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Experimental analysis of Electro Hydraulic Active Suspension System for Commercial Vehicle.

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Abstract—The purpose of the suspension in a vehicle is to keep the tires in contact with the road and to prevent the shocks from the roads. Adjustable Variable damping suspension is an automotive technology that controls the vertical movement of the wheels with an automated system rather than the movement being determined entirely by the condition of the road surface. The system attempts to minimize the body roll and pitch variation in many driving situations including cornering, accelerating, and braking. This technology allows vehicle to achieve a greater degree of ride comfort and maneuver by keeping the wheels perpendicular to the road in corners, allowing better traction and control. This paper proposes an electro hydraulic active suspension system which overcome the limitations of passive suspension. This paper also illustrates the experimental results for variable damping. The theoretical and experimental results are validated.

Index Terms—.Active suspension, Variable damping, Electro hydraulic suspension

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1. INTRODUCTION

Suspension system is one of the critical components in the present of vehicle system. Ride Safety and the handling Capabilities of the vehicle are mainly determined by its suspension system, which transmits the forces between the vehicle and the roads. Suspension consists of the system of springs, shocks absorbers and linkages that connects a vehicle to its wheels. In other meaning, suspension system is a mechanism that physically separates the car body from the car wheel. The main function of vehicle suspension system is to minimize the vertical acceleration transmitted to the passenger which directly provides road comfort. In the past few decades, many researchers have paid considerable attention to the issues on how to guarantee the stability of the suspension systems and how to improve the required suspension performances, namely, ride comfort, road handling and suspension deflection.

Generally, there are three types of systems namely; passive, semi – active and active suspension system that have been widely investigated by many researchers with different techniques and algorithms.

1.1 Passive Suspension System

Passive suspension systems are the most common systems that are used in commercial passenger cars. They are composed of conventional springs, and single or twin-tube oil dampers with constant damping properties. Traditional springs and dampers are referred to as passive suspensions most vehicles are suspended in this manner. The passive suspension system showed lack of performance of vehicle stability as compared with semi-active and active suspension system. The dynamic behavior of passive automotive suspension system is determined by the selection of the spring stiffness and the damper coefficient. The fixed damper and spring component of the passive system has not well enough for energy absorption to sustain the load or road disturbance acted into the vehicle system.

1.2Semi Active Suspension System

The semi active suspension system uses a variable damper or other variable dissipation component in the automotive suspension. An example of a variable dissipater is a twin tube viscous damper. Another example of semi- active dissipater is a Magneto rheological (MR) damperswhich used MR fluid. The MR fluids are materials that responds to an applied magnetic field with a change in rheological behavior. Typically, this change is manifested by the damper can be controlled by controlling the electromagnetic field. Semi- active suspension systems have been investigated in various literatures in order to achieve lower energy consumption and as good performances as full – active suspension system. The semi active system can adjust the damping and thus improve either ride comfort or ride safety compared to the passive system.

1.3 Active Suspension System

Active suspension system refers to a system that uses an active power source to actuatethe suspension links by extending or contracting them as required. In an active suspension, controlled forces are introduced to the suspension by means of hydraulic or electric actuators, between the sprung and unsprungmass of the wheel assemblies. A variable force is provided by the active suspension at each wheel to continuously modify the ride and handling characteristics.

2. CONCEPT OF ELECTRO HYDRAULIC ACTIVE SUSPENSION SYSTEM

The fig.1 shows the concept of electro hydraulic suspension system for the vehicle.

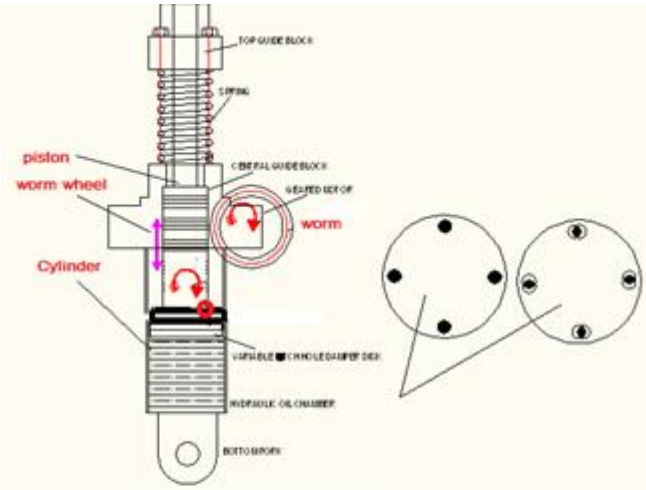

Fig -1: Electro hydraulic active suspension system

2.1 Working of Electro Hydraulic Active Suspension System

Fig 2 : Experimental setup

Geared motor is used to drive the worm which will rotate the worm wheel which will drive the damper disk to change damping orifice size. Another geared motor is used to drive the pinion which will cause the linear motion of rack. Rack and pinion is attached to wedge plate which will describe the road conditions.

The system consists of spring and a damper. Deflection of the spring is to change the spring rate as per condition of the road i.e. for large bumps spring length will be maximum and for short bumps but in series the spring length will be short. The up and down motion of the spring will be as per the road conditon. To change the damping coefficient of the systemby changing the piston orifice size using the variable pitch disk. The rotation motion of the worm wheel makes the disk to open the holes of piston hereby allowing oil to easily pass through system disks thus enable a smooth descent of the suspension in large bump. The piston orifice opening can be reduced to allow the reduced oil flow to provide better damping in case of short but series of bump.

3. EXPERIMENTATION

The experiment was carried out on a test rig shown in Fig 1. The experiment mainly focuses on the characteristics of damping ratio and damping coefficient. The theoretical and experimental results are validated for the different orifice opening. The experiment is carried out for different cases.

Case I : 50% orifice opening

Mass of the system is 25 kg. Damped period of vibration limited to 3.5 sec.

Damped frequency (ω_d) is given by,

$$(\omega_d) = \frac{2\pi}{t_d} = \frac{2\pi}{3.5}$$

$$\omega_d = 1.7951 \text{ rad/sec}$$

Considering δ= 0.5,

$$\delta = \frac{1}{n}\ln\left(\frac{x_1}{x_2}\right)$$

$$\frac{x_1}{x_2} = 1.65$$

Damping ratio (ζ) is given by

$$\zeta = \frac{\delta}{\sqrt{4\pi^2+\delta^2}} = \frac{0.5}{\sqrt{4\pi^2+0.5^2}}$$

ζ = 0.079326

As we know, $\omega_d = \omega_n\sqrt{1-\zeta^2}$

ω_n = 1.80067 rad/sec

Where, ω_n is natural frequency

Spring stiffness (k) = mx ω_n^2 = 25x1.80067^2

k = 81.0603 N/m

Damping ratio is the ratio of damping coefficient to critical damping coefficient we can find damping coefficient by using the equation,

$$C = \zeta \, x \, C_c$$

Where, C = damping coefficient

C_c = critical damping coefficient

$$C = \zeta x 2\sqrt{k \, x \, m} = 0.079326 \, x 2\sqrt{81.0603 \, x \, 25}$$

$$C = 7.141$$

Further the calculations were carried for the different excitation and the theoretical values are shown in table-1.

Table-1: Theoretical calculations for 50% orifice opening

Excitation (mm)	x1/x2	Damping factor	Natural frequency (rad/sec)	Stiffness (N/m)	Damping coefficient
0.5	1.65	0.079	1.800	81.07	7.14
1	2.72	0.157	1.817	82.60	14.28
1.5	4.48	0.232	1.845	85.15	21.42
2	7.38	0.303	1.883	88.73	28.57
2.5	12.18	0.369	1.932	93.32	35.71

The vibrometer is used to measure the amplitude. Vibrometer is connected to the system. Excitation readings were taken from vibrometer. Similarly the calculations were performed and shown in table-2.

Table-2: Experimental calculations for 50% orifice opening

Excitation (mm)	x1/x2	Damping factor	Natural frequency (rad/sec)	Stiffness (N/m)	Damping coefficient
0.42	1.52	0.066	1.799	80.94	6.0
0.92	2.5	0.144	1.814	82.31	13.14
1.28	3.6	0.199	1.832	82.93	18.28
1.84	6.3	0.281	1.870	87.49	26.28
2.28	9.8	0.341	1.909	91.19	32.57

The plot of thetheoretical and experimental damping ratio and damping coefficient with respect to excitation amplitude for 50% orifice opening is shown in chart-1 and chart-2 respectively

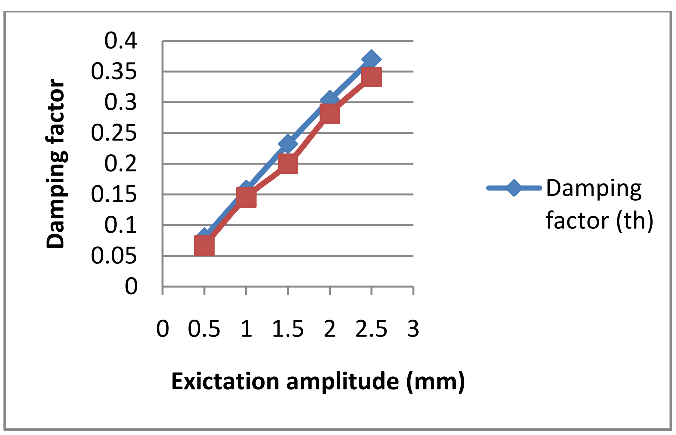

Chart-1: Comparisonplot of theoretical and experimental damping ratio versus excitation amplitude (50% orifice opening)

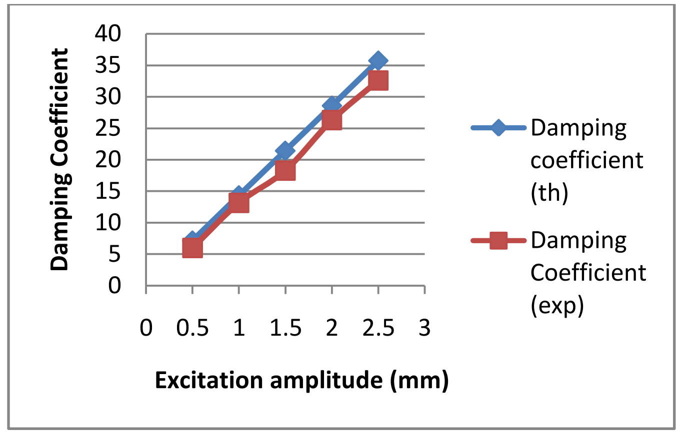

Chart-2: Comparisonplot of theoretical and experimental damping coefficient vs excitation amplitude (50% orifice opening)

Case II: 90% orifice opening

Mass of the system is 25 kg. Damped period of vibration limited to 2 sec.

Damped frequency (ω_d) is given by,

$$(\omega_d) = \frac{2\pi}{t_d} = \frac{2\pi}{2}$$

$$\omega_d = 3.142 \text{ rad/sec}$$

Similarly the further theoretical and experimental calculations are performed and the values are shown in table-3 and table-4 respectively.

Table-3: Theoretical calculations for 90% orifice opening

Excitation (mm)	x1/x2	Damping factor	Natural frequency (rad/sec)	Stiffness (N/m)	Damping coefficient
0.5	1.65	0.079	3.151	248.3666	12.5
1	2.72	0.157	3.181	253.0541	25
1.5	4.48	0.232	3.230	260.8666	37.5
2	7.38	0.303	3.297	271.8041	50
2.5	12.18	0.369	3.381	285.8666	62.5

Table-4: Experimental calculations for 90% orifice opening

Excitation (mm)	x1/x2	Damping factor	Natural frequency (rad/sec)	Stiffness (N/m)	Damping coefficient
0.45	0.45	0.071	3.150047	248.069	11.25
0.96	0.96	0.151	3.178461	252.565	24.00
1.39	1.39	0.216	3.217966	258.552	34.73
1.92	1.92	0.292	3.285422	269.849	48.00
2.38	2.38	0.354	3.359854	282.170	59.50

The plot of thetheoretical and experimental damping ratio and damping coefficient with respect to excitation amplitude for 90% orifice opening is shown in chart-3 and chart-4 respectively.

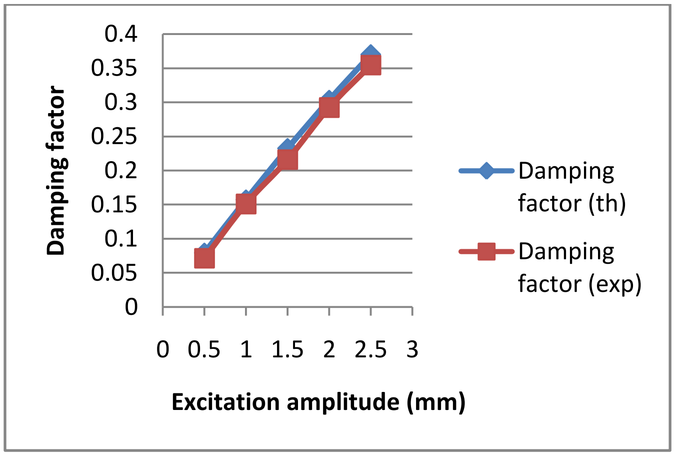

Chart-3: Comparisonplot of theoretical and experimental damping ratio versus excitation amplitude (90% orifice opening)

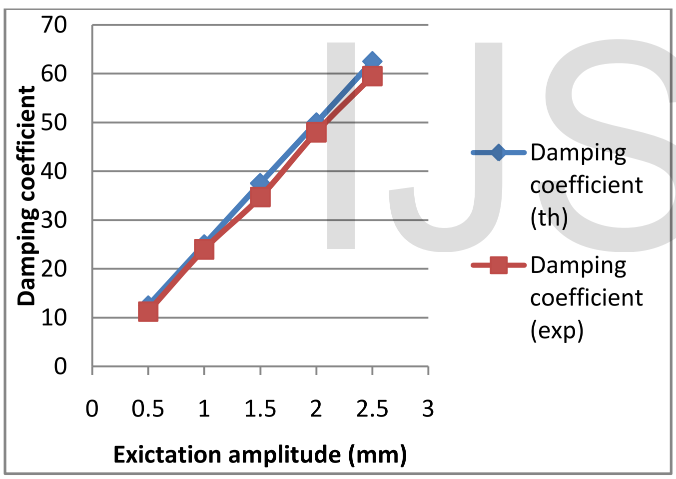

Chart-4: Comparisonplot of theoretical and experimental damping coefficient versus excitation amplitude (90% orifice opening)

4. CONCLUSIONS

The active suspension maintains the required comfort and stability due to the ability of adaptation in correspondence with the state of the vehicle. In this paper, the experiment was performed on the electro hydraulic active suspension system. Experimental results were examined and following conclusions were made,

1. Damping coefficient varies with respect to damper orifice size.
2. Damping coefficient increases with increase in excitation amplitude.
3. The theoretical and experimental damping coefficient shows the same nature and are within 5 percent variance the experimental results are validated
4. Damping ratio is less than 1 hence it is an under damped condition.
5. Electro hydraulic active suspension has the potential to change according to road conditions.

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[12] Intelligent optimal Controller of Active Suspension System Based on Particle Swarm Optimization

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