Sudan University of Science and Technology
College of Graduate Studies

Adaptive Vehicle Suspension System Using Microcontroller and Electro-hydraulic actuator

نظام التعليق المتكيف للسيارة باستخدام المعالج الدقيق والمشغل الكهروهيدروليكي

A thesis Submitted in Partial Fulfillment of the Requirement for the Degree of M.Sc. in Mechatronics Engineering

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الإستهلال

بِسْمِ ٱللَّهِ ٱلرَّحْمَٰنِ ٱلرَّحِيمِ

وَاللَّهُ أَخْرَجَكُمْ مِنْ بُطُونِ أُمَّهَاتِكُمْ لا تَعْلَمُونَ شَيْئًا وَجَعَلَ لَكُمُ السَّمْعَ وَالْأَبْصَارَ وَالْأَفْئِدَةَ لَعَلَّكُمْ تَشْكُرُونَ
﴿٧٨﴾

سورة النَّحْلْ

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I wish to express my sincere appreciation to honorable Associate Dr. Ebtihal Haider Gismalla my supervisor of this project. Thank you for the encouragement, guidance and critics. Without her continued support, idea and knowledge received, this thesis would not possible to complete. I thank my parents, siblings for their trust they had in me, by supporting me mentally and financially throughout my Master's degree. my appreciations and gratitude for DAL Motor for its generous assistance on my stage of collecting vehicle specifications. Finally I would like to thank all those who have contributed with their help, guidance, motivation and assistance during my thesis work, I truly love all of you.

Author

Ahmed Adam Mohamed

Abstract

The design of suspension system of today's light vehicles is getting very critical, owing to unbalancing in sharp turns and irregular road pattern which affect the comfort of the passengers and safety. This project deals with the design and development of an active suspension system for a light car. The idea of this adaptive system is to compensating the rough road profiles be adding a hydraulic damper controlled by microcontroller. The system in this thesis had been intendeded for Mitsubishi Triton GLX Plus (4×4), and includes the mechanical assembly, hydraulic circuit, control circuit and the controller program. Kinematics theory is been used to set up the active suspension system dynamic model, Through a Simulink software, simulation model was constructed. Different road surfaces has been modeled for the conventional and the modified systems models. The outputs displacement response comparisons between the old suspension and the modified system had recorded, and the obtained results had taken to the design stage. The closed-loop control algorithm in the Simulink model is interfaced to an Arduino Mega 2560 microcontroller through Deploy-To-hardware feature on the matlab software, to be used in the implementation stage at future works.

المستخلص

أصبح تصميم نظام التعليق للمركبات الخفيفة اليوم حرجا للغاية ، بسبب عدم التوازن في المنعطفات الحادة ونمط الطرق غير المنتظم الذي يؤثر على راحة الركاب والأمان. يتعامل هذا المشروع مع تصميم وتطوير نظام التعليق النشط لسيارة خفيفة. فكرة هذا النظام التكيفي هو تعويض ملامح الطريق الخشنة عن طريق إضافة المثبط الهيدروليكي التي تسيطر عليها متحكم دقيق. النظام المقترح في هذه الرسالة يستهدف ميتسوبيشي تريتون جي إل اكس بلس (٤ × ٤)، ويشمل التجميع الميكانيكي والدائرة الهيدروليكية ودائرة التحكم وبرنامج التحكم. تم استخدام نظرية الحركة في إعداد نموذج ديناميكي لنظام التعليق النشط، من خلال برنامج سيمولينك، تم بناء نموذج المحاكاة. تم تصميم أسطح الطرق المختلفة لنماذج الأنظمة التقليدية والمعدلة. سجلت مقارنات استجابة نزوح المخرجات بين التعليق القديم والنظام المعدل ، والنتائج التي تم الحصول عليها قد انتقلت إلى مرحلة التصميم. كانت خوارزمية التحكم في الحلقة المغلقة في نموذج سيمولينك تتفاعل مع متحكم اردوينو ميجا ٢٥٦٠ من خلال ميزة النشرالالعتاد المادي في برنامج ماتلاب، ليتم استخدامها في مرحلة التنفيذ في الأعمال المستقبلية.

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List of Abbreviations

DCV	Directional control valve
ECU	Electronic Control Unit
FASS	Fully active suspension system
MC	MicroController
PID	Proportional-Integral-Derivative
PWM	pulse width modulation
SASS	semi-active suspension System

List of Symbols

$(\cdot)^T$	matrix transpose operator
$\dot{(\cdot)}$	first derivative operator
$\ddot{(\cdot)}$	second derivative operator
s	Laplace complex variable
m_1	Sprung Mass (Vehicle mass)
m_2	Unsprung Mass (Tire mass)
k_1	Spring Stiffness
k_2	Tire Stiffness
D	Damping Coefficient
$\mathbf{X}$	state variable matrix in SS model
$\mathbf{A}$	state (or system) matrix in SS model
$\mathbf{B}$	input matrix in SS model"
$\mathbf{U}$	input (or control) vector in SS model
$\mathbf{Y}$	output variable matrix in SS model
$\mathbf{C}$	output matrix in SS model
$\mathbf{D}$	feedthrough (or feedforward) matrix in SS model
$e(t)$	error value in PID controller
$r(t)$	given value in PID controller
$c(t)$	actual value in PID controller
K	controller proportional gain
K_p	proportional gain
K_I	integral gain
K_D	differential gain
T	sample time
T_I	controller integral time
T_D	controller differential time

u	active force
l	proportionality coefficient
$x_i(t)$	road input
$w(t)$	white noise generator
D_p	Piston diameter
A_p	Area of Piston
X_{act}	the actuation stroke
P	actual value in PID controller
P_o	Applied Pressure
V	Volume
M	Mass

Chapter one
Introduction

1.1 Introduction

All of vehicles moving on the different profiles road are exposed to vibrations which are harmful both for the passengers in terms of comfort and for the durability of the vehicle itself. Therefore the main task of a vehicle suspension is to ensure ride comfort and road holding for a variety of road conditions. Any suspension system in the vehicle must be soft against road disturbances and hard against load disturbances [1]. Therefore the Suspension system is the most significant part which heavily affects the vehicle handling performance and ride quality. Traditional suspension systems use a combination of sets of springs and sets of dampers to decrease the vehicle vibration [2].

A Basic automobile suspension that is known as a *passive suspension system* consists of an energy storing element normally a spring and an energy dissipating element normally a shock absorber. The main weakness of the passive suspension is that it is unable to improve both ride comfort and safety factor simultaneously. In the passive suspension system, there is always trade-off between vehicle ride comfort and safety factor. To improve the ride comfort, the safety factor must be sacrificed, and vice versa. One way to overcome such a problem, the car suspension system must be controlled [3, 4].

1.2 Problem Statement

A major issue is that the vehicle suspension system is responsible for driving comfort and safety as the suspension carries the vehicle body and transmits all forces between body and road. The suspension systems have traditional mechanical structure it consists of springs, dampers and oriented institutions. Passive suspension cannot adjust stiffness and damping, according to the random vibration theory, it can only ensure the specific operating conditions to achieve optimal spring and damping effects, it is difficult to adapt to different road and tough use, also difficult to acquire good ride comfort and handling stability, and a ride is considered to be comfort when the natural frequency is around 1-1.5 Hz. When the natural frequency has reached 2 Hz, it is considered as an uncomfortable ride.

1.3 Proposed Solution

Previous studies, e.g., [5], put an actuator to the suspension system mathematically by adding PID controller to the model the thesis Development of MatLab Simulink model for dynamics analysis of passive suspension system for lightweight vehicle. Second, an analytical design of PID controller for enhancing ride comfort of active vehicle suspension system was proposed in [6]. In the same track, this thesis takes further steps by building a structural design for adaptive suspension based on a microcontroller used to control the level of the vehicle using a hydraulic actuator.

1.4 Objectives

- Reducing the settling time and over shoot of overall vehicle body of the conventional suspension, by adding a adaptive hydraulic actuator to the suspension system.
- The hydraulic actuator controlled by a Microcontroller, it programed by the mathematical models. The results monitored by simulation software. When the settling time be reduced the vehicle vibrations will scaled down, and that will allow vehicle to more comfort ride, and reduced the hazard of losing the balance while driving.

1.5 Methodology

- Determine a vehicle which using spring suspension System, And generate a mathematical model from its suspension System.
- Using a simulation software apply the vehicle suspension model for rough road profile inputs and record the response.
- Modifying the conventional suspension by adding an actuator controlled by Controller

- Using simulation software for effect rough road and record the response.

1.6 Thesis Outline

The rest of this thesis is organized as follows. Chapter two presents preliminary material about the suspension systems types and discusses its comparisons. Chapter three make the a quarter mathematical model of the conventional and the modified suspension systems, at chapter four describes the simulation stage and the comparison of the old and the modified suspension systems. chapter five describes the design circuits and specifications. Chapter six the thesis conclusion and recommendations.

Chapter Two
Background and Literature Review

2.1 Background

The roles of a suspension system are to support the vehicle weight, to isolate the vehicle body from road disturbances, and to maintain the traction force between the tire and the road surface. The purpose of suspension system is to improve the ride comfort, road handling and stability of vehicles. For vehicle suspension system design, it is always challenging to maintain simultaneously a high standard of ride, handling, and body attitude control under all driving conditions. The problems stem from the wide range of operating conditions created by varying road conditions, vehicle speed, and load. In general, during cornering, braking, and bumping, a high stiffness and damping is needed to provide good handling properties, and to satisfy workspace limitations of the suspension system [7].

2.2 Suspension Systems

A car suspension system is the mechanism that physically separates the car body from the wheels of the car. The purpose of suspension system is to improve the ride comfort, road handling and stability of vehicles [8].

The suspension systems is used to isolate road vibrations from passengers to improve ride comfort. Its history begins with the passive suspension system. Which only absorbs vibration energy and its characteristics cannot be modified to suit different road conditions. This deficit led to the development of semi-active and active suspension systems. A semi-active suspension system consists of a spring and a damper, which can absorb vibration energy as the passive one does, but its damper's characteristics can vary depending on the road conditions. An active suspension can generate additional force to improve ride comfort and road holding. Active suspensions are more efficient in reducing vibration, it is, however, more expensive than a semi-active

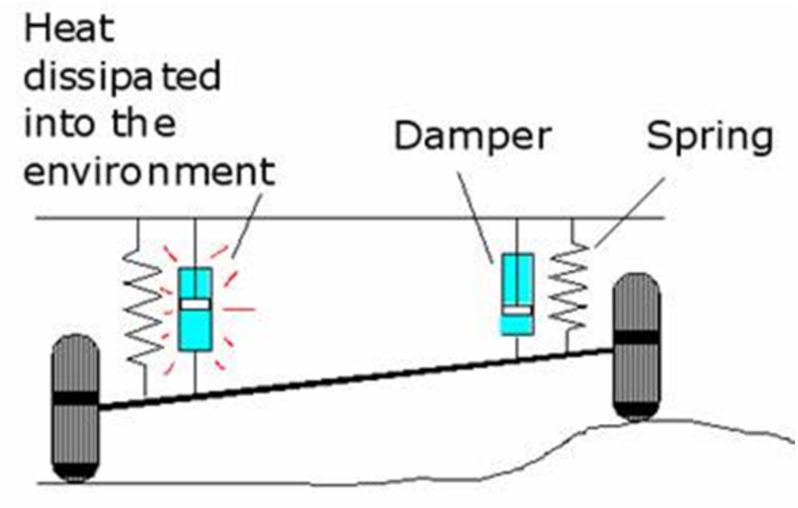

Figure 2.1: Suspension System of Light Vehicle

suspension system [9]. the suspension system can be divided into passive suspension, semi-active suspension and active suspension, these suspension system is very different in the performance aspects.

2.2.1 Passive Suspension System

Passive suspension systems are widely used to reduce and isolate road vibrations. A passive suspension system consists of a passive spring and a damper.

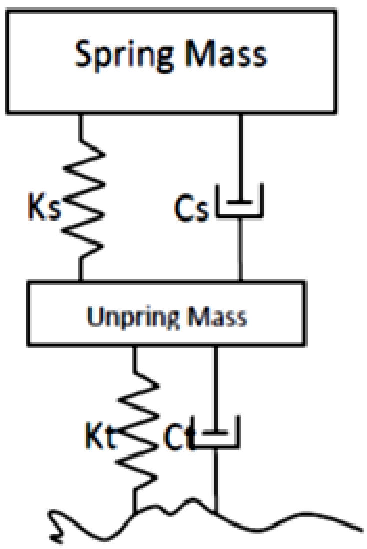

Figure 2.2: Quarter Vehicle Passive Suspension System

The spring could be coil, leaf or air spring [9]. There is either gas or oil fluid inside a damper, which goes through a constriction valve every time the damper moves. Each damper has its own damping coefficient. The characteristics of the fluid and restriction give the damper dynamic properties.

2.2.2 Semi Active Suspension System

A semi active suspension consists of a spring and a damper but, unlike a passive suspension, the value of the damper coefficient f can be controlled and updated. In some types of suspensions that it continuously adjusts damping levels according to road conditions and vehicle dynamics, such as speed, turning and cornering, delivering comfort without sacrificing the safety of sure handling [10].

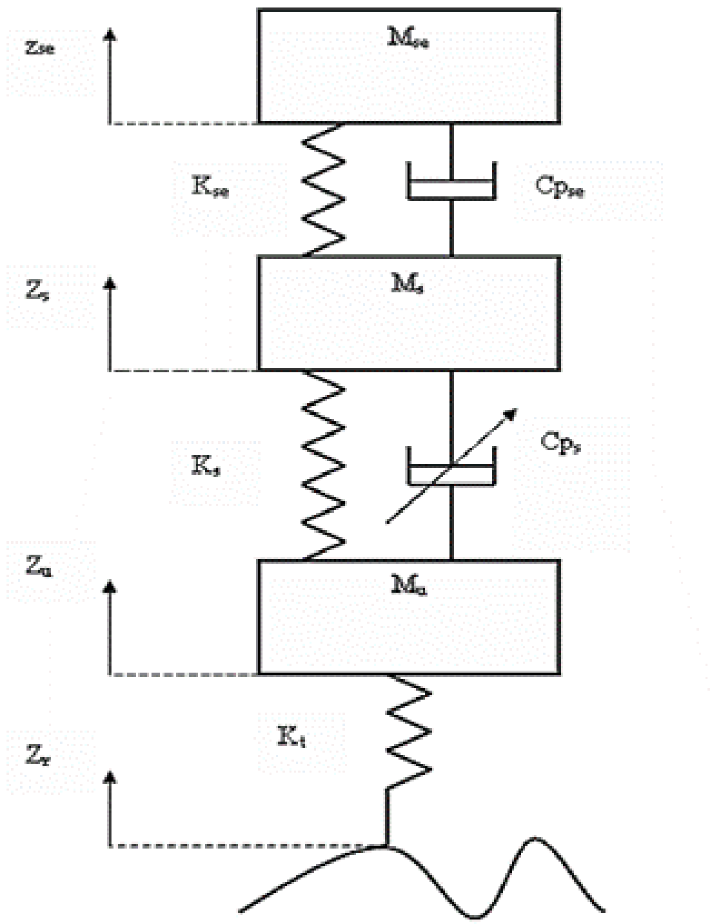

Figure 2.3: Semi Active Suspension System

Semi-active suspension system has no specifically dynamic control components. It passed the velocity sensors data to the vehicle ECU controller

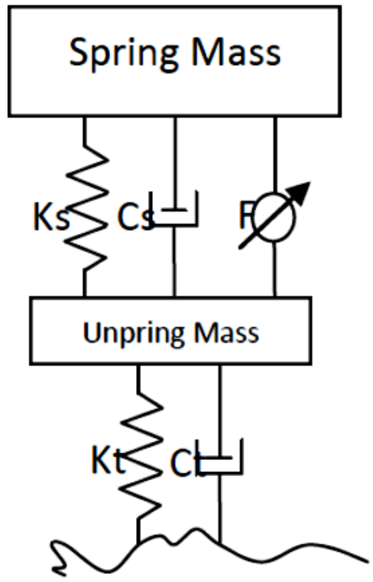

Figure 2.4: Active Quarter Car Model

to calculate the required control force, and then adjust the shock absorber damping to attenuate vibration of the vehicle body. The study of semi-active suspensions are also focused on two aspects, one is the research on actuator, which is the damping adjustable shock absorbers; the other hand, the control strategy. Magneto-rheological damper is the semi-active suspension is new technology It has a quick response, which is lower than 1ms [2].

2.2.3 Active Suspension System

Active suspension systems add hydraulic actuators to the passive components of suspension system. The advantage of such a system is that even if the active hydraulic actuator or the control system fails, the passive components come into action [10]. The advantage of such a system is that even if the active hydraulic actuator or the control system fails, the passive components come into action [8].

In recent years, considerable interest has been generated in the use of active vehicle suspensions, which can overcome some of the limitations of the passive suspension systems. Demands for better ride comfort and controllability of road vehicles has motivated many automotive industries to consider the use of active suspension electronically. Controlled active suspension systems can potentially improve the ride comfort as well as the road handling of the vehicle simultaneously [2], the active suspension is to control body movement. When the car is braking or turning, inertia will generate caused by the spring deformation, active suspension will produce an inertia force against this force, to reduce the changes of body position [2]. Fully active suspension system (FASS) is differentiated from semi-active suspension that System (SASS) consists of a separate active force generator. The physical Implementation of (FASS) is usually provided with a hydraulic actuator and power supply.

2.2.4 Suspension Systems Comparison

In active suspension systems, sensors are used to measure the acceleration of sprung mass and unsprung mass and the analog signals from the sensors are sent to a controller. The controller is designed to take necessary actions to improve the performance abilities already set. The controller amplifies the signals and the amplified signals are fed to the actuator to generate the required forces to form closed loop system (active suspension system), which is schematically depicted [10]. Vehicle suspension system performance is typically rated by its ability to provide improved road handling and improved passenger comfort.

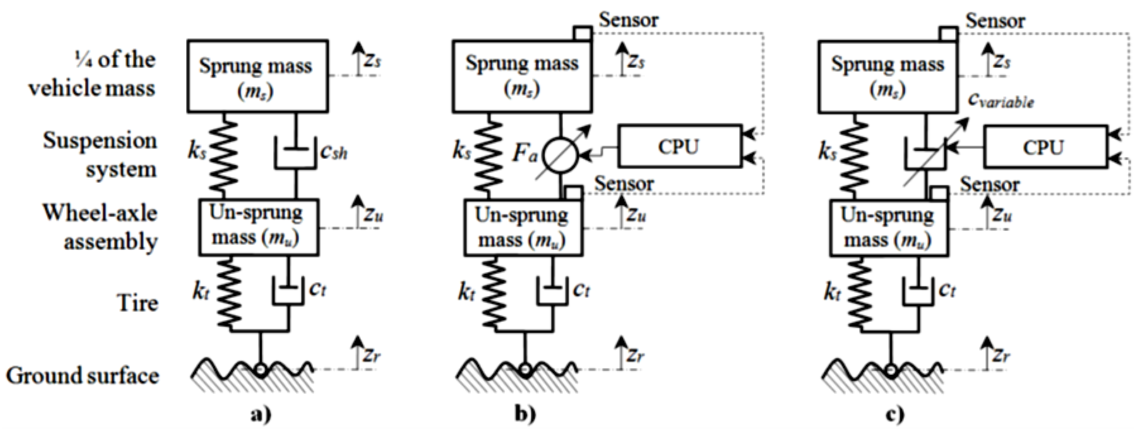

Figure 2.5: Quarter vehicle models; a) passive, b) active, and c) semi-active suspension systems [11]

Current automobile suspension systems using passive components can only offer a compromise between these two conflicting criteria by providing spring and damping coefficients with fixed rates . Poor road handling capability and decreased passenger comfort are due to excess car body vibrations resulting in artificial vehicle speed limitations, reduced vehicle-frame life, biological effects

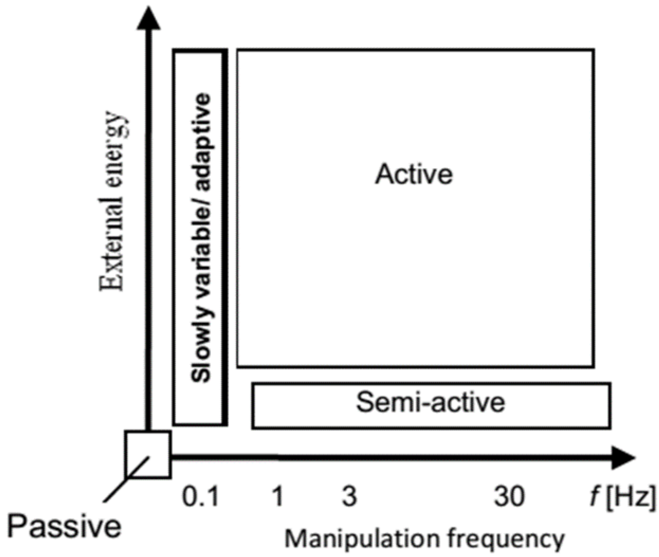

Figure 2.6: Comparison Between Passive, Adaptive, Semi-Active and Active Systems [7].

on passengers, and detrimental consequences to cargo. Active suspension control systems aim to ameliorate these undesirable effects by isolating the car body from wheel vibrations induced by uneven terrain [12, 13].

This case is not considered at the semi active suspension , and that's due its ability to control the elastic constant ¸s of the spring. A semi active suspension is a valid engineering solution when it can reasonably approximate the performance of the active control because it requires a low power controller that can be easily realized at a lower cost than that of a fully active one [14].

At another side the active suspension is taking the action by prevent vehicle shocks nearly zero even if the road profile is so hard and that due its close loop control system.

Table 2.1: Performance Comparison of Three Type Suspensions [2].

Suspension system type	Passive suspension system	Semi-active suspension system	Active suspension system
Regulatory element General	shock absorber	Adjustable damper	Hydraulic system or Servo motor system
Action principle	Damping constant	Damping Continuously adjust	Adjust the force between wheel and vehicle body
Control method	No	Electronic-hydraulic automatic	Electronics or magnetic or fluid control
Bandwidth Middle Highest	Unknown	Up to 20Hz	>15Hz
Energy consumption	Zero	Very small	Big
Lateral dynamics characteristics	No	Middle	Good
Vertical dynamics characteristics	No	Middle	Good
Cost	Lowest	Middle	Highest

Chapter Three
Mathematical Modeling and System Design Approach

3.1 Introduction

The mathematical model and the simulation made by following chapters are discussing the behaviors of the passive and active suspension. Most of the researchers chose to use quarter vehicle dynamic vibration model when they focus on the vehicle body vertical vibration caused by the input of pavement roughness. Although quarter vehicle dynamic vibration model has not included the entire vehicle geometrical information, it has nearly included most of the essential feature, such as the change of the load and suspension system's stress information.

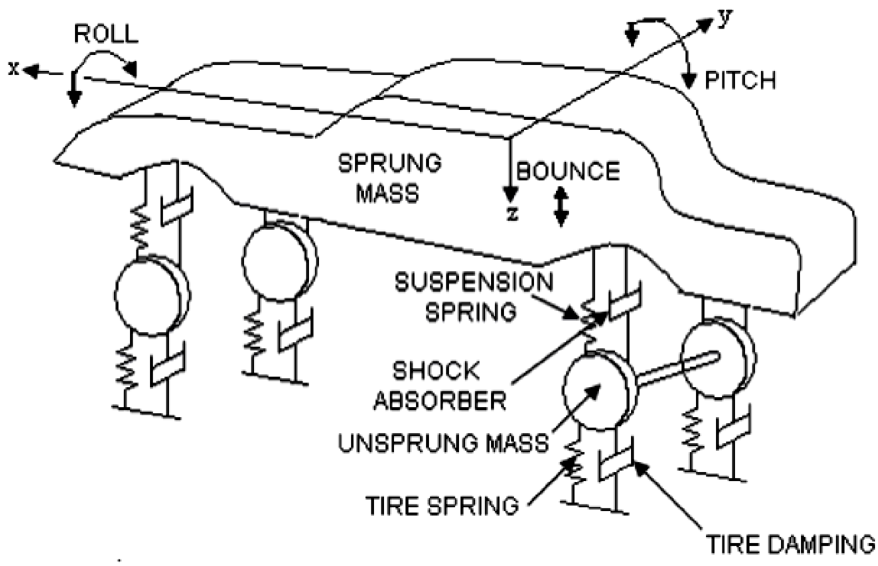

Figure 3.1: Suspension system of a passenger car

3.1.1 Passive Suspension Modeling

Physical models for the investigation of vertical dynamics of suspension systems are most commonly built on the quarter-car model as shown in figure

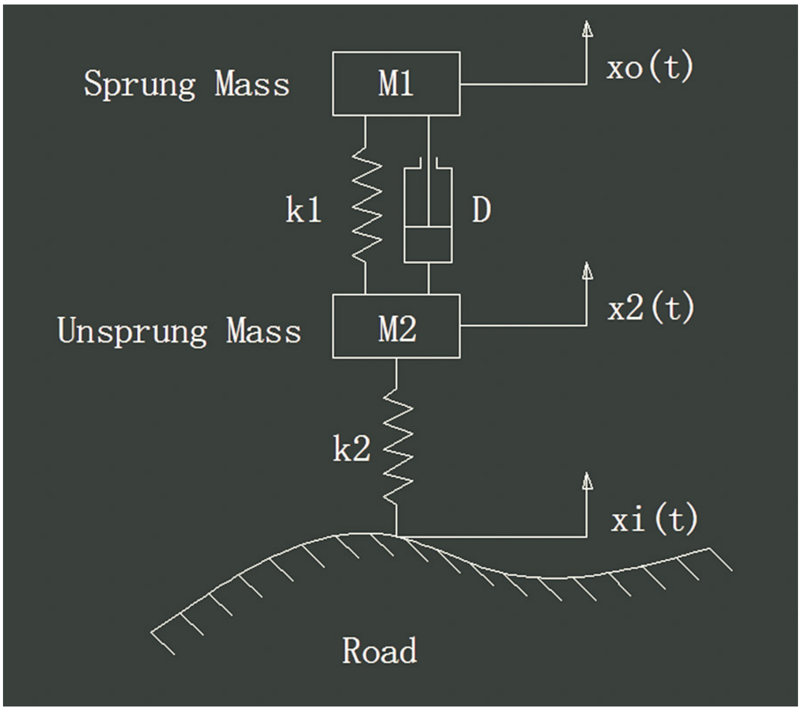

Figure 3.2: Quarter vehicle model of the passive suspension system

3.2,where M_1 is Sprung Mass (Vehicle mass), M_2 is Unsprung Mass (Tire mass), K_1 is Spring Stiffness K_2 is Tire Stiffness, D is Damping Coefficient

The system can be described mathematically as

$$m_1\ddot{x}_0(t) + D\left[\dot{x}_0(t) - \dot{x}_2(t)\right] + k_1\left[x_0(t) - x_2(t)\right] = 0 \tag{3.1}$$

$$m_2\ddot{x}_2(t) - D\left[\dot{x}_0(t) - \dot{x}_2(t)\right] + k_1\left[\dot{x}_2(t) - \dot{x}_0(t)\right] + k_2\left[\dot{x}_1(t) - \dot{x}_0(t)\right] = 0 \tag{3.2}$$

If we make a Laplace transformation to the above equation, we can get equation:

$$\frac{X_0}{X_1} = \frac{k_2\left(Ds + k_1\right)}{M_1M_2s^4 + \left(M_1 + M_2\right)Ds^3 + \left(M_1k_1 + M_1k_2 + M_2k_1\right)s^2 + Dk_2s + k_1k_2} \tag{3.3}$$

The above equation is the passive suspension system transfer function.

Quarter model of passive suspension system can simulate the vehicle suspension system when it drives on the road. It can better shows the influence which make by springs and damping of the suspension system.

3.1.2 Active Suspension Modeling

Transfer-functions of the quarter-car model for the normalized body acceleration and tire load in respect to the road excitation are shown in Figure

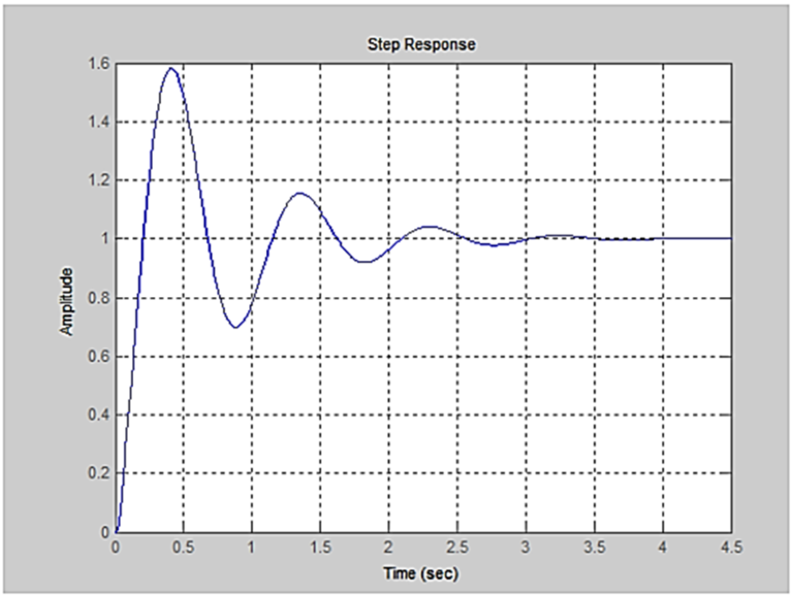

Figure 3.3: The passive suspension time domain response with unit step signal

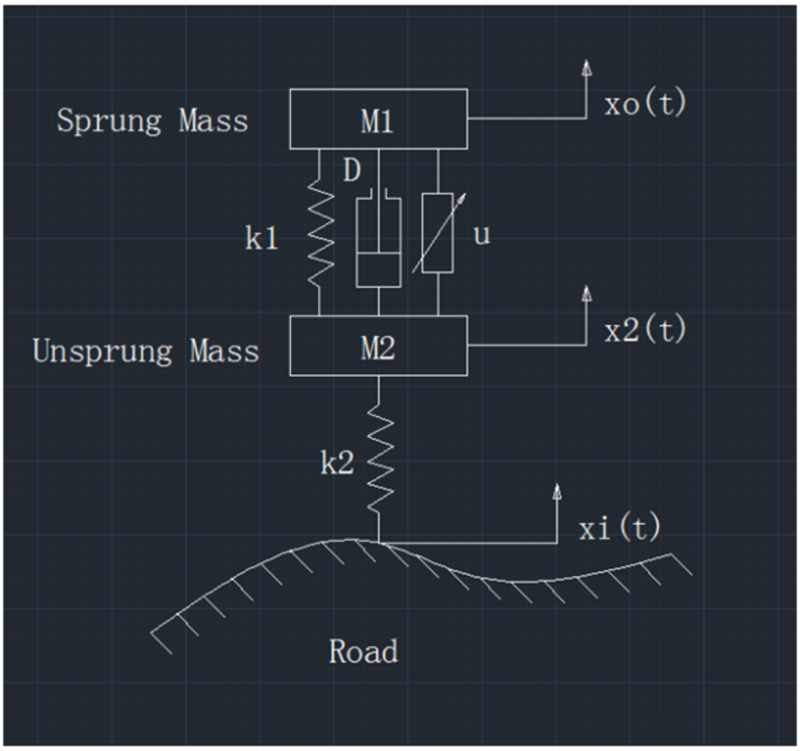

Figure 3.4: Quarter Model of active suspension system with actuating force (u) between sprung and unsprung mass.

According to the above assumption, figure 5 shows a vehicle ¼ model of active suspension system, the parameters M_1, M_2, K_1, K_2, D, $x_i(t)$, $x_o(t)$, $x_2(t)$ represent as same as in the passive suspension model, while u is the active control force, which created by the active suspension actuator. From this model, the can be analyzed by the vehicle suspension system dynamics and establish two degrees of freedom motion differential equations as follow:

$$m_1\ddot{x}_0(t) + D\left[\dot{x}_0(t) - \dot{x}_2(t)\right] + k_1\left[x_0(t) - x_2(t)\right] = u \tag{3.4}$$

$$m_2\ddot{x}_2(t) - D\left[\dot{x}_0(t) - \dot{x}_2(t)\right] + k_1\left[\dot{x}_2(t) - \dot{x}_0(t)\right] + k_2\left[\dot{x}_1(t) - \dot{x}_0(t)\right] = -u \tag{3.5}$$

We can set:

$$x_1 = x_2(t) \tag{3.6}$$

$$x_2 = x_0(t) \tag{3.7}$$

$$x_3 = \dot{x}_2(t) \tag{3.8}$$

$$x_4 = \dot{x}_0(t) \tag{3.9}$$

The system state space equation can be express as:

$$\frac{d\mathbf{X}}{dt} = \mathbf{AX} + \mathbf{BU}, \tag{3.10}$$

where in this equation, state variable matrices are

$$\mathbf{X} = \begin{bmatrix} x_1 & x_2 & x_3 & x_4 \end{bmatrix}^T \tag{3.11}$$

Constant matrices $\mathbf{A}$ and $\mathbf{B}$ are shown as

$$\mathbf{A} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_1+k_2}{m_2} & \frac{k_1}{m_2} & -\frac{D}{m_2} & \frac{D}{m_2} \\ \frac{k_1}{m_1} & -\frac{k_1}{m_1} & \frac{D}{m_1} & -\frac{D}{m_1} \end{bmatrix} \tag{3.12}$$

$$\mathbf{B} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ \frac{k_2}{m_2} & \frac{1}{m_2} \\ 0 & -\frac{1}{m_2} \end{bmatrix} \tag{3.13}$$

The system input variable matrix is

$$\mathbf{U} = \begin{bmatrix} x_i(t) & U(t) \end{bmatrix} \tag{3.14}$$

The vehicle suspension system output matrix equation is

$$\mathbf{Y} = \mathbf{CX} + \mathbf{DU} \tag{3.15}$$

In above equation, the output variable matrix $\mathbf{Y}$ is:

$$\mathbf{Y} = \left\{ k_2 \left[x_i(t) - x_2(t) \right] \quad \ddot{x}_0(t) \quad x_0(t) \right\}, \tag{3.16}$$

and $\mathbf{C}$ and D in (3.15) can be expressed as

$$\mathbf{C} = \begin{bmatrix} -k_2 & 0 & 0 & 0 \\ \frac{k_1}{m_1} & -\frac{k_1}{m_1} & \frac{D}{m_1} & -\frac{D}{m_1} \\ 0 & 1 & 0 & 0 \end{bmatrix} \tag{3.17}$$

$$\mathbf{D} = \begin{bmatrix} k_2 & 0 \\ 0 & -\frac{1}{m_2 n} \\ 0 & 0 \end{bmatrix} \tag{3.18}$$

The simulink software blocks can be used for modeling passive suspension system by means of state space approach. The various matrices are entered in to the simulink block and the response for the given input is obtained. The above equations are state space formulas, and we can use Matlab/Simulink State-Space function module. As same as the test we have done in the part of Vehicle Passive Suspension System Mathematical Model Establishment, we still use step signal to test our system, the Simulink structure form is as shown in Figure 3.5.

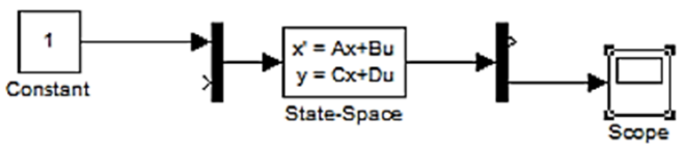

Figure 3.5: State Space Description Simulink Model

In this test, we do not add U matrix, so it is a passive suspension system, which is as same as the system built in section 3.1, we can see the result in Figure 3.6 which is as same as Figure 2.3, but by using different method.

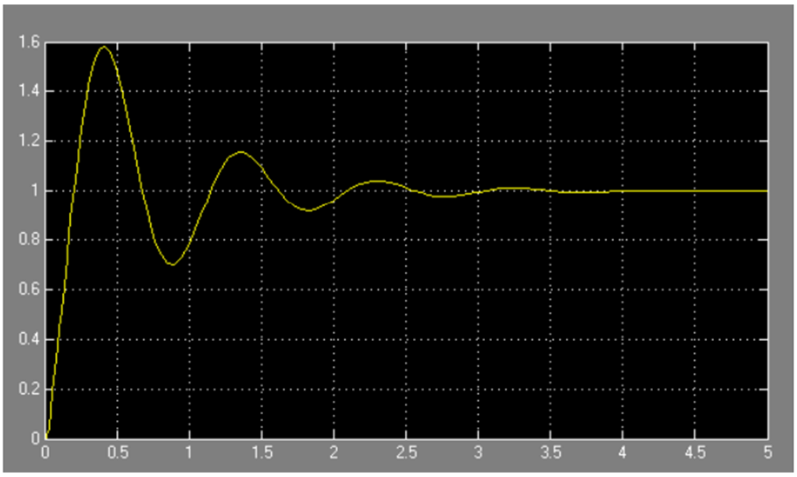

Figure 3.6: The active suspension time domain response with unit step signal

3.2 PID Model Considerations

PID controller is a classical loop-shaping with a proportional-integral-derivative controller. It is widely used in industrial control systems. The PID controller creates an input signal to the system process by attempting to correct the error between a demanded reference signal and the actual output signal [15,16].

There are lots of aspects that vibration affects people, such as acceleration, amplitude, frequency and action directions. Among them, the top reasons that vibration causes people uncomfortable is acceleration and amplitude [17]. this chapter applying evaluate standard using The vehicle body vertical displacement, which is $x_o(t)$. and make comparing of the conventional suspension system (Passive) and the modified suspension system (Active) with different Road inputs by outputting their vehicle body vertical displacement.when The lower $x_o(t)$ means the better performance.

Digital PID algorithm implemented by the computer, due to the flexibility of the software system, the PID algorithm has been further revised and improved. In industrial process control, although automatic control theory and the rapid technological development, especially the development of modern control theory and computer 34 technology, contributed greatly to the industrial automation process, but the PID control technology is still a dominant position, the current control PID type accounts for 84.5%, and optimization of the PID type occupies 6.8%. Conventional PID control principle framework shown in Figure 16, the system consists of the PID controller and the controlled object.

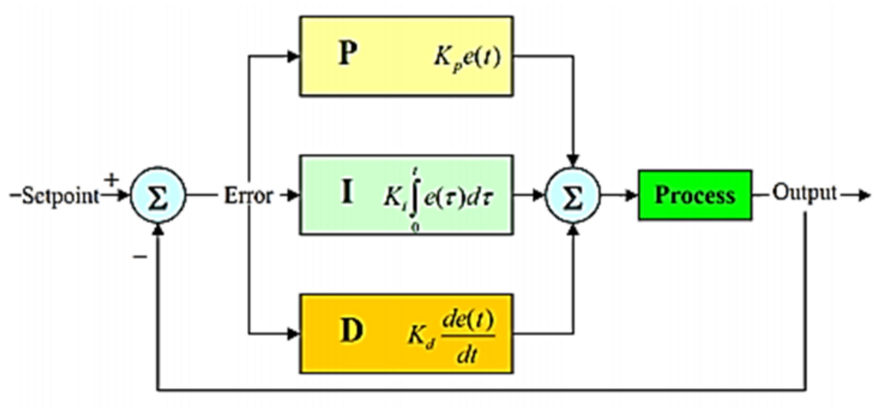

Figure 3.7: Typical case PID control chart

The PID controller is a linear controller, which according to the given value $r(t)$ and the actual value $c(t)$ to create of the control deviation [17],

$$e(t) = r(t) - c(t) \tag{3.19}$$

The amount of deviation constitute with the proportional (P), integral (I) and differential (D) linear combination, to control the control object, so it is called a PID controller. In continuous-time analog controller, proportion, integral and differential movement's characteristics can be express by the following formula:

$$u(t) = K\left[e(t) + \frac{1}{T_I}\int e(t)\,dt + T_D\frac{d}{dt}e(t)\right] \tag{3.20}$$

where $u(t)$ is the output of the controller, K is controller proportional gain; $e(t)$ is control error, which is the input of controller; T_I is controller integral time; T_D is controller differential time. In computer PID control, the computer can only handle discrete signal, hence we need to transfer PID control algorithm into a discrete form which the computer can be achieved, the discretization of the difference method form as follows:

$$u(k) = k\left[e(k) + \frac{T}{T_I}\sum_{j=0}^{K} e_j + \frac{T_D}{T}\left[e(k) - e(k-1)\right]\right] \tag{3.21}$$

$$= k_p e_{(n)} + k_i \sum_{j=0}^{K} e_j + k_D\left[e(k) - e(k-1)\right] \tag{3.22}$$

where $K_p = K$, which is proportional gain; $K_i = kT/T_i$, which is integral gain; $K_d = kT_d/T$, which is differential gain. T is sample time, $e(k)$ is the number

k sampling deviation value; k is natural number, which $k = 0, 1, 2, 3, \ldots$. We need to find K_p, K_i and K_d to let the system achieve its best working condition. The three parameters have some rules can be followed, which is shown in the following table. This table gives us a guide to find K_p, K_i and K_d.

Table 3.1: Characteristics of P,I and D Controllers

CL Response	Rise Time	Overshoot	Settling Time	S-S Error
K_p	Decrease	Increase	Small Change	Decrease
K_i	Decrease	Increase	Increase	Eliminate
K_d	Small Change	Decrease	Decrease	No change

3.3 Pavement Monitoring Method

Pavement monitoring method its main theoretical idea is to collect the road surface changing information before the wheel get through it, so the vehicle ECU has enough time to make right decisions to control the active actuator. It based on the road surface wave signal collection system. We assume that the vehicle installs this system and it continuously collects road surface changing information [18]. For unit step and sine wave signal, the active force can be expressed as the follow equation:

$$u = -l x_i(t) \tag{3.23}$$

From above equation we can see that the active force u is relative to the road input signal $x_i(t)$, l is a proportionality coefficient. For white noise signal, above equation can be transform to the following equation:

$$u = -l \int_0^{\Delta t} w(t) d\Delta t \tag{3.24}$$

the Delta t is assigned to 0.01s, which is as same as the system sample time. $w(t)$ is the white noise generator.

3.4 Proposed Control System Design

The modification of the suspension system is designed for determining whether the ride height is at the proper level, and when the circuit detects that the

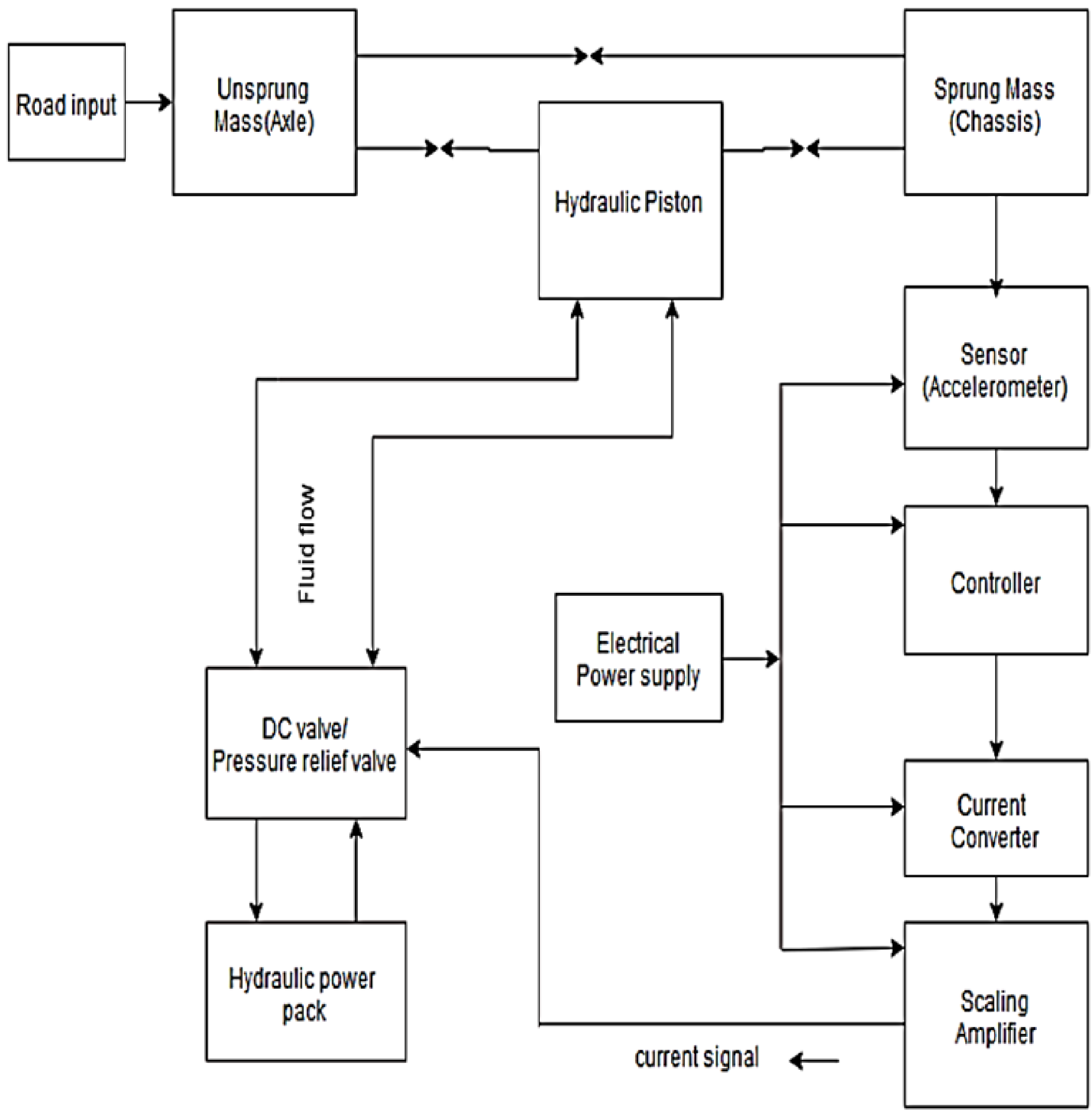

Figure 3.8: Block diagram of control system

vehicle height is not within the allowable range, it varies the hydraulic actuator level gradually using by sending a signal to the solenoid Directional control valve to correct the vehicle height to the proper level, the figure below is describing the control system

3.5 Hydraulic System

A hydraulic power unit is used to drive the active suspension system. It contains five main components namely hydraulic pump, hydraulic accumulator, electro hydraulic directional control valve, piston-cylinder, reservoir and pip-

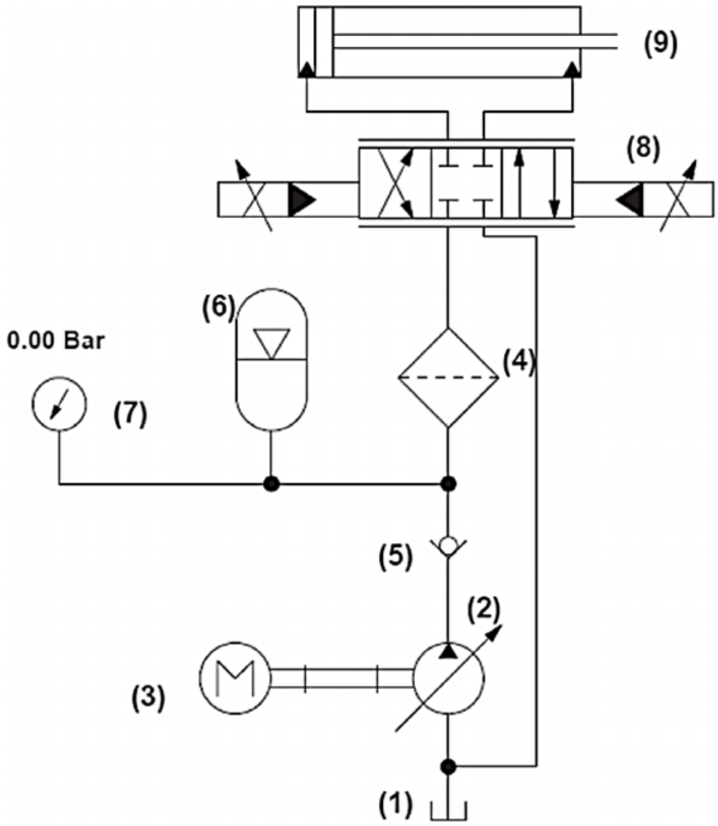

Figure 3.9: Active Suspension System

ing system. A schematic for it with the active suspension system is designed by Automation Studio Program as shown in figure below:

- Hydraulic oil Reservoir (40 l)
- Variable Displacement Pump (21.17 LPM)
- Electrical Motor (2.27 kW)
- Medium Pressure Filter
- Check Valve (6) Accumulator
- Pressure Gauge
- 4/3 Directional Control Valve DCV
- Double Acting Cylinder

The power supply of the hydraulic system is using the same hydraulic pump of the car steering which drove by the engine. The variable displacement hydraulic pump will keep the supply pressure to the both the car steering and hydraulic system of the suspensions. the hydraulic accumulator is parallel connected with the pump to enhance the pump feeding (flow rate) to the suspension system actuators. the simulation results show potentially low system response for the small road inputs and that occurs due. The spool valve position will control the fluid to come-in or come-out to the piston-cylinder.

The hydraulic actuators are governed by electro hydraulic Directional Control valve which control the fluid to come-in or come-out to the piston-cylinder then allowing for the generation of displacement between the sprung (Tire) and unsprung (vehicle) masses. the time required for the mechanical actuating by the valve spool 10 ms (valve response), for that reason the system had been designed to activate only at a high road inputs which happens through rough road profiles or unbalancing turning. The control program includes PID algorithm and the mathematical models had conceded, the control Program had implemented for Arduino microcontroller

3.5.1 Hydraulic System Parameters

The hydraulic pump discharge rate is determined by calculating the necessary flow rate on basis of changes in the vertical travel of the vehicle.

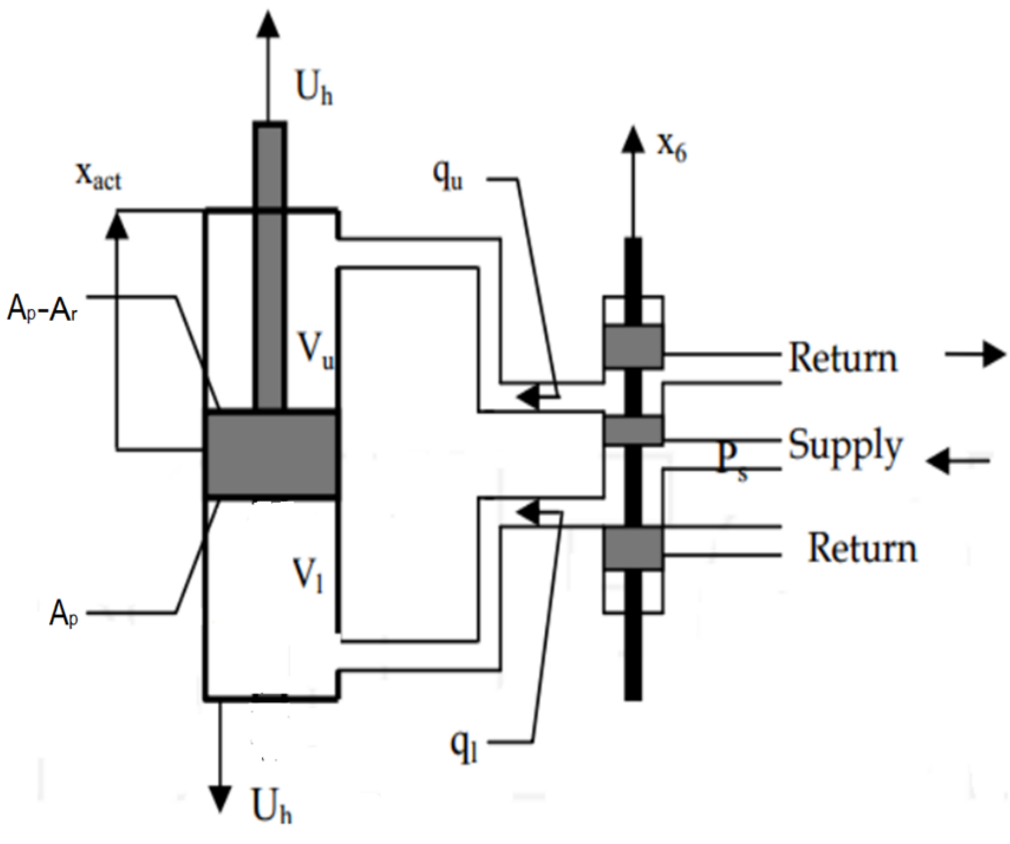

Figure 3.10: Hydraulic valve and cylinder

3.5.2 Hydraulic Valve and Cylinder

For there are four Hydraulic cylinders that means the unsprung mass for one cylinder is

$$\text{Unsprung mass}/4 = 2500/4 = 625 \quad kg \tag{3.25}$$

Other parameters are defined as follows.

$$D_p = 2.5cm \qquad \text{(Piston diameter)} \tag{3.26}$$

$$A_p = \frac{\pi}{4}D_p^2 = \frac{\pi}{4}\left(2.5 \times 10^{-2}\right)^2 = 4.9 \times 10^{-4} \quad m^2 \tag{3.27}$$

$$\text{Rod Diameter} = 1.5cm \tag{3.28}$$

$$A_r = \frac{\pi}{4}D_p^2 = \frac{\pi}{4}\left(1.5 \times 10^{-2}\right)^2 = 1.76 \times 10^{-4} \quad m^2 \tag{3.29}$$

The actuation stroke $X_{act} = 5$ cm. The value of F is given by

$$F = \text{mass} \times \text{Acceleration} = 650 \times 10 = 6500 \quad N \tag{3.30}$$

The value of P is given by

$$P = F/A \tag{3.31}$$

The flow rate need can be computed as follows

$$Q = \text{Cylinder Speed} \times A_p \tag{3.32}$$

The cylinder speed is 0.1 m/s, i.e., the level change will be 1 cm for every 10 m second. A_p is 4.9×10^{-4} m^2. The values for v and Q_{th} are given by

$$v = \frac{h}{1000t} = \frac{Q}{6A} \quad m/s \tag{3.33}$$

$$Q_{th} = 6Av = 60\frac{V}{t} \quad I/min \tag{3.34}$$

Hence, $v = 20km/60 \times 60$, therefore $v = 0.1/1500 = 6.66 \times 10^{-5}$ m/s and

$$Q = 6 \times 4.9 \times 10^{-4} \times 6.66 \times 10^{-5} \tag{3.35}$$

3.5.3 Calculations For Hydraulic Cylinder

Applied Pressure

$$P_o = F/A = 3000N/(\pi/4)d^2 \tag{3.36}$$

$$= 3000/3.8 \times 10^{-3} = 779534.6 \; N/m^2 \tag{3.37}$$

$$= 779.4 \; KPa \tag{3.38}$$

Volume

$$V = \pi r^2 h = \pi \times (0.035)2 \times 0.007 \tag{3.39}$$
$$= 2.6939 \times 10^{-5} \quad m^3 \tag{3.40}$$

Mass

$$m = d \times V = 8000 \times 2.6939 \times 10^{-5} = 0.2155\ Kg \tag{3.41}$$

Now Area of Piston

$$A_p = 2 \times rh = 2 \times \pi \times 0.035 \times 0.138 = 0.0303\ m^2 \tag{3.42}$$

Now Pressure of Oil inside the cylinder to sustain the load is

$$P_o + \frac{\text{Mass of Piston} \times g}{textAreaofPiston} \tag{3.43}$$
$$= 779.4 + (0.2155 \times 9.81)/0.0303 \tag{3.44}$$
$$= 849.3\ KPa \tag{3.45}$$
$$= 8.493\ Bar \tag{3.46}$$
$$= 0.849\ N/mm^2 \tag{3.47}$$

3.5.4 Control System

The control system is responsible of take the corrections of the deflections of the system outputs from the desired values [19]. Figure 3.11 shows the control system illustration.

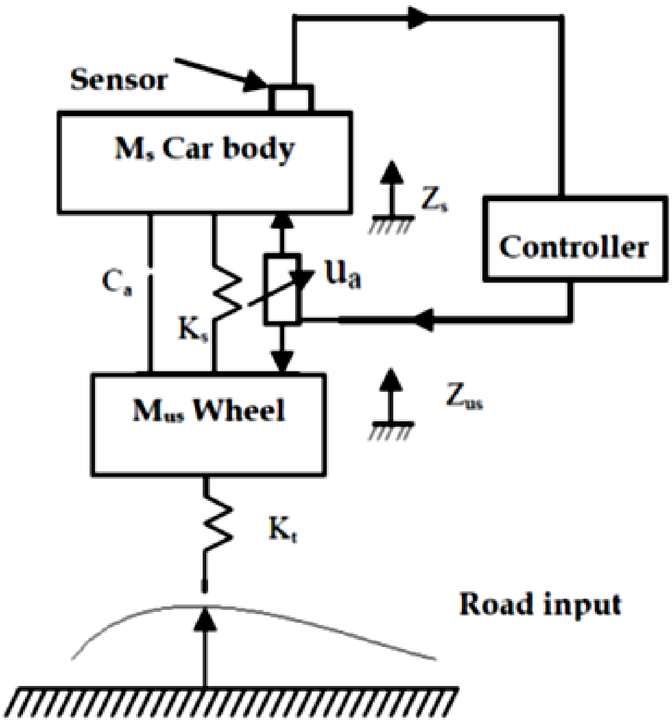

Figure 3.11: The control system

The hydraulic cylinder correcting the vehicle vertical position at the optimum level of the vehicle which controlled by the solenoid directional control valve, shown in figure 3.12. Also, the figure 3.13 showing the DCV parts. The solenoid DCV structure consists of the (1)valve body; (2)spool; (3)solenoid; (4)spring and (5)spool position sensor. The directional control valve DCV input signal generated by the PWM output pin from the Arduino microcontroller the program had used a PID controlling algorithm.

Arduino Mega 2560 used for processing and control of the digital and analogue I/O signals via serial communications, a universal multioutput power supply with input 220 V_{AC} and outputs +12, 12, +5, 5, +3.3 V_{DC}, a Rexroth and a MOOG NG6 Driver Card with integrated Signal Generator and On-Board Relay Module, circuit breakers used as protection for the main supply and the two induction motors.

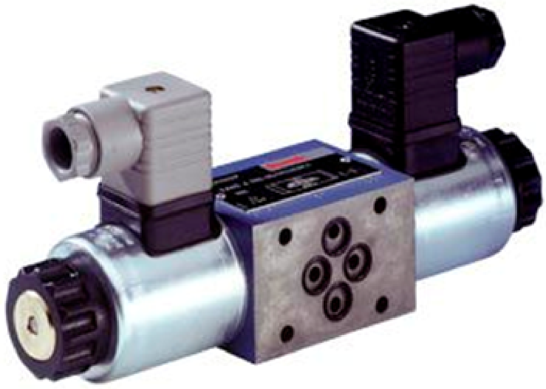

Figure 3.12: Solenoid Directional Control Valve

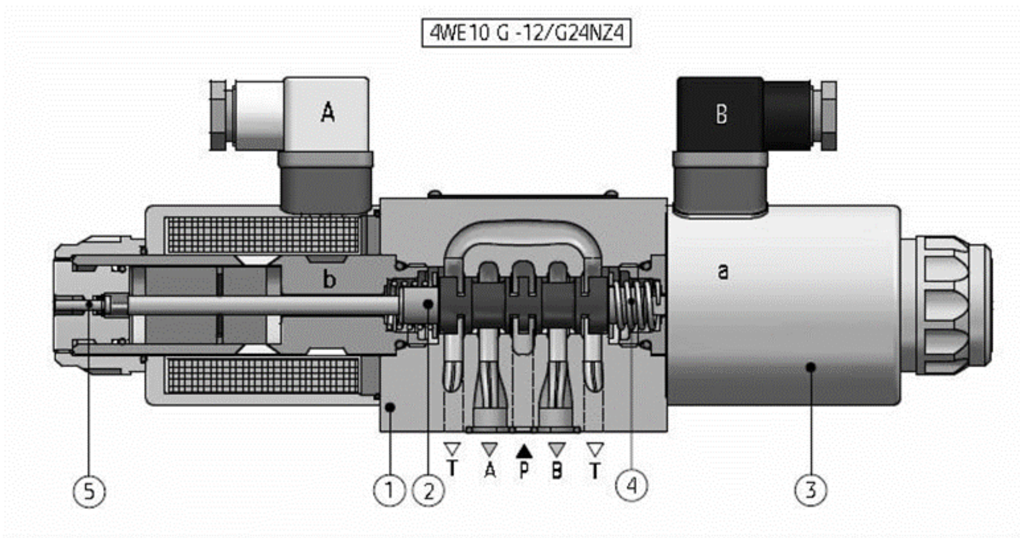

Figure 3.13: Solenoid DCV structure: (1)valve body; (2)spool; (3)solenoid; (4)spring and (5)spool position sensor

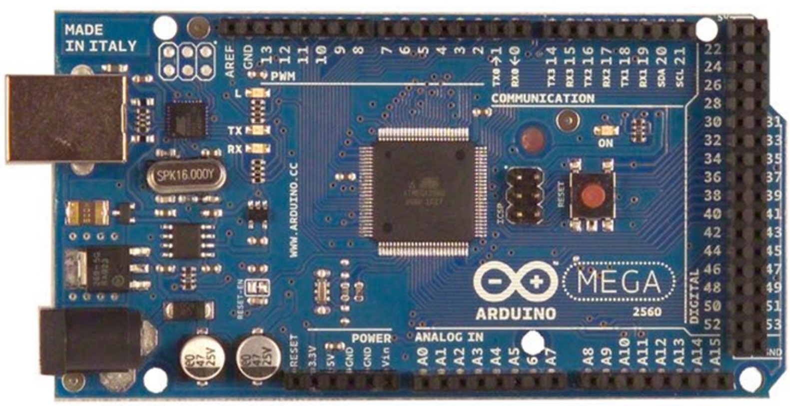

Figure 3.14: Arduino Mega 2560 Microcontroller

The human interface is through start and stop push buttons on the front panel, two potentiometers for manual control of the two servo valves and a main switch. Proximity sensors are used to measure the displacement between the vehicle body mass and tires and the analog signals from the sensors are sent to a controller. The controller is designed to take necessary actions to improve the performance abilities already set. The controller amplifies the signals and the amplified signals are fed to the solenoids of the Directional Control Valves to form closed loop system.

Figure 3.15: Linear Position Sensor -10cm From Formula SAE® (FSAE) Organization

LCD display which is specifically manufactured to be used with microcontrollers which means that it cannot be activated by standard IC circuit .it is used for displaying different messages on a miniature liquid crystal display. The model described here is for its low price and great capabilities most frequently used in practice. it can display messages in two lines with 16 characters each. it can display all the letters of alphabet , Greek letters, punctuation marks, mathematical symbols etc. It is also possible to display symbols made up by the user and it include automatic message shift (left and right) Cursor appearance [14].

This thesis considered the use of a microcontroller for two reasons bellow:

1. At the mechanical control systems There is a slight difference between

the experimental and simulation results due some facts the friction between the guide bars and both sprung and unsprung masses, The response of the directional control valve DCV spool 50 mill second ms. The little amount of time required for the hydraulic cylinder to arrive the energized pressure all those deflections wasn't taken into consideration during the mathematical modeling phase, and the simulation theoretical setups' performance had enhanced by adding the response specifications of the mechanical components at the microcontroller simulator PROUTOS.

2. using a microcontroller for the control system is preparing the invention to be implemented in practical stage. the changes on vehicle vertical position is measured by approximaty sensor. The hydraulic actuator cylinder getting the action and change the vehicle vertical displacement for compensating the variation of the level of vehicle due the road profile

The hydraulic pump connection with the that means at the minimum revolution speed of the engine so that the working fluid pressure to be supplied to the working chamber of the hydraulic cylinder can be satisfactorily high at any engine driving range. As will be appreciated, the output pressure of the fluid pump increases according increasing of the engine revolution speed. Therefore, at high engine revolution speed range.

The system assures levels of energy consumption and control valve response. The task of suppressing vehicle vibrations is shared among the accumulators and level control valves. This works to reduce the amount of oil required and to lessen the load on the control valve response. The use of coil springs in parallel with the system also serves to reduce the required hydraulic pressure. Another element contributing to lower energy consumption is the use of a variable capacity oil pump Mechanical structure The kit is from Superior which is develop, manufacture and retail suspension systems located in Australia We have developed this kit to be easy to install with full instructions and have done all the hard work and testing as well as performing accredited lane change tests and the Design pre-approval under the VSB LS Codes. This kit needs to be welded by a qualified boilermaker or welder (AS1554.1 certified) and fitted by a suitably qualified suspension fitter. Once this coil conversion is fitted the vehicle must be produced at your local transport approved modification plate officer, they will inspect that the vehicle has the suspension fitted in accordance with the supplied instructions and is welded correctly and to an acceptable standard, original invoice will need to be provided for the coil kit at time of inspection to prove it is a Superior Engineering kit . The inspector/ officer will then plate the vehicle with an LS6 approval code. As with all modification and aftermarket accessories, it is recommended to confirm all modifications in writing with your insurance company as some policies or providers may not accept such modifications although street legal.

The features and highlights are as follows.

- Street Legal and Compliant, Approval Under VSB LS Codes
- Weld-In Kit Featuring 4 x Chassis Side Strengthening Braces
- Heavy Duty Sway bar and Coil Tower Brace
- Uses Factory Mounting Locations For A Precise Fit
- Supplied Pre-Fabricated For An Easy Installation
- Improved Roll Axis For Tyre Clearance
- Increased Wheel Travel and Articulation
- Adjustable Upper Control Arms
- 40mm OD Fixed Lower Control Arms
- Reduced Bush Bind Over A 5-Link
- Compatible with Coil Air Bags
- 4-Link Rear Suspension Setup
- Fully Adjustable Pinion Angle
- Provides More Precise Handling
- Improved Comfort and Ride
- 350 Grade Steel Mounts
- Supplied with Shocks, Coils and Sway Bar
- Will work on up to 5” lift

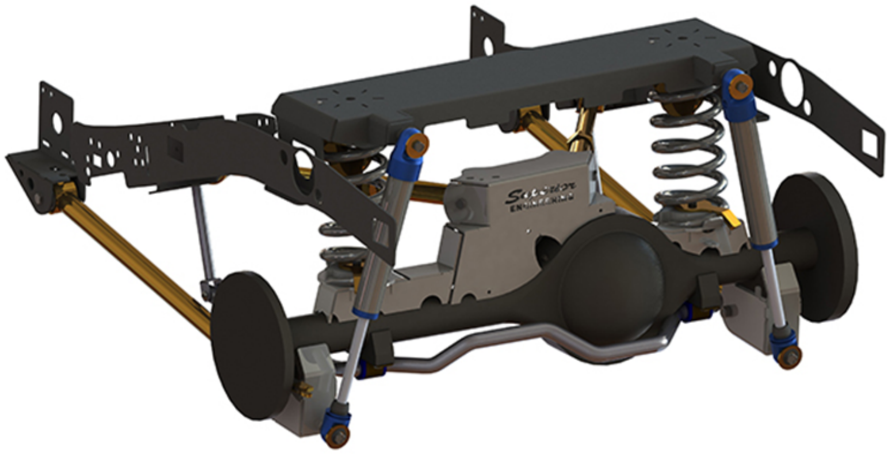

Figure 3.16:

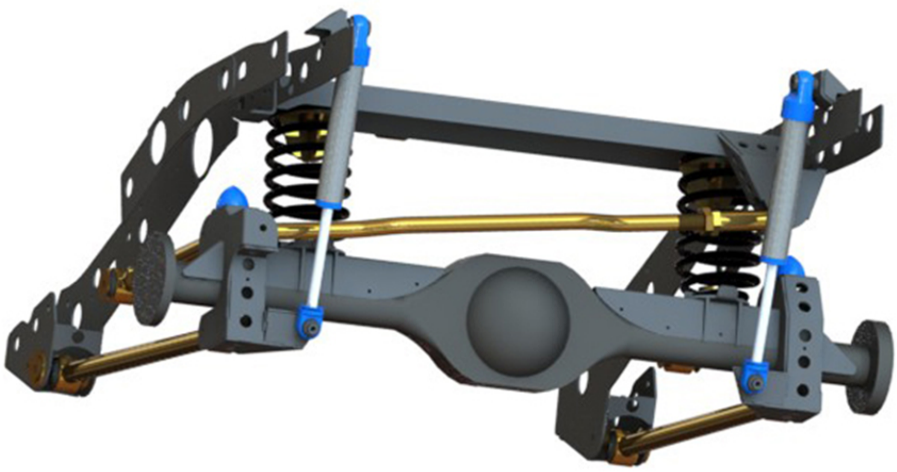

Figure 3.17:

Figure 3.18:

Chapter Four
Simulation Results and Discussion

4.1 Vehicle Simulation Parameters

Here, Matlab/Simulink is used as a computer aided-control system tool for modeling the non-physical quarter car with its modeling as, all included in one analysis loop passive system, and active system. The vehicle parameters considered for the analysis are given in Table 4.1.

4.2 Road Input

Signals In this thesis, three types of road input signal will be used to simulate different kinds of road condition. They are step input signal, sine input signal and white noise road input signal. These inputs are the prerequisite to simulate the vehicle suspension system, and they should be accurately reflecting the real road condition when a vehicle drives on the road. Precise signal is crucial to the result of the simulation. We assume the vehicle is a linear system.

Table 4.1: Vehicle Parameters Considered for the Analysis

Vehicle Model Parameters Suspension	Symbol	Numerical Value	Unit
Sprung Mass	M_1	300	kg
Unsprung Mass	M_2	40	kg
Suspension Stiffness	K_1	15000	N/m
Tire Stiffness	K_2	150000	N/m
Damping Coefficient	D	1000	Ns/m

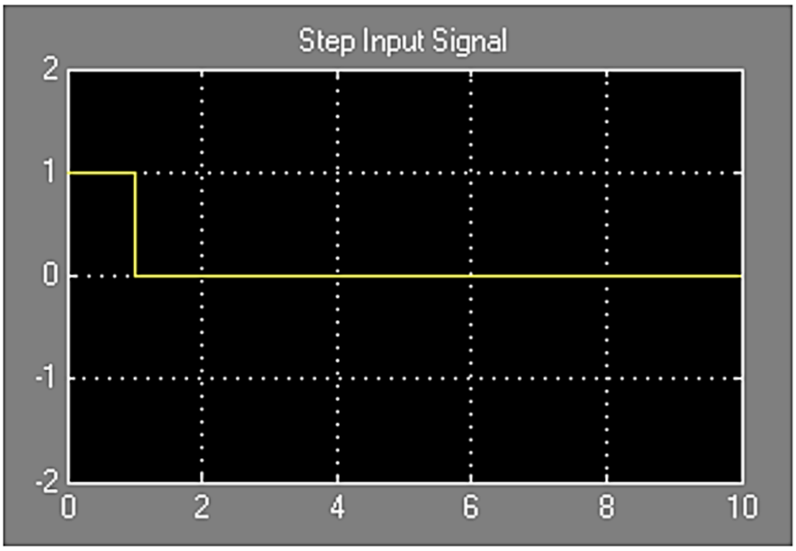

Figure 4.1: Unit step input signal

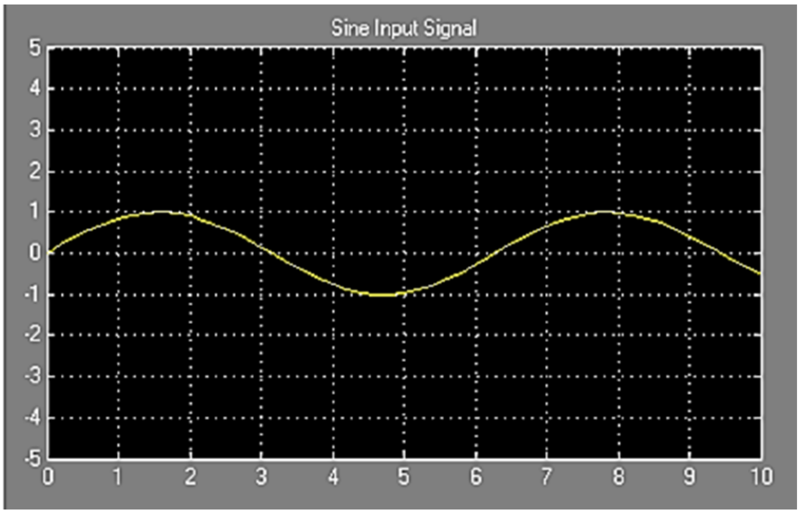

Figure 4.2: Sine wave input signal

4.2.1 Step and Sine Inputs

Step input signal is a basic input to research the suspension system. It simulated a very intense force for a very short time, such as a vehicle drive through a speed hump.

Sine wave input signal can be used to simulate periodic pavement fluctuations. It can test the vehicle suspension system elastic resilience ability while the car experiences a periodic wave pavement. Sine input pavement test is made by every automotive industries before a new vehicle drives on road [16].

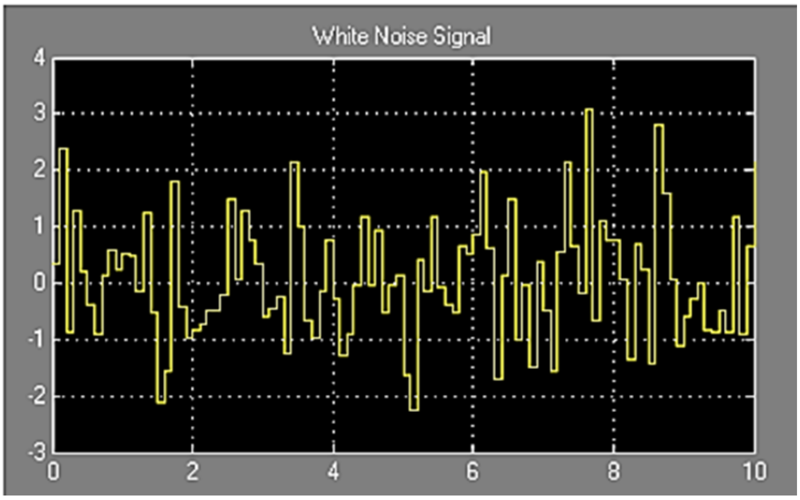

Figure 4.3: White noise input signal

4.2.2 White Noise Road Input Signal

Numerous researches show that when the vehicle speed is constant, the road roughness is a stochastic process which is subjected to Gauss distribution, and it cannot be described accurately by mathematical relations. The vehicle speed power spectral density is a constant, which correspond with the definition and statistical characteristic of the white noise, so it can be simply transformed to the road roughness time domain model. An example for the creation of white noise by using Matlab/ Simulink is shown as below: (noise power is 0.1, sample time is 0.1)

The transformation of white noise road input signal can perfectly simulate the actual pavement condition. It has random character when it is used for the vehicle vibration input of road roughness, which is always use the road power spectral density to describe its statistical properties.

4.3 Passive Suspension System Simulation

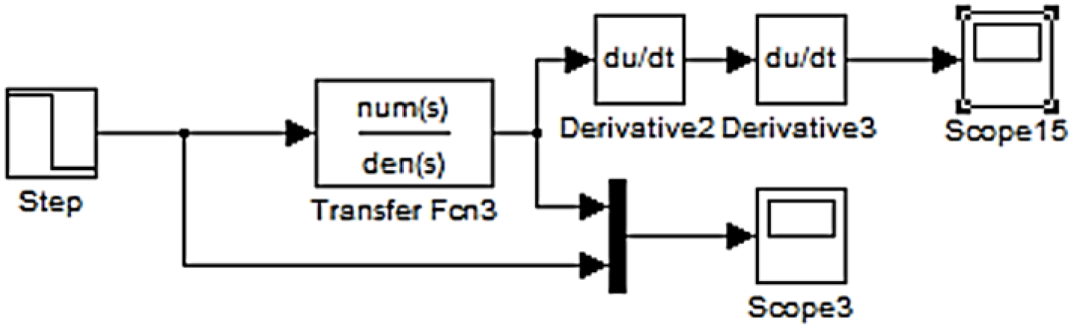

Figure 4.4: Passive suspension system Simulink structure with the unit step input

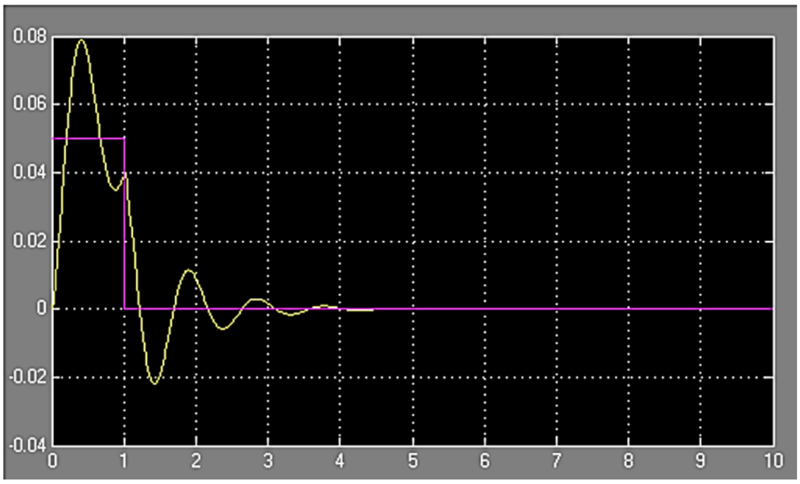

Figure 4.5: The Passive suspension system output displacement time response with the input of unit step signal

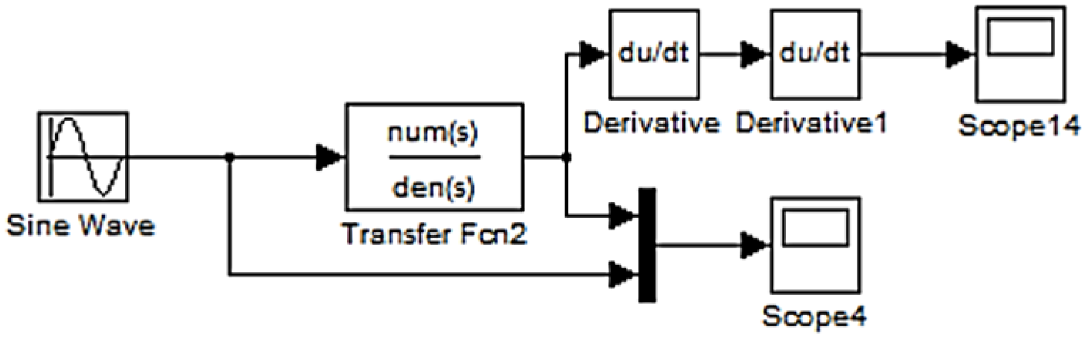

Figure 4.6: Passive suspension system Simulink structure with the sine wave input

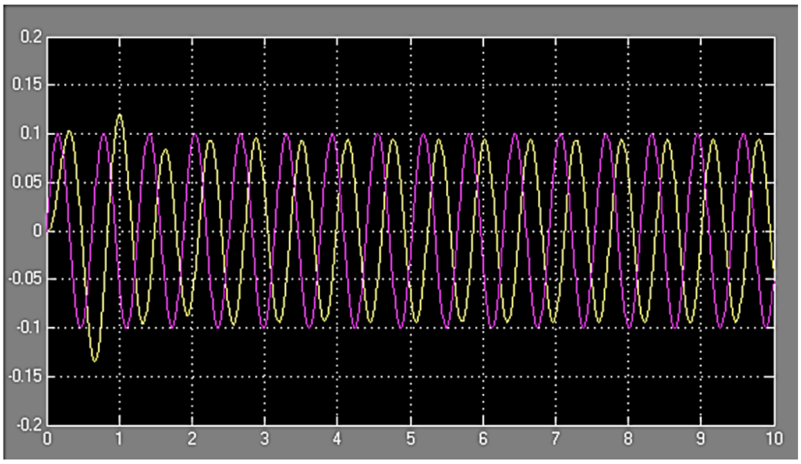

Figure 4.7: The passive suspension system output displacement time response with the input of sine wave signal (purple line is sine wave signal)

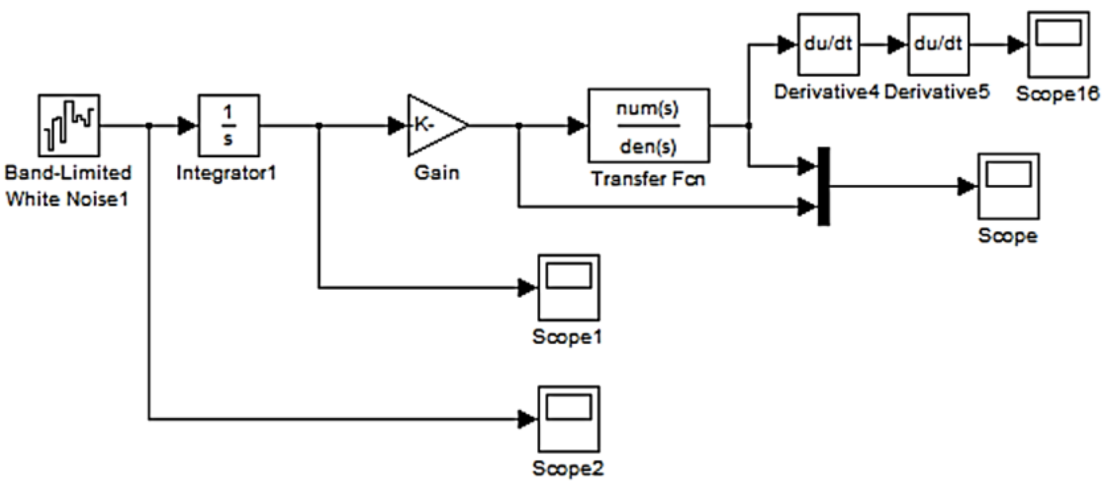

Figure 4.8: Passive suspension system with white noise input

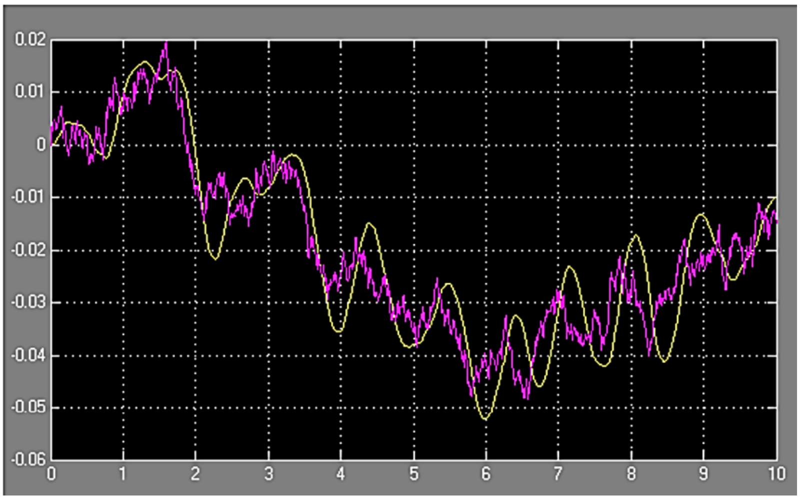

Figure 4.9: The passive suspension system output displacement time response with the input of white noise signal (purple line is white noise signal)

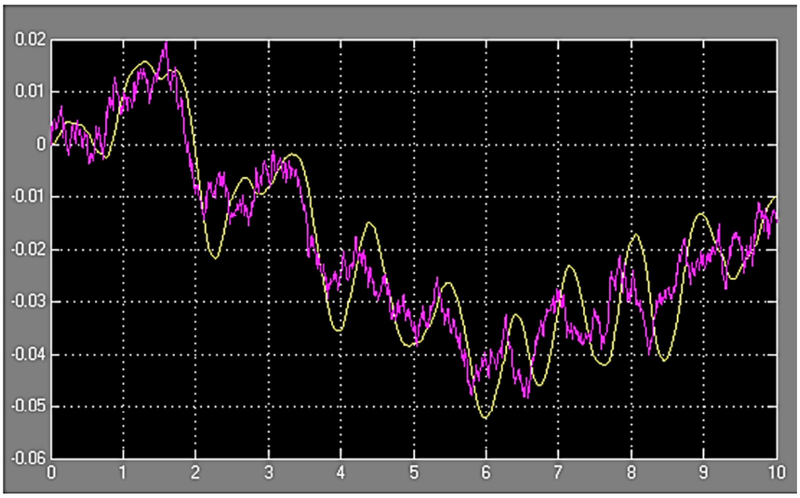

Figure 4.10: The passive suspension system output displacement time response with the input of white noise signal (purple line is white noise signal)

From above pictures, we can see that passive active suspension system cannot reduce vibration. It can only control the vibration to a limit extent. The suspension travel strokes are even larger than the road surface wave amplitude.

4.4 Active Suspension Simulation

Simulink Structure of the Active suspension system is shown in figure below:

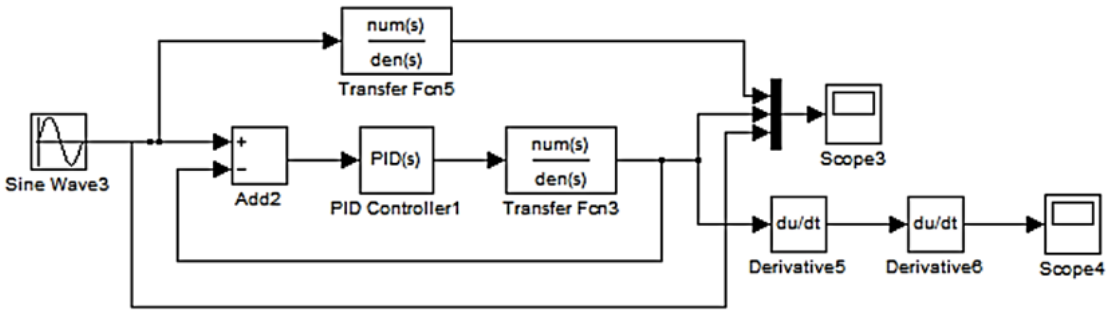

Figure 4.12: PID control active suspension system Simulink structure with the sine wave input

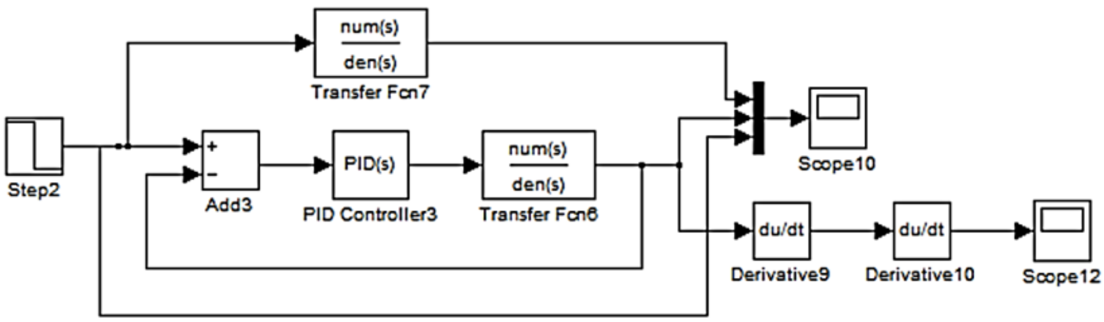

Figure 4.11: PID control active suspension system Simulink structure with the unit step

Figure above shows a set of PID control active suspension system Simulink structure with the unit step input. A PID controller is added into the system and the displacement feedback is added with the origin signal. The above transfer function module is passive suspension system, which is the experiment control group. Two signal output is displacement

4.5 Response Comparative Results

Figure 27 shows the passive suspension system and PID control active suspension system displacement change over time. The yellow line is passive system, and the purple line is PID control system. We can see that, compare to the passive suspension system, the PID control active suspension system successful reduce the amplitude of the vibration. The PID active suspension system's output displacement amplitude is lower than the unit step signal's amplitude.

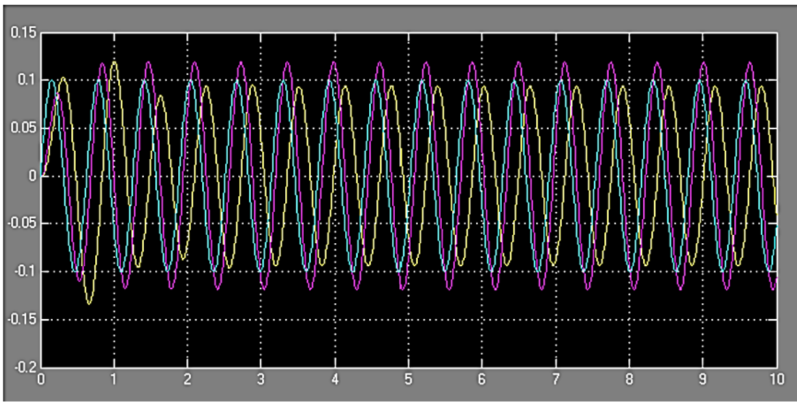

Figure 4.13: PID Control Active Suspension System compare with Passive Suspension system output displacement time response with the input of Sine Wave signal

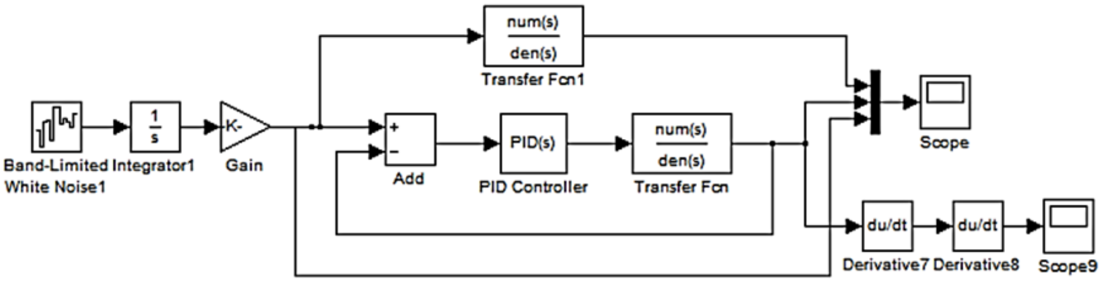

Figure 4.14: PID control active suspension system Simulink structure with the white noise input

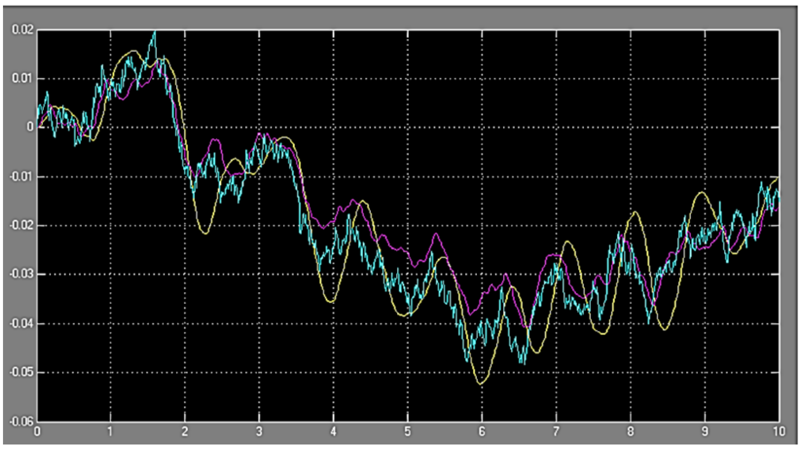

Figure 4.15: PID Control Active Suspension System compare with Passive Suspension system output displacement time response with the input of White Noise signal

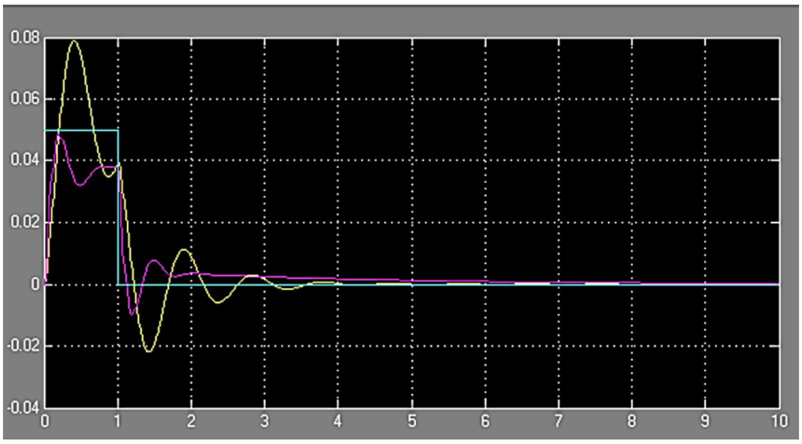

Figure 4.16: PID control active suspension system compare with passive suspension system output displacement time response with the input of unit step signal

The sprung mass settling time for passive is 51.13% more as compared to the active system. The unsprung mass settling time under the passive is 53.33% more as the semi active controllers.

The output displacement and acceleration is further reduced. The reliability and stability is further increased. The Curves is more flat than passive suspension control method curve.

Chapter Five
Conclusion and Recommendations

5.1 Conclusions

The passive suspension system is an open loop control system. It only designs to achieve certain condition only. The characteristic of passive suspension fix and cannot be adjusted by any mechanical part. The problem of passive suspension is if it designs heavily damped or too hard suspension it will transfer a lot of road input or throwing the car on unevenness of the road. Then, if it lightly damped or soft suspension it will give reduce the stability of vehicle in turns or change lane or it will swing the car. Therefore, the performance of the passive suspension depends on the road profile. In other way, active suspension can gave better performance of suspension by having force actuator, which is a close loop control system. The hydraulic actuator is a mechanical part that added inside the system that control by PID controller in the Simulink model. Controller will calculate either add or dissipate energy from the system, and send the out puts to the microcontroller pins to perform the actuation. Also Sensors will give the data of road profile to the controller.

Compared with passive control, active control can improve the performance over a wide range of frequencies controlled active suspension systems can potentially improve the ride comfort as well as the road handling of the vehicle simultaneouslythe designed system is using a hydraulic components which take more time to get action (low response) when compared with the electromagnetic actuating damper , although the system in this thesis didn't use electromagnetic damper and that due its energy extremely consumption, therefore the design However, active vibration control has the disadvantages of complexity and high-energy consumption. To solve the problems low response and high energy consumption the system had designed to activate at high rates of road profiles or on unbalancing states.

5.2 Recommendations

The technology of hydraulic systems is growing rapidly forward increasing its rabidity factors due ongoing metal production revaluation, though the further studies could implement the system and record experimental results. The thesis had used a quarter body mathematical models; therefore further works could work on full body mathematical models for further accuracy and complexity.

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