MODELING AND PERFORMANCE ANALYSIS OF AN ACTIVE SUSPENSION SYSTEM

THESIS
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DEDICATED

TO

MY PARENTS

DECLARATION OF ORIGINALITY AND COMPLIANCE OF ACADEMIC ETHICS

I hereby declare that this thesis contains literature survey and original research work done by the undersigned candidate as part of his Master of Engineering studies.

All information in this document have been obtained and presented in accordance with academic rules and ethical conduct.

I also declare that as required by these rules and conduct I have fully cited and referred all materials and results that are not original to this work.

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CERTIFICATE OF RECOMMENDATION

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*The foregoing thesis, entitled as "*Modeling and performance analysis of an active suspension system" is hereby approved by the committee of final examination for evaluation of the thesis as a creditable study of an engineering subject carried out and presented by **Nilarghya Debnath** (Examination Roll No.- M4AUT22021B, Class Roll No.- 002011204021 Registration No.- 154354 of 2020-2021) in a manner satisfactory to warrant its acceptance as a prerequisite to the degree of **Master Of Engineering in Automobile Engineering(Department of Mechanical Engineering).** It is understood that by this approval, the undersigned do not necessarily endorse or approve any statement made, opinion expressed or conclusion drawn therein, but approve the thesis only for the purpose for which it is submitted.

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CHAPTER – 1

INTRODUCTION

1.1 GENERAL

The suspension system is the device that is physically positioned between the vehicle's body and wheels of a car. Performance of the suspension systems has improved dramatically over time thanks to the enhanced vehicle capabilities. A variety of performance parameters must be considered in order to design a suitable suspension system for a car. Controlling force distribution, suspension movement, and body movement are all aspects of these characteristics. The vehicle will veer forward when the brake is applied because of the way that vehicles move longitudinally. This is because inertia will cause the centre of gravity of the car to change, shifting weight from the back tyres to the front tyres. Similar to this, when the throttle is input, the vehicle will squat to the back. Weight shifting from the front to the back is the cause of this. Vehicle pitching refers to both the dive and squat undesired vehicle motions. This motion will make the car unstable, impair its ability to handle, send it careening out of control, and it could even result in an accident.The purposes of a vehicle's suspension, within the range of suspension travel, are (a) to support the weight of the vehicle's body and give rides that are comfortable and safe on a variety of road surfaces, (b) to shield the vehicle body from outside interference resulting from internal instabilities and uneven road surfaces caused by a vehicle's acceleration, deceleration, or cornering, in order to have a comfortable ride, (c) be able to adapt to changes in load, which whether they were brought on by variations in the number of travelers and baggage, or from internal problems; and (d) to maintain a firm contact between the road and the tires to ensure that the wheels follow the road profile in order to achieve good handling performance and safe drive of the car. Key design constraints are presented by maximum allowable relative displacement between the vehicle body and various suspension components. Additional constraints are imposed by the overall system robustness, reliability and cost requirements. Different suspensions satisfy the above requirements to differing degrees. Although significant improvements can result from a designer's ingenuity, on the average, suspension performance mainly depends on the type of class of suspension used. Today, the majority of manufacturers use a passive suspension system that combines hydraulic or pneumatic shock absorbers with some kind of spring. Despite the broad variety of designs currently available, passive suspensions will always represent a tradeoff between passenger ride comfort, handling, and suspension stroke over the operational range because they can only store and release energy in a pre-determined manner. It is clear that for good and safe ride, body handling, cargo handling, comfort of vehicle mainly depend on the suspension system. Therefore research and development on the suspension system for vehicle should be focused with delivering high-caliber results to standards for both safe drive and riding quality. Active suspensions have been a topic of discussion for more than a decade. The advantage of active suspensions is that they offer dynamic compensation as compared to passive suspensions. The objectives of adopting an active suspension system are to increase passenger comfort, maneuverability, and stability. The passive suspension system is enhanced by pneumatic, hydraulic, or electro-magnetic actuators that provide additional external force in the active suspension system. It is commonly assumed that the hydraulic actuator is an ideal force generator and able to carry out the commanded force accurately. The capacity of active suspension to introduce energy into the system sets it apart from traditional passive suspension. An actuator is often attached in parallel with the spring and shock absorber in an active suspension system. The fundamental benefit of using an active suspension is the associated adaptation capability, which enables the suspension parameters to be changed while driving to correspond with the profile of the road being travelled. Depending on the varied road conditions, the active suspension system should be able to deliver diverse behavioural characteristics. The fundamental concept of active suspension control is to apply a desired force between the automobile body and wheel axle using an active element (the actuator).This desired force is computed by the car’s control unit to achieve certain performance objectives under external disturbances, such as passenger’s comfort under road imperfections [1].

While detrimental vibrations induced by road irregularities and on-board excitation sources impinge upon the vehicle, active suspensions are utilizing to give the vehicle good handling qualities and to increase ride comfort. Passenger comfort, suspension deflection, tire load fluctuation, and energy consumption are some of the parameters used to quantitatively evaluate the performance of suspension systems (see [2,3]. It's crucial to minimize the effects of road disturbance on the heave acceleration in order to increase passenger comfort. The amount of suspension deflection that can be used to slow down the acceleration of the vehicle body is strictly limited by structural elements of the vehicle. Therefore, it's crucial to keep the disturbance's impact on the suspension deflection minimal. The effects of the disturbance on tire deflection should also be kept to a minimum in order to minimize the dynamic tire load deflection. To prevent using huge control forces, the control force limitation is built into the design process. The outcome also shows that, usually speaking, restrictions placed on suspension deflection also limit tire deflection and vice versa (see [3, 4]). The fact that the model’s component attributes are unclear presents another challenge to the control design. Neglected dynamics, high order modes, limited component knowledge, or altered component utilization as a result of alterations in operating conditions are the causes of the uncertainty. The suspension problem is analyzed in fundamental papers such as [5–7]. The majority of the time, a precise system model is required before the controller can be created. Most active suspensions employ state space techniques which require complete measurements of the states for optimal performance. This necessitates the use of costly transducers for the measurement of the states. The usual option that is available to circumvent the use of such costly transducers is to employ state observers, but this calls for the use of a microcontroller with superior computing capability. The other alternative is to use a suboptimal constrained feedback controller [8,9]. Normally the performance of these constrained feedback controllers does not match that of the full state feedback controllers. But with some ingenuity and proper choice of the feedback structure it can be shown that the performance of these controllers can be enhanced so as to match a full state feedback controller.

The suspension should, in a perfect world, shield the body from road disturbances and inertial disturbances brought on by braking and cornering maneuvers. Additionally, the suspension should be able to lessen excessive vertical wheel travel and force distribution from the tires to the road. In a passive suspension system, these requirements are frequently incompatible and unmet under a variety of circumstances [10]. The low-frequency response could be improved by making the suspension damper stiffer, but at higher frequencies, the response is subpar. Reducing the suspension damper's stiffness can enhance response at high frequencies, but at the expense of worse low-frequency behavior. Intelligent control techniques are used to address this contradicting behavior and allow the suspension system to adjust to irregularities of varied driving situations.

1.2 LITERATURE REVIEW

To avoid the unwanted pitching motion, a considerable amount of works has been carried out to solve the problem. Through the combination of mechanical, electrical and hydraulic components, a wide range of controllable suspension systems have been developed varying in cost, sophistication and effectiveness. Passive, semi-active, and active damping suspension systems are the three fundamental types of suspension system used for automobiles. The least complicated system, passive suspension offers many benefits. The limitation of passive suspension, however, is its inability to eliminate undesirable oscillations brought on by uneven road conditions [11]. Passive performance is constrained by typical characteristics because it has a fixed spring and damper. Similar to how certain car specifications have a defined standard, so does its efficiency. Thus, semi-active suspension systems with variable dampers are implemented in order to attain superior performance results. Semi-active damping systems are simpler and more practical than passive damping systems and use significantly less energy. As a result, it can significantly improve riding performance. Karnopp introduced the skyhook control methodology, which is a popular control strategy for semi-active suspension systems because it can lower a vehicle's "resonant peak" and improve ride quality [12]. Due to their minimal power consumption and protection requirements, magneto-rheological dampers (MR) are one of the most explored approaches in semi-active suspension systems [13], but because they only dissolve energy and act as passive suspension systems when the power fails, their performance is likewise constrained. Therefore, hydraulic, air, or electric power actuator-based active suspension systems are introduced. The power source that produces the desired force is part of the active suspension system. Additionally, it doesn't use any energy to generate the forces that prevent suspension deflection [14]. Active suspension has the added advantage of providing dynamic correction, in contrast to passive and semi-active suspension systems [15].

The first serious effort to apply an active suspension to automobiles was at Citroe¨n Automotive, which developed a mechanical type of active system based on a hydro-pneumatic suspension [30]. Fully active suspensions use electronic monitoring of vehicle conditions, coupled with the ability to assess the system behaviour in real-time to directly control the motion of the car. Lotus Cars developed a number of prototypes and introduced them to Formula One (F1) racing cars, where they have been found to be fairly effective. Nissan introduced a low bandwidth active suspension around 1990. Pneumatic suspension has also been developed for the Porsche Cayman through the Porsche Active Suspension Management system [31].

Hydraulic, pneumatic, or electromagnetic actuators are frequently positioned in tandem with a spring and a damper to apply active control forces [16]. The main benefit of pneumatic actuators is their simplicity. The efficiency of hydraulic actuators in high-power applications is widely utilized. Accuracy in electromagnetic systems is consistently substantially higher than in hydraulic and pneumatic systems [17]. Reference [18] proposes a brand-new cascade control method for the active suspension of forest machinery combining parallel mechanical construction and pneumatic actuators. However, the suspension force cannot function as the suspension system itself; instead, it is applied in [19] utilizing the hydraulic actuator dynamics. In [20], the electromagnetic actuator takes the place of the damper and the hydraulic actuator, creating an oil-free suspension with the spring. In [21], a redesigned hybrid electromagnetic actuator-based skyhook for active suspension is created. Reference[22] presents an active vehicle suspension control strategy using electromagnetic and hydraulic actuators.

It is widely acknowledged that an active damping suspension is a highly effective means of improving suspension performance [23]. Simulations of these controllers were frequently done without considering actuator dynamics, or with highly simplified hydraulic actuator dynamics. In real implementation, actuator dynamics can be quite complicated, and the interaction between the actuator and the vehicle suspension cannot be ignored. It is also difficult to produce the actuator force close to the target force without implementing force tracking controller. This is due to the fact that hydraulic actuator exhibits non-linear behavior resulted from spool-valve dynamics, residual structural damping, and the unwanted effects of back-pressure due to the interaction between the hydraulic actuator and vehicle suspension system. A few previous works on the force tracking controller of hydraulic actuator can be found on [26-29].On the basis of control techniques, various active control solutions are provided to enhance ride performance. The performance will be better the more efficient the control approach is. Therefore, the choice of control strategy for suspension has a significant impact on the effectiveness of the system. PID controllers have held the centre stage for a long time in the field of control engineering. The advantages of these controllers are that they are cheap and easy to operate. PID controllers not only have the ability to improve both the transient as well as the steady state responses, but also can take care of the constant disturbing forces [24] acting on the system. These factors are very essential for a good active suspension system. This influences the choice of a PID structured controller for active suspensions. The normal passive suspensions are basically PD controllers; including an integral term to these controllers should give all the advantages of the PID controllers. This addition increases the cost of the suspension only by a marginal amount. A PID controller for active suspensions whose parameters were optimised stochastically was proposed earlier [25], but a detailed study of the performance of this controller was not taken up at that stage. It was in principle shown that the performance of this controller is comparable to that of an optimal feedback controller at that stage. The present work focuses on a detailed study of the performance of this controller, so as to prove its capability.

1.3 OBJECTIVE OF THE THESIS

MATLAB-SIMULINK software is chosen as a computer simulation tool to simulate the vehicle dynamics behavior and evaluate the performance of the control structure. In order to verify the effectiveness of the proposed controller, passive system and active system with PID controller are selected as benchmark.

1.4 ORGANISATION OF THE THESIS

The thesis is divided into individual chapters to understand, demonstrate and implement the various aspects apropos the scope of work as proposed. Necessary efforts have been put across to proceed in a sequential manner of operation, in developing the model to be proposed.

In chapter-2 extensive idea about the vehicle dynamics and the basic idea about modeling of vehicle systems have been described. In chapter-3 basics of vehicle suspension system have been described followed by basic suspension system elements. And then suspension dynamics is being discussed via lumped parameter approach. A full car model and a quarter car model with mathematical expressions for simulation purpose is being described also. In chapter-4 hydraulic actuator with necessary dynamic equations is being introduced. The chapter-5 is dedicated to control system design and PID controller theory. In chapter-6 various SIMULINK model of quarter car with different input road profile and control strategy is described. Chapter-6 is dedicated for the result and discussions of the SIMULINK models. Finally in chapter-7 few future aspects and a conclusive statement is being delivered.

CHAPTER –2

VEHICLE DYNAMICS

2.1 INTRODUCTION TO VEHICLE DYNAMICS

It has often been said that the primary forces by which a high-speed motor vehicle is controlled are developed in four patches – each the size of a man's hand – where the tires contact the road. This is indeed the case. Knowledge of the forces and moments generated by pneumatic (rubber) tires at the ground is essential to the understanding of the vehicle dynamics.

Insofar as a vehicle's performance—the actions taken during acceleration, braking, cornering, and riding—is a reaction to forces applied, vehicle dynamics includes research into how and why the forces are generated. The tire's friction with the road generates the primary forces that regulate a vehicle's performance. Thus, it is essential to have a thorough grasp of how tyres behave, which is demonstrated by the forces and moments produced over the wide range of situations in which they work.

Although turning, cornering, and directional reaction are usually used interchangeably with the word "handling," there are small distinctions between these concepts. When a vehicle changes direction and encounters lateral acceleration, it has certain objective properties that are referred to as cornering, turning, and directional reaction. For instance, cornering ability can be assessed by the amount of lateral acceleration that can be sustained in a steady state, and directional reaction can be assessed by the time it takes for lateral acceleration to emerge following a steering input. On the other hand, handling improves aspects of a vehicle that are advantageous to the driver by making driving simpler or by aiding in maintaining control. The performance of the driver/vehicle combination as a system, as well as the vehicle's stated capabilities, are all included in handling.

There are two ways to comprehend the dynamics of a vehicle: analytically and empirically. The empirical understanding comes from trial and error method, which is how one learns which factors, in what ways, and under what circumstances affect vehicle performance. However, the empirical approach frequently fails. Without a mechanistic understanding of how modifications to a vehicle's design or physical characteristics affect performance, extrapolating prior results to new circumstances may involve unknown aspects that could provide a new outcome that defies the accepted conventions. Engineers prefer the analytical approach for this reason as well as the fact that they are meticulous by nature. The goal of the analytical approach is to build an analytical model by attempting to characterize the relevant mechanics using the existing physics rules. These models can be represented, in the simpler circumstances, by algebraic or differential equations that link forces or motions of interest to control inputs and tire or vehicle characteristics. The role played by each vehicle property in the phenomenon of interest can then be evaluated using these equations. Thus, the presence of the model provides a way to identify the key components, how they work, and in what situations. The model also has the ability to forecast, making it feasible to identify the modifications needed to achieve a particular performance goal. However, it is also true that analytical techniques are not infallible since they typically only provide approximations of reality. As many people have discovered, the assumptions that must be made in order to produce manageable models can frequently be deadly to the application of the analysis, and engineers have occasionally been proven to be in error. In order to avoid these mistakes, it is crucial for engineers to understand the presumptions that have been made while modeling any part of dynamics. For road vehicles the topic of vehicle dynamics can be broken into following sections:

Tires - The development of mathematical models to predict tire forces, often based on empirical or semi-empirical techniques, i.e., not derived from first principles, but rather measurements of tire properties and data from experiments.

Longitudinal dynamics - Acceleration and braking capability, coupling between engine and vehicle through the drive train, and weight transfer effects on tire traction.

Ride quality – A study of vertical dynamics and the ability of the vehicle's suspension to accommodate varying terrain while maintaining passenger comfort. In other words the vehicle goes straight on a bumpy road, with constant forward speed.

Handling - The behavior of the vehicle with respect to its motion in the plane of the road, particularly its directional stability and its response to steering inputs. In other words the vehicle makes turns on a flat road, usually with an almost constant forward speed.

***Performance* -** The vehicle goes straight on a flat road, possibly braking or accelerating (non constant forward speed).

Suspension kinematics - The geometry of the suspension and steering components, and its effects on vehicle motion; sometimes expanded to include *elastokinematics*, the small changes in the geometry due to deflection under load.

For the purpose of this thesis paper work, model based analytical method will be followed.

2.2 MODELING OF THE VEHICLE SYSTEM

The first stage of modern vehicle development is now entirely virtual, meaning the entire vehicle only exists on different computers as a CAD model, a model for judging the stiffness of the body, a model for the driving dynamics of the vehicle, a model for crash safety, a model for vibration in the powertrain, and, depending on the OEM strategy in the virtual development process, there may be other models as well. This implies that models can be used to evaluate the ride quality, one facet of comfort, during the early stages of development.

In the past, the mathematical constraints on problem-solving were the cause of many analytical methodologies' flaws. Analysis was only deemed successful before the invention of computers if the "problem" could be reduced to a closed form solution. That is, only if it was possible to alter the mathematical expression into a shape that permitted the analyst to extrapolate relationships between the relevant variables. This significantly reduced the analytical approach's ability to solve problems related to vehicle dynamics. Comprehensive modeling of automobiles was almost impossible due to the sheer amount of parts, systems, subsystems, and nonlinearities present, and the only useful models that could be produced were overly simplified representations of specific mechanical systems. Even while they were helpful, the models' simplicity frequently resulted in flaws that made the engineering approach to vehicle development difficult.

A significant drawback of the analytical method has been fixed, thanks to the processing capability of modern desktop and mainframe computers. In order to simulate and evaluate a vehicle's behavior before it is physically represented in hardware; it is now possible to put together models (equations) for the behavior of specific vehicle components that may be merged into thorough models of the entire vehicle. These models are able to calculate performance that was previously unsolvable. When an engineer is unsure of the significance of a certain property, that property might be incorporated in the model and its significance determined by analyzing how it affects simulated behavior. This gives the engineer a strong new tool to assess our comprehension of a complex system and look into ways to boost performance. In the end, we are compelled to face every factor that can have an impact on the performance of interest and identify everything that is crucial.

The movement of cars on a road surface is the topic of "vehicle dynamics." The movements of interest are riding, turning, accelerating and braking. The forces applied to the vehicle by the tires, gravity, and aerodynamics defines its dynamic behavior. In order to ascertain what forces will be generated by each of these sources at a specific maneuver and trim condition, as well as how the vehicle will react to these forces, the vehicle and its components are analyzed. Establishing a strict methodology for modeling the systems and the conventions that will be applied to describe motions is crucial for achieving this goal.

The modeling involved starts simple and becomes more complex. The preparation of all the models necessary to completely parameterize all suspensions from all points of view is too large a task for this paper work. The approach followed is the ‘lumped parameter method’, and this will be introduced in the following chapter.

CHAPTER –3

SUSPENSION SYSTEM

3.1 SUSPENSION BASICS

This paper treats automotive suspension design issues primarily from the points of view of ride comfort. Attention is concentrated on vibratory motions in the vertical direction which normally dominate the motions in other directions. Lateral and longitudinal suspension kinematic properties are not dealt with, the influence of these being on higher frequency vibrations. Contemporary car suspension systems predominantly contain passive control elements, namely springs and dampers, and have evolved to the point at which it seems reasonable to suppose that they will not improve much without changes in principle. Recently such changes in principle have become commercially viable with advances in transducers, processors and force producers and new systems of several different types are of interest.

Passive systems will be considered to contain only springs and dampers with the possible addition of a self-leveling system which typically involves time delays of many seconds and which is designed to compensate for variations in static load. Active systems will be thought of as, at least in part, replacing the springs and dampers of passive systems by actuators which act as force producers according to some control law. The actuators operate with force transducers providing inner loop feedback signals to their controllers and are imagined to track faithfully a force demand signal determined by the control law. The control law may contain information of any kind obtained from anywhere in the system, and an important part of the active system design problem is the determination of the control law which will give a good system performance. It is implied in the description that the actuator control bandwidth extends to substantially beyond the wheel hop frequency, determined by the unsprung mass and tyre stiffness and typically near to 10 Hz. Actuators can be of either of two basic types. They can be flexible like a spring when they become inactive in a control sense (beyond the bandwidth), in which case they can support the body weight or act in parallel with a spring, or they can be rigid when inactive, in which case they must be mounted in series with a spring. Pneumatic actuators would be of the former type while spool valve controlled hydraulic and electric motor/irreversible leadscrew actuators would be of the latter type. This intrinsically rigid type of actuator is better treated as a displacement producer rather than a force producer. Its inner control loop would involve displacement feedback and the actuator demand signal will notionally be a desired displacement. It is indisputable that the capital cost of an active vehicle suspension system will be considerably greater than that of a conventional passive arrangement, and the active system may also involve significant running costs since energy supply is, in general, necessary. To determine the commercial viability of an active system, we need to quantify its performance advantages as well as its cost disadvantages.

Preview involves the acquisition and use for actuator control purposes of information relating to the road profile ahead of the vehicle wheels. It will be taken to include the use of front axle derived information in order to improve the rear axle actuator control laws under the assumption that the displacement input to the rear wheels is a time delayed version of that to the front wheels. Adaptation refers to the possibility with passive systems that the parameters of the system will be alterable in response to some kind of (relatively simple) information which indicates the running conditions, and correspondingly that the parameters of active systems will be changeable according to the conditions. An essential difference between passive and active system is that the latter types contain a measurement system and a control law such that some adaptive capability is virtually implicit in their design.

3.2 BASIC ELEMENTS OF A SUSPENSION SYSTEM

Shock Absorbers

The automotive industry most commonly uses single and twin-tube shock absorbers. Figure 1 shows a schematic diagram of a monotube shock absorber. Usually the gas is under a pressure of up to 25 bar. The high pressure in the interior of the shock absorber is intended to prevent cavitation and foaming. The movement of the piston rod causes the shock absorber piston to move up and down. During these movements, the oil flows through either the compression valve (2) or the rebound valve (1), with the two valves usually having different characteristics for compression and rebound. The separating piston between the oil and the gas seals off the gas from the oil. The gas volume is necessary to compensate for the piston rod volume, which moves inside the working cylinder during compression. Due to the high internal pressure, special demands are placed on the seal. One advantage of the monotube shock absorber is that it can be installed in any orientation. The disadvantages are the additional costs compared to a twin-tube shock absorber due to the higher manufacturing precision and the necessary tightness. A further advantage is the efficient cooling of the oil. The static pressure acting on the piston rod means that a static force acts permanently. If we assume a pressure of 25 bar and a radius of the piston rod of 6 mm, this force is about 280 N.

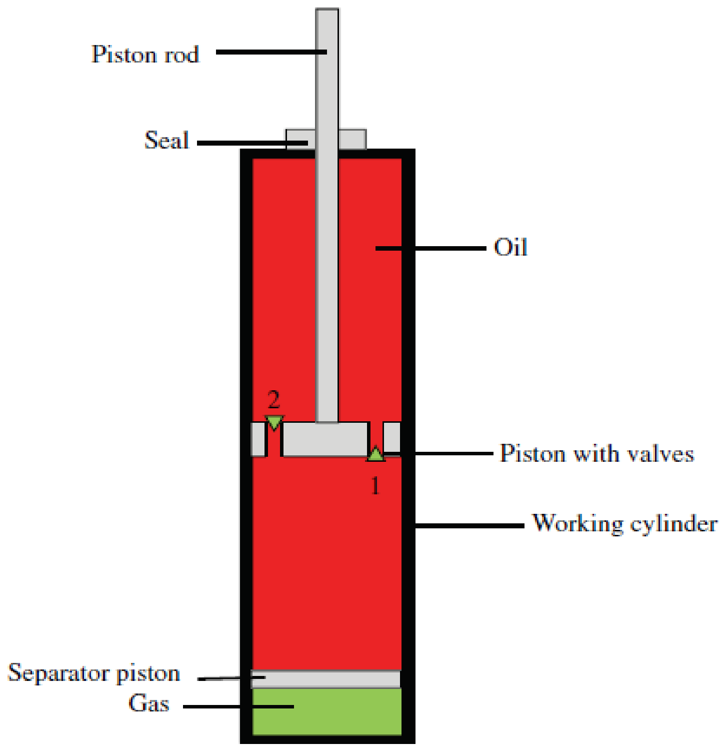


Figure 1: Monotube shock absorber.

Figure 2 shows the basic structure of a twin-tube shock absorber with the compression valve (1) and the rebound valve (2). Here the oil is not under a static pressure. Consequently, the demands placed on the seals and manufacturing precision are not as high as for the monotube shock absorber. Between the outer and inner cylinders is the compensation cylinder. The compensation cylinder is necessary for volume compensation of the rod. The oil flows through either the compression valve (4) or the rebound valve (3) into or out of the compensation cylinder, which is about half-filled with oil. The remaining part of the compensation cylinder is used for absorption of oil during expansion (temperatures of up to 120 ◦C are possible). Twin tube shock absorbers may not be installed in any arbitrary orientation; otherwise, air from the compensating cylinder would be drawn into the working chamber during rebound.

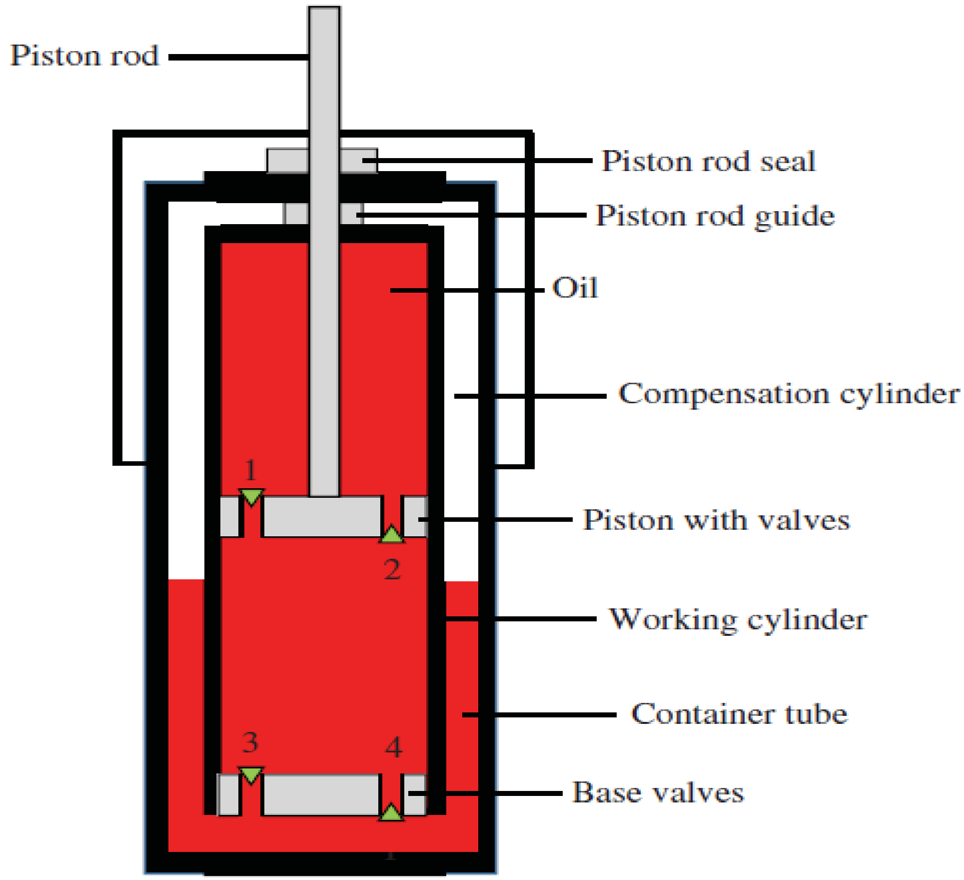


Figure 2: Twin-tube shock absorber.

Some special types of shock absorbers are also in use. One is the suspension strut module, which comprises the suspension spring and the shock absorber. This strut is used in the so-called McPherson wheel suspension. As the coil spring cannot significantly transfer forces or moments in a radial direction, the shock absorber has to do this. As a result, the diameter of the piston rod has to be larger than that of a conventional shock absorber.

Suspension Springs

Different kinds of springs are used in automobiles for connecting the wheel carrier and body. Springs support the weight of the vehicle. In passenger vehicles, a spring travel of ±100 mm may occur. In vehicle suspensions, coil springs (seldom leaf springs) are used mainly, but air springs, torsion bars, and leaf springs are also used, figure 3. Coil springs, torsion bars, and leaf springs absorb additional load by compressing. Thus, the ride height depends on the loading condition. Air springs are rubber cylinders filled with compressed air. They are becoming more popular on passenger cars, light trucks, and heavy trucks because here the correct vehicle ride height can be maintained regardless of the loading condition by adjusting the air pressure.

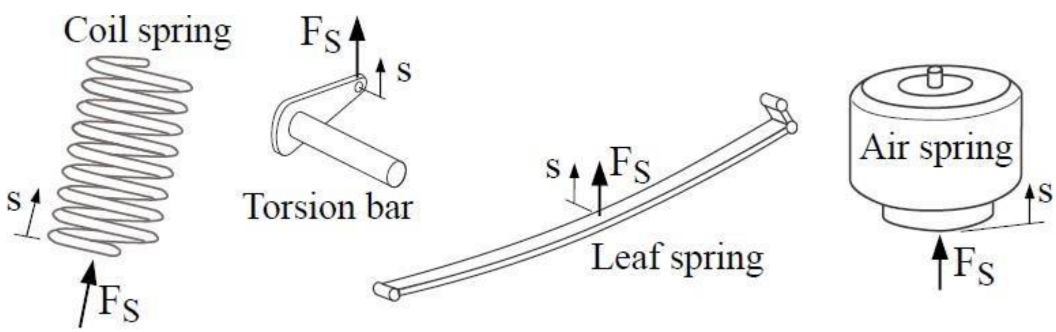
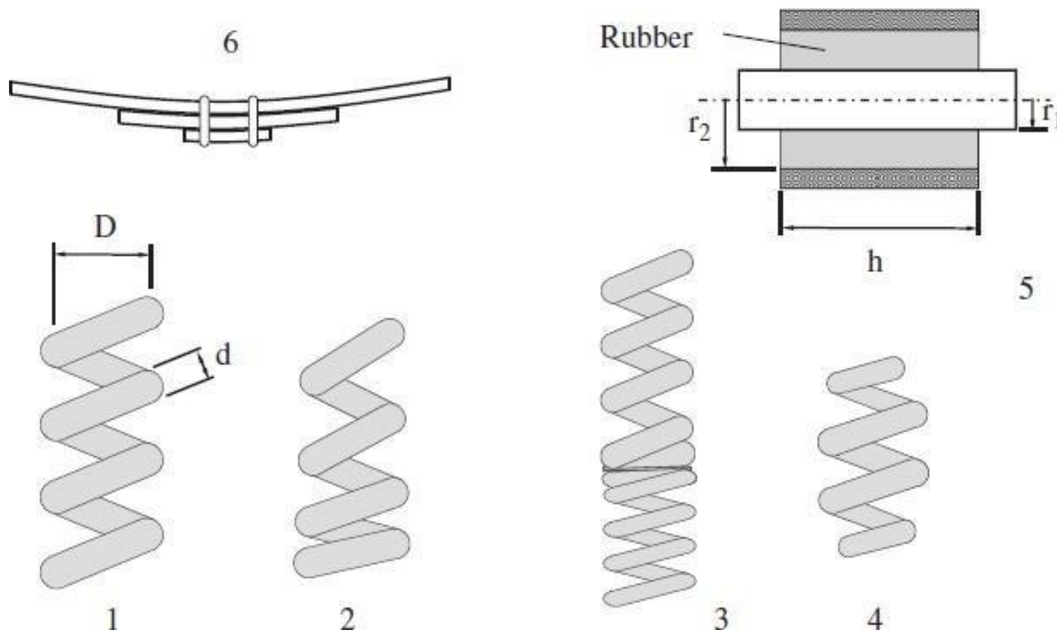


Figure 3: Vehicle suspension springs.

Leaf springs and torsion bars play a special role because they can also be used as suspension links. Leaf spring or torsion bar suspensions can be found in heavy-duty vehicles, but seldom in new passenger cars. Coil springs and air springs are not suitable for replacing suspension links. In addition to these springs (air, leaf, coil and torsion bar), rubber bushings are also used in the suspension in order to reduce noise and vibrations.

Leaf springs (cf. spring 6 in Figure 4) are bending beams with a very low second moment of area, in which the deflections are large. Laminated leaf springs of different lengths or leaf springs with a variable cross-section (parabolic profile) reduce the bending stresses at the clamping end points. Figure 4 shows a leaf spring and coil springs. The coil spring is a coiled torsion bar. Figure 4 shows some different designs. The basic shape is cylindrical (1) with constant wire diameter and mean spring diameter. The working characteristic (i.e. the force vs.

deflection) is linear. A non-linear, progressive characteristic is achieved with constant wire and coil diameters but variable pitch. When the spring is deflected, the number of active coils decreases due to variable pitch, and hence a progressive characteristic is obtained (cf. spring 2 in Figure 4). A progressive characteristic curve also results from a series arrangement of two different springs (spring 3). The largest number of degrees of freedom for creating a spring characteristic is provided by a barrel spring, where the pitch, the wire diameter and the mean spring diameter are not constant (spring 4).



<u>**Figure 4**: Vehicle suspension springs</u>

The spring stiffness of a spring with constant wire diameter *d*, mean spring diameter *D* and *n* active coils is (where *G* is the shear modulus) is given by,

$$c = \frac{Gd^4}{8nD^3}$$

A linear coil spring may be characterized by its free length L_F and the spring stiffness c, Figure5. The force acting on the spring is then given by,

$$F_s = c\ (L_F - L) = c\ \Delta L$$

Where L denotes the actual length and ΔL denotes the overall deflection of the spring.

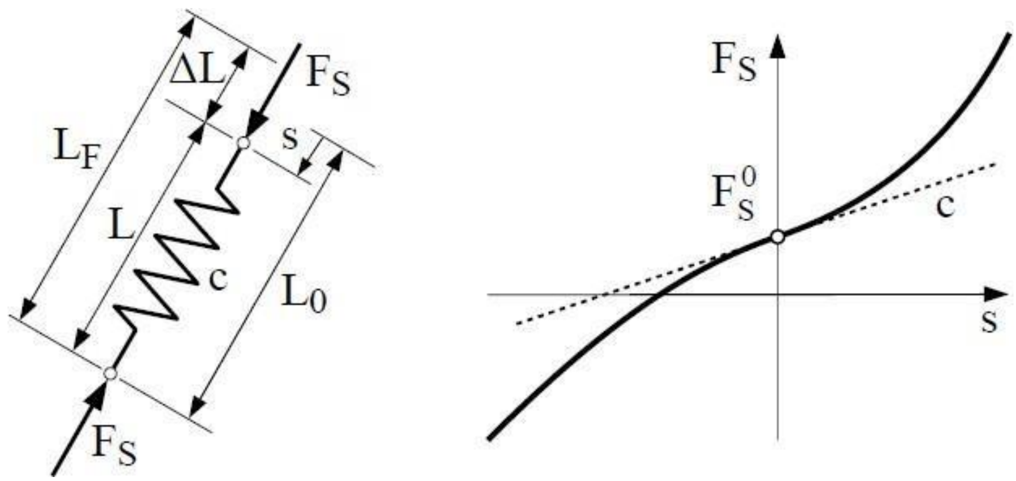


Figure 5: Linear coil spring and general spring characteristics.

Mounted in a vehicle suspension, the spring has to support the corresponding chassis weight. Hence, the spring will be compressed to the configuration length $L_0 < L_F$. Now, Equation (6.1) can be written as

$$F_s = c(L_F - (L_0 - s)) = c\,(L_F - L_0) + cs = F_s^0 + cs$$

Where F_s^0 denotes the spring preload and s = $L_0 - L$ describes the spring displacement measured from the spring's configuration length. Note, s > 0 indicates compression. In general, the spring force F_s can be defined by a nonlinear function of the spring displacement s,

$$F_s = F_s(s) = F_s(L_0 - L)$$

Now, arbitrary spring characteristics can be approximated by elementary functions, like polynomials, or by tables that are then inter- and extrapolated by linear functions or cubic splines. The complex behavior of leaf springs and air springs can only be approximated by simple nonlinear spring characteristics, $F_s = F_s(s)$.

Rubber bushings (cf. spring 5 in Figure 18.13) can be found as additional elastic components in vehicles, with examples being in the joints between the suspension arms and the body or subframe, in engine or transmission mounts, in anti-roll bar mounts or shock absorber mounts (e.g. for McPherson struts). The main stiffnesses of rubber bushings are axial c_a , radial c_r and torsional $\boldsymbol{C}_\varphi$:

$$c_a = \frac{2\pi h G}{\ln\left(\frac{r_2}{r_1}\right)}$$

$$c_r = \frac{k 7.5\pi h G}{\ln\left(\frac{r_2}{r_1}\right)}$$

$$\boldsymbol{C}_\varphi = \frac{4\pi h G}{\left(\frac{1}{r_1^2} - \frac{1}{r_2^2}\right)}$$

The correction factor, k, depends on the ratio between the height, h, and the thickness, $s = r_2 - r_1$, of the rubber. For $h/s = 0$ the factor is 1, and k increases progressively up to 2.1 for $h/s = 5$. G is the shear modulus of the rubber material.

3.3 FUNCTIONS OF SUSPENSION SYSTEM

The automotive suspension on a vehicle typically has the following tasks:

1) To isolate a car body from road disturbances in order to provide good ride quality
   Rode quality in general can be quantified by the vertical acceleration of the passenger locations. The presence of a well-designed suspension provides isolation by reducing the vibratory forces transmitted from the axle to the vehicle body. This in turns reduces vehicle body acceleration. In the case of the quarter car suspension, sprung mass acceleration can be used to quantify ride quality.
2) To keep good road holding
   The road holding performance of a vehicle can be characterized in terms of its cornering, braking and traction abilities. Improved cornering, braking and traction are obtained if the variations in normal tire loads are minimized. This is because the lateral and longitudinal forces generated by a tire depend directly on the normal tire load. Since a tire roughly behaves like a spring in response to vertical forces, variations in normal tire load can be

directly related to vertical tire deflection. The road holding performance of a suspension can therefore be quantified in terms of the tire deflection performance.

3) To provide good handling

   The roll and pitch accelerations of a vehicle during cornering, braking and traction are measures of good handling. Half-car and full-car models can be used to study the pitch and roll performance of a vehicle. A good suspension system should ensure that roll and pitch motions are minimized.

4) To support the vehicle static weight

   This task is performed well if the rattle space requirements in the vehicle are kept small. In the case of the quarter car model, it can be quantified in terms of the maximum suspension deflection undergone by the suspension.

3.4 SUSPENSION DYNAMICS

Lumped Parameter Approach

A common and very fruitful approach in dynamics modeling is to divide the system in question up into discrete lumped parameters that offer a simplified model when compared with the continuous system seen in the real life. For example, in the figure below, the encastred beam on the right is subjected to a sinusoidally varying force F. This system could be modeled by considering the beam as a continuous elastic solid with internal damping and preparing complex analytical equations for the resulting motion. A numerical approach could be taken using dynamic finite element methods. However, by considering the system to be equivalent to the lumped parameter system on the right, a relatively swift analysis can be prepared using classical methods (Fig.1).

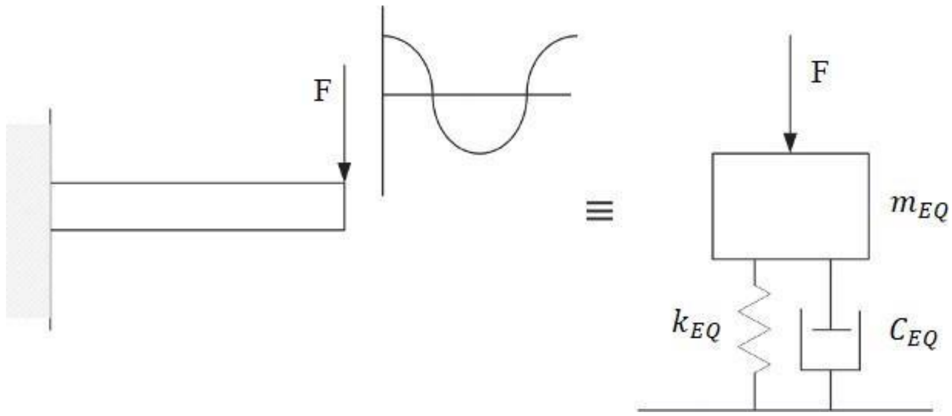
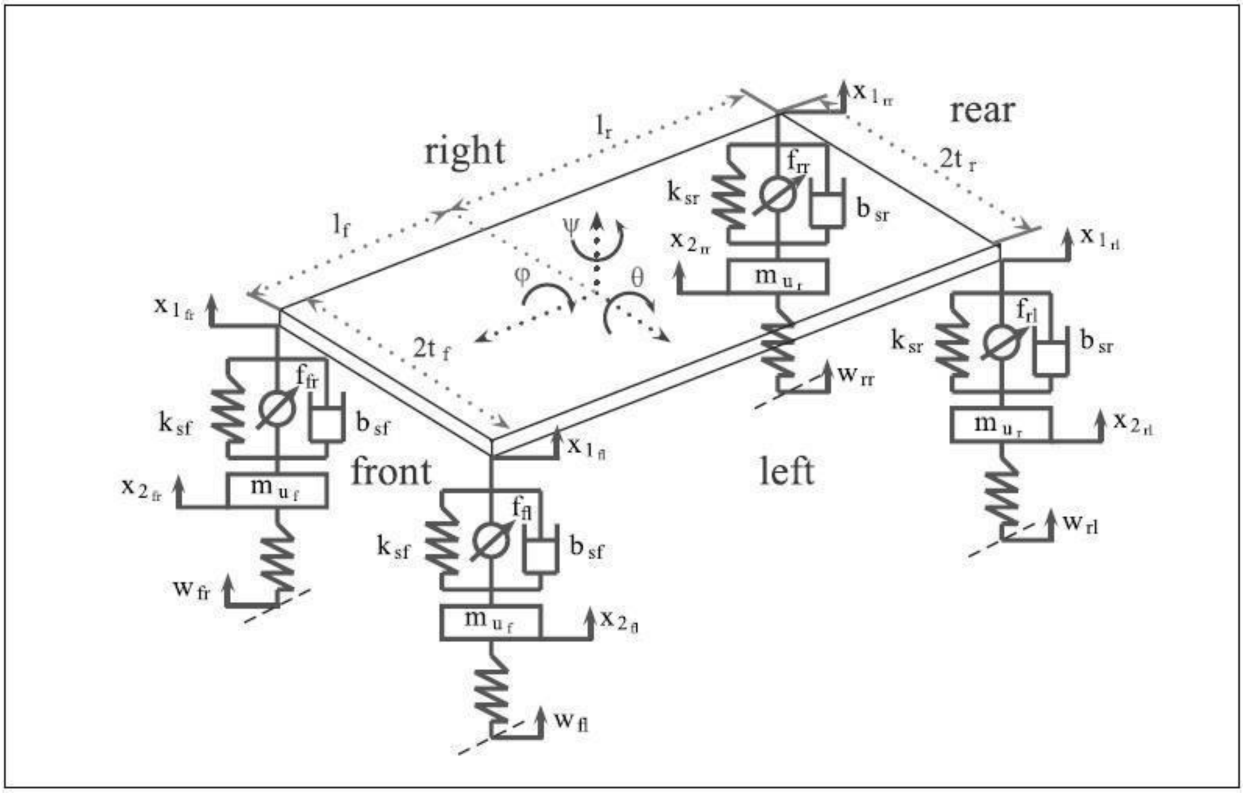


Figure 6: The lumped parameter approach to modeling dynamic systems.

In the example above, we would need to find a way to model the elastic displacement under the load in the beam as a single spring constant, k_{EQ}, in the lumped parameter model on the right. In a similar way, we would need to find an equivalent damping value C_{EQ} to represent the damping distributed evenly within the beam and similarly for the mass m_{EQ}. Once these equivalent parameters are found, then a solution to the model can be prepared, based on equations of motion for the lumped parameter model, and from these, optimization can begin. Now the lumped parameter model for a whole car will be addressed.

Fig. 2 shows a seven-degree-of-freedom-lumped parameter model of a vehicle and its suspension. This level of detail provides a very useful model, and much can be learned and usefully applied though its consideration.



<u>**Figure 7:** Seven-degree-of-freedom approximation for a vehicle suspension system.</u>

In the model, the road provides a vertical displacement input that the contact patch is required to follow. The displacement of the contact patch with respect to the unsprung mass causes compression in the tyre, which develops a force. The unsprung mass is free to displace vertically. The unsprung mass is linked to the car body by a spring to model the suspension stiffness and normally a damper to model the damper unit. The body is free to move vertically in heave and to rotate about the centre of gravity in the side view, causing pitch, and in the front view causing roll.

In the figure, we can see that the road inputs might conceivably conspire to produce vertical motion only. Indeed, there will be a particular natural frequency in heave at which the road heave input will produce a maximum body heave output. Alternatively, the inputs might happen to oscillate by rising at the rear whilst falling at the front or by rising at the left whilst falling at the left. These possibilities give rise to a heave, pitch or roll motion, respectively. Again, there will be a natural frequency for each of these motions influenced by the stiffness damping and mass moment of inertia about each axis. One final possibility exists called warp, when the diagonal inputs move in antiphase, the left rear and right front rising whilst the right rear and left front fall. There is no resulting large-scale movement of the body in warp because it is a stiff structure designed to resist such deformation. A warp input for a symmetrical car with equal dynamic parameters all around would result in no body motion because the moment rotating the body in one direction is cancelled by a moment rotating it in the other, from the other end. However, such symmetry is rare, and although the displacements resulting from warp are generally smaller than other modes, because even if the moments don't cancel they oppose each other, it is still significant. On top of all this, there are the motions of the unsprung mass to consider, each of which will have their natural frequencies.

For a real car travelling over a real road surface, the input waveforms would clearly consist of many different frequency inputs, and there would be an input frequency spectrum. The inputs would very likely be a random mixture of displacements.

Thus, we see that for the seven-degree-of-freedom system, the output response is going to be a complex affair. The road will provide inputs over a wide frequency range, and the body will respond in four modes and have seven degrees of freedom. The overall response of the body would then be a superposition of these four modes. The uprights will have their own resonant frequencies that will affect the body response, for example, if the front uprights are being driven at their natural frequency, then their displacements will be large, and this will be communicated to the body that itself will show an increased amplitude of vibration at this frequency. So the seven-degree-of- freedom system is complicated. Instead, we shall approximate the car to a simpler version of all such systems, the two-degree-of-freedom system, and start there.

Quarter Car Modeling

In this paper approximation of the suspension to a two-degree-of-freedom system is being done, featuring either a harmonic input of constant amplitude or a step input. The quarter car model for passive suspension system consists of one-fourth of vehicle body mass, suspension components and one wheel as shown in Figure 8. It is constructed on the basis of a four-wheel independent limited bandwidth suspension. In this approximation, the whole car is assumed to move vertically together, and there is no roll or pitch. The input from the road is assumed to be entirely in heave, with all four wheels moving as one. This is clearly a large approximation, but it gives a starting point and yields useful results despite its simplicity. Heave, is in fact, the largest of the road inputs.

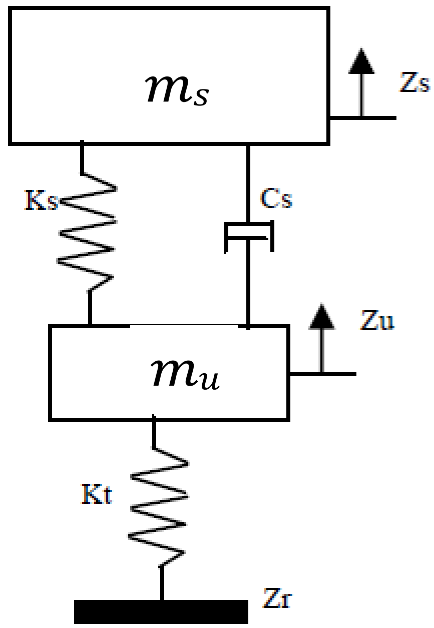


Figure 8: Quarter car model.

3.5 PERFORMANCE CRITERIA

Suspension system performance criteria include discomfort, working space, wheel load variation, static and dynamic attitude control, contributions to good steering behaviour, and for active systems, actuator force levels and power consumption. Using the simple quarter car model, discomfort, working space, wheel load variation, static attitude variations, actuator forces and energy consumption can all be calculated, provided that a relationship between discomfort and vehicle body motion be established. However, dynamic attitude control and contributions to steering behaviour depend on many factors and would only be calculable using comparatively very detailed models. It must suffice here to note that statically stiff systems have advantages for dynamic attitude control and handling, and that active systems clearly have great potential in respect of these qualities if control laws are suitably devised.

Much research effort has gone into the objective measurement of discomfort. One line of attack, involving the testing of human subjects with sinusoidal disturbances of different frequencies in a laboratory environment, is well reviewed by Pollard and Simons [32]. Another, based on determining the correlation between objective criteria and subjective opinion in vehicle tests, was pursued by Aspinall and Oliver [33] and later by Smith et al. [34]. Smith's experimental design is subject to the criticism that it was bound to lead to very high levels of correlation between almost all of the objective measures which were of interest, and that consequently if the subjective ratings correlated with any of the objective measures, they would correlate with all of them, as indeed happened. The main conclusion of Aspinall and Oliver was that the root mean square (rms) vertical acceleration of the vehicle body at the passenger location was a good measure of discomfort and Smith's results confirm that finding. Laboratory testing has meanwhile led to standardized reduced comfort boundary curves [35] and it is now widely accepted that acceleration data should be frequency weighted before rms values are formed as objective measures of discomfort.

CHAPTER –4

HYDRAULIC ACTUATOR

A complete set of a hydraulic actuator consists of five main components namely electro hydraulic powered spool valve, piston-cylinder, hydraulic pump, reservoir and piping system as shown in Figure 9. The actuator shown in the figure is a translational double acting hydraulic actuator driven by a three-land four-way spool valve. Power supply is needed to drive the hydraulic pump through AC motor and to control the spool valve position. The hydraulic pump will keep the supply pressure at the optimum level of about 10342500N/m^2. The spool valve position will control the fluid to come-in or come-out to the piston-cylinder which determines the amount of force produced by the hydraulic actuator.

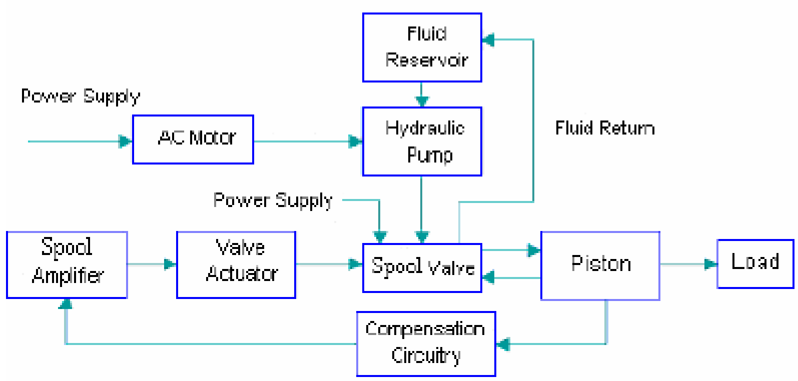


Figure 9: Diagram of a complete set of hydraulic actuator.

The hydraulic actuators are governed by electro hydraulic servo valve allowing for the generation of forces between the sprung and unsprung masses. The electro hydraulic system consists of an actuator, a primary power spool valve and a secondary bypass valve. As seen in Figure 10, the hydraulic actuator cylinder lies in a follower configuration to a critically centered electro hydraulic power spool valve with matched and symmetric orifices. p_s and p_r are the supply and return pressures going into and out of the spool valve, respectively, x_v is the spool valve position, p_u and p_l are the oil pressures in the upper and lower cylinder chambers respectively and $(x_w - x_c)$ is the hydraulic piston displacement. Positioning of the spool (x_v) directs high pressure fluid flow to either one of the cylinder chambers and connects the other chamber to the pump reservoir. This flow creates a pressure difference P_L across the piston. This pressure difference multiplied by the piston area A_P is what provides the active force F_a for the suspension system.

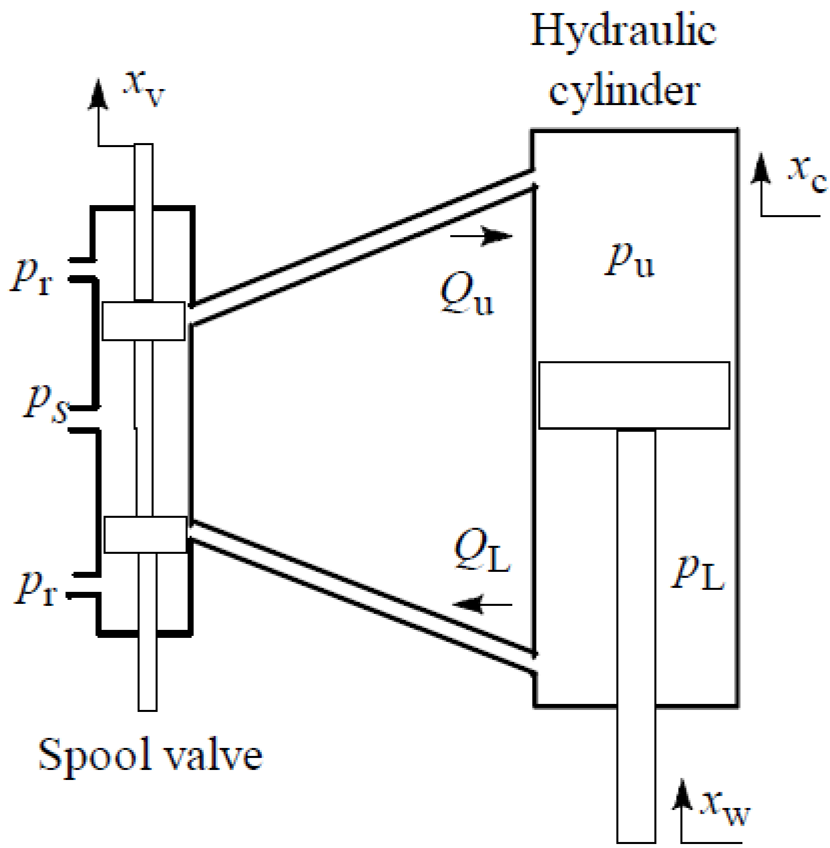


Figure 10: Schematic diagram of a double acting hydraulic actuator.

The actuator works as follows. As the spool valve move upward (positive x_v), the cylinder upper chamber is connected to the supply line p_s and its pressure increases. At the same time, the lower chamber is connected to the return line p_r and its pressure decreases. The pressure difference then make the hydraulic cylinder extend or compress.

The differential equation governing the dynamics of the actuator is given in [37, 38] as follows:

$$\frac{V_t}{4\beta}\dot{P}_L = Q_L - C_{tm}P_L - A(\dot{z}_s - \dot{z}_u)$$

where F_a, V_t, β , A, C_{tm} are force, total actuator volume, effective bulk modulus, actuator ram area and coefficient of total leakage due to pressure, respectively. Note that the actuated force, F_a could be easily computed using the general equation:

$$F_a = P_L A$$

where F_a is the actuator force, P_L is the pressure and A is the cross sectional area of the cylinder (pipe). Using the equation for hydraulic fluid flow through an orifice, the relationship between spool valve displacement x_v and the total flow Q_L is given by:

$$Q_L = C_d w x_v \sqrt{\frac{p_s - sgn(x_v)P_L}{\rho}}$$

where C_d and w are the discharge coefficient and spool valve area gradient respectively. The spool valve displacement x_v can be related to the input current i_v through the following linear dynamic equation:

$$\dot{x_v} = \frac{1}{r}(-x_v + k_v i_v)$$

where k_v is the valve gain and τ is time constant of the servo valve.

CHAPTER –5

PID CONTROL THEORY

5.1 DESIGN OF CONTROL SYSTEM

Controlling system behavior is an important necessity for practical systems. No matter how well a system is designed, its response (or system output) is not always exactly what was desired or expected. Also, external disturbances can affect the system; and its behavior, as a result, may change from what was desired. Hence, it is necessary to have a control system that will adjust system behavior by altering the input so that the desired output is achieved. There are some simple control techniques, such as a “bang-bang” (on/off) control, which are of limited use. The most well-known and useful technique is feedback control, where the response of the system is monitored and compared with the expected response, and the error in the response is used to alter the input dynamically to achieve the desired result. The block diagram in Figure shows a schematic of the feedback control system. The system’s (or plant’s) response is subtracted from the set point to obtain the error. The error signal is used in a control algorithm to determine the system input that is fed into the system, and the response is adjusted, as a result, to achieve the desired output.

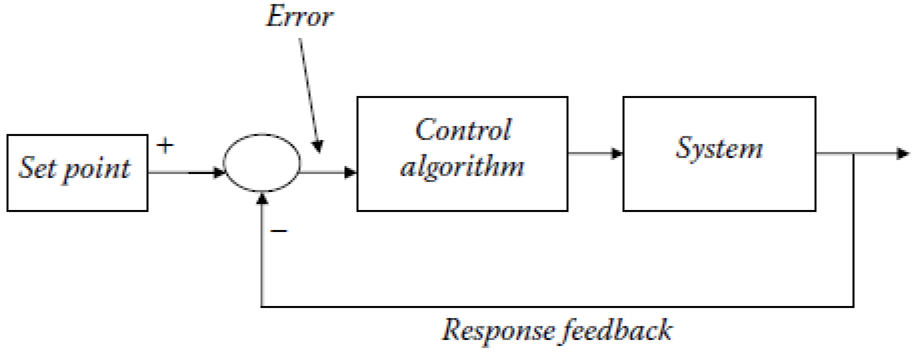


Figure 11: Schematic showing of a feedback control loop.

Control system designers use several different measures of system performance. These measures include:

Stability: Initial condition disturbances should die off quickly.

Speed: The system need to react quickly.

Sensitivity: System sensitivity to noise should be low and to control inputs should be high.

Accuracy: Error should be low.

Dynamic coupling: Reduced coupling among system variables.

When designing control systems, one has to judge effectiveness using some of these criteria.

5.2 DESIGN WITH PID CONTROLLER

In all feedback control algorithms, the actual output is fed back into the control system so that an error measure (the difference between expected and actual output) is computed, and the error measure is used to set the change in the input needed to minimize the error. More than 90% of all control strategies revolve around the use of PID controls in some form. PID stands for proportional, integral, and derivative control. PID control is made of three different control strategies that are a function of the error, the rate at which this error changes (derivative), and the accumulation of error (integral).

Proportional Control

This is a control strategy where the control signal is proportional to the error.

We could write the control signal as:

$$\textit{Control Signal} = Kp\,(\textit{error})$$

Where error = (expected output – actual output)

The parameter that needs to be adjusted in this case is the multiplier parameter *Kp*. The proportional control module receives the error information (the difference between set point output and actual output). The model for the proportional control module is expressed through the following statements:

Equations

Output = kp×error;

So, the difference between the expected value and the actual value is multiplied by the *Kp* factor to determine the command variable that will be used for the system. The proportional control method is simple and works relatively fast. However, it has poor stability characteristics and may end up giving offset errors. There is a steady state or offset error that remains. We can try increasing the *Kp* factor. The response is significantly better, but a smaller offset error between the set point and the actual output still persists. We can try and correct this by increasing *Kp* even more, the error has been reduced even more, but the system response then becomes oscillatory.

So some of the important behavioral characteristics of a proportionally controlled system are, specifically, a steady state error usually persists; its magnitude can be reduced by increasing the *Kp* value, but beyond a certain limit, the higher *Kp* will make the system behavior oscillatory or less stable.

Proportional Integral Control

PI control stands for proportional integral control where the control signal can be written as:

$$\text{Control Signal} = K_p\left(error + \frac{1}{T_i}\int (error)\, dt\right)$$

This means that the control signal is dependent not only on the actual error but also on how much error is accumulating over time (i.e., the integral of error). Two parameters in this control algorithm are to be adjusted in order to tune the control signal: the *Kp* factor and the T_i factor. While *Kp* factor is the same error multiplier as before, T_i is the integral time constant of the error, and its reciprocal is sometimes referred as integral rate. The special characteristics of the integral controller are that it is good at eliminating offset errors and reducing noise. However, response is slow and may have stability issues. But in general, it complements the proportional controller very well and is often used in conjunction with the proportional controller.

Proportional Derivative Control

There is another mode of feedback control that is called the PD control. PD stands for proportional and derivative control. In this case, the control signal or control input may be written as:

$$Control\ Signal = K_p\left(error + T_d\frac{d}{dt}(error)\right);$$

For the PD control, the control signal is a function of both the error itself as well as the derivative of the error or the rate at which the error is changing. The two user-defined factors that control the behavior of this algorithm are Kp and T_d. The parameter T_d is known as the derivative time constant. The response of a derivative controller is very fast and this generally improves stability of the controller. But this controller also tends to amplify noise. The offset error can be reduced, and the speed of the controller is improved without it becoming oscillatory.

Proportional Integral Derivative Control

The last and most frequently used control algorithm is the PID control, which stands for proportional, derivative, and integral control all together. Obviously, all three control algorithms that make up the PID control have some advantages and some disadvantages. With this approach of combining all three, the control system designers attempt to maximize the advantages and minimize the disadvantages. This process is carried out in reality by adjusting the three parameters in the PID controller, namely Kp, T_i , and T_d . The process of adjusting these parameters is also known as tuning.

The control signal in this case can be written as a sum of three components, the proportional, integral, and the derivative. The control signal in this case is represented as:

$$Control\ Signal = K_p\left(error + \frac{1}{T_i}\int(error)dt + T_d\frac{d}{dt}(error)\right)$$

The described PID control algorithm works very well once the adjustable parameters are properly set. One way of doing this is by trial and error method. It is believed that obtaining the right control parameters requires some experience and practice working with the particular system. Control experts have come up with different algorithms of tuning these control parameters. The most well known among these is the Ziegler–Nichols criteria for PID control.

Ziegler–Nichols Closed Loop Method

This technique provides a systematic approach to tuning the three parameters for the PID control. First, the controller is set to proportional mode. Next, the gain of the controller (K_p) is set to a small value. The load or set point is changed a little, and the response of the controlled variable is tracked. If K_p is low, the response will be quite slow. Now K_p is increased by a factor of 2 and the set point or the load is changed a little. At every trial, K_p is increased by a factor of 2 until the response becomes oscillatory. Finally, K_p is adjusted until the response is completely oscillatory. This particular value of Kp is called the ultimate gain (K_u). For this value of K_p, the period of oscillation is also recorded as T_u. The control law settings are then obtained in terms of K_u and $T_u$and are listed in the table below:

Parameters for Tuning the PID Controller

	K_p	T_i	T_D
P	$K_u/2$		
PI	$K_u/2.2$	$T_u/1.2$	
PID	$K_u/1.7$	$T_u/2$	$T_u/8$

It is unwise to force the system into a situation where there are continuous oscillations because this represents the limit at which the feedback system is stable. Generally, it is a good idea to stop at the point where some oscillation has been obtained. It is then possible to approximate the period (T_u), and if the gain at this point is taken as the ultimate gain (K_u), it will provide a more conservative tuning regime.

CHAPTER –6

SUSPENSION SYSTEM MODELING

6.1 GENERAL

A linear two degree of freedom quarter-car model shown in figure 12 is considered here. The most general and useful automotive suspension system design information derives from the single wheel station, quarter car representation. The real problem approximates to this form only for long wavelength, low frequency inputs but for higher frequencies it has been argued [37] that coupling of the wheel stations through the body motion is not so important because the body motions are not so large. The quarter car model contains no representation of the geometric effects of having four wheels (wheelbase filtering and its lateral equivalent) and offers no possibility of studying longitudinal interconnection, the use of front suspension state information to improve the performance at the rear, or the resonant excitation of the body roll motion. However, it does appear to contain the most basic features of the real problem and to give rise to design thinking which accords with experience. Important features of the quarter car model are that it includes a proper representation of the problem of controlling wheel load variations and contains suspension system forces which are properly applied between wheel mass and body mass. It is the simplest model which has these features and possesses particular advantages over more complex models in terms of

(a) being described by few design parameters,

(b) having few performance parameters,

(c) having only a single input, leading to ease of computation of performance and ease of application of optimal control theory to derive control laws, and

(d) ease of mapping and understanding of the relationships between design and performance.

When full advantage of the simplicity of the quarter car model has been taken and studying it further offers no benefits, more elaborate models will become cost beneficial and of course are essential at any stage to assess features which are implicitly omitted from simpler approaches.
Six different quarter car models are being prepared, one is passive suspension and the others are with different control strategies. They are as follows:

- Quarter car model with passive suspension system.
- Quarter car model with sinusoidal road profile and PID controller.
- Quarter car model with step-up road profile and PID controller.
- Quarter car model with a single sinusoidal wave front of low amplitude road profile and PID controller.
- Quarter car model with a single sinusoidal wave front of higher amplitude than the previous road profile and PID controller.
- Quarter car model with sinusoidal road profile and PID controller to control the actuator force for an active suspension system.
- Quarter car model with step-up road profile and PID controller to control the actuator force for an active suspension system.

The above dynamic suspension systems of a quarter car model are formulated and analyzed using the SIMULINK (MATLAB).

The following assumptions have been considered for the effective operation of the above mentioned vehicle models:

• The modeling is carried out with the assumption that the system has two degrees of freedom, and only the vertical dynamics' displacements and other components are examined.

• There is no rotational motion in wheel and body.

•The tire is modeled as an undamped linear spring. The behavior of the springs and dampers is linear, and neither the wheel nor the chassis have any rotational motion. The springs and dampers also have very little mass.

• The road surface is always in contact with the tire. Friction's impact is disregarded. The centre of gravity of the masses is where the forces act.

• Minor forces like linkage flex, vehicle body flex, and backlash etc are disregarded.

6.2 PASSIVE SUSPENSION SYSTEM

A passive suspension system is a mechanical system of springs and shock absorbers. A conventional passive suspension system composed of the non-controlled spring and the shock-absorbing damper. Both components work mechanically in parallel and are fixed between the wheel supporting structure (unsprung mass) and the vehicle body (sprung mass). The spring constant and damping coefficient values are fixed (as defined for respective application); spring stores vibration energy in form of strain energy and damper dissipates energy due to compressive action over fluid. Resistance to free vibration gives damping effect and such system is called damper. The damper is a cylinder filled with hydraulic oil or compressed gas. Inside the cylinder there is a piston driven by a rod. Furthermore, the fluid or gas may pass between the components inside the cylinder. This fluid or gas flow generates a reaction force that is proportional to the relative speed between sprung and unsprung masses. The damping is achieved by converting the energy of the oscillations in heat.

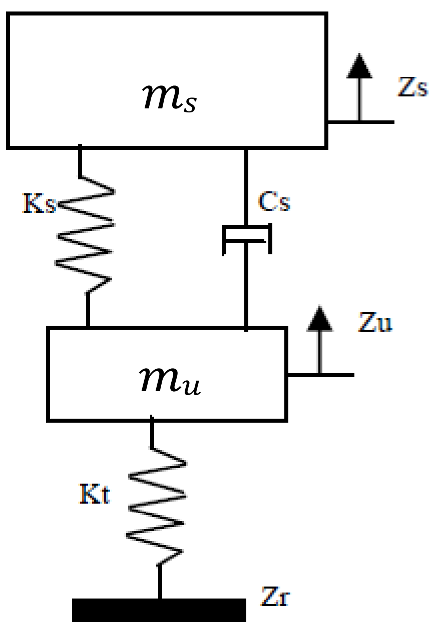


Figure 12: Quarter car model of a passive suspension system

Applying Newton's Second Law of Motion to the masses, the governing equations (neglecting k_t) are obtained as follows:

$$m_s \ddot{Z_s} = k_s (Z_u - Z_s) + c_s(\dot{Z_u} - \dot{Z_s})$$

$$m_u \ddot{Z_u} = k_s (Z_s - Z_u) + c_s(\dot{Z_s} - \dot{Z_u}) + k_t(Z_r - Z_u)$$

Where,

m_s = Sprung mass

m_u = Unsprung mass

Z_s = Displacment of sprung mass

Z_u = Displacement of unsprung mass

k_s = Spring stiffness of suspension

k_t = Spring stiffness of tyre

c_s = Damping coefficient of the suspension

Z_r = Excitation of road surface

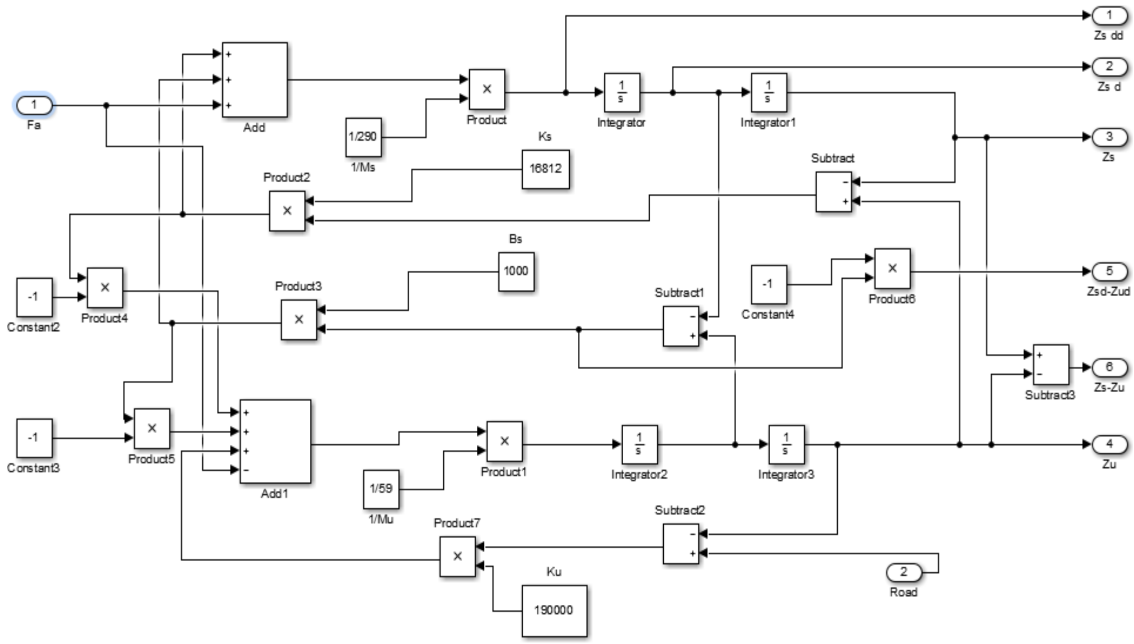


Figure 13: The quarter car model in SIMULINK

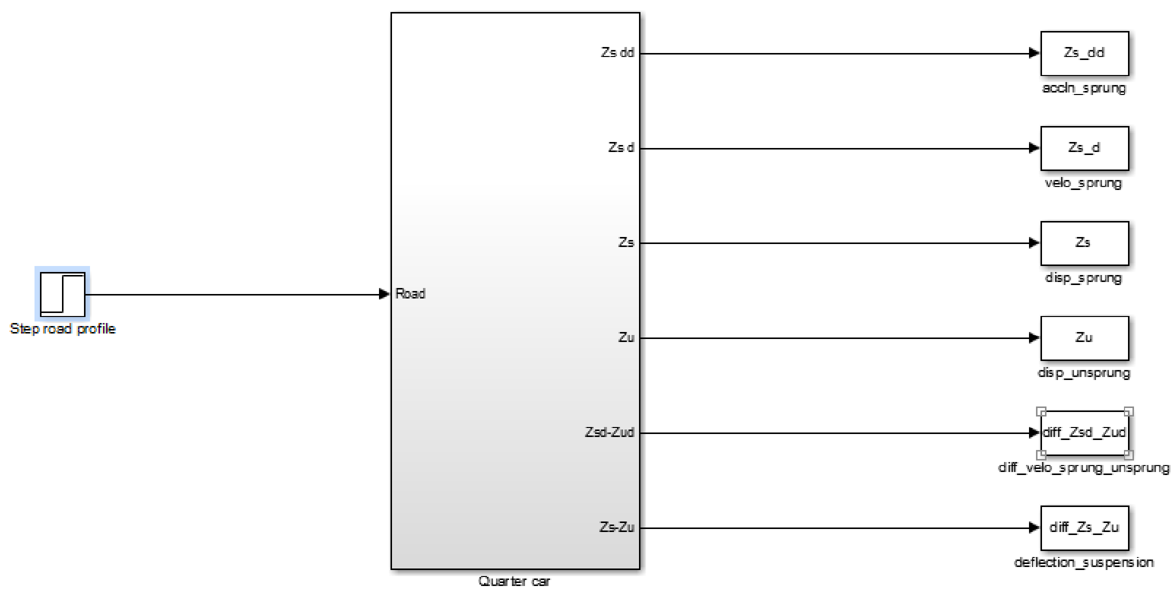


Figure 14: Passive suspension system in SIMULINK

6.3 FIRST MODEL OF ACTIVE SUSPENSION SYSTEM USING PID CONTROLLER

In the very first model of an active suspension system we have used PID controller without considering an actual hydraulic actuator, instead we have used a gain block in the simulation process to mimic the role of a hydraulic actuator. The quarter car model with active suspension system based on PID controller is modeled in SIMULINK according to the figure 15 as shown below:

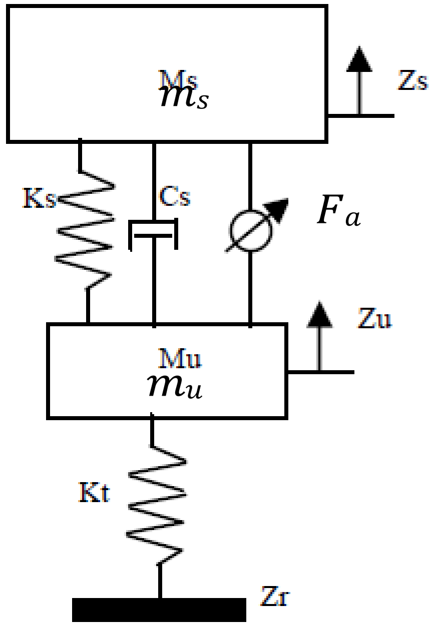


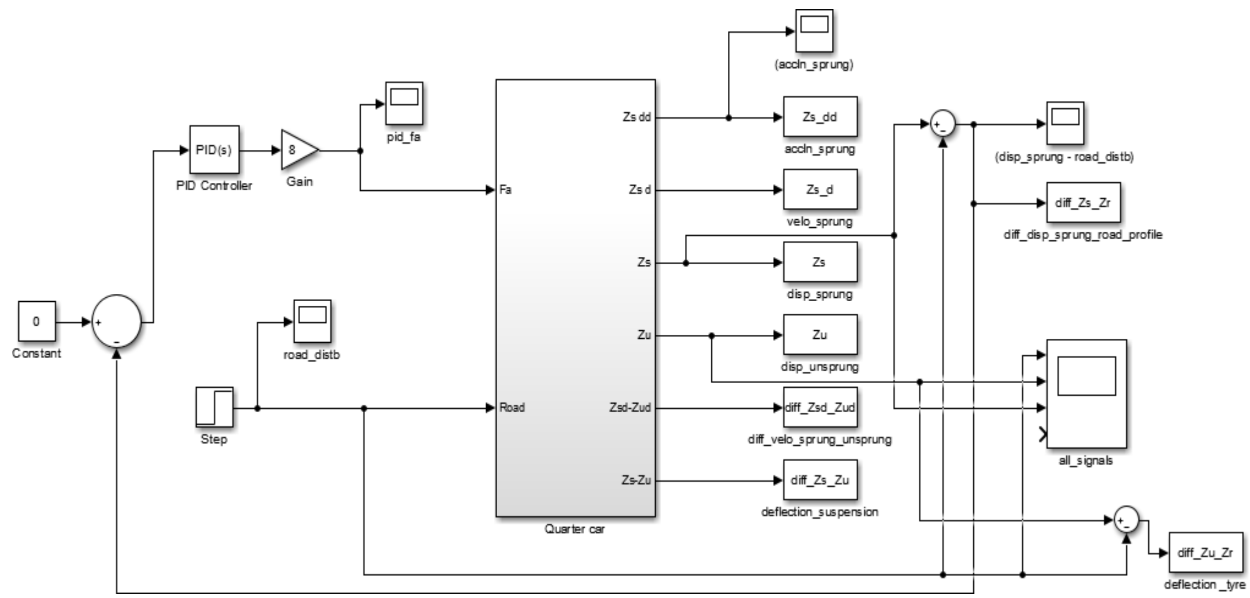
Figure 15: Active system configuration of a quarter car model

By applying Newton's second law, the equations of motion of m_s and m_u are:

$$m_s \ddot{Z}_s = k_s (Z_u - Z_s) + c_s(\dot{Z}_u - \dot{Z}_s) + F_a$$

$$m_u \ddot{Z}_u = k_s (Z_s - Z_u) + c_s(\dot{Z}_s - \dot{Z}_u) + k_t(Z_r - Z_u) - F_a$$

The Proportional Integral Derivative (PID) is a very useful and common controller in suspension system which is usually used in order to increase the stability and ride handling performance in automobile system. This controller has three parameters (K_P, K_I and K_D) which normally help the system to decrease rise time, reduce the steady state error, and decrease the settling time and overshoot of the system. The basic PID controller structure is connected to the Quarter car plant. And the PID controller is tuned to give an output force to the quarter car system to stabilize the plant. PID controller is tuned using trial and error method to get the better value of controller parameters K_P , K_I and K_D which assists to increase the stability of the system, improve transient response as well as eliminate steady state error. The controller reference is used as a target of control design which keeps the body flat at any external disturbance coming. In our quarter car case, ‘0’ constant value is set as a reference and quarter car model is used as the plant in the closed-loop system. For the input signal to the PID controller the displacement signal of the sprung mass is subtracted from “zero” which is the reference considered here. By doing so the PID control strategy actually comparing the output force signal with the actual given road profile. In practice this is similar to maintaining the suspension deflection constant. The PID model in SIMULINK is shown in figure 16.



<u>**Figure 16:** SIMULINK model of an Active suspension system with PID controller and step input as road profile</u>

The controller parameters and the compensator formula used in the above mentioned SIMULNIK model is shown in the figure 17.

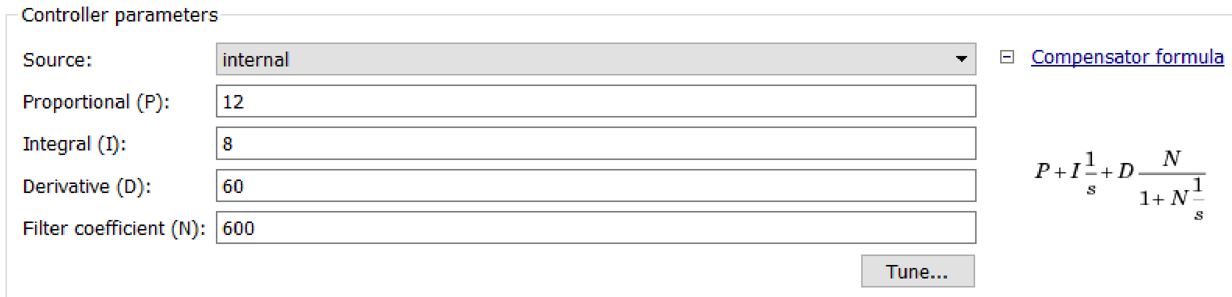


Figure 17: PID control parameters and compensator formula.

Another model of the same configuration as above except the road profile has also been studied. Here a sinusoidal road profile is used for the analysis. As any driver in real world condition will go slowly over a bumpy road, the approach velocity of the car towards the bump in this model is considered to be 1m/s. The SIMULINK model is shown in the figure 18 below:

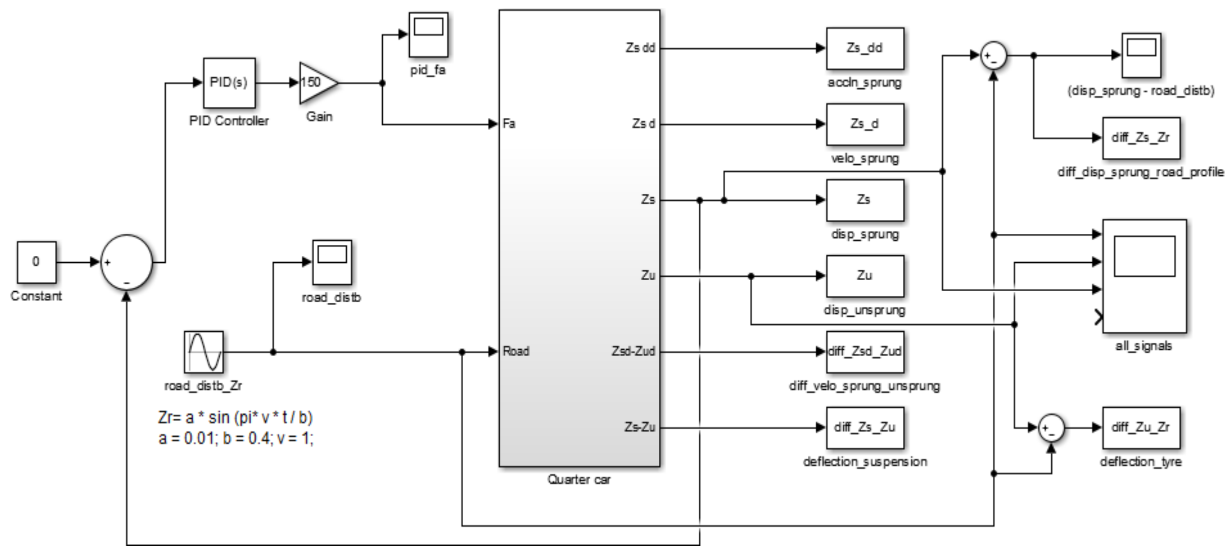


Figure 18: SIMULINK model of an Active suspension system with PID controller and sinusoidal road profile.

The controller parameters and the compensator formula used in the above mentioned SIMULNIK model is shown in the figure.

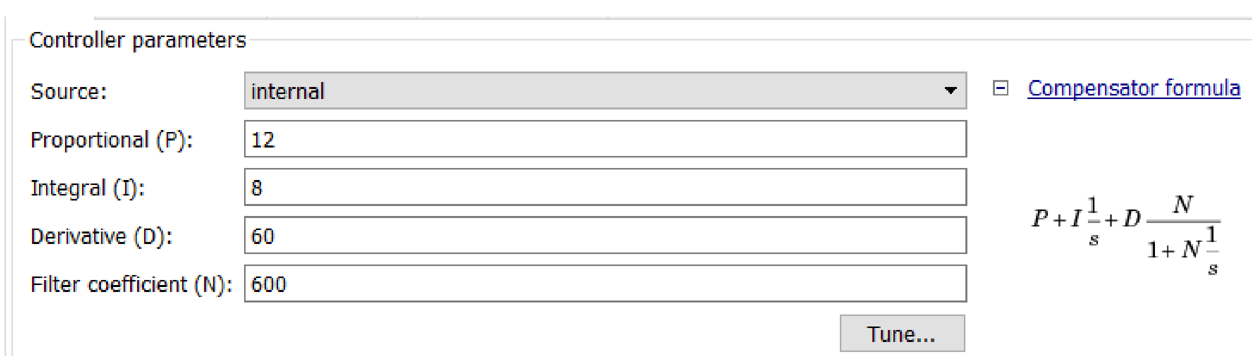


Figure 19: PID control parameters and compensator formula.

For both the simulation models mentioned above the values of K_P, K_I and K_D were adjusted by trial and error method.

6.4 SECOND MODEL OF ACTIVE SUSPENSION SYSTEM USING PID CONTROLLER

In this model also PID controller is being used to control the vertical dynamics of the quarter car model and here also no actual actuator is considered for the simulation. In this second model of the thesis work the road profile is modeled as a discrete sine curve which will represent a bump in real road condition. The approach velocity of the car towards the bump in this model is considered to be 1m/s.

The equation for the sinusoidal wave representing the road bump is as follows:

$$y = a\ \sin(\omega t)$$

$$\Rightarrow y = a\ \sin(\frac{2\pi v}{b} t)$$

Where,

a = amplitude of the wave (here in this study it is the height of the road bump)

ω = angular velocity of the wave

t = time period

v = linear velocity

b = wavelength (here in this study it is the width of the road bump)

The road profiles used in this simulation model are sine curves, as follows:

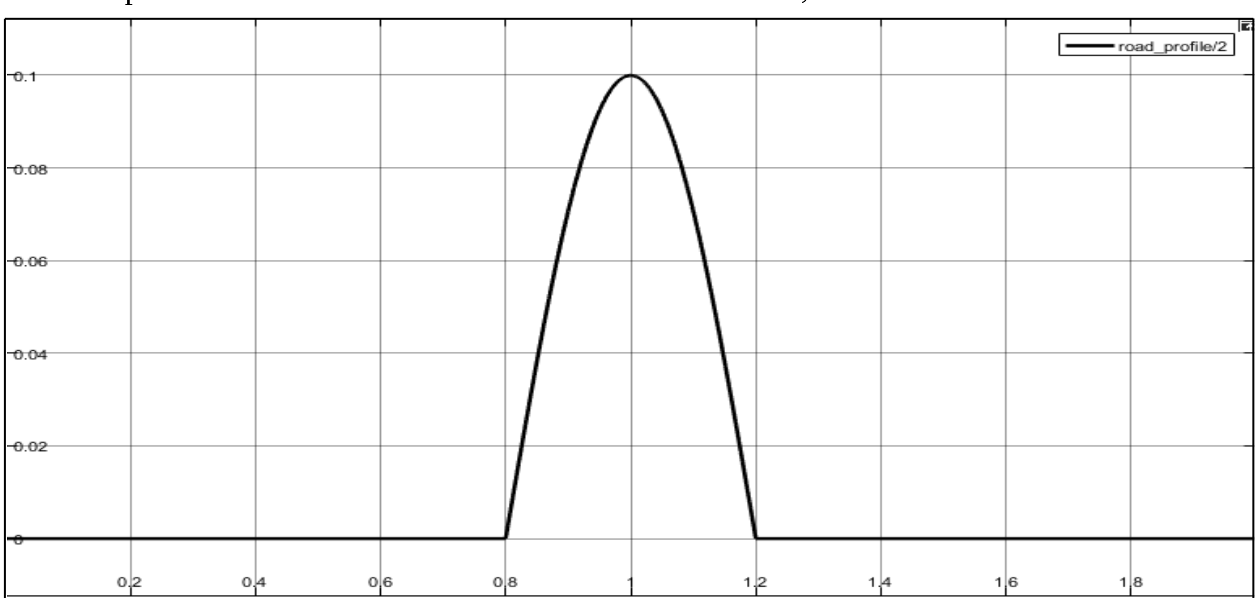


Figure 20: A discrete sinusoidal wave representing bump in real road condition

For the above mentioned road profile the values of the parameters are as follows:

a = 0.1, b = 0.4 and v = 1

Here "v" is actually the approach velocity of the quarter car model, the value of which we can optimize.

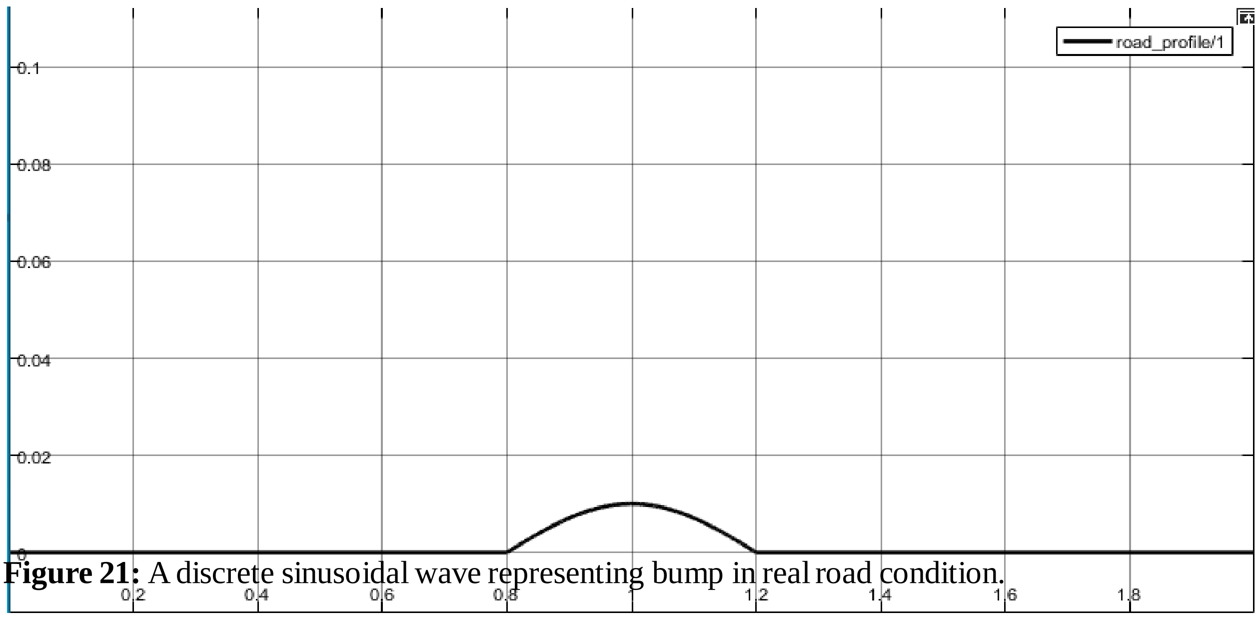


Figure 21: A discrete sinusoidal wave representing bump in real road condition.

For the above mentioned road profile the values of the parameters are as follows:

a = 0.01, b = 0.4 and v = 1

The SIMULINK model is as shown in the figure 22 below:

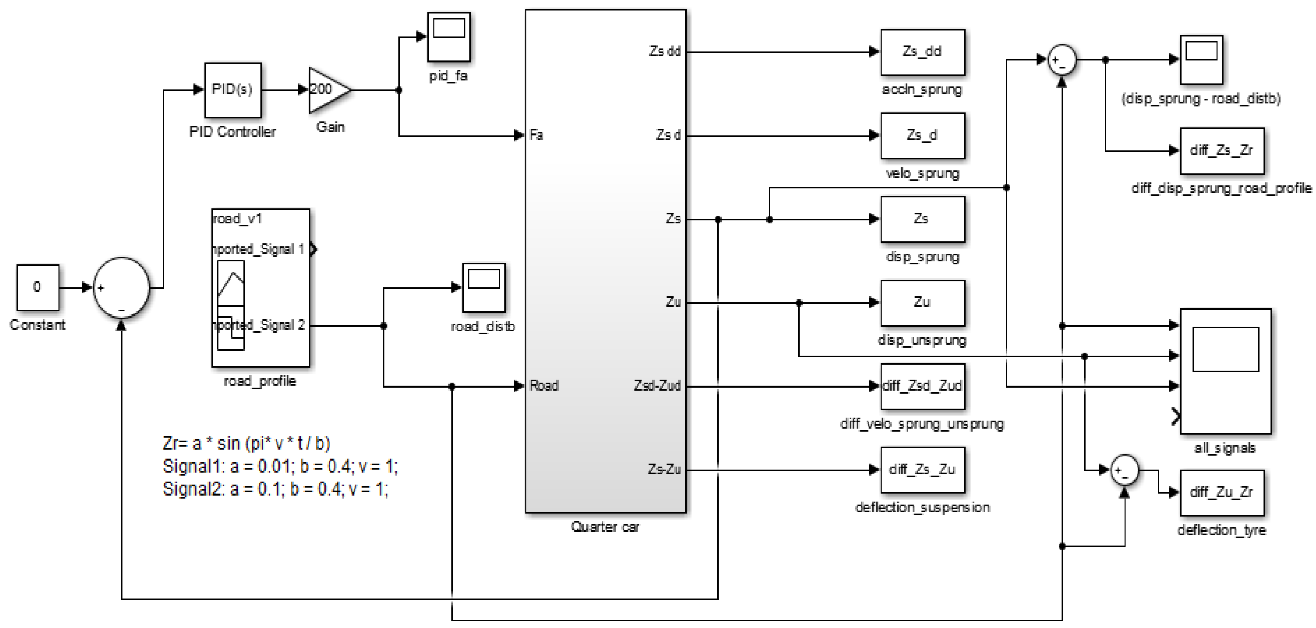


Figure 22: SIMULINK model of an Active suspension system with PID controller and sinusoidal road profile.

The controller parameters and the compensator formula used in the above mentioned SIMULNIK model is shown in the fig

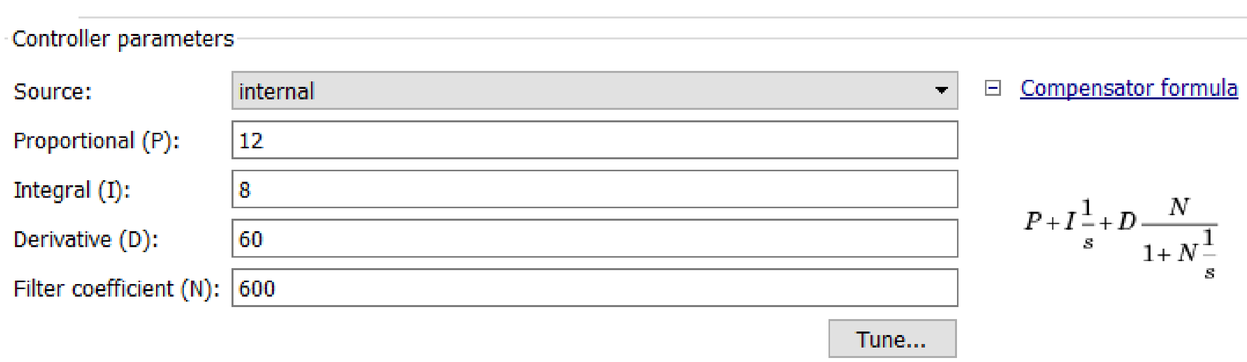


Figure 23: PID control parameters and compensator formula.

6.5 THIRD MODEL OF ACTIVE SUSPENSION SYSTEM USING PID CONTROLLER

This third model is same as the second one except the road profile. In this model the road profile is sinusoidal as the second model but here the slope is higher than the previous one. This road profile is being used to analyze the dynamic behavior of the quarter car in case of a steep bump in real road conditions. The model is analyzed for two road profile input, one with low amplitude and the other with higher than that. The approach velocity of the car towards the bump in this model is considered to be 10m/s.

The equation for the sinusoidal wave representing the road bump is as follows:

$$y = a \ \sin(\omega t)$$

$$\Rightarrow y = a \ \sin(\frac{2\pi v}{b} t)$$

Where,

a = amplitude of the wave (here in this study it is the height of the road bump)

ω = angular velocity of the wave

t = time period

v = linear velocity

b = wavelength (here in this study it is the width of the road bump)

The road profiles used in this simulation model are sine curves, as follows:

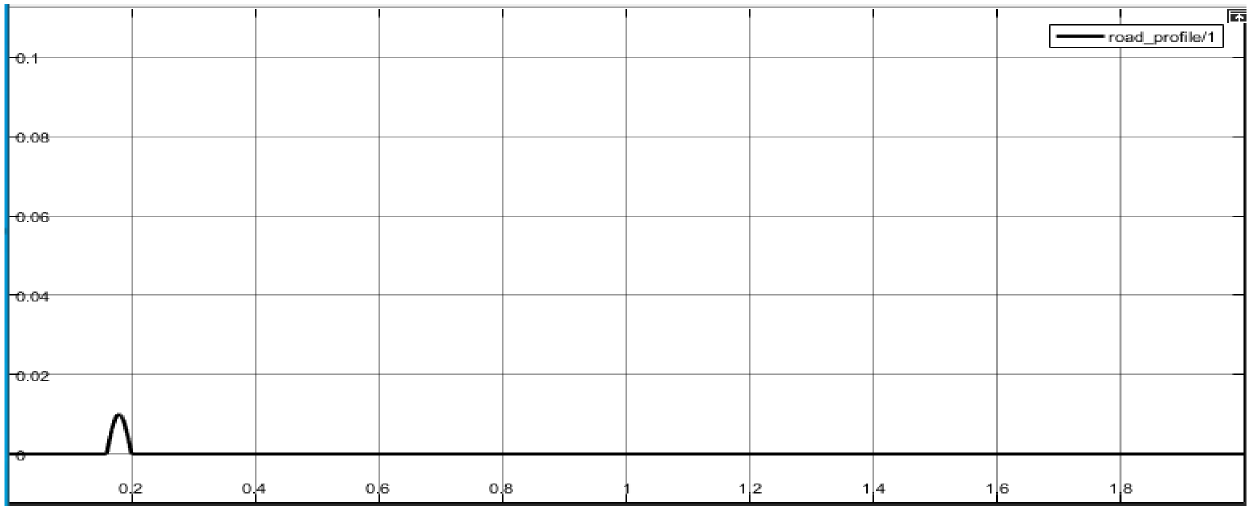


Figure 24: A discrete sinusoidal wave representing bump in real road condition.

For the above mentioned road profile the values of the parameters are as follows:

a = 0.01, b = 0.4 and v = 10

Here "v" is actually the approach velocity of the quarter car model, the value of which we can optimize.

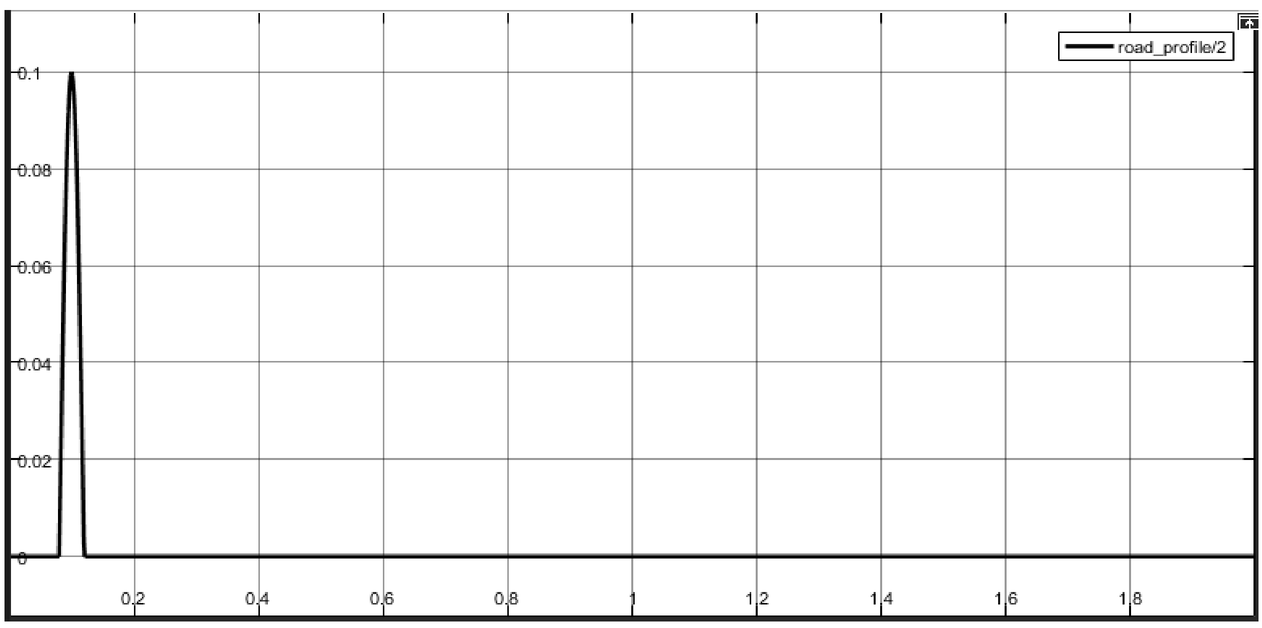


Figure 25: A discrete sinusoidal wave representing bump in real road condition.

For the above mentioned road profile the values of the parameters are as follows:

a = 0.01, b = 0.4 and v = 10
The SIMULINK model is as shown in the figure below

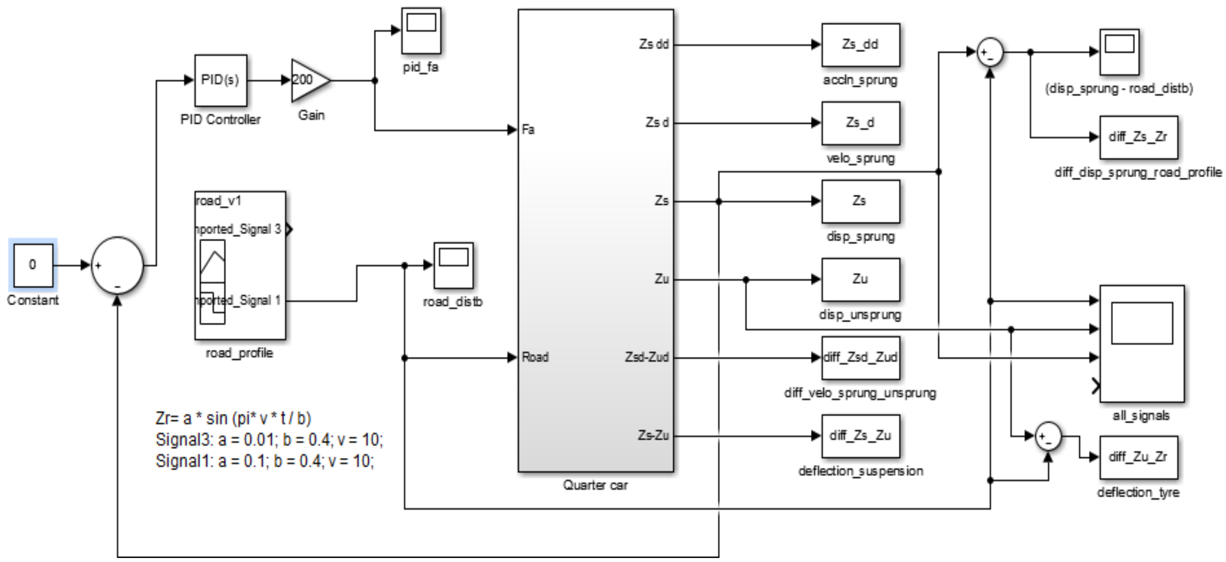


Figure 26: SIMULINK model of an Active suspension system with PID controller and sinusoidal road profile.

The controller parameters and the compensator formula used in the above mentioned SIMULNIK model is shown in the fig

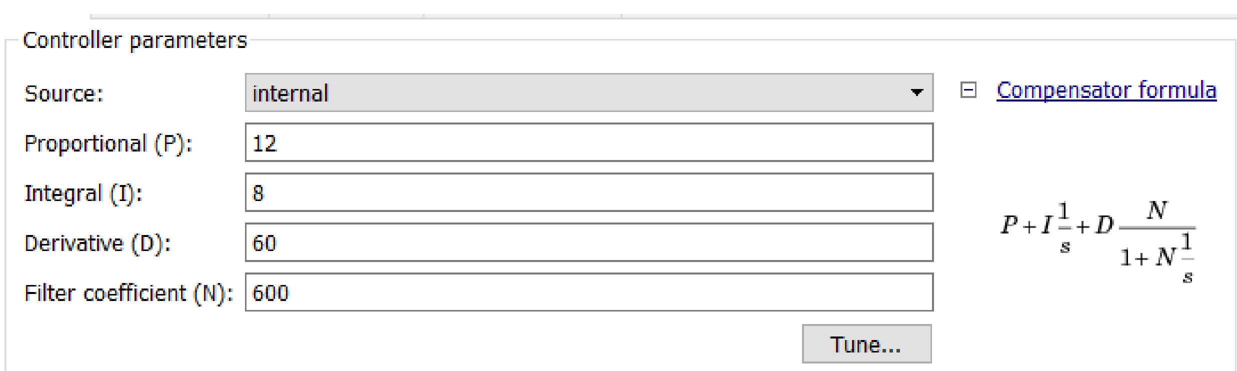


Figure 27: PID control parameters and compensator formula.

6.6 FOURTH MODEL OF ACTIVE SUSPENSION SYSTEM USING PID CONTROLLER

In this SIMULINK model a simulation of an actual hydraulic actuator is being incorporated. To control the vertical dynamics of the quarter car an active force generating element is necessary which a hydraulic actuator is in this case.

The dynamic equations of the hydraulic actuator used in the simulation are as follows as depicted in the chapter 4 already:

$$\frac{V_t}{4\beta}\dot{P}_L = Q_L - C_{tm} P_L - A(\dot{z}_s - \dot{z}_u)$$

$$Q_L = C_d\, w x_v \sqrt{\frac{p_s - sgn(x_v) P_L}{\rho}}$$

$$\dot{x_v} = \frac{1}{r}\left(-x_v + k_v i_v\right)$$

$$F_a = P_L A$$

The SIMULINK models of the above equations are shown in the figures below:

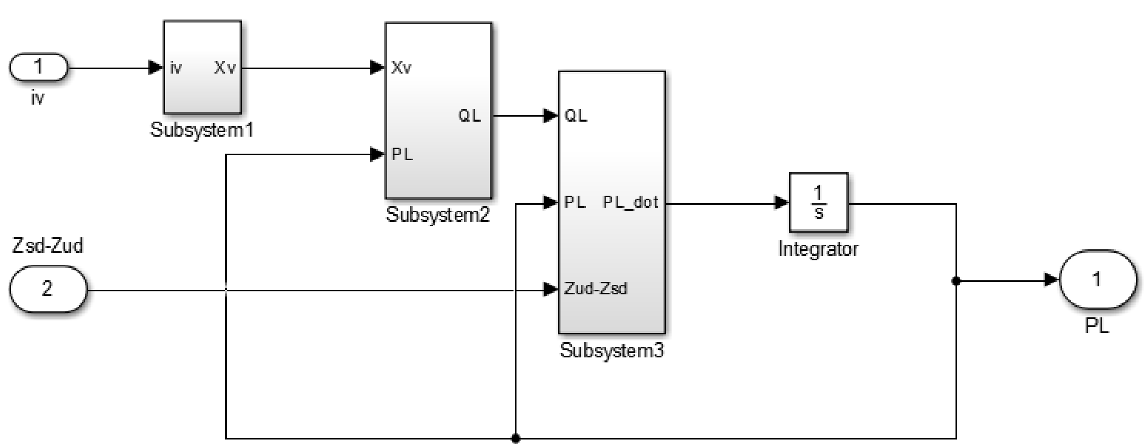


Figure 28: SIMULINK model of Hydraulic Actuator

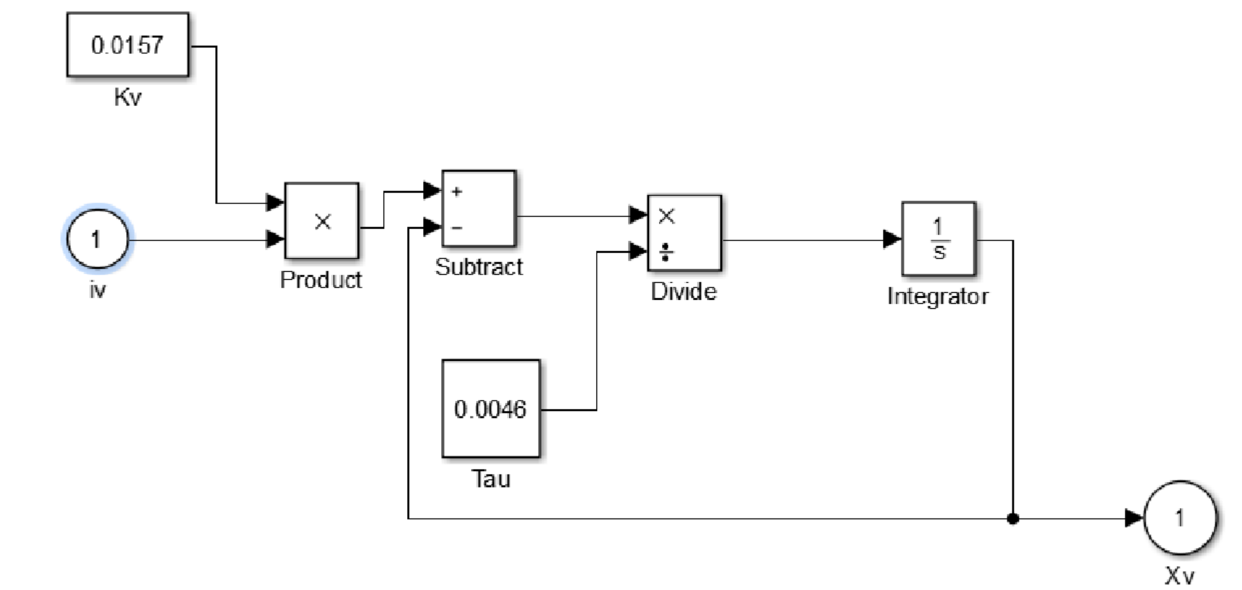


Figure 29: Subsystem1 of Hydraulic Actuator in SIMULINK model

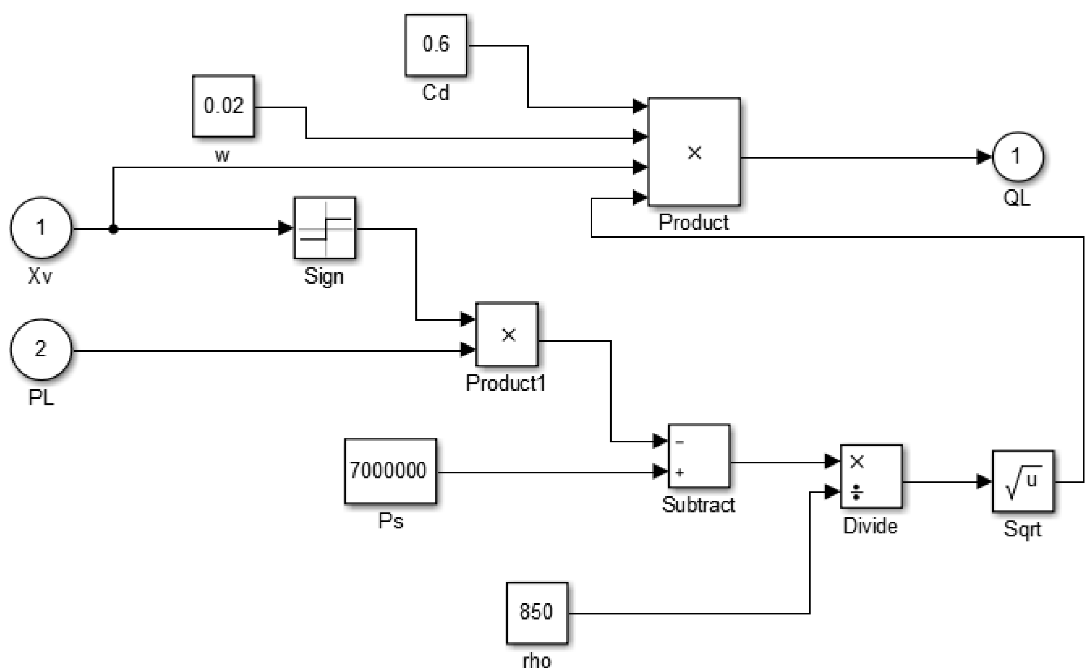


Figure 30: Subsystem2 of Hydraulic Actuator in SIMULINK model

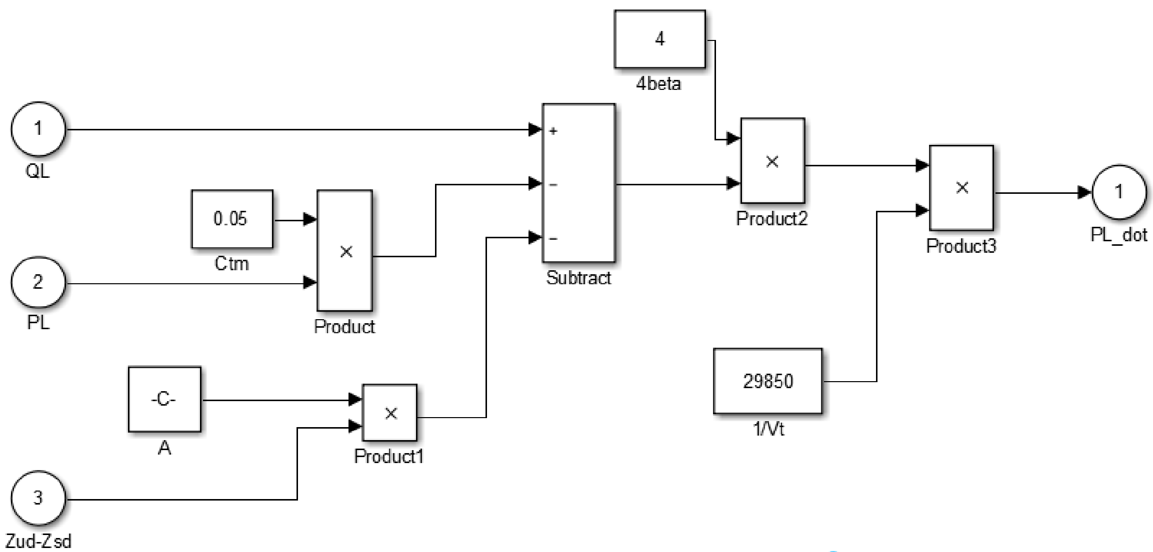


Figure 31: Subsytem3 of Hydraulic Actuator in SIMULINK model.

The SIMULINK models of the quarter along with the hydraulic actuator is shown in figures below,

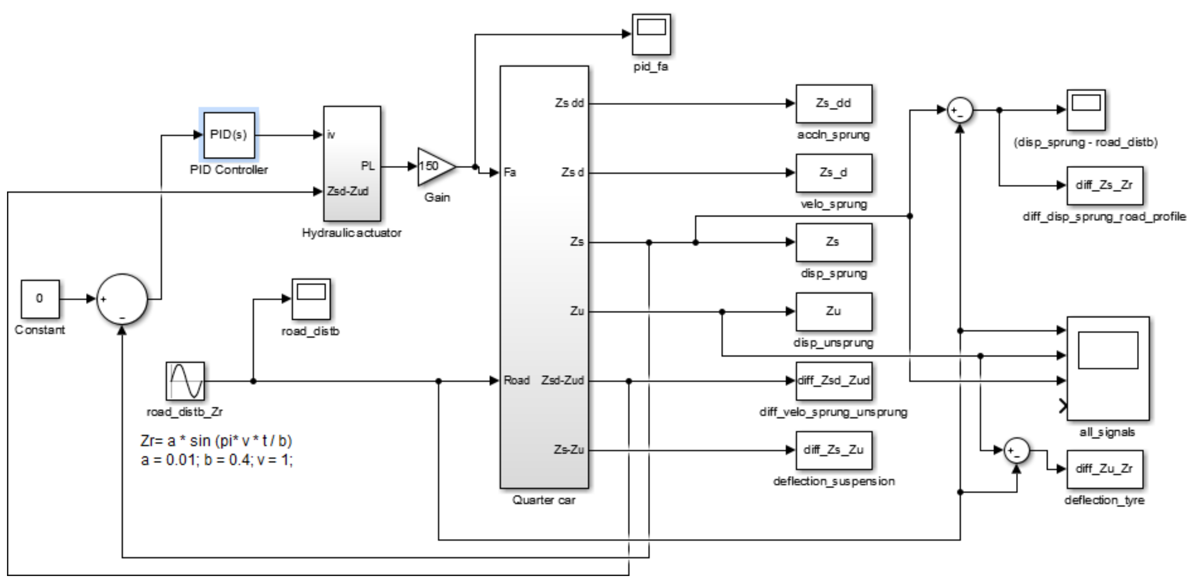


Figure 32: SIMULINK model of an Active suspension system with PID controller and sinusoidal road profile.

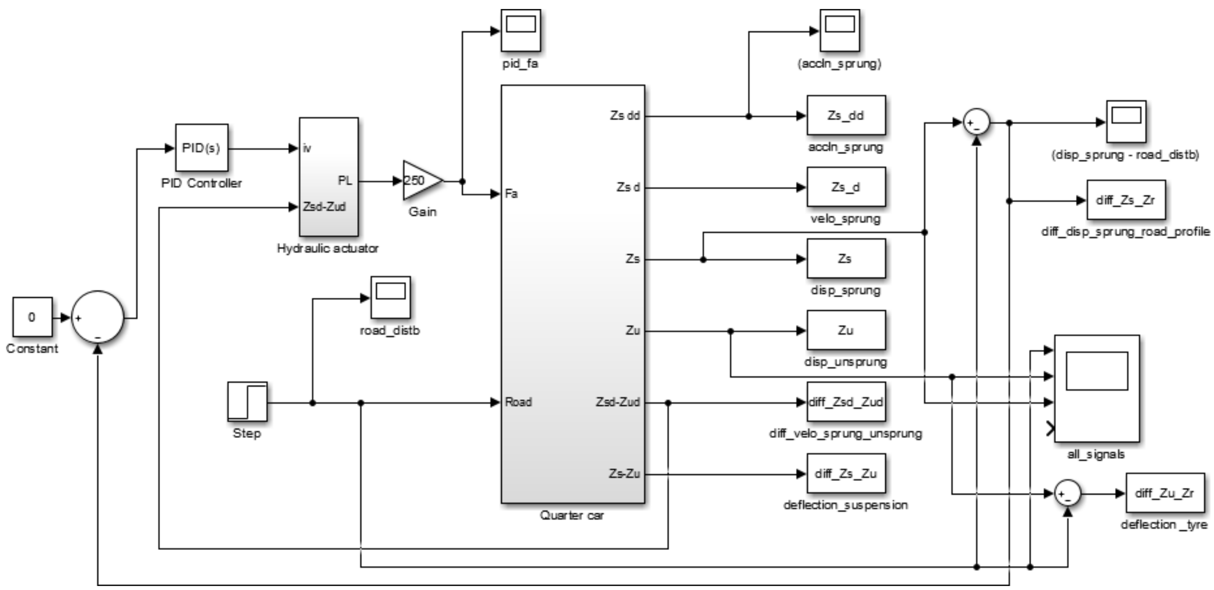


Figure 33: SIMULINK model of an Active suspension system with PID controller and step input as road profile

The controller parameters and the compensator formula used in the above mentioned SIMULNIK model is shown in the fig

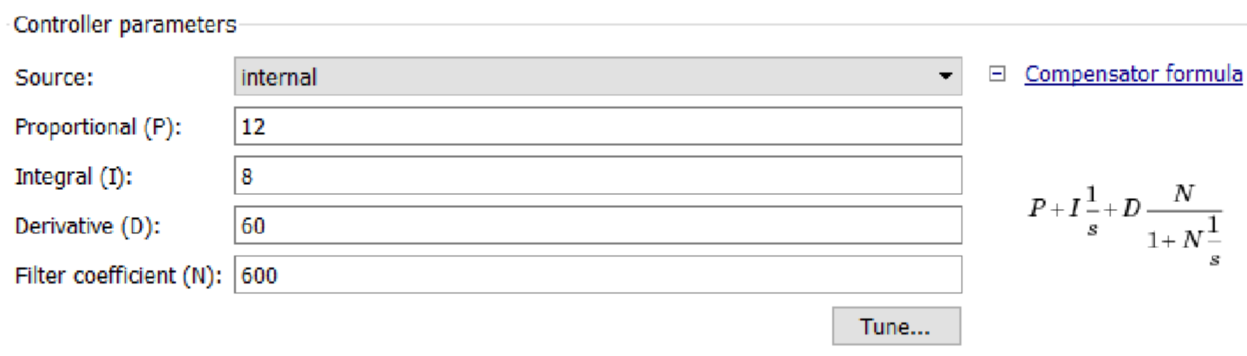


Figure 34: PID control parameters and compensator formula.

CHAPTER –7

RESULT AND DISCUSSION

7.1 GENERAL

Withthe knowledge ofthe mathematical models, simulation schemes, model parameters and PID controller development as described in Chapters 2, 3, 4 and 5, a series of simulation runs were conducted on MATLAB SIMULINK environment. The Quarter Car Model (QCM) simulated with the PID controller developed is optimized first to get the best results based on the requirements of this research. The Optimized Quarter Car Model with its application is hence the proposed work in this thesis. An extensive comparison of results of the proposed model was conducted in the scope of this work. Numerous simulations were conducted for a standard passenger car on irregular road surface conditions like step-up, bumps and wavy roads and they are represented graphically. Also the application of hydraulic actuator to build-up an active suspension system and its comparison with the passive suspension system is represented in this work. The results of these simulations and the necessary details are discussed, in the following sections of work.

The parameters of the quarter car model and hydraulic actuator are listed below:

Parameter	Description	Value
m_s	Sprung mass	290 kg
m_u	Unsprung mass	59 kg
k_s	Spring stiffness of suspension of suspension	16812 N/m
k_t	Spring stiffness of tire	190,000 N/m
c_s	Damping coefficient of the suspension	1000 N/m/sec
V_t	Total actuator volume	$0.89 \times 10^{-3}\ m^3$
β	Effective bulk modulus	$1 \times\ 10^9\ Pa$
C_{tm}	Total leakage coefficient	2×10^{-14}
A	Total actuator ram area	$3.35 \times 10^{-4}\, m^2$
C_d	Discharge coefficient	0.6

w	Spool valve area gradient	0.02 m^2
P_s	Supply pressure	$0.7 \times 10^7\ Pa$
ρ	Fluid mass density	$850\ kg/m^2$
r	Time constant of servo valve	0.0046
k_v	Spool valve gain	0.0157 m/A

7.2 MODEL OPTIMISATION

The passive suspension system based QCM as developed was not optimized for best results and did not have any scalar gains for the inputs of the PID controller. Tuning these values does the necessary purpose of optimization of the PID controller model developed, and yields better results. This was done manually by trial-and-error method. In this process, a particular scalar gain value was designated arbitrarily for a scalar gain variable, and the results analyzed. This process was repeated until the best results were achieved as per the requirements of this study. A multitude of value sets were obtained for the gains for different road conditions, however, the optimal values were selected on the basis of the requirement of this thesis which is nothing but the reduction of the vertical oscillation of the QCM.

7.3 OPTIMISED RESULTS FOR STEP-UP ROAD PROFILE

The passive suspension system and the active suspension system with PID controller models are compared for the step-up road profile input. The results for the vertical oscillations of the QCM for both the cases are compared in the given figure 35 below.

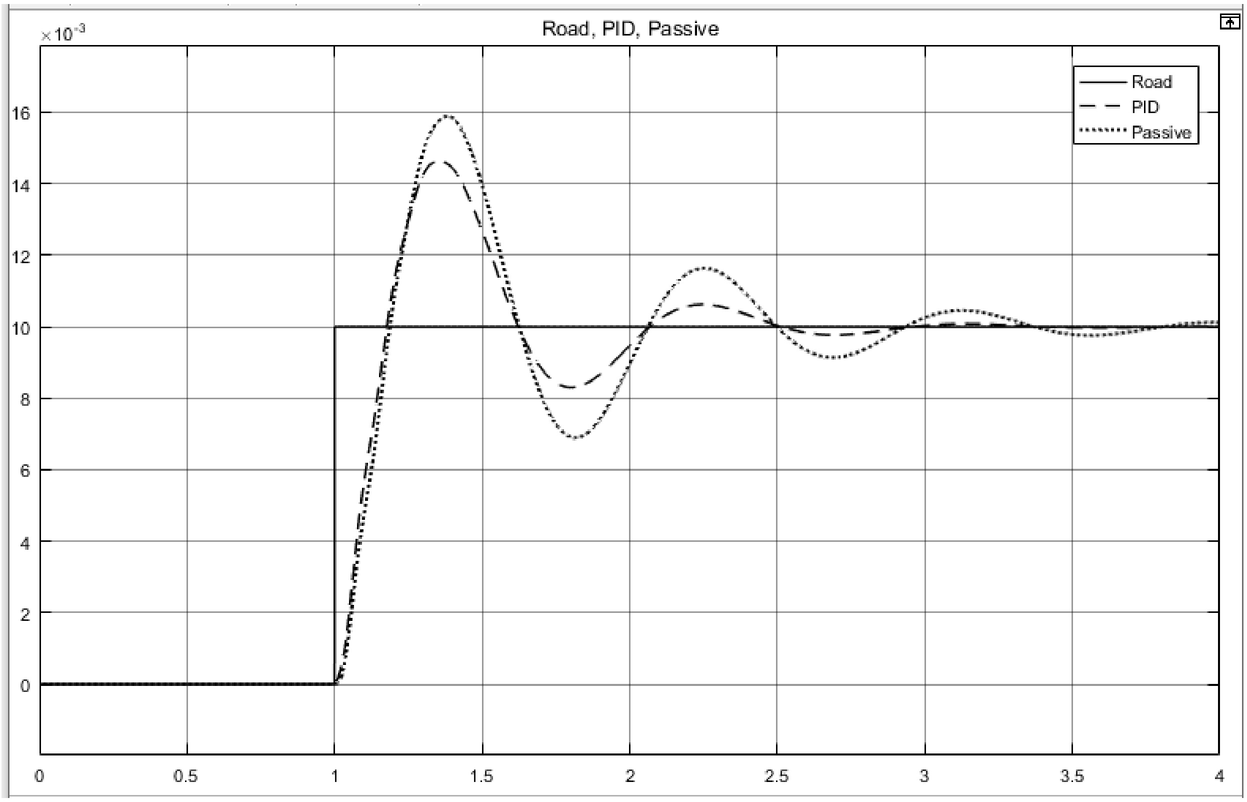


Figure 35: Comparison of passive and active suspension system for vertical displacement of the sprung mass with step input.

7.4 DISCUSSION

In this optimized model we can see that the PID controller has done its job of reducing the vertical vibrations of the QCM. It can be readily observed from the graphical representation of variations in the sprung mass displacements that the optimization operations results in better outputs in terms of reducing the vertical oscillations of the car body. This happens to be our prime objective in terms of vertical displacement control of a car body incorporation of an active suspension system. The optimization facilitates the vertical displacement control with lesser overshooting and settling time, which helps the sprung mass reach equilibrium position faster and hence better comfort is available for a greater period of time of travel in a car.

7.5 OPTIMISED RESULTS FOR SINUSOIDAL ROAD PROFILE

The passive suspension system and the active suspension system with PID controller models are compared for the sinusoidal road profile input. The results for the vertical oscillations of the QCM for both the cases are compared in the given figure below.

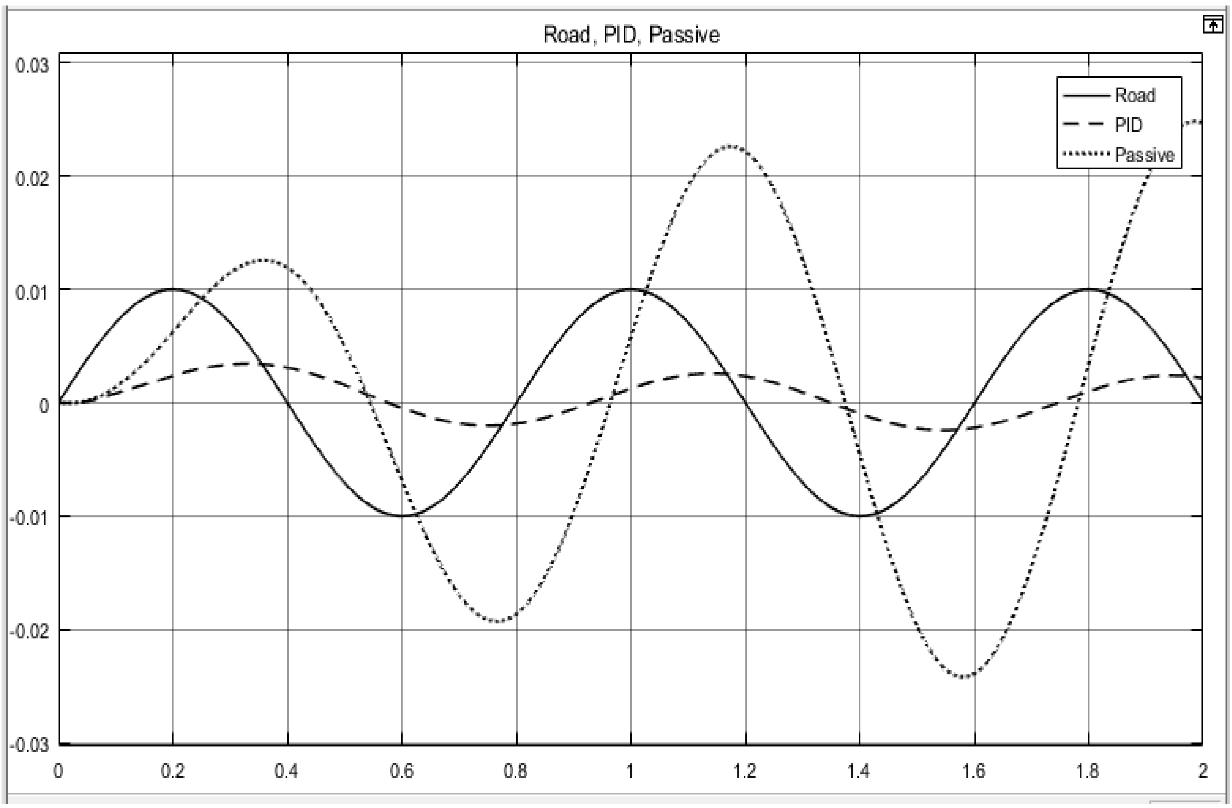


Figure 36: Comparison of passive and active suspension system for vertical displacement of the sprung mass with sinusoidal road profile.

7.6 DISCUSSION

In this optimization also we can notice the same effect of the PID controller as the previous one. Here also the controller has successfully kept the vertical displacements of the sprung muss less as compared to the passive suspension system. As in this simulation model for the whole time period of the simulation run the sinusoidal curve was continuous we cannot make any conclusion on the settling time of the oscillations.

7.7 OPTIMISED RESULTS FOR A LOW AMPLITUDE BUMP IN THE ROAD

The passive suspension system and the active suspension system with PID controller models are compared for a low amplitude bump in the road. The results for the vertical oscillations of the QCM for both the cases are compared in the given figures below.

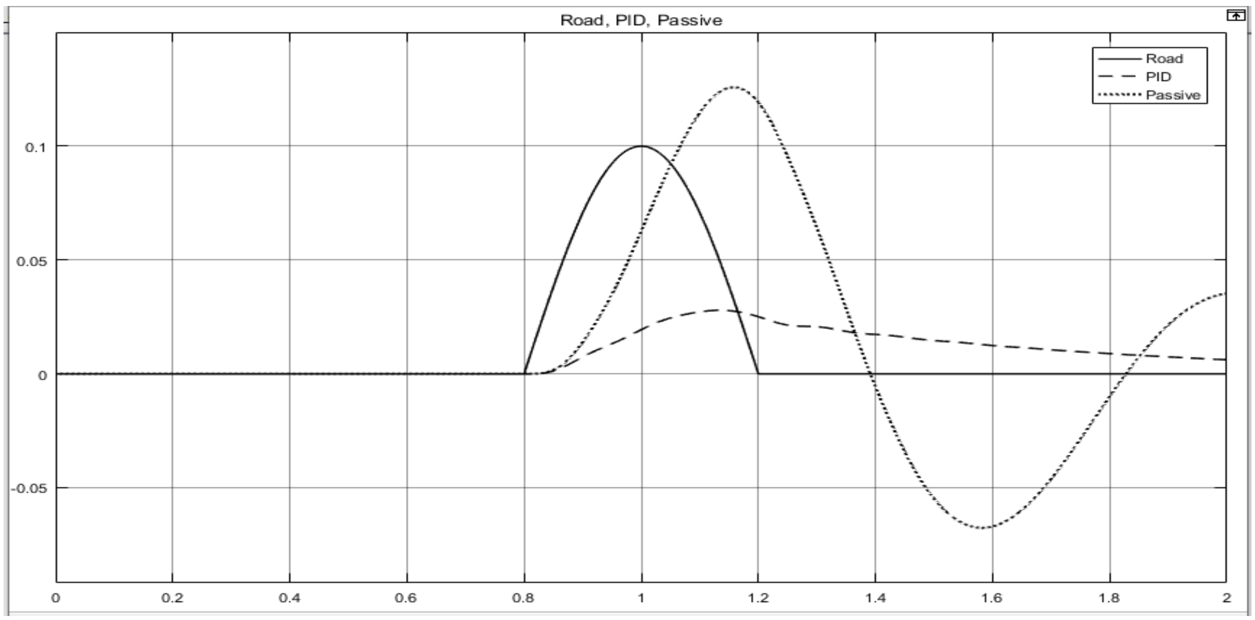


Figure 37: Comparison of passive and active suspension system for vertical displacement of the sprung mass with a low amplitude bump-up road profile.

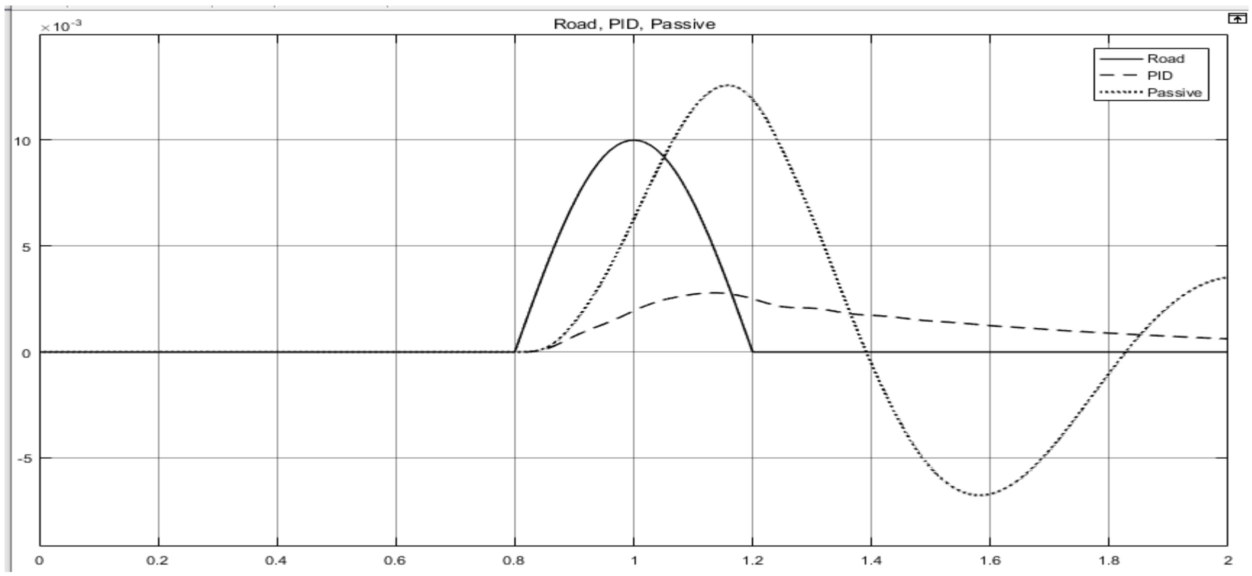


Figure 38: Comparison of passive and active suspension system for vertical displacement of the sprung mass with a low amplitude bump-up road profile.

7.8 DISCUSSION

It can be readily observed from the graphical representation of variations in the sprung mass displacements that the optimization operations results in better control on the reduction of the vertical displacement of the sprung mass. In this part of the model simulation two road profiles are used. One with an amplitude of 0.1m and the other with 0.001m. Here one more input variable is added in the control loop, which is approach velocity of the car. The approach velocity of this SIMULINK model is kept as 1m/s. The optimization facilitates the vertical displacement control with lesser overshooting and settling time, which helps the sprung mass reach equilibrium position faster and hence better comfort is available for a greater period of time of travel in the car.

7.9 OPTIMISED RESULTS FOR A SHARP BUMP IN THE ROAD

The passive suspension system and the active suspension system with PID controller models are compared for a sharp bump in the road. The results for the vertical oscillations of the QCM for both the cases are compared in the given figures below.

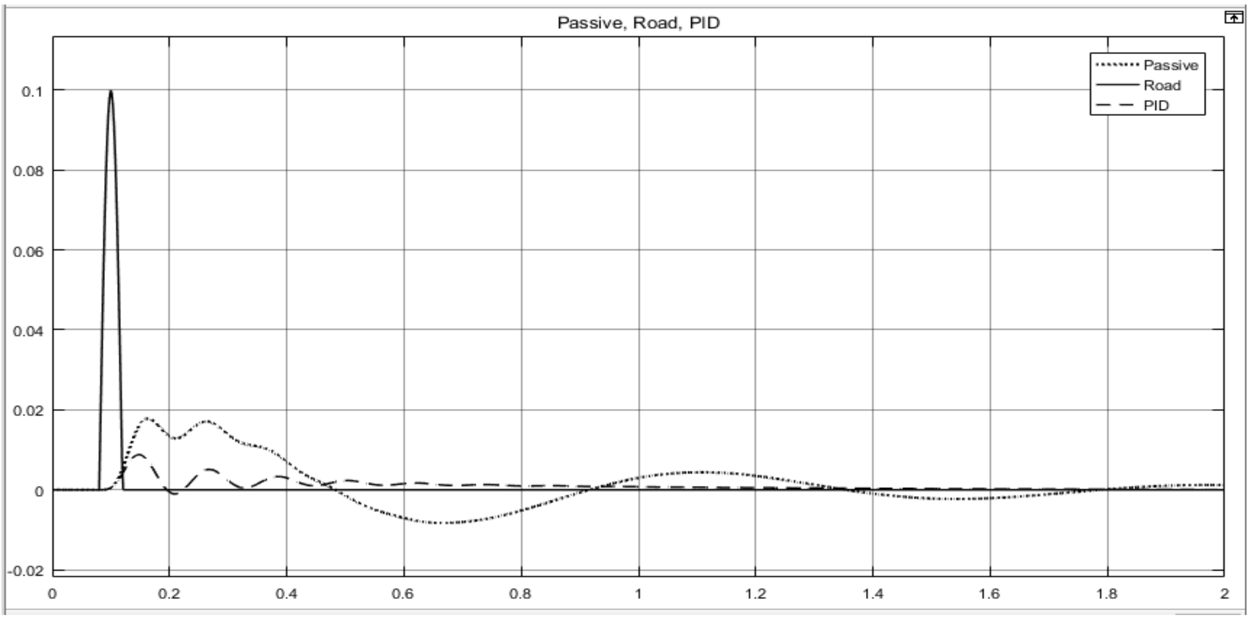


Figure 39: Comparison of passive and active suspension system for vertical displacement of the sprung mass with a sharp bump-up road profile.

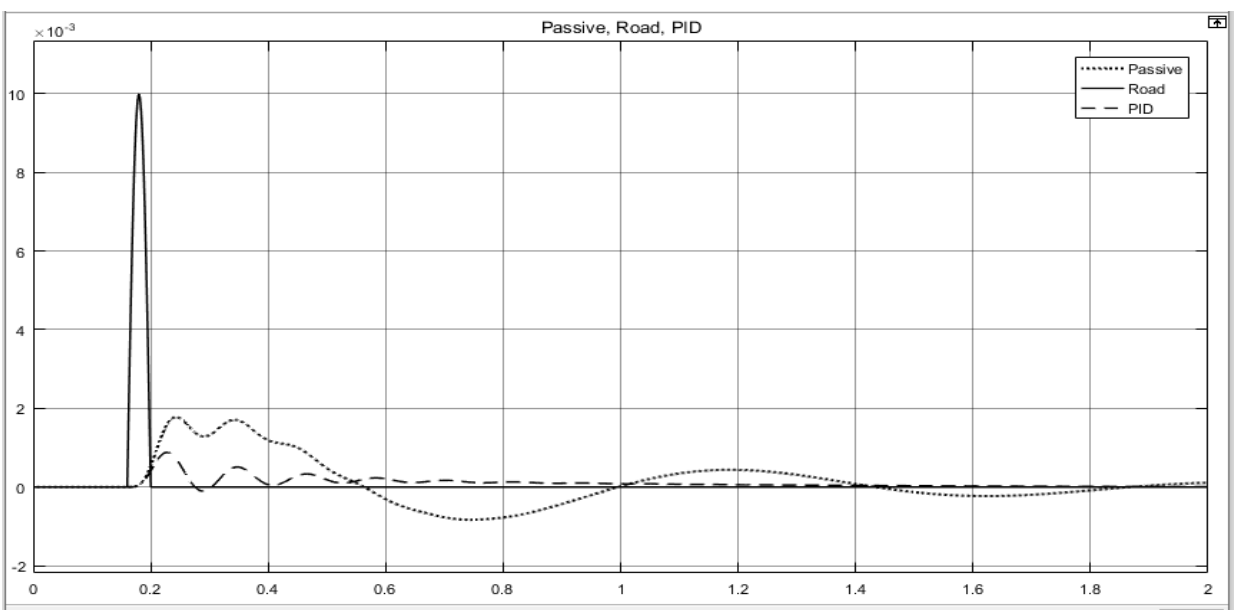


Figure 40: Comparison of passive and active suspension system for vertical displacement of the sprung mass with a sharp bump-up road profile.

7.10 DISCUSSION

The results of this simulation model are almost same as the previous one. Here also the approach velocity of the car model is incorporated in control loop of the simulation model. The approach velocity for this model is considered to be 10m/s. It can be readily observed from the graphical representation of variations in the sprung mass displacements that the optimization operations results in better control on the reduction of the vertical displacement of the sprung mass. The optimization facilitates the vertical displacement control with lesser overshooting and settling time, which helps the sprung mass reach equilibrium position faster and hence better comfort is available for a greater period of time of travel in the car.

7.11 OPTIMISED RESULTS FOR THE FINAL MODEL WITH HYDRAULIC ACTUATOR

Here the passive suspension system and the active suspension system with PID controller models are compared for a step input and sinusoidal road profile. Now instead of a scalar gain block the simulation model of an actual hydraulic actuator as a force generating element is incorporated in this particular model and the output force generated by the actuator is controlled by the PID controller. The results for the vertical oscillations of the QCM for both the cases are compared in the given figures below.

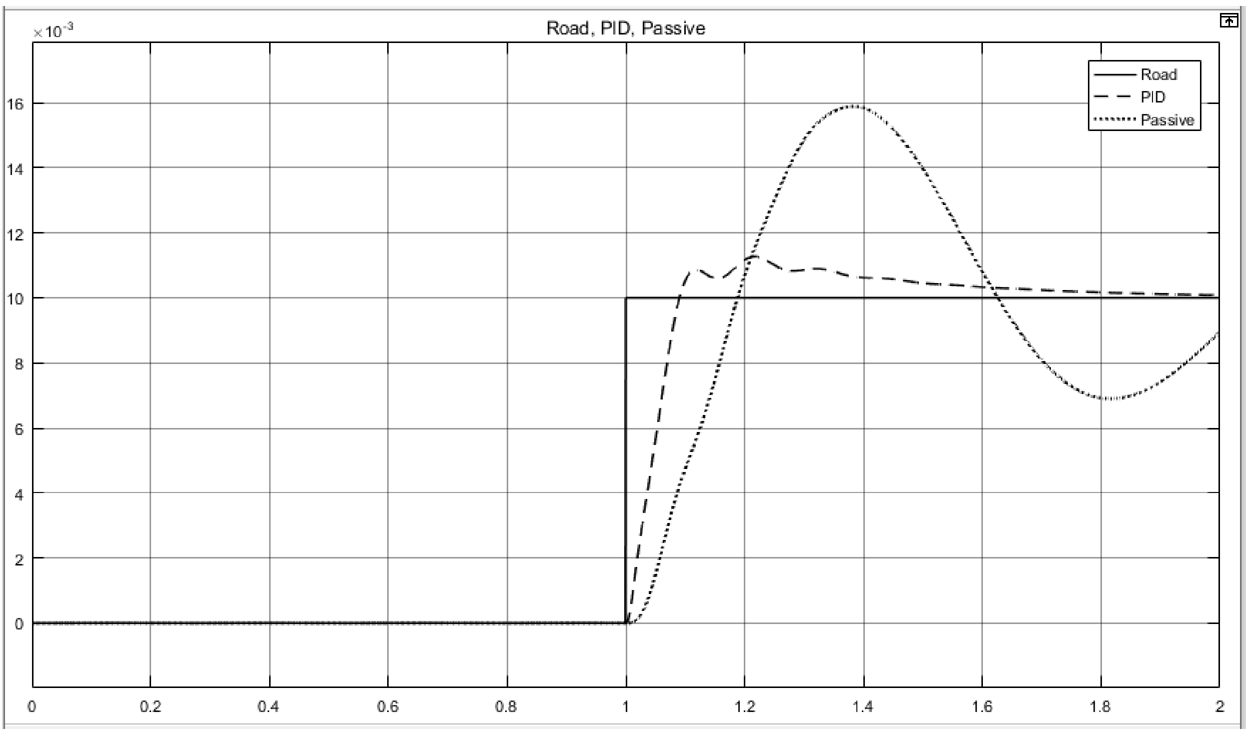


Figure 41: Comparison of passive and active suspension system for vertical displacement of the sprung mass with a low step-up road profile.

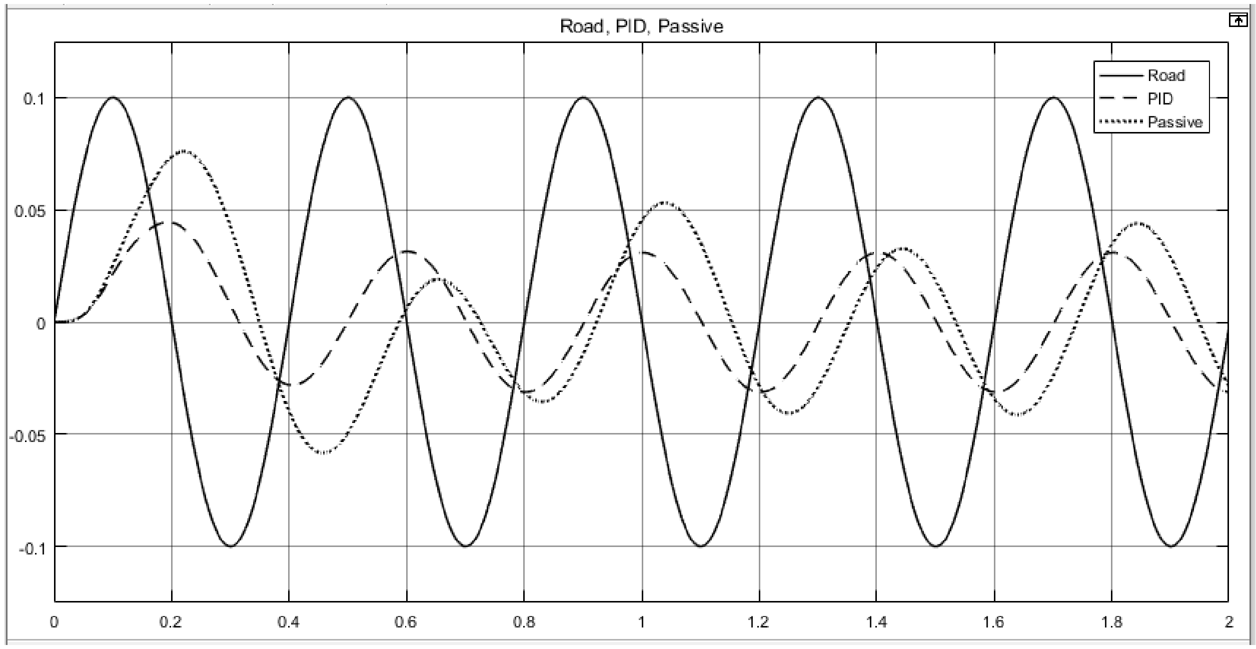


Figure 42: Comparison of passive and active suspension system for vertical displacement of the sprung mass with a sinusoidal road profile.

7.12 DISCUSSION

From the information derived from all the six SIMULINK models without any force generating element it can be concluded that the active suspension system with only PID controller gives a better result if we consider the vertical vibrations of the car body as a parameter to decide the comfort factor while travelling by car. Now in this model we can observe the effectiveness of the hydraulic actuator in the active suspension system SIMULINK model. Here also it must be noted that the hydraulic actuator has successfully kept the vertical oscillations low as compared to the passive suspension system. In case of step input road profile the active system has kept the overshoot almost negligible in comparison to the passive one and also the settling time very less. In case of continuous sinusoidal road profile input we can definitely say that the active system has successfully controlled the overshooting of the sprung mass.

7.13 CONCLUSION

From the comparison of the passive suspension system and active suspension system with hydraulic actuator as a force generating element we can conclude that, for the given configuration the active suspension system performs better as compared to the passive one but both are unable to completely diminish the overshooting of the sprung mass. We can see that the active suspension system cannot eliminate the vertical oscillations corresponding to the road profile completely, but it surely can reduce the overshoot and the settling time. We can surely make the control system more robust to make the suspension system more useful to increase the ride comfort. Also we have used multiple values of the controller parameter to generate active force by the actuator to reach an optimum level of ride comfort so that the sprung mass does not vibrate with high amplitude oscillations and settles the vibratory motion fast.

We have previously discussed that the PID control strategy used in this study is of a primitive level control strategy, so we can use some other advance and more updated control strategies for the suspension system of vehicle. Also the hydraulic actuator used is just one type of force generating element available for practical purposes. There are various different kinds of other actuators also which can be used for active force generation. We can extend this study with multiple loops in the control design along multiple control theories for different input variables.

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