CHAPTER 1
INTRODUCTION

1.1 Background

A car suspension system is a mechanism which physically separates the car body from wheels of car. Performance of the suspension systems has been greatly increased because of increasing vehicle capabilities. To achieve a good suspension system of a car, we have to consider several performance characteristics which deal with the regulation of body movement, suspension movement and the force distribution. The ideal suspension system should isolate the body from inertial disturbances and road disturbances associated with cornering and acceleration or braking. Suspension system should also be able to minimize the vertical force which is transmitted to the passengers for their comfort.

Mainly this objective can be achieved by minimizing the vertical acceleration of car body. An excessive travel of wheel will results in non-optimum attitude of tire relative to the road which will cause poor adhesion and handling. Further, to maintain a good handling characteristic of a car, the optimum tire-to-road contact on four wheels must be maintained. These characteristics are conflicting in conventional suspension system and do not meet all the conditions. Many researchers have studied the suspension system extensively through both experiments and analysis. The main aim of the study is to improve the traditional design trade-off between ride and road handling by controlling the suspension forces directly to suit with the performance characteristics.

Suspension systems can be classified as passive, semi-active, and full-active suspension system. Passive Suspension system consists of conventional components with damping (shock absorber) and spring properties which are time-invariant. Passive element can only store energy for some portion of a suspension cycle (springs) and dissipate energy (shock absorbers). No external energy is directly supplied, in this type of suspension system. In Semi-active suspension system damping elements and spring is used, for which properties can be changed by an external control. An external power or signals are supplied to these systems to change the properties. In Fully-active suspensions system, actuators are used to generate the desired forces in the suspension. These actuators are normally hydraulic cylinders, which require external power to operate the system. The next subsection will present the various types of suspension system.

1.2 Types of suspension system

Now a days following type of suspension system are extensively used in automobile industries.

1.2.1 Passive suspension system

A passive suspension system generally consists of the components (i.e. dampers and springs) which are fixed. The characteristics of passive suspension are calculated and determined by the designer according to the design goals and application of the suspension. Basically passive suspension design is a compromise between vehicle handling and ride comfort as shown in Figure 1.1

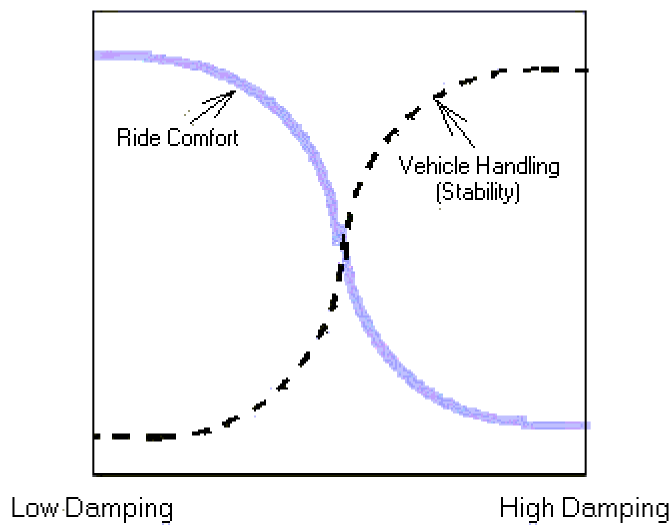


Figure 1.1: Damping compromise for passive dampers

A good vehicle handling can be achieved by a heavily damped suspension, but as a result vehicle body will be transferred much of the road input. When vehicle is moving on rough road at low speed or in a straight line at high speed, then this will be termed as a harsh ride. The vehicle owner may find the harsh ride unpleasant, or it can also damage cargo. Whereas, more comfortable ride can be achieved by a lightly damped suspension, but this will significantly reduce the stability of the vehicle in lane change maneuvers, turns, or in negotiating exit ramp. A good design of a passive suspension generally optimize ride and stability to some extent, but it cannot eliminate this compromise.

1.2.2 Active suspension system

In an active suspension, both the passive damper and spring or the passive damper is replaced with a force actuator, as shown in Figure 1.2.

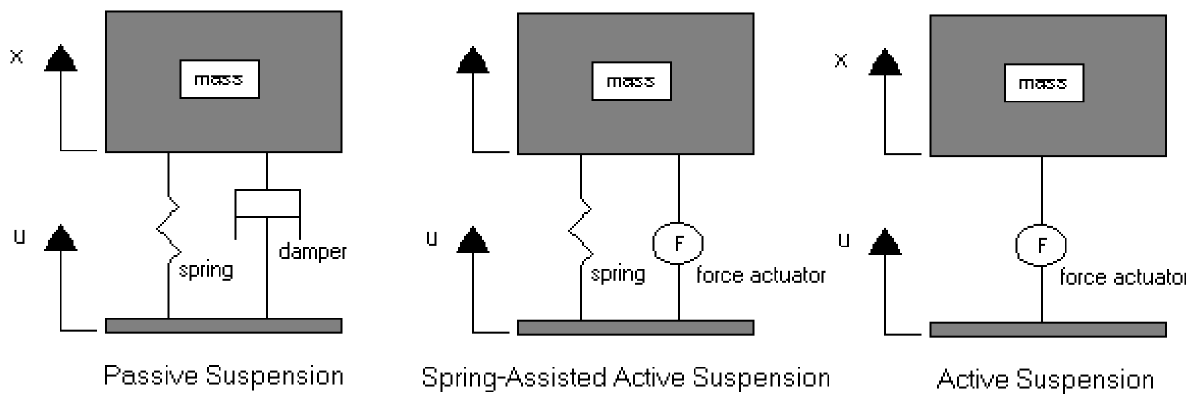


Figure 1.2: Passive and active suspensions

Unlike a passive damper, which can only dissipate energy, the force actuator is able to both add and dissipate energy from the system. The force actuator can apply force independent of velocity across the suspension or the relative displacement in an active suspension system. With the correct control strategy, compromise between ride comfort and vehicle stability as compared to a passive system will have better results, as shown in Figure 1.3 for a quarter-car model.

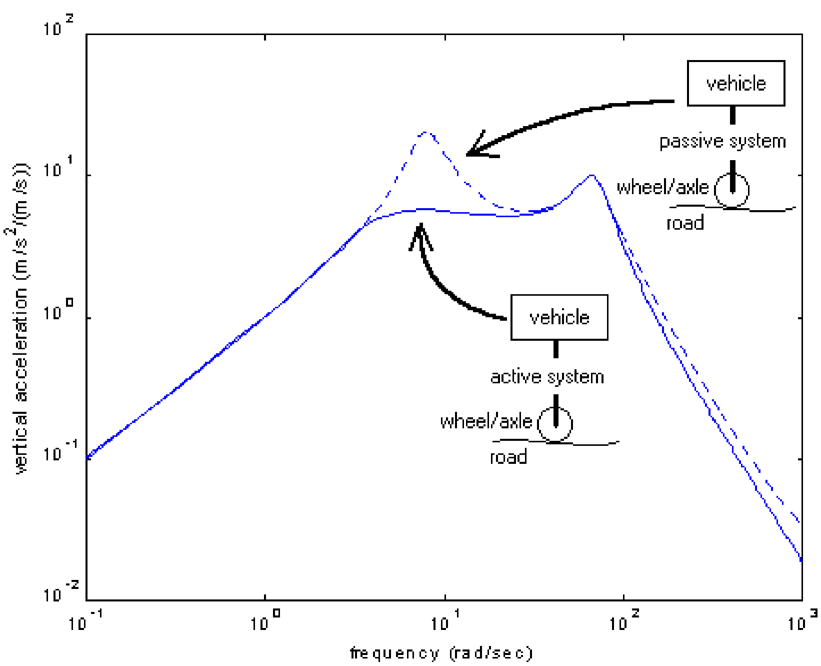


Figure 1.3: Comparison of passive and active suspension

A quarter-car model shown in Figure 1.4 has two-degree-of-freedom that emulates the vehicle body and axle dynamics at single time (i.e., one-quarter of a car).

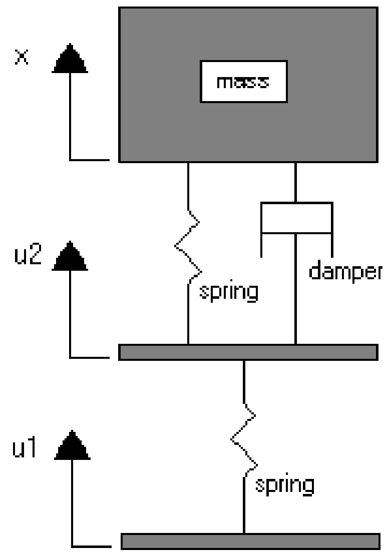


Figure 1.4: A Quarter car model.

A quarter car model is used to investigate an active suspension system to find the possible gain in performance, where road input was modeled as white noise input velocity. However, an active suspension can reduce the RMS (root mean square) acceleration of the sprung mass by 20%, within practical design limitations. This suspension configuration exhibited approximately the same level of suspension travel and wheel-hop damping ratio as lightly damped soft passive suspension. Furthermore, similar simulations and analysis were performed for half car model, while the active suspensions could reduce the RMS value of the sprung mass acceleration by 15%.

Active suspension systems have the added advantage of controlling the attitude of a vehicle. They can reduce the braking effect, which can causes a vehicle to nose-dive, or acceleration, which causes a vehicle to skid and also reduce the vehicle roll during cornering maneuvers. Active suspension systems, are capable to improve the ride and roll stability of the road vehicle. The force actuators necessary in an active suspension system typically have large power requirements (typically 4-5 hp). The power requirements decrease the overall performance of the vehicle and have unacceptable failure modes, which cannot be acceptable.

1.2.3 Semi-active suspension system

In early 1970's active suspension systems was first proposed. In this type of system, damper is replaced by a controllable damper, but the conventional spring element is retained as shown in Figure 1.5.

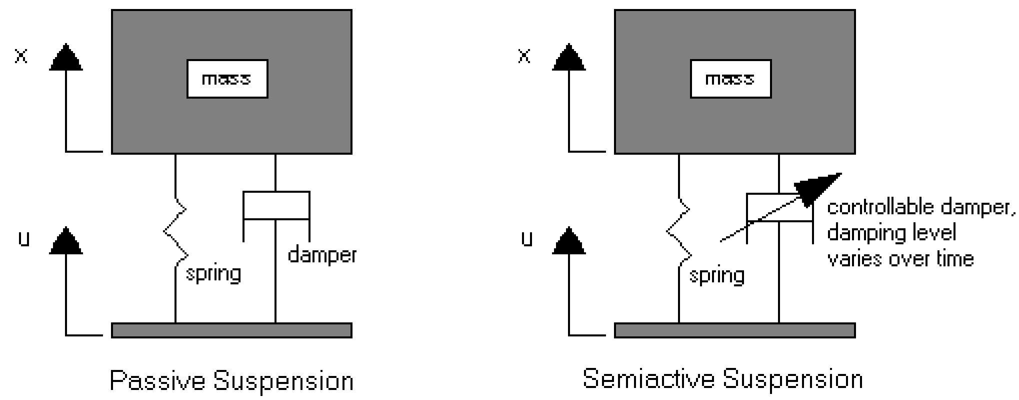


Figure 1.5: Passive and semi active suspensions

Whereas an active suspension system requires an external energy source to power an actuator which controls the vehicle, a semi active system only adjust the damping levels by using external power, which operates a set of sensors and an embedded controller. A Controller helps in determining the level of damping based on a control strategy, which automatically adjusts the damper to achieve the required damping.

One of the most common semi active control policy is skyhook control which adjusts the damping level to emulate the effect of a damper which is connected from the vehicle to a stationary ground as shown in Figure 1.6

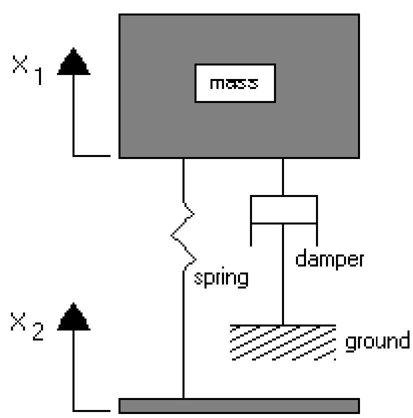


Figure 1.6: Quarter car model with skyhook damper

It has been shown that a continuously variable semi-active suspension system is able to achieve performance comparable to that of a fully active system. It is also possible to develop a control policy in which the damper is not just switched between a high and low state, but it also have an infinite number of positions between high and low state. This system is called a continuously variable semi active system. The ranges of damping values used in these two systems are illustrated in Figure 1.7.

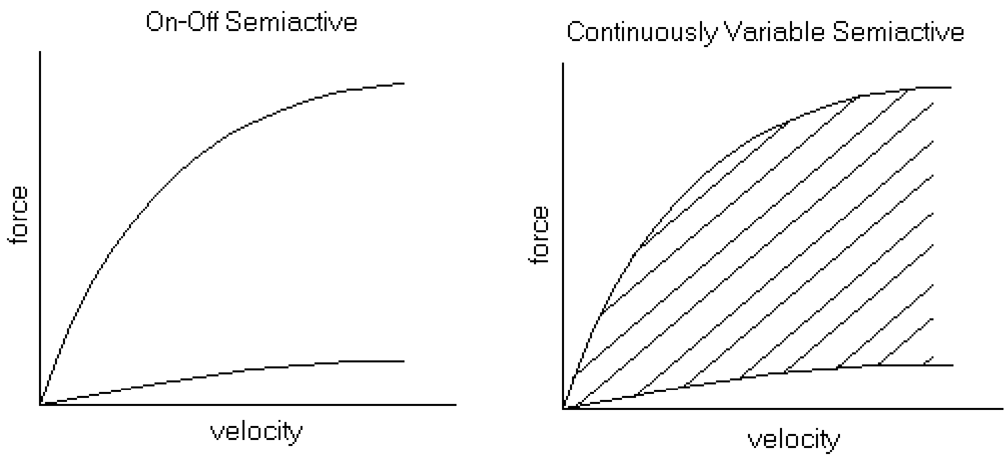


Figure 1.7: Range of damping values

Further the performance of an on-off semi active suspension system would be very close to the performance of a continuously variable semi active system. In the case that the controllable damper necessary in a semi active suspension fails, it will simply revert to a conventional damper. Semi active systems have a less dangerous failure mode, less complex, mechanical failure, and less power requirements are much less as compared to active systems.

1.3 Hydraulic actuators

An actuator is the mechanism by which an agent acts upon an environment. The electro hydraulic actuator comprises a servo-valve and a hydraulic cylinder as shown in Figure 1.8. The supply pressure and the return pressure are denoted by P_s and P_r respectively. Spool valve displacement Z_{sp}. The fluid pressures in the upper and lower cylinder chambers of the actuator are P_u and P_l, respectively. It is noted that P_r is approximately zero. A servo valve is to control the flow of hydraulic fluid to each suspension strut assembly. The servo valve controls flow rate, either directly proportional to a control signal or proportional to the difference between the downstream pressure and the control signal. The servo valve system is formulated as:

$$\dot{Z}sp = \frac{Z - Zsp}{\tau} \qquad (1.1)$$

where Z_{sp} is the valve current , and τ is the mechanical time constant of the servo valve system.

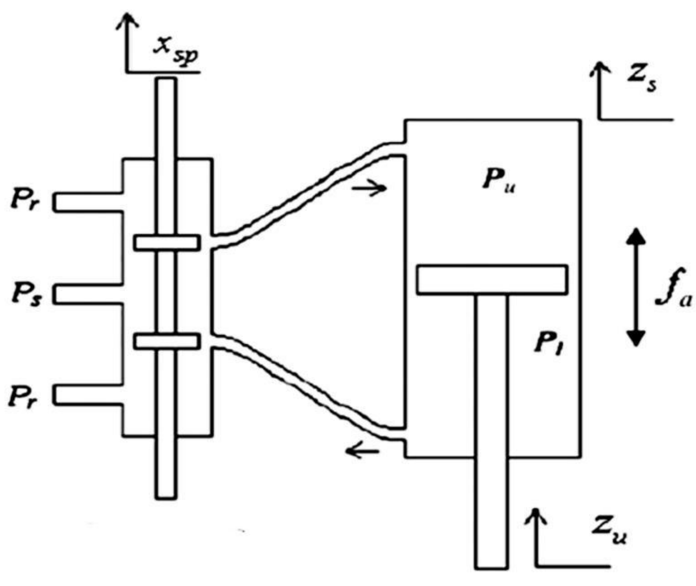


Figure 1.8: The electro hydraulic actuator system

The dynamic equation of the hydraulic actuator is given by

$$\dot{f}_a = -\alpha A_P^2\left(\dot{Z}_S - \dot{Z}_u\right) - \beta f_a + \gamma A_p X_{SP}\sqrt{P_S - \mathrm{sgn}\left(\frac{X_{SP}\, f_a}{A_p}\right)} \qquad (1.2)$$

Where A_p is the piston area, $\alpha = 4\frac{4\beta_c}{v}$, $\beta = \alpha C_{tp}$, $\gamma = \alpha C_d \omega\sqrt{\frac{1}{\rho}}$, β_c bulk modulus of hydraulic fluid, V_t is the total volume of actuator cylinder chamber, C_{tp} is the leakage coefficient, C_d is the discharge coefficient, w is the spool valve area gradient and ρ is the hydraulic fluid density.

1.4 Control system

A control system is set of devices to command, manage, regulate or direct the behavior of systems or other devices. The two common types of control systems, with many combinations and variations are sequential or logic controls, and linear or feedback controls. Also there is fuzzy logic, which combines the utility of linear control with some of the design simplicity of logic. Some systems or devices are not controllable. The term "control system" is applied to the essentially manual controls which allow operator to for example, open or close a hydraulic press, where logic requires that it should not be operated unless safety guards are in appropriate place. To perform such task, an automatic sequential control system will trigger a series of mechanical actuators in correct sequence. For example, various pneumatic and electric transducers may perform fold and glue a cardboard box operation, fill it with required product and after that seal it in with an automatic packaging machine.

In case of linear feedback systems, control loop, including actuators, control algorithms and sensors, are arranged in such a fashion to regulate a variable at a reference value or set point. For example the fuel supply may be increased to a furnace when temperature drops from desired temperature. In such cases, Proportional Integral Derivative (PID) controllers are commonly and effectively used. Control systems, which include some sensing of the results and are trying to achieve the desired result make use of feedback system. The open-loop control systems run only in pre-arranged ways and does not directly make use of feedback system.

1.5 Literature review

In this chapter the background of various investigation and the literature review are briefly presented. There are various studies, which are contributed by many researchers regarding the dynamic analysis, however very few researches are carried out for analyzing a dynamic behavior of road car, which is further validated through experimental analysis through OROS system, which justifies the project proposal.

1.5.1 Vehicle modeling

The dynamic analysis of vehicle requires a mathematical model of the car (quarter/half/full model), suspension and the road excitation. There are various researchers who have studied about quarter/half/full car model.

Optimum design of road-friendly vehicle suspension systems subjected to rough pavement surfaces has been proposed by Lu Sun[1] . Further this work presented an optimum concept to design road-friendly vehicles with the recognition of pavement loads as a primary objective function of vehicle suspension design. S. J. Chikhale et al.[2] presented comparative analysis of vehicle suspension system in MATLAB-SIMULINK and MSc- ADAMS with the help of a quarter car model, which prepared a quarter car model of vehicle suspension system in MATLAB-SIMULINK and MSc-ADAMS. Vibration analysis was done by giving step input. Li Xueying et al. [3] presented the study on accurate modeling of suspension based on ADAMS. The torque vibration derived from in-wheel-motor transmits to body frame through suspension system without the absorption of mechanical transmission parts, which influenced the quality of the vehicle and also aimed to build an accurate suspension system model to analyze the vibration transmission property.

Yogesh Sanjay Pathare et al. [4] presented design and development of quarter car suspension test rig model and its Simulation. He developed a simplified and cost effective setup for testing suspension system. A quarter car setup was made, which was reduced from a full car model, to reduce the complications and cost of development in its design and manufacturing process. Hadi Adibi-asl et al. [5] presented bond graph modeling and simulation of full car model with active suspension. Results showed a significant reduction in pitch and bounce acceleration as well as some improvement in roll acceleration of a body for both random and deterministic road profiles. Simulations were performed using commercial software which allows hybrid bond graph and block diagram models. A control matrix elements are chosen to reduce the sprung mass acceleration (roll, pitch, and bounce) and consequently improve road holding (less variation in tire deflection) and ride comfort of passengers. Ashraf Emam. et al. [6] A semi-active suspension system was designed for quarter vehicle model using fuzzy logic control (FLC). Performance of a fuzzy logic controller system designed to control a semi-active suspension system for a quarter vehicle model was studied. All simulations were performed using single bump road profile. The control performance criteria were evaluated in the time domains. The designed fuzzy logic controllers are simple and suitable for real time implementation. Simulation results demonstrated the ability of active suspension system to reduce the level of body acceleration, tire deflect ion and suspension travel.

Luis Silva et al. [8] had discussed bond graph based fault diagnosis of 4w-vehicles suspension systems. He discussed the model-based ARR technique, implemented on diagnostic bond graph, to the problem of detecting and isolating faults in vehicle suspensions. The main contribution was the proposition of simplified diagnostic bond graph that, allow solving a FDI problem on a reduced subsystem decoupled from the wheel dynamics. The simulation results presented illustrate the method ability of monitoring and isolating all the possible suspension faults considered. Faraz Ahmed Ansari et al [10] presented modeling, analysis and control of active suspension system using sliding mode control and disturbance observer. An active suspension control was constructed for a quarter car model subject to excitation from a road profile using an improved sliding mode control with observer design. Sliding mode is chosen as control strategy, and road profile was estimated by using observer design. To achieve the desired road handling, ride comfort and to solve the uncertainties, a sliding mode control technique was presented. A nonlinear surface was used to ensure fast convergence of vehicle's vertical velocity and to changes the system's damping. Sliding surface selection effect in the proposed controller was also presented. Bart L. J. Gysen et al. [11] presented Active Electromagnetic Suspension System to Improve Vehicle Dynamics. He offers motivations for an electromagnetic active suspension system that provides both additional stability and maneuverability by performing active roll and pitch control during cornering and braking, as well as eliminating road irregularities, hence increasing both vehicle and passenger safety and drive comfort. He compared Various technologies with the proposed electromagnetic suspension system that used a tubular permanent-magnet actuator (TPMA) with a passive spring.

In another study Padraig Dowds et al. [12] presented modeling and control of a suspension system for vehicle applications. He discussed the modeling of active and passive suspension system in car, and design of appropriate feedback controller for the active suspension system. The models were investigated using a quarter car model and a full car model approach. Modeling of passive and active suspension systems was done, for both quarter and full car model. It was shown that active suspension system facilitates improved regulator response when compared to passive suspension system. Controlling element of an active suspension system is generally based on actuator. The main practical difficulty in implementing active suspension is power consumption of the actuator. Ayman A. Aly et al. [13] presented a review on vehicle suspension systems control. In order to improve comfort and handling performance, semi-active and active systems are being developed, instead of a conventional static spring and damper system. An active suspension system has been proposed to improve the ride comfort. A quarter-car 2 degree-of-freedom (DOF) system was designed and constructed on the basis of the concept of a four-wheel independent suspension to simulate the actions of an active vehicle suspension system. M. D. Emami et al. [14] presented modeling and simulation of active hydro-pneumatic suspension system through bond graph. His aim was to use BG method in modeling of hydro-pneumatic suspension. A hydro pneumatic spring consists of two fluids acting upon each other, usually gas over oil. This system consists of mechanical, hydraulic, thermo-fluid and control subsystems. Therefore, it was a multi-domain system and BG is a good candidate to model it.

Cătălin et al [16] presented a comparative analysis between the vehicles passive and active suspensions. He presented comparative analysis between the passive and active suspension systems. A study was performed for a half-car model, which corresponds to the guiding suspension system of a rear axle of motor vehicle. An active suspension system was obtained by introducing a force actuator in parallel to passive suspension system. The goal was to minimize the effect of the road disturbances on vehicle. The passive and active suspensions were analyzed in the passing over bumps dynamic regime. The response of the active suspension was compared with the passive suspension, important improvements in the dynamic behavior (in terms of comfort and stability) being observed for the active suspension. Gilberto Gonzalez-A et al. [17] discussed steady state of passive and active suspensions in the physical domain. Steady state response of bond graphs representing passive and active suspension was presented. S. Segla et al. [18] presented optimization and comparison of passive, semi-active and active suspension systems of vehicle. He discussed that not only active, but also semi-active suspension system is able to improve the ride comfort significantly compared with the passive suspension systems. Using the vibration absorber (split axle) and additional dampers between the axle and engine, had improved the ride comfort slightly in this paper. Germán Filippini et al. [19] presented vehicle dynamics simulation using bond graph construction of a four-wheel, nonlinear vehicle dynamic bond graph model and its implementation in the modeling and simulation environment was presented.

Nonlinear effects arising from coupling of vertical, longitudinal and lateral vehicle dynamics and geometric nonlinearities coming from the suspension system were also taken into account. T. Ram Mohan Rao et al. [20] presented analysis of semi active and passive controlled suspension systems for ride comfort in an omnibus passing over a speed bump. Control performance of a 3 DOF quarter car semi active suspension systems was investigated using Matlab/Simulink, model. Ben Creed et al. [21] presented development of full car vehicle dynamics model for use in the design of an active suspension control system. The creation of a full car model for a standard road going vehicle was discussed. This model was equipped with suspension force actuators which allow future development of an active suspension control system to improve the ride comfort. An active suspension is concerned with controlling the vertical movement in response to the road inputs for each wheel. This was accomplished by actively applying vertical forces in the suspension to counteract some of the effects of road surface. However, these systems can be used to minimize vertical accelerations, vehicle body roll experienced by the passengers and improving overall handling of vehicle. Bharat Raj Singh et al. [22] presented dynamic modeling and application using bond graph. Technique for presenting quantitative and systematic modeling and simulation using Bond Graph for the vehicle dynamic system was discussed. For accurate and efficient simulation towards the design of the vehicle dynamic system, portioning algorithm can be employed to the existing automated modeling techniques.

1.5.2 Modeling

Modeling is a process of producing model, whereas a model is representation of the working and construction of some system. Purpose of a model is to enable the analyst to predict the effect of changes to the system and it should be a close approximation to the real system and incorporate most of its salient features. Model should not be so complex that it is impossible to understand and experiment on it. A model is said to be good if it is a judicious tradeoff between realism and simplicity. Model validity is an important issue in modeling. Validation techniques of model include simulating the model under known input conditions and comparing the output with system output or some experimental data available.

Most of the modern mechanical system form a part of multidisciplinary system and are closely coupled with the magnetic, hydraulic, electrical or other kind of energy domains. Hierarchical approach is advantageous in modeling of an integrated dynamic system. Systematic approach minimizes the modeling mistakes by breaking the system model into small components and subsystems, which are manageable in size and complexity. Same components can be used repeatedly without building models of the same part again. For example, an axle component in the vehicle dynamics subsystem was built once and can be used for both rear and front axle. With hierarchical modeling, models of same component with different complexity can easily be interchanged without changing the remaining structure of the model. The simulation process steps are shown in figure 1.9

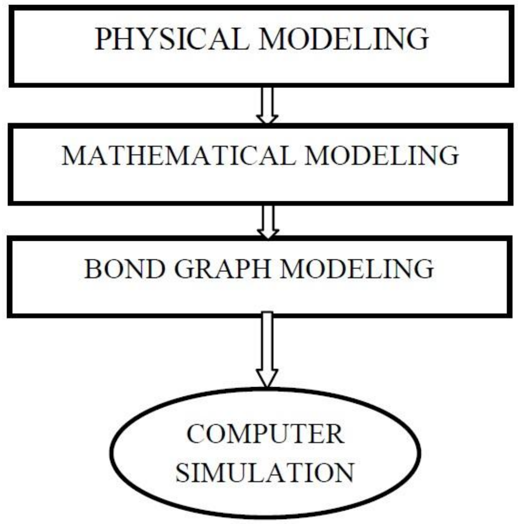


Figure 1.9: Simulation process steps

The various approaches adopted in vehicle modeling are stated as follows:

a) Ignoring body flexibility using a lumped mass model.

b) Modeling the frame as a regular free beam and calculating, measuring or estimating modal masses and stiffness.

c) Modeling an entire vehicle using the finite element method technique.

d) Modeling the dynamic behavior of vehicle using bond graph technique.

1.6 Contribution of the research work

The contribution of this work is to develop a precise computational model of a road car, which is able to analyze a dynamic behavior at different uneven road conditions. Moreover, dynamic behavior of car is analyzed while passing through bumps and recording the vertical acceleration of a quarter car model using OROS setup. These obtained vertical acceleration are used to validate the simulation result given by bond graph modeling.

1.7 Significance of bond graph modeling

In the 1950s, H.M. Paynter of MIT worked on various interdisciplinary engineering projects including digital and analog computing, hydroelectric plants, nonlinear dynamics and control. He observed that the similar forms of equations were generated by the dynamic systems in a wide variety of domain (for example mechanical, fluid and electrical). He incorporated the notion of an energy port into his methodology and thus bond graph was invented in 1959. Later, bond graph theory has been further developed by many researchers.

Bond graph method is a graphical approach of modeling, in which component energy parts are connected by bonds which supply the transfer of energy between the systems components.

The bond graphs language aspires to express general class of physical systems through power interactions. Factors of power, i.e., Flow and Effort, have different interpretations in different physical domains. Hence, the power can be used as the generalized co-ordinate, to model coupled system residing in several energy domains in bond graph. One only needs to recognize four groups of basic symbols in bond graphs, i.e., the three basic one port passive elements inertance (I), capacitance (C), and resistance (R); two basic active elements source of effort (SE), and source of flow (SF); two basic two port elements gyrator (GY), and transformer (TF); and two basic junctions i.e., constant flow junction (1) and constant effort junction (0). The basic variables are flow (f), effort (e), the time integral of flow (Q) and time integral of effort (P). A physical system can be represented by lines and symbols, identifying the power flow paths by this approach. The lumped parameter elements of inertance, capacitance and resistance are interconnected in the energy conserving way by junctions and bonds resulting in a network structure. The deviation of system equation is so systematic that it can be algorithmzed, from the pictorial representation of bond graph. Whole procedure of simulation and modeling of the system can be performed with some of the existing software e.g., **Camp- G, ENPORT, 20sim, SYMBOL-shakti, , COSMO** etc.

Main advantages of bond graph techniques are-

- Bond graphs can be used to describe simple linear and non linear systems.
- Bond graphs provide a useful notation for the purpose of modeling physical systems.
- Systems with diverse energy domain are treated in a unified manner.
- Easiest way to communicate the description of energy flows in dynamic systems.
- By using power conservation properties of bond graphs one may need to constraint velocities only and the forces will automatically be balanced.
- Graphical representation document complex models very clearly and unambiguously.

- Model subsystems independently.

The present dissertation work explores the ability of the bond graphs to obtain dynamic behavior and modeling of the vehicles based on the physical paradigm of the system. Bond graph technique generally offers flexibility in modeling and formulation of system equations. A very large system may also be modeled in a modular form by creating sub -system models and then joining them together at their interaction port to create an integrated system model. mode is may be easily modified making it a powerful tool for system synthesis and consolidation of innovative ideas. Bond graph equations normally use generalized displacement and generalized moment as state variables. The bond graph modeling, their simulation and animation is performed using SYMBOLS Shakti®, a bond graph modeling software.

1.8 Objectives

This research objective is:

- To create the dynamic model of a cars by using bond graph technique.
- To create it's mathematical modeling.
- Simulation of the Bond graph model for variable parameters to obtain behavior of car
- Perform experimental work Using OROS on Car (XUV 500) suspension system
- Validating results of Bond Graph and Experimental Work.
- Modeling of full car model on bond graph and simulating on various speed.

1.9 Organization of Thesis

The chapters of the thesis are arranged in the following manner. Chapter 1 discusses about the background of this project along with a summary of previous works. The primary goal of this chapter is to provide the reader with a rough idea about this thesis with some literature reviews. Chapter 2 presents the mathematical modeling of the various car models (quarter/full). Chapter 3 deals with the introduction of bond graph. Chapter 4 presents the bond graph modeling of a car. Chapter 5 presents the simulation study whereas Chapter 6 deals with the experimental study. The Chapter 7 provides results, and finally concludes the thesis and also suggests some scope for future work.

CHAPTER 2
MATHEMATICAL MODELLING

2.1 Introduction

A mathematical model of a dynamic system is defined as a set of equations that represents dynamics of the system accurately or at least, fairly well. A mathematical model is not unique to a given mechanical system. A system may be represented in many different ways and therefore may have many mathematical models, depending on one's perspective.

The dynamics of a mechanical system can be described in terms of differential equations. Such differential equations may be obtained by using physical laws governing particular system, for example, Newton's laws are used in case of a mechanical system. Mathematical models may assume many different forms. Depending on the particular system and the particular circumstances, one mathematical model may be better suited than other models. Once a mathematical model of a system is obtained, various analytical and computer tools can be used for analysis and synthesis purposes.

This chapter presents the mathematical modeling of the car structure. Vertical translation and rotation about X- and Y-axis (roll and pitch) have included for the car body. The mathematical model of the system can be derived from Newton's second law of motion. The basic form of the equation is presented as:

$$M\{\ddot{X}\}+C\{\dot{X}\}+ K\{X\}=\{F\} \tag{2.1}$$

Here, mass is represented by the sprung mass (M), the damping ratio of the damper(C), and the stiffness of the spring (K), F represent the dynamic force.

2.2 Mathematical modeling of various models of car.

The main objective of suspension systems is to reduce the sprung mass (vehicle body) to road disturbances motions. The conventional vehicle suspension systems achieve this through passive means i.e. by using dampers and springs.

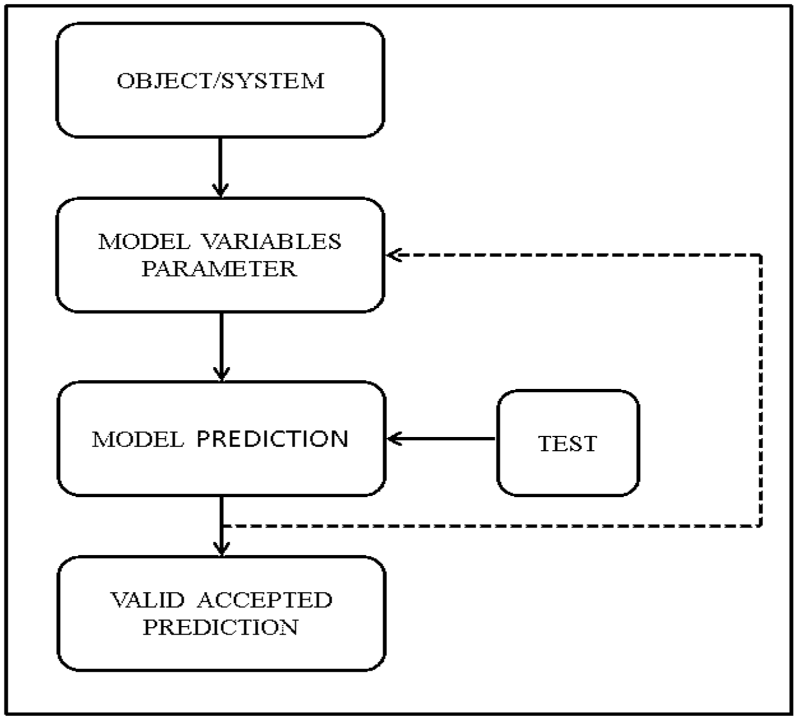


Figure 2.1: Principle of mathematical modeling

When designing vehicle suspension system, the dual objective is to maximize the tyre-to-road contact for handling and safety, and to minimize the vertical forces transmitted to the passenger. While the traditional passive suspension systems can negotiate this trade-off effectively, while active suspension systems have potential to improve both handling performance and ride quality, with the important benefits of better braking and cornering because of reduced weight transfer. This improvement is conditional on the use of feedback control of the actuators in the active suspension system.

2.2.1 Quarter car model for active and passive suspension system

A quarter car model is set up by using interconnections of springs, dampers and masses. Figure 2.2 and 2.3 shows a passive and active quarter car suspension model respectively. In Figure 2.2 and 2.3, M_s is the car body mass, M_{us} is the unsprung mass of the axle and wheel assembly, the spring constant in the suspension system is represented by k_s, b_s is dashpot constant (representing the shock absorber) and the spring constant of the tire is k_t; the road input force is represented by r, x_s represents the force which is acting on the mass M_s, f represents the control element as shown in Figure 2.3 and x_{us} represents the force acting on the mass M_{us}.

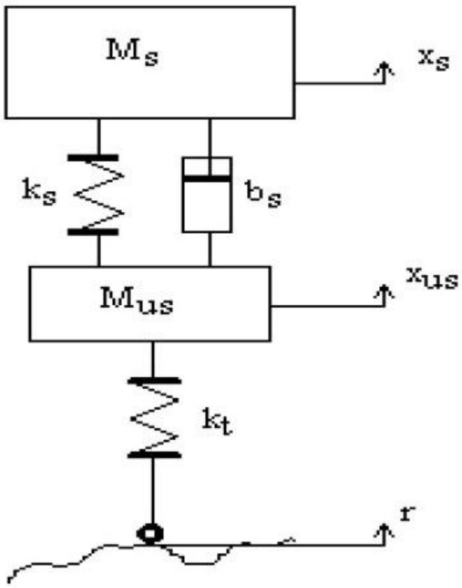


Figure 2.2: Quarter car model – passive

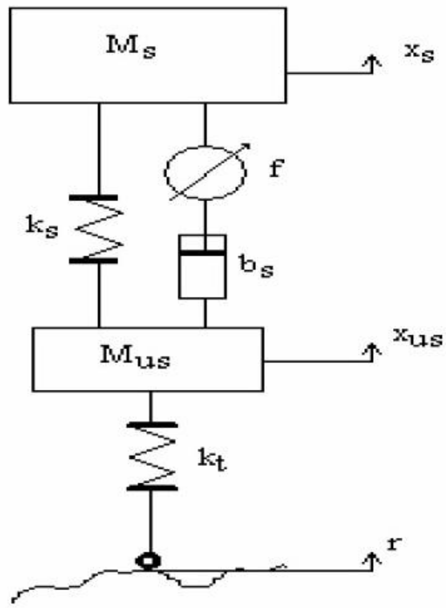


Figure 2.3: Quarter car model – active

The state equations for the passive quarter car model are:

$$M_s\ddot{x}_s = -k_s(x_s - x_{us}) - b_s(\dot{x}_s - \dot{x}_{us}) \tag{2.1}$$

$$M_{us}\ddot{x}_{us} = -k_s(x_s - x_{us}) + b_s(\ddot{x}_s - \dot{x}_{us}) - k_t(x_{us} - r) \tag{2.2}$$

These equations give the following transfer function relating x_s to r:

$$\frac{x_s}{r} = \frac{k_t(b_s s + k_s)}{M_s M_{us} s^4 + (M_s + M_{us})b_s s^3 + \left((M_s + M_{us})k_s + M_s k_t\right)s^2 + b_s k_t s + k_s k_t} \tag{2.3}$$

The state equations for the active quarter car model are:

$$M_s\ddot{x}_s = -k_s(x_s - x_{us}) - b_s(\dot{x}_s - f) \tag{2.4}$$

$$M_{us}\ddot{x}_{us} = -k_s(x_s - x_{us}) + b_s(f - \dot{x}_{us}) - k_t(x_{us} - r) \tag{2.5}$$

These equations give the following transfer function relating s x to r and f:

$$x_s = \frac{k_s k_t}{M_s M_{us} s^4 + (M_s + M_{us})b_s s^3 + \left((M_s + M_{us})k_s + b_s^2 + M_s k_t\right)s^2 + (2k_s + k_t)b_s s + k_s k_t} . r$$

$$+ \frac{M_{us} b_s s^2 + b_s^2 s + (2k_s + k_t)b_s}{M_s M_{us} s^4 + (M_s + M_{us})b_s s^3 + \left((M_s + M_{us})k_s + b_s^2 + M_s k_t\right)s^2 + (2k_s + k_t)b_s s + k_s k_t} . f \tag{2.6}$$

2.2.2 Full car model for passive suspension system

Figures 2.4 and 2.5 shows two full car passive suspension system models, with Figure 2.5 including the effects of the roll and pitch motions.

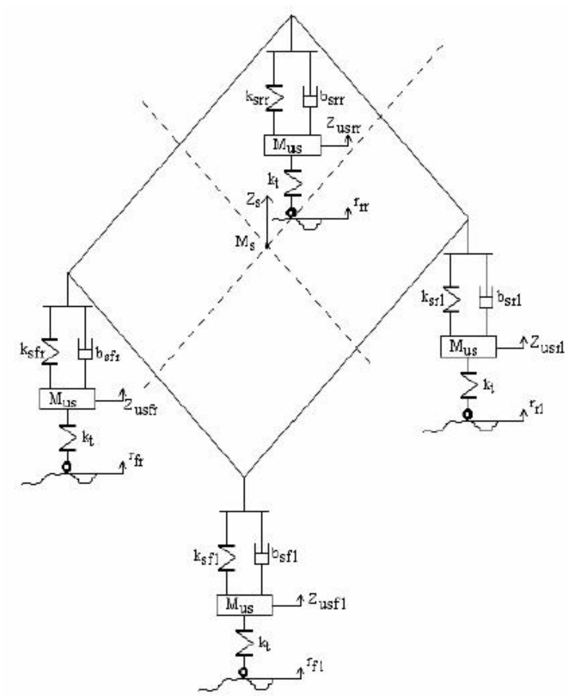


Figure 2.4: Full car model- passive

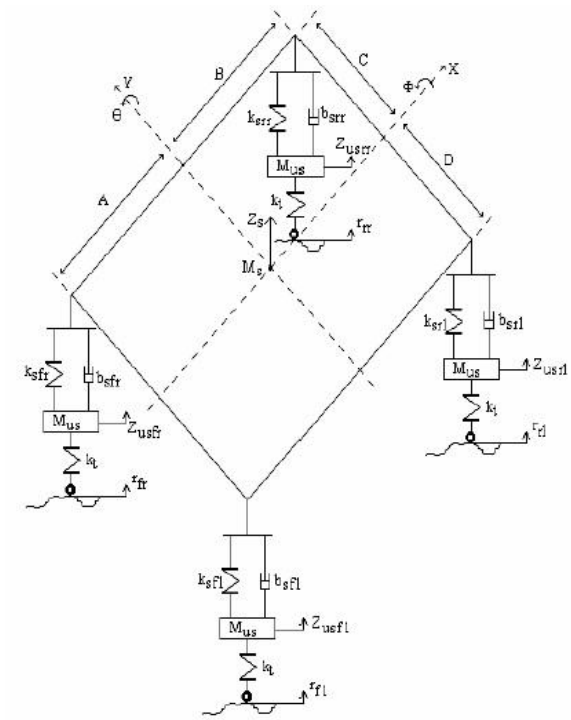


Figure 2.5: Full car model- passive with Pitch and roll

In Figure 2.4 and 2.5, the coefficient labels mirror those of Figures 2.2 and 2.3. For Figure 2.5, the pitch angle of the car is represented by θ and the roll angle of the car is represented by φ.

The state equations for Figure 2.4, may be written as

$$\dot{x}_2 = \frac{1}{M_s}[-(2k_{sf} + 2k_{sr})x_1 - (2b_{sf} + 2b_{sr})x_2 + k_{sf}x_3 + b_{sf}x_4$$

$$+k_{sf}x_5 + b_{sf}x_6 + k_{sr}x_7 + b_{sr}x_8 + k_{sr}x_9 + b_{sr}x_{10}] \qquad (2.7)$$

$$\dot{x}_4 = \frac{1}{M_{us}}[k_{sf}x_1 + b_{sf}x_2 - (k_{sf} + k_t)x_3 - b_{sf}x_4 + k_t r_{f1}] \qquad (2.8)$$

$$\dot{x}_6 = \frac{1}{M_{us}}[k_{sf}x_1 + b_{sf}x_2 - (k_{sf} + k_t)x_5 - b_{sf}x_6 + k_t r_{fr}] \qquad (2.9)$$

$$\dot{x}_8 = \frac{1}{M_{us}}[k_{sf}x_1 + b_{sr}x_2 - (k_{sr} + k_t)x_7 - b_{sr}x_8 + k_t r_{r1}] \qquad (2.10)$$

$$\dot{x}_{10} = \frac{1}{M_{us}}[k_{sf}x_1 + b_{sr}x_2 - (k_{sr} + k_t)x_9 - b_{sr}x_{10} + k_t r_{rr}] \tag{2.11}$$

With $\dot{x}_1 = x_2$, $\dot{x}_3 = x_4$, $\dot{x}_5 = x_6$, $\dot{x}_7 = x_8$ and $\dot{x}_9 = x_{10}$.

To develop these equations, $k_{sf1} = k_{sfr} = k_{sf}$, $k_{sr1} = k_{srr} = k_{sr}$, $b_{sf1} = b_{sfr} = b_{sf}$ and $b_{sr1} = b_{srr} = b_{sr}$. The state variables are assigned as follows; $x_1 = z_s$, $x_2 = \dot{z}_s$, $x_3 = z_{usf1}$, $x_4 = \dot{z}_{usf1}$, $x_5 = z_{usfr}$, $x_6 = \dot{z}_{usfr}$, $x_7 = z_{usrt}$, $x_8 = \dot{z}_{usrt}$, $x_9 = z_{usrr}$ and $x_{10} = \dot{z}_{usrr}$

2.2.3 Full car model for active suspension system.

Similarly, Figures 2.6 and 2.7 shows two full car active suspension system models, with Figure 2.7 including the effects of the roll and pitch motions. Coefficient labels mirror those of Figures 2.2 to 2.5.

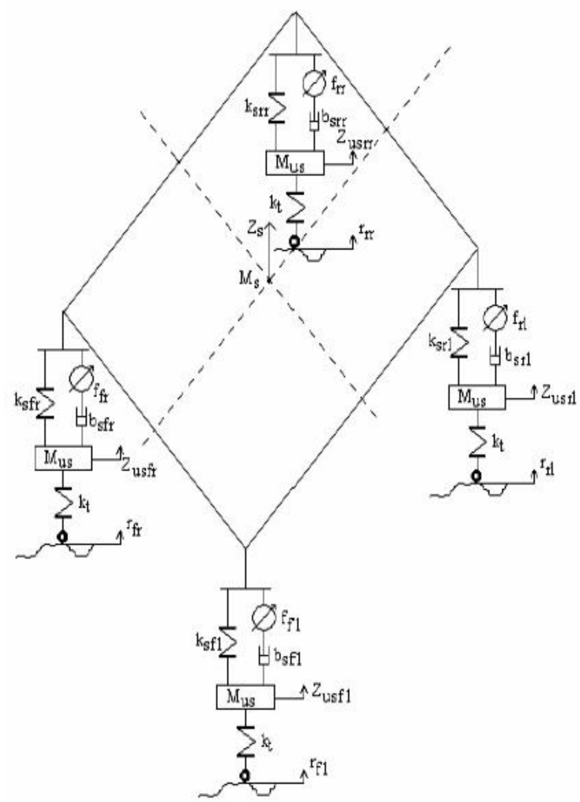


Figure 2.6: Full car model- active

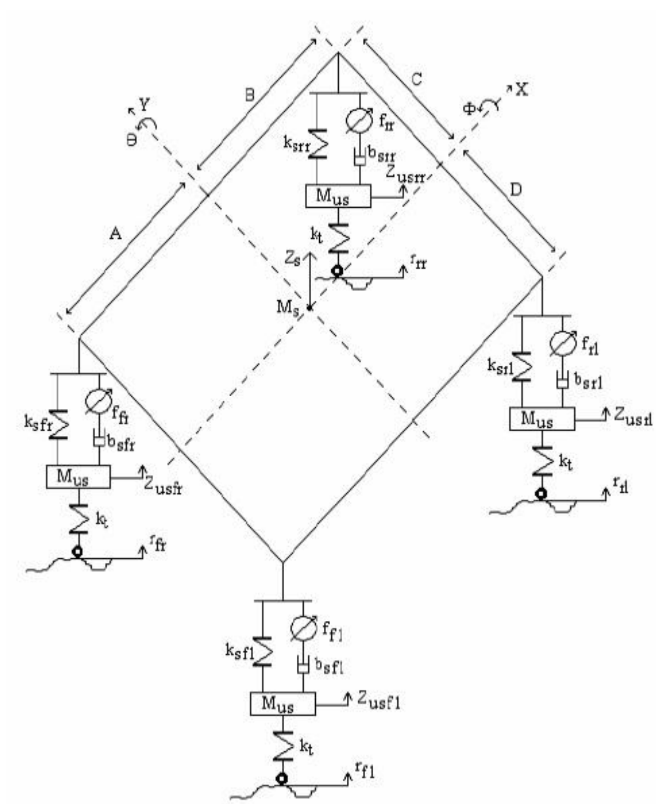


Figure 2.7: Full car model- active with roll and pitch

The state equations for Figure 2.6 may be deduced to be

$$\dot{x}_2 = \frac{1}{M_s}[-(2k_{sf} + 2k_{sr})x_1 - (2b_{sf} + 2b_{sr})x_2 + k_{sf}x_3 + k_{sf}x_5$$

$$+k_{sr}x_7 + k_{sr}x_9 + b_{sf}f_{f1} + b_{sf}f_{fr} + b_{sf}f_{r1} + b_{sr}f_{rr}] \qquad (2.12)$$

$$\dot{x}_4 = \frac{1}{M_{us}}[k_{sf}x_1 + b_{sf}f_{f1} - (k_{sf} + k_t)x_3 - b_{sf}x_4 + k_t r_{f1}] \qquad (2.13)$$

$$\dot{x}_6 = \frac{1}{M_{us}}[k_{sf}x_1 + b_{sf}f_{fr} - (k_{sf} + k_t)x_5 - b_{sf}x_6 + k_t r_{fr}] \qquad (2.14)$$

$$\dot{x}_8 = \frac{1}{M_{us}}[k_{sf}x_1 + b_{sr}f_{r1} - (k_{sr} + k_t)x_7 - b_{sr}x_8 + k_t r_{r1}] \qquad (2.15)$$

$$\dot{x}_{10} = \frac{1}{M_{us}}[k_{sf}x_1 + b_{sr}r_{rr} - (k_{sr} + k_t)x_9 - b_{sr}x_{10} + k_t r_{rr}] \qquad (2.16)$$

with $\dot{x}_1 = x_2$, $\dot{x}_3 = x_4$, $\dot{x}_5 = x_6$, $\dot{x}_7 = x_8$ and $\dot{x}_9 = x_{10}$. State equations are developed in a corresponding manner to that of Figure 2.4.

2.3 Summary of the Chapter

In this chapter, mathematical model of quarter car and full car have been presented for active and passive suspension system. The nomenclature presents all notations used in the mathematical model. Next chapter will present the modeling of a quarter/full car using bond graph techniques.

CHAPTER 3
EXPERIMENTAL STUDY

3.1 Introduction

In this work vibration analyses of suspension system had been done using Vibration Analyser OROS 36, which operates with NV Gate 8.30 version software, and record the signals of component in the form of acceleration, velocity and displacement.

Figure 3.1: Experimental equipment (OROS 36) for vibration analysis.

3.1.1 About Equipment:

OR36/OR38 is designed for high channel count capacity without comprising the analyzer geographies. All the channels are handled in real-time whatever the analysis mode: FFT, 1/3rd Octave, CPB or Synchronous order analysis. OR36 & OR38 keep these real-time capabilities up to 20 kHz. There are LCD screen controls on the OR36 and OR38 hardware that allow you to run, stop the analyzer, change the fan speed etc.

Basic procedures of NV Gate follow a simple sequence:

- Input connection to the plug-in analyzers.
- Format of Front-end and plug-in analyzers.
- Range of results to be displayed and/or saved.
- Result management (manager) and report generation.

NV Gate platform offer a comprehensive sets of tools for noise and vibration acquisition, recording and analysis.

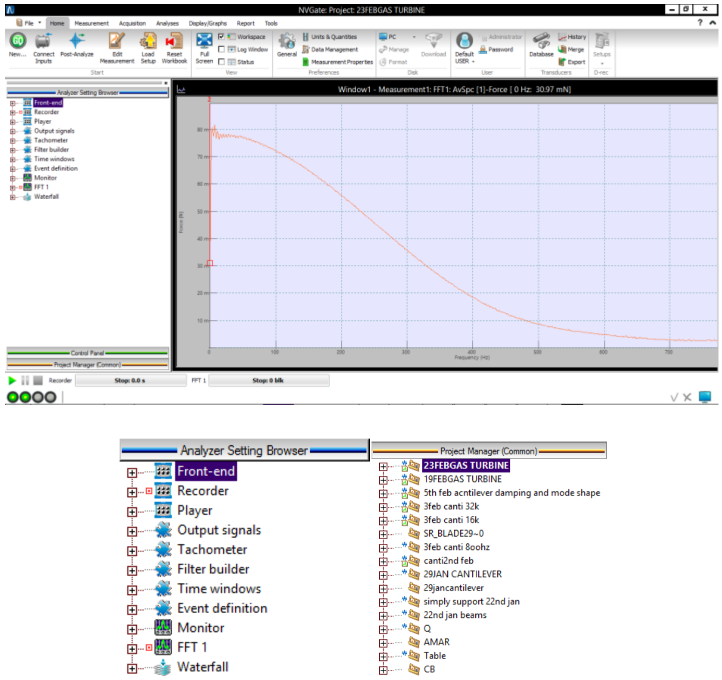
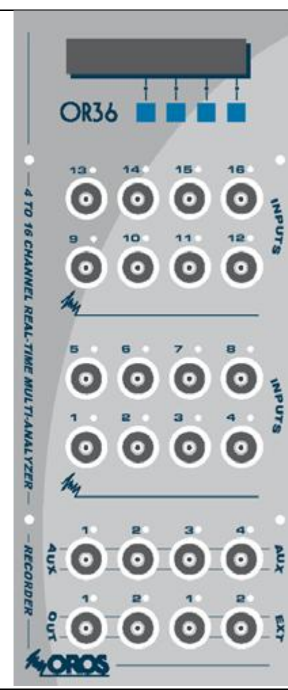


Figure3.2: Graphic user interface of NV Gate 8.30

3.1.2 Specifications

3.1.2.1 Description of OROS Hardware

Table 3.1: OROS Hardware specification

Inputs	1 to 16 Dynamic Inputs DC Inputs	
Out.	Generators 1 & 2	
Ext.	External Sync.1 & 2	
Aux.	Auxiliary connectors 1 to 4	

3.1.2.2 Transducers

Table 3.2: Accelerometer specification

Transducer Type	Acceleration Sensor	
Unit/ Magnitude	Acceleration (m/s^2)	
Identifier	PCB-78534	
Model	356A16	
Coupling	ICP	
Sensitivity	1×10^{-2} V/g or V/(m/s^2)	

Table 3.3: Hammer specification

Transducer Type	Force Sensor	
Unit/ Magnitude	Force (N)	
Identifier	PCB	
Model	SN-25679	
Coupling	ICP	
Sensitivity	2.25×10^{-3} V/N	

3.1.2.3 Integrated components of OROS

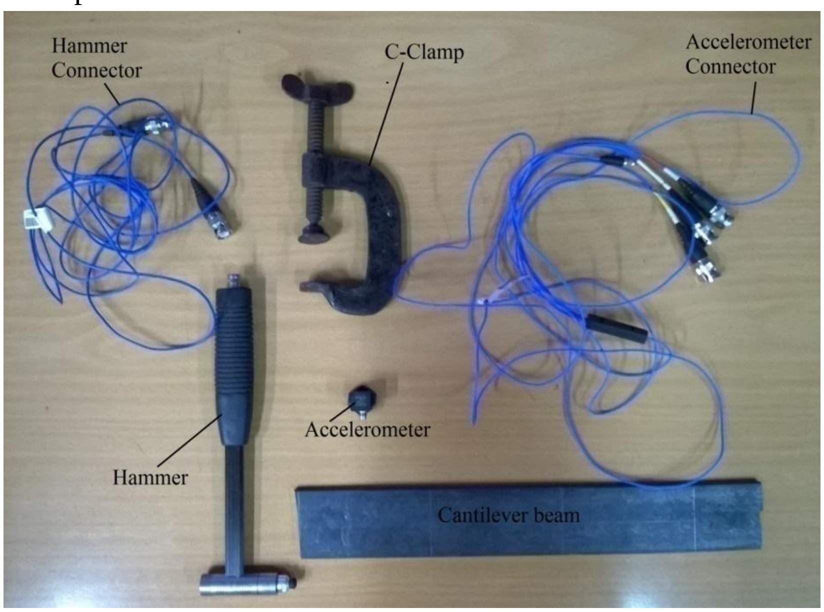


Figure3.3: Nomenclature of components

3.2 Dynamic analysis using OROS & NV Gate software:

Dynamic analysis is a process, where one may describe the structure in terms of its natural characteristics which are the frequency, damping and mode shapes i.e. its dynamic properties.
FRF based modal testing started in the early 1970's with the commercial availability of the digital FFT analyzer and has grown steadily in popularity since then.

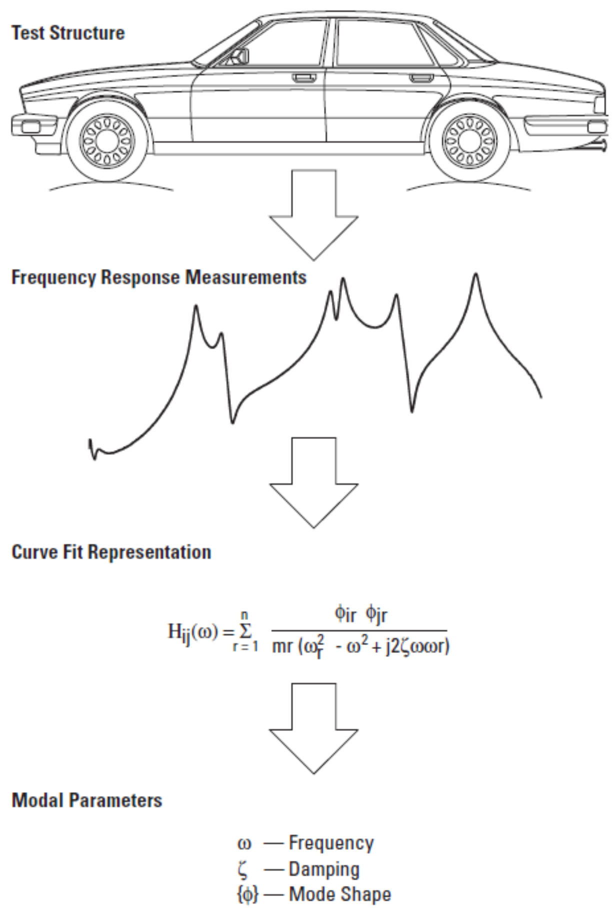


Figure 3.4: Phases of dynamic test

3.2.1 Operating deflection shape (ODS)

An operating deflection shape (ODS) is basically termed as any forced motion of two or extra points at a structure. Stipulating the motion of two or more points expresses a shape. Stated in a different way a shape is the movement of one point relative to all others. Motion is a vector quantity, which has both direction and position associated with it. Motion at a point in a direction is called a degree of freedom. All the experimental dynamic constraints is obtained from dignified ODS’s.

3.2.2 Single Input (or SIMO) Testing

The most mutual type of modal testing is completed with either a single static input or a single static output. A moving hammer impact test using a single static motion transducer is a mutual example of single reference testing. The single static output is called the allusion in this case. When a single static input is used, this is termed as SIMO (Single Input Multiple Output) analysis. In this case, the single static input is called the reference.

3.2.3 Multiple Input (or MIMO) Testing

When two or more static inputs are used, FRFs is considered between each input and multiple outputs then, FRFs as of multiple columns of the FRF matrix are found. This is termed as Multiple Reference or MIMO (Multiple Input Multiple Output) analysis. The inputs are the references in this case.

3.3 Experimental Procedure

Several experiments has been performed by taking account of various factors. These are discussed below.

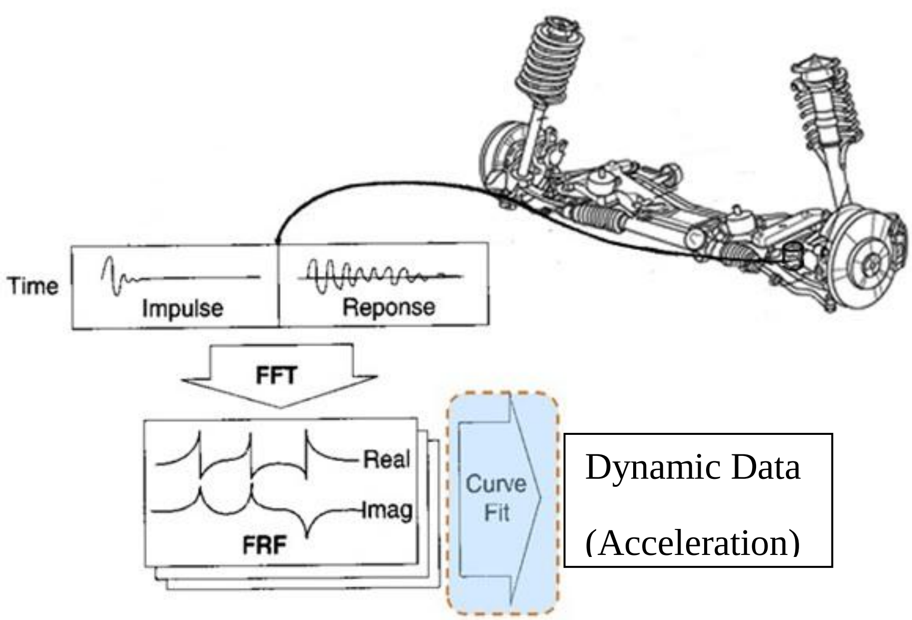


Figure 3.5: Experimental procedures

The following equipment is required to perform a dynamic test:

- Accelerometer to measure the response acceleration at fixed point & direction.
- A two or four channel FFT analyzer to compute FRFs.
- Post-processing modal software for identifying modal parameters and displaying the mode shapes in animation.

3.4 Experimental arrangement

The experimental results have been taken by fixing accelerometer in the suspension system of car. Here, the car used for experimental work is XUV 500 and the suspension system used is Mcpherson Suspension System. The experiment is performed while running a car on University road and passing the car on bumps to obtain the required acceleration.

3.4.1 Mcpherson suspension system

The McPherson suspension system is currently employed in the vast majority of medium and small sized cars. In its common configuration as shown in Figure 3.6, the suspension consists of a knuckle (K) or a strut (S) rigidly connected to the wheel support. The upper part of the strut is connected to the body (B) by means of a flexible union which is formed by an elastic element and a thrust ball bearing which allows the rotation of strut. The lower part of the suspension consists of a wishbone (W), which joins the knuckle to the body of a car. The union between the wishbone and the knuckle is made via a spherical joints, the wishbone is connected to the body by means of two bushings (R1 and R2), which allow the relative rotation between both the elements. The tie rod is connected to the knuckle or the damper by means of a spherical joint in order to transmit the turn of the steering wheel to the wheel as shown in Figure 3.6.

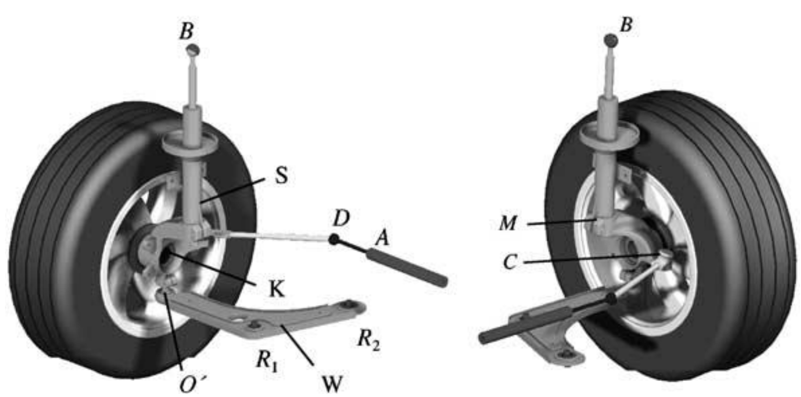


Figure 3.6: Front and rear view of the characteristic parts of the left front wheel.

3.4.2 Experimental set up

Here the accelerometer is placed in on the suspension system as shown in Figure 3.7 to measure the vibration on bumps at different speed conditions

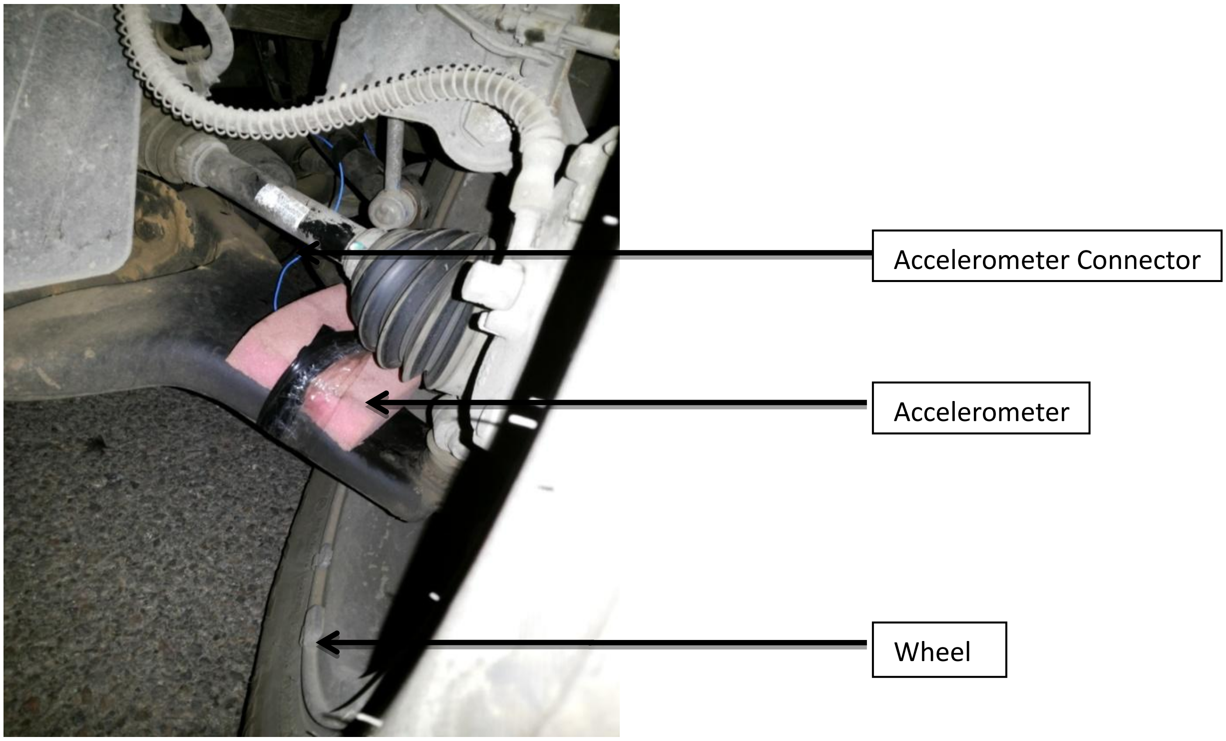


Figure 3.7: Experimental setup

The OROS Set up is placed in car and the accelerometer is placed on suspension system and fixed with safety form and cello tape. This accelerometer measure the vertical vibration which is produced on bumps at different speed condition.

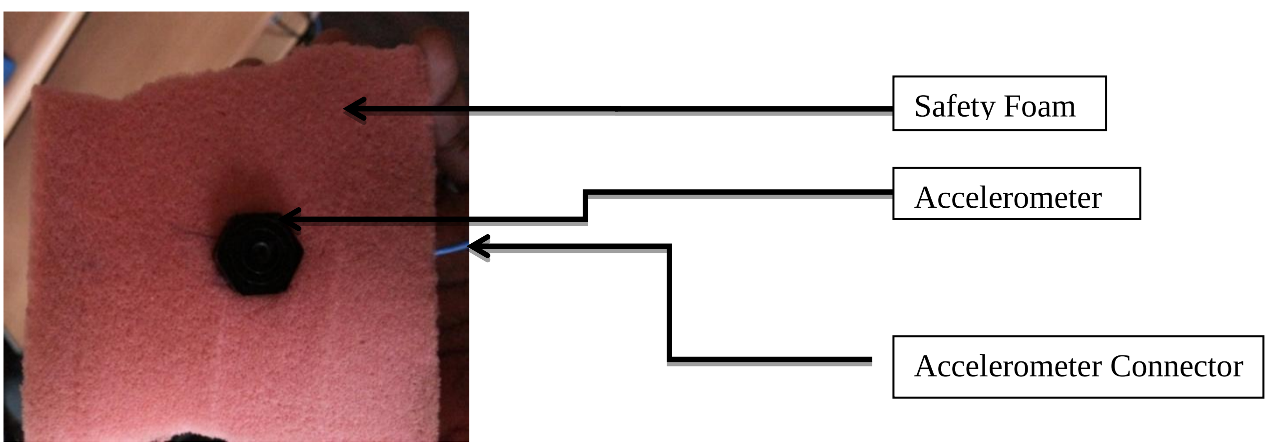


Figure 3.8: Accelerometer

Accelerometer used in the experiment is shown in Figure 3.8, which is placed in between the safety foam to protect the accelerometer.

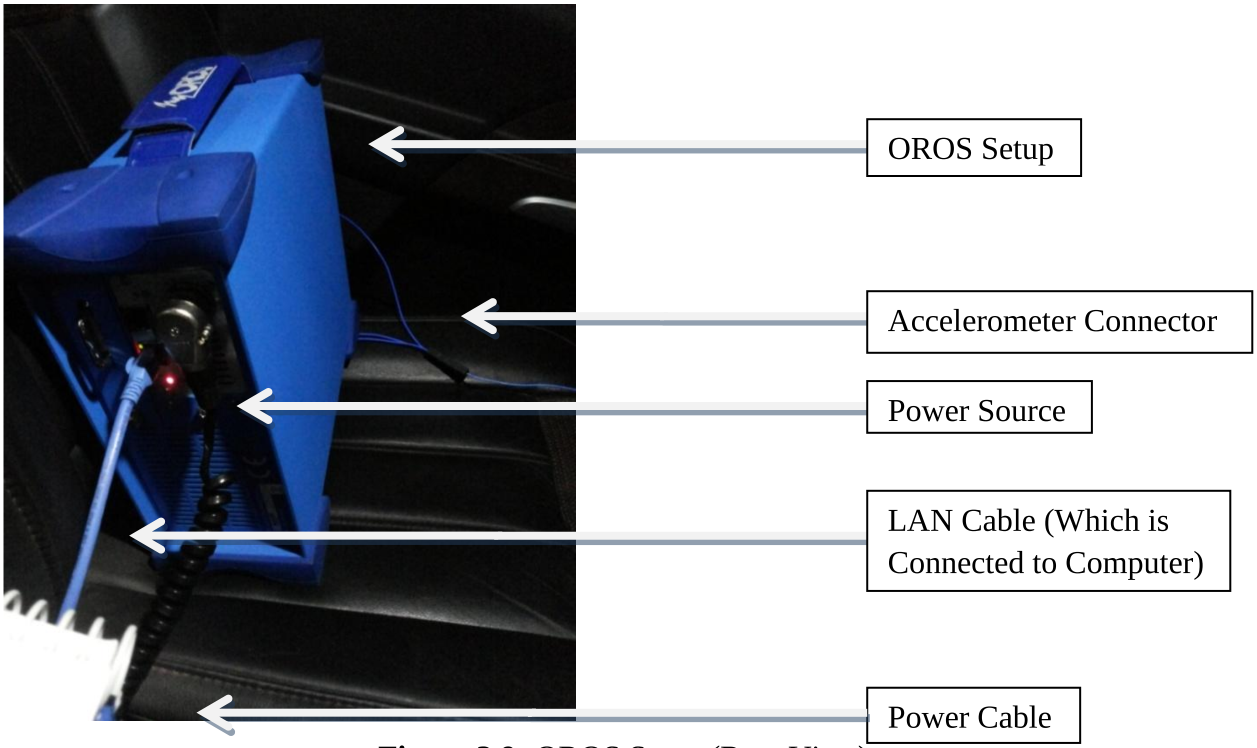


Figure 3.9: OROS Setup (Rear View)

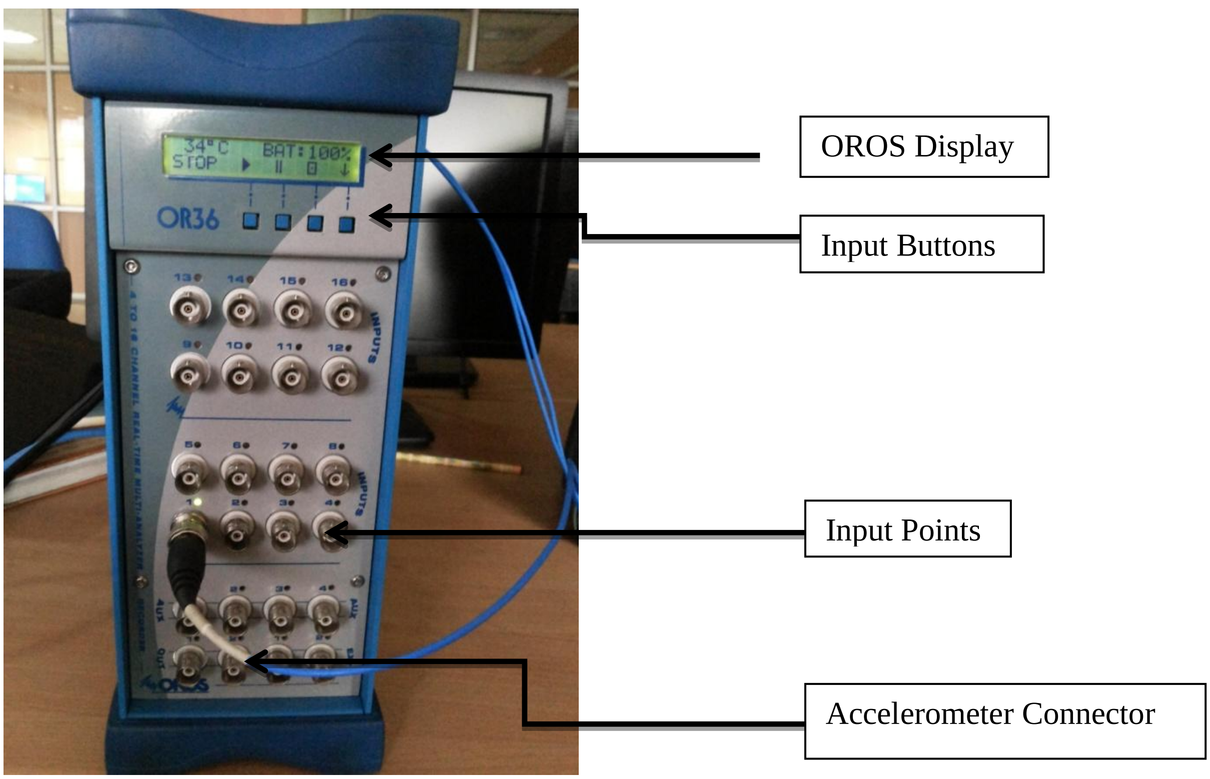


Figure 3.10: OROS Setup (Front View)

3.5 Experimental results

Here, one may evaluate the vertical acceleration of a suspension system of a car on bumps at different speed conditions in order to validate the findings of computational analysis and also the variation from the analytical result. Different observation has been taken at different speeds of the car. This experimental analysis is done for frequency of 800 Hz

Following results shows that one may obtain the maximum acceleration at bump with length of 0.4 meter and height of 0.05 meter.

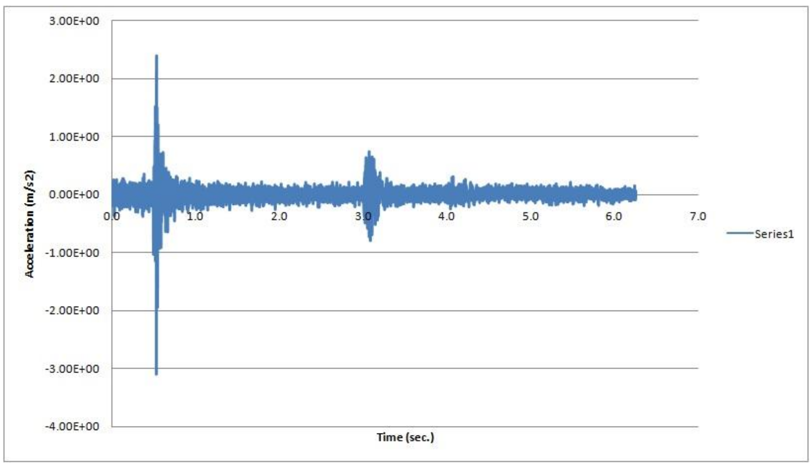


Figure 3.11: Vertical acceleration at speed of 20 Km/hr

The maximum acceleration of a quarter car model is 2.4 m/s^2 at a speed of 20 Km/hr on a bump as shown in Figure 3.11.

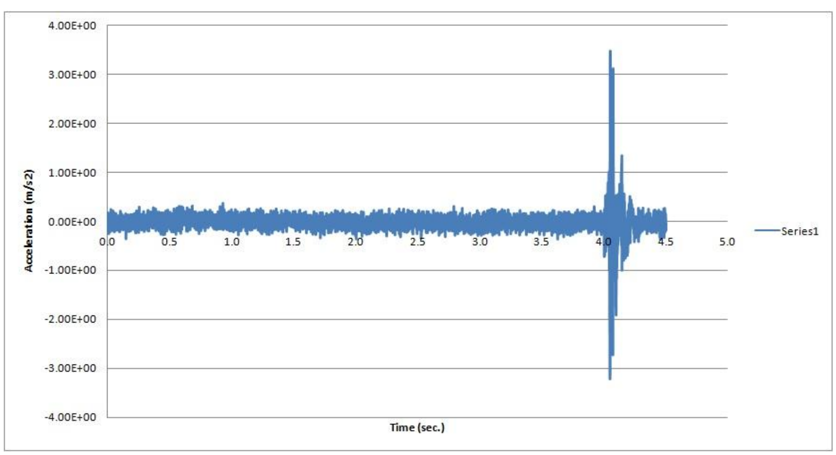


Figure 3.12: Vertical acceleration at speed of 30 km/hr

The maximum acceleration of a quarter car model is 3.34 m/s^2 at a speed of 30 km/hr on a bump as shown in Figure 3.12.

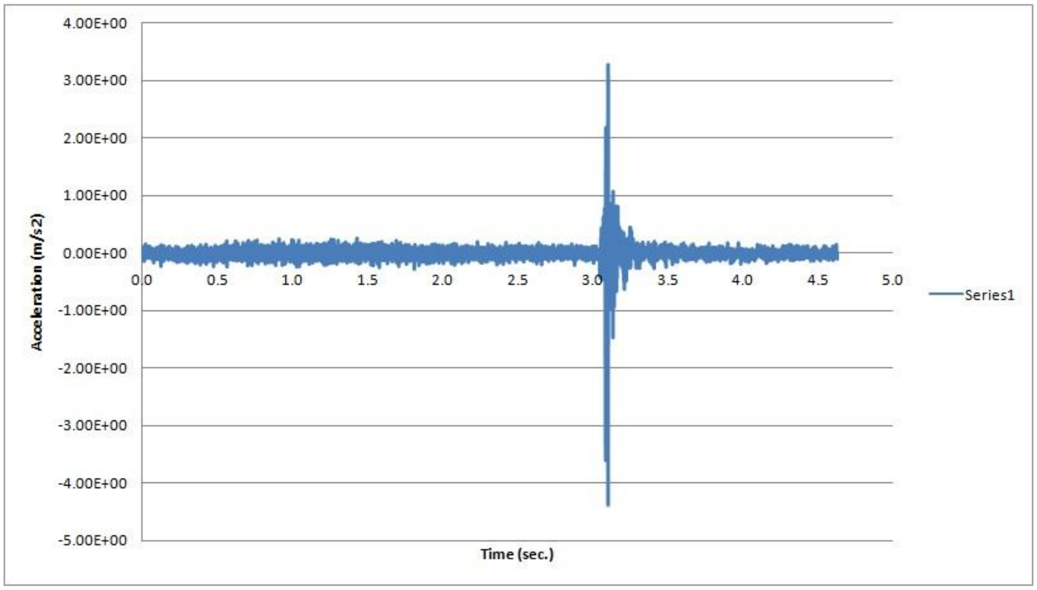


Figure 3.13: Vertical acceleration at speed of 40 Km/hr

The maximum acceleration of a quarter car model is 3.25 m/s^2 at a speed of 40 Km/hr on a bump as show in Figure 3.13.

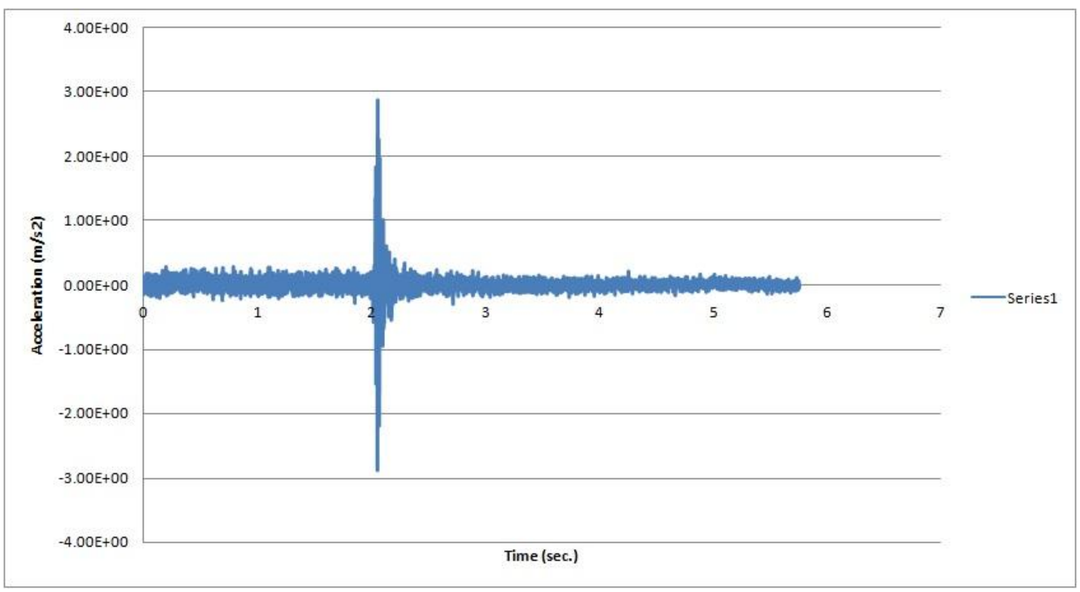


Figure 3.14: Vertical acceleration at speed of 50 Km/hr

The maximum acceleration of a quarter car model is 2.82 m/s^2 at a speed of 50 Km/hr on a bump as shown in Figure 3.14.

CHAPTER 4
BOND GRAPH MODELING

4.1 Introduction

Bond graph is a graphical representation of the physical dynamic system, which is similar to the better known signal-flow graph and block diagram, with a major difference that the arcs in bond graphs represents the bi-directional exchange of physical energy. While those in signal-flow graphs and block diagrams represents the uni-directional flow of information. Bond graphs are multi-energy domain (e.g. electrical, mechanical, hydraulic, etc.) and domain neutral, which means that a bond graph can incorporate multiple domains seamlessly.

Bond graph is composed of the "bonds" which link together "single port" elements, "double port" elements and "multi port" elements. Each bond represents the instantaneous flow of power or energy (dE/dt). The flow in each bond is denoted by a pair of variables which is called as 'power variables' whose product is the instantaneous power of the bond. For example, the bond of an electrical system will represent the flow of electrical energy and the power variables would be current and voltage, whose product is power. In bond graph each domain's power variables is broken into two types: "flow" and "effort". An effort multiplied by flow produces power, thus it is termed as power variables. Every domain has a pair of power variables with a corresponding flow and effort variable. Effort examples include torque, force, pressure or voltage; while flow examples include current, volumetric flow and velocity. The table 4.1 contains the most common energy domains and the corresponding "flow" and "effort".

A bond graph has two other features which is described briefly here, and further discussed in more detail. One is the "half-arrow" sign convection, which defines the assumed direction of positive energy flow. As with free-body diagrams and electrical circuit diagrams, the choice of positive direction is arbitrary, with a caveat that analyst must be consistent throughout with the chosen definitions. The other feature is the "causal stroke", which is a vertical bar placed on only one end of the bond and it is not arbitrary. There are rules for assigning the proper causality to a given port and rules for the precedence among ports. In bond graph any port (single, double or multiple) attached to the bond will specify either "flow" or "effort" by its causal stroke, but not both. The port attached at the end of the bond with a "causal stroke" specifies the "flow" of the bond and the bond imposes "effort" upon that port. Similarly, the port at the end without the "causal stroke" imposes "effort" to the bond, whereas the bond imposes "flow" to that port.

Table 4.1: Power variables in some energy domains

Energy Domain	**Effort**	**e symbol**	**e unit (metric)**	**e unit (imperial)**	**flow**	**f symbol**	**f unit (metric)**	**f unit (imperial)**
Mechanical, translation	Force	F	N	lb	Linear velocity	v	m/s	ft/s, mph
Mechanical, rotation	Torque	T	N·m	ft·lb	Angular velocity	ω	rad/s	rad/s
Electrical	Electromotive force	V or u	V	V	Current	I or i	A	A
Magnetic	Magnetomotive force	e_m			Flux rate	ϕ		
Hydraulic	Pressure	P	Pa	psi	Volumetric flow rate	Q	m^3/s	ft^3/s
Thermal	Temperature	T	°C or K	°F	entropy flow rate	S	W/°C	ft·lb/s·°F

4.1.1 Basics of Bond graph modeling

The fundamental idea of the bond graph is that power is transmitted between connected components by a combination of "flow" and "effort" (generalized flow and generalized effort). Refer to the Table 4.1 for examples of flow and effort in different domains can be ascertained. If an engine is connected to wheel through a shaft, the power is transmitted in the rotational mechanical domain, which means that the flow and the effort are angular velocity (ω) and torque (τ) respectively. The word bond graph is a first step towards the bond graph, in which words define the components. As the word bond graph, this system will look like:

$$\text{engine} \; \frac{\tau}{\omega} \; \text{wheel}$$

To provide a sign convention a half-arrow is used, if the engine is doing work when the τ and ω are positive, then the diagram would be drawn as:

$$\text{engine} \; \overset{\tau}{\underset{\omega}{\rightharpoonup}} \; \text{wheel}$$

To indicate a measurement a full arrow is used and is referred as signal bonds, because the amount of power flowing through the bond is insignificant. However, it may be useful to certain physical components. For example, the power required to activate a relay is orders of magnitude smaller than the power through the relay itself; making it relevant only to convey whether the switch is on, not the power consumed by it.

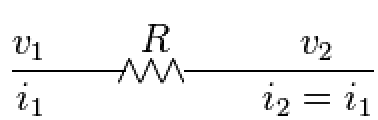
$$\text{wheel} \xrightarrow[\omega]{} \text{tachometer}$$

4.1.2 Junctions structure in bond graph modeling

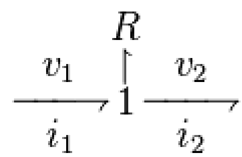
Power bonds can join at one of two kinds of junctions i.e. a 1 junction and a 0 junction.

- In a 1 junction, the flows are equal and the efforts sum to zero. This corresponds to a force balance at a mass in a mechanical system or to an electrical loop.
- In a 0 junction, the efforts are equal and the flow sums to zero. This corresponds to a mechanical "stack" in which all forces are equal or to a node in an electrical circuit (where the Kirchhoff's current law applies).

Consider a resistor in series, for an example of a 1 junction:



In this case, the flow (current) is constrained to be same at all the points, and the efforts sum to zero when the implied current return path is included. Power can be computed at points 1 and 2, whereas in general some power will be dissipated in resistor. Bond graph of this system becomes



This diagram may seem counterintuitive in that flow which is not preserved in the same way across the diagram, from an electrical point of view. It will be helpful to consider the 1 junction as a daisy chaining the bonds it connects to and the power bond up to R as a resistor with a lead returning down. Bond graph modeling proceeds from the identification of key 0 and 1 junctions associated with identifiable flows and efforts in the system, after that identifying the storage elements (I and C) and dissipative (R), power sources, and drawing bonds wherever the information or power flow between the junctions, sources, and dissipative/storage components. Then the sign conventions (arrow heads), and after that the causality are assigned, and finally the equations which is describing the behavior of a system can be derived using the graph as a kind of map or guide.

4.1.3 Concept of causality in bond graph modeling

Bond graphs have a notion of causality that indicate which side of bond determines the instantaneous flow and which determines the instantaneous effort. While formulating the dynamic equations which describe the system, causality for each modeling element, which variable is independent and which is dependent. Analysis of a large scale model becomes easier by propagating the causation graphically from one modeling element to other. In a bond graph model, completing causal assignment will allow the detection of a modeling situation where an algebraic loop exists i.e. the situation when a variable is defined recursively as the function of itself.

Consider a capacitor in series with a battery, as an example of causality. To charge a capacitor instantly is not physically possible therefore anything connected in parallel with a capacitor should necessarily have the same voltage (effort variable) as that of capacitor. Similarly, an inductor cannot change flux instantly therefore any component connected in series with an inductor should necessarily have the same flow as that of inductor. Because inductors and capacitors are passive devices, therefore they cannot maintain their respective flow and voltage indefinitely the components to which they are connected will affect their respective flow and voltage, but only indirectly by affecting their voltage and current respectively. Causality is basically a symmetric relationship. When one side causes flow, the other side causes effort. Active components such as an ideal current or voltage source are also causal.

In a bond graph notation, a causal stroke can be added to one end of the power bond which indicates that the opposite end is defining the effort. For example, consider a constant torque motor which is driving a wheel, that is, a source of effort (SE). Which may be drawn as follows:

$$\begin{matrix}\text{motor} & \\ SE & \xrightarrow[\omega]{\tau}| \text{ wheel}\end{matrix}$$

Similarly, the side with the causal stroke (in this case the wheel) defines the flow for the bond.

Causality results in the compatibility constraints. It is clear that only one end of a power bond can define the effort therefore only one end of a bond can have a causal stroke. The two passive components with time dependent behavior, C and I, can only have one sort of causation i.e. a C component defines effort and I component defines flow. Therefore from a junction J, the only legal configurations for C and I are

$$J \longrightarrow| I \qquad \text{and} \qquad J \vdash\!\longrightarrow C$$

A resistor has no time-dependent behavior therefore a voltage can be applied to get flow instantly, or a flow can be applied to get a voltage instantly. Hence a resistor can be at either end of a causal bond.

$$J \longrightarrow| R \qquad \text{and} \qquad J \vdash\!\longrightarrow R$$

Sources of effort (SE) define effort, sources of flow (SF) define flow. Transformers are passive, neither storing energy nor dissipating, so the causality passes through them.

$$-\!\!-\!\!-\!\!\dashv \dot{T}\dot{F} -\!\!-\!\!-\!\!\dashv \qquad \text{or} \qquad \vdash\!\!-\!\!-\!\!- \dot{T}\dot{F} \vdash\!\!-\!\!-\!\!-$$

A gyrator transforms effort to flow and flow to effort, so if effort is caused on one side, flow is caused on the other side and vice versa.

$$\vdash\!\!-\!\!-\!\!- \dot{G}\dot{Y} -\!\!-\!\!-\!\!\dashv \qquad \text{or} \qquad -\!\!-\!\!-\!\!\dashv \dot{G}\dot{Y} \vdash\!\!-\!\!-\!\!-$$

4.1.4 Junction and Causality

In a 1-junction, flows are equal and in a 0-junction, efforts are equal. Thus, with causal bonds only one bond can cause the flow in a 1-junction and only one can cause the effort in a 0-junction. Therefore if the causality of one bond of the junction is known, the causality of the others is also known. That one bond is known as a strong bond

$$\text{strong bond} \rightarrow \dashv \overset{\top}{\underset{\bot}{0}} \dashv \qquad \text{and} \qquad \text{strong bond} \rightarrow \vdash \overset{\bot}{\underset{\top}{1}} \vdash$$

Using the above rules one can continue to assign the causality. If any model which results in inconsistent causality therefore, it is not physically valid. For example, consider an inductor in series with an ideal current source which is a physically impossible configuration and the bond graph would look like.

$$SF -\!\!-\!\!-\!\!- 1 -\!\!-\!\!-\!\!- I$$

Assigning causality to the source bond we get,

$$SF \vdash\!\!-\!\!-\!\!- 1 -\!\!-\!\!-\!\!- I$$

Propagating the causality through the junction will give,

$$SF \vdash\!\!-\!\!-\!\!- 1 \vdash\!\!-\!\!-\!\!- I$$

But assigning causality to the inductor will give,

$$SF \vdash\!\!\!-\!\!\!-\!\!\!-\!\!\!-1 \vdash\!\!\!-\!\!\!-\!\!\!-\!\!\!-\!\!\!-| \; I$$

The causality on the right bond is redundant, therefore this is invalid. This ability which helps to automatically identify impossible configurations is a major advantage of bond graph.

4.2 Car Modeling

In this section, full car model is treated for a standard road-going vehicle. This model has been equipped with suspension force, actuators to improve the vehicle's ride comfort. Active suspension is concerned with controlling the vertical movements of the vehicle in response to the road inputs to each of the wheels. This is accomplished by actively applying vertical forces in the suspension to counteract some of the effects of the road surface. As a result, these systems can be used to minimize vehicle body roll, vertical accelerations experienced by the passengers, and improve overall vehicle handling.

4.2.1 Modeling assumptions

The following assumptions were made to simplify the model as model of vehicle considers three dynamics i.e. pitching, rolling and bouncing which significantly affect dynamic behavior of the vehicle.

- The body of the vehicle is rigid.
- The lateral and longitudinal motion of the tires is negligible compared to their vertical motion.
- The vehicle is a neutral steer car.
- The vehicle is not skidding.
- The vehicle is moving with a constant velocity on rigid and constant gauge.
- The spring damper system is assumed to be mass less.
- The spring and damper of the suspension system element have linear characteristics.
- The tires of the vehicle remain in contact with the road all the time.
- The positions of the two ends of the spring connecting two rigid or connecting one rigid body and one contact point are required as input data.
- Each rigid body connected to each other by spring damper system.

- Straight road is assumed.

4.2.2 Vehicle structure

Figure 4.1, illustrates the vehicle model considered in this study conforming to Indian road vehicles. Body and base frame are treated as rigid bodies. The body (sprung mass) is modeled as the rigid body having a mass M_b with three actions pitching, rolling and yawing about the lateral, longitudinal and axes respectively. The spring and shock absorber in the suspension system are characterized by the damping coefficient C_s and spring stiffness K_s respectively. The various displacements of the vehicle are described with respect to the equilibrium positions. As the vehicle is assumed to be rigid thus its motion can be described by the vertical displacement (bounce or Z_c), about the longitudinal horizontal axis (roll or Φ_c) and the rotation about the transverse horizontal axis (pitch or θ_c). Figure 4.1 shows a schematic diagram of a car representing the motion in three directions.

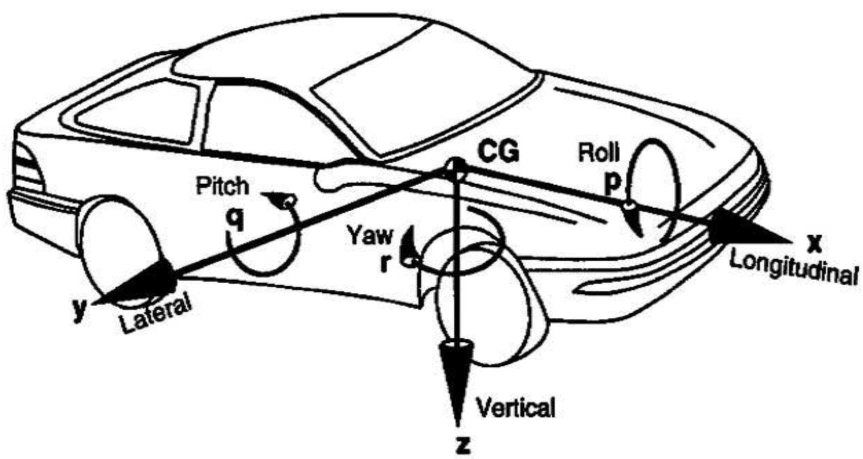


Figure 4.1: Schematic diagram of a complete car

4.2.3 Detailed description of elements of suspension system

Suspension system can be disassembled into the following parts, i) mass, ii) spring, iii) damper. The next subsection will highlight these in detail.

4.2.3.1 Mass

Mass of a vehicle is subdivided into two parts- a) sprung mass, b) unsprung mass.

a) Sprung mass

Sprung mass is the portion of the vehicle's total mass that is supported above the suspension which also includes half of the weight of the suspension itself. Sprung weight typically includes the internal components, body, cargo, frame and passengers; but it does not include the mass of the components which is suspended below the suspension components including the wheel bearings, wheels, calipers, brake rotors, and continuous tracks (also called caterpillar tracks), if any.

b) Unsprung mass

Most of the vehicle's weight is supported by its suspension system, which suspends the body and associated parts so that they are insulated from vibrations and road shocks that would otherwise be transmitted to the vehicle and the passengers itself. However, the other parts of vehicle are not supported by the suspension system, such as the tires, steering, brakes and suspension parts are also not supported by the springs. All these parts are called unsprung weight, which should be kept as low as possible in most of the application.

4.2.3.2 Spring

Spring is an elastic object used to store mechanical energy, which is usually made of spring steel. Large springs are made from annealed steel and hardened after fabrication, whereas Small springs can directly be wound from pre-hardened stock. Some non ferrous metals are also used including titanium and phosphor bronze for springs requiring corrosion resistance and beryllium copper for the springs carrying electrical current (because of spring low electrical resistance).

When a spring is stretched or compressed, then the exerted force is proportional to its change in length. The rate or spring constant of a spring is the change in the force it exerts, which may be divided by the change in the deflection of spring. Basically, it is the gradient of force versus deflection curve. A compression or extension spring has unit of force divided by distance, for example N/m or lbf/in. Torsion springs has unit of force multiplied by distance divided by angle, for example ft·lbf/degree or N·m/rad. Inverse of spring rate is compliance, i.e. if the spring has a rate of 10 N/mm, then it has a compliance of 0.1 mm/N. The stiffness or rate of springs in parallel is additive, whereas the compliance of springs in series.

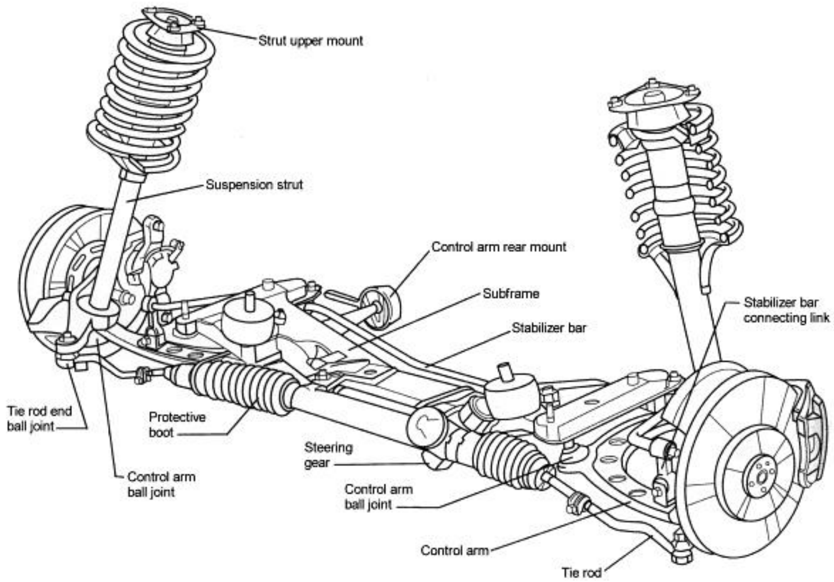


Figure 4.2: Schematic diagram of a suspension

4.2.3.3 Damper

It is a mechanical device designed to smooth out or damp shock impulse, and dissipate kinetic energy. In a vehicle, it reduces the effect of traveling over rough ground leading to improved ride quality and increase in comfort. While it serves the purpose of limiting excessive suspension movement and their intended sole purpose is to dampen spring oscillations. It uses valving of oil and gasses to absorb excess energy from the springs. Spring rates are chosen by the manufacturer based on the weight of the vehicle, loaded and unloaded. Some people use shocks to modify spring rates but this is not the correct use. Along with hysteresis in the tire itself, they dampen the energy stored in the motion of the unsprung weight up and down. Effective wheel bounce damping may require tuning shocks to an optimal resistance. Spring-based dampers commonly use coil springs or leaf springs, whereas torsion bars are used in torsional shocks as well. Ideal springs alone, however, are not dampers, as springs only store and do not dissipate or absorb energy in any form.

4.3 Dynamic model of road car

Dynamic model of Quarter and full car is shown in Figure 4.3 and Figure 4.4.

4.3.1 Quarter car model

This model consists of single wheel and consists of two degree of freedom. The labeled diagram is shown in Figure 4.3.

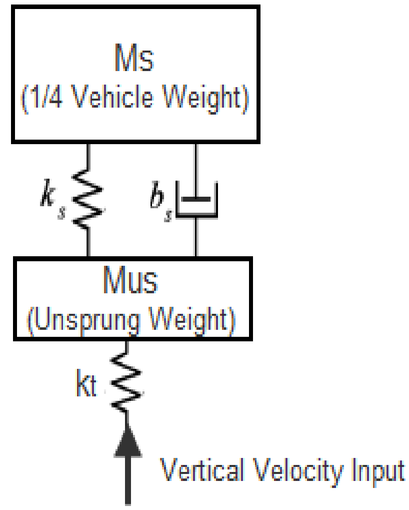


Figure 4.3: Schematic diagram of a quarter car

4.3.2 Full car model

The vehicle model includes a suspension unit at each corner of the vehicle which consists of a spring, damper and a force actuator as shown in Figure 4.4

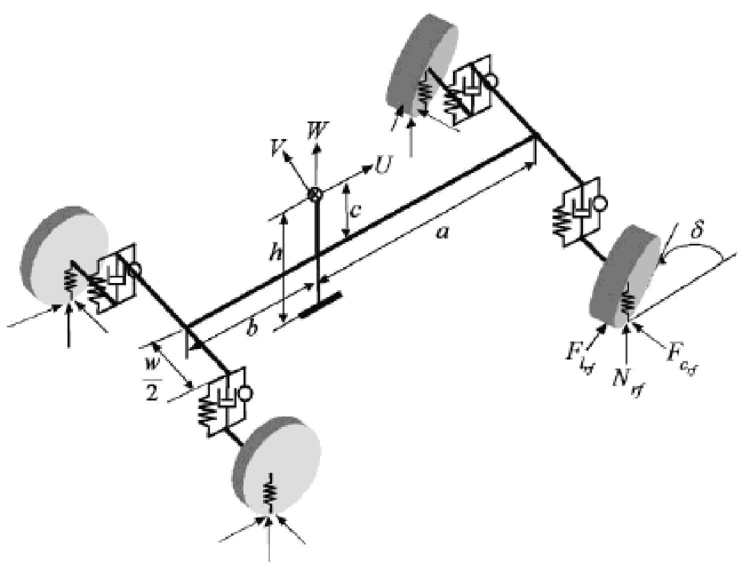


Figure 4.4: Schematic diagram of a full car

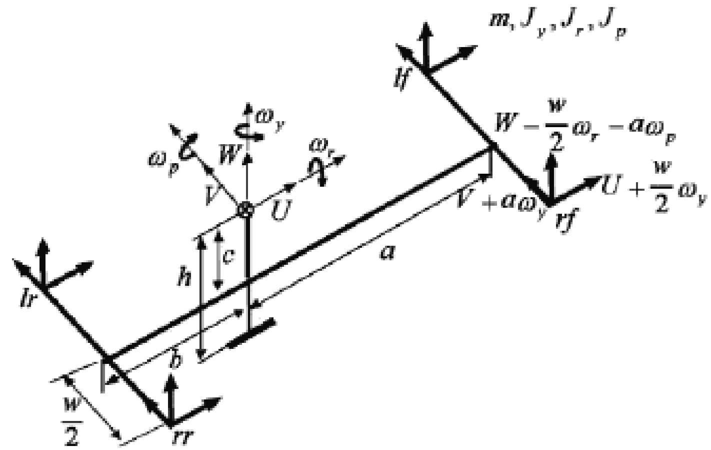


Figure 4.5: Coordinate system for a rigid body

The vehicle chassis is modeled as a rigid body with body fixed coordinates, U, V, and W attached at the Center of Gravity (CG) and aligned in its principal directions as shown in Figure 4.5. The body has mass m, and moments of inertia J_r (roll) about the U-axis, J_p (pitch) about the V-axis, and J_y (yaw) about the W axis. The CG is located a distance 'a' from the front axle, 'b' from the rear axle, and 'h' from the ground. The half-width of the vehicle is w/2. The suspension actuators are implemented simply as controllable force inputs. This will allow more flexibility once the control system has been designed for selecting the most appropriate actuator.

4.4 Bond Graph modeling

A vehicle suspension system is composed of components i.e. springs, dampers and tires. When the dynamic system is put together from these components, it must interconnect translating and rotating inertial elements with rotational and axial spring's dampers and must also appropriately accounts for the kinematics of the system structure. The bond graphs are well suited for this task.

Modeling the mentioned system began with the creation of a 'bond graph' of the system. Bond graphs are a concise pictorial representation of all types of interacting energy domains, and are an excellent tool for representing vehicle dynamics with associated control hardware. Each bond represents a pair of signals (flow and effort), whose product is an instantaneous power of the bond. In the case of a mechanical system, effort and flow translate into force and velocity respectively. The 'half arrow' sign convention defines the direction of energy flow. The energy storing elements in the bond graph define the number of state variables in the system and using the established methods in bond graphing. State equations can also be derived directly from the bond graph.

4.4.1 Bond graph model of quarter car

Whole bond graphs are divided into two parts- sprung and unsprung. Node 3, node 6 and node 7 represent inertia, stiffness and resistance of sprunged parts whereas node 9, node 12 represent inertia and stiffness of unsprunged parts. SF is source of flow which correspond velocity of vehicle and node 2 shows total weight of vehicle. The bond Graph of quarter car model is shown in Figure 4.6

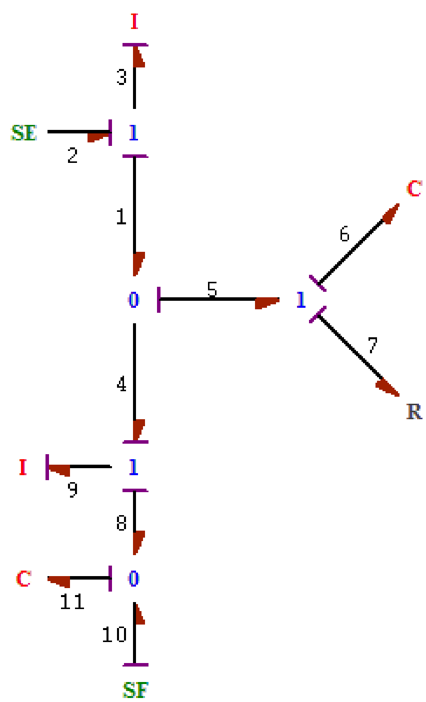


Figure 4.6: Bond graph of a quarter car model

4.4.3 Bond graph model of full car

Like quarter car model's nomenclature a full car model can also be created. Only difference from quater car model is that, it consists of four wheels and so every part will be four times**.** Node 10 represents total weight of vehicle which is in form of source of energy. TF is used for showing the distance from particular point. The remaining nomenclatures is similar to quarter car model.

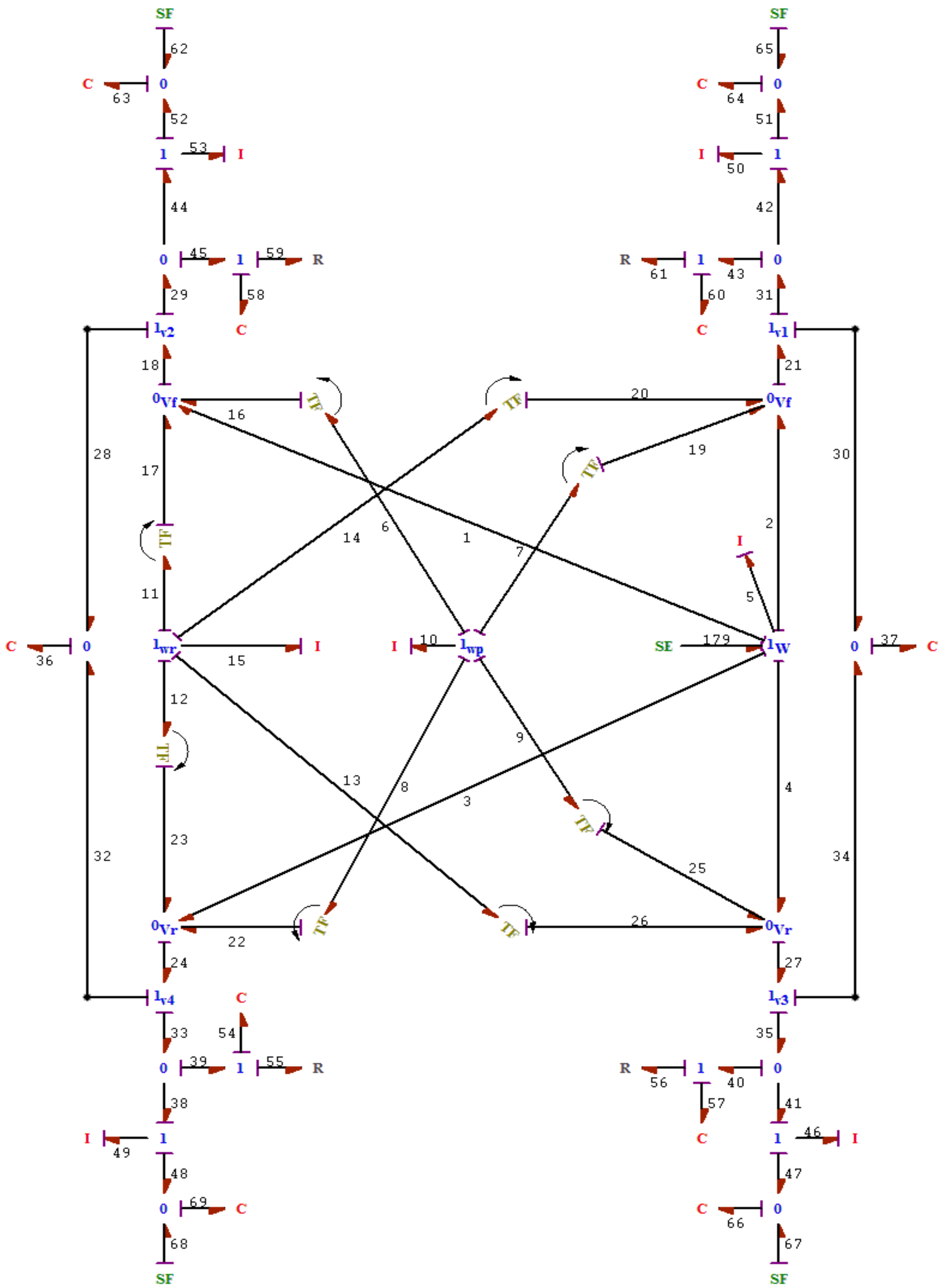


Figure 4.7: Bond graph of a full car model

4.5 Summary of the chapter

In this chapter, bond graph model of quarter car and full car suspension system have been created. The nomenclature presents all notations used in the bond graph model. Next chapter will present the simulation study of the bond graph model.

CHAPTER 5
SIMULATION STUDY

5.1 Introduction

Computer simulation can compress the performance of a system over years into a few minutes of computer running time. Simulation models are comparatively flexible and can be modified to accommodate the changing environment to the real situation. There is no area, where the technique of system simulation cannot be applied as the complexities of the problem increase the scope of application of simulation equations. At present, most of the simulation models are made by means of differential equations. In this research, quarter car and full car model suspension system is investigated at different operating speed on bumps using Bond Graphs technique and simulator of SYMBOLS-Sonata software are used.

5.2 Simulation environment

The Simulator of Symbols sonata, which is the base post-processing module of SYMBOLS Sonata, is used for the simulation quarter and full car model on bumps at different operating speed.

SYMBOLS Sonata software

SYMBOLS Sonata is the next generation of SYMBOLS software (System Modeling by Bond graph Language and Simulation) running in Microsoft Windows 95/98/XP/NT 4.0 environment. It is a modeling, simulation and control systems software for a variety of scientific and engineering applications. Being a powerful research tool, it can help avoid unaffordable, sophisticated fabrications. Yet, one may know precisely the response characteristics of the simulated system. A model in SYMBOLS Sonata may be created using combination of bond graphic elements, block diagram elements in capsulated forms or others capsules. Even model can be created purely using capsules. Sub-model capsules can be imported from the huge capsules library or can even be created by the modeler. The pre-cast capsules are not Pandora's boxes. They can be opened using the Bond pad editor and customized according to modeler need. Modeler may personalize and organize capsules created by them to separate their capsule group from other users.

Key Features of SYMBOLS Sonata Software is as follows

- Drawing a Bond graph model.
- Augmenting the model by numbering the bonds, assigning power direction.
- Causality, module of 2-port elements, bond activation etc.
- Validation of the Bond graph. Bond graph's integrity is validated after the model is created.
- Creation of non-integrated observers in form of detectors.
- Equations can be generated and displayed on single pallets.
- Creation of expressions.
- Generation of program code.
- Creation of sub-system models henceforth called as Capsules and incorporation of Capsules in a Bond graph model.
- Fault diagnosis.
- Preparing models for simulator and control modules.
- Export of sub-system and system models to MAT lab/Simulink environment.
- Integrated simulation environment.
- Easy to access control panels.
- Multiple and intelligent entry mode.
- Online plotting, pause, stop and resume option.
- Online parameter variation through slider during simulation.
- Continuous run simulation, multiple simulations of different systems at the same time.
- Simulation extension facility.
- Advance post simulation plotting facilities.
- Online code editing and compilation.
- Different integration methods for stiff equations.
- Multi-run facility with interpolated or discrete parameter values.
- Event handlers and notification messages.
- Direct debugging and variable tracking.
- Improved external date and chart interpolation routines.
- Export routines for Microsoft Excel Datasheet.

5.3 Simulation properties

The bond graph model of the vehicle is simulated for 10 sec to obtain different output responses. Total 1024 records are used in the simulation and simulation error is kept in the order of 5.0 x 10^{-4}. Runge-Kutta Gill method of fifth order is used in present work to solve the differential equations generated through bond graph model.

Runge-Kutta method

Runge-Kutta methods propagate a solution over an interval by combining the information from several Euler-style steps (each involving one evaluation of the state equations), and then using the information obtained to match a Taylor series expansion up to some higher order. This method treats every step in a sequence of steps in an identical manner. This is mathematically correct, since any point along the trajectory of an ordinary differential equation can serve as an initial point. Fifth-order Runge-Kutte method, is used in present simulation work.

5.4 Road inputs

Bump is taken into consideration which have height H and length L

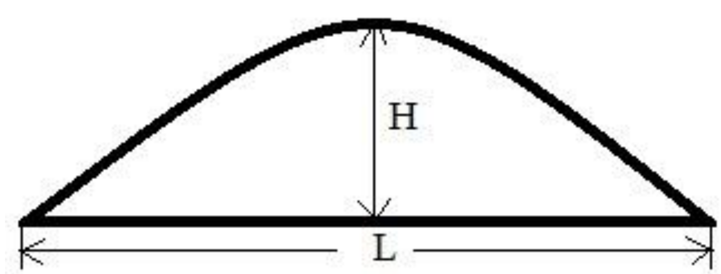


Figure 5.1: Schematic diagram of road bump

Equations involved for bump simulation are as follows

V_2=-PI*H*V/L*cos(PI*V/L*(t-u/V))*swi(t,u/V)*swi((u+L)/V,t) (5.1)

V_2=-PI*H*V/L*cos(PI*V/L*t)*swi(t,0)*swi(L/V,t) (5.2)

These eqations are used in bond graph to simulate a car suspension on bump at different speeds.

5.5 Simulation parameters

Table.5.1 presents the linear parameters used for the vehicle simulation. These parameters are used to populate the state-space representation of the bond graph model.

Table 5.1: Parameter values for simulation

Parameter	**Value**
Vehicle	
Distance from Cg to front axle (a)	1.17m
Distance from Cg to rear axle (b)	1.68m
Height of Cg above the road (h)	0.55m
Track (w)	1.54m
Mass of the car (m_s)	1513 kg
Roll moment of inertia (J_r)	637.26 kgm2
Pitch moment of inertia (J_p)	2443.26 kgm2
Anti-roll bar stiffness (k_a)	1.5 x 106 N/m
Tire	
Unsprung mass (m_{us})	38.42 kg
Tire stiffness (k_t)	150,000 N/m
Suspension	
Suspension stiffness (k_s)	14,900 N/m
Damper coefficients (b_s)	475 Ns/m

Table 5.2: Bump parameter

Bump Parameters	
Height	0.05m
Length	0.4m

Following Simulation studies are carried out for the vehicle at varying speed:

i. Simulation Study of Quarter car Model
ii. Simulation Study of Full car model

5.6 Simulation study for quarter car model

Acceleration of suspensions on bump at different speeds for quarter car model are presented in this section, which are obtained through simulation of bond graph model.

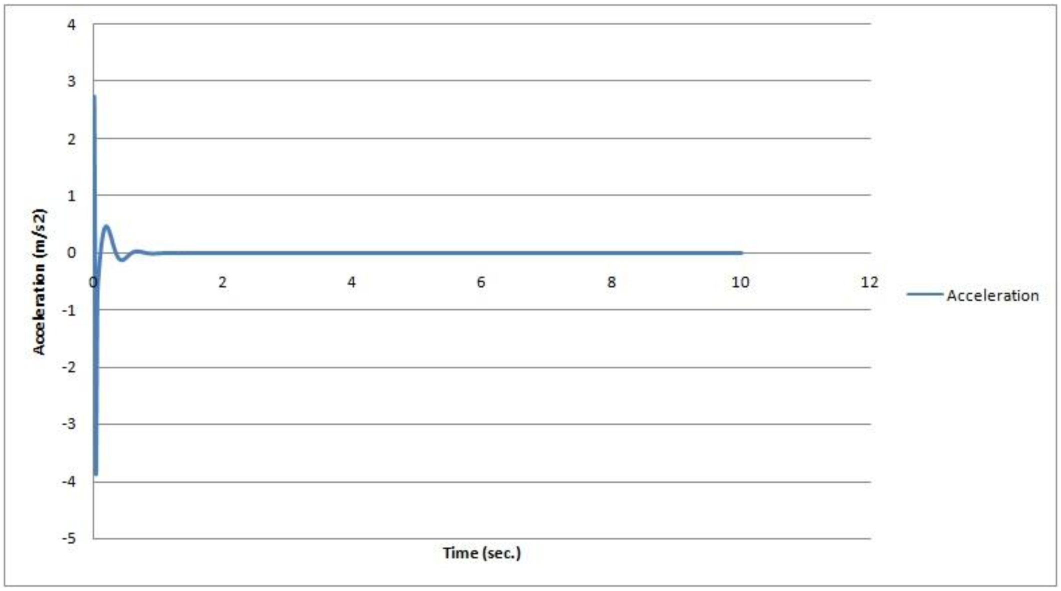


Figure 5.2: Acceleration of quarter car on bump at speed of 20 Km/hr

Maximum acceleration of quarter car model given out in Figure 5.2 is 2.7153 m/s^2 at a speed of 20 Km/hr on a bump, when the simulation is run for 12 seconds.

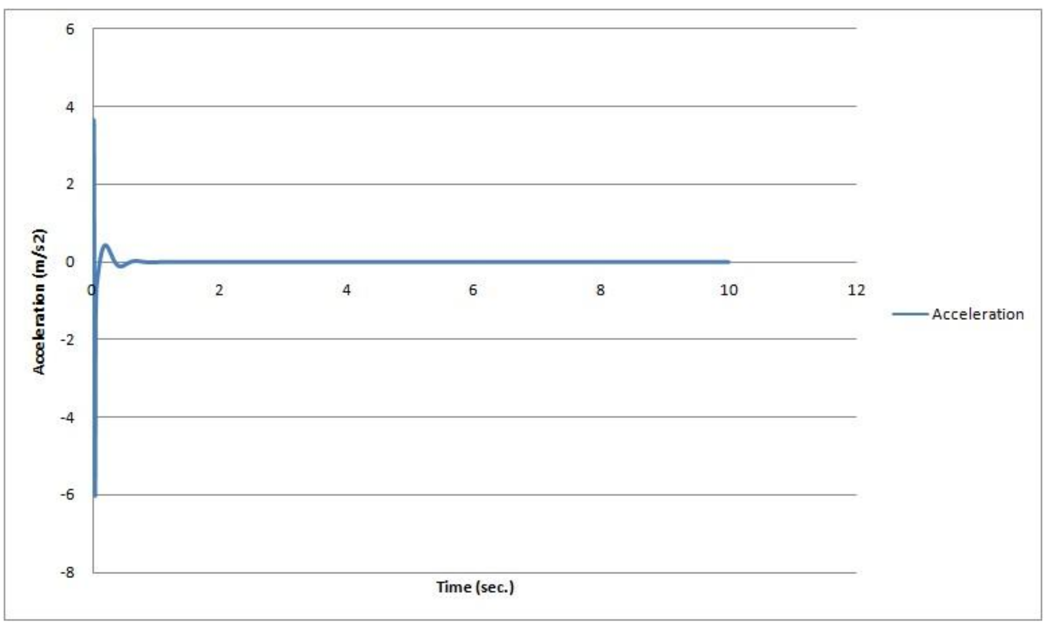


Figure 5.3: Acceleration of quarter car on bump at speed of 30 Km/hr

Maximum acceleration of quarter car model given out in Figure 5.3 is 3.6473m/s^2 at a speed of 30 Km/hr on a bump, when the simulation is run for 12 seconds.

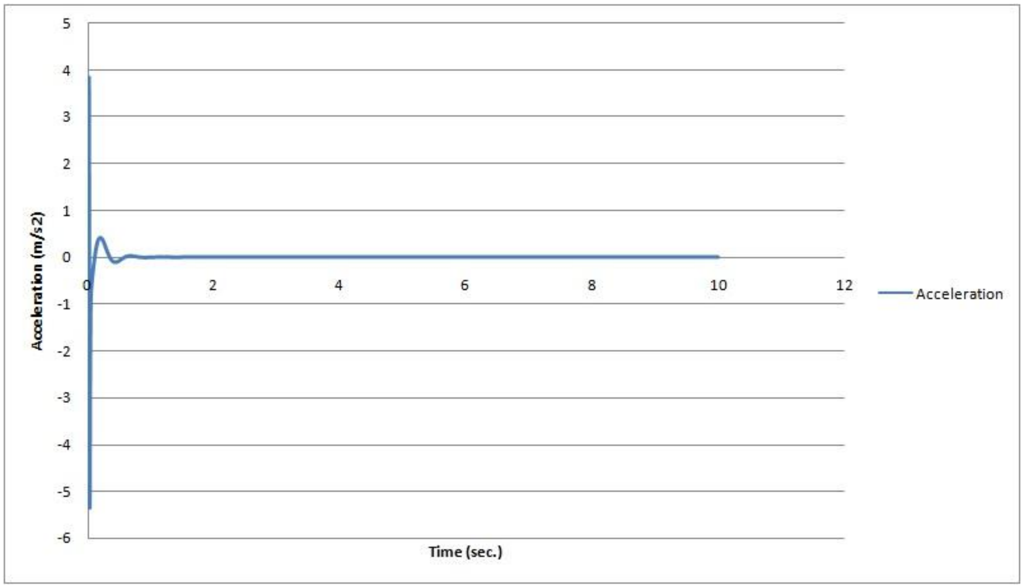


Figure 5.4: Acceleration of quarter car on bump at speed of 40 Km/hr

Maximum acceleration of quarter car model given out in Figure 5.4 is 3.7067 m/s^2 at a speed of 40 Km/Hr on a bump, when the simulation is run for 12 seconds.

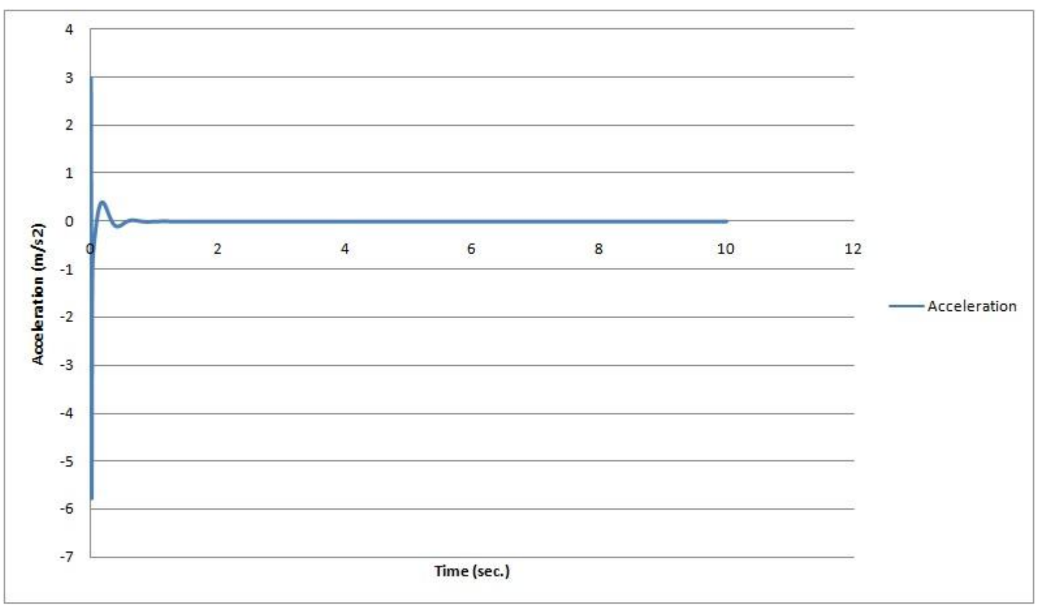


Figure 5.5: Acceleration of quarter car on bump at speed of 50 Km/hr

Maximum acceleration of quarter car model given out in Figure 5.5 is 2.7960 m/s^2 at a speed of 50 Km/hr on a bump, when the simulation is run for 12 seconds.

5.7 Suspension deflection at various speed for full car model

Deflection of suspensions over the bump at different speeds for a full car model are shown below

5.7.1 Suspension deflection at speed of 20 Km/hr.

The deflection of suspension system of full car at 20km/hr for front and rear wheels are shown in Figures 5.6 to 5.9.

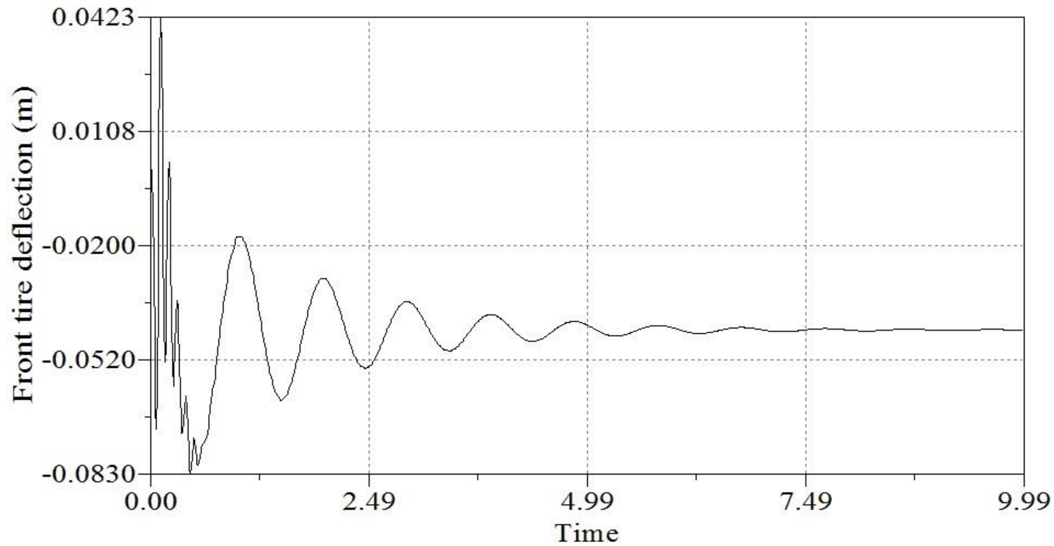


Figure 5.6: Deflection of front wheel (right) on bump at speed of 20 Km/hr

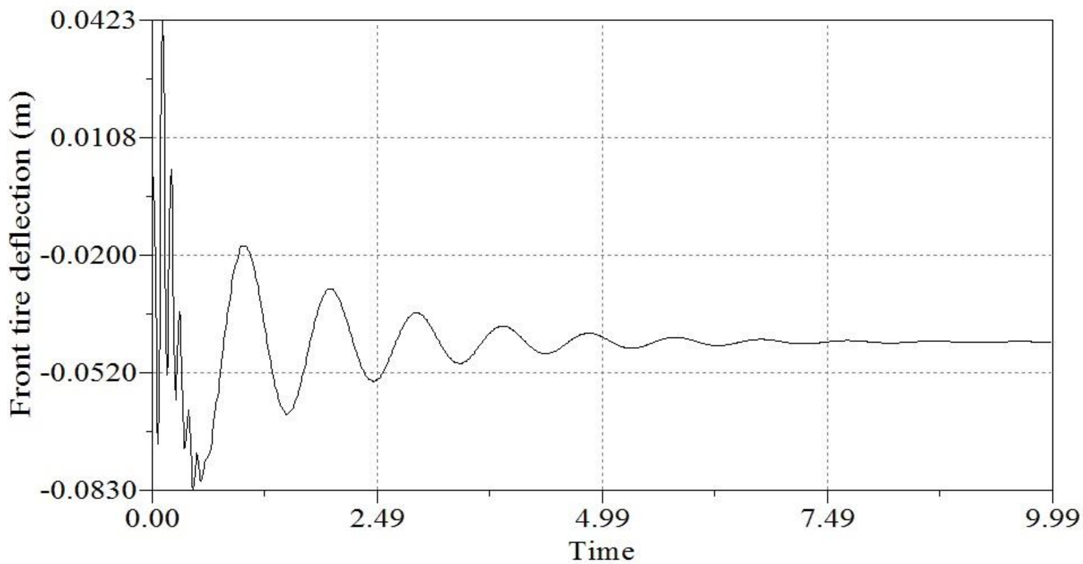


Figure 5.7: Deflection of front wheel (left) on bump at speed of 20 Km/hr

Figure 5.6 and Figure 5.7 show the deflection of a car suspension over the bump at a speed of 20 Km/hr for front right and front left wheel respectively

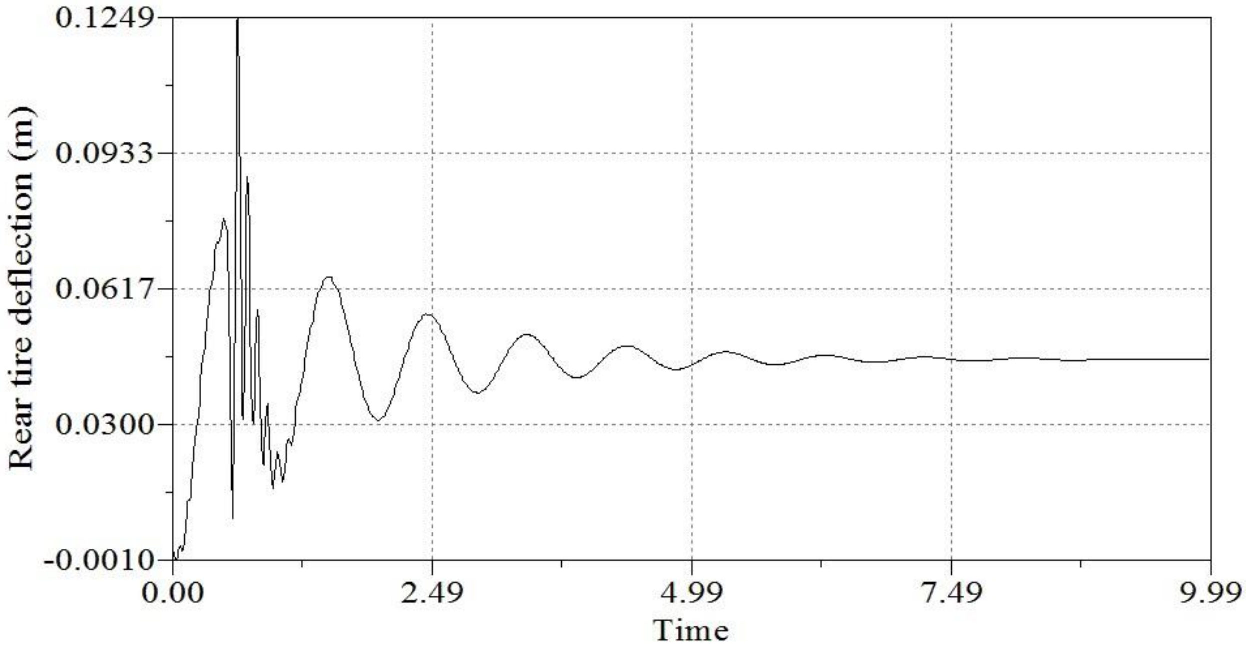


Figure 5.8: Deflection of rear wheel (right) on bump at speed of 20 Km/hr

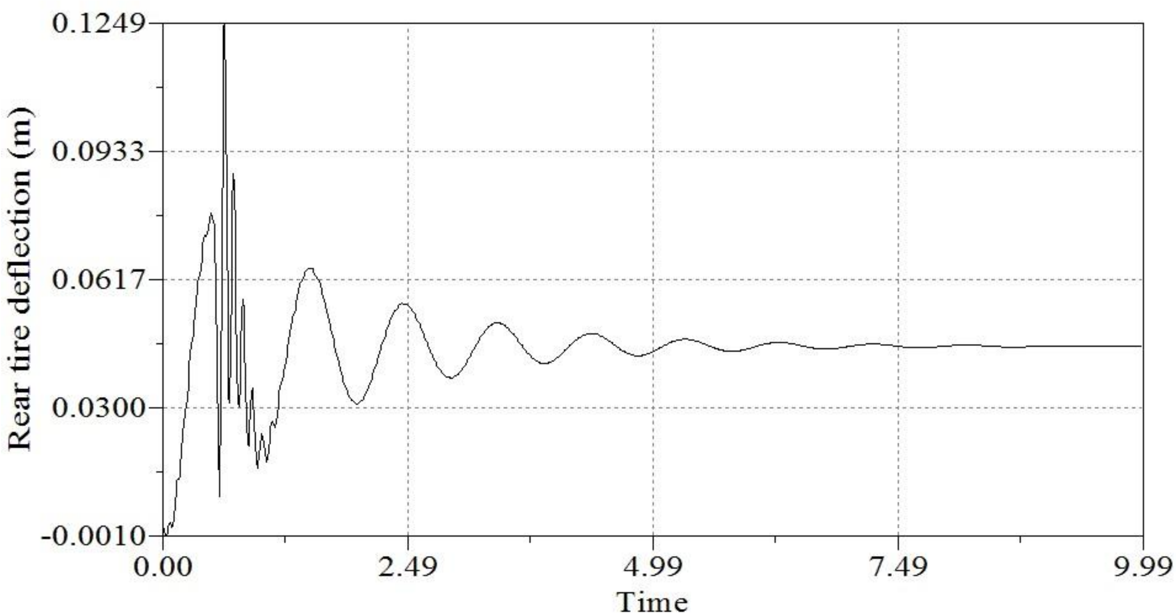


Figure 5.9: Deflection of rear wheel (left) on bump at speed of 20 Km/hr

Figure 5.8 and Figure 5.9 show the deflection of a car suspension over the bump at a speed of 20 Km/hr for rear right and rear left wheel respectively.

5.7.2 Suspension deflection at speed of 40 Km/Hr.

The deflection of suspension system of full car at 40km/hr for front and rear wheels are shown in Figures 5.10 to 5.13

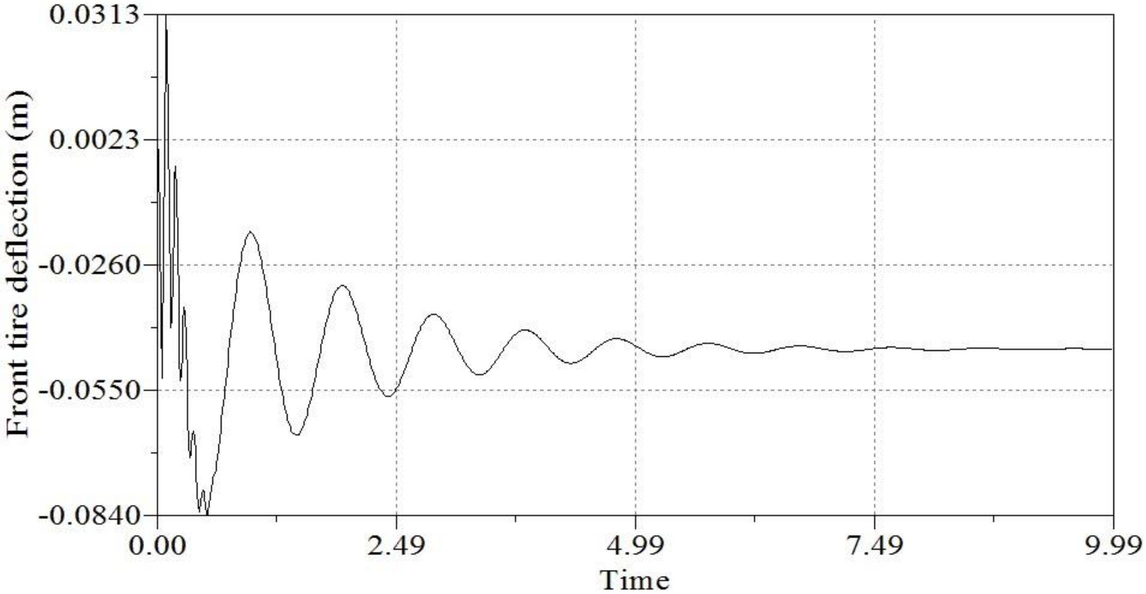


Figure 5.10: Deflection of front wheel (right) on bump at speed of 40 Km/hr

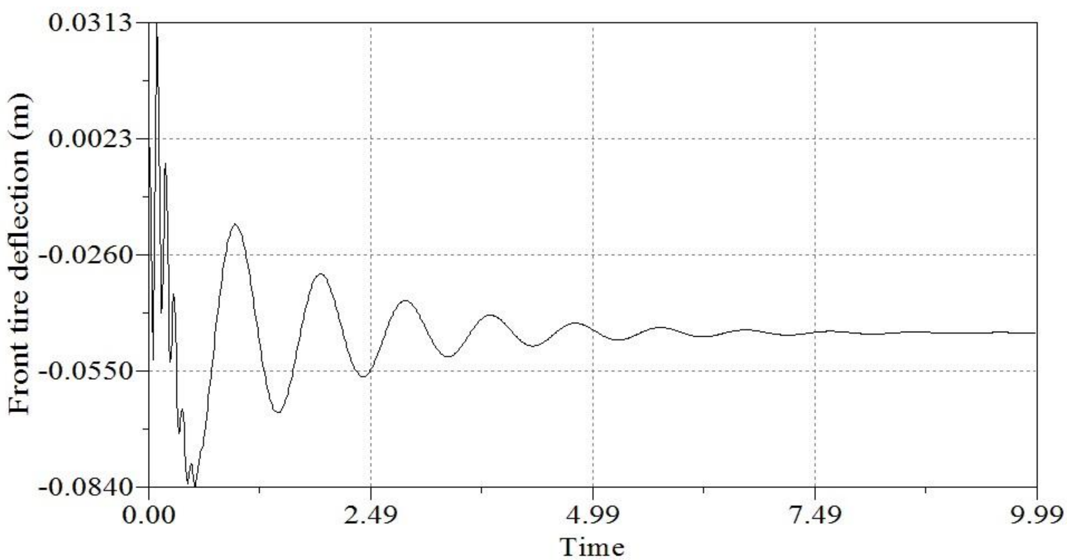


Figure 5.11: Deflection of front wheel (left) on bump at speed of 40 Km/hr

Figure 5.10 and Figure 5.11 show the deflection of a car suspension over the bump at a speed of 40 Km/hr for front right and front left wheel respectively.

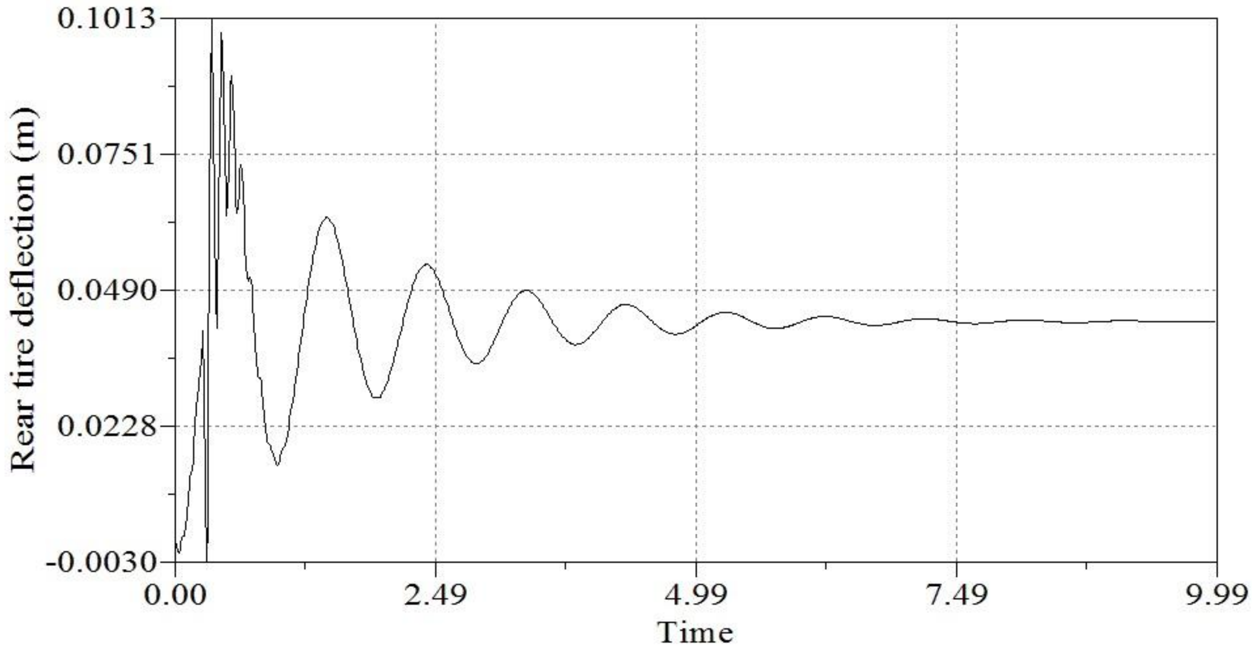


Figure 5.12: Deflection of rear wheel (right) on bump at speed of 40 Km/hr

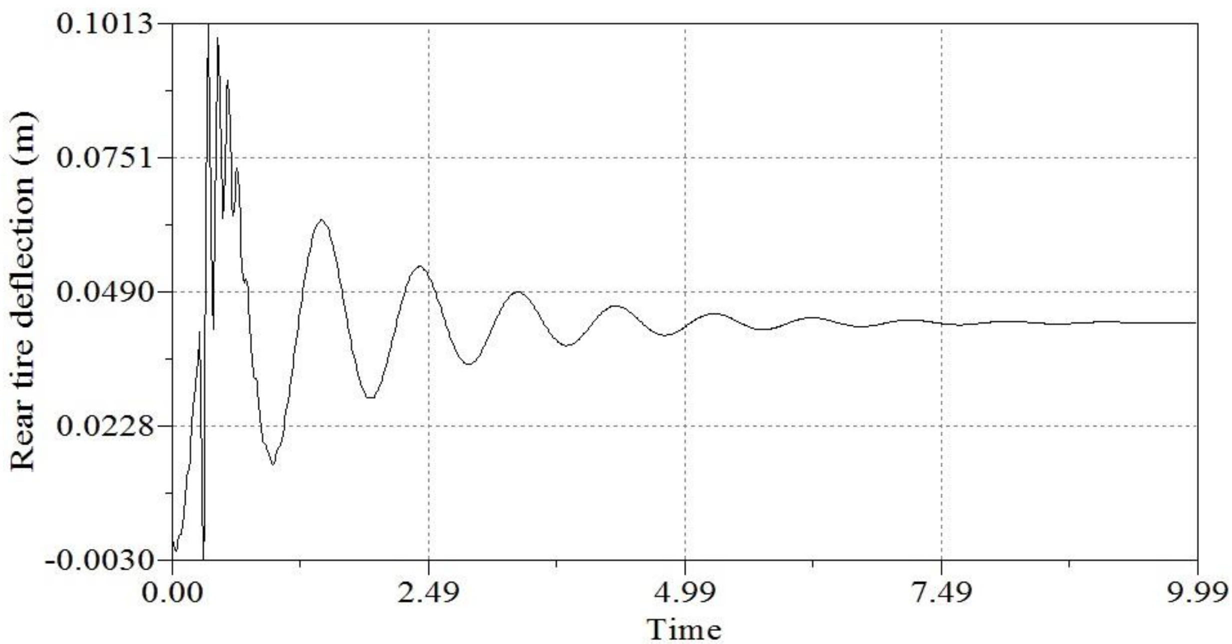


Figure 5.13: Deflection of rear wheel (left) on bump at speed of 40 Km/hr

Figure 5.12 and Figure 5.13 shows the deflection of a car suspension over the bump at a speed of 40 Km/hr for rear right and rear left wheel respectively.

5.7.3 Suspension deflection at speed of 60 Km/Hr.

The deflection of suspension system of full car at 60 km/hr for front and rear wheels are shown in Figures 5.14 to 5.17.

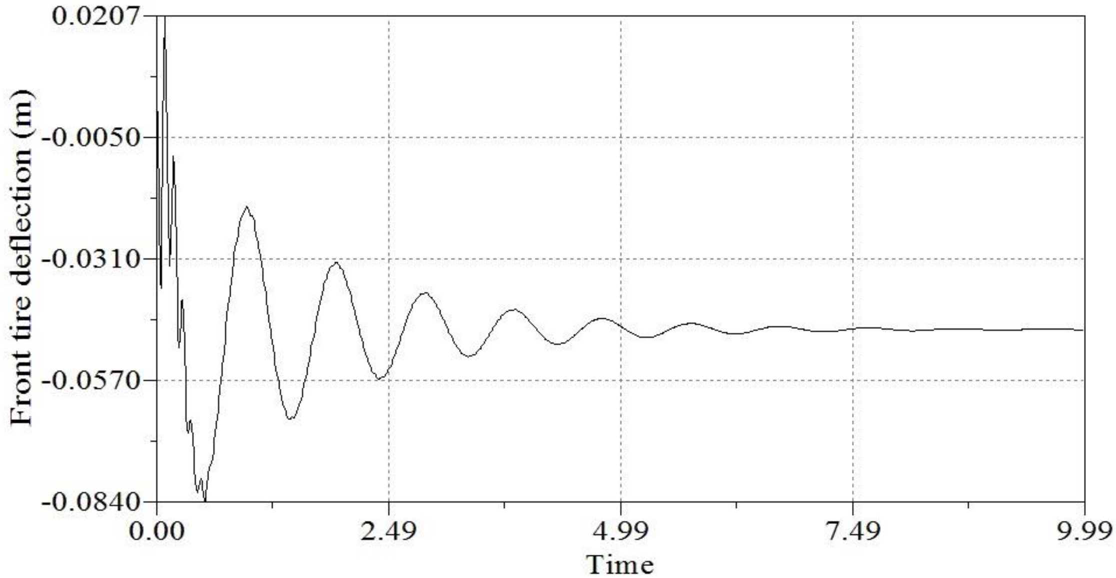


Figure 5.14: Deflection of front wheel (right) on bump at speed of 60 Km/hr

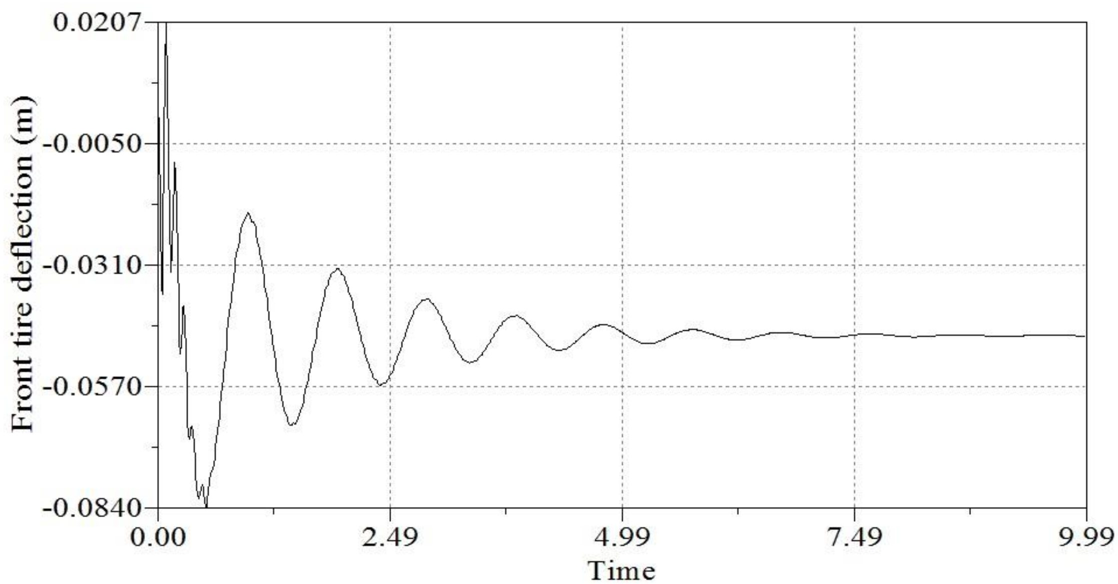


Figure 5.15: Deflection of front wheel (left) on bump at speed of 60 Km/hr

Figure 5.14 and Figure 5.15 show the deflection of a car suspension over the bump at a speed of 60 Km/hr for front right and front left wheel respectively.

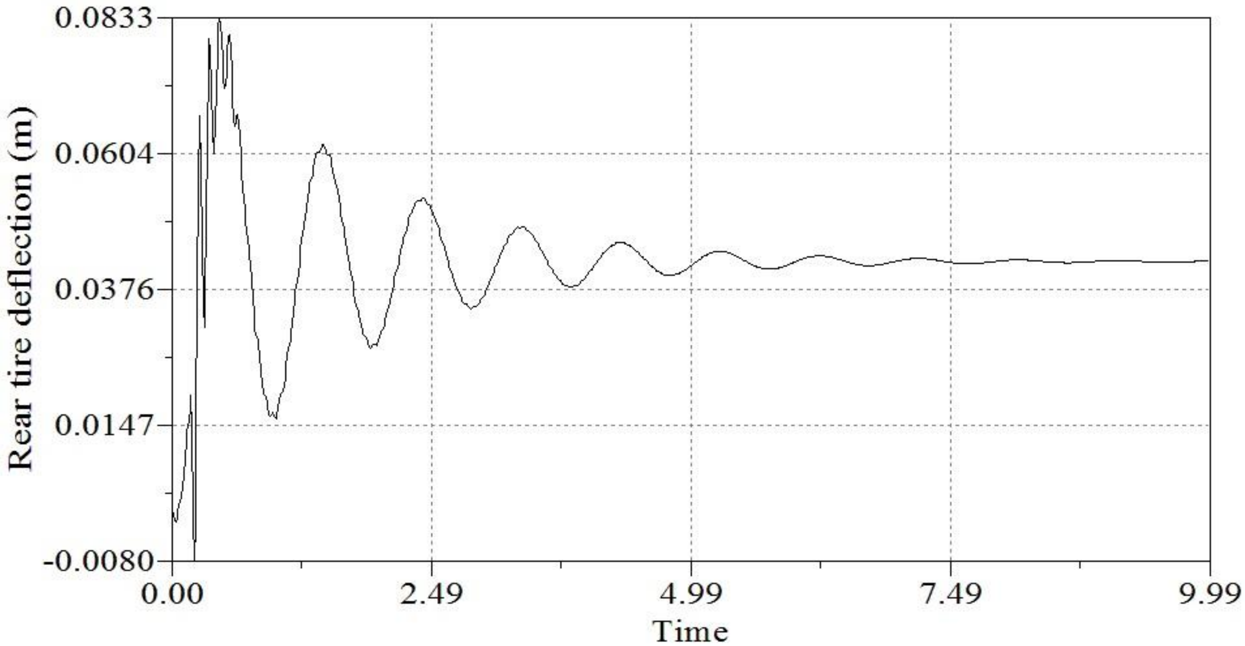


Figure 5.16: Deflection of rear wheel (right) on bump at speed of 60 Km/hr

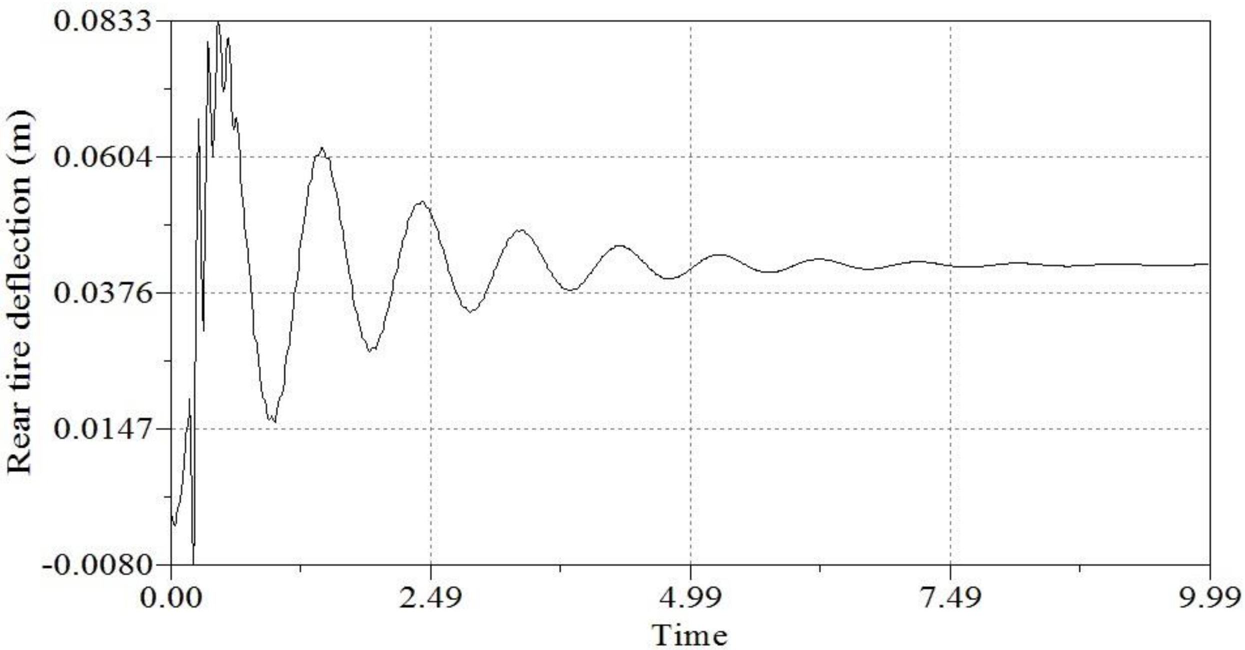


Figure 5.17: Deflection of rear wheel (left) on bump at speed of 60 Km/hr

Figure 5.16 and Figure 5.17 shows the deflection of a car suspension over the bump at a speed of 60 Km/hr for rear right and rear left wheel respectively.

5.7.4 Suspension deflection at speed of 80 Km/hr.

The deflection of suspension system of full car at 80 km/hr for front and rear wheels are shown in Figures 5.18 to 5.21.

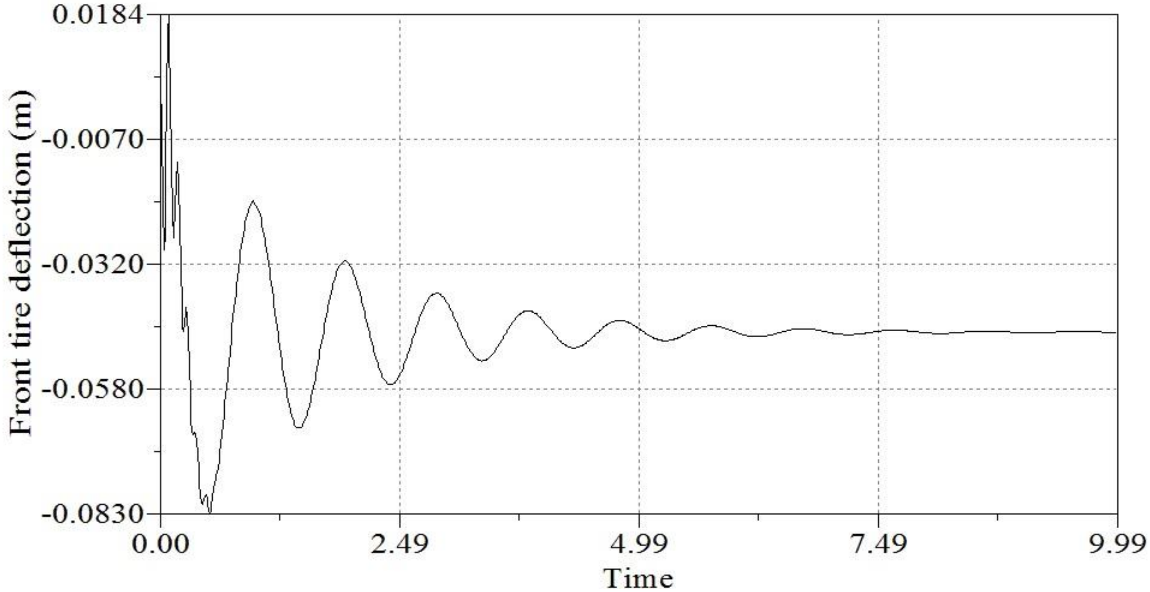


Figure 5.18: Deflection of front wheel (right) on bump at speed of 80 Km/hr

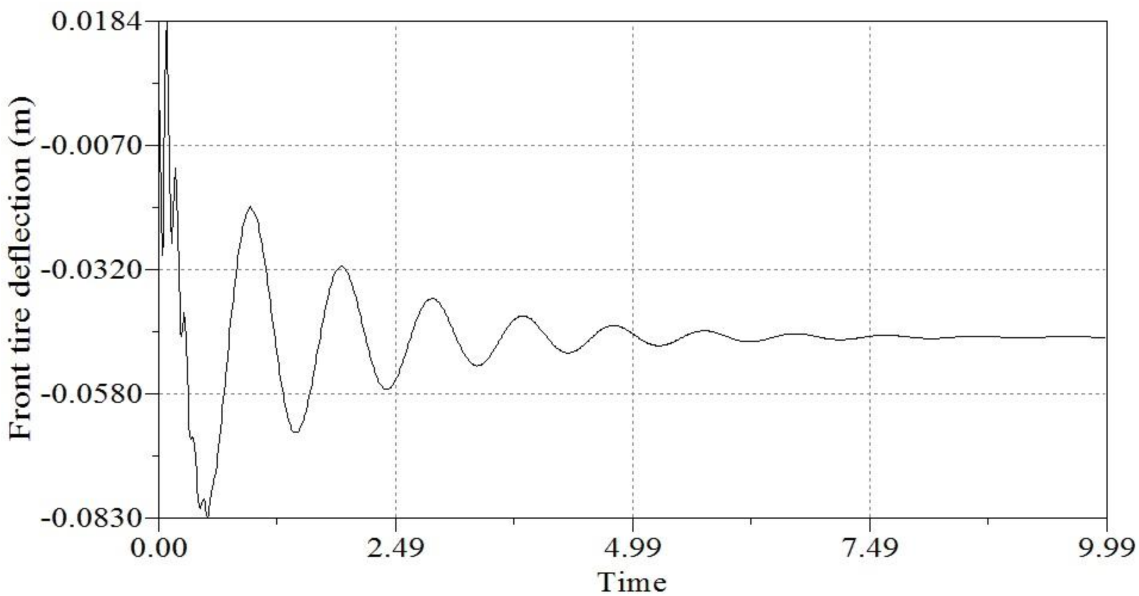


Figure 5.19: Deflection of front wheel (left) on bump at speed of 80 Km/hr

Figure 5.18 and Figure 5.19 show the deflection of a car suspension over the bump at a speed of 80 Km/hr for front right and front left wheel respectively.

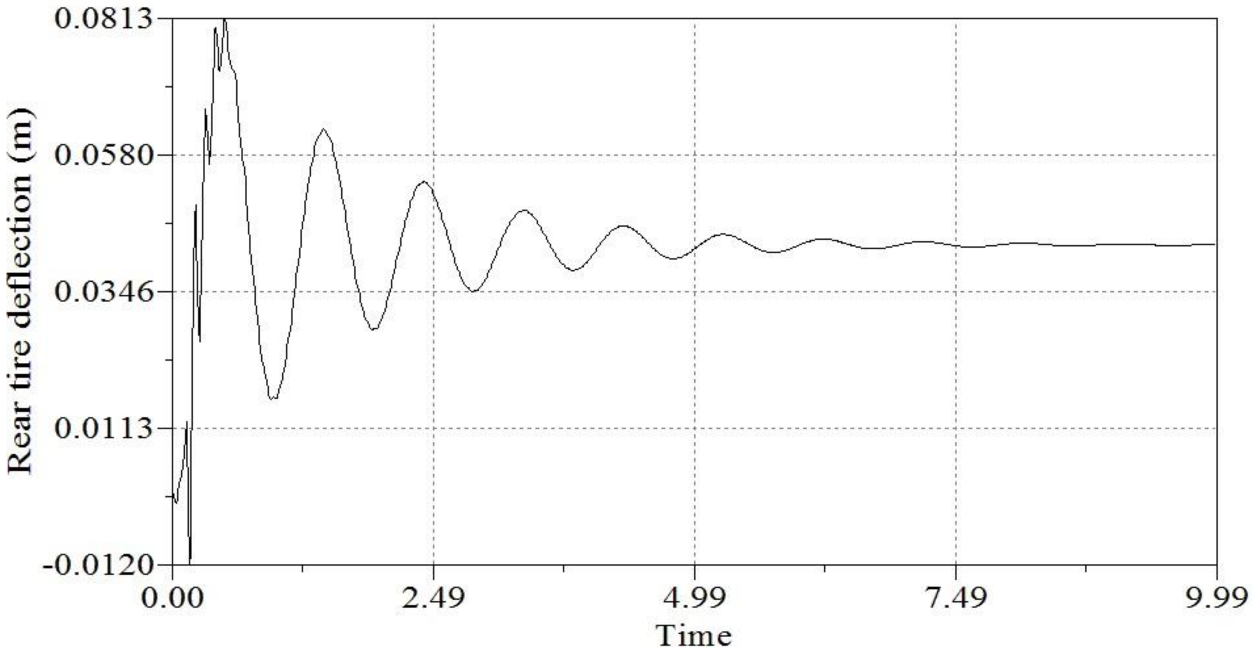


Figure 5.20: Deflection of rear wheel (right) on bump at speed of 80 Km/hr

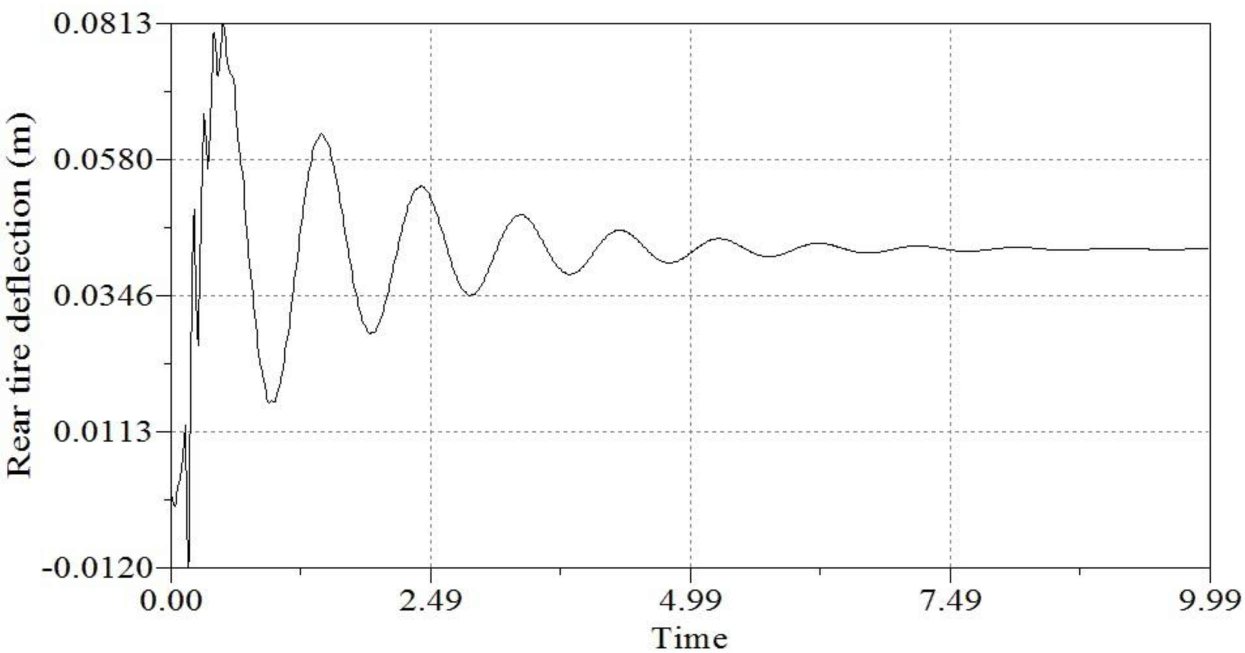


Figure 5.21: Deflection of rear wheel (left) on bump at speed of 80 Km/hr

Figure 5.20 and Figure 5.21 show the deflection of a car suspension over the bump at a speed of 80 Km/hr for rear right and rear left wheel respectively.

5.7.5 Suspension deflection at speed of 100 Km/hr.

The deflection of suspension system of full car at 100 km/hr for front and rear wheels are shown in Figures 5.22 to 5.25

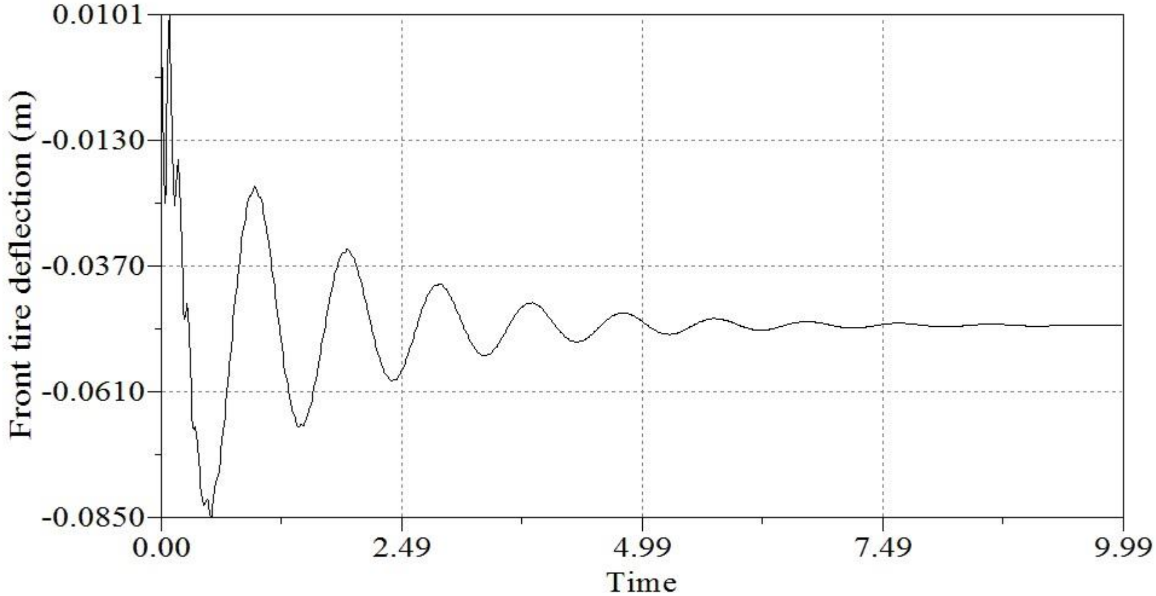


Figure 5.22: Deflection of front wheel (right) on bump at speed of 100 Km/hr

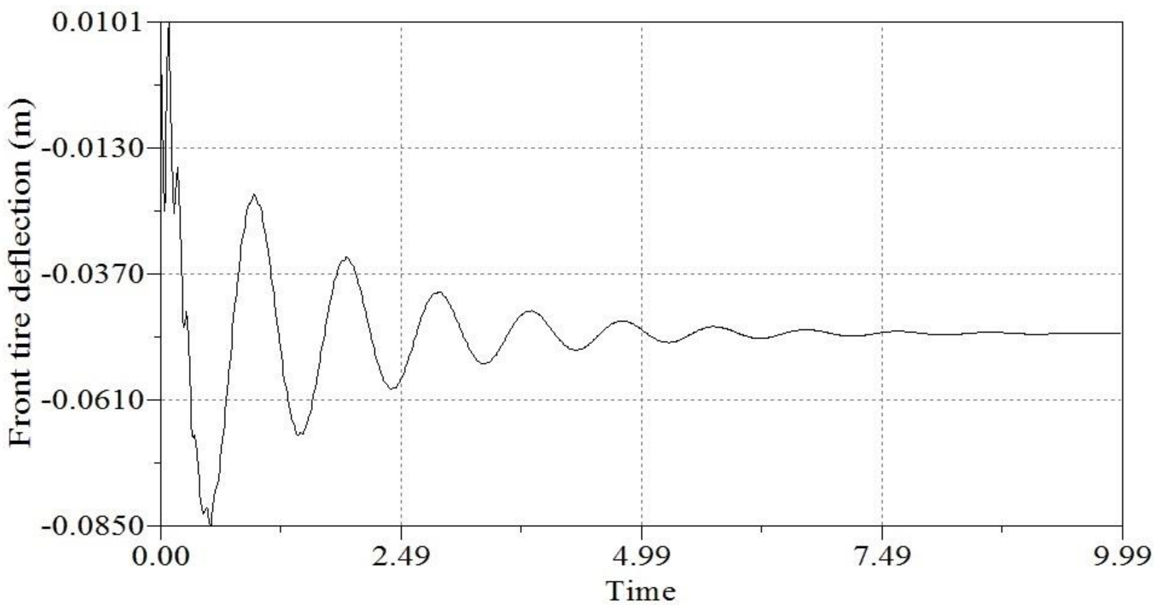


Figure 5.23: Deflection of front wheel (left) on bump at speed of 100 Km/hr

Figure 5.22 and Figure 5.23 show the deflection of a car suspension over the bump at a speed of 100 Km/hr for front right and front left wheel respectively

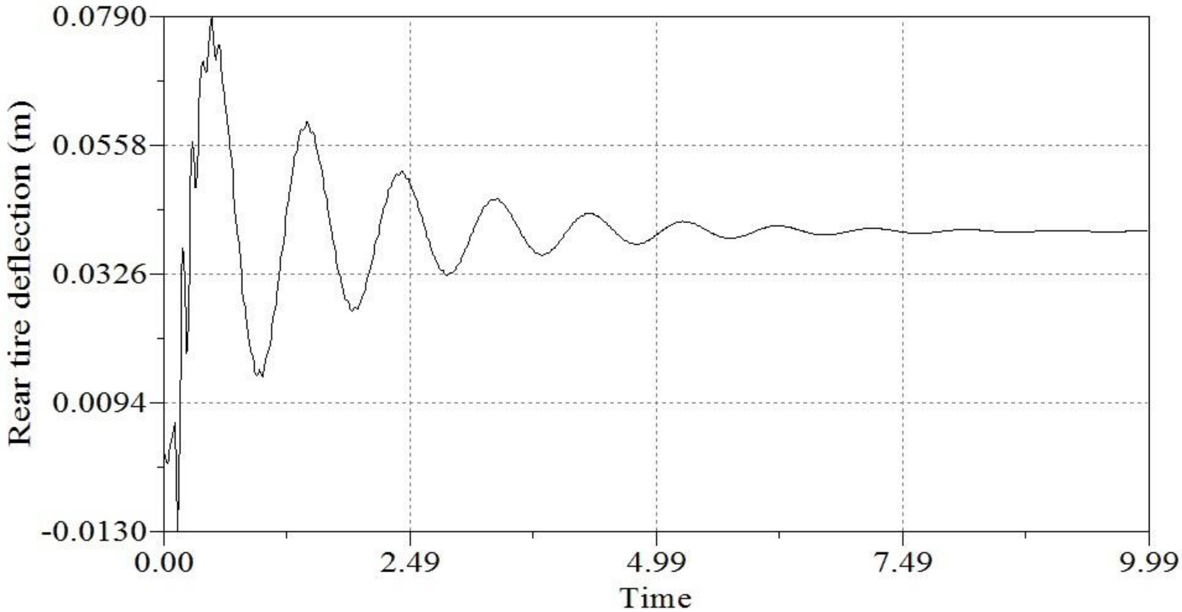


Figure 5.24: Deflection of rear wheel (right) on bump at speed of 100 Km/hr

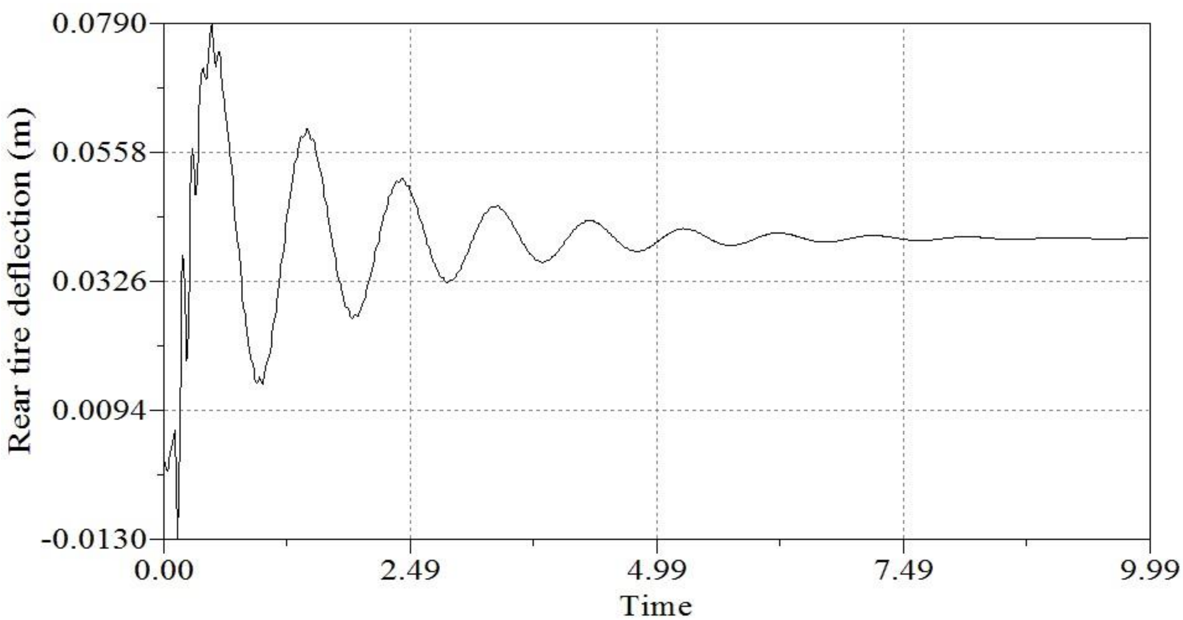


Figure 5.25: Deflection of rear wheel (left) on bump at speed of 100 Km/hr

Figure 5.24 and Figure 5.25 show the deflection of a car suspension over the bump at a speed of 100 Km/hr for rear right and rear left wheel respectively.

5.8 Vertical acceleration of suspension at various speed for a full car model

Vertical acceleration of suspensions on bump at different speeds for full car model are shown in Figures 5.26 to 5.29

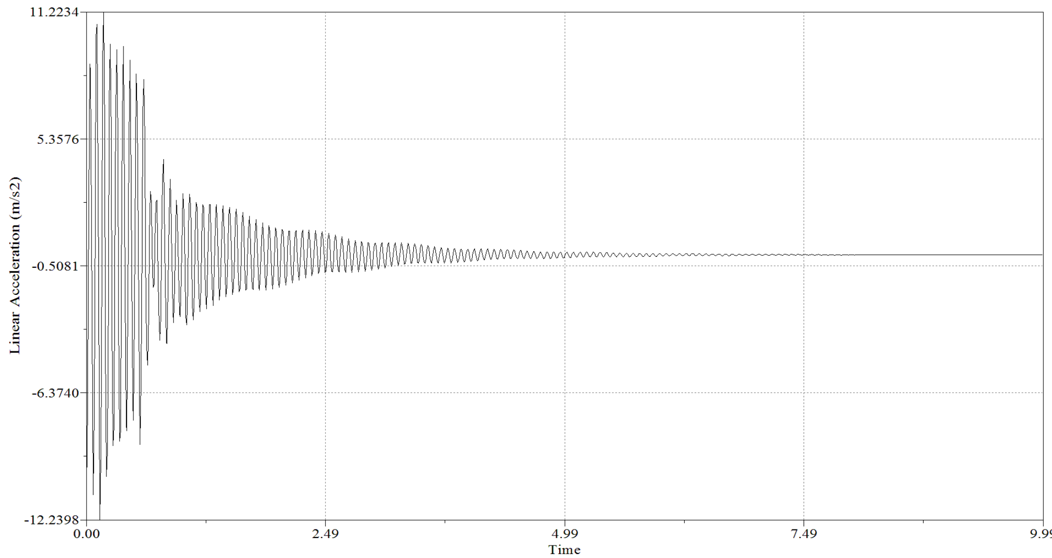


Figure 5.26: Vertical acceleration of a car on bump at speed of 20 Km/hr

Figure 5.26 shows the vertical acceleration of a car suspension when it runs on a bump with a speed of 20 Km/hr. The value of vertical acceleration obtained is 11.2234 m/s^2.

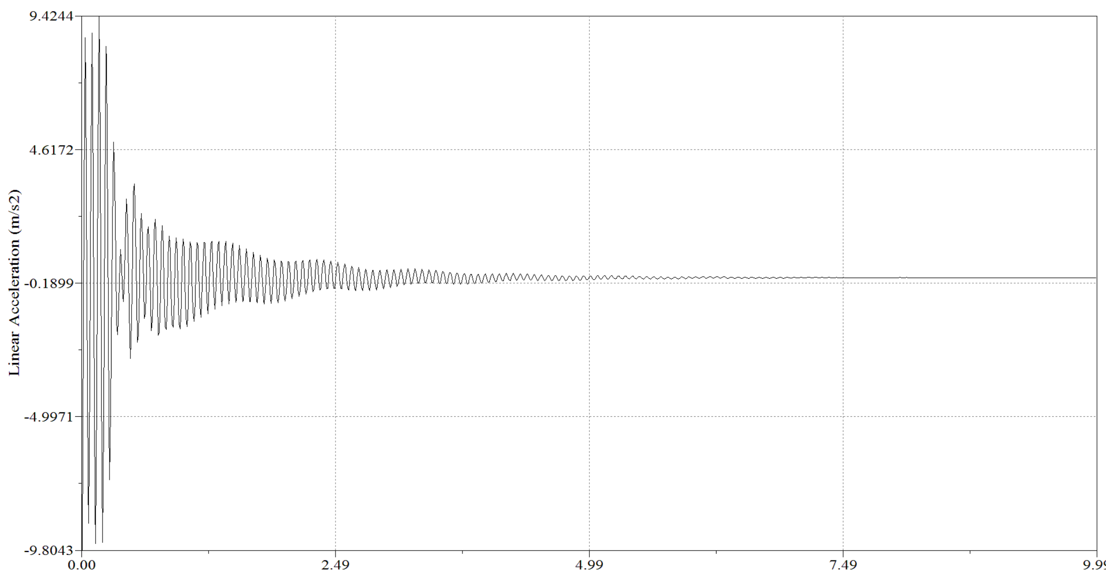


Figure 5.27: Vertical acceleration of a car on bump at speed of 40 Km/hr

Figure 5.27 Show the vertical acceleration of a car suspension when it runs on a bump with a speed of 40 Km/hr. The value of vertical acceleration obtained is 9.4244 m/s^2.

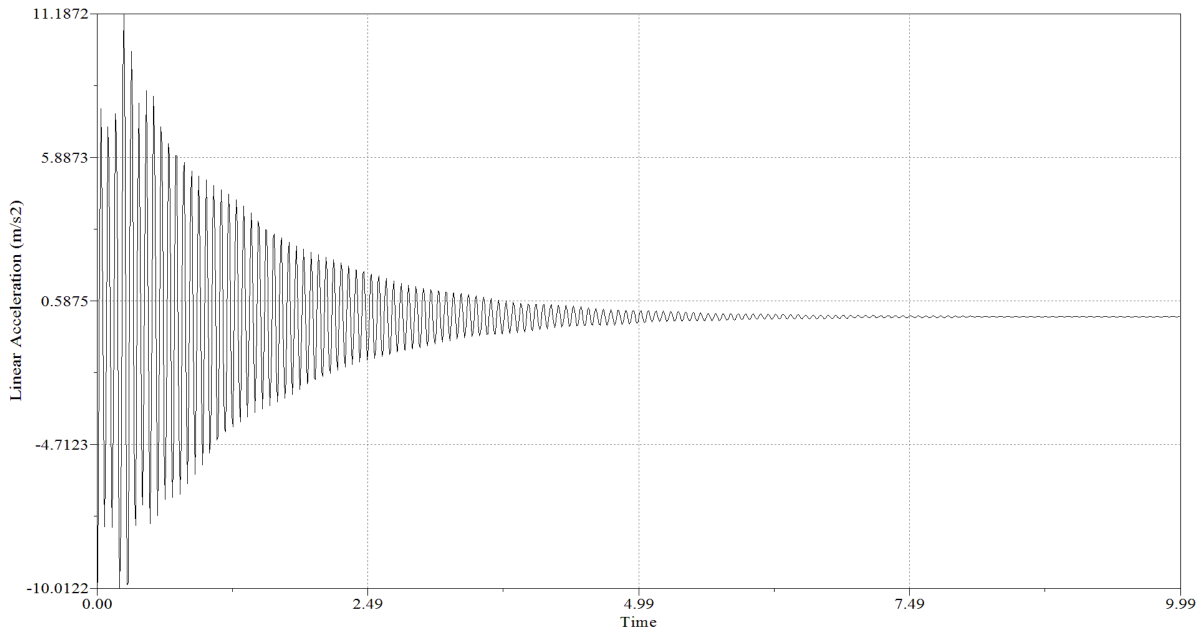


Figure 5.28: Vertical acceleration of a car on bump at speed of 60 Km/hr

Figure 5.28 shows the vertical acceleration of a car suspension when it runs on a bump with a speed of 60 Km/hr. The value of vertical acceleration obtained is 11.1872 m/s^2.

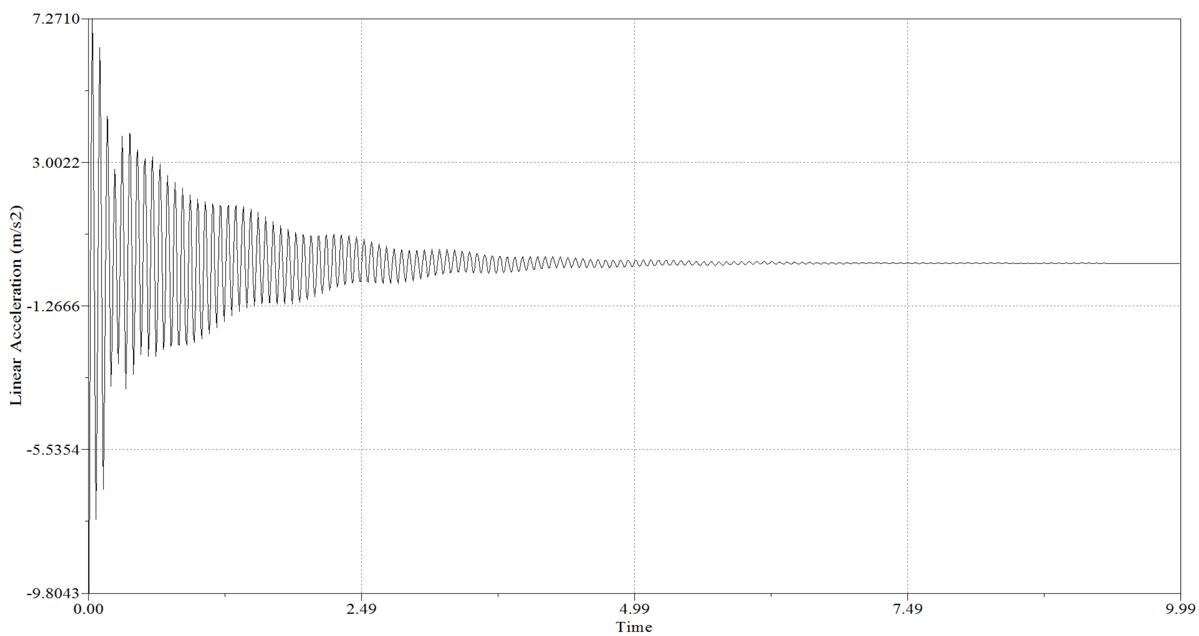


Figure 5.29: Vertical acceleration of car on bump at speed of 80 Km/hr

Figure 5.29 shows the vertical acceleration of a car suspension when it runs on a bump with a speed of 80 Km/hr. The value of vertical acceleration obtained is 7.2710 m/s^2.

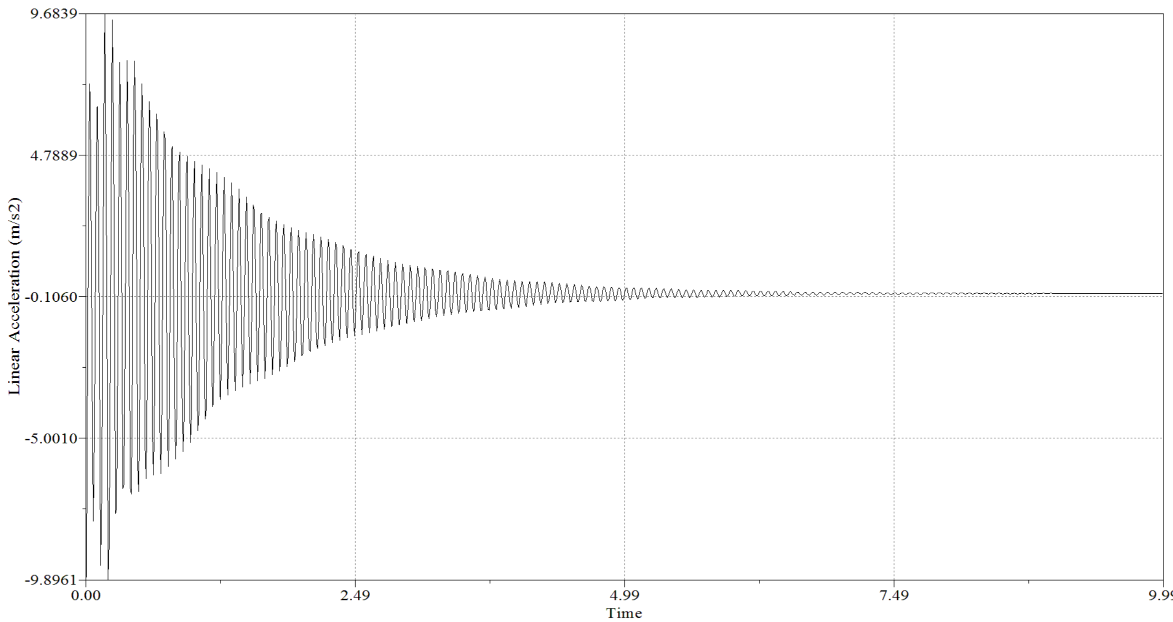


Figure 5.30: Vertical acceleration of car on bump at speed of 100 Km/hr

Figure 5.30 shows the vertical acceleration of a car suspension when it runs on a bump with a speed of 100 Km/hr. The value of vertical acceleration obtained is 9.6839 m/s^2.

The next chapter will present the overall results and discussion.

CHAPTER 6

RESULTS AND DISCUSSIONS

6.1 Introduction

This chapter presents the simulation results of the bond graph model of car on bumps at various speeds, which is further validated with experimental results.

6.2 Vertical acceleration of a quarter car model obtained through experimental analysis.

Maximum vertical acceleration of a quarter car model at various speeds is presented in Table 6.1

Table 6.1: Maximum acceleration at different speed through experimental work

S.no.	Speed (Km/hr)	Maximum Acceleration (m/s^2)
1	20	2.40
2	30	3.34
3	40	3.25
4	50	2.82

The values of the maximum acceleration recorded through experimental setup of OROS at various speed are shown in Figure 6.1

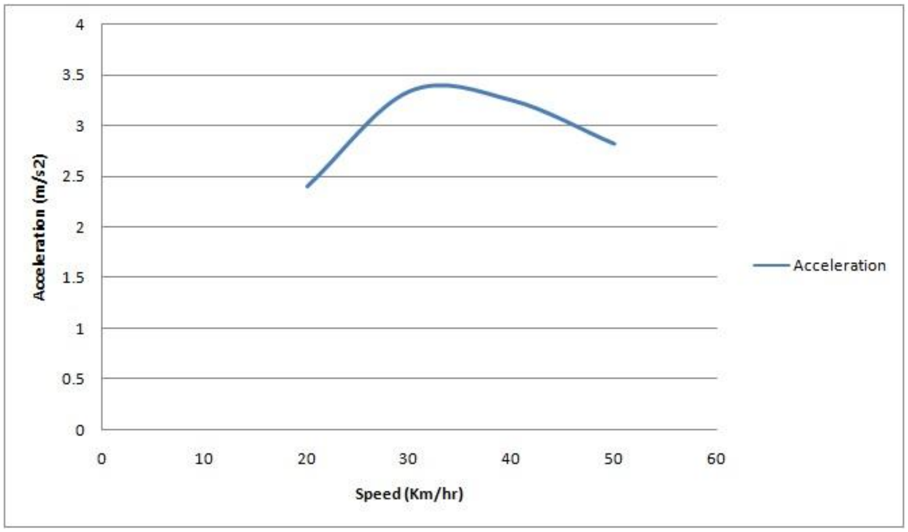


Figure 6.1: Maximum acceleration at various speeds (Experimental)

6.3 Vertical acceleration of a quarter car model obtained through simulation study.

Maximum vertical acceleration of suspension system at various speeds for a quarter car model is shown in Table 6.2

Table 6.2: Maximum acceleration at various speeds through simulation stydy

S.no.	Speed (Km/hr)	Maximum acceleration (m/s^2)
1	20	2.7153
2	30	3.6473
3	40	3.7067
4	50	2.7960

The values of the maximum acceleration received through simulation study at various speeds are presented in Figure 6.2

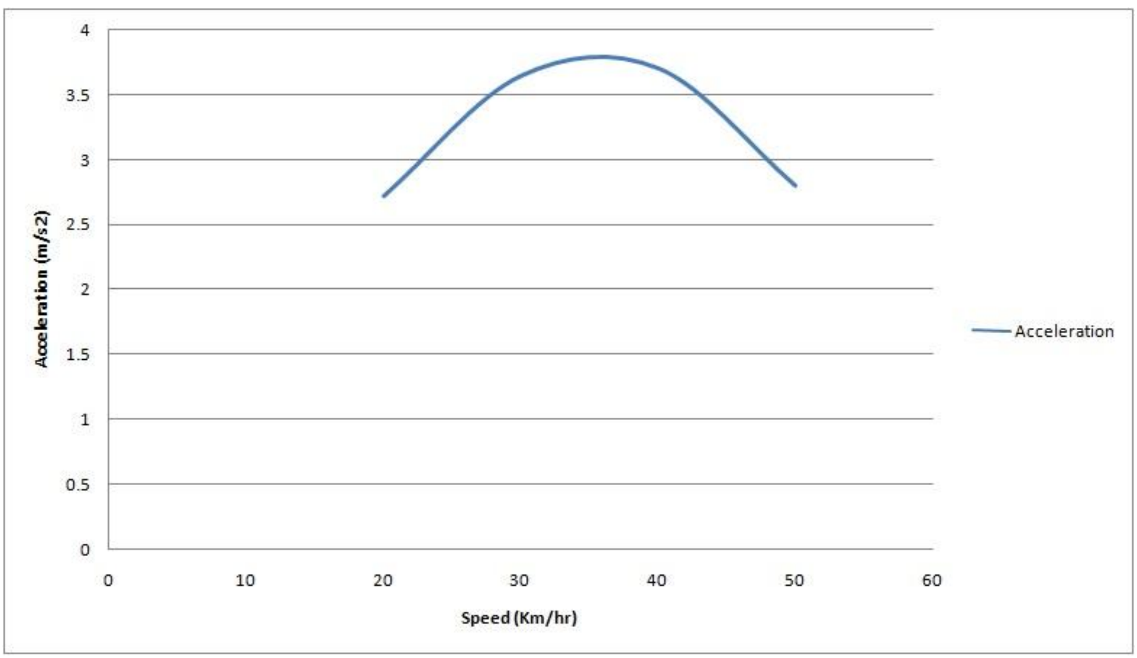


Figure 6.2: Maximum acceleration at various speeds (Simulation).

6.4 Result validation

Experimental data for a quarter car suspension system obtained from OROS 36 is validated with the simulation data obtained simulation study. Table 6.3 gives the maximum acceleration (simulation) and maximum acceleration (experimental) at different speeds.

Table 6.3: Comparing experimental and simulation results

S. no.	Speed (Km/hr)	Maximum acceleration through simulation	Maximum acceleration through experiment
1	20	2.7153	2.40
2	30	3.6473	3.34
3	40	3.7067	3.25
4	50	2.7960	2.82

The experimental results and simulated results for maximum vertical accelerations and speed is presented in Figure 6.3

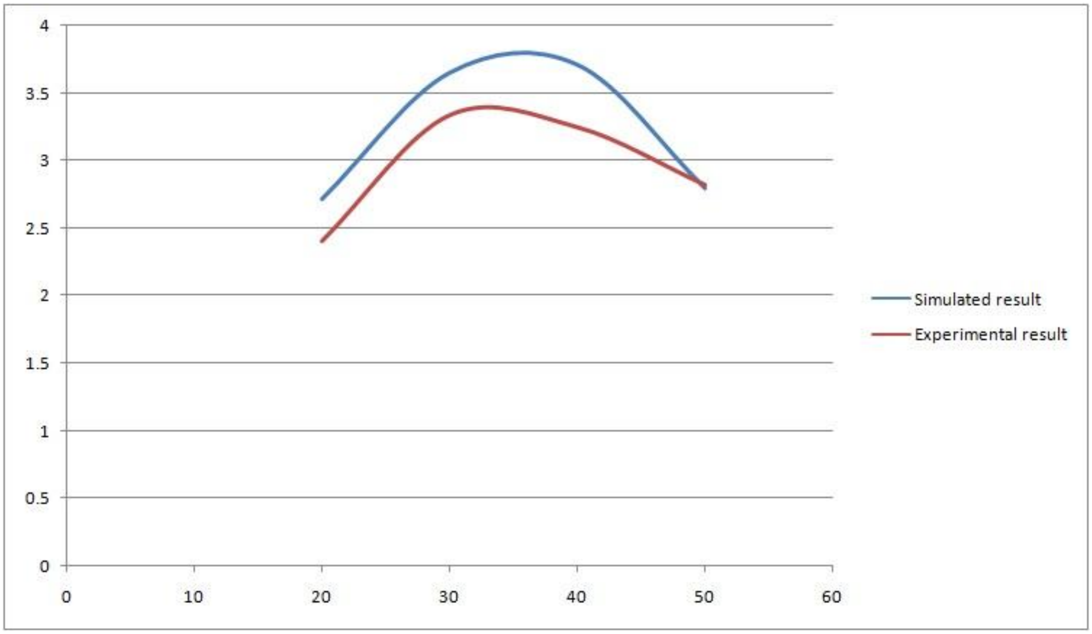


Figure 6.3: Vertical acceleration at various speeds through experiments and simulations

It is evident that the vertical accelerations shows the same behavior pattern at different speeds. Although there is a marginally difference between these values but the reasons lies that during experiments full car model is evaluated rather than quarter car, which ultimately makes difference.

CHAPTER 7

CONCLUSION AND FUTURE SCOPE

7.1 Conclusion

This dissertation work has been attempted to obtain the dynamic behavior of a car through bond graph modeling and also to evaluate various parameters through simulation.

The simulation study of road car model has been validated with experimental study, which was conducted on defined tracks as per Indian road conditions.

The following conclusions are made:

- The dynamic model of car (XUV 500) has been modeled using bond graph modeling.
- Nine degree of freedom model has been analyzed through modeling.
- Simulation results were compared with experimental data which was evaluated through OROS system.

7.2 Future scope

Following future scope has been proposed for further analysis.

- In this model only vertical dynamics has been investigated.
- The lateral and longitudinal dynamics may also be considered in future.
- In present work, experimental study has been successfully accomplished for analyzing a vertical acceleration, which may be further evaluated for a ride comfort of a vehicle.

APPENDIX A

State equations of Quarter car model

Please note: 'd' represents the time derivative of the state variable within the first parenthesis.

d(P9)= -K11*Q11+K6*Q6+R7*(-P9/M9+P3/M3)

d(P3)= -K6*Q6-R7*(-P9/M9+P3/M3)+SE2

d(Q11)= SF10+P9/M9

d(Q6)= -P9/M9+P3/M3

State equations of Full car model

Please note: 'd' represents the time derivative of the state variable within the first parenthesis.

d(P53)= -K63*Q63+K58*Q58+R59*(-P53/M53+P5/M5+m1*P10/M10+m2*P15/M15)

d(P50)= K60*Q60+R61*(P5/M5+m3*P10/M10+m4*P15/M15-P50/M50)-K64*Q64

d(P49)= K54*Q54+R55*(P5/M5+m5*P10/M10+m6*P15/M15-P49/M49)-K69*Q69

d(P46)= R56*(P5/M5+m7*P10/M10+m8*P15/M15-P46/M46)+K57*Q57-K66*Q66

d(P5)= -K36*Q36-K58*Q58-R59*(-P53/M53+P5/M5+m1*P10/M10+m2*P15/M15)-K37*Q37-
K60
Q60-R61(P5/M5+m3*P10/M10+m4*P15/M15-P50/M50)-K36*Q36-K54*Q54-
R55*(P5/M5
+m5*P10/M10+m6*P15/M15-P49/M49)-K37*Q37-
R56*(P5/M5+m7*P10/M10+m8*P15/M15
-P46/M46)-K57*Q57+SE179

d(P10)= -m1*(K36*Q36+K58*Q58+R59*(-P53/M53+P5/M5+m1*P10/M10+m2*P15/M15))-
m3*(K37
*Q37+K60*Q60+R61*(P5/M5+m3*P10/M10+m4*P15/M15-P50/M50))-
m5*(K36*Q36+K54
Q54+R55(P5/M5+m5*P10/M10+m6*P15/M15-P49/M49))-

m7*(K37*Q37+R56*(P5/M5+m7

*P10/M10+m8*P15/M15-P46/M46)+K57*Q57)

d(P15)= -m2*(K36*Q36+K58*Q58+R59*(-P53/M53+P5/M5+m1*P10/M10+m2*P15/M15))-

m6*(K36

*Q36+K54*Q54+R55*(P5/M5+m5*P10/M10+m6*P15/M15-P49/M49))-

m8*(K37*Q37+R56

*(P5/M5+m7*P10/M10+m8*P15/M15-P46/M46)+K57*Q57)-

m4*(K37*Q37+K60*Q60+R61

*(P5/M5+m3*P10/M10+m4*P15/M15-P50/M50))

d(Q60)= P5/M5+m3*P10/M10+m4*P15/M15-P50/M50

d(Q58)= -P53/M53+P5/M5+m1*P10/M10+m2*P15/M15

d(Q36)= P5/M5+m1*P10/M10+m2*P15/M15+P5/M5+m5*P10/M10+m6*P15/M15

d(Q37)= P5/M5+m7*P10/M10+m8*P15/M15+P5/M5+m3*P10/M10+m4*P15/M15

d(Q57)= P5/M5+m7*P10/M10+m8*P15/M15-P46/M46

d(Q54)= P5/M5+m5*P10/M10+m6*P15/M15-P49/M49

d(Q69)= P49/M49+SF68

d(Q66)= P46/M46+SF67

d(Q64)= P50/M50+SF65

d(Q63)= SF62+P53/M53

APPENDIX B

Various expression used in a quarter car modeling

Variable	Assignment
K_sus (double)	
K6 (double)	K6=K_sus;
R_sus (double)	
R7 (double)	R7=R_sus;
L (double)	
V (double)	
PI (double)	PI=3.1427;
H (double)	
SF10 (double)	SF10=-PI*H*V/L*cos(PI*V/L*t)*swi(t,0)*swi(L/V,t);
Mass_vehicle (double)	
M (double)	M=Mass_vehicle;
G (double)	G=9.81827;
SE2 (double)	SE2=-M*G;
M3 (double)	M3=M;
M_tire (double)	
M9 (double)	M9=M_tire;
K_tire (double)	
K11 (double)	K11=K_tire;

Various expression used in a full car modeling

Variable	Assignment
m1 (double)	m1=a;
a (double)	
m3 (double)	m3=a;
m5 (double)	m5=-b;
b (double)	
m7 (double)	m7=-b;
m2 (double)	m2=w/2;
m6 (double)	m6=w/2;
w (double)	
m4 (double)	m4=-w/2;
m8 (double)	m8=-w/2;
M15 (double)	M15=Jwr;
Jwr (double)	
Jwp (double)	
M10 (double)	M10=Jwp;
Mass_of_base_frame (double)	
M5 (double)	M5=Mass_of_base_frame;
Mtv3 (double)	
M46 (double)	M46=Mtv3;
Mtv4 (double)	
M49 (double)	M49=Mtv4;
Mtv1 (double)	

Variable	Assignment
M50 (double)	M50=Mtv1;
Mtv2 (double)	
M53 (double)	M53=Mtv2;
Kt (double) : tire stiffness	
K63 (double)	K63=Kt;
K64 (double)	K64=Kt;
K66 (double)	K66=Kt;
K69 (double)	K69=Kt;
Ks (double)	
K54 (double)	K54=Ks;
K57 (double)	K57=Ks;
Kabr (double)	
K37 (double)	K37=Kabr;
K36 (double)	K36=Kabr;
L (double)	L=Length_of_bump;
g (double)	g=9.81827;
PI (double)	PI=3.1421;
H (double)	H=Height;
u (double)	u=a+b;
Vr (double)	Vr=-PI*H*V/L*cos(PI*V/L*(t-u/V))*swi(t,u/V)*swi((u+L)/V,t);
V4 (double)	V4=Vr;
SF68 (double)	SF68=V4;
V3 (double)	V3=Vr;
SF67 (double)	SF67=V3;

Variable	Assignment
Height (double)	
V (double)	
Length_of_bump (double)	
Vf (double)	Vf=-PI*H*V/L*cos(PI*V/L*t)*swi(t,0)*swi(L/V,t);
V1 (double)	V1=Vf;
SF65 (double)	SF65=V1;
V2 (double)	V2=Vf;
SF62 (double)	SF62=V2;
K58 (double)	K58=Ks;
K60 (double)	K60=Ks;
Rs (double)	
R59 (double)	R59=Rs;
R61 (double)	R61=Rs;
R56 (double)	R56=Rs;
R55 (double)	R55=Rs;
SE179 (double)	SE179=-Mass_of_base_frame*g;

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