

# A Novel Semi-active Suspension System for Automobiles Using Jerk-Driven Damper (JDD)

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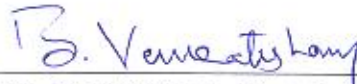
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# Approval Sheet

This thesis titled "A Novel Semi-active Suspension for Automobiles Using Jerk-Driven Damper (JDD)" by Mr. Chandrakant Kumar Nirala is approved for the degree of Master of Technology.



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**Chandrakant Kumar Nirala**

*To*  
*The God and My Loving Parents*

## Abstract

With the new advancements in the vibration control, the control strategies for the controllable semi-active dampers are finding their way as an essential part of vibration isolators, particularly in vehicle suspension systems. An analysis of frequency response for single degree of freedom (1DOF) system gives an attribute to the fact that in a semi-active suspension system, the damping coefficients can be adjusted to improve ride comfort and road handling performances.

The systems study includes various type of semi-active suspension systems, employing nonlinear magnetorheological(MR) dampers that are controlled to provide improved vibration isolation. The currently available control strategies for semi-active dampers can be divided into two main groups. The first one is 'On-Off' control and second one is 'continuous' control of variable dampers. Available control strategies are either proportional to the relative velocity of sprung mass or the acceleration of sprung mass. A new control strategy which is proportional to the jerk produced in sprung mass called Jerk Driven Damper (JDD) is proposed and analyzed by the use of two state 'On-Off' damper. The control strategy for 'JDD' system is extremely simple and it involves very common logic. 'JDD' system requires a two state controllable damper and jerk sensor. A brief study on controllable damper and jerk sensors are presented in this thesis.

Two types of positive amplitude half sinusoidal speed breakers (severe and smooth) with same height are considered as input to the vehicle. The better performance of JDD is examined over SH and 'ADD' control which is observed with both 'severe' and 'smooth' speed breakers.

Later, the acceleration response (comfort objective) of JDD control strategy is compared with various types of well-studied control strategies and shown that it has better isolation. The work ends with the suggestion of future scope as combination of two or more semi-active suspension systems.

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# Chapter 1

## Introduction

Suspension system is a mechanical system comprising of either a spring or a spring and damper system connecting the wheels and axles to the chassis of a vehicle. The mass supported over the suspension system is called sprung mass. A suspension system is one of the important components of a vehicle, which plays a crucial role in handling performance and the ride comfort characteristics of a vehicle. The fundamental role of a vehicle suspension system is to isolate the vehicle body from the force transmitted by the external excitation. It is well known that the suspension system performs multiple tasks such as maintaining contact between vehicle tires and the road, addressing the stability of the vehicle, and isolating the frame of the vehicle from road-induced vibration and shocks. With the development of mechanical and electronics technology, the requirements of ride comfort and driving performance have been major development objectives of modern vehicles to satisfy the expectations of passengers. Hence, the design of an appropriate suspension system is always an important research topic for achieving the desired ride quality.

### 1.1 Motivation

It can be seen from the frequency response of a 1DOF system explained in chapter 3, a constant damping as used in passive suspension system can not fulfill the requirement of maximum isolation at all excitation frequencies. A passive suspension system has no means of adding external energy to the system because it contains only passive elements such as a damper and a spring. Therefore, its damping force cannot be varied by external signal. To avoid this disadvantage of passive suspension system, active suspension system is introduced. An active suspension system has a means of adding external energy to the system by the

use of actuator. But, adding external energy requires actuator and controllers and these makes the suspension system more complex. As discussed above, controlling the active suspension system is not at all easy, semi-active suspension system is introduced. Though, in a semi-active suspension there is no force actuator as in active suspension, it is still possible to continuously vary the rate of energy dissipation using a controllable damper, but it is not possible to add energy. In the semi-active suspension system it is also possible to change the stiffness of the spring, but conventional implementation of variable stiffness device is complicated. On other hand, the variable damping can easily be produced by a controllable damper, such as a fluid damper with variable orifices or a magnetorheological (MR) damper [1]. So, by manipulating the damping force of a controllable damper, we can get the required damping. Semi-active suspension systems have shown a significant improvement over the passive systems. Due to this fact, semi-active dampers have been designed and made commercially available; the control strategies have been adopted and implemented to offer superior ride quality to passenger vehicles. However, the technology is still an emerging one, an elaboration and more research work on different theoretical and practical aspects are required. This thesis is an attempt to develop an understanding of some of those aspects, such as the effect of the semi-active dampers response-time on the performance of the control strategies through analytical and numerical methods. On the other hand, the technology has not yet been adopted for heavy vehicles. This attributes to two reasons: firstly, the un-availability of semi-active actuators (dampers) suitable for a particular requirement of the heavy and off-road vehicles; secondly, lack of interest in the manufacturing sector, given that the superior advantages of such systems.

## **1.2 Suspension Systems and Suspension Control**

### **1.2.1 Passive Suspension System**

A passive suspension system is the simplest way to protect the vehicle from vibrations caused by road disturbances. It consists of an energy dissipating element, which is the damper, and an energy-storing element, which is the spring. Since these two elements can not add energy to the system this kind of suspension systems are called passive. Passive suspension systems are subject to various trade-offs when they are excited across a large frequency range. The trade-offs associated with the passive suspension systems can easily be understand by the frequency response of a 1DOF system spring mass damper system. Requirement of a higher damping coefficient at a lower excitation frequency and a very low damping coefficient at a

higher excitation frequency, motivates us to vary the damping coefficient according to the excitation frequency. The study of frequency for 1DOF system and 2DOF suspension system is presented in chapter 3.

### **1.2.2 Active Suspension System**

The suspension system must support the vehicle, provide directional control during handling and provide effective isolation of passengers/payload from road disturbances [2]. It can be seen from the frequency-response curve that a good ride comfort requires a higher damping at lower excitation frequency, whereas the higher excitation frequency requires low a damped suspension. Good handling requires a suspension setting somewhere between the two. A passive suspension system has the ability to store energy via a spring and to dissipate it via a damper. Its parameters are generally fixed, being chosen to achieve a certain level of compromise between road holding, load carrying and comfort. To withstand with the demanded damping for the suspension system it is required to store, to dissipate and to introduce energy to the system. This is how the active suspension system can be defined. Active suspension system may vary its parameters depending upon operating conditions where the passive system is limited to.

### **1.2.3 Semi-Active Suspension System**

Semi-active suspension system is introduced by ‘Karnopp’ in the early 1970s [3]. Semi-active suspension system is most often been studied and used in vehicle. The semi-active suspension system is meant to reduce the vibration transmitted to the vehicle body from the axle in a vehicle by adjusting its damping [3]-[5]. This adjustment of the damping coefficient may be termed as closed-loop regulation of the semi-active suspension [Anh Lam]. This adjustment of damping to its maximum and minimum value and in between maximum and minimum value can be termed as tuning of variable damper. The vibration of the base excited by the road disturbance and the sprung mass are measured and fed into a controller, which tunes the damping coefficient such that the damping force, which is proportional to the relative velocity can be varied as a function of time. This is how semi-active damper works.

semi-active dampers are may be of the ‘On-Off’ type or the continuous variable type. A damper of ‘On-Off’ type is switched between ‘On’ and ‘Off’ damping states according to the control algorithm. In the ‘On’ state of damper the damping coefficient is relatively high and equal to its maximum value, and in ‘Off’ state, it is relatively low and equal to its minimum

value. Ideally the 'Off' state damper should give zero damping, but in practical situations it is not possible. So, the minimum value of damper coefficient is chosen for its 'Off' state. A continuous variable semi-active damper is switched between its maximum and minimum value of damping. However, in 'On' state the damper coefficient and corresponding damping force are varied and it is relatively high as compare to 'Off' state 'continuous' variable damper. The concept of 'On-Off' two state damper and 'continuous' variable damper are illustrated in Fig. 1.1 [6].

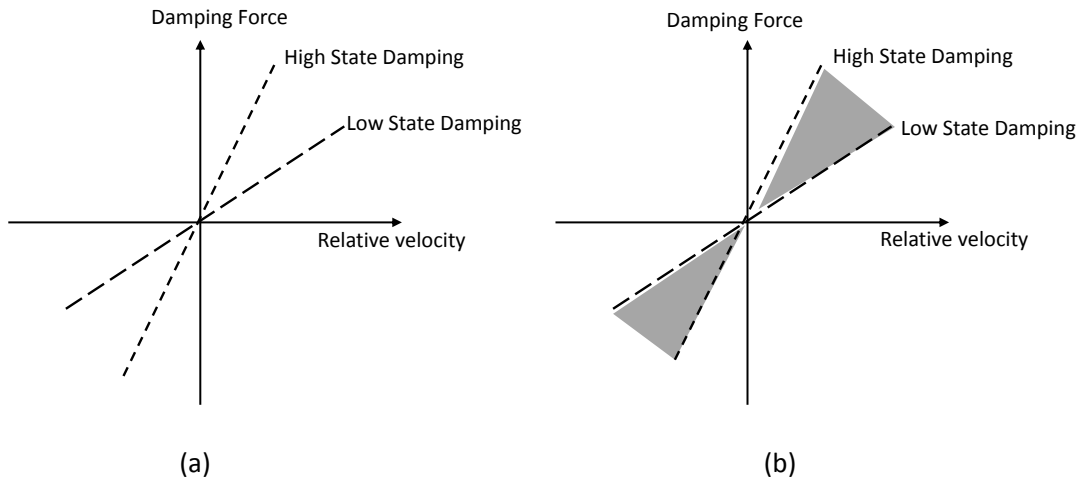


Figure 1.1: Semi-active damper concepts (a) 'On-Off' damper and (b) 'Continuous' variable damper (the shaded part in (b) represents the range of the achievable damping coefficients)

### 1.3 Semi-Active Dampers

Dampers are an integral part of any suspension system. They are also the least understood and most confusing part of the suspension. The main function of the dampers is to control the transient behavior of the sprung and unsprung masses of the vehicle. This is accomplished by damping the energy stored in the springs from suspension movement. The damper generates a force which is defined by the piston velocity of the damper. Dampers are also called shock absorbers.

Semi-active dampers are electro-mechanical control devices which have the capability to vary the damping coefficient to its maximum and minimum value and in between this two values. This variation in damping coefficient results in variation of dissipated energy

with supplying a small amount of power. The semi-active dampers can be categorized into three types [1]. (a) Conventional semi-active damper (b) Electro-rheological (ER) and (c) Magneto-rheological (MR) damper.

The common function of a conventional semi-active damper is the variation of bypass cross-section area that connects the two chambers of the damper's piston obtaining multiple performance curve from a single damper.

### **1.3.1 Electro-rheological and Magneto-rheological Dampers**

'ER' and 'MR' are basically a property of damper fluid which alters its property by changing viscosity after supplying an electric or magnetic field to the damper. The advantage of 'ER' damper over conventional semi-active damper is its fast response to an electric field and hence a wide control bandwidth [7]. However, 'MR' damper became more popular because of its high yield strength over a wide temperature range 'MR' fluid is a non-Newtonian fluid that changes its properties in the presence of a magnetic field. 'MR' fluid contains micron-size iron particles suspended in a carrier fluid (water, petroleum-based oil, or silicon-based oil) align in chain-like structures along the flux lines. Supply of magnetic field causes change in the density of fluid as well as rheological property of fluid [8]. 'MR' fluid can react within 1 ms which is five times faster than 'ER' damper fluid [9]. In order to achieve a damping force required by a semi-active 'ER' damper, a high electric field about 5kV/mm and a high voltage up to 6kv has to be applied. Whereas, 250kA/m magnetic field with 2-25V power is required to achieve the required damping with MR damper [10].

## **1.4 Objective of the Work**

The primary objective of the present work is to realize the virtual semi active suspension control strategies such as 'Sky-hook'(SH) and 'Ground-hook'(GH) in more sophisticated way. The detailed study of an existing semi-active suspensions control gives us an idea to think of other possible control logic. The main objective of this thesis is, to propose an improved control method, called jerk driven damper (JDD) for a semi-active suspension system on the basis 'On-Off' control and 'Continuous' control of a variable damper. To achieve the work objectives, this thesis makes effective use of different analysis methods available as control logics. The road disturbance (input to the vehicle) is considered as a very common half sinusoidal speed breaker with positive amplitude. The simulation for the time-response of sprung mass acceleration has been carried out with two different types of road inputs.



The road inputs are chosen and modeled on its practical existence in residential areas. The performance of commonly existing suspension systems has been seen with the predicted road inputs and then the result is compared with proposed ‘JDD’ suspension. Later the work has been extended up to the comparison of the performance of different control strategies on the basis of ‘comfort objective’ (the comfort objective of a suspension system is to have minimum vertical acceleration of the sprung mass). At the end, the trade-offs related to JDD suspensions are discussed.

## 1.5 Working Methodology

This work will first discuss the trade-offs associated with a typical passive 1DOF(degree of freedom) base excited suspension systems. The frequency response of this 1DOF system will be discussed and in addition, disadvantage of a passive suspension and motivation towards semi-active suspension system is also discussed. The conceptual basis for control strategies of existing semi-active suspension system is studied and simulated. The computer simulation for nonlinear vibration problem is done in MATLAB and it is simulated in such a way that its practical implementation is possible. Optimality of the ‘SH’ and ‘ADD’ semi-active suspension is noticed over other well studied suspension systems and the result makes us to think of other possible control law. The logic behind the all three control strategies is explained in the report such that its practical realization is possible. Later, a control logic based on sprung mass jerk and relative acceleration of sprung mass is proposed and simulated for time response of sprung mass acceleration and vertical displacement of sprung mass. The proposed semi-active suspension system is explained in the easiest way to avoid the complexity. Thesis will conclude with a discussion and comparison of the performance of all existing strategies with the proposed control strategy on the basis of comfort objective. The road input which is a disturbance to the vehicle is predicted and modeled in such a way that it can be realized. All the simulation has been done for two type of road disturbances, severe speed breaker and smooth speed breaker with standard dimension.

## 1.6 Thesis Outline

To fulfill the proposed approaches, the thesis has been structured as follows.

**Chapter 2** produces comprehensive view of the previous research on the topic. In this section a well-structured literature review is characterized by a logical flow of ideas. The

ultimate goal is to bring the reader up to date with current literature on a topic and forms the basis for another goal, such as future research that may be needed in the area. The section contains a talk about previous research which happened in the area of control suspension after passive suspension. The overview of the research which has been carried out up to the time on the nonlinear semi-active suspension system are discussed. Discussion and comments are provided based on the reviewed theoretical and simulation results. The review is done in such a way that the study of previous literatures can provide a basis for the new idea.

**Chapter 3** proposes an improved quarter-vehicle passive suspension model and its equivalent of semi-active quarter-vehicle model by replacing the passive component by semi-active component. The modeling of the semi-active suspension system is done with more realistic road disturbance. The road profile which is the cause of disturbance to the vehicle is modeled in such a way that its practical existence is possible. Later, this chapter gives a detail study of frequency response of 1DOF and 2DOF system and trade-offs associated with the passive suspension systems. The main objective of this chapter is to give the motivation towards the use of variable damper in a suspension system.

**Chapter 4** presents a study of all important well existing semi-active suspension system control strategies. The previously studied semi-active suspension systems are simulated with the proposed road disturbance, and suspension performance in the accordance of comfort objective is noticed. The computer simulations of semi-active suspension for all the control strategies are done in MATLAB.

**Chapter 5** is the core work of the thesis. Continuous control of ‘JDD’ suspension system is proposed and its optimality over two state ‘On-Off’ ‘JDD’ control is noticed. The proposed semi-active suspension system ‘JDD’ is introduced and its simulation has made in such a way that its physics can easily be understood.

**Chapter 6** and **Chapter 7** summarized the work carried out in the thesis and optimality of the ‘JDD’ semi-active suspension is shown on the basis of vertical acceleration of the sprung mass. Chapter ends with the section of practical implementation of the ‘JDD’ semi-active suspension.

# Chapter 2

## Literature Survey

As it has been said in the chapter 1 that a suspension system is one of the important components of a vehicle, which plays a crucial role in handling performance and the ride comfort characteristics of a vehicle. The handling performance is basically related to maintaining contact between the road and the tyres to provide guidance along the track and ride comfort means isolate the vehicle body with its passengers from external disturbance inputs which mainly comes from irregular road surfaces. Passive suspension systems are able to isolate the vehicle body from the road disturbance but some trade-offs are always with this system. To come across these trade-offs a lot of controlled suspension systems are studied in last two decay. Detailed classification can be seen in [11].

### 2.1 Background

The study of ‘balance logic’, ‘sky-hook’, ‘ground-hook’, ‘ADD’ suspensions ‘MR’ and ‘ER’ dampers, are the background of this thesis. The work is carried out after a detailed study of all important existing suspension system and its two state ‘On-Off’ control. In this section the literature review is presented on the basis of key words (variable damper, control strategies etc) used in the semi-active suspension system. The majority of the studies in the area of semi-active suspensions used a two-degree-of freedom (2DOF) model representing single suspensions [12]-[21]. There were a few studies in the area of one-degree-of-freedom (1DOF) systems [22]-[24]. However, semi-active suspension system and the function of its variable component i.e damper and fixed component i.e spring can easily be understand by 1DOF system. The simplest control strategy in semi-active suspension system is ‘balance logic’. The semi-active suspension system based on ‘balance logic’ is basically cancelation of

energy stored by spring [25]. It is a simple logic and it can be observe by Eqn. 2.1

$$m\ddot{x}(t) = -b(\dot{x}(t) - \dot{u}(t)) - k(x(t) - u(t)) \quad (2.1)$$

Eqn. 2.1 is equation of motion(EOM) of a 1DOF base excited system. Here,  $m$  is the sprung mass of the suspension whereas, ‘ $x$ ’ and ‘ $u$ ’ are the vertical displacement of sprung mass and base respectively.  $b$  and  $k$  are the damping coefficient of damper and stiffness of the suspension respectively. For semi-active suspension the damping force in the right hand side of Eqn. 2.1 is variable and it should be varied in such a way that it can cancel the spring force  $k(x(t) - u(t))$  [25].

An excellent review of many of the past efforts in the area of semi-active suspension design is carried out in [26]. He also presents a very good background of the information that is required to understand semi-active suspension systems. Finally, he discusses several semi-active suspension applications. A different sky-hook method on the basis of ‘On-Off’ control strategies in order to reduce both the tire force and body acceleration of a heavy truck is presented in [27]. This is a modified ‘sky-hook’ control with 2DOF system and they compare the results of their mathematical simulations to their experimental testing using a hardware in-the-loop test method. They conclude that, compared to passive suspensions, the full state feedback methods works the best in reducing tire loads and body acceleration according to simulation and experimental results. However, along with the ‘On-Off’ control of semi-active suspensions a ‘Continuous control’ has been introduced and proven a better performance in the perspective of comfort objective [28]. The road input which is the cause of disturbance to the vehicle is considered as random profile. Widely used road profile for the simulation is the response of a first-order filter to white noise [29]. Finally, A practical realization of a fully active sky-hook suspension system on a passenger vehicle is done by [30].

The number of hit counted is very less for ‘ground-hook’ control of semi-active suspension system. Its practical realization is very common with ‘sky-hook’ but the damper is assumed to be fixed to the ground instead of sky [25]. The ground-hook is developed in [31][32]. The ideal concept and its realization is presented and a comparison with ‘sky-hook is done’ [33]. Later, sky-hook is proven a better control scheme on the basis of its performance and its practical realization.

The above discussed semi-active suspension systems capable to deliver the damping force proportional to sprung mass velocity and relative velocity of sprung mass. But, a new control law is introduced where the damping force is proportional to relative velocity of sprung mass and vertical acceleration of sprung mass [34]. The semi-active suspension system with this

new scheme is named as Acceleration-Driven-Damper(ADD). A comparative study of ‘ADD’ and ‘SH’ control for sprung mass vertical acceleration is done with a random input to the suspension. A significant improvement performance in ‘JDD’ is noticed over ‘SH’ control of semi-active suspension system.

The first few studies in the area of magnetorheological damper deal with characterizing the properties of ‘MR’ fluids. A detail study on the properties of MR fluids that are based on barium and strontium ferrites and iron oxides is done [35]. The fluids were prepared using various combinations of the materials, and their properties, such as the MR effect, could have studied. later, the composition of the fluid is tried for ‘MR’ damper it is optimized such that the fluid can have desired property [36]. After the study of ‘MR’ fluid, ‘MR’ device is studied widely [37]. The advantage of ‘MR’ fluid over ‘ER’ fluid is observed for yield strength, the required working volume of fluid, and the required power. Later, the working of ‘MR’ devices is examined for different fluid and compositions of fluid [38].

Away from all the above suspension systems some new concepts is tried in this field. A new passive damper is introduced with variable damping force in 2009 [39]. A cylindrical piston in conical dash pot is introduced and damping is modeled in the function of stroke length. The simulation is done with random input to the suspension and it is realized with experimental study. One more semi-active suspension system with variable damping and variable stiffness is study and its disadvantage is discussed [40]. Before this variable damping and stiffness is discussed where the author were able to vary the stiffness by using ‘voigt’ element [41].

A mixed ‘SH’ and ‘ADD’ semi-active suspension control is discussed in 2006 and a slight improvement is noticed over ‘SH’ control. A gain parameter is suggested on the basis of ‘invariant point’ which comes from the frequency response of ‘SH’ and ‘ADD’ system. The control law for this configuration was more complex than ‘SH’ and ‘ADD’ configuration alone.

# Chapter 3

## Modeling of Suspension System

This chapter reviews the necessary background which provides a fundamental for study of existing and newly proposed semi-active suspension system in later chapters. A vehicle body is generally a rigid body with 6 DOF motions shown in Fig 3.1 [42]. The 6 DOF can be realized as, the longitudinal, lateral, heave, roll, pitch and yaw motion of vehicle. The study of vehicle suspension system can be done for half and full car model and this is carried out by many researchers [43]. A good understanding and analysis of vehicle suspension system can be done by quarter vehicle model. A quarter car linear models with 1DOF and 2DOF systems are used for frequency response of the suspension system. An equivalent non-linear 1DOF system replacing the passive element by semi-active element is used for further work. A appropriate set of parameter is used through out the thesis.

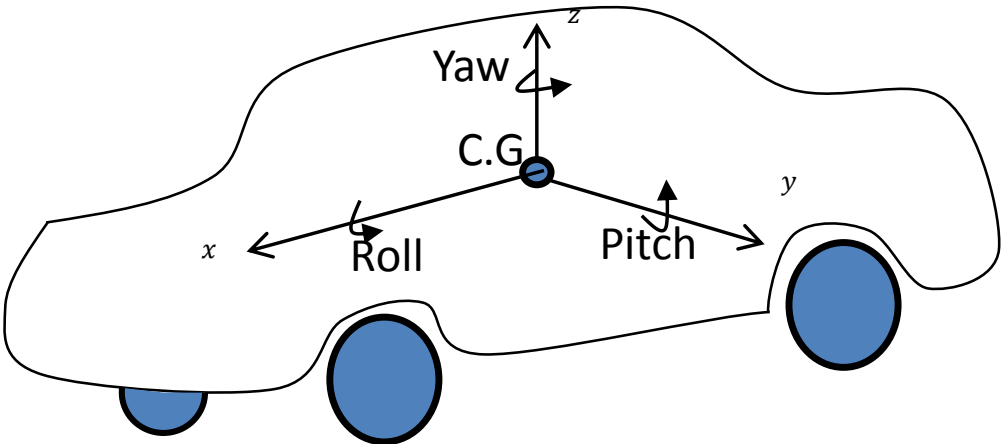


Figure 3.1: 6 DOF vehicle model

### 3.1 Quarter Vehicle Model

The 2DOF quarter car model is more realistic and it is shown in Fig 3.2 (A). But, under the assumption of very high stiffness very low damping of tyre, it can be considered as rigid body. As we have considered tyre as rigid body, the suspension system may be assumed as 1DOF system. Fig 3.2 (B) shows the the equivalent 1DOF vehicle suspension. The following set of parameter is taken from the benchmark of car.

Sprung mass  $m_s = 290$  kg

Unsprung mass  $m_{us} = 59$  kg

Sprung mass stiffness  $k_s = k = 16182$  N/m

Unsprung mass stiffness  $k_{us} = 190000$  N/m

Sprung mass damping coefficient  $b_s = b = 1000$  Ns/m

Unsprung mass damping coefficient  $b_{us} = 300$  Ns/m

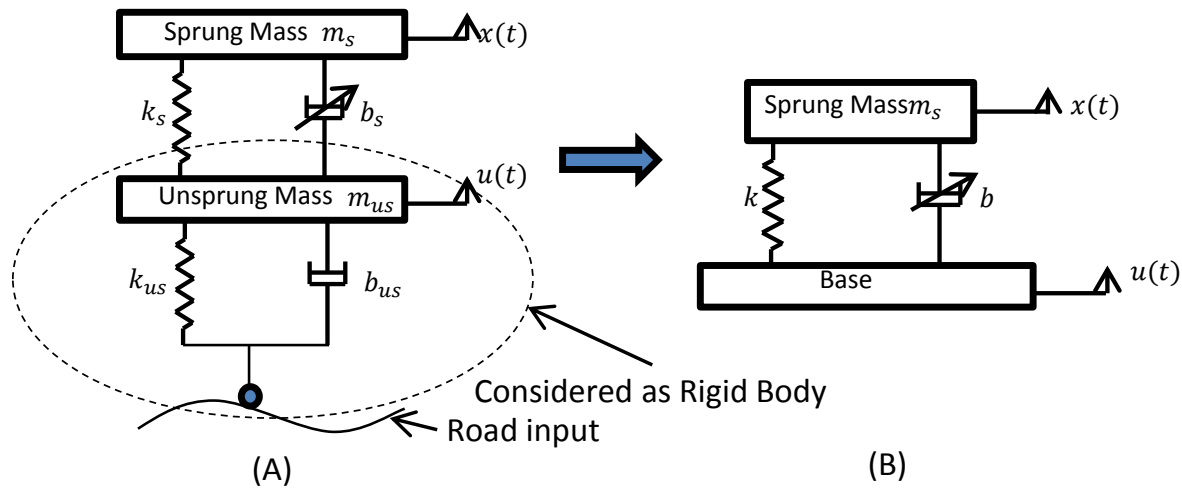


Figure 3.2: 2 DOF quarter car model and its equivalent 1 DOF model under the assumption of very high stiffness and very low damping of tire.

#### 3.1.1 1DOF Passive Suspension System

This subsection covers the frequency response of a linear 1DOF system typically used in the car suspension system, which can be modeled as shown in Fig 3.3. The value of parameters is used as mentioned above. The transfer function for the system shown in Fig 3.3 in Laplace transform is:

$$\frac{X(s)}{U(s)} = \frac{bs + k}{m_s s^2 + bs + k} \quad (3.1)$$

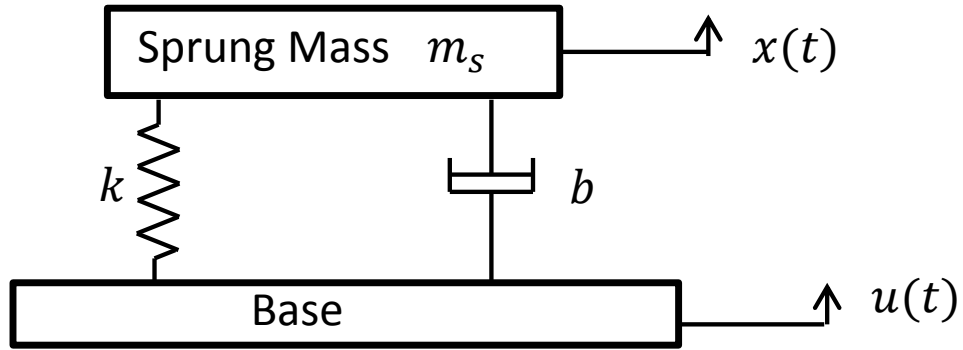


Figure 3.3: 1DOF Quarter car model.

The eqn 3.1 represents the transfer function of 1DOF passive suspension model and replacement of  $s$  by  $i\omega$  gives the transfer function in  $\omega$ .  $\omega$  is the input frequency to the 1DOF system. Now, on the basis of eqn 3.1, we can derive the transmissibility (frequency response) of the system by taking modulus both sides. The frequency response of the system is shown in Fig 3.4. Fig 3.5 shows the closer view. The insight study of both the plot shows the frequency response for varying damping coefficient. The cross point 'C' in the plot shows the cut-off point of frequency from where the alteration of plot takes place. Before the cut-off point, system requires a higher damping coefficient to avoid resonance whereas lower damping is suitable beyond the cut-off point. Next subsection it the study of frequency response of 2DOF passive suspension system.

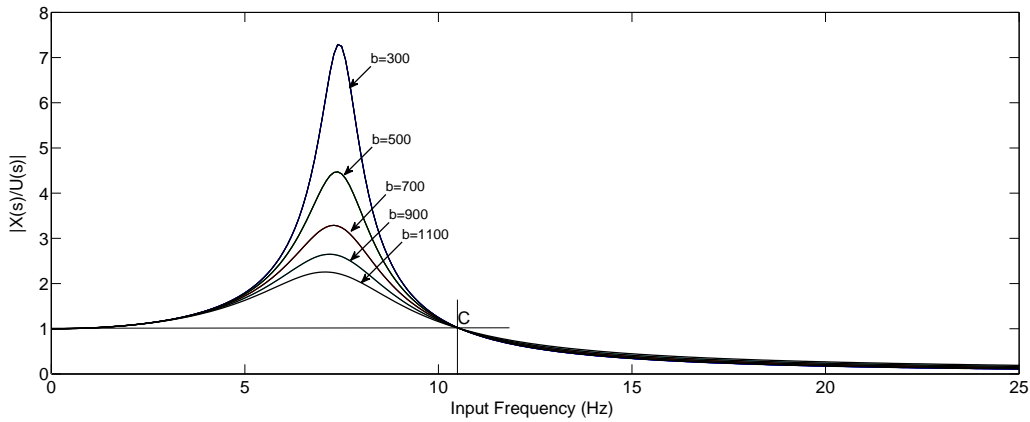


Figure 3.4: Frequency response of the 1DOF quarter car model for different damping.



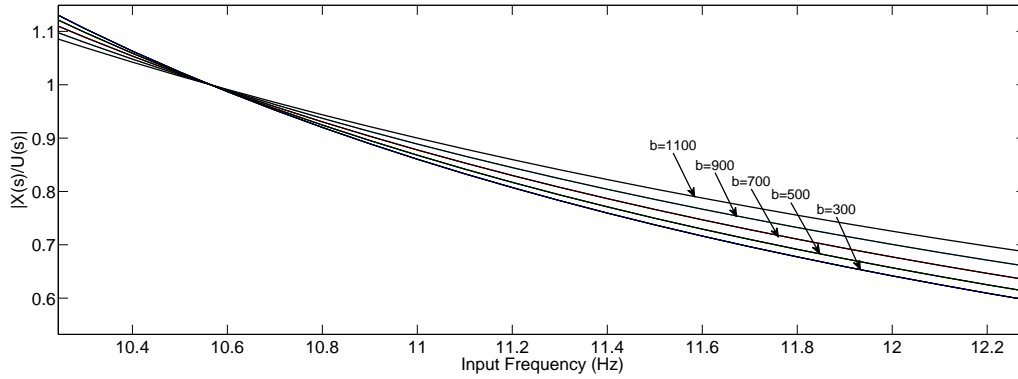


Figure 3.5: Closer view of frequency response of the 1DOF quarter car model for different damping

### 3.1.2 2DOF Passive Suspension System.

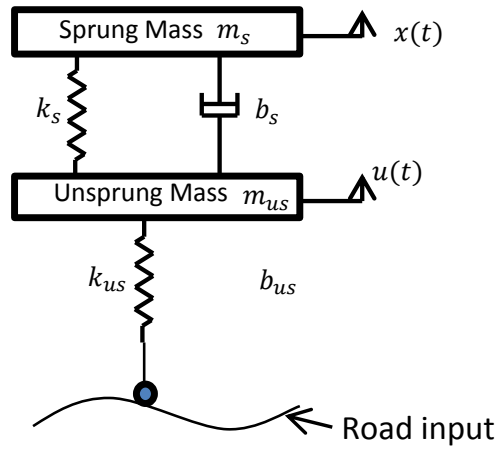


Figure 3.6: 2DOF passive suspension model

Fig 3.6 is the equivalence of Fig 3.2 (A) with the assumption of very low damping of tyre. The variable damper shown in Fig 3.2 is replaced by passive element.

$$\frac{X(s)}{U(s)} = \frac{k_{us} + k_s k_{us}}{m_s m_{us} s^4 + (m_s b_s + m_{us} b_s) s^3 + (m_s k_{us} + k_s m_{us}) s^2 + b_s k_{us} s + k_s k_{us} - 1} \quad (3.2)$$

Now, similar to the previous subsection, the transfer function of 2DOF suspension can be seen in eqn 3.2. Replacing the  $s$  by  $i\omega$  gives the transmissibility of 2DOF system. We can see the demonstration of this frequency response in Fig 3.7 and can discuss on the trade off of passive suspension system. If we choose a low damping value, we can gain superior

high frequency isolation but poor resonant frequency control, and at resonant it will harsh passengers as well as vehicle. However, as we increase the damping value, the resonant frequency control is good but its contribution at higher frequency is not beneficial and it need to be avoided at higher frequency. This is the main point from where the idea for need of varying damping comes. Fig 3.8 gives closer view about cut-off point.

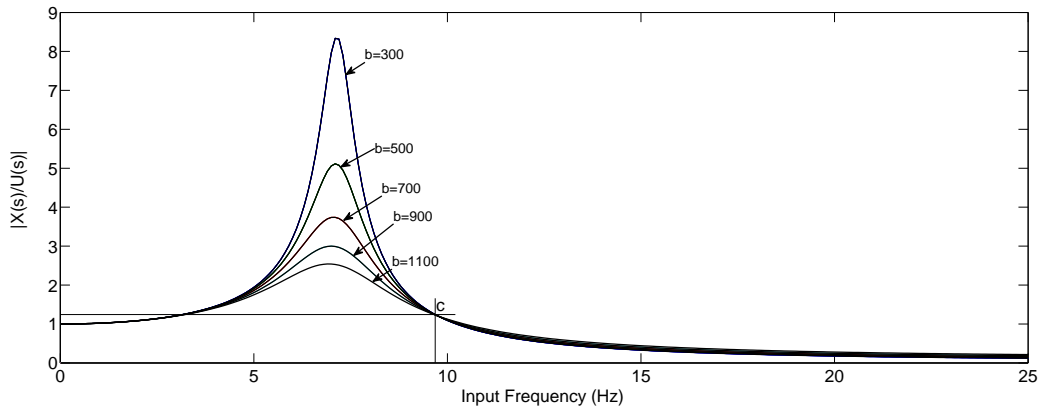


Figure 3.7: Frequency response of the 2DOF quarter car model for different damping

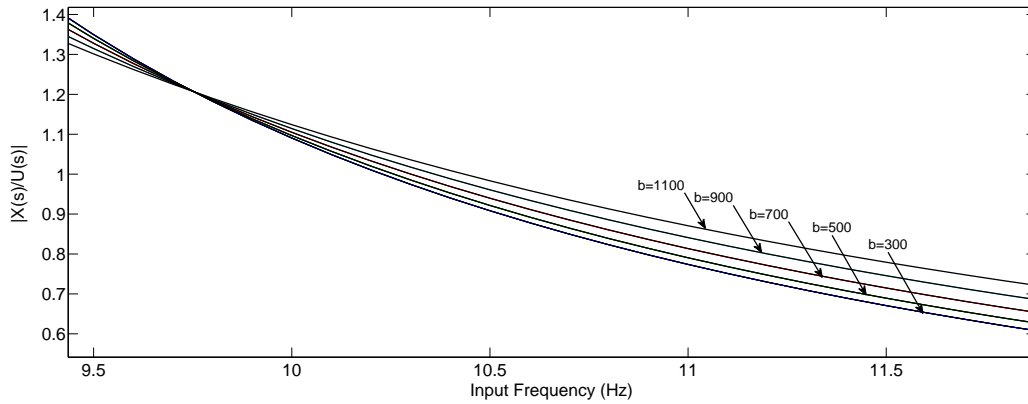


Figure 3.8: Closer view of frequency response of the 2DOF quarter car model for different damping

With the help of Fig 3.7 and Fig 3.4 we conclude the following points:

1. At a certain frequency i.e cut-off frequency,  $\omega$  for all values of damping coefficient the amplitude is same.
2. The amplitude ratio is more then the amplitude at cut-off point for all values of  $\omega$  less than cut-off frequency and as the value of damping increases the amplitude comes

close to the amplitude at cut-off point, in this range of  $\omega$ . It means at lower value of input frequency we require a higher damping value.

3. The amplitude ratio is less than the amplitude at cut-off point for all values of damping coefficient when  $\omega$  greater than cut-off frequency. It means at the high frequency we required the damping coefficient to be low as much as possible. At higher frequency damping coefficient does not have any important role.
4. From the above conclusion we can say that for the different value of input frequency we require different damping coefficient, Which is not possible in passive suspension system because of fixed damper. So, because of this reason variable damper is being used.

## 3.2 Road Profile Estimation

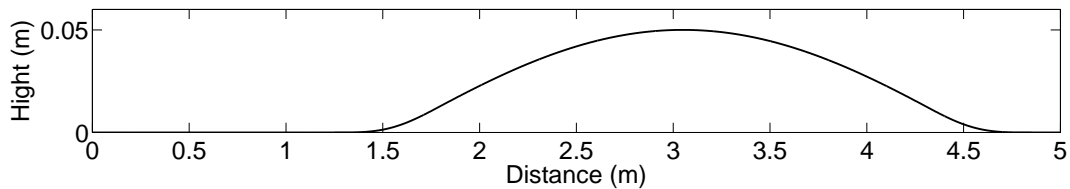


Figure 3.9: Smooth speed breaker

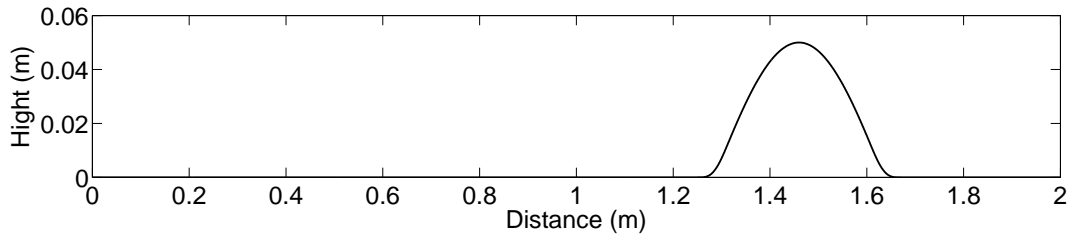


Figure 3.10: Severe speed breaker

Disturbance to the vehicle caused by road is not predictable and it can be random. Widely used road profile for the simulation is the response of a first-order filter to white noise [29]. The road profile used for simulating the existing semi-active suspension system and JDD suspension is a very common half sinusoidal speed breaker with positive amplitude. The dimensional data of the breaker has been taken from the residential road of ordinance factory estate Hyderabad.

A modification is done in the sinusoidal road bump by introducing a polynomial at the both end of the sinusoidal bump. This is done to meet the realistic geometry of the road bump. Fig 3.9 and Fig 3.10 shows the smooth and severe bump physics. The road bumps is created by using a sinusoidal curve of 0.05m amplitude and two polynomial at the each end of the sinusoidal curve. The polynomial starts before the meeting point of sinusoidal profile and road by 10% of width and ends after the meeting point of sinusoidal profile and road by same amount.

# Chapter 4

## Control Strategies in Semi-Active Suspension System

### 4.1 Introduction

The amplitude curves for different damping for a semi-active suspension system has been observed in the chapter 3, but apart from that most of the research has been concentrated on the use of control technology in order to provide controllable suspension force. This section deals with the analysis of all important existing semi-active suspension systems and its simulation. The control logic behind these semi-active suspension systems are tried to make simple. The simulation for the sprung mass displacement and acceleration is done with all three type of speed breakers as it is modeled in chapter 3. As we know that the heart of semi-active suspension system is controllable damper, This force has been controlled as a function of the relative velocity and relative displacement between the attachment point of sprung and unsprung masses. But, this is not true for all the semi-active suspensions. In ‘ADD’ and proposed ‘JDD’ control, sprung mass acceleration and its relative acceleration with base is considered to control the damper force. The semi-active control strategies which are studied in the background of the work covered in this theses are:

- Balance Logic
- Sky-hook Logic
- Ground-hook Logic and
- ADD Control

Preceding sections of this chapter will give a detailed study and understanding of all above explained controlled strategies.

## 4.2 Balance Logic Analysis

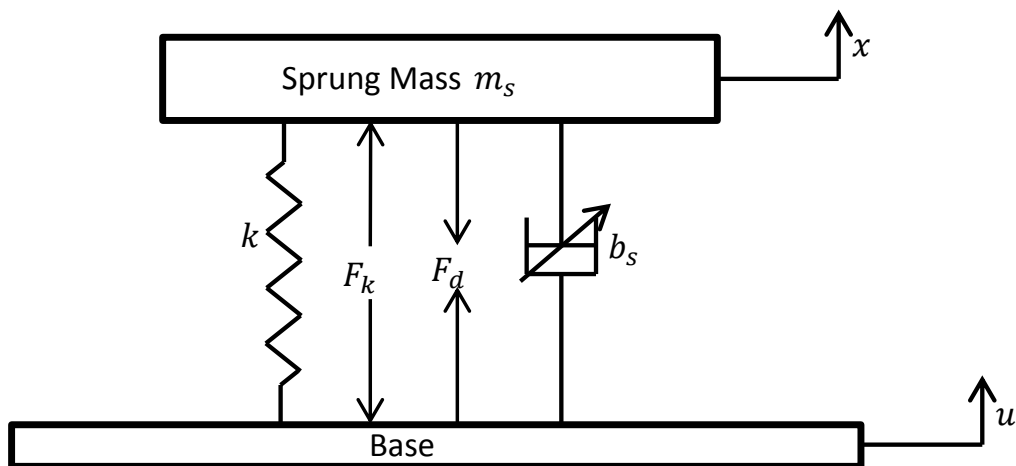


Figure 4.1: Force analysis of 1DOF vehicle suspension model

A 1DOF system with variable damper is installed at the place of conventional passive damper with the excited base. The motion of the sprung mass  $m_s$  is described by:

$$m_s \ddot{x} + Kx_1 + F_d(x_1, \dot{x}_1) = 0 \quad (4.1)$$

Where  $x$  and  $x_1$  are the absolute and relative displacements of the sprung mass respectively where,  $x_1 = (x - u)$  (Refer to Fig 4.1).  $Kx_1$  is the passive elastic force which is caused by the spring and  $F_d(x_1, \dot{x}_1)$  is semi-active damping force. This damping force is the function of relative velocity and relative displacement of the sprung mass so it will make the damper nonlinear. The damper coefficient is directly proportional to the relative velocity  $(\dot{x} - \dot{u})$  and it can be tuned in the function of time according to the vibration measured from the sprung mass and base. In the eqn 4.1, to make the system linear we need to cancel the spring force  $Kx_1$  (Refer to Fig 4.1). Once this spring force will be canceled sprung mass will not be disturbed by any force as we can see in eqn 4.1. But we can see from the further sections canceling the spring force is not possible always, in that case we can minimize the damping force as much as possible.

### 4.2.1 Balance Logic ‘On-Off’ Control

Balance Logic is widely used control strategy in the semi-active suspension system. The semi active damper is considered as two-state damper with On-Off characteristic. When the relative velocity and relative displacement of the sprung mass and has the opposite sign in that case we will be able to generate a damping force in the direction same as relative velocity of sprung mass. In this case we will be able to cancel the spring force between the base and sprung mass. Now, we can consider the case when the sign of relative velocity and relative displacement of the sprung mass is same, in this case it will always generate a damping force which will have positive sign. Whenever this positive force will be added to the system it will increase the acceleration of the sprung mass. So, better way to make this force zero so that its contribution will be zero to the system.

The control law for Balance Logic on the basis of above discussion is given by:

$$F_d = F_{balance} = \begin{cases} k|x_1|sgn\dot{x}_1 & \text{if } x_1\dot{x}_1 \leq 0 \\ 0 & \text{if } x_1\dot{x}_1 > 0 \end{cases} \quad (4.2)$$

But, it is not possible to make the damping force zero as stated in the case of when  $x_1\dot{x}_1 > 0$ , because the provided semi-active damper bonded with some maximum as well as minimum value of damping. So, we can write control balance logic algorithm for damping coefficient as:

$$b_{balance} = \begin{cases} b_{max} & \text{if } x_1\dot{x}_1 \leq 0 \\ b_{min} & \text{if } x_1\dot{x}_1 > 0 \end{cases} \quad (4.3)$$

Where  $b_{max}$  and  $b_{min}$  is the maximum and minimum damping coefficient of the semi-active damper. Now, on the basis of above discussion we can prepare a table for the different cases of relative position and relative velocity of sprung mass.

**Case(1)and case(2)** are the cases when relative position  $x_1$  is positive. For positive value of  $x_1$  relative velocity  $\dot{x}_1$  can be positive or negative. If  $\dot{x}_1$  is positive, it causes the positive damping force to the system. This positive damping force will be added to spring force and causes a large amount of disturbance to the sprung mass refer to eqn 4.1. So, to avoid this large disturbance, better to minimize the damper force by keeping the damper in ‘Off’ condition. But, if  $\dot{x}_1$  is negative, there is a probability that the damper force will be canceled out with the addition of spring force (because the spring force is positive). So, for better isolation of sprung mass from the disturbance damper should be kept in ‘On’ condition.

**Case(3)and case(4)** are the cases when relative position  $x_1$  is negative. Again for negative value of  $x_1$  relative velocity  $\dot{x}_1$  can be positive or negative. if  $\dot{x}_1$  is positive, there is a probability that the damper force will be canceled out with the addition of spring force (because the spring force is negative). So, for better isolation of sprung mass from the disturbance damper should be kept in ‘On’ condition. But, if  $\dot{x}_1$  is negative, it causes the negative damping force to the system. This negative damping force will be added to spring force and causes a large amount of disturbance to the sprung mass (a large negative quantity) refer to eqn 4.1. So, to avoid this large disturbance, better to minimize the damper force by keeping the damper in ‘Off’ condition.

So, on the basis of above discussion, a table can be prepared for the easy understanding of the all explained cases.

Table 4.1: Illustration of the balance logic:

Case	Relative Position	Relative Velocity	Damper force on mass	Switch
1	$x_1 > 0$	$\dot{x}_1 > 0$	Not possible to cancel	Off
2	$x_1 > 0$	$\dot{x}_1 < 0$	$F_d = -k x_1 $	On
3	$x_1 < 0$	$\dot{x}_1 > 0$	$F_d = k x_1 $	On
4	$x_1 < 0$	$\dot{x}_1 < 0$	Not possible to cancel	Off

## 4.2.2 Balance Logic ‘On-Off’ Control Simulation

On the basis of above explained control law for balance logic, MATLAB simulation for sprung mass vertical displacement, sprung mass vertical acceleration can be done. Simulation is done for three type of possible speed breaker i.e smooth speed breaker, severe speed breaker, and continuous bump speed breaker as explained with dimension and modeled in chapter 3. Below are the plots for sprung mass vertical displacement, sprung mass vertical acceleration at different speed breakers.

## 4.2.3 Balance Logic ‘Continuous’ Control

‘On-Off’ control of balance logic gives an understanding to the working of two state ‘On-Off’ semi-active damper. But, when the damper is said to be in ‘On’ condition we can choose a maximum value damper. The explanation of this statement is given below.

We know the damping force of 1DOF suspension system is given by eqn 4.4(refer to fig 4.1).



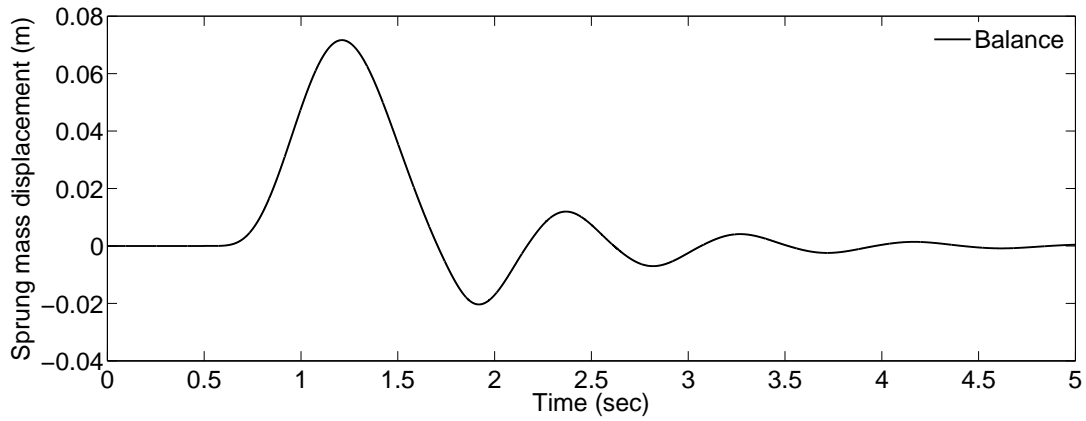


Figure 4.2: Sprung mass displacement for ‘On-Off’ balance logic with smooth breaker

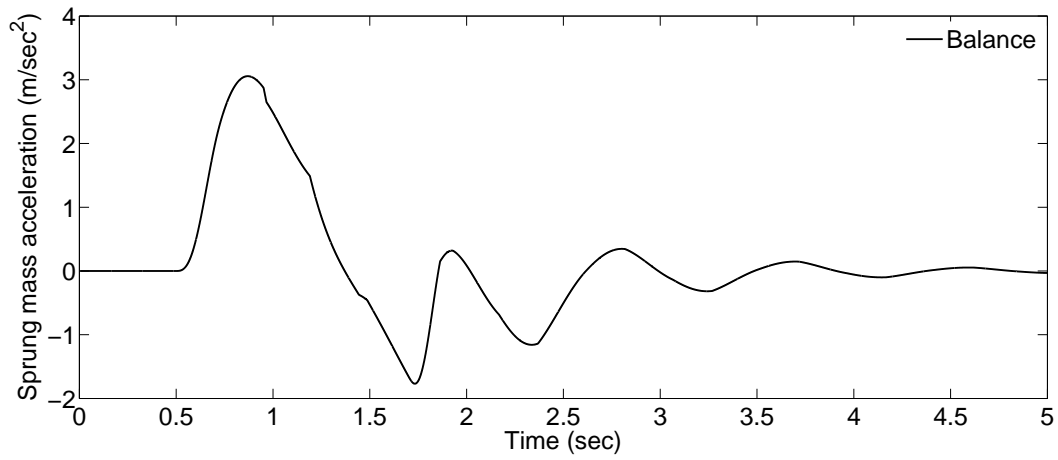


Figure 4.3: Sprung mass acceleration for ‘On-Off’ balance logic with smooth breaker

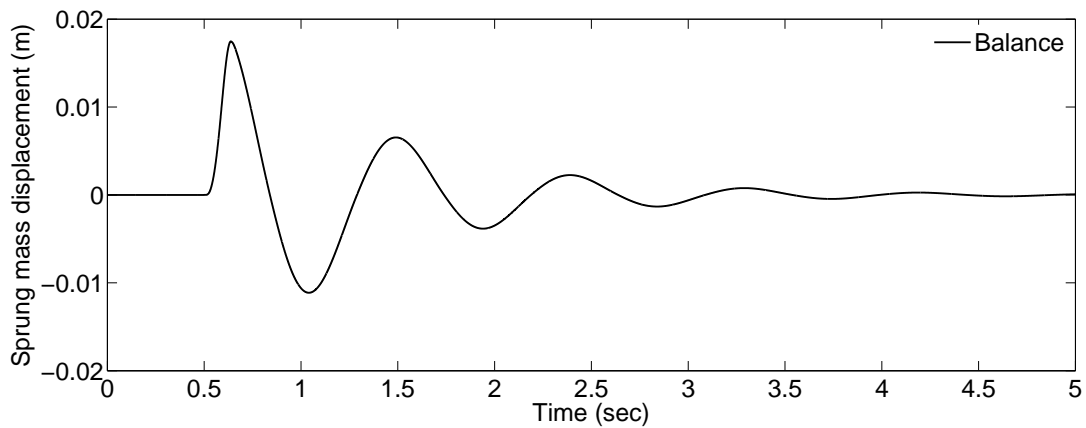


Figure 4.4: Sprung mass displacement for ‘On-Off’ balance logic with severe breaker

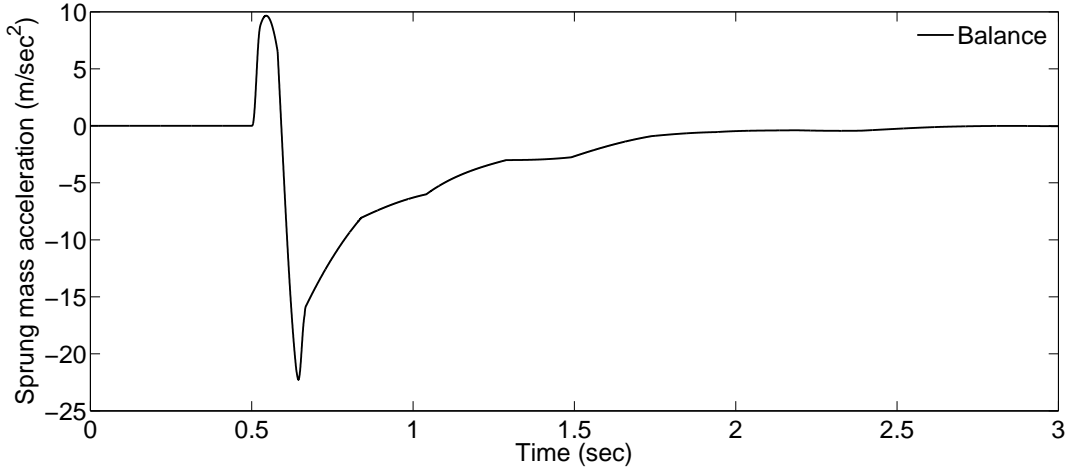


Figure 4.5: Sprung mass acceleration for ‘On-Off’ balance logic with severe breaker

$$F_d = b(\dot{x} - u) \quad (4.4)$$

If eqn 4.4 is equated with eqn 4.2, the damping coefficient become  $b_s$  will be:

$$b = \begin{cases} \frac{k|x_1|sgn\dot{x}}{(\dot{x}-\dot{u})} & \text{if } x_1\dot{x}_1 \leq 0 \\ 0 & \text{if } x_1\dot{x}_1 > 0 \end{cases} \quad (4.5)$$

Now, we can see from the equation 4.6, when the denominator  $(\dot{x} - \dot{u})$  is minimum of very less the semi-active damping will tends to infinity. But, we have the upper and and lower limit of damping in practice, so for this reason of physical constraints minimum and maximum damping can be term as  $b_{min}$  and  $b_{max}$  and algorithm for the damping can be written as eqn 4.6

$$b = \begin{cases} \max[b_{min}, \min[\frac{k|x_1|sgn\dot{x}}{(\dot{x}-\dot{u})}, b_{max}]] & \text{if } x_1\dot{x}_1 \leq 0 \\ b_{min} & \text{if } x_1\dot{x}_1 > 0 \end{cases} \quad (4.6)$$

### 4.3 Ground-hook Logic Analysis

This section will discuss the ground-hook policy of the semi-active suspension control. This study will examine the velocity based, on-off ground-hook control. In the ground-hook control the passive damper is regarded as being hooked to a fixed point in the ground hence, the name ground-hook. However this ideal ground-hook configuration cannot be realize in

practice because the damper cannot be fixed to a non-moving inertia frame. The controllable damper of semi-active suspension system will implement the ground-hook control by modulating the damper such a way that it can play the role of ground-hook damper. For implementation of ground-hook policy in the damper an input current to the damper is only applied when the relative velocity between the base and the sprung mass and the absolute velocity of base are in same direction. The ultimate objective of the ground hook control of semi-active suspension system is to minimize the force between the tire and road and to minimize the road damage [Y. Liu, T.P.Waters, 2005].

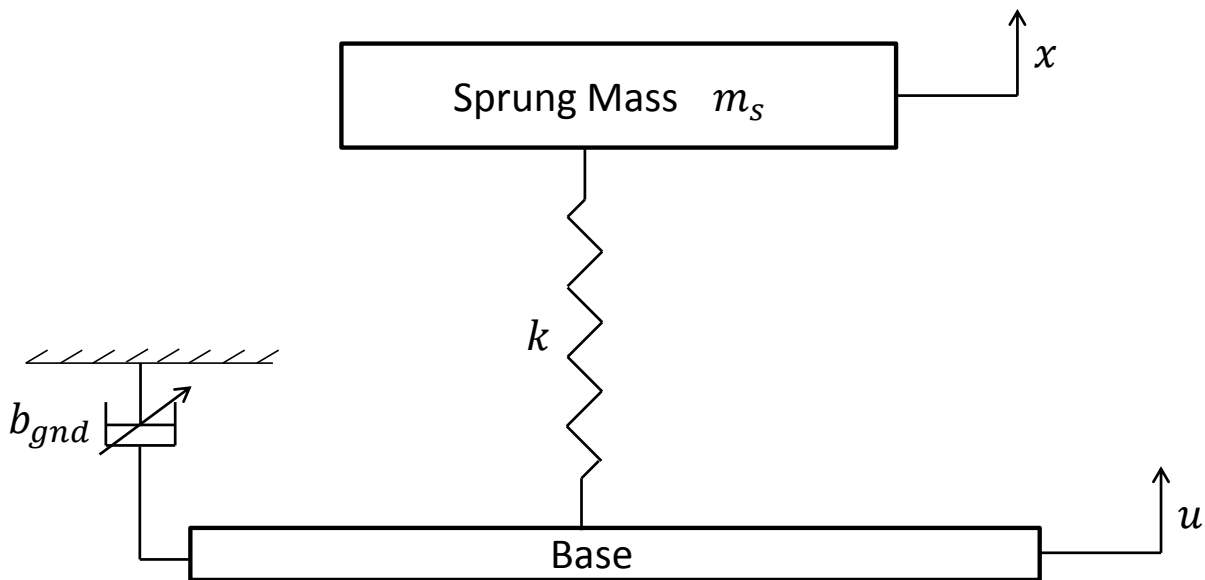


Figure 4.6: 1DOF vehicle suspension model with groundhook(GH) logic

### 4.3.1 Ground-hook ‘On-Off’ Control

Consider a one degree of freedom model as indicated in fig 4.1 The motion of the sprung mass  $m$  is described by:

$$m\ddot{x} + k(x - u) + F_d \quad (4.7)$$

Where  $x$  and  $(x - y)$  are the absolute and relative displacements of the sprung mass,  $k(x - u)$  is the passive elastic force which is caused by the spring and  $F_d = (\dot{x} - \dot{u})$  is semi-active damping force. Fig. 4.4 shows the ideal ground-hook configuration of a one degree of freedom model. The most comprehensive way to determine the equivalent ground-hook

damping force is to examine the forces acting on the base under several conditions. For all the cases the mass moving upward is considered as positive and the mass moving downward considered negative.

Let us **case1** in which the relative velocity of the base mass is positive and the two masses separates. In this case when we will consider the ground-hook equivalent configuration, there is tension placed on the damper, and the damper force pulls the base mass from its equilibrium position. For this reason, minimum damping is required. In this case:

$$b = b_{off}$$

The above equation can also be satisfied when we will consider **case2** as, base is moving in negative direction and relative velocity of the base is positive. In this case damper will be in tension and it will pull the sprung mass.

Now, consider the **case3** as, base mass is moving with positive velocity and relative velocity of base is also positive. In this case Damper is in compression and pushing down base to its equilibrium. In this case:

$$b = b_{on}$$

The above equation can also be satisfied when we will consider the **case4** as, base is moving in negative direction and its relative velocity is also in the negative direction. In this case the damper will be in compression and will push the sprung mass to its equilibrium.

The above discussion helps us to give an ideal ground-hook logic as:

$$F_d = F_{gnd} \begin{cases} b_{gnd} & \text{if } \dot{u}(\dot{u} - \dot{x}) \leq 0 \\ 0 & \text{if } \dot{u}(\dot{u} - \dot{x}) > 0 \end{cases} \quad (4.8)$$

From the above discussion the control logic for the ground-hook control made clear and the on-off control policy for the ground-hook control is illustrated in table 4.2.

Table 4.2: Illustration of the GH control:

Case	Base velocity	Relative Velocity of base	Damper force on sprung mass	Switch
1	$\dot{u} > 0$	$(\dot{u} - \dot{x}) < 0$	0	Off
2	$\dot{u} > 0$	$(\dot{u} - \dot{x}) > 0$	$b_{gnd}\dot{u}$	On
3	$\dot{u} < 0$	$(\dot{u} - \dot{x}) < 0$	$b_{gnd}\dot{u}$	On
4	$\dot{u} < 0$	$(\dot{u} - \dot{x}) > 0$	0	Off

### 4.3.2 Ground-hook ‘On-Off’ Control Simulation

On the basis of above explained control law for ‘GH’ logic, MATLAB simulation for sprung mass vertical displacement, sprung mass vertical acceleration can be done. Simulation is done for three type of possible speed breaker i.e smooth speed breaker, severe speed breaker, and continuous bump speed breaker as explained with dimension and modeled in chapter 3. Below are the plots for sprung mass vertical displacement, sprung mass vertical acceleration at different speed breakers.

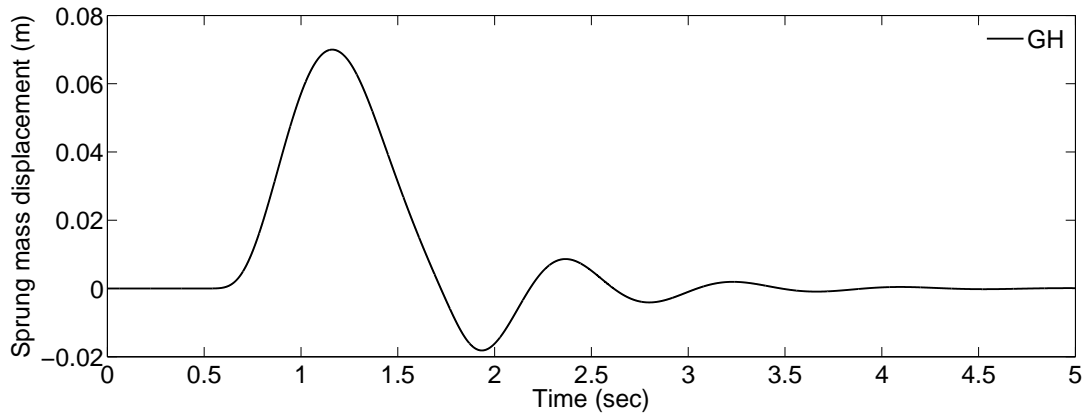


Figure 4.7: Sprung mass displacement for ‘On-Off’ GH control with smooth breaker

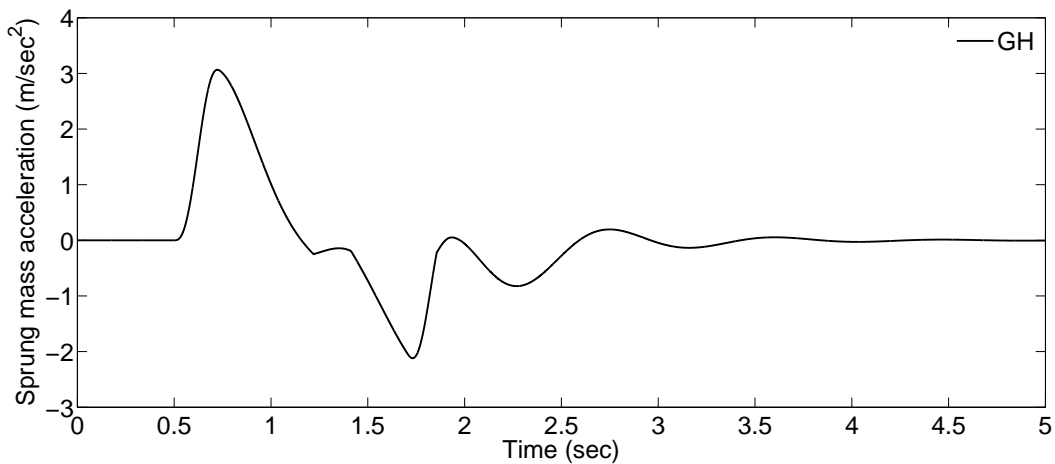


Figure 4.8: Sprung mass acceleration for ‘On-Off’ GH control with smooth breaker

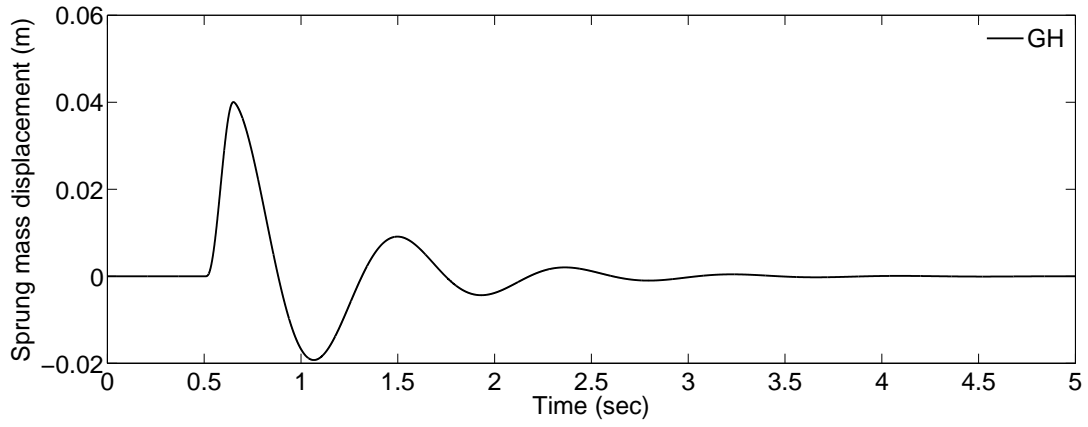


Figure 4.9: Sprung mass displacement for ‘On-Off’ GH control with severe breaker

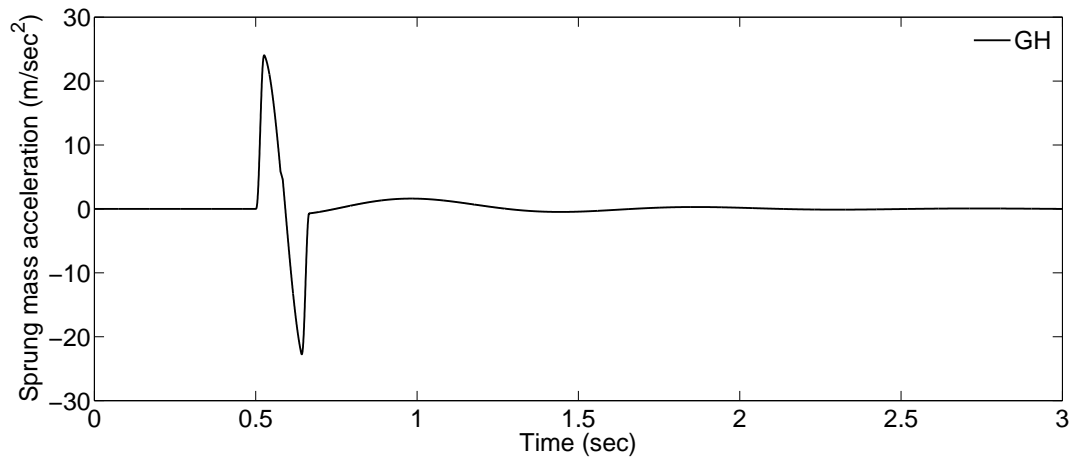


Figure 4.10: Sprung mass acceleration for ‘On-Off’ GH control with severe breaker

### 4.3.3 Ground-hook ‘Continuous’ Control

‘On-Off’ control of GH control gives an understanding to the working of two state ‘On-Off’ semi-active damper. But, when the damper is said to be in ‘On’ condition, we can choose a maximum value damper. The explanation of this statement is given below.

We know the damping force of 1DOF suspension system is given by eqn 4.9(refer to fig 4.1).

$$F_d = b(\dot{x} - \dot{u}) \quad (4.9)$$

If eqn 4.9 is equated with eqn 4.8, the damping coefficient  $b$  will be:

$$b = \begin{cases} b_{gnd} \frac{\dot{u}}{(\dot{u} - \dot{x})} & \text{if } \dot{u}(\dot{u} - \dot{x}) \leq 0 \\ 0 & \text{if } \dot{u}(\dot{u} - \dot{x}) > 0 \end{cases} \quad (4.10)$$

Now, we can see from the eqn 4.10, when the denominator  $(\dot{u} - \dot{x})$  is minimum of very less the semi-active damping will tends to infinity. But, we have the upper and lower limit of damping in practice, so for this reason of physical constraints minimum and maximum damping can be term as  $b_{min}$  and  $b_{max}$  and algorithm for the damping can be written as eqn 4.11

$$b = \begin{cases} \max[b_{min}, \min[b_{gnd} \frac{\dot{u}}{(\dot{u} - \dot{x})}, b_{max}]] & \text{if } \dot{u}(\dot{u} - \dot{x}) \leq 0 \\ b_{min} & \text{if } \dot{u}(\dot{u} - \dot{x}) > 0 \end{cases} \quad (4.11)$$

## 4.4 Sky-hook Logic Analysis

The motion of the sprung mass  $m_s$  from Fig 3.3 is:

$$m_s \ddot{x} + k(x - u) + F_d = 0 \quad (4.12)$$

Where  $x$  and  $(x - u)$  are the absolute and relative displacements of the sprung mass,  $k(x - u)$  is the passive elastic force caused by the spring and  $F_d$  is semi-active damping force. We can also define the relative velocity of the sprung mass as  $(\dot{x} - \dot{u})$ . Consider a semi-active suspension with sky-hook damper as shown in fig 4.11, the damping force can be written as:

$$F_{sky} = b_{sky} \dot{x} \quad (4.13)$$

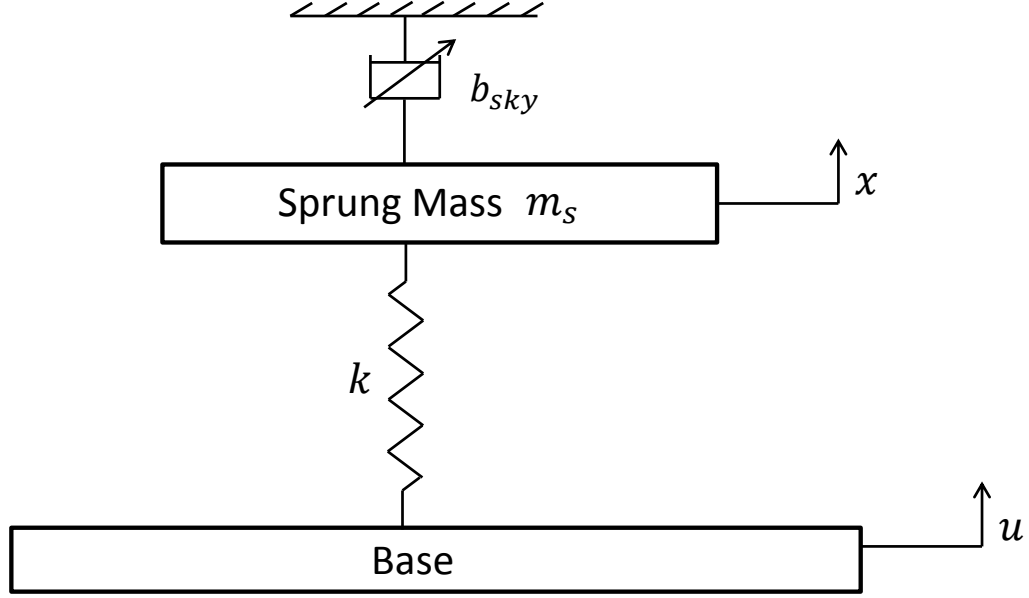


Figure 4.11: 1DOF vehicle suspension model with skyhook(SH) logic

#### 4.4.1 Transmissibility of ‘Sky-hook’ Configuration

Consider a one degree of freedom model as indicated in Fig 4.11. We can derive the transmissibility of the model as given in eqn 4.14.

$$\left| \frac{X}{U} \right| = \frac{k^2}{(k - m\omega^2)^2 + (\omega b_{sky}^2)} \quad (4.14)$$

where, in this case,  $b_{sky}$  is the sky-hook damping coefficient. Once again, if we plot the transmissibility for various values of  $b_{sky}$ , we find the result shown in Fig 4.12. As in the passive case, as the sky-hook damping ratio increases, the resonant transmissibility decreases. Increasing the sky-hook damping, however, does not increase the transmissibility above the resonant frequency. For sufficiently large sky-hook damping we can drive even at the resonance frequency. If we compare the plot in Fig 3.4 and Fig 4.12, at the higher damping the amplitude ratio is more closer to the step input given to the suspension, and it is reaching close to zero amplitude ratio at the lower frequency as compare to the passive suspension transmissibility. This improvement in the transmissibility is caused by the change in the position of damper from its original position to new position. It is obvious that the new position of the damper is hypothetical and its realization not possible, though the performance result of sky-hook configuration will be in between the passive transmissibility and sky-hook transmissibility. where ‘1,’2,’3,’4 and ‘5 are the plot for different damping



coefficient as:  $b_{sky} = 200, 600, 1000, 1400,$  and  $1800$  Ns/m respectively.

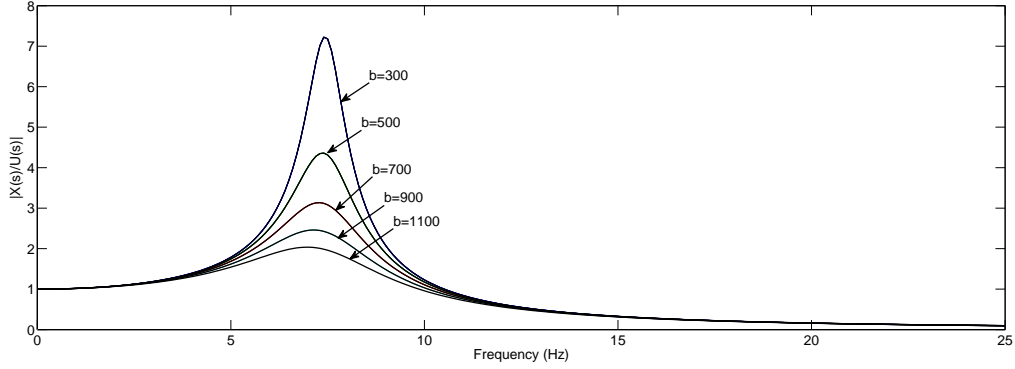


Figure 4.12: 'Skyhook' Suspension Transmissibility

#### 4.4.2 Sky-hook 'On-Off' Control

The intention is to replace such a sky-hook damping force with a conventional semi-active damper mounted between the sprung mass and unsprung mass. The desired force is  $b_{sky}\dot{x}$ , but the semi-active damper can only achieve this force when  $\dot{x}$  and  $(\dot{x} - \dot{u})$  have the same sign. When  $(\dot{x} - \dot{u})$  are of opposite sign, the semi-active damper can only provide a force opposite to the desired force. In this situation, it is better to supply no force between sprung mass and unsprung mass. Now, we assume that the mass  $m_s$  of the fig 4.11 is moving upwards with a positive velocity  $\dot{x}$ . If we consider the force applied by the sky-hook damper to the mass, we notice that it is in negative direction.

$$F_{sky} = -b_{sky}\dot{x} \quad (4.15)$$

Next, we need to determine if the semi-active damper is able to provide the same force. If the base and the suspended mass in Fig 3.3 is separating, then the semi-active damper is in tension. Thus, the force applied to the suspended mass (i.e  $m_s$ ) is in negative direction.

$$F_d = -b(\dot{x} - \dot{u}) \quad (4.16)$$

This is the maximum damping force generated by the damper and we can termed it as damper is in on state. Since, we are able to generate a force in the proper direction, the only requirement to match the sky-hook suspension. To match the requirement the sky-hook damping force can be equated with the conventional damping force. So, equating eqn 4.16 and eqn 4.15 we can get as:

$$b = b_{sky} \frac{\dot{x}}{(\dot{x} - \dot{u})} \quad (4.17)$$

We can see from equation eqn 4.17,  $b$  can be defined only when  $\dot{x}$  and  $(\dot{x} - \dot{u})$  is same sign, and it shows that the state of damper is decide by the sign of  $\dot{x}$  and  $(\dot{x} - \dot{u})$ . Now, consider the case in which base and suspended mass is moving downward and suspended mass with a negative velocity  $\dot{x}$ . In the sky-hook configuration, the damping force will now be applied upwards, or positive direction. In the semi-active configuration, however, the semi-active damper is still in tension, and the damping force will still be applied in the downward, or in negative direction. Since the semi-active damping force cannot possibly be applied in the same direction as the sky-hook damping force, the best way to achieve the good solution is to minimize the damping force. In this case minimizing the damping the damper force as much as possible is more beneficial or, the semi-active damper is desired to be set so that there is no damping force, in real there is some small damping force present and it is in the same direction as the sky-hook damping force. Thus, if  $(\dot{x} - \dot{u})$  is positive and  $\dot{x}$  is negative, we need to minimize the semi-active damping force. Similarly we can apply the same analysis to the other two possible combination of  $\dot{x}$  and  $(\dot{x} - \dot{u})$ .

When the sprung mass is moving upward and base too in same direction but velocity of base is more then the sprung mass. Then, in the sky hook system the damping force will be in negative direction or downward and damping force of semi-active suspension is upward on sprung mass. In this case it is not possible to achieve the sky-hook damping force for semi-active suspension. So, better way is to minimize the damping as explained above. Now we can consider last possible case in which motion of the suspension system is downward but velocity of sprung is more then the base. In this case sky-hook damper will give a force is in upward direction and semi-active suspension will also give in upward direction. So, in this case it is possible to get sky-hook damping for semi-active suspension system, and the force will be same as eqn 4.17. Then on the basis of above discussion we can write the control algorithm as:

$$F_d = F_{sky} \begin{cases} b_{sky} \dot{x} & \text{if } \dot{x}(\dot{x} - \dot{u}) \geq 0 \\ 0 & \text{if } \dot{x}(\dot{x} - \dot{u}) < 0 \end{cases} \quad (4.18)$$

Now, we can also write the control algorithm for the semi-active damping with the help of eqn 4.17.

$$b = \begin{cases} b_{sky} \frac{\dot{x}}{(\dot{x} - \dot{u})} & \text{if } \dot{x}(\dot{u} - \dot{x}) \leq 0 \\ 0 & \text{if } \dot{x}(\dot{u} - \dot{x}) > 0 \end{cases} \quad (4.19)$$

Table 4.3: Illustration of the SH control:

Case	sprung mass vel.	Relative velocity	Damper force on sprung mass	Switch
1	$\dot{x} > 0$	$(\dot{x} - \dot{u}) > 0$	$b_{sky}\dot{x}$	On
2	$\dot{x} > 0$	$(\dot{x} - \dot{u}) < 0$	0	Off
3	$\dot{x} < 0$	$(\dot{x} - \dot{u}) > 0$	$b_{sky}\dot{x}$	On
4	$\dot{x} < 0$	$(\dot{x} - \dot{u}) < 0$	0	Off

#### 4.4.3 Sky-hook ‘On-Off’ Control Simulation

On the basis of above explained control law for ‘On-Off’ SH control, MATLAB simulation for sprung mass vertical displacement, sprung mass vertical acceleration can be done. Simulation is done for two type of possible speed breaker i.e smooth speed breaker and severe speed breaker. Below are the plots for sprung mass vertical displacement, sprung mass vertical acceleration at different speed breakers. The road profile physics is described in chapter 3. Speed of the vehicle at the bump is considered as  $2.5 \text{ m/sec}$  and it is used for all simulations.

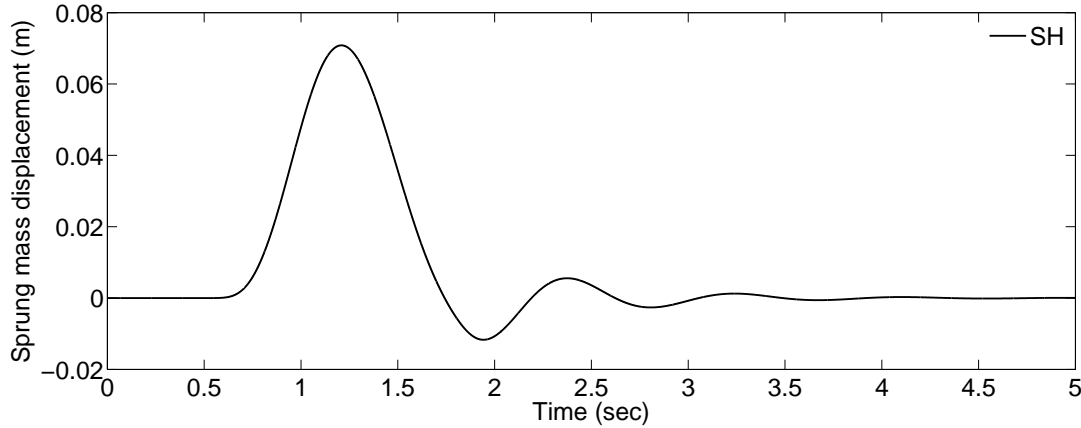


Figure 4.13: Sprung mass displacement for ‘On-Off’ SH control with smooth breaker

#### 4.4.4 Sky-hook ‘Continuous’ Control

Now, we can see from the eqn 4.19, the minimum value of  $(\dot{x} - \dot{u})$  will cause the abrupt change in the value of semi-active damping coefficient and it tends to zero as this value

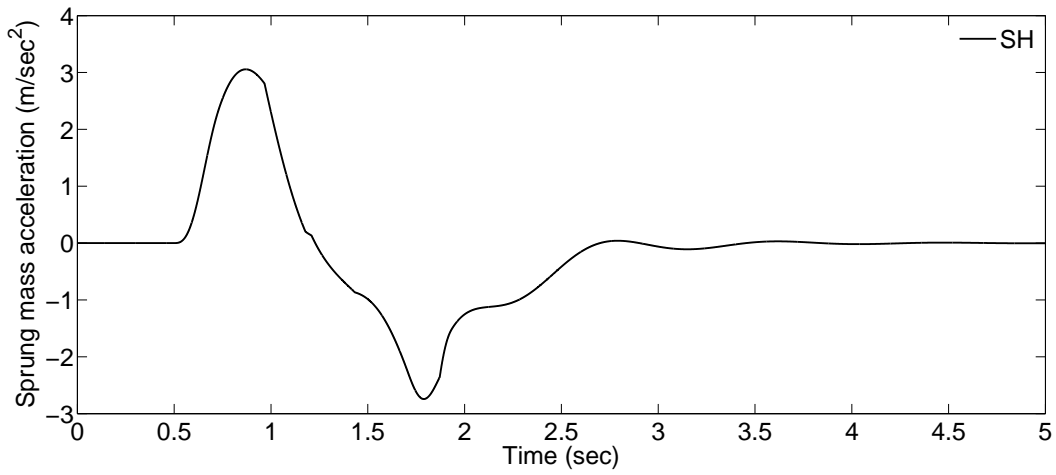


Figure 4.14: Sprung mass acceleration for ‘On-Off’ SH control with smooth breaker

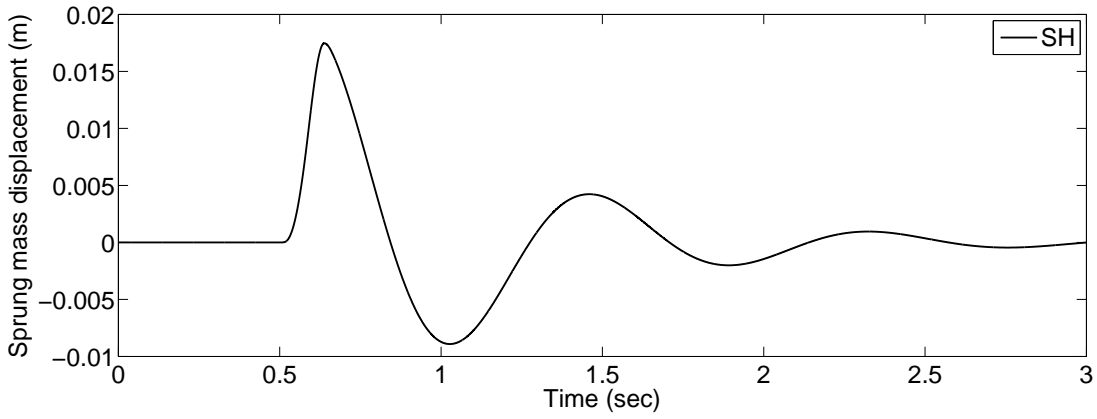


Figure 4.15: Sprung mass displacement for ‘On-Off’ SH control with severe breaker

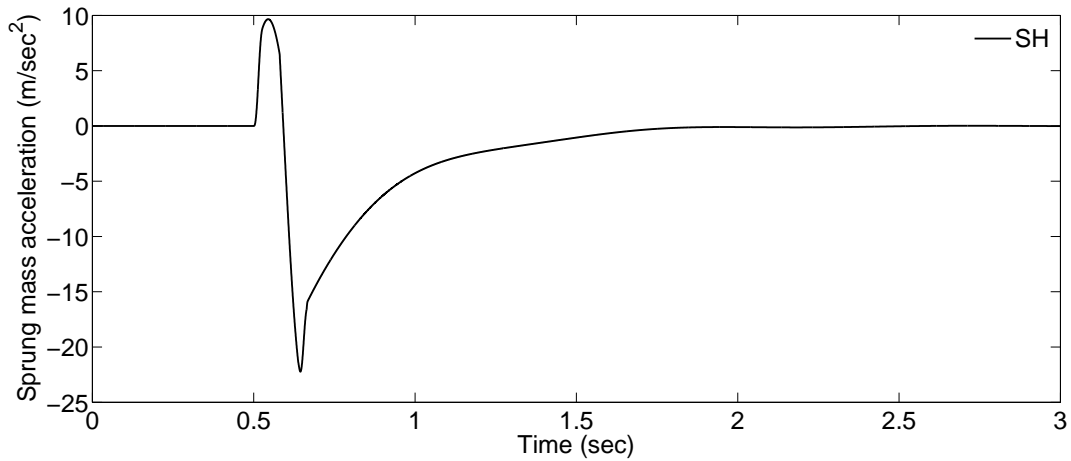


Figure 4.16: Sprung mass acceleration for ‘On-Off’ SH control with severe breaker

reaches zero. But, we have the maximum and minimum limit of damping and we cant go beyond this limit. Also we cant get the zero damping even in off condition of damper, there will be some damping which can be termed as  $b_{min}$ . It means that we have a upper and lower bond of damping coefficient as  $b_{max}$  and  $b_{min}$  respectively. So, we can write the damping coefficient in more suitable way as eqn 4.20:

$$b = \begin{cases} \max[b_{min}, \min[b_{sky} \frac{\dot{x}}{(\dot{x}-\dot{u})}, b_{max}]] & \text{if } \dot{x}(\dot{x} - \dot{u}) \geq 0 \\ b_{min} & \text{if } \dot{x}(\dot{x} - \dot{u}) < 0 \end{cases} \quad (4.20)$$

## 4.5 ADD Control Analysis

The earlier studied control strategies are either proportional to sprung mass displacement or relative velocity of sprung mass. In this chapter, a different type of semi-active suspension control is explained which a latest study in semi-active suspension system called acceleration driven damper(ADD) control. The damping force generated by the variable damper is proportional to sprung mass acceleration and relative velocity instead of sprung mass displacement and velocity. The control law for ADD is quite simple and it can be understand easily by the analysis of equation of motion(EOM) of 1DOF suspension system. Similar to other semi-active suspension system, a continuous control of ADD is proposed in the next chapter. This chapter will explain the ‘On-Off’ control of ADD suspension system.

### 4.5.1 ADD ‘On-Off’ Control

The EOM od 1DOF suspension system according to Fig 3.3 can be written in eqn 4.21.

$$m_s \ddot{x} = -b(\dot{x} - \dot{u}) - k(x - u) \quad (4.21)$$

The equation has to be analyze by using the term present in the RHS of eqn 4.21. For this purpose let us break the RHS of eqn 4.21 in two terms.

$$m_s \ddot{x} = [-b(\dot{x} - \dot{u})][-k(x - u)] \quad (4.22)$$

To have a better understanding of ADD working principle and control logic, eqn 4.22 can be analyze for the different cases of acceleration force  $m_s \ddot{x}$ . Eqn 4.22 is the a second order differential equation in the term of acceleration force. Right hand side of the equation may considered as two part of algebraic term named ‘A’ and ‘B’. For the better understanding of ADD working principle it is advice to remember the RHS terms of eqn 4.22 as ‘A’ and ‘B’

**Case (1):** At the any stage of the motion, let us assume that the acceleration force is positive or zero and for this value of acceleration force, the value of relative velocity can be positive or negative at any instant. If relative velocity is positive, then ‘A’ will be negative quantity and ‘B’ has to be a relatively big positive quantity. But our goal is not only to satisfy the equation, it is also to minimize the acceleration force acting on the sprung mass. The goal can be achieve by increasing the magnitude of ‘A’ and can be done by replacing  $b$  by its maximum value.

Now, if relative velocity is negative, then ‘A’ will be positive quantity and ‘B’ has to be either any positive quantity or a small negative quantity to satisfy the equation. If ‘B’ is any positive quantity, then to achieve the goal we need to minimize the damping value so that A and B together can give a less positive value. And if, ‘B’ is relatively a small negative quantity then also to achieve the goal, damping should be minimized. So to achieve this minimum value of damping, damper should be in off condition.

**Case (2)** Similar to the first case, at any instance of motion if the acceleration force is negative, the value of relative velocity can be negative or positive. If relative velocity is negative, then ‘A’ will be positive and to satisfy the equation, ‘B’ has to be a negative quantity and comparatively large in magnitude. To minimize the acceleration force, we require a large value of ‘A’ and it can be achieve by switching damper in its ‘on’ condition. If the relative acceleration is positive, ‘A’ will be a negative quantity and to satisfy the equation, ‘B’ has to be any negative quantity or relatively a small positive quantity. Similar to the first case, to achieve the goal minimum value of damping is required and damper can be set in its off state.

On the basis of above discussion a table can be prepare for the different cases. It is shown in the table 4.4. In table 4.4, items in ‘relative position of sprung mass’ column are compared with relative velocity of base.

Table 4.4: Illustration of the ADD control:

Case	Acc force	Relative velocity	Relative position of sprung mass	Damping value
1	$m\ddot{x} \geq 0$	$(\dot{x} - \dot{u}) \geq 0$	Negative and large in magnitude	On
2	$m\ddot{x} \geq 0$	$(\dot{x} - \dot{u}) < 0$	Any negative or small positive	Off
3	$m\ddot{x} < 0$	$(\dot{x} - \dot{u}) < 0$	Positive and large in magnitude	On
4	$m\ddot{x} < 0$	$(\dot{x} - \dot{u}) \geq \leq$	Any positive or small negative	Off

## 4.5.2 ADD 'On-Off' Control Simulation

The MATLAB simulation of 'ADD' suspension is seen for the time response of sprung mass vertical displacement and time response of sprung mass vertical acceleration. Fig 4.17 to Fig 4.20 shows plots for both the road profiles.

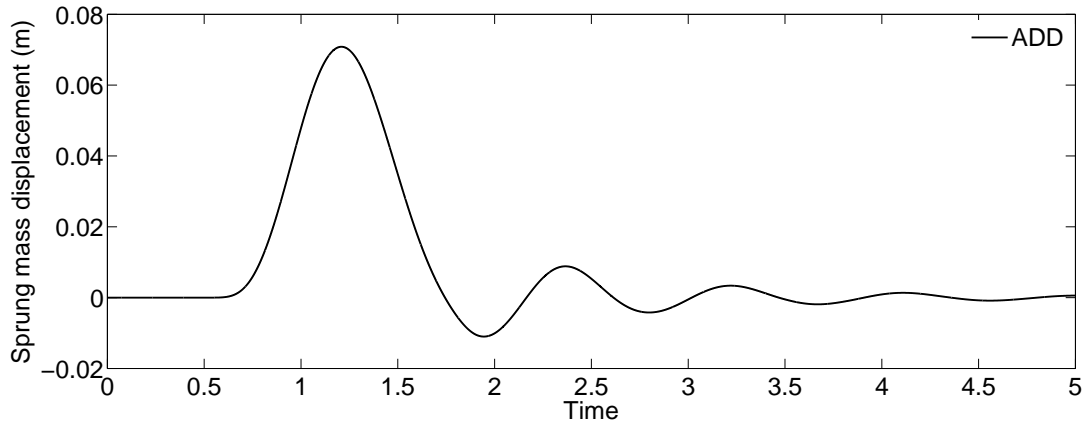


Figure 4.17: Sprung mass displacement for 'On-Off' ADD control with smooth breaker

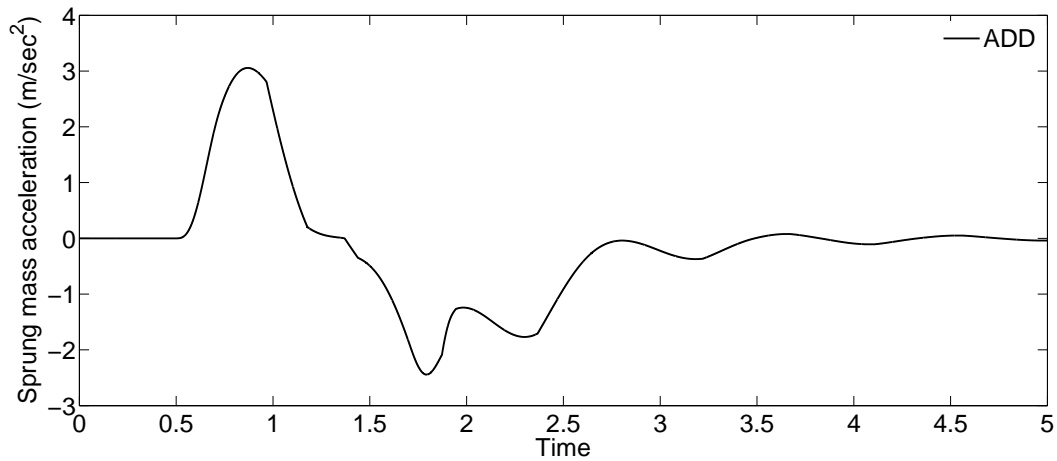


Figure 4.18: Sprung mass acceleration for 'On-Off' ADD control with smooth breaker

## 4.6 Conclusion

Table 4.5 shows the comparison of 'Balance', 'GH', 'SH' and 'ADD' control for sprung mass acceleration and displacement.

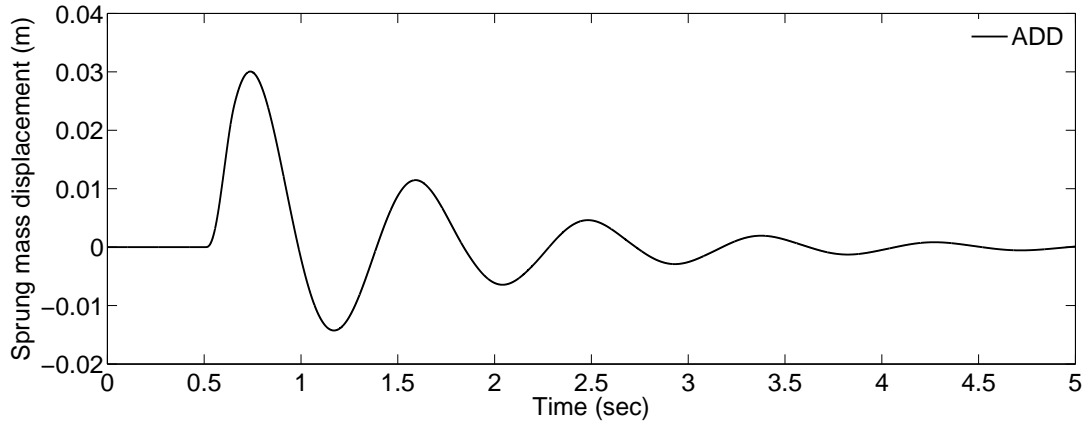


Figure 4.19: Sprung mass displacement for ‘On-Off’ ADD control with severe breaker

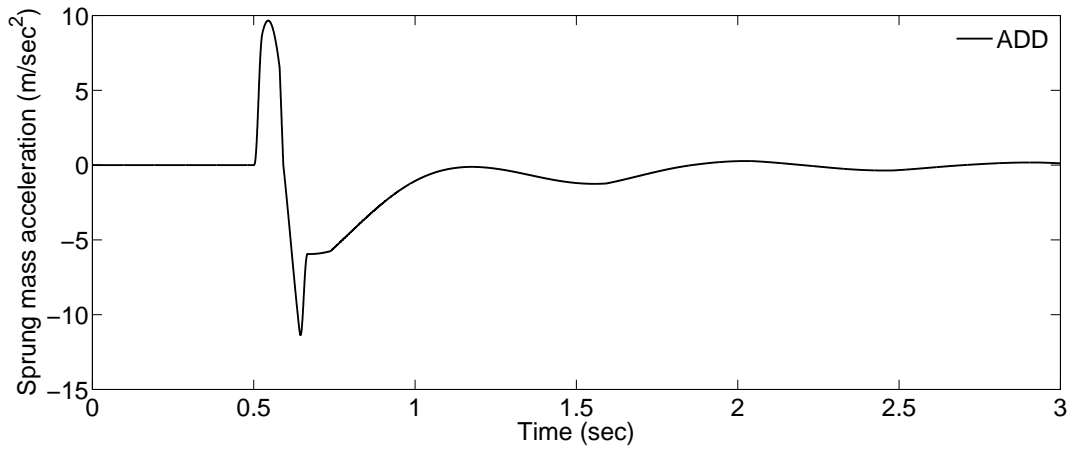


Figure 4.20: Sprung mass acceleration for ‘On-Off’ ADD control with severe breaker



Table 4.5: Comparison of different parameters for various two state ‘On-Off’ suspension systems

Type	Disp( $m$ ), smooth	Disp( $m$ ), severe	Acc( $m/sec^2$ ), smooth	Acc( $m/sec^2$ ), severe
Balance	0.0716	0.0174	3.056	23.26
GH	0.070	0.040	3.066	24.03
SH	0.074	0.019	3.056	22.23
ADD	0.074	0.030	3.056	11.36

From the insight of table we can conclude that the maximum amplitude of acceleration in ‘ADD’ suspension is lesser when bump is considered severe whereas, ‘SH’ and ‘ADD’ perform approximately same when bump is considered smooth. So, the ‘Balance logic’ and ‘GH’ control is not considered in the ‘Result and Discussion’ chapter.

# Chapter 5

## Improved Vehicle Semi-Active Suspension Model

### 5.1 ADD ‘Continuous’ Control

As it is said that ‘ADD’ control is similar to ‘SH’ control. ‘SH’ as well as ‘ADD’ follows the same ‘SH’ configuration to be realized with 1DOF suspension system with different control law. As it is said that the ‘SH’ configuration is used to realized the ‘ADD’ control scheme, damping value for the ‘ADD’ control will be same as ‘SH’ damping. i.e

$$b = \begin{cases} b_{sky} \frac{\dot{x}}{(\dot{x}-\dot{u})} & \text{if } \ddot{x}(\dot{x} - \dot{u}) \geq 0 \\ 0 & \text{if } \ddot{x}(\dot{x} - \dot{u}) < 0 \end{cases} \quad (5.1)$$

Now, we can see from the eqn 5.1, the minimum value of  $(\dot{x} - \dot{u})$  will cause the abrupt change in the value of semi-active damping coefficient and it tends to zero as this value reaches zero. But, we have the maximum and minimum limit of damping and we cant go beyond this limit. Also we cant get the zero damping even in off condition of damper, there will be some damping which can be termed as  $b_{min}$  It means that we have a upper and lower bond of damping coefficient as  $b_{max}$  and  $b_{min}$  respectively. So, we can write the damping coefficient in more suitable way as eqn 5.2. This is may be termed as the continuous control scheme of ‘ADD’ suspension system.

Simulation of continuous controlled ‘ADD’ suspension for sprung mass displacement and

sprung mass acceleration with three type of speed breaker is plotted below.

$$b = \begin{cases} \max[b_{min}, \min[b_{sky} \frac{\dot{x}}{(\dot{x}-\dot{u})}, b_{max}]] & \text{if } \ddot{x}(\dot{x} - \dot{u}) \geq 0 \\ b_{min} & \text{if } \ddot{x}(\dot{x} - \dot{u}) < 0 \end{cases} \quad (5.2)$$

## 5.2 JDD Suspension

### 5.2.1 Introduction

As it is discussed in the introduction chapter, ‘JDD’ is very similar to ‘SH’ control and uses same logic behind this. Only the difference found in ‘JDD’ is, it is a control based on the jerk force of sprung mass instead of velocity based control as it is in ‘SH’ control. Jerk is third time derivative of position and some time it is also refer as jolt. Some common terminology for jerk are, pulse, impulse, bounce, surge, shock and super-acceleration[46].

### 5.2.2 JDD ‘On-Off’ Control

Eqn 4.21 shows the second order differential equation in term of acceleration force. Let us differentiate the eqn 4.21 with respect of time to make it a third order differential equation in term of jerk force. Eqn 5.4 is shown as said above and it can be analyze for the different cases of jerk force  $m\ddot{x}$ .

$$m_s \ddot{x} = -b(\dot{x} - \dot{u}) - k(x - u) \quad (5.3)$$

$$m_s \ddot{\ddot{x}} = -b(\ddot{x} - \ddot{u}) - k(\dot{x} - \dot{u}) \quad (5.4)$$

Let us consider the first term of RHS as ‘A’ and second term of RHS as ‘B’. For two different possibility of jerk force (positive and negative) there will be four possibility (two for positive jerk force and two for negative jerk force) of relative acceleration. Eqn 5.4 shows the third order differential equation in term of jerk force and it can be use for the analysis as said above.

**Case(1):** At any stage of the motion, let us assume that the jerk force is positive or zero and for this value of jerk force, the value of relative acceleration can be positive of negative at any instant. If relative acceleration is positive, then ‘A’ will be negative quantity and ‘B’ has to be a relatively big positive quantity. But our goal is not only to satisfy the equation, it is also to minimize the jerk force acting on the sprung mass. The goal can be achieved by increasing the magnitude of ‘A’ and can be done by replacing  $b$  by its maximum value.

Now, if relative acceleration is negative, then ‘A’ will be positive quantity and ‘B’ has to be either any positive quantity or a small negative quantity to satisfy the equation. If ‘B’ is any positive quantity, then to achieve the goal we need to minimize the damping value so that ‘A’ and ‘B’ together can give a less positive value. And if, ‘B’ is relatively a small negative quantity then also to achieve the goal, damping should be minimized. So to achieve this minimum value of damping, damper should be in ‘Off’ condition.

**Case(2):** Similar to the first case, at any instance of motion if the jerk force is negative, the value of relative acceleration can be negative or positive. If relative acceleration is negative, then ‘A’ will be positive and to satisfy the equation, ‘B’ has to be a negative quantity and comparatively large in magnitude. To minimize the jerk force, we require a large value of ‘A’ and it can be achieved by switching damper in its ‘on’ condition. If the relative acceleration is positive, ‘A’ will be a negative quantity and to satisfy the equation, ‘B’ has to be any negative quantity or relatively a small positive quantity. Similar to the first case, to achieve the goal minimum value of damping is required and damper can be set in its ‘Off’ state.

A table can be prepared on the basis of above discussion for the different possibility of jerk force (refer to table 5.1). In table 5.1, items in ‘relative position of sprung mass’ column are compared with relative velocity of base.

Table 5.1: Illustration of the JDD control:

Case	Jerk force	Relative acc.	Relative vel. of sprung mass	Damping value
1	$m\dot{\ddot{x}} \geq 0$	$(\ddot{x} - \ddot{u}) \geq 0$	Negative and large in magnitude	On
2	$m\dot{\ddot{x}} \geq 0$	$(\ddot{x} - \ddot{u}) < 0$	Any negative or small positive	Off
3	$m\dot{\ddot{x}} < 0$	$(\ddot{x} - \ddot{u}) < 0$	Positive and large in magnitude	On
4	$m\dot{\ddot{x}} < 0$	$(\ddot{x} - \ddot{u}) \geq 0$	Any positive or small negative	Off

On the basis of above discussion presented in table 5.1 a control law for ‘JDD’ can be given in eqn 5.5.

$$b = \begin{cases} b_{max} & \text{if } \ddot{x}(\ddot{x} - \ddot{u}) \geq 0 \\ b_{min} & \text{if } \ddot{x}(\ddot{x} - \ddot{u}) < 0 \end{cases} \quad (5.5)$$

### 5.2.3 JDD ‘On-Off’ Control Simulation

The MATLAB simulation for the sprung mass displacement and sprung mass acceleration are done for different road input. Fig 5.1 to Fig 5.4 shows the time response of sprung mass displacement and sprung mass acceleration for ‘On-Off’ JDD suspension system.

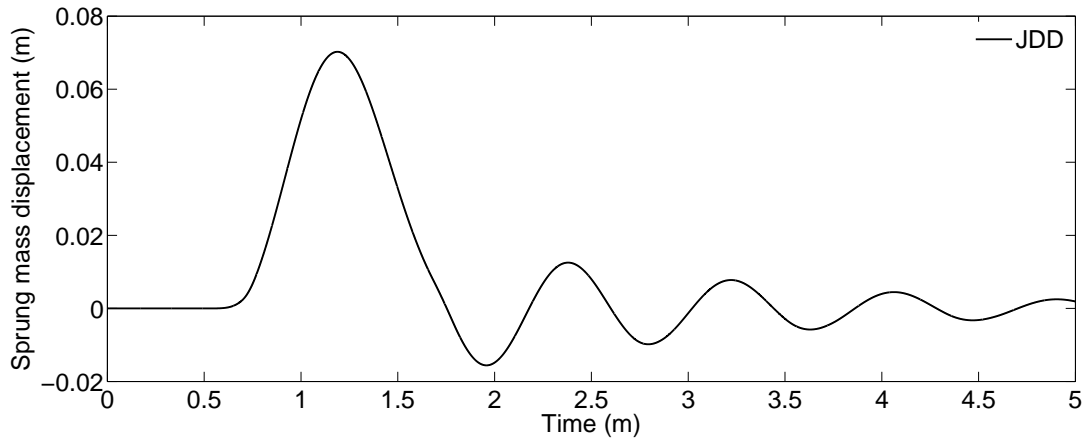


Figure 5.1: Sprung mass displacement for ‘On-Off’ JDD control with smooth breaker

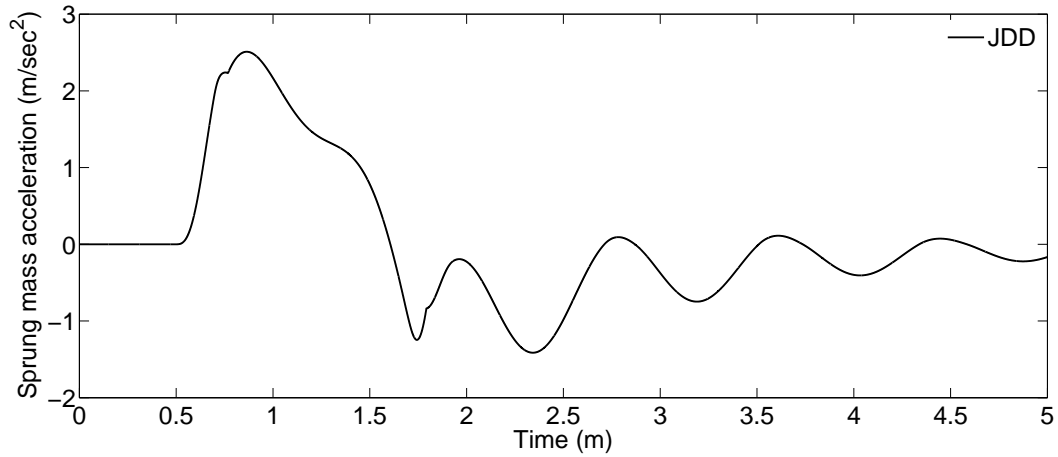


Figure 5.2: Sprung mass acceleration for ‘On-Off’ JDD control with smooth breaker

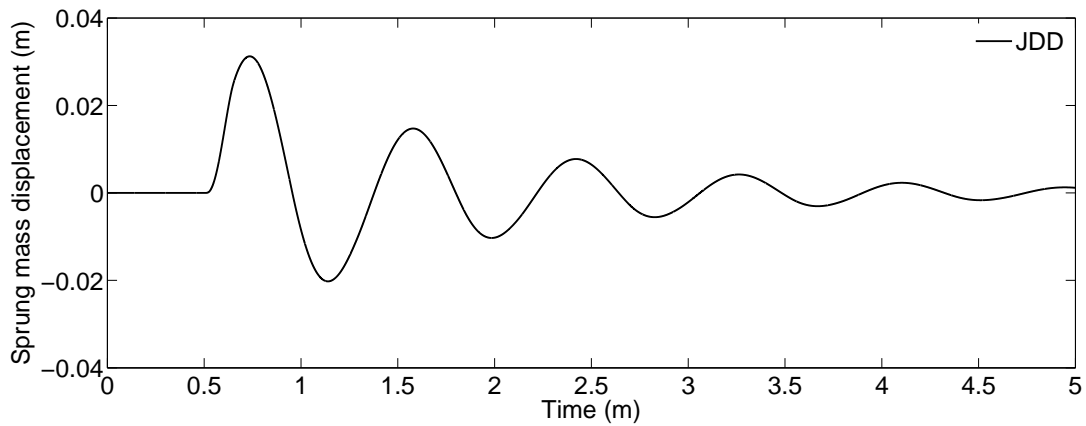


Figure 5.3: Sprung mass displacement for ‘On-Off’ JDD control with severe breaker

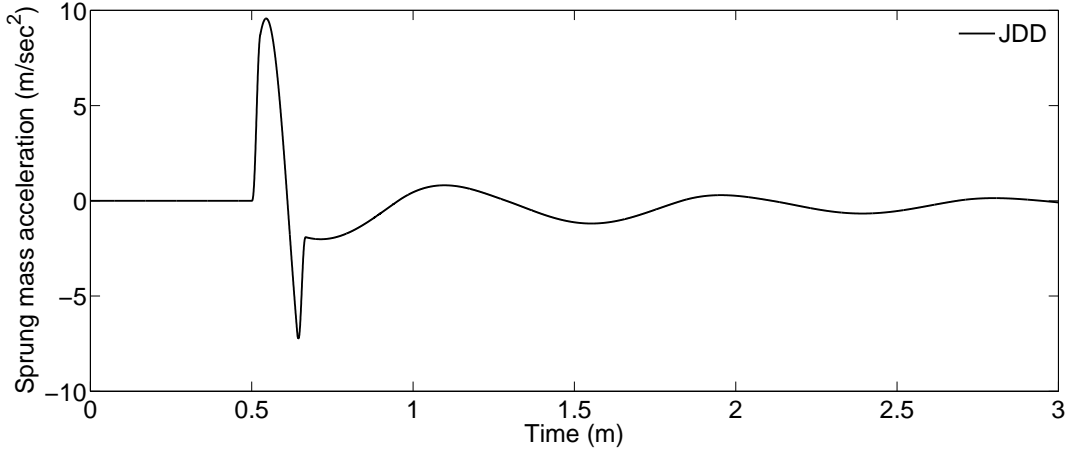


Figure 5.4: Sprung mass acceleration for ‘On-Off’ JDD control with severe breaker

### 5.2.4 JDD ‘Continuous’ Control

This is very similar to the continuous control of ‘ADD’ configuration. Now, let us write the third order differential equation of the skyhook configuration.

$$m_s \ddot{x} = -b\dot{x} - k(x - u) \quad (5.6)$$

Eqn 5.6 is the third order EOM of sky-hook configuration and we know the sky-hook configuration shows better frequency response than a passive configuration. Equating the eqn 5.6 and eqn 5.4 gives the required damping which is:

$$b = \begin{cases} b_{sky} \frac{\ddot{x}}{(\ddot{x} - \ddot{u})} & \text{if } \ddot{x}(\ddot{x} - \ddot{u}) \geq 0 \\ 0 & \text{if } \ddot{x}(\ddot{x} - \ddot{u}) < 0 \end{cases} \quad (5.7)$$

Now, we can see from the eqn 5.7, the minimum value of  $(\frac{\ddot{x}}{(\ddot{x} - \ddot{u})})$  will cause the abrupt change in the value of semi-active damping coefficient and it tends to zero as this value reaches zero. But, we have the maximum and minimum limit of damping and we cant go beyond this limit. Also we cant get the zero damping even in ‘Off’ condition of damper, there will be some damping which can be termed as  $b_{min}$ . It means that we have a upper and lower bond of damping coefficient as  $b_{max}$  and  $b_{min}$  respectively. So, we can write the damping coefficient in more suitable way as eqn 5.8:

$$b = \begin{cases} \max[b_{min}, \min[b_{sky} \frac{\ddot{x}}{(\ddot{x} - \ddot{u})}, b_{max}]] & \text{if } \ddot{x}(\ddot{x} - \ddot{u}) \geq 0 \\ b_{min} & \text{if } \ddot{x}(\ddot{x} - \ddot{u}) < 0 \end{cases} \quad (5.8)$$

### 5.3 Implementation of JDD

As described in the introduction chapter, ‘JDD’ control is based on two signals, relative acceleration and absolute jerk. The ‘JDD’ suspension can be controlled in the same way as SH suspension is controlled. A detailed study of SH suspension control for seat suspension is mentioned in [44]. In the ‘JDD’ suspensions, a jerk sensor and two accelerometers are used to measure the sprung mass jerk and the absolute acceleration of sprung mass and unsprung mass, respectively. A jerk sensor has been introduced to measure the jerk directly instead of using an accelerometer with rate filter [45]. Relative accelerations can be calculated from the absolute acceleration of sprung mass and unsprung mass. As shown in Fig 5.5, relative acceleration signals and jerk signals are used to JDD control to determine the control voltage in the power stage circuits. Through a power stage circuit, the control voltage is transferred to the corresponding current ‘ $I$ ’ for the MR damper. The damping can be changed according to the current ‘ $I$ ’ supplied to the damper. The current ‘ $I$ ’ which is determined by the JDD control logic can be decided by eqn 5.9:

$$I = \begin{cases} On, & \text{if } \ddot{x}(\ddot{x} - \ddot{u}) \geq 0 \\ Off, & \text{if } \ddot{x}(\ddot{x} - \ddot{u}) < 0 \end{cases} \quad (5.9)$$

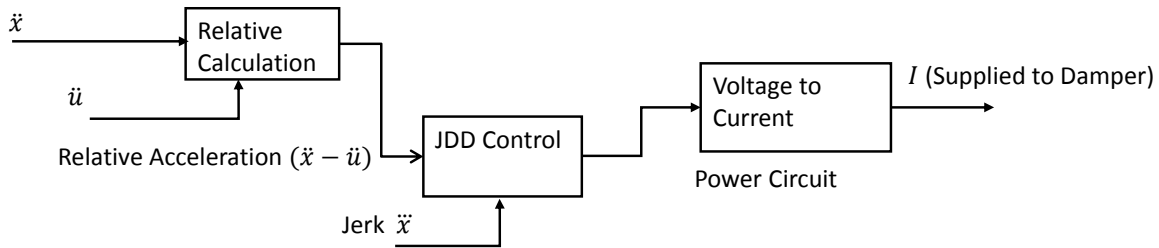


Figure 5.5: Implementation of JDD control in suspension system

# Chapter 6

## Result and Discussion

Fig 6.1 to Fig 6.4 represents the comparison of 'Passive', 'SH', 'ADD' and 'JDD' suspension systems for sprung mass acceleration and displacement. The comparison is presented for both two state 'On-Off' control and 'Continuous' control of semi-active suspensions. Later, a tabulated study is given for better understanding. Comparison plot of 'GH' and 'Balance logic' with other suspension are not given because of the bad performance of 'GH' and 'Balance logic'(refer to table 4.5)

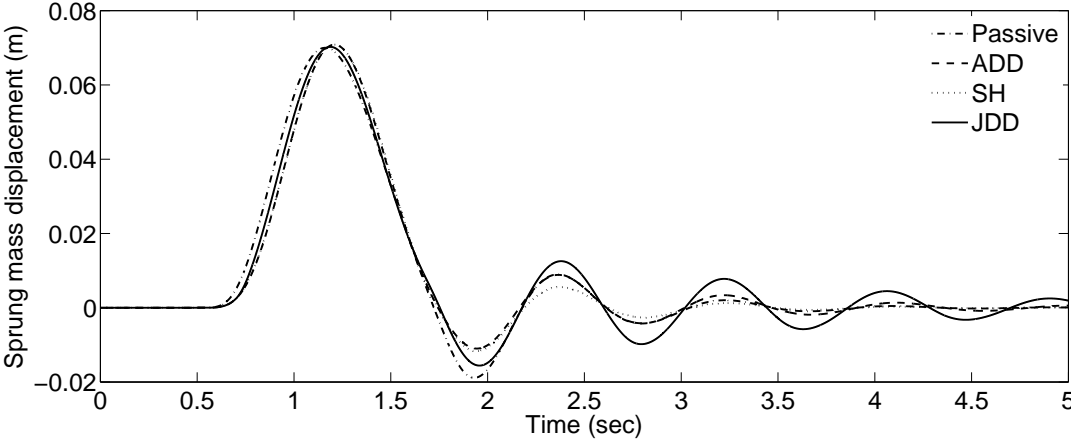


Figure 6.1: Comparison of sprung mass displacement for different 'On-Off' semi-active suspensions on smooth breaker

From the analysis of above plots from Fig 6.1 to Fig 6.4, a tabulated study can be presented. Table 6.1 comparison of different two state 'On-Off' semi-active suspension control and passive suspension for time response of sprung mass displacement and acceleration.



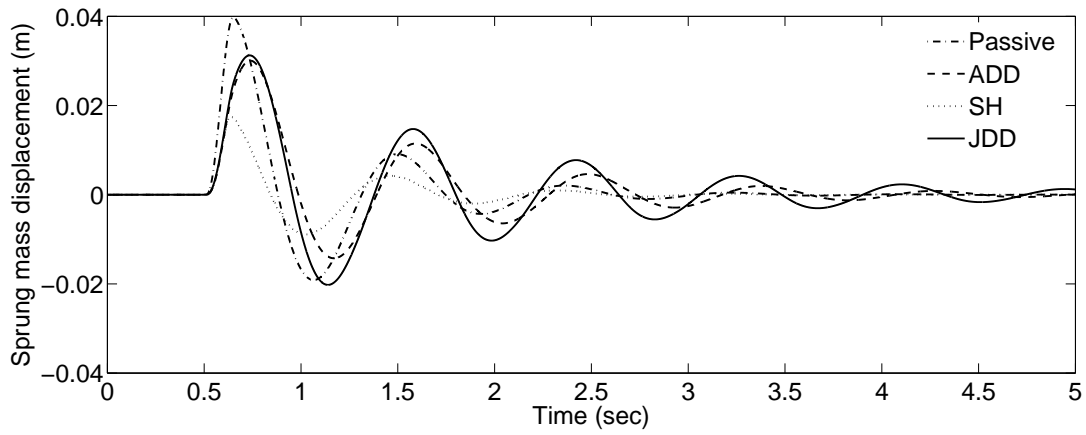


Figure 6.2: Comparison of sprung mass displacement for different ‘On-Off’ semi-active suspensions on severe breaker

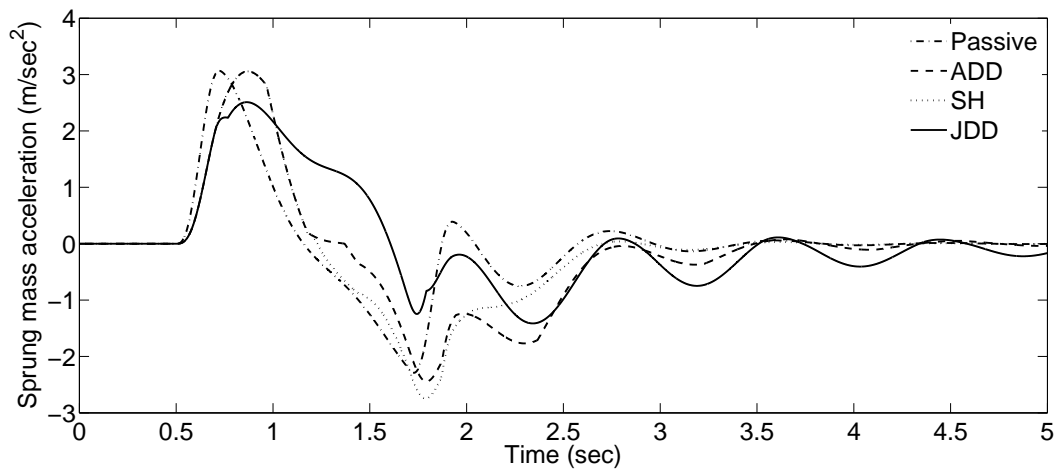


Figure 6.3: Comparison of sprung mass acceleration for different ‘On-Off’ semi-active suspensions on smooth breaker

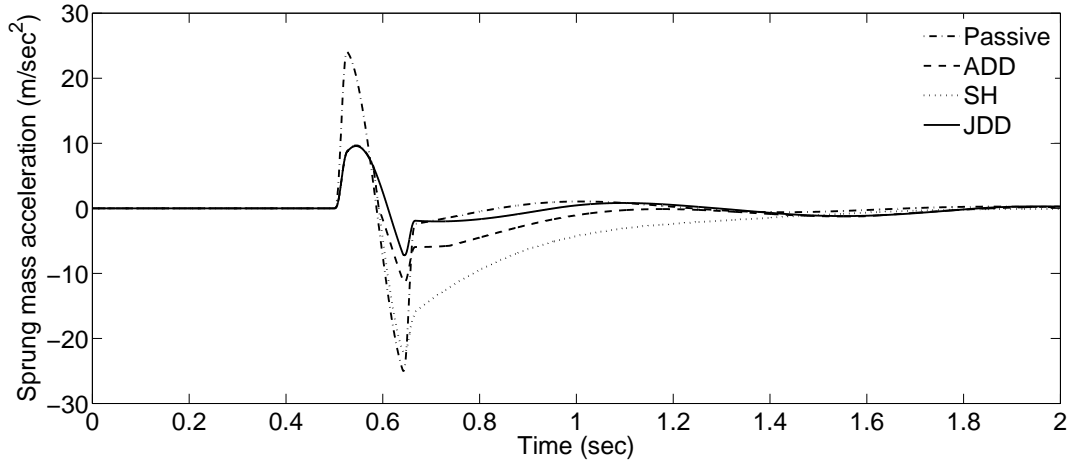


Figure 6.4: Comparison of sprung mass acceleration for different ‘On-Off’ semi-active suspensions on severe breaker

Table 6.1: Comparison of different parameters for various two state ‘On-Off’ suspension systems

Type	Disp( $m$ ), smooth	Disp( $m$ ), severe	Acc( $m/sec^2$ ), smooth	Acc( $m/sec^2$ ), severe
Passive	0.072	0.040	3.066	25.01
SH	0.074	0.019	3.056	22.23
ADD	0.074	0.030	3.056	11.36
JDD	0.070	0.031	2.51	9.57

# Chapter 7

## Conclusion and Future Scope

**Conclusion of the two state ‘On-Off’ control:** From the table 6.1, we can conclude for the sprung mass acceleration in both cases of road inputs. JDD suspension gives least peak value of sprung mass acceleration for both road inputs. The sprung mass acceleration of JDD with smooth breaker is 17.87% lesser than SH and ADD whereas, it is 15.76% lesser than ADD and 16.97% lesser than SH control when severe bump is input to the vehicle.

Peak value of displacement is same in all the suspensions (Refer to Fig 6.1) when smooth breaker is input to the vehicle. If we consider severe breaker as input, time response of sprung mass displacement shows better isolation in SH configuration. The peak value of displacement in this case is same for JDD add ADD and better than passive suspension.

But, our comfort objective is to minimize the peak value of sprung mass vertical acceleration. So, we can make a conclusion in one sentence as: Proposed ‘On-Off’ control of ‘JDD’ suspension perform better than ‘SH’ and ‘ADD’ control on both the speed breakers.

**Future scope:** Since the base of the semi-active vibration control is found in the early of 1970s, it has been advanced with the progress of computer technologies, nonlinear dynamic analysis techniques and smart materials. An advancement can be seen by a deep analysis of transmissibility plot for different semi-active suspensions and switching one control scheme to another one.

We have seen a better performance of ‘JDD’ control on the basis of its lower value of time response for sprung mass acceleration. But, Comparison plot for acceleration in chapter 5 shows that the settling time for acceleration is more in ‘JDD’ control. Though, the ‘SH’ control shows the minimum settling time for time response of sprung mass acceleration, a combination of ‘JDD’ and ‘SH’ control can be suggested. The frequency response plot of 1DOF suspension gives an idea to switch the damper from its minimum to maximum value

of damping. A cross point of frequency response for 'SH' and 'JDD' control can be seen from the transmissibility plot and according to this point need for switching from 'SH' to 'JDD' and vice versa can be suggested.

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