

Performance of semi-active cab suspension system with different control methods

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Abstract. In this study, based on adaptive control methods, the semi-active suspension system of the heavy truck cab is researched and controlled to improve the ride comfort of the heavy truck. A dynamic model of the vehicle is established for simulation. Matlab/Simulink software is used to simulate and calculate the root mean square (RMS) accelerations of the driver's seat and cab pitch angle under different operation conditions. Proportional-integral-derivate controller with its parameters optimized by the genetic algorithm (GA-PID controller) and Fuzzy logic control combined with PID (FLC-PID controller) are used to control the semi-active cab suspension system of the heavy truck. The obtained results show that the ride comfort of the vehicle using FLC-PID is better improved in comparison with using GA-PID under different operating conditions. Especially, when the vehicle moves at a speed of 72 km/h, the RMS accelerations of the driver's seat and cab pitch angle are greatly reduced by 26.45 % and 26.07 % respectively. Therefore, the FLC-PID control should be applied to the suspension system of the vehicles to improve the vehicle's ride comfort.

Keywords: FLC, PID controller, GA, cab's suspension system, dynamic model.

1. Introduction

To improve the ride comfort and road friendliness of the heavy trucks, the heavy truck's dynamic parameters were researched by some authors. Mahmoodabadi et al. [1] applied the genetic algorithm (GA) for improving the ride quality of the model of half-vehicle, while Li and Wang [2-3] built the dynamic model of the half-vehicles for simulating and optimizing the parameters of the cab's suspension systems to decrease the seat's vibration in the vertical direction. Chen et al. [4] also applied measurement and simulation methods to optimize the cab's isolation system for heavy commercial vehicles. In addition, Zhang et al. [5] had researched a dynamic model of the multi-body for the isolating systems of the cab to reduce the RMS accelerations of the vertical driver's seat and cab pitch angle. The ride comfort of the vehicle is significantly improved by the cab's suspension system optimized. To further improve the ride comfort of vehicles, the combined control methods were used to control the vehicle's suspension systems including the fuzzy logic control (FLC) [6-7], FLC-skyhook damper, FLC-PID controller [8-9], and FLC-H_{inf} control [10]. Most of the authors applied the above control methods for the car and bus to improve the ride comfort. The study results indicated that ride comfort of the vehicle with the vehicle's suspension controlled is better than that of the vehicle's suspension optimized.

Based on the above control methods, the passive suspension system of the heavy trucks was also studied and improved to reduce less power and low cost as well as increase the ride comfort by applying the air spring systems, active and semi-active suspension systems. In Refs. of Michele and Nguyen [9, 11], a 5-axle heavy truck using the half-truck model was established, and Adams/Matlab simulation was then applied to control parameters of the suspension system of the tractor driver. Georgios and Yi [12-13] applied the hybrid balance control of a magnetorheological to control the suspension systems of heavy trucks. Xie and Peter [14-15] built a vibration

model of half-heavy trucks to control the semi-active suspensions. The PDI, Fuzzy-Logic-Control, and Wheelbase-Preview combined were applied for controlling the active damping force of the suspension systems of the heavy trucks. The research results showed that ride comfort and road friendliness were significantly reduced in comparison without the control. The above studies also showed that the cab's suspension system is also effect not small to the driver's ride comfort and health. However, the control of the cab's suspension system has not yet been considered in these studies. Additionally, the control performance of the control methods of the GA-PID or FLC-FLC has been also studied in the existing research yet.

In this study, to evaluate the performance of the semi-active suspension system of the heavy truck cab using FLC-PID and GA-PID controllers, a dynamic model of the heavy truck is established. Matlab/ Simulink software is used to simulate and calculate the root mean square (RMS) accelerations of the driver's seat and cab pitch angle under different operation conditions. Proportional-integral-derivate controller with its parameters optimized by the GA (GA-PID controller) and Fuzzy logic control combined with PID (FLC-PID controller) are used to control the semi-active suspension system of the heavy truck cab. This goal study is to improve the ride comfort of the heavy truck.

2. Vibration model of the heavy truck

2.1. Vehicle dynamic model

In this study, a heavy truck including the seat of the driver, cab, vehicle's floor, and two front/rear wheel axles of the heavy truck is used. The suspension of the seat of the driver is used by the stiffness coefficient k_s and damping coefficient c_s ; the suspension of the cab is used by the stiffness coefficients k_{ci} and damping coefficients c_{si} ; the suspension system of the vehile body are also used by the stiffness coefficients k_i and damping coefficients c_i ; and the wheels are characterized by the stiffness coefficients k_{ti} and damping coefficients c_{ti} , respectively. The dynamic model of the heavy truck is established in Fig. 1, where z_s , z_c , z_b , and z_i are the displacements of the seat, cab, vehicle body, and wheel axles inthe vertical derection; ϕ_c and ϕ_b are angular displacements of the cab and vehicle body; m_s , m_c , and m_b are the mass of the seat of the driver, cab and vehicle; m_i is the unsprung mass of the wheel axles; u_i are the control forces of the cab's suspension systems; q_i is the excitation of the random road surface; l_n are the vehicle's distances ($i = 1-2; n = 1-7$).

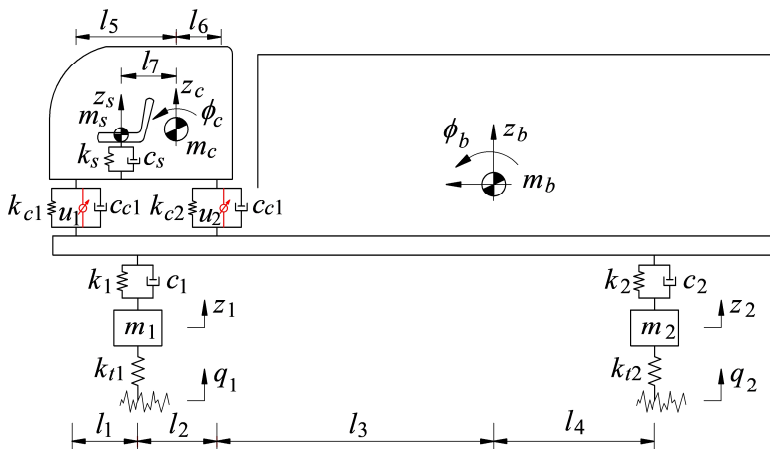


Fig. 1. A dynamic model with the cab's active suspension system of the heavy truck

Based on the heavy truck's dynamic model, the motion equations of the driver's seat, cab, and

the vehicle body are given as follows:

The vibration equation of the driver's seat:

$$m_s \ddot{z}_s - c_s \dot{z}_{cs} - k_s z_{cs} = 0, \quad (1)$$

where z_{cs} is the deformation of the driver's seat suspension determined as in Eq. (2):

$$z_{cs} = z_c - l_7 \phi_c - z_s. \quad (2)$$

The vibration equation of the cab:

$$\begin{aligned} m_c \ddot{z}_c + c_s \dot{z}_{cs} + k_s z_{cs} - (c_{c1} \dot{z}_{bc1} + k_{c1} z_{bc1} + u_1) - (c_{c2} \dot{z}_{bc2} + k_{c2} z_{bc2} + u_2) &= 0, \\ I_c \ddot{\phi}_c - (c_s \dot{z}_{cs} + k_s z_{cs}) l_8 + (c_{c1} \dot{z}_{bc1} + k_{c1} z_{bc1} + u_1) l_6 - (c_{c2} \dot{z}_{bc2} + k_{c2} z_{bc2} + u_2) l_7 &= 0, \end{aligned} \quad (3)$$

where z_{bc1} and z_{bc2} are the front and rear deformations of the cab suspension determined as in Eq. (4); u_1 and u_2 are the front and rear active control of the cab suspension system:

$$\begin{aligned} z_{bc1} &= z_b - z_c - (l_1 + l_2 + l_3) \phi_b + l_5 \phi_c, \\ z_{bc2} &= z_b - z_c - l_3 \phi_b - l_6 \phi_c. \end{aligned} \quad (4)$$

The vibration equation of the vehicle body:

$$\begin{aligned} m_b \ddot{z}_b &= F_1 + F_2 - F_{c1} - F_{c2}, \\ I_b \ddot{\phi}_b &= F_{c1} (l_1 + l_2 + l_3) + F_{c2} l_3 + F_2 l_4 - F_1 (l_2 + l_3), \\ m_1 \ddot{z}_1 &= F_{t1} - F_1, \\ m_2 \ddot{z}_2 &= F_{t2} - F_2, \end{aligned} \quad (5)$$

where the force responses of the cab suspension (F_{c1} and F_{c2}), force responses of the vehicle body suspension (F_1 and F_2), and force responses of the wheels (F_{t1} and F_{t2}) are determined as in Eq. (6), respectively:

$$\begin{aligned} F_{c1} &= k_{c1} z_{bc1} + c_{c1} \dot{z}_{bc1}, \\ F_{c2} &= k_{c2} z_{bc2} + c_{c2} \dot{z}_{bc2}, \\ F_1 &= k_1 [z_1 - z_b + (l_2 + l_3) \phi_b] + c_1 [\dot{z}_1 - \dot{z}_b + (l_2 + l_3) \dot{\phi}_b], \\ F_2 &= k_2 (z_2 - z_b - l_4 \phi_b) + c_1 (\dot{z}_1 - \dot{z}_b - l_4 \dot{\phi}_b), \\ F_{t1} &= k_{t1} (q_1 - z_1), \\ F_{t2} &= k_{t2} (q_2 - z_2). \end{aligned} \quad (6)$$

2.2. Vibration excitation of the road surface roughness

The roughness of the highway road surface greatly affects the interaction between the wheel and road surface and the road damage. It plays an important role in evaluating dynamic interaction between vehicles and roads. The random excitation of the road surface can be represented with a periodic modulated random process. The general form of the power spectrum density (PSD) of the road surface roughness is determined by [3, 16]:

$$S_q(n) = S_q(n_0) \left(\frac{n}{n_0} \right)^{-\omega}, \quad (7)$$

where n is the reciprocal space frequency of the wavelength; n_0 is the reference space frequency ($n_0 = 0.1 \text{ m}^{-1}$); $S_q(n)$ is the PSD of road surface under the reference space frequency n_0 known as the road surface roughness coefficient; and ω is the frequency index that decides the frequency

configuration of PSD of road surface ($\omega = 2$).

The road surface roughness is assumed to be a zero-mean stationary Gaussian random process. It can be generated through an inverse Fourier transformation:

$$q = \sum_{i=1}^N \sqrt{2S_q(n_i)\Delta n} \cos(2\pi n_k t + \phi_i), \quad (8)$$

where ϕ_i is the random phase uniformly distributed from 0 to 2π .

According to the standard ISO 8068 [16], a type of road surface roughness of ISO level C is applied to evaluate the ride comfort of the heavy truck as well as the control methods of the semi-active cab suspension system. The simulation result of the typical road surface roughness of ISO level C is shown in Fig. 2.

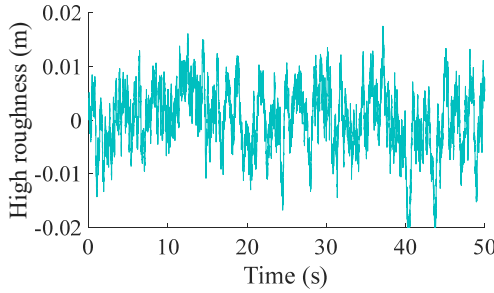


Fig. 2. The vibration excitation of the random road surface of ISO level C

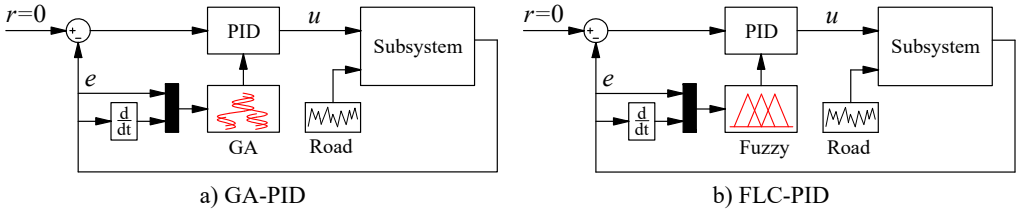


Fig. 3. The control models of semi-active cab suspension.

3. Semi-active cab suspension system with different control methods

Proportional-integral-derivate (PID) controller not only a simple structure but also robust performance. It is the most used in industrial process control. Its transfer function can be written as follows:

$$u(t) = k_p e(t) + k_i \int_0^t e(t) dt + k_d \dot{e}(t), \quad (9)$$

where k_p , k_i , and k_d are the proportional, integral, and derivative parameters, respectively. The performance of the PID controller depends on the appropriate choice of the parameters of k_p , k_i , and k_d . The well-known Ziegler-Nichols technique is used to choose these parameters. However, it is efficient only when the system works at the designed operating condition [3, 9]. To solve this problem, the GA and FLC combined with the PID are then applied to optimize the PID's parameters for the semi-active suspension system of the heavy truck cab. The control models of the semi-active cab suspension system using the GA-PID and FLC-PID are shown in Figs. 3(a) and 3(b). Where u and $q(t)$ are the input parameters of the vehicle model while error e (the relative displacement between the cab and vehicle body) and de (its relative velocity) are the input

parameters of the control models.

3.1. The GA-PID controller

The GA is applied to optimize the parameters of the vehicle suspension and engine [3, 17]. The researchers used the optimal suspension parameters of the vehicle, cab, and driver's seat to reduce the RMS accelerations of the vehicle body, the cab, and the driver's seat. In this study, to use the GA for optimizing the PID's parameters, it is assumed that the operating parameters of the PID are from k_j^{min} to k_j^{max} . Based on the vehicle dynamic model and cab suspension system, the GA's program is simulated to choose the optimal parameters k_j^g for the PID to reduce the RMS accelerations of the cab pitch angle and driver's seat. The GA-PID controller is designed in Fig. 3(a). In which error e and de are two input values while three output values are the optimal k_j^g of the GA for the PID controller to control the vehicle system model. The optimization condition of the GA is defined as follows:

$$\begin{aligned} x_{1init} &= e(t), \\ x_{2init} &= \dot{e}(t), \\ k_j^{min} &\leq k_j^g \leq k_j^{max}, \\ f_{min} &= \min\{a_{ws}, a_{w\phi c}\}, \end{aligned} \quad (10)$$

where subscript j denotes p , i , and d , respectively; a_{ws} and $a_{w\phi c}$ are the RMS accelerations of the driver's seat and cab pitching angle determined by [18]:

$$a_{ws} = \sqrt{\frac{1}{T} \int_0^T \ddot{z}_s^2 dt}, \quad a_{w\phi c} = \sqrt{\frac{1}{T} \int_0^T \ddot{\phi}_c^2 dt}. \quad (11)$$

3.2. The FLC-PID controller

In section 3.1, the parameters of the PID controller are optimized by the GA. However, under various operating conditions of the vehicle, the control forces are also changed. This means that the optimal parameters of the PID controller also need to be changed to optimize the control forces. Thus, the FLC is applied to control the parameters of the PID controller. The FLC-PID controller model is plotted in Fig. 3(b). Where error e and de are two input values of the FLC; three output values of the FLC are k_j^f and these values are the input values of the PID controller. It is also assumed that the PID's parameters are changed in a range from k_j^{min} to k_j^{max} . The PID's parameters in the FLC are changed in a range between 0 and 1 by the following linear transfer function [9]:

$$k_j^f = \frac{k_j - k_j^{min}}{k_j^{max} - k_j^{min}} = \frac{k_j - k_j^{min}}{\Delta k_j} \Rightarrow k_j = k_j^f \Delta k_j + k_j^{min}. \quad (12)$$

The design of the FLC is as follows: The cab's suspension system deflection e and its derivation ec are the two input values while the variable k_j^f are three output values. The shape of the membership function for the variables of in-output is the Triangular function and their values are between 0 and 1. The nine linguistic terms are defined in Table 1 [12, 14].

The if-then rules in the FLC are applied to calculate the control forces according to expertise experiences and the designer's knowledge. There are 49 control rules listed in Table 2 and if-then rules are described as follows:

- (1) if $e = nb$ and $ec = nb$ then $u = pb$;
- (2) if $e = nb$ and $ec = nm$ then $u = pb$;
- (3) if $e = pb$ and $ec = pb$ then $u = nb$.

The fuzzy inference system of Mamdani [12] which uses the minimum function and the centroid method has been mainly chosen to calculate the output values of the control force. Thus, the Mamdani's fuzzy inference in Matlab/Fuzzy is applied in this study.

Table 1. Fuzzy linguistic values

Linguistic value	Description	Linguistic value	Description
pb	Positive medium	ns	Negative medium
pm	Positive small	nm	Negative big
ps	Zero	nb	Positive big
ze	Negative small		

Table 2. Rule base for fuzzy control

k_j^f		e						
		nb	nm	ns	ze	ps	pm	pb
ec	nb	pb	pb	pb	pb	ps	ze	ze
	nm	pb	pm	pm	ps	ps	ze	ze
	ns	pb	pm	ps	ps	ze	ze	ns
	ze	pb	ps	ps	ze	ns	ns	nb
	ps	ps	ps	ze	ns	ns	nm	nb
	pm	ze	ze	ns	ns	nm	nm	nb
	pb	ze	ze	ns	nb	nb	nb	nb

Table 3. The design parameters of the different heavy trucks

Parameter	Value	Parameter	Value
m_s (kg)	120	c_1 (Ns/m)	7.029×10^3
m_c (kg)	500	c_2 (Ns/m)	14.09×10^3
m_b (kg)	19 000	k_{t1} (N/m)	1.38×10^6
m_1 (kg)	450	k_{t2} (N/m)	1.38×10^6
m_2 (kg)	1025	c_{t1} (Ns/m)	2.00×10^3
k_s (N/m)	0.20×10^5	c_{t2} (Ns/m)	2.00×10^3
c_s (Ns/m)	0.20×10^3	l_1 (m)	1.60
k_{c1} (N/m)	1.00×10^5	l_2 (m)	0.50
k_{c2} (N/m)	1.00×10^5	l_3 (m)	4.68
c_{c1} (Ns/m)	0.75×10^5	l_4 (m)	0.62
c_{c2} (Ns/m)	0.75×10^5	l_5 (m)	1.10
k_1 (N/m)	1.02×10^5	l_6 (m)	1.00
k_2 (N/m)	5.45×10^5	l_7 (m)	0.20

4. Results and analysis

To evaluate the control performance of the GA-PID and FLC-PID and the ride comfort of the vehicle, based on the vehicle's dynamic parameters listed in Table 3, the semi-active cab suspension systems of the vehicle is controlled based on the GA-PID and FLC-PID.

4.1. Performance of the GA-PID and FLC-PID controllers

The GA-PID and FLC-PID controllers are applied to control the semi-active cab suspension system of the heavy truck under a random surface roughness of ISO level C at speed of 20 m/s. The simulation results are shown in Fig. 4. Fig. 4 shows that the acceleration responses of the driver's seat and cabin pitch angle controlled by the GA-PID and FLC-PID are significantly decreased compared to the passive suspension of the heavy truck cab. Besides, the simulation results also show that the acceleration responses of the driver's seat and cabin pitch angle

controlled by the FLC-PID are smaller than that of the GA-PID. Especially, the calculation results of the RMS accelerations of the driver's seat and cab pitch angle in Fig. 5(a) with a semi-active cab suspension system using the FLC-PID are reduced by 26.45 % and 26.07 % in comparison with the GA-PID; and reduced by 48.16 % and 34.32 % in comparison with the passive cab suspension system. This means that the ride comfort with the FLC-PID is better than that of the GA-PID.

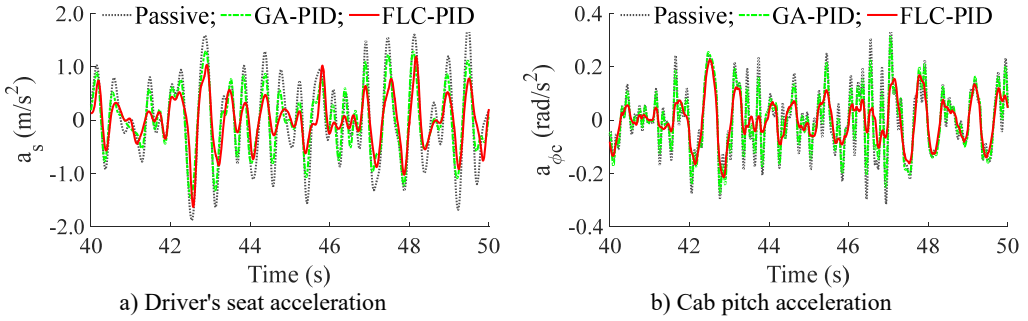


Fig. 4. The acceleration responses of the cab and driver's seat

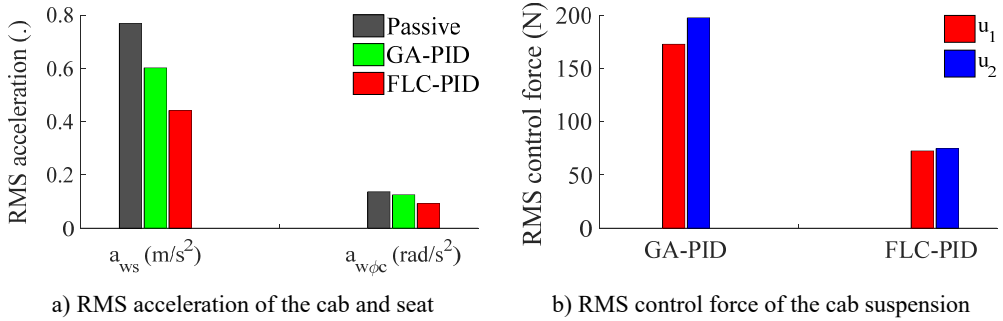


Fig. 5. The RMS values of the accelerations and control forces

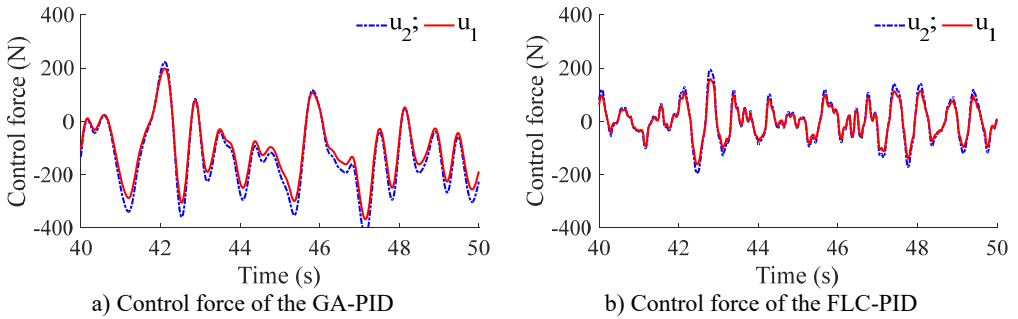


Fig. 6. The control forces of semi-active cab suspension

Additionally, the active control forces of the GA-PID and FLC-PID are also shown in Figs. 6(a) and 6(b). The results indicate that the maximum value of control forces generated by the GA-PID is an approximation of 420 N, while the maximum value of the control forces generated by the FLC-PID is an approximation of 250 N. Besides, the calculation results of the RMS values of the control forces with the GA-PID and FLC-PID in Fig. 5(b) indicate that the RMS control forces of the FLC-PID are smaller than that of the GA-PID. Especially, the RMS control forces of u_1 and u_2 with the FLC-PID are respectively reduced by 57.96 % and 62.02 % in comparison with the GA-PID. Thus, the FLC-PID not only better improves the ride comfort but also reduces the control forces and energy consumption.

4.2. Performance under the different velocities of the vehicle

In this part, a range of the vehicle velocities from 5 km/h to 30 km/h are simulated to compare the performance of the semi-active cab suspension system of the heavy truck with the GA-PID and FLC-PID under the excitation of the random road surface of ISO level C. The simulation results of the RMS accelerations of the driver's seat and cabin pitch angle are shown in 7(a) and 7(b). Besides, the comparison results of the RMS accelerations of the driver's seat and cabin pitch angle between the GA-PID and FLC-PID are also listed in Table 4.

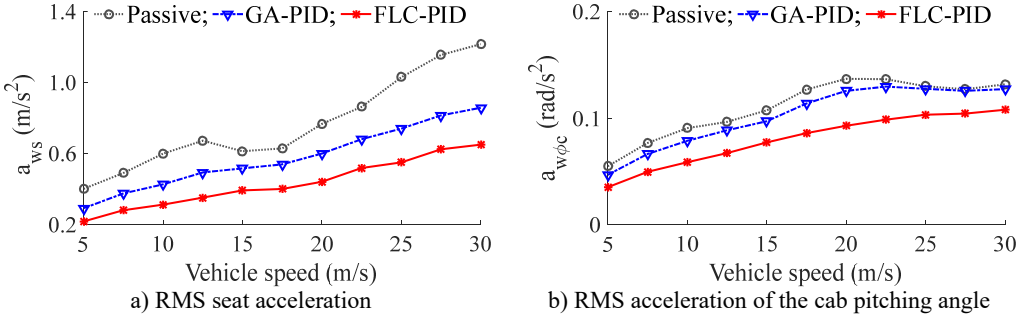


Fig. 7. The RMS accelerations of the cab and driver's seat

Table 4. The RMS acceleration of the driver' seat and the cab pitch angle

Speeds (m/s)	a_{ws} (m/s ²)			$a_{w\phi c}$ (rad/s ²)		
	GA-PID	FLC-PID	Reduction (%)	GA-PID	FLC-PID	Reduction (%)
5	0.2942	0.2199	25.25	0.0470	0.0354	24.68
10	0.4286	0.3160	26.27	0.0789	0.0589	25.35
15	0.5185	0.3945	23.92	0.0973	0.0775	20.35
20	0.6020	0.4428	26.45	0.1258	0.0930	26.07
25	0.7425	0.5524	25.60	0.1277	0.1034	19.03
30	0.8598	0.6517	24.20	0.1274	0.1080	15.23

Figs. 7(a-b) and Table 4 show that in the speed range of the vehicle, the RMS accelerations of the driver's seat and cab pitch angle with a semi-active cab suspension system are significantly decreased in comparison with the passive cab suspension system. Besides, the results with the FLC-PID are also reduced compared to the GA-PID. Therefore, the ride comfort of the heavy truck with the FLC-PID is better improved compared to the GA-PID. Especially, at a range of the vehicle velocities from 20 m/s to 30 m/s, the RMS acceleration of the driver's seat with the FLC-PID is greatly decreased in comparison with both GA-PID and without control; and the RMS accelerations of the driver's seat and cab pitch angle with the FLC-PID at the vehicle speed of 20m/s are greatly reduced by 26.45 % and 26.07 % compared to the GA-PID. Thus, the driver's ride comfort is strongly improved by the FLC-PID.

5. Conclusions

In this study, the combined control methods of the GA-PID and FLC-PID are researched and applied to control the semi-active suspension system of the heavy truck cab. The control results of the FLC-PID are better than that of the GA-PID in improving the ride comfort of the heavy truck under different operating conditions. Especially, when the vehicle moves at a velocity of 20 m/s, the RMS acceleration of the seat of the driver and cab pitching angle with the FLC-PID are greatly reduced by 26.45 % and 26.07 % in comparison with the GA-PID, respectively.

In addition, the control forces and energy consumptions used for the FLC-PID are also lower than that of the GA-PID. Therefore, the FLC-PID control should be applied to the suspension system of the vehicles to improve the vehicle's ride comfort.

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Data availability

The datasets generated during and/or analyzed during the current study are available from the corresponding author on reasonable request.

Conflict of interest

The authors declare that they have no conflict of interest.

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