DEVELOPMENTS IN SUSPENSION – A REVIEW

L.M. Jugulkar*

Dr. Shankar Singh**

Nitin Satpute***

Dr. S. M. Sawant****

ABSTRACT

Function of a Shock Absorber is to isolate occupants from road-induced vibrations. Vehicle undergoes vibrations due to uneven road profile, corner turning and braking. Hydraulic telescopic Shock Absorbers used since inception of Automobiles, are very robust and very reliable means of attenuating vibrations. With this type of Shock Absorber, damping and spring properties do not change. Nowadays increased comfort requirement form the passenger has force automobile manufacturers to explore for Semi active and Active suspension, that changes it damping properties and uses an external active for attenuating road induced vibrations.

In this paper, review has been taken from latest development in Automotive Suspension System.

*Assistant Professor, Rajarambapu Institute of Technology, Rajaramnagar, Sangli, Maharashtra.

**Associate Professor, Mechanical Engineering Department, (SLIET), Longowal, Dist. Sangrur, Punjab

****Professor, Rajarambapu Institute of Technology, Rajaramnagar, Sangli, Maharashtra

INTRODUCTION

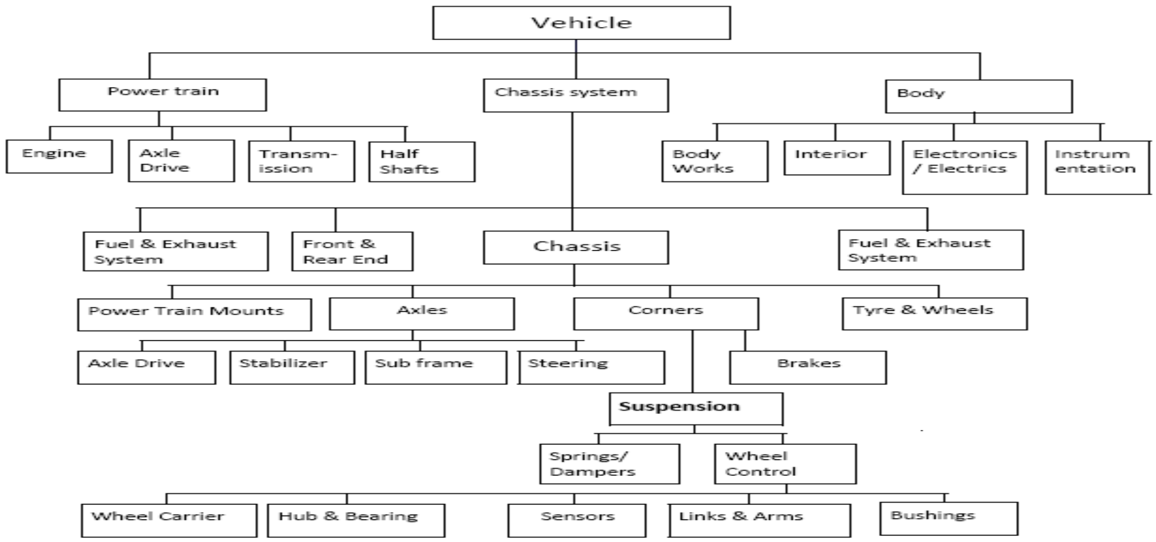

Fig 1 Vehicle Components

The complete vehicle is traditionally divided into three groups as power train, chassis/suspension, and body. The power train contains elements, which propel the vehicle, the body provides room for people, and the chassis and suspension allow the vehicle to ride, turn, and stop. Modern vehicles integrate the body and chassis into a structure known as unibody or monocoque. The chassis and suspension are main components of the automobile and together are made up of the wheels, wheel carriers, wheel bearings, brakes, wheel suspension, sub frames, springs (including stabilizers), dampers, steering gear, steering linkage, steering column and wheel, pedal cluster, motor mounts, drive shafts, differential.

In designing suspension of vehicle, increasing attention is being given to comfort of the passenger. Task of the suspension system is to isolate the vehicle from road-induced vibrations. The conventional suspension consists of fluid damping and spring connected in parallel. For conventional passive dampers, properties of fluid damping and spring are constant.

A piston is attached to the end of the piston rod and works against hydraulic fluid in the pressure tube. As the suspension travels up and down, the hydraulic fluid is forced through tiny holes, called orifices, inside the piston. However, these orifices let only a small amount of fluid through the piston. This slows down the piston, which in turn slows down spring and suspension movement. The amount of resistance a shock absorber develops depends on the speed of the suspension and the number and size of the orifices in the piston. All modern shock absorbers are velocity sensitive hydraulic damping devices - meaning the faster the suspension moves, the more resistance the shock absorber provides. Because of this feature, shock absorbers adjust to road conditions. As a result, shock absorbers reduce the rate of:

Bounce

Roll or sway

Brake dive and Acceleration squat

Shock absorbers work on the principle of fluid displacement on both the compression and extension cycle. A typical car or light truck will have more resistance during its extension cycle then its compression cycle. The compression cycle controls the motion of a vehicle's unsprung weight, while extension controls the heavier sprung weight. During the compression stroke or downward movement, some fluid flows through the piston from chamber B to chamber A and some through the compression valve into the reserve tube. To control the flow, there are three valve stages each in the piston and in the compression valve. At the piston, oil flows through the oil ports, and at slow piston speeds, the first stage bleeds come into play and restrict the amount of oil flow. This allows a controlled flow of fluid from chamber B to chamber A. At faster piston speeds, the increase in fluid pressure below the piston in chamber B causes the discs to open up away from the valve seat.

At high speeds, the limit of the second stage discs phases into the third stage orifice restrictions. Compression control, then, is the force that results from a higher pressure present in chamber B, which acts on the bottom of the piston and the piston rod area. As the piston and rod move upward toward the top of the pressure tube, the volume of chamber A is reduced and thus is at a higher pressure than chamber B. Because of this higher pressure, fluid flows down through the piston's 3-stage extension valve into chamber B. However, the piston rod volume has been withdrawn from chamber B greatly increasing its volume. Thus, the volume of fluid from chamber A is insufficient to fill chamber B. The pressure in the reserve tube is now greater than that in chamber B, forcing the compression intake valve to unseat. Fluid then flows from the reserve tube into chamber B, keeping the pressure tube full. Extension control is a force present because of the higher pressure in chamber a, acting on the topside of the piston area.

There are several shock absorber designs in use today

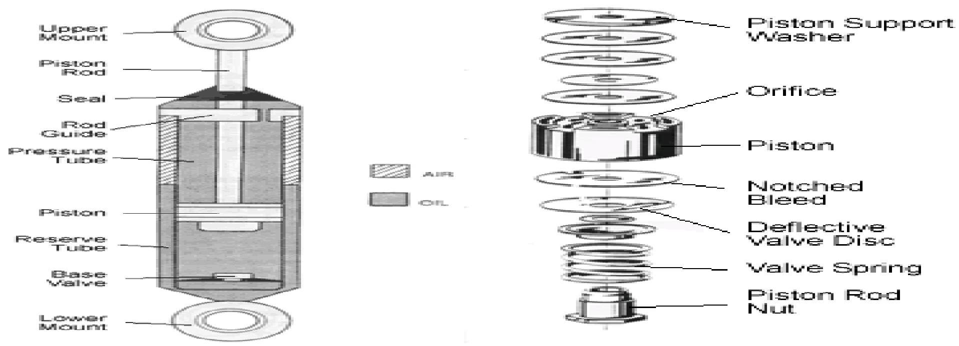

Figure 2-Basic Twin Tube Design

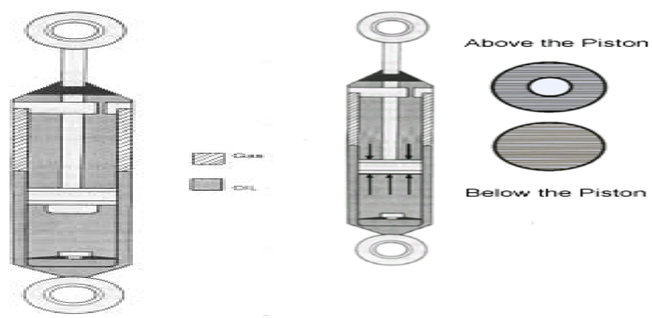

Figure 3-Twin Tube - Gas Charged Design

The twin tube design has an inner tube known as the working or pressure tube and an outer tube known as the reserve tube. The outer tube is used to store excess hydraulic fluid. There are many types of shock absorber mounts used today. Most of these use rubber bushings between the shock absorber and the frame or suspension to reduce transmitted road noise and suspension vibration. The rubber bushings are flexible to allow movement during suspension travel. The upper mount of the shock absorber connects to the vehicle frame. Notice that the piston rod passes through a rod guide and a seal at the upper end of the pressure tube. The rod guide keeps the rod in line with the pressure tube and allows the piston to move freely inside. The seal keeps the hydraulic oil inside and contamination out. The base valve located at the bottom of the pressure tube is called a compression valve. It controls fluid movement during the compression cycle. Bore size is the diameter of the piston and the inside of the pressure tube. Generally, the larger the unit, the higher the potential control levels because of the larger piston displacement and pressure areas. The larger the piston area, the lower the internal operating pressure and temperatures. This provides higher damping capabilities. Ride engineers select valve values for a particular vehicle to achieve optimal ride characteristics of balance and stability under a wide variety of driving conditions. Their selection of valve springs and orifices control fluid flow within the unit, which determines the feel and handling of the vehicle. The development of gas charged shock absorbers was a major advance in ride control technology. This advance solved many ride control problems which occurred due to an increasing number of vehicles using uni-body construction, shorter wheelbases and increased use of higher tire pressures. The design of twin tube gas charged shock absorbers solves many of today's ride control problems by adding a low-pressure charge of nitrogen gas in the reserve tube. The pressure of the nitrogen in the reserve tube varies from 100 to 150 psi, depending on the amount of fluid in the reserve tube. The gas serves several important functions to improve the ride control characteristics of a shock. The prime function of gas charging is to minimize aeration of the hydraulic fluid. The pressure of the nitrogen gas compresses air bubbles in the hydraulic fluid. This prevents the oil and air from mixing and creating foam. Foam affects performance because it can be compressed - fluid cannot. With aeration reduced, the shock is able to react faster and more predictably, allowing for quicker response time and helping keep the tire firmly planted on the road surface. An additional benefit of gas charging is that it creates a mild boost in spring rate to the vehicle. This does not mean that a gas charged shock would raise the vehicle up to correct ride height if the springs were sagging. It does help reduce body roll, sway, brake dive, and acceleration squat. This mild boost in spring rate is also caused by the difference in the surface area above and below the piston. With greater surface area below the piston than above, more pressurized fluid is in contact with this surface. This is why a gas charged shock absorber would extend on its own. The final important function of the gas charge is to allow engineers greater flexibility in valve design. In the past, such factors as damping and aeration forced compromises in design.

Recent Developments in Shock Absorber

Cristiano Spelta et al. (2010) has described the controller does not consider mass supported by the vehicle. Variation of mass will affect performance of the shock absorber. While choosing damping level of the shock absorber in addition to the velocity, mass supported by the vehicle should be considered. [1]

In this, the Semi active suspension system for Motorcycle is proposed. It works on input from single velocity sensor; variable damping is achieved by using an electro hydraulic valve. This valve will vary flow area between two chambers of the shock absorber, resulting in variable damping effect. Two current driven solenoid valves are used to vary damping factor. Control algorithms decided so that they transmit very low acceleration to the occupants and maintaining road contact of the tire. Skyhook controller is used that will decide the optimum damping level based on relative velocity measurement. There are two damping levels as high or low, the controller will choose one of the level and suitable amount of current (300-1200 MAmp) will be send to the solenoid valve. This valve will decide extend of damping provided by the shock absorber. Various methods of control strategies are discussed with these highlights.

M. Zapateiro et al. (2009) This paper descries methodology to compute control voltage that has to be applied to the MR fluid damper to produce optimal damping force necessary to reduce vibrations. Current methodologies are based on Binghamand Bouc-Wen models, but it has limitations due to non-linear model. In addition, present models give damping force created by specified voltage, but reverse is not possible. [2]

MR fluids create high force in compact size; they need low energy to operate. But their only drawback is that it has hysteresis force velocity loop of which it depends on many factors. Due to these non-linear control methods needs to be applied.

In this paper, a neural network-back stepping controller for a class of semi active vehicle suspension systems equipped with MR dampers is proposed. Back stepping is a recursive design technique that consists in designing virtual controllers for each state of the system until the actual control input is reached. Each controller takes into account the previous one and must be asymptotically stable in the Lyapunov’s sense. After the formulation of control law, a neural network is used as the inverse model of the MR damper, i.e., as the model that takes the desired damping force and calculates the damper control signal that generates the needed damping force.

Dahl model is used for MR modeling that uses control voltage & other variables affecting shape of control loop. Vibrations of the sprung mass, i.e. car body are reduced by reducing its angular velocity (i.e. velocity).

Formula for control voltage is derived using back stepping approach. Later on neural networks can be trained to learn MR fluid dynamics in order to apply control voltage for optimum damping. The second test consists of a neural network, which was designed to predict the control voltage given the piston displacement and velocity and the desired MR damper force. The output, i.e., the voltage, is fed back to the input.

Future work will investigate the performance of other controllers for this system based on other techniques such as the frequency-based QFT or mixed H2/HN control. Various models are presented in the paper for MR damper application. Work presented in the paper can be used for accurate modeling of MR damper performance.

W.L.Wang et al. (2009) propose use of low cost solenoid control valve for changing damping factor of the damper. The damper is simple in construction and is easy to control. For damping lateral vibrations, two interchangeable dampers are installed between the bogeys. These two dampers are controlled by three high-speed solenoid valves and two unloading valves. Combination of orifice and relief valve is obtained for getting desired damping. By changing position of three solenoid valves and two relief valves, flow passage and hence damping factor is varied. Skyhook control logic is used to decide control algorithm for this shock absorber. Totally six modes hard obtain ranging from very soft. This paper describes simple and low cost method of obtaining variable damping, rather than using costly and complicated method like MR dampers. In the proposed research work low cost solenoid valve can be used for changing cross sectional area of flow between two chambers, to vary damping factor of the shock absorber. This paper will be useful for designing such variable damper. [3]

M.J. Thoresson et al. (2009) Gradient based optimization method is used for optimizing shock absorber. Gradient-based method is least time consuming and all the design variables are efficiently considered. Design factors are damper scale factor and static gas volume. Objective function is written in the form of comfort and safety criteria. For this acceleration at sprung mass and tire displacement are considered. Optimization with four design variable sis carried out in ADAMS software. The results of two design variable optimization using MATLAB software are also discussed. Result shows that combined optimization is compromise between ride and handling. The optimization procedure discussed in this paper may be useful in performing optimization of the proposed semi active shock absorber. [4]

Yanqing Liu et al (2008). This paper proposes use of Voigot element to achieve variable spring stiffness. Spring stiffness can be varied up to three times its initial value. This uses using variable damper along with springs connected in series and parallel. In addition, if variable damper is connected in parallel, the system with variable stiffness as well as variable damping can be achieved. Formulas for equivalent stiffness are written by assuming single degree of freedom system. Since changing stiffness will change natural frequency o the system, significant change in performance of the system can be obtained using the system with variable stiffness. Control laws are written to decide level of stiffness required at the particular instant. In the paper, change in damping properties is achieved using MR damper. However, any other means of changing the damping factor can also be used. The paper describes system that changes its stiffness using variable MR damper connected in series and in parallel with spring. This method can be extended for using in automotive shock absorbers for changing equivalent stiffness of the supporting spring. For changing damping factor, methods other than using MR fluid can also be used. [5]

Semi active damping only expends a small amount of energy to change system parameters, such as damping and stiffness. They are simple to implement. Spring with variable stiffness is not fully explored by the researchers. By changing damping C_2, equivalent stiffness of sub damper changes and so equivalent stiffness, k will change. Results are obtained for various settings of the various elements. Peak displacement is found to be significantly reduced with optimum settings of c1 & c2.

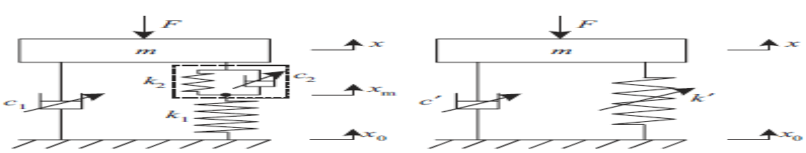

Fig 4: Mechanical configuration of variable stiffness and damping: (a) original model and (b) equivalent model (after Yanqing Liu et al., 2008)

Three distinct settings for damper are chosen based on soft, low & high. This system offers excellent control over stiffness and stiffness can be varied up to three times the given value. It is very easy & quick to control with MR dampers. But this paper does not give its application for use in automobile suspension design. This research paper will be useful for designing a system with variable stiffness for use in the proposed shock absorber.

Ping Yang et al (2008). Describes new type of shock absorber consisting of rubber ball, viscous fluid is explained. This absorber is used for delicate electronics items. It has non-linear characteristics. Energy is dissipated through forcing of oil by piston on rubber air balls.

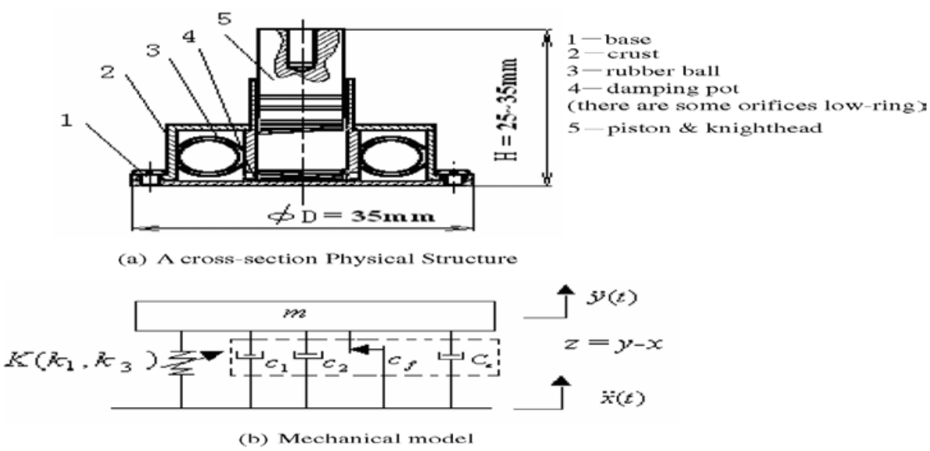

Fig 5: Physical structure and mechanical model of the prototype shock absorber (after Ping Yang et al 2008)

The shock absorber performance can be distributed in three bands as, low transmission, resonance transmission & again low transmissibility isolation band. The components considered to compute damping force are - throttling damping force, laminar flow force, inertia damping force, structural damping force in the components & friction force. Based on test & analytical data, there is some light difference between the two. Mathematical model constructed is not very accurate. Factors affecting performance of damper are - oil viscosity, number of orifices in the damper, frequency (force increase with increase in frequency & vice versa). The analysis can be used as basis for design since performance varies as per frequency, etc. also mathematical model is not very accurate. Non-linear damping effect of rubber ball is discussed in the paper. [7]

C. Lauwerys J et al (2004) given presently available control methods for active shock absorbers are complex, & hence difficult to employ. They have either very simple mathematical model (than actual system), or the parameters are so many that they are difficult to implement. To avoid complexity of mathematical models, a model-free control structure and design approach is developed. This approach is based on physical principles of semi-active shock absorbers and cars in general, but does not require a model of its dynamics. Therefore, it is applicable to any semi-active or active suspension system and any type of car.

The control structure incorporates many physically interpretable parameters. The tuning of these parameters is based on the principles of passive shock absorber tuning. Skyhook method is used to calculate desired damping force, and then control module will decide the required current flow through circuit. Accelerometers at four corners of vehicle are used to; compute velocities, which are used to calculate vehicle roll, pitch & bounce. These three velocities are used to compute proper damping factor in the above three directions. Non-linear skyhook model is used. This controller will change damping intensity based on feedback from this algorithm. [8]

Choon Tae Lee et al (2006) studied displacement sensitive shock absorber has variable flow passage between compression & rebound chamber. For lesser displacement of suspension flow area is greater & as suspension displacement increases flow area reduces accordingly. It is achieved by using displacement sensitive orifice at the cylinder wall. Such a DSSA improves ride comfort on the paved road driving conditions because of low damping force caused by small piston stroke. Displacement-sensitive orifices can be divided into three zones such as the soft, transient and hard zone. The flow continuity equation was used that takes in to account piston velocity, coefficient of discharge, pressure, discharge rate areas of all the orifice & valves at various locations. Based on this equation damping force is calculated, this force is same as of the experimental readings taken. Detailed mathematical model constructed in this paper gives exact value of damping force developed by the shock absorber. This model can be used to determine response of the shock absorber for various operating parameters. The developed mathematical model does not take in to account loss of laminar flow and temperature effect. In addition, it is not applicable to piston with disc valves. [9]

T. Pranoto et al (2005) have developed two types of dampers are discussed as rotary type & with two degree of freedom. For applications like bed in ambulance or any other, in order to avoid motions, the damping force has to be large in usual usage, but it has to be small when the shock occurs. I.e. at start, damping should be less & there should be deflection, but later damping should be more. [10]

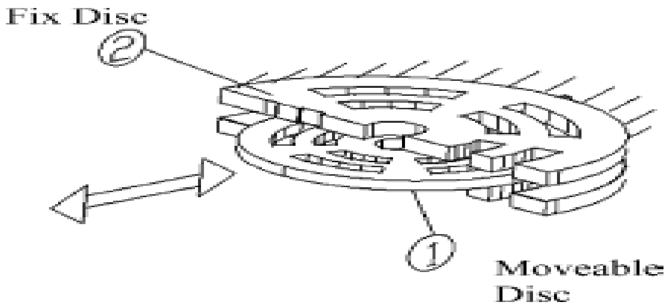

Fig 6 : The disc of 2DOF-type.(after T. Pranoto et al,2005)

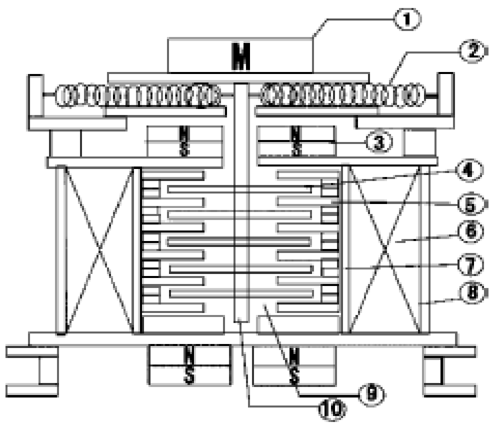

Fig 7: 2DOF-type Damper. Notes: (1)mass; (2) spring; (3) permanent magnet; (4) inner disc; (5) outer disc; (6) coil; (7) inner pipe; (8) outer pipe; (9) MRF; (10) shaft.(after T. Pranoto et al, 2005)

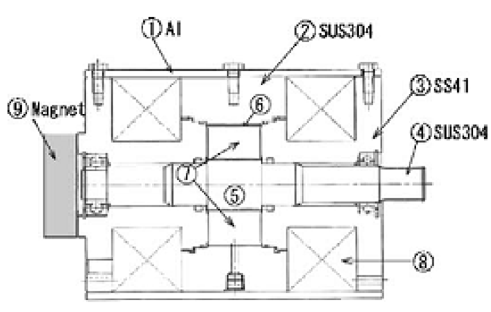

Fig 8: Rotary-type damper. (1) Outer Cylinder, (2) inner Housing, (3) outer housing, (4) shaft, (6) grooves to fix outer disc, (7) MRF, (8) coils, (9) permanent magnet. (after T. Pranoto et al, 2005)

Rotary type gives better efficiency. Shock is improved significantly. MR fluid damper can be made in rotary or reciprocating configuration. The work discusses various configurations their effectiveness. Formulas presented in the work can be useful for damper modeling.

G.Z. Yao et al, (2002) MR fluid offer much higher change in viscosity than ER fluid. It takes place very quickly, needs very low energy. Modified Bouc–Wen model to describe the MR damper behavior, which consists of 16 parameters. Skyhook control method is adopted here. A machine with load cells, velocity sensors is used for experimentation.

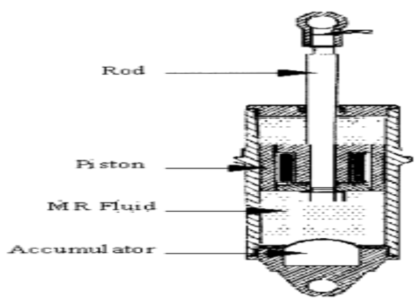

Fig 9 :Schematic drawing of MR damper.(after G.Z. Yao et al, 2002)

By controlling the parameters in the shape hysteresis loop, linearity & shape of the loop in loading & unloading can be controlled. Error function between estimated & obtained damping force is used as objective function. Optimization is carried out in Mat lab simulink. This paper will be useful in designing a sub system with variable stiffness. [11]

Darin Kowalski et al, (2001) in this paper proposes new testing and analysis methodology for shock absorbers. Effect of variation of temperature and nominal length is discussed. Also effect of various types of inputs is discussed. Stepped sine wave and sine on sine wave type input is discussed. A linear model is constructed base on stiffness and damping factor. The parameters are then found using data fitting methods. In earlier research on shock absorber researchers have found 14 to 84 parameters on which function of the shock absorber depends. These methods may not be suitable if multiple wave input is given. For pure sine wave, testing effect of frequency of excitation on stiffness and damping are found from experimentation. In case of multiple wave input of high frequency wave over low frequency wave, effect of low frequency wave is less dominant once high frequency wave is introduced. The paper discusses linear parametric model for sine wave input of shock absorber. [12]

J. A. Tamboli et al, (2001) describes in the earlier literatures, power spectral density (PSD) will treated as white noise & velocity of vehicle as constant to calculate Root mean square acceleration (RMSA). But in actual practice, PSD varies exponentially & velocity is not same. The output based on the above assumption Root mean square acceleration (RMSA) varies significantly from actual conditions. RMSA for highway road conditions is calculated for actual road input and vehicle velocity. Using FFT power PSD is calculated at rear wheel. Effect of vehicle velocity on PSD is studied. For this, a new variable in form of ratio of amplitude at rear & front suspension is introduced based on a time lag between two suspension displacements. Optimum values of front & rear suspension are obtained based on criteria of minimum RMSA. Half car model is constructed for the analysis. q0i (t) =q (t + time lag), in this manner displacement at front and rear wheel are interrelated. The time lag depends on velocity of the vehicle. Factor corresponding to the time lag is included in the analysis. Method of transfer function is used to find RMSA. Again, by using method of power spectral density, values of RMSA are found and compared with the earlier method. Optimization is done for minimizing objective function corresponding to acceleration of the sprung mass. Humans are most sensitive to vibrations of the frequency range of 4-8 Hz. Therefore the optimization is done for frequency of 5 Hz. Non linear optimization techniques are used for optimizing damping parameters for reduction in RMSA subjected to some boundary conditions .[14]

CONCLUSIONS:

-Even though conventional hydraulic telescopic Shock Absorber provides very robust and reliable solution for Automobile Shock Absorber, it has some limitations since its damping and stiffness properties cannot be changed.

-Ideally damping properties and spring rate should be varied depending on vibration velocity. For lower velocities if spring rate and damping are kept low lesser acceleration is transmitted to the occupants, increasing comfort, whereas at higher velocities damping and spring rate needs to be increased to ensure that the tires are not getting lifted off the ground. Cost of Semi active suspension is very high as compared to that of passive suspension.

-In case of semi active suspension damping properties can be varied by using MR fluid damper or by varying flow area between compression and rebound chamber. Magnetic circuits can be used in both the cases, but inductance of the magnetic circuits has to be kept at minimum to avoid phase lag between the signals.

-In using the semi active suspension, damping rate is kept lower for lower velocities and vice versa. No major attempt has been made until date to change stiffness of the suspension. Change in stiffness changes natural frequency of the suspension, hence effect of change in stiffness is affecting the performance of the Shock Absorber.

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