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Analysis and Design of Light Vehicles for Rural Roads Considering Vibration and Its Performance

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Abstract: This paper investigates light vehicle vibration due to rural road geometry. Normally due to bumps, potholes, uneven road conditions, the life of such vehicles are getting reduced. The mathematical model with higher degrees of freedom was developed and simulated on Matlab/Simulink software. Various independent/dependent parametric variables are considered such as: bumps, pitching, bounce, suspension, tire stiffness coefficient and damping etc. The simulation is performed over a range of bump heights (in the range of 0.025 m, 0.050 m, 0.075 m, 0.100 m). , 0.125 m and 0.150 m) and the vehicle speed is kept from 25 km/h to 125 km/h. Simulation results show that vehicle performance and comfort is achieved at higher speed of 65 km/h in the very poor rural road conditions in India. It is also observed that the vehicle ride can be improved by using proper tyre stiffness coefficient, suspension, damping devices and sprung mass and unsprung mass when designing a car model is done with seven degrees of freedom.

Keywords*: Automobile, Mathematical model, Vibration, Matlab, Comfort ride, Environment*

1. Introduction

As on 31 March 2020, India has the second largest road network in the world with a road network of 6,215,797 km after the United States of America with 6,853,024 km. India has developed about 138,531 km of National Highways and Expressways, 176,818 km of State Highways and has a total length of 3,15,349 km. Remaining 3,06,448 kms; approx. 50% of the road network is in rural areas. This shows that still around 50% of the road network is rural based and is neither made of cement pavement nor with bituminous of the required strength, though it is an important mode of transport for rural population. Due to unevenness, potholes on the roads and sometimes due to damaged roads, the vehicles plying on these roads are causing inconvenience to the passengers as well as the driver. This causes them severe stress and fatigue. It is observed that light vehicles for rural roads such as: transport carriers or passenger cars/jeeps need proper spring, damper and leaf suspension as well as proper tyre stiffness to maintain comfort and speed above 60 km/h. is required.

2. Literature Review

Detailed literatures of various researchers work done in this field were studied. It is found that researchers formulated a mathematical model of light vehicle (car) with a non-linear equation, linear quadratic regulator method etc. They call their model algorithms, Matlab, 20sim, Adams, Bond graph, 20sim mechanism or fuzzy control etc. These studies are described as follows:

An active suspension of a full car model with seven degrees of freedom was investigated by Hadi Adibi and Geoff Rideout et al. [1] for the reduction of bounce, pitching and rolling effects on random road profiles. Their hybrid bond graph simulation results for buoyance acceleration pitch acceleration and roll acceleration. Bond graph concept was invented by H.M. Poynter et al. [2] Physical System Modeling in 1961. This approach supports 0 and 1 junction element. It means the object as a system of interconnected elements. Vertical dynamics of passenger car using Bond graph with eight degree of freedom full car model of passenger car assuming rolling motion of spring, unsprung mass, pitching, spring mass was done by Vivek Kumar et al. [3]. A study on quarter motor vehicle about the comfortable ride with eleven degrees of freedom mathematical model through MATLAB/Simulink vehicle was done by Liqiang Jin et al. [4].

The mathematical model of quarter car evaluated in spring mass, displacement and acceleration was simulated through MATLAB for comfortable ride of quarter car speed not exceeding 6.75km/h by Galal Ali Hasan et al. [5] and a complete car model with seven degrees of freedom was built with a virtual prototype by Yeqing Lu et al. [6] and the simulation result was obtained by using ADRC controller for ride comfort and

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Ji Gao et al. [7] also developed a virtual model of the ride comfort of the vehicle. D. Karnop and Rosenberg et al. [8] parameterized mathematical systems modeled by the Kirchhoff bond graph approach to electrical networks and Yazan M. Al. Rawshad et al. [9]. A chassis of the car was studied and the position of its centre of gravity was fixed with a swarm optimization technique. The simulation mechanism was used to create a full car active suspension using the laws of motion.

Radionova L.V. et al. and other researchers [10–13] presented for building mathematical models using Matlab/Simulink. Cheng Cheng et al. [14] described the road ride comfort and road-holding performance enhancements to the Active Variable Geometry Suspension (SAVGS) concept. R S. Sharp et al. [15], J.B. Ashtekar et al. [16], L.Dahil et al. [17], Shink Jen Wu et al. [18] and Anil Shirhatt et al. [19] introduced that potential road performance of active suspension limited control bandwidth is obtained with theoretical analysis when using the primary function of vehicle suspension by the tires to the transmitted passenger. The results were bounced back and passenger acceleration and displacement reduced by 74.2%, 82.7% and 28.5% respectively. Vivek Kumar et al. [20] studied high-speed rail transport passengers which creates a hunting problem. Because of this inconvenience and physical damage, the Bond Graph model was used in the 31 Degrees of Freedom railway vehicle.

Junoh, AK et al. [21] said that comfort vibration of the passenger car cabin is a comfortable driving environment. The disturbance depends on the magnitude, frequency direction and duration of the vibration. Vinay R Varude et al. [22], studied that suspension system provides a compromise between both ride comfort and road holding. This paper focuses on a two degree freedom passive suspension system model. German Filippini et al. [23] evaluated a four-wheel nonlinear vehicle dynamic bond graph model. Modeling and Simulation 20sim software utilizes the chassis, transmission, pneumatic tyres and bond graph models of the vehicle to achieve through 20sim simulation, Kum-Gil Sung et al. [24] evaluated a robust vibration control using an electro rheological (ER) suspension system passenger vehicle by Fuzzy Moving Sliding Morse Controller (FMSMC) and experimentally realized that the vibration level of sprung mass acceleration can be increased by ER suspension. By using the body resonance can be significantly reduced.

Keith J Wakeham et al. [25] tested the vehicle's active suspension controller using the linear quadratic regulator (LQR) method. It was found that the acceleration of pitch is 40% higher, compared to a 90% increase in the decoupled model. Brendan J. Chan et al. [26] obtains acceleration versus time and displacement versus time using Matlab to evaluate their design simulation results for ride control systems. Compare the results of the modified MCVD system versus the idle system of chassis acceleration, chassis displacement and axle displacement versus time.

Guangqiang Wu et al. [27] deal with rigid and rigid flexible coupling vehicle multibody mode. The Finite Element Method (FEM) makes for a flexible rear suspension. Saayan Banerjee, et al. [28] mentioned that 17 degrees of freedom trailing arm is related to the mathematical model of a full track with hydro-gas suspension. Amalendu Mukherjee et al. [29] developed using the Symbols2000 software for modelling, simulation and design creation. It incorporates a feature called the Encapsulation Subsystem Model called a capsule. Rafal Budzik et al. [30] states that the information in the vibration signal allows for driving safety and comfort on the road. Motor engines as vibration consider rotating machinery as vibration generator.

Sihem Dridi et al. [31] show that the tubular permanent magnet linear synchronous actuator (TPMLSA) dynamic actuator is modeled by the Bond graph formalism. Minimizing wheel vibration problem for comfort ride vehicle, A Seygin et al. [32] study the effect of vibration using a simulation program of a complete vehicle model. The road roughness is used as an input to the system. If a driver travels at a speed of 72 km/h (20 m/s) for 5 to 6 hours on a smooth road, he feels uncomfortable. Mohd. Huff and Rajeev Srivastava et al. [33], proposes for an active suspension system of automobiles to improve ride comfort for passengers and vehicle stability by reducing the vibration effect on the suspension system.

Amar Majid et al. [34], describes Modeling, Simulation and Control of Linear Half Car Suspension System with Algorithms using Matlab/Simulink Study of Pitch, Heave Motion of Sprung Mass Active Suspension with Fuzzy PID. Kyung-Tak Hong et al. [35] dealt improvements passenger car ride comfort using air cells. The use of air cells for various road turbulences creates optimal pressure between the human body and the seat surface. The experimental method obtains the spring constants and damping coefficients of an air cell with 3 degrees of freedom of a quarter cars.

By Manoj K. Mahala et al. [36], lumpy parameter mathematical models have been described for the study of vehicle dynamics. In this paper, different models are studied under different road conditions. Loukas S Louka et al. [37] presents an integrated model of vehicle subsystems using bond graphs. An energy-based model approach is applied to improve vehicle system performance. Ashish R Patil et al. [38] describes about quarter car models of non-linear spring force, Hyundai's property, and the Electra model suspension spring.

A. Mitra et al. [39] states that the full car model for different road profiles is analytically validated with Matlab/Simulink. M.Hammed et al. [40] provides a Suspension system for comfortable ride of seven degrees of freedom of a complete car using Matlab. The main function of the suspension is to protect the driver and passenger from vibration. The spring stiffness defects are 25%, 50% and 80% using the simulation.

This paper deals with vibration damping of a full car with seven degrees of freedom using Matlab/Simulink. A mathematical model was developed considering bouncing, pitching and rolling conditions with an active suspension system of full car and seven degrees of freedom and results were found using Matlab / Simulink. Based on simulation results, a light and durable car is proposed to be developed on cost effective model instead of expensive vehicles to ply on inconvenient, uneven, pothole, and damaged rural roads, which will have better performance, longevity as well as economical. This will also boost the country's economy and energy saving to curb carbon foot prints.

3. Development of Vehicle Model

3.1. **Dynamic Model**

The full car model develops linear equation of mass, spring and damper with seven degree of freedom suspension for comfort ride. To examine and optimize the vibration of a vehicle, full car vibrating model must be used. Full car model can be seen in **Fig. 1** Full car dynamic model. This model includes the body bounce the full car model may be different for the front and rear suspension and mass distribution.,unsprung mass are $m_2/2$,$m_1/2$, $m_1/2$ and $m_2/2$ respectively,damping coefficient are c_1, c_2, c_3 and c_4 respectively,stiffness are k_1, k_2, k_3 and k_4 are respectively,displacement are x_{21},x_{12},x_{11} x_{22}.

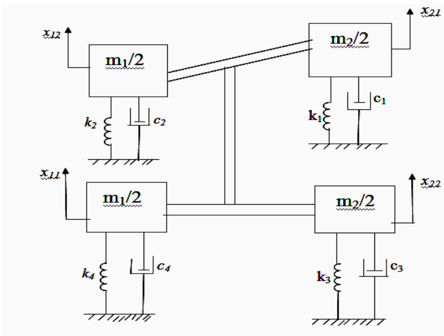


Fig. 1: Dynamic model

Using differential and law of motion authors develop the following linear equation.

Equations of motions of unsprung mass are given as:

$$\frac{m_2}{2}\ddot{x}_{21}+k_1\left(x_{21}-x_{12}\right)+c_1\left(\dot{x}_{21}-\dot{x}_{12}\right)=0 \quad (1)$$

$$\frac{m_1}{2}\ddot{x}_{12}-k_1\left(x_{21}-x_{12}\right)-c_1\left(\dot{x}_{21}-\dot{x}_{12}\right)+k_2x_{12}+c_2\dot{x}_{12}=0 \quad (2)$$

$$\frac{m_2}{2}\ddot{x}_{22}+k_3x_{22}+c_3\dot{x}_{22}=0 \quad (3)$$

$$\frac{m_1}{2}\ddot{x}_{11}+k_4x_{11}+c_4\dot{x}_{11}=0 \quad (4)$$

When we consider stiffness and damping coefficient are equal k_1=k_3 and c_1=c_3

Applying in equation (3), we get

$$\frac{m_2}{2}\ddot{x}_{22}+k_1x_{22}+c_1\dot{x}_{22}=0 \quad (5)$$

Applying Equation (5) into Equation (1) if both are equal

$$\frac{m_2}{2}\ddot{x}_{21}+k_1\left(x_{21}-x_{12}\right)+c_1\left(\dot{x}_{21-}\dot{x}_{12}\right)=\frac{m_2}{2}\ddot{x}_{22}+k_1x_{22}+c_1\dot{x}_{22}$$

$$\frac{m_2}{2}\left(\ddot{x}_{21}-\ddot{x}_{22}\right)+k_1\left(x_{21}-x_{12}-x_{22}\right)+c_1\left(\dot{x}_{21-}\dot{x}_{12}-\dot{x}_{22}\right)=0 \quad (6)$$

Similarly we consider stiffness and damping coefficient are equal k_2=k_4 and c_2=c_4

Appling in equation (4), we get;

$$\frac{m_1}{2}\ddot{x}_{11}+k_2x_{11}+c_2\dot{x}_{11}=0$$

(7)

Applying equation (7) into equation (2) if both are equal

$$\frac{m_1}{2}\ddot{x}_{12}-k_1\left(x_{21}-x_{12}\right)-c_1\left(\dot{x}_{21-}\dot{x}_{12}\right)+k_2x_{12}+c_2\dot{x}_{12}=\frac{m_1}{2}\dot{x}_{11}+c_2\dot{x}_{11}+k_2x_{11}$$

$$\frac{m_1}{2}\left(\ddot{x}_{11}-\ddot{x}_{12}\right)+k_1\left(x_{21}-x_{12}\right)+c_1\left(\dot{x}_{21-}\dot{x}_{12}\right)+k_2\left(x_{11}-x_{12}\right)+c_2\left(\dot{x}_{11}-\dot{x}_{12}\right)=0$$

(8)

Consider displacement, stiffness, damping coefficient are equal

$x_{21}=x_{22}$, $k_1=k_3$, $c_1=c_3$, $c_2=c_4$, $k_2=k_4$

Putting equation (6) and (8) we get;

From equation (6)

$$\frac{m_2}{2}\left(\ddot{x}_{21}-\ddot{x}_{22}\right)+k_1\left(x_{21}-x_{12}-x_{22}\right)+c_1\left(\dot{x}_{21-}\dot{x}_{12}-\dot{x}_{22}\right)=0$$

$$\frac{m_2}{2}\left(\ddot{x}_{21}-\ddot{x}_{22}\right)+k_1\left(x_{21}-x_{12}-x_{21}\right)+c_1\left(\dot{x}_{21-}\dot{x}_{12}-\dot{x}_{22}\right)=0$$

$$\frac{m_2}{2}\left(\ddot{x}_{21}-\ddot{x}_{22}\right)-k_1x_{12}+c_1\left(\dot{x}_{21-}\dot{x}_{12}-\dot{x}_{22}\right)=0$$

(9)

From equation (8)

$$\frac{m_1}{2}\left(\ddot{x}_{11}-\ddot{x}_{12}\right)+k_3\left(x_{22}-x_{12}\right)+c_3\left(\dot{x}_{21}-\dot{x}_{12}\right)+k_4\left(x_{11}-x_{12}\right)+c_4\left(\dot{x}_{11}-\dot{x}_{12}\right)=0$$

$$\frac{m_1}{2}\left(\ddot{x}_{11}-\ddot{x}_{12}\right)+k_1\left(x_{21}-x_{12}\right)+c_1\left(\dot{x}_{21}-\dot{x}_{12}\right)+k_2\left(x_{11}-x_{12}\right)+c_2\left(\dot{x}_{11}-\dot{x}_{12}\right)=0$$

(10)

We consider velocity, stiffness; damping coefficient and displacement are equal

$\dot{x}_{21}=\dot{x}_{22}$, $k_1=k_3$, $c_1=c_3$, $c_2=c_4$, $k_2=k_4$, $x_{21}=x_{22}$

Putting in equation (9) and (10) we get;

From equation (9)

$$\frac{m_2}{2}\left(\ddot{x}_{21}-\ddot{x}_{22}\right)-k_1x_{12}+c_1\left(\dot{x}_{21-}\dot{x}_{12}-\dot{x}_{22}\right)=0$$

$$\frac{m_2}{2}\left(\ddot{x}_{21}-\ddot{x}_{22}\right)-k_1x_{12}+c_1\left(\dot{x}_{21-}\dot{x}_{12}-\dot{x}_{21}\right)=0$$

$$\frac{m_2}{2}\left(\ddot{x}_{21}-\ddot{x}_{22}\right)-k_1x_{12}-c_1\dot{x}_{12}=0$$

(11)

From equation (10)

$$\frac{m_1}{2}\left(\ddot{x}_{11}-\ddot{x}_{12}\right)+k_3\left(x_{22}-x_{12}\right)+c_3\left(\dot{x}_{21-}\dot{x}_{12}\right)+k_4\left(x_{11}-x_{12}\right)+c_4\left(\dot{x}_{11}-\dot{x}_{12}\right)=0$$

$$\frac{m_1}{2}\left(\ddot{x}_{11}-\ddot{x}_{12}\right)+k_1\left(x_{22}-x_{12}\right)+c_1\left(\dot{x}_{21-}\dot{x}_{12}\right)+k_2\left(x_{11}-x_{12}\right)+c_2\left(\dot{x}_{11}-\dot{x}_{12}\right)=0$$

(12)

3.2 Dynamic Model with Pitching

To excellent examine and optimize the full car model with pitching vibration of a vehicle, Full car vibrating model must be used.

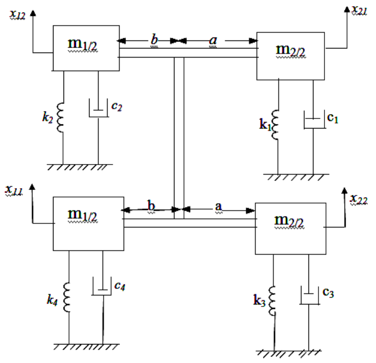


Fig. 2: Dynamic model with pitching

This model includes the body bounce and body roll. The full car model may be different for the front and rear full car due to different suspension and mass distribution. Sprung mass is m,sprungmass displacement are z_1,z_2,z_3 and z_4 respectively, rolling in x direction pitching in y direction and bouncing in z direction respectively, unsprung mass are $m_2/2$,$m_1/2$, $m_1/2$ and $m_2/2$ respectively, damping coefficient are c_1, c_2, c_3 and c_4 respectively, stiffness are k_1, k_2, k_3 and k_4 are respectively,displacement are x_{21},x_{12},x_{11} x_{22} However, vibration model of vehicle must be expanded for including pitch and other modes of vibrations a and b are distance from mass centre to front and rear axle full car model includes body bounce and body pitch Full car model with pitching can be seen in **Fig. 2**

Using Newton's second law, the dynamic equation of bouncing are given as

$$m\ddot{z} - c_1(\dot{z}_1 - \dot{x}_{12}) - k_1(z_1 - x_{21}) - c_2(\dot{z}_2 - \dot{x}_{12}) - k_2(z_2 - \dot{x}_{12}) - c_3(\dot{z}_3 - \dot{x}_{22})$$
$$- k_3(z_3 - x_{22}) - c_4(\dot{z}_4 - x_{11}) - k_4(z_4 - x_{11}) = 0$$

(13)

Using newtons second law, the dynamic equation of pitching are as

$$I\ddot{\theta} - bk_2(x_{12} - b\theta) - bc_2(\dot{x}_{12} - b\dot{\theta}) - bk_4(x_{11} - b\theta) - bc_4(\dot{x}_{11} - b\dot{\theta})$$
$$+ ak_1(x_{21} - a\theta) + ac_1(\dot{x}_{21} - a\dot{\theta}) + ak_3(x_{22} - a\theta) + ac_3(\dot{x}_{22} - a\dot{\theta}) = 0$$

(14)

Using newtons second law , the dynamic equation of rolling are given as

$$I\ddot{\varphi} + bk_2(x_{12} - b\varphi) + bc_2(\dot{x}_{12} - b\dot{\varphi}) + bk_4(x_{11} - b\varphi) + bc_4(\dot{x}_{11} - b\dot{\varphi}$$
$$- ak_1(x_{21} - a\varphi) - ac_1(\dot{x}_{21} - a\dot{\varphi}) - ak_3(x_{22} - a\varphi) - ac_3(\dot{x}_{22} - a\dot{\varphi}) =$$

(15)

3.3 Simulink Model

Simulink model for the same road excitation: System needs to simulate the entire suspension system derive from equation 1 to equation 15 respectively for sprung mass, unsprung mass , unsprung wheel, pitching, rolling and bouncing. The mathematical model of 4-wheeled vehicle with driver seated on cushion seat is simulated with Simulink Software.

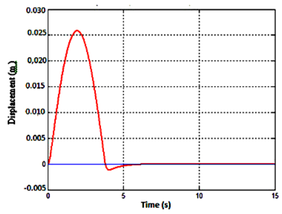
Table-1: Parameter of full car model

Symbol	Parameter Description	Value
m_s	Mass of the sprung mass	1300 kg
m_{uf}	Front mass of the wheel or unsprung mass	65 kg
m_{ur}	Rear mass of the wheel or unsprung mass	60 kg
Ip	Pitch moment of inertia	2391.08 kgm^2
Ir ()	Roll moment of inertia	391.08 kg m^2
k_f	Stiffness of vehicle for front	36300 N/m
k_r	Stiffness of vehicle for rear	19600 N/m
c_f	Front damping coefficient	4000 kNs/m
c_r	Rear damping coefficient	3000 kNs/m
a	Distance from centre of sprung mass to front wheel	1.6 m
b	Distance from centre of sprung mass to rear wheel	0.9 m

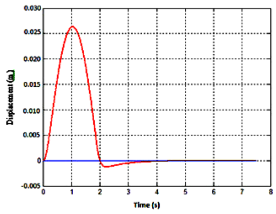
4. Simulation

Simulation results are presented in this paper. Results are also presenting in summarized and graphical form in **Table-1** is obtained after simulating the development model. The different figures in the simulated results obtained at various speeds are 25 km/h to125 km/h taken for simulation of parameter analyzed suspension displacement. I have found representation of simulation result presented easier to review and read. It is clear to understand with logical order through Simulink. Bump height 0.025 m, 0.050 m, 0.075 m, 0.100 m, 0.125 m and 0.150 taken for different vibration in a different time interval in real simulation for comfort ride.

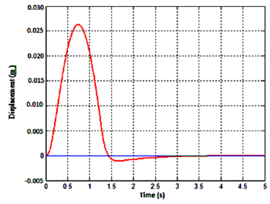
4.1. Simulation results at Bump height 0.025m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below :



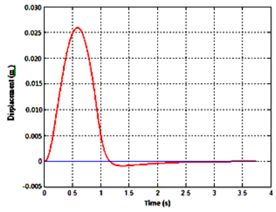
(a)



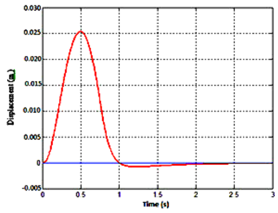
(b)



(c)



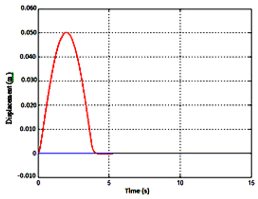
(d)



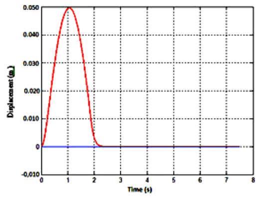
(e)

Fig. 3: Sprung-Mass Displacement vs. Time at Bump height 0.025m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

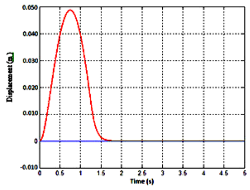
4.2. Simulation results at Bump height 0.050m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below :



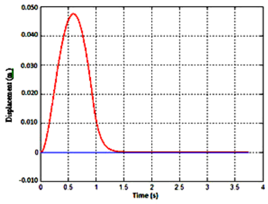
(a)



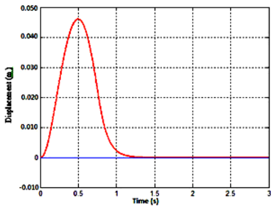
(b)



(c)



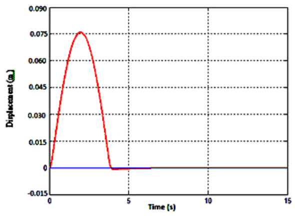
(d)



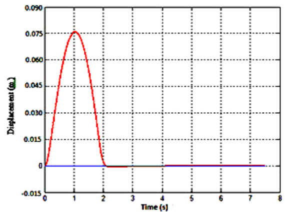
(e)

Fig. 4: Sprung-Mass Displacement vs. Time at Bump height 0.050m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

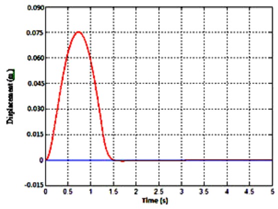
4.3. Simulation results at Bump height 0.075m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below :



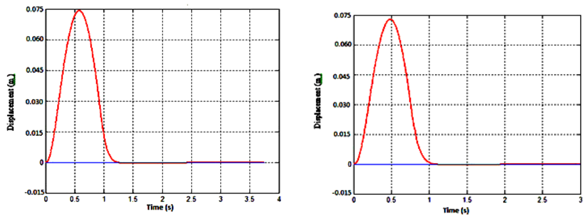
(a)



(b)



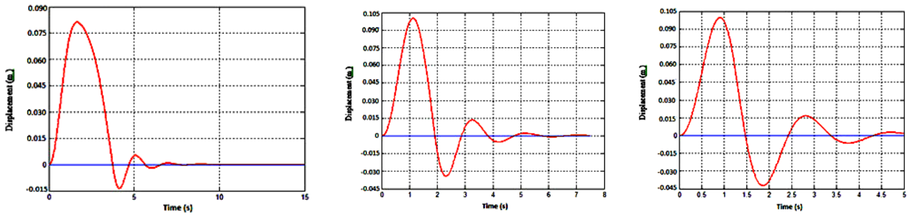
(c)



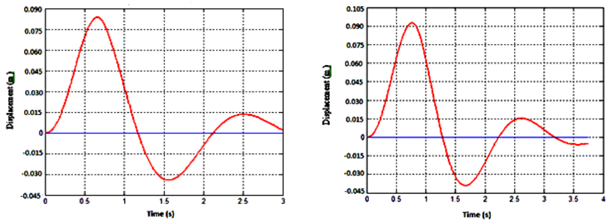
(d) (e)

Fig. 5: Sprung-Mass Displacement vs. Time at Bump height 0.075m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

4.4. Simulation results at Bump height 0.100m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below :



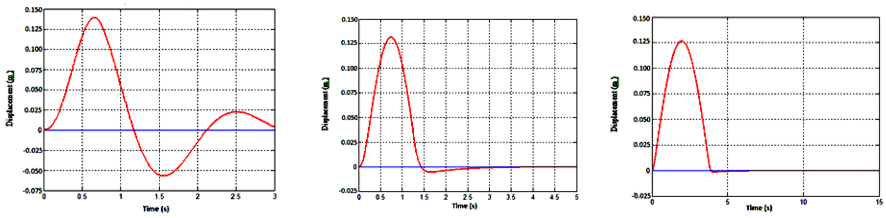
(a) (b) (c)



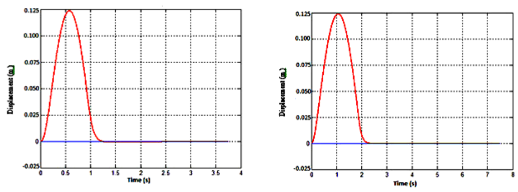
(d) (e)

Fig. 6: Sprung-Mass Displacement vs. Time at Bump height 0.100m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

4.5. Simulation results at Bump height 0.125m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below :



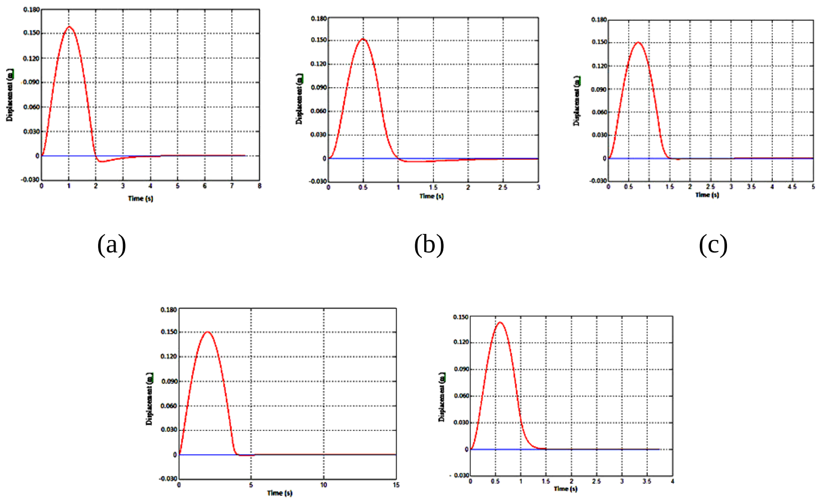
(a) (b) (c)



(d) (e)

Figure 7: Sprung-Mass Displacement vs. Time at Bump height 0.125m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

3.6. Simulation results at Bump height 0.150m and at different speeds of 25 km/h, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr are shown below :



(a) (b) (c)

(d) (e)

Fig. 8: Sprung-Mass Displacement vs. Time at Bump height 0.150m and a speed of (a) 25 km/h, (b) 50 km/h, (c) 75 km/h, (d) 100 km/h, and (e) 125 km/h.

5. Results and Discussion

5.1 From the simulated graphs shown in Fig. 3(a), 3(b), 3(c), 3(d) and 3(e)

At different speeds like: 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr when bump height is kept 0.025 m, maximum amplitudes of vibration and its die out of time periods are listed below in the **Table 2**.

Table 2: Displacement vs. Time at 0.025m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.025 m	25 km/hr	0.026 m	4 s
0.025 m	50 km/hr	0.027 m	2 s
0.025 m	75 km/hr	0.027 m	1.4 s
0.025 m	100 km/hr	0.026 m	1.1 s
0.025 m	125 km/hr	0.025 m	1 s

It is observed that at ***bump height 0.025 m*** vibration amplitudes are 0.026 m, 0.027, 0.027, 0.026 and 0.025 and its corresponding vibration die out time 4 sec, 2sec, 1.4 secs, 1.1 secs and 1.0 sec at vehicle speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr respectively. This indicates that at low vehicle speed vibration gets die out abruptly and time taken was found larger up to 4 seconds whereas at 125 km/hr speeds time taken is 1 sec only for the vehicle under examination for its comfort ride. This situation creates comfort ride at low speed.

5.2 From the simulated graphs shown in Fig. 4(a), 4(b), 4(c), 4(d) and 4(e)

At different speeds like: 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr when bump height is kept 0.050m, maximum amplitudes of vibration and its die out of time periods are listed below in the **Table 3**.

Table 3: Displacement vs. Time at 0.050m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr.

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.050 m	25 km/hr	0.050 m	4 s

0.050 m	50 km/hr	0.050 m	2.2 s
0.050 m	75 km/hr	0.048 m	1.7 s
0.050 m	100 km/hr	0.047 m	1.5 s
0.050 m	125 km/hr	0.046 m	1.3 s

It is observed that at ***bump height 0.050 m***, vibration amplitudes are 0.050 m, 0.050., 0.048, 0.047 and 0.046 and its corresponding vibration die out time 4 sec, 2.2sec, 1.7 sec, 1.5 sec and 1.3 sec at vehicle speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr respectively. This indicates that at low speed (i.e., 25 km /hr), vehicle vibration gets die out abruptly and time taken was found larger up to 4 seconds and at 125 km/hr vehicle vibration gets die out and time taken is found least up to 1.3 seconds for the vehicle under examination for its comfort ride. This situation creates comfort ride at low speed and at larger speeds discomfort for less time.

5.3 The simulation graphs Fig. 5(a), 5(b), 5(c), 5(d) and 5(e)

At different speeds like: 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr when bump height is kept 0.075m, maximum amplitudes of vibration and its die out of time periods are listed below in the **Table 4**.

Table 4: Displacement vs. Time at 0.075m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr.

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.075 m	25 km/hr	0.075 m	4 s
0.075 m	50 km/hr	0.075 m	2 s
0.075 m	75 km/hr	0.075 m	1.5 s
0.075 m	100 km/hr	0.074 m	1.2s
0.075 m	125 km/hr	0.073 m	1 s

It is observed that at ***bump height 0.075 m***, vibration amplitudes are 0.075m, 0.075, 0.075, 0.074 and 0.073 and its corresponding vibration die out time 4 sec, 2 sec, 1.5 sec, 1.2 sec and 1 sec at vehicle speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr respectively. This indicates that at low speed (i.e., 25 km /hr), vehicle vibration gets die out abruptly and time taken was found larger up to 4 seconds and at 125 km/hr vehicle vibration gets die out and time taken is found least up to 1.0 seconds for the vehicle under examination for its comfort ride. This situation creates comfort ride at low speed and at larger speeds discomfort for less time.

5.4 From the simulated graphs shown in Fig. 6(a), 6(b), 6(c), 6(d) and 6(e)

At different speeds like: 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr when bump height is kept 0.100m, maximum amplitudes of vibration and die out of time periods are listed below in the **Table 5.**

It is observed that at ***bump height 0.100 m***, vibration amplitudes are 0.080m, 0.098, 0.097, 0.082 and 0.097 and its corresponding vibration die out time periods are found 3.9 sec, 1.9sec, 1.4 sec, 1.2 sec and 1.3sec at vehicle speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr respectively. This indicates that at low speed (i.e., 25 km /hr), vehicle vibration gets die out in harmonic motion and time taken was found larger up to 5.0 seconds, at 50km/hr vibration also die out in harmonic motion with 4.5 seconds, and at 125 km/hr vehicle vibration gets die out in pulse and time taken was found least up to 1.3 second for the vehicle under examination for its comfort ride. This situation creates comfort ride at low speed and at larger speeds discomfort for less time.

Table 5: Displacement vs. Time at 0.100m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.100 m	25 km/hr	0.080 m	3.9 s
		-0.001 m	4.1s

		0.003 m	5.0s
0.100 m	50 km/hr	0.098 m	1.9 s
		-0.022m	2.3 s
		0.015m	3.2 s
		-0.002m	4.5 s
		0.000m	5.5s
0.100 m	75 km/hr	0.097 m	1.4 s
0.100 m	100 km/hr	0.082 m	1.2 s
0.100 m	125 km/hr	0.092 m	1.3 s

5.5 From the simulated graphs shown in Fig. 7(a), 7(b), 7(c), 7(d) and 7(e)

At different speeds of the 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr at bump height 0.125m; maximum amplitudes of vibration and die out of times are listed below in the **Table 6.**

Table 6: Displacement vs. Time at 0.125m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.125 m	25 km/hr	0.137 m,	1.2 s,
		0.55 m,	-0.55s,
		0.025 m	2.5 sec
		0.000m	4.5sec
0.125 m	50 km/hr	0.131 m	1.4 s
		-0.016 m	1.6 s
		0.00 m	3.0s
		0.001m	5.0s
0.125 m	75 km/hr	0.126 m	4 s
		-0.040m	1.7s
		0.015m	2.7s
		-0.001m	3.7s
		0.001m	5.5s
0.125 m	100 km/hr	0.080 m	0.6 s
		-0.035m	1.6s
		0.015m	2.5s
		0.000m	3.6s
0.125 m	125 km/hr	0.095 m	0.7 s
		-0.040 m	1.6 s
		0.015m	2.6s

	-0.005m	3.5s
	0.000m	4.2s

It is observed that at ***bump height 0.125 m***, vibration amplitudes are 0.137m, 0.131m, 0.126m, 0.124m and 0.125m and its corresponding vibration die out time periods are found 1.2 sec, 1.4sec, 4.0sec, 1.3sec and 2sec at vehicle speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr respectively. This indicates that at low speed (i.e., 25 km /hr), vehicle vibration gets die out in harmonic motion and time taken was found larger from 1.2 to 4.5 seconds and at 125 km/hr vehicle vibration gets die out in harmonic motion and time taken was found least up to 4.2 second for the vehicle under examination for its comfort ride. This situation creates comfort ride at low speed and at larger speeds comfort in little less time.

5.6 From the simulated graphs shown in Fig. 8(a), 8(b), 8(c), 8(d) and 8(e)

At different speeds of the 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr at bump height **0.150m;** maximum amplitudes of vibration and die out of times are listed below in the **Table 7.**

Table 7: Displacement vs. Time at 0.150 m bump height at different Speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr

Bump height	Speed of Vehicle	Vibration amplitude	Time
0.150 m	25 km/hr	0.161 m	2 s
0.150 m	50 km/hr	0.151 m	1 s
0.150 m	75 km/hr	0.150 m	1.5 s
0.150 m	100 km/hr	0.150 m	4 s
0.150 m	125 km/hr	0.142 m	1.5 s

It is observed that at ***bump height 0.150m***, vibration amplitudes are 0.161m, 0.151m, 0.150m, 0.150m and 0.42m and its corresponding vibration die out time periods are found 2 sec, 1sec, 1.5sec, 4sec and 1.5sec at vehicle speeds of 25 km/hr, 50 km/hr, 75 km/hr, 100 km/hr and 125 km/hr respectively. This indicates that at low speed (i.e., 25 km /hr), vehicle vibration gets die out in 2 sec and time taken was found larger up to 4 seconds and at 125 km/hr vehicle vibration gets die out and time taken was found least up to 1.5 second for the vehicle under examination for its comfort ride. This situation creates comfort ride at low speed and at larger speeds discomfort for less time.

6. Conclusion

The 4-wheeled vehicle design model with seven degree of freedom was developed and performance was evaluated. The mathematical model of the vehicle for rural roads was prepared on the basis of road conditions in India with bumps taken as: 0.025m, 0.050m, 0.075m, 0.100m, 0.125m and 0.150m for the comfort ride of vehicles moving at different speeds of: 25km/hr, 50km/hr, 75km/hr, 100km/hr and 125km/hr. The fixed and variable parameters such as: spring, damping, tyre stiffness coefficient, sprung mass and unsprung mass etc., is listed in **Table-1.**

The simulation analysis of the model is carried out at various conditions taking vehicle speeds from 25 km/h to 125 km/h and sprung mass displacement 0.025mm to 0.150 mm, from the Fig. 3 to Fig. 8. It is found that when tyre stiffness coefficient is considered constant and vehicle speeds are kept 25km/hr to 125 km/hr on rural roads:

1) With bumps 0.025m to 0.075m, vibrations of vehicle are found as shown in Fig. 3 to Fig.5. This indicates that vibration of vehicle gets die out with first spike in 2 seconds to 4 seconds at a speed of 25km/hr to 125 km/hr. This situation creates discomfort to the rider at low speed and at larger speeds too.
2) With bumps 0.100m to 0125m, vibrations of vehicle are found as shown in Fig.6 to Fig.7. This indicates that vibration of vehicle gets die out in harmonic condition in 4 sec to 5 seconds at speeds from 25km/hr to 125km/hr. This situation creates comfort to the rider at 50km/hr to 75km/hr speeds and at lower and larger speeds gives little discomfort.
3) With bumps 0.150m, vibrations of vehicle are shown in Fig.8. This indicates that vibration of vehicle gets die out in harmonic condition in 3 seconds to 3.5 seconds at speeds of 25km/hr and 50

km/hr. This situation creates comfort to the rider at 25km/hr to 50 km/hr vehicle speeds and at larger speeds between 75km/hr to 125km/hr no comfort situation is seen.

In Indian Rural Road conditions, it is therefore found from the simulations that 4-wheeled vehicles can be designed with a constant tyre stiffness coefficient for speeds ranging from 25km/hr to 125km/hr with comfort ride and best comfort is obtained at 50km/hr to 75km/hr. At these conditions vehicles would also have considerably longer life and will be cost effective due to less fatigue.

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