Analysis of Vibration in Pickup Truck with Modified Rear Suspension under Heavy Load

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***Abstract*— This paper aims to study the vibration of a pickup truck when adding the leaf spring and shock absorbers in the rear suspension system under heavy load. The pickup trucks have been widely used for the logistics system because it is convenient to distribute goods and access to limited areas such as downtowns, and narrow roads. For cost reduction, it is necessary to transport heavy goods. Therefore, the car suspension must be modified, and the vibration responses should be considered. This paper used a haft-car dynamic model to simulate the vibration responses under various loads. The numerical results show that the amplitude of vibration for the car body reduces when adding the leaf spring and shock absorbers in the rear suspension system.**

***Index Terms*—pickup truck, vibration, heavy load, leaf spring, shock absorber, bump**

I. Introduction

The pickup trucks have been widely used to distributes goods such as agricultural products, consumer products, and so on. Because using a pickup truck is very convenient to access limited areas. However, the normal pickup truck is design for carrying the limited load to be about 1.5 tons. In practice, the weight of goods which transported is an overload for the normal pickup truck. The car suspensions need to be modified such as adding leaf spring and shock absorbers. Because of safety, the modification of suspension, as mentioned above without permission from the Department of transport becomes breaking the law, especially in Thailand. Therefore, studying the vibration of the car body before and after modification of suspension under heavy load is useful for safety. The previous research articles which presented about the vibration in vehicles can be summarized as follows.

The vehicle model with four-degrees-of-freedom was used to analyze the nonlinear dynamics and stability for the mono-wheel inclined vehicle-vibration platform coupled system [1]. The nonlinear ordinary differential equations were obtained using Lagrange's equation. In the same way, the articles as found in [2] also presented the four-degrees-of-freedom for studying the vibration of the vehicle. Because of considering comfort, the driver seat model was included in the mathematical model. Therefore, the five-degrees-of-freedom model was used to predict the vibration behavior for obtaining the value of design variables in the optimization problem [3]. The quarter car-seat-suspension model that includes the human model was proposed and considered as the eight-degrees-of-freedom model [4].

For numerical solving the equation of motions, there were some previous works can be described as follows. The numerical methods which called the incremental harmonic balance method (IHBM) and the Newmark method (NMM) were used to solve the differential equation for the nonlinear vehicle system [5]. The results obtained by the IHBM were verified very well with those obtained by NMM. The computation time for IHBM was less than that of NMM. The vibration response for the full car model with ten degree-of-freedoms was presented [6]. The software was known Simulink used for the vibration simulation and the equations of motion with 10-DOF were solved using the function as called ode-45.

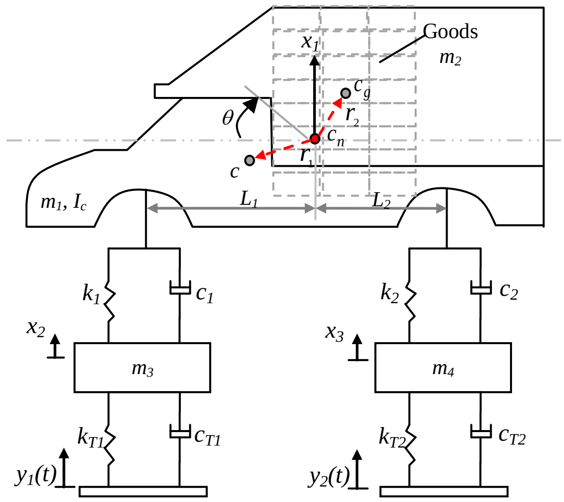

Figure 1. A haft-car dynamic model.

This paper focuses on the effect of changing the car center of gravity due to heavy loads on a pickup truck. Both the normal and modified rear suspensions also are investigated. The haft-car model is used to obtain the vibration response. The equations of motion with 4 DOF was solved using the function as known ode45 in Matlab software. Section II presents the model of a pickup truck, the equations of motion, and the base excitation respectively. The numerical procedure, results, and conclusions are presented in sections III, IV and V respectively.

Manuscript received December 2, 2019; revised June 15, 2020.

doi: 10.18178/ijmerr.9.7.973-978

II. Mathematical Model of Pickup Truck

A. Equation of Motions and Parameters Studied

This section presents the mathematical model of a pickup truck. In order to simplify a simulation for the first step, the half-car dynamic model as shown in Fig.1 was used to analyze the vibration. The front and rear suspension systems and tires are the spring and damper models. The symbols k_1, k_2, and c_1, c_2 denote the stiffness and damping parameters of a suspension system. In the same way, k_{T1}, k_{T2}, c_{T1}, and c_{T2} are the stiffness and damping parameters of wheels. While m_1 is the mass of car body, m_2 is the mass of goods, m_3 and m_4 denote the mass of wheels, and I_c is the mass moment of inertia in case of no goods. When the system is under base excitations (y_1, y_2), the system will vibrate. The vibration response which expressed as x_1, x_2 and x_3 are denoted as vertical displacements of a car body and wheels respectively while θ is a pitch angle of a car body.

As mentioned above, the vibration of a pickup truck is considered to be four degrees of freedom and the equation of motion for a matrix form is presented in (1).

$$[M]\ddot{\vec{x}}+[C]\dot{\vec{x}}+[K]\vec{x}=\vec{F}(t) \quad . \tag{1}$$

The matrices [M], [C], and [K] are expressed in (2), (3), and (4) respectively [2].

$$[M]=\begin{bmatrix} m_1+m_2 & 0 & 0 & 0 \\ 0 & I_c+\sum_{i=1}^{2} m_i r_i^2 & 0 & 0 \\ 0 & 0 & m_3 & 0 \\ 0 & 0 & 0 & m_4 \end{bmatrix} \tag{2}$$

$$[C]=\begin{bmatrix} c_1+c_2 & L_1c_1-L_2c_2 & -c_1 & -c_2 \\ L_1c_1-L_2c_2 & L_1^2c_1+L_2^2c_2 & -L_1c_1 & L_2c_2 \\ -c_1 & -L_1c_1 & c_1+c_{T1} & 0 \\ -c_2 & L_2c_2 & 0 & c_2+c_{T2} \end{bmatrix} \tag{3}$$

$$[K]=\begin{bmatrix} k_1+k_2 & L_1k_1-L_2k_2 & -k_1 & -k_2 \\ L_1k_1-L_2k_2 & L_1^2k_1+L_2^2k_2 & -L_1k_1 & L_2k_2 \\ -k_1 & -L_1k_1 & k_1+k_{T1} & 0 \\ -k_2 & L_2k_2 & 0 & k_2+k_{T2} \end{bmatrix} \tag{4}$$

The parameters studied in this paper explained as follows. While the goods are loading into the cabinet, the center of gravity position for a system changes from the point c to point c_n. The point c is the center of gravity for a car body. Because of knowing positions of c and c_g (center of gravity for goods), the position of c_n can be found out. The distances c_n to c and c_n to c_g are defined as r_1 and r_2 respectively. Therefore, the mass moment of inertia for a system with loading goods can be calculated based on the parallel-axis theorem. For the center of gravity for the pickup truck without goods, this article used the data that studied the center of gravity of pickup truck by using IPG CARMAKER program [7]. When the pickup truck carries the heavy load, the performance of normal leaf springs which produced from the factory is insufficient for supporting the load. Therefore, the spring stiffness of a rear suspension (k_2) needs to modify by adding leaf spring. Then, the value of spring stiffness increases. The spring stiffness for leaf spring depends on the number of leaves and the configuration of the cross-section area. The value of linear spring stiffness can be approximated by (5) [8].

$$k_2=\frac{8nEbh^3}{3L^3} \tag{5}$$

The symbols n, E, b, h, and L denote the number of leaves, Young modulus, the width of the leaf, the thickness of the leaf, and the span respectively. Moreover, the value of the damping coefficient (c_2) increases by adding the number of shock absorbers. For this study, the number of shock absorbers is four cylinders.

B. Base Excitation from Single Bump

During the pickup truck is moving on the road with a bump, the front wheel, and rear wheel are under the excitation. The excitations at the front wheel y_1 and rear wheel y_2, shown in Fig. 2, are expressed in (6) and (7).

$$y_1(t)=\begin{cases} R\sin(\omega t) & 0\le t\le t_1 \\ 0 & t>t_1 \end{cases} \tag{6}$$

$$y_2(t)=\begin{cases} R\sin(\omega[t-t_2]) & t_2\le t\le (t_1+t_2) \\ 0 & t>(t_1+t_2) \end{cases} \tag{7}$$

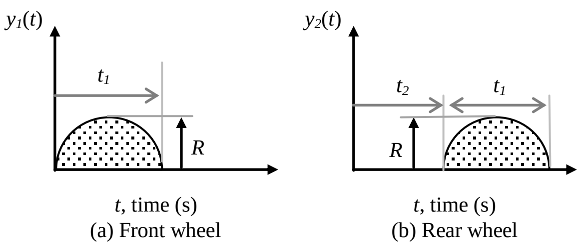

Figure 2. A model of base excitation.

While R is the amplitude of bump, t_1 is the time that front wheel is passing the bump, t_2 is time between the first excitation at the front wheel and second excitation at the rear wheel, and ω is excitation frequency respectively. The times t_1, t_2 and ω depended on the pickup truck velocity (V) can be estimated as:

$$t_1=\frac{2R}{V},\quad t_2=\frac{L_1+L_2}{V},\quad \omega=\frac{2\pi}{t_1} \tag{8}$$

From (6) to (8), the force vector $\vec{F}(t)$, subject to base excitation, can be presented as:

$$\vec{F}_i(t)=k_{Ti}y_i+c_{Ti}\frac{\partial y_i}{\partial t} \quad ,i=1,2 \tag{9}$$

III. Numerical Procedure

Equations (1) to (9) as mentioned in the previous section, the vibration responses are numerically studied using the Runge-Kutta algorithm provided by MATLAB (ode45). The sub-function file used in an ode45 code can be written defining the first derivative expressions [9]. The vector, used for the algorithm, shown in (10).

$$\dot{q}_{8\times1} = \begin{Bmatrix} \ddot{\vec{x}} \\ \dot{\vec{x}} \end{Bmatrix} = \begin{Bmatrix} [M]^{-1}\left[\vec{F}(t) - [C]\dot{\vec{x}} - [K]\vec{x}\right] \\ \dot{\vec{x}} \end{Bmatrix} \tag{10}$$

The vectors $\vec{x}$, $\dot{\vec{x}}$ and $\ddot{\vec{x}}$ are shown by (11).

$$\vec{x} = \begin{Bmatrix} x_1 \\ \theta \\ x_2 \\ x_3 \end{Bmatrix}, \dot{\vec{x}} = \begin{Bmatrix} \dot{x}_1 \\ \dot{\theta} \\ \dot{x}_2 \\ \dot{x}_3 \end{Bmatrix}, \ddot{\vec{x}} = \begin{Bmatrix} \ddot{x}_1 \\ \ddot{\theta} \\ \ddot{x}_2 \\ \ddot{x}_3 \end{Bmatrix} \tag{11}$$

The initial condition for this study is presented by (12).

$$q_{8\times1}(0) = \begin{Bmatrix} \dot{\vec{x}}(0) \\ \vec{x}(0) \end{Bmatrix} = (0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0)^T \tag{12}$$

The flowchart of the numerical procedure shows in Fig. 3 and the modification present in a red line.

IV. Results and Discussions

In this section, the numerical results present in 4 case studies. The results of vibration response under varying weights are presented. The goods in the cabinet are empty (Case-1), one-third of the cabinet (Case-2), two-thirds of the cabinet (Case-3), and the full load (Case-4) respectively. The vehicle specifications of this study show in Table I. The car velocity is 30 km/hr for simulation, the amplitude of bump is 0.1 m, and the goods, examined in this paper, are the pack of plastic water bottles. The weights of goods for case-1, case-2, case-3, and case-4 correspond to 0, 864 kg, 1,728 kg, and 2,592 kg respectively. Therefore, the centers of gravity for the cases study, measured from the rear of a pickup truck and the ground, are presented in Figs. (4) and (5).

The results in Fig. 4 and Fig. 5 show that the center of gravity position is changed from the front position to rear position and changed from below position to above position when the weights of goods increase. Therefore the distances L_1 and r_1 in the system as shown in Fig. 1 increases and the mass moment of inertia needs to be updated for simulation.

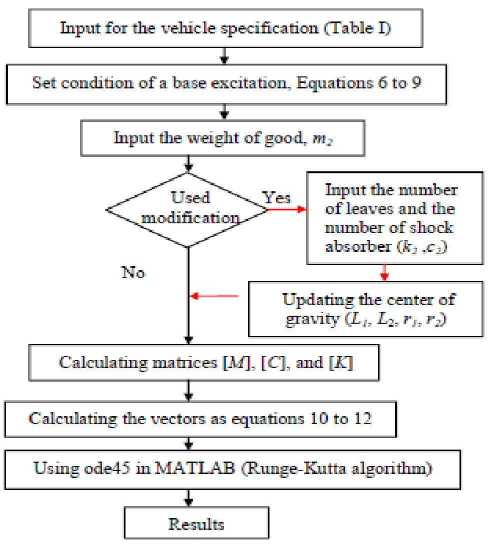

Figure 3. Flowchart for a numerical calculation

TABLE I. Vehicle Specifications

Dimension (Length x Width x Height)	5.26 x 1.76 x 2.5 m
Wheelbase, L_1+L_2	3,085 mm
Front and rear tread	1,510 mm
Mass of car body, m_1	1,460 kg
Pitch axis mass moment of inertia, I_c	1,660 kg.m^2
Front un-sprung mass, m_3	60 kg
Rear un-sprung mass, m_4	80 kg
Front suspension spring stiffness, k_1	135,000 N/m
Rear suspension spring stiffness, k_2	245,000 N/m
Stiffness of front wheel, k_{T1}	190,000 N/m
Stiffness of rear wheel, k_{T2}	190,000 N/m
Damping coefficient of front suspension, c_1	2,000 N.s/m
Damping coefficient of rear suspension, c_2	1,000 N.s/m
Damping coefficient of tire, c_{1T} , c_{2T}	500 N.s/m

Fig.6 shows the response of a normal rear suspension for case-1, case-2, case-3, and case-4. Figs.6(a) to (d) show the vertical displacement of a car body, the pitch angle of a car body, the vertical displacement of a front wheel, and the vertical displacement of a rear wheel respectively.

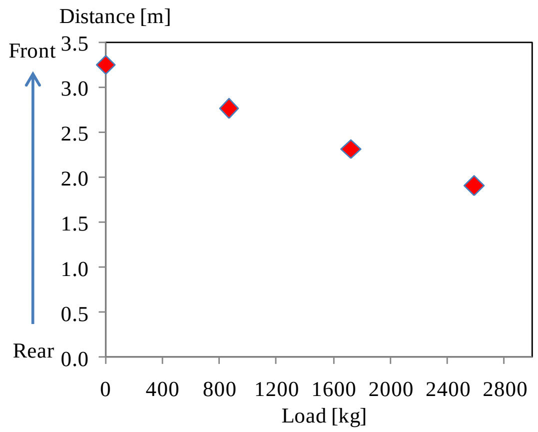

Figure 4. Longitudinal center of gravity changed.

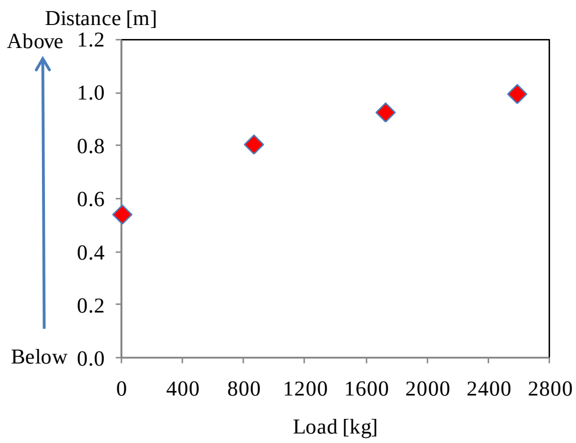

Figure 5. Vertical center of gravity changed

While the front wheel is rolling on the bump, the first transmitted force is generated and presses on the front wheel. Therefore, the peak displacement of a front wheel increases suddenly at the time period from 0.0 s to 0.05 s as shown in Fig.6(c). In the same way, while the rear wheel is rolling on the bump, the second transmitted force is generated and presses on the rear wheel. Therefore, the peak displacement of a rear wheel increases suddenly at the time period from 0.35 s to 0.40 s as shown in Fig. 6(d). As causes already mentioned above, the car body vibrates in the vertical direction together with rolling about the center of gravity as shown in Figs.6(a) and (b). The results show that while the car is undergoing to excitation due to passing a bump and the time period is about 0.0 s to 0.45 s, the car body with heavy load vibrates under amplitude that is lower than that of the car body with a light load. However, after the car passes a bump especially time is more than 2.0 s, it is very clear that the car body with heavy load vibrates under amplitude that is higher than that of the car body with a light load. Fig. 7 shows the amplitude of vibration in the vertical direction and in the pitch angle after excitation (time >2.0 s) under varying loads. The results show that when the weight of goods increases, an amplitude of vibration for the vertical of a car body in the vertical direction increases exponentially. Considering a pitch angle, the amplitude of vibration increases when the weight of goods increases. Because the moment of inertia for the pickup truck under heavy load is large, the vibrations of a car body decay difficultly. The time for a steady condition is longer than that of the lightweight.

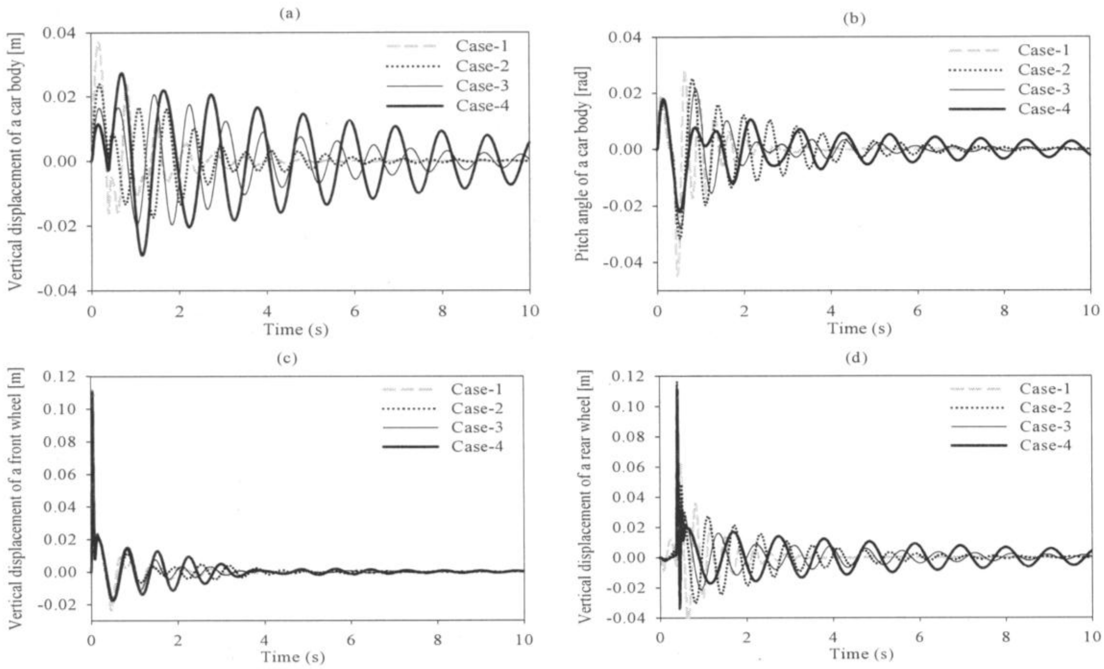

Figure 6. Vibration response of a pickup truck for the normal rear suspension: (a) x_1, (b) θ, (c) x_2 and (d) x_3

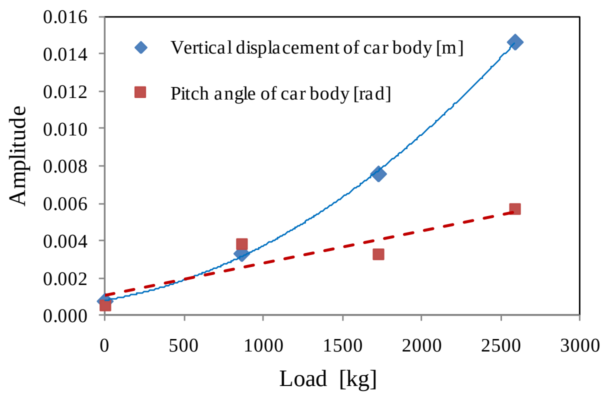

Figure 7. Amplitude of vibration for a car body after excitation under varying weights of goods (case-1 to case-4)

As from results mentioned above, the pickup truck with a heavy load is difficult to control when it undergoes to excitation such as the rough or hole roads. For that condition, the pickup truck with a heavy load is very risky to be accidents. Therefore, the rear suspension of a car needs to be modified.

Fig.8 shows the vibration response under a full load (case-4) for both the normal and modified rear suspensions. In case of modification, the number of leaves is ten (n=10), and the number of shock absorbers is four cylinders. The vertical displacement of a car body, the pitch angle of a car body, the vertical displacement of a front wheel, and the vertical displacement of rear wheel present in Figs.8(a) to (d) respectively. For the first excitation at the front wheel, the peak displacement of a front wheel increases suddenly at the time from 0.0 s to 0.05 s as shown in Fig.8(c). The peak displacement for the modified suspension is a little higher that of the normal suspension. After that, the second excitation acts at a rear wheel. The peak displacement of a rear wheel increases suddenly at the time from 0.35 s to 0.40 s as shown in Fig.8(d). As causes already mentioned above, the car body is vibrating in the vertical direction together with rolling about the center of gravity as shown in Figs.8(a) and (b). For during excitation, the car body with modified rear suspension vibrates with an amplitude that is little higher than that of the normal rear suspension. However, if comparing with the amplitudes of vibration at the same time for case-1 as seen in Figs.6(a) and (b), the amplitudes of vibration for modification is still less than that of the normal suspension. Considering after the excitation that time is more than 2.0 s. The advantage of a modified rear suspension is reducing the amplitudes of vibration fast comparing with the case of a normal rear suspension.

As well known, the systems vibrating depend on the natural frequencies. The resulting free vibration will be a superposition of the four vibration modes for 4-DOF. For this study, the vibration modes checked can be found by solving the eigenvalue problem. In case of a normal rear suspension under heavy load (case-4), the frequencies of mode are 5.97 rad/s, 9.66 rad/s, 56.48 rad/s, and 70.69 rad/s respectively. For a modified rear suspension under heavy load (case-4), the frequencies of mode are 8.38 rad/s, 9.72 rad/s, 56.47 rad/s, and 94.97 rad/s respectively. For the first, second and fourth modes, the frequencies of modifying are higher than that of a normal suspension.

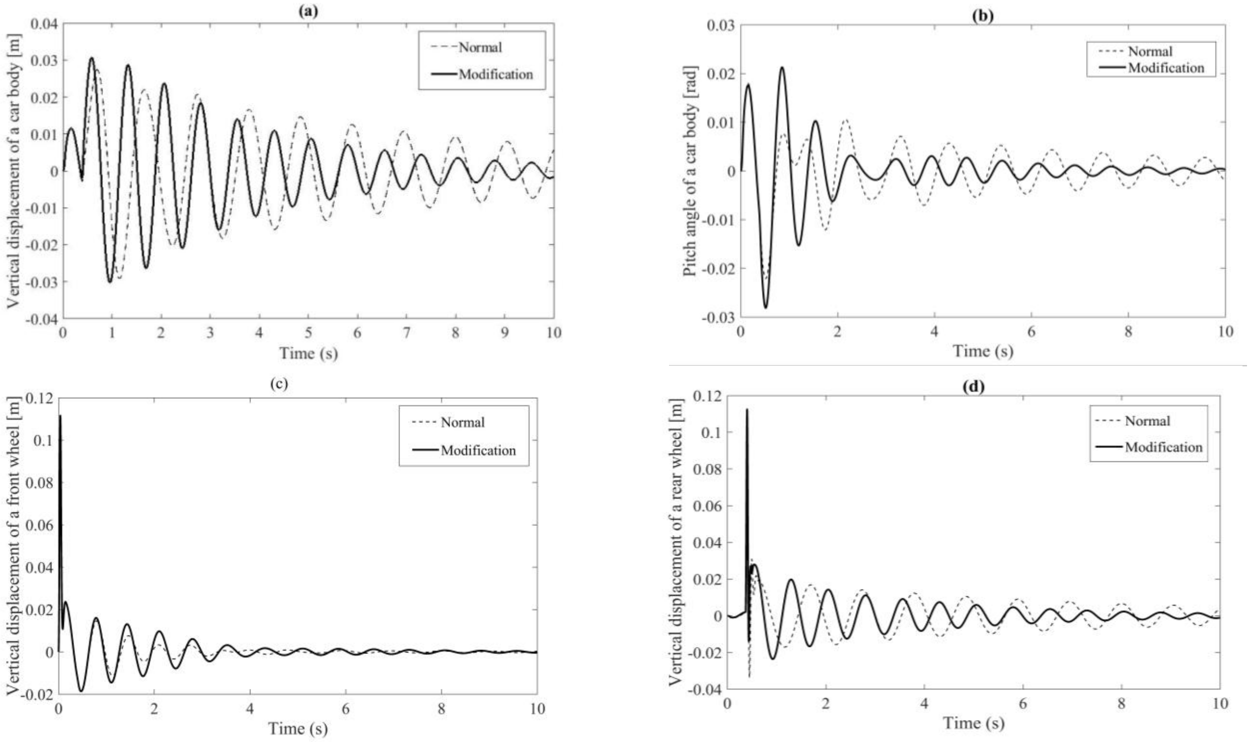

Figure 8. Vibration response of a pickup truck for the normal and modification: (a) x_1, (b) θ, (c) x_2 and (d) x_3

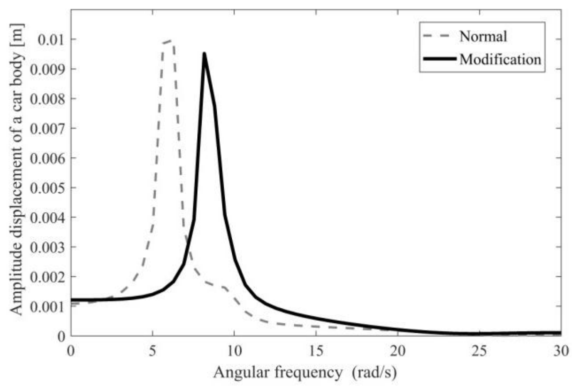

Figure 9. Spectrum analysis for a vertical displacement of a car body

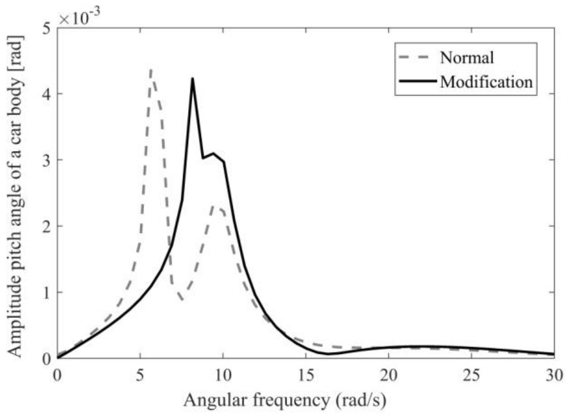

Figure 10. Spectrum analysis for a pitch angle of a car body

However, although the number of vibration modes is four sequences as mention above, the characteristic of vibration in this study is dominated by the first and second modes when using spectrum analysis as shown in Fig. 9 and Fig.10. Considering the spectrum analysis for a vertical displacement of a car body, as shown in Fig.9, the results show vibrating in the first mode. The second mode of vibration occurs clearly for a pitch angle of a car body especially in the case of a normal suspension as shown in Fig.10.

As from results mentioned above, when the pickup truck, being under heavy load, undergoes the rough or hole roads, the vibrations of car body decay fast for the modified rear suspension. The control of the car is better than that of without modified rear suspension. Therefore, the risk of accident due to those vibrations can reduce.

V. Conclusions

This paper explained the vibration response of a pickup truck under varying the weights of goods. The main conclusions can be summarized as follows:

The pickup truck, used to transport the heavy goods, should modify the rear suspension for reducing the amplitudes of vibration

This study recommends adding leaf springs and shock absorbers for improving the vibration of a pickup truck transporting heavy goods.

Conflict of Interest

The authors declare no conflict of interest.

Author Contributions

Puttha Jeenkour and Kittipong Boonlong conducted the research, analyzed the data and wrote the paper. All authors had approved the final version.

Acknowledgment

The authors would like to acknowledge Department of Mechanical Engineering, Faculty of Engineering, and Burapha University for their financial support.

References

[1] S. Zhou, G. Song, M. Sun, Z. Ren, and B. Wen, "Dynamic interaction of monowheel inclined vehicle-vibration platform coupled system with quadratic and cubic nonlinearities," *Journal of Sound and Vibration*, vol.412, pp.74-94, 2018.

[2] E. H. Choi, J. B. Ryoo, J. R. Cho, and O. K. Lim, "Optimum suspension unit design for enhancing the mobility of wheeled armored vehicles," *Journal of Mechanical Science and Technology*, vol. 24, pp. 323-330, 2010.

[3] K. Boonlong, "Multiobjective optimization of a vehicle vibration model using the improved compressed-objective genetic algorithm with convergence detection," *Advances in Mechanical Engineering*, 2013.

[4] M. P. Nagarkar, G. V. Patil, and R. Z. Patil, "Optimization of nonliner quarter car suspension-seat-driver model," *Journal of Advanced Research*, vol. 7, pp. 991-1007, 2016.

[5] S. Zhou, G. Song, M. Sun, and Z. Ren, "Nonlinear dynamic analysis of a quarter vehicle system with external periodic excitation," *International Journal of Non-Linear Mechanics*, vol. 84, pp. 82-93, 2016.

[6] S. K. Sharma, V. Pare, M. Chouksey, and B. R. Rawal, "Numerical studies using full car model for combined primary and cabin suspension," *Procedia Technology*, vol. 23, pp. 171-178, 2016.

[7] P. Paksupho, and E. Wirojsakunchai, "A simulation study of the position-changed effects on the center of gravity of natural gas pickup truck by IPG CARMAKER software package," in *Proc. 54th Kasetsart University Annual Conf.*, 2016, pp.303-310.

[8] R. Karwa, *Textbook of Machine Design*, 2 ed. New Delhi, India.: Laxmi Publications, 2006.

[9] A. Palazzolo, *Vibration Theory and Applications with Finite Elements and Active Vibration Control*, 1st. Chichester, U.K.: Wiley, 2016, ch.2, pp.50-54.

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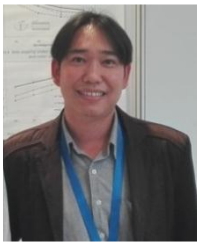

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