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Shock Absorber Testing using FFT Analyzer with DEWESoft

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Abstract - *This paper is based on Shock absorber testing on different road texture by varying spring stiffness using FFT Analyzer. Shock absorber is an important part of automotive suspension system which reduce shock impulse and used for driving safety. The road disturbance is generated in the model by giving speed brakes fixed on drum which is rotated by using motor. In this paper study and analysis of single DOF spring-mass-damper system (Hero Splendor and Honda Shine i.e. Shock Absorber) and plotted its dynamic characteristics curve for different values of spring stiffness and different oils for optimum motion transmissibility.*

***Key Words*: FFT Analyzer, Shock Absorber, Motion Transmissibility, Equation of absolute motion, Dewesoft software, Design and calculation etc.**

1. INTRODUCTION

Now-a-days Modern vehicles come along with strong shock absorbers to tolerate any type of bouncy conditions. Modern motorcycle allows the driver to fine tune the machine to give him/her better control over it when driving. The vehicle is subjected to bump at the bottom end of the rear wheel.

The primary function of the suspension system is to isolate the vehicle structure from shocks and vibration due to irregularities of the road surface. The Suspension system is 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 turns in dynamic state of balance, continuously compensating and adjusting for changing driving conditions according to road profile. Suspension of vehicle need to analyse before the manufacturing. This is because to make sure components in shock absorber system remain in good conditions.

The mechanism or setup should be able to create a vehicular situation in which one end of the suspension test rig is subjected to jerks and bounce producing unsteady vibrations. While on the other side it should face loading conditions representing the weight of passenger and vehicle itself. So this encompasses the design and development of shock absorber test rig in which the above mentioned conditions are simulated and performance is evaluated.

2. PROBLEM STATEMENT

The aim of the project is to study and analyze single degree of freedom spring-mass-damper system and plot its dynamic characteristics curve for different values of spring stiffness for various speed conditions using FFT Analyzer

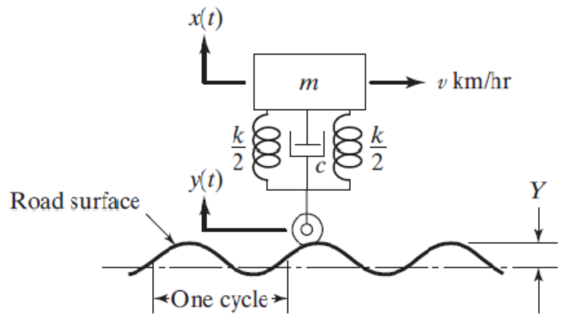

Fig -1: Problem Specification

3. OBJECTIVES

1) To determine dynamic characteristic of shock absorber.
2) To test suspension on different types of oils and stiffness to find out optimum motion transmissibility.
3) Suspension will be test for multiple stiffness by varying loads, speed and different oils.

4. SPEED BRAKER PROFILE

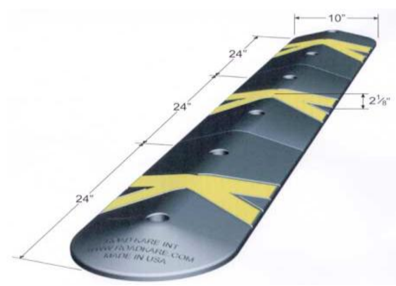

Fig -2: Speed Breaker Profile

Speed breaker profile is 250mm X 54mm selected. Standard available size of rumbler is taken according to Indian government standard. Suspension will be tested on this speed breaker profile. It's made by Lake Traffic Solutions.

5. THEORETICAL BACKGROUND

In case of locomotives or vehicles the wheels act as base or support for the system. The wheels can move vertically up and down on the road surface during the motion of the vehicle.

At the same time is relative motion between the wheels and the chassis is having motion relative to the wheels and the wheels are having motion relative to the road surface. The amplitude of vibration in case of support motion depends on the speed of vehicle and nature of road surface. The vibration measuring instruments are designed on the support motion approach. Such systems are supposed to have single degree of freedom for the simplicity of mathematical expression. In a vibratory system where the support is put to excitation absolute and relative motion become important from subject point of view.

5.1 Terminology Used

1) Natural frequency (ωn)

When no external force acts on the system after giving it an initial displacement, the body vibrates. These vibrations are called free vibrations and their frequency as natural frequency. It is expressed in rad/sec or Hertz.

2) Damping

The external force which is provided to reduce the vibrations.

3) Critically damping co-efficient

The critical damping co-efficient Cc is that value of damping coefficient c at which the frequency of free damped vibration is zero and the motion is a periodic.

4) Damping co-efficient (ξ)

It is defined as the ratio of damping coefficient to critical damping coefficient. Mathematically, $\xi = C/Cc$

5) Amplitude

The maximum displacement of vibrating body from the equilibrium position.

6) Time period (tp)

The time required to complete one cycle.

7) Absolute Motion

Absolute motion of a mass means its motion with respect to the coordinate system attached to the earth. As shown in figure, the absolute displacement if support is y=B sin ωt and the absolute displacement of the mass m from its equilibrium position is x. The displacement of mass m relative to the support is z. The net elongation of the spring is (x‟-y‟) and the relative motion between the two ends of the damper is (x-y). Then z= x-y and z‟= x‟-y‟.

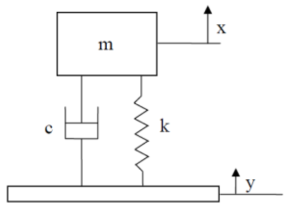

Fig-3: Absolute Motion

The equation of motion can be written as,

$m\ddot{x}+c\,((x\dot{-})\dot{y})+k(x-y)=0$

Or $m\ddot{x}+c\dot{x}+kx=c\dot{y}+ky.$

Solve the equation, we get, Steady state amplitude can be written as:

$$\frac{X}{Y}=\frac{\sqrt{\left\{1+\left[\frac{2\xi\omega}{\omega n}\right]^2\right\}}}{\sqrt{\left\{\left[1-\left(\frac{\omega}{\omega n}\right)^2\right]^2+\left[\frac{2\xi\omega}{\omega n}\right]^2\right\}}}$$

The ratio of **X/Y** is called the **displacement transmissibility** which is the ratio of amplitude of the body to amplitude of the support.

6. DESIGN AND ANALYTICAL CALCULATIONS

6.1 Spring Stiffness Calculation

Fig-4: Spring Stiffness Measurement of Splendor Suspension and Honda Shine Suspension

As initially we don't have spring stiffness value for two suspensions those are used for experimentation .For this one small experiment is done to calculate the stiffness. Initial length of the spring is measured with scale. Then 60 Kg load is applied on spring of one of the shock absorber. Thus spring gets compressed and now again spring length is measured. Now by using the formula for calculating spring stiffness,

K= (F/X) 9.81 N/mm
Where,
K-Spring stiffness in N/mm F-Load applied in Kg

X-Displacement due to loading=
(Free length -Compressed length) in mm. Sample Calculation For Splendor,
K= (60 ▯9.81)/ (230 -205)
K= 23.54 N/mm

Table -1: Spring Stiffness of Splendor and Honda Shine Shock Absorber

Sr. No	Shock Absorber	Load on Spring (Kg)	Free Length (mm)	Comp. Length (mm)	Spring Stiffness (K) (N/mm)
1	Splendor	60	230	205	23.540
2	Honda Shine	60	240	206	17.31

Natural frequency:

$$\omega n = \sqrt{\frac{k}{m}} = \omega n = \sqrt{\frac{23540}{343.35}} = \omega n = 8.28 \text{ rad/sec.}$$

Critical damping coefficient:

$$Cc = 2*\sqrt{k*m}$$

$$Cc = 2*\sqrt{23540*343.35} = Cc = 5685.93 \text{ Ns/m.}$$

Damping coefficient:

ξ = C/Cc

$$0.25 = \frac{C}{5685.93} = C = 1421.48 \text{ Ns/m}$$

Time period:

$$t\rho = \frac{\lambda}{v} = t\rho = 0.5/1.39 = t\rho = 0.3597\text{sec}$$

Excitation frequency:

$$\omega = \frac{2\pi}{t\rho} = \omega = \frac{2\pi}{0.2632} = \omega = 17.4678 \text{ rad/sec}$$

Motion Transmissibility:

$$\frac{X}{Y} = \frac{\sqrt{\left\{1+\left[\frac{2\xi\omega}{\omega n}\right]^2\right\}}}{\sqrt{\left\{\left[1-\left(\frac{\omega}{\omega n}\right)^2\right]^2+\left[\frac{2\xi\omega}{\omega n}\right]^2\right\}}}$$

$$\frac{X}{0.054} = \frac{\sqrt{1+\left(\frac{(2*0.25*17.4678)}{8.28}\right)^2}}{\sqrt{\left\{\left[1-\left(\frac{17.4678}{8.28}\right)^2\right]^2+\left[\left(\frac{(2*0.25*17.4678)}{8.28}\right)^2\right]\right\}}}$$

X = 0.1714 m

6.2 Shaft material and Calculation

EN 19 Alloy Steel used for shaft. It is a high quality, high tensile steel usually supplied readily machineable in any temperature condition, giving good ductility and shock resisting properties combined with resistance to wear.

6.2.1 Applications:

EN19T was originally introduced for the use in the machine tool and motor industries for gears, pinions, shafts, spindles and the like. Later its applications became much more extended and it is now widely used in areas such as the oil and gas industries. EN19T is suitable for applications such as gears, bolts, studs and a wide variety of applications where a good quality high tensile steel grade is suited.

6.2.2 Calculation:

Perimeter of Drum
P=2πr ; P=2π ×0.305 ; P=1.91637 m

Volume of Drum
V=2πr^2 h ; V=2▯3.14▯0.305▯0.305▯0.005
V=0.00292246 m3

Mass of Drum
M=V×ρ ; M=0.00292246▯ 860 ; M=22.98 kg

Weight of Drum
W=m▯ ; W=22.98▯9.81 ; W=225.48 N

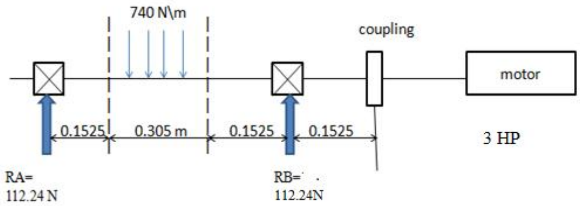

Fig- 5: Force Diagram

By maximum shear stress theory
Support reaction (RA, RB)
∑MA=0
[(-RB▯0.6096) + (740▯0.305▯0.15252)]=0 RB=112.74N
RA= -RB+ (740▯0.305) RA=112.74 N
Bending Moment (M)
M=RA▯ - w ▯(0.15252)/2

M=112.74×0.305 -740×(0.1525) 2/2 M=25.78 N -m
Torque
T=F× ; T= (149×9.81) ×0.405 ; T= 592 N-m

Equivalent Torque
Te = √ (M2+T2) ; Te = √ (25.782 + 5922); Te = 593 N-m

Diameter of Shaft
d= ∛ (16×Te/ π τ) ; d = ∛ [(16×593×1000)/ (π×45)]
d=40.69 mm~50 mm

By maximum principal stress theory
Equivalent Moment Me= [M +√ (M2+T2)] /2
Me= [25.78+√ (25.782+5922)] /2
Me=309.17 N-m

Diameter of Shaft
d= ∛ (32×Me/πσ); d=∛((32×309.17)/(π×75)) ;d=41 mm
Selecting maximum diameter & after select the roller bearing
Find Te < T
T= (60×1000×P/2πN)
T= (60×1000×2.238/2 π×240) T=89.047 N-m

Velocity
V= πDN/60 V=π×0.05×240/60 V= 628.31 m/s
V=2.2611 Kmph

6.3 Bearing

Bearing is mechanical element which locates two machine parts relative to each other and permits a relative motion between them. It has two or more contacting surfaces through which a load is transmitted. UCP210 bearing used for shaft. According to required torque of 89 Nm & internal diameter of 50mm bearing selected is Plummer block.

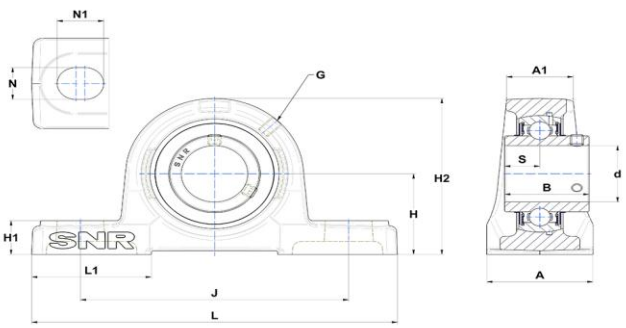

Fig-6: Bearing UCP210

6.4 Key

Key is a mechanical element used on shafts to secure rotating elements like gears, pulleys, or sprocket and prevent relative motion between two. The key transmits torque from the shafts to shaft supported element or vice versa. It is always inserted parallel to the axis of shaft.

Carbon steels have carbon as the key alloying element in their composition. They also contain up to 0.4% silicon and 1.2% manganese. In addition, the residual elements such as copper, molybdenum, aluminium, chromium and nickel are present in these steels.

TMax=1.25×593 TMax=741.25 N-m 741.25×103= (π/16) ×3×45
d=50mm w=h=50/4≈13mm l=75mm

Shear Stress
TMax= w×l× (d/2) × (τ per) Key
741.25×103= 13××(50/2) ×45
l= 50.68 mm

Crushing Stress
TMax= (h/2) ×l× (d/2) × (σ per) Key
741.25×103= (13/2) ××(50/2) ×80
l=57mm
Thus key dimensions should be taken as 13mm, 13mm, and 75mm in breadth, width and length respectively.

7. EXPERIMENTAL SETUP

Fig-7: Experimental Setup (CAD Model)

It consist following parts

1) Frame - It is Base structure of setup. It is made of MS bars in C-Section. Total material used is about 35 Feet. Frame gives the support to all the assembly components.
2) Drum - It is made of MS sheet having thickness 4mm. It is manufactured by rolling of sheet metal. Standard speed breaker profiles are also made by sheet metal by giving radius and welded to drum. Drum is supported by 3 spokes.
3) Wheel Assembly **-** It is wheel assembly of Hero Splendor Bike. Wheel is fitted in swing arm. Shock absorbers lower point is mounted on swing arm. Swing arms are assembled to Frame.
4) Motor - 1440 RPM 3 HP single phase motor is coupled to shaft. It rotates drum and ultimately drum

5) Dimmerstat - Dimmerstat is auto transformer having continuously variable voltage. 20ampere dimmerstat is used to control the motor speed.
6) FFT Analyzer - FFT-Fast Fourier Transform. It is a noise & vibration measurement instrument. Time domain data is converted into frequency domain. We will take reading by using accelerometer. DEWEsoft is used to display the results.
SPECIFICATION
•Small USB- based system
•8 analogue input channels (strain, voltage; with MSI adapters any input)
•200 kS/s aliasing-free 24bit-ADC
•8 precise real time counters
•2 CAN bus ports isolated
7) Accelerometer -The accelerometers consist of a piezoelectric crystal which has a mass attached to one of its surfaces. When the mass is subjected to a vibration signal, the mass converts the vibration (acceleration) to a force, this then being converted to an electrical signal. Accelerometer output may then processed to provide the instantaneous velocity and displacement signals.
8) The DEWE soft Software - The analysis is carried out in DEWESoft Software. Various methods of dynamic signal analysis are present in the software such as Sound level, Torsional vibration, Human Vibration and Order Tracking.
9) Tachometer - A tachometer is a sensor device for measuring the rotation speed of an object such as the engine shaft in a car. This device indicates the revolutions per minute (RPM) performed by the object. The device comprises of a dial, a needle to indicate the current reading, and markings to indicate safe and dangerous levels.

8. WORKING

Fig-8: Experimental Actual Setup

1. Shaft is mounted in bearing on which drum is mounted.
2. Speed breaker profiles is welded on drum.
3. On drum wheel assembly is mounted.
4. Shaft is coupled to motor. Motor shaft rotates the Drum shaft which simultaneously rotates the wheel which in on drum.
5. Motor speed is controlled by using Dimmer stat.
6. As wheel and drum rotates wheel reaches to speed beaker profile it create bump on shock absorber.
7. Shock absorber will get compress.
8. FFT analyzers sensors will attached to Upper and lower point of shock absorbers and readings displayed on computers screen.

9. OBSERVATION TABLE

Table -2: Splendor Suspension (Oil 1)

Sr. No.	Spring Stiffness (K) (N/mm)	Load (Kg)	Peak(RMS)		Transmissibility T_R=A/B
			Top (A)	Bottom (B)	
1	23.54	5	11.085	23.347	0.4748
2	23.54	10	14.420	24.230	0.5951
3	23.54	15	14.224	22.955	0.6196
4	23.54	20	12.066	23.544	0.5125

Table -3: Honda Shine Suspension (Oil 1)

Sr. No.	Spring Stiffness (K) (N/mm)	Load (Kg)	Peak(RMS)		Transmissibility T_R=A/B
			Top (A)	Bottom (B)	
1	17.31	5	13.34	24.32	0.5485
2	17.31	10	16.57	27.66	0.5993
3	17.31	15	24.52	26.19	0.9364
4	17.31	20	10.30	25.70	0.4019

Table -4: Splendor Suspension (Oil 2)

Sr. No.	Spring Stiffness (K) (N/mm)	Load (Kg)	Peak(RMS)		Transmissibility T_R=A/B
			Top (A)	Bottom (B)	
1	23.54	5	12.1644	25.407	0.4787
2	23.54	10	12.1644	24.525	0.4960
3	23.54	15	14.8131	21.876	0.6771
4	23.54	20	9.81	21.876	0.4484

Table -5: Honda Shine Suspension (Oil 2)

Sr. No.	Spring Stiffness (K) (N/mm)	Load (Kg)	Peak(RMS)		Transmissibility T_R=A/B
			Top (A)	Bottom (B)	
1	17.31	5	11.085	25.8984	0.4280
2	17.31	10	16.3827	27.664	0.5922
3	17.31	15	9.4176	24.721	0.3809
4	17.31	20	7.651	23.093	0.33134

10. RESULTS

10.1 Result of oil 1 (MOTUL) and oil 2 (YAMALUBE)

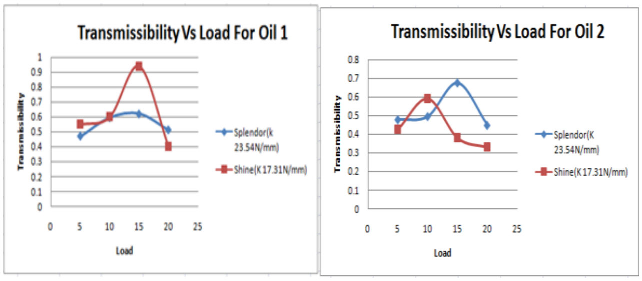

Chart -1: Transmissibility Vs Load for oil 1 and oil 2

From above graph for oil 1 we conclude that at 15 Kg load motion TR of Shine suspension(K=17.321N/mm)is high and it is low at 20kg,which is not continuous. Splendor suspension (K=23.54N/mm) have less difference in motion TR at different loads. So, splendor suspension have good motion TR at different loads. And for oil 2 we conclude that at initially for low load motion TR of Shine suspension (K=17.31N/mm) is intermediate and it is decreasing with respect to increase in loads. Splendor suspension (K=23.54N/mm) have high motion TR for this oil than Shine suspension

10.2 Result for Splendor and Honda shine using both oils

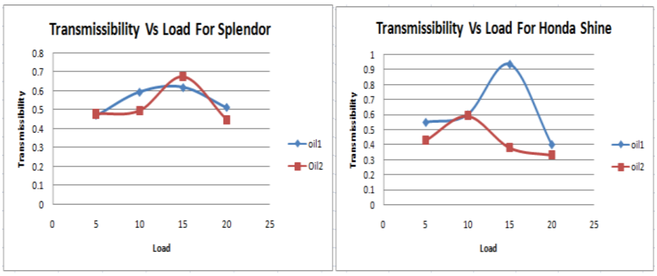

Chart -2: Transmissibility Vs Load for both oils (Splendor and Honda Shine)

From above graph, for oil 1 motion TR of Splendor suspension (K=23.54N/mm) have less difference for different loads. And for oil 2 motion TR of Honda Shine suspension (K=17.31N/mm) have less fluctuations for different loads.

10.3 FFT Result

Fig-9: Splendor Bottom for weight 10Kg for Oil 1

Fig-10: Splendor Top for weight 10Kg for Oil 1

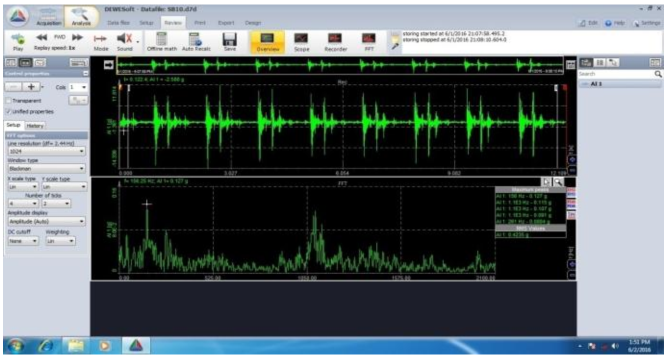

Fig-11: Splendor Bottom for weight 10Kg for Oil 2

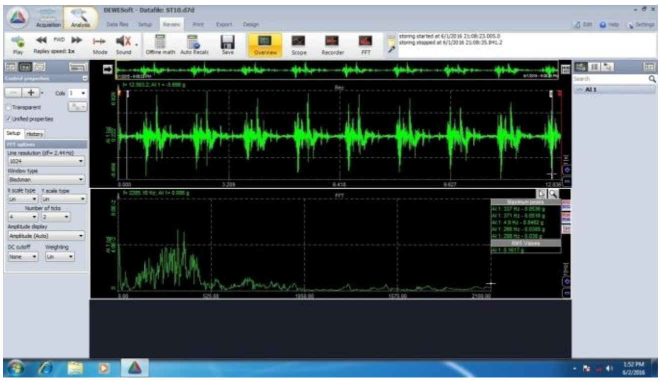

Fig-12: Splendor Top for weight 10Kg for Oil 2

11. CONCLUSIONS

From this Suspension testing setup we can test multiple number of suspensions at different loads, different angles and different speeds. Also we can use suspensions of different height.

By changing different suspensions and oils we can find out optimum motion transmissibility. With ultimate objective of studying and plotting dynamic characteristics for Hero Splendor suspension and Honda Shine suspension using single wheel model of suspension analysis to produced large number of results. However it concludes the project work with following points:

1. Input and output graph shows transmissibility which is in limit.
2. The suspension system gives best performance when designed to be slightly under-damped.
3. From experimental results and graphs we can conclude that for good ride, transmissibility should be as low as possible and this can be attained by using low damping constant and high spring stiffness and Honda Shine suspension gives the better results as compared to Splendor suspension

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