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ANALYZING THE EFFECT OF VEHICLE SPEED AND CLASS OF RANDOM ROAD PROFILE ON A 4-AXLE TRUCK VEHICLE VIBRATION

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ABSTRACT

The vehicle will generate vibration due to the impact of road surface, wind, steering, accelerating, braking...as it moves on road surfaces. These vibrations directly affect people and goods. A 14 DOF full vehicle dynamic model of the 4-axle truck vehicle with the torsion of the chassis and the “Walking Beam” tandem suspension is developed to analyze the effect of vehicle speed and a class of random road profile conditions on the vibration of the 4-axle truck vehicle which is developed based on Newton Euler equations and solved via Matlab/Simulink software. The study results have shown that the road surface and vehicle speed conditions have a great effect on vehicle vibrations. In addition, the results of this study could provide useful analysis results for proposing solutions to optimize the structure of the suspension system to improve the ride comfort of the vehicle and reduce the dynamic load of multi-axle truck vehicles.

Keywords: 4-axle truck vehicle, dynamic model, random road profile, Newton Euler equations, ride comfort, dynamic load coefficient.

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1. INTRODUCTION

The vehicle will generate vibration due to the impact of road surface, wind, steering, accelerating, braking... as it moves on road surfaces. These vibrations directly affect people and goods in the literature [1, 2]. When researching vehicle vibration, it is necessary to consider the vibration affected: (1) Humans in the vehicle: the vibrations which affect humans in the form of mechanical waves are transmitted directly to the human body. These waves cause the whole body or each part of the body to oscillate accordingly. For the health of the human, especially of the driver, the vibration of the vehicle can cause fatigue, and stress affects the behavior of the driver in the literature [3, 4, 5]. (2) Goods: Vibrations can crush, break, or change the nature of the goods. (3) Structure of vehicle: the vibrations cause bumps to the part or assembly of the vehicle which affects the durability of the part or assembly. In addition, vibrations also affect the capacity of the powertrain system as well as the stable motion of the vehicle. (3) Road: the vibrations of the vehicle cause loads (static and dynamic) that damage roads and bridges. In addition, vibrations change the traction force of the wheel. This change can affect vehicle movement in the literature [6, 7, 8]. Especially, the study results on the influence of truck vibrations on the ride comfort of the vehicle and friendliness of the road surface are presented in the literature [17], [18], [19], [20]. For heavy truck vehicles, in addition to affecting people and goods, it is necessary to evaluate the effect of vibrations on roads. A 14 DOF full vehicle dynamic model of the 4-axle truck vehicle with the torsion of the chassis and the “Walking Beam” tandem suspension is developed in this study is developed based on the Newton-Euler equations.

2. DYNAMIC MODEL OF THE 4-AXLE TRUCK VEHICLE

2.1 Equations of Motion

Assume that the vehicle moves at constant speed on a random road surface. A 14 DOF full vehicle dynamic model of the 4-axle truck vehicle without braking, accelerating, steering…and with the torsion coefficient of the frame of the body is developed, as shown in Figure-1 [2].

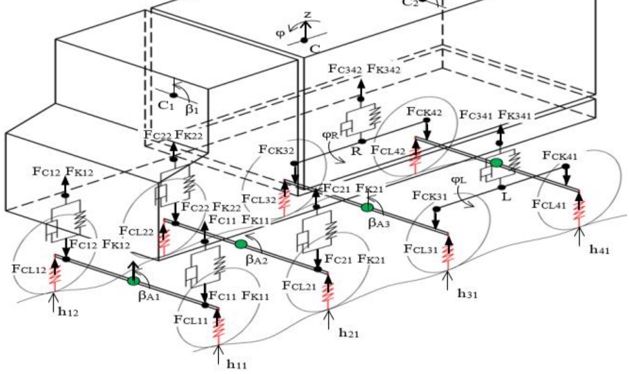

Figure-1. Schematic of a 4-axle truck vehicle full dynamic model.

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Interpretation of symbols on Figure-1: The motions of the sprung mass m are described by the vertical displacement of the center of gravity z, the pitch angle φ, the roll angle of the front mass β_1, and the roll angle of the rear mass β_2. The unsprung mass (axles), m_{Ai} is described by the displacement of the center of gravity z_{Ai} and the roll angle β_{Ai}. The axles are associated with the sprung mass by the suspension systems with the spring forces F_{Cij}, damper forces F_{Kij}, and stabilizing moment M_{Ti}. The wheel contact with the road surface has a road profile h_{ij}. With these assumptions, a 14 DOF full vehicle dynamic model of the 4-axle truck vehicle is developed based on Newton Euler's equation as follows:

$$\begin{cases}
m\ddot{z} = F_{C11}+F_{C12}+F_{K11}+F_{K12}+F_{C21}+F_{C22}+F_{K21}+F_{K22}+F_{C341}+F_{C342}+F_{K341}+F_{K342} \\
J_y\ddot{\varphi} = -l_1(F_{C11}+F_{K11}+F_{C12}+F_{K12})+-l_2(F_{C21}+F_{K21}+F_{C22}+F_{K22})+ \\
(l_3+\frac{a}{2})(F_{C341}+F_{K341}+F_{C342}+F_{K342}) \\
J_{x1}\ddot{\beta}_1 = w_1(F_{C11}+F_{K11}-F_{C12}-F_{K12})+w_2(F_{C21}+F_{K21}-F_{C22}-F_{K22}) \\
-M_{T1}-M_{T2}-M_{kx} \\
J_{x2}\ddot{\beta}_2 = w_{34}(F_{C341}+F_{K341}-F_{C342}-F_{K342})-M_{T3}-M_{T4}+M_{kx} \\
m_{A1}\ddot{z}_{A1} = (F_{CL11}+F_{CL12})-(F_{C11}+F_{C12}+F_{K11}+F_{K12}) \\
J_{Ax1}\ddot{\beta}_{A1} = (F_{CL11}-F_{CL12})b_1+(F_{C12}+F_{K12}-F_{C11}-F_{K11})w_1+M_{T1} \\
m_{A2}\ddot{z}_{A2} = (F_{CL21}+F_{CL22})-(F_{C21}+F_{C22}+F_{K21}+F_{K22}) \\
J_{Ax2}\ddot{\beta}_{A2} = (F_{CL21}-F_{CL22})b_2+(F_{C22}+F_{K22}-F_{C21}-F_{K21})w_2+M_{T2} \\
m_{A3}\ddot{z}_{A3} = (F_{CL31}+F_{CL32})-(F_{CK31}+F_{CK32}) \\
J_{Ax3}\ddot{\beta}_{A3} = (F_{CL31}-F_{CL32})b_3+(F_{CK32}-F_{CK31})w_3+M_{T3} \\
m_{A4}\ddot{z}_{A4} = (F_{CL41}+F_{CL42})-(F_{CK41}+F_{CK42}) \\
J_{Ax4}\ddot{\beta}_4 = (F_{CL41}-F_{CL42})b_4+(F_{CK42}-F_{CK41})w_4+M_{T4} \\
J_{AyL}\ddot{\varphi}_L = (F_{CK41}-F_{CK31})\frac{a}{2} \\
J_{AyR}\ddot{\varphi}_R = (F_{CK42}-F_{CK32})\frac{a}{2}
\end{cases} \quad (1)$$

In order to calculate the distribution of suspension force on the “Walking Beam” tandem suspension system, it is necessary to describe two rigid beams joining between the 3rd axle and the 4th axle. These beams are connected to the suspension spring by a pin joint in its center. The beam has mass and moment of inertia for left and right m_{BL}, m_{BL}, J_{AyL}, and J_{AyR}, respectively [9]. It moves with vertical displacement and pitch angle z_{AL}, z_{AR}, φ_L, φ_R, respectively. These pitch angles are calculated according to equations (13) and (14) in equation system (1).

The spring forces F_{Cij} and the damper force F_{Kij} are determined from the relative vertical displacement and relative velocity of the suspension according to the following:

$$F_{Cij} = \begin{cases} C_\infty\left(z_{ij}-z_{Aij}+f_{ij}^n\right) & when\ f_{dij}^n < z_{ij}-z_{Aij} \\ C_{ij}\left(z_{ij}-z_{Aij}\right) & when\ f_{ij}^t \le z_{ij}-z_{Aij} \le f_{ij}^n \\ -C_\infty\left(z_{ij}-z_{Aij}-f_{ij}^n\right) & when\ z_{ij}-z_{Aij} < f_{ij}^n \end{cases} \quad (2)$$

$$F_{Kij} = K_{ij}\left(\dot{z}_{Aij}-\dot{z}_{ij}\right)$$

where: f is the deflections of the suspension systems

The vertical tire force F_{CLij} is calculated as follows:

$$F_{CLij} = \begin{cases} C_{Lij}\left(h_{ij}-z_{Aij}\right) & when\ \ h_{ij}-\left[z_{Aij}-\dfrac{F_{ztij}}{C_{Lij}}\right] \ge 0 \\ -F_{ztij} & when\ \ h_{ij}-\left[z_{Aij}-\dfrac{F_{ztij}}{C_{Lij}}\right] < 0 \end{cases} \quad (3)$$

where: z_i is the vertical displacement of the upper of the suspension. These motions are calculated from the kinematic relationship with motions of sprung mass (z, φ, β_1, β_2). z_{Aij} is the vertical displacement of the lower of the suspension. These motions are calculated from the kinematic relationship with the motions of the axle (z_{Ai}, β_{Ai}). F_{ztij} is the static load on the wheel.

2.2 Random Road Profile According to ISO 8608:2016

ISO 8608: 2016 [10] proposes a methodology for the class of road profiles based on the Power Spectral Density (PSD). The PSD allows for representing a road surface as the sum of waves of different wavelengths and amplitudes, which are added one to the other by a mathematical transformation called the Fourier transform. The PSD for different classes of roads is shown in Table-1.

Table-1. Classification of random road profile.

Road class	$G_d(n_0)$ ($10^{-6}m^3$), n_0=0,1 cycles/m			$G_v(n)$ (10^{-6}m)
	Lower limit	Geometric mean	Upper limit	
A	-	16	32	6,3
B	32	64	128	25,3
C	128	256	512	101,1
D	512	1024	2048	404,3
E	2048	4096	8192	1617
F	8192	16384	32768	6468
G	32768	65536	131072	25873
H	131072	262144	-	103490

If the vehicle is assumed to move with speed over a given road segment with length L, a random profile of a single track can be approximated by a simple harmonic [11,12] according to:

$$h(x) = A_i cos(2\pi.n_i.x+\varphi_i) \ \ (i = 1 \div N) \quad (4)$$

Where

$$A_i = \sqrt{2.G_d(n_i).\Delta n} \quad (5)$$

The random road profile can be described as:

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$$h(x)=\sum_{i=1}^{N}\sqrt{2.G_d(n_i).\Delta n}.cos(2\pi.i.\Delta n.x+\varphi_i) \tag{6}$$

2.3 Assessment Criteria

The Root Mean Square of the vertical acceleration of the sprung mass z'' is used as an assessment criterion of ride comfort [13]. RMS (z'') is determined according to the following formula as

$$RMS(z'')=\sqrt{\frac{1}{T}\int_0^T z''^2(t)dt} \tag{7}$$

Where z'' is the vertical acceleration of sprung mass in the time domain (m/s^2); T is the impact time (s).

The dynamic load coefficient (DLC) was developed by Sweatman [14]. This coefficient is determined according to the following formula as

$$DLC=\frac{\mathrm{RMS}(F_{zdyn})}{F_{zt}} \tag{8}$$

where: RMS (F_{zdyn}) is the root mean square of the dynamic load on the wheel (N); F_{zt} is the static load on the wheel (N). The DLC value is in the range of 0.05 ÷ 0.3 under normal conditions [15].

3. RESULTS AND DISCUSSIONS

A 14 DOF full vehicle dynamic model of a 4-axle truck vehicle described in the above section is modeled using MATLAB/Simulink software with the structural parameters according to Table-2. The paper generates the random road profiles using the sinusoidal method for random road profiles from ISO class A to ISO class F [12, 16], as shown in Figure-2.

Table-2. The structure parameter of a 4-axle truck vehicle.

Parameters	Symbol	Values	Unit
Sprung mass	m	27290	kg
Mass of the first and second axles	m_{A1}, m_{A2}	570	kg
Mass of the third and fourth axles	m_{A3}, m_{A4}	785	kg
Roll moment of the inertia of front and rear mass	J_{x1}, J_{x2}	12240, 16979	kgm^2
Pitch moment of inertia of sprung mass	J_y	118850	kgm^2
Pitch moment of inertia of the rigid beam	J_{AyL}, J_{AyR}	600	kgm^2
Suspension stiffness of the suspension of the first and second axles	C_{1j}, C_{2j}	250000	N/m
Suspension stiffness of the third axle and fourth axle	C_{34j}	1400000	N/m
Damper coefficient of the suspension of the first axle and the second axle	K_{1j}, K_{2j}	15000	Ns /m
Damper coefficient of the third axle and fourth axle	K_{34j}	30000	Ns/m
Torsion coefficient of frame	C_{kx}	4000000	Nm/rad
Torsion coefficient of stabilizing bar in the first axle	C_{T1}	28648	Nm/rad
Torsion coefficient of stabilizing bar in the second axle	C_{T2}	171890	Nm/rad
Vertical stiffness of the tire of the first axle and the second axle	C_{L1j}, C_{L2j}	980000	N/m
Vertical stiffness of the tire of the third axle and fourth axle	C_{L3j}, C_{L4j}	1960000	N/m
Distance from CoG to the first axle	l_1	3.674	m
Distance from CoG to the second axle	l_2	1.874	m
Distance from CoG to the third axle	l_3	1.326	m
Distance from CoG to the fourth axle	l_4	2.676	m

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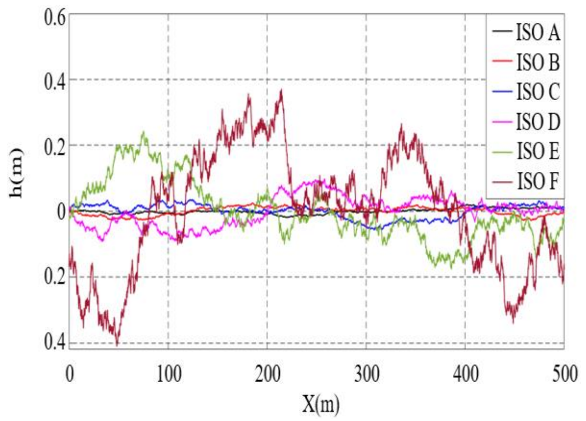


Figure-2. Random road profile (ISO class A to ISO class F).

Figure-3 and Figure-4 show the vertical displacement and acceleration of the sprung mass when the vehicle moves at a constant speed of 40 (km/h) with the random road profile from ISO class A to ISO class F. From the results of Figure-3 and Figure-4, we show that the amplitude values of the peak of the vertical acceleration responses of the sprung mass increase very quickly when the vehicle moves on a very bad road surface. The value of the root mean square of vertical acceleration of sprung mass, RMS (z''), and the value of DLC of four axles of the vehicle at vehicle speed range from 10 km/h to 60 km/h with the random road profile from ISO class A to ISO class F are shown in Figure-5 and Figure-6. From the results of Figure-5 and Figure-6, we show that the values of RMS (z'') of sprung mass and DLC at wheels of different vehicle axles increase rapidly when road surface conditions are turned bad and the vehicle speed condition increases.

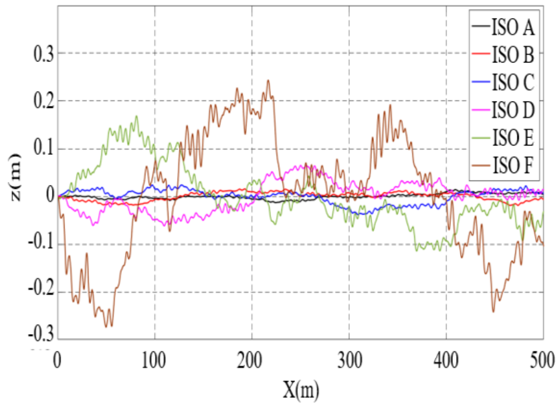


Figure-3. Vertical displacements of the sprung mass with the random road profile from ISO class A to ISO class F.

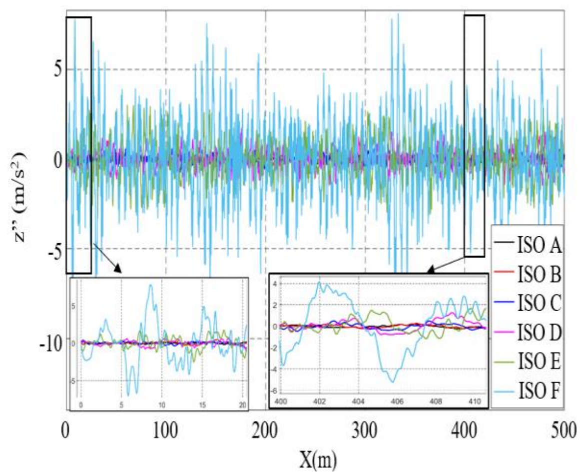


Figure-4. Vertical acceleration responses of the sprung mass with the random road profile from ISO class A to ISO class F.

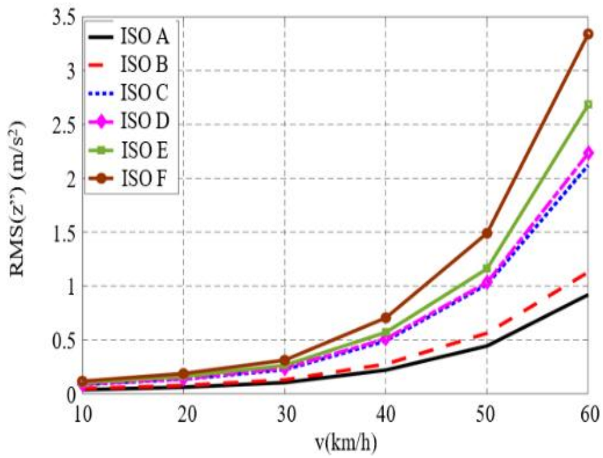
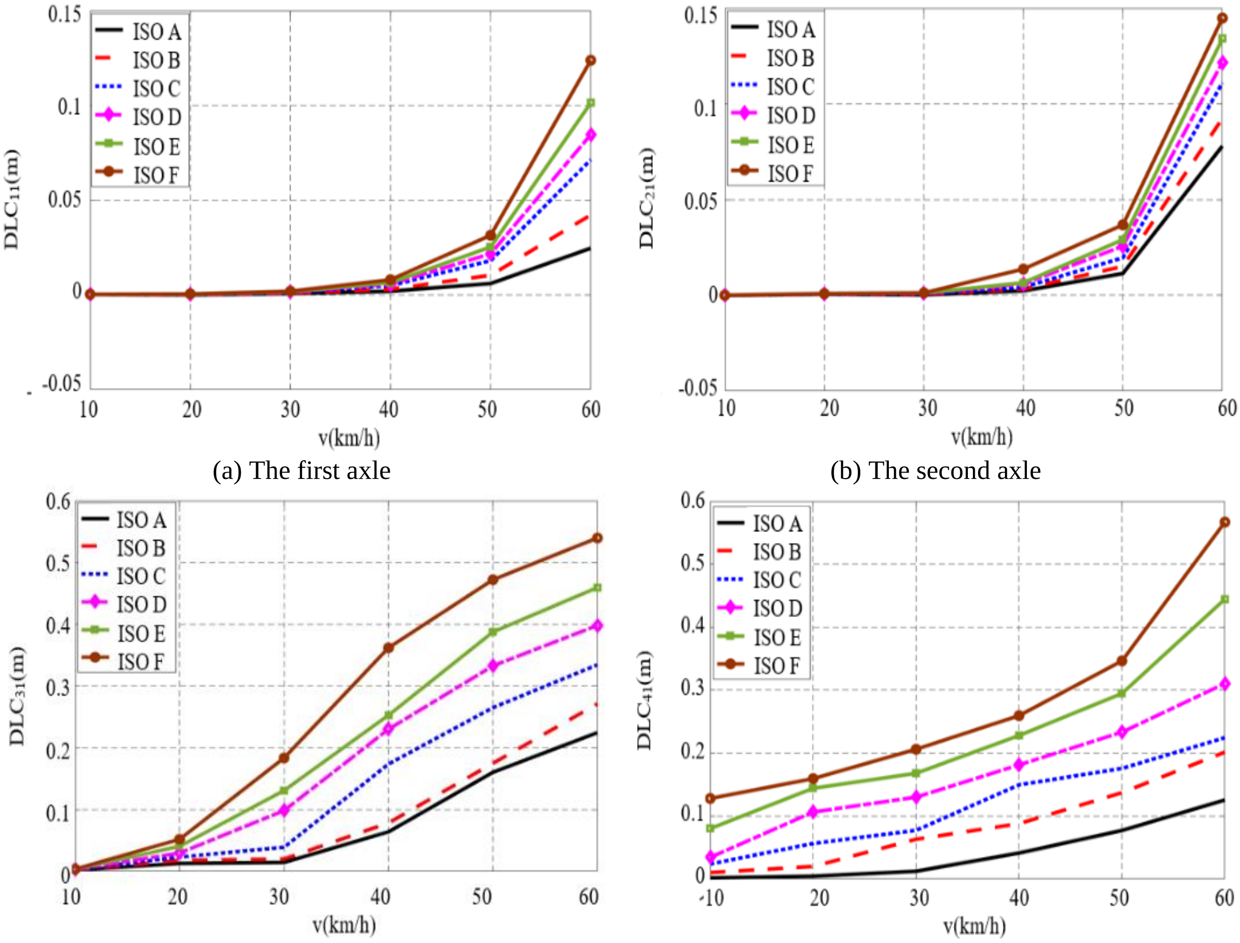


Figure-5. RMS (z'') value of sprung mass when the road surface and vehicle speed conditions change.

The results of Figure-5 and Figure-6 show the effect of different operating conditions such as the vehicle speed and class of random road profiles on the values of RMS (z'') and DLC. For single axles of vehicles (including the first axle and second axle), the values of the DLC change incrementally when the road surface level turns bad and the value of vehicle speed increases. For axles with the "Walking Beam" tandem suspension system, the DLC values in some cases are higher than the normal threshold of DLC [15], [18]. This could be explained by the structure of the "Walking Beam" tandem suspension system. The structural parameters of the rigid beam can greatly affect the load distribution between the third axle and the fourth axle of the vehicle. It is necessary to have structural solutions, such as air suspension or control solutions, to reduce the dynamic load on the road surface.



(a) The first axle

(b) The second axle

(c) The third axle

(d) The fourth axle

Figure-6. DLC values at wheels of different vehicle axles when the road surface and vehicle speed conditions change.

4. CONCLUSIONS

A 14-DOF full-vehicle model of the 4-axle truck vehicle was established to analyze the effect of the different operating conditions such as the road surface level and vehicle speed with the torsion of the chassis and the "Walking Beam" tandem suspension on vehicle vibrations. This dynamic model is developed based on Newton Euler equations and solved via Matlab/Simulink software. The analysis results of this study showed that the amplitude values of the peak of the vertical acceleration responses of the sprung mass increase very quickly when the vehicle moves the very bad road surface and the values of RMS (z'') of the sprung mass and DLC at wheels of different vehicle axles increase rapidly when road surface condition is turned bad and the vehicle speed condition increases. In addition, the results of this study are provided useful analysis results for proposing solutions to optimize the structure of the suspension system of the vehicle to improve the ride comfort of the vehicle and reduce the dynamic load of multi-axle truck vehicles.

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