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Analysis of Real Size Variable Stiffness and Variable Damping Shock-Absorber for Vehicular Application

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Abstract — Variable stiffness and damping systems for shock-absorber has been studied in past by many researchers. Apart from variable stiffness and damping most of the authors have used MR dampers and effective length changing techniques to achieve variable stiffness. In this paper previously developed prototype reference has been taken and real size shock-absorber is designed. This will be fitted in the vehicle to investigate whether it follows the trend of performance given by prototype. For this purpose spring and damper arrangement has been changed by keeping mathematical arrangement same as prototype. The performance parameters investigated are force transmissibility and acceleration for different inputs of frequency and sprung mass. The fabrication work will be completed very soon and testing will be carried out.

Index Terms— Semi-active, Sprung mass, Suspension, Variable Damping, Variable Stiffness, Voigt element

1 INTRODUCTION

Fourth part of ISO 2631 defines methods for the measurement of periodic, random and transient whole-body vibration. It indicates the principal factors that combine to determine the degree to which a vibration exposure will be acceptable. Informative annexes in ISO 2631 indicate current opinion and provide guidance on the possible effects of vibration on health, comfort and perception and motion sickness. The frequency range considered is

— 0.5 Hz to 80 Hz for health, comfort and perception, and

— 0.1 Hz to 0.5 Hz for motion sickness. [11]

It is seen that the human body is not uniformly tolerant of different frequencies.

Road irregularities always cause vibration in ground vehicles. This vibration is unwanted because it can cause discomfort to passengers and drivers, and even damage to the vehicle systems. For these reasons, developing a well-performed vehicle suspension system has become necessary.

The basic suspension using the simple spring and damper is not ideal, but it is good enough for most purposes. For low-cost vehicles, it is the most cost-effective system. Therefore much emphasis remains on improvement of operating life, reliability and low-cost production rather than on refinement of performance by technical development. The variable damper, in several forms, has now found quite wide application on mid-range and expensive vehicles. On the most expensive passenger and sports cars, magetorheologically controlled dampers are now a popular fitment, at significant expense. [12]

Development of the adaptive damper has occurred rapidly. Although there will continue to be differences between commercial units, such systems are now effective and can be considered to be mature products. Fully active suspension offers some performance advantages, but is not very cost effective for passenger cars. Further developments can then be expected to be restricted to rather slow detail refinement of design, control strategies and production costs. Fast acting control, requiring extra sensors and controls, will continue to be more expensive, so simple fixed dampers, adjustable and slow adaptive types will probably continue to dominate the market numerically for the foreseeable future .[12]

Liu used variable stiffness and variable damping system for isolation purpose. It was initial days in this field and he shown that the can be controlled independently. Anubi has been working on variable stiffness concept for years. He has demonstrated various configurations over these years. Sun demonstrated the same idea with air springs as smart components and found that system suppress vibrations better for high speed train. Greiner-Petter proposed a mechanism which has three different selectable stiffness values with continuously variable damping coefficient.

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Different from all these Tchamna used variable stiffness suspension system for attitude control of ground vehicle travelling in cornering. In recent years researchers like Jugulkar, Twafik, Wu etc. come up with the experimental prototypes. But these prototypes are larger in size and could not be incorporated in vehicles. The necessity is to develop real scale suspension system with variable stiffness and variable damping. This shall be similar to conventional shock-absorbers but work better in terms of performance.

This paper presents a novel semi-active shock absorber. It is comparable with the conventional McPherson strut systems and hence can be fitted into four wheeled passenger car.

2 Variable Stiffness And Variable Damping System

2.1 Mechanical Configuration

This new semi-active suspension system consists of two springs and two dampers as shown in Fig. 1. Spring 1 and 2 has fixed stiffness k_1 and k_2. Dampers 1 and 2 have corresponding variable coefficients c_1 and c_2. Spring 2 and damper 2 are in series. Spring 1 is in parallel with these series elements. Even when the stiffness of both the springs is fixed, these three units together can produce varying stiffness value with variation in damping coefficient of damper 2. Again these three elements are in parallel with damper 1 forming Voigt element. If damping coefficient is very large, the resulting stiffness approaches to parallel stiffness of spring 1 and spring 2. If the damping coefficient of damper 2 is small enough, the total stiffness approaches to stiffness of Spring 1.

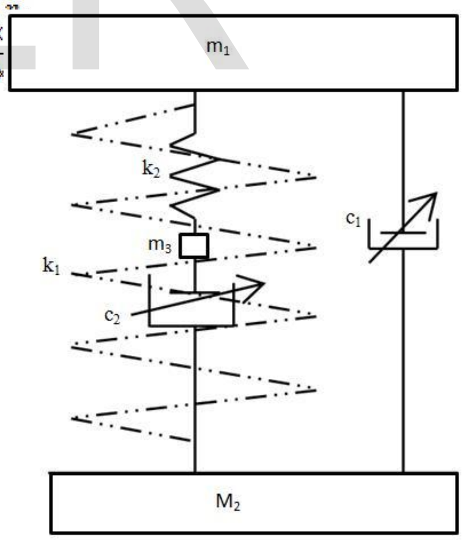

Fig. 1. Mechanical Configuration of proposed system

2.2 Equations of Motion

In Fig. 1, F represents excitation force. x, y and z corresponds to displacement of mass *m*, point between spring 2 and damper 2 and base respectively. The equations of motion for the system given in Fig. 2 is given a follows

$$m_1\ddot{x} = -k_1(x - z) - k_2(x - y) - c_1(\dot{x} - \dot{z}) \ldots\ldots\ldots \text{(1)}$$

$$k_2(x - y) = c_2(\dot{y} - \dot{z}) \ldots\ldots\ldots\ldots\ldots\ldots \text{(2)}$$

Equivalent stiffness of the system is determined by comparing potential energy of the actual and equivalent system, [9] that is

$$k_{eq} = (k_1 + k_2) - \left(\frac{4\pi^2 f^2 m_1 k_2}{k_2 + 2\pi f c_2}\right) - \left(\frac{k_2}{k_2 + 2\pi f c_2}\right) \ldots\ldots (3)$$

Where,

k_{eq} = Equivalent stiffness

k_1 & k_2 = Stiffness of inner & outer spring

c_2 = Damping coefficient

f = frequency of excitation

3 Design

3.1 Design of Springs

It is possible to design a number of springs for a given application by changing the three basic parameters, viz., wire diameter, mean coil diameter and the number of active turns. Before proceeding to design calculations, the designer should specify the limits on these diameters. Before going to design some assumptions are also needed to be made clear. It included assumption regarding maximum sprung mass for quarter car model, spring stiffness desired, spring index and spring material etc. [13]

- *Spring 1-*

Active no of turns,

$$n_a = \frac{G \times d}{8 \times C^3 \times k_1}$$

Where,

n_a = no. of active turns

G = modulus of Rigidity (N/mm^2) = 81370 N/mm^2 for all types of steel wires [13]

d = wire diameter (mm) C = spring index

k_1 = stiffness of spring (N/mm) Actual spring stiffness,

$$k_1 = \frac{G \times d}{8 \times C^3 \times n}$$

$$k_1 = \frac{81370 \times}{8 \times 10^3 \times}$$

$$k_1 = 13.56\ ^{N}/_{mm}$$

The actual deflection,

$$\delta = \frac{8 \times P \times C^3 \times n_a}{G \times d}$$

Where,

δ = deflection of spring (mm)

P = max. load applied (N)

$$\delta = \frac{8 \times 3237.3 \times 10^3 \times 6}{81370 \times 8}$$

$$\delta = 238.7096\ mm$$

Free length of Spring,

L = deflection in spring + coil diameter × No. of turns

= 238.7096 + 8 × 8

= 302.70 mm

The pitch will be calculated as

$$Pitch = \frac{Length\ of\ Spring\ (Active\ turns\ only)}{No.of\ Active\ Turns}$$

$$Pitch = \frac{286.7096}{6} = 47.78mm$$

- *Spring 2* –

Active no of turns,

$$n_a = \frac{G \times d}{8 \times C^3 \times k_2}$$

$$n_a = \frac{81370 \times 4}{8 \times 7^3 \times 26.6475}$$

$$n_a = 4.4 \sim 4$$

Actual spring stiffness,

$$k_2 = \frac{G \times d}{8 \times C^3 \times n_a}$$

$$k_2 = \frac{81370 \times 4}{8 \times 7^3 \times 4}$$

$$k_2 = 29.6538\ {}^{N}/_{mm}$$

Total deflection,

$$\delta = \frac{8 \times P \times C^3 \times n_a}{G \times d}$$

$$\delta = \frac{8 \times 3237.3 \times 7^3 \times 4}{81370 \times 4}$$

$$\delta = 109.1699\ mm$$

TABLE I. List of specifications of Spring

Parameter	**Value**	
	Spring 1	***Spring 2***
Spring Stiffness	13.56 N/mm	29.65 N/mm
No. of Turns	6+2=8	4+2=6
Wire Diameter	8 mm	4 mm
Mean Coil Diameter	80 mm	28 mm
Free Length	302.70 mm	133.17 mm

Free length of Spring,

L = deflection in spring + coil diameter × No. of turns

= 109.1699 + 4 × 6

= 133.17 mm

3.2 Damper Design

- Damping Coefficient –

The critical damping (cc) is defined as the value of damping coefficient c for which the mathematical term [14] [12] $\left(\frac{c}{2m}\right)^2 - \frac{k}{m}$ in differential equation of damped free vibration is zero,i.e.

$$\left(\frac{c}{2m}\right)^2 - \frac{k}{m} = 0$$

Critical damping cr = 2× $\sqrt{k \times m}$ = 2× $\sqrt{10659 \times 330}$

cr = 3750.983 Ns/m

Damping Ratio = ⊚ = $\frac{c_1}{c_r}$ = 0.4

c1 = 0.4 × cr = 1500.39 Ns/m

1. Damping Force –

Fd = c1×V Damper Velocity = V = 0.2m/s

Fd = 1500.39×0.2 = 300.07 N

Consider factor of safety F.O.S. = 5 Buckling Load (P) –

P = F.O.S. × Fd

= 1500.39 N

$$I = \frac{\pi}{64} \times (D^4 - d^4)$$

also

$$P = \frac{\pi^2 \times E \times I}{4 \times L^2}$$

Where

E = Modulus of elasticity (N/mm2)

I = Polar Moment of inertia (kg-m2)

So, Buckling Load is

$$P = \frac{\pi^2 \times E \times I}{4 \times L^2}$$

D = Outer diameter of Piston Rod (mm)

d = Inner diameter of Piston Rod = 2mm

L = Piston Rod length = 500mm

$$1500.39 = \frac{\pi^2 \times 2 \times 10^5 \times \left({}^{\pi}/_{64}(D^4 - 16^4)\right)}{4 \times 160^2}$$

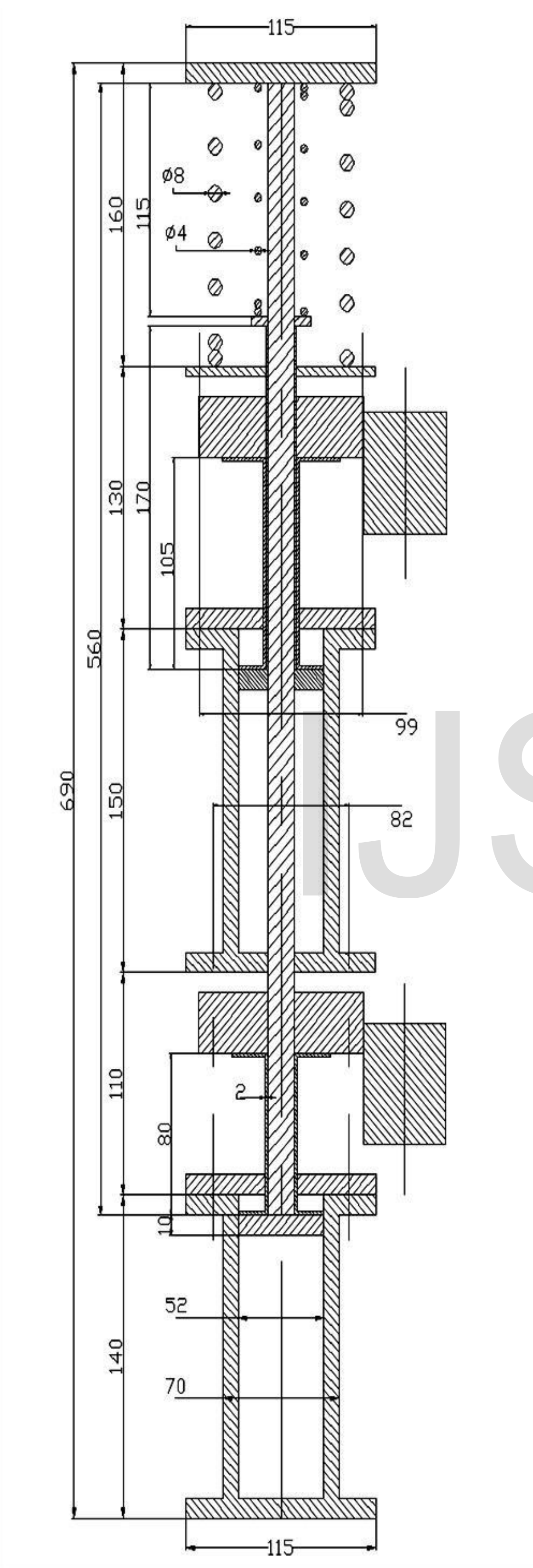

Fig. 2.Detailed Assembly Drawing

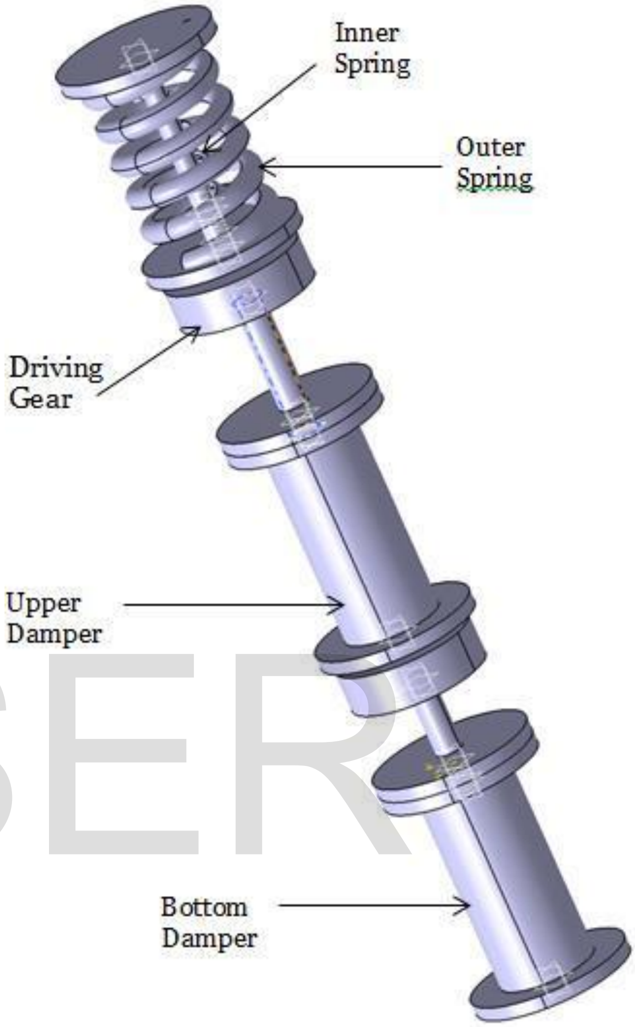

Fig. 3. 3D Assembly Drawing

D = 16.09 mm   16 mm

Consider Piston diameter is 3 – 4 times of piston rod diameter.

Dp = 52 mm

2. No. of orifices and size

$$Max.\ pressure\ applied, P = \frac{Force}{Area\ of\ Piston}$$

$$P = \frac{3732.3}{\left[\pi/_4 \times 0.052^2\right]}$$

$$P = 1525\ {}^{kN}/_{m^2}$$

For single orifice, Where,

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P = Drop in pressure across piston (kN/m2) ϱ = Density of fluid (kg/m3)
Vd = Velocity of fluid through orifice (m/s) f = friction factor
dp = thickness of piston (m)
d = diameter of orifice hole (m)

Velocity of fluid through the orifice holes is given by, from equation of conservation of mass
Where,
Vd = Velocity of fluid through orifice (m/s)

TABLE II. List of specifications of Dampers

Parameter	Value	
	Damper 1	***Damper 2***
Outer Diameter of Cylinder	70mm	70mm
Inner Diameter of Cylinder	52.3mm	52.3mm
Piston Rod Diameter	D_i =16.5mm D_o= 18mm	D =16mm
Piston Diameter	52mm	52mm
Width of Piston	20mm	20mm
Length of Cylinder	170mm	170mm
Length of Piston Rod	140mm	450mm
No. of Orifices	8	8
Orifice Diameter	2mm	2mm

V_p = Velocity of piston (m/s)
A_d = Total area of orifice holes (m^2)

For single orifice –

$$V_d = \frac{[\pi/_4 \times 0.052^2] \times 0.2}{[\pi/_4 \times 0.016^2]}$$

Therefore,

$$V_d = 211.25 \; {}^{m}/_{s}$$

$$\Delta P = \frac{0.809}{2} \times 211.25^2 \times \frac{0.3 \times 0.010}{0.0016}$$

$$\Delta P = 33.846 \; {}^{kN}/_{m^2}$$

Force = [P-] A_P

$$Force_{(1)} = [1525 - 33.846] \times \left[\frac{\pi}{4} \times (0.0052^2 - 0.0016^2)\right]$$

$$Force_{(1)} = 3163.79 \; N$$

Similarly, for 8 no. of holes

$$Force_{(8)} = 2643.459 \; N$$

A_p = Area of piston (m^2)

The minimum load on system will be 1000×9.81÷4 = 2452.5 N. in addition to this the orifice diameters will be taken as 2 mm as standard from designed 1.6 value.

Hence, force with 8 orifices open can sustain the minimum load and for maximum load all orifices will be closed. Therefore 8 no. of orifices in both pistons are finalized. The calculations for damper 2 are not repeated here as both of these are identical.

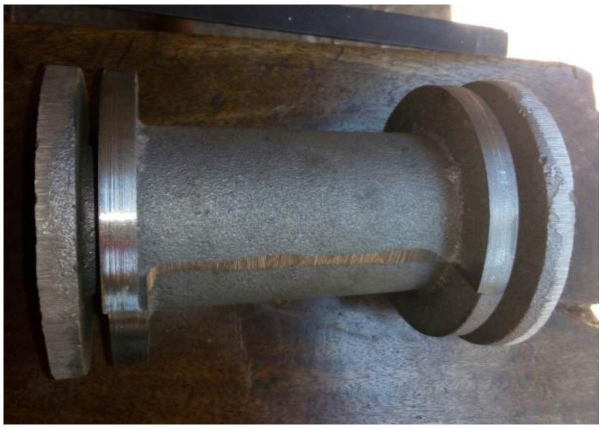

Fig. 4. Components under fabrication

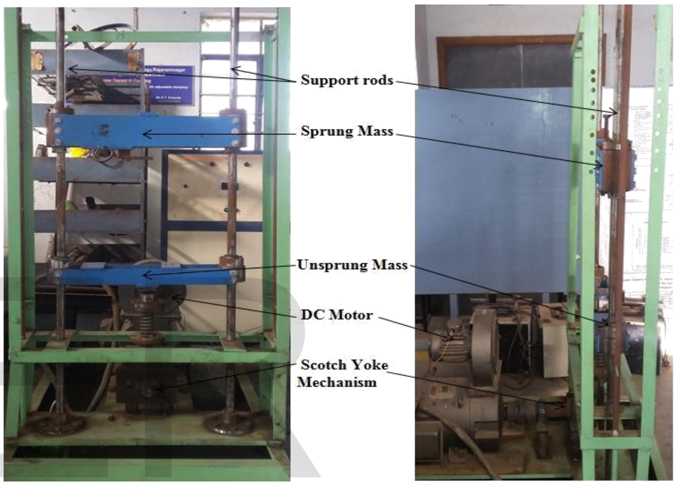

Fig. 5. Shock-absorber Test Rig.

4 EXPERIMENTATION

Test rig for testing shock absorber had been developed in labs. This test rig, as shown in Fig. 4, contains scotch yoke mechanism to convert rotary motion into reciprocating motion, where 7.5 kW power bars are supported with the help of sprung (33kg) mass and unsprung (23 kg) mass In between sprung and unsprung mass load cell are mounted to calculate force transmissibility. Designed shock absorber is mounted in between sprung and unsprung mass. Two accelerometers will be used for calculating acceleration velocity of sprung and unsprung mass. DC motor rpm will be varied with help of dimmer so that required frequency can be achieved. The fabrication work has been started as per the design and manufactured Damper cylinders are as shown in Fig.4.

5 Conclusion

Semi-active suspensions are being chosen over fully Active Suspension due to its reduced cost and less complexity in operation. This paper presents a novel configuration of shock-absorber for variable stiffness and damping configuration which will be fitted in vehicle along with modification in the McPherson strut. Total length of shock-absorber is 0.69m and radial dimensions are maximum 0.115m. The components have been manufactured by casting and machining processes. Low carbon Cast Iron and Aluminum alloy materials are used for components with casting, turning, honing and grinding operations. Mechanical design has been checked with standard procedures. The proposed system is analyzed for design and development according to standards and with the help of previous researches.

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