Sudan University of Science and Technology

College of Graduate Studies

School of Mechanical Engineering

Developing and Analyzing an Active Suspension System of an Automobile Using (PID) Controller

تطوير وتحليل نظام التعليق الفعّال لسيارة بإستخدام متحكم (**PID**)

A thesis submitted in partial fulfillment of the requirements for the Degree of M.Sc. in mechanical engineering (power)

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DEDICATION

The price of success is passion, hard work, self-discipline, dedication to the job at hand and the determination that whether we win or lose, we have applied the best of ourselves to the task at hand…

We should have no borders; believe in our competencies by those whom we love is all what needed to achieve the incredible end of the journey …

I dedicate this work My family who triggered me up when I lost my confident; then, I must express my very profound gratitude to them for providing me with unfailing support and continuous encouragement throughout my years of study and through the process of researching and writing this thesis. This accomplishment would not have been possible without them.

My supportive work partners who were always there when I need…

My colleagues in MRC for the good days we have, and still will…

Thank you all…

Author

Mohammed Salah Mohammed Abd-alal

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I am taking the advantage of this opportunity to express my gratitude to everyone who supported me throughout my preparation of this research paper. I am thankful for the aspiring guidance, invaluably constructive criticism and friendly advice during the project work.

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Thank you all.

Author

Mohammed Salah Mohammed Abd-alal

ABSTRACT

Recent huge progression in science and technology upshots in an effective designs and newly advanced ways of manufacturing in order to fulfill the customers' needs and expectations and to provide them better life experiences. Alongside those massive developments the technological innovation in automobiles sector has improved the ride performance through many aspects that include suspension systems, which has the responsibility of safety of both the vehicle and its occupants by providing stability and comfort ride during driving maneuvers. Without the help of suspension system, it would have been extremely hard for the driver to control a vehicle since all the shocks and vibrations would have been directly transmitted to steering chamber without any damping.

The main objective of this thesis is to build up a computerized mathematical and physical model of a PID controller by utilizing MATLAB Simulink / SimMechanics programs that will upgrade a quarter car passive suspension system of a passenger car into an active suspension system in order to improve the performance of the vehicle while facing different road profile. The Dynamic system used is imported from a Solidwork CAD model.

The simulation system can capture the basic performances of the vehicle suspension such as body acceleration, body displacement and settling time. Performance of suspension system is determined by the ride comfort and vehicle handling. Two types of road profiles are used as inputs for the system such as step and sinusoidal signal.

The results obtained show that the active suspension system oscillation time reduced compare to passive suspension system as well as both the body vertical displacement and acceleration.

المستخلص

أدت الطفرة التقنية والعلمية الكبيرة في السنوات الأخيرة إلى تطوير عمليات التصميم والتصنيع الهندسية بصورة أكثر كفاءة وذلك تلبيةً للاحتياجات والتوقعات المتزايدة من قبل العملاء والمستهلكين. ومن المعلوم أن من أكثر هذه المجالات تطوراً وتأثراً بهذه الطفرة مجال صناعة السيارات والذي شهد تقدماً كبيراً في أحد أهم أجزائه والمتمثلة بنظم التعليق والتي تؤثر بشكلٍ مباشرٍعلى تحسين جودة الرحلة وضمان سلامة الركاب من حيث إضفاء مزيد من الراحة والسلاسة على المناورات التي يقوم بها السائق والتي بغيابها ستتعرض المركبة لاهتزازات عنيفة كون جميع العوائق والمؤثرات على الطريق ستنتقل مباشرة إلى هيكل السيارة وبالتالي لركابها.

تهدف هذه الدراسة بصورة أساسية إلى تطوير نموذج محاكاة رياضية وفيزيائية لنظام تعليق سيارة ربعي باستخدام برنامج MATLAB Simulink / SimMechanics وبتطبيق متحكم (تناسبي تكاملي تفاضلي) على عربة ركاب اعتيادية لتحويلها من العمل بنظام تعليق محايد إلى آخر فعال وقياس أدائها عند أنماط مختلفة من المعوقات على الطرق تشمل منحنى الخطوة و منحنى الدالة الجيبية ، باستخدام نموذج مصمم بأداة (التصميم بمساعدة الحاسوب) باستخدام برنامج Solidwork، كما تم تحديد مستوى تسارع هيكل السيارة، الإزاحة الرأسية للهيكل و زمن استقرار المركبة على الطريق كمؤشرات لقياس أداء السيارة تحت نظام التعليق المطور، والمقارنة بين أداء النظامين.

بعد دراسة القيم والمنحنيات الناتجة من نظام المحاكاة، أظهرت هذه النتائج أن نظام التعليق الفعال كان أكثر سرعة في إخماد التذبذب من نظام التعليق المحايد كما قلل من مستوى الإزاحة والعجلة الرأسيتين لهيكل السيارة ...

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CHAPTER I

INTRODUTION

1.1 Preface

A system which support a load from above an isolate the occupants of a vehicle from the road disturbances is called a suspension system. Any suspension system comprises many elements, some are flexible elements which include spring and damper, others are fixed which are wheel assembly, bushes, locating links and anti-roll bars. Suspension system elements are shown in figure (1.1) below:

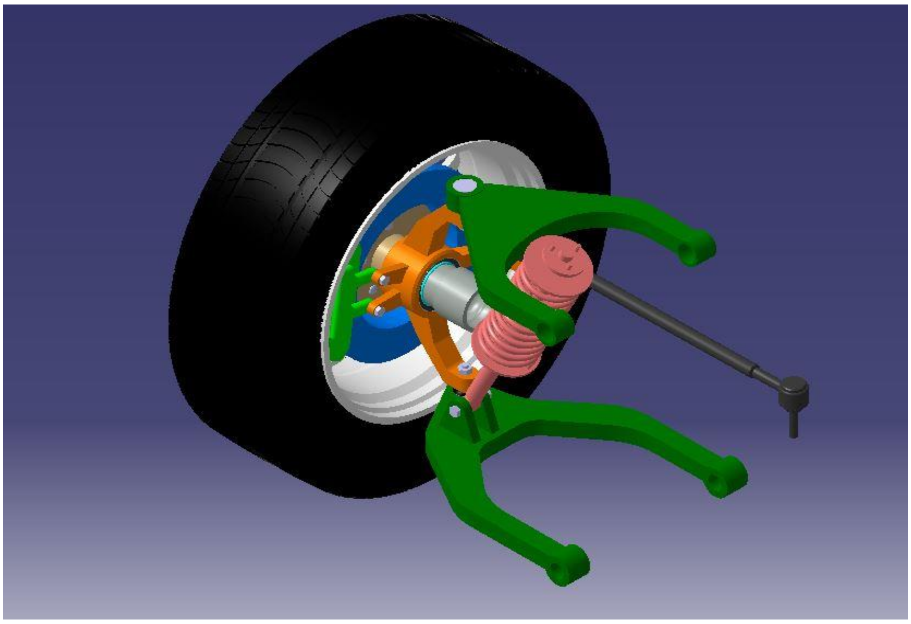


Figure 1.1 suspension system

The flexible elements in the suspension system are springs, these components have the ability of overstoring the energy applied in a form of loads and deflections. The spring is able to absorb energy and bend when it is compressed to a shorter length. When a tire meets an obstruction, it is forced upward and the spring absorbs energy of this upward motions, this energy is absorbed by the spring for a short time and it will be released by extending back to its original condition. When stored energy is released by the spring, it does so with such quickness and momentum that cause the end of the spring to extend too far. Until all of the energy in the spring is released it will go through a series of oscillations; contractions and extensions. The speed of the oscillations will be determined by the natural frequency of the spring and the load that the suspension system is subjected to.

The energy that the spring released, will be converted into heat then dissipated partially by friction from the system via damper. Generally, dampers are in the form of a piston working in a cylinder which filled with hydraulic fluid. The force which they applied is proportional to the square of the piston velocity. The function of damper is to restrain undesirable bounce characteristic of the sprung mass. Furthermore, it is used to ensure that the wheel assembly will be always in contact with the road.

Other mechanical elements in a suspension system are the wheel assemblies and control geometry of their movement. Some of these elements are simple links and multi-role members such as transverse torsion bars used to stabilize the vehicle in corners by restricting roll.

Vehicle suspensions are mainly classified into three major genus: passive, semi active and active suspensions, which depend on the operation mode to provide proper steering control, improve vehicle ride comfort by minimizing the vertical acceleration of the vehicle body that is conveyed to the passengers, increase vehicle safety, minimize road damage and promote overall performance.

Current automobile suspension systems typically use passive components which consist of springs and dampers that can only offer a conciliate between these two contradicting norms by providing damping and spring coefficients (stiffness) with fixed rates. A good suspension system should provide good vibration isolation via physical separation between the vehicle body and the wheels, i.e. slight acceleration of the body mass, and a slight “rattle space”, that can be defined as the maximum permissible relative displacement between the vehicle body and various suspension components.

A passive suspension has the ability to store energy via the spring and to disperse it via the damper. Its parameters are mostly fixed, being elected to achieve a certain level of concession between road handling, load carrying and ride comfort. Semi active and active suspension systems have the ability to store, disperse and to generate energy to the system. they may alter their parameters according to operating conditions due to involvement of external controlling element (Ahmed et al., 2015).

A PID controller is an instrument used in industrial control applications to regulate temperature, flow, pressure, speed and other process variables. PID (proportional integral derivative) controllers use a feedback control loop mechanism to control process variables and are the most accurate and stable controller.

The PID controller is a well-established way of driving a system towards a target position or level. It is a practically ubiquitous as a means of controlling temperature and finds application in myriad chemical and scientific processes as well as automation. PID control uses a feedback closed-loop control to keep the actual output from a process as close to the target or setpoint output as possible.

1.2 Problem statement

Develop and analyze a PID controller for a quarter car-model of a passenger car to promote its performance by improving safety, ride comfort and reduce the redundancy of fractures while encountering road disturbances.

1.3 Study objectives

The principle objective of this thesis is to build up a computerized program of a PID controller that will upgrade a passive suspension system of a passenger car into an active suspension system. while bumping into different road profiles and obstructions.

1.4 Study methodology

The methodology of this thesis will proceed with the following sequence, firstly to study the various types of cars suspension systems including the passive suspension system of a quarter car model, as it is the basis of the other revolutionary types of suspension systems, secondly is to study the active suspension system of a quarter car model, in accordance to its optimum suspension characteristics, then to establish a derivation of a mathematical model of a quarter car suspension system, after that a PID controller for a quarter suspension system will be developed using MATLAB/SIMULINK and MATLAB/SIMMECHANICS programs, finally a comparison between suspension system response in both passive and active systems will be made in terms of the results obtained from simulation programs.

1.5 Study restrictions

Here some obstacles that affected the studying effort will be appointed:

1. Limited monitory fund required to provide genuine copy of MATLAB program, the available cracked copy of the program results in difficulties in importing CAD model, simulating some parts of the system and automatic tuning in the PID controller which led to undesirable possessions on simulation process results.

2. Poor performance of the used computer device, in terms of processor properties that affect the simulation process adversely.

3. Limited availability of data and previous studies in SimMechanics simulation tool that results in the necessity of building up the model used for the study from the undergird.

1.6 Study layout

This research paper consists of the following five chapters that comprise the introduction which split into preface that includes the background of the research field, and the statement of problem discussed throughout the research, then the objectives or outcomes of carrying out this research were denoted, formerly the methodology or algorithm that followed in modeling and analysis processes, then the obstacles and restrictions that faced in the preparation of the research paper, and the layout or outline of the study.

The second chapter is the literature review that lights the major points in the developing and evolution of different types of suspension systems, and also their parts and subsystems and the PID controlling schemes. In addition, it represents the historical review of all study aspects.

Chapter three or the methodology and approach indicates the process followed in the simulation of both passive and active suspension systems; the software utilized, road profiles applied and the detailed procedure followed in simulation of the system.

The fourth chapter that is results and discussion or analysis which is a comprehensive graphical representation of the aspects of the simulation process results, combined with the analysis and explanation of those findings.

Finally, the fifth chapter is conclusion and recommendations, is about final recommendations about the utilized system characteristics and the software used in the process of simulation in order to provide a base for future studies in the same discipline.

CHAPTER II

LITERATURE REVIEW

2.1 Introduction

The suspension systems must supply proper ride quality to maintain passengers' satisfaction and reduce driver fatigue, as well as provide proper wheel and tire position to maintain directional stability while driving. Proper wheel position also ensures normal tire tread life.

Generally, the vehicle suspension models are divided into three types namely quarter car, half car and full car models (Agharkakli et al., 2012). In the quarter car model, the model is based on the interaction between the quarter car body and the single wheel. The motion in the quarter car model is only in the vertical direction. For the half car model; the interactions are between the car body and the wheel and also between both ends of the car body. The first interaction in the half car models caused the vertical motion and the other interaction produces an angular motion. In full car model, the interactions are between the car body and the four wheels which generates the vertical motion, and also between the car body and the left and right wheels which generated an angular motion called rolling and the last interaction is between the car body and the front and rear wheels which produced the pitch motion (Sam and Osman, 2006).

2.2 Historical background

2.2.1 History of suspension system

The first suspension system has been designed for the light chariots of Ramses II almost at the year of 1296 B.C. But this suspension system was deficient due to instability. It was still at that time, the only suspension system that has been found as comfortable as can be for light chariots. However, there were many problems in that design that include; the reduction of the speeds and the rapid wear of the component that results in the need of being changed frequently. The year 1886 is regarded as the birth of the modern automobile with the Benz Patent Motorwagen, the German inventor Karl Benz. In 1901 Mors of Germany first fitted an automobile with shock absorbers. With the advantage of having a damped suspension system in his 'Mors Machine'. In 1901, British inventor Frederick William Lanchester patented the disc brakes (M. Ravindran, 2018).

By that time, the history witnessed a rapid evolution of suspension system design with several problems have been discovered and identified. Regardless, of the positive development of suspension systems design, there were still problems pertaining to the system such as noise arises at an iron chain suspension.

A new suspension system design was found by William Brush at 1906, after his brother had a car accident at unpaved road with the speed of 30 mph. The impression is that the car's right wheel started shimmy violently and the entire car vibrated furiously. Brush has designed a suspension system for the Brush Two-Seat Run about car model. The feature of the model was different with front coil springs and devices at each wheel that dampened spring bounce (shock absorber) mounted on a flexible hickory axle.

In 1920, Leyland used torsion bars in a suspension system. In 1922, independent front suspension was pioneered on the Lancia Lambda and became more common in mass market cars from 1932. The Hotchkiss drive invented by Albert Hotchkiss was the most popular rear suspension system used in American cars from the 1930s to the 1970s (M. Ravindran, 2018).

It worth to notice that the passive suspension system itself has been passed through different phases until the year of 2002 and the introduction of passive interconnected suspension, that provide greater freedom to independently specify bounce, pitch, roll, and warp dynamics than conventional (passive) suspension arrangements (Smith and Walker, 2005).

Generally, there are two categories of vehicle's suspension i.e. conventional suspension and advanced suspension systems. Conventional suspension system refers to the passive suspension system whereas advanced suspension system indicates semi-active suspension or active suspension system (Amit, 2013). The latter mentioned active suspension system which is characterized by employing certain kinds of suspension force generation, such as pneumatic, magnetorheological, or hydraulic actuators has been used in practical applications facilitated by the development of microprocessors and electronics since the middle 1980s (Cao et al., 2008). It was proven by two or three decades of analytical developments, the vehicle models with increasing complexity had a useful evolutionary role and contributed to subsequent advancements. Quarter-car model established and active suspensions do have a potential to substantially improve ride and handling when compared with their conventional, passive counterparts. The resulting optimal structure gave birth to the concepts of skyhook damping and fast load leveling which are now being developed toward actual, large-scale production applications (Hrovat, 1997). those trends pass through many scientific research and contributions that eventually resulted in high performance, effective and efficient active suspension systems

2.2.2 History of PID Controller

The first evolution of the PID controller was developed in 1911 by Elmer Sperry. However, it was not until really industrially useful 1933 that the Taylor Instrumental Company (TIC) introduced the first pneumatic controller with a fully tunable proportional controller. A few years later, control engineers attempted to eliminate the steady state error which is calculated as the value of the difference between a measured process variable and a desired set point. The controller attempts to minimize the error by adjusting the process control inputs found in proportional controllers by resetting the point to some artificial value as long as the error was not zero. This resetting “integrated” the error and became known as the proportional-Integral controller. Then, in 1940, TIC developed the first PID pneumatic controller with a derivative action, which reduced overshooting issues. However, it was not operated optimally until 1942, when Ziegler and Nichols tuning rules were introduced that engineers were able to find and set the appropriate parameters of PID controllers. By the mid-1950’s, automatic PID controllers were widely adopted for industrial use (Omega, 2019).

2.3 Suspension system

Suspension system is one of the important components of a vehicle, which plays a crucial role in handling the performance and ride comfort characteristics of a vehicle. A suspension system acts as a bridge between the occupants of a vehicle and the road on which it rides. It has two main functionalities. One is to isolate the vehicles' body with its passengers from external disturbance inputs, which mainly come from irregular road surfaces. It always relates to riding quality. The other is to maintain a firm contact between the road and the tires to provide guidance along the track that is relevant to safety issues (Cao et al., 2008, Munaim Akhtar et al.).

2.3.1 Types of suspension systems

Generally, there are three types of suspension system. Those types are going to be discussed extensively in the following threads.

1. Passive suspension: passive suspension system is the conventional suspension system. However, it is still to be found on majority of cars production. It consists two elements namely dampers and springs. The function of the dampers in this passive suspension is to dissipate the energy and the springs are installed to store the energy. If a load exerted to the spring, it will compress until the force produced by the compression is equal to the load force. When the load is disturbed by an external force, it will oscillate around its original position for a period of time.

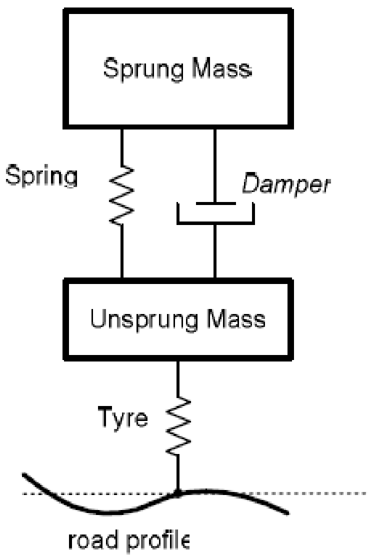


Figure 2.1 Passive suspension system

Dampers will absorb this oscillation so that it would only bounce for a short period of time. Damping coefficient and spring stiffness for this type of suspension systems are fixed so that this is the major weakness as parameters for the ride comfort and good handling vary with different road surfaces, vehicle speed and disturbances (Manap, 2007).

2. Semi-active suspension: semi-active suspension system employs the same elements that used in the passive suspension system and it uses the same application of the active suspension system where external energy is needed in the system. The difference is the damping coefficient can be controlled. The fully active suspension is modified so that the actuator is only capable of dissipating power rather than supplying it as well. The actuator then becomes a continuously variable damper which is theoretically capable of tracking force demand signal independently of instantaneous velocity across it. This suspension system exhibits high performance while having low system cost, light system weight and low energy consumption (Ahmad et al., 2005, Manap, 2007).

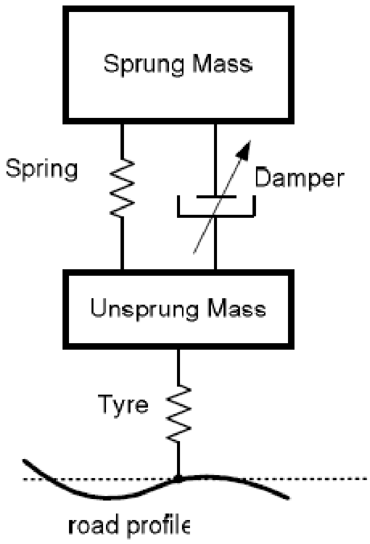


Figure 2.2 Semi-active suspension system

3. Active suspension: the concept of active suspension system was introduced as early as 1958. The major difference occurs while comparing it to conventional suspension systems is that active suspension system is able to inject energy into vehicle dynamic system via actuators rather than dissipates that energy. Active suspension can make use of more degrees of freedom in assigning transfer functions and thus improve performance. The active suspension system contains an extra element above the conventional suspension which is basically an actuator that is controlled by a high

frequency response servo valve and which involves a force feedback loop. The demand force signal, typically generated in a microprocessor, is governed by a control law which is normally obtained by application of various forms of optimal control theory. Theoretically, this suspension provides optimum ride and handling characteristics. It is done by maintaining an approximately constant tire contact force, maintaining vehicle geometry level and by minimizing vertical accelerations to the vehicle. However due to its complexity, cost and power requirements, but it has not yet put into mass production (Ahmad et al., 2005, Manap, 2007).

Most of the active suspensions, which have reached the stage of hardware development and production, have used some form of electrohydraulic actuator. Two forms of active suspension are commonly recognized. The first is the fast-active suspension or high-bandwidth system (HB), often referred to as fully active. The second is the slow active suspension or low-bandwidth system (LB). In an active suspension, the passive damper or both the passive damper and spring are replaced with a force actuator (Elattar et al., 2016).

a. Fully active suspension system (High bandwidth): Also known as high bandwidth has actuator placed between the sprung and unsprung masses. The main function of high bandwidth system is controlling the system over the full bandwidth of the whole system. Specifically, this implies that it is aimed to enhance the suspension response around both of the rattle-space frequencies (from10 to 12 Hz) and tire-hop frequency (from 3 to 4 Hz) (Elattar et al., 2016).

b. Slow active suspension system (Low bandwidth): It is applicable to operate in low bandwidth operations. In this system the actuator is placed in series with spring and/or a damper. Slow active suspension system (operating in low band width less than 3 Hz) is designed to fulfill a control policy of the suspension over the lower frequency range, and specifically around the rattle space frequency. At higher frequencies the actuator effectively locks-up and hence the wheel hop motion is controlled passively. Low bandwidth systems can achieve a worthy reduction in both body roll and pitch during maneuvers with lower energy consumption than a high bandwidth system (Elattar et al., 2016).

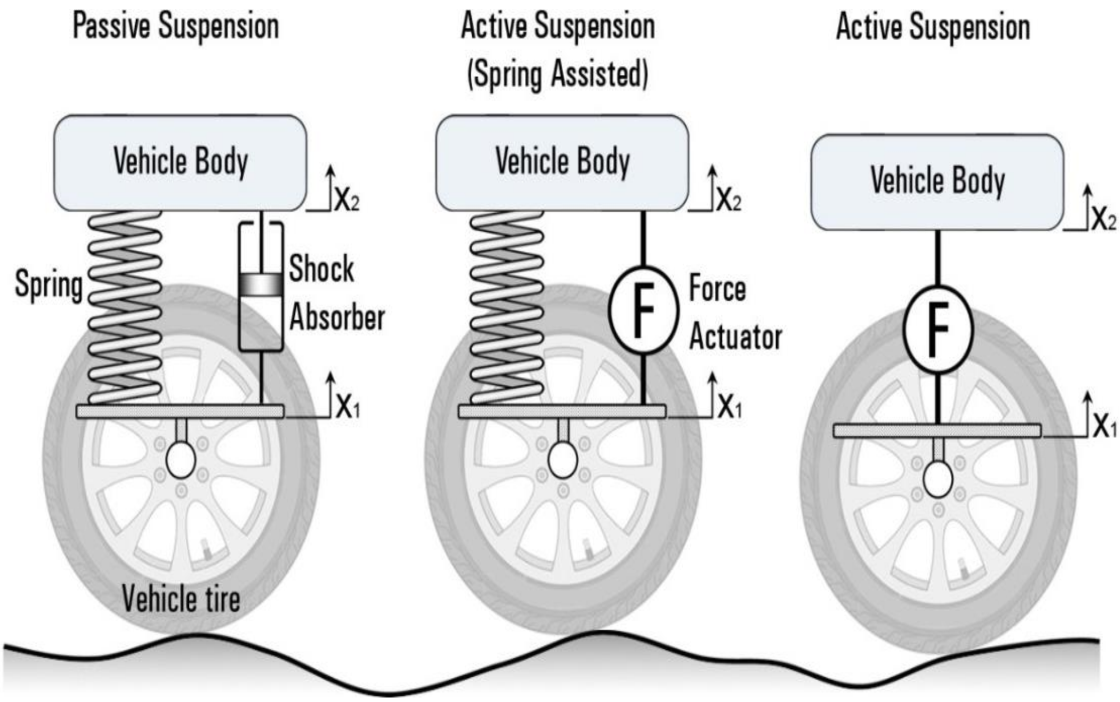


Figure 2.3 Active suspension system (a) Passive (b) Low bandwidth and (c) High bandwidth

The following table shows the differences between passive, semi active and active suspension systems.

Table 2.1 Comparison between different types of suspension systems

Comparisons Parameters	**Active**	**Semi-active**	**Passive**
Spring	Controlled	Uncontrolled	Uncontrolled
Shock absorber (Damper)	Controlled	Controlled	Uncontrolled
Spring coefficient (stiffness)	Varied	Fixed	Fixed
Damping coefficient	Varied	Varied	Fixed
Cost	High	Moderate	Low
Time of response	Slow	Fast	-
Performance	High	Moderate	Poor

2.3.2 Design Process of Suspension Systems

The design optimization process of vehicle suspension systems consists of pre-processing analysis and post-processing stages. In pre-processing stage, design equations for analysis are derived (analysis of displacement, velocity and acceleration).

In post-processing design stage, design sensitivity analysis and optimization of the static design factor and reactions with respect to the design variable are carried out (Lee et al., 2009).

The basic components in any suspension system are springs and dampers, those two elements will be discussed briefly in the subtitles below:

i. Springs: springs are resilient members that act as reservoirs of energy. They absorb road shocks or impacts due to bumps in roads via oscillation. They store the energy due to the sudden force which results when the vehicle encounters a bump or an obstruction; this energy is converted into heat and bounce is avoided. Tires also provide spring effect, but to a smaller extent.

   As mentioned above the main function of springs is to absorb road shocks quickly and to return to the original position as slow as possible in order to provide comfort riding experience, then, commonly there are five types of springs that can be stated as follows:

   - Leaf spring: referred to as laminated springs since they use steel strips or lamination one over the other with reducing length (SAE, 2019).
   - Coil spring: A section of spring steel rod wound in spiral pattern or shape. It is widely used in both front and rear suspension systems (SAE, 2019), most of the cars were introduced with the coil spring front suspension sprung independently and started using hydraulic shock absorbers. However, after several years, manufacturers have switched back and forth from model to model between leaf and coil spring. Eventually most of them equipped the leaf spring for a heavy car while the coil spring for a light car.
   - Air and gas spring: In these springs compressed air or gas is filled in the cylinder or bellows against which the wheel movement is transmitted through diaphragm. As soon as the wheel passes over a road irregularity the compressed air returns the system to its original position (Moon, Sep 27, 2014).
   - Torsion bars: A torsion bar is a solid bar of steel which is connected to the car chassis at one end, and free to move at the other end. They can be

mounted across the car or along the car. The springing motion is provided the metal bars resistance to twisting (SAE, 2019).

- Rubber spring: As rubber can store more energy per unit mass than any other type of spring material, considerable weight can be saved with rubber suspension. Rubber springs, if works on compression or shear, can be used as the main suspension spring, otherwise can be fitted along with metal springs to improve the suspension characteristics. Large rubber 'bump' stops used in many suspension layouts stiffens the suspension spring against maximum deflection (singh, 2019).

Figure 2.4 Leaf spring

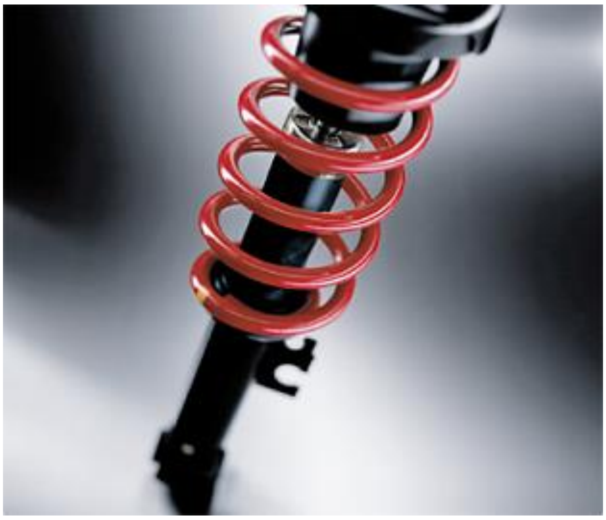


Figure 2.5 Coil spring

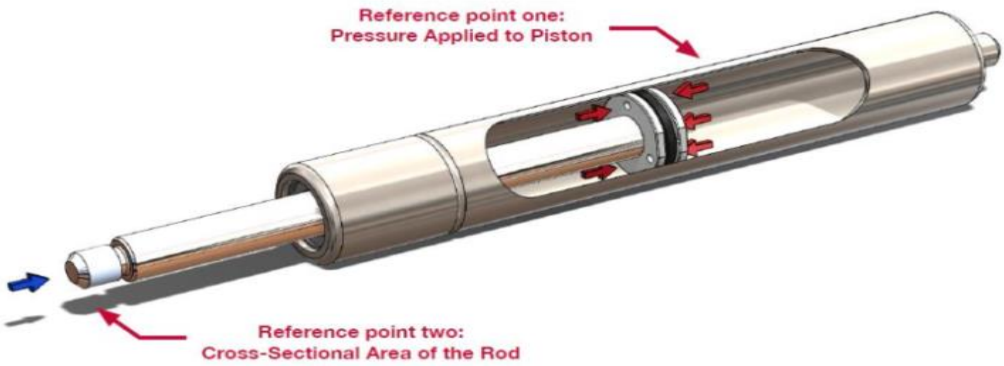


Figure 2.6 Air and gas spring

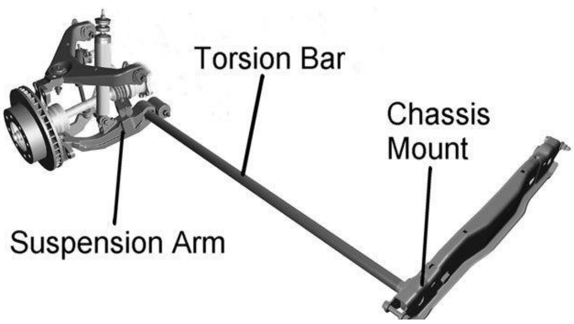


Figure 2.7 Torsion bar

Figure 2.8 Rubber spring

ii. Dampers: dampers used to reduce the tendency of the carriage unit to continue to "bounce" up and down on its springs. Oscillations due to road shocks are restricted to a reasonable level by damper engagement.

iii. Shock absorbers and struts: each corner of the vehicle has a shock absorber or strut connected from the suspension system to the chassis. Shock absorbers control spring action and wheel oscillations also improve vehicle safety because the struts help to keep each tire tread in contact with the road surface. If the struts are worn out, excessive wheel oscillations when driving on irregular road surfaces can cause the driver to lose control on the vehicle. Struts also reduce body sway and lean while turning a corner. Struts reduce the tendency of the tire tread to lift off the road surface. This action improves tire tread life, traction, steering control, and directional stability. Struts are also containing a sealed lower chamber filled with special oil.

Many shock absorbers have a nitrogen gas charge on top of the oil. This gas charge helps to prevent the shock absorber oil from foaming. A circular steel mount containing a rubber bushing is attached to the bottom end of the lower chamber and this lower mounting is bolted to the suspension system. The upper strut housing is connected to a piston rod that extends into the lower chamber. A piston valve assembly is attached to the lower end of the piston rod. The upper mount is bolted to the chassis. When a wheel strikes a road irregularity, the wheel and suspension move upward, and the spring in the suspension system is compressed. This action forces the lower shock absorber to move upward and the oil must flow from below the shock absorber piston and valve to the area above the valve. Upward wheel movement is called jounce travel (SAE, 2019).

Figure 2.9 Shock absorbers

2.4 PID controller

PID control is a particular control structure that has become almost universally used in industrial control. The letters 'PID' stand for Proportional, Integral and Derivative. They have proven to be quite robust in the control of many important applications for specific operating conditions. Its structure is simple but very effective feedback control method applied to dynamical systems. PID also most conveniently integrated with other more advanced control techniques which more often than not results in better overall performance. Pure PID control is excellent for slow speed operation and with very small or no disturbances, the performance severely degrades in the adverse conditions (Omega, 2019).

However, the simplicity of these controllers is also their weakness where it limits the range of plants that they can control satisfactorily. Indeed, there exists a set of unstable plants that they cannot be stabilized with any member of the PID family. Nevertheless, the versatility of PID control ensures continued relevance and popularity for this controller.

The PID method is error driven and largely relies on the proper tuning of the controller gains and accurate information from the feedback element (sensor). The basic algorithm of the PID is expressed as follows:

$$m(t) = K_p + K_i \int_0^t e(t)dt + K_d \frac{de(t)}{dt} \qquad (2.1)$$

Where:

m (t) ≡ control signal

K_p ≡ proportional controller gain

K_i ≡ integral controller gain

K_d ≡ derivative controller gain

e (t) ≡ error (output – input)

$\frac{de(t)}{dt}$ ≡ derivative error

A simple PID controller applied to a vehicle suspension system can be illustrated as shown in Figure (2.10).

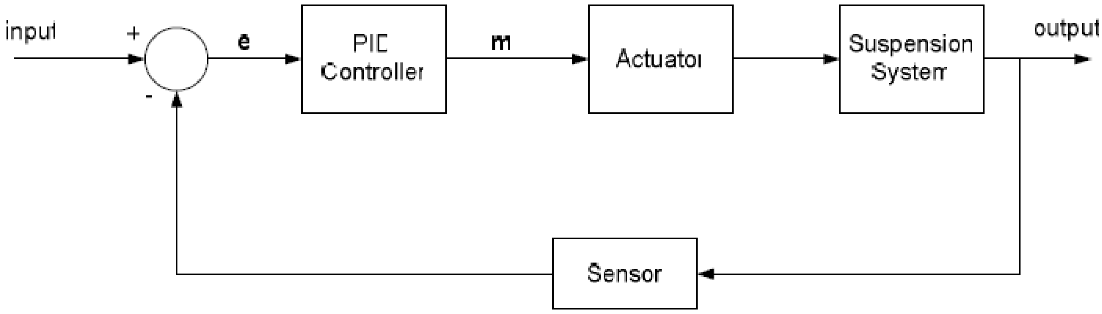


Figure 2.10 A block diagram of suspension system using PID controller

The effects of the P, I and D parameters to the system are as follows:

i. Proportional (P) action: this parameter provides a contribution which depends on the instantaneous value of the control error. A proportional controller can control unstable plant but it provides limited performance and non-zero steady-state errors. This later limitation is due to the fact that its frequency response is bounded for all frequencies.

ii. Integral (I) action: integral parameter gives a controller output that is proportional to the accumulated error, which implies that it is a slow reaction mode. This characteristic is also evident in its low-pass frequency response. The integral mode plays a fundamental role in achieving perfect plant inversion at zero frequency. This forces the steady-state error to zero in the presence of a step reference and disturbance.

iii. Derivative (D) action: derivative action acts on the rate of change of the control error. Consequently, it is a fast mode which ultimately disappears in the presence of constant errors. It sometimes referred to as a predictive mode because of its dependence on the error trend. The main limitation of the derivative mode is its tendency to yield large control signals in response to high-frequency control errors, such as errors induced by set-point changes or measurement noise (Goodwin et al., 2001, Manap, 2007).

2.5 Modeling suspension systems using Simulink tool

Simulink is a block diagram environment for multi-domain simulation and model-based design available in MATLAB software. It supports system-level design, simulation, automatic code generation, and continuous test and verification of embedded systems. Simulink provides a graphical editor, customizable block libraries, and solvers for modeling and simulating dynamic systems. And therefore, due to its useful characteristics mentioned it has been widely used in modeling different dynamic systems which include suspension systems (Matlab, 2019a).

According to A. Tandel, A. Deshpande, S. Deshmukh, K. Jagtap in their piece of article named Modeling, analysis and PID controller implementation on double wishbone suspension using SimMechanics and Simulink (2014) (Tandel et al., 2014). Simulink was used to module both passive and active suspension systems, the wishbone suspension includes stiffness of spring and damping coefficient of damper. These parameters affect the ride comfort of passengers and road handling of vehicle. The model of wishbone suspension is analyzed for different combinations of spring stiffness and damping coefficient to study the behavior of suspension during the simulation. The vertical acceleration of car body for each of the combinations will be obtained by simulation. Table (2.2) given below shows the different quarter car suspension system parameters values.

Table 2.2 Quarter car suspension system parameters values

Parameter	**Value**	**Unit**
Sprung mass (M_s)	200	Kg
Unsprung mass (M_u)	40	Kg
Tire stiffness (K_t)	20,000	N/m

More above different combinations of stiffness and damping coefficient will be applied to the suspension model during simulation are given in table (2.3) below.

Table 2.3 Combinations of stiffness and damping coefficient

S.N.	**1**				**2**				**3**				**4**			
Spring stiffness, K_a (N/m)	1500				1600				1700				1800			
Damping coefficient, C_a $(^{N}/_{m/s})$	30	40	50	60	30	40	50	60	30	40	50	60	30	40	50	60

Using Simulink tool in MATLAB software a quarter car suspension system model that employ a PID controller is developed to control the vertical acceleration and improve ride comfort and handling of vehicle. As shown in figure (2.11) below:

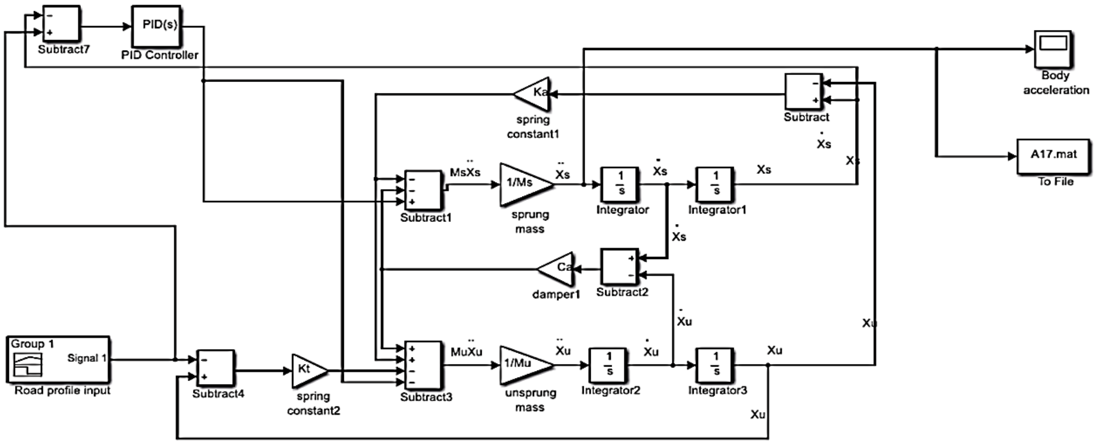


Figure 2.11 Simulink model of active quarter car suspension

After simulating the suspension system, the main concern of the analysis process was the response of the body or chassis acceleration. The results for body acceleration of active and passive suspension are obtained from the Simulink model of suspension. The results for four different combinations of spring stiffness and damping coefficient as shown in table (2.3) and figures (2.12), (2.13), (2.14) and (2.15). It is observed from the prementioned figures that with the implementation of PID controller, there is a drastic change in body acceleration of active suspension compared to passive suspension. It also observed that, with increase in damping coefficient, the body acceleration decreases for both passive and active suspension. Also, with increase in spring stiffness from figure (2.12) to figure (2.15), body acceleration increases.

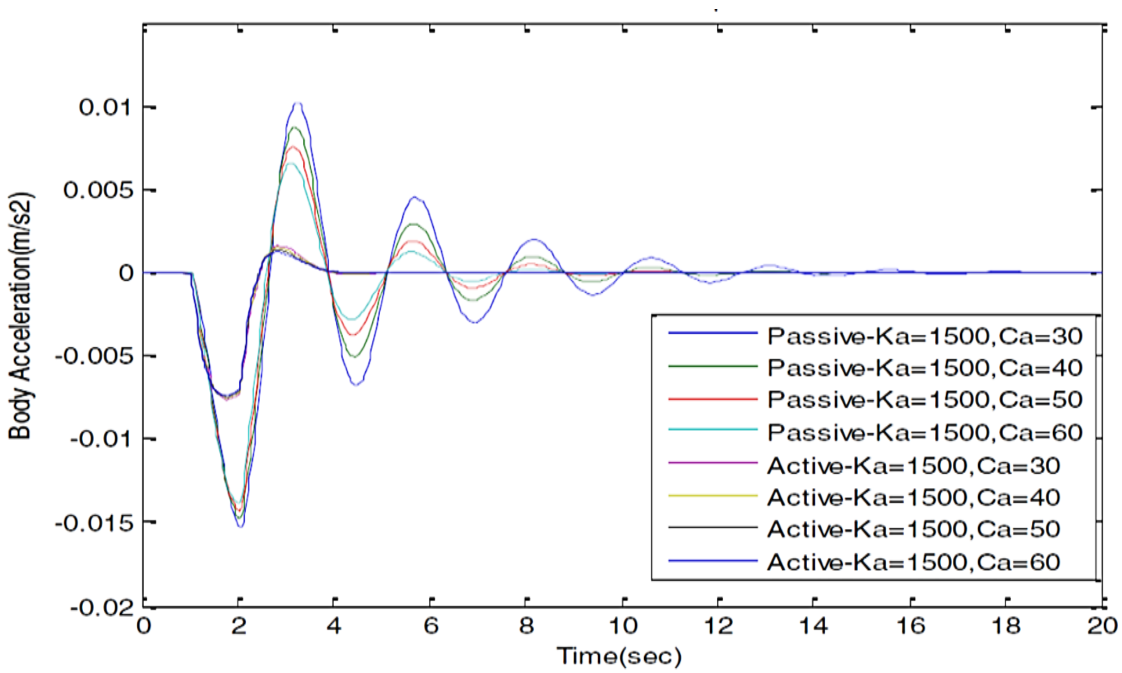


Figure 2.12 Body acceleration of active and passive suspension for (Ka=1500 and Ca=30 - 60)

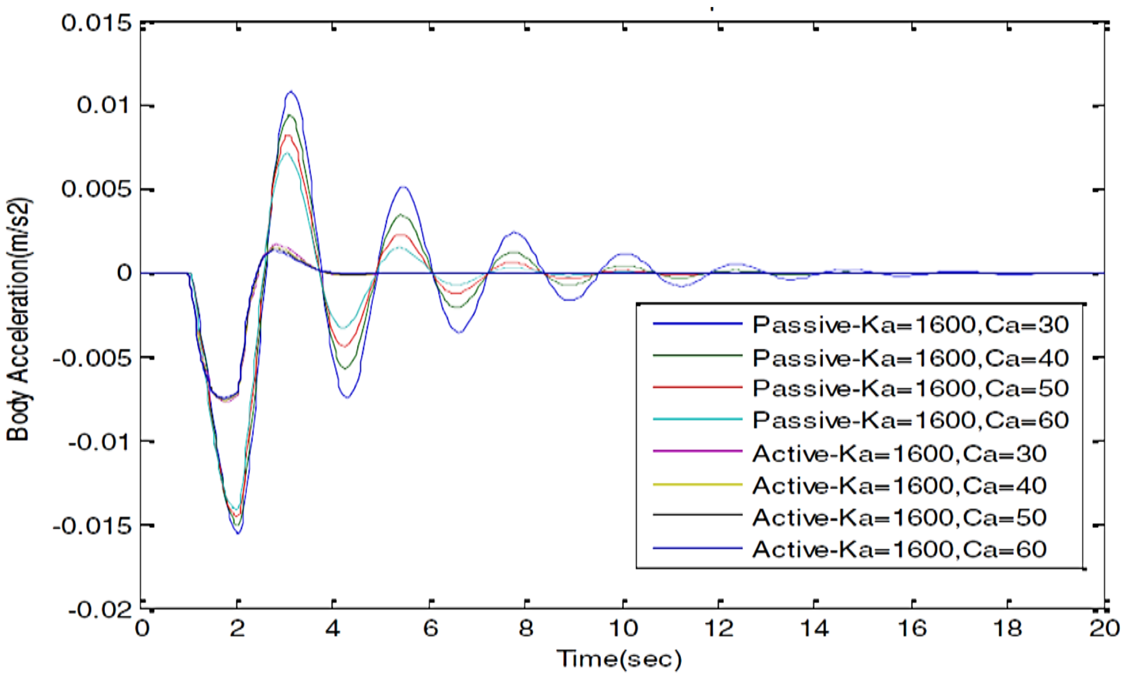


Figure 2.13 Body acceleration of active and passive suspension for (Ka=1600 and Ca=30 - 60)

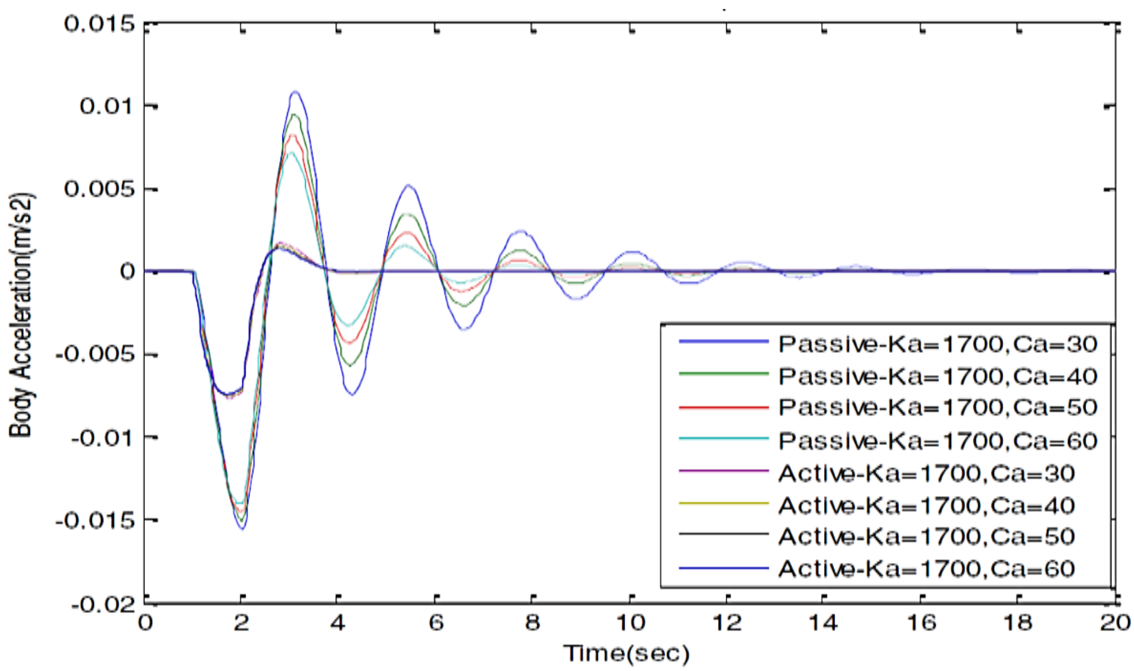


Figure 2.14 Body acceleration of active and passive suspension for (Ka=1700 and Ca=30 - 60)

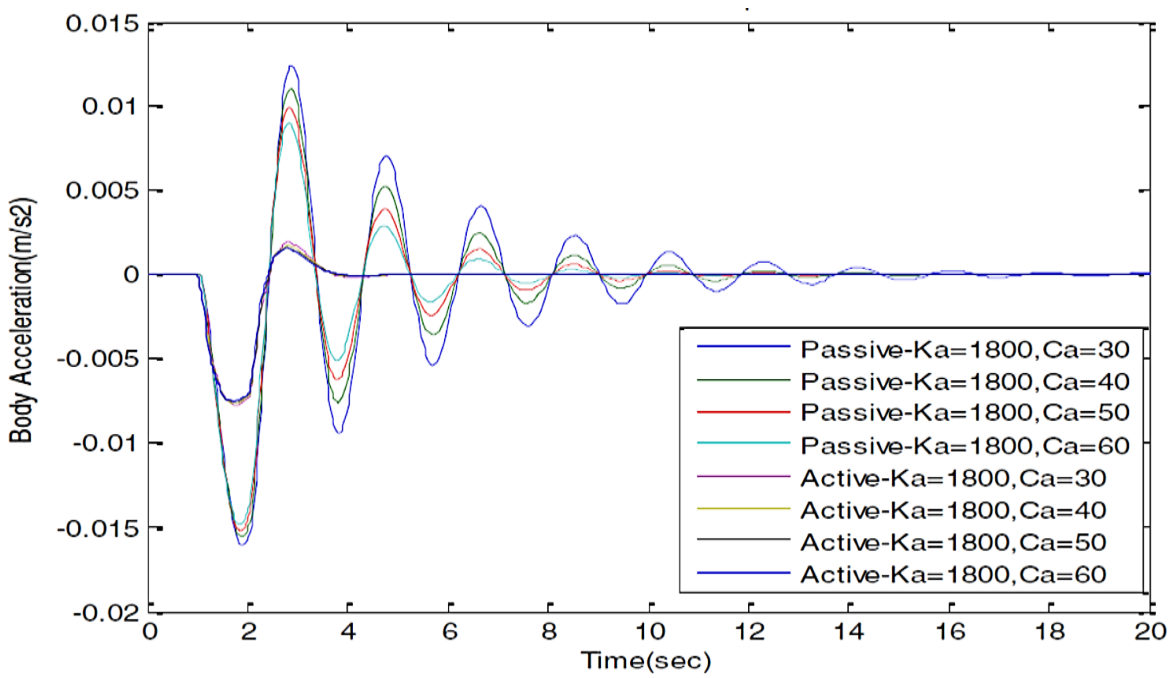


Figure 2.15 Body acceleration of active and passive suspension for (Ka=1800 and Ca=30 - 60)

A.E.-N.S. Ahmed, A.S. Ali, N.M. Ghazaly, G. El-Jaber in their published thesis PID controller of active suspension system for a quarter car model (2015), have reported in their state of art a Simulation attempt based on a mathematical model for quarter car adopted by using MATLAB/SIMULINK and the active suspension system model is illustrated in figure (2.16) below (Ahmed et al., 2015).

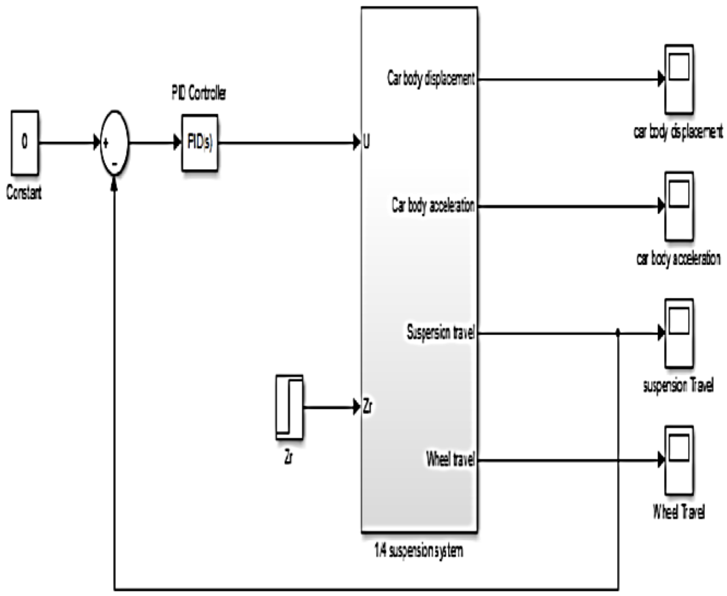


Figure 2.16 Simulink Model of active Suspension system using PID Controller

Performance of the suspension system in term of ride quality and car handling was observed, where road disturbance is assumed as the input for the system. Parameters were observed are the suspension travel, wheel travel, the car body acceleration and displacement of a quarter car. The objective is to minimize the amplitude value for suspension travel, wheel deflection and the car body acceleration. The steady state for each part also should be fast. Suspension travel, wheel travel, the car body acceleration and displacement for quarter car are obtained as shown in the following Figures (2.17), (2.18), (2.19) and (2.20) for a Step input, Z_r equal to 0.1.

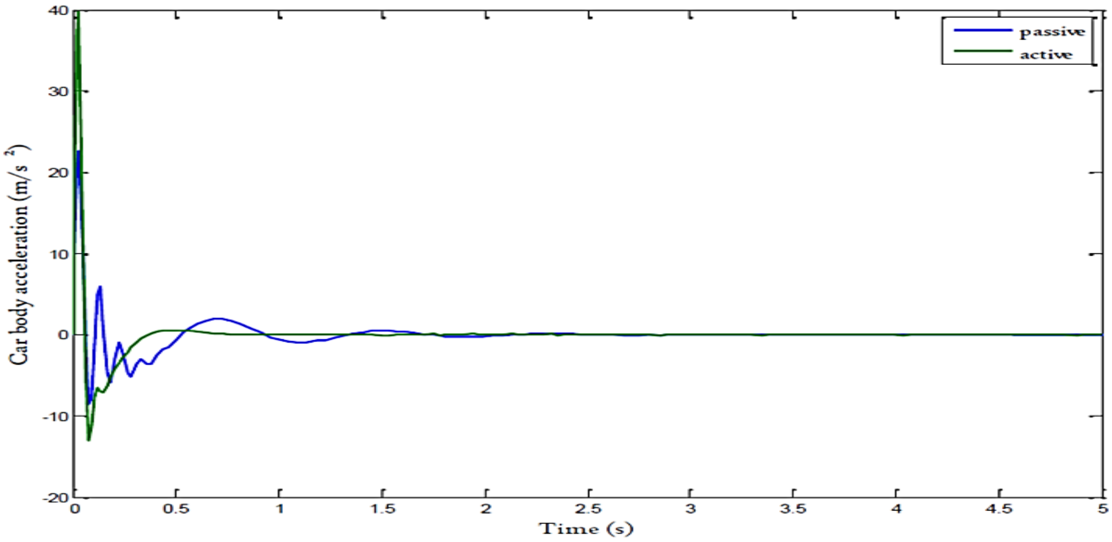


Figure 2.17 Car body acceleration for step input Zr = 0.1

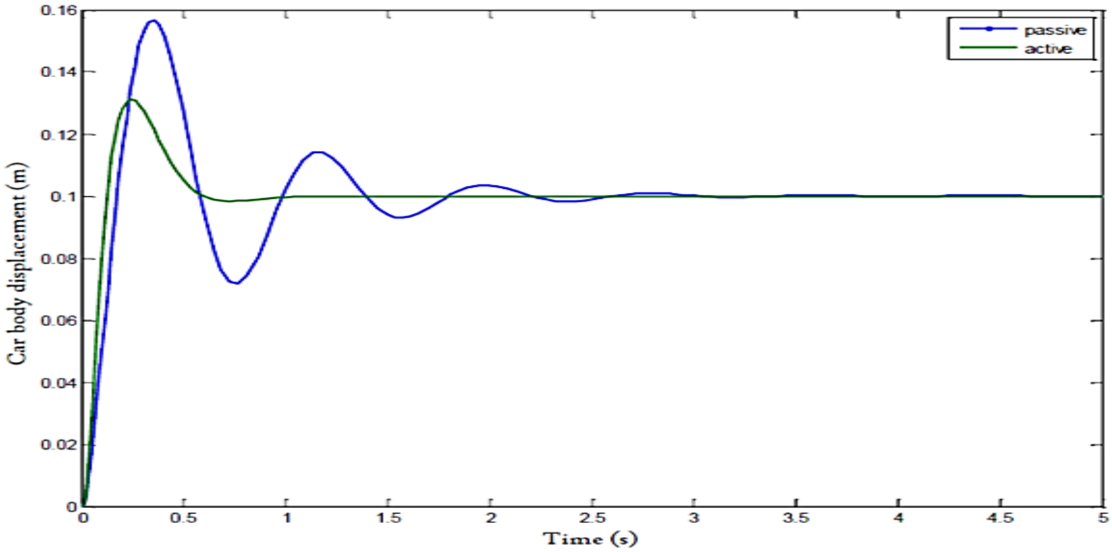


Figure 2.18 Car body displacement for step input Zr = 0.1

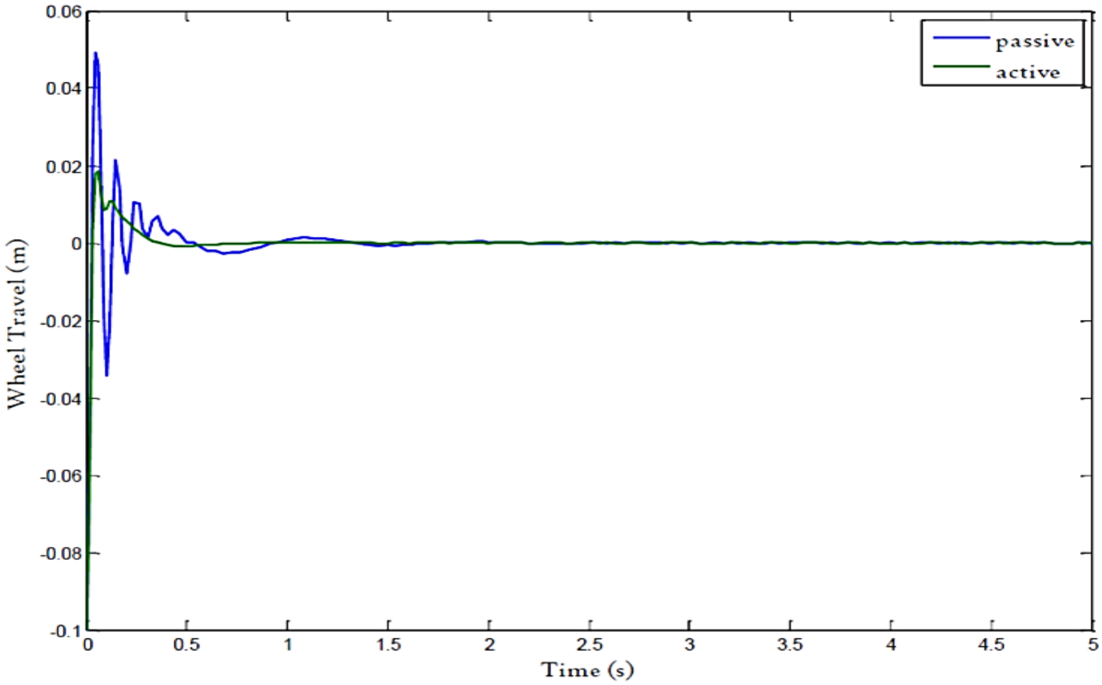


Figure 2.19 Wheel Travel for step input Zr = 0.1

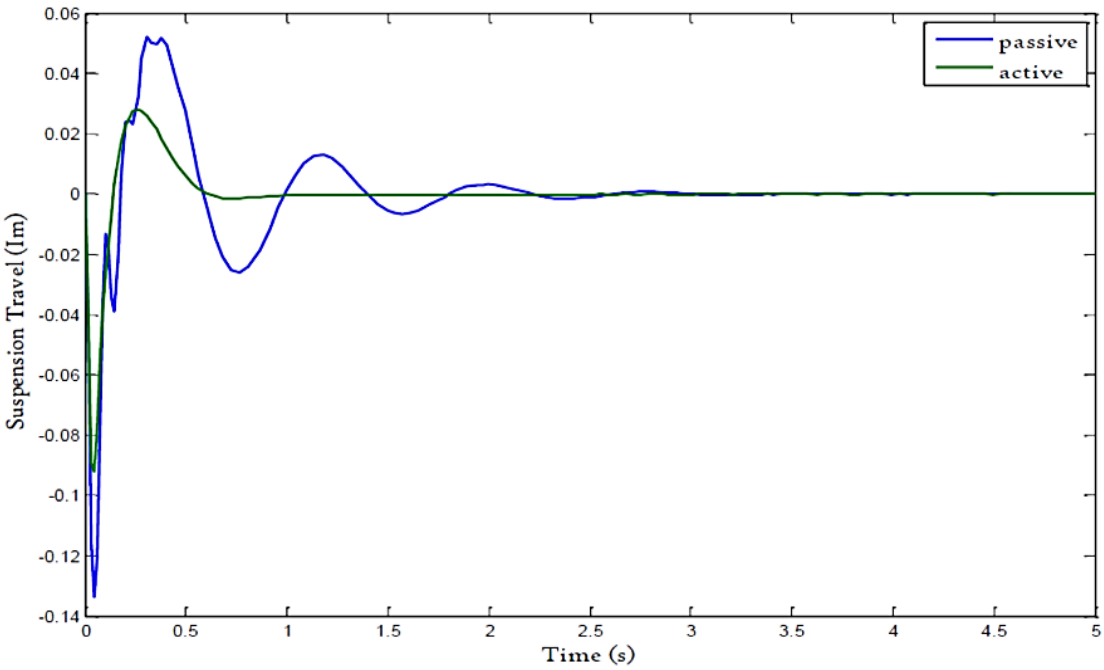


Figure 2.20 Suspension travel for step input Zr = 0.1

Then, as shown in the figures above both peak (overshoot) value and settling time have been reduced by the active suspension system compared to the passive system for all the parameters of sprung mass acceleration (passenger comfort), suspension deflection (road holding) and tire deflection. In additions, Table (2.4) shows the percentage of reduction in peak values of the various parameters for the step road input, the percentage of reduction can be calculated as represented in equation (2.2) below:

$$\% \, Reduction = \frac{passive\ response - active\ response}{passive\ response} \times 100 \qquad (2.2)$$

Table 2.4 Reduction in overshoot values for step road input

S.N.	Parameter	Passive	Active	% Reduction
1	Car body acceleration (m/s^2)	2.00	0.55	72.50
2	Suspension travel (m)	0.052	0.0281	46.00
3	Wheel travel (m)	0.0491	0.0186	62.11

The car body acceleration, suspension travel and wheel travel parameters that measure the performance of quarter car are obtained as shown in the following Figures (2.21), (2.22) and (2.23) for a sinusoidal input (bumpy road).

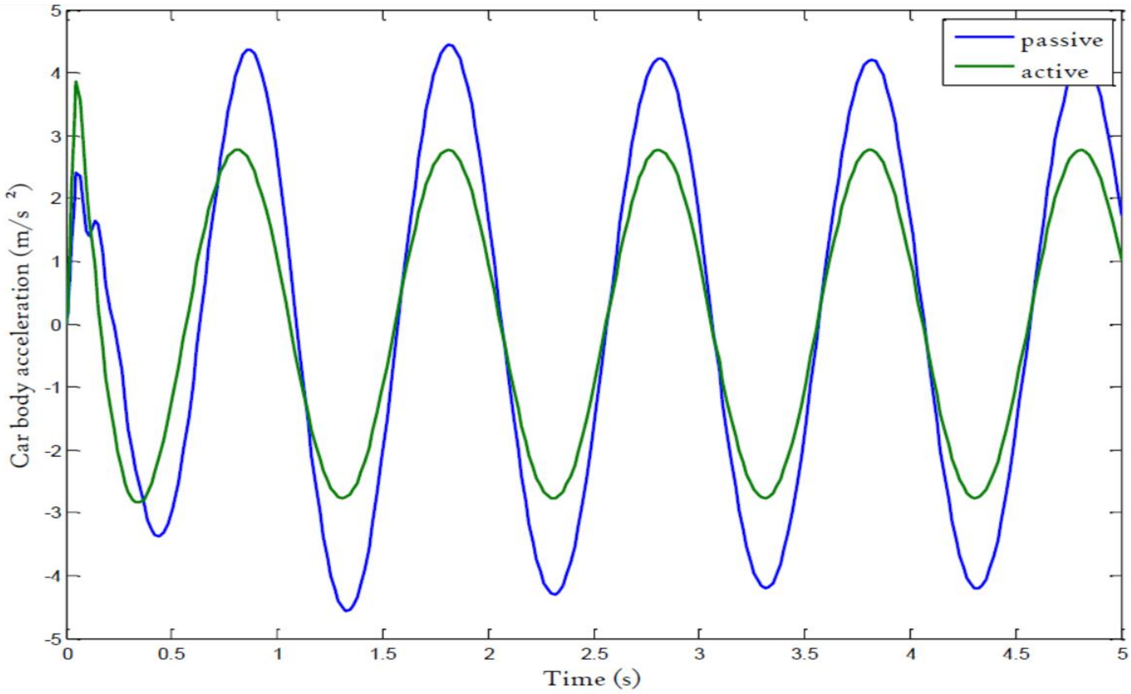


Figure 2.21 Car body acceleration for bumpy road

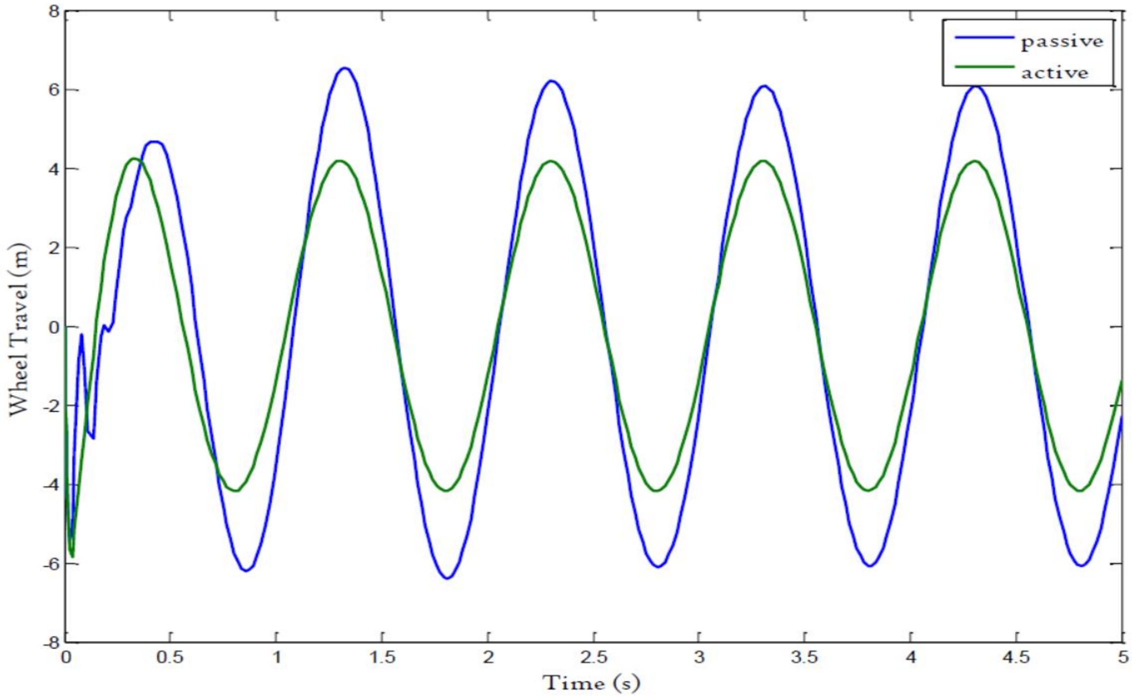


Figure 2.22 Wheel Travel for bumpy road

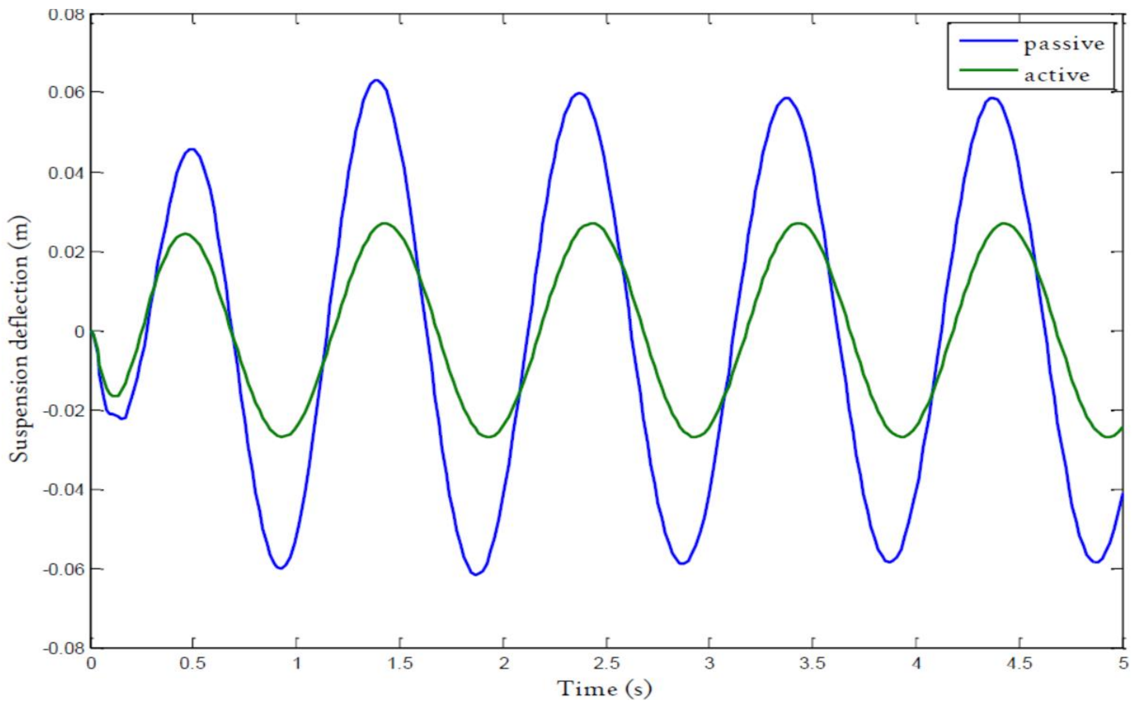


Figure 2.23 Suspension deflection for bumpy road

Figures (2.21), (2.22) and (2.23) illustrate that overshoot values and settling time have been reduced by the active system compared to the passive system for all the parameters of sprung mass acceleration (passenger comfort), suspension deflection (road holding) and tire deflection. Table (2.5) gives the percentage reduction in peak values of the various parameters for the sinusoidal input (bumpy road).

Table 2.5 Reduction in overshoot values for bumpy road

S.N.	Parameter	Passive	Active	% Reduction
1	Car body acceleration (m/s^2)	4.5628	2.772	39.24
2	Suspension travel (m)	0.0632	0.027	57.3
3	Wheel travel (m)	0.00654	0.004176	36.15

The car body acceleration, suspension travel and wheel travel parameters that measure the performance of quarter car are obtained as shown in the following Figures (2.24), (2.25) and (2.26) for a random road input signal.

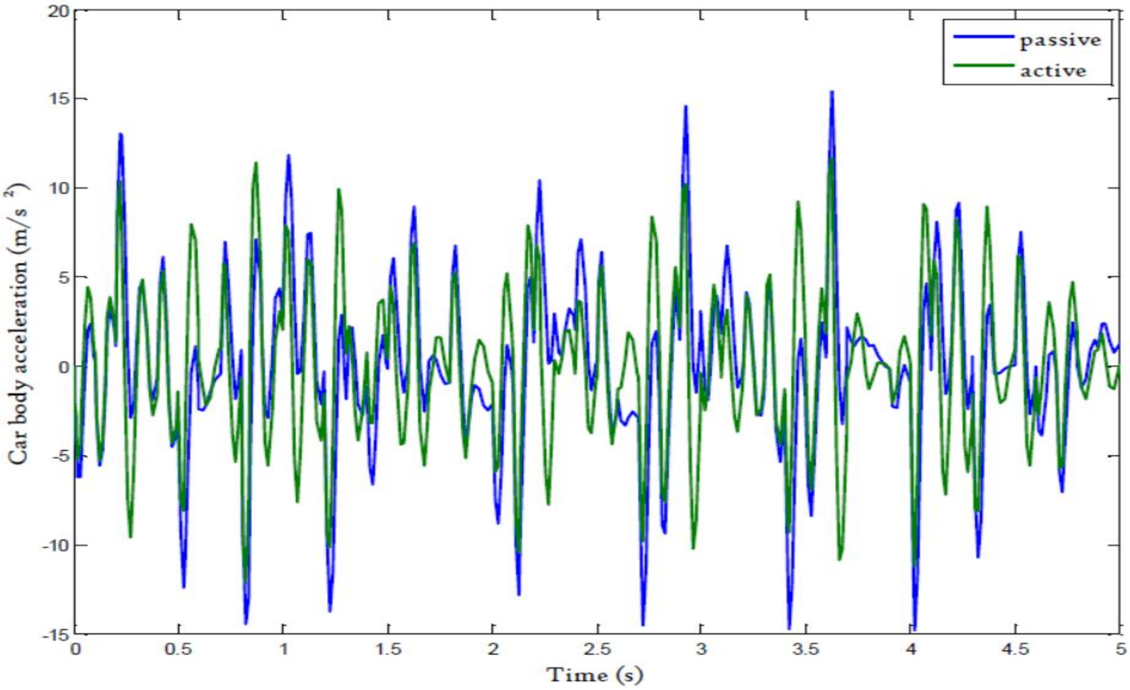


Figure 2.24 Car body acceleration for random road

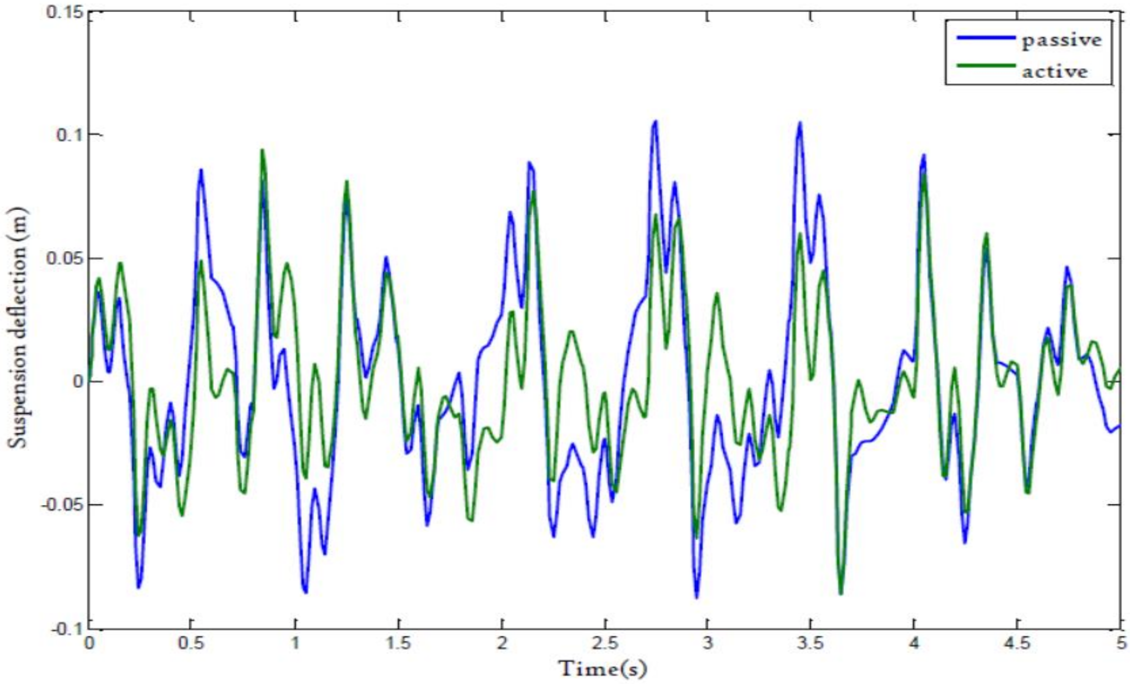


Figure 2.25 Suspension deflection for random road

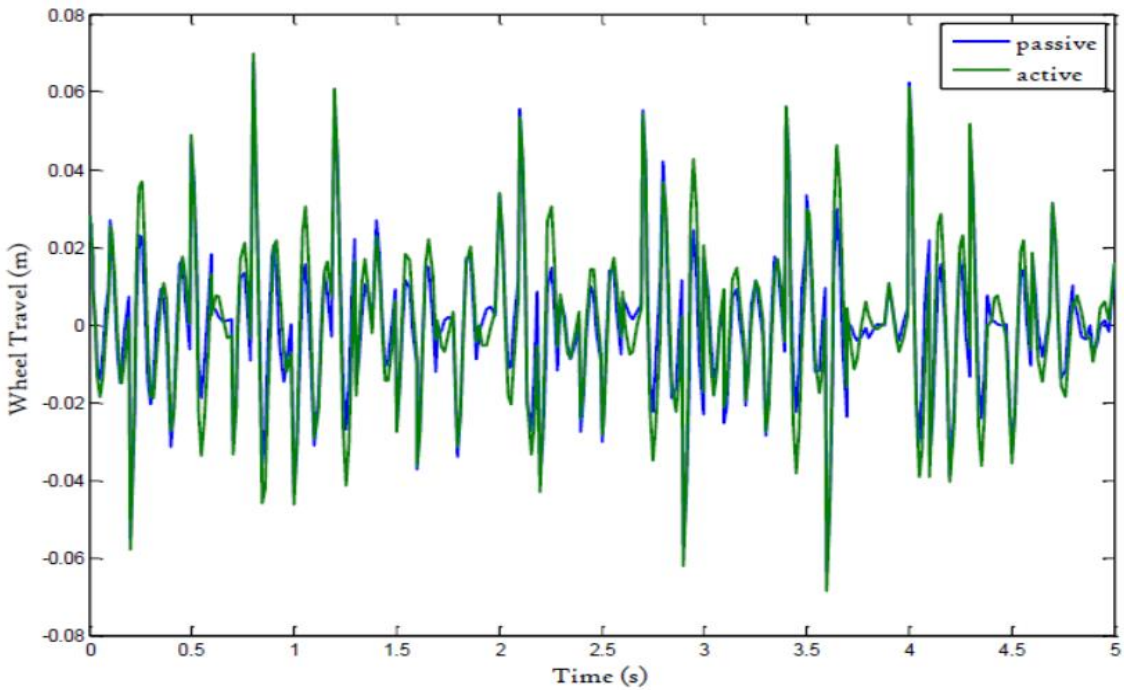


Figure 2.26 Wheel Travel for random road

Figures (2.24), (2.25) and (2.26) illustrate that overshoot values and settling time have been reduced by the active system compared to the passive system for all the parameters of sprung mass acceleration (passenger comfort), suspension deflection (road holding) and tire deflection. Table (2.6) gives the percentage reduction in peak values of the various parameters for the random input (random road).

Table 2.6 Reduction in overshoot values for random road

S.N.	Parameter	Passive	Active	% Reduction
1	Car body acceleration (m/s^2)	15.4569	11.68	24.44
2	Suspension travel (m)	0.1052	00676	35.74
3	Wheel travel (m)	0.0625	0.061	2.40

2.6 Modeling suspension systems using SimMechanics tool

SimMechanics tool can be prescribed briefly as the MATLAB software multibody simulation environment for 3D mechanical systems, that can provide different analysis modes and dynamic visualization options by using preset blocks. SimMechanics blocks do not directly model mathematical functions but yet have the potential to represent the definite physical (e.g. mechanical) nature and behavior of those systems. The block set consists of block libraries for bodies, joints, sensors, actuators, constraints and force elements. It facilitates the modeling of complex systems enormously via two essential features, the first is to eliminate the necessity of solving non-linear mathematical equations to bound or define the system, and the latter is to determine the initial states of the system.

Blocks can be configured by the user via graphical user interface (GUI), then models can be generated, adjusted and changed with certain commands from MATLAB programs (Matlab, 2019b).

According to A. Tandel, A. Deshpande, S. Deshmukh, K. Jagtap in their piece of article named Modeling, analysis and PID controller implementation on double wishbone suspension using SimMechanics and Simulink (2014). The quarter car suspension is modeled and assembled using SimMechanics 2^{nd} generation toolbox in MATLAB software. The modules shown in figure (2.21) are used for modeling and assembly. Using body block from the body elements module, extrusion and revolution can be given to the geometry of the part, where the geometry is specified using the coordinates. Rigid transform block from the frames and transforms module is used to rigidly connect the different parts of the component. It specifies the position of one part with respect to other using two types of motion, translational and rotational. There are different joints in the joints module and depending on the degree of freedom required, appropriate joint is selected for connecting the two components during assembly of the model. The input to the model is given using forces and torques module. The utility module contains the block Simulink-PS converter, which converts the Simulink signals to physical system signals. This converter is used to connect the input block, which is the signal builder block from source module available in Simulink. The output to the model is obtained using scope block from sink module of Simulink. The PS-Simulink converter is used to connect output block with the model. figure (2.27) shows the SimMechanics model of active wishbone suspension prepared in SimMechanics 2^{nd} generation in MATLAB software. The blocks indicate different components of suspension, like chassis, upper wishbone, lower wishbone, kingpin, damper, rim and tire, which are connected to each other by means of constrained joints. The step input equivalent to bump height of 2 cm is given to the wheel using signal builder block. The spring stiffness and damping coefficient of spring and damper is varied to find out their effect on the acceleration of chassis. Active suspension uses a controller, which controls the actuator of suspension

(Tandel et al., 2014). It generates the force which helps in suppressing the body acceleration. PID controller is used in the active suspension model shown in figure (2.27). The PID block is available in Simulink library. The Proportional, integral and derivative gain is tuned automatically by the block.

After simulating suspension system, the main concern from the responses is the body or chassis acceleration. Responses for body acceleration of active and passive suspension for four different combinations as given in table (2.3) are shown in figures (2.28), (2.29), (2.30) and (2.31). It is observed from the figures mentioned before that, with the implementation of PID controller, body acceleration of active suspension reduces to almost half of passive suspension. Also, with the increase in damping coefficient, the body acceleration decreases for both active and passive suspension. It is also observed that, with the increase in spring stiffness, body acceleration increases, which is not good for rider's comfort and life of vehicle.

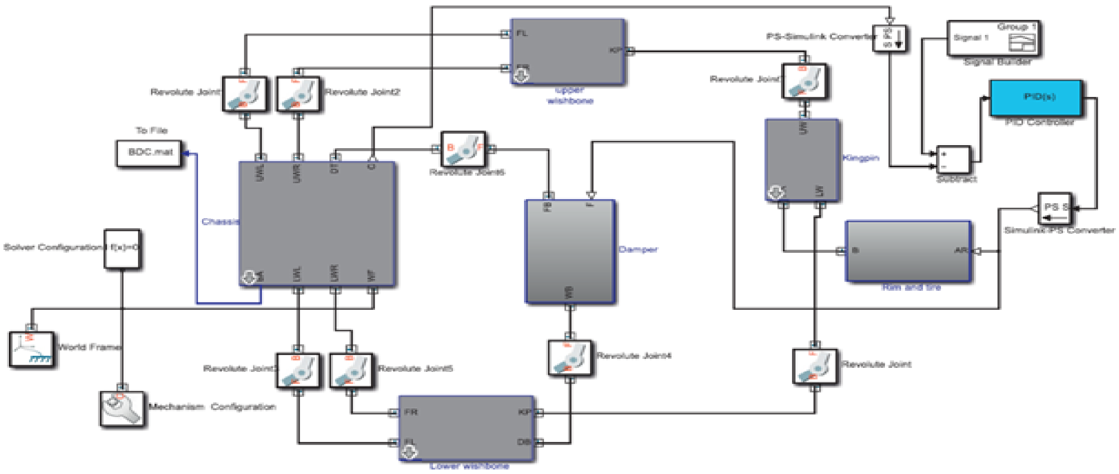


Figure 2.27 SimMechanics model of active quarter car suspension

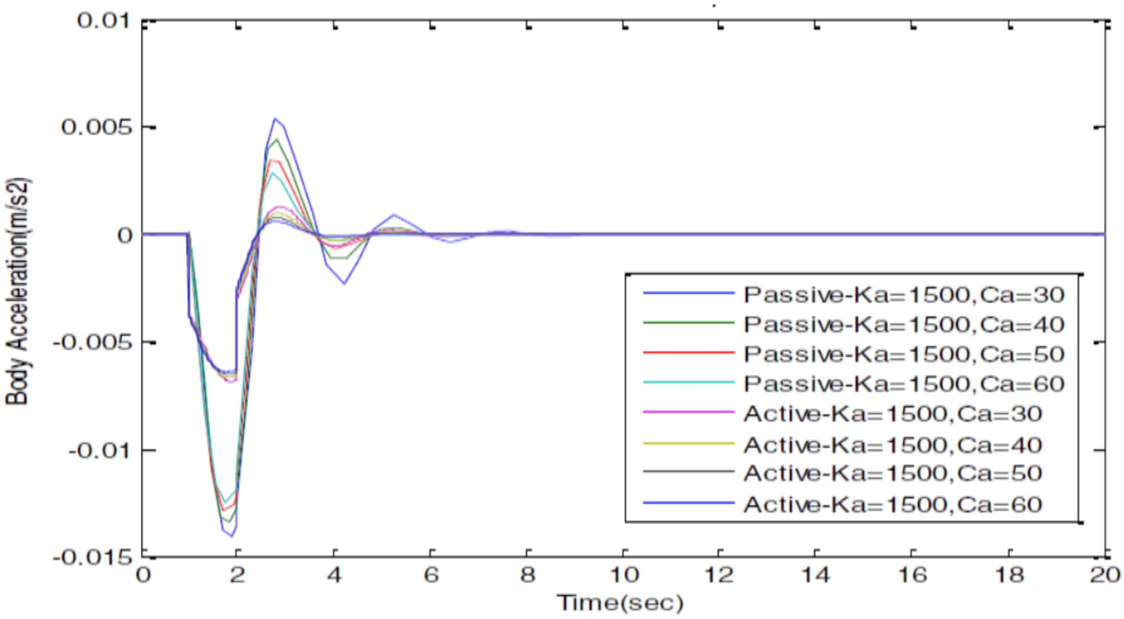


Figure 2.28 Body acceleration of active and passive suspension for (Ka=1500 and Ca=30 - 60)

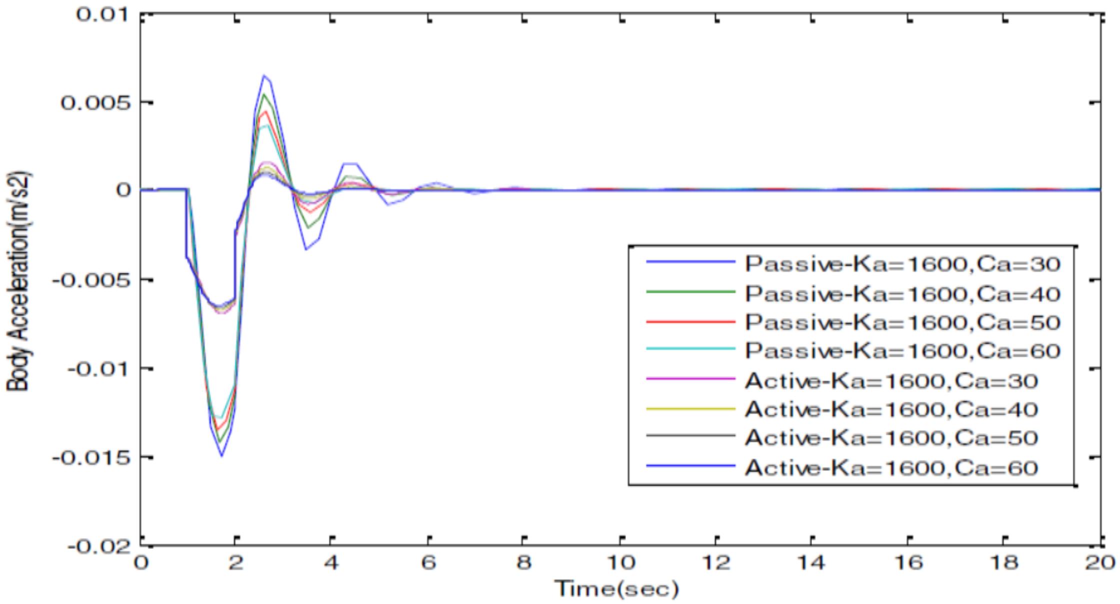


Figure 2.29 Body acceleration of active and passive suspension for (Ka=1600 and Ca=30 - 60)

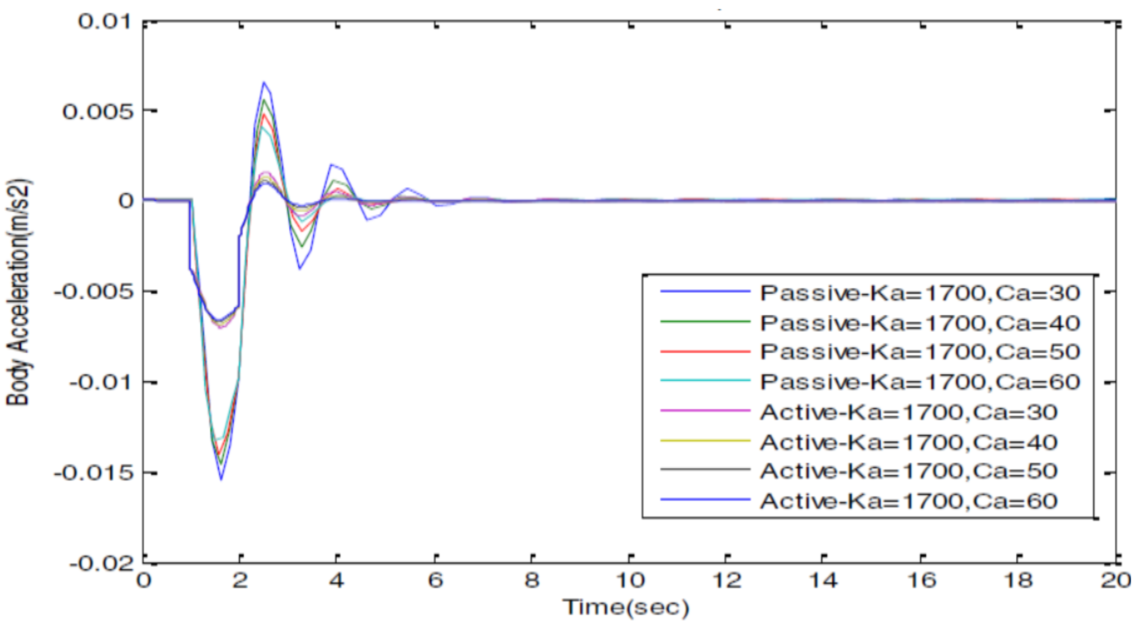


Figure 2.30 Body acceleration of active and passive suspension for (Ka=1700 and Ca=30 - 60)

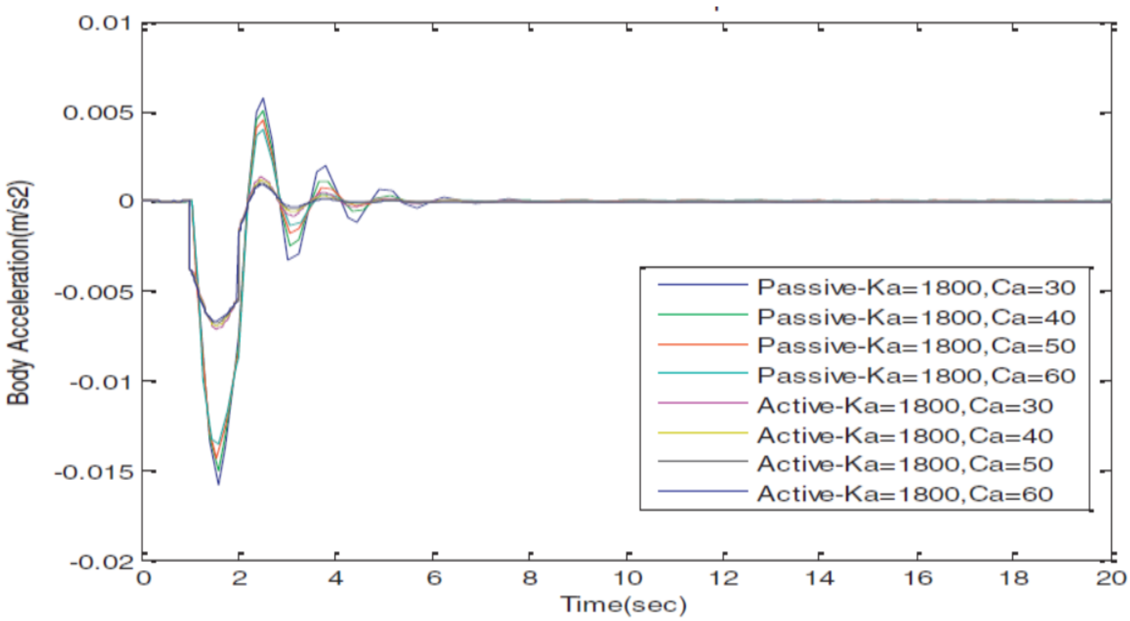


Figure 2.31 Body acceleration of active and passive suspension for (Ka=1800 and Ca=30 - 60)

CHAPTER III

METHODOLOGY AND APPROACH

3.1 Introduction

In concern with methodology and approach of performing this project, a full mathematical, graphical and physical modeling of the system dynamics related to the vehicle suspension system, proposed PID controlling strategy and road disturbances are described. This shall provide the basis for the rigorous computer simulation study to be carried out using MATLAB Simulink and SimMechanics software package. The mathematical modeling of the dynamic system is performed using the Newtonian mechanics. The suspension system is modeled based on a quarter car configuration. The active suspension system is specifically designed and modeled with the feedback control element embedded into the system. A number of assumptions that are made throughout the modeling and simulation study will be also described.

3.2 Quarter car model

A Quarter car model is used to derive the mathematical model of both passive and active suspension systems. The quarter car model is commonly used in the analysis and design processes of suspension systems due to its simplicity, but it is yet has the potential of capturing many significant characteristics of the full car model. It is also representative adequately to validate the suspension simulations.

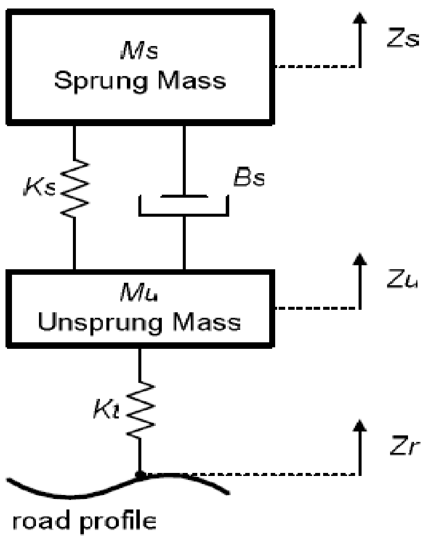


Figure 3.1 Quarter car vehicle passive suspension

Figure (3.1) above shows a quarter car vehicle passive suspension system. Single wheel and axle are connected to the quarter portion of the car body (Sprung mass) through a passive spring and damper. The tire (Unsprung mass) is assumed to have only the spring feature and is in contact with the road terrain at the other end. The road terrain serves as an external disturbance input to the system.

The equations of motion for the passive system are based on Newtonian mechanics and they will be provided as follows (Priyandoko and Mailah, 2007):

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - b_s(\dot{z}_s - \dot{z}_u) \quad (3.1)$$

$$m_u\ddot{z}_u = k_s(z_s - z_u) + b_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) \quad (3.2)$$

Where:

m_s and m_u	≡	Sprung mass and Unsprung mass respectively
b_s	≡	Damping coefficient
k_s and k_t	≡	Stiffness of spring and tire respectively
z_s and z_u	≡	Displacement of sprung mass and Unsprung mass respectively
z_r	≡	Displacement of road
$z_s - z_u$	≡	Deflection of suspension
$z_u - z_r$	≡	Deflection of tire
$\dot{z}_s$ and $\dot{z}_u$	≡	Velocity of sprung mass and Unsprung mass respectively
$\ddot{z}_s$ and $\ddot{z}_u$	≡	Acceleration of sprung mass and Unsprung mass respectively

Active suspension system for a quarter car model can be erected by adding an actuator parallel to spring and damper. Figure (3.2) shows a scheme of a quarter car vehicle active suspension system.

The equations of motion for an active system are as follows:

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - b_s(\dot{z}_s - \dot{z}_u) + f_a \quad (3.3)$$

$$m_u\ddot{z}_u = k_s(z_s - z_u) + b_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) - f_a \quad (3.4)$$

Where:

f_a	≡	Actuator force

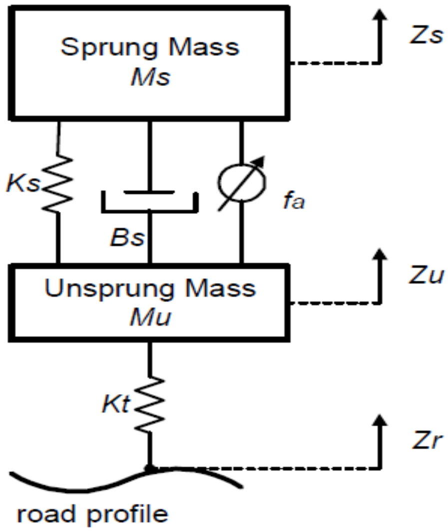


Figure 3.2 Quarter car vehicle active suspension

Some assumptions were made in the modeling process of the active suspension system. those assumptions are summarized as follows:

i. The behavior of the vehicle can be represented accurately by a quarter car model.
ii. The suspension spring stiffness and tire stiffness are linear in their operation ranges and tire does not leave the ground. The displacements of both the body and tire can be measured from the static equilibrium point.
iii. The actuator is assumed to be linear with a constant gain.

3.3 Suspension system parameters

The suspension parameters have been used through the modeling process are adopted from the previous study (Ahmed et al., 2015). Detailed suspension model parameters, simulation software parameters and PID controller parameters are represented in tables (3.1), (3.2) and (3.3) below respectively.

Table 3.1 Suspension model parameters

No.	Parameter	Value	Unit
1	Sprung mass (m_s)	250	Kg
2	Unsprung mass (m_u)	50	Kg
3	Spring stiffness(K_s)	16812	N/m
4	Damping coefficient(b_s)	1000	Ns/m
5	Tire stiffness(K_t)	190000	N/m

Table 3.2 Simulation software parameters

No.	Parameter	Value
1	Solver	(ode45) Domand - Prince
2	Type	Variable step
3	Simulation time	10 seconds
4	Minimum step size	Auto
5	Maximum step size	1/120
6	Initial step size	Auto
7	Relative tolerance	1e-3
8	Absolute tolerance	Auto
9	Zero crossing control	Use local setting

Table 3.3 PID controller parameters

No.	Parameter	Value
1	Control method	Ziegler-Nichols method
2	Proportional gain K_P	1759.04
3	Integral gain K_I	28195.40
4	Derivative gain K_D	1500
5	Filter coefficient	100

3.4 Disturbance Models

There are two types of disturbances introduced to the vehicle suspension system in this study. They are the step input and bump and hole (sinusoidal) disturbance, that is called road disturbance which represents the irregular road profile.

Figures (3.3) and (3.4) show the disturbances introduced to the system, that are 1 cm step input signal and the bump and hole (sinusoidal) signal respectively.

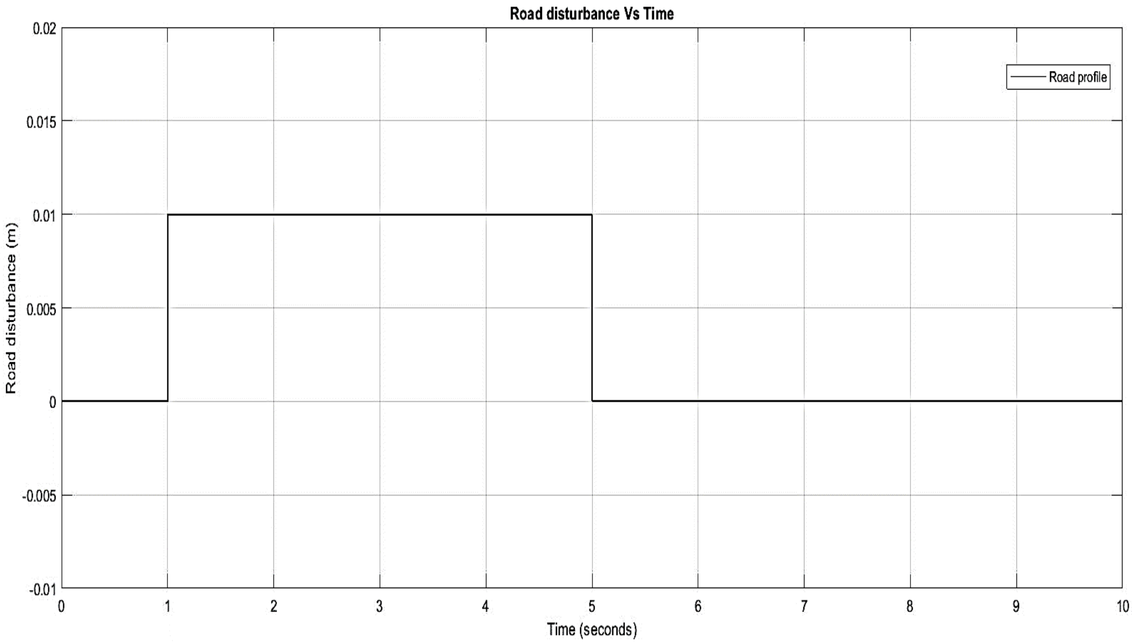


Figure 3.32 Step input

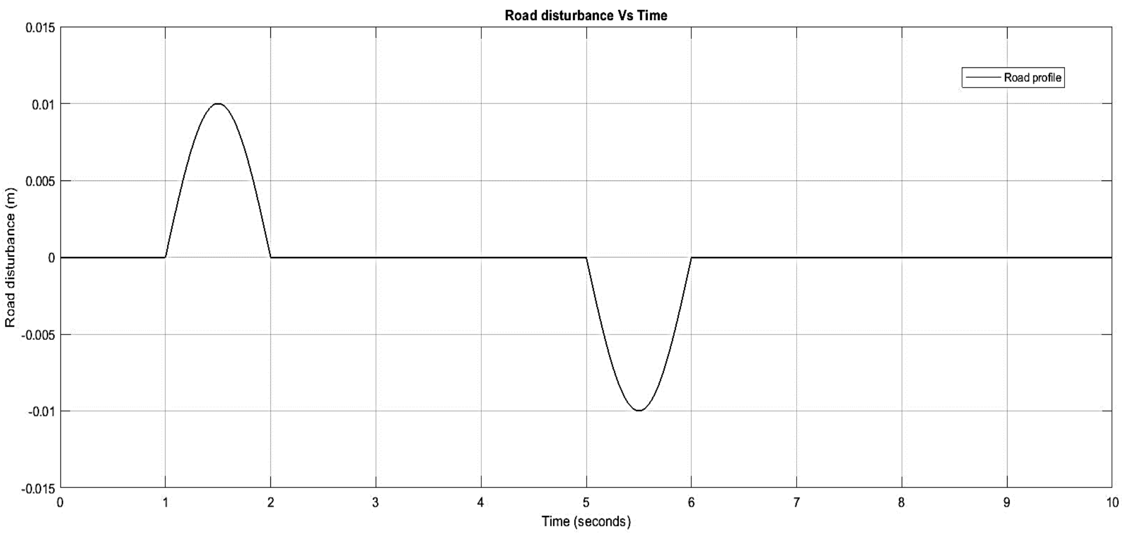


Figure 3.4 Bump and hole input

3.5 Modeling suspension System via Simulink tool

3.5.1 Passive Suspension System Simulink model

The suspension system Simulink modeling was started basically with developing the passive suspension system of a quarter car model. The dynamical system is separated into two systems as the suspension system involves two degrees of freedom. This passive suspension model was modeled in a Simulink form as shown in Figure (3.5). This model was built based on the equations (3.1) and (3.2). There is an open loop system with no feedback element for appropriate adjustment of parameters.

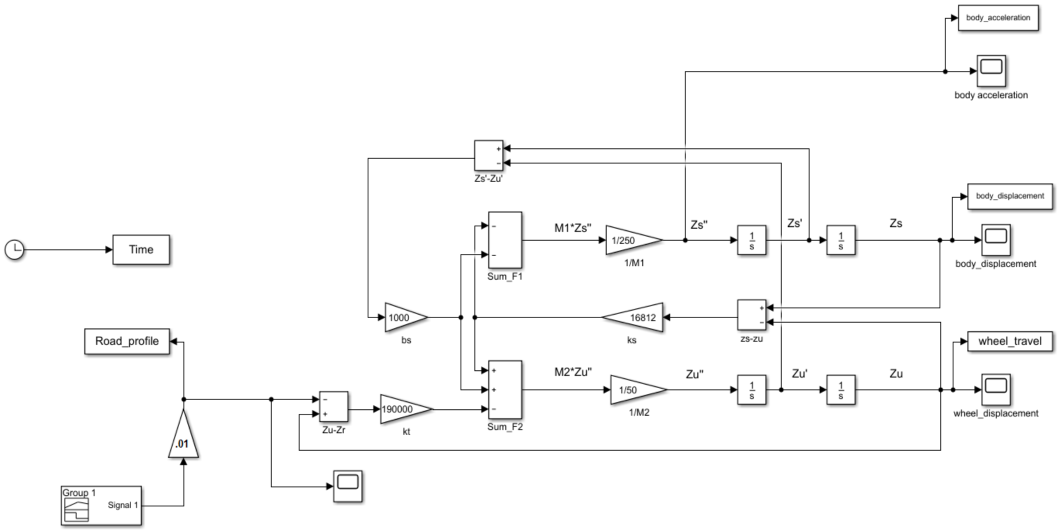


Figure 3.5 Simulink model of passive suspension system

3.5.2 Active Suspension System Simulink model

Active suspension system requires an actuator force to provide a better ride and handling than the passive suspension system. The actuator force, f_a is an additional input to the suspension system model. The model in Simulink was built based on equations (3.3) and (3.4). As will be shown in Figure (3.6). The actuator force is controlled by the PID controller which involves a feedback loop.

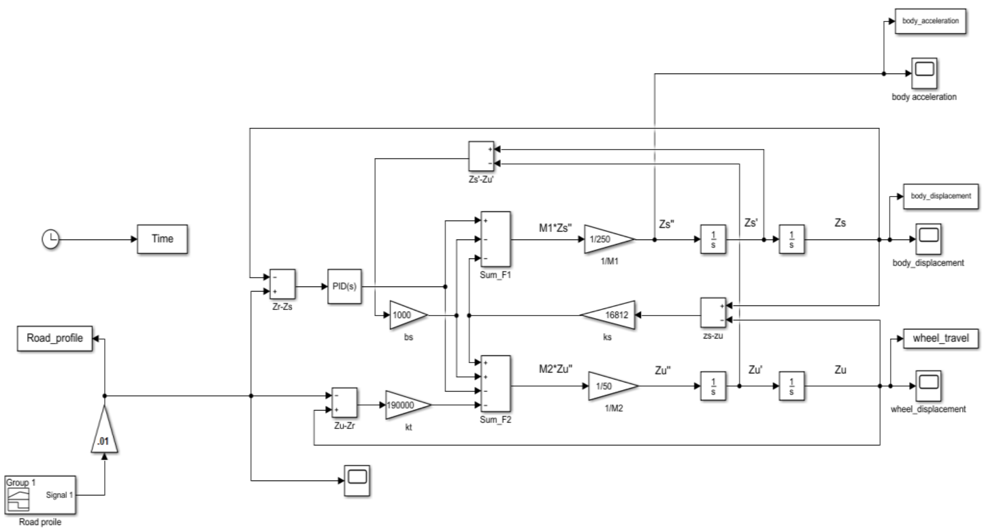


Figure 3.6 Simulink model of active suspension system

3.6 Modeling suspension System via SimMechanics tool

Suspension system computer aided design (CAD) model was imported from Solidwork, then modeled and assembled using SimMechanics 2nd generation toolbox in MATLAB software. The modules shown in figure (3.7) are used for modeling and assembly. Using body block from the body elements module, extrusion and revolution can be given to the geometry of the parts, where the geometry is specified using the coordinates.

Rigid transform block from the frames and transforms module is used to rigidly connect the different parts of the component. It specifies the position of one part with respect to other using two types of motion, translational and rotational. There are different joints in the joints module and depending on the degree of freedom required, appropriate joint is selected for connecting the two components during assembly of the model.

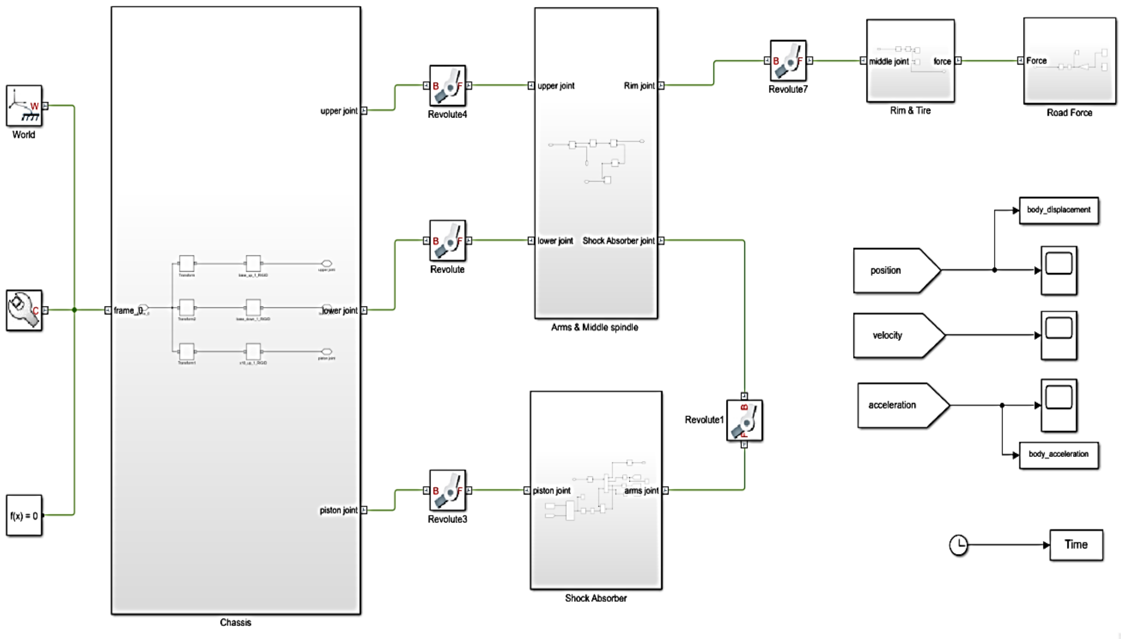


Figure 3.7 SimMechanics model of proposed suspension system

The input to the model is given using forces and torques module. The utility module contains the block Simulink-PS converter, which converts the Simulink signals to physical system signals. This converter is used to connect the input block, which is the signal builder block from source module available in Simulink.

The output to the model is obtained using scope block from sink module of Simulink. The PS-Simulink converter is used to connect output block with the model.

Figure (3.7) shows the SimMechanics model of active wishbone suspension prepared in SimMechanics 2nd generation in MATLAB software. The blocks indicate different components of suspension, like chassis, arms and middle spindle, shock absorber (e.g. damper and spring), rim and tire, which are connected to each other by means of constrained joints. The step input equivalent to bump height of 1 cm is given to the wheel using signal builder block. The spring stiffness and damping coefficient of spring and damper is varied to find out their effect on the acceleration of chassis.

3.6.1 Passive suspension system SimMechanics model

The breakdown of the SimMechanics model described in the above section for the passive suspension system will be represented in the figures below.

The proposed road profile (e.g. step and sinusoidal inputs) are introduced to the system through the road force block that employee the external force and torque represented in figure (3.8).

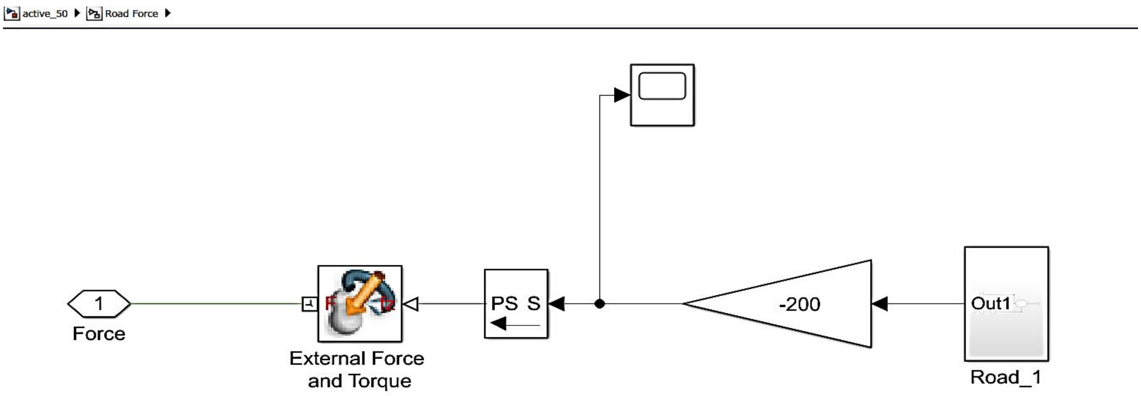


Figure 3.8 Road force

Rim and tire block connection details are shown in figure (3.9) below

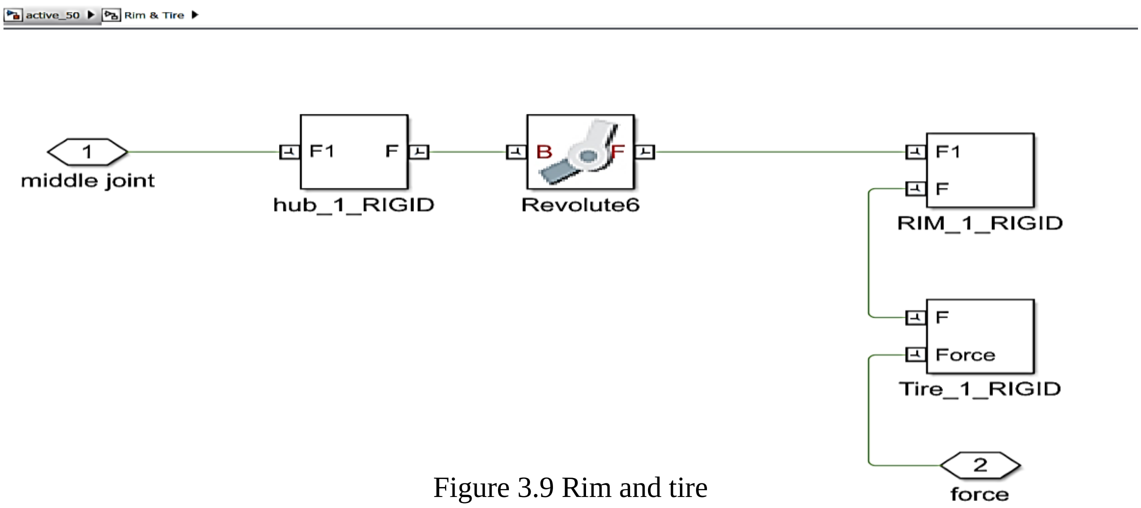


Figure 3.9 Rim and tire

Arms and middle spindle which consist of the upper wishbone, lower wishbone and kingpin are shown in figure (3.10).

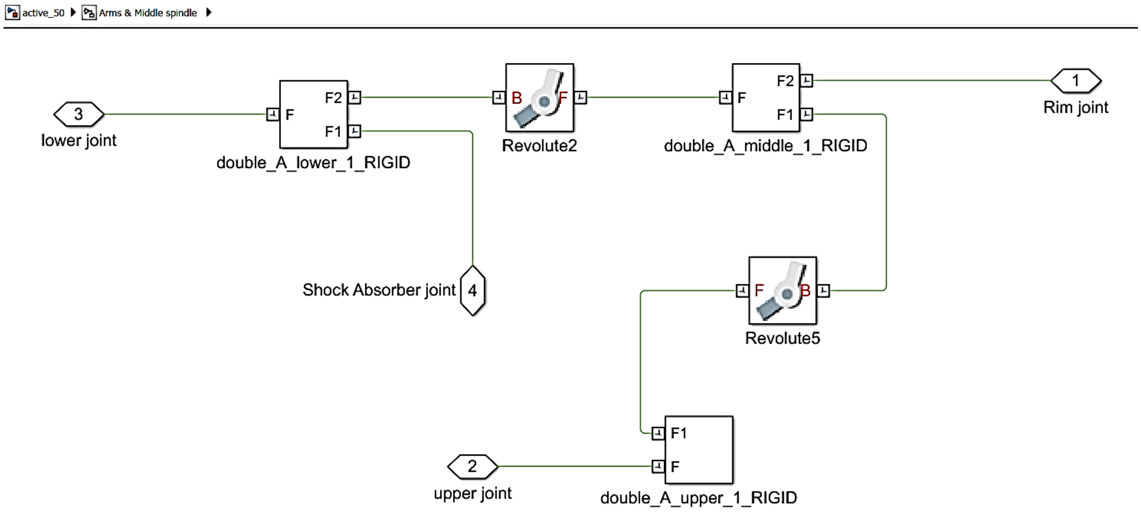


Figure 3.10 Arms and middle spindle

In figure (3.11) in the shock absorber block the (cylindrical 1) joint represents the spring and damper and evolve determining the numerical values of the spring stiffness and damping coefficient.

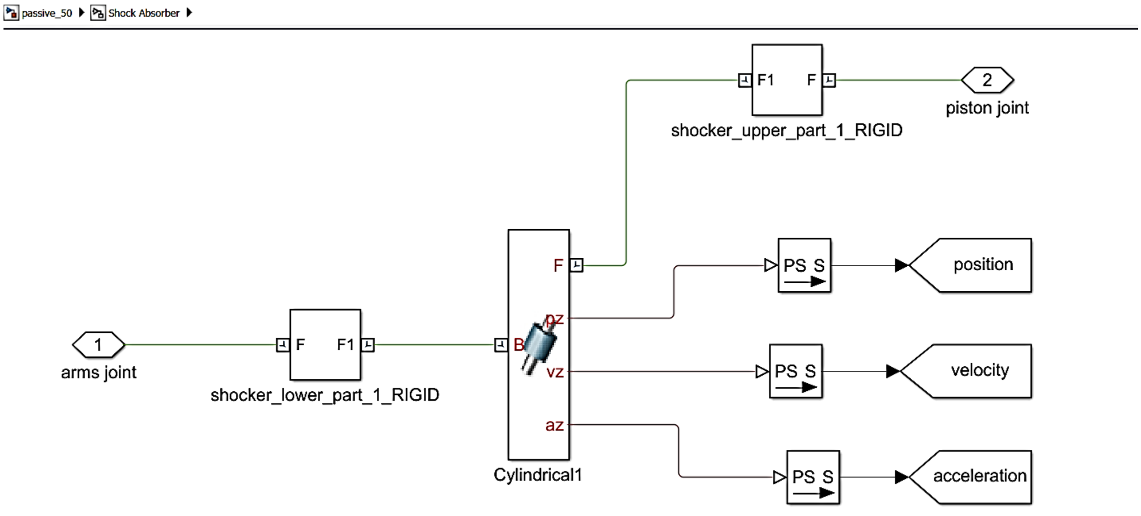


Figure 3.11 Shock absorber

3.6.2 Active Suspension System SimMechanics model

Active suspension system is in fact share all the elements and components of the passive system with an extra actuating element (hydraulic actuator) that adopt a PID controller, which controls the actuator of suspension. It generates the force which helps in suppressing the body acceleration. The PID controller used in the active suspension model is shown in figure (3.12). The PID block is available in Simulink library. And the Proportional, integral and derivative gains are tuned automatically by the block to achieve the best controlling performance.

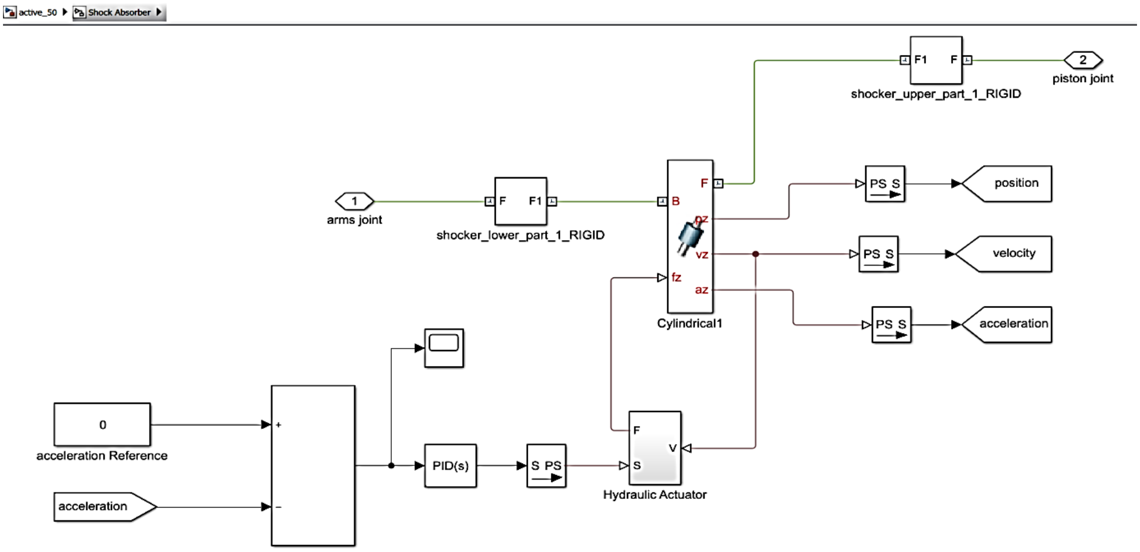


Figure 3.12 Hydraulic actuator and PID controller in active suspension system

3.7 Conclusion

The mathematical equations of vehicle suspension system that have been derived using quarter car model based on Newtonian mechanics were modelled using Simulink and SimMechanics tools for both passive and active suspension systems. Disturbances were also modelled in Simulink and SimMechanics. The active suspension control system in particular were fully modelled complete with the control scheme with intelligent element to be simulated to observe their responses. The simulation results of all models pointed out above will be represented and analyzed in chapter four.

CHAPTER IV

RESULTS AND DISCUSSION

4.1 Introduction

Simulated suspension systems with both passive and active configurations responses results will be represented throughout the following sections of this chapter. The major solicitude of the simulation process for suspension systems will be the responses results in form of body acceleration and body displacement. Comparisons of the results between types of the suspension systems and different types of disturbances (road profiles) are also going to be discussed extensively.

4.2 Suspension system results for Simulink tool model

4.2.1 Passive suspension system results

The response of the passive suspension system in terms of body acceleration and body displacement is shown for the step and sinusoidal inputs aforesaid in the previous chapter in Figures (4.1), (4.2), (4.3) and (4.4) represented below respectively.

i. Passive suspension system responses for step input

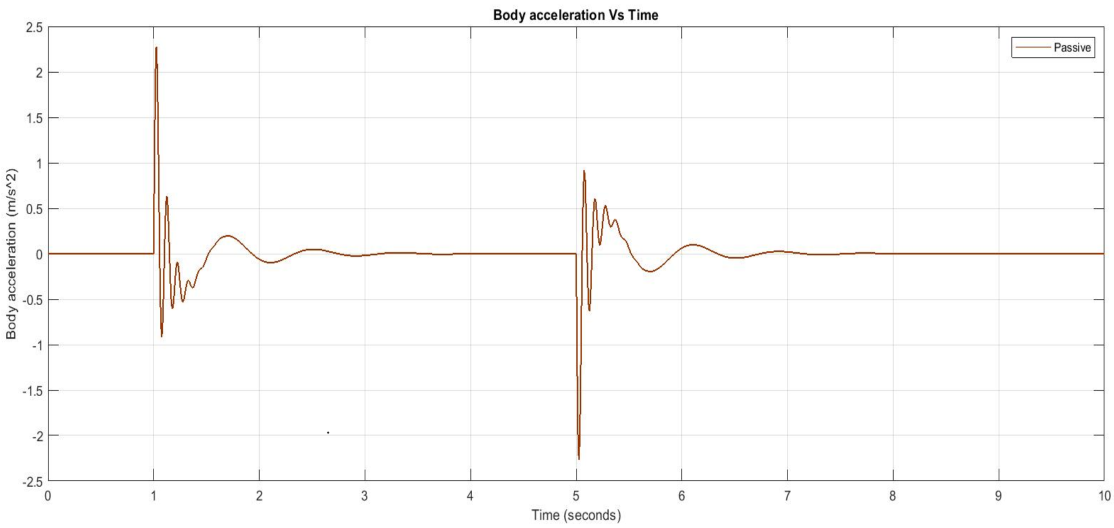


Figure 4.1 Body acceleration of passive suspension system for the step input

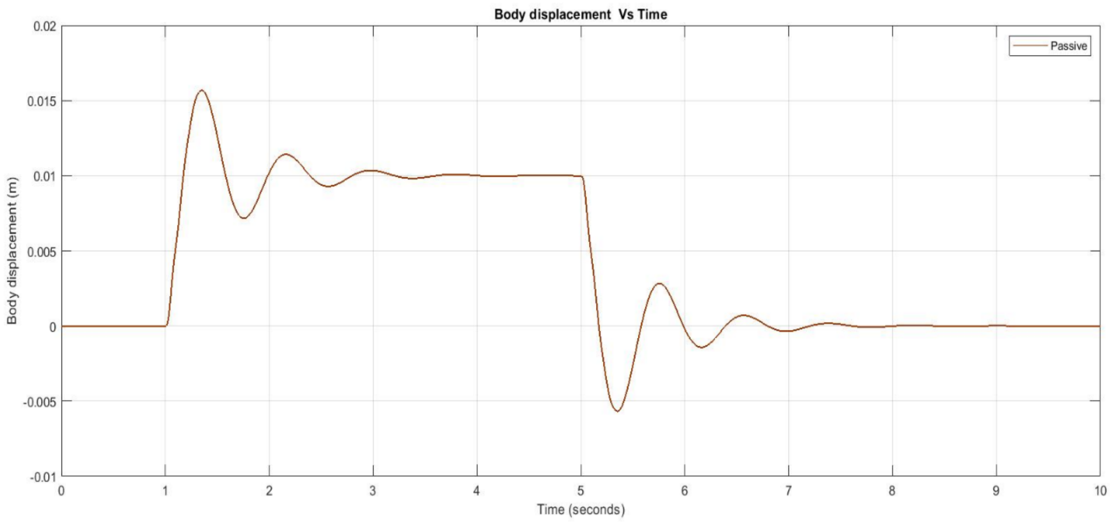


Figure 4.2 Body displacement of passive suspension system for the step input

ii. Passive suspension system responses for sinusoidal

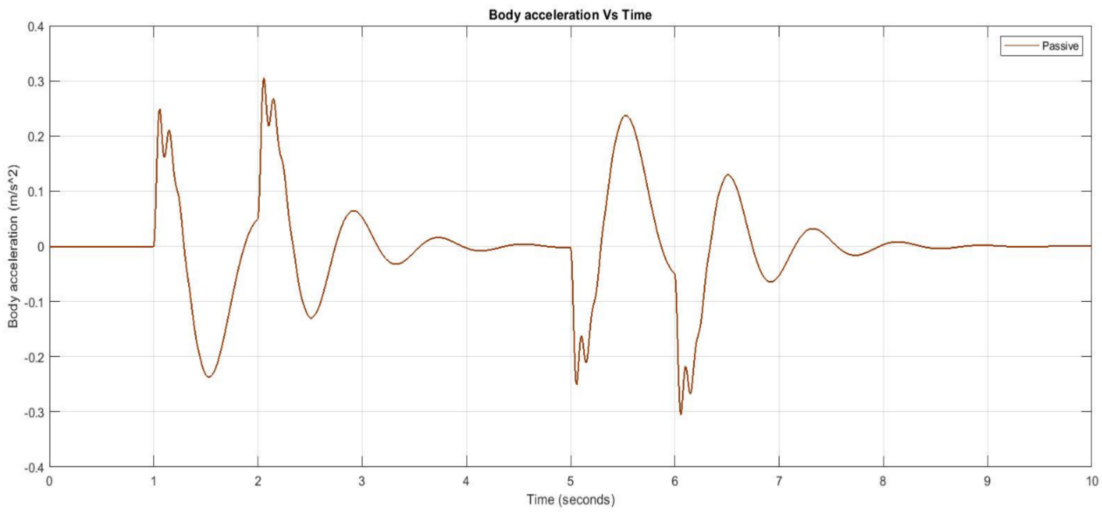


Figure 4.3 Body acceleration of passive suspension system for the sinusoidal input

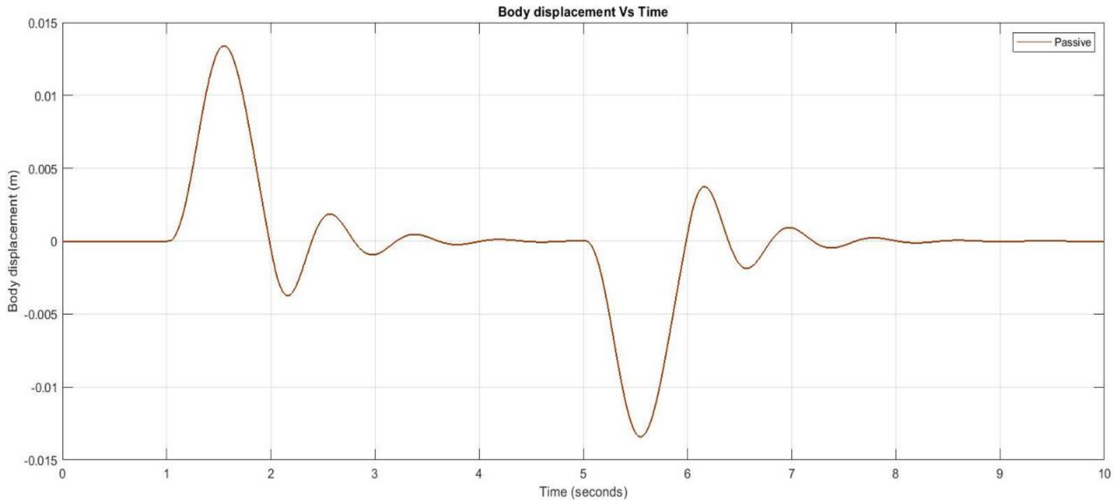


Figure 4.4 Body displacement of passive suspension system for the sinusoidal input

From the figures above we observe that, for step input the maximum body acceleration is 2.27 m/s^2, the maximum body displacement is 0.016 m and the oscillation dissipated in 2.5 seconds.

For sinusoidal input the maximum body acceleration is 0.305 m/s^2, the maximum body displacement is 0.013 m the fluctuation dissipated in 2.5 seconds.

4.2.2 Active suspension system results

The response of the active suspension system in terms of body acceleration and body displacement is shown for the step and sinusoidal inputs aforementioned in the previous chapter in Figures (4.5), (4.6), (4.7) and (4.8) respectively.

i. Active suspension system responses for step input

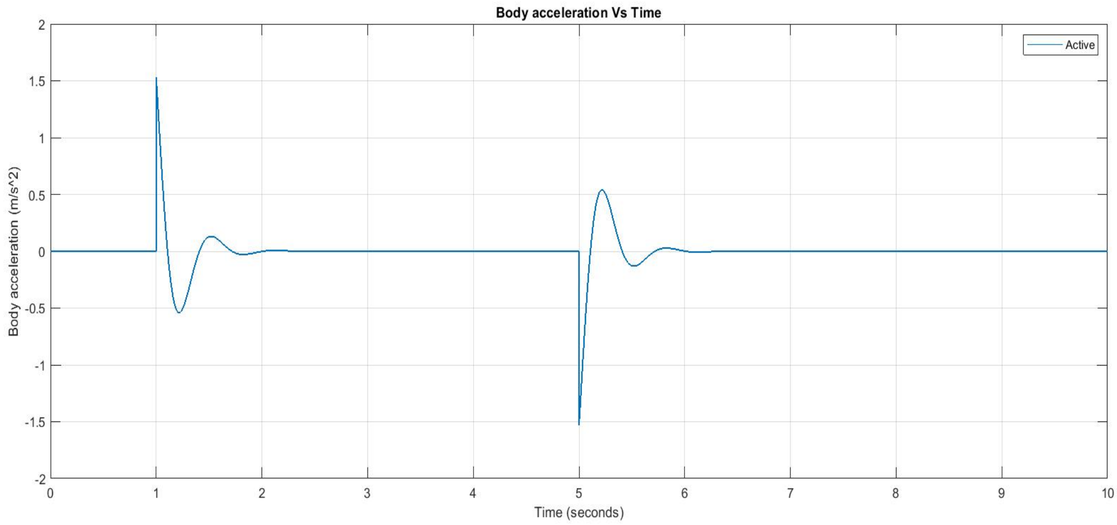


Figure 4.5 Body acceleration of active suspension system for the step input

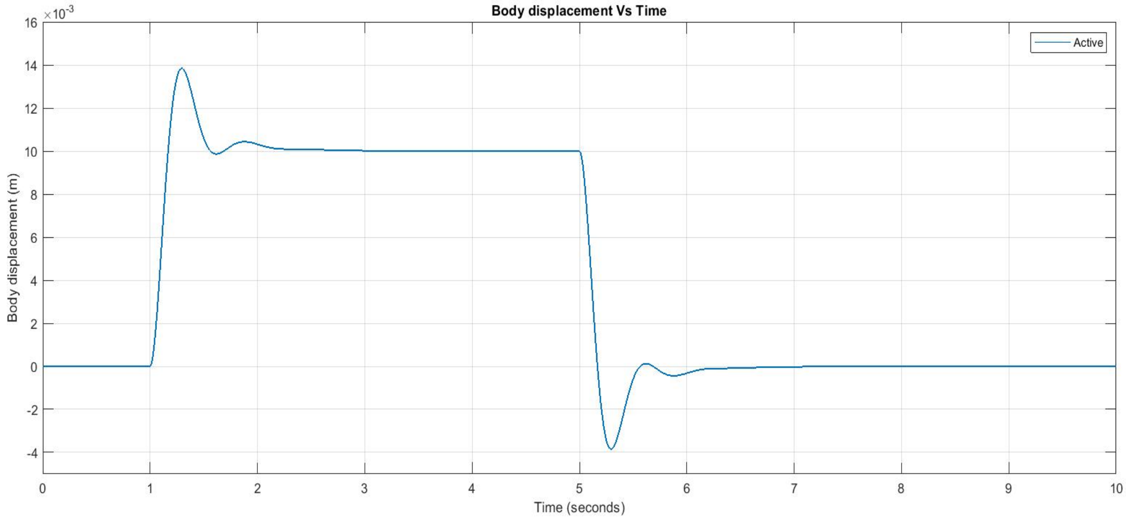


Figure 4.6 Body displacement of active suspension system for the step input

ii. Active suspension system responses for sinusoidal

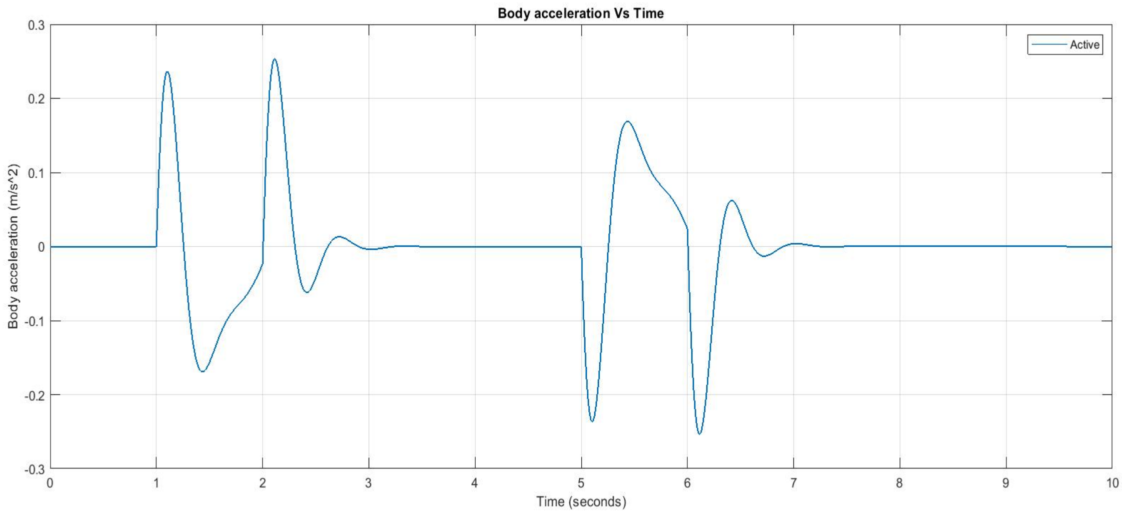


Figure 4.7 Body acceleration of active suspension system for the sinusoidal input

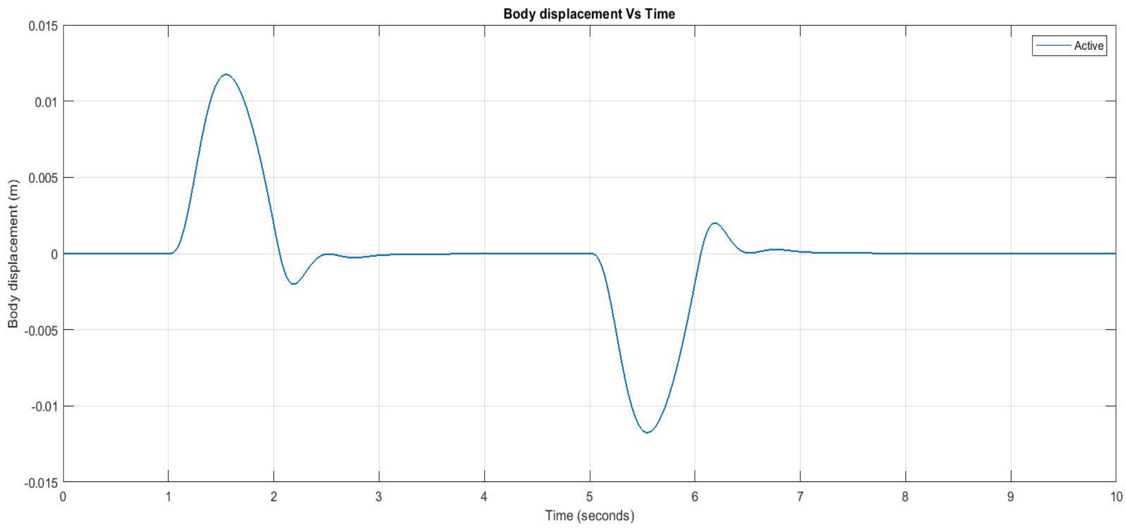


Figure 4.8 Body displacement of active suspension system for the sinusoidal input

From the figures above we observe that, for step input the maximum body acceleration is 1.53 m/s^2, the maximum body displacement is 0.0138 m and the fluctuation dissipated in 1.2 seconds.

For sinusoidal input the maximum body acceleration is 0.253 m/s^2, the maximum body displacement is 0.011 m and the oscillation dissipated in 1 second.

4.3 Suspension system results for SimMechanics tool model

4.3.1 Passive suspension system results

The response of the passive suspension system in terms of body acceleration and body displacement is shown for the step and sinusoidal inputs aforesaid in the previous chapter in Figures (4.9), (4.10), (4.11) and (4.12) represented below respectively.

i. Passive suspension system responses for step input

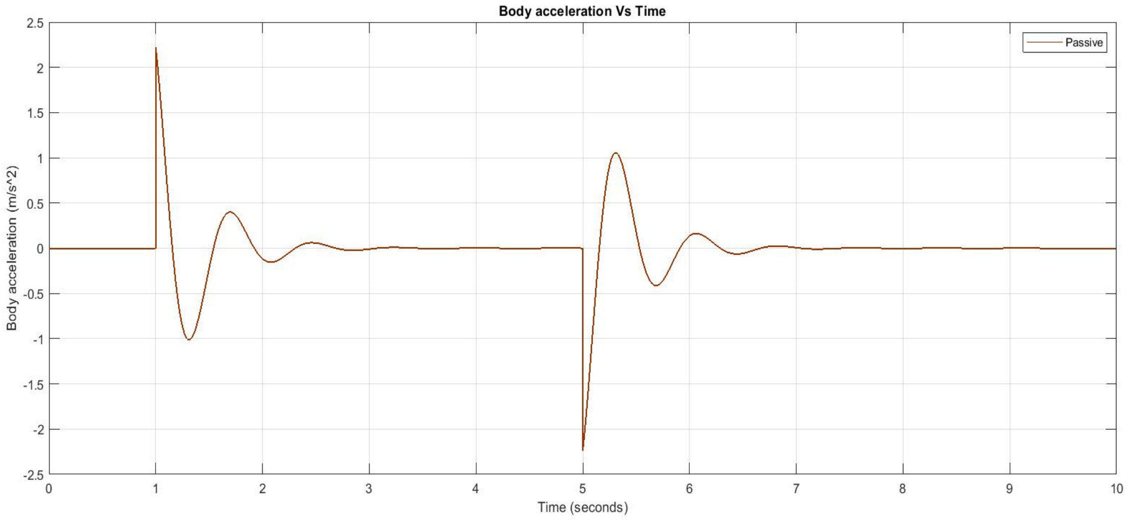


Figure 4.9 Body acceleration of passive suspension system for the step input

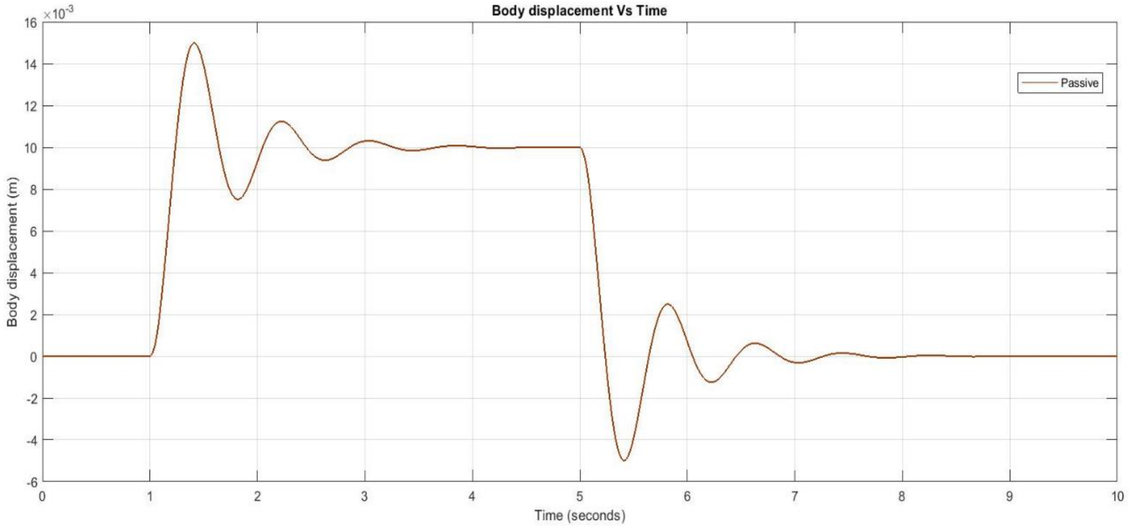


Figure 4.10 Body displacement of passive suspension system for the step input

ii. Passive suspension system responses for sinusoidal input

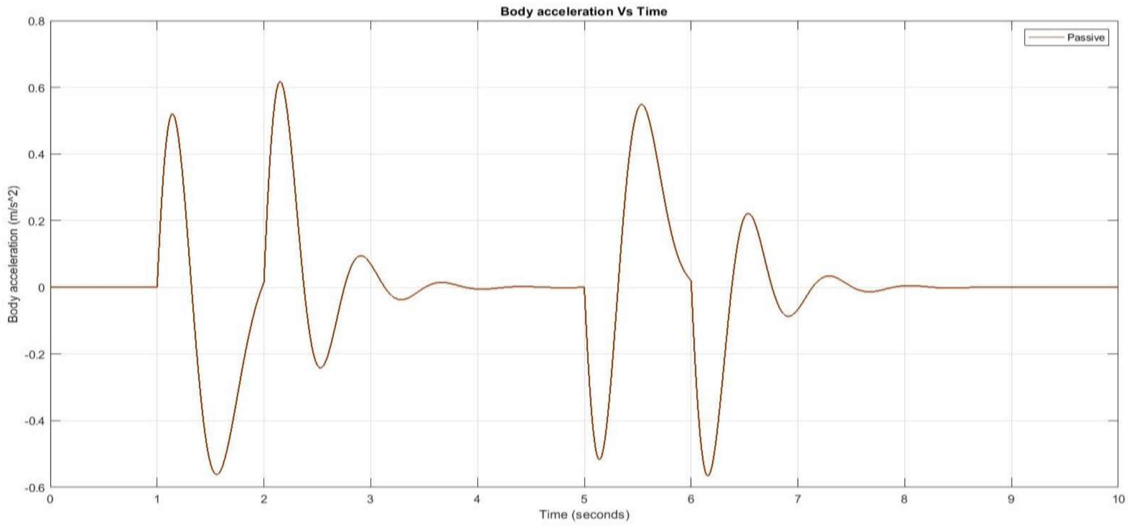


Figure 4.11 Body acceleration of passive suspension system for the sinusoidal input

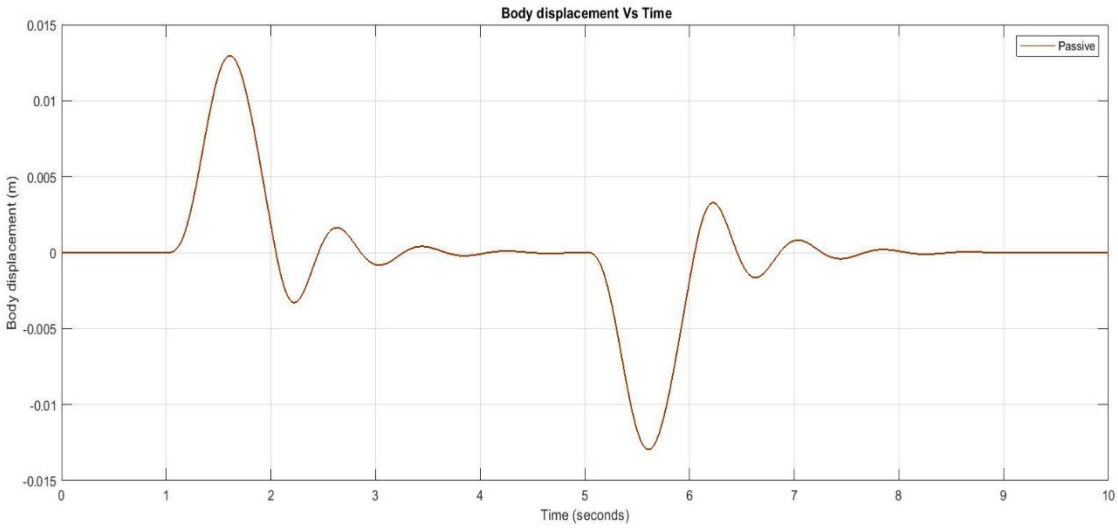


Figure 4.12 Body displacement of passive suspension system for the sinusoidal input

From the figures above it could be seen that, for step input the maximum body acceleration is 2.22 m/s^2, the maximum body displacement is 0.015 m and the fluctuation dissipated in 2 seconds.

For sinusoidal input the maximum body acceleration is 0.618 m/s^2, the maximum body displacement is 0.013 m and the oscillation dissipated in 1.5 seconds.

4.3.2 Active suspension system results

The response of the active suspension system in terms of body acceleration and body displacement is shown for the step and sinusoidal inputs aforementioned in the previous chapter in Figures (4.13), (4.14), (4.15) and (4.16) respectively.

i. Active suspension system responses for step input

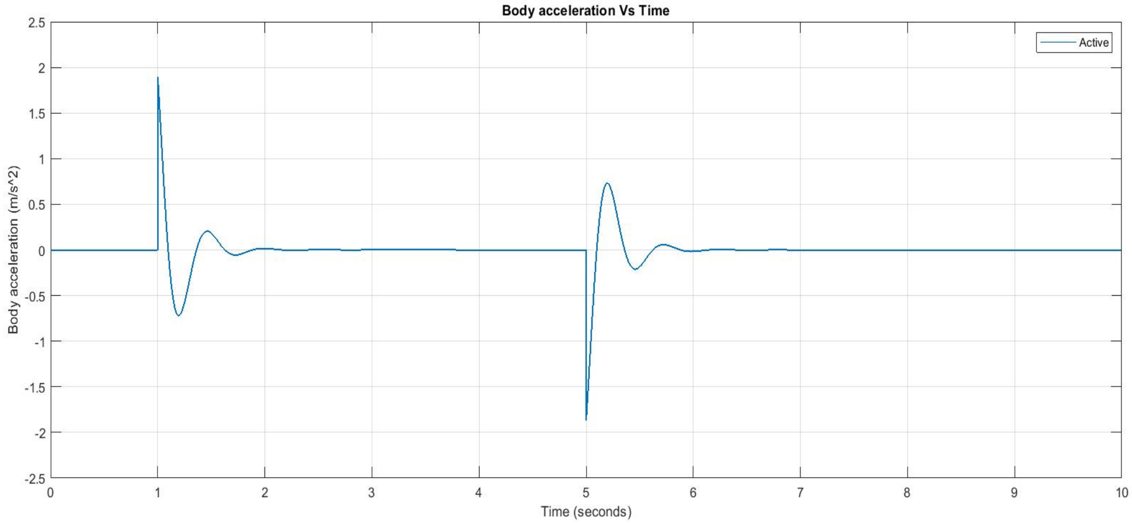


Figure 4.13 Body acceleration of active suspension system for the step input

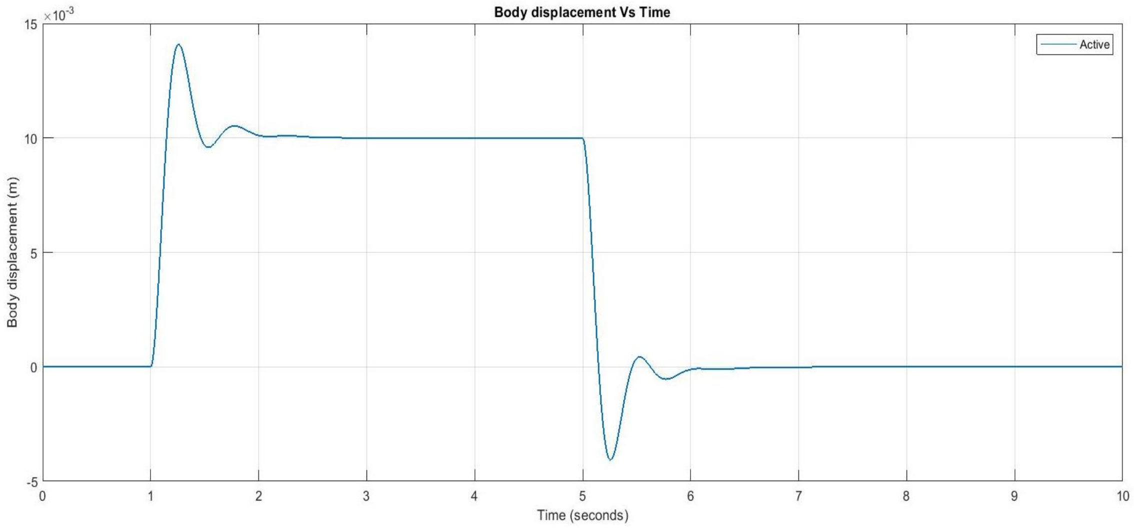


Figure 4.14 Body displacement of active suspension system for the step input

ii. Active suspension system responses for sinusoidal input

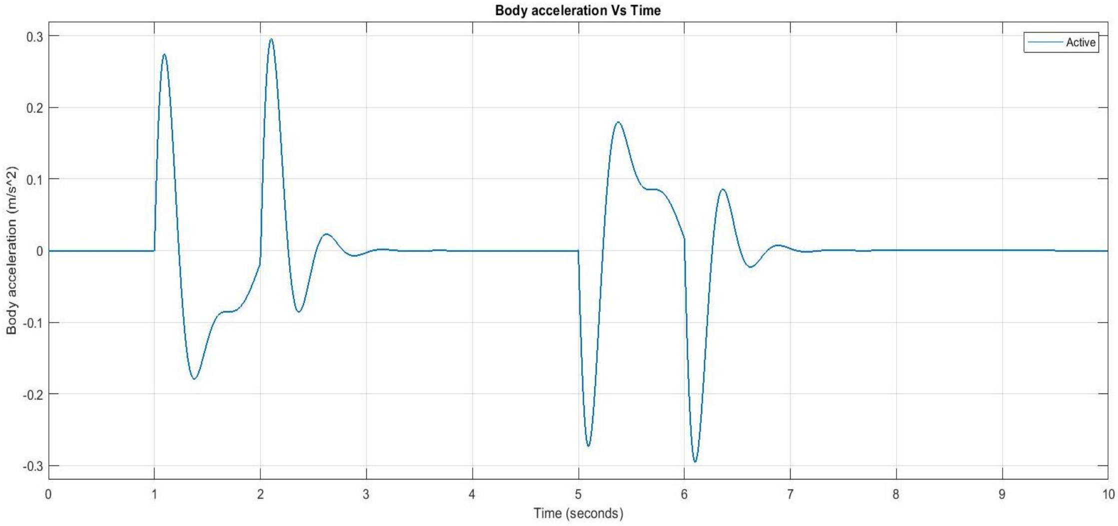


Figure 4.15 Body acceleration of active suspension system for the sinusoidal input

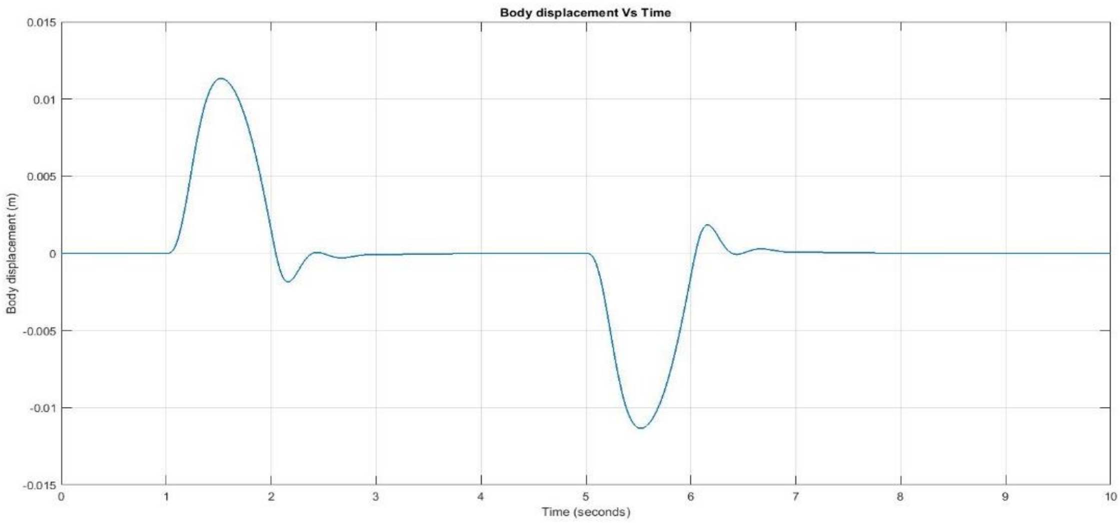


Figure 4.16 Body displacement of active suspension system for the sinusoidal input

From the figures above it could be observed that, for step input the maximum body acceleration is 1.89 m/s^2, the maximum body displacement is 0.014 m and the oscillation dissipated in 1 second.

For sinusoidal input the maximum body acceleration is 0.296 m/s^2, the maximum body displacement is 0.011 m and the oscillation dissipated in 1 second.

4.4 Discussion of passive and active suspension systems performance

4.4.1 System performance in Simulink tool

Bearing in mind the vehicle response under the proposed step and sinusoidal road profiles in both passive and active configurations, it can be noticed that settling time have been reduced by the active suspension system compared to the passive system for all the parameters of body acceleration (passenger comfort) and body displacement suspension deflection (road holding). Those results are represented in figures (4.17), (4.18), (4.19) and (4.20) below.

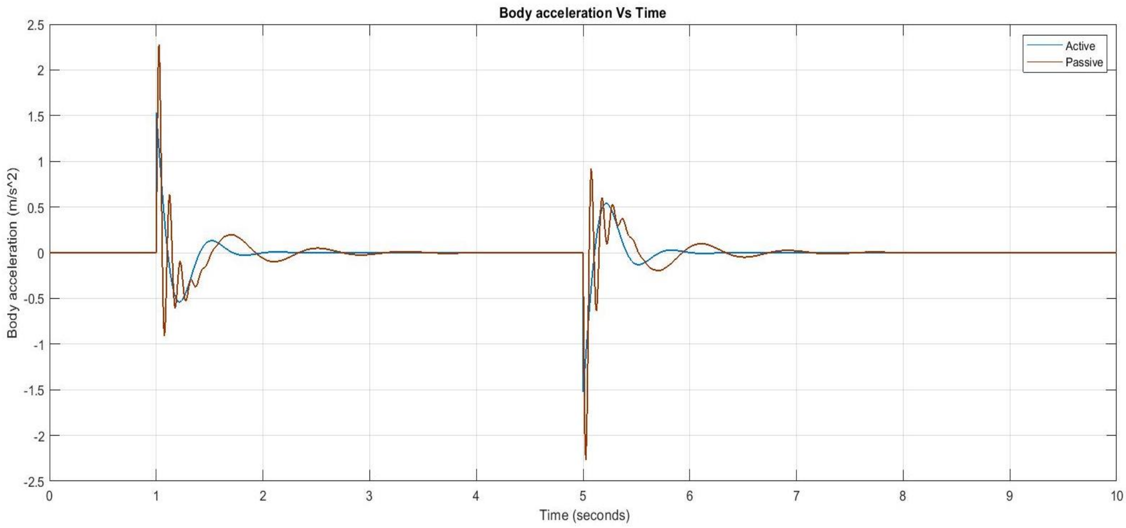


Figure 4.17 Body acceleration of suspension systems for the step input in Simulink

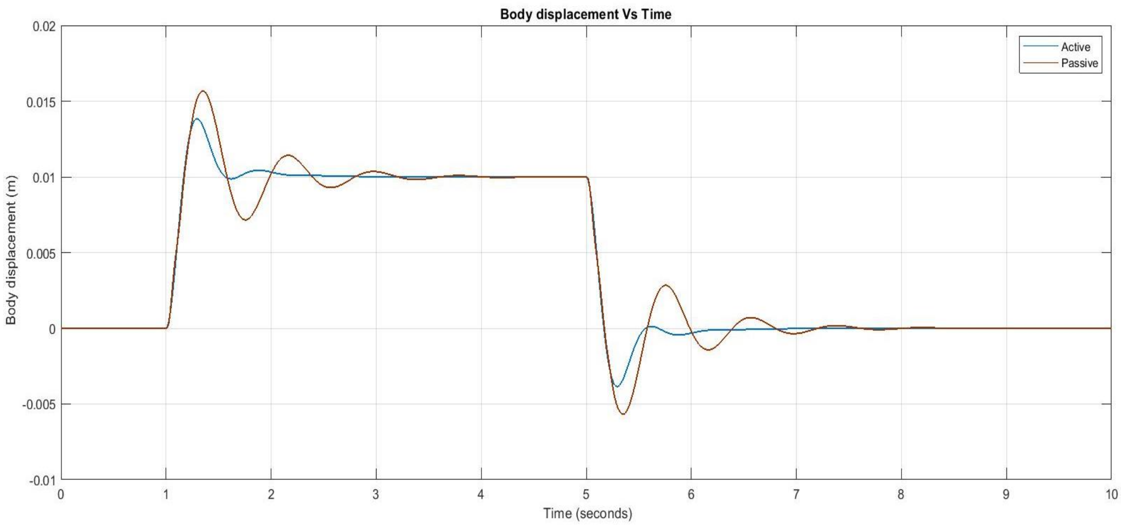


Figure 4.18 Body displacement of suspension systems for the step input in Simulink

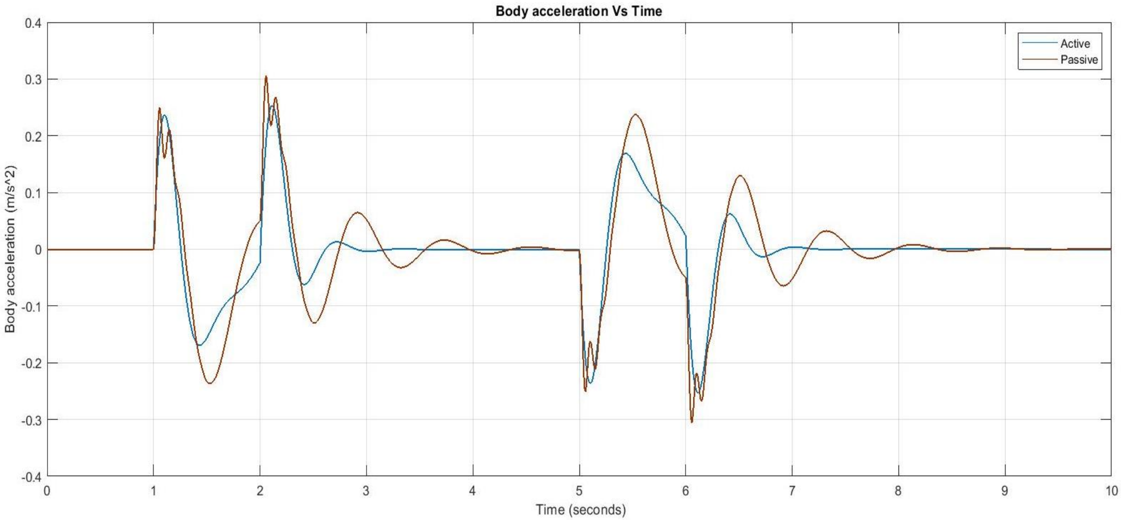


Figure 4.19 Body acceleration of suspension systems for the sinusoidal input in Simulink

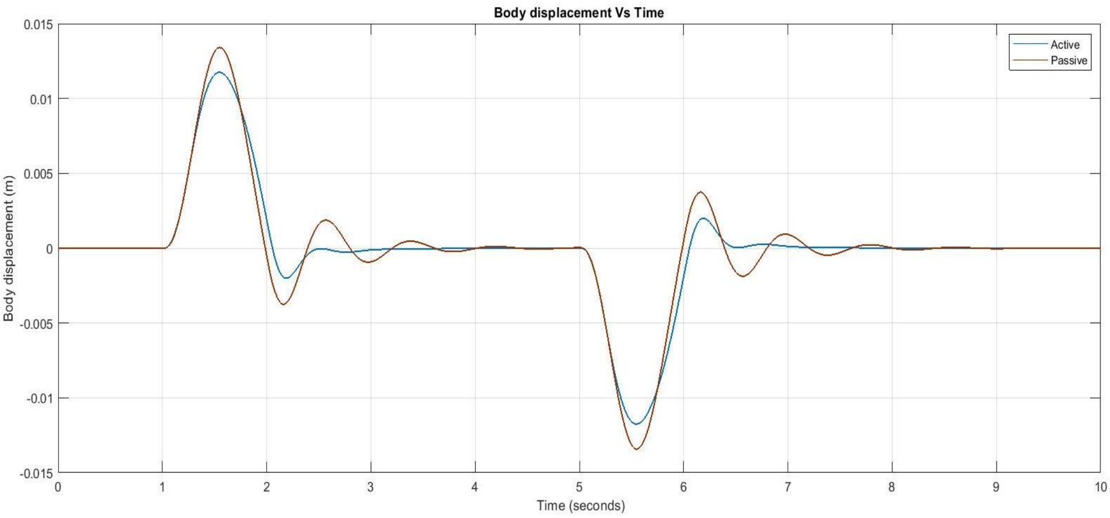


Figure 4.20 Body displacement of suspension systems for the sinusoidal input in Simulink

The results represented in figures (4.17), (4.18), (4.19) and (4.20) above are summarized by means of comparison in table (4.1) and figure (4.21) that shows the difference between passive and active suspension systems performance and how the vehicle response to the road disturbance improved under the application of the PID controlling scheme.

Table 4.1 Evaluation of performance of passive and active suspension systems in Simulink

No.	Parameter	Passive		Active		Reduction %	
		Step	Sin.	Step	Sin.	Step	Sin.
1	Body acceleration (m/s^2)	2.270	0.305	1.530	0.253	32.599	17.049
2	Body displacement (m)	0.016	0.013	0.014	0.011	13.750	15.385
3	Settling time (seconds)	2.500	2.500	1.200	1.000	52.000	60.000

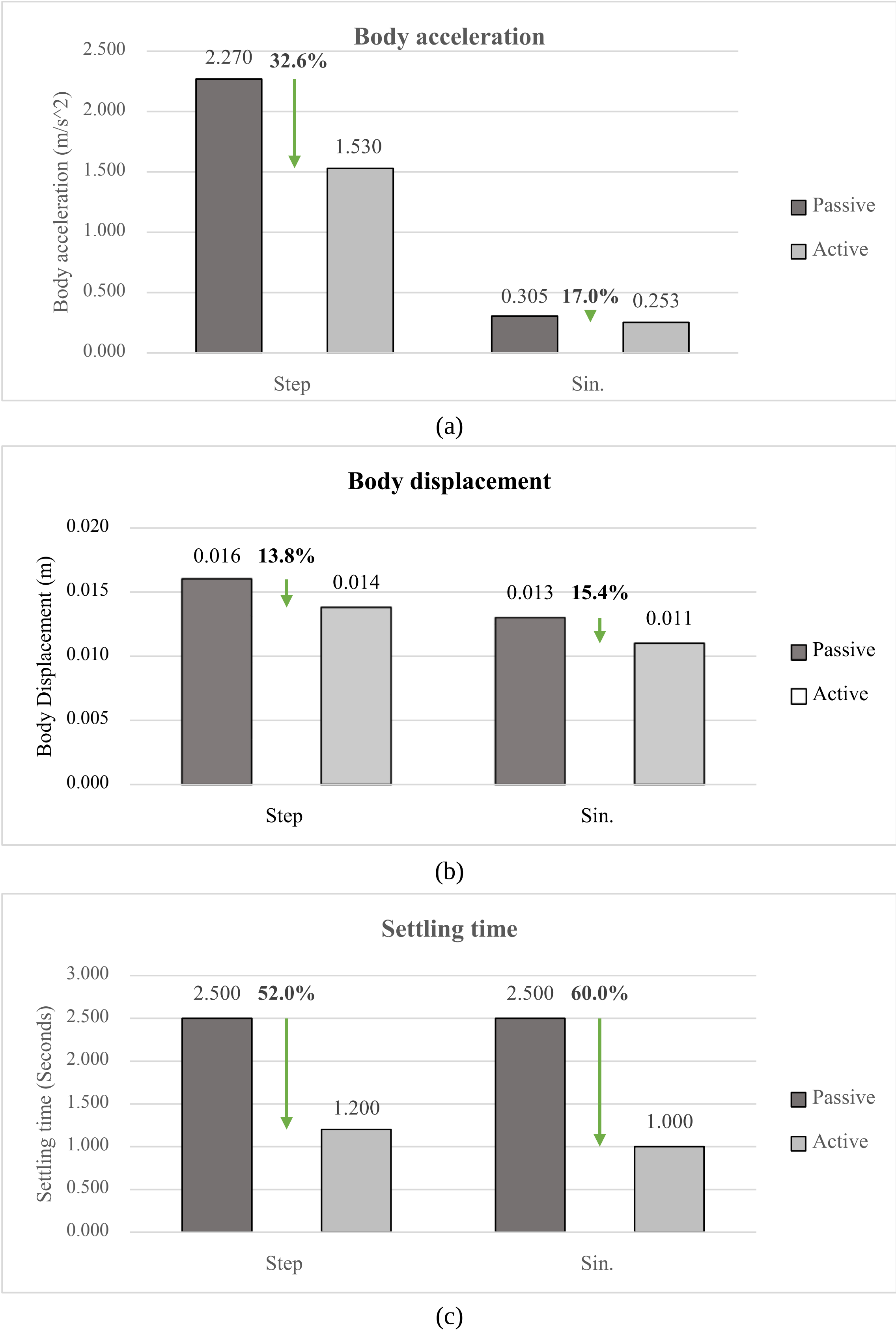


Figure 4.21 Comparison between passive and active suspension systems performance in Simulink (a) Body acceleration, (b) Settling time and (c) Body displacement

4.4.2 System performance under SimMechanics input

Taking into consideration the vehicle response under the proposed sinusoidal road profile in both passive and active arrangements, it can be observed that settling time have been reduced by the active suspension system compared to the passive system for all the parameters of body acceleration (passenger comfort) and body displacement suspension deflection (road holding). Those results are represented in figures (4.22), (4.23), (4.24) and (4.25) below.

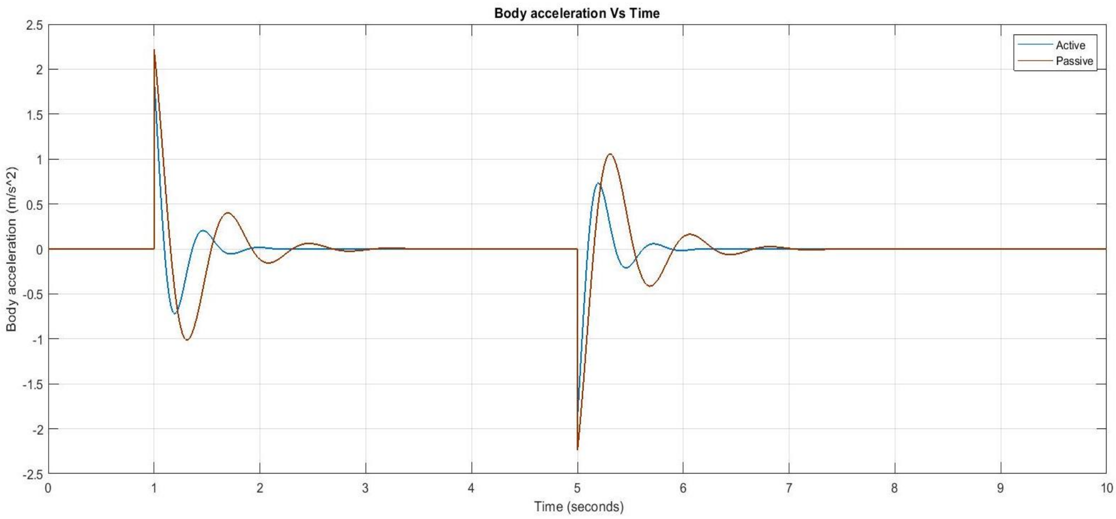


Figure 4.22 Body acceleration of suspension systems for the step input in SimMechanics

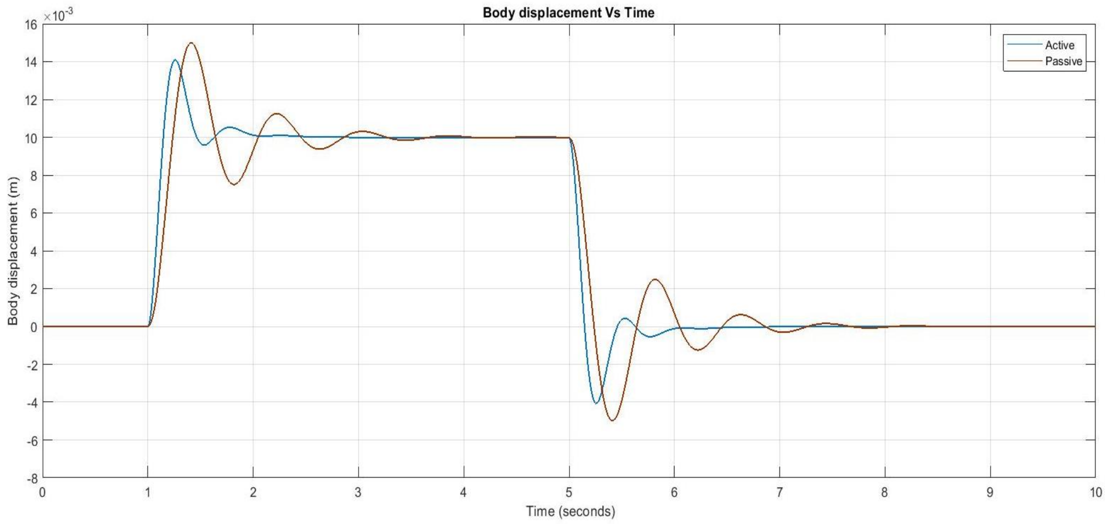


Figure 4.23 Body displacement of suspension systems for the step input in SimMechanics

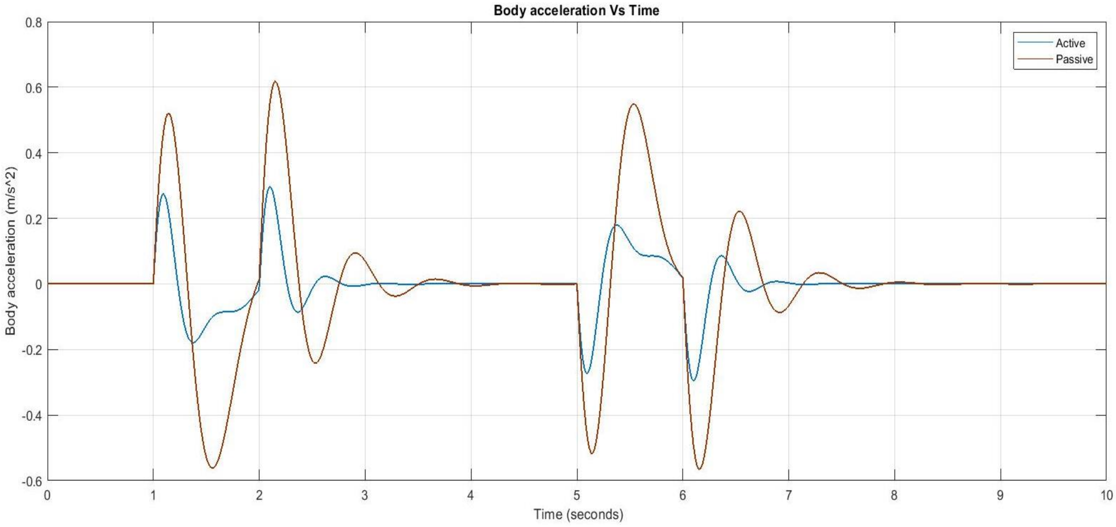


Figure 4.24 Body acceleration of suspension systems for the sinusoidal input in SimMechanics

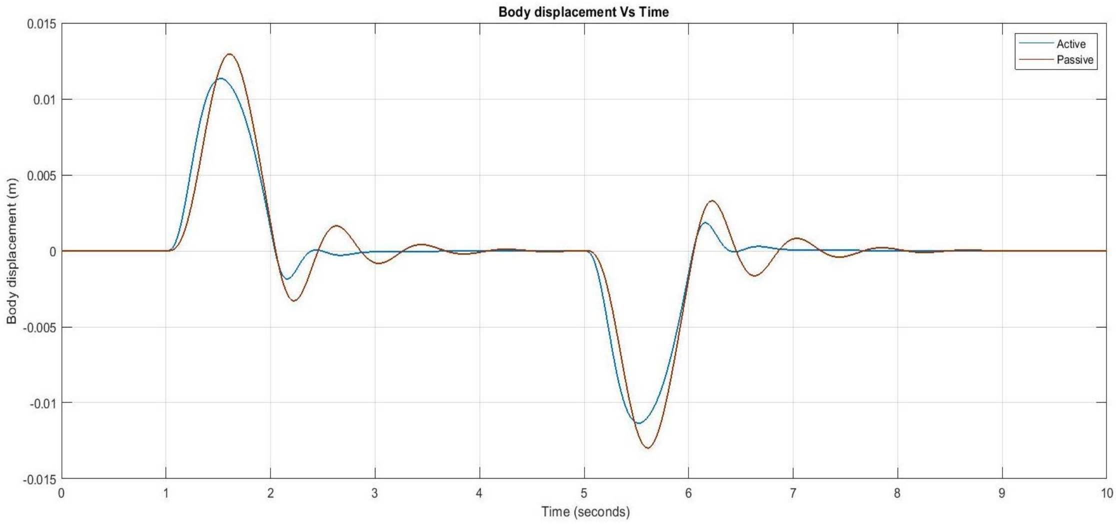
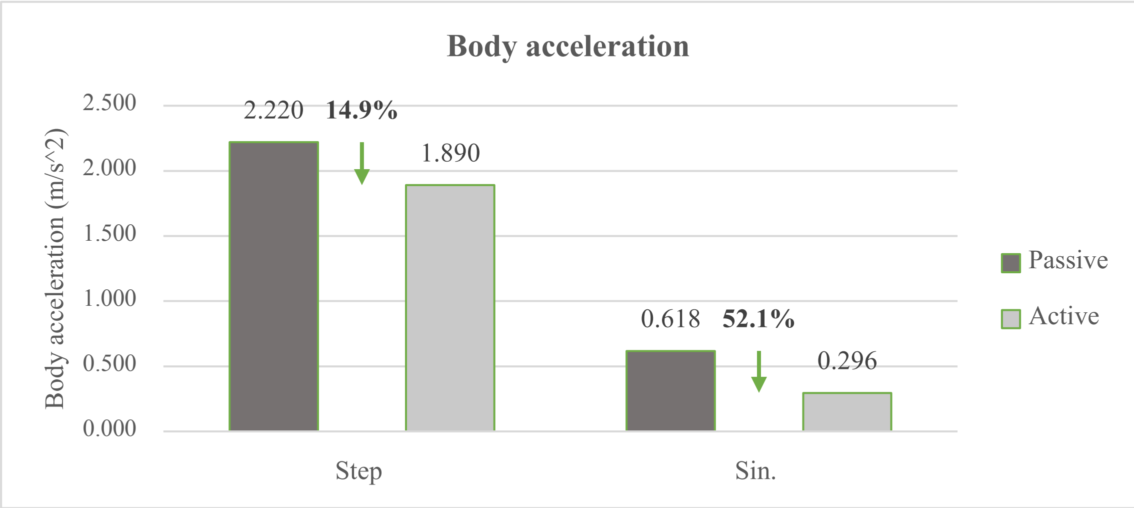


Figure 4.25 Body displacement of suspension systems for the sinusoidal input in SimMechanics

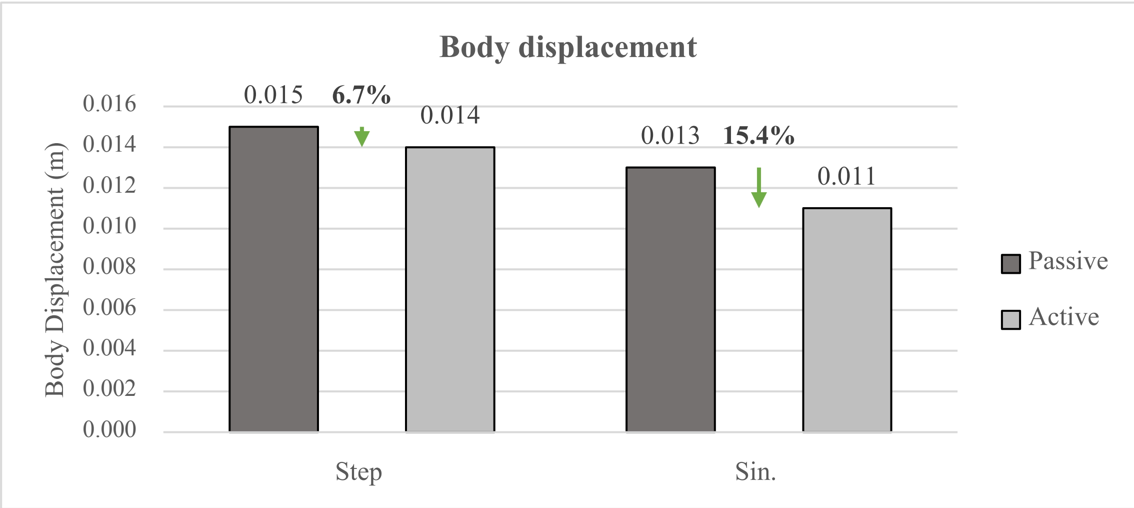
The results represented in figures (4.22- 4.25) above are summarized by means of comparison in table (4.2) and figure (4.26) that shows the distinction between passive and active suspension systems performance and how the vehicle response to the road disturbance improved under the application of the PID controlling scheme.

Table 4.2 Evaluation of performance of passive and active suspension systems in SimMechanics

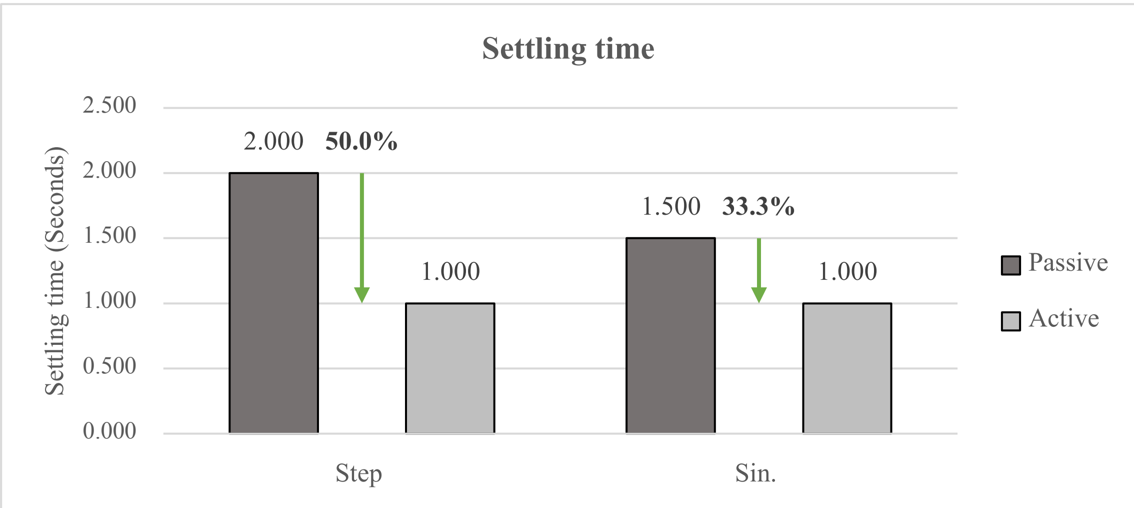
No.	**Parameter**	**Passive**		**Active**		**Reduction %**	
		Step	Sin.	Step	Sin.	Step	Sin.
1	Body acceleration (m/s^2)	2.220	0.618	1.890	0.296	14.865	52.104
2	Body displacement (m)	0.015	0.013	0.014	0.011	6.667	15.385
3	Settling time (s)	2.000	1.500	1.000	1.000	50.000	33.333



(a)



(b)



(c)

Figure 4.26 Comparison between passive and active suspension systems performance in SimMechanics (a) Body acceleration, (b) Settling time and (c) Body displacement

CHAPTER V

CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion

The main function of vehicle suspension system is to minimize the vertical displacement transmitted to the passengers. This research has studied both types of suspension systems passive and active, modelling and analyzing the response of each type using MATLAB program, through Simulink and SimMechanics tools.

From the results represented in the previous chapter the following conclusions can be remarked:

- Simulink model findings:

i. The active suspension system was showing a better performance criterion than passive suspension system represented by a substantial reduction of vehicle settling time as for the step input with 52%, also a reduction in the vertical displacement by 13.75% was observed. similarly, a reduction in the magnitude of body acceleration by 32.6% occurred.

ii. The active suspension system is representing a better performance than the passive suspension system in settling time for the sinusoidal input with a noteworthy reduction by 60%, also the reduction was observed in the vertical displacement by 15.39%. While a reduction in the body acceleration magnitude by 17.05% was observed.

- SimMechanics model verdicts:

i. The active suspension system performs better than the passive suspension system by taking the settling time as a bar for comparison, the step input shows a significant reduction of 50% than the passive system, also a reduction was detected in the vertical displacement by 6.67%, and decreasing in the body acceleration magnitude by 14.87% was noticed.

ii. The active suspension system has improved the passive suspension system in settling time for the sinusoidal input by reducing it with about 33.33%, also a reduction occurred in the vertical displacement by 15.39%. Moreover, a major decrease of the body acceleration magnitude by 52.1% was perceived.

5.2 Recommendations

i. It is suggested to use the original versions of MATLAB software with the full licenses kit in the simulation processes, to prevent the cracked programs issues, that include defective or odd results.

ii. For further simulation attempts to build up dynamic systems, it is recommended to start the process of simulation using Simulink tool due to the simplicity of building the system, and the reliability of the results obtained by it.

iii. SimMechanics is a very useful and realistic method that can be utilized in order to understand the behaviors and responses of mechanical systems through exploring its dynamics and motions, therefore, it is recommended as a simulation tool in further projects.

iv. This project adopts the modeling for a light weight passenger car, and the values of spring stiffness and damping coefficient were selected based on that assumption, then caution is required while modeling other types of automobiles (heavy trucks and vehicles) based on this thesis hypothesis.

v. Conduct research with other type of control systems instead of the suggested (PID) controller, the type of controlling system adopted will have a direct impact on the active suspension system performance, and it can affect results in different manners.

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