MODELLING AND SIMULATION OF HYDROPNEUMATIC SUSPENSION FOR A CAR

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Abstract

The main functions of a vehicle suspension system is to isolate the road excitations experienced by the tyres from being transmitted to the passengers; to create anti roll, anti-squat and anti-dive effects that happen due to dynamic load transfer and to provide road holding. Passive suspensions have constant spring stiffness and damping coefficient limiting the suspension system unable to adapt to the dynamic conditions of a vehicle leading to deterioration of ride and handling performance of a vehicle. This necessitates search for adaptive suspension technologies and at present hydropneumatic suspension technology is promising and becoming popular in high end passenger cars. The present dissertation work has been on modelling and analysis of hydropneumatic suspension for passenger car.

A hydropneumatic suspension will have a metal chamber supporting the weight of the vehicle, and the chamber is partitioned by a diaphragm to accommodate air and oil on its either side. The suspension stiffness is varied by varying the pneumatic pressure in the chamber and the pneumatic pressure can be controlled by controlling hydraulic pressure on the other side of the diaphragm in hydropneumatic chamber. The variable damping coefficient of the suspension is achieved by allowing the liquid to flow through orifices. The oil pressure is varied depending on the longitudinal, lateral and vertical acceleration the wheel experiences as the vehicle travels.

In the present work, a hydropneumatic suspension of a quarter car is built using Matlab/simulink. The suspension model includes pump, valve, hydraulic cylinder, piston, orifice and gas compression and expansion. A PID controller operates the valve to achieve the desired suspension performance. It has been observed from the solution of the suspension model of a selected car, the acceleration of the sprung mass coming down by 79.5% compared to traditional suspension system.

Keywords: Hydropneumatic suspension, Matlab/Simulink, PID control, Quarter car

Nomenclature

A_0	Orifice area, m^2
A_p	Piston area, m^2
B	Bulk modulus of oil, N/m^2
g	Acceleration due to gravity, N/s^2
k	Polytrophic constant
k_d	Differential coefficient
k_i	Integral coefficient
k_p	Proportionality coefficient
K	Gas stiffness, N/m
P_a	Gas spring pressure, Pa
P_{sys}	System pressure, Pa
Q_s	Orifice flow rate, m^3/s
Q_v	Servo valve flow rate, m^3/s
y_{rel}	Piston displacement, m
ρ	Density of oil, kg/m^3

Abbreviations

DOF	Degrees of Freedom
PID	Proportional, Integral and Derivative
RMS	Root Mean Square

1. INTRODUCTION

It is always challenging to design a vehicle suspension system to maintain simultaneously a high standard of ride and body attitude control under all driving conditions. The requirements for suspension systems of modern automobiles are continuously increasing. The primary function of a suspension system is isolating the chassis from the roughness of the road. The suspension system should also react to the control forces produced by the forces acting on the palm sized patches at the tires. It should also keep the tires in contact with the road with minimal load variations and resist roll, squat and dive [1]. Safety and ride comfort characteristics of a vehicle mainly depend on the suspension system.

In passive suspension system the spring and damping coefficients have fixed rates. Hence passive suspension system usually consists of a non-controlled spring and a damper which can only offer a compromise between ride and handling.

The spring and damper characteristics required to design a suspension system to achieve superior handling is not the same as those to achieve superior ride. The problem of passive suspension is if it is designed to achieve critical dampening road holding will be lost and if underdamped system is designed the oscillations will be periodic and takes more time to dampen the vibrations. If suspension is stiffer, superior handling will be achieved and for superior ride suspension should be softer. Therefore, the performance of the passive suspension depends on the road profile [2, 3].

Since there is a limitation for passive suspension system there is a need for suspension system that could significantly reduce the ride comfort vs. handling compromise. The system should be capable of switching safely and predictably between a stiff spring and high damping mode (for handling) as well as a soft spring and low damping mode (for ride comfort). The inherent limitations of passive suspension system have led to the modeling of hydropneumatic suspension system.

Shirahat and Prasad [4] theoretically and practically determined the optimal design of passenger car suspension for ride and road holding for a full car. They have found that the driver's vertical displacement is reduced approximately by 74.2% in case of active suspension as compared to car with passive suspension. The settling time is also reduced from 6 sec to 3.5 sec. The vertical weighted RMS acceleration of seat and sprung mass is also reduced from 0.3032 m/s^2 to 0.0534 m/s^2 and from 0.2834 m/s^2 to 0.0492 m/s^2 using active LQR controller design since more weightage is given to ride comfort.

M. Senthilkumar and S. Vijayarangan [5] in their analytical and experimental studies on active suspension system of light passenger vehicle to improve ride comfort using PID controller found that ride comfort is improved by 78.03%, suspension travel is reduced by 71.05% and road holding ability is improved by 60% with active suspension system when compared with passive suspension system.

Haitao Wang and J.T. Xing [6] carried out an investigation on hydropneumatic landing gear system to reduce aircraft vibrations caused by landing impacts and runway excitations. It was demonstrated that the impact loads and the vertical displacement of the aircraft's center of gravity caused by landing and runway excitations are greatly reduced using this controlled suspension system, which result in improvements in the performance of landing gear systems, increases aircraft's fatigue life, taxiing performance, crew/passenger comfort and reduces requirements on the unevenness of runways.

Margolis and Nobles [7] in their research work concluded that by using semi active suspensions for heavy vehicles, vibrations control, such as roll and heave control, of the body of these vehicles were improved. In addition, this improvement affected the handling capability of vehicles positively. Yi and Hedrick [8] investigated on the influence of semi active suspensions on the dynamic tire force of vehicles. Dynamic tire force was considered as the criterion of handling capability. Using prototype of a vehicle, some experiments were performed on a suspension test rig. The results of these tests showed that improvement in ride comfort of the vehicle was achieved without rise in the dynamic tire force. In other words, improvement in ride comfort was accomplished without drop in the handling capability of the vehicle.

Hydropneumatic suspensions were introduced first in the 1950's. The hydropneumatic struts were installed on the prototype of a heavy tracked vehicle. This type of suspension system is popular due to its nonlinear characteristic and versatility. Due to nonlinear characteristic the spring rate increases as the load increases. The body roll and pitch is also reduced. Many controllable suspension systems make use of hydropneumatic springs because the hydraulic fluid can easily be channeled through ducts, orifices and valves. The ride height can be altered by adding or removing hydraulic fluid [9].

2. HYDROPNEUMATIC SUSPENSION SYSTEM SETUP

The principle of a Hydropneumatic suspension is illustrated in Figure 1. A double acting hydraulic cylinder is placed between the chassis and wheel of the vehicle instead of the spring and damper in conventional suspensions.

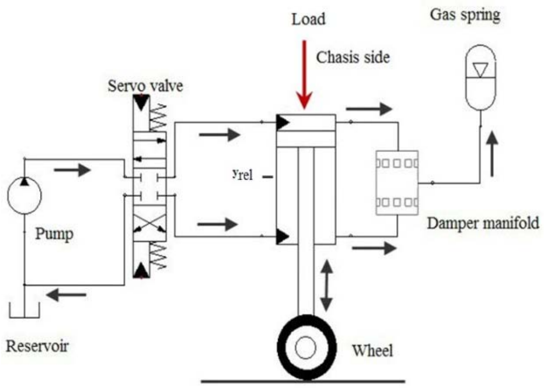

Fig. 1 Hydropneumatic suspension system setup

2.1 Setup for hydropneumatic system

One side of the hydraulic cylinder is connected to a 4/3 servo valve which controls the in and out flow of oil to and from the system. Whereas on the other side of cylinder, an accumulator or gas spring is connected through a damping manifold or orifice in between. Hydraulic capacitor or gas spring, consisting of two chambers, one connected to the oil circuit, the other, separated by a membrane, contains a gas (nitrogen). Cylinder's piston is connected directly to the wheel and vehicle body is placed on the top of the cylinder as shown in Figure 1. A pump is connected to the servo valve which pressurizes the fluid. The pump will produce constant fluid pressure to the system and a reservoir is also connected which is at atmospheric pressure. The valve is controlled by the input signal (z) from the PID controller. The actuator of the system is placed between chassis and wheel in place of conventional spring and damper. Parameter values for the different components in the system is shown in the Figure 2 [10-14].

2.2 Working Principle

As the cylinder is compressed with the activation of servo valve, oil is forced to the accumulator through the damping manifold. This condenses the gas inside the accumulator and produces a pressure in the accumulator that appears as the stiffness force on the cylinder.

The energy that is transferred into the suspension by external excitations needs to be dissipated to achieve a decay of the resulting oscillation amplitude and to avoid increasing amplitudes due to resonance. Therefore additional elements in the suspension system are necessary to transform the kinetic and/or potential energy of the suspension. In most cases kinetic energy is transformed into heat by application of a retarding force during the motion of the suspension elements. This retarding damping force usually is based upon the principle of friction. A flow resistor or damping manifold is placed in the flow path of a fluid and causes internal fluid friction which therefore causes a pressure increase in upstream of the resistor. This additional pressure is acting upon the active areas of the cylinder thus creating a retarding force, a damping force. Thus the damping force and spring force can be generated in the hydropneumatic suspension system.

```
%%Parameter Values of Hydropneumatic suspension system%%
Ap =19.95e-4            % Piston area m^2
Ri =1e20                % Resistance to internal leakage
Xlim =0.001             %Relative Spool displacement limit  m
Omiga =0.001            %Spool valve opening width
c=4e-6                  %Radial clearance area
Ko =0.03                % Constant = Cd*(2/Ro)^0.5
Re =1e20                %Resiatance to internal leakage
fv =2000                % Piston friction coefficient  Ns/m
Vo =100e-6              %Initial volume of oil in the cylinder
KL=40000                %Equivalent loas stiffness
m1=500                    % in kg
m2=100
g=9.81                  % acceleration due to gravity in m/s^2
Ps =2e7                 %Supply Pressure in  pa
Pt=0                    %Return pressure in  pa
B=1.5E9                 %Bulk modulus of oil   N/m^2
r=2e12;
Pa=7e5;                 %precharge pressure pa
Va=1e-2;                %precharge volume m^3
k=1.36;                %adiabatic exponent

row=900;               % density of oil kg/m^3
Cd2=0.3;               %coefficient of discharge
Ao=6.412e-4;           %orifice area
%%Parameter Values of 2-DOF Passive suspension system %%
m3= 500;
m4= 100;               %sprung & unsprung masses in kg
k = 800;               %stifness of spring N/m
c = 60;                %damping coefficient Ns/m
k2= 221306             %tire stiffness N/m
c2= 3000;              %tire daamping coefficient Ns/m
```

Fig. 1 Values for the different parameters in the system

3. MATHEMATICAL MODELLING

The components used in the Hydropneumatic system are modeled mathematically and a Simulink model is created according to the equations.

3.1 Dynamic equilibrium equations for Quarter car

Using Newton's second law of motion and examining the dynamic equilibrium of the two masses, the dynamic equations describing the system can be written as

$$m_1 \ddot{y}_{1rel} = A_p(P_B - P_A) - f_v \dot{y} - F_L - m_1 g \quad (1)$$

$$m_2 \ddot{y}_{2rel} = -A_p(P_B - P_A) + f_v \dot{y} - F_L - m_2 g + F_{ts} + F_{tc} \quad (2)$$

3.2 Double acting hydraulic cylinder

The oil inside the cylinder is slightly compressible and a simple linear assumption, that the current pressure is proportional to the current amount of oil in the cylinder relative to some nominal oil volume, was made. The proportionality factor here corresponds to the bulk modulus of elasticity of the oil used. Piston end of the cylinder is connected to the wheel, as shown in Figure 1. The output of the cylinder gives the pressure. Application of continuity equation to the cylinder chambers yields the equations for double acting cylinder.

$$Q_B - A_p \frac{dy}{dt} - Q_i - Q_{eB} = \frac{V_{o1} + A_p y}{B} \frac{dP_B}{dt} \quad (3)$$

$$A_p \frac{dy}{dt} - Q_A + Q_i - Q_{eA} = \frac{V_{o1} - A_p y}{B} \frac{dP_A}{dt} \quad (4)$$

3.3 Hydraulic capacitor or Gas spring

As oil enters to an accumulator, the gas inside the accumulator becomes compressed. As the cylinder is compressed, oil is forced to the accumulator through the damping manifold. Due to this the gas inside the accumulator is condensed and a pressure in the accumulator is produced that appears as the stiffness force on the cylinder. The bladder type of accumulators is employed in vehicle suspension [10]. A simple way of modelling this element would be to assume a polytrophic state change of the gas in the chamber. When oil is forced into the capacitor, the gas gets compressed, which results in an increase of pressure. This compression is assumed to be adiabatic relative to some initial or nominal state. The equation 5& 6 shows the final pressure and gas stiffness. Differentiating the spring force with respect to displacement gives the gas stiffness equation. The output of the gas spring is gas stiffness and final pressure.

$$P_a = \frac{P_o * V_0^k}{(V_o - (A_p * y))^k} \quad (5)$$

$$K = \frac{dF_a}{dy} = \frac{P_o * V_0^k}{(V_o - (A_p * y))^k} A^2{}_p \quad (6)$$

3.4 Damping manifold or Orifice

The hydraulic fluid in a hydropneumatic suspension is used as a medium to transfer the pressure on the active areas of the piston to the accumulator. Due to the suspension movement and therefore the displacement of the piston, the hydraulic fluid steadily flows between cylinder and accumulator with regularly changing flow direction. The flow resistor placed in the fluid flow, the kinetic energy of the fluid is transformed into heat due to shear flows inside the fluid. The flow resistor creates a pressure loss, which causes, via the active areas of the piston, a force which counteracts the motion of the piston. This force is therefore taking energy out of the oscillation and hence is a damping force. The damping force depends on the energy dissipated by of the oil flowing through the orifice connected between the cylinder and accumulator. It is assumed that the oil is incompressible and ΔP represents the difference between the pressures of the lower and upper chambers. From the mass conservation law and Bernoulli's equation [15] the equations 7 & 8 are derived. As oil flows through this orifice, a pressure drop proportional to the flow rate is created. This pressure appears on the piston as the damping force, which is proportional to the piston velocity. The equation 9 gives the damping coefficient which is obtained by differentiating damping force with piston displacement.

$$A_p * \dot{y} = C_{d2} * A_o * V \qquad (7)$$

$$\Delta P = \frac{\rho * A_p^{\,3}}{2 * C_{d2}^{\,2} * A_o^{\,2}} \dot{y}^2 \qquad (8)$$

$$\frac{dF_o}{dy} = \frac{\rho * A_p^{\,3}}{2 * C_{d2}^{\,2} * A_o^{\,2}} \dot{y} \qquad (9)$$

The flow rates through the orifice are given by equations 10 & 11.

$$Q_{s1} = C_{d2} * A_o * \sqrt{2(P_A - P_a)/\rho} \qquad (10)$$

$$Q_{s2} = C_{d2} * A_o * \sqrt{2(P_B - P_a)/\rho} \qquad (11)$$

3.5 Servo valve

Servo valves are used to start, stop, or change the direction of fluid flow. A 4/3 servo valve is used for the model. A servo valve has two input connections, one is connected to pump and another is connected directly to the reservoir. The two output connections are connected to cylinder ports. The output of the servo valve is flow rate. The equation 12 to 15 gives the flow rates from the valve and equations 16 & 17 gives the area generated in the valve due to input signal.

$$Q_{v1} = C_d * \mathrm{A}_a(x)\sqrt{2(P_A - P_{res})/\rho} \qquad (12)$$

$$Q_{v2} = C_d * \mathrm{A}_b(x)\sqrt{2(P_{sys} - P_A)/\rho} \qquad (13)$$

$$Q_{v3} = C_d * \mathrm{A}_c(x)\sqrt{2(P_{sys} - P_B)/\rho} \qquad (14)$$

$$Q_{v4} = C_d * \mathrm{A}_d(x)\sqrt{2(P_B - P_{res})/\rho} \qquad (15)$$

$$\begin{aligned} &\mathrm{A}_a = A_c = \omega * \sqrt{(x^2 + c^2)} \\ &A_b = A_d = A_r \qquad For - x \geq 0; z \geq y \qquad (16) \end{aligned}$$

$$\begin{aligned} &A_a = A_c = A_r \\ &\mathrm{A}_b = A_d = \omega * \sqrt{(x^2 + c^2)} \qquad For - x \leq 0; z \leq y \qquad (17) \end{aligned}$$

3.6 PID controller

The displacement of the servo valve is controlled by Proportional Integral Derivative (PID) control. The PID controller combines system motion information, allowing generation of a synthesized control signal. The PID controller is chosen to complete the mathematical model and to investigate the hydropneumatic quarter car model.

$$K(s) = K_p + \frac{K_i}{s} + K_d s \qquad (18)$$

Here, k_p represents a proportionality coefficient, k_i an integral coefficient and k_d a differential coefficient. These feedback coefficients can be adjusted to obtain the best control efficiency. In Figure 3, r(s) is the input signal, K(s) is the controller, G(s) is the plant and y is the feedback signal which is the displacement of the piston.

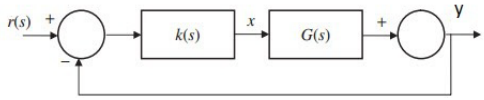

Fig. 2 Schematic diagram of the Proportional Integral Derivative controller

Figure 4 shows the simulink model of PID controller. Through numerical simulation experiments adopting a wide range of control parameters, found that the approximate optimum set kp = 4.6, ki = 2 and kd = 0 produced the best control efficiency for the system.

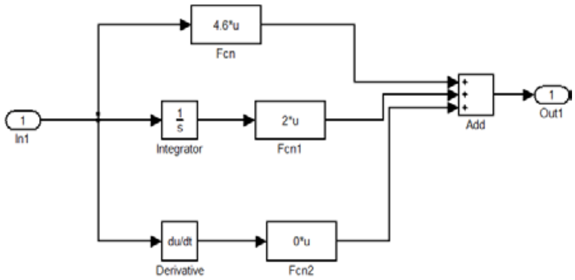

Fig. 3 Simulink model of PID controller

4. SIMULATION & RESULTS

4.1 Signals interconnection for simulation

Signals, interconnections and action of the different parts of the system are shown in Figure 5. The pump is connected directly to car engine, which produces constant system pressure (p_{sys}). The reservoir pressure (p_{res}) on the contrary is always low, roughly equal to atmospheric pressure, usually around 1 bar. The control input (z), for which positive z injects oil into the hydraulic cylinder, negative z allows oil to leave hydraulic cylinder. Q_v is the flow rate output from servo valve. P_A & P_B are piston and rod side pressures of hydraulic cylinder and P_a is pressure in capacitor.

The flow variables in the system are Qv and Qs respectively corresponding to the flows of oil from valve to cylinder and from cylinder to capacitor. The position of the plunger is y_{rel} (Figure 1); it is zero at the neutral (middle) position, positive if it is “above” that position, negative when it is “below” it.

Variable damping coefficient of the suspension is achieved by allowing the liquid to flow through orifices. An orifice or damping manifold is placed in the flow path of the fluid. This causes internal fluid friction which in turn causes pressure increase upstream of the resistor. This additional pressure acts on the active areas of the cylinder, creating damping force. The orifice is placed in between the cylinder and gas spring. The suspension stiffness is varied by varying the pneumatic pressure in the chamber and the pneumatic pressure can be controlled by controlling hydraulic pressure on the other side of the diaphragm in hydropneumatic chamber.

4.2 Hydropneumatic and Passive suspension simulink models for quarter car

From the formulas derived, the overall simulink model is created for simulation (Fig. 6).

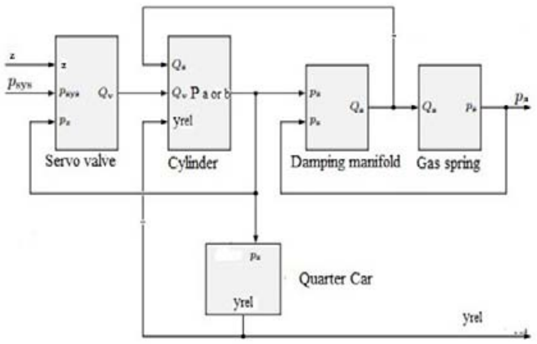

Fig. 4 Signals and interconnections in the system

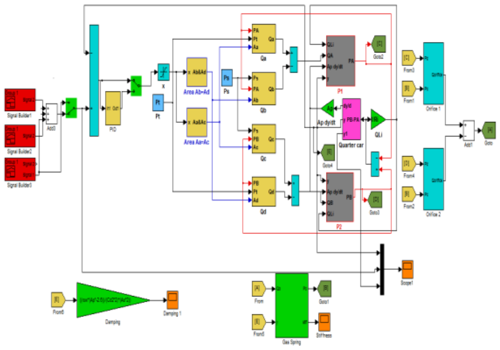

Fig. 5 Hydropneumatic suspension simulink model

4.3 Results

Performance of quarter car hydropneumatic suspension system and its corresponding passive system are compared using results of the Simulink model developed.

4.4 Input signal

Input signal in terms of displacement of spool of servo valve is modeled according to the road input. The signal is modeled by using signal builder in Simulink. Figure 7 shows the input signal. The input signal is modeled with one pot hole and two bumps in sine and square form.

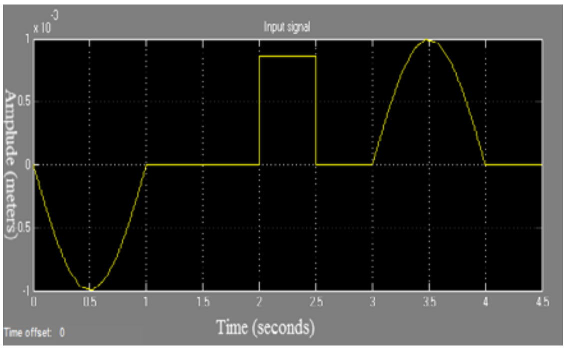

Fig. 6 Input signal

4.5 Sprung and unsprung mass displacements time history plots from simulink model

Figure 8 shows the unsprung mass displacement of the quarter car without PID controller. In general the suspension should follow the road profile to achieve road holding. Figure 8 shows the unsprung mass is not following the path of input signal and there is overshoot and error in the displacement of mass. The overshoot and error can be reduced by using the PID controller. Figure 9 shows the variation of displacement of unsprung mass with time. The suspension system is maintaining a permanent contact between tire and road surface which provide good road holding capability which intern provides good ride stability for the vehicle.

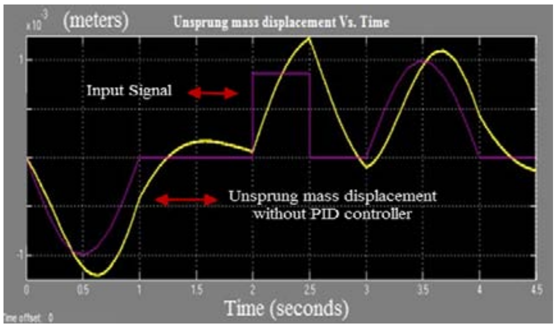

Fig. 7 Unsprung mass displacement Vs. Time without PID controller

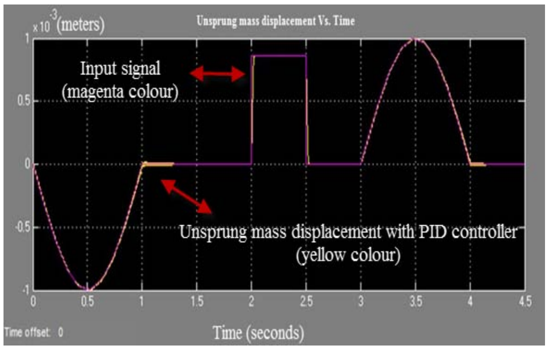

Fig. 8 Unsprung mass displacement Vs. Time with PID controller

Figure 10 shows the variation of displacement of sprung and unsprung masses with time The input signal is also shown in figure. Variation of sprung mass displacement clearly shows the decrease in amplitude levels of displacement. Maximum amplitude of displacement for sprung mass is about 1.1×10^{-4} m whereas maximum amplitude for unsprung mass is about 1×10^{-3} m. Figure 11 shows the sprung mass displacement of the passive system which shows higher amplitudes of 1.1×10^{-3} m compared with hydropneumatic system.

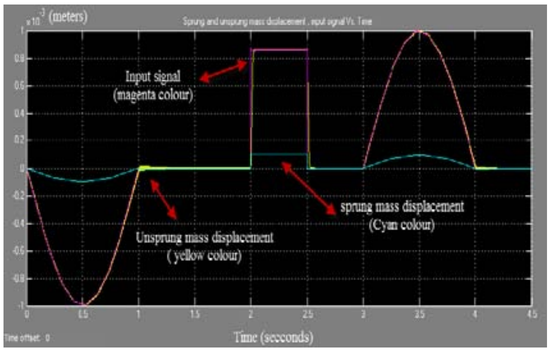

Fig. 9 Sprung and unsprung mass displacement, input signal vs. Time

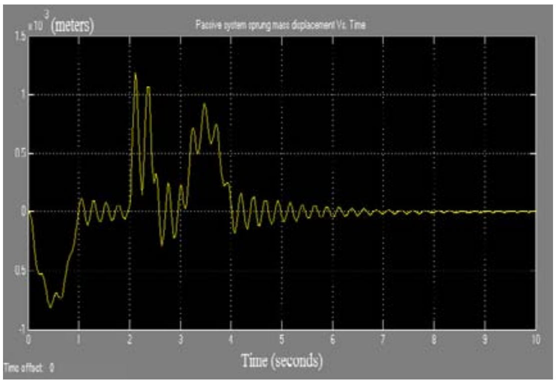

Fig. 10 Passive system sprung mass displacement vs. Time

4.6 Sprung mass acceleration time history plots from simulink model

Figure 12 shows the acceleration of sprung mass with hydropneumatic suspension system. The maximum acceleration of the sprung mass is about 0.08 m/s^2. Figure 13 shows the sprung mass acceleration obtained from the passive suspension system. The peak acceleration is about 0.399 m/s^2.The acceleration of the sprung mass is greatly reduced by using hydropneumatic suspension system and fluctuations in acceleration are also reduced by variable damping coefficient of the system.

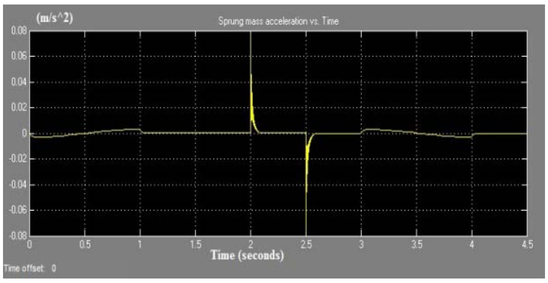

Fig. 11 Sprung mass acceleration vs. Time of hydropneumatic suspension system

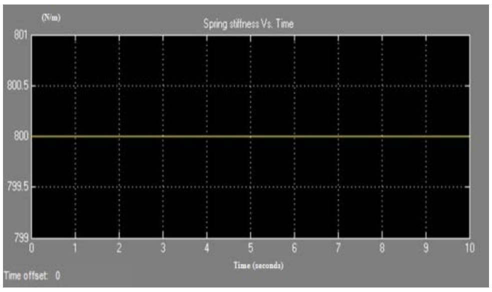

Fig. 12 Sprung mass acceleration vs. Time of Passive suspension system

4.7 Spring and Damping coefficients time history plots from simulink model

A passive suspension will store energy through spring and dissipate the energy through damper. Its parameters are fixed, being chosen to achieve a certain level of compromise between road handling, load carrying and ride comfort. Figure 14 shows the spring stiffness vs. time for the passive system. Since the stiffness of the spring is kept constant the curve appears linear. The stiffness of spring for passive system is 800 N/m.

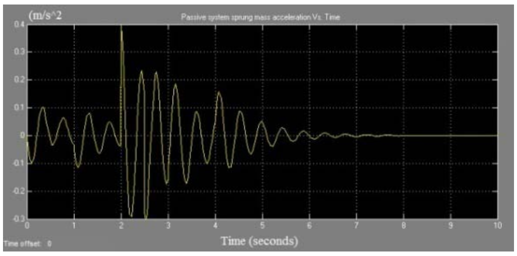

Fig. 13 Spring stiffness vs. Time of Passive suspension system

Figure 15 shows the gas pressure variation inside the accumulator with respect to time. The peak gas pressure generated due to the input signal is about 1184 Pa.

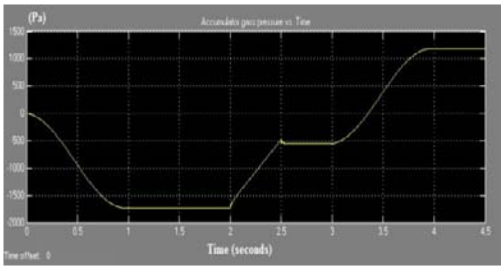

Fig. 14 Gas pressure variation vs. Time

Figure 16 shows the stiffness variation of the gas spring. As oil enters the accumulator, the gas inside the accumulator is compressed depending on the pressure of the oil coming out from the hydraulic cylinder. Stiffness property of a Hydropneumatic suspension is created based on the gas compression. Unlike steel springs, the stiffness is not linear, and it rises progressively with increase in suspension displacement. Maximum stiffness available with respect to the input signal is about 380 N/m. The hydropneumatic suspension system is able to produces different stiffness values depending on the road surface.

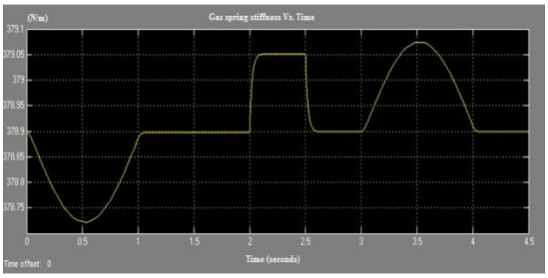

Fig. 15 Gas spring stiffness vs. Time for hydropneumatic suspension

Figure 17 shows the damping coefficient vs. time for hydropneumatic suspension system. Since flow resistor creates a pressure loss, which causes, via the active areas of the piston, a force which counteracts the motion of the piston. This force is therefore taking energy out of the oscillation and hence is a damping force. Unlike passive system damping coefficient is not linear, as the pressure inside the hydraulic cylinder changes the damping coefficient values are also varying and the maximum damping coefficient is about 43 Ns/m.

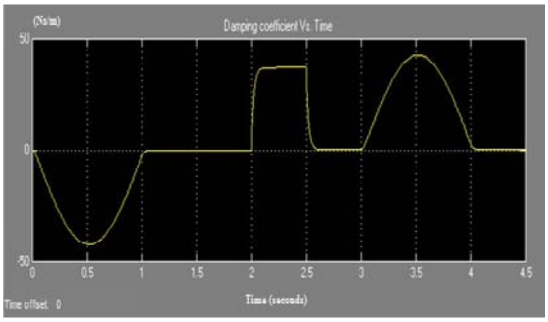

Fig. 16 Damping coefficient vs. Time for hydropneumatic suspension

5. CONCLUSION

The mathematical model of hydropneumatic suspension using simulink created and using it response of a quarter car to road inputs was studied. Unsprung mass of the system follows the road profile which significantly provides better road holding capability.

Results of simulation show significant improvement in the suspension behaviour. From the simulation results the maximum displacement of passive suspension system sprung mass is about 1.11×10^{-3} m, whereas the maximum displacement of hydropneumatic suspension system sprung mass is about 1.1×10^{-4} m. The displacement of the sprung mass is reduced by 90%.

It is further demonstrated that by using hydropneumatic suspension model, a reduction in the time length responses to return to static equilibrium positions is achieved, thus improving the performance suspension system and passenger comfort.

With the proposed hydropneumatic suspension model the reduced displacements and accelerations due to variable spring stiffness and variable damping coefficient will provides good ride comfort and handling to the vehicle.

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