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Design and Modeling of Passive Hydro-pneumatic Suspension System for Car

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Abstract :*Suspension system is most important in a vehicle which counters the disturbances generated from the road. It contributes significantly on the vehicle's stability, safety and control. The hydro-pneumatic system uses an accumulator to generate spring force and a remote valve block to generate damping force. A hydraulic cylinder replaces the damper strut and springs of the vehicle. The cylinder generates oil volume displacement towards the accumulator. The oil is assumed to be incompressible and the volume of the air chamber inside the accumulator is diminished which creates a pressure increase by means of the "ideal gas law". Higher pressure results in a higher reaction force and so a spring is established. The flows from the piston chambers and rod chambers of the cylinders are led through the tubing system and a flow resistor. Due to the pressure losses energy is dissipated and damping is generated. This paper is an over view on design and modeling of hydro-pneumatic suspension system. A quarter cars approach with a passive Hydro-pneumatic suspension have been presented in this paper in which static and dynamic load is calculated considering all arbitrary conditions to design the system. The initial pressure and volume of an accumulator and dimensions of a single acting cylinder without preload is derived.This hydro-pneumatic system is built as multi-body SIMULINK model of the suspension used in the vehicle and translated to the same in a prototype.*

Keywords-*Accumulator, single acting cylinder, Hydro-pneumatic suspension, flow resistor, static and dynamic load, initial pressure.*

I. INTRODUCTION

The main components of Hydro-pneumatic suspension system are accumulators, cylinders, flow resisters, lines and fittings.

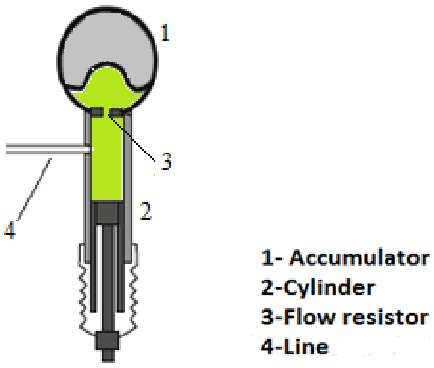

Fig.1.1 Elements of hydro-pneumatic suspension

The basic layout of the hydro-pneumatic suspension system is shown in the Figure 1.1. The Cylinders are load carrying elements in the system, transfers the forces between the input side and isolated side. It also provides travel of the suspension. The Accumulators are the elements which provide the spring function through an elastic medium. Generally the accumulators are preloaded with gas, as gas is compressible and it provides increasing pressures with increasing loads. The Flow Resistors (orifice) are important elements for the system as they provide the required damping to absorb the disturbances. The Hydraulic lines and fittings are the elements which transfer mechanical power from one place to other of the circuit in form of hydraulic/ pneumatic energy. They also act as controlling devices [1].

II. IMPORTANCE OF HYDRO-PNEUMATIC SUSPENSION SYSTEM

1. Spring characteristics

The Hydro-pneumatic system is developed for better comfort and handling while driving over uneven roads which require no additional damping system. The damping in this system is formed by friction in the suspension joints and in the hydraulic cylinder and pressure loss created over the tubing system. The damping coefficient can be freely selected by the driver as well as the ride height and stiffness can be varied also. Hydro-pneumatic Suspension system has progressive spring-rate and spring-rate is completely adaptive to the load acting on it. The stiffness can be varied from soft to very hard all by its own. The continuous Self leveling system allows the passenger his safety and comfort which is an added advantage. The wheel travel is fixed no matter what load is acting on it [2].

2. Damping characteristics

The damping coefficient can be freely selected by the driver as well as the ride height and stiffness can be varied also. The characteristic equation for the system is,

$$\mathrm{mx}^2 + \mathrm{c_s} + \mathrm{k} = 0$$

This equation determines the two independent roots for the damped vibration problem. The roots to the characteristic equation fall into one of the following 3 cases:

If $[\mathrm{c}_s^2 - 4\mathrm{mk} < 0]$ the system is termed under-damped. The roots of the characteristic equation are complex conjugates, corresponding to oscillatory motion with an exponential decay in amplitude.

If $[\mathrm{c}_s^2 - 4\mathrm{mk} = 0]$ the system is termed critically-damped. The roots of the characteristic equation are repeated, corresponding to simple decaying motion with at most one overshoot of the system's resting position.

If $[\mathrm{c}_s^2 - 4\mathrm{mk} > 0]$ the system is termed over-damped. The roots of the characteristic equation are purely real and distinct, corresponding to simple exponentially decaying motion [3].

III. OPERATING PRINCIPAL

The hydraulic pressure should be adjusted for the static loads to a required level by adding or releasing the hydraulic fluid from the accumulator. As the piston moves towards the piston side due to the static load, the fluid volume in the accumulator changes and hence the pressure also changes. Gas, which is the other fluid in the accumulator, gets compressed and exerts a force on the piston rod. This defines the spring rate of the system. The force acting on piston is always equal to the forces resulting from the pressures acting. When the force is increased due to the road profile, and the piston is displaced by a distance ‘x’, the hydraulic fluid is displaced into the accumulator which changes the pressure. This change proceeds until the pressure in the accumulator has reached a certain level which again provides a balance for the system.The Stiffness parameter given by [2];

$$F_s = A_p \times P_s; \quad F_d = A_p \times P_{sys}$$
$$K = \frac{F_s - F_d}{\Delta x}$$

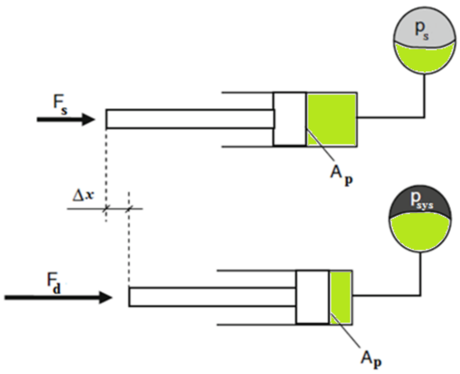

Fig.3.1. Operating principle of hydro-pneumatic suspension

IV. DYNAMIC LOAD CALCULATIONS

1. Specifications of car selected
   The basic specifications of the car selected (HONDA CIVIC) are collected and is represented in the table [5]. These values are applied to calculate some of the basic loads [4].

Table 4.1: Car specifications

Wheel Base (L)	2621 mm
Front Track-width (T_f)	1471 mm
Rear Track-width (T_r)	1468 mm
Total Kerb Weight (W)	1350 Kg
Fore length (b)	1038 mm
Aft length (c)	1583 mm
Height of CG (h)	513 mm
0 -100 kmph	11.2 sec
Wheels	Alloy wheels, 195/ 65 R15

2. Assumed Parameters
   - Front suspension: McPherson suspension
   - Tires: 195/65 R15 tubeless tires
   - Speed of the vehicle (V): 100 Kmph
   - Weight of car including driver: 1600 Kg
   - Turning Radius (R): 80 m
   - Front Slip angle (αf): 4.010 [Slalom test@40 Kmph]
3. Calculations of forces acting on front individual wheel

$$\begin{aligned}\mathit{Total\ load}\ \mathrm{F_d} &= \mathit{Static\ load}\ W_z + \mathit{Load\ transfer\ due\ to\ acceleration}\mathrm{W_x} \\ &+ \mathit{Lateral\ weight\ transfer}\ W_y + \mathit{Load\ transfer\ due\ to\ braking}\mathrm{F_b} \\ &+ \mathit{Force\ due\ to\ road\ hump}\ \mathrm{F}_h + \mathit{Load\ transfer\ due\ to\ downhill}\ \mathrm{W_{dh}}\end{aligned}$$

$$\mathrm{F_d} = \frac{W\times c}{2L}\times\cos\theta - \frac{W\times a_x\times h}{2\times g\times l} + \frac{W\times a_y\times c}{g\times l} + \mu\times\frac{W\times D_x\times h}{2\times g\times l} + \mathrm{F_h} + \frac{w\times h}{2\times l}\sin\theta$$

$$\mathrm{F_d} = 4.740 - 0.388 + 9.322 + 0.5376 + 4.16 + 0.267 = 18.64\ \mathrm{KN}$$

Note: All conditions discussed above will not be present at same time so, dynamic load value is not fixed. The total load on one front wheel is calculated considering worst case [4].

V. DIMENSIONING OF THE HYDRO-PNEUMATIC SUSPENSION HARDWARE

4.1. Cylinder-piston diameter

As per Standard sizes of piston seal available; seal of 63 mm (So, the diameter of piston $d_p = 0.063$ m) diameter is selected, so system pressure P_{sys} can be;

$$P_{sys} = \frac{F_d}{A_p} = \frac{F_d}{\frac{\pi}{4}d_p^2} = \frac{4\times 18.64\times 10^3}{3.14\times 0.063^2} = 59.83\ bar = 5.983\ MPa$$

Cylinder thickness:

Material used for manufacturing: Mild steel **UNS G10180**
(Density 7.87 g/cm3; Tensile strength 440 MPa; Yield strength 370 MPa)

$$6 = \frac{370\ \mathrm{MPa}}{\mathrm{Design\ \ stress}}$$

Design stress = 61.67 MPa

The Hoop stress is always maximum than axial and radial stresses. Considering Hoop stress = 61.67 MPa, then outer diameter is found by [6];

$\sigma_c = [(p_i r_i^2 - p_o r_o^2) / (r_o^2 - r_i^2)] - [r_i^2 r_o^2 (p_o - p_i) / (r^2 (r_o^2 - r_i^2))]$

Where
σ_c = stress in circumferential direction (MPa)
r = radius to point in tube or cylinder wall (mm) ($r_i < r < r_o$)
Maximum stress when $r = r_i$ (inside pipe or cylinder)

$61.67=[\frac{(5.983*0.0315^2-0.101325*r_0^2)}{(r_0^2-0.0315^2)}]-[\frac{0.0315^2*r_0^2(0.101325-5.983)}{0.0315^2(r_0^2-0.0315^2)}]$

$r_0^2=\frac{(5.983*0.0315^2+61.67\times0.0315^2)}{(61.67+0.101325-5.983)}$

r_0= 34.69 mm

t = 34.69 – 31.5 = 3.19 mm

4.2. Accumulator

Maximum volume of oil displaced in cylinder $=A_p\times$ Stroke length $= 3.117\times10^{-3}\times0.120$
$= 3.74\times10^{-4}$ m^3
= 0.374 litre

From this value we can select accumulator close to required value is of 0.5 liter from the standard capacities available [2].

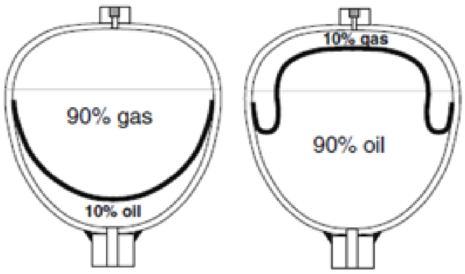

Fig.4.1. Limits for diaphragm deformation for the application in a suspension system

$$P_{max} = 8.21p_0 \times 0.9 = 7.39p_0$$
$$P_{min} = 1.26p_0 \times 1.05 = 1.32p_0$$

Good pressure ratio = $\frac{82.69\text{ bar}}{14.78\text{ bar}}$ = 5.6 (Considering all factors)

As, P_{min} = 14.78 bar & P_{max} = 59.83 bar

Design pressure ratio$=\frac{p_{max}}{p_{min}}=\frac{59.83\text{ bar}}{14.78\text{ bar}}$= 4.048 (Safe side)

VI. SIMUINK MODEL

Using a basic Spring-Damper system, the governing equations are taken [5], [6];

Acceleration of the sprung mass

$$M_s\ddot{Z} = -K_s(Z\text{-}Z_1) - C_s(\dot{Z}\text{-}\dot{Z}_1)$$
$$\ddot{Z} = \frac{[-K_s(Z-Z_1)-C_s(\dot{Z}-\dot{Z}_1)]}{M_s}$$

Acceleration of Un-sprung mass

$$M_u\ddot{Z}_1 = -K_s(Z_1-Z) - C_s(\dot{Z}_1-\dot{Z}) - K_t(Z_1\text{-}Q_1)$$
$$\ddot{Z}_1= \frac{[-K_s(Z_1-Z)-C_s(\dot{Z}_1-\dot{Z})-K_t(Z_1-Q_1)]}{M_u}$$

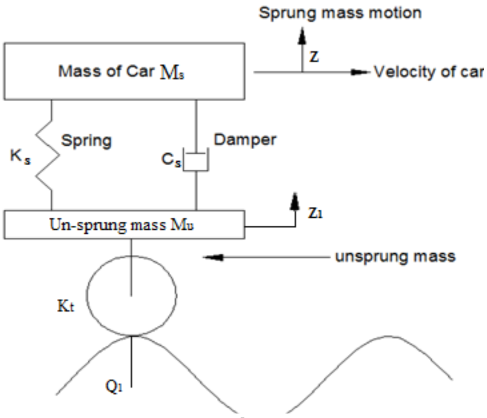

Fig.5.1. Spring damper system

4.3. The SIMULINK Model of System with All Subsystems

The model basically has tire subsystems; spring damper subsystem system. The input acceleration is plotted just after the tire, which gives the un-sprung mass acceleration and the output acceleration plot is after spring system which gives the sprung mass acceleration. A step input Q_1 is given to the system.

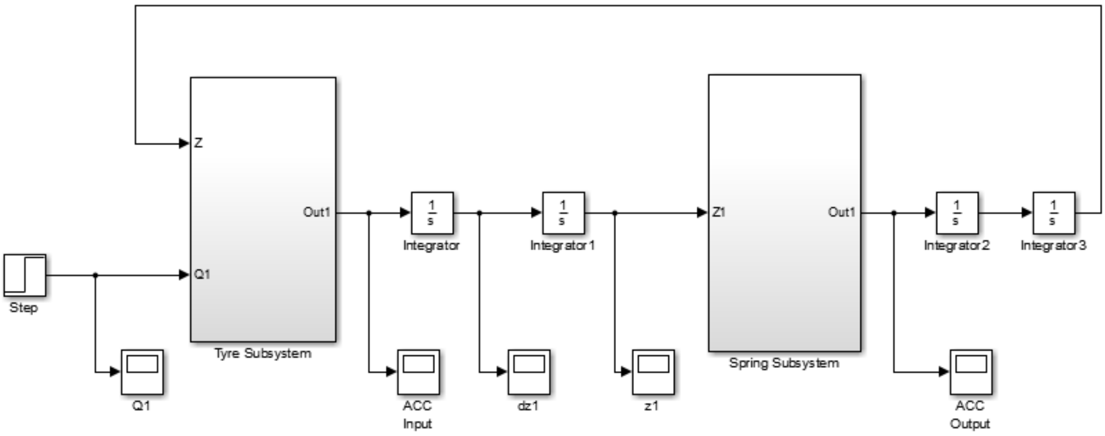

Fig.5.2. Simulink model of system with all sub-systems

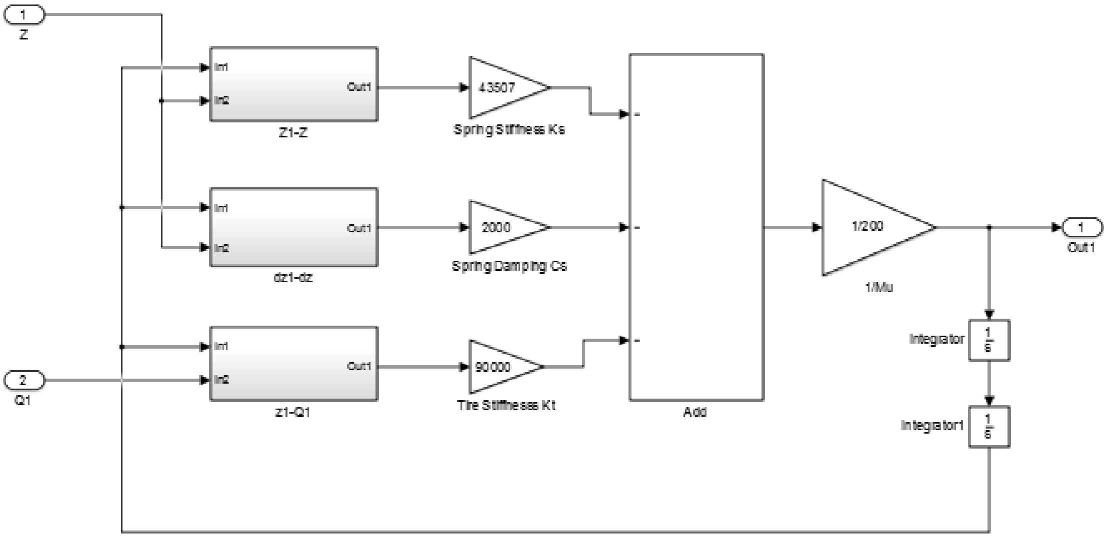

Fig.5.3. Tire sub-system

The stiffness of the tire is very much influential for the suspension behaviour. The tire is stiffness is assumed to be 90000 N/m. The spring stiffness has been changed according to the case.In a dynamic mode the tire forces are also affected by the forces coming out of spring system. So a provision was given to include the output forces from the spring system to add to the tire system.

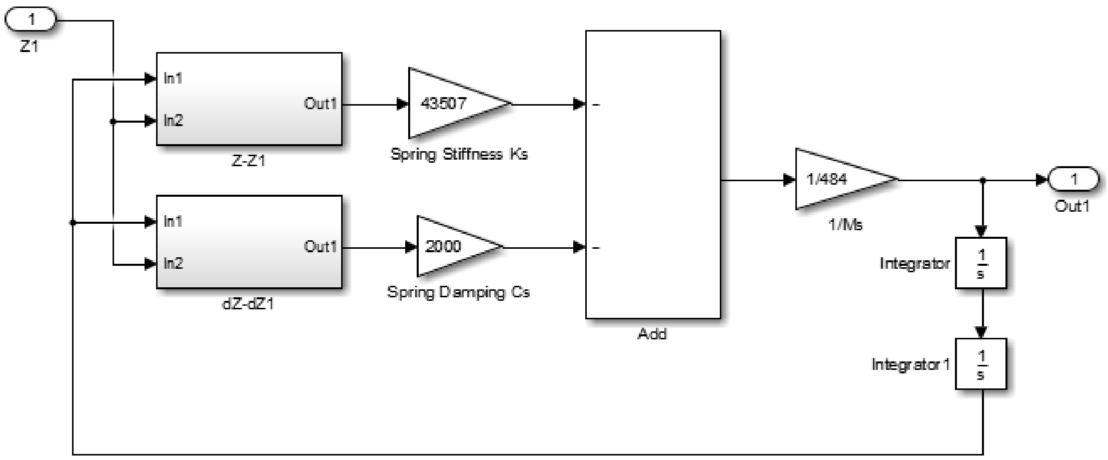

Fig.5.4. Spring Damper -Simulink

The basic model of spring damper system is taken as reference for the modelling of the particular subsystem. The load acting on the single strut is assumed to be nearly 484 Kg.

VII. CONCLUSION

System is designed as per the general design procedure that is first need definition as importance of hydro-pneumatic suspension system explained above. The total load acting on the system is calculated according to vehicle dynamics considering all arbitrary loads acting on individual front wheel of car. For that load piston and cylinder assembly is designed using maximum principle stress theory of failure. According to maximum oil volume displaced the capacity of diaphragm type accumulator is selected from olaer catalogue. Detailed calculation of initial (pre-charge) pressure and volume is done and good pressure ratio is found 5.6. Minimum and maximum pressure values calculated to find design pressure ratio which is 4. The next step isbuilding a SIMULINK model for performance evaluation of the system.

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