SHOCK ABSORBER TEST SET-UP

Sagar D. Shinde[1], Jeevan J. Jagdale [2], Ashish A. Salunkhe [3], Aditya V. Shinde[4], Sushant R. Koli[5]

[1,2,3,4,5] *Department of Mechanical Engineering, D.I.E.T. Sajjangad, Satara, Maharashtra, India.*

Abstract — *Shock absorber is one of important component in a vehicle suspension system. The shock control spring motion by damping energy from the spring. This paper was focused on the dynamic characteristics of an automotive shock absorber. The design of interchangeable shock absorber test rig was developed and fabricated for the dynamics measurement system. This test setup integrated with the computer systems to record the signal. An experiment was conducted to identify the stiffness and damping parameter for 850 cc and 1600 cc shock absorber. Simulation study was performed utilizing the COSMOS Motion software. It can be seen from the results that there is a good agreement between the experimental and simulated results in terms of stiffness and damping value except few discrepancy. The acquired results show that the range of discrepancy within 10%. The good range of stiffness of the passenger vehicle shock absorber is 20 N/mm to 60 N/mm while the damping of passenger vehicle shock absorber is 1 Ns/mm to 6 Ns/mm.*

***Keywords**-Shock absorber, COSMOS motion software, Structural Steel, Pedastal Bearing, Load cell.*

I. INTRODUCTION

Suspension system is an assembly used to support weight, absorb and dampen road shock, and help maintain tire contact as well as proper wheel to chassis relationship. A vehicle in motion is more than wheels turning. As the wheel revolves, the suspension system is in a dynamic state of balance, continuously compensating and adjusting for changing driving conditions. Suspension of vehicle need to analyse before be manufacturing. This is because to make sure components in shock absorber system remain in good conditions. On the other hand, shock absorber system need to analyse how shock to see how they going to perform in worst-case scenario.

A safe vehicle must be able to stop and man over a wide range of road conditions. Good contact between the tires and the road will able to stop and man oeuvre quickly. Suspension is the term given to the system of springs, shock absorbers and linkages that connects a vehicle to its wheels. Shock absorber is an important part of automotive suspension system which has an effect on ride characteristics. Shock absorbers are also critical for tire to road contact which to reduce the tendency of a tire to lift off the road. This affects braking, steering, cornering and overall stability. The removal of the shock absorber from suspension can cause the vehicle bounce up and down. It is possible for the vehicle to be driven, but if the suspension drops from the driving over a severe bump, the rear spring can fall out. Basically, the shock absorbers must be replaced after driving exceeds certain distance. But this actually not should have been followed if there are no defective. To ensure there are no defective, the consideration to check the condition of the shock absorber is the best way. There are several methods to test the condition of the shock absorber. One of the tests is endurance test use frequency response method which applied to the suspension and tire. But this test is very complicated and costly because the machine is very expensive.

The other method to check the shock absorber without removing from the vehicle is by using the bounce test. In the workshop, some mechanics will perform bounce test which is push down each corner of the car several times to check the condition of shock absorber. However, the result of this test is not very accurate to indicate the condition of shock absorbers. The objectives of this paper are to determine the dynamic characteristics of automotive shock absorber systems. Running vehicles are exposed to almost constant vibration excitation; shock (i.e. vibration) absorbers are consequently required for reasons of driving safety and riding comfort. These aims partly conflict, because a taut suspension prevents wheel hopping and thus a loss of road contact, whereas a soft suspension is supposed to reduce body vibration and thus the annoying effects of acceleration on the occupants. For many years, the semi-elliptic leaf springs used on carts and carriages were the most common method of providing suspension springing. This was a neat and simple arrangement, but it unfortunately produced a number of problems such as a tendency of the axle to wind up around the springs on braking or acceleration. Pneumatic variable height suspension was developed for family cars by Citroën, and has also been used on commercial vehicles. Now-a-days the dampers used in almost all the passenger vehicle are hydraulic. This is because the damping effect due to hydraulic oil is much more effective than the conventional methods of damping.

II. DESIGN OF COMPONENT

2.1 Design of structural frame:

The frame is fabricated from C-channels of dimensions 3.5'X 1.7'and solid circular rod having O.D 50mm. The design consists of rectangular base having dimensions 1200 X 896mm. Two opposite pillars are connected by a

horizontal member. The top member i.e. channel is not fixed therefore it provides coarse adjustment to accommodate any type and length of dampers.

Table 2.1: Standard C-Channel dimensions

Imperial Size	Depth (d)	Flange Width (b)	Flange Thick (t)	Web Thick (w)	Metric Size
C 3 x 4.1	3	1.41	0.273	0.170	C 75 x 6
C 3 x 5	3	1.50	0.273	0.258	C 75 x 7
C 3 x 6	3	1.60	0.273	0.356	C 75 x 9
C 4 x 5.4	4	1.58	2.96	0.184	C 100 x 8
C 4 x 6.25	4	1.65	2.96	0.247	C 100 x 9
C 4 x 7.25	4	1.72	2.96	0.321	C 100 x 11
C 5 x 6.7	5	1.75	0.320	0.190	C 130 x 10
C 5 x 9	5	1.88	0.320	0.325	C 130 x 13
C 5 x 11.5	5	2.03	0.320	0.472	C 130 x 17
C 6 x 8.2	6	1.92	0.343	0.200	C 150 x 12
C 6 x 10.5	6	2.03	0.343	0.314	C 150 x 16
C 6 x 13	6	2.16	0.343	0.437	C 150 x 19
C 7 x 9.8	7	2.09	0.366	0.210	C 180 x 15
C 7 x 12.25	7	2.19	0.366	0.314	C 180 x 18
C 7 x 14.75	7	2.30	0.366	0.419	C 180 x 22
C 8 x 11.5	8	2.26	0.390	0.220	C 200 x 17
C 8 x 13.75	8	2.34	0.390	0.303	C 200 x 21
C 8 x 18.75	8	2.53	0.390	0.487	C 200 x 28
C 9 x 13.4	9	2.43	0.413	0.233	C 230 x 20
C 9 x 15	9	2.48	0.413	0.285	C 230 x 22
C 9 x 20	9	2.65	0.413	0.448	C 230 x 30

2.2 Material Selection:

We have selected C-channel made from Structural Steel. C-channels made of **St 30 Grade** have been selected.

2.3 Cam and follower:
The specifications of cam and follower are as follows:
   1. Eccentric cam and roller follower.
   i. Maximum displacement = 50 mm
   ii. Angle of rise = 180 degree
   iii. Angle of return = 180 degree
   iv. Diameter of base Circle = 50 mm
   v. Diameter of Roller Follower = 30 mm
   2. Rise and return considering SHM.
   vi. Thickness of Cam disc = 16 mm
   vii. Eccentric radius of cam = 25 mm
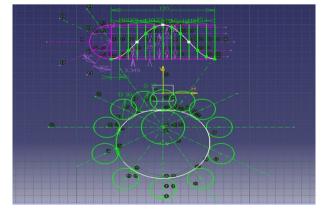

Figure 2.1 Cam Profile and Displacement Diagram

2.4 Pedastal Bearing and Jaw coupling:
Based upon the maximum load (Radial load) acting on the shaft i.e. 1000N and assumed, rating life of bearing to be 12000 hours and inner race factor to be 1.2. Based upon load and assumptions we selected a bearing (P204) having I.D of 20 mm.

Figure 2.2 Pedestal Bearing

As the maximum torque to be transmitted by the motor and gearbox is 50Nm Based upon which we selected a jaw coupling from the catalogue having dimensions as follows:

 i.   Input shaft diameter   =   14.5 mm
 ii. Output shaft diameter   =   20 mm
 iii. Outer diameter of hub   =   32 mm

2.5 Drive System:
The Drive system is an integral part of the suspension test rig setup. The major function of the drive system is to produce the exciting force and to transmit this force to the Cam mechanism.
The tasks required to be performed by the drive system are:
 i.   To provide the driving force with the help of prime mover.
 ii. To couple the output shaft of motor and the input shaft of the gearbox.
 iii. To convert the speed of the motor to the required speed with the help of the gearbox.
 iv. To connect the output shaft of the gearbox to the shaft of the cam disc.

2.5.1 Drive System Requirements:-
Taking into consideration different factors, the required speed from the gearbox has been chosen as 300 RPM and the Motor speed has been chosen as 3000 RPM.
Torque requirement:-
Torque: $T = F \times r$
$\qquad = (mg) \times r$
Where,
T - Torque required.
F - Force at the damper.
r - Moment arm or torque arm.
Thus, for a 100 kg force acting at a moment arm of 0mm the torque required is:-
$T = F \times r$
$\quad = (mg) \times r$
Power requirement:
$$\mathbf{P} = \frac{2 \Pi N T}{60}$$
$$= 1.08678 \ KW$$
Where,
P - Minimum motor power.
N - Motor speed in RPM.
T - Torque required.
For maximum motor speed of 300 rpm we get,
Based on the above requirements the different components selected in the drive system are as follows:-

2.5.2 DC Motor:
Taking into consideration the possibility of testing dampers with a greater damping force at higher velocities the motor with the following specifications has been selected.
Speed:-Maximum speed available 3000 rpm which can be controlled using speed regulator (dimmer stat)
Power:-1.5 HP (1.12 KW). Shaft size: 15 mm.
Advantages:
 i.   DC motor provides us with more Flexibility as we can operate the test rigs at a required speed.
 ii. Due to higher power capacity (1.5 HP) motor, Dampers with high Damping coefficient can also be tested with ease.

2.6 Reduction System:

The main function of the reduction system is to provide an efficient speed reduction as well as to achieve required torque from the specific motor speed to the required actuation speed. In our case we require a reduction system to reduce the motor speed of 3000 RPM to the maximum actuation speed of 300 RPM. To accomplish this task, there are various types of reducers available, some of these are:-

  i.   Chain sprocket
  ii. Belt and pulley
  iii. Gearbox

2.6.1 Features of gear reducer:

Advantages:
  i.   Compact drive requirements demand the shortest possible distance between shaft centers.
  ii. High speed ratios are required.
  iii. High rotating speeds (RPM) are required.
  iv. High horsepower AND high speed loading is required.
  v.   High Efficiency.

2.7 Riverted gear train:

When the axes of the first gear and the last gear (i.e. last driven or follower) are co-axial then, the gear train is known as riverted gear train.

Spur gear terminology used in calculation:
T1 = Number of teeth on gear no 1
R1 = Pitch circle radius of gear no1
N1 = Speed of gear no 1 in rpm

Similarly,
T2, T3, T4   = Number of teeth on respective gears.
R2, R3, R4   = Pitch circle radii of respective gear.
N2, N3, N4   = Speed of respective gear in rpm.
Pc   = Circular pitch
M   = Module
Φ   = Pressure angle
X   = Distance between two shafts.

Input Data:
  i.   Shaft diameter   = 20 mm
  ii.   Pressure angle of spur gear   = Φ20
  iii. Input speed = N1   = 1000 rpm
  iv.   Output speed =N4   = 200 rpm
  v.   Module of gears   = 1.5 mm
  vi.   Distance between two shaft=R = 120 mm

Solution:
We know that circular pitch,
$$Pc = \frac{2\pi R}{T} = \pi M \text{ or } R = \frac{MT}{2}$$
$$R1 = \frac{MT1}{2} \quad, \quad R2 = \frac{MT2}{2}, \quad R3 = \frac{MT3}{2}, \quad R4 = \frac{MT4}{2}$$
$$\frac{MT1}{2} + \frac{MT2}{2} = \frac{MT3}{2} + \frac{MT4}{2}$$

Therefore,   T1 + T2 = T3 + T4
Since the speed ratio between the gears a & b and between the gears c & d are to be same., therefore ,

$$\frac{Na}{Nb} \quad \frac{Nc}{Nd} = \frac{\sqrt{Na}}{\sqrt{Nd}} = \frac{1000}{200} = 2.24$$

We know that, $\frac{Nb}{} = \frac{Ta}{}$

$$Na \qquad Tb$$

Since, $Na = 1$ revolution therefore, $Nb = \dfrac{Ta}{Tb}$

Also the speed ratio of any pair of gear in mesh is the inverse of their Number of teeth therefore,

$$\frac{Tb}{Ta} = \frac{Td}{Tc} = 2.24$$

We know that the distance between the shafts:

$$X = Va + Vb = Rc + Rd = 120 \text{ mm}$$

OR

$$\frac{Ma\,Ta}{2} + \frac{Mb\,Tb}{2} = \frac{Mc\,Tc}{2} + \frac{Md\,Td}{2} = 120$$

$$1.5\,(Ta + Tb) = 1.5\,(Tc + Td) = 240$$

$$Ta + Tb = \frac{240}{1.5} = 160$$

$$Tc + Td = \frac{240}{1.5} = 160$$

And From equation above,

$$Tb = 2.24$$

Since,

$$Ta + 2.24\ Ta = 160$$

$$3.24\,Ta = 160$$

$$Ta = 49.38 \quad \text{say} \quad 50$$

$$Tb = 160\text{-}50 = 110$$

And From equation $Td = 2.2\ Tc$

$$Tc + 2.24\ Tc = 160$$

$$Tc = \frac{160}{3.24}$$

$$Tc = 49.36 \ \text{Say} \ 50$$

$$Td = 160\text{ -}50 = 110$$

By Checking,

$$\frac{Na}{Nd} = \frac{Tb * Td}{Ta * Tc}$$

$$Ra = \frac{m\,Ta}{2}$$
$$= \frac{1.5 \times 50}{2}$$
$$Ra = 37.5 \text{ mm}$$

$$Da = 75 \text{ mm}$$

Therefore,

$$Rb = 82.5 \text{ mm}$$
$$Rc = 37.5 \text{ mm}$$
$$Rd = 82.5 \text{ mm}$$

III. EXPERIMENTAL SETUP

Displacement sensitive shock absorber is actuated with the help of D.C. Motor. The apparatus and experimental strategy are shown in figure Control unit is connected to D.C motor to control its rpm. D.C. motor is coupled to Gear Box. The output shaft of the gear box is rotating about vertical axis at same speed as that of the D.C motor. The Optimization of Displacement Sensitive Twin Tube Shock Absorber 25 output shaft of the gearbox is coupled with cam disc on which cam profile is welded. The lower mount of the shock absorber is placed on the cam profile with the help of ball bearing. The shock absorber is constrained to move in only one direction in order to justify the assumption of single degree-of-freedom (SDOF) behavior.
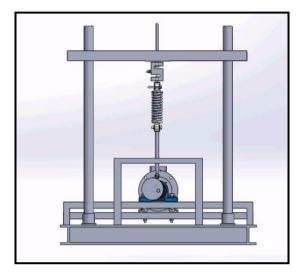

Figure 3.1 Cad model of the shock absorber test rig

The top of the absorber is attached to a load cell so that the internal force could be measured directly (it was found that inertial forces were negligible). S-type of load cell is used having 500kg capacity. Accelerometer is also connected at the top. The displacement transducer is connected at lower mount of shock absorber. Data Acquisition system records the data from load cell, accelerometer and displacement transducer. To prove that dampers are strongly nonlinear and also frequency dependent, it shock is decided to carry out the first set of tests at various frequencies in order to see the effects of the nonlinearity and frequency dependency. However, if the displacement response of a system is a single harmonic, force data are only available above the corresponding phase trajectory that is simply an ellipse. For this reason cam profile is made to provide ramp excitation signals at 50 mm amplitude as shown in figure.
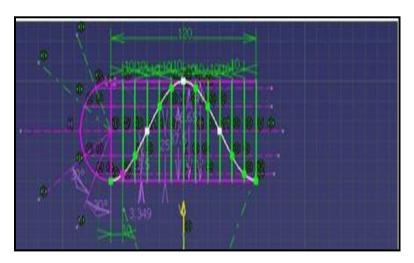

Figure 3.2 Schematic diagram of the cam profile

Ideally, force data are required that are evenly distributed in the phase plane. In order to meet this requirement without violating the requirement of a single response frequency, several tests were carried out at each frequency, each subtest was for a different response amplitude. Once the subtest data were assembled at each frequency, this procedure gave force data over a set of concentric curves in the phase plane. This allowed the construction of a force values for each test frequency, an iso-frequency curves.

Comparison of such surfaces then indicates if the absorber under test is frequency dependent. Because automotive dampers or shock absorbers are designed to be significantly nonlinear, it is untenable to model them as linear systems. In order to correctly simulate the behavior of such shock absorbers, it is essential that one characterizes the actual nonlinearities present in the structure. A traditional approach to characterization of the nonlinearities present in the shock absorber is accomplished by obtaining a force–velocity or characteristic diagram. The force data from a test are simply plotted against the corresponding velocity values. These diagrams show hysteresis loops, that is, a finite area is enclosed within the curves.

The comprehensive approach illustrated here is to use measured data to construct the restoring force surface for the absorber. This simultaneously displays the position and velocity dependence of the restoring force in the absorber without a priori knowledge of the structure. The shock absorber test facility essentially took the same form as Fig. 5.3.The piezoelectric load cell provided a measurement of x(t). The other signal measured is displacement, the required velocity and acceleration being arrived at by data acquisition system. The shock absorber incorporates an LVDT displacement transducer that produces a high quality signal.

3.1 Data Acquisition System
Load cell:

A load cell is a transducer which converts force into a measurable electrical output. Although there are many varieties of load cells, strain gauge based load cells are the most commonly used type. In the project a S type load cell with a capacity of 500 kg (tension and compression) has been used.
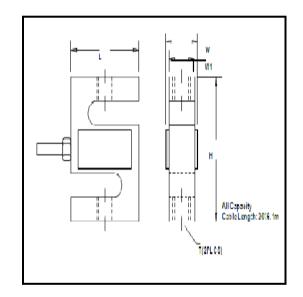

Figure 3.3 S- Load Cell

Main Function: - To measure the damping force values from the damping force testing setup

Table 3.1 S-Load Cell Specifications Table

Full scale output	mv/v	2.001
Combined error	% FS	0.02
Non Linearity	% FS	0.02
Repeatability	% FS	0.015
Hysteresis	% FS	0.02
Creep	% FS/30 min	0.02

IV. OBSERVATIONS
4.1 Transmissibility:
By plotting the curve of displacement as shown in figure we can plot the transmissibility curve for different values of speed.
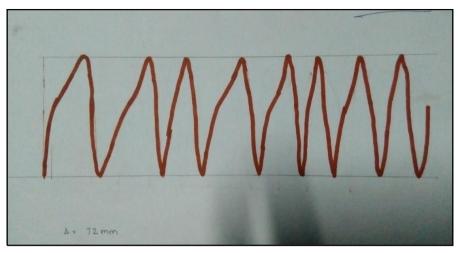
Figure 4.1 Displacement Curve at 70 RPM

Table 4.1 Observation table for different values of speed

Sr No.	Speed (rpm)	ω	MASS (KG)	STIFFNESS(K) N/m	ωn	X	Y	X/Y	ω/ωn
1	100	10.467	62	12210.58	14.034	36	50	0.72	0.746
2	110	11.513	62	12210.58	14.034	37	50	0.74	0.820
3	120	12.560	62	12210.58	14.034	42	50	0.84	0.895
4	130	13.607	62	12210.58	14.034	42.5	50	0.85	0.970
5	140	14.653	62	12210.58	14.034	46	50	0.92	1.044
6	150	15.700	62	12210.58	14.034	46.5	50	0.93	1.119
7	160	16.747	62	12210.58	14.034	47	50	0.94	1.193
8	170	17.793	62	12210.58	14.034	48	50	0.96	1.268
9	180	18.840	62	12210.58	14.034	49	50	0.98	1.342
10	190	19.887	62	12210.58	14.034	50.1	50	1.002	1.417
11	200	20.933	62	12210.58	14.034	48.7	50	0.974	1.492
12	210	21.980	62	12210.58	14.034	48	50	0.96	1.566

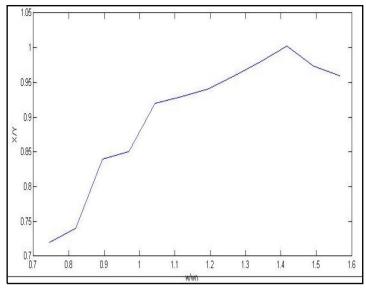**4.2 TRANSMISSIBILITY CURVE:**

Figure 4.2 Transmissibility Curve

The above graph is plotted against the ratio of $\omega/\omega n$ to the transmissibility ratio i.e. x/y. From the above curve we get to know that at $\omega/\omega n$ equal to $\sqrt{2}$ or 1.

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