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Automotive suspension with variable damping system A review

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Abstract - *The intent of present paper is to study the automotive suspension with variable stiffness system. The basic function of suspension system is to reduce the deviation of vehicle body from its mean position, besides giving to the passengers the best possible ride comfort. In current scenario much attention has been given to the problem of vehicle suspension system as it plays an important role in improving safety of vehicle and ride comfort. Now a day's automotive industry is very keen regarding the passengers comfort. And development of faster vehicles required smoother and more comfortable rides have led to fitment of efficient damper. Many researchers have studied the automotive suspension system in depth and developed a new kind of system called as variable stiffness system, which works accordingly the load on the vehicle and tries to give best ride comfort to passenger.*

***Keywords**:* Automotive suspension, Variable Stiffness, Deviation, Ride comfort, Damper.

1.INTRODUCTION

Automobile suspension device is the general term for all the parts and components, which mainly consists of three parts: spring (such as spring, spiral springs, torsion bars, etc.), shock absorber, and guiding mechanism. When the vehicle is traveling on different road surfaces so that the wheels are random excitation, Due to elastic support achieved by suspension between body and wheels, which can effectively curb and reduce dynamic load and vibration between body and wheels so as to ensure vehicle traveling comfort and handling stability and therefore achieve the purpose of raising the average speed. [1]. Function of the suspension system is reducing the deviation of the vehicle body from the mean line of travel to a minimum, and the same time, thereby giving the passengers the best possible ride and making best use of contact with ground via tires to provide good adhesion during cornering, acceleration and braking.

The development of faster vehicles and also the requirements of smoother and more comfortable rides have led to the fitment of dampers almost on all present day vehicles. Shock absorbers have a significant influence on handling performance and riding comfort. Shock absorber plays an important role not only for comfort of the riders of the vehicle but also in the performance and life of the vehicle. However, no further reduction of vehicle vibration can be expected for using the optimum values of damping coefficient and spring stiffness for the shock absorber. Thus it is necessary to make modification to improve the functions of shock absorber.

2. AUTOMOTIVE SUSPENSION WITH VARIABLE STIFFNESS SYSTEM

SHI Ying, et.al. [1] Described the 1 / 4 of a suspension model simulation and analysis and had taken 1 / 4 vehicle model as the object, makes vertical displacement vehicles in time-domain analysis in Matlab, and compares the vehicle with the bidding vehicle.

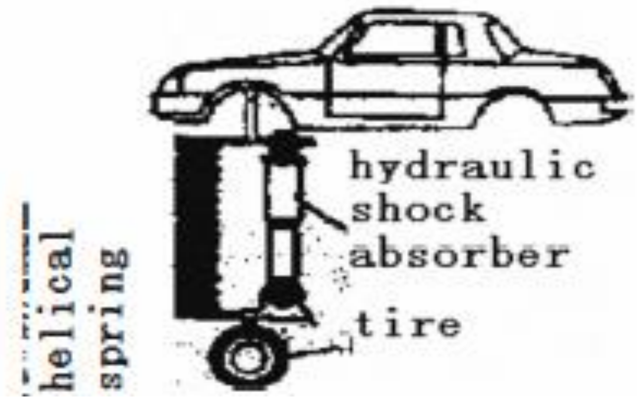

Figure-1: Schematic diagram of vehicle suspension

Also he made some assumption as follows
1) Vehicle near the equilibrium position to make a slight vibration, the spring elastic element is a linear function of its displacement; the damping force is a linear function of its speed.
2) The operating conditions is driving at a constant speed on the flat straight road.
3) Symmetrical to its longitudinal axis line of cars.

And did mechanical analysis on the model and used the method of precise integration. Made a simulation comparison between models and bidding models using Matlab software. In this study they concluded that keeping suspension parameters of the research vehicle and the bidding vehicle are the same and changing the quality of spring set, he observed that because the quality of suspension spring set of bidding vehicle is relatively small, the displacement of in the vertical direction is relatively small and the settling time is relatively short, So that we need to optimize the parameters of the suspension.

Yanhai Xu et.al. [2] Described variable stiffness and damping suspension system via MR damper as shown bellow,

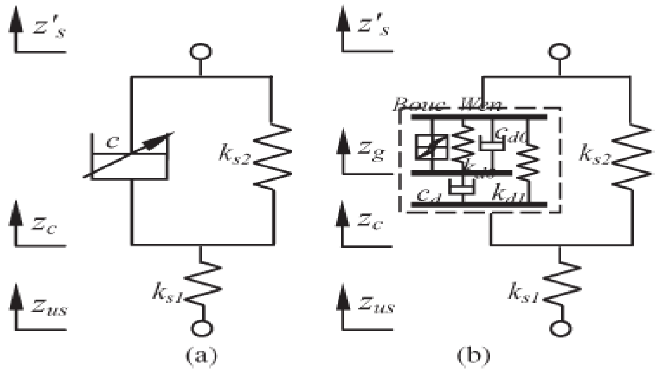

Figure -2: VSVD model. (a) Variable damping damper. (b) MR damper.

Fig. 2(a) presents a variable suspension and damping (VSVD) suspension system controlled by variable damping. This system is composed of two springs, i.e., the upper spring ks2 and the lower spring ks1, and a damper c with variable damping characteristics. The VSVD suspension system can be treated as a conventional semi active suspension system (ks2, c) with an additional spring ks1 in series.

If damping coefficient c is large enough, the equivalent stiffness of the system k' will be ks1, and the equivalent damping coefficient of the system will tend to be zero. If damping coefficient c is close to zero, the equivalent stiffness of the system k' will be the value of springs ks1 and ks2 in series. Therefore, the stiffness of VSVD suspension can be changed with the use of an adjustable damper. The MR damper will be used in VSVD suspension for improving vehicle performance, as shown in Fig. 1(b).

Olugbenge Moses Anubi et.al. [3] Described a detailed description of the variable stiffness concept, the nonlinear energy sink (NES) and the overall suspension system.

A.Variable Stiffness Concept

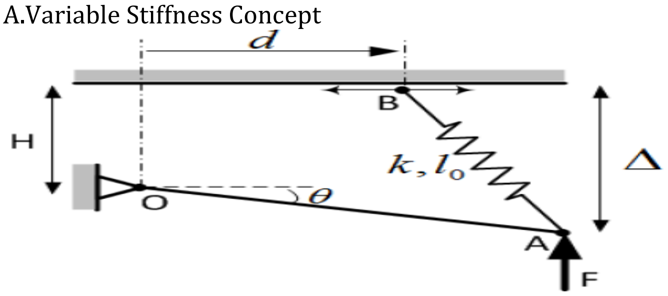

Figure-3: Variable stiffness mechanism.

The variable stiffness mechanism concept is shown in Fig. 3. The lever arm OA is pinned at a fixed point O and free to rotate about O. The spring AB is pinned to the lever arm at A and is free to rotate about A. The other end B of the spring is free to translate horizontally, as shown by the double headed arrow. Without loss of generality, the external force F is assumed to act vertically upward at point A. d is the horizontal distance of B from O. The idea is to vary the overall stiffness of the system by letting d vary actively under the influence of a horizontal linear force generator (not shown in Fig. 3). The variables k and l0 denote the spring constant and the free length of the spring AB, respectively, and Δ the vertical displacement of the point A. The overall free length l0 of the mechanism is defined as the value of Δ when no external force is acting on the mechanism.

B. Orthogonal NES

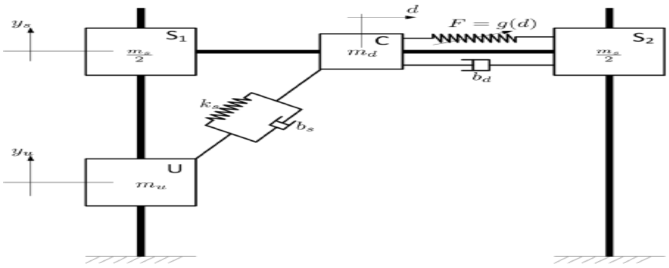

Figure -4: Orthogonal NES.

Fig. 4 shows the NES considered in this paper, the term orthogonal NES is used to describe the concept, because the direction of motion of the secondary system is orthogonal to the primary system. This is suitable, structurally, for the application considered in this paper. The subsystems S1, C, and S2 constitute the primary subsystem, and are allowed to slide vertically together as a unit of total sprung mass ms + md. The subsystem C is termed the control mass (or control subsystem). It, together with the nonlinear spring and the dashpot of damping coefficient bd, constitutes the secondary subsystem.

C. Semiactive VSS System

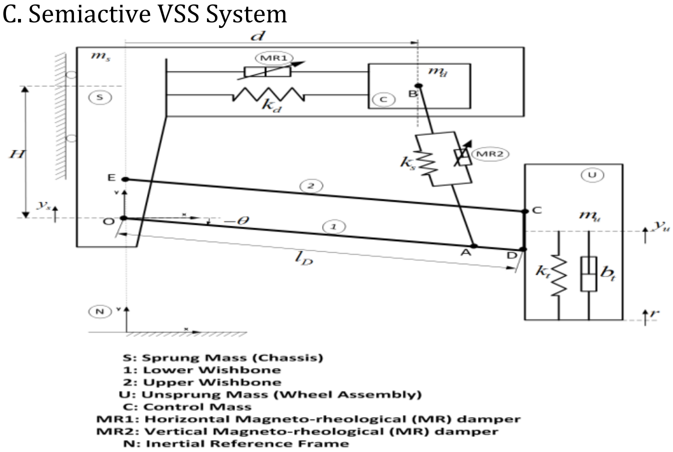

Figure-5: Quarter car model.

The quarter car model of the suspension system considered is shown in Fig. 5. It is composed of a quarter car body, wheel assembly, two-spring-MR damper systems, road disturbance, and lower and upper wishbones. The points O, A, and B are the same, as shown in the variable stiffness mechanism in Fig. 3. The motion of the control mass, which in turn determines the effective stiffness of the suspension system, is influenced by the MR1. The MR1 damper force is designed in subsequent sections to mimic the orthogonal NES introduced in the previous section and the MR2 damper force is designed to mimic the traditional skyhook damping force. The tire is modeled as a linear spring of spring constant kt.

The assumptions adopted in Fig. 5 are summarized as follows.

1) The lateral displacement of the sprung mass is neglected, i.e., only the vertical displacement ys is considered.

2) The wheel camber angle is zero at the equilibrium position and its variation is negligible throughout the system trajectory.

3) The springs and tire deflections (TDs) are in the linear regions of their operating ranges.

The damping characteristics of the considered semi active device can be changed by a control current. However, there is no corresponding energy input into the system as a result of the control current. This implies a passivity constraint on the MR-damper model. The control current is designed to mimic a desired force as close as possible, while enforcing the passivity constraint. This approach has been used in the past for semi active control design.

With this study and comparison he concluded that a better semi active suspension performance, in terms of comfort, can be achieved using an additional semi active device to control the horizontal location of the point of attachment of the vertical strut to the car body, and controlled to mimic a fictitious NES attached between the control mass and the vehicle body.

Yarub Omer Naji Al-Azzawi [4] had described the mechanism of changing the spring stiffness in order to enhance vibration damping is as follow:-

Taking two springs of the extension type as show in figure (6) a, by changing the length of this type of spring, it is possible to change its stiffness, and taking two springs in order to get the action of the helical spring, since the extension spring can extend only and cannot be compressed, at the same time, the type of the spring required in the vibration systems must be of the helical type. Changing the length of these types of springs must be occur without changing the setting location of the mass of the vibration system, and that may be happened by connecting the two mentioned springs as shown in figure (6) b.

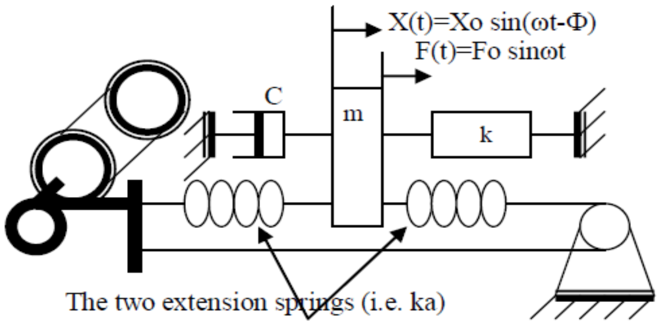

Figure-6: (a). A schematic of an extension spring.

Figure 6. (b). Spring-mass-damper system showing the two additional extension springs. From Figure (6) (b), the purpose of the servomotor is to change the length of the two extension springs (i.e., the developed helical spring), and thus change the spring stiffness as it is required to get damping required. Note here, that each of extension springs must be initially stretched to have an initial length (that mean initial stiffness) to prevent the spring looseness cases as show in Figure (7).

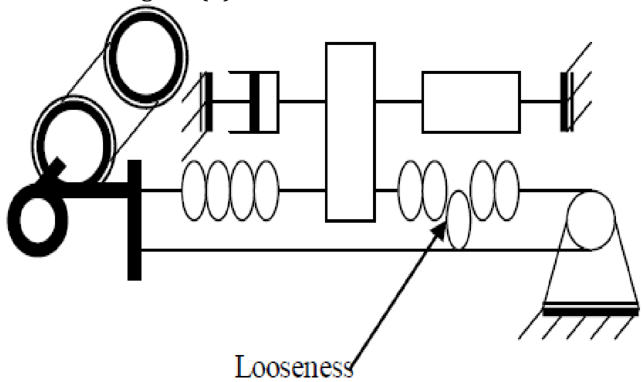

Figure-7: Schematic of the system showing the looseness.

Also there must have mechanical stops to prevent the vibration responses from moving or rotate the servo-motor.

Now the overall semi-active damping system can be explained in the simulation flow diagram shown in Fig. (8)

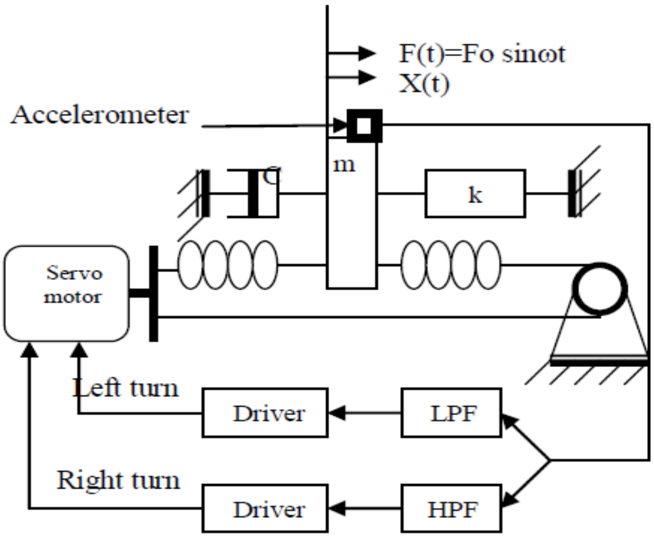

Figure-8: Simulation flow diagram of semi-active damping system.

From Figure(8), the accelerometer senses the harmonic signal of the vibration, and this signal which is assumed as sine wave signal with its frequency will be input signal to the two active filters, and depending on the cut-off frequency of each filters, the signal will passed through either low-pass-filter (LPF) or high-pass-filter (HPF), where if the frequency is low respectively, the signal will pass through the LPF and through the driver shown that will turn the motor left to increase the length of the extension spring (i.e. increasing its stiffness). And if the frequency of the signal is high respectively, it will pass through the HPF and through other driver that will turn the motor right to decrease the length of the extension spring (i.e. decrease its stiffness).

In this he concluded that new idea of using the active filters approved its ability to pick-up the signal of the required frequency, and this idea may be develop to enhance the overall system performance. The flexibility of the system here is limited due to the hardware design rather than software, but at the same time the system is more durable.

Yanqing Liu et.al. [5] described a new variable stiffness and variable damping system in which the stiffness and damping can be independently and easily controlled is proposed. The responses of the proposed systems to the sinusoidal and random excitations are studied in numerical simulations and experiments.

A new model of 1-dof vibration isolation system with two controllable dampers (dampers 1 and 2 corresponding damping coefficients of c1 and c2) and two springs (springs 1 and 2 corresponding stiffness's of k1 and k2) shown in Fig. 9(a) is proposed. Damper 2 and spring 2 comprise a Voigt element. The Voigt element and spring 1 are in series. The stiffness values of the two springs are constant; however, the effective stiffness of the net system can be varied by the controllable damper 2. If the damping coefficient of damper 2 is small enough, the total system stiffness approaches the series stiffness's of springs 1 and 2. However, if the damping coefficient of damper 2 is large enough, the total stiffness approaches the stiffness of spring 1. Damper 1 provides variable damping for the system.

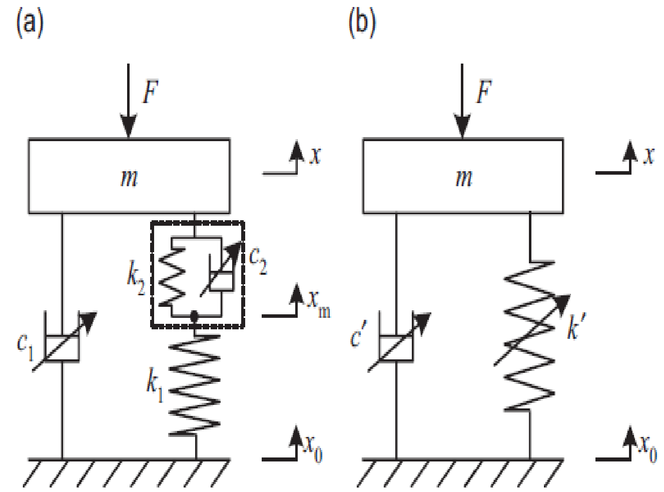

Figure-9: Mechanical configuration of variable stiffness and damping: (a) original model and (b) equivalent model.

Fig. 9(b) shows the equivalent model of the system. Here k' and c' are equivalent stiffness and damping coefficient, respectively.

In this paper he compared eight types of control schemes shown in Table 1

Table -1: Control scheme of damper

	Name	Damper 1	Damper 2
Type 1	Soft system	0ff	0ff
Type 2	Low damping	0ff	On
Type 3	High damping	On	On
Type 4	D on-off (soft)	On-off	0ff
Type 5	D on-of (stiff)	On-off	On
Type 6	S on-off (low)	0ff	On-off
Type 7	S on-off (high)	On	On-off
Type 8	D+S on-off	On-off	On-off

In Type 1 system, dampers 1 and 2 are always in the off-sate and the total stiffness is the small ("Soft suspension"). In the Type 2 system, damper 1 is in the off-state and damper 2 is in the on-state ("Low damping"). In the Type 3 system, dampers 1 and 2 are both in the on-state ("High damping"). Because damper 2 is always in the on-state and the total stiffness is large in the low and high damping systems, they are typically called "stiff suspension". In the Type 4 system, damper 1 is on–off controlled and damper 2 is in the off-state ("D on–off (soft)"). In the Type 5 system, damper 1 is on–off controlled and damper 2 is in the on-state ("D on–off (stiff)"). In the Type 6 system, damper 1 is in the off-state and damper 2 is on–off controlled ("S on–off (low)"). In the Type 7 system, damper 1 is in the on-state and damper 2 is on–off controlled ("S on–off (high)"). In the Type 8 system, dampers 1 and 2 are on–off controlled ("D+S on–off"). Types 1–3 are passive systems, while Types 4–8 are semi-active control systems. After experimentation and test he concluded that the stiffness is controlled by changing the damping coefficient, this system is very simple and easy to apply in practical systems. The system is experimentally investigated using the MR damper that the damping can be changed easily.

Yanhai Xu et.al. [6] Studied many research paper regarding suspension system and variable stiffness and damping. His main purpose of using variable stiffness and damping (VSVD) suspension system was not to mitigate excitation on ride comfort but to improve the capacity of tire normal force and then ameliorate vehicle lateral stability.

In order to illustrate the basic performance of variable stiffness and damping suspension system, a system with single degree of freedom is given in Fig. 10. In the model, the suspension system consists of a set of parallel spring and damper in upper part and a spring located in lower part which is in series with the upper set. The set of parallel spring and damper can be treated as conventional suspension system such as a suspension strut. The lower spring that is in series with the upper set is an additional one. The aim of using the additional one is to obtain a suspension system with variable stiffness performance. So the suspension system is a simple modification of a conventional suspension system.

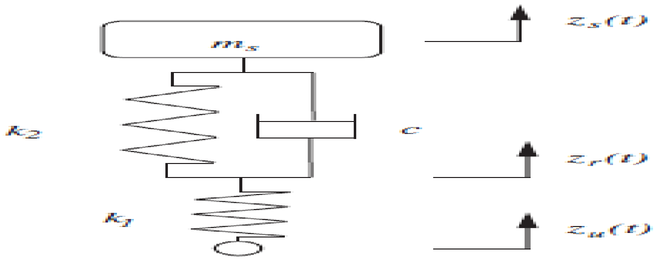

Figure-10: Suspension system model.

Where, ms is sprung mass of the system. zu(t), zr(t) and zs(t) are the displacement of unsprung mass, the reference displacement and the response of sprung mass, respectively. k1 and k2 define the stiffness's of the upper and lower springs as shown in Fig. 10. c is the damping coefficient of the damper in the upper part.

He described Effects of variable stiffness and damping suspension system on tire load transfer. When cornering especially severe turning, load transfer from inside to outside tires caused by lateral acceleration is large enough to impair vehicle characteristics. Normal forces at inside tires are sharply lowered and they probably can no longer provide needed side force. All of these will harm vehicle lateral stability because of the shortage of holding ability. When assuming a steady state maneuver of turning left, sprung mass will roll around rolling axis with a rolling angle in counter clockwise under certain lateral acceleration. The resistance moment will generate via the deformation of suspension system in which the inside is rebounded and the outside is compressed.

The following conclusions he drawn from the study on tire normal force through the use of suspension system with variable stiffness and damping behavior. It is feasible and viable to get a much larger physical limit or saturation of tire than the role of conventional suspension system does. The adjustment of variable stiffness and damping behavior is feasible by the use of MR damper because MR damper is capable to change damping coefficient rapidly.

3. CONCLUSIONS

This paper has reviewed technologies for variable stiffness suspension system. The technologies include the use of arrangement of springs and damper in series and parallel. By making one of the dampers on or off, stiffness of the system may vary. MR dampers are also used to vary the stiffness of suspension system because of its faster response.

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