SIMULATION OF VIBRATION OF THE AUTOMOTIVE POWERTRAIN SUSPENSION

PHÂN TÍCH DAO ĐỘNG HỆ THỐNG TREO CỤM TRUYỀN LỰC TRÊN Ô TÔ

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ABSTRACT

Vibration in automotive powertrain is one of the main causes low frequencies vibration of vehicle. This paper studies the vibration analysis of the powertrain's four-point suspension system of light truck. The method of reseash is simulation by using MATLAB Simulink. The relationship between the system's physical parameters (spring stiffness and damping coefficient) and the vibration parameters is then analyzed to find the best combination of the physical parameters that yields the optimized vibration behavior. An experimental of measuring the vibration frequencies at the four-point of powertrain's suspension with the engine working with different rotation. The result of this reseach is to reduce vibration that includes changing the physical properties of parts of suspension and modifying the overall setup. A proposal to the engineers with another simple and cost-effective approach is to properly design a vibration-isolation system.

Keywords: *Powertrain, Vibration, Stiffness of Suspension.*

TÓM TẮT

Dao động của hệ thống truyền lực là một trong những nguyên nhân chính gây ra dao động ở tần số thấp của xe. Bài báo này trình bày kết quả phân tích dao động của hệ thống truyền lực treo bốn điểm của xe tải nhẹ. Phương pháp nghiên cứu được sử dụng công cụ mô phỏng trong phần mềm Matlab Simulink. Quan hệ giữa các tham số vật lý của hệ thống treo (độ cứng và hệ số giảm chấn) với các thông số của dao động đã được phân tích. Thực hiện một thí nghiệm đo tần số dao động tại bốn điểm treo của hệ thống truyền lực khi động cơ đang hoạt động với các chế độ số vòng quay khác nhau. Kết quả của nghiên cứu này là đã xác định được thông số vật lý của hệ thống treo và kết cấu tổng thể để có thể giảm được dao động của hệ thống truyền lực. Một đề xuất cho các kỹ sư với cách tiếp cận khác đơn giản và hiệu quả là thiết kế đúng hệ thống cách ly dao động.

Từ khóa: *Truyền lực, dao dộng, độ cứng của hệ thống treo.*

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1. INTRODUCTION

The powertrain of a vehicles (acronym is PWT) includes an engine, a clutch, a gearbox, a transmission shaft, a differential, and two half shafts connected to the two proactive wheels, which altogether result in a complex operation and basically there are two types of vibrations that appear. The twisting oscillations in the powertrain system occur in the frequency's range from a few Hz to a few kHz as the load varies due to engine torque, clutch clearance, gears, helical or hypoid gears. These vibrations damage the gears in the system. The bending vibrations occur at low frequencies depend on the resistance to bending of the powertrain system, which causes the system to move both horizontal and longitudinally along the direction of the vehicle's moving. These vibrations damage the bearings and shafts in the system [1,2].

One of the causes of PWT oscillation is failure of damping details of the hanging rack. The low frequency vibration sources in PWT are due to the inertial force imbalance of the mass of the clusters when moving the rotation, which will cause damage to the details of PWT. High-frequency vibrations of PWT are transmitted to the vehicle body causing vehicle noise to vibrate. To overcome this phenomenon, people have used racks with reasonable hardness to reduce vibration. Depending on the types of suspension and influence of external factors of the vehicle, it will lead to fluctuations in the frequency of PWT. At the vehicle starting, sudden acceleration and braking will cause dynamic loads in PWT and cause overload of parts [3].

One of solutions to reduce vibration is to design a reasonable installation of PWT's suspension system with damping and proper hardness to reduce random impulses. Usually design damping mounds and stiffness so that when PWT oscillates at low frequencies below 20Hz and the amplitude varies from 0.3 ~ 15mm, the requirement of the sputum should be large and variable strong form; When PWT fluctuates at high frequencies above 20Hz (20 ~ 200Hz), the amplitude of 0.05 ~ 0.15mm of the mounds should be small (softer) and less deformed. To verify this problem, a experiments for intuitive results, can be used for many types of PWT of many vehicles, directly changing the experimental conditions to draw direct conclusions for the results from which there are faster decision on design [4].

On the dynamic characteristic analysis of the magneto rheological hydraulic suspension based on the experimental study of dynamic characteristic test for samples, and analyze the characteristics of suspension dynamic simulation results for conclusions. At the range of 0 - 35Hz, the magneto rheological suspension dynamic stiffness and hysteresis angle have obvious nonlinear characteristics. The dynamic stiffness reaches the peak at about 15Hz; the damping angel reaches the peak at about 12Hz lag angle. With the applied current becomes larger, the peak frequency slightly decreases [5].

A model six degree of freedom designed mixed new type of magneto rheological mode (MR) mount to mitigate vibration. It has been the implementation of the semi-active optimal controller and fuzzy controller that the displacement and velocity at a certain MR mount can be both suppressed [6].

2. MODEL OF POWERTRAIN SUSPENSION

Model of PWT is characterized as rigid body mode with seven degree-of freedom, three rotational modes on the body and four translational modes could exist in suspensions. The simplified PWT suspension model is shown in Figure 1. The symbols are as follows: α, β, λ is rotary movement around the axis of PWT at the mass center. The translational modes of suspensions at the four mount 1, 2, 3, 4, where each mount can be represented by three mutually perpendicular sets of spring stiffness (k) and viscous damping (b) coefficient of the three principal directions, which defined are as x, y and z.

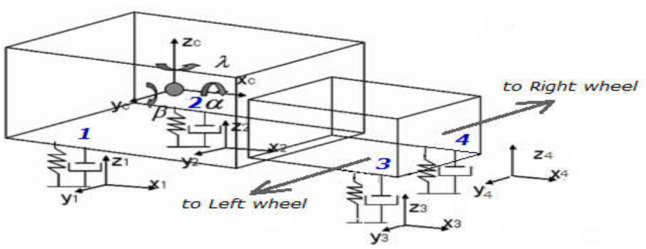


Figure 1. Model of powertrain suspension

In the process vehicle moving as well as the stops on the road, always the stimulating factors arising from the engine, from the vehicle itself and the bumps of the road surface. So on, the powertrain oscillation process is always a process of forced vibration. By applying the Lagrangian equations of the second kind, equation of motion of this 7-DOF system shown in equation (1).

$$M\ddot{X} + C\dot{X} + KX = F \quad (1)$$

M,C,K,F are mass matrix, damping matrix, stiffness matrix and loading matrix.

The loading matrix will be analyzed at each mount tripod of powertrain. Equation movement of each mount tripod shown in equation (2).

$$\begin{cases} M\ddot{x} = F_x \\ M\ddot{y} = F_y \\ M\ddot{z} = F_z \\ I_{xx}\ddot{\alpha} - I_{xy}\ddot{\beta} - I_{xz}\ddot{\gamma} = M_x \\ I_{xy}\ddot{\alpha} - I_{yy}\ddot{\beta} - I_{yz}\ddot{\gamma} = M_y \\ I_{xz}\ddot{\alpha} - I_{yz}\ddot{\beta} - I_{zz}\ddot{\gamma} = M_z \end{cases} \quad (2)$$

Writing at matrix form as equation (3)

$$\begin{bmatrix} M_x \\ M_y \\ M_z \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{y} \\ \ddot{z} \end{bmatrix} = \begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix}$$
$$\begin{bmatrix} I_{xx} & I_{xy} & I_{xz} \\ I_{xy} & I_{yy} & I_{yz} \\ I_{xz} & I_{yz} & I_{zz} \end{bmatrix}\begin{bmatrix} \ddot{\alpha} \\ \ddot{\beta} \\ \ddot{\gamma} \end{bmatrix} = \begin{bmatrix} M_x \\ M_y \\ M_z \end{bmatrix} \quad (3)$$

3. DETERMINING PARAMETERS STIFFNESS OF THE MOUNT

Stiffness is the ratio k between the variations of static load and the displacement, which could be calculated by the equation (4).

$$k = \frac{\Delta F}{\Delta S} \quad (4)$$

where ΔF and ΔS can be tested by using Electronic universal testing machine (EUT). The schematic diagram of testing is shown in Figure 2. Here (1) is machine (EUT) Instron 3365, (2) is mount of powertrain.

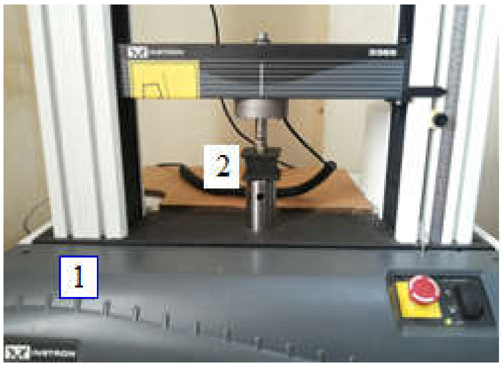


Figure 2. Stiffness testing of the mount

Take stiffness values of each mount into the equation (4), the ratio of stiffness of powertrain mounting system can be obtained. The results calculated are listed in Table 1.

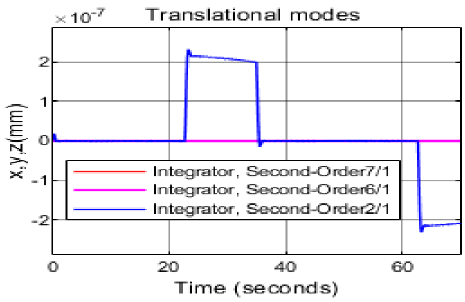
Table 1. Ratio of stiffness of mounts

No	Position of the mount	Values measurmented (kg/mm)	Ratio of stiffness k (N.s/mm)		
			k_x	k_y	k_z
1	Front left	100	100	120	120
2	Front right	100	100	120	120
3	Rear left	100	60	90	90
4	Rear right	100	60	90	90

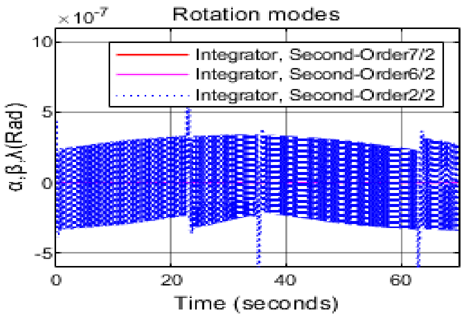
4. RESULTS ANALYSIS

Stiffness of suspension depends on specification of mount, that is rubber, bracket and control method. Frequencies vibration of suspension depends on three factors: Stiffness of suspension, preload and excitation frequency, the amplitude of dynamic load. Using Matlab Simulink to survey vibration of powertrain. Based measurmented on stiffness of suspension, the curve of the vibration of the suspension shown in Figure 3.

In Figure 3a are the displacements in the straight direction x, y, z in mm length. The consider in Second-Order, in the direction of z there will be oscillation, remaining in the horizontal direction x, y the oscillations are very small, not worthwhile, this does not lead to horizontal flip but only bumps up. In Figure 3b are the rotations modes α, β, λ is rotary movement around the axis x, y, z in radian unit. The consider in second order, in the axis of z there will be oscillation, along with straight oscillation will cause great vibration of powertrain. In the axis x, y the oscillations are very small, not significant.



a)



b)

Figure 3. Curve of the vibration of the suspension

5. EXPERIMENTAL

In the experimental, the modal parameters of the system can be identified through the curve fitting of the Frequency Response Function (FRF) expressed in frequency domain. Excitation force F(*) acting on a tripod mount determined by empirical equation (5).

$$F^{(*)}(\omega)=K(\omega)\left[X_s(\omega)-X_t(\omega)\right] \quad (5)$$

Where: ω is vibration of frequency, K is stiffness matrix of tripod mount, X_t is displacements matrix in the straight direction x, y, z at the top of mount suspension, X_S is displacements matrix in the straight direction x, y, z at the under of mount suspension, shown in Figure 4.

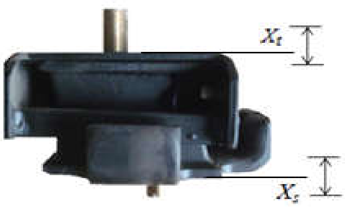


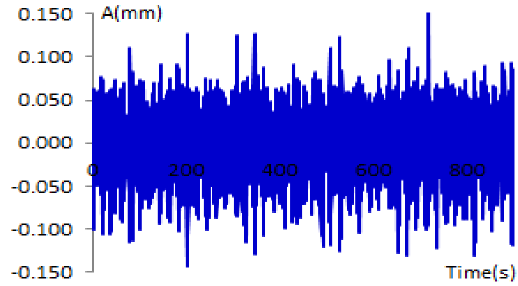
Figure 4. Displacements in top and under of mount suspension

a) b)

Figure 5. Accelerometer sensors and install in the tripod mount suspension

Using two 3 dimention accelerometer sensors shown in Figure 5a. The two sensors installed at the top (Xt) and under (Xs) of the tripod mount suspension shown in Figure 5b.

Figure 6 shows the result of measuring the total displacement amplitude over time at each tripod mount. We see that in the second tripod (front right) shown in Figure 6.b, there is less fluctuation than the other tripods shown in Figure 6.a,c,d, the amplitude of the general movement at this is also higher. This is due to PWT's inertia moment deviated from the center of each component.



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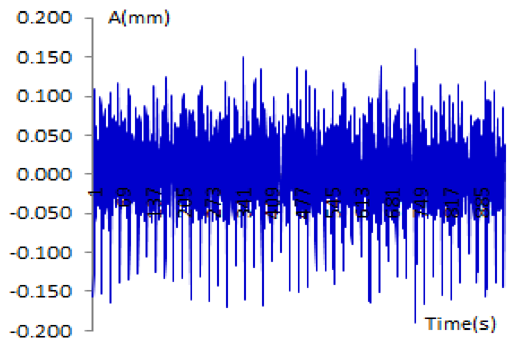

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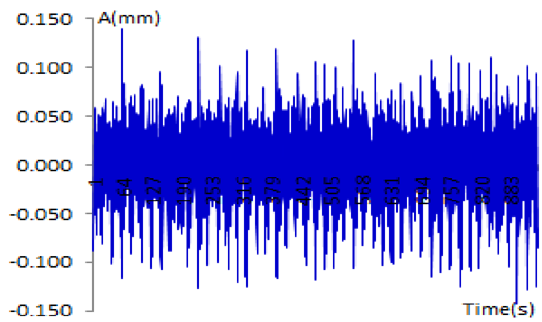

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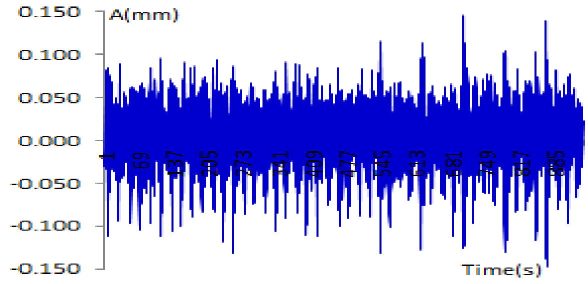
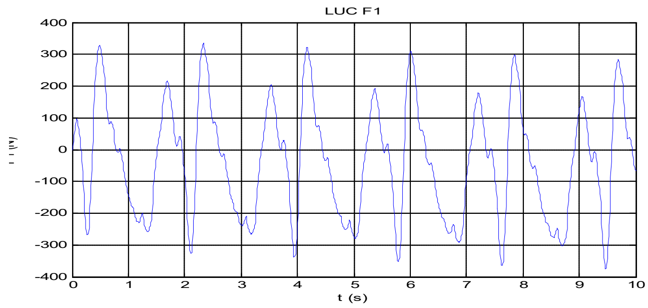


d)

Figure 6. Total displacement amplitude in time domain



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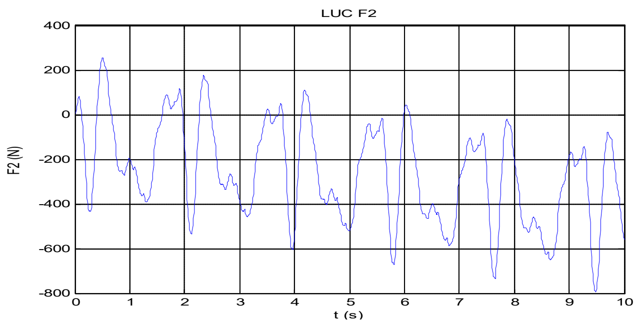

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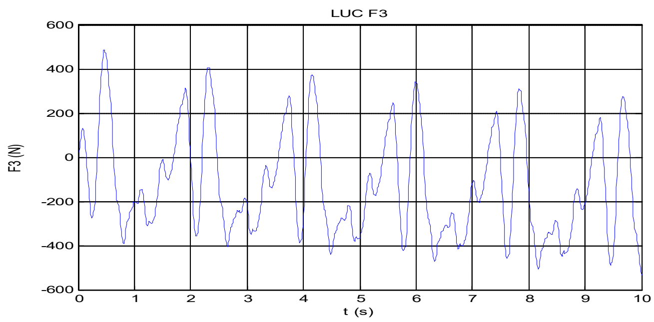

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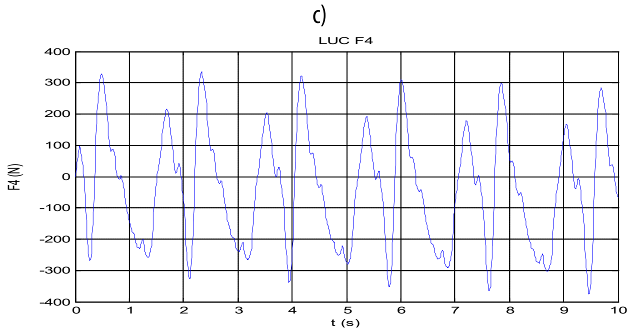

d)

Figure 7. Total displacement amplitude in frequency domain

Analysis of results in frequency domain determine frequency and amplitude of excitation force. The results get the graphs corresponding shown in Figure 7.

6. CONCLUSIONS

Vibration of the automotive powertrain suspension system is process complex. The stiffness of the mount trip of suspension has a nonlinear nature. In this paper, using Principle of Superposition we get general equation of loading on the each mount tripod of powertrain. Simulation in Matlab Simulink has results vibration of the suspension. Analysis the displacements modes in the straight direction x, y, z we see the direction of z there will be oscillation highest, remaining in the horizontal direction x, y the oscillations are very small. The rotations modes α, β, λ is rotary movement in the axis of z there will be cause great vibration to powertrain, in the axis x, y the oscillations rotations modes are very small.

By the experimental results, stiffness matrix of the mounting powertrain suspension system should be constructed based on total stiffness of each tripod mount suspension. The input parameters required to test the stiffness, such as preload should also be measured under the actual working conditions corresponding.

From the relationship between the system's physical parameters (spring stiffness and damping coefficient) and the vibration parameters analyzed willbe to hepl find the best combination of the physical parameters for optimination vibration behavior of vehicles.

REFERENCES

[1] Automotive Handbook, 2018. *SAE Society of Automotive Engineers*. Bosch, 10th Edition, ISBN: 978-1-119-53081-7.

[2] John Fenton, 1996. *Handbook of Vehicle Design Analysis, SAE Society of Automotive Engineers*. ISBN: 1-5609-903-5. SEA Order No. R-191

[3]. Gang Shen Chen, 2012. *Vehicle Noise, Vibration, and Sound Quality*. SAE Internation Warrendale, Pennsylvania, USA. SAE Order No. RF-400.

[4]. Pavel Brabec, 2017. *Methodology for the Experimental Determination of the Powertrain's Inertia Ellipsoid and its Verification*. MATEC Web of Conferences 89, 01005.

[5]. Zuo-yue Chen, 2016. *Experimental Study on Vibration System of Vehicle Engines Based on Magneto Rheological Mounts*. Advances in Engineering Research, volume 104, 7th International Conference on Mechatronics, Control and Materials (ICMCM 2016), Published by Atlantis Press.

[6]. Zhaoxue Deng, Ling Zheng, 2012. *Vibration Control and Design of Vehicle Engine Mount Activated by manetorheological Fluid.* Advanced Engineering Forum Vols. 2-3 (2012), pp 1007-1081, Trans Tech Publications, Switzerland. Doi:10.4028/www.scientific.nnet/AFF.2-3.1077

[7]. Y.Zhang,W.Shangguan, 2006. *A novel approach for lower frequency performance design of hydraulic engine mounts.* Computers & Structures, Vol. 84, No. 8, 2006, pp. 572-584.

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