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Performance analysis of air suspension system of heavy truck with semi-active fuzzy control

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Abstract: In order to analyze and evaluate the performance of the air suspension system of heavy trucks with semi-active fuzzy control, a three-dimensional nonlinear dynamical model of a typical heavy truck with 16-DOF (degree of freedom) is established based on Matlab/Simulink software. The weighted root-mean-square (RMS) acceleration responses of the vertical driver's seat, the pitch and roll angle of the cab, and the dynamic load coefficient (DLC) are chosen as objective functions, and the air suspension system is optimized and analyzed by the semi-active fuzzy control algorithm when vehicles operate under different operation conditions. The results show that the influence of the roll angle of the cab on the heavy truck ride comfort is clear when vehicles move on the road surface conditions of the ISO level D and ISO level E at a velocity over 27.5 m/s. The weighted RMS acceleration responses of vertical driver's seat, the pitch and roll angle of the cab are decreased by 24%, 30% and 25%, respectively, when vehicles move on the road surface condition of the ISO level B at a velocity of 20 m/s. The value of the DLC also significantly decreases when vehicles operate under different operation conditions. Particularly, the DLC value of the tractor driver axle is greatly reduced by 27.4% when the vehicle operates under a vehicle fully-loaded condition on the road surface condition of ISO level B at a velocity of 27.5 m/s.

Key words: heavy truck; dynamic model; air suspension; fuzzy logic control; dynamic load coefficient

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One of the most important requirements of the heavy vehicles is to improve the ride comfort and road friendliness. In order to solve these problems, the heavy vehicle suspension system ought to be able to isolate the sprung mass from road-induced disturbances as well as reducing the dynamic load coefficient (DLC) of tire forces from the heavy vehicle axles within the limit of the working space under different operation conditions.

The passive suspensions were widely used in the heavy vehicles before. The air suspensions and semi-active suspensions are widely studied and improved due to their capability to consume less power and provide better ride quality. Many researchers have reported that using the semi-active suspensions for the vehicles can improve ride comfort and reduce dynamic tire forces[1–2].

Some control methods, such as the fuzzy logic control (FLC)[3–4], FLC-skyhook and FLC-PID control[5–6], FLC-H_{inf}, MR fluid damper and skyhook-NFLC control[7–9], are applied to adjust the damping coefficient of the vehicle suspension system. However, the quarter-car dynamic models are used in most of the research.

Yagiz et al.[10] used a half vehicle with 5-DOF and applied the FLC to control vehicle suspension. Yoshimura et al.[11] used a half vehicle with 6-DOF and applied the FLC-Skyhook damper to control active suspensions. Ieluzzi et al.[12] established a half-heavy truck model and applied the skyhook method to control the suspension system of the tractor driver. All the above research aims to improve vehicle ride comfort.

The air suspension system with air springs is chosen instead of conventional steel springs since air springs can effectively reduce the effect of the disturbance from the road input and also easily adjust to the ride height onboard. Buhari et al.[13] studied the effect of the air suspension of heavy trucks by comparing the dynamic load coefficient of tire forces with steel suspension, and showed that the DLC's value was greatly decreased by air suspension. Xie et al.[14] applied the FLC-PID controller to control the semi-active air suspension of heavy vehicles by using the half-vehicle model and showed that the ride comfort and tire loads are significantly improved. All the proposed results show that the performance of the air suspension control system can not only increase ride comfort but also reduce road damage. However, the effect of the roll vibration of the full vehicle on ride comfort has not yet been considered in these studies.

This paper proposes a 3D dynamic model of a typical heavy truck with 16-DOF; the weighted RMS acceleration responses of vertical driver's seat, the pitch and roll angle of the cab, and the dynamic load coefficient (DLC) val-

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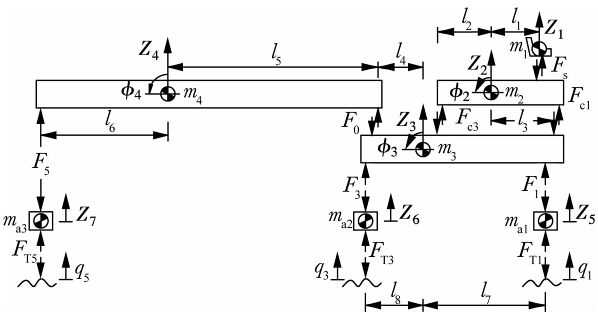
ue are chosen as the objective function; and a FLC program is developed based on Matlab/Simulink software. The semi-active fuzzy controller is then applied to control the damping coefficients of the air suspension system and the performance of the air suspension system is analyzed based on the objective functions when vehicles operate under different operation conditions.

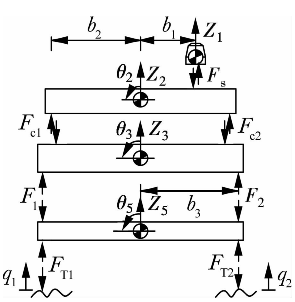
1 Heavy Truck Model

1.1 Heavy truck dynamic model

A three-axle heavy truck with the dependent suspension systems for the steering axle, the tractor driver axle and the trailer axle is selected for vehicle dynamic analysis. A 3D model of vehicle with 16-DOF is established to analyze the performance of the air suspension system with semi-active fuzzy control (see Fig. 1).



(a)



(b)

Fig. 1 3D dynamic models of the three-axle heavy truck. (a) Side view; (b) Front view

In Fig. 1, Z_i are the vertical displacements at the centre of gravity of the driver's seat, the cab, the tractor driver, the trailer and the axles; ϕ_2, ϕ_3, ϕ_4 and θ_k are the angular displacements at the centre of gravity of the cab, the tractor driver, the trailer and the axles; m_j are the sprung masses of the driver's seat, the cab, the tractor driver and the trailer; m_{a1}, m_{a2} and m_{a3} are the unsprung masses of the steering axle, the tractor driver axle and the trailer axle; F_s, F_{cj} and F_u are the dynamic reaction forces of the suspension systems of the driver's seat, the cab, the tractor driver and the trailer; F_0 is the dynamic reaction force of the articulation connection between the tractor driver and trailer; F_{Tu} are the dynamic reaction forces of the wheels; q_u is the random road surface; l_m, b_1, b_2 and b_3 are the distances of the vehicle ($i = 1, 2, \ldots, 7$; $j = 1, 2, 3, 4$; $k = 2, 3, \ldots, 7$; $u = 1, 2, \ldots, 6$; $m = 1, 2, \ldots, 8$).

Based on the heavy truck dynamic model in Fig. 1, Newton's second law is chosen in this study. The general dynamic differential equation for the three-axle heavy truck is given by the following matrix form:

$$\boldsymbol{M}\ddot{\boldsymbol{Z}} + \boldsymbol{C}\dot{\boldsymbol{Z}} + \boldsymbol{K}\boldsymbol{Z} = \boldsymbol{C}_{\mathrm{T}}\dot{\boldsymbol{Q}} + \boldsymbol{K}_{\mathrm{T}}\boldsymbol{Q} \tag{1}$$

where $\boldsymbol{Z}$ is the vector of displacement; $\boldsymbol{M}$ is the mass matrix; $\boldsymbol{C}$ is the damping matrix of the suspension system; $\boldsymbol{K}$ is the stiffness matrix of the suspension system; $\boldsymbol{C}_{\mathrm{T}}$ is the damping matrix of the wheel system; $\boldsymbol{K}_{\mathrm{T}}$ is the stiffness matrix of the wheel system; $\boldsymbol{Q}$ is the vector of excitation of the road surface.

1.2 Semi-active controlled air suspension model

The suspension systems of the tractor driver and the trailer are used by the rolling lobe air springs and viscous dampers which are controlled by semi-active fuzzy control, as shown in Fig. 2.

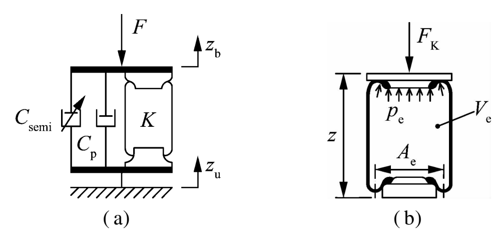


Fig. 2 Air suspension model. (a) Semi-active air suspension; (b) Air spring

In Fig. 2(a), the equation of dynamic reaction forces of the air suspension system can be written as

$$F = K(z_{\mathrm{u}} - z_{\mathrm{b}}) + (C_{\mathrm{p}} + C_{\mathrm{semi}})(\dot{z}_{\mathrm{u}} - \dot{z}_{\mathrm{b}}) \tag{2}$$

where K is the air spring stiffness coefficient; C_{p} is the passive damping coefficient; C_{semi} is the semi-active damping coefficient; $z = z_{\mathrm{u}} - z_{\mathrm{b}}$ and $\dot{z} = \dot{z}_{\mathrm{u}} - \dot{z}_{\mathrm{b}}$ are the relative displacement and velocity of the air suspension.

The air spring stiffness coefficient K is determined based on the experimental data and the laws of the thermodynamics method[14]. In this study, the air spring stiffness coefficient is calculated by a new method based on the variation of volume, area, and other structural parameters of the interior air spring.

The air spring stiffness can thus be obtained by the derivative of ratio elastic force and the displacement of the air spring, as shown in Fig. 2(b), as follows:

$$K = \frac{\mathrm{d}(p_{\mathrm{e}}A_{\mathrm{e}})}{\mathrm{d}z} = p_{\mathrm{e}}\frac{\mathrm{d}A_{\mathrm{e}}}{\mathrm{d}z} - nA_{\mathrm{e}}(p_0 + p_{\mathrm{a}})\frac{1}{V_{\mathrm{e}}}\frac{\mathrm{d}V_{\mathrm{e}}}{\mathrm{d}z} \tag{3}$$

The effective volume and area are defined as

$$V_{\mathrm{e}} = V_0 - \alpha_1 z, \quad A_{\mathrm{e}} = A_0 + \alpha_2 z \tag{4}$$

Based on the laws of thermodynamics[15], if the compression or expansion stroke of the air spring changes quickly, it can be regraded to be an adiabatic process. Thus, the air state of the air spring can be defined as

$$(p_e + p_a) V_e^n = (p_0 + p_a) V_0^n \tag{5}$$

where V_e, A_e and z are the effective volume, area and the instantaneous height variation of air spring; V_0 and A_0 are the initial effective volume and area; α_1 and α_2 are the change of V_e and A_e with respect to z; p_0, p_e are the air pressures of initial and final state; p_a is the standard atmospheric pressure (0.1 MPa); and n is the specific heat ratio ($n = 1.33$).

Substituting Eqs. (4) and (5) into Eq. (3) and applying Eq. (3) for the stiffness of air springs, we have

$$K_i = \left[(p_{0i} + p_a) \left(\frac{V_{0i}}{V_{0i} - \alpha_l z_i} \right)^n - p_a \right] \alpha_{l+1} + \left[n(p_{0i} + p_a) \left(\frac{A_{0i} + \alpha_{l+1} z_i}{V_{0i} - \alpha_l z_i} \right) \left(\frac{V_{0i}}{V_{0i} - \alpha_l z_i} \right)^n \right] \alpha_l \tag{6}$$

where $l = 1$ when $i = 1, 2$; $l = 3$ when $i = 3, 4, 5, 6$.

2 Design of Fuzzy Logic Control Model

The FLC was created by Zadeh in 1965. It has been widely applied in various situations and fields. The fuzzy logic-based control for semi-active suspension of vehicle is proposed and the capability for the improvement of ride comfort are studied by simulation. In this study, in order to control the semi-active air suspension system, six passive air suspension systems should be controlled separately. Thus, six different fuzzy controllers are designed. However, the design process of these controllers are the same, thus a specific fuzzy control is designed and applied to control the air suspension system.

The FLC consists of a fuzzification interface, a fuzzy inference system and a defuzzification interface. First, the crisp values in fuzzification are transformed into linguistic variables. The fuzzy inference system is then used by the fuzzy rule in accordance with the inference rule. Finally, the linguistic variables are transformed back to crisp values through defuzzification[3-4]. In this study, the relative displacement z and the relative velocity $\dot{z}$ are considered as two input variables, while the damping coefficient C_{semi} is the output of the semi-active fuzzy controller.

The nine linguistic variables of input and output variables are defined, such as the positive very big (PVB), positive big (PB), positive medium (PM), positive small (PS), zero (Z), negative small (NS), negative medium (NM), negative big (NB), negative very big (NVB) and y_i ($i = 1, 2, \ldots, 9$).

The membership functions for input and output variables of the semi-active air suspension system are represented by a fuzzy set. The shape of membership functions is the triangular function and the value of the degree of memberships (DOM) is between 0 and 1, as shown in Fig. 3.

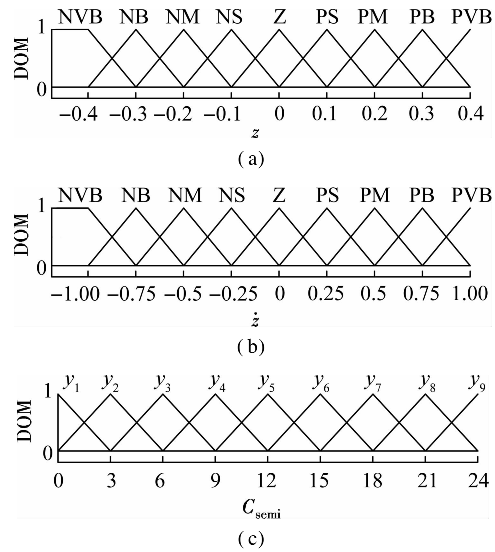


Fig. 3 Membership functions for input and output variables of the air suspension system. (a) z; (b) $\dot{z}$; (c) C_{semi}

In this fuzzy controller, the if-then rules are applied to describe the relationship of z, $\dot{z}$ and C_{semi} according to the designer's knowledge and experience. There are at most 81 possible rules, and the fuzzy rules are given in Tab. 1.

Tab. 1 Rules for fuzzy control

$\dot{z}$	z								
	NVB	NB	NM	NS	Z	PS	PM	PB	PVB
NVB	y_5	y_9	y_7	y_6	y_5	y_4	y_4	y_3	y_3
NB	y_3	y_5	y_8	y_6	y_5	y_4	y_3	y_3	y_3
NM	y_1	y_2	y_5	y_7	y_5	y_3	y_3	y_3	y_2
NS	y_1	y_1	y_1	y_5	y_5	y_3	y_2	y_2	y_2
Z	y_1	y_1	y_1	y_1	y_5	y_1	y_1	y_1	y_1
PS	y_2	y_2	y_2	y_3	y_5	y_5	y_1	y_1	y_1
PM	y_2	y_3	y_3	y_3	y_5	y_7	y_5	y_2	y_1
PB	y_3	y_3	y_3	y_4	y_5	y_6	y_8	y_5	y_3
PVB	y_3	y_3	y_4	y_4	y_5	y_6	y_7	y_9	y_5

The fuzzy inference system is selected by the minimum function and the centroid method of Karray and Mamdani et al[16-17]. In this paper, the fuzzy inference system of Mamdani is used to control the air suspension systen model.

3 Evaluation Criteria

3.1 Basic evaluation method

According to the international standard ISO 2631-1[18], the effect of vibration on driver ride comfort is evaluated based on the weighted root-mean-square (RMS) acceleration response, which is defined as

$$a_{wz} = \left[\frac{1}{T} \int_0^T a_{wt}^2 \mathrm{d}t \right]^{1/2} \tag{7}$$

where a_{wt} is the acceleration (translational and rotational) depending on the time of measurement T.

The weighted RMS acceleration responses of the vertical driver's seat a_{ws}, the pitch and roll angle of the cab, a_{wpc} and a_{wrc}, can be calculated from Eq. (7), and the results are compared with a_{wz}, as given in the international standard ISO 2631-1.

3.2 Dynamic load coefficient

The DLC is frequently used to characterize dynamic loading of axles and it is defined as[13]

$$\mathrm{DLC} = \frac{F_{\mathrm{T,rms}}}{F_{\mathrm{st}}} \tag{8}$$

where $F_{\mathrm{T,rms}}$ is the RMS of the vertical dynamic wheel load, and F_{st} is the average vertical wheel load.

The DLC's value is in the range of 0.05 to 0.3 under normal operation conditions. The DLC's value can reduce to zero when the wheel moves on a special smooth road or increases up to 0.4 when the wheel-road contact is broken[19]. In this study, the dynamic load coefficient is used to analyze the vehicle-road interaction.

4 Road Surface Roughness

The road surface is a random excitation, which impacts not only on vehicle-road interaction but also on the vehicle's fatigue life. The random excitation of road surface roughness can be represented by a randomly modulated periodic. The general form of the displacement power spectral density (PSD) of road surface roughness is determined by the experimental formula[20]:

$$G_{\mathrm{q}}(n) = G_{\mathrm{q}}(n_0)\left(\frac{n}{n_0}\right)^{-\omega} \tag{9}$$

where n is the space frequency; n_0 is the reference space frequency, $n_0 = 0.1\ \mathrm{m}^{-1}$; $G_q(n)$ is the road roughness constant or PSD of the road surface under the reference space frequency n_0; ω is the frequency index which determines the frequency configuration of the PSD of the road surface ($\omega = 2$ rad/s). The random road is assumed to be a zero-mean stationary Gaussian random process, and it can be generated by the Fourier transformation as

$$q(t) = \sum_{i=1}^{N} \sqrt{2G_{\mathrm{q}}(n_i)\Delta n}\cos(2\pi n_k t + \phi_i) \tag{10}$$

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

According to the standard ISO/TC 8068[20], the road surface roughness is established in this study.

5 Results and Analysis

The purpose of this study is to analyze the performance of the air suspension system based on the objective functions when vehicles operate under different operation conditions. The parameters of the three-axle heavy vehicle is given in Tab. 2.

5.1 Simulation control air suspension system

In order to control the air suspension system of heavy trucks with the semi-active fuzzy control, simulations are carried out when vehicles operate under different operation conditions. The simulation results of the acceleration responses of driver's seat, the pitch and roll angle of the cab when vehicles move on the road surface condition of ISO level B at $v = 20$ m/s are shown in Fig. 4.

With the passive air suspension system of heavy trucks, the weighted RMS acceleration response of the cab roll angle is increased by 16.6% in comparison with the cab pitch angle (see Fig. 4(c) and Tab. 3). Thus, the roll angle of the cab clearly affects ride comfort and driver's health.

Tab. 2 Parameters of the three-axle heavy truck[21]

Parameter	Value	Parameter	Value
m_{s}/kg	85	b_2/m	0.4
m_{c}/kg	500	b_3/m	1.0
m_{b}/kg	3 600	$A_{01,2}$/m^2	0.028 1
m_{t}/kg	12 500	$A_{03,4}$/m^2	0.090 6
m_{a1}/kg	270.1	$A_{05,6}$/m^2	0.090 6
m_{a2}/kg	520.4	$V_{01,2}$/m^3	0.007 8
m_{a3}/kg	340.0	$V_{03,4}$/m^3	0.032 2
l_1/m	0.2	$V_{05,6}$/m^3	0.032 2
l_2/m	0.8	$p_{01,2}$/MPa	0.134
l_3/m	1.3	$p_{03,4}$/MPa	2.860
l_4/m	4.1	$p_{05,6}$/MPa	1.035
l_5/m	6.9	α_1	0.059 0
l_6/m	4.0	α_2	0.018 6
l_7/m	1.2	α_3	0.127 0
l_8/m	4.8	α_4	0.030 7
b_1/m	0.3		

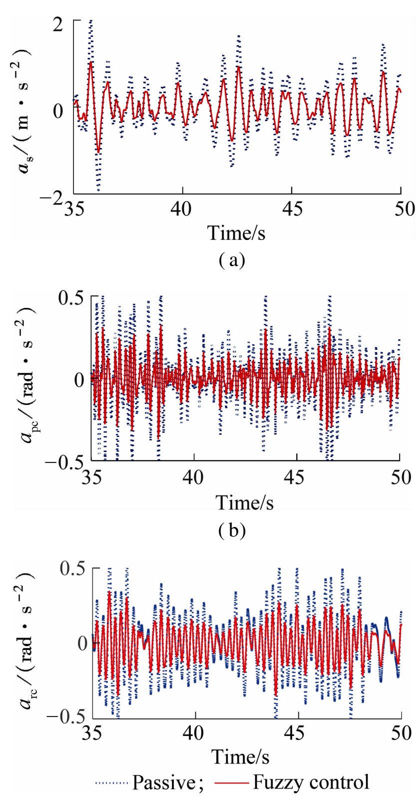


Fig. 4 The accelerations of the driver's seat and cab. (a) Driver's seat heave; (b) Cab pitch angle; (c) Cab roll angle

Tab. 3 Control performance with the fuzzy control

Parameters	Passive	Fuzzy control	Reduction/%
$a_{ws}/(\mathrm{m \cdot s^{-2}})$	0.42	0.32	24.0
$a_{wpc}/(\mathrm{rad \cdot s^{-2}})$	0.10	0.07	30.0
$a_{wrc}/(\mathrm{rad \cdot s^{-2}})$	0.12	0.09	25.0
DLC	0.09	0.07	22.2
$F_{T,rms}$/kN	4.62	3.51	24.0

The air suspension system is controlled by the semi-active fuzzy control, as shown in Figs. 4(a), (b) and (c), the acceleration responses of the driver's seat, the pitch and roll angle of the cab are reduced in comparison with the passive air suspension system. The results in Tab. 3 show that the weighted RMS acceleration responses of the driver's seat, the pitch and roll angle of the cab are significantly reduced by 24%, 30% and 25%. According to the standard ISO 2631-1[18], the driver feels a little uncomfortable. The DLC's value and the RMS dynamic tire forces on the tractor driver axle are also decreased by 22.2% and 24.0% in comparison with the passive air suspension system. Consequently, the air suspension system of heavy truck with the semi-active fuzzy control not only increases the ride comfort but also reduces the road damage.

5.2 Effect of road surface roughness

Five road conditions from Level A (very good) to Level E (very poor) in ISO/TC 8068[20] are chosen to simulate and analyze the effect of road surface roughness on the air suspension system of heavy trucks at $v = 20$ m/s.

Fig. 5 shows that compared with the cab pitch angle, the weighted RMS acceleration response of the cab roll angle is increased with the passive air suspension system. Specifically, the weighted RMS acceleration response of the roll angle of the cab is increased by 25.5% and 30.8% on the road surface conditions of ISO level D and ISO level E. Thus, the influence of the roll angle of the cab on the heavy truck ride comfort is very clear.

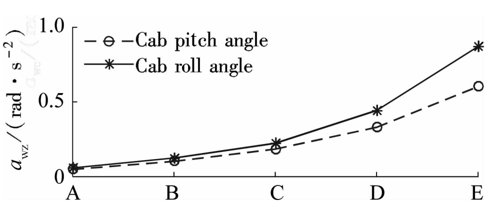


Fig. 5 Weighted RMS acceleration responses of cab pitch and roll angle with the passive air suspension system

The simulation results in Tab. 4 show that the weighted RMS acceleration responses of the vertical driver's seat, the pitch and roll angle of the cab are significantly reduced in the air suspension system of heavy trucks with semi-active fuzzy control; particularly, when vehicles move on the road surface condition of ISO Level B. They decrease by 24%, 30% and 25%, respectively. However, the weighted RMS acceleration responses of driver's seat on ISO Level E road surfaces is increased by 3.8%.

Tab. 4 Percentage reduction of weighted RMS acceleration responses %

Parameters	Road surface roughness				
	Level A	Level B	Level C	Level D	Level E
a_{ws}	12.3	24.0	12.0	23.8	−3.8
a_{wpc}	24.0	30.0	17.0	16.4	10.7
a_{wrc}	25.0	25.0	16.7	13.9	3.2

Fig. 6 shows that the DLC's values of dynamic tire forces at each axle are significantly reduced in comparison with those of the passive air suspension system, particularly at the tractor driver axle. However, the values at the tractor driver axle are the maximum and thus the impact on road damage is also the greatest.

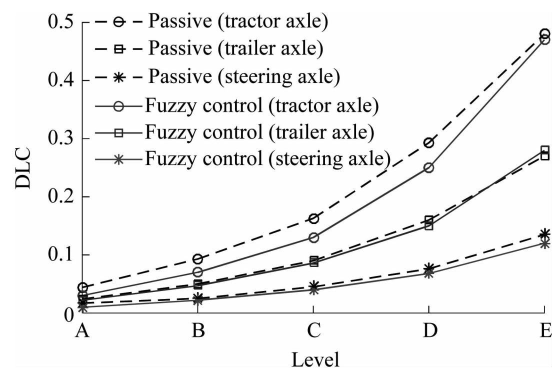


Fig. 6 The DLC's values of dynamic tire forces on different road surfaces

The results in Tab. 5 also show that the maximum DLC's values at the tractor axle of the air suspension system with semi-active fuzzy control on road surface conditions from ISO level A to ISO level D are 0.03 to 0.25. Thus, the DLC's value is under normal operation conditions[18-19]. However, the DLC's value is greatly increased by 0.47 when vehicles move on road surface under the condition of ISO Level E, and when the wheels-road contact of the tralor driver axle is broken.

Tab. 5 Results of DLC on the tractor axle

Road surface	DLC		Reduction/%
	Passive	Fuzzy control	
Level A	0.04	0.03	25.0
Level B	0.09	0.07	22.2
Level C	0.18	0.13	27.7
Level D	0.29	0.25	13.8
Level E	0.52	0.47	9.6

5.3 Effect of vehicle velocity

A range of vehicle velocities under different loading conditions (half-loaded, fully-loaded and over-loaded vehicles) on the road surface of ISO Level B is chosen to simulate and analyze the performance of FLC.

Figs. 7(a), (b) and (c) show that the weighted RMS acceleration responses of the driver's seat, the pitch and roll angle of the cab are significantly decreased in comparison with those of the passive air suspension system under different loading conditions. When the vehicle velocities range from 10 to 12. 5 m/s, the weighted RMS acceleration response of the roll angle of the cab is greatly decreased. Meanwhile, the weighted RMS acceleration response of the pitch angle is slightly decreased. It is found that the weighted RMS acceleration responses of the driver's seat, the pitch and roll angle of the cab are significantly reduced at velocities from 17. 5 to 20 m/s. This implies that dirver's ride comfort is significantly improved. In fact, the weighted RMS acceleration responses of the pitch and roll angle of the cab increase at a velocity of 27. 5 m/s in both the passive and semi-active air suspension systems.

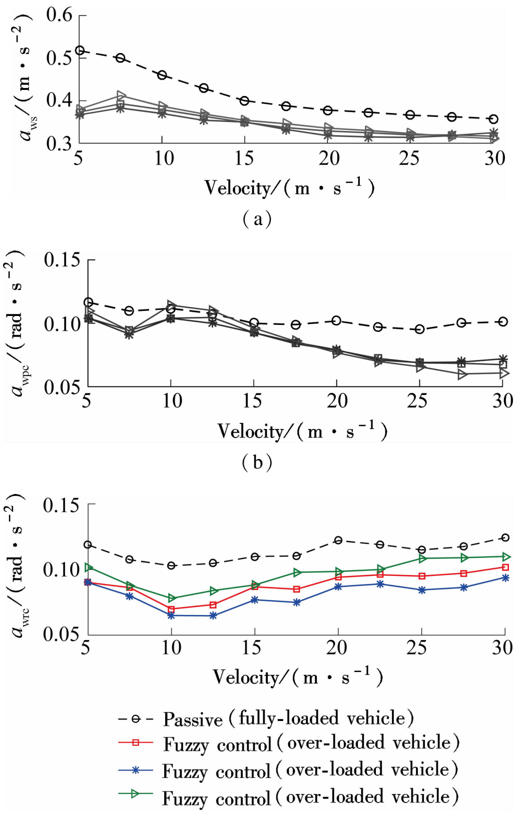


Fig. 7 Weighted RMS acceleration responses of the vertical driver's seat and the angles of the cab. (a) Driver's seat heave; (b) Cab pitch angle; (c) Cab roll angle

As shown in Fig. 8, the DLC's value is increased when the vehicle operates under the fully-loaded condition with a passive air suspension system.

When vehicles operate under the same condition, the DLC's value is significantly reduced by the air suspension system with the semi-active fuzzy control. The maximum DLC's are reduced by 17. 9% , 22. 2% and 27. 4% , respectively, when vehicles move on the road surface condition of ISO Level B at 7. 5, 20 and 27. 5 m/s. In addition, the DLC's value strongly depends on the loading conditions. The value of DLC is greatly increased in

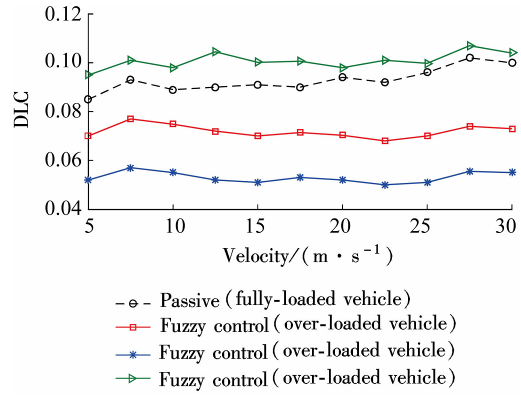


Fig. 8 The DLC's values on the tractor driver axle

the case of the half-loaded vehicle and it is also greatly reduced in the case of the over-loaded vehicle in comparison with the fully-loaded vehicle.

6 Conclusions

1) The influence of the cab roll angle on the heavy truck ride comfort is clear when vehicles move on the road surface conditions of ISO Level D and ISO Level E at a velocity of over 27. 5 m/s.

2) The weighted RMS acceleration responses of the vertical driver's seat, the pitch and roll angle of the cab are greatly decreased by 24% , 30% , 25% and slightly decreased by −3. 8% , 10. 7% and 3. 2% , respectively, when vehicles move on the road surface conditions of ISO Level B and ISO Level E at a velocity of 20 m/s.

3) The DLC's value is also significantly decreased when vehicles operate under different operation conditions. Specifically, the DLC's value of the tractor driver axle is reduced by 27. 4% when vehicles operate under full-loading conditions on the road surface condition of ISO Level B at a velocity of 27. 5 m/s.

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基于半主动模糊控制的重型卡车空气悬架系统性能分析

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摘要:为了分析和评价具有半主动模糊控制的重型卡车空气悬挂系统性能,基于 Matlab/Simulink 软件建立了具有 16 自由度的三维非线性动力学模型. 以座椅的平均垂直加速度响应、驾驶室的俯仰和倾斜角及动载系数(DLC)为目标函数,用半主动模糊控制方法对不同工况下的车辆空气悬挂系统进行了优化分析. 结果表明:在 ISO D 级和 ISO E 级路面上,当车速超过 27.5 m/s 时,驾驶室侧倾角对重型卡车乘坐舒适性影响非常明显;在 ISO B 级路面车速为 20 m/s 时,车辆座椅的平均垂向加速度、驾驶室俯仰角及驾驶室倾侧角分别降低了 24% ,30% 和 25% . 此外,在不同路况条件下,车辆的动载系数均有较大的降低. 特别地,在 ISO B 级路面车速为 27.5 m/s 且满载时,车辆驱动轴处的动载系数降低了 27.4% .

关键词:重型卡车;动力学模型;空气悬架;模糊逻辑控制;动载系数

中图分类号:U461.3