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Dynamic Analysis of Instability in Three-Wheeler Automobile Vehicle

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Authors' contributions

This work was carried out in collaboration between both authors CCI and PSE. Author PSE designed the study, performed the statistical analysis, wrote the protocol and wrote the first draft of the manuscript. Authors CCI and PSE managed the analyses of the study. Author CCI managed the literature searches. Both authors read and approved the final manuscript.

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ABSTRACT

At pre-design stage of three wheeler automobile vehicle, engineers are posed to use simple models to show the fundamental and functional characterizations of the vehicle. Wobbling instability is consider as one of the major problems of a three wheeled automobile vehicle, and are of great interest to industry and academia. This study outlines the effect of applying camber angles on the handling and lateral stability of a three wheeled automobile vehicle and the use of a rigid body model with two variables, the sprung mass and un-sprung mass. This study investigates the effect of applying camber angles on the vehicle handling, lateral stability, ride comfort and skid of a three-wheeled automobile vehicle. Due to the unbalance nature of the vehicle, controlling the instability especially during turning, cornering and harsh maneuver is very challenging. Hence the stability is ensured by the introduction of an active camber system to the vehicle suspension system. Which are determined by modeling the (quarter car model) vehicle movement using harmonic and transient responses over isolated road bump model and the forces generated on each part of the suspension system are analyzed.

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Keywords: Structural and fundamental characterization; three wheeled automobile vehicle; camber angle, lateral stability; ride comfort and quarter car model.

1. INTRODUCTION

In lure of high costs of petroleum products, significant pressure has put on automotive industries to develop more fuel-efficient passenger's vehicles. Due to safety regard, gas emissions and economy requirements, automobile transportation are made to undergo vast changes in the next few decades [1]. Obviously, the automobile vehicle are made to become smaller and lighter in the search for better fuel efficiency. As a result of these considerations, the designs based on three-wheeled auto vehicles are likely to be the most popular mode of public and private transportation not only in India but in other countries [2]. For instance, three wheelers automobile vehicle have already been used as a means of public transportation system in several countries such as India, Thailand, Peru, China, Nigeria and even Italy as well [3,4].

Three wheeler automobile vehicles also has the advantage of being a compromise between two wheeled and four wheeler automobile vehicles in different aspects like cost, load carrying capacity, fuel consumption, space occupied and weight of the vehicle. Efforts in fixing these problems will directly depict how the vehicle dynamics of three-wheeler is of great significance and its increasing importance to the transportation systems [1]. The three-wheeled vehicles operating in India have their front steering with one wheel similar to those of motor cycles and motor scooters, and the two rear wheels are the driving wheels with a differential and a suspension system, which are the same to those automobiles [1,5].

2. REVIEW OF RELATED WORK

Several authors have been working for innovative approaches as well as models for optimal simplification of Body-in-White (BIW) at early stage of the design [6,7,8,9]. [10] proposed an engineering approach for replacement of beam-like structures (BLS) and joints in a vehicle model. The proposed replacement techniques were based on the reduction of structural modeling approach, which involved a performance analysis of beam-member cross-sections and a static analysis of frame joints [7,8]. [11] presented a beam element model applying a recurrent neural network to extract the input/output relationship between the crash dynamic characteristics and beam element features which shows that the predicted beam element model enables to generating essential crash dynamic characteristics for the concept of BIW design evaluation at a reasonable level of accuracy [12,13].

[14] presented a reduced beam and joint modeling approach to analyze and optimize the low-frequency vehicle performance. They also analyzed the concept modifications in the body beam-like structures (BBLS) and in the joints using the body reduced modal system [12]. [13] investigates a semi-rigid beam element (SRBE) that consists of a beam element with two semi-rigid connections and finally simulate the reliability of member joints. They also developed special a finite element software for structural modeling and analysis of Body-in-White (BIW) in .NET framework [12,5,13]. [4] introduced object-oriented graphics interface design and optimization software for cross-sectional shape of automobile vehicle body which can be used at the conceptual phases of automotive design. This software was designed to optimize the weight and the stiffness of the thin-walled beams using genetic equations [12].

Despite the popularity of three-wheeler automobile vehicle and its attractive features, their typical light-weight and narrow design have one major drawback [15]. They are not very stable in harsh maneuvers and different methods have been proposed to improve their stability among all is the application of camber angle to the wheels and tilting the body are the most considering approaches. Choosing between them completely depends on the purpose of the vehicle. That is, for maneuvering and fast driving, usually tilting system is preferable, while for normal commuting in cities, cambering system is recommended [16,12].

[17] investigate this instability using a multi-body dynamic (MBD) model and with experiments conducted on a prototype of three wheeled vehicle (TWV) on a test track. The Multi-Body-Dynamic (MBD) model of a three wheeled vehicle is developed using the commercial software known as ADAMS-CAR. In an initial model, all components including main structures like frame, steering column and rear forks are assumed to be rigid bodies. A linear eigenvalue analysis that was carried out at different speeds,

reveals a mode that has a predominantly steering oscillation, also called a Wobble mode, with frequency around 5 to 6 Hz [17]. The analysis result depict that the damping of this mode is low but positive up to the maximum speed of the TWV as in Fig. 1a to 1c respectively.

Fig. 1a. Depiction of a three wheeled vehicle [15]

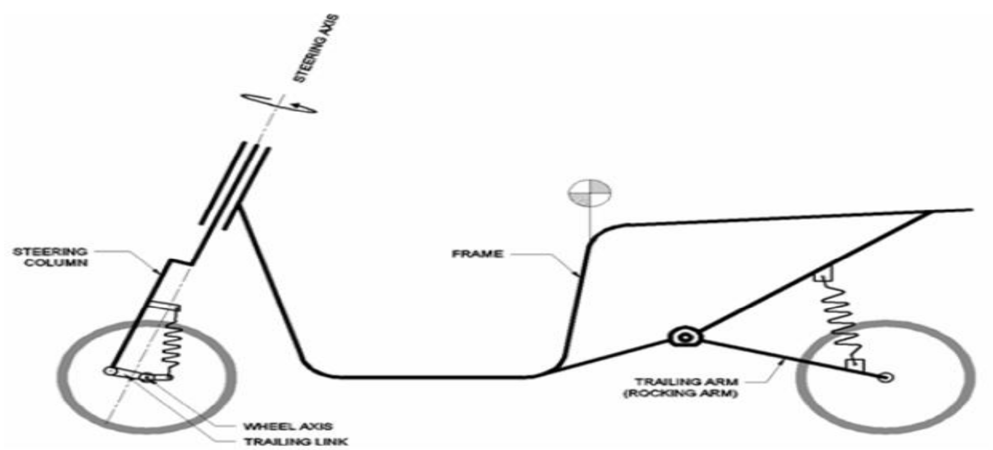

Fig. 1b. A schematic of a three wheeled vehicle [15]

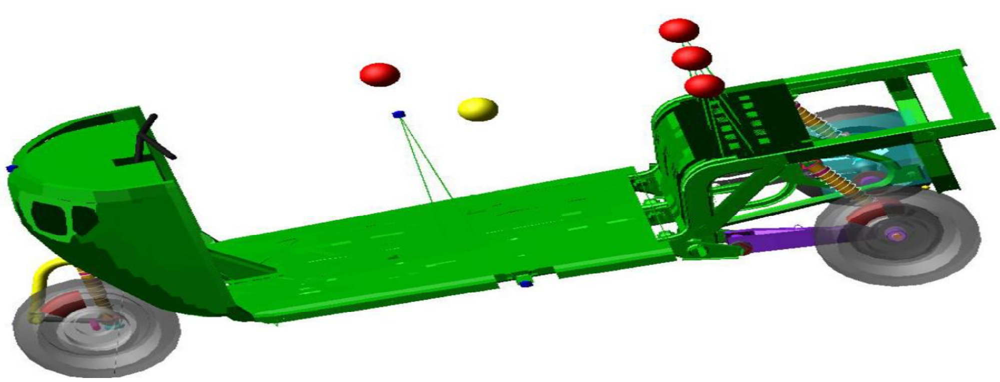

Fig. 1c. Multi-body dynamic model of a three wheeler, represent the center of gravity location of the driver, three passengers and the chassis [15]

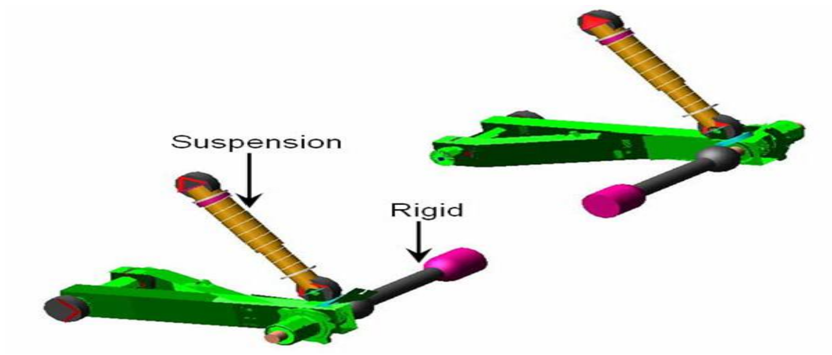

Fig. 1d. Schematic of rear suspension system with flexible model of trailing arm [15]

3. STUDY BACKGROUND

3.1 Spring Force Analysis

This section analyses the ride comfort of three wheeler automobile vehicle (Suspension wheels). The ride comfort is considered as the satisfaction of the vehicle occupant (persons) in motion. Here the ride comfort of a vehicle is determined by the performance of the suspension system (dampers, springs, links and chassis of the vehicle [18]. Basically, the study employs a two degree-of-freedom (2-DOF) quarter car model for analyzing the ride comfort for the sprung mass (m_s) (car body) (see Fig. 2), and can be defined as [16,5].

Where, M_s, M_{us}, K_z , C_t, y_s, y_{us} and s are the sprung mass, the un-sprung mass, the Suspension stiffness, the suspension damping coefficient, the sprung mass displacement, the un-sprung mass displacement and the Road input respectively.

For sprung mass:

$$M_s\frac{d^2y_s}{dt^2}+c_s\left(\frac{dy_s}{dt}-\frac{dy_{us}}{dt}\right)+k_s(y_s-y_{us})=0 \tag{1}$$

For un-sprung mass:

$$M_{us}\frac{d^2y_{us}}{dt^2}+c_s\left(\frac{dy_{us}}{dt}-\frac{dy_s}{dt}\right)+k_s(y_{us}-y_s)+k_t(y_{us}-s)=0 \tag{2}$$

That is;

$$M_{us}\frac{d^2y_{us}}{dt^2}+c_s\left(\frac{dy_{us}}{dt}-\frac{dy_s}{dt}\right)+k_s(y_{us}-y_s)+k_ty_{us}=k_ts \tag{3}$$

$$\begin{cases} M_s\dfrac{d^2y_s}{dt^2}=-c_s\left(\dfrac{dy_s}{dt}-\dfrac{dy_{us}}{dt}\right)-k_s(y_s-y_{us}) \\ M_{us}\dfrac{d^2y_{us}}{dt^2}=-c_s\left(\dfrac{dy_{us}}{dt}-\dfrac{dy_s}{dt}\right)-k_s(y_{us}-y_s)-k_t(y_{us}-s) \end{cases} \tag{4}$$

$$\begin{cases} M_s\dfrac{d^2y_s}{dt^2}+c_s\left(\dfrac{dy_s}{dt}-\dfrac{dy_{us}}{dt}\right)+k_s(y_s-y_{us})=0 \\ M_{us}\dfrac{d^2y_{us}}{dt^2}+c_s\left(\dfrac{dy_{us}}{dt}-\dfrac{dy_s}{dt}\right)+k_s(y_{us}-y_s)+k_ty_{us}=k_ts \end{cases} \tag{5}$$

Then, in matrix form;

$$\begin{bmatrix} M_s & 0 \\ 0 & M_{us} \end{bmatrix}\begin{bmatrix} \dfrac{d^2y_s}{dt^2} \\ \dfrac{d^2y_{us}}{dt^2} \end{bmatrix}+\begin{bmatrix} c_s & -c_s \\ -c_s & c_s \end{bmatrix}\begin{bmatrix} \dfrac{dy_s}{dt} \\ \dfrac{dy_{us}}{dt} \end{bmatrix}+\begin{bmatrix} k_s & -k_s \\ -k_s & k_s+k_t \end{bmatrix}\begin{bmatrix} y_s \\ y_{us} \end{bmatrix}=\begin{bmatrix} 0 \\ k_ts \end{bmatrix}$$

$$\mathbf{M}\ddot{\mathbf{y}}+\mathbf{c}\dot{\mathbf{y}}+\mathbf{k}\mathbf{y}=\mathbf{f} \tag{6}$$

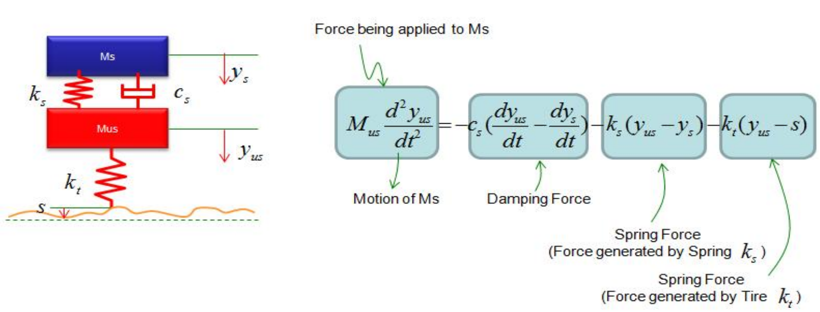

Fig. 2. Summary of governing equations for sprung and un-sprung masses

3.2 Skidding Study

Skidding is one of the major problem experiences in a three-wheeler automobile vehicle as a result of lateral instability. It usually occurs on slippery roads condition or as a result of fast or quick turning [19]. This consequence can lead to rollover of the vehicle by hitting an obstacle during skidding [19]. So during pre-design stage, skidding is very important for engineers for safe car design, and also it is very crucial for accident reconstruction specialists who are trying to design for safer roads [19]. Analyzing the skidding on a dry road, different parameters such as vehicle weight, type of tires, longitudinal speed, are put into consideration [20]. In all considerations, the type of tires and road/tire coefficient of friction is of great important factors which influence the process of skidding [19]. To reduce or prevent skidding, the lateral acceleration must be bounded by the road/tire friction coefficient (μ) as stated in Equation (7) [19,20,5].

$$\alpha_{y\text{-}CG} \leq \mu g \tag{7}$$

In consideration of all parameters, in order to reduce or prevent skidding, the value of road/tire coefficient of friction should be kept high, especially in winter, when the road/tire coefficient of friction is naturally low; this can be achieved by using winter tires and other anti-skidding devices. In case of winter tires, it has been shown that they can significantly prevent skidding on snowy roads, although they are not very beneficial for driving on polished ice, particularly when the temperature is between -10 and 3ºC [21,20]. Hence in this work, the impact of cambering on skidding is considered. According to [21], skid lateral acceleration threshold, based on Fig 3. and can be evaluated using equation (8) and (9) respectively [19,22].

$$\sum F_z = ma_{y(\text{skidding})} \tag{8}$$

Expanding as,

$$\mu F_{z1} + \mu F_{z2} + \mu F_{z3} = ma_{y(\text{skidding})} \tag{9}$$

Where μ is the road/tire coefficient of friction while F_{z1} , F_{z2}, and F_{z3} are vertical forces on the rear-right, front, and rear-left wheel, respectively . This equation can simplify in equation (10) and (11) [19].

$$\mu F g = ma_{y(\text{skidding})} \tag{10}$$

This gives,

$$\mu g = a_{y(\text{skidding threshold})} \tag{11}$$

Using equation (11), the skidding lateral acceleration threshold is directly depends on the road/tire friction coefficient, μ, and gravity acceleration (g), irrespective of the geometry of the vehicle and its weight. Skidding for various road/tire friction situations is listed (Table 1) [19].

Table 1. Friction coefficient for different types of roads [19]

Road type	Road/tire friction coefficient (μ)
Dry	0.90
Wet	0.60
Snowy	0.20
Icy	0.05

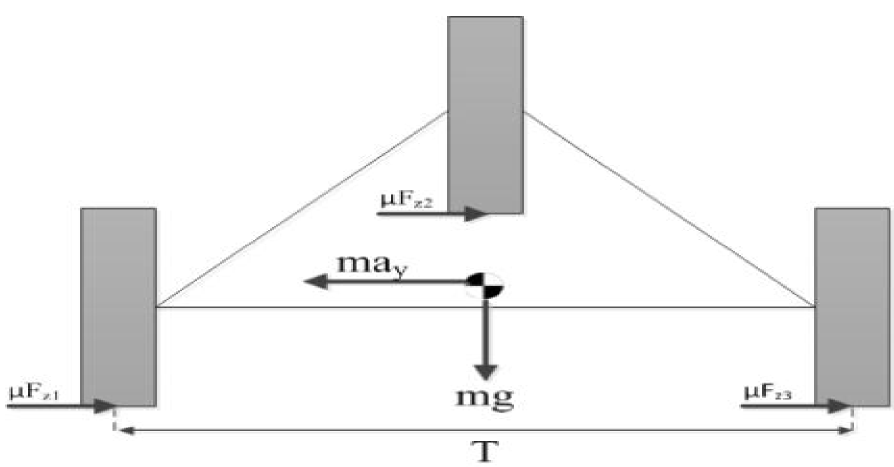

Fig. 3. Free body diagram for skidding [19]

4. METHODOLOGY

4.1 Handling and Stability Analysis using (3-DOF) Model

Using a three Degree of Freedom (3-DOF) model, the lateral velocity, yaw rate, and roll angle are the degrees of freedom and the transfer functions of yaw rate/camber angle and those of yaw rate/*Mz* of each wheel are obtained and discussed accordingly. The handling model of the three-wheeled vehicle is shown in Fig. 4. And all the equations are extracted from Newton's law [19]. All wheels are independently steerable while camber angles can be applied to all wheels separately.

And the general equations of motion with respect to the axes fixed to the vehicle body can be considered as [19] (see Fig. 4).

$$F_{y1} + F_{y2} + F_{y3} = m(\dot{v + ur}) \quad (12)$$

And

$$a(F_{y1} + F_{y3}) – bF_{y2} + M_z = I_z\dot{r} \quad (13)$$

Also,

$$I_s\ddot{\varphi}+c_1\dot{\varphi}+(k_1-m_sgh)\varphi = -m_sh'(\dot{v}+ur) \quad (14)$$

And, lateral forces on rear tires are

$$F_{y1} = c_{\alpha1}\alpha_1 + c_{y1}y_1 \quad (15)$$

$$F_{y3} = c_{\alpha3}\alpha_3 + c_{y3}y_3 \quad (16)$$

Also, lateral force on the front wheel is

$$F_{y2} = c_{\alpha2}\alpha_2 + c_{y2}y_2 \quad (17)$$

F_{y1}, F_{y2}, and F_{y3} are lateral forces on each tires, *m* is the total weight of the vehicle and *ms* is the sprung mass. Slip angles, α_1, α_2, and α_3, are derived according to the corresponding steer angles of each tire (δ_1, δ_2, and δ_3), yaw rate (*r*), lateral speed (*v*), and longitudinal speed (*u*). I_z which can be seen in Equation (13) and I_x, appeared in equation (14), are the mass moment of inertia of the vehicle, respectively [19]. M_z is the control moment which can be produced by torque vectoring, active steering, or their combination [19]. Side slip angles are defined by the vehicle motion variables such as yaw rate, longitudinal, and lateral velocity, and also according to their corresponding steer angles. Slip angle for the rear wheels can be evaluated as in equation (18) and (19), respectively [19].

$$\alpha_1 = \partial_1-(\frac{v+ar}{u}) \quad (18)$$

And,

$$\alpha_3 = \partial_3-(\frac{v+ar}{u}) \quad (19)$$

Also, front wheel can be obtained as shown in equation (20)

$$A_2 = \partial_2-(\frac{v+br}{u}) \quad (20)$$

By plugging Equations (15), (16), (17), (18), (19), and (20) into Equations (12) and (13), we have

$$C_{\alpha1}\partial_1- c_{\alpha1}\,\partial_1-(\frac{v+ar}{u})+c_{y1}y_1+c_{\alpha3}\partial_3-c_{\alpha3}\,\partial_3-(\frac{v+ar}{u})+c_{y3}y_3+c_{\alpha2}\partial_2-c_{\alpha2}\,\partial_3-(\frac{v+br}{u})+c_{y2}y_2= M\,(\dot{v + ur}) \quad (21)$$

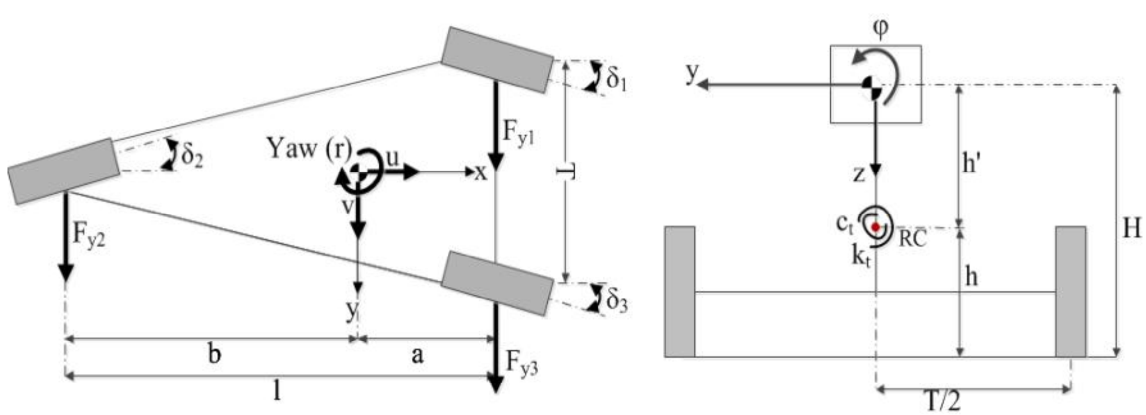

Fig. 4. Handling model of the three-wheeled vehicle by showing degrees of freedom (lateral speed, yaw rate, and roll angle) [19]

While,

$C_{\alpha1}\partial_{1\alpha}$- $c_{\alpha1}$ -($\frac{v+ar}{u}$)α +$c_{y1}y_{1\alpha}$+$c_{\alpha3}\partial_{3\alpha}$-$c_{\alpha3}$ ($\frac{v+ar}{u}$)α+$c_{y3}y_3\alpha$-$c_{\alpha2}\partial_2 b$ +$c_{\alpha2}$ ($\frac{v+br}{u}$)b-$c_{y2}y_2 b$+M_z = $I_z\dot{r}$ (22)

Equations (14), (21), and (22) are basically the expanded forms of equations of motion model for the handling analysis of the three-wheeled vehicle where c_{y1},c_{y2} *and* c_{y3} are the camber coefficients for the rear left wheel, front wheel, and rear right wheel, respectively, and $c_{\alpha1}$, $c_{\alpha2}$ and $c_{\alpha3}$ are the cornering coefficients for the rear left wheel, front wheel, and rear right wheel, respectively [23].

4.2 Isolated Road Bump Model

This is an important aspect that determines the ride comfort of an auto vehicle during motion. This section employs a single and double haversine road profile to study the ride comfort of the three-wheeled automobile vehicle by determining the ride comfort parameters, the sprung mass, the vertical displacement and the acceleration [19]. Using a single haversine road bump, the model shows that the springs and dampers reduce the vibrations which are transmitted to the body, and the input for the rear-left wheel and the rear-right wheels can be determined and modeled using equation (23) [19].

$$y_{rl} = y_{rr} = \text{d.sin}(\frac{2\pi u}{L}t) \quad (23)$$

considering the front wheel, since there is a phase difference between front wheel and the rear wheels which is directly proportional to the length of the car, the input is also calculated by using equation (24) [19].

$$y_f = \text{d.sin}\ (\frac{2\pi u}{L}(t+\frac{l}{u})) \quad (24)$$

Where, *d, L, u, l and* t *are* the height of the bump in meters, the length of the bump in meters, the longitudinal speed (assumed 20 km/h), the length of vehicle 1.8 in meters and the time in seconds [19].

5. RESULTS AND DISCUSSION

Performances analysis of the suspension system in term of ride quality and vehicle handling were carried-out, and the road disturbance is taken as the input for the system. Two type of road disturbance is assumed as the input for the system model. The road profile1 is taken as a single road bump. And the disturbance input model is cited in equation (30), where a, is taken as the bump amplitude. And the sinusoidal bump with frequency of 8H_z has been characterized as in Fig. 5. using equation (25) [19].

$$r(t) = \begin{cases} \frac{a(1-cos8\pi t}{2}, 0.5 \leq t \leq 0.75 \\ 0 \end{cases} \quad (25)$$

Where, a=0.05, and the road bump height equels10 cm.

Similarly, the road profile 2 is assumed to have 3 bumps as shown in equation (31), where a, denotes the bump amplitude. The sinusoidal bump with frequency of 8H_z, 4H_z and 8H_z has been characterized as in Fig. 6. using equation (26) [19].

$$r(t) = \begin{cases} \frac{a(1-cos8\pi t}{2}, 0.5 \leq t \leq 0.75\ ;\ 3 \leq t \leq 3.25\ ;\ 5 \leq t \leq 2.5 \\ 0 \end{cases} \quad (26)$$

Where, a=0.05 and road bump height were 10cm, 5cm, and 10cm).

- **Single haversine bump profile:** Already indicated in section 3.2, the conventional method to analyze the ride comfort is to observe the vertical displacement of the sprung mass (car body) of the three-wheeled automobile vehicle, when it passes over the haversine road bumps. The input for the rear left and right wheels are characterized as shown in Fig. 7 using equation (23) [19].

As shown in Fig. (7), the vertical displacement of sprung mass is damped approximately after two seconds while the maximum amplitude of oscillation transmitted to the car body is 42% of the road amplitude.

- Vertical acceleration of the sprung mass (car body), is another parameter that should be considered when studying the ride comfort of a vehicle, is Fig. 8 shows the imposed acceleration to the sprung mass of the three-wheeled vehicle when it is passing over the haversine road bump. The input for the front wheel is characterized as in Fig. 8, using equation (24) [19].

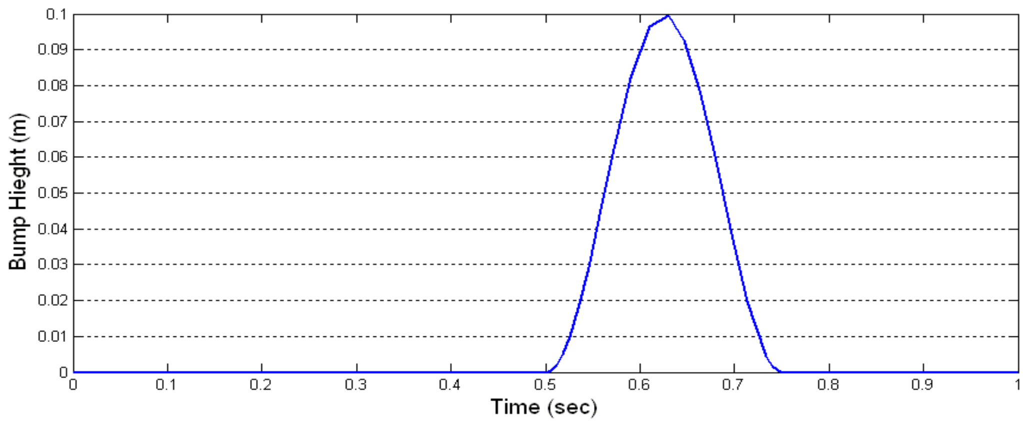

Fig. 5. Road profile 1, depicts sinusoidal bump with frequency of 8Hz

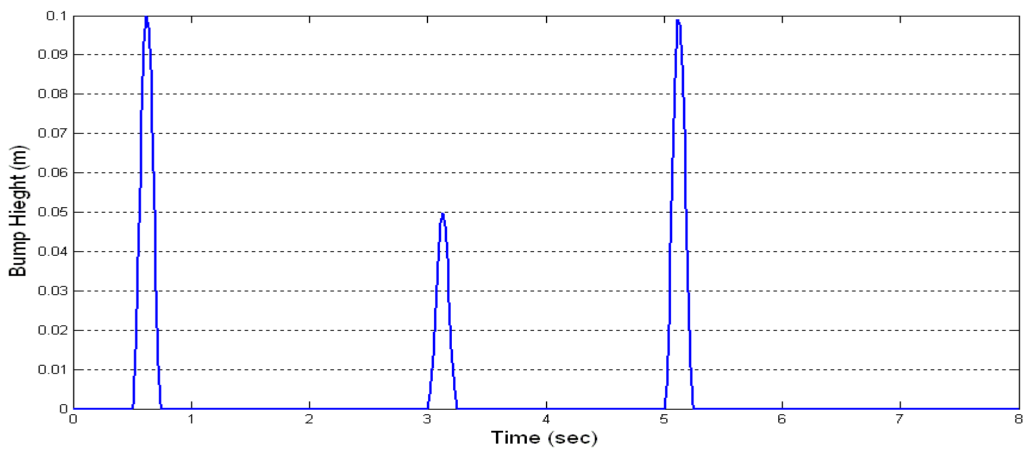

Fig. 6. Road Profile 2, depicts sinusoidal bump with frequency of 8Hz, 4Hz and 8Hz

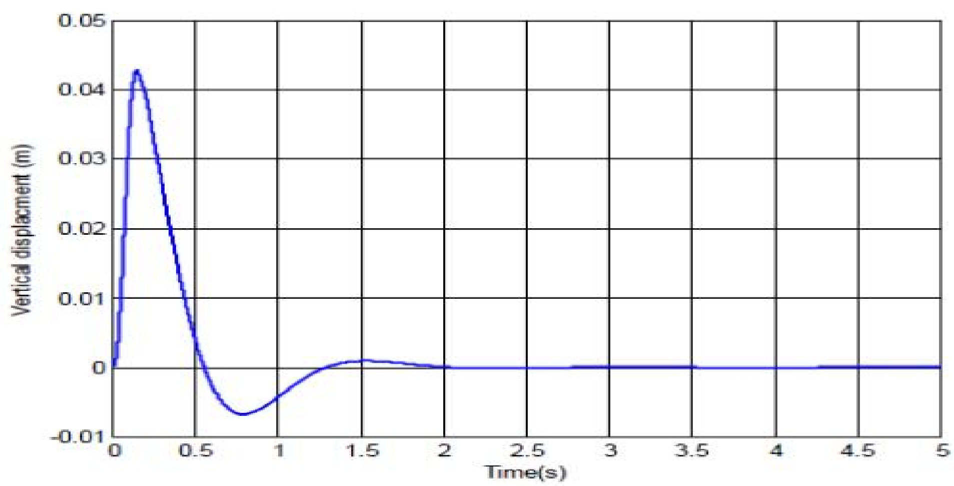

Fig. 7. Vertical displacement of the sprung mass when goes over a haversine road bump

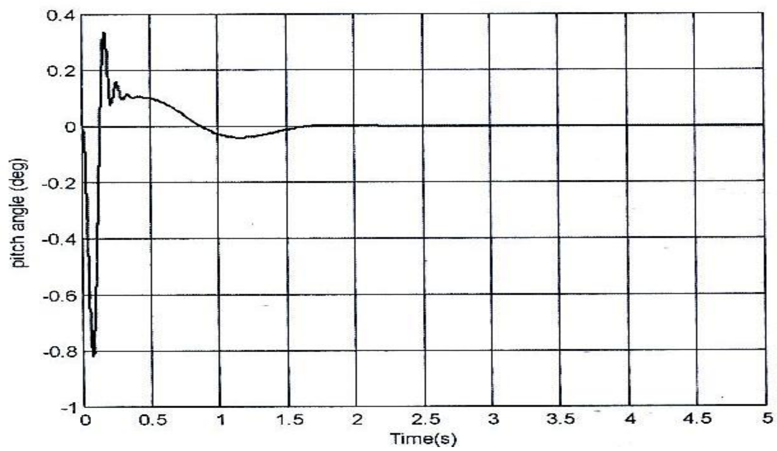

Fig. 8. Vertical acceleration when the vehicle passes over a single haversine bump.

Fig. 8 depicts the maximum acceleration imposed to the car body at the specific speed of 40 km/h is approximately 13.1 m/s2.

6. CONCLUSIONS

This work evaluates the dynamic of three-wheeled automobile vehicle design. And the goals were to observe the importance of active camber system addition to obtain a good ride quality on the vehicle dynamic performance. The conclusions are:

- That the maximum stress occurs at the frequency of 8Hz, because there is a frequency overlap between road load and suspension system, especially when passing-over a single and double haversine road bump, as indicated in transient result analysis.
- That because the vertical displacement of the sprung mass is damped after some seconds the passenger seating is not affected by the increasing and decreasing the values of front suspension damping, rear suspension stiffness.
- That the transient analysis, when passing-over a double haversine road bump, it is observed that by decreasing the values of rear suspension damping and rear tire stiffness, most displacements are on the lower side resulting in comfortless for the passenger.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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