A STUDY ON SEMI-ACTIVE SUSPENSION SYSTEM IN APPLICATION OF RIDE COMFORT OPTIMIZATION OF A BUS

NGHIÊN CỨU HỆ THỐNG TREO BÁN TÍCH CỰC ỨNG DỤNG NÂNG CAO ĐỘ ÊM DỊU CHUYỂN ĐỘNG CỦA Ô TÔ KHÁCH

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Abstract - Nowadays, the study on advancing safety factor of automobile, especially in the bus is concerned by many scientists. One of the factors for optimizing safety coefficient that deserve to be mentioned is related to the research, design and perfect construction of suspension system, steering system and brake system to ensure smooth, high safety in motion. This paper presents the results of applied research on the theoretical basis of the linear quadratic regulator (LQR) to control the semi-active suspension system for bus to enhance the smooth movement on rough road. At the same time, the authors set up mathematical models and surveys in the time domain of semi-active suspension system in different working modes, through which the results of a bus ride comfort in using semi-active suspension system will be optimized in comparison with the passive suspension.

Key words - linear quadratic regulator, semi-active suspension system, ride comfort, automobile, steering system.

Tóm tắt - Ngày nay, việc nghiên cứu nâng cao hệ số an toàn trong ô tô đặc biệt là ô tô chở khách được các nhà khoa học quan tâm. Một trong những yếu tố để nâng cao hệ số an toàn phải kể đến việc nghiên cứu, thiết kế, chế tạo hoàn thiện các hệ thống treo, hệ thống lái, hệ thống phanh đảm bảo độ êm dịu, độ an toàn cao khi chuyển động. Bài báo này trình bày kết quả nghiên cứu ứng dụng cơ sở lý thuyết bộ điều chỉnh toàn phương tuyến tính để điều khiển hệ thống treo bán tích cực cho ô tô khách nhằm nâng cao độ êm dịu khi chuyển động trên đường mấp mô. Đồng thời nhóm tác giả thiết lập mô hình toán học và khảo sát trong miền thời gian của hệ thống treo bán tích cực ở các chế độ làm việc khác nhau, thông qua đó thấy được kết quả độ êm dịu chuyển động của ô tô khi sử dụng hệ thống treo bán tích cực sẽ tăng lên so với hệ thống treo bị động kinh điển.

Từ khóa - bộ điều khiển toàn phương tuyến tính, hệ thống treo bán tích cực, độ êm dịu, ô tô chở khách, hệ thống lái.

1. Introduction

Ride Comfort is the general sensation of noise, vibration and motion inside a driving vehicle, experienced by both the driver as well as the passengers. Ride comfort optimization goes beyond the pure ISO2631 Whole body vibration certification testing as it affects the comfort, safety and health of the passengers subjected to it. Semi-active systems can only change the viscous damping coefficient of the shock absorber, and do not add energy to the suspension system. Though limited in their intervention (for example, the control force can never have different direction than the current vector of velocity of the suspension), semi-active suspensions are less expensive to design and consume far less energy. In recent times, research in semi-active suspensions has continued to advance with respect to their capabilities, narrowing the gap between semi-active and fully active suspension systems.

The most important criterion of the ride comfort is weighted root – mean – square (RMS) acceleration of the body mass. Because the dependent suspension system is often used on bus, so the writer made survey on the vibration of bus with half car model on sine wave road with two different suspension systems: Semi-active suspension and passive suspension in time domain. From these, it can be seen that the ride comfort of semi-active suspension is much more than the classic passive suspension.

2. Survey the ride comfort of semi-active suspension of bus using LQR

2.1. Half car model for semi-active suspension system

The half car model is shown in Fig.1. Where:

- Z- Vertical displacement of the car body at the center of gravity [m];

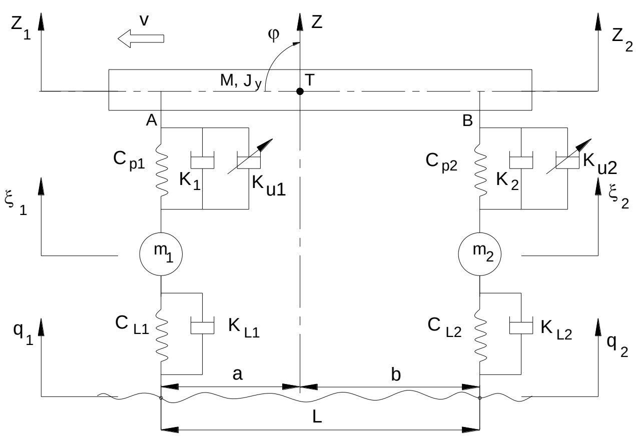

***Fig 1.** Half car model for semi – active suspension system*

- Z_1, Z_2- Vertical displacement of the car body at the front and rear location [m];
- ξ_1, ξ_2 - Vertical displacement of the car wheel at the front and rear wheel [m];
- q_1, q_2- An irregular excitation from the road surface at the front and rear car [m];
- M - Mass of the car body [Kg];
- m_1, m_2- Mass of the front and rear wheel [Kg];
- J_y- Moment of inertia for the car body [Kg.m^2];
- φ - Rotary angle of the car body at the centre of gravity [rad];
- C_{p1}, C_{p2}- Stiffness of the front and rear car body spring [N/m];
- C_{L_1}, C_{L_2}- Stiffness of the front and rear car tire [N/m];
- K_1, K_2- Damping of the front and rear car damper [Ns/m];
- K_{u_1}, K_{u_2} - Damping of the front and rear car damper controlled [Ns/m];
  $0 \le K_{u1}, K_{u2} \le K\max$

- K_{L_1}, K_{L_2} - Damping of the front and rear car tire [Ns/m];
- A, B - Point of junction between body and wheel mass at the front and rear car;
- T - Centre of the gravity of the body mass;
- a, b - Distance of the front and rear suspension location with reference to centre of the gravity of the body mass [m];
- L - Ground length of the bus [m];
- v - Speed of the bus [m/s];

2.2. Building differential equations of the motion

Using d'Alembert principle, tire damping is assumed to be zero; the set of equations of motion can be derived as follow:

$$\begin{cases} M.\left(\dfrac{a.\ddot{Z}_1 + b.\ddot{Z}_2}{L}\right) + K_1(\dot{Z}_1 - \dot{\xi}_1) + C_1(Z_1 - \xi_1) \\ + K_2(\dot{Z}_2 - \dot{\xi}_2) + C_2(Z_2 - \xi_2) - F_1 - F_2 = 0 \\ J_y.\left(\dfrac{\ddot{Z}_1 - \ddot{Z}_2}{L}\right) + [K_1(\dot{Z}_1 - \dot{\xi}_1) + C_1(Z_1 - \xi_1) - F_1]a \\ -[K_2(\dot{Z}_2 - \dot{\xi}_2) + C_2(Z_2 - \xi_2) - F_2]b = 0 \\ m_1\ddot{\xi}_1 + C_{L1}(\xi_1 - q_1) - K_1(\dot{Z}_1 - \dot{\xi}_1) - C_1(Z_1 - \xi_1) \\ + F_1 = 0 \\ m_2\ddot{\xi}_2 + C_{L2}(\xi_2 - q_2) - K_2(\dot{Z}_2 - \dot{\xi}_2) - C_2(Z_2 - \xi_2) \\ + F_2 = 0 \end{cases} \quad (1)$$

Where:

F_1, F_2: Control force [N];

$$F_1 = -K_{u1}(\dot{Z}_1 - \dot{\xi}_1),\ F_2 = -K_{u2}(\dot{Z}_2 - \dot{\xi}_2)$$

The state space representation of the motion equations is written in the following form:

$$\begin{aligned} \dot{x} &= Ax + Bu + Gw \\ y &= Cx + Du + Hw \end{aligned} \quad (2)$$

Where:

x – State vector;

$$x = \begin{bmatrix} x_1 & x_2 & x_3 & x_4 & x_5 & x_6 & x_7 & x_8 \end{bmatrix}^T \quad (3)$$

The state variables are chosen to be: $x_1 = \dot{z}_1,\ x_2 = \dot{\xi}_1,\ x_3 = \dot{z}_2,\ x_4 = \dot{\xi}_2,\ x_5 = z_1,\ x_6 = \xi_1,\ x_7 = z_2,\ x_8 = \xi_2;$

u – Input vector,

$$u = \begin{bmatrix} F_1 & F_2 \end{bmatrix}^T \quad (4)$$

w – White noise vector,

$$w = \begin{bmatrix} q_1 & q_2 \end{bmatrix}^T \quad (5)$$

y – Output vector

A– State matrix;

B – Input matrix

C – Output matrix;

D – feed through matrix ($D = 0$);

G, H – White noise matrix.

$$A = \begin{bmatrix} a_{11} & a_{12} & a_{13} & a_{14} & a_{15} & a_{16} & a_{17} & a_{18} \\ a_{21} & a_{22} & 0 & 0 & a_{25} & a_{26} & 0 & 0 \\ a_{31} & a_{32} & a_{33} & a_{34} & a_{35} & a_{36} & a_{37} & a_{38} \\ 0 & 0 & a_{43} & a_{44} & 0 & 0 & a_{47} & a_{48} \\ 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \end{bmatrix};$$

$$G = \begin{bmatrix} 0 & 0 \\ \dfrac{C_{L1}}{m_1} & 0 \\ 0 & 0 \\ 0 & \dfrac{C_{L2}}{m_2} \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \end{bmatrix};\ B = \begin{bmatrix} b_{11} & b_{12} \\ b_{21} & 0 \\ b_{31} & b_{32} \\ 0 & b_{42} \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \end{bmatrix};\ H = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ -1 & 0 \\ 0 & -1 \end{bmatrix};$$

$$C = \begin{bmatrix} a_{11} & a_{12} & a_{13} & a_{14} & a_{15} & a_{16} & a_{17} & a_{18} \\ 1 & -1 & 0 & 0 & 0 & 0 & 0 & 0 \\ a_{31} & a_{32} & a_{33} & a_{34} & a_{35} & a_{36} & a_{37} & a_{38} \\ 0 & 0 & 1 & -1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & -1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix};$$

Where:

$$a_{11} = -\frac{a.b.K_1}{J_y} - \frac{K_1}{M};\ a_{12} = -a_{11};\ a_{13} = \frac{b^2.K_2}{J_y} - \frac{K_2}{M};$$

$$a_{14} = -a_{13};\ a_{15} = -\frac{a.b.C_1}{J_y} - \frac{C_1}{M};\ a_{16} = -a_{15};$$

$$a_{17} = \frac{b^2.C_2}{J_y} - \frac{C_2}{M};\ a_{18} = -a_{17};\ a_{21} = \frac{K_1}{m_1};$$

$$a_{22} = -\frac{K_1}{m_1};\ a_{25} = \frac{C_1}{m_1};\ a_{26} = -\frac{C_{L1} + C_1}{m_1}$$

$$a_{31} = \frac{a^2.K_1}{J_y} - \frac{K_1}{M};\ a_{32} = -a_{31};\ a_{33} = -\frac{a.b.K_2}{J_y} - \frac{K_2}{M};$$

$$a_{34} = -a_{33};\ a_{35} = \frac{a^2.C_1}{J_y} - \frac{C_1}{M};\ a_{36} = -a_{35};$$

$$a_{37} = -\frac{a.b.C_2}{J_y} - \frac{C_2}{M};\ a_{38} = -a_{37};\ a_{43} = \frac{K_2}{m_2};$$

$$a_{44} = -a_{43};\ a_{47} = \frac{C_2}{m_2};\ a_{48} = -\frac{C_{L2} + C_2}{m_2};$$

$$b_{11} = \frac{1}{M} + \frac{a.b}{J_y};\ b_{12} = \frac{1}{M} - \frac{b^2}{J_y};\ b_{21} = -\frac{1}{m_1};$$

$$b_{31} = \frac{1}{M} + \frac{a^2}{J_y};\ b_{32} = \frac{1}{M} + \frac{a.b}{J_y};\ b_{42} = -\frac{1}{m_2}$$

2.3. Designing Linear Quadric Regulator

For the system in Fig. 1**,** irregular excitation from the road surface is considered to be white noise to control system. The semi-active suspension system is described by equations (6):

$$\begin{aligned} \dot{x} &= Ax + Bu \\ y &= Cx + Du \end{aligned} \quad (6)$$

The structure diagram of the system is shown in Fig. 2.

From equation (6), it is necessary to find $u = -k.x$ that minimizes the performance index:

$$J = \int_0^\infty (x^T Q x + u^T R u) dt \qquad (7)$$

Where: $Q \geq 0$ and $R \geq 0$

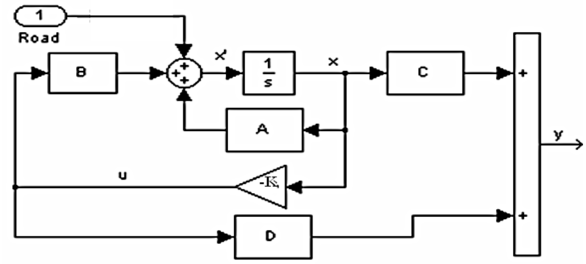

Fig 2. Structure diagram of LQR

Matrix Q and R are defined in the expression (8) and (9):

$$Q = \begin{bmatrix} q_1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & q_2 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & q_3 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & q_4 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & q_5 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & q_6 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & q_7 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & q_8 \end{bmatrix} \qquad (8)$$

$$R = \begin{bmatrix} r_1 & 0 \\ 0 & r_2 \end{bmatrix} \qquad (9)$$

2.4. Simulation result

2.4.1. Input parameter

- Characteristics of the bus

  G = 6670 kg, m_1 = 245 kg, m_2 = 343 kg, Cp_1 = 92100 N/m, Cp_2 = 123160 N/m, CL_1 = 902520 N/m, CL_2 = 1805040 N/m, K_1 = 5644 N.s/m, K_2 = 3420 N.s/m, L = 4,085 m.

- Irregular excitation from the road surface: choose the sine wave rough road, amplitude is q_0 = 0,05m, road surface wavelength is S = 5 m.

- Weighted matrix Q and R are chosen as follow:

$$Q = diag\ q_1, q_2 ... q_3\ ,$$

Where: $q_1 = q_2 = ... = q_8 = 1000$

And $R = diag[r_1, r_2]$,

Where: $r_1 = r_2 = 1e-3$

2.4.2. Testing result

To compare the ride comfort of the bus having passive suspension system and semi-active suspension system, the writers made survey on vibration accelerator and weighted RMS acceleration of the suspension in three regulations of movement speed of the bus: v = 40 Km/h, v = 60 Km/h, v = 80 Km/h.

Using control theory in state space in combination with the Matlab/Simulink software, the diagram of acceleration variables in three regulations is shown as follows:

- 1st regulation: v = 40 km/h, q_0 = 0,05m, S = 5 m.

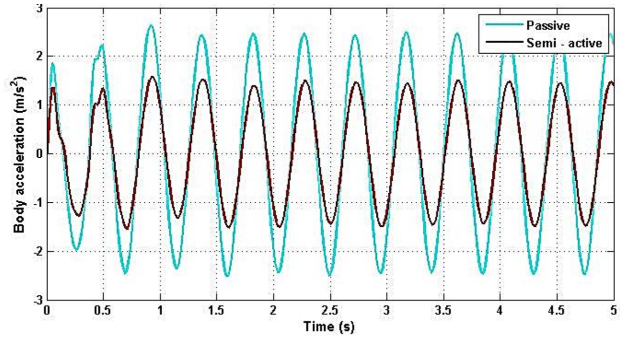

***Fig 3.** Body acceleration (Front) in 1st regulation*

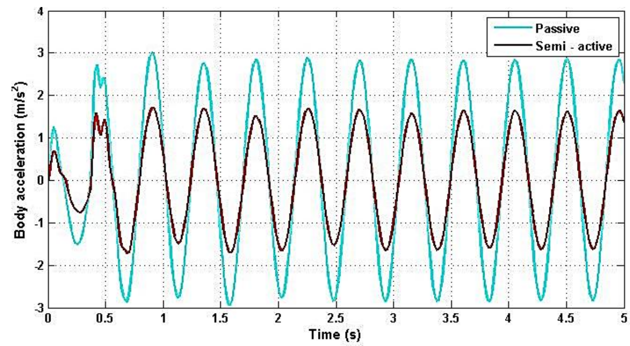

***Fig 4.** Body acceleration(Rear) in 1st regulation*

In the first regulation, the comparison weighted RMS acceleration between semi-active suspension system and passive suspension system is shown as follow:

Position	Weighted RMS acceleration (m/s^2)		Increase + /decrease -
	Passive	Semi-active	
Front wheel	1,72	1,032	- 40 %
Rear wheel	1,968	1,124	- 42,4 %

- 2nd regulation: v = 60 km/h, q_0 = 0,05m, S = 5 m.

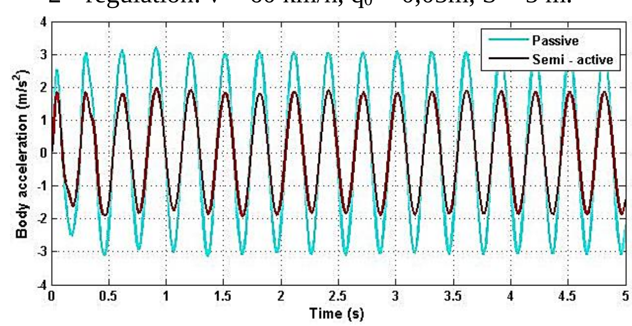

***Fig 5.** Body acceleration(Front) in 2nd regulation*

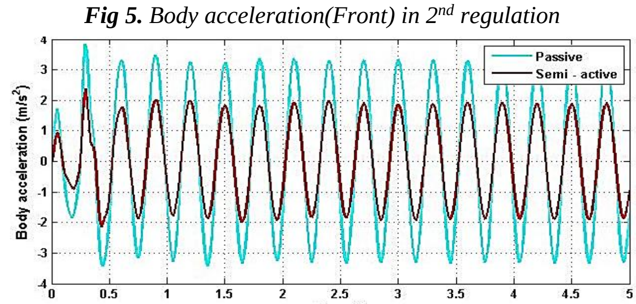

***Fig 6.** Body acceleration(Rear) in 2nd regulation*

In the second regulation, comparison weighted RMS acceleration between semi-active suspension system and passive suspension system is shown as follow:

Position	Weighted RMS acceleration (m/s^2)		Increase + /decrease -
	Passive	Semi-active	
Front wheel	2,163	1,318	- 39,1 %
Rear wheel	2,321	1,328	- 42,8 %

- 3rd regulation: v = 80 km/h, q_0 = 0,05m, S = 5 m.

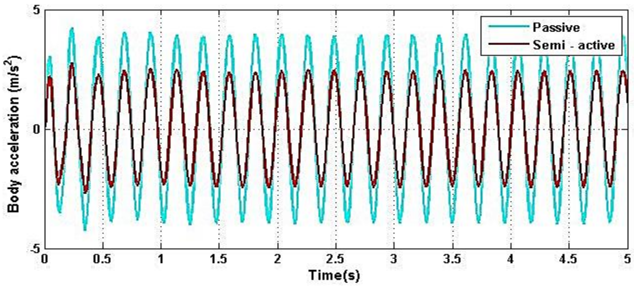

Fig 7. *Body acceleration(Front) in 3rd regulation*

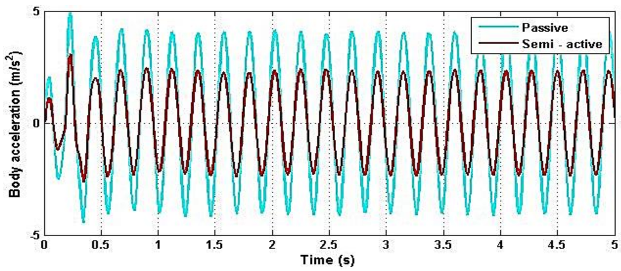

Fig 8*. Body acceleration(Rear) in 3rd regulation*

In the third regulation, the comparison weighted RMS acceleration between semi-active suspension system and passive suspension system is shown as follow:

Position	Weighted RMS acceleration (m/s^2)		Increase + /decrease -
	Passive	Semi-active	
Front wheel	2,782	1,726	- 36,7 %
Rear wheel	2,868	1,651	- 42,4 %

According to the ISO 2631 standard (Maximum RMS value is 2,5 m/s^2), the ride comfort of this semi- active suspension system is positive satisfaction.

3. Conclusion

From result of vibration survey of the bus having passive suspension system and semi-active suspension system of a half car model in time domain using Matlab/Simulink, it can be seen that weighted RMS acceleration of the body mass decreases significantly when using semi-active suspension system. This proves that when using the semi-active suspension the ride comfort of the bus increases significantly. RMS criterion is built on the basis of the statistics, so the evaluation ensures objectivity. Therefore, using semi-active suspension on bus is completely applicable and this helps to improve working life when using bus.

Below are a few recommendations that flow from this work:

- Continued design and create of controlled semi-active suspension of the bus.

- To carry out a test of the ride comfort when the bus is moving on roads.

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