ISSN 2395-1621

Design and Fabrication of Non-linear Damper for Vehicle Suspension

#1Mr. Vijaysingh M Chavan,#2Prof. Chandrashekhar S. Dharankar

1vmchavan@aissmscoe.com

2csdharankar@aissmscoe.com

#1 Research Scholar -Mechanical Design Engineering, AISSMS College of Engineering, Pune-01.

#2 Assistant Professor -Mechanical Engineering Department, AISSMS College of Engineering, Pune-01

ABSTRACT

Vehicle dynamics has been a key research area due to its important role in ride comfort, vehicle handling, vehicle safety and vehicle performance. Increased competition in automotive market has forced industries to research alternative strategies to classical passive suspension system. Most passive suspensions are designed with linear spring and dampers. This work presents a novel design, fabrication methodology for an automotive suspension using concept of position dependent damping. Ride comfort perceived by passenger has been shown to depend on the r.m.s. acceleration of chassis and also on rate of change of acceleration. A good suspension system would stiffen up for larger rates and soften for smaller rates of acceleration input from the road when the car moves over bumps. This is related to certain desirable non-linearity that can be tuned actively. But, passive systems could also be tuned non-linearly, although it would not be possible to change the algorithm continuously as per road input as in active systems. Still, some significant improvements could bring about if the response of the system could depend upon rate of acceleration felt by chassis. This theoretical possibility has been realized in this paper with a practically possible passive suspension design with non-linear damper. This study focuses on a concept of Position-Dependent Damping (PDD) which was proposed in which the damping coefficient varies with position and the damper force is a function of both relative velocity and position or relative displacement across the damper. A suitable damping law is considered from literature for variation of damping coefficient with position for improving performance of passive vehicle suspension. Various parameters are considered for the complete specification of PDD. A new parameter 'Energy Dissipation Index' (EDI) is identified from literature for the performance analysis of dampers. The optimum parameters of the damping law are obtained on the basis of EDI. Literature and simulation results showed that a properly designed PDD reduces gap between the conflicting requirements the ride comfort and the road holding.

Keywords— **Non-linear damper, Position Dependent Damper, Energy Dissipation Index**

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

Ride comfort perceived by passenger has been shown to depend on the r.m.s. acceleration of chassis and also on rate of change of acceleration. A good suspension system would stiffen up for larger rates and soften for smaller rates of acceleration input from the road when the car moves over bumps. This is related to certain desirable non-linearity that can be tuned actively. But, passive systems could also be tuned non-linearly, although it would not be possible to change the algorithm continuously as per road input as in active systems. Still, some significant improvements could bring about if the response of the system could depend upon

rate of acceleration felt by chassis. This theoretical possibility has been realized in this paper with a practically possible passive suspension design with non-linear damper [1-2].

A literature search reveals that when the damper force depends on relative displacement along with relative velocity the performance is better. A very few literature is available on the PDD and there is no analysis available regarding the damping law for variation of damping coefficient with position. Dixon suggests that the position dependent damping can improve the ride performance of passive vehicle suspension [3]. Therefore passive hydraulic dampers with stroke sensitivity and similar refinements remain an attractive option for cost-conscious vehicle manufacturers. Fukushima et al. suggested optimum characteristics of automotive shock absorbers under various driving conditions and road surfaces. They showed that having the damper force as a function of velocity only was a serious limitation, and that variation with stroke was highly desirable [4].

Lee and Moon proposed a mathematical dynamic model of a displacement-sensitive shock absorber (DSSA) to predict the dynamic characteristics of an automobile shock absorber. A typical twin-tube type passive shock absorber with a longitudinally grooved pressure cylinder to relax the damping around the central position is considered in order to study the operating principles of the DSSA [5].

1.1 Energy Dissipation Index (EDI)

The Energy Dissipation Index (EDI) of a damper can be defined as "The ratio of energy dissipated by a damper (E_d) to the maximum possible energy (E_m) that can be dissipated with the maximum damping force generated by the damper in one cycle of sinusoidal excitation." It can be expressed as,

$$EDI = \frac{E_d}{E_m} \qquad (1)$$

$$\% \, EDI = \frac{E_d}{E_m} * 100$$

The EDI as defined above can be used for this purpose. A high value of EDI (more than 78.5%) signifies that the damper has good energy dissipation capacity and can be considered as a good damper for vibration isolation [7]. Since, in vibration isolation it is desirable to have lower damping force with higher energy dissipation so that acceleration and force transmission are low [7].

Now, If F_d = Damping force generated across the damper

$F_{d(max)}$ = Maximum damping force generated across the damper

Z = Amplitude of excitation across the damper

ω = Circular frequency of excitation

For sinusoidal excitation, the relative position, displacement and velocity between two ends of the damper are,

$$z = Z\sin(\omega t), dz = Z\omega\cos(\omega t) = \dot{z} \qquad (2)$$

1.2 Concept of PDD Coefficient

In vehicle suspension system, any kind of road excitation produces relative motion across the damper such that relative velocity decreases towards end of stroke and zero at the end. This generates smaller damping force near the end of stroke resulting in lower energy dissipation in case of conventional damper. The position dependent damper (PDD) can be designed such that it has low damping coefficient at equilibrium position and slowly increases to maximum towards the end of stroke. When the vehicle suspension is subjected to low amplitude excitations (high frequency) the PDD woks like a conventional soft damper to provide ride comfort. On the other hand near resonance as amplitude tries to build it provides high damping to attenuate the vibration.

The PDD coefficient can be expressed as,

$$C_z = C_0 + C_p|Z|^d$$

Where, z = Position of damper piston from static or mean equilibrium position with respect to cylinder C(z) = Damping coefficient at position z from mean position

Co = Minimum damping coefficient or damping coefficient at MEP i.e. at z = 0

Cp = Position dependency coefficient

d = Position dependency index

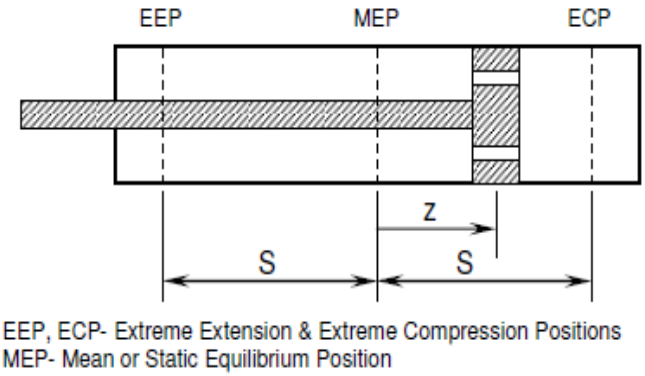


Fig. 1 Schematic of Piston and Cylinder [7]

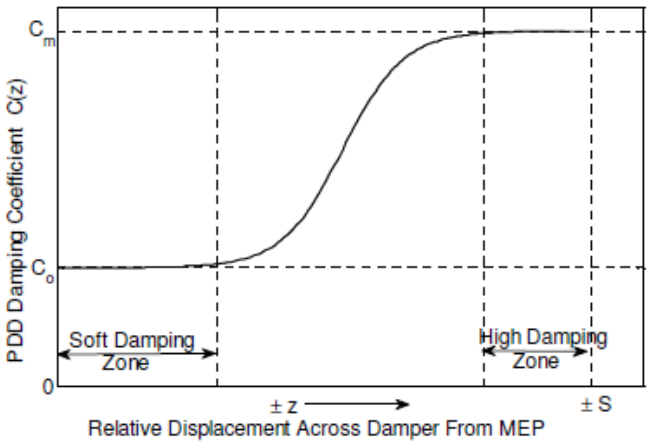


Fig. 2 Variation of damping coefficient in PDD [7]

Equation 3 gives the simplest law for the variation of damping coefficient with relative displacement (z). But this has limitation of giving smooth variation between minimum and maximum damping coefficients. A curve shown in Figure-9 is desirable [7]. Therefore a law for the variation of damping coefficient is proposed using a hyperbolic tangent function as follows.

$$C_z = \frac{C_0}{2} * \left\{(h+1) + (h-1)tanh\left[\frac{a\,|Z|}{b} - b\right]\right\} \qquad (3)$$

S = Maximum stroke of damper piston from mean position

Co = Minimum damping coefficient i.e. damping coefficient at mean position z = 0
Cm = Maximum damping coefficient i.e. damping coefficient at extreme position z = +/- S
h = Factor defining maximum damping coefficient Cm.
a = Slope Parameter defining the slope or steepness of Damping Coefficient curve.
b = Soft Zone Parameter defining the length of soft damping zone.

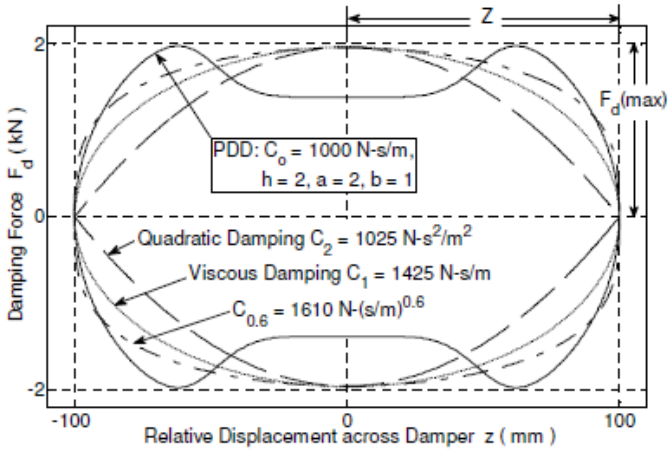


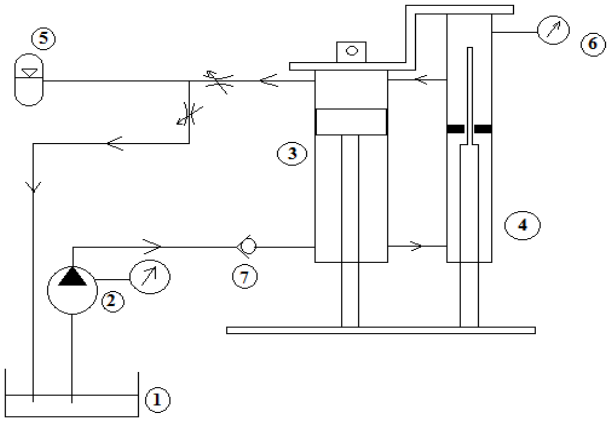
Fig. 3 Work diagram for different types of damping under sinusoidal excitation (f = 2.2 Hz) [7]

Figure 3 shows the effect of various types of damping on the work-diagram. It is possible for PDD to increase the area of work-diagram (Energy Dissipation) and therefore EDI by proper selection of its parameter. It is to be noted for PDD that, the energy dissipation index (EDI) is independent of excitation frequency (ω), minimum damping coefficient (Co) and maximum stroke of damper piston (S) but depends on amplitude of excitation (Z) and parameters h, a, d.

II. EXPERIMENTAL POSITION DEPENDENT DAMPER

In a Position Dependent Damper (PDD) the damping coefficient varies with position and the damper force is a function of both relative velocity and position or relative displacement across the damper. The position dependent damping can be achieved practically, by providing the bypass in the form of longitudinal tapered grooves on the cylinder of hydraulic damper and allowing the oil to flow through these grooves around the piston. This varies the area of passage to oil flow during displacement of piston. When the piston is near the mean position there is maximum area for the flow of oil and therefore lowest damping coefficient. As piston moves towards the end of stroke bypass area decreases and damping coefficient increases.

Fig. 4 Schematic flow diagram of Experimental Damper



1.) Reservoir 2.) Oil Pump 3.) Main cylinder 4.) Auxiliary cylinder 5.) Accumulator 6.) Pressure gauge. 7.) Non Return valve

Fig. 5 Pictorial View of Non-linear Damper

It was decided to design the damper using a standard hydraulic cylinder with an auxiliary valve arrangement. The auxiliary valve provides the orifice for the damper and also gives change in area available for oil flow. This way is adopted because of simplicity for identification of different parameters.

Force acting on piston when accumulator is connected to cylinder above piston side. Therefore, accumulator is connected to cylinder above piston in order to reduce the force acting on the piston.

Fig. 6 Determination of orifice plate thickness

Flow of oil in an experimental damper is as shown in figure 4. Non-return valve is used to prevent back flow of oil towards pump. Accumulator is used to compensate volume of rod entering into cylinder.

Allowable tensile stress of copper (σ) = 73.33 Mpa

Inner diameter of the cylinder (d) = 25mm

Maximum pressure subjected to the cylinder (P) = 100 Bar

To determine the Thickness (t),

$\sigma_x = \sigma_y = 73.33 N/mm^2$

d= 25mm

P= 100Bar= $10 N/mm^2$

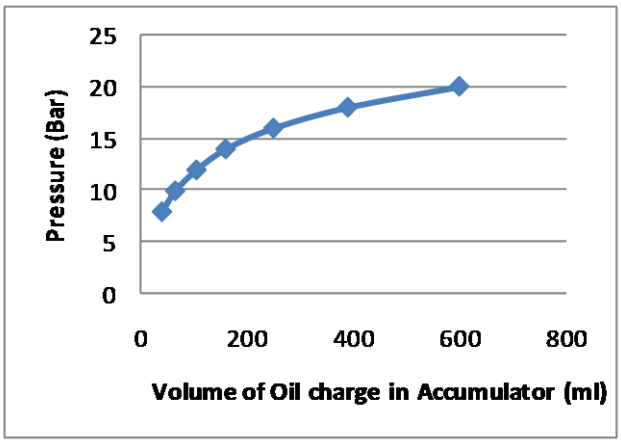
Hoop's Stress is given by,

$\sigma_y = P * d/2t$

73.33= 10*25/2t

t = 1.704mm

Longitudinal Stress is given by,



$\sigma_x = P * d/4t$

73.33= 10*25/4t

t = 0.8523mm

Hence, the required thickness is 1.704mm

Thus thickness is selected from the point of manufacturing,

As t= 4mm.

2.1 Buckling Calculation of Auxiliary Cylinder Rod:

Length of the rod under consideration, l = 230mm

Effective Length, Le = 2L = 2*230 = 460mm.

Diameter of rod, d = 12mm.

Young's Modulus (Mild Steel), E = 210GPa = $210*10^3$ N/mm^2

Area of rod (A) = $\pi/4*d^2$

$A = \pi/4*12^2$

$A = 113.09\ mm^2$

Moment of Inertia (I) = $\pi/64*d^4$

$I = \pi/64*12^4$

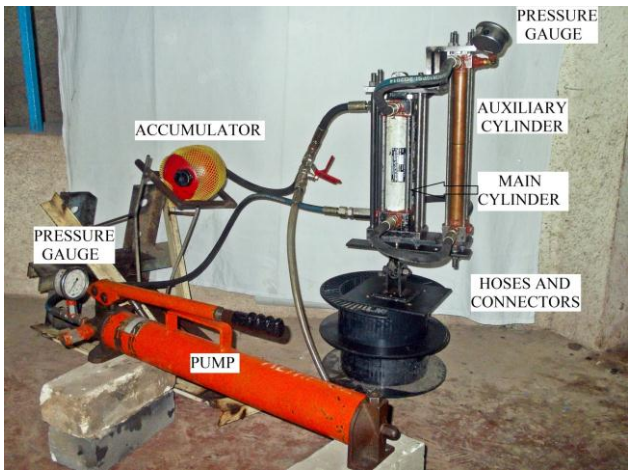
$I = 1017.87mm^4$

For one fix and one end free,

Critical load, $P_{cr} = \pi^2 EI/Le^2$

$P_{cr} = (\pi^2*210*10^3*1017.87)/(460^2)$

$P_{cr} = 9970N$



Critical Stress, $\sigma_{cr} = P_{cr}/A$

σ_{cr}=9970/ 113.09

σ_{cr}= 88.159 N/mm^2

Max. Operating Pressure in the System = 200Bar = 20N/mm^2

Thus, FOS = 88.159/20

FOS = 4.4

Now, S_{ut}= 430 N/mm^2 (For mild Steel)

∴Allowable stress = 430/ 4.4

= 97.72 N/mm^2

As $\sigma_{cr}<\sigma_{all}$, Rod is safe in Buckling.

2.2 Calculation for Dimensions of Orifice Plate:

2.2.1 Calculation for Thickness:

Assume,

- The orifice plate as a cantilever beam.
- Pressure differential across the orifice plate in dynamic condition = 20Bar

Shearing Area, $A_s = \pi*D*t*\tau_s = (P_1 - P_2)*A$

$A = \pi/4\ (D^2-d^2)$

$A = \pi/4\ (24^2-6.3^2)$

$A = 421.21\ mm^2$

$\pi * 24* t* 115 = (20*10^5*421.21)/ 10^6$

t = 2.15mm

Thus, we selected, t = 3mm

2.2.2 Calculation for Inner Diameter of the orifice plate:

Outer diameter of Orifice, d_o= 28mm

Area of opening, $A = \pi/4*(d_i^2-d_r^2)$

Where, d_i = inner diameter of orifice

d_r = rod diameter

By varying the orifice area (A), we have obtained various orifice diameters d_i for calculating damping force for varying orifice area:

TABLE I

SIZES OF ORIFICE PLATES

Area of Opening, A (mm^2)	Inner Diameter, d (mm)
3	6.31
6	6.606
9	6.89
16	7.51
22	8
26	8.31
32	8.76

Fig. 7 Pictorial view of some orifice plates

Figure 7 shows pictorial views of orifice plates manufactured for use in auxiliary cylinder of non-linear damper.

III. RESULTS AND DISCUSSIONS

Study of the pressure volume characteristics for determination of pre-charge pressure of accumulator is carried out. Accumulator is precharged at different pressure levels. Results are represented in fig 8 to 11.

Fig.8 Pressure Vs Volume of oil (Precharge=5 Bar)

Oil is pressurised into the system. Then slowly pressure is released and volume of oil coming out is measured. This procedure is repeated for different conditions of precharge pressures.

Fig.9 Pressure Vs Volume of oil (Precharge=10 Bar)

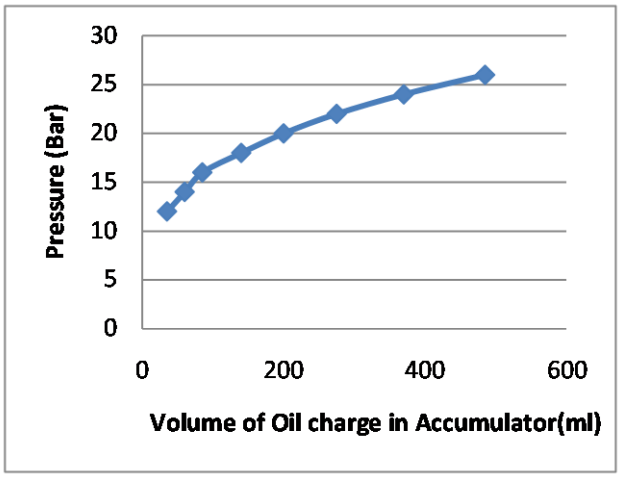
Fig.10 Pressure Vs Volume of oil (Precharge=15 Bar)

Study of the pressure volume characteristics gives basis for determination of pre-charge pressure of accumulator. According to these characteristics, Accumulator is precharged at different pressure levels.

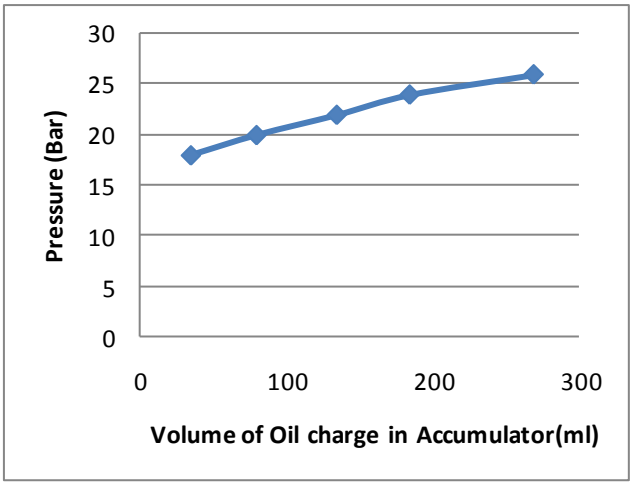
Fig.11 Pressure Vs Volume of oil (Precharge=18 Bar)

According to the specifications of the exciter the maximum force exerted is 50KN. It is recommended that the maximum amount of oil to be accumulated after the displacement of the rod. Interpretation from graph shows that increase in pressure increases volume of oil accommodated. For a force of 50KN, the required pre charge pressure should be maximum up to 18bars. It is required that the maximum amount of oil to be accumulated after the displacement of the rod.

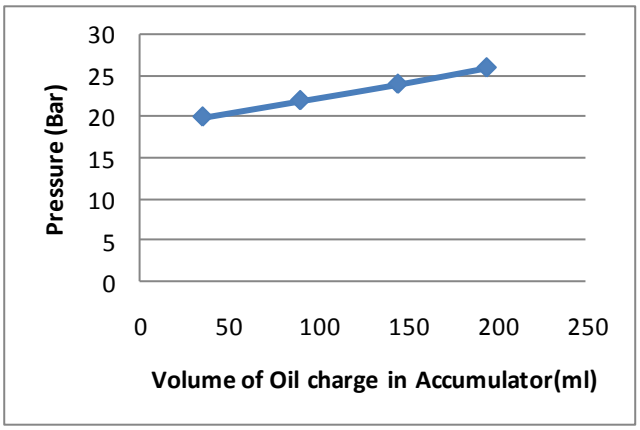
IV. CONCLUSIONS



The objective of this proposed work is to identify the



parameters for complete specification of position dependent damping (PDD) and to show the potential of PDD concept for improving the performance of passive vehicle suspension system. The PDD parameters are obtained on the basis of maximization of EDI. A suitable damping law is considered from literature for variation of damping coefficient with position for



improving performance of passive vehicle suspension. Various parameters are new parameter 'Energy Dissipation Index' (EDI). A conventional passive suspension, which is designed for ride comfort (soft damping) has a tendency of suspension bottoming and large fluctuation of road holding force (normal force between tire and road) near resonance. On the other hand a passive suspension, which is designed for optimum handling (heavy damping), has lower suspension travel and less fluctuation of road holding force near resonance but higher acceleration transmissibility with discomfort at high excitation frequencies above resonance. Higher acceleration transmissibility is associated with the high fluctuation of road-holding force and therefore reduced vehicle handling quality. A passive suspension with PDD works like a conventional soft damper at high excitation frequencies which are usually of low amplitude, on the other hand if resonance gets excited and amplitude tries to build it provides more damping to control it. Therefore a properly designed PDD has ability to reduce the gap between conflicting requirements of the ride comfort and the road holding.

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