Design and Performance Analysis of Mechanical-Hydro-Pneumatic Suspension System for Motorbikes

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Abstract - *During Travelling on normal motorbikes like Hero Splender, Bajaj Discover, CT-100, and Honda Shine etc, The comfort level is not good as like Royal Enfield, Harley-Davidson, Ducati, Kawasaki, BMW, Honda Gold, Benelli, Norton, Hayabusa, Triumph, Tork because the one reason of such issue is that available suspension system in Normal Motorbikes is not up to the mark of comfort and safety, Suspension perceived as most comfortable when the natural frequency is in the range of 60 to 90 cycles per minute (CPM), or about 1 Hz to 1.5 Hz. When the frequency approaches 120 CPM (1.5-2Hz), occupants perceive the ride as harsh. A high-performance sports motorbike will have a stiffer suspension with a natural frequency of about 120 to 150 CPM (2 to 2.5 Hz). So the available suspension system of these vehicles must be redesigned to achieve the comfortable ride as per the standard range of comfort frequency of suspension i.e. 0.5-1.5 Hz.*

Key Words: **Smooth ride, high level of comfort, reduced jerk transfer, modified frequency of suspension in range 0.5-1.5 Hz, Improved comfort level.**

1. INTRODUCTION

During passing over the speed breakers, bumps and obstacles of the roads the jerk is directly transferred to the passengers, which is too high even traction of tires because of non – standard roads causes the jerk in the vehicle which is also transferred to the passengers. This shows that the available suspension system does not give effective comfort to the rider. The cemented RCC roads are also a reason behind the discomfort during the ride, as medically they as suitable for the passengers, as in case RCC roads the jerks and bumps are directly transferred to the passengers which may lead to back pain, spinal injury etc. So available suspension system of the vehicles must be redesigned to achieve the comfortable ride as per the standard range of comfort frequency of suspension i.e. 0.5 - 1.5 Hz. The desired effect can be achieved by the modification of the available suspension system for normal bikes and i.e. Mechanical Hydro-Pneumatic Suspension System.

2. LITERATURE SURVEY

For the analysis, part researchers employ the Theoretical, Numerical and Finite Element Methods (FEM). The study concludes FEM is the best method for calculating the stress, life cycle and shear stress of helical compression spring Lavanya et al. [1].

The objective of Dr Dhananjay et al [2] is the performance analysis of a shock absorber spring by varying stiffness can be obtained by doing optimization to obtain maximum comfortable ride with Pro-E and ANSYS.

Based on numerous investigations, studies and developments of automotive suspensions structures, models, and features of various automotive suspensions, weight, cost, ride comfort, handling performance, reliability, dynamic performance, energy regeneration, and commercial maturity, the concept of the suggested suspension system can provide desirable results which are supported by the Ansar Allauddin Mulla and Deepak Rajendra Unaune et al. [3].

Ali M. Abd-El-Tawwab et al [4] performed an experiment which reveals that, by using active suspension systems compared with a passive suspension system a significant improvement in ride performance can be obtained.

The article provided on the theoretical and experimental foundations which determine parameters of gas springs with constant gas mass, provides good results by L. KONIECZNY, R. BURDZIK, T. WĘGRZYN et al. [5].

The details of dimensional calculation for designing the suspension system found good from “The Shock Absorber Handbook,” by John C. Dixon et al. [6].

The type of suspension can be decided from the basic three models, the study done by Babak Ebrahimi et al [7], gives a better and clear idea to consider the passive suspension system will be the suitable design of suspension in the considered case.

The detailed work done by Karthikeyan S.S, Karthikeyan V, Leoni Praveen C, Manigandan G and Rathish R, et al [8], on the designing of helical spring with variable pitch, to consider the facts for designing the mechanical element for the desired load.

The design and performance analysis done by Dr. Ashesh Tiwari and Atreya Pathak et al [9], shows that the hybrid system can be easily applied to any vehicle to increase the comfort from the jerks and shocks of the roads.

3. RESEARCH METHODOLOGY

Based on the above studies, references and researches, we conclude that for designing the "Mechanical-Hydro-Pneumatic Suspension System for Motorbikes" the designing is required to work with the approximate vehicle weight (150 kg gross + 150 kg passenger+ 60 kg luggage = 360 Kg) and to achieve the required designed system and its components must be capable to work under the desired parameters in designing Points considered are as below (Control Arm Type Semi Active Suspension)

Total Length of system

From the study and observations of present design of suspension, the system is having length of 300 mm

Length and type of mechanical Component (spring) it's designing procedure and material with testing results

From different analysis and studies of previous work and researches in this system the helical spring with variable pitch is used, which will be made up of Oil tempered chrome silicon (ASTM A401) material with variable pitch having outer diameter of coil as 80 mm, inner diameter of coil as 70 mm, spring wire diameter as 10 mm, total length of the spring as 250 mm, total number of rounds 14, cap thickness on coil is 20 mm.

Length of cylinder and Piston and its Material

Based on design requirements the length of pneumatic cylinder is 210 mm, bore dia. Outer 60 mm, inner 45 mm wall thickness 7.5 mm, piston length 100 mm, piston rod dia.20 mm , the length of hydraulic cylinder is 150 mm, bore dia. Outer 45 mm, inner 35 mm wall thickness 5 mm , piston length 100 mm with stroke length 112 mm. The material chosen for the cylinders is Type 304 stainless steel. This is a widely used material for engineering purposes.

Type of oil with its properties

According to American Society for Testing and Materials (ASTM) oil referred for damping in the suspension is high performance synthetic fork oil extra heavy 20WT with preferable American Petroleum Institute (API) gravity 35.4 and reservoir pressure will be at 8.5 bar.

Gas spring (nitrogen gas) and required pressure to sustain the weight of system

As per the reference studies and researches nitrogen gas will be considered good to serve the purpose as a gas spring. Due to the negligible influence of temperature the volume of Nitrogen and the fact that it does not exert any aggressive impact on the material of membrane, which is the case when air is used. Pressure to absorb the shock after testing and analysis is found to be **13** bar which is very less from the available design used in Audi A-6 (24 Bar).

Design Procedure

For Helical Spring

Diameter of Spring Wire = 10 mm

ID of spring = 70 mm

OD of spring = 80 mm

Mean diameter = (70+80)/2 = 75 mm

Spring Index C = D/d = 75/10 = 7.5

Length of spring = 250 mm

No. of Rounds = 14 with variable pitch

No. of Active turns = 14-2 = 12

No. of Inactive turns = 14-12 = 2

Thickness of Supporting cap for the element = 20 mm

Deflection in the spring = $8PD^3N/Gd^4$

$= 8 \times 1350 \times (75)^3 \times 12 / 79289.709 \times (10)^4 = 68.96$ mm

Stiffness $k = P/\delta = 1350/68.96 = 19.576$ N/mm

Wahl factor K = 4C-1/4C-4 + 0.615/C

$= (7.5 \times 4 - 1)/ (7.5 \times 4 - 4) + 0.615/7.5$

$= 29/26 + 0.082 = 1.197$

Final Deflection = $\delta \times K = 82.54$ mm

During Calculation load on the element is 1350 N

For Pneumatic Cylinder

Applied Pressure, $P_o = F/A = 1350\ N/ (\pi/4)\ d^2$

$P_o = 1350/ 0.0028274$

$P_o = 477.465$ kPa

Volume $V = \pi r^2 h = \pi \times (0.03)^2 \times 0.007 = 5.938 \times 10^{-7}\ m^3$

Mass $m = \rho \times V = 7000 \times 5.938 \times 10^{-7} = 4.156 \times 10^{-3}$ Kg

Now Area of Piston $A_p = 2\pi r h = 2 \times \pi \times 0.03 \times 0.205$

$= 0.0375\ m^2$

Now Pressure of N_2 inside the cylinder to sustain the load $= P_o$ + (Mass of Piston x g)/ Area of Piston

$= 477.465 + (4.156 \times 10^{-3} \times 9.81)/0.0375$

$= 478.546$ kPa = 4.785 Bar = 0.479 N/mm^2

For Hydraulic Cylinder

Applied Pressure $P_o = F/A = 1350\ N/ (\pi/4)\ d^2$

$P_o = 1350/ 1.5904 \times 10^{-3}$

$P_o = 848826.363\ N/m^2 = 848.826\ kPa$

Volume $V = \pi r^2 h = \pi \times (0.00225)^2 \times 0.070$

$V = 1.1133 \times 10^{-4}\ m^3$

Mass $m = \rho \times V = 7000 \times 1.1133 \times 10^{-4} = 0.07839$ Kg

Now Area of Piston, $A_p = 2\pi r h = 2 \times \pi \times 0.0225 \times 0.138$

$A_p = 0.0195\ m^2$

Now Pressure of Oil inside the cylinder to sustain the load $= P_o$ + (Mass of Piston x g)/ Area of Piston

$= 848.826 + (0.07839 \times 9.81)/0.0195$

$= 888.243\ kPa = 8.882\ Bar = 0.88\ N/mm^2$

4. ANALYSIS OF SUSPENSION ON SOLID-WORKS

Analysis type: Static

Description

The assembly of suspension is tested with load of 1350 N taking factor of safety 1.5 under static load

Assumptions

Design will work under the applied static load with all considered facts during actual working

Mesh type -Solid Mesh

Solver type- FFE Plus

Mesh Information

Mesh type- Solid Mesh

Jacobian points- 4 Points

Maximum element size - 28.51 mm

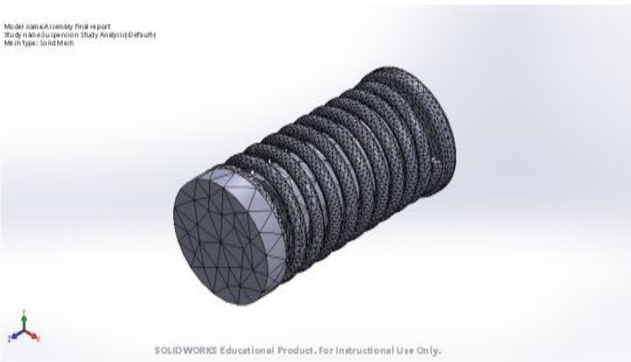
Minimum element size-5 .7 mm

Mesh Quality- High

Mesh Information- Details

Total Nodes - 68974

Total Elements – 43385

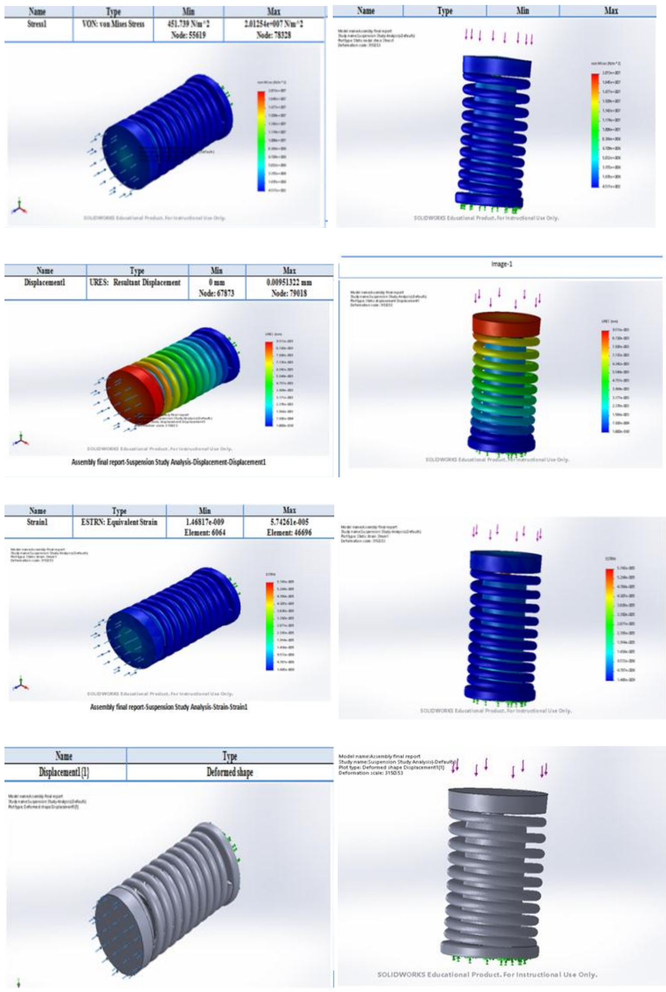


Resultant Forces

Reaction Forces

Selection Set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N	0.133479	0.0417101	2500.57	2500.57

Study Results and Conclusions



From The analysis the results are

Von Mises Stress – 381.73 N/m^2 (Minimum)

– $1.812 \times 10^7\ N/m^2$ (Maximum)

Resultant Displacement – 0.007425 mm

Equivalent strain – 3.24×10^{-5}

5. ANALYSIS CALCULATIONS

Damping ratio calculations

Damping Ration ς = Actual damping / Critical Damping

$\zeta = C / C_c$

Where, $C = 2\, C_c / 3$

And $C_c = 2 \sqrt{(K \times m_s)}$

Here K (Total Suspension stiffness) = P / δ

K = 900 N / 0.069 m

K = 13043.748 N/m

And m_s (Sprung mass) = 360 Kg = 3600 N

Now $C_c = 2 \sqrt{(K \times m_s)} = 2 \sqrt{(13043.748 \times 3600)}$

$C_c = 2 (6852.483) = 13704.966$

And C = (W x t) L_u

And $t = 2\pi/\sqrt{\{b-(a/2)^2\}}$

Now a = f x g/W = (900 x 9.81) / 90 = 98.1

And b = k x g/W = (P x g) / (δ x W)

b = (900 x 9.81) / (0.034 x 90)

b = 2885.3

Therefore, $t = 2\pi/\sqrt{\{2885.3 - (98.1/2)^2\}}$

$t = 2\pi/\sqrt{\{2885.3 - 2405.9\}}$

$t = 2\pi/21.89 = 0.287$

Now C = (1350 x 0.287)/0.25 = 1549.8

So Damping Ration of Assembly $\varsigma = C / C_c$

ς = 1549.8/ 13043.748

ς = 0.12

The value for Damping ratio $\varsigma < 1$ condition of under damped which is desirable.

Dynamic Load Calculations Due to Bump

Impact Load = Jerk / Δt, where

Jerk = force = m x Δv (for dynamic Loading) and

Δt = Time to cross a bump

Now Actual load on suspension during bump

$= 900 / \sqrt{2} = 636.39$ N

(Here $\sqrt{2}$ is taking for load profile in sin wave form)

As we know that

f = m x a

Therefore

a = f / m = 900/360 = 2.5 m/s^2

Here let us consider that v = 120 Km/hr = 33.33 m/s to cross a bump

Now

t = v/acc. = 33.33/2.5 = 13.33 s.

Now change in velocity

$v^2 = u^2 + 2x$ acc. x s

Here velocity $u^2 = 0$,

So

$v^2 = 2x\ 2.5 \times 1 = 5$

v = 2.236 m/s

Now dynamic load during impact force = (m x δv) / δt

= (360 x 2.236) / 13.33 = 60.38Kg = 603.8 N

Now Natural Frequency of the Suspension (For the spring and dashpot mass system) is

$\omega_n = \sqrt{K/m}$

Where K = Load actual / Deflection, here m =5400 N (with FOS = 1.5)

K = 900/0.069 = 13043.478

So

$\omega_n = \sqrt{13043.478/5400} = 1.5$ Hz

6. CONCLUSIONS

Following conclusions from the study and analysis

1. Deflection in the suspension is 75.5 mm.

After testing on Solid Works it is clear that the total deflection in suspension is found to be 75.5 mm, in this case it is considered good as the deflection up to 90 mm considered as good.

2. Frequency of the suspension system is found to be 1.5 Hz.

In dynamic condition the analysis of the actual load on the element is 636.39 N which shows that the actual frequency of the suspension is 1.5 Hz, i.e. considered as good for comfortable ride as the range is 0.5-1.5 Hz

3. Damping Ratio $\varsigma = 0.12$, i.e. under damped condition.

Calculations show that for under damped condition, the damping ratio value will be $\varsigma < 1$, and that will be good for a perfect suspension system, for a smooth ride.

4. Actual load on system is 636.39 N

After getting actual deflection from analysis calculations shows the total dynamic impact load on suspension during crossing a bump at a speed of 120 Km/hr is 603.8 N which is very less than the designed load i.e. 900 N, and shows the design is safe during ride.

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