INFLUENCE OF THE ENGINE MOUNTING SYSTEM ON THE AUTOMOTIVE RIDE COMFORT

Le Van Quynh*, Hoang Anh Tan, Nguyen Khac Minh
College of Technology - TNU

SUMMARY

Nowadays, automotive ride comfort is one of the most important performances of automobile, the research of automotive ride comfort is getting more and more important. The aim of this study is to evaluate the influence of the parameters of engine mounting system on automotive ride comfort. To achieve this goal, a 3-D vibration model for automobile with 10 DOF is established and Matlab/Simulink is used to simulate and calculate the impact factor. The the parameters of engine mount system such as stiffness and damping coefficients are analyzed respectively according to the international standard ISO 2631-1(1997-E) for the assessment of the impact of noise and vibration to human health. The results show that the damping coefficient of engine mount system has the greatest influence on automotive ride comfort when engine operates at road surface conditions in Viet Nam. This study can provide a theoretical basis for the semi-active mounting system for engine.

Keywords: *engine mounting system, vibration model, stiffness, damping, ride comfort*

INTRODUCTION

Engine mount is purposed to control an excessive motion generated from powertrain system and to isolate vibration and noise to be transmitted to main system. As vibration design of engine mount is one of the main items on the phase of vehicle development, the design should be optimized considering various design variables and uncertainties. In recent years, in research on engine mount system, there have been a lot of papers to mention aspects such as application of ANSYS, A.DAM software... etc for the design of engine mount system with the source of excitation by itself[3-5]. The torque roll axis for a mounting system with non-proportional damping (under oscillating torque excitation) is indeed decoupled when one of the damped modes lies in the torque roll axis direction and the study has proposed the design optimization for engine mounting system[6].

Study on the effects of the vibration vehicle on ride comfort movement of the vehicle using a linear vibration model with 8 d.o.f. is mentioned by references[7]. A 3-D linear vibration vehicle model with 10 DOF was presented to analyze the effects of vehicle parameters on ride comfort[8].

The major goal of this study is to improve a 3-D non-linear vibration models with 10 DOF. Matlab-simulink software is applied to simulate the automotive vibration model under the conditions of roads in Vietnam. The weighted r.m.s acceleration responses of the vertical automotive body, pitch and roll angles of automotive body are chosen as objective functions. The the parameters of engine mount system such as stiffness and damping coefficients are analyzed respectively according to the international standard ISO 2631-1(1997-E) for the assessment of the impact of noise and vibration to human health[9].

AUTOMOTIVE VIBRATION MODEL

Physical model

The arrangement of engine mount system is choosed four mounts in this study, so a vibration model engine with 6 DOF is shown in Fig.1.

Many studies indicate that the vertical engine body, pitch and roll angles of engine body have the most impact on automotive ride comfort, so a 3-D non-linear automotive vibration model with 10 is established to evaluate the influence of the engine mounting system on ride comfort, as shown in Fig.2.

* *Tel: 0943 141653, Email: lequynhdl@yahoo.com*

In Fig. 2, K_{ij} are the suspension stiffness coefficients; C_{ij} are the suspension damping coefficients; K_{Tij} are the stiffness coefficients of tires; C_{Tij} are the damping coefficients of tires; K_{ek} are the stiffness coefficients of the engine mount system; C_{ek} are the damping coefficients of the engine mount system; M and M_e are the sprung mass of the automobile and engine; m_{Aij} are the unspung mass of the axles; I_X, I_Y, I_{eX}, I_{eY} are sprung moment of inertia about X/Y-axis; L and $L_{e1,2}$ are wheelbase of automobile and engine; a, b are distance between the centre gravity of automobile body and the centre gravity of the front/rear tires; B_f and B_r are distance between the centre gravity of automobile body and the centre gravity of the left/right tires; B_{e1} and B_{e2} are distance between the centre gravity of engine and the left/right mount system of engine; ξ_{ij}, Z_{ij}, Z and Z_e are the vertical displacements; φ, θ and φ_1, θ_e are the angle deflection at the centre gravity of the automobile body and engine; x_1, x_2 are distance between the font and rear mounting system of engine and the centre gravity of automobile body; q_{ij} are road surface roughnesses; v is the speed of automobile ($i=1,2$ and $j=left, right$; $k=1 \div 4$).

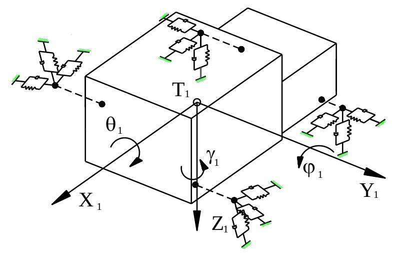

Fig 1. *Vibration model of engine mount system*

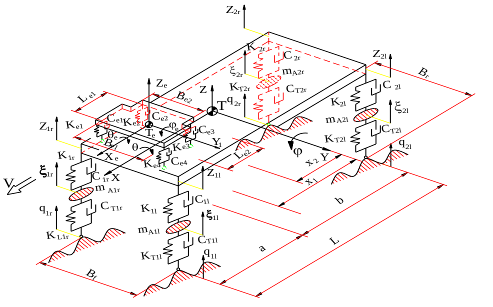

Fig 2. *Vibration model of four – wheel vehicle*

Mathematical model

The combined method known as the multi-body system theory and D'Alembert's principle are applied to set up differential equations to describe vehicle dynamics for the facilitate simulation where the object is separated into subsystems linked by the force and moment equations based on multi-body system theory and D'Alembert's principle is used to set up force and moment equations to describe automotive dynamic system. Diagram of forces and torques are shown in Fig.3.

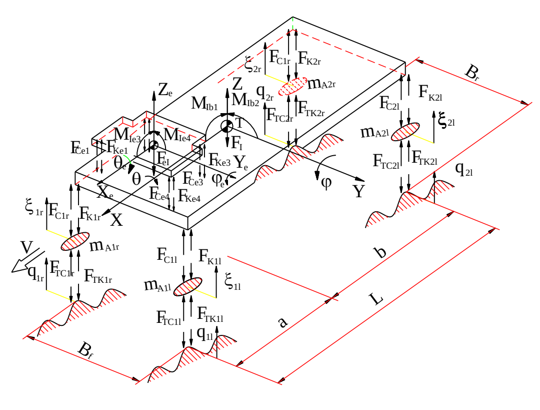

Fig 3. *Diagram of forces and torques*

$$
\left\{
\begin{array}{l}
m_{A1F}\,\ddot{\xi}_{1l} = [F_{K1f} + F_{C1f}] - [F_{TK1f} + F_{TC1f}] \\
m_{A1r}\,\ddot{\xi}_{1r} = [F_{K1r} + F_{K1r}] - [F_{TK1r} + F_{TC1r}] \\
m_{A2l}\,\ddot{\xi}_{2l} = [F_{K2l} + F_{C2l}] - [F_{TK2l} + F_{TC2l}] \\
m_{2r}\,\ddot{\xi}_{2r} = [F_{K2r} + F_{C2r}] - [F_{TK2r} + F_{TC2r}] \\
M\,\ddot{Z} = [F_{Ke1} + F_{Ce1}] + [F_{Ke2} + F_{Ce2}] + \\
+ [F_{Ke3} + F_{Ce3}] + [F_{Ke4} + F_{Ce4}] - [F_{K1l} + F_{C1l}] - [F_{K1r} + F_{C1r}] \\
- [F_{K2l} + F_{C2l}] - [F_{K2r} + F_{C2r}] \\
I_Y\,\ddot{\varphi} = a[F_{K1l} + F_{C1l}] + a[F_{K1r} + F_{C1r}] - b[F_{K2l} + F_{C2l}] - \\
- b[F_{K2r} + F_{C2r}] - x_1.[F_{Ke1} + F_{Ce1}] - x_2.[F_{Ke2} - F_{Ce2}] - \\
- x_1[F_{Ke3} + F_{Ce3}] - x_2[F_{Ke4} + F_{Ce4}] \\
I_X\,\ddot{\theta} = \dfrac{B_f}{2}[F_{K1l} + F_{C1l} - F_{K1r} - F_{C1r}] + \\
+ \dfrac{B_r}{2}[F_{K2l} + F_{C1l} - F_{K2l} - F_{C2r}] + \\
+ \dfrac{B_{e1}}{2}[F_{Ke1} + F_{Ce1} - F_{Ke3} - F_{Ce3}] + \\
+ \dfrac{B_{e2}}{2}[F_{Ke2} + F_{Ce2} - F_{Ke4} - F_{Ce4}] \\
M_e\,\ddot{Z}_e = -\left(\sum_{i=1}^{i=4} F_{Kei} + \sum_{i=1}^{i=4} F_{Cei}\right) \\
I_{eY}\,\ddot{\varphi}_1 = \dfrac{L_{e1}}{2}.[F_{Ke1} + F_{Ce1}] + \dfrac{L_{e2}}{2}.[F_{Ke3} + F_{Ce3}] + \\
- \dfrac{L_{e1}}{2}[F_{Ke2} + F_{Ce2}] - \dfrac{L_{e2}}{2}[F_{Ke4} + F_{Ce4}] \\
- I_{eX}\,\ddot{\theta}_1 = \dfrac{B_{e1}}{2}.[F_{Ke3} + F_{Ce3} - F_{Ke1} - F_{Ce1}] + \\
+ \dfrac{B_{e2}}{2}.[F_{Ke4} + F_{Ce4} - F_{Ke2} - F_{Ce2}]
\end{array}
\right. \qquad (1)
$$

In Fig. 3, F_{Kek} and F_{Cek} are the spring forces and the damping forces of the engine mount system; F_{TKij} and F_{TCij} are the spring forces and the damping forces of tires; F_{Kij} and F_{Cij} are the spring forces and the damping forces of the suspension systems; F_{eI}, F_{bI} and M_{eI1}, M_{eI2}, M_{bI1}, M_{bI2} are forces and moments of inertia about *X/Y*-axis of engine and automotive body.

The general dynamic differential equation for the typical four-wheel vehicle is given by Eq.1.

A difficulty in establishing the dynamic equations of automotive system is to find the nonlinear properties for suspension systems and tires which always appear two types of the nonlinear (nonlinear physics and nonlinear geometry), when vehicle moves on road surfaces. Both nonlinear geometry and nonlinear physics are considered in this study and these nonlinear factor could be described by the nonlinear mathematic function and the independent module-based programming. Then, there will be no difficulty in finding the solution for that.

For suspension system, the spring forces of the suspension systems could be determined by the following formula:

$$F_{Kij} = K_{ij}\left(Z_{ij} - \xi_{ij}\right) \tag{2}$$

The damping forces of the suspension systems could be determined by the following formula:

$$F_{Cij} = \begin{cases} C_{sc\,ij}.(\dot{\xi}_{ij} - \dot{Z}_{ij}) & when \ \ (\dot{\xi}_{ij} - \dot{Z}_{ij}) \le -0.3 \\ C_{lcij}.(\dot{\xi}_{ij} - \dot{Z}_{ij}) & when -0.3 < (\dot{\xi}_{ij} - \dot{Z}_{ij}) \le 0 \\ C_{srij}.(\dot{\xi}_{ij} - \dot{Z}_{ij}) & when \ 0 < (\dot{\xi}_{ij} - \dot{Z}_{ij}) \le 0.3 \\ C_{lrij}.(\dot{\xi}_{ij} - \dot{Z}_{ij}) & when \ \ (\dot{\xi}_{ij} - \dot{Z}_{ij}) > 0.3 \end{cases} \tag{3}$$

Meanwhile, the dynamic reaction forces of the suspension systems in vertical direction is defined as follows:

$$F_{ij} = F_{Kij} + F_{Cij} \tag{4}$$

For tire, a quarter of automotive model is selected for anlyzing the nonlinear properties of the tire, as shown in Fig.4(a). We are known that when vehicle moves on the roughness road, the wheel's motion in the vertical direction could be described in two stages: compression processes (static compression and dynamic compression) and rebounded processes (rebounded processes and wheel left-off leaving processes). As is shown in Fig.4(b) and Fig.4(c).

The radial spring force of the front right wheel could be determined by the following formula:

$$F_{TK1r} = \begin{cases} K_{T1r}.(q_{1r} - \xi_{1r}) \\ 0 \end{cases}$$
$$when[q_{1r} - (\xi_{r1} + \frac{(M_{1r} + m_{A1r}).g}{K_{T1r}})] \ge 0$$
$$when[q_{1r} - (\xi_{1r} + \frac{(M_{1r} + m_{A1r}).g}{K_{T1r}})] < 0 \tag{5}$$

The radial damping force of the front right wheel is determined by the following formula:

$$F_{TC1r} = \begin{cases} C_{T1r}\left(\dot{q}_{1r} - \dot{\xi}_{1r}\right) & when[q_{1r} - (\xi_{1r} + \frac{(M_{1r} + m_{A1r}).g}{K_{T1r}})] \ge 0 \\ 0 & when[q_{1r} - (\xi_{1r} + \frac{(M_{1r} + m_{A1r}).g}{K_{T1r}})] < 0 \end{cases} \tag{6}$$

Meanwhile, the dynamic reaction force of the front right wheel in vertical direction is defined as follows:

$$F_{T1r} = F_{TK1r} + F_{TC1r} \tag{7}$$

Eq.(4) and Eq.(7) are very important in creating subsystems for simulating which will be presented in the following paragraph.

Road surface roughness

Road surface roughness plays an important role in evaluating vehcle ride comfort. In this study, the random excitation of road surface roughness is selected the road surface highway1 in Hanoi-Lang Son section which is measured by equipment laser ARRB Profiler[8]. The measuring results are processed by Matlab 7.0 software and the processing results are shown in Fig.5.

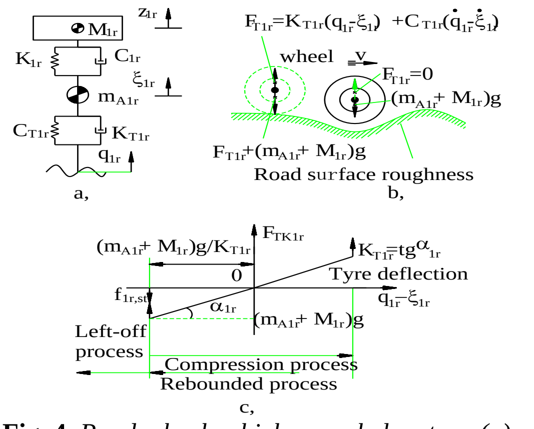

Fig. 4 *Road-wheel-vehicle compled system. (a) Quarter of automotive vibration model; (b)Wheel moving on road;*
(c) elastic properties of radial tire

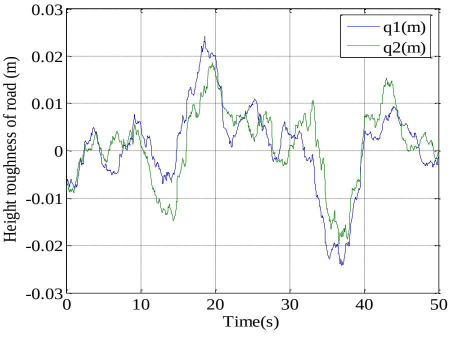

Fig 5. *Random function of road surface*

Table 1. *Comfort levels related to a_w threshold values*

a_{WZ} values	Comfort level	a_{WZ} values	Comfort level
Less than 0.315 $m.s^{-2}$	Comfortable	0.8 $m.s^{-2}$ to 1.6 $m.s^{-2}$	Uncomfortable
0.315$m.s^{-2}$ to 0.63$m.s^{-2}$	A little uncomfortable	1.25 $m.s^{-2}$ to 2.5 $m.s^{-2}$	Very uncomfortable
0.5$m.s^{-2}$ to 1 $m.s^{-2}$	Fairly uncomfortable	Greater than 2 $m.s^{-2}$	Extremely uncomfortable

INTERNATIONAL STANDARD ISO 2631

The most widely used international standard for whole-body vibration (WBV) is ISO 2631-1:1997E. This standard defines the methods to quantify WBV in relation to human comfort and health, perception and motion sickness. The standard has given two methods for evaluation human body comfort and health. In this study is selected a methods for evaluation human body comfort, vibration evaluation based on the basic evaluation method always includes measurements of the weighted root-mean-square (r.m.s) acceleration dened by:

$$a_w = \left[\frac{1}{T} \int_0^T a_w{}^2(t)dt \right]^{1/2} \qquad (8)$$

where, $a_w(t)$ is the weighted acceleration (translational and rotational) as a function of time, m/s^2; T is the duration of the measurement, s.

In this way, a_{wz}, $a_{w\varphi}$ and $a_{w\theta}$ values can be calculated from formula Eq.(8) and the r.m.s. value of the vertical acceleration in vehicle would be compared with the values in Tab 1, for indications of likely reactions to various magnitudes of overall vibration in the public transport.

SIMULATION AND ANALYSIS RESULTS

In order to solve the nonlinear differential equations which presented in section 2 for evaluating influence of the parameters of engine mounting system on automotive ride comfort, Matlab-Simulink software is used to simulate with a specific set of parameters of a 8-seat minibus "MEFA5-Lavi-304" manufactured in Vietnam[2], the diagram of simulation is shown in Fig. 6.

Simulations are carried out under the conditions of the different road surfaces, vehicle speeds and structural parameters of the vehicle to acquire the impact factors, For example, the simulation results of the vertical acceleration of automotive body when three vertical damping values of engine mounting system conditions of $0C_{ek}$, $0.5C_{ek}$, $1C_{ek}$ are applied and the vehicle moves on the road surface highway1 in Hanoi-Lang Son section condition at $v=80km/h$ (where C_{ek} is used to designate the vertical damping values of engine mounting system[2]) which is shown in Fig.7. From Fig.7 shows that the vertical acceleration of automotive body (a_z) values increase while C_{ek} values reduce which makes the automotive ride comfort bad.

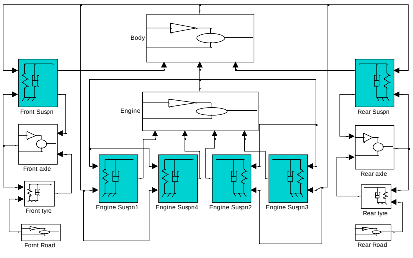

Fig 6. *Simulation diagram*

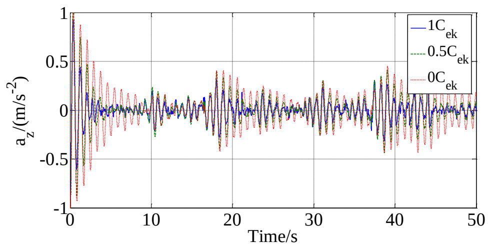

Fig7. *a_z when the damping coefficients of engine mount system change*

Effects of the stiffness coefficients of the engine mount system

The stiffness coefficients of the engine mount system are important factor that influence the automotive ride comfort. To analyze its effect on a_{wz}, $a_{w\varphi}$ and $a_{w\theta}$ values, the stiffness conditions of $K_e = [0.8\div2]xK_{ek}$ values and the damping conditions of $C_e=[0.5\ 1\ 1.5]xC_{ek}$ values have been studied when the vehicle moves on the road surface highway1 in Hanoi-Lang Son section condition at $v=80km/h$ and , where K_{ek} and C_{ek} is used to designate the stiffness and damping coefficients of engine mounting system shown in reference[2]. The effects of the stiffness coefficients of the engine mount system are shown in Fig.8. The a_{wz} increases as the engine mounting system stiffness increases which makes the automotive ride comfort bad. When the damping coefficient increases from $0.2C_{ek}$ to $1.6\ C_{ek}$, the a_{wz} value increases by 4.2 times. However, $a_{w\varphi}$ and $a_{w\theta}$ values increase as the K_{ek} value decreases which is the direct cause of the growth of pitch and roll vehicle vibration [10].

Effects of the damping coefficients of the engine mount system

The damping coefficients of the engine mount system are another important factor that influence the automotive ride comfort. To analyze its effect on a_{wz}, $a_{w\varphi}$ and $a_{w\theta}$ values, the stiffness conditions of $C_e = [0.8\div2.6]xC_{ek}$ values and the damping conditions of $K_e=[1\ 1.5]xK_{ek}$ values have been studied, where C_{ek} and K_{ek} is used to designate the stiffness and damping coefficients of engine mounting system shown in reference[2]. Effects of the damping coefficients of the engine mount system are shown in Fig.9. The a_{wz} value decreases as the engine mounting system stiffness increases which makes the automotive ride comfort good. When the damping coefficient increases from $0.8C_{ek}$ to $2.4C_{ek}$, the a_{wz} value decreases by 1.4 times.

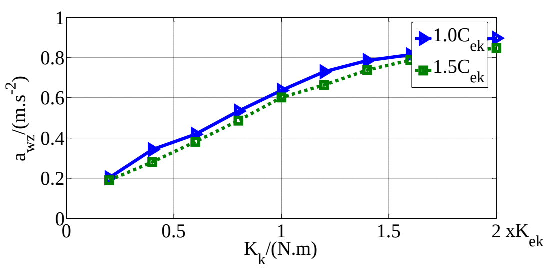

Fig 8. *Effects of stiffness*

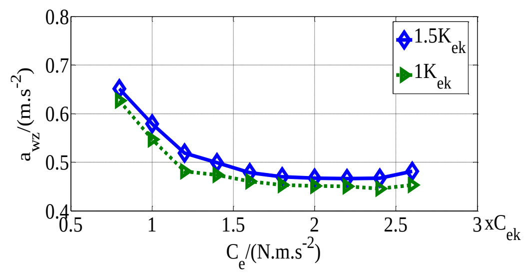

Fig 9. *Effects of damping*

CONCLUSION

The 3-D non-linear vibration model for 8-seat minibus "MEFA5-Lavi-304" manufactured in Vietnam is developed for simulating and analyzing the influence of the parameters of engine mounting system on automotive ride comfort. The major conclusions that can be drawn from the analysis results as follows:

(i) The damping coefficient of engine mount system has the greatest influence on automotive ride comfort when engine operates at road surface conditions in Viet Nam.

(ii) The combination of the stiffness and damping coefficients of engine mount system have implications for improving the automotive ride comfort.

(iii) In order to optimize the structural parameters of vehicle engine mount system, one can offer an engine mount system structure with intelligent control, and the study results in the paper can therefore serve as a basis for designing the control-intelligent engine mount system through controlling the range of stiffness, damping and road roughness.

REFERENCES

1. Du Quoc Thinh(2008), Production of mini-car assembly between the two governments of China and Vietnam and the contents of this paper is also part of the study in the test project "KC.05.DA.13".

2. Nguyen Tan Chinh(2010), Study on the effect of engine vibration on automobile's ride comfort, Master thesis, Tnu.edu.vn.

3.Chang Yong Song(2010). Design Optimization and Development of Vibration Analysis Program for Engine Mount System, http://mscsoftware.co.kr/.

4. Zhang Junhong, Han Jun (2006). CAE process to simulate and optimise engine noise and vibration, Mechanical Systems and Signal Processing,Vol.20, pp.1400–1409.

5. Jun Lan (2004). Multi-Body Nonlinear Analysis for Engine Vibration Simulation, WCCM VI in conjunction with APCOM'04, Sept. 5-10, 2004, Beijing, China.

6. Jae-Yeol Park, Rajendra Singh (2008). Effect of non-proportional damping on the torque roll axis decoupling of an engine mounting system. Journal of Sound and Vibration, Vol.313. pp.841–857.

7. Le Van Quynh, Zhang Jianrun, L.D Dat, N.K Binh, L.V Tuan. Research on the Optimal Parameters of Suspension Systems for Improved Ride Comfort Movement of the Vehicle by Vibration Model with 8 D.O.F(2010). Thai Nguyen, Viet Nam, the 10th National Conference on Solid Mechanics, 2010, 615-621.

8. Dao Manh Hung (2005). Effect of the maximum load of vehicle on road surface, Ministerial-level Reports, Hanoi, Vietnam.

9. ISO 2631-1; Mechanical vibration and shock-Evanluation of human exposure to whole-body vibration, Part I: General requirements, The International Organization for Standardization.

10. Le Van Quynh, Zhang Jianrun, Liu Xiaobo, Wang Yuan, Nguyen Van Liem(2013). Influence of heavy truck dynamic parameters on ride comfort using 3D dynamic model, Journal of Southeast University(Natural Science Edition), vol.43 (4), pp. 763-770.

TÓM TẮT

NGHIÊN CỨU ẢNH HƯỞNG HỆ THỐNG ĐỆM ĐỘNG CƠ ĐẾN ĐỘ ÊM DỊU CHUYỂN ĐỘNG CỦA Ô TÔ

Lê Văn Quỳnh*, Hoàng Anh Tấn, Nguyễn Khắc Minh

Trường Đại học Kỹ thuật Công nghiệp - ĐH Thái Nguyên

Ngày nay độ êm dịu chuyển động của ô tô là một trọng chỉ tiêu quan trọng nhất của ô tô, vì vậy nghiên cứu độ êm dịu càng ngày càng trở nên quan trọng. Mục tiêu chính nghiên cứu này là đánh giá ảnh hưởng của thông số hệ thống đệm động cơ đến độ êm dịu của ô tô. Để đạt được mục đích đó, một mô hình dao động phi tuyến không gian của ô tô với 10 bậc tự do được thiết lập và phần mềm Matlab/Simulink được sử dụng để mô phỏng và tính toán các hệ số ảnh hưởng. Các thông số của hệ thống đệm động cơ như độ cứng và hệ số cản lần lượt được phân tích dựa vào tiêu chuẩn Quốc tế ISO 2631-1(1997-E) về đánh giá ảnh hưởng của dao động và tiếng ồn đến sức khỏe con người. Kết quả nghiên cứu chỉ ra rằng hệ số cản của đệm cách dao động động cơ có ảnh hưởng rất lớn đến độ êm dịu khi xe hoạt động trong điều kiện mặt đường quốc lộ Việt Nam. Ngoài ra, kết quả nghiên cứu cung cấp một cơ sở lý thuyết để thiết kế hệ thống đệm bán tích cực cho động cơ đốt trong.

Từ khóa: *Hệ thống đệm động cơ, mô hình dao động, độ cứng, hệ số cản, độ êm dịu*

Ngày nhận bài:20/6/2015; Ngày phản biện:06/7/2015; Ngày duyệt đăng: 30/7/2015

Phản biện khoa học: *PGS.TS Vũ Ngọc Pi - Trường Đại học Kỹ thuật Công nghiệp - ĐHTN*

* *Tel: 0943 141653, Email: lequynhdl@yahoo.com*