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Changing Stiffness Shock Absorber for Automotive Application

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Abstract—*It is apparent that the design of a passive suspension system always involves a compromise between ride comfort and handling. However, most semi-active suspension systems are designed to only vary the damping coefficient of the shock absorber while keeping the stiffness constant. To address this issue, achanging stiffness suspension system has been introduced. The changing stiffness suspension system which varies both the stiffness and damping of the suspension element as per mass supported on the vehicle could provide more flexibility in ride comfort. This paper presents, design and theoretical analysis of changing stiffness suspension system. To validate feasibility of improved ride comfort, simulation of the system has been carried out in MATLAB Simulink. The sprung mass displacement and velocity parameters are selected as the indicators of ride comfort. The results indicated that, the system provides minimum percentage overshoot and rapid stabilizing time.*

1. INTRODUCTION

Shock absorber is an important part of automotive system. Task of the suspension system is to provide isolation to the vehicle body from road-induced vibrations and to ensure good road holding. Suspension systems are classified into three categories namely passive, semi-active and active suspensions. For passive system, governing properties like damping coefficient and spring stiffness are constant.In case of passive shock absorber, spring and damper are placed parallel with each other which results in neither change of damping coefficient nor the stiffness. It results in compromise between ride and handling. For active system, an actuator is integrated which provides the controlling force to dampen the road induced vibrations irrespective of the relative velocity damper. However, the complexity and large power requirement results it in too expensive for commercial use. A semi-active system has an ability to vary damping coefficient of the damper. A semi-active suspension system performs better in improving the ride comfort and road handling keeping the complexity and cost at minimum [4]. The stiffness was changed only when the required control force could not be generated by variable damping alone. A vehicle system with variable stiffness demonstrated good performance compared to a semi-active system with variable damping and fixed stiffness. [1]

In this paper, a changing stiffness suspension system is proposed. The responses of the proposed system to sinusoidal harmonic excitations and random excitations are studied with help of MATLAB Simulink.

2. CHANGING STIFFNESS SUSPENSION SYSTEM

2.1 Mechanical Structure

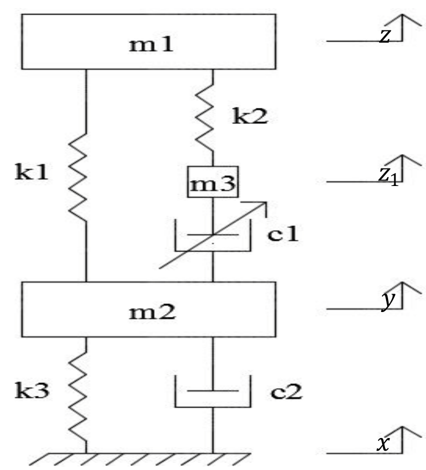

Fig. 1: Quarter car model of changing stiffness suspension system

In this system, series combination of spring 2 and damper 1(spring 2 having stiffness of k_2 damper 1 having damping coefficient ofc_1) are placed in parallel with spring 1(spring 1 having stiffness ofk_1). Damper 1 is having variable damping coefficient $c_1$which is achieved by means of changing the flow area between the compression and the rebound chamber.

In the case of a vehicle suspension, x corresponds to the road input excitation. z_1, y and$z$ are the displacements of the point between damper 1 and spring 2,unsprung massm_2 and sprung mass m_1 respectively. Mass m_3 is having less mass as compared to sprung mass m_1 and the unsprung massm_2. The tire stiffness k_3 and tire damping coefficient c_2 are shown in Fig. 1.

2.2 Equations of motion

A quarter car model of changing stiffness suspension system is shown in Fig. 1. Quarter car model is always shown with 2-dof but the model shown in Fig. 1 is having 3-dof to show the displacement of the point between damper 1 and spring 2. The equations of motion for the systemshown in the Fig. 1 are as follows:

$$m_1.\ddot{z} = -k_1(z-y) - k_2(z-z_1) \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

$$m_3.\ddot{z_1} = -c_1(\dot{z_1} - \dot{y}) + k_2(z-z_1) \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

$$m_2.\ddot{z} = k_1(z-y) - k_3(y-x) - c_2(\dot{y} - \dot{x}) + c_1(\dot{z_1} - \dot{y}) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

3. NUMERICAL ANALYSIS AND SIMULATIONS

The equations presented abovecan be solved numerically using MATLAB’s dynamic system simulation software, SIMULINK.A SIMULINK model shown in Fig. 2 for these equations of motions is built-up using SIMULINK library in MATLAB.

Two important characteristics of a vehicle suspension are its ride comfort and handling ability. The ride comfort can be inferred by analysing the sprung body dynamics. Several factors can adversely affect the ride comfort. The important parameters such as sprung mass displacement and sprung mass velocity have severe impact on the vehicle dynamics. A changing stiffness suspension system’s SIMULINK model is shown in Fig. 2. The input road profiles are defined from the SIMULINK library blocks such as sinusoidal road excitation and step road excitation.

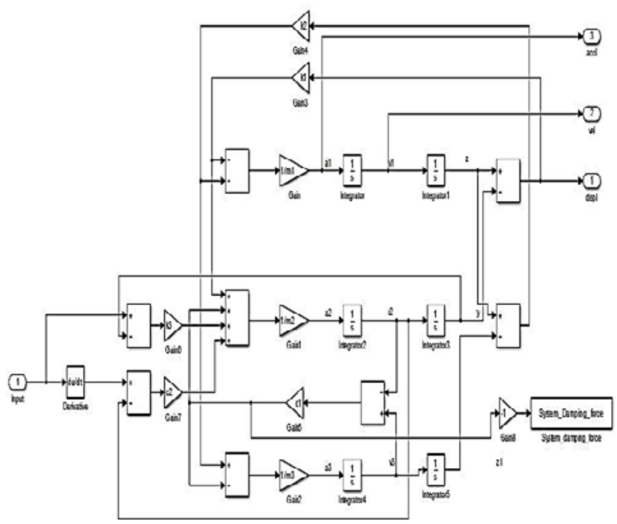

Fig. 2: Simulink model of changing stiffness suspension system

To study performance of the changing stiffness suspension system, it has been compared with passive suspension system. The quarter car model of passive suspension system is shown in Fig. 3.

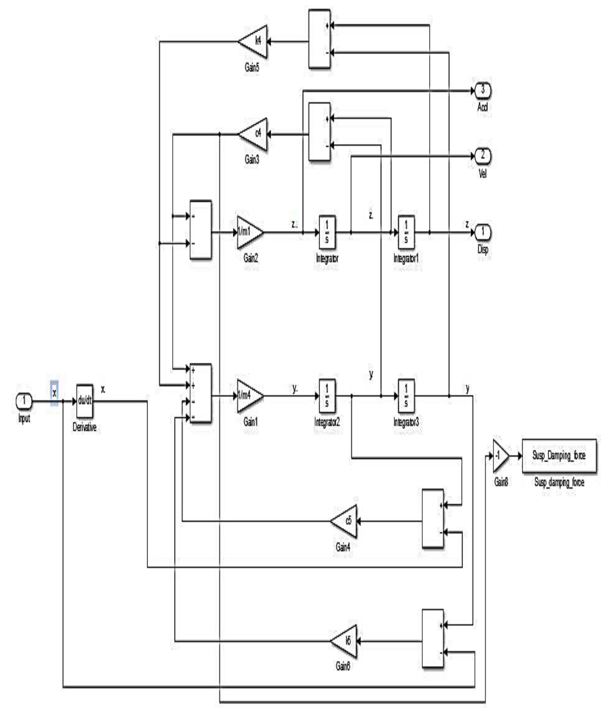

Fig. 3: Passive suspension quarter car model

Basic parameters used for simulations are given in Table 1 below:

Table 1: Changing stiffness suspension system parameters

Parameter	Value	Description
m_1	200	Sprung-mass; Kg
m_2	12	Unsprung-mass; Kg
m_3	1	Mass between spring 2 and damper 1; Kg
m_4	37.5	Unsprung-mass for suspension; Kg
k_1	5190	Peripheral spring stiffness; N/m
k_2	49050	Central spring; N/m
k_3	365600	Tire rate; N/m
k_5	270800	Tire rate for suspension; N/m
c_1	2239.2128	Suspension damping; Ns/m
c_2	100	Tire damping; Ns/m
c_4	1697	Suspension damping; Ns/m

The value of the suspension damping coefficient is varied with the help of orifice opening and closing mechanism. The values of the damping coefficients for number of orifice openings are given as shown in Table 2:

Table 2: Damping coefficient values for number of orifice openings

No. of orifice openings	Damping coefficient c_1 Ns/m
2	2239.2128
4	1700
6	1000
8	550
10	330

The value of suspension damping coefficient c_1 is varied in the MATLAB Simulink and model is simulated to observe performance of the vehicle.Input road profile is taken as sinusoidal road excitation having amplitude of 0.01 m. The frequency of the input signal is varied from 0.5 to 10 Hz. The time response of the system has been calculated for harmonic excitation. To study transient analysis, the step input of 0.01 m has been implemented as shown in Fig. 4.

To change the input road excitation, the manual switch is provided. Depending on the position of the contact switch the input road excitation is implemented to the both changing stiffness suspension system and to the passive suspension system.

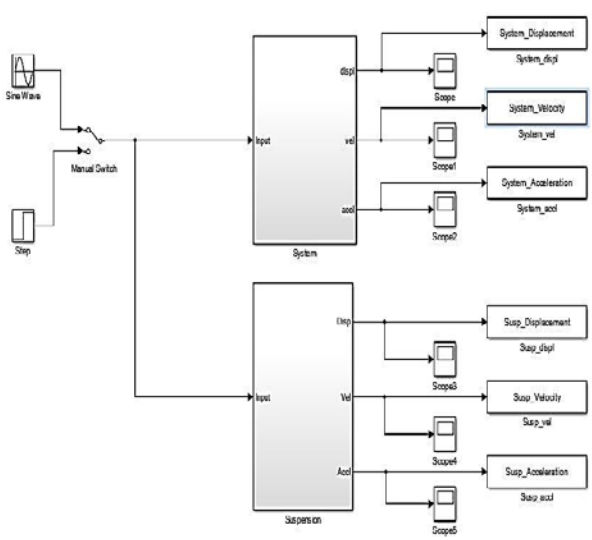

Fig. 4: Road input excitations

4. RESULTS OF SIMULATION

Fig. 5, 6 and 7 shows damping force, sprung mass displacement and sprung mass velocity plot varying against time for the sinusoidal excitation. It has been shown that, the changing stiffness suspension system has rapid stabilizing time and less overshoot as compared to passive suspension system. The spring 2 is placed in series with damper 1 this result inas damping force is changed, the stiffness of the spring is changing accordingly. The sinusoidal road excitation can be considered as the harmonic analysis of the changing stiffness suspension system.

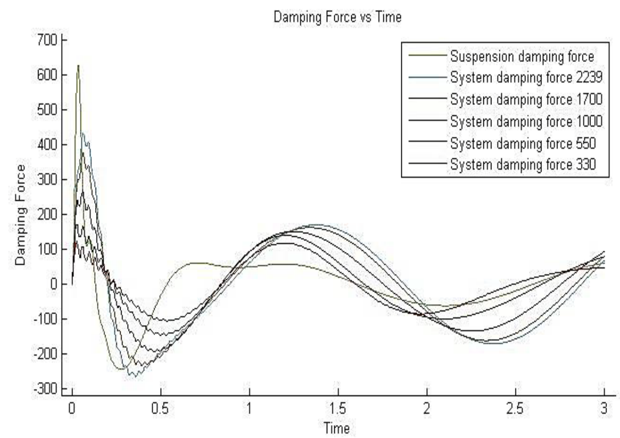

Fig. 5: Damping force plot for sinusoidal input

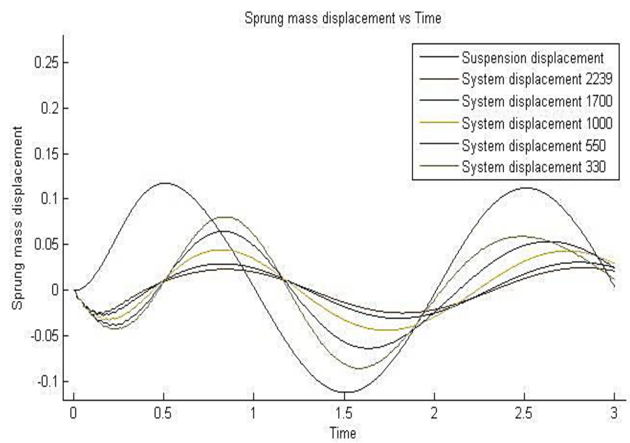

Fig. 6: Sprung mass displacement plot for sinusoidal input

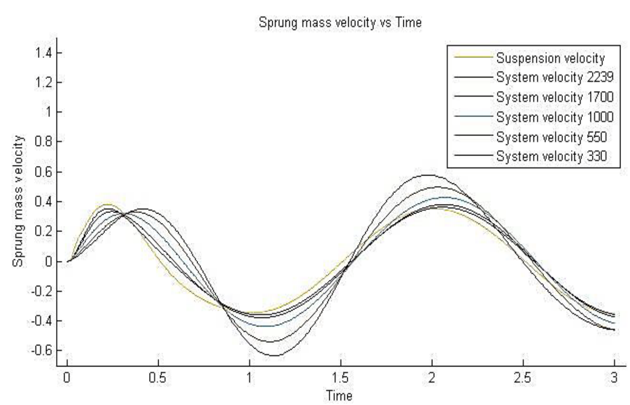

Fig. 7: Sprung mass velocity plot for sinusoidal input

Fig. 8, 9 and 10 shows damping force, sprung mass displacement and sprung mass velocity plot varying against time for the step excitation. It has been shown that, the changing stiffness suspension system has rapid stabilizing time and less overshoot as compared to passive suspension system. The step road excitation can be considered as the transient analysis of the changing stiffness suspension system.

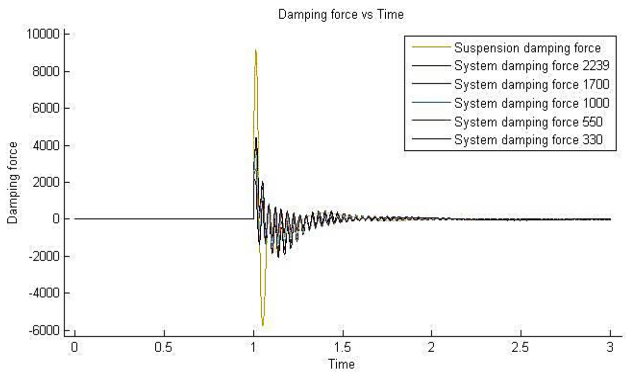

Fig. 8: Damping force plot for step input

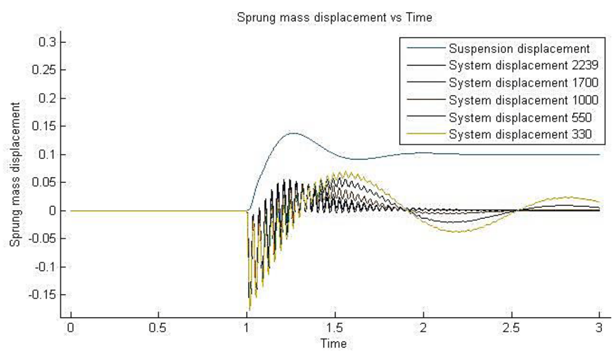

Fig. 9: Sprung mass displacement plot for step input

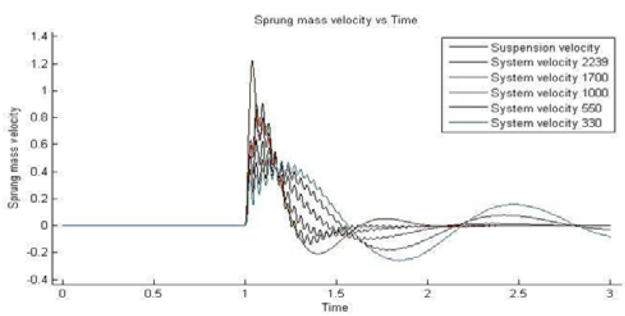

Fig. 10: Sprung mass velocity plot for step input

5. CONCLUSION

The changing stiffness suspension system performs best in terms of ride comfort. The system shows minimum stabilizing time and less overshoot as compared to passive suspension system. The system is validated for its performance in relation to two different road excitations input. It has been observed that,

- Although changing stiffness suspension system is having less damping force as compared to passive suspension system, its stabilizing time is rapid as compared to passive suspension system.
- The sprung mass displacement and velocity are reduced considerably as compared to passive suspension system with less overshoot.

6. ACKNOWLEDGEMENT

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