International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 07 Issue: 12 | Dec 2020 www.irjet.net p-ISSN: 2395-0072

Effectiveness Evaluation of Passive and Semi-active Cab Suspension Systems for the Improvement of A Semi-Trailer Truck Ride Comfort

Vu The Truyen[1], Ho Thi Thanh Mai[2]

[1,2] Department of Basic Studies, Thai Nguyen Training Facility, University of Transport Technology, Ha Noi, Vietnam

---***---

Abstract – *To evaluate the performance of semi-active cab suspension system for the improvement of a semi-trailer truck ride comfort, a half-vehicle dynamic model of semi-trailer truck is established for the effectiveness evaluation of passive and semi-active cab suspension systems for the improvement of vehicle ride comfort. The fuzzy logic controller (FLC) is designed to control the damping coefficient of cab suspension system. The weighted r.m.s acceleration responses of the vertical driver's seat and pitch angle of the cab according to the ISO 2631:1997(E) standard are chosen as objective functions. The results show that* the peak acceleration amplitudes with semi-active cab suspension system respectively *reduce in comparison with passive cab suspension system under different operating conditions.* Especially, *the a_{ws} and $a_{w\phi c}$ values with semi-active cab suspension system significantly decrease by 17.4 % and 25.4 % in comparison with passive cab suspension system when vehicle moves on the ISO class B road surface at v=70 km/h and full load.*

Key Words: Semi-trailer truck, Cab, Suspension system, Ride comfort.

1. INTRODUCTION

In order to improve the ride comfort and road - friendliness of heavy trucks, the suspension systems play an important role in vehicle ride performance. The suspension parameters of the heavy truck were analyzed for its effects on road surface friendliness based on a nonlinear full-vehicle model[1]. The effects of the heavy truck dynamic parameters such as vehicle suspension, cab suspension and driver suspension on vehicle ride comfort were analyzed based on 3D dynamic model[2]. The ride performance of the hydro-pneumatic suspension system on vehicle was analyzed and compared to two other suspension systems such as rubber and leaf springs of suspension systems when vehicle operates under different operating conditions[3]. A 3D dynamic model with 14 degrees of freedom was developed with the dynamic models of the traditional and new air suspension systems to compare the performance of the air suspension systems for reducing the negative impacts on the road surface when vehicle moves on the different road conditions[4]. A full dynamic model of a heavy vehicle equipped with three different suspension systems was established to evaluate the effect of suspension characteristics on dynamic load coefficient (DLC)[5]. In order to improve the performance of the suspension systems in the direction of vehicle ride comfort and road surface friendliness. The design parameters for the air suspension systems were optimized using genetic algorithm (GA)[6, 7]. The ride performance of the air suspension system of heavy trucks with semi-active fuzzy control was analyzed and evaluated based on a three-dimensional nonlinear dynamical model of a typical heavy truck with 16-DOF(degree of freedom)[8]. The ride performance of heavy truck's semi-active isolation systems was evaluated based on a half-vehicle dynamic model with three control cases[9]. The cab suspension system of a semi-trailer truck was controlled to improve the vehicle ride comfort based on a nine-degree of freedom model[10]. The electromagnetic actuator was used as an active element in cab suspension system of a heavy truck based on a dynamic modeling of a three-axle heavy vehicle[11]. The Fuzzy Logic Controller (FLC) was applied to control the cab's isolation system of heavy truck, and the program was developed based on a 3-D dynamic model with 13 DOF and Matlab/Simulink software to simulate and calculate the r.m.s acceleration of the vertical driver's seat, the cab pitch angle and roll angle under different road surface conditions[12].

The main objective of this study is to evaluate the ride performance of semi-active cab suspension system of a semi-trailer truck ride in comparison with passive cab suspension system in the direction of vehicle ride comfort. A half-vehicle dynamic model of semi-trailer truck is established for the effectiveness evaluation of passive and semi-active cab suspension systems for the improvement of vehicle ride comfort. The fuzzy logic controller (FLC) is designed to control the damping coefficient of cab suspension system. The weighted r.m.s acceleration responses of the vertical driver's seat and the cab's pitch angle according to the ISO 2631:1997(E) standard[12] are chosen as objective function.

2. VEHICLE DYNAMIC MODEL

A semi-tractor-trailer truck with the dependent leaf spring of suspension systems is selected for the effectiveness evaluation of passive and semi-active cab suspension systems for the improvement of vehicle ride comfort. A semi-trailer truck dynamic model with 12 degrees of freedom is established for comparing the ride performance of the passive and semi-active suspension systems, as shown in Fig. 1.

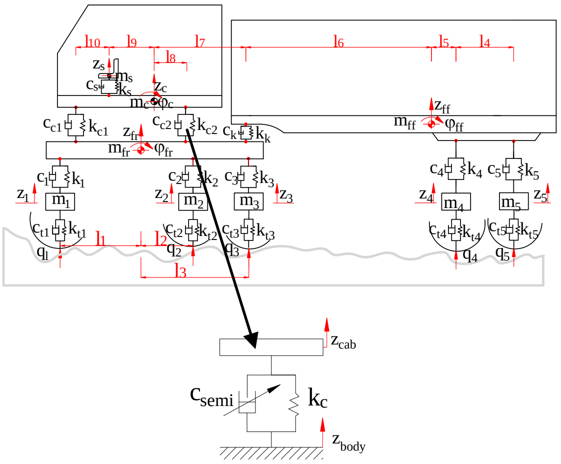


Fig-1: Half-vehicle dynamic model of a semi-trailer truck

In Fig-1, m_i are the wheel masses of the semi-trailer truck axles, respectively; m_{fr}, m_{ff}, m_c and m_s are the masses of the tractor body, trailer body, cab body, and driver seat, respectively; l_k are the calculation distances; k_{ti} and c_{ti} are the tire stiffness and damping coefficients, respectively; k_i and c_i are the vehicle suspension stiffness and damping coefficients, respectively; k_{cj} and c_{cj} are the cab suspension stiffness and damping coefficients, respectively; k_s and c_s are the driver seat stiffness and damping coefficients; z_i are the vertical body displacement of the axles, respectively; z_{fr} and z_{ff} are the vertical body displacement of the tractor and trailer; z_c is the vertical body displacement of cab; z_s is the vertical body displacement of driver seat, respectively; φ_{fr} and φ_{ff} are the pitch response of the tractor and trailer; φ_c is the pitch response of cab; q_i is the excitation of the road surfaces (i=1÷5,j=1÷2, k=1÷10).

The equations of motion for a half-vehicle dynamic model of a semi-trailer truck using Newton's second law of motion are written as follows:

$$m\ddot{z} + c\dot{z} + kz = c_t\dot{q} + k_t q \tag{1}$$

Where, m, c, k, c_t and k_t are the mass matrix of the vehicle, damping matrix of the suspension system, stiffness matrix of the suspension system, damping matrix of the wheel system and stiffness matrix of the wheel system, respectively; z is the vector of displacement; q is the vector of excitation of road surface.

Road roughness random excitation: In this study, the white noise method is used to generate the time domain road surface[13]. Random road roughness is typically assumed to be homogeneous and isotropic Gaussian random process and its statistical characteristics can be described by power spectral density (PSD). According to the International Standards Organization (ISO) 8608 [14], PSD of road roughness can be defined as Eq. (13).

$$G_q(n) = G_q(n_0)\left(\frac{n}{n_0}\right)^{-w} \tag{2}$$

where, n is spatial frequency in m^{-1}, n_0 is reference spatial frequency with a value of $0.1m^{-1}$, $G_q(n_0)$ is PSD value for reference spatial frequency in m^3, w is termed waviness, and reflects approximate frequency structure of the road profile, commonly taken as w=2. Also, $G_q(n_0)$ represents different road levels ranging from A (very good) to H (very poor), depending on its values.

Eq. (2) leads to estimation errors (overrated phenomenon) especially at low frequencies. To deal with this problem, PSD of road roughness based on rational white noise signal is proposed [16], and Eq.(3) can be modified as follows.

$$G_q(\omega) = \frac{\alpha\rho^2}{\pi(\alpha^2 + n^2)} \tag{3}$$

where, α is constant related to road feature, ρ^2 represents variance of road roughness.

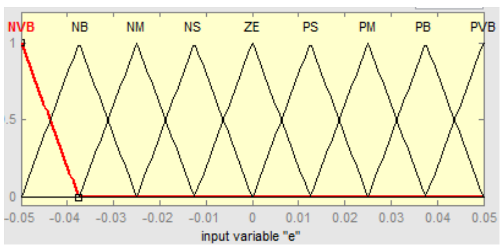
Time domain representation of the road can be given as

$$\dot{q}(t) + \alpha v q(t) = \mathrm{w}(t) \tag{4}$$

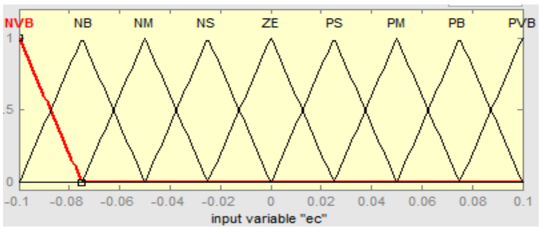
where, q(t) is the random road excitation, m; v represents velocity of the vehicle, m/s; w(t) is the white noise sequence.

3. FUZZY LOGIC CONTROLER (FLC)

The design of a FLC is as follows: the relative displacement of cab suspension systems "e" and its relative velocity "ec " are the two input variables while the damping coefficient "c_{semi}" is the output value of semi-active cab suspension system. The linguistic variables of input and output variables are defined by the positive very big (PVB), positive big (PB), positive medium (PM), positive small (PS), zero (ZE), negative very big (NVB), negative big (NV), negative medium(NM), negative small (NS) and the membership functions for their variables are represented by a fuzzy set. The shape of membership functions is the Triangular function, shown as Fig.2.



(a)



(b)

Fig- 2: The membership functions for input variables

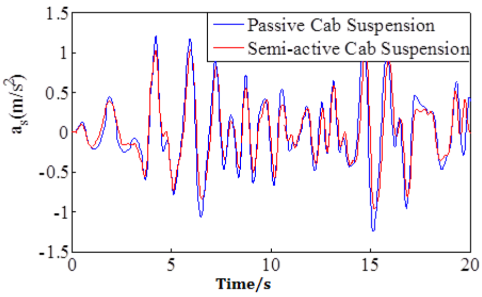
The if-then rules are applied to describe the e, ec and c_{semi} according to the designer's knowledge and experience. There are at most 81 possible rules, and fuzzy rules are given in Tab.1.

Table- 1: Rules for fuzzy control

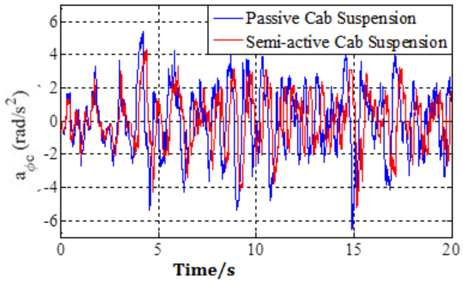
c_{semi}		e								
		NVB	NB	NM	NS	ZE	PS	PM	PB	PVB
ec	MVB	y_5	y_9	y_7	y_6	y_5	y_4	y_4	y_3	y_3
	NB	y_3	y_5	y_8	y_6	y_5	y_4	y_3	y_3	y_3
	NM	y_1	y_2	y_5	y_7	y_5	y_3	y_3	y_3	y_2
	NS	y_1	y_1	y_1	y_5	y_5	y_3	y_2	y_2	y_2
	ZE	y_1	y_1	y_1	y_1	y_5	y_1	y_1	y_1	y_1
	PS	y_2	y_2	y_2	y_3	y_5	y_5	y_1	y_1	y_1
	PM	y_2	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

4. SIMULATION AND DISCUSSION

In order to solve the general dynamic differential equation of a semi-tractor-trailer truck in section 2, Matlab/Simulink software is used with a set of parameters of vehicle in the reference[17] to control the damping coefficient of the cab suspension system with fuzzy rules in Tab.1. The simulation results of the time domain acceleration responses of the vertical driver's seat and the cab's pitch angle when vehicle moves on the ISO class B road surface at v=70 km/h and full load are shown in Fig.3.



(a)The acceleration of cabin pitch angle



(b) The vertical acceleration of driver's seat

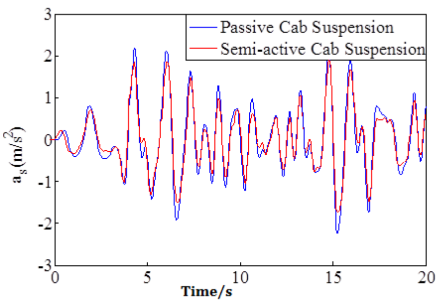
Fig-3: The time domain responses of the cab vertical, pitch angular accelerations when vehicle moves on the ISO class B road surface at v=70 km/h and full load

From the results of Fig.3 we see that the peak acceleration amplitudes with semi-active cab suspension system respectively reduce in comparison with passive cab suspension system. The values of the weighted r.m.s acceleration responses of the vertical driver's seat (a_{ws}) and the cab's pitch angle($a_{w\phi c}$) accoding to the standard ISO 2631-1[15] are determined and shown in Tab.2. The results in Tab.2 showed that the a_{ws} and $a_{w\phi c}$ values with semi-active cab suspension system significantly decrease by 17.4 % and 25.4 % in comparison with passive cab suspension system when vehicle moves on the ISO class B road surface at v=70 km/h and full load.

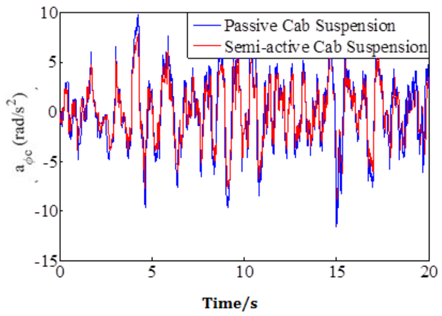
Table-2: The a_{ws} and $a_{w\phi c}$ values when vehicle moves on the ISO class B road surface at v=70 km/h and full load

	a_{ws} (m/s^2)	$a_{w\phi c}$ (rad/s^2)
Passive	0.4664	2.0420
Semi active	0.3973	1.6279
Decrease %	17.4%	25.4%

The simulation results of the time domain acceleration responses of the vertical driver's seat and the cab's pitch angle when vehicle moves on the ISO class C road surface at v=70 km/h and full load are shown in Fig.4.



(a) The acceleration of cabin pitch angle



(b) The vertical acceleration of driver's seat

Fig-4: The time domain responses of the cab vertical, pitch angular accelerations when vehicle moves on the ISO class C road surface at v=70 km/h and full load

Table-3: The a_{ws} and $a_{w\phi c}$ values when vehicle moves on the ISO class C road surface at v=70 km/h and full load

	a_{ws} (m/s^2)	$a_{w\phi c}$ (rad/s^2)
Passive	0.8443	3.6967
Semi active	0.7507	3.0723
Decrease %	12.5%	20.3%

Similarly, from the results of Fig.4, the values of the weighted r.m.s acceleration responses of the vertical driver's seat (a_{ws}) and the cab's pitch angle($a_{w\phi c}$) are determined and shown in Tab.3. The results in Tab. 3 showed that the a_{ws} and $a_{w\phi c}$ values with semi-active cab suspension system significantly decrease by 12.5 % and 20.3 % in comparison with passive cab suspension system when vehicle moves on the ISO class C road surface at v=70 km/h and full load.

5. CONCLUSIONS

In this study, a half-vehicle dynamic model of a semi-trailer truck is established for the effectiveness evaluation of passive and semi-active cab suspension systems for the improvement of vehicle ride comfort. The fuzzy logic controller (FLC) is designed to control the damping coefficient of cab suspension system. From the analysis results, the conclusions may be drawn as follow: (1) The a_{ws} and $a_{w\phi c}$ values with semi-active cab suspension system significantly decrease by 17.4 % and 25.4 % in comparison with passive cab suspension system when vehicle moves on the ISO class B road surface at v=70 km/h and full load and (2) The a_{ws} and $a_{w\phi c}$ values with semi-active cab suspension system significantly decrease by 12.5 % and 20.3 % in comparison with passive cab suspension system when vehicle moves on the ISO class C road surface at v=70 km/h and full load.

REFERENCES

[1] Le V. Q., Zhang J., et al (2011), "Nonlinear dynamic analysis of interaction between vehicle and road surfaces for 5-axle heavy truck". J. of South. Univer., Vol. 27, p.405-409.

[2] Van Quynh, L., Zhang, J., et al (2013), "Influence of heavy truck dynamic parameters on ride comfort using 3D dynamic model". Dongnan Daxue Xuebao (Ziran Kexue Ban)/J. Southeast Univ. (Nat. Sci. Ed.) 43(4), 763–770.

[3] Le Xuan Long, et al (2020), "A Comparison of Ride Performance of Hydro-Pneumatic Suspension System with Those of Rubber and Leaf Suspension Systems". IOP Conf. Ser.: Mater. Sci. Eng. 914 012037.

[4] Le V. Q (2017). "Comparing the performance of suspension system of semi-trailer truck with two air suspension systems". Vibroengineering Procedia, Vol. 14, p. 220-226.

[5] NV Tuan, LV Quynh, VTP Thao, LQ Duy (2020). "Optimal design parameters of air suspension systems for semi-trailer truck. Part 1: modeling and algorithm" Vibroengineering PROCEDIA, Vol. 33, p. 72-77.

[6] NV Tuan, LV Quynh, VTP Thao, LQ Duy (2020). "Optimal design parameters of air suspension systems for semi-trailer truck. Part 2: results and discussion" Vibroengineering PROCEDIA, Vol. 33, 2020, p. 147-152.

[7] Long L. X., Hong T. T., et al(2018), "Performance analysis of the hydro-pneumatic suspension system of heavy truck". International Journal of Mechanical Engineering and Technology, Vol. 9, p. 1128-1139.

[8] Liem N., Zhang I., Le V., Jiao R., Liao L(2017). "Performance analysis of air suspension system of heavy truck with semi-active fuzzy control". Journal of Southeast University, Vol. 33, Issue 2, p. 159-165.

[9] Van Liem Nguyen, Van Quynh Le (2019), "Ride comfort performance of heavy truck with three control cases of semi-active isolation systems". Vibroengineering PROCEDIA, Vol. 22, pp. 93-98.

[10] Soliman, A. M. A. (2008). Improvement of the Truck Ride Comfort Via Cab Suspension. SAE Technical Paper Series.

[11] Basaran, S., & Basaran, M. (2020). "Vibration Control of Truck Cabins With the Adaptive Vectorial Backstepping Design of Electromagnetic Active Suspension System", IEEE Access, 8, 173056–173067.

[12] Nguyen Van Liem, Zhang Jianrun, et al (2016). "Study of fuzzy control for cab's isolation system of heavy

truck", Vibroengineering Procedia, Vol. 10, pp. 309-314.

[13] Van Quynh L (2017). "Influence of semi-trailer truck operating conditions on road surface friendliness", Vibroengineering Procedia, Vol. 16, 2017, p. 67-72.

[14] ISO 8068. Mechanical Vibration-Road Surface Profiles-Reporting of Measured Data. International Organization for Standardization, 1995.

[15] ISO 2631-1 (1997). Mechanical Vibration and Shock-Evaluation of Human Exposure to Whole-Body Vibration, Part I: General Requirements. The International Organization for Standardization, 1997.

[16] Le, V.: Vibration Study and Control for Cab of Vibratory Roller. Southeast University (2013)

[17] Trinh Ngoc Tu (2020), "Study on semi-active control of cab's suspension system of a semi-trailer truck", Master's thesis: Thai Nguyen University of Technology.