Modeling and analysis of Polyurethane filled automotive coil spring for enhanced damping application

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Thesis Supervisor:

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FEBRUARY, 2021

<u>**Declaration**</u>

I certify that this research work titled "*Modeling and analysis of polyurethane filled automotive coil spring for enhanced damping application"* is my own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources it has been properly acknowledged / referred.

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2017(Fall)-NUST-MECH-203849

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MASTER THESIS WORK

We hereby recommend that the dissertation prepared under our supervision by ABDUL MUIZ (FALL 2017-MS-ME-00000203849).

Titled: Parametric optimization and analysis on robotic based fiber laser welding of HSLA steel by varying Gap width be accepted in partial fulfillment of the requirements for the award of MS Mechanical Engineering degree with (_____________Grade).

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DEDICATION

Dedicated to my exceptional parents and adored siblings whose tremendous support and cooperation led me to this wonderful accomplishment.

Abstract

Ameliorating advancement, innovation and competition urges an automotive industry to reengineer the older items with novel composite materials. Several advancements on Suspension system of a vehicles are being carried out as it renders a feasible gateway for the reduction of vehicle weight to obtain improved efficiency. Helical Spring, being a vital part in the structure of suspension system is engaging the focus of researchers from a long time ago. As composite materials having numerous points of interest over steel materials make it concern full for an automobile industry and research on composite materials for helical springs is holding the researcher's attention from a few past years. This paper breaks down the plausibility of composite material adoption for the helical spring and its feasibility in a suspension system with respect to conventional SSRS (Solid Steel Reference Spring). Purposed composite material Spring, RFS (Rubber Filled Spring), which is comprised of Polyurethane filled inside a hollow steel spring is designed and incorporating SSRS behavior of both springs upon a specified road profile have been studied. Rubber filled spring entailed improved damping along with suitable stiffness, lower weight and acceptable manufacturing process which resulted in mandatory drive comfort, improved road handling and improved efficiency of vehicle.

Key Words Helical spring, QCM (Quarter Car Model), Passive suspension system, RFS (Rubber Filled Spring), HSS (Hollow Steel Spring)

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CHAPTER 1: INTRODUCTION

The research on the feasibility of adoption Rubber Filled Spring instead of Conventional solid steel spring in automobiles has been summarized in two sections in this thesis. First section deals with the selection of optimal diameters for SRS (Solid Rubber Spring) "d_i" and HSS (Hollow Steel Spring) "d_o" fulfilling constraints such as Spring allowable shear stress, deflection and Stiffness. The second section deals with the comparative analysis of both springs in QCM (Quarter Car Model) Simulink to analyze the feasibility of RFS in suspension with characteristics such as acceleration, velocity and displacement for sprung and upsprung masses.

1.1 Background, Scope and Motivation

In an automotive industry from a past few decades' fuel efficiency is a prime concern. Thus, weight reduction provides contented gateway for improvement of vehicles efficiency. As compromise cannot be done on vehicles cheesy, steering system, braking system, and wheel assembly for safety purposes so research on suspension system opened a suitable gateway to decrease the weight of a vehicle as approximately every suspension system involves spring as a common part so spring can be considered as the backbone of suspension system. Hence optimized spring with improved damping and lower weight can not only result in mandatory drive comfort and improved road handling but also in improved efficiency of vehicle. So is the reason research on spring is withholding the attention of researches from a long time ago.

So, the focus of researchers has trended towards optimization of helical spring which is the mostly used spring for countering vehicle suspension system from a few past decades. The research on helical spring is phenomenal. Several methods were adopted for optimization of damping in suspension system. Several parameters have variable effects on Spring, such as Sprung and un sprung mass, stiffness, damping coefficient, load bearing capacity and deflections. Suitable stiffness and Damping coefficient are required for spring where it is being used.

Modern revolutionary era demands creative research works for optimization. The widely technique to improve damping include use of several materials combined known as composite materials instead of adopting steel in helical spring e.g. composites of Epoxy along with Kevlar and E glass , carbon epoxy ,(Glass fiber reinforced plastic)GFRP ,Fiber Reinforced Polymer Composite etc.,[2] Many composites were tested and analyzed numerically and practically to observe their tendency to replace solid steel spring and in many results composite springs resulted improved mechanical behavior than that of conventional steel springs with some drawbacks.

This research breaks down the plausibility of adoption of a Polyurethane and Steel composite spring referred as Rubber Filled Spring RFS. In order to reduce the weight and to improve the damping of spring in a vehicle RFS is designed to install in a passive suspension system instead of SSS as Passive shock absorbers are designed for standard load condition. These give better vibration isolation performance only for the standard load condition [2]. Performance of both springs is analyzed in QCM in Simulink with Specified Road profile as an input. The other types such as active, semi active suspension system are upgraded versions of passive suspension system via some external gadgets So improvement in passive suspension system means improvement in these both too as spring is used as a common material in all the types mentioned.

1.2 Objectives and Major Challenges

The prime objective in research is Introduction of novel Composite Spring and configuring its feasibility in a suspension system whereas other objectives include weight reduction, improvement of damping and withdrawal of results from comparative analysis of RFS and SSS from simulation.

To satisfy the objectives the research countered some challenges such as figuring out optimal diameter “d_i” for SRS (Solid Rubber Spring), optimal “d_o” for HSS (Hollow Steel Spring) and to satisfying constraints such as spring allowable shear stress, deflection

Fundamental Concept

1.3 Areas of Application

Having a basic function to apply, store, control, measure and absorb forces make it versatile and suitable for various application from small scale items such as Brakes, clutches, spring balances, watches, toys, robotics, shock absorbers, vibration reducers in machine foundations etc. to big scale items like in automotive, railways, aircrafts, helicopters, chemical processing equipment's, civil infrastructures like bridges buildings etc.

CHAPTER 2: LITERATURE REVIEW

2.1 Introduction

2.2.1 Spring

Spring has an elastic body which regains its dimensions and shape upon load removal.
Thus, spring can be considered as an elastic member that tend to apply torque, force and can store energy and afterward discharged. Linear, radial and tensile or compressive forces can be produced by spring

2.2.2 Applications of spring

From the past the utilization of springs is unavoidable in countless areas from small scale to big scale machinery due to its effective functions, materials and types.
In general, the material of spring is chosen in such a way that upon removal of load it regains its shape and dimensions. The material of spring is usually alloyed steel, comprising of different ranges of carbon content and high valued yield strength. Spring could be composed of metals and alloys even upon light loads plastics springs are also used. The concept of composite spring is also on a trending note. Compression Spring wire could be composed of widely used materials such as High Carbon steel Spring, Alloy steel, Copper or Nickle based alloys, Phosphorus bronze, Hard drawn, Nonferrous alloys and stainless steel, titanium etc. [1]

2.2.3 Functions of Spring

Springs have various functions in multiple fields of applications but some basic functions are listed below [2]

- To mitigate vibration, Road shocks in car and railway buffers
- Force application in clutches and brakes to bring vehicle to stop
- Storage of energy in toys and clocks
- For spring balance Force measurement
- Motion control of followers and cams via sustaining contact of two mutual elements

2.2 Classification of spring

Classification of springs results in following types.[3]

- Helical Spring
- Leaf Spring
- Spring of conical or volute shape
- Laminated or leaf Spring
- Disc or Belleville Spring
- Spring used for special motives

Out of listed above classification Helical and leaf springs are widely used whereas other types are shown in Figure 2.1.

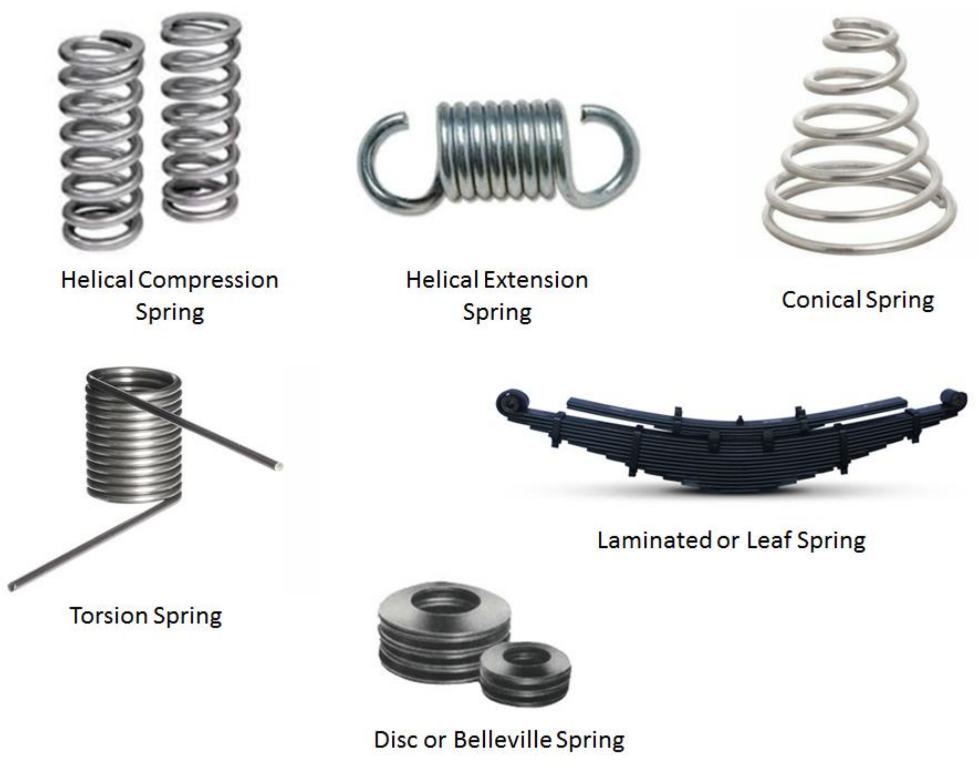


Figure 2.1: Classification of Springs

2.3 Types of Helical spring

Helical Spring encapsulate following types

2.3.1 Tension Helical Spring

To produce load, tension helical spring stretches apart. The distance is small between the successive coils as shown in figure 2.2. The wire is coiled in a series in which the spring elongates in motion by applying load. In a mechanism these are used to restrain their pulling power.

2.3.2 Torsion Spring

This form of helical spring tends to rotate on its own axis as shown in figure 2.2 to generate force. Ends are connected to utilized objects so that if it rotates about the spring center, it tends to thrust the spring to recover its original position. The torsion springs are widely used for the purpose of transmitting torque

2.3.3 Spiral Spring

It has a geometrical shape as shown in Figure 2.2 which constitute of steel band wrapped a number of times around itself. Spiral spring inner end is tied to an arbor and a retaining drum is fixed to the outer end. Upon few rotations it releases power that it stores.

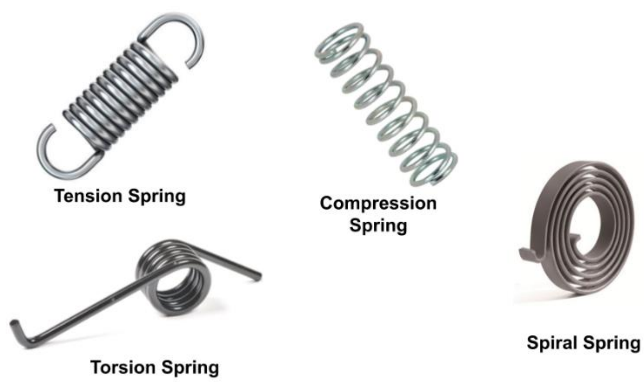


Figure 2.2: Types of Springs

2.4 Helical spring

A wire is coiled in a form of helix design to absorb or eject an energy or force is called helical spring. It is shown in the Figure 2.2

Figure 2.3: Helical compression spring

Several advantages that ameliorate its usage in applications such as Torque or force transfer, energy absorption, reduction or absorption of shock and mitigation of vibrations etc. [AH1] The helical spring which is one of the most widely utilized springs winds up in different industrial and vehicle equipment's, for example, "brakes"," Horns", "balances", "vehicle suspensions" to counter the demanding functions.

2.5 Types of compression spring

Now there are various types of compression springs whereas classification and design consideration are done upon.[4]

- body shape
- Springs forms.

2.6 Compression spring body shapes

Usually, four different body shapes of helical springs exist that are shown in the figure 2.4.

2.6.1 Constant Outside Diameter

The helical spring of constant outside diameter has a shape of Straight Cylindrical with constant or varying pitch. Basically, the distance between centers of adjacent coils of the wire is called pitch. Instead of pitch recommended practice is to state the number of active coils as upon load the behavior of active coil is free towards deflection. Whereas loads and spring constants are also determined by turns of active coil.

2.6.2 Variable outside diameter

The types lying in this category have variable outside diameter with constant and variable pitches. [5]

- Hourglass shaped
- Barrel shaped
- Conical shaped

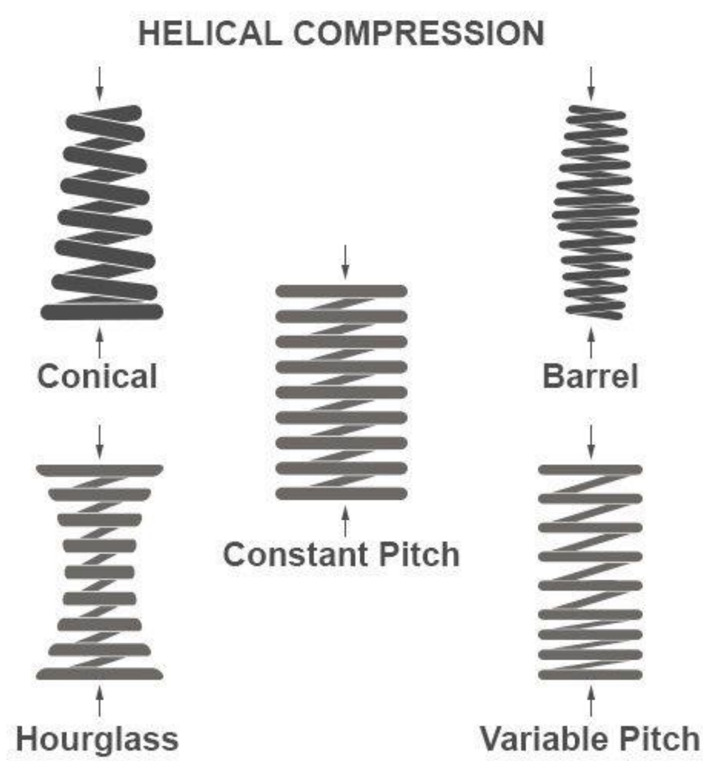


Figure 2.4: Helical compression spring body shapes

2.7 Forms of compression helical Spring

Generally, four forms of compression springs exist Other forms of springs include, cylindrical, non-cylindrical, conical, barrel Hyperboloidal etc. also exists but these are not as common and widely used as the following forms are used. [6]

As compression springs constitutes of various sizes and shapes in addition with variation of their ends. The adjustment in the ends of the spring can tend to ameliorate its position and stability. It is point of concern to understand the pros and cons and application of each end type for correct selection. compression springs can have grounded or cannot have grounded ends as shown in the figure 2.5. Ends are grounded to facilitate the spring for its firm stand up position or to reduce its solid height.

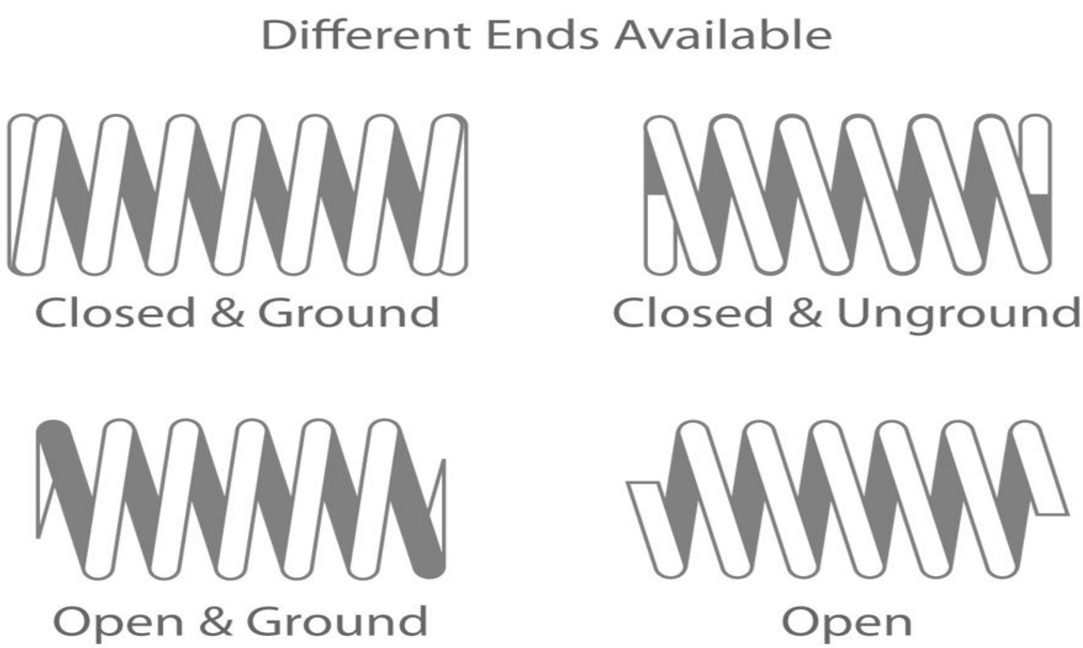


Figure 2.5: Ends available for helical compression spring

2.7.1 Open end/Plain end

In open end or plain end compression helical spring each coil of whole spring consists of constant pitch. It can be found in applications where the need of lower solid height and absence of tolerance of exact height is required However placement of rod or placing it inside in a shaft is necessary for its stability.

2.7.2 Open and ground end/Plain and ground end

The geometry of Open and Ground end /Plain and ground end is same as Open end/Plain end instead of a difference that its ends are grounded to ameliorate its position, storage and stability and to counter buckling.

2.7.3 Close and square end

Withstanding most popularity in all spring end types this spring type provides vertical stability as its end coil closed during manufacturing process. They are economical and widely used as common springs in every application. However, the spring of this type with higher slenderness ration requires a rod inside it or a shaft as an outer cage for it for countering buckling and stability

2.7.4 Close and ground end type/Square and ground end

Rather being second most common in end types of springs its costs is higher because of end grinding process as an additional work during manufacturing. With permittable slenderness ratio it eliminates presence of shaft or rod

The all described ends of spring are shown in the Figure 2.6.

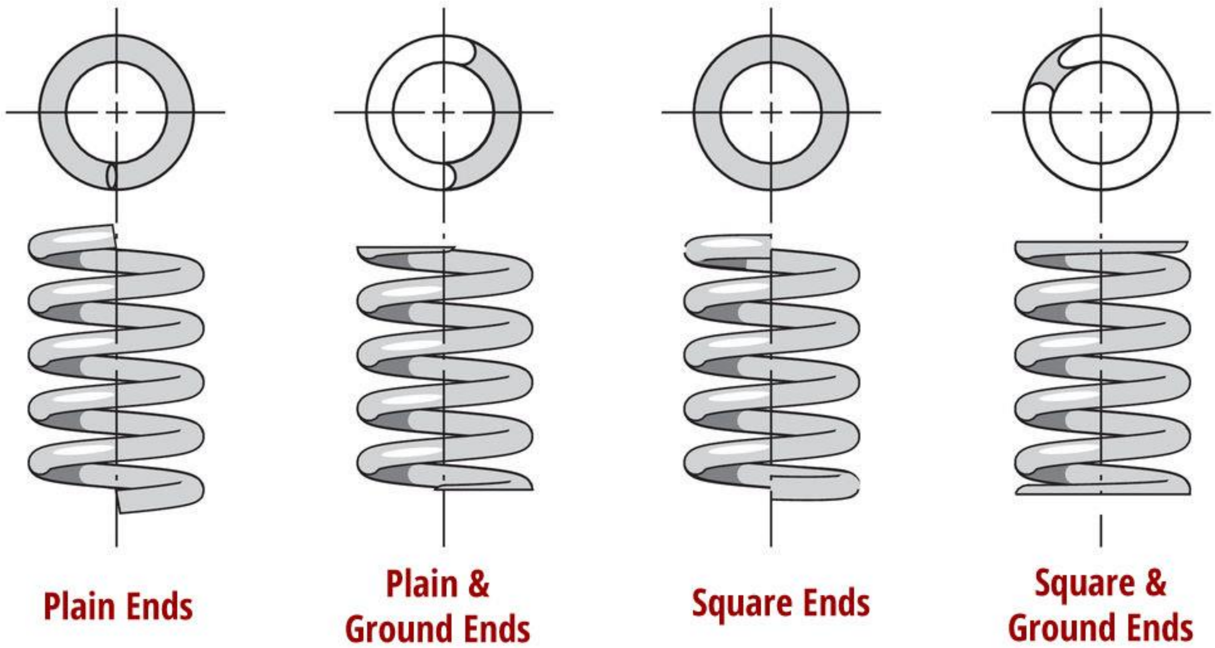


Figure 2.6: Forms of compression helical spring

To counter higher slenderness ratio and stability with effective cost the double closed end springs as shown in figure 2.7 are used.

Figure 2.7: Double closed end spring

Each form has its own class of application. The most common and economical is squared end as due to misalignment they are resistant to buckling and widely used in automotive suspension [7, 8]

2.8 Leaf spring

Leaf spring is additionally called as a semi-elliptical spring because it reflects the shape of calendrical arc form having a rectangular cross section. The position for the axle is given by the middle of the arc while the tie holes are given for connection to the body of the vehicle at either end. To encapsulate strength and rigidity leaves are piled over each other in heavy vehicles. [9] The lea spring is shown in the Figure 2.8.

Figure 2.8: Leaf spring

2.9 Leaf spring vs Helical spring

Arc-shape Leaf springs are used in heavy duty vehicles, mini trucks hauling medium range of load and passenger vehicles, SUVs, and vans because of withstanding high load because of higher stiffness/higher load bearing capacity and simple construction .whereas helical springs are utilized in light vehicles like bikes, domestic cars, passengers cars, trucks etc. having an advantage of performance, ride comfort and better driving stability than leaf spring [10,11,12]

Carrying advantages and disadvantages, out of these springs types helical and leaf springs have been widely utilized in automotive, railways, aircrafts etc. from the past decades. As the automotive industry retains very strong influence and importance and due to its significant global industrial and economic force around the world urges the researcher to tend their heads towards research in automotive industry

2.10 Suspension system

Combination of springs and shock absorber joining the wheels of a vehicle to its body is called suspension system.

2.10.1 Function of suspension system

Smooth ride is provided by vehicle suspension system over rough and tough roads hence ensuring the capability of wheel ground contact and mitigation of vehicle roll.

2.10.2 Parts of suspension system

Suspension system involves three main parts

- Structure holding the weight of a vehicle and determining the geometry of suspension
- Spring for the conversion of kinematic energy into KE energy.
- A mechanical device, shock absorber for KE dissipation

2.11 Classification of automotive suspension system

2.11.1 Passive suspension system

Mostly vehicles use passive suspension system for vehicle dynamics control e.g. roll, pitch and vertical motion etc. As the term indicates passive, the energy supply is absent in that kind of suspension hence. Proper ride comfort needs a specific rate of relative velocities between wheel and body. passive suspension system ensures it via use of some sort of damping material in between them known as hydraulic shock absorber. Damper properties are required to mitigate the vertical acceleration of the body to achieve the comfortable ride but on the other hand vehicles Handling is reduced due to enhanced variation between tire ground contact. Allowable deflection also got limited because of displacement of sprung mass. Enhancement in relative velocity is desirable which demands a high rate or stiffer shock absorber. Upon that stiffness tends to reduce Ride performance and enhance body acceleration at the same time. Hence lacking the better characteristics of ride. The desirability of lower unsprung mass and lower stiffness and appropriate damping ratio for improved control were focused during early design of suspension system. Approaching the values which were optimal, the passive suspension system provided an alluring choice for automotive suspension system enabling it to be widely used. Controlling only displacement of wheel and vehicle body, determined sprung and unsprung velocity limitation the energy supply to suspension system is absent in. conventional shock absorber. Thus, passive suspension optimization varies as road profile varies. The fig 2.8 depicts Passive suspension system [13]

2.11.2 Active suspension system

Gathering information from the attached sensors of vehicle an actuator can regulate the active force through algorithm as compared to the other classes of suspension. Hence this suspension system contains a mechanical spring along with an actuator. AS spring is motion is controlled via an actuator so it can respond to the vertical irregular profile road more smoothly. The actuator controller has various types so upon proper selection of suitable controller the ride comfort and road handling can be improved. Due to its complicated model design and expensive production it is used only by a few cars and trucks.[13] The model is shown in the picture 2.8.

2.11.3 Semi suspension system

In semi suspension system shock absorbers coefficient of viscus damping is varied. The energy addition to the system is absent in this suspension system. Low Cost and lower energy demand make it suitable choice. Simple mechanical design, Lower Powdered, Lower production cost, having high damping force, magneto-rheological (MR) dampers are suitable option of interest. Figure 2 .8 depicts this suspension type [13].

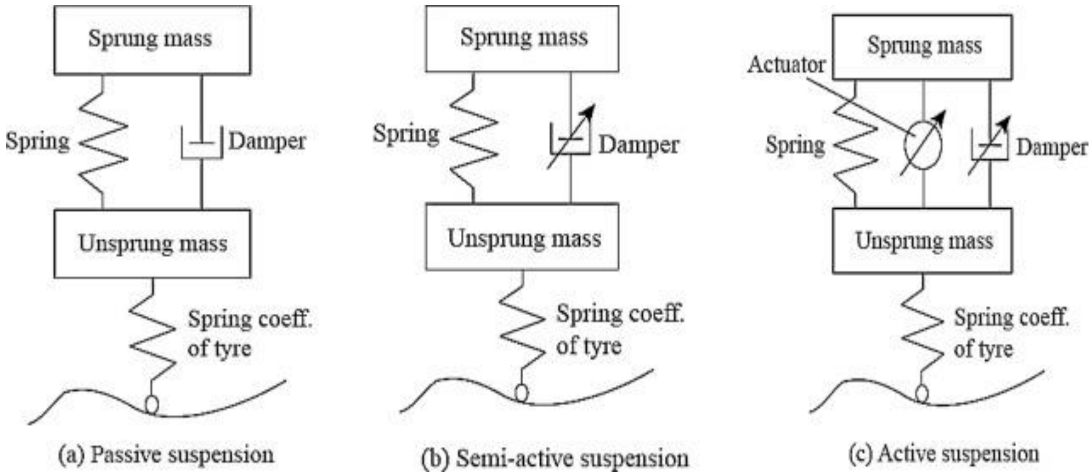


Figure 2.9: Passive, semi active and active suspension system

So, from all the types we figured that spring is a common material thus improvement in spring damping results in improved efficiency of passive, semi and active suspension system.

Now to Improve the efficiency of spring researchers have figured a way via introduction of composite material or turning spring hollow but these approaches have several pros and cons due to which research is still attracting several researcher's attention.

2.12 Literature review on composite spring

Modern revolutionary era demands creative research works for optimization. As composite materials are proving themselves as an alternate and improved characteristics quality than that of conventional materials So, approach of composite spring was brought forwarded by researchers. Instead of adopting conventional steel material for the manufacturing of helical spring, composites of Epoxy along with Kevlar and E glass, carbon epoxy, reinforced plastic of glass fiber and Composite of reinforced fiber were presented in several researches. These composites were tested and analyzed numerically and practically to observe their tendency. Composite springs resulted improved mechanical behavior than that of conventional steel springs and hammered a barriers of costly manufacturing processes, lower stiffness and dimensional constraints hence make it vulnerable choice for heavy vehicles.

Holding prime focus on weight reduction of spring [14] Abdul Budan and T.S.Manjunathathe took Glass, Carbon and their both combined fiber made helical springs and manufactured them via filament winding technique. Manufacturing and experiment analysis accounted carbon fiber spring most befitting option to replace conventional ones because of adequate stiffness and less weight but on the other hand costly manufacturing process and its compatibility with only light vehicles appends to its downsides.

Directing his research on three composite materials, E-glass/Epoxy, Carbon/Epoxy and Kevlar/Epoxy upon static conditions of loading using FEM approach [15] Mehdi Bakhshesh and Majid Bakhshesh done the comparison of composite Spring properties with solid steel spring. Upon three different fiber angles, properties such as weight, deflection, maximum shear stress and Safety factor were studied. Carbon/Epoxy composite depicted least shear stress and higher factor of safety when fiber positioning is done perpendicular to applied loading and E-glass/epoxy composite depicted higher longitudinal displacement than Carbon/Epoxy composite and Solid steel spring.

composite springs have lower weight and high strain energy. Accounting this [16] Md. MUSTHAK and M. MADHAVI carried his research on Carbon/Epoxy composite helical spring. Considering deflection and axial stress as designed constrains of composite spring under 3 different degree of orientation were analyzed on ANSYS. Mechanical behavior of manufactured Carbon Epoxy Composite spring having carbon fiber at 45-degree orientation were observed upon testing and results revealed lower axial deformation for composite spring as compared to solid steel spring.

Dr. Pushpendra Sharma et al. [17] modified suspension system via insertion of parallel combination of two springs as compared to single spring and tested deflection upon loading. Results reveals lower deflection values of modified spring hence providing better suspension system.

Mr Ganesh Jadhav and Prof. Vipin Gawande [18] analyzed feasibility of helical composite spring of Glass fiber epoxy resin. Considering four designs, two solid steel and two composite springs experiments were conducted on them and after analyzing properties like deflection, maximum shear stress, stiffness and weight results revealed composite springs as suitable option but at a sake of lower stiffness and expensive manufacturing making it appropriate for only light weight vehicles.

bulky and heavy spring carry Larger stiffness. So, keeping that in mind, Karthikeyan.S.S et al. [19] made his approached towards metal matrix composite material (Steel, Copper and magnesium composite) to attain lower weight and higher stiffness. Utilizing Ansys simulation composite spring depicted weight reduction up to 60 percent hence making it useful for heavy vehicles to enhance their fuel efficiency but at a sake of higher spring cost.

In order to replace conventional spring, Anil et al. [20] directed his research on composite materials. Steel, Carbon Fiber, Kevlar Fiber springs were designed to investigate the mechanical properties like stiffness, weight carrying capacity etc. Having the same design parameters as solid steel spring and utilizing ANSYS Software the mechanical behavior of springs is studied. Their research reflects that highest and lowest specific modulus belongs to CFRP composite and Kevlar FRP (KFRP) whereas in the characteristics such as deflection and load carrying composite springs were unable to beat solid steel springs because of lower stiffness.

Reviewing manufacturing approaches like, Filament welding method, Resin Transfer molding, laminated composite helical spring for FRP composite helical spring, I Balaguru, et al. [21] focused his target to draw out the effects that were imposed on the composite spring mechanical properties via utilizing braided outer layer and rubber core structures. Encasing them BUR structure overall accounted greater mechanical properties and enhanced failure load capacity upon compression whereas BU structure accounted 6% more than this by 16 % improvement of K.

Taking Composite Springs results in Lower weight and unsprung masses of vehicle into consideration Kirill B. Evseev, et al. [22] Proceeded his research on other features like hysteresis effect and viscoelastic properties, beneficial for having lower vibration and Noise properties. Via conducting tests on servo-hydraulically test bench his results depicted lower dissipation of energy of composite spring than solid steel spring and due to independent frequency independent hysteresis damping the composite springs revealed better properties at higher frequencies. The dissipated energy got its percentage equivalence with the tire damping reflecting the significance of viscoelastic properties for fiber-reinforced composite spring

Some researchers also conducted their research by adoption of hollow springs and composite such as rubber and other metals in the spring of vehicles suspension system.

Advancing research on constrained layer visco-elastic damping Eric R. Ponslet and William O. Miller [23] worked on design development & optimization of tubular coil spring that reveals its compatibility in vaccume along with its fabrication and testing. optimization of designed spring that was going to be used in LIGO (Laser Interferometer Gravity Wave Observatory) project as a prime component was carried out on MATLAB to obtain dimensional parameters. After Fabricating Prototype its thorough testing was carried out to ensure its capabilities in passive seismic isolation stacks. Optimized results and carried tests revealed desired Quality factor value and mechanical properties.

Constraining the parameters like strength, stiffness etc. I. SPINELLA and E. DRAGONI [24] made an efficient Shape memory alloys (SMAs) spring via turning solid spring into hollow spring and provided mathematical design and analysis. Presenting the results of both solid spring and hollow spring the results revealed lower cooling time, mass and heating energy of hollow spring hence enhancing its electrical, thermal and mechanical performance than solid spring

Abhijit Mukhopadhyay, et al. [25] Carried his research on comparison of metal disc springs with Rubber metal spring (combination of rubber washers and metal disc). After Manufacturing testing is carried out in UTM. Lower hysteresis results of rubber metal spring over metal springs hence made them suitable choice for the application of Buffing and draw gear of railway vehicle

The Composite material we used in the research is basically the combination of sarbothane and the conventional steel that is adopted in suspension compression helical springs. The idea imparted in this research is the insertion of polyurethane sarbothane inside a hollow steel tube of spring in such a way that it can satisfy the constraints of the spring that is being replaced by the composite one.

2.13 Sarbothane

Sarbothane is a polyurethane material which is polyether based.

Polyurethane material due to wide countless applications is amazingly versatile. Unlike any other material it creates unique opportunities to resolve problems with higher performance characteristics as polyurethane allows the isolation and manipulation of its mechanical properties via innovative chemistry

Basically, sarbothane is thermoset and polymer having viscoelastic characteristics. Having a higher value of damping coefficient, it holds characteristics such as vibration absorption, damping and isolation introducing it as a very good shock absorbing material. Having a lower rate of creep and longer fatigue life it is found to be better than other polymers such as neoprene, silicon and rubber etc.

In comparison to another polymer sarbothane holds a higher damping coefficient upon a large range of temperature from 160 to -20-degree Fahrenheit. Comparative to other foam products sarbothane can withstand millions of shock cycles efficiently. It reduces the requirement of metal springs after absorbing a shock to restore the device to its equilibrium state. Over the bulk of the working temperature spectrum and ranging frequency from 10-30000 hertz the absorption percentage of sarbothane is above 50.[34]

2.13.1 Applications of sarbothane

Sarbothane is widely used in the field of electronic, industrial, medical and athletic for the isolation of vibration, damping of vibration, shock absorption. Satisfying the need of low cost, it serves its application in colour plastic molding. Other fields like acoustic barriers and absorbers of high space efficiency etc. holds the usage of sarbothane.

So the approach of sarbothane inside the helical steel spring is brought forward in this research and mathematical modeling is carried out to extract the proper volume ratio of steel and rubber to satisfy the constrains of conventional solid steel spring that we took as a reference spring in our case .Thus upon proper selection of inside and outside diameters of rubber filled spring its stiffness and damping coefficient is computed for further simulation in MATLAB Simulink.[34]

2.14 Quarter Car Model (QCM)

The platform of Quarter Car Model (QCM) is brought forward to analyze the vehicle behavior with instability under random road input excitations. In brief QCM is useful for the study of vehicles suspension system. Random variables to considered in a QCM is suspension and tire stiffness, damping coefficient, sprung and unsprung mass.

Assumptions involve in QCM is tire as linear spring with no damping, absence of body and wheels rotational motion, linear behavior of spring and damper, absence of friction and continuous connection of tire with the surface of road.

2.15 Literature review on Quarter Car Model (QCM)

Abdolvahab Agharkakli, et al. [26] compared the common passive PSS and active suspension system ASS having LQR (Linear Quadratic Control technique) by Putting their performance values obtained from QCM on Simulink upon two different road profiles and obtained mathematically designed model for both passive & active suspension types. Results revealed ASS better and comfortable approach than PSS.

Abhijeet S. Taksale [27] constructed an ASS controller for a QCM and tested it on a nonlinear inbuilt Simulink signal as road profile and found out that ASS with improved regulated response leading towards designed a QCM with minimum amplitude and time of oscillation via alternating system constants.

As passenger comfort under vibration is related to parameters like vehicle's settling time and overshoot time Pankaj Sharma, et al. [28] used 2DOF ¼ vehicle model MATLAB programme for passive suspension system to analyze parameters described for a step function. Displacement and acceleration vs time graphs of sprung and unsprung masses revealed higher overshoot leading towards reduced comfortability of passengers. However, these problems can be overcome via active suspension system.

Galal Ali Hassaan [29] examined the vehicle speed and suspension damping effect on acceleration and sprung mass displacement of passive suspension. Using QCM having input as a single circular bump graphs are obtained at vehicle velocity (5- 25km/h) and damping coefficient of (1-15 kNs/m). Considering ISO 2631 Comfort range criteria the limitation on vehicle velocity depending on suspension damping has found to be 6.75 km/h.

Trupti P. Phalke, et al. [30] investigated vehicles performance considering (RC) and road holding (RH). Modeling quarter car model on Simulink having single sinusoidal road profile bump, Relation of RC and RH with Stiffness, damping, sprung mass and vehicle velocity is obtained. Results revealed that RC is better for low damping coefficient values and soft stiffness of spring whereas RH is different at different damping coefficient and better at greater stiffness of spring. Sprung mass depicted appropriate value for RH and RC. Considering RMS standard discussed four cases depicted that passengers are found to be in comfortable zone.

Tejas P. Turakhia, et al. [31] analyzed the ride performance and handling of passive suspension by approaching QCM having a step signal input. Behaviors between variable sprung, unsprung masses and their displacement revealed that for both sprung and unsprung cases variation of mass ameliorate the oscillations. Obtained Transient response table revealed reduced and increased peak for sprung and unsprung mass respectively at variable unsprung and sprung masses.

Ankit Chumbar et al. [32] revealed designing of QCM via two approaches, Matlab & Simulink methods to inspect the damping vs time reaction. linear and nonlinear passive sprung mass Simulation revealed a suitable difference.

Lalitkumar Maikulal Jugulkar, et al. [33] presented optimized suspension system containing fluid damper that shows response with variable force along with two helical springs. Algorithm is designed for stiffness and damping variation to obtain desired performance at variable mass. Results obtained from Simulink QCM model and Practical porotype test revealed 15% higher improvement (acceleration transmissibility and tire discomfort) than the passive one.

So, after rendering the desired portion of literature survey, for comparison purpose we tested our designed spring and reference spring on a specified road profiles upon specific velocities. The simulation was adopted on Simulink MATLAB and results were obtained after running simulation and results were compared for the sake of validity of rubber filled spring with the conventional solid steel spring.

CHAPTER 3: MATHIMATICAL MODELING

3.1 Methodology

Methodology used for the Investigation on the feasibility and comparative analysis of designed Rubber Filled Spring with conventional spring via quarter car model design in a passive suspension system is resolved in two sections

- 1^{st} Section
- 2^{nd} Section

3.1.1 1^{st} Section

1st section deals with

- Configuring the optimal diameters for SRS (Solid Rubber Spring) “d_i” and HSS (Hollow Steel Spring) “d_o”
- Acquiring the desired Variables from tables and satisfying constraints such as spring allowable shear stress, deflection and mass

3.1.2 2^{nd} Section

2^{nd} section deals with

- Deriving equations of motion for designed model.
- Generating code for two types of customized Road Profiles (RP)
- Construction of QCM (Quarter Car Model) in MATLAB Simulink for RP1 and RP2.
- Comparative analysis of both springs

Analyzing characteristics such as acceleration, velocity and displacement for sprung and upsprung masses for reference and designed Spring.

3.2 1st Section

1st section deals with the mathematical modeling of geometrical parameters related to the springs defined in this research.

3.2.1 Mathematical modeling of geometrical parameters

Coming towards the first section first of all geometric properties such as mean diameter, free length, installed height diameters, mass etc. of reference spring which is named as Solid Steel Reference Spring (SSRS) depicted in Figure 3.1 is carried out and listed in a tabular form as shown in the Table 3.1 [35]

The section view displayed in Figure 3.2 is showing the necessary geometric parameters of hollow helical springs which is quite as same as of SSRS spring excluding d_i whereas Spring terminologies for hollow section and solid section Helical Spring is shown in the Figure 3.3.[36]

Figure 3.1: Solid Steel Reference Spring

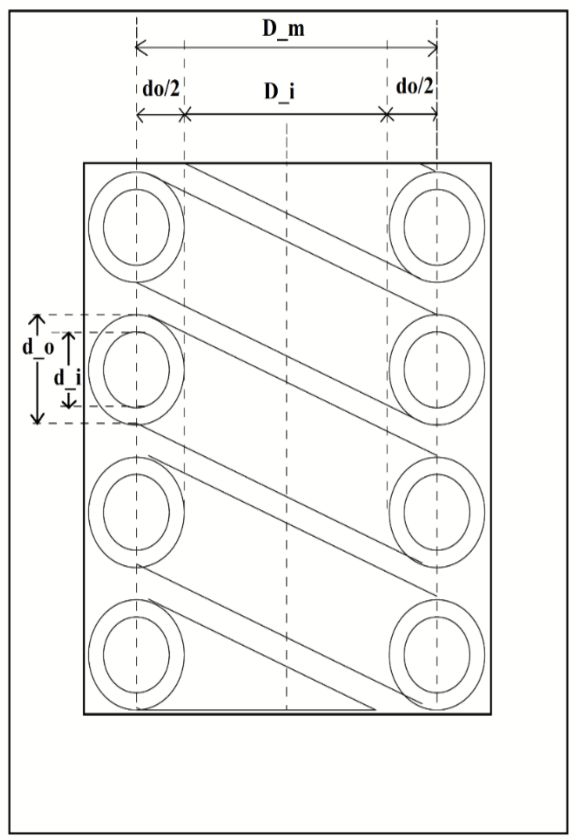


Figure 3.2: Geometrical parameters of Spring

	Descriptions	Values
$\boldsymbol{D_m}$ (m)	**Spring mean Diameter**	**0.1277**
$\boldsymbol{l_f}$ (m)	**Free length of Spring**	**0.3855**
$\boldsymbol{l_i}$ (m)	**Installed Height**	**0.3238**
$\boldsymbol{K_{ssrs}}$ (N/m)	**Solid Steel Reference Spring**	**278451.66**
$\boldsymbol{d_o}$ (m)	**Outer Diameter of the wire**	**0.0254**
$\boldsymbol{d_i}$ (m)	**Inside Diameter of the Spring**	**0.1023**
$\boldsymbol{m_{ssrs}}$ (kg)	**mass of the solid steel reference spring**	**15**

Table 3.1: SSRS geometrical parameters

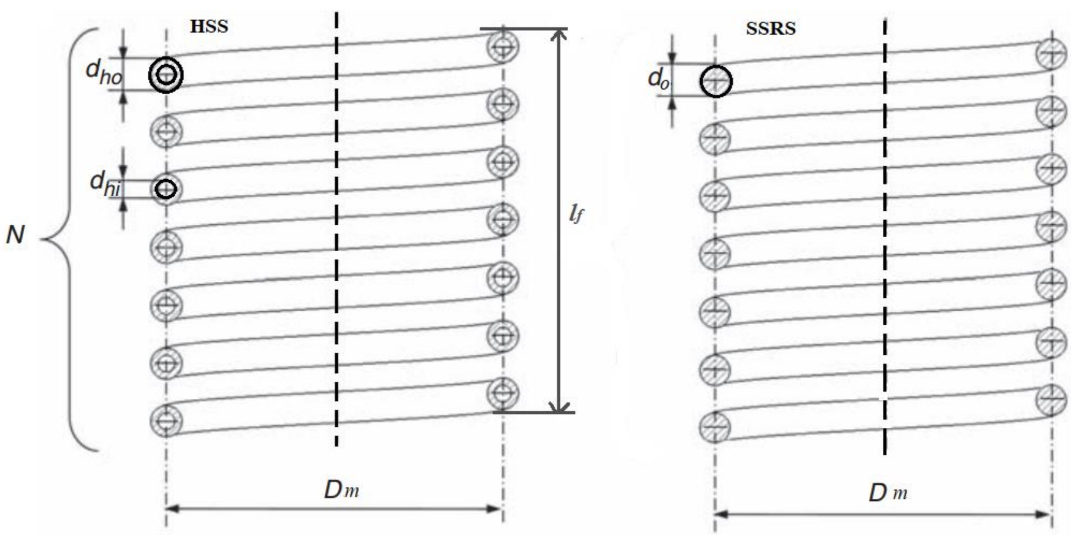


Figure 3.3: Spring terminologies for hollow section and solid section Helical Spring

3.2.2.2 Solid Steel Reference Spring (SSRS) Geometric Properties Equations

SSRS geometric properties are drawn by following equations [36] whereas some vital variables as an input and calculated are shown in the Table 3.2

$\mathrm{N} =$ Number of turns of Spring

$$\mathrm{N} = \frac{G_{steel} * d_o}{8 * K_{ssrs} * C^3}$$

Ϛ(zeta) = Damping Ratio

G_{steel} =Shear modulus of the steel or modulus of rigidity

$\triangle_{max\ \_ssrs}$ Maximum Deflection of Solid steel Reference Spring

$$\triangle_{max\ \_ssrs} = l_f - (d_o * N)$$

μ_{steel} Poison ratio or lateral strain of steel

$C =$ spring index of Solid Steel reference Spring and hollow spring $= \frac{D_m}{d_o}$

$K_{c(ssrs)}$=stress concentration factor for Solid steel reference spring

$K_{c(ssrs)} = \frac{(4C-1)}{4C-4} + \frac{1}{C}\{\frac{(1+2\mu_{steel})+\xi^2(3+2\mu_{steel})}{2(1+\mu_{steel})}\}$ for Solid Steel reference spring ξ =0 [1]

$\tau_{max\ \_ssrs} = \frac{8 * F_{max} * C * K_{c(ssrs)}}{\pi * d^2}$ $\therefore F_{max} = \triangle_{max\ \_ssrs} * K_{ssrs}$ [2]

$\tau_{max\ \_ssrs} =$ Maximum allowable shear stress of solid steel reference spring

Variables	Values
N (turns)	**7.1050**
ϛ(zeta)	**0.3**
$\tau_{max\ \_ssrs}$ (Gpa)	**1.48362**
μ_{steel}	**0.3**
G_{steel} Pa	$\mathbf{79.3 * 10^9}$

Table 3.2: SSRS variables

3.2.2.3 Hollow Steel Spring (HSS) Geometric Properties Equations

$C =$ spring index of Solid Steel reference Spring and hollow spring $= \frac{D_m}{d_o}$

ξ =Void Ratio $= \frac{dh_i}{dh_o}$ For Solid Steel reference spring/for solid rubber spring ξ =0

μ_{steel} Poison ratio or lateral strain of steel 0.3

$K_{c(hss)}$ stress concentration factor for hollow steel spring

$K_{c(hss)} = \frac{(4C-1-\xi^2)}{4C-4} + \frac{1}{C}\{\frac{(1+2\mu_{steel})+\xi^2(3+2\mu_{steel})}{2(1+\mu_{steel})}\}$

K_{hss} =Stiffness of the hollow steel spring

$$K_{hss} = \frac{G_{steel} * d_o(1-\xi^4)}{8 * N * C^3}$$

C_{hss} =Damping of hollow steel spring

$$C_{hss} = 2 * \xi\sqrt{m * K_{ss}}$$

Figure 3.4: Hollow Steel Spring

$\tau_{max\ \_hss}$ =Shear stress of hollow steel Spring

$$\tau_{max\ \_hss} = \frac{8 * F_{max\ \_hss} * C * K_{c(hss)}}{\pi * d^2 (1 - \xi^4)}$$

Hollow Steel Spring (HSS) is shown in the Figure 3.4

3.2.2.4 Solid Rubber Spring (SRS) Geometric Properties Equations

SRS properties are drawn by following equations whereas some vital input variables are listed in the Table 3.3 whereas SRS is depicted in Figure 3.5 geometric

Variables	***Values***
G′ (Mpa)	***0.6205***
G″ (Mpa)	***0.3102***
$\boldsymbol{\eta}$	***0.50***
$\boldsymbol{\mu_{rubber}}$	***0.48***

Table 3.3: SRS variables

Figure 3.5: Solid Rubber Spring

η=Etta $= \frac{G''}{G'}$ [37]

G' = storage modulus, here denoted as G_{rubber} [37]

G''= loss modulus

μ_{rubber} Poison ratio or lateral strain of rubber [38]

$K_{c(srs)}$ stress concentration factor for Solid rubber spring

$K_{c(srs)} = \frac{(4C_s-1)}{4C_s-4} + \frac{1}{C_s}\{\frac{(1+2\mu_{steel})+\xi^2(3+2\mu_{steel})}{2(1+\mu_{steel})}\}$ for Solid rubber spring $\xi = 0$

K_{srs} =stiffness of the solid rubber spring

$$K_{srs} = \frac{G_{rubber} * {d_i}^4}{8 * N * {D_m}^3}$$

C_{srs} =Damping of the solid rubber spring

$$C_{srs} = \eta\sqrt{m * K_{srs}}$$

τ_{max_srs} =Shear stress of Solid Rubber Spring

$$F_{max\_srs} = \triangle_{max\_ssrs} * K_{srs}$$

$$\tau_{max\_srs} = \frac{8 * F_{max\_srs} * C_s * K_{c(srs)}}{\pi * {d_i}^2}$$

3.2.2.4 Rubber Filled Spring (SRS) Geometric Properties Equations

Rubber Filled Spring (RFS) is shown in the Figure 3.5

$C_s =$ spring index of the RFS $= \frac{D_m}{dh_o}$

$C_{(rfs)} =$ Damping of rubber filled Spring

$C_{(rfs)} = C_{srs} + C_{hss}$

$K_{(rfs)} =$ Stiffness of rubber filled Spring

Figure 3.6: Rubber Filled Spring

$K_{(rfs)} = K_{srs} + K_{hss}$ [39]

$\tau_{max\ (rfs)}$ =Shear stress of Rubber filled Spring

$\tau_{max\ (rfs)} = \tau_{max\ \_hss} - \left(\frac{v_f}{f}\right) \tau_{max\ \_srs}$

$v_f = \left(\frac{d_i}{d_o}\right)^2$ f = fitting factor

$\triangle_{actual\ \_rfs}$ Actual Deflection of the RFS

$\triangle_{actual\_rfs} = \frac{F_{max}}{K_{(rfs)}}$

Figure 3.5 summarized all the spring types upon which mathematical equations for geometric parameters were derived

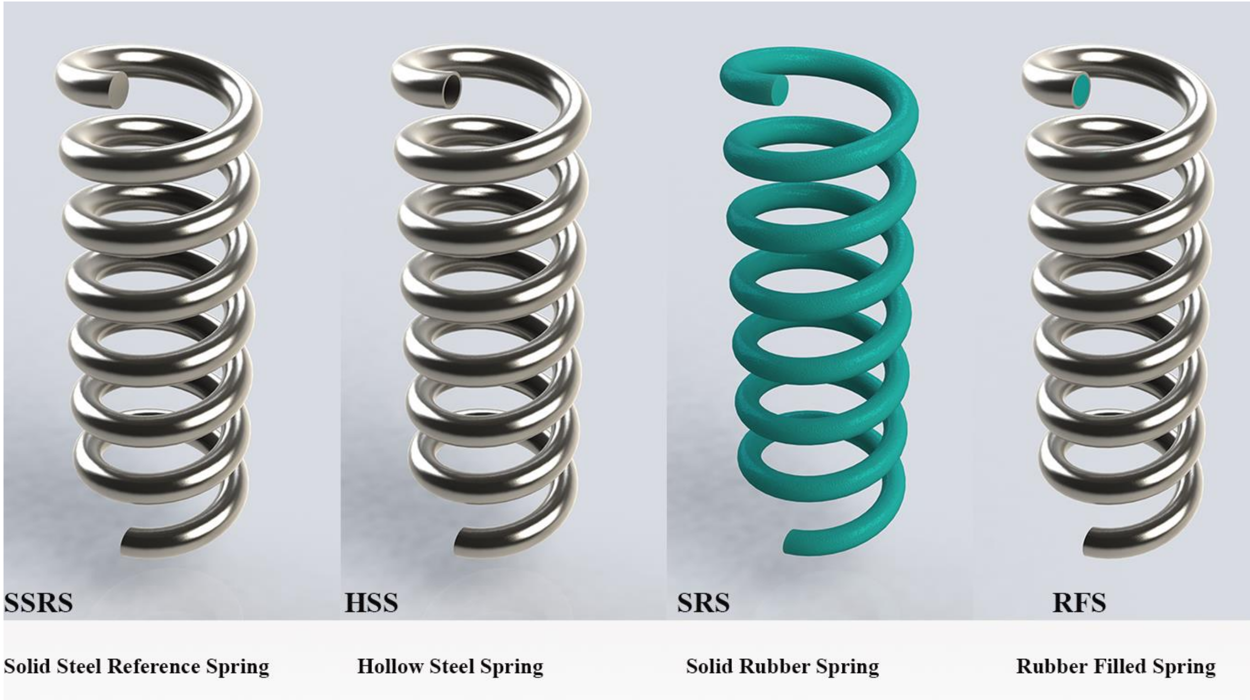


Figure 3.7: SSRS/HSS/SRS/RFS

3.2.2 Input and Output Variables

To obtain the suitable d_i (internal diameter of wire)and d_o (Outer diameter of the wire) 6 Cases were setup having d_o ranging from 0.0254m-*0.0304m with an internal diameter of the wire ranging from* $d_o - 0.001m$ *for every case* .

In the first scenario for Case_1 d_o *,0.0254m is set to fix and d_i range is taken from 0 - 0.0244m with difference of* $0.001m$ *thus providing 1000* $1000\ d_i$ *values.*

Same criteria have been carried out for all other remaining 5 cases as shown in Table 3.4.

Cases	D_0
Case 0	**0.0254**
Case 1	**0.0264**
Case 2	**0.0274**
Case 3	**0.0284**
Case 4	**0.0294**
Case 5	**0.0304**

Table 3.4: Wire outer diameter (d_o) Cases

d_{int}	C_{rfs}	C_{ssrs}	$\tau_{max\ \_rfs}$ GPa	$\tau_{max\ \_ssrs}$ GPa	K_{rfs} N/m	K_{ssrs} N/m	$\triangle_{actual\ \_rfs}$	$\triangle_{max\ \_rfs}$
		1.325891641e+04		**1.483623**		**278647.15**		

Table 3.5: Obtained table

Encoding the listed equations on MATLAB the required tables as shown in Table 3.5 are obtained which are ordered in the pattern as shown in table

Thus from 6 Cases 6 tables are obtained and the rows are selected which satisfied the following constrains

$\tau_{max\ \_rfs} \leq \tau_{max\ \_ssrs}$

$\triangle_{actual\ \_rfs} \leq \triangle_{max\ \_rfs}$

Case_0 violated this constraint so from Case_1 to Case_5 Crfs, Krfs and d _i values are obtained from table.

After obtaining d_i from every Case in the next scenario d_o is varied

$d_{new} = d_o + \triangle_d$

$\triangle_d = 0.001m$

Encoding the listed equations on MATLAB the required tables are obtained which are ordered in the pattern as shown in Table 3.6.

$\mathbf{d_{int}}$	$\mathbf{C_{rfs}}$	$\mathbf{C_{ssrs}}$	$\boldsymbol{\tau}_{\mathbf{max\ \_rfs}}$ **GPa**	$\boldsymbol{\tau}_{\mathbf{max\ \_ssrs}}$ **GPa**	$\triangle_{\mathbf{actual\_rfs}}$	$\triangle_{\mathbf{max\_rfs}}$
		13258.91		**1.483623**		

Table 3.6: Obtained table Pattern

In this scenario the constrained that had to be fulfilled are listed below

$m_{rfs} \leq m_{ssrs}$

$\tau_{max\ \_rfs} \leq \tau_{max\ \_ssrs}$

$\triangle_{actual\_rfs} \leq \triangle_{max\_rfs}$

$$m_{rfs} = \left[3701(d_i^2) + \left\{(1 - V_f)24211\left(d_{o\_new}\right)^2\right\}\right]$$

$$m_{ssrs} = 24211\left(d_{o\_ssrs}^2\right)$$

3.3 2nd Section

2nd section deals with the derivation of the equations of motion for designed model, generation of the code for two types of customized Road Profiles (RP), construction of QCM (Quarter Car Model) in MATLAB Simulink for RP1 and RP2.

3.3.1 Mathematical modeling QCM.

Mathematical modeling of quarter car model involves its physical diagram to derive its equation of motion and constructing a layout for the customized signal of road profile

3.3.2 parameters for mathematical model

To analyze the common performance parameters involved in vertical vehicle dynamics upon road excitation Passive QCM with two degree of freedoms is used. It contains Sprung, Unsprung masses and suspension, comprising of damper and spring. Two degree of freedoms depicts two mass motions. Sprung and unsprung ones. Parameters involved in the physical model of quarter car model as our input and output variables are listed below

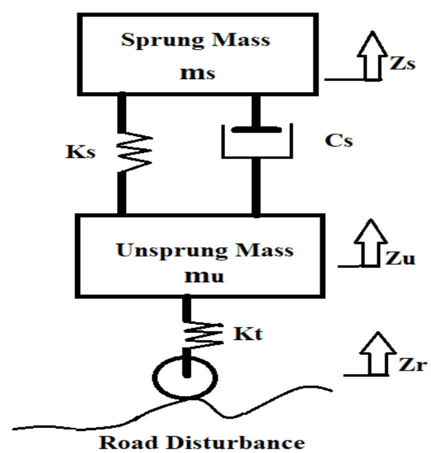


Figure 3.8: Physical Model of the Quarter Car

m_s=Body or sprung weight

m_u=Unsprung Mass

c_s =Damping coefficient for suspension

z_s= displacement of Vehicles body

$\dot{z_s}$=Vehicles body Velocity

$\ddot{z_s}$= acceleration of vehicles body

k_s=Stiffness of the Sprung mass

g=Gravitational constant

z_u= displacement of the Unsprung mass

$\dot{z_u}$= Velocity for Unsprung mass

$\ddot{z_u}$= acceleration for unsprung mass

RP=Customized Road Profile Signal

k_t=Stiffness of the Tire

The Equations of motions derived from schematic of passive QCM under vertical plane are

$$m_s\ddot{z_s} + c_s(\dot{z_s} - \dot{z_u}) + k_s(z_s - z_u) + m_s g = 0$$

$$\ddot{z_s} = \frac{-c_s}{m_s}(\dot{z_s} - \dot{z_u}) - \frac{K_s}{m_s}(z_s - z_u) - \frac{m_s}{m_s}g$$

$$m_u\ddot{z_u} - c_s(\dot{z_s} - \dot{z_u}) - k_s(z_s - z_u) + k_t(z_u - z_r) + m_u g = 0$$

$$\ddot{z_u} = \frac{-c_s}{m_s}(\dot{z_s} - \dot{z_u}) - \frac{K_s}{m_s}(z_s - z_u) - \frac{K_t}{m_u}(z_u - z_r) - \frac{m_u}{m_u}v$$

3.3.3 Input and output simulation parameters of RFS and SSRS

Input parameters are listed in the below Table 3.7

Parameters	Descriptions	Values
M_s (Kg)	**Vehicle Body weight**	**1350**
M_u (Kg)	**Vehicle unsprung weight**	**335**
K_t (KN/m)	**Tire Spring constant**	**760**
K_s (KN/m)	**RFS Spring constant**	**278**
C_s (KNs/m)	**Damping Coefficient of RFS**	**132**
K_s (KN/m)	**RFS Spring constant**	**345**
C_s (KNs/m)	**Damping Coefficient of RFS**	**202**

Table 3.7: Input Parameters

Whereas output parameters are

$\ddot{z}_s$=Road Comfort /Passenger Comfort / Body acceleration/Sprung Mass acceleration

$\dot{z}_s = \text{Body velocity, Sprung mass velocity}$

$z_s = \text{Body displacement, Sprung mass displacement}$

$z_s - z_u$ = Suspension Deflection

$z_u - z_r = \text{Tire Deflection , Road holding (RH)}$

3.3.4 Road Profile Signal (RP1, RP2)

In the research two road profiles are used one to counter general behavior whereas other to observe the vehicle response at different velocities

3.3.4.1 Road Profile Signal RP1

Customized ground excitation signal RP1 has been given to QCM where this road profile signal acts as input of the tire thus transferring this signal to unsprung and sprung mass respectively through suspension. The RP1 signal [40] is shown in the Figure 3.7.

$$\mathrm{RP1} = \begin{cases} \dfrac{a(1-\cos(8\pi t))}{2} \end{cases}$$

if $5.50 \leq t \leq 5.75 \quad a = 11\mathrm{cm} \quad$ and

$8.00 \leq t \leq 8.25 \;\; a = 5.5\mathrm{cm}$

0 otherwise [6-8]

$a = \text{maximum amplitudeof bump}$

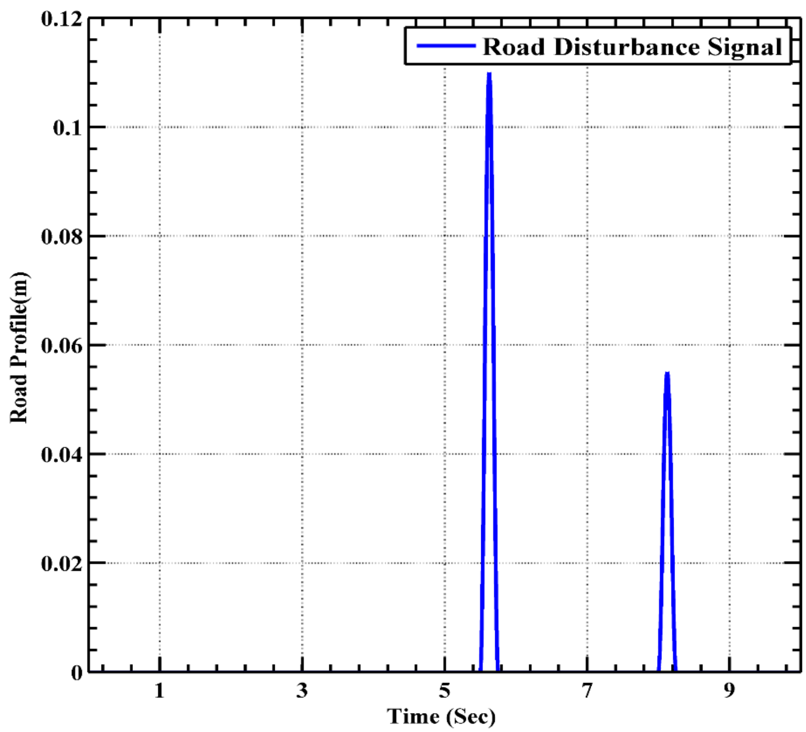


Figure 3.9: Road Profile (RP1) for QCM

3.3.4.2 Road Profile Signal RP2

The RP2 [41] is a circular hump road profile signal as it is common out of parabolic and flat topped bumps.

Height (Y): 100 mm

Length (L): 5.2 m

$y = Y\sin(\omega t)$ for $0 \leq t \leq T$

Velocities and desired output parameters that were taken into account in this RP2 profile QCM are

- 20 km/hr.
- 45 km/hr.
- 90 km/hr.

$\ddot{z}_s$=Road Comfort /Passenger Comfort / Body acceleration/Sprung Mass acceleration

$\dot{z}_s$ = Body velocity, Sprung mass velocity

z_s = Body displacement, Sprung mass displacement

z_u = Wheel Displacement

$z_u - z_r$ = Tire Deflection , Road holding (RH)

3.3.4 Simulink QCM for passive suspension system (RP1)

Constructed QCM Simulink MATLAB block diagram for passive suspension system upon RP1 is shown in the Figure 3.7.

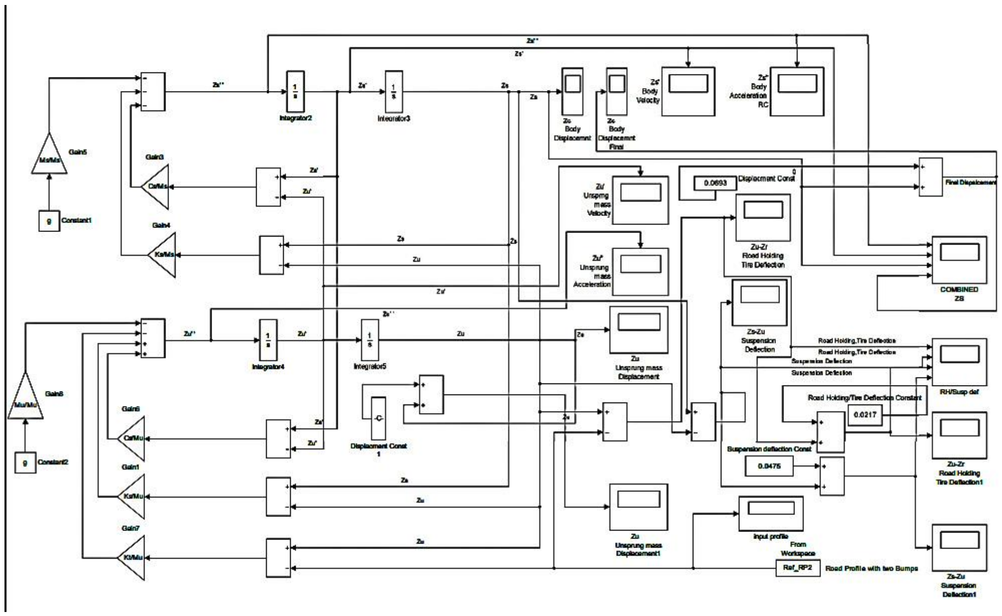


Figure 3.10: Passive suspension system QCM (RP1)

3.3.5 Simulink QCM for passive suspension system (RP2)

Constructed QCM Simulink MATLAB block diagram for passive suspension system upon RP2 is shown in the Figure 3.8.

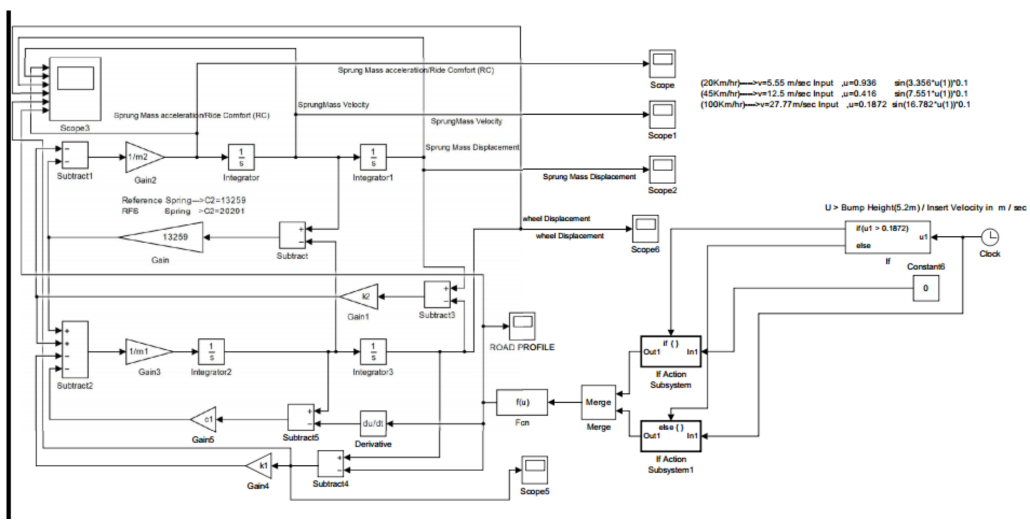


Figure 3.11: Passive suspension system QCM (RP2)

CHAPTER 4: RESULTS & DISCUSSION

4.1 Introduction

This chapter surrounds the review of obtained data based on mathematical modeling and simulation. This chapter withholds the suitable diametric selection from the obtained table upon constraints and revealed simulation results from MATLAB Simulink QCM.

4.2 Results and Discussion

4.2.1 Stopping Criteria for RFS

Now as the obtained values in the tables revealed increased damping values and decreased deflection values respectively so intersection point on the graph as shown in fig provided the suitable required case which was case3 which is highlighted in the Table 4.1.

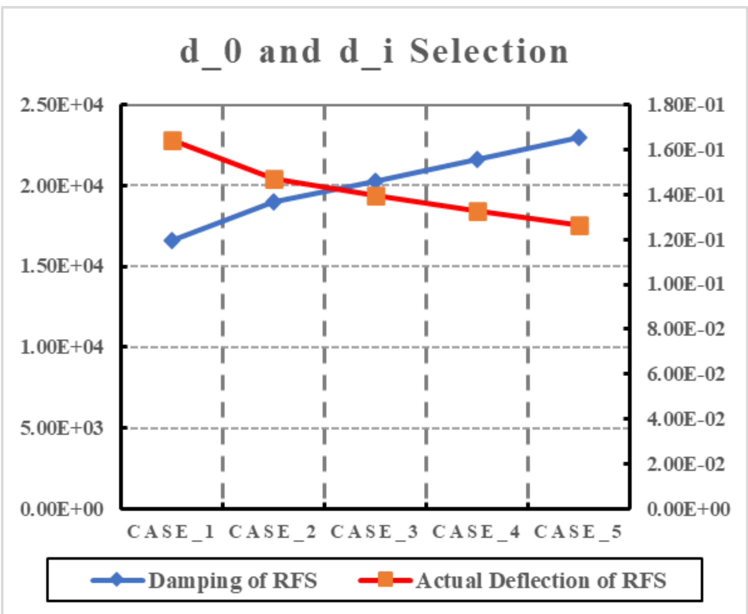


Figure 4.1: Stopping criteria for Rubber Filled Spring

So as from obtained values of wire diameters upon 5 cases as shown in Table 4.2. Case 3 will be the choice of selection as shown in the Table 4.1.

Cases	d_0	d_i
Case1	0.0284	0.01904
Case2	0.0304	0.0228
Case3	0.0314	0.0244
Case4	0.0324	0.0260
Case5	0.0334	0.0274

Table 4.1: Obtained values of wire diameters upon 5 cases

Obtained input and output variables for every 5 Cases after satisfaction of constraints is shown in the table 4.2 whereas Case 3 will be the selected one.

Cases	d_0	d_i	C_{rfs}	$\triangle_{max\ \_rfs}$	m_{RFS}	m_{SSRS}
Case1	**0.0284**	**0.01904**	**1.66E+04**	**1.65E-01**	**14.60**	**15.62**
Case2	**0.0304**	**0.0228**	**1.90E+04**	**1.47E-01**	**13.8**	**15.62**
Case3	**0.0314**	**0.0244**	**2.03E+04**	**1.39E-01**	**13.48**	**15.62**
Case4	**0.0324**	**0.0260**	**2.16E+04**	**1.33E-01**	**13.26**	**15.62**
Case5	**0.0334**	**0.0274**	**2.30E+04**	**1.26E-01**	**13.15**	**15.62**

Table 4.2: Obtained input and output variables for every case

d_o =0.0314 d_i =0.0244

$C_{RFS} = 20291\ Ns/m \quad C_{SSRS} = 13258\frac{Ns}{m}$

$m_{RFS} = 13.48\ kg \qquad m_{SSRS} = 15.62\ kg$

Hence the final obtained variables after satisfying constraints and stopping criteria are listed below in the Table 4.4.

d_{int}	C_{rfs}	C_{ssrs}	τ_{max_rfs} GPa	τ_{max_ssrs} GPa	K_{rfs} N/m	K_{ssrs} N/m	$\triangle_{max\_rf}$	$\triangle_{max\_ss}$
0.02449	**2029.15**	**13258.91**	**1.480e+09**	**1.483e+09**	**3455.81**	**2786.15**	**0.1394**	**0.1624**

Table 4.3: Obtained Parameters for RFS

4.2.2 Sprung mass acceleration upon 20/45/100 km/hr. upon RP2

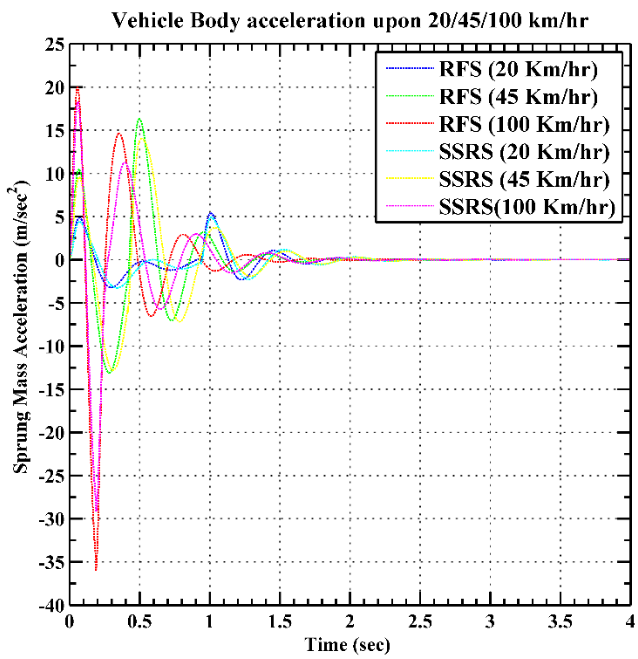
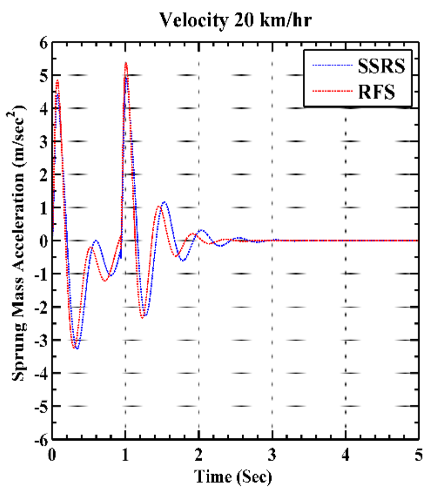


Figure 4.2 Sprung mass acceleration vs time

As shown in the Figure 4.2, obtained Body acceleration graph from QCM reflects peaks of RFS as compared to SSRS which is obvious due to increased damping value [42] however it can be countered by its installation in an active suspension system [43] however earlier stability of vehicle upon installation of RFS as compared to SSRS is increased by 20%.Now according to ISO 2631-1 [44] if RMS acceleration < 0.315 $\mathrm{m/sec}^2$ it will be in comfortable zone and if RMS acceleration 0.315-0.63 $\mathrm{m/sec}^2$ it will be in a little uncomfortable whereas RMS acceleration value for uncomfortable zone >0.8 $\mathrm{m/sec}^2$. As our RMS for both RFS and SSRS are in comfortable zone at 20KM/hr. whereas a little uncomfortable zone at other velocities, as it is clear to notice that higher speed on such type of circular hump will be resulted in absence of comfortable ride.

Isolated results of body acceleration upon 20/45/100 km/hr. velocities are shown in the Figure 4.3,4.4 and 4.5.



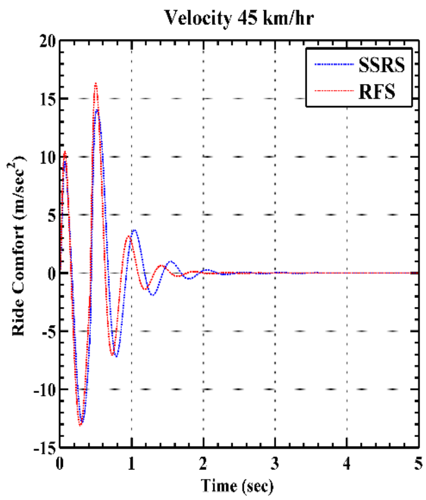


Figure 4.3 Body acceleration at 20 km/hr. Figure **4.4** Body acceleration at 45 km/hr.

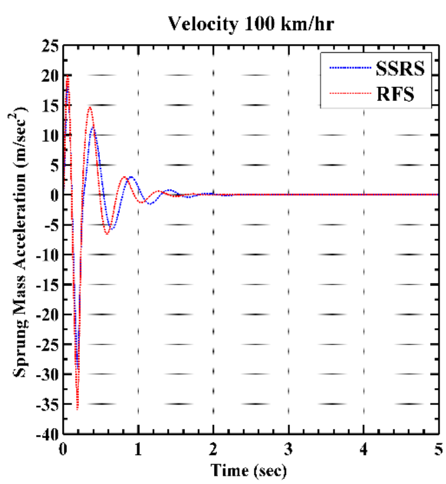


Figure 4.5 Body acceleration at 100 km/hr.

4.2.3 Tire Deflection/ Road Holding (RH) upon 20/45/100 km/hr. upon RP2

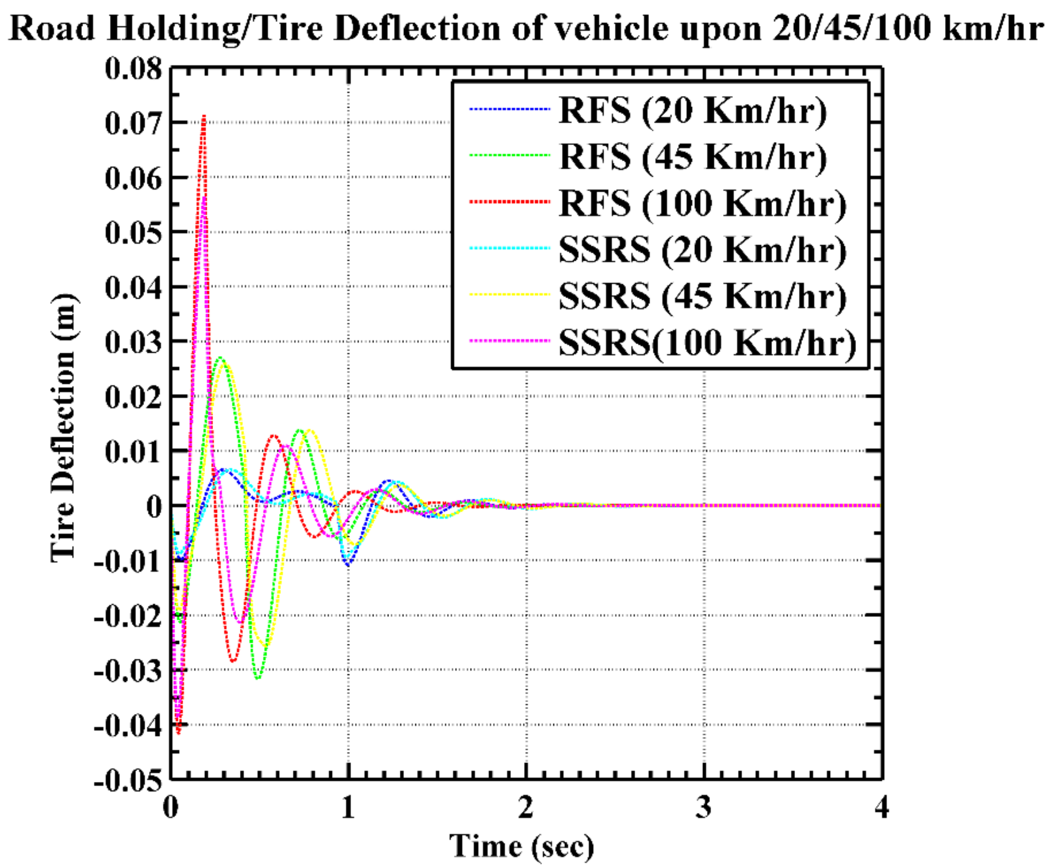


Figure 4.6 Tire Deflection vs time

As shown in the Figure 4.6 increased velocity tend to increase road holding of vehicle upon installation of RFS [42]. Road holding decrease as velocity is increased but due to improved damping coefficient value and subtle stiffness improvement provides these improved results as compared to SSRS. There is 27% rise in first peak value of road holding is observed for RFS as compared to SSRS at 100Km which is a positive turn for a vehicle demanding higher road holding at higher velocities.

Isolated results of tire deflection upon 20/45/100 km/hr. velocities are shown in the Figure 4.7,4.8 and 4.9.

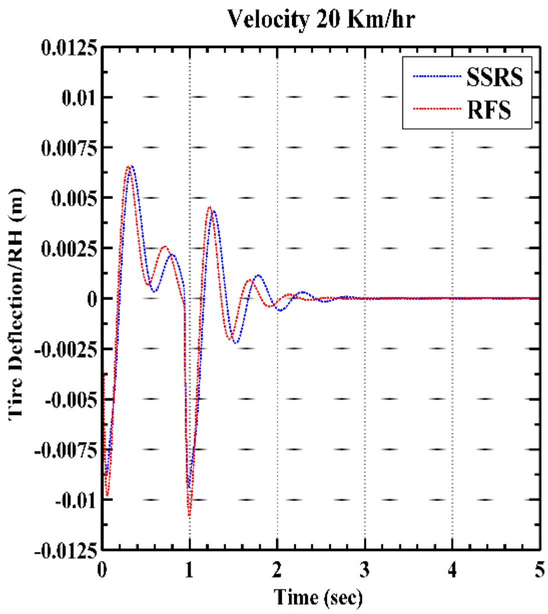


Figure 4.7 Tire Deflection at 20 km/hr.

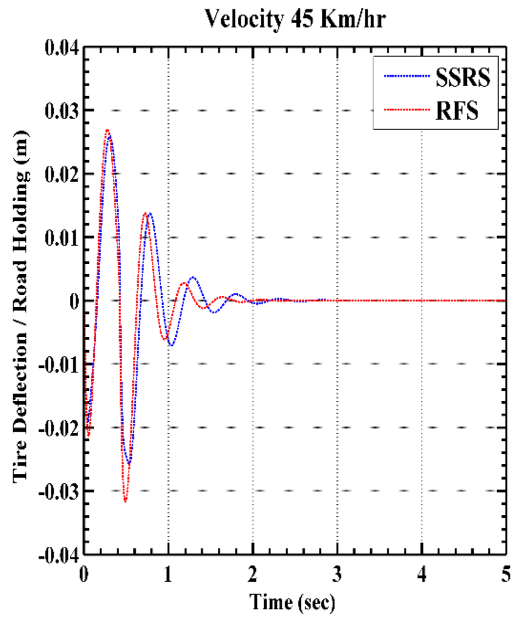


Figure 4.8 Tire Deflection at 45 km/hr.

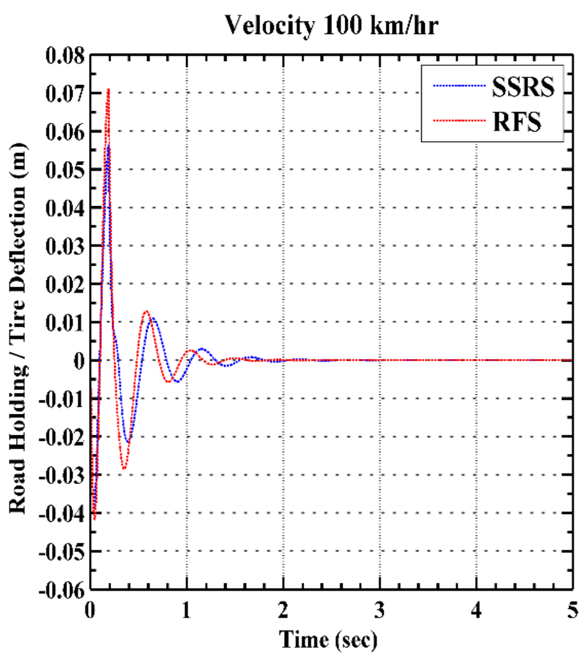


Figure 4.9 Tire Deflection at 100 km/hr.

4.2.4 Wheel Displacement upon 20/45/100 km/hr. upon RP2

The peak value of the wheel displacement tends to rise at medium velocity but reduces at higher velocity as shown in the Figure 4.10. In case of r stability of wheel RFS is an absolute winner as compared to SSRS results.

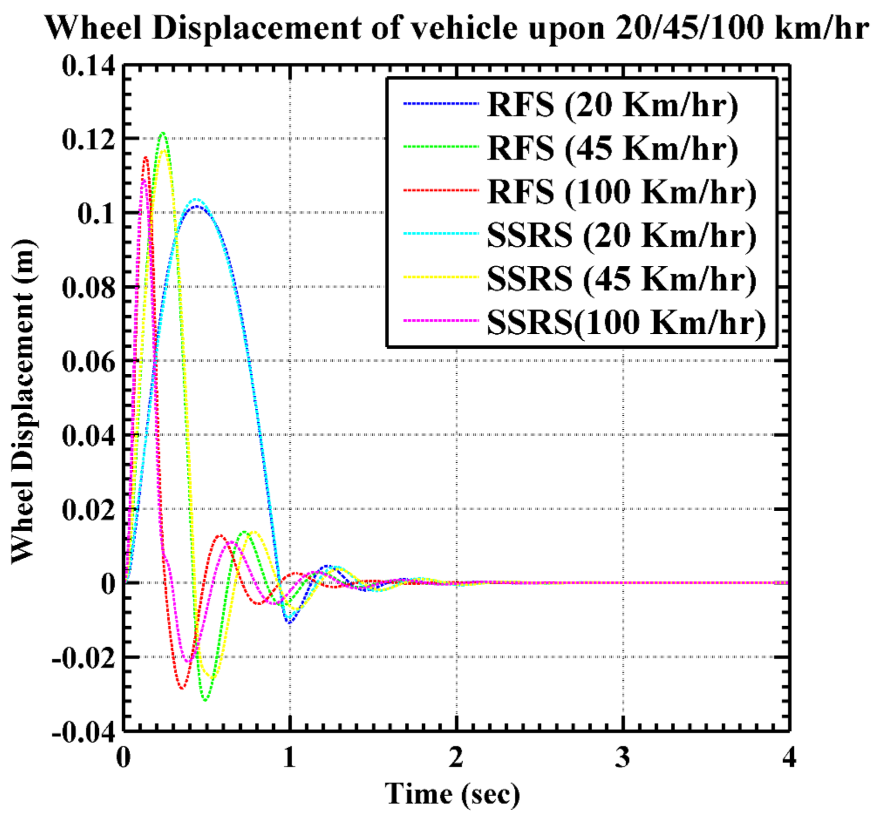


Figure 4.10 Wheel displacement vs time

4.2.5 Sprung mass velocity upon 20/45/100 km/hr. upon RP2

As acceleration is a derivative form of velocity so velocity depicts same results as acceleration but with improved earlier stability in RFS case as shown in the figure 4.11.

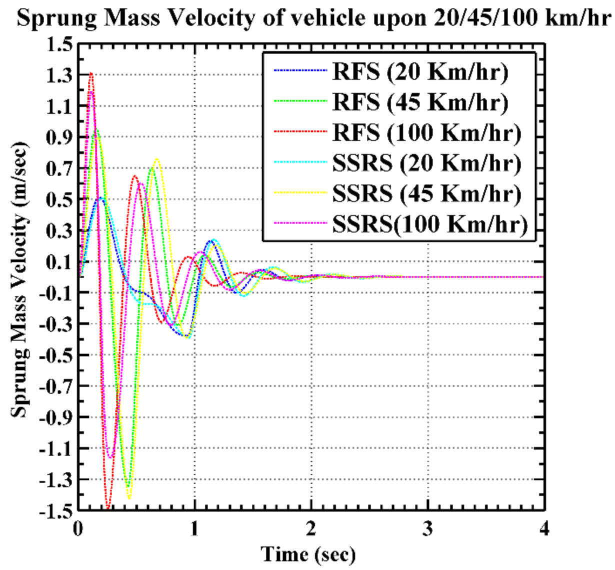


Figure 4.11 Sprung mass velocity vs time

4.2.6 Sprung mass acceleration upon RP1

As shown in the Figure 4.12 the sprung mass acceleration of the RFS installed in a vehicle provides higher peak as compared to SSRS but these higher peaks can be mitigated via installation of spring in a passive suspension system as the simulation is carried out in passive suspension system which is quite conventional .Whereas on the other hand RFS due to improved damping coefficient value provides 6% earlier stability which is beneficial for human's comfortable ride.

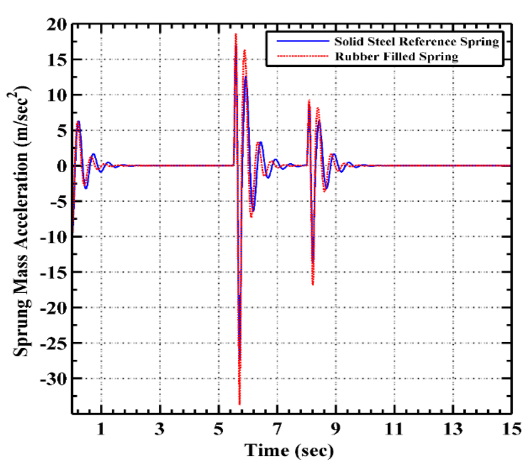


Figure 4.12 Sprung mass acceleration vs time

4.2.7 Tire deflection/Road holding, body velocity, displacement and wheel travel upon RP1

As can be seen from the Figure 4.13. that road holding or tire deflection of the vehicle upon installation of RFS as compared to SSRS is providing much better result in addition to earlier stability with improvement of 6%. when the car is passed from the Road Profile1.

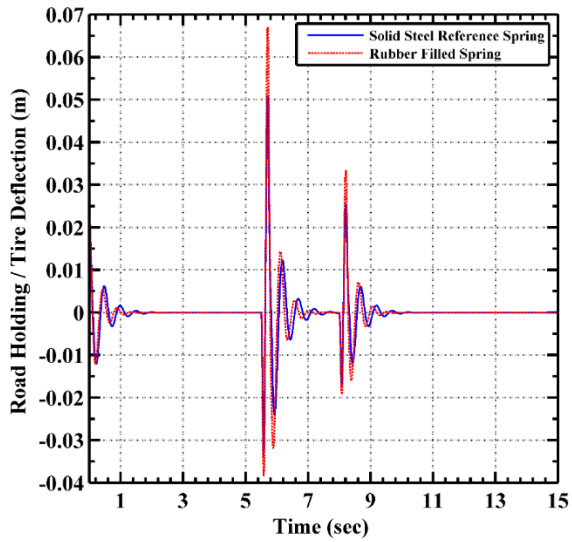


Figure 4.13 Tire deflection vs time

Vehicles body displacement ,velocity and wheel displacement are reflecting the same behavior as shown in the Figure 4.14, 4.15, 4.16 by sprung mass acceleration as acceleration is the successive derivative of displacement and velocity so to mitigate these peaks for RFS active suspension system installation is a gateway whereas on the other hand earlier stability of the vehicle is the prime advantage of RFS**.**

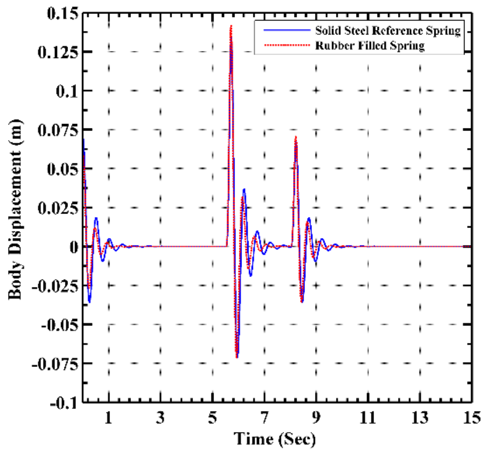


Figure 4.14 Body displacement vs time

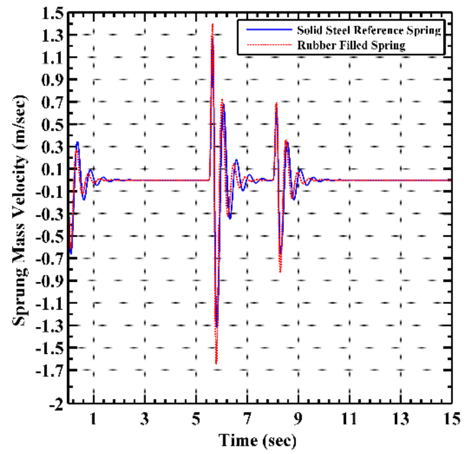


Figure 4.15 Body Velocity vs time

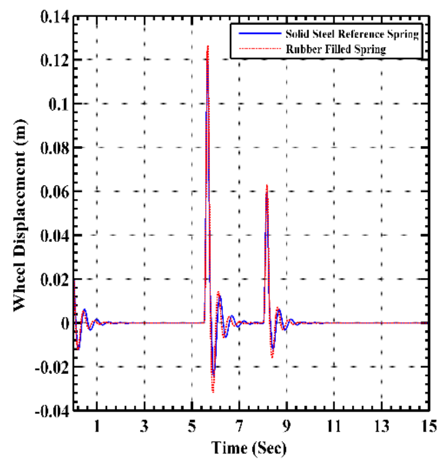


Figure 4.16 Wheel displacement vs time

4.2.8 Suspension deflection upon RP1

Coming towards the suspension deflection improvement in suspension is also related with the suspension deflection ,the peak in the curves should be minimum in improved vehicle suspension [45].Designed RFS in this research fulfill this criteria by reducing the suspension deflection by 10 % in addition to 0.4 sec earlier stability resulting 6% improved earlier stability as shown in the Figure 4.17.

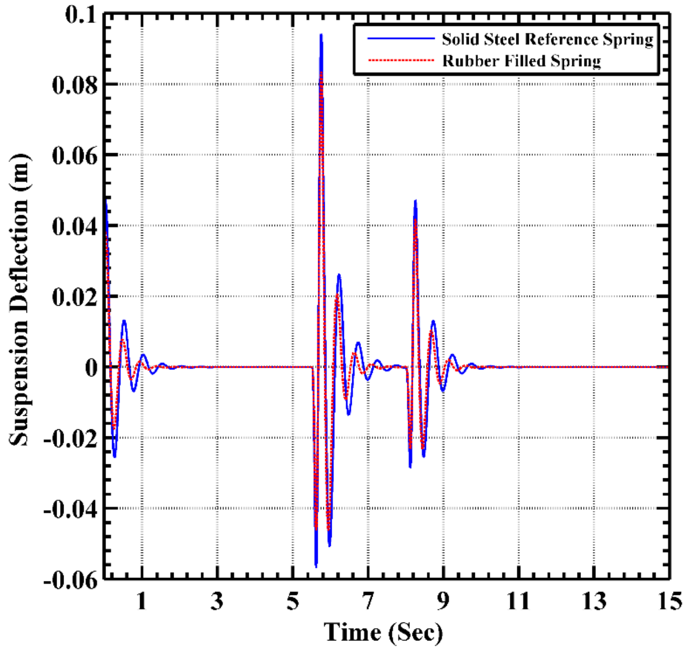


Figure 4.17 Suspension deflection vs time

CHAPTER 5: CONCLUSIONS

5.1 Conclusions

- RFS feasibility inside a suspension system as compared to conventional SSRS underlines composites potential gains as compared to conventional steel.
- RFS highlights its soundness in those area of fields where a compromise between damping and lightness is considered a primary design goal as rise in damping coefficient is 53.04% and weight is decreased by 13.70%.
- RFS represented up to 20% rise in earlier stability as compared to SSRS whereas the peaks in sprung mass acceleration can be countered by its installation in active suspension system
- RFS reflected up to 27% and 14% rise in Road Holding and Ride Comfort peak values respectively at higher Speed whereas a subtle improvement of Road Holding at medium speeds and equivalent response at lower speed along with earlier stability as its response is bracketed with the SSRS Spring
- RFS represented 10% reduction in suspension deflection peak values in addition to 0.4 sec earlier stability as compared to SSRS.

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