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Vehicle Dynamics Response to Road Hump using a 10 Degrees of Freedom Full-Car Model

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Abstract:

A mathematical model with *3D 10-DOF* vehicle model is developed to simulate and study the effect of changing the suspension parameters (*stiffness and damping coefficient*) on the vehicle dynamics in case of a vehicle crossing a sinusoidal road hump.

Lagrange's equation is used to derive the equation of motion and system matrices and then *MATLAB* is used to solve the differential equation of motion by using the central difference method. The model includes 10 motions (*10 DOF*) and it can be used to reduce cost , time of trial and optimize the vehicle suspension parameters. The maximum bounce of the driver, passenger and vehicle-chassis increases by 85.1 %, 107.1 % and 64.5 % respectively as the vehicle speed increases from 25 to 50 % during crossing a sinusoidal hump.

Keywords — **Vehicle dynamics , Mathematical modelling , 10 DOF full-car model, Driver bounce, Passenger bounce, Chassis bounce.**

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I. INTRODUCTION

Krebs (2001) presented a genetic toolbox of vehicle that can be used to assemble and simulate in real-time any kind of wheeled or tracked vehicles. He calculated the vehicle dynamics for a 6 DOF system, type of the ground and the aerodynamic forces [1]. Scare (2002) developed and validated a vehicle dynamic model to predict the maximum acceleration rates of passenger vehicles. His model predicted speed and acceleration profiles in all domains for a variety of vehicle types [2]. Duni et. al. (2003) described a numerical methodology based on the finite element method used for the dynamic simulation of a full vehicle rolling on different obstacles. Their methodology is based on integration of Abaqus Implicit and Explicit codes and applied to Fiat Punto car passing over a comfort and pothole obstacle [3]. Kruczek and Stribrsky

(2004) designed a full-car model with passengers.

They used four quarter-car suspension models connected to produce the full-car model [4]. Mueller (2005) created a full vehicle ADAMS/Car model incorporating an empirical tire-road force model and validating the longitudinal performance of the model by using vehicle responses recorded at the track. He measured mass and inertia properties of each vehicle part, locations of all the kinematic joints, damping and stiffness, engine torque and modeled the tire behavior [5].

Patricio, Becker and Lander (2006) introduced a vehicle 3D model with 7 DOF to study the vehicle dynamics. They used the D'Alembert approach and anti-roll bar in the model. They used the MATLAB command 'ode45' to get the response of the vehicle [6]. Rodonyi and Gasper (2007) a full vehicle model built in the MATLAB/Simulink environment. Their model covered the brake, steering and suspension systems [7]. Lee and Heo (2008) developed a full vehicle model with 14 DOF providing analytical and experimental analyses. They used 3 DOF for the horizontal vehicle model, 7 DOF for the vertical model and 4 DOF for the tire model [8]. Smoker (2009) presented the evolution of a full-car model into virtual environment to virtually demonstrate the car dynamics resulting from various inputs controlled passively and actively. He used a 7 DOF model excited by a bump or harmonic inputs [9]. Zaheh, Salehpour, Jamalir and Haghgoo (2010) used a multi-objective uniform-diversity genetic algorithm to optimize a 5

DOF vehicle vibration model considering five conflicting functions. The optimization process allowed more choices for the optimal design of the vehicle model [10].

Sun and Cui (2011) studied the influence of parameter variations on the full-car model in the frequency domain. They performed system identification of the full-car model using MATLAB [11]. Kamalakkannan, Elayaperumal and Managlaramam (2012) created a full-car model fulfilling the guidelines of BAJA SAEINDIA rules. Their model has 7 DOF incorporating the road excitation modeling [12]. Obialero (2013) refined an existing vehicle dynamic model for a driving simulation in order to make the driving experience closer to reality. He focused on the development of vertical dynamics to extend the DOF of the model from 10 to 14 [13]. Tukaya and Akcya (2014) studied a multi-objective control of a full-car suspension excited by random road disturbances. They synthesized controllers for a range of orders less than the vehicle model order using the HIFOO toolbox. They examined the efficiency of their proposed procedure using several design examples [14].

II. ANALYSIS

2.1 Full-Car Model

In the model as shown in Fig.1, the vehicle body (chaises) is represented by the sprung mass (M_1) while the mass due to the axles and tires are represented by unsprung masses (m_1, m_2, m_3, m_4). The suspension elements (springs and dampers) lie between the sprung and unsprung masses. The vertical stiffness of each of the four tires is represented by the springs (k_{11}, k_{22}, k_{33}, k_{44}). The driver and passengers seats stiffness and dampers are represented by (k_5, k_6) & (c_5, c_6) respectively and (m_5, m_6) represent driver and passengers with seats masses.

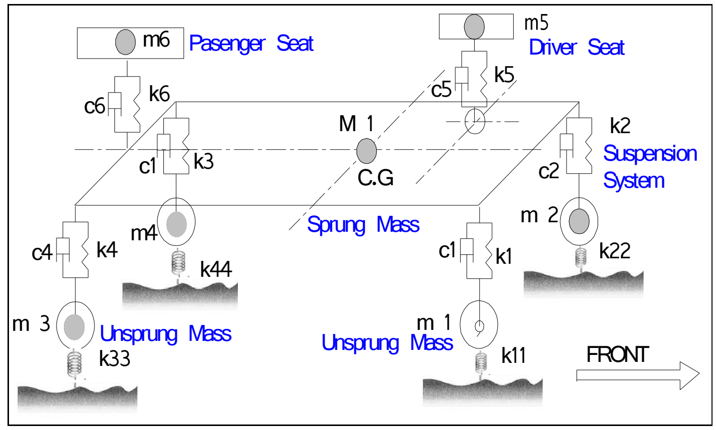

Fig.1 A Simplified Physical Model of the Vehicle

The base point of the main Cartesian coordinate system is set to the CG of vehicle sprung mass. The directions of the coordinate system are set as follows: the Z-axis points upward vertically (positive upward); X-axis is set along the vehicle running direction (forward direction); Y-axis perpendicular to the X-Z plan and pointed to the right which represent the lateral direction as shown in Fig.2. The vertical axis, Z, is used to study ride, pitch and roll of the system.

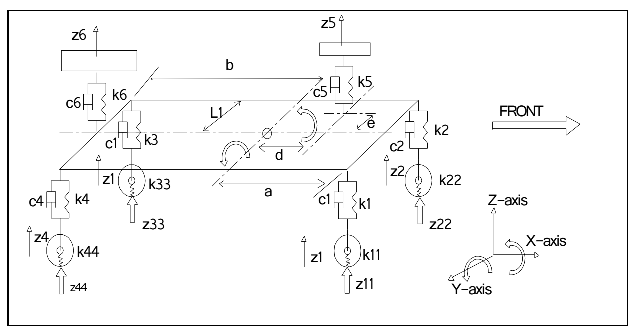

Fig.2 Vehicle dimensions, motions and parameters.

2.2 model assumptions

The model is derived using the following assumptions to simplify the equations and calculations.

1. The chassis frame, vehicle body and engine are rigidly connected.
2. Aerodynamics effect is neglected.

3. The suspension system is passive.
4. The effect of camber angle is neglected.
5. The road inclination angle is zero and smooth road surface except the hump surface.
6. The tire contact with the ground is assumed to be a point.
7. All springs have linear characteristics and the damping elements have a linear damping coefficients.
8. Vehicle coordinate system used is according to SAE J670e except for Z-axis positive pointed upwards vertically.
9. The reference axes (*X, Y, Z*) are selected to coincide with the principal inertial axes of the sprung mass ($I_{xy}=I_{xz}=I_{yz}=0$).
10. The principal elastic axes of suspension elements are orthogonal with the main reference axes *X, Y, Z.*
11. The effect of heat on dampers characteristics is neglected.

2.3 System Degree Of Freedom

There are 10 degrees of freedom for the vehicle system of Fig.1 as follows:

1. Vehicle chassis vertical motion in Z-axis.
2. Vehicle chassis pitching (about Y-axis) , Θ .
3. Vehicles chassis rolling (about X-axis) , Ø.
4. Vehicle chassis lateral motion, y.
5. Driver bounce (vertical motion in Z-axis), z_5.
6. Passengers bounce (vertical motion in Z-axis), z_6.
7. Front right wheel bounce , z_1.
8. Front left wheel bounce , z_2.
9. Rear left wheel bounce , z_3.
10. Rear right wheel bounce , z_4.

The model is exited at wheels by motions z_{11}, z_{22}, z_{33} and z_{44} which result from the crossing of vehicle over the sinusoidal hump.

2.4 Vehicle Model Parameters

The system parameters of the vehicle dynamic system , vehicle dimensions and hump profile dimensions are shown in Table I [16].

TABLE I
MODEL PARAMETERS [16]

Parameter	***Value***	***Parameter***	***Value***
M_1(Kg)	1100	c_3,c_4 (Ns/m)	2500
m_1,m_2 (Kg)	25	c_5 (Ns/m)	150
m_3, m_4 (Kg)	45	c_6 (Ns/m)	300
m_5 (Kg)	90	k_{11}, k_{22} (N/m)	250000
m_6 (Kg)	180	k_{33}, k_{44}	250000
I_x (kg.m^2)	550	k_{1t}, k_{2t} (N/m)	5250
I_y (kg.m^2)	1848	k_{3t} , k_{4t}	5250
k_1,k_2 (N/m)	1500	a (mm)	1200
k_3,k_4 (N/m)	1700	b (mm)	1400
k_5 (N/m)	15000	L_1 (mm)	500
k_6 (N/m)	30000	c (mm)	300
c_1,c_2 (Ns/m)	2500	d (mm)	250

2.5 Hump Dimensions

The road disturbance is assumed sinusoidal with the following values:

- Wave length (L_h) equal *4.5* meter, Wave peak amplitude (H) equal *50* mm.

Fig.3 shows the hump.

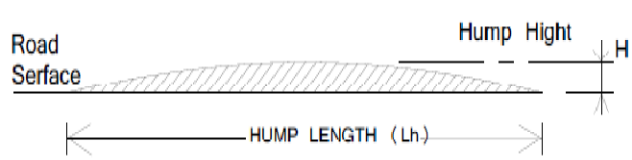

Fig.3 Sinusoidal hump.

2.4 The Model Equations:

The equations representing the full-car model are derived the following Lagrange's equation.

Kinetic energy (T):

$$T=\frac{1}{2}M\dot{Z}^{2}+\frac{1}{2}M\dot{Y}^{2}+\frac{1}{2}I_{x}\dot{\phi}^{2}+\frac{1}{2}I_{y}\dot{\Theta}^{2}+\frac{1}{2}m_{1}\dot{z}_{1}^{2}+\frac{1}{2}m_{2}\dot{z}_{2}^{2}+\frac{1}{2}m_{3}\dot{z}_{3}^{2}+\frac{1}{2}m_{4}\dot{z}_{4}^{2}+\frac{1}{2}m_{5}\dot{z}_{5}^{2}+\frac{1}{2}m_{6}\dot{z}_{6}^{2} \quad (1)$$

Potential energy (U):

$$\begin{aligned}U=&\frac{1}{2}k_{1}\left(Z-l\varphi+a\Theta-z_{1}\right)^{2}+\frac{1}{2}k_{2}\left(Z+l\varphi+a\Theta-z_{2}\right)^{2}\\&+\frac{1}{2}k_{3}\left(Z+l\varphi-b\Theta-z_{3}\right)^{2}\\&+\frac{1}{2}k_{4}\left(Z-l\varphi-b\Theta-z_{4}\right)^{2}+\frac{1}{2}k_{11}\left(z_{1}-z_{11}\right)^{2}\\&+\frac{1}{2}k_{22}\left(z_{2}-z_{22}\right)^{2}\\&+\frac{1}{2}k_{33}\left(z_{3}-z_{33}\right)^{2}+\frac{1}{2}k_{44}\left(z_{4}-z_{44}\right)^{2}\\&+\frac{1}{2}k_{1t}Y^{2}+\frac{1}{2}k_{2t}Y^{2}\\&+\frac{1}{2}k_{3t}Y^{2}+\frac{1}{2}k_{4t}Y^{2}+\frac{1}{2}k_{5}\left(z_{5}-e\varphi-d\Theta-Z\right)^{2}\\&+\frac{1}{2}k_{6}\left(z_{6}+b\Theta-Z\right)^{2}\end{aligned} \quad (2)$$

Damping (D):

$$\begin{aligned}D=&\frac{1}{2}c_{1}\left(\dot{Z}-l\dot{\phi}+a\dot{\Theta}-\dot{z}_{1}\right)^{2}+\frac{1}{2}c_{2}\left(\dot{Z}+l\dot{\phi}+a\dot{\Theta}-\dot{z}_{2}\right)^{2}\\&+\frac{1}{2}c_{3}\left(\dot{Z}+l\dot{\phi}-b\dot{\Theta}-\dot{z}_{3}\right)^{2}\\&+\frac{1}{2}c_{4}\left(\dot{Z}-l\dot{\phi}-b\dot{\Theta}-\dot{z}_{4}\right)^{2}+\frac{1}{2}c_{5}\left(\dot{z}_{5}-e\dot{\phi}-d\dot{\Theta}-\dot{Z}\right)^{2}\\&+\frac{1}{2}c_{6}\left(\dot{z}_{6}+b\dot{\Theta}-\dot{Z}\right)^{2}\end{aligned} \quad (3)$$

Taking the derivatives of Eqs.2, 3 and 4 according to Lagrange's equation gives the differential equations of the 10 DOF full-car model [17].

III. VEHICLES DYNAMICS SIMULATION

3.1 MATLAB Simulation

The MATLAB software program is used to solve the differential equations of motion by using the central difference method as a numerical integration method.

(a) Wheels response:

Fig.4 shows the wheels bounce when the vehicle is crossing the hump with 25 km/h speed.

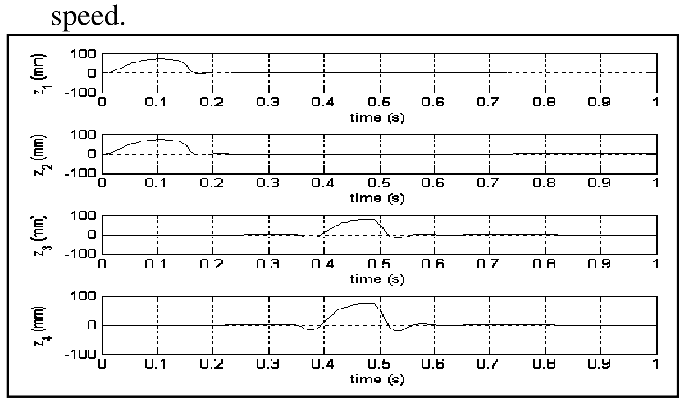

Fig.4 Wheels bounce at 25 km/h speed.

In Fig.3, the front wheels (unsprung masses) have the same reaction for the hump and preceding the rear wheels with a time gap related to the vehicle speed.

3.2 Effect of vehicle speed on its dynamics

Driver bounce:

- On crossing the hump, the driver response at vehicle speed of 25 and 50 km/h is shown respectively in Figs.5 and 6.

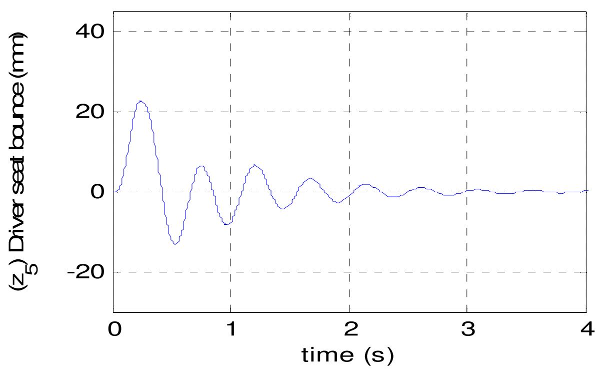

Fig.5 Driver bounce at 25 km/h speed.

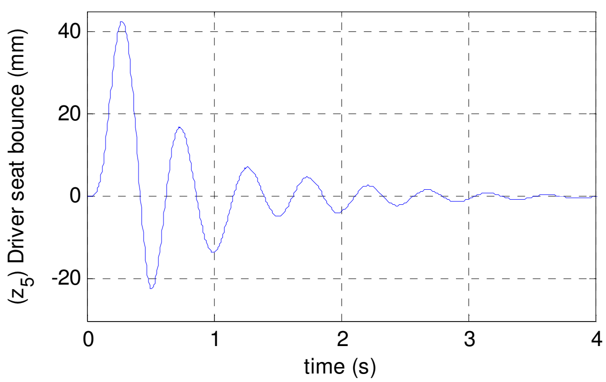

Fig.6 Driver bounce at 50 km/h speed.

Passenger bounce:

- The passenger bounce (on the back seats of the vehicle) have the vibration response shown in Figs 7 and 8 as excited by the hump at 25 and 50 km/h speeds.

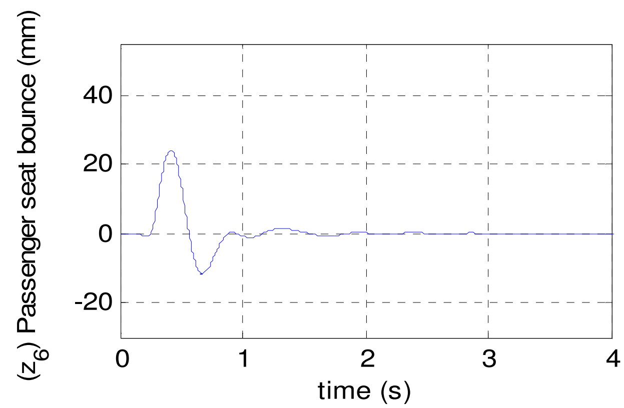

Fig.7 Passenger bounce at 25 km/h speed.

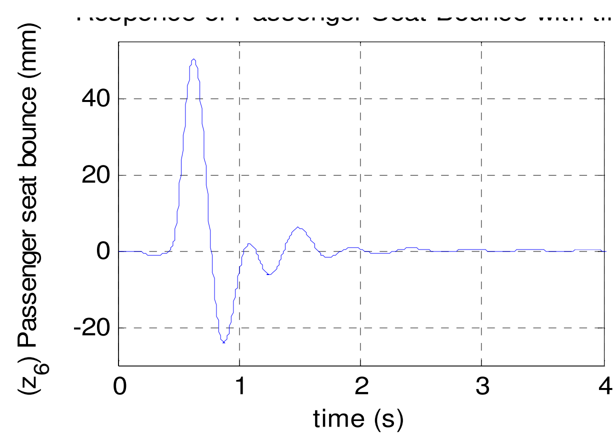

Fig.8 Passenger bounce at 50 km/h speed.

Chassis bounce:

- The chassis bounce is the time response of the sprung mass as the vehicle crosses the hump.
- This is the vertical dynamic motion of the vehicle centre of mass.
- Figs.9 and 10 show this bounce for the 25 and 50 km/h vehicle speed respectively.

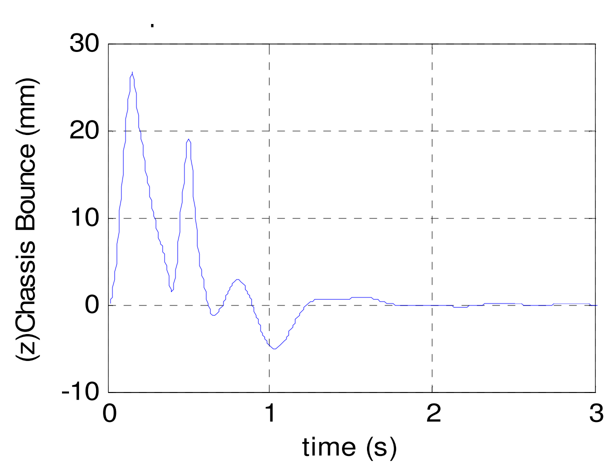

Fig.9 Chassis bounce at 25 km/h speed.

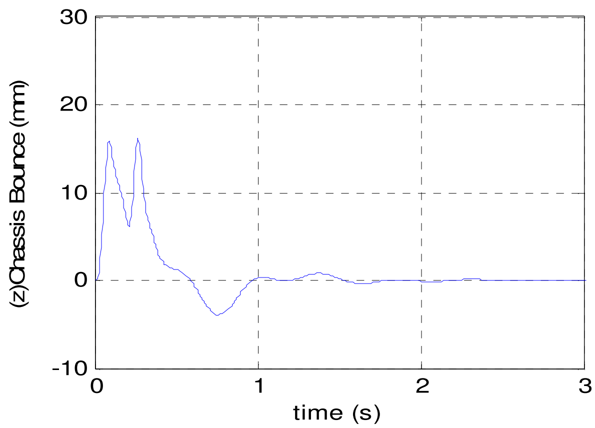

Fig.10 Chassis bounce at 50 km/h speed.

IV. CONCLUSIONS

- A 10 DOF full-car model with passive elements was considered in this study to investigate the car dynamics during passing a sinusoidal hump.
- The vehicle dynamics were simulated using the MATLAB computer program for a vehicle crossing a sinusoidal hump.
- Two vehicle speeds of 25 and 50 km/h ware considered during crossing the sinusoidal hump.
- The bounce of the four vehicle wheels was investigated when crossing the hump.
- The bounce of the vehicle chassis represented by its centre of mass was investigated.
- The bounce of the vehicle driver and the passengers on the back vehicle-seat was also demonstrated upon crossing the hump.
- The maximum driver bounce increased by 85.1 % as the hump crossing speed increased from 25 to 50 km/h.
- The maximum passenger bounce increased by 107.1 % as the hump crossing speed increased from 25 to 50 km/h.
- The maximum chassis bounce increased by 64.5 % as the hump crossing speed increased from 25 to 50 km/h.

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