mode controller (FOSMC) are designed to detach the vibration from the passenger under step, sinusoidal and random road profile. Literature reveals that the FOSMC can eliminate steady state error and chattering effect unlike SMC and PID. The dynamic analysis of designed controllers is examined in terms of time domain, frequency domain and ride comfort analysis as per ISO 2631. The stability of SMC and FOSMC is confirmed by Lyapunov stability for three different road profiles. The FOSMC shows satisfactory simulation results than SMC, PID and passive suspension system in reducing suspension deflection and body acceleration of HDV to upgrade the ride quality. Further, the efficacy of proposed FOSMC is proved by comparing the simulation results against existing results of PID controller.

Keywords FOSMC · HDV · Ride comfort analysis · Stability · VASS

1 Introduction

In the transportation field, tractor trailers, trucks, heavy trucks, trailer trucks and military trucks are predominantly used as heavy duty vehicles (HDVs). The continuous rise in vehicle loads and road traffic, there is a damage of vehicle parts. The passenger safety and ride comfort are greatly affected by vibration of the vehicle. In the earlier days, Passive Suspension System (PSS) is used and it does not provide travel comfort and safety for the passenger due to fixed values of spring and damper. The complications raised from PSS are moderately dealt with Semi ASS using variable damper. Hence the automotive industry introduces Active Suspension System (ASS) to enhance the ride comfort and suppresses the vehicle vibration by the actuator force. The front suspension in the midi bus reduces the

intensity of rollover and improves the stability of the vehicle [1]. In mining vehicles, ride comfort can be achieved with hydraulic interconnected suspension system which provides soft bounce and decreases the pitch motion [2]. The electro hydraulic damper is utilised to improve the handling performance of heavy trucks [3]. Further, the off road vehicle dynamics are studied to forecast the vibration of the vehicle [4]. For better safety and passenger comfort, real time damped shock absorber is introduced in off-road vehicle [5].

The seat vibration in HDV can be suppressed with static output feedback H-infinity controller and it replaces conventional linear actuator by active seat suspension system [6]. The semi active and active damper is introduced to analyse the dynamics of tractor for road-3 random road profile [7]. The higher order sliding mode controller (SMC)

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is developed to avoid the risk of rollover and twisting algorithm is implemented to guarantee the stability of the vehicle [8]. The active rollover control technique is highlighted in HDV to reduce accidents [9]. The payload changes can significantly affect the dynamics of HDV. Hence the Linear Quadratic Regulator (LQR) is designed to upgrade the dynamics of front and rear loaded vehicle [10].

The various control strategies such as bat optimized proportional integral and derivative (PID) controller [11], fuzzy PID controller [12] and single neuron PID [13] are presented to reduce the effects of vibration in the HDV. The SMC [14], Fractional Order SMC (FOSMC) [15] and fractional order terminal SMC [16, 17] are designed to arrest the vibration for quarter car and full car model which enhances the travel comfort of the passenger under irregular road profile. The speed control of HDV is foremost criteria to avoid the risk of accidents. To achieve superior speed regulation performances in HDV, an improved internal model controller [18] and neural based fuzzy inference control systems [19] are presented. Based on the driving environment, drivability is evaluated by frequency weighted function and Root Mean Square (RMS) values [20]. On the other hand, available control techniques [21, 22] can improve steady state performance, while their transient behaviour such as overshoot and settling time cannot be dictated. To achieve safety and ride comfort, it is more desirable to guarantee both steady state and transient response.

Influenced by the literature survey, this paper is focussed on the design of FOSMC for HDV. The prominent factor of the proposed controller is to assure both steady state and transient response. Another key factor is to eliminate chattering effect and guarantees the stability for HDV. The prerequisites of suspension performances such as Body Acceleration (BA) and Suspension Deflection (SD), Frequency Weighted RMS (FWRMS) and Vibration Dose Value (VDV) are also discussed.

The organisation of the proposed work is as follows; in Sect. 2, the modelling equations of HDV are derived. The control strategies are presented in Sect. 3. In Sect. 4, the performances of the designed controllers are evaluated in terms of numerical and simulation results. Finally, conclusions are outlined in Sect. 5.

2 Mathematical modelling of heavy duty vehicle

A HDV is a complex vibration system. It is quite troublesome to develop a perfect model to reflect the complete vibration characteristics of the HDV. Further analysis, the mathematical model of HDV is accomplished to simplify

Fig. 1 Heavy duty vehicle model

Table 1 HDV parameters

Symbol	Description	Values
m_1	Body mass	2500 kg
m_2	Suspension mass	320 kg
k_1	Suspension system spring constant	80,000 N/m
k_2	Spring constant of wheel and tyre	500,000 N/m
c_1	Suspension system damping constant	350 N s/m
c_2	Damping constant of wheel and tyre	15,020 N s/m
w	Wheel	
f_a	Control force	

the vibration system based on the research objective. The mathematical model of any system describes the relationships between the input and output along with the elements present in the system. A basic quarter HDV model with 2 DOF [22] is shown in Fig. 1. The tyre stiffness and suspension spring stiffness are assumed to be linear in their operating range and that the tyre does not leave the ground. The mathematical modelling of the HDV is done by using MATLAB.

The differential equations of quarter HDV with 2 DOF [22] are as follows;

$$m_1 \ddot{y}_1 + k_1(y_1 - y_2) + c_1(\dot{y}_1 - \dot{y}_2) = f_a \quad (1)$$

$$m_2 \ddot{y}_2 - k_1(y_1 - y_2) - c_1(\dot{y}_1 - \dot{y}_2) - c_2(\dot{w} - \dot{y}_2) - k_2(w - y_2) = -f_a \quad (2)$$

The parameters of HDV [21, 22] are given in Table 1.

3 Controller design

To suppress vibration in HDV, PID, SMC and FOSMC are designed for SASS.

3.1 PID controller design

The basic form of PID control is given as

$$u(t) = K_p e(t) + K_i \int e(t) dt + K_d de(t)/dt \quad (3)$$

where K_p , proportional gain; K_i , integral gain; K_d , derivative gain; $e(t)$, error and $u(t)$, controller output.

The aim of PID controller combines the useful characteristics of Proportional and Integral (PI) controller and Proportional Derivative (PD) controller. The output response of PID controller has minimum overshoot and return back to set point more shortly than the other forms of PI and PD controller. For higher order process, PID is necessary to prevent large overshoot and long settling time [23] and the PID controller tuning values are considered from [22]. The advantage of using this controller is to reduce the vibration to zero and overshoot.

3.2 Sliding mode controller design

To achieve desired ride quality and stability, SMC is designed for HDV. The designed SMC is tested for step, random and sinusoidal road disturbances. The Lyapunov function used for SMC is $V = \frac{1}{2} \dot{s}^2$. A necessary condition for existence and reachability of a sliding surface is $S(X, t)$ in the system state space is to choose the control variable $u(t)$ such that the following condition is satisfied

$$\dot{V} = s\dot{s} < 0 \quad (4)$$

The suspension deflection (x_1) and car body velocity (x_2) is chosen as state variables for HDV to design SMC [24, 25]. The vibration of the HDV is decreased by applying control force generated from actuator. This control force produces opposite force to suppress vibration developed due to road disturbances.

The following sliding surface is chosen as to design SMC,

$$s = x_2 + \lambda x_1 \quad (5)$$

where λ is the gain of sliding surface. By differentiating the Eq. (5),

$$\dot{s} = \dot{x}_2 + \lambda \dot{x}_1 \quad (6)$$

The system moves on sliding state at $\dot{s} = 0$,

$$0 = \dot{x}_2 + \lambda \dot{x}_1 \quad (7)$$

The following state variables are used to obtain the control force,

$$x_1 = y_1 - y_2 \quad (8)$$

$$x_2 = \dot{y}_1 \quad (9)$$

The state variables are substituted in the above Eq. (6)

$$\dot{f}_a = k_1(y_1 - y_2) + c_1(\dot{y}_1 - \dot{y}_2) - \lambda m_1(\dot{y}_1 - \dot{y}_2) - k \text{sign}(s) \quad (10)$$

where k is a positive constant and the switching function is $\text{sign}(s)$

The main drawback incorporated with SMC is the discontinuity of the controller around the sliding surface $s(X, t)$. Hence the chattering effect is developed which may leads to unpredictable degradation in the system performance.

3.3 Fractional order sliding mode controller design

To overcome the chattering phenomenon, fractional order is included with SMC. To decrease the maximum peak overshoot and steady state error, the FOSMC is developed to improve the transient behaviour of the HDV. The state variables for designing FOSMC are chosen from SMC. The FOSMC is designed by including fractional derivative term in the SMC equation [24]. The sliding surface for FOSMC is chosen as

$$s = D^{\alpha-1} x_2 + \lambda x_1 \quad (11)$$

where D represents the differentiation, and α is the fractional order of the D .

$$\dot{s} = D^{\alpha-1} \dot{x}_2 + \lambda \dot{x}_1 \quad (12)$$

As per Eq. (3),

$$0 = D^{\alpha-1} \dot{x}_2 + \lambda \dot{x}_1 \quad (13)$$

The control force for FOSMC is

$$f_a = k_1(y_1 - y_2) + c_1(\dot{y}_1 - \dot{y}_2) - D^{\alpha-1} \lambda m_1(\dot{y}_1 - \dot{y}_2) - k \text{sign}(s) \quad (14)$$

4 Results and discussion

In this section, the simulation results of designed controllers for HDV are studied. The HDV model and the controllers are simulated in the MATLAB/Simulink environment. The parameters used for simulation are chosen by trial and error method. The chosen parameters are $\alpha = 0.85$, $\lambda = 0.9$, $K = 1$. The performances of the vehicle suspension under step, random and sinusoidal disturbances are investigated with PSS, PID, SMC and FOSMC. The HDV parameters are used for simulation which is listed in Table 1. The HDV runs at a speed of 30 kmph. The three different road profiles [11] are shown in Fig. 2.

Fig. 2 Road profiles

Fig. 4 Passive response of SD-step input

Fig. 3 Passive response of BA-step input

Fig. 5 Active response of BA-step input

4.1 Performance analysis of SMCs with step input

The suppression of BA and SD deteriorate the life span of the ride quality and suspension system of the HDV. For further assessment of potency of the proposed controller with regards to the above mentioned performance, this study is used the Frequency Weighted Root Mean Square (FWRMS) and transient response analysis as performance criteria.

The BA and SD of PSS with step input [11] are shown in Figs. 3 and 4 respectively. It is noted from the Fig. 3, the maximum peak amplitude of BA is 0.4 m/s^2 and settling time is 32 s. Similarly, the maximum peak amplitude and settling time of SD (observed from the Fig. 4) are $2.2 \times 10^{-5} \text{ m}$ and 38 s respectively. Similarly, the transient responses of ASS with step input are analysed. It can be seen from the Fig. 5, the maximum peak amplitude and

Fig. 6 Active response of SD-step input

Fig. 7 Control force-step input

settling time of BA with FOSMC are 0.03 m/s^2 and 0.2 s whereas the BA for SMC are 0.04 m/s^2 and 0.25 s (Fig. 5).

The settling time of SD (Fig. 6) with FOSMC and SMC are 0.8 s and 1.2 s . Also the overshoot and settling time of PID are listed in the Table 5. From the above-mentioned time domain analysis shows that the maximum peak amplitude and settling time of FOSMC are reduced appreciably than SMC and PID. The control force of the designed controllers is shown in Fig. 7. From this response; the FOSMC takes more control force to attain desired response.

Fig. 8 Passive response of BA-RR input

Fig. 9 Passive response of SD-RR input

4.2 Performance analysis of SMCs with random road

To evaluate the performances of the SMCs, it is very essential to examine the real road as the disturbance to the HDV model. The real road profile is random road (RR) in nature and it is modelled as per ISO 8608. There are more deformities in the real road and varies continuously with distance. The suspension performances are validated by using RR input.

The SMCs parameters designed for step input is utilised for RR profile. The passive responses of BA and SD are shown in Figs. 8 and 9 respectively [11]. The active responses of BA and SD with RR are shown in Figs. 10 and 11.

Fig. 10 Active response of BA-RR input

Fig. 12 Control force-RR input

Fig. 11 Active response of SD-RR input

Fig. 13 Passive response of BA-SR input

It can be observed from the Figs. 10 and 11, the vibration level of BA and SD with FOSMC is significantly decreased when compared to SMC, PID and PSS. The vibration produced by RR is decreased with FOSMC than PID, SMC and PSS. The corresponding control forces of FOSMC, SMC and PID are shown in Fig. 12.

4.3 Performance analysis of SMCs with sinusoidal road

The sinusoidal road (SR) is considered to examine the performances of the designed controllers and it is periodic in nature. Figures 13 and 14 shows the passive response

of HDV model [11]. The active responses of SR input are shown in Figs. 15 and 16.

The corresponding control force is shown in Fig. 17. The magnitude of passive response is increased due to SR. Therefore the more control force is required to reduce the vibration of HDV. The designed controller performances are analysed from the Figs. 15 and 16 and the FOSMC has the effective control over BA and SD. Hence the desirable output can be achieved with FOSMC which enhances the ride comfort by suppressing the vibration due to SR input than SMC, PID and PSS.

Fig. 14 Passive response of SD-SR input

Fig. 16 Active response of SD-SR input

Fig. 15 Active response of BA-RR input

Fig. 17 Control force-SR input

4.4 Frequency domain and stability analysis

The linearised model of HDV is analysed in terms of frequency domain using bode plot. The bode plot of BA is plotted for PID and PSS with Step, RR and SR Inputs. It can be observed from the Figs. 18, 19 and 20, the resonant peak of PSS is comparatively higher than PID controller.

The stability of SMC and FOSMC has been proved by Lyapunov stability criteria for step, RR and SR. The system trajectories are plotted for SMCs with respect to state variables of HDV under step, RR and SR (shown in Figs. 21, 22, 23, 24, 25 and 26). The trajectories of SMC and FOSMC are converging to origin for step (Figs. 21, 22) and SR input

(Figs. 25, 26). Therefore, the FOSMC and SMC for step and SR input are stable as per Lyapunov stability criteria [26].

Similarly it is tested for RR (Figs. 23, 24) to prove the potential of the designed controllers. The system trajectories neither converge nor diverge to infinity. It indicates the marginal stability of HDV with RR.

4.5 Ride comfort analysis

To analyse the ride comfort of the passenger, FWRMS values and vibration dose value (VDV) [27] are computed and it is tabulated in Tables 2, 3 and 4.

Fig. 18 Frequency response of BA-step input

Fig. 20 Frequency response of BA-SR input

Fig. 19 Frequency response of BA-RR Input

Fig. 21 Phase plane of SMC-step input

As per ISO-2631 [27], the proposed FOSMC enhances ride comfort to NU level for step input, FU level for RR and LU level for SR input which is noted from the Tables 2, 3 and 4. The FWRMS and vibration dose value (VDV) may cause discomfort for the passengers. Hence the cumulative values of VDV are measured to analyse the intermittent vibration level received over an 8-h or 16-h period. The unit

of VDV is taken from safety sciences. The ISO2631-1 Standard uses the exposure duration and Vibration Dose Values (VDV) to calculate the equivalence, being those terms a power of 2 or 4, respectively, in the equivalence formula [27]. It is evaluated for step, RR and SR input. Hence the numerical values of FWRMS and VDV are decreased for

Fig. 22 Phase plane of FOSMC-step input

Fig. 24 Phase plane of FOSMC-RR

Fig. 23 Phase plane of SMC-RR input

Fig. 25 Phase plane of SMC-SR input

FOSMC when compared to SMC and PID and it improves the quality of ride comfort.

4.6 Power spectrum density analysis

The human sensitivity range lies in the range of 4–8 Hz. It is observed from the Figs. 27, 28 and 29, the designed FOSMC lies in the human sensitivity range than PSS. Hence, the frequency range of PSD of BA with FOSMC improves the ride comfort of the passenger for three different road profiles.

4.7 Comparative analysis with existing results

The comparative analysis of HDV is made with existing PID [22] controller. The time responses of designed FOSMC and SMC are compared with existing PID controller with step input and the time domain specifications are listed in Table 5.

It can be observed from the Table 5, the proposed FOSMC settles quickly within 1 s and there is no overshoot compared to existing PID controller. The numerical and simulation results prove that the efficacy of the proposed FOSMC than existing PID controller.

Fig. 26 Phase plane of FOSMC-SR input

Table 2 Ride comfort analysis with step input

Controllers	Step input		
	BA (rad/s ²)	UL	VDV (m/s ^{1.75})
PSS	1.82	VU	2.13
PID	1.423	U	1.75
SMC	0.745	FU	1.1
FOSMC	0.296	NU	0.636

Extremely uncomfortable (EU), very uncomfortable (VU), uncomfortable (U), not uncomfortable (NU), not uncomfortable (NU), little uncomfortable (LU), fairly uncomfortable (FU)

Table 3 Ride comfort analysis with RR input

Controllers	RR input		
	BA (rad/s ²)	UL	VDV (m/s ^{1.75})
PSS	2.52	EU	2.82
PID	1.81	VU	2.11
SMC	1.2	U	1.53
FOSMC	0.95	FU	1.31

Table 4 Ride comfort analysis with SR input

Controllers	SR input		
	BA (rad/s ²)	UL	VDV (m/s ^{1.75})
PSS	1.98	VU	2.61
PID	1.31	U	1.73
SMC	0.75	FU	1.12
FOSMC	0.33	LU	0.632

Fig. 27 PSD of BA-step input

Fig. 28 PSD of BA-RR input

5 Conclusion

In this paper, the FOSMC is proposed for HDV with ASS. The 2 DOF HDV linear model is considered for controller design. The Proposed FOSMC can suppress the BA and SD while travelling over the step, RR and SR. The FWRMS and VDV are computed based on ISO-2631 to confirm the effectiveness of the proposed FOSMC. The time domain analysis of HDV with PSS, PID, SMC and FOSMC is validated using MATLAB/Simulink software. The step, RR and SR inputs are taken for the investigation of the FOSMC. The simulation results with step input is compared to existing PID controller which proves the effectiveness of the FOSMC than PID, SMC and PSS. Further discussion

Fig. 29 PSD of BA-SR input

Table 5 Time domain analysis

Controllers	Suspension deflection (m)	
	Overshoot (m)	Settling time (s)
PSS	2.2×10^{-5}	38
PID	0.81	2
SMC	–	1.2
FOSMC	–	0.9

confirms that the stability of the vehicle is guaranteed by FOSMC and SMC.

Compliance with ethical standards

Conflict of interest On behalf of all authors, the corresponding author states that there is no conflict of interest.

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