

Design and Development of Semi Active Variable Stiffness Hydro-Pneumatic Suspension System



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ABSTRACT

The majority population in India depends on vehicle- 3 wheeler for transportation due to their poor economic condition. These vehicles are exposed to vibrations due to irregular surface of road or soil profile, engine vibration and condition of vehicle which affects the health as well as discomfort of the driver and passengers. These oscillations are transferred into the body of driver and passenger through the body tissues, organs and systems of the individual causing various effects on the structures within the body before it is dampened and dissipated. The literature review reveals that the vibrations are most hazardous to the health if it exceeds the limit. The experimental analysis is carried out to measure the magnitude of the vibrations acting on driver as well as passenger for the different road profile at different speed. The methodology adopted is as per the International Organization for Standardization (ISO) guidelines for whole body vibration (WBV) exposure having frequency ranges from 0 to 100 Hz. The study emphasis on vibration tests for different road and speed conditions by referring, ISO 2631 and human comfort charts.

Keywords— FFT analyzer, ISO-2631, acceleration level

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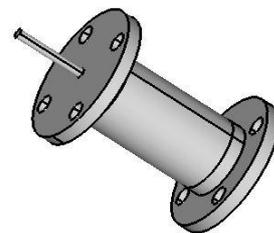
I. INTRODUCTION

Ride comfort is the general sensation of noise, vibration and motion inside a driven vehicle, experienced by both the driver as well as the passenger's ride quality is the response of the vehicle when it goes over a bump or over a series of bumps, the vibrations of the vehicle should be damped completely.

There are various problems encountered if there is no good ride quality. Ride comfort ensures that the passenger feels good when inside, especially in dynamic motion. Ride quality keeps the noise levels much lower which relaxes the driver and reduces the risk of accidents. In the longer run, when a driver experiences continuous vibrations for much longer periods, this may lead to severe back pain problems. The ride optimization problem could be viewed as the problem where one would attempt to eliminate the

negative effects of vibrations caused by road roughness. With these issues taken care of, the possibility of sales of the vehicle increases since most of the drivers care much about ride comfort.

II. DESIGN



The Hydro-pneumatic system is developed for better comfort and handling while driving over uneven roads which require no additional damping system. The damping is formed by friction in the suspension joints and in the hydraulic cylinder and pressure loss created over the tubing system. The principle of damping is discussed later in the thesis. The damping coefficient can be freely selected by the driver as well as the ride height and stiffness can be varied also. In order to analyse the behaviour of the vehicle using an active suspension system, it is easier to make a model first and to perform simulations using the control system for the active suspension. There is a vast area of application for the Hydro-pneumatic suspension system. These are mostly used in military tankers which integrate a nitrogen spring within a mono-tube damping unit. The nitrogen gas is stored in a hollow piston rod, which is separated from the damping fluid by a floating piston. Citroen invented this system for passenger cars which provides a better ride quality along with sensitive, dynamic and high-load bearing capacity. Semi-active suspension is an alternative suspension system, utilizing active damping as the control mechanism. These systems are termed semi-active because they cannot supply a force but only control the dissipation of energy from the system. Several sensors measure the vehicles state, such as velocity and the spring deflection, and according to these values the damping force of the actuator is adjusted [1]. Semi-Active suspension uses solenoids to dampen the vibrations. The rider can adjust the map to his requirements like soft, medium or hard modes. The main advantage of the semi-active suspension systems over the passive suspensions is that by altering the damping or stiffness a better compromise between handling and ride comfort can be achieved. Additionally, the semi-active suspension components have been designed so that they are not affected by external power supply failures and the implementation of mechatronic control using semi-active actuators is very robust. There are several classifications for semi-active suspension, namely

Slow-active: Suspension damping and/or spring rate can be switched between some discrete levels in response to alterations in driving or braking circumstances.

Low-bandwidth: Spring rate and damping are adjusted continuously in response to the low frequency sprung mass motions. The suspension system is controlled over the low-frequency range by use of mechatronic actuator.

High-bandwidth: Spring rate and/or damping are adjusted continuously in response to both the low frequency sprung mass motions and the high frequency axle motions. This requires the use of actuators with relatively higher bandwidth.

A. Operating principle

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 damping effect can be calculated using Stiffness as a parameter given by

$$C = \frac{F_s - F_d}{x}$$

Where, F_s – initial static force

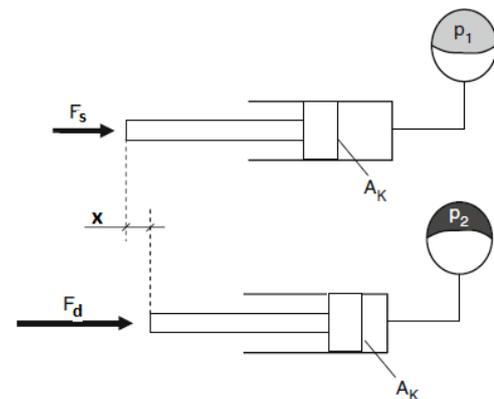
F_d – Dynamic load

x – Displacement

To allow for additional damping, a flow resistor is placed between cylinder and accumulator. It converts the kinetic energy of the hydraulic fluid into heat and thus providing the damping effect.

The damping of the system happens in combination with the boundary friction between the cylinder sealing and guiding elements and the viscous friction within the fluid.

The typical combination of mechanical spring and damper unit can be replaced by this system consisting of cylinder, accumulator, flow resistor and hydraulic fluid. One more advantage of using Hydropneumatic suspension system is level control. By adding an additional level control unit, a constant position can be maintained independent of the static spring load.



Basic Hydropneumatic Model

B. Flow (non-linear) modeling:

It is assumed that leakages in the cylinder are negligible (due to the tightly sealing piston), the compression of the cylinder causes a flow out of the piston. The flow rate depends on the piston area A_p and the rate of compression. By combining the flow through the pump Q_m with the flow in the cylinder we get the velocity of flow going towards the accumulator. \dot{Z} going towards the accumulator.

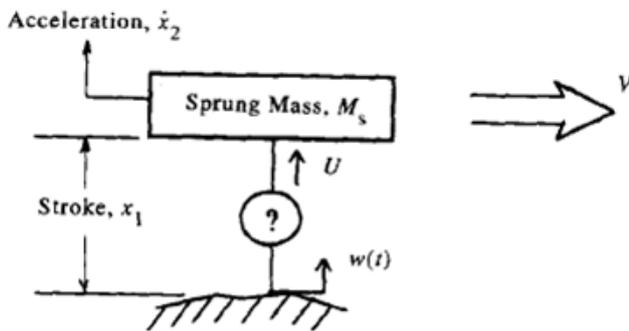
$$Q_{ks} = Q_m + (A_p * \dot{Z})$$

In order to make a linearized model of the system, it is important to estimate some of the most nonlinear parameters with simpler versions. These parameters include the Orifice damping values C_0 and the friction torque T_v of the pump.

The damping values of orifice is highly nonlinear since it depends on both the amount of flow going through the valve and the user defined suspension stiffness.

C. DOF model

As we review different models from the design of most optimal suspension system for better ride control, it is found that the design of any model starts with a simple 1DOF model with vertical motion. The simple 1D model has a sprung mass, which is nothing but the vehicle which is supported by a suspension placed between the vehicle and the ground. The figure shows the simple 1D model of a car. it is most common practice to assume the car is moving forward with a velocity V.



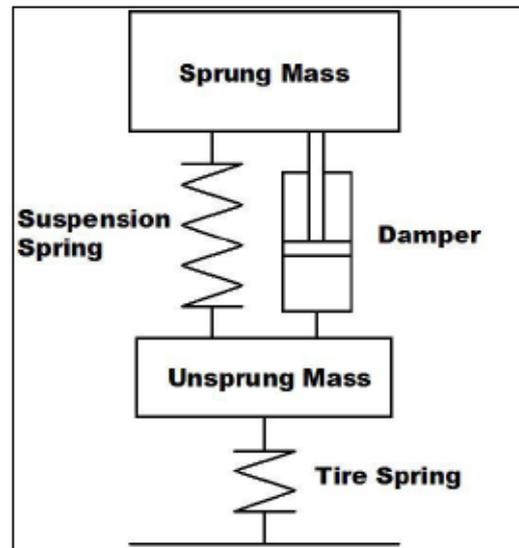
1DOF Model

The vertical disturbances or the heave is directly proportional to the velocity of the car in this model. By just adding the wheel to this model a 2 DOF quarter car model can be

defined. The wheel is nothing but the unsprung mass being added to the sprung mass model. The unsprung mass mode is often referred to as the “wheel-hop” mode, and is characterized by relatively light damping and a natural frequency between 8 and 12Hz.

D. Quarter car model

Single degree of freedom or two degrees of freedom quarter car models subjected to road excitations are used to study the suspension behavior of vehicle. Using these models the dynamic behavior of the model is studied and the



Quarter car model

The basic specifications of the car selected are collected and is represented in the table . These values are applied to calculate some of the basic loads.

Specifications of car selected [15]

Wheel Base (L)	2621 mm
Front Track-width (Tf)	1471 mm
Rear Track-width (Tr)	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

E. Assumed Parameters [15]

- 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(af) : 4.010 [Slalom test@40 Kmph]

F. Weight distribution and suspension data

The weight acting on front individual wheel is calculated using the formula

$$W = \frac{W}{2} = \frac{1600}{2} = 800 \text{ kg}$$

$$= 800 \times 9.81 = 7848 \text{ N}$$

G. Lateral forces

The lateral force depends on the front axle load and centrifugal force acting on the vehicle. The lateral acceleration can be calculated using the formula,

$$a_y = \frac{V^2}{R}$$

$$= \frac{(27.77)}{10} = 7.711 \text{ m/s}$$

Lateral weight transfer

$$L = W \frac{ay}{g} = \frac{1600 * 27.77 * 5}{9.81 * 80 * 147} = 5346 \text{ N}$$

Load transfer due to Acceleration

$$\frac{W \cdot ax}{g \cdot 2} = \frac{1600 * 2}{9.81 * 2} = 201.42$$

H. Calculation of 'ax'

From the table we can see that 0-100 kmph (27.77 m/s) in 11.2 secs

So, using Newton's laws of motion, the acceleration is

$$V = U + a * t$$

$$27.77 = 0 + a * (11.2)$$

$$ax = 2.47 \text{ m/s}^2$$

I. Total Load

Now, the total Dynamic load acting on a vehicle is calculated by

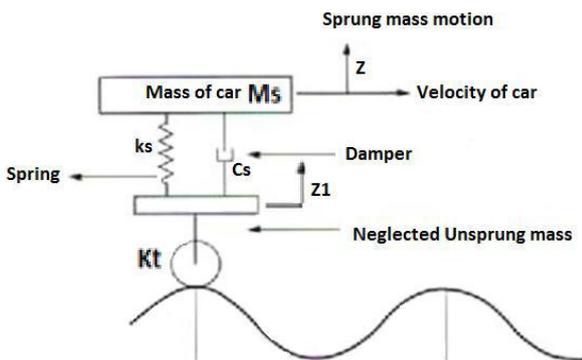
$$L = \frac{W}{g} + \frac{W \cdot ax}{g} + W \cdot \frac{ax}{2}$$

$$= 483.201.42 + 545$$

$$= 12 \text{ KN (approx)}$$

J. Simulink model

Using a basic Spring-Damper system, the governing equations are taken:



Spring damper system

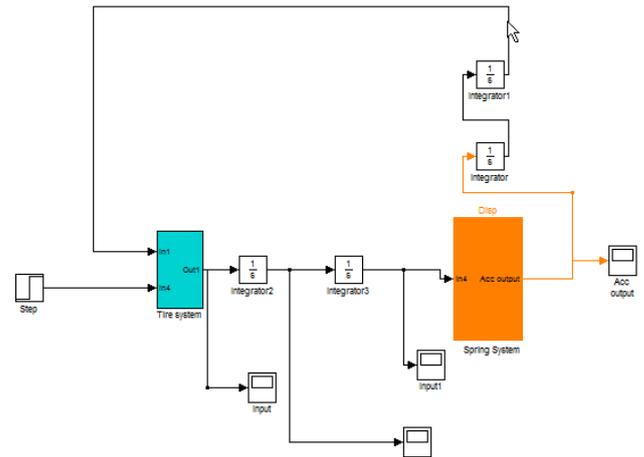
Displacement of the sprung mass

$$Z = I + \dots (Z - \dots) + \dots (- \dots)$$

Displacement equation of Un-sprung mass:

$$= M_u + \dots (Z_1 Z) + \dots (Z - \dots) + \dots (\dot{Z}_1 - \dots)$$

The Simulink model of system with all subsystems:

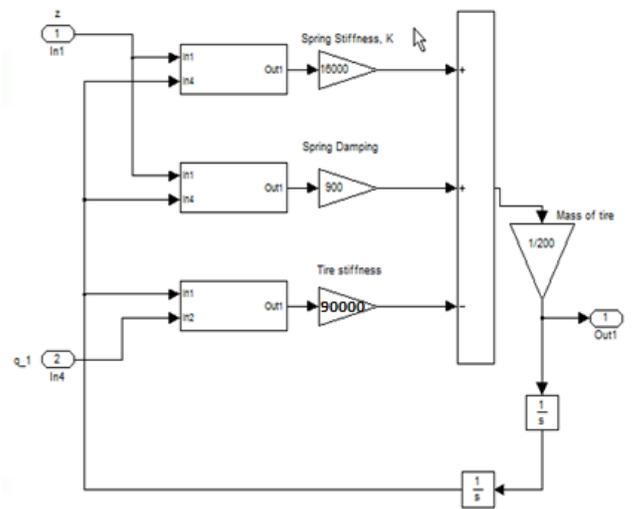


Simulink model

The model basically has tire subsystems; tire and spring damper system. The input acceleration is plotted just after the tire, which gives the unsprung mass acceleration and the output acceleration plot is after spring system which gives the sprung mass acceleration. A step input of 12 cm was given to the system

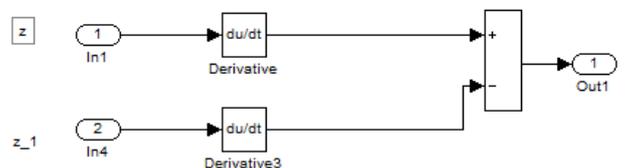
K. The subsystem of tire:

The stiffness of the tire is very much influential for the suspension behavior. The tire stiffness is assumed to be 90000 N/m. The spring stiffness has been changed according to the case.



Tire system

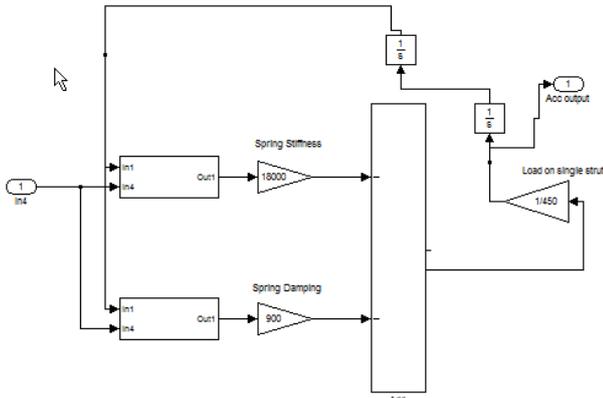
In a dynamic mode the tire forces are also affected by the forces coming out of spring system. So a provision was given to include the output forces from the spring system to add to the tire system.



Spring forces into tire system

The subsystem of spring damper system:

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 500 Kg.



Spring Damper –Simulink

L. Cylinder

The dimension of the cylinder piston diameter is important as it ensures the maximum utilization of the system pressure. For the required calculations to be carried out, we require certain information

- Available maximum pressure
- Maximum static spring load, Ff1
- Preload which acts on the cylinder, Fv

By equating the balance of forces the diameter of the piston can be calculated by

$$d_k = \sqrt{\frac{4 \cdot (F_{f1} + F_v)}{\pi \cdot P_{sys}}}$$

Accumulator

Accumulators are the elements which provide the elastic medium for the spring function to occur. The use of accumulator separates the damping effect from the spring effect unlike the original damper. The oil volume displacements and velocities are derived

$$\begin{aligned} & \square \quad \square \\ & (\square \quad \square \quad \square \quad \square \end{aligned}$$

The “Accumulator” models the relation between gas pressure and oil volume displacement and the “Damping Manifold and tubing system” models the relation between pressure losses and oil flows from the piston chamber of the cylinder. Spring behavior relates to the oil volume displacement at the one hand, damping behavior relates to oil volume flow at the other hand.

The pressures applied are converted to forces by multiplying it by the area of the piston.

$$= A \cdot P$$

The volume of the oil displaced into the accumulator is

$$= - (A_p \cdot x)$$

Using the polytropic equation,

$$P_0 V_0^n = P_a V_a^n$$

$$= \frac{P_0}{V}$$

$$= \frac{P_0 V_0^n}{(V_0 - A_p \cdot x)^n}$$

The unloaded gas states of the accumulator; pressure and volume as initially in steady state can be found in the air chamber of the accumulator. The pressure increases by compression and decreases by extension but always returns to the original unloaded situation; in conclusion the accumulator acts as a spring.

To derive at the equation for initial volume of the accumulator, three considerations are taken into account.

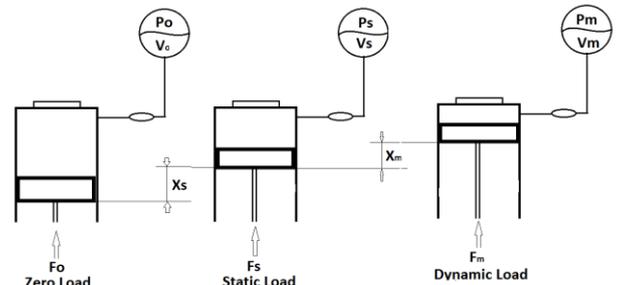
1. Zero Load (Case1)

2. Static Load (Case 2)
3. Maximum Load (Case 3)

The three cases are represented in the below figure, as we can observe the position of the piston and the displacement of the piston at each case.

We can observe in the figure that the piston displacement from X₀ to X_s

$$Z = x_0 - x_s$$



Representation of various cases

Stiffness of the system is force per the displacement

$$K_s(Z) = \frac{F(x)_0 - F(x)_s}{x_0 - x_s}$$

To arrive at the equations we have to compare different cases,

Comparing Case1 and Case2:

$$V_s = V_0 - A_c X_s$$

Where,

- V₀ is the initial volume
- (A_p * x) is the volume displaced

Pressure increase due to the compression is calculated by

$$P_s = P_0 + \frac{F_s}{A_p}$$

But, Type equation he

$$P_0 V_0^n = P_s V_s^n$$

$$P_0 = \left[P_0 + \frac{F_s}{A} \right] (V_0 - A_p \cdot X_s)^n$$

Comparing Case1 and Case3:

$$V_m = V_0 - A_p X_m$$

$$P_m = P_0 + \frac{F_m}{A_p}$$

$$P_0 V_0^n = P_m V_m^n$$

$$P_0 V_0^n = \left[P_0 + \frac{F}{A} \cdot (V_0 - A_p \cdot X_m) \right]^n$$

$$\left[P_0 + \frac{F}{A} \cdot (V_0 - A_p \cdot X_s) \right]^n = \left[P_0 + \frac{F}{A} \cdot (V_0 - A_p \cdot X_m) \right]^n$$

$$\left[P_0 + \frac{F}{A} \right] \left[1 - \frac{A_p \cdot X_s}{V_0} \right]^n = \left[P_0 + \frac{F}{A} \right] \left[1 - \frac{A_p \cdot X_m}{V_0} \right]^n$$

By Binomial Approximation and by neglecting the higher terms

So the equation becomes,

$$\left[P_0 + \frac{F}{A} \right] \left[1 - \frac{n A_p X_s}{V_0} \right] = \left[P_0 + \frac{F}{A} \right] \left[1 - \frac{n A_p X_m}{V_0} \right]$$

By simplifying,

$$\left[P_0 + \frac{F}{A} \right] \cdot [V_0 - n A_p \cdot X_s] = \left[P_0 + \frac{F}{A} \right] \cdot [V_0 - n A_p \cdot X_m]$$

$$P_m = \frac{F_m}{A_p}$$

$$V_0 = \frac{A_p P_m P_s X_m}{P_0 (P_m - P_s)}$$

M. Natural frequency of Non-Preloaded system

The frequency f of for the non-preloaded hydropneumatic suspension system can be calculated by using the formula

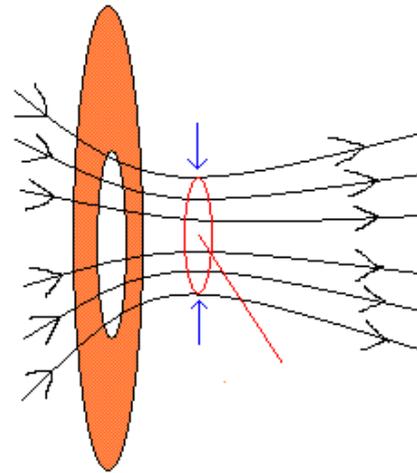
$$f = \frac{1}{2\pi} \sqrt{\frac{nF}{P_0}}$$

The frequency of suspension should be controlled to limit to constraint the vehicle from rolling uncontrollably. Therefore it is also required for the spring rate to be increased for constant bounce frequency. The curves of natural frequency with various static loads provide essential details about the suspension properties and behavior of a particular suspension system. The curves regarding the natural frequency and static load are plotted in the later part of the thesis.

N. Orifice

The main damper system represents the damper manifold with an orifice. the fluid undergoes turbulence which results in internal fluid friction and hence the transformation of kinetic energy to heat, resulting in damping.

Sharp edged orifice plate with a coefficient of .6



Orifice

The damping effect is calculated by using the formula

$$Q = a C_d \sqrt{\frac{2}{\rho}}$$

$$a = \frac{Q}{C_d \sqrt{\frac{2}{\rho}}}$$

Where,

- a = area of orifice
- Cd = Coefficient of discharge
- ρ = pressure difference
- ρ = Density of fluid

From the area of the orifice the diameter of the orifice is calculated.

O. Setup

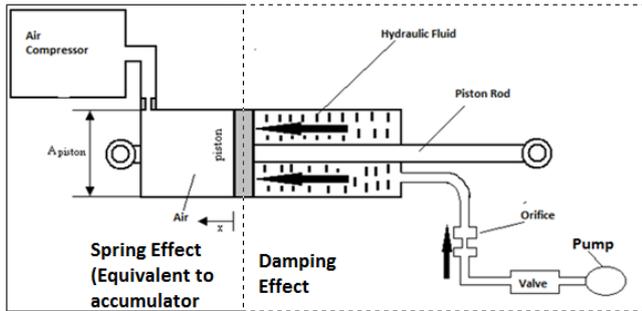
Since the force can't be applied directly to the piston rod, the force is applied in the form of pressure. Pressure when multiplied by area gives force. In this scenario, one end of the accumulator is connected to a 1HP pump which applies pressure over the piston of the cylinder and hence displaces the piston. The pump and the accumulator is connected via a 3 way valve and an orifice. The 3way valve enables us to control the flow through the pipe to the system. The orifice, as discussed earlier, is the primary damping unit for the system. a relief valve has been incorporated just prior to accumulator, for the added safety. The other end of the accumulator is connected to an Air compressor, which is used to vary the stiffness of the system. As we vary the stiffness, the resistive force against the piston displacement increases. Since gas is incompressible it gives the spring effect to the system.

These basic details of the mode as calculated are discussed below

- The dimensions of the cylinder calculated is shown in the figure
- Diameter of the piston =75 mm
- Cylinder inner Diameter = 75 mm
- Cylinder Outer diameter=85 mm
- Cylinder Length =200 mm
- Area of the Piston Ac=A= π * 75² =4.41x 10⁻³ m²
- Stroke length= 130 mm
- Material used for fabrication- Mild steel

P. Proposed design for Hydropneumatic suspension

The proposed Hydropneumatic actuator utilizes the two working fluids: compressed air and Hydraulic fluid. Air is used as working gas in this device and the pressure of gas could be adjusted easily. Stiffness of the device is adjusted by changing the gas pressure or the length of gas reservoir. The flow through the orifices creates a pressure loss over the orifices; multiplied by the area it is acting on, this loss is responsible for the damper force of the strut.



Proposed Model

The stiffness of the suspension can be altered changing the gas pressure by compression/expansion of the gas. Since we know the area of the Piston, the volume of the gas V is calculated from the piston position.

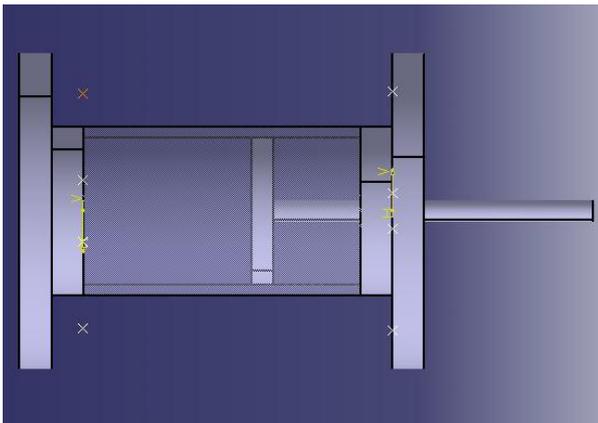
Using the Polytrropic law, we get the pressure.

$P.V^n = \text{Constant}$ (where n is the polytrropic co-efficient.)

i.e. $P_1.V_1^n = P_2.V_2^n$

The force on the cylinder block is computed by multiplying the pressure P with the piston cross section area. The force-deflection relationship yields the gas spring characteristics corresponding to the movements of the piston. Therefore, hydraulic conductance of an orifice, which is proportional to the orifice area, is defined. The instantaneous rate of flow through a damper orifice can also be written as

$$Q = c_d.A_0.\sqrt{(2\Delta P/\rho)}$$



CATIA model of Hydropneumatic Suspension

Therefore the stiffness of the suspension system is controllable. The mechanical section of the suspension is build-up in an equal way for the hydropneumatic suspension system as for an ordinary suspension system. The only difference is the replacement of the ordinary linear spring and damper strut by a hydraulic cylinder. Where the oil remains in the strut at an ordinary damper, here all oil is led outside through the damper manifold towards the accumulator. The motive is to split the spring behavior from

the damping behavior of the vehicle and to form two different relations which do not influence each other. Spring behavior relates to the oil volume displacement whereas, damping behavior relates to oil volume flow at the other hand.

III. CONCLUSION

The semi-active suspension system proves its importance over conventional suspension system in all the aspects. It varies its stiffness as per road conditions quickly which helps us to get a smooth and safe ride. Thus it improves the life of vehicle, health of the passengers.

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