Design and Performance Analysis of Mechanical Hydro Pneumatic Suspension System

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Abstract - *During Travelling in passenger cars like Alto-800, The level of comfort is not good as like Mercedes-Benz, Audi, BMW because the one reason of such issue is that available suspension system in Light Commercial Vehicle 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 car have a stiffer suspension with a natural frequency of about 120 to 150 CPM (2 to 2.5 Hz) So to achieve the comfortable ride in small vehicles suspension must be redesigned.*

***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

Available suspension system in Light Commercial Vehicle is not up to the mark of comfort, means during passing over the obstacles and bumps of the roads the jerk transferred to the passengers is too high even traction of tires because of nonstandard roads causes the jerk in the vehicle which is also transferred to the passengers. So to achieve the comfortable ride the available suspension system of these vehicles must be redesigned as per the standard range of comfort frequency of suspension i.e. 0.5-1.5 Hz Desired effect could be achieved by modifying the available suspension system for normal cars and i.e. Mechanical Hydro Pneumatic Suspension System.

2. LITERATURE SURVEY

Researchers employ the Theoretical, Numerical and FEM methods. 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 to analyze the performance of Shock absorber spring by varying stiffness, which is obtained by doing optimization obtain maximum ride comfort with pro-E and ANSYS

Based on study of suspensions, structures, models, and features for the aspects of weight, cost, ride comfort, and commercial maturity, the concept of suggested suspension system could be good to provide desirable results which is supported by the Deepak Rajendra Unaune et al **[3]**

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

The theoretical and experimental results were in good agreement. Base of this system where gas spring is used found best suited by L. KONIECZNY, R. BURDZIK, T. WĘGRZYN, et al. **[5],** during testing and analysis.

The details given in "The Shock Absorber Handbook," by John C. Dixon et al **[6]** for the Dimensional calculation found good for designing the suspension system.

To decide the type of suspension system from the basic three models, the study done by Babak Ebrahimi et al **[7],** thesis chapter 1 gives a better and clear idea to find the suitable design of suspension which will be passive suspension system in the considered case.

For designing of helical spring with variable pitch work done by Karthikeyan S.S, Karthikeyan V, Leoni Praveen C, Manigandan G and Rathish R, et al **[8]** "is in a very detailed manner to consider the facts for designing the mechanical element for the desired load.

3. RESEARCH METHODOLOGY

Based on above study, references and researches , The conclusion is that for designing the "Mechanical Hydro Pneumatic Suspension System" the design is required to work with the vehicle of approx weight (675 kg gross+350 kg passenger+ 50 kg luggage = 1075 Kg) and to achieve that system and its components must have the capability to work under the desired parameters. In designing Points considered are as below (Control Arm Type Semi Active Suspension)

Total Length of system

As per study and observation of present suspension design the system is having length of 300 mm

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

Based on the study of previous work and research in this system helical coil spring with varying pitch is used made up of Oil tempered chrome silicon (ASTM A401) material with variable pitch and total length of 250 mm, no of rounds 12, spring wire dia. 16 mm, outer dia. of coil 132 mm, inner dia. of coil 100 mm, cap thickness on coil is 30 mm.

Length of cylinder and Piston and its Material

Based on design requirements the length of pneumatic cylinder is 210 mm, bore dia. Outer 90 mm, inner 80 mm wall thickness 5 mm , piston length 86 mm, piston rod dia.30 mm , the length of hydraulic cylinder is 138 mm, bore dia. Outer 80 mm, inner 70 mm wall thickness 12 mm , piston length 86 mm stroke length 112 mm. The material for cylinders is Type 304 stainless steel. This is a widely used material for engineering purposes.

Type of oil with its properties

Based on ASTM (American Society for Testing and Materials) referred oil for damping in the suspension is high performance synthetic fork oil extra heavy 20wt with API gravity 35.4 is preferred were API (American Petroleum Institute) and reservoir pressure will be 8.5 bar.

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

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

Designing of helical spring

Wire diameter = 16 mm

ID of spring = 100 mm

OD of spring = 132 mm

Mean diameter = 100+132/2 = 116 mm

Spring Index C = D/d = 116/16 = 7.25

Length of spring = 250 mm

No. of Rounds = 12 with variable pitch

No. of Active turns = 12-2 = 10

No. of Inactive turns = 12-10 = 2

Thickness of Supporting cap for the element = 30 mm

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

$= 8 \times 3000 \times (102)^3 \times 10 / 7928.970 \times (16)^4 = 72.09$ mm

Stiffness k = P/delta = 3000/72.09 = 41.62 N/mm

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

= (5.66 x 4 - 1)/(5.66 x 4 - 4) + 0.615/7.25

= 28/25 + 0.84 = 1.204

Final Deflection = Delta x K = 86.79 mm

During Calculation load on the element is 3000 N

For Pneumatic Cylinder

Applied Pressure Po= F/A = 3000 N/ $(\pi/4)\, d^2$

= 3000/ 0.0254469

=471.698 KPa

Volume V = $\pi r^2 h = \pi \times (0.045)^2 \times 0.007 = 4.45 \times 10^{-5}$ m^3

Mass m = d x V =8000 x 4.45 x 10^{-5} = 0.35625 Kg

Now Area of Piston Ap = $2 \pi r h = 2 \times \pi \times 0.045 \times 0.205$

= 0.05796 m^2

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

= 471.698 + (0.35625 x 9.81)/0.05796

= 531.91 KPa = 5.319 Bar = 0.531 N/mm^2

For Hydraulic Cylinder

Applied Pressure Po= F/A = 3000 N/ $(\pi/4)\, d^2$

= 3000/ 3.8 x 10^{-3}

=779534.6 N/m^2 = 779.4 KPa

Volume V = $\pi r^2 h = \pi \times (0.035)^2 \times 0.007$

= 2.6939 x 10^{-5} m^3

Mass m = d x V =8000 x 2.6939 x 10^{-5} = 0.2155 Kg

Now Area of Piston Ap = $2 \pi r h = 2 \times \pi \times 0.035 \times 0.138$

= 0.0303 m^2

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

= 779.4 + (0.2155 x 9.81)/0.0303

= 849.3 KPa = 8.493 Bar = 0.849 N/mm^2

4. Analysis of Suspension on Solid-Works

Analysis type: Static

Description

The assembly of suspension is tested with load of 3000 N taking factor of safety 2 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 - 28.5178 mm

Minimum element size-5 .70356 mm

Mesh Quality- High

Mesh Information- Details

Total Nodes - 82714

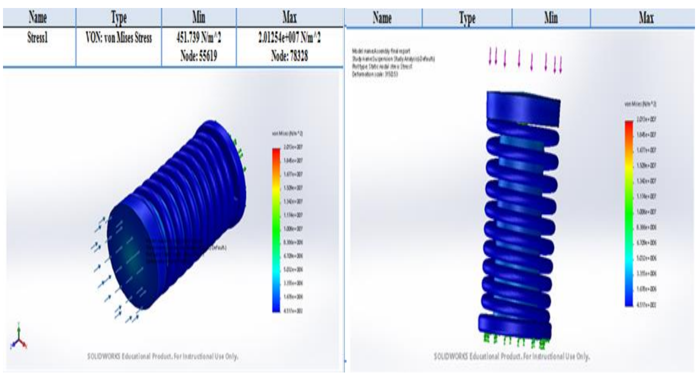
Total Elements – 47565

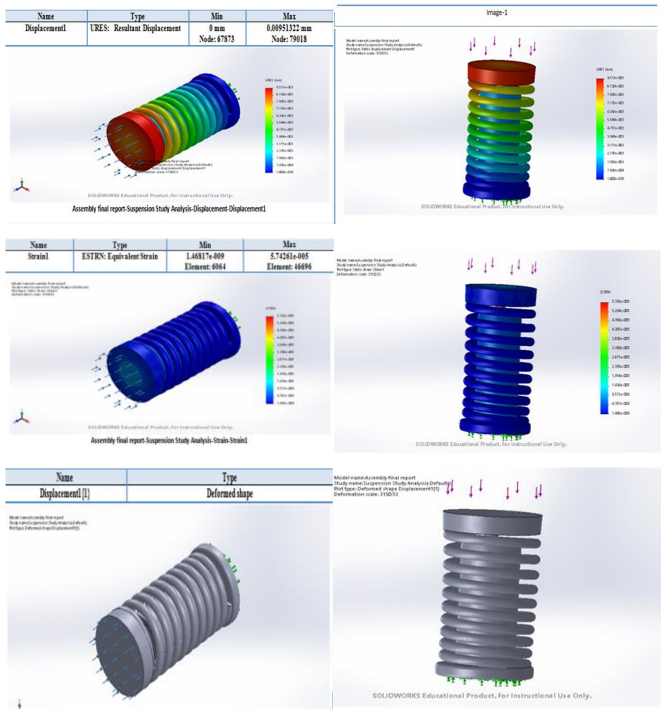
Resultant Forces

Reaction Forces

Selection Set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N	0.143479	0.0517101	3500.57	3500.57

Study Results and Conclusions





From The analysis the results are

Von Mises Stress – 451.739 N/m² (Minimum)

– 2.01 x 10^7 N/m² (Maximum)

Resultant Displacement – 0.00951322 mm

Equivalent strain – 5.7 x 10^{-5}

Analysis Calculations

Damping ratio calculations

Damping Ration ς = Actual damping / Critical Damping

= C / Cc

Where C = 2 Cc / 3

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

Here K (Total Suspension stiffness) = P / delta

= 3000 N / 0.086 m

= 34883.7 N/m

And m_s (Sprung mass) = 1075 Kg = 10542.1 N

Now Cc = 2 $\sqrt{(K \times m_s)}$ = 2 $\sqrt{(34883.7 \times 10542.1)}$

= 2 (19176.7) = 38353.48

And C = 2 Cc / 3 = 2 x (38353.48) / 3 = 25568.9

So Damping Ration of Assembly ς = C / Cc

= 25568.9/ 38353.48

ς = 0.66

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

$= 3000 / \sqrt{2} = 2121.3256$ N

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

As f = m x a so

a = f / m = 216.31 / 1075 = 0.20

Where a = Δv / t

Here taking v = 30 Km/hr = 8.33 m/s

Now t = Δv / a = 8.33 / .20 = 41.44

Now change in velocity $v^2 = u^2 + 2as$ here velocity

$u^2 = 0$

So $v^2 = 2as = 2 \times 9.8 \times .1$

v = 1.4

Now dynamic load during impact

Force = m x Δv / Δt = 1075 x 1.4 / 41.44 = 36.31 Kg

= 356 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 =10542 N

= 2160/0.08 = 27000

So $\omega_n = \sqrt{27000/10542} = 1.5$ Hz

5. CONCLUSIONS

Following conclusions from the study and analysis

1. Deflection in the suspension is 80.1 mm.
It's clear after testing on Solid works that the total deflection in suspension is found 80.01 mm which is good in this case as deflection up to 90 mm considered as good.
2. Frequency of the system is found 1.5 Hz
Getting results from analysis about actual load on the element in dynamic condition is 2121.32 N which shows the actual frequency of the suspension 1.5 Hz, i.e. considered very good for comfortable ride as the rang is 0.5-1.5 Hz
3. Damping Ratio $\varsigma = 0.66$ under damped condition
Calculations shows damping ratio value $\varsigma < 1$, which is under damped condition and counts and good for a perfect suspension, for a smooth ride.
4. Actual load on system is 2121.325 N
After getting actual deflection from analysis calculations shows the total dynamic impact load on suspension during crossing a bump at a speed of 30 Km/hr is 316 N which is very less than the designed load i.e. 3000 N, and shows the design is safe during ride.

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