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Original Article

Design, Analysis and Simulation of a Hydro-Pneumatic Suspension System for a Quarter Car Model with Two Degrees of Freedom

Prem Narayan Vishwakarma[1], Ajay Sharma[2], K Kedar[3], Abhishek Jha[4], Bharat Kumar[5]

[1,2,3,4,5]*Dept. of Mechanical Engineering, Amity University, Uttar Pradesh, India.*

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Abstract: *Hydro pneumatic suspension enables smooth suspension as the hydraulic fluid to the bladder containing the dinitrogen gas transmits the force due to uneven road. In the present work, design has been developed in such a manner that the isolation is possible to maximum extent, as four chambers of hydro pneumatic systems have been incorporated for each wheel. The dimensions have been fixed with respect to the already existing accumulator. The spacing of four chambers is done keeping in mind the same dimension so that more space is not required. Chambers of hydro pneumatic systems have been incorporated for each wheel for which the piston rod has been designed such that the impact of uneven road will be shared among the four chambers equally for individual wheel and increasing the smoothness of the suspension. The mathematical modelling is developed and the differential equations of motion are derived from it. Simulation is performed for quarter car model for two degree of freedom. The results shows the variation of displacement of the sprung and unsprung mass with respect to time.*

Keywords: *Hydro-pneumatic Suspension, Accumulator, Quarter Car Model, Sprung Mass, Unsprung Mass.*

I. INTRODUCTION

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

There is a need to develop a smooth and even suspension system due to lack of which, the passengers can develop various spine problems and the automobile can be damaged. When compared with spring suspension systems, hydro pneumatic suspension systems would provide more smoother isolation from the sudden forces due uneven road as the force is transmitted via piston to hydraulic fluid and further to bladder containing Dinitrogen gas. The bladder would be made of Desmopan, a material with high wear resistance as it needs to withstand the compression due to the force transmitted by the piston due to uneven road and simultaneously the reaction force of Dinitrogen as a result of compression.

LHM (Liquid Hydraulique Minerale), which has a density of 891 kg/m3, will be utilised as the hydraulic fluid. This substance is utilised to convey the pressure created by the piston's displacement. The following are benefits of hydropneumatic suspension in military vehicle applications: Progressive elasticity offers excellent driving comfort with light loads as well as the ability to load the vehicle heavily (the needs are mutually incompatible in classical suspension). Compressed nitrogen is roughly six times more flexible than steel spring components at static load of the suspension and is enclosed in the sphere by an elastic diaphragm, providing a high level of driving comfort. - the capacity to change the road clearance to reduce the height of the vehicle, increasing, for example, availability for aerial transport;- the potential for raising the road clearance, which would improve cross-country driving performance; - There is a chance that the suspension will level itself; - the hydro pneumatic column's small housing, - suspension servicing is rather straightforward (but maintenance staff must be trained), - Mass manufacturing costs that are often inexpensive; - the vehicle frequently has a lower unsprung mass; - The vehicle's roll-over resistance while moving in a curvilinear manner is determined by the suspension's capabilities (greater resistance to roll-over).

The hydro pneumatic column's potential horizontal location in the vehicle's rear suspension is crucial when it comes to big trucks since it might free up room for other special equipment or increase the loading or assault space capacity. [1-3]. HPSS consists of double acting cylinder, two oil chambers, damping valve and one gas accumulator. The two chambers are connected through a flow control valve. The accumulator is connected to the chamber through a damping valve. The results showed that the proposed HPSS gives improvement over the passive suspension respectively, the acceleration by over 50%, the suspension working space by 12% and the dynamic tyre deflection by 3% when compared to the passive suspension system[4-6].

II. DESIGN OF HYDRO-PNEUMATIC SUSPENSION SYSTEM

A Hydro Pneumatic Suspension System works by transmitting the force resulting because of an uneven road to the Desmopan bladder containing Dinitrogen via a hydraulic fluid (LHM). The hydraulic fluid is incompressible while Dinitrogen is compressible.

As far as the designing of Hydro pneumatic suspension system is concerned, we have to prepare a sketch (figure 1) which elucidates where all the system parts are going to be placed in the new design.

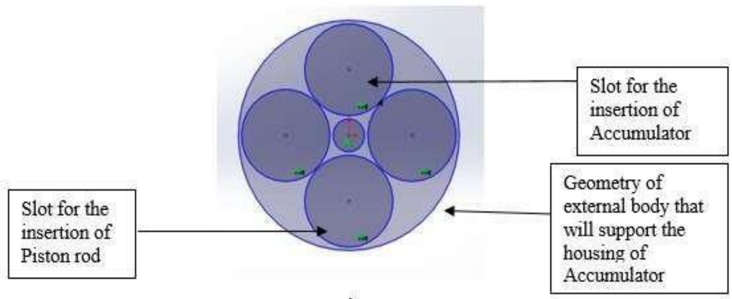

Figure 1: Basic Sketch of Hydro-pneumatic Suspension System

A. Components Design

It houses the four accumulators, the piston cylinder and the piston rod. The designed piston rod central component will be inserted into this slot and simultaneously the piston assembly for the 4 chambers is aligned as per the requirement. The four accumulators containing the Desmopan bladder will be inserted into these slots in such a manner that they are in the same level and the impact due to uneven road is transmitted equaly among the four chambers equally. Figure 2 shows the design of cylindrical accumulators along with their dimensions.

The accumulator has to bear high pressure as the hydraulic fluid exerts the force on the bladder and at the same time, thereis some reaction forces acting on the walls as well[7-10]. Thus, it needs to be manufactured with High tensile chrome Molybdenum Steel. The forced exerted by the hydraulic fluid acts on this bladder containing Dinitrogen gas which is compressible.

Figure 2: Cylindrical Accumulator with Dimensions

This piston rod is designed in such a manner that the force due to the uneven road gets distributed among the 4 pistons and similarly the hydraulic fluid gets pushed into the 4 accumulators at a time equally (Figure 3). The piston should be made of Al-Si alloy.

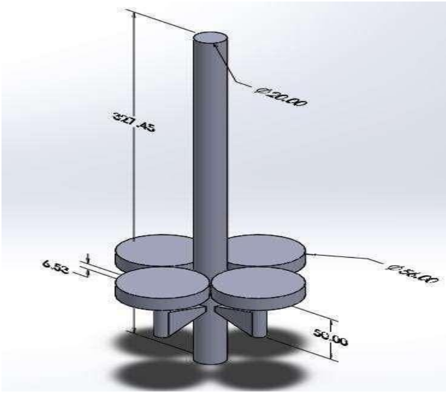

Figure 3: Design of Piston Rod along with Dimensions (Isometric View)

B. Assembly Design:

The individual accumulators need to be assembled in the main body (Figure 4). Four slots are present into which the accumulators are inserted. The accumulators have been fixed into the slots present on the Main body.

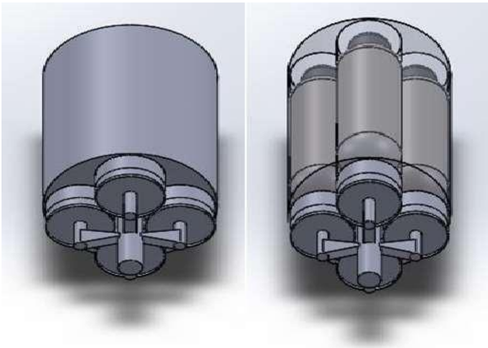

Figure 4: Complete Assembly

III. SPRING AND DAMPING CHARACTERISTICS OF SUSPENSION SYSTEMS

After the installation all the parts and adjusting the hydraulic pressure to the required level (by charging or discharging hydraulic fluid), the system now provides a suspension function. As the piston rod moves, the volume of liquid in the accumulator changes and with it the pressure (p1 → p2). This causes a change in the force at the piston rod, which, together with the change in position, determines the spring rate k.

The piston's position changes(s) as the applied force rises to a specific level (F to F*), which causes the hydraulic fluid to be transported to the accumulator. This shift continues until the system is balanced, at which point the pressure in the accumulator (and therefore on the piston's active surface) stabilises. The suspension system's operation and comprehension are based on this balance of forces (Figure 5). It will be applied to other computations in the parts that follow.

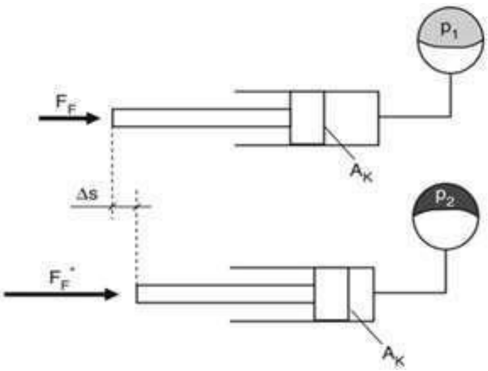

Figure 5: Forces Representations

The hydraulic fluid's kinetic energy is converted into heat (viscous friction) by a flow resistance that is used as dampening. This dampens the vibrations to the proper level and slows them down.

C. Calculation of Stiffness:

The flexibility of the entire arrangement is due to the gas that is present inside the accumulator. Due to the fact that it handles the majority of the work (spring stiffness), its characteristics are crucial to how the overall suspension system behaves.

Vo is defined as the volume of the accumulator under initial charging pressure Po and no hydraulic pressure is applied.

Pre-pressure is established during manufacture at a constant temperature of 293.15K (about 200C).

The gas volume in the accumulator does not change once the accumulator has been connected to the hydraulic system and the system has been pressurised, as long as the hydraulic pressure is less than or equal to the precharge pressure. The gas volume is compressed until force equilibrium or, for equal active areas on the gas and liquid side as in the diaphragm accumulator, is attained to pressure balance as soon as the hydraulic pressure surpasses the pre-charge pressure. For instance, adding a suspended mass to the suspension can raise the hydraulic pressure. Since the air is compressed relatively slowly and the new pressure level is held for a longer period of time, an isothermal change in state takes place in this situation. The temperature stays constant throughout the procedure because the heat produced during compression is dispersed to the surroundings. Calculating the initial load on the suspension and all future gradual load changes may be done using this isothermal change.

When suspension move on the uneven road the movement of the suspension when absorbing the shocks during operation is quite quick. These high rate state changes allow less time to dissipate heat. As a result the gas will change in temperature. Assuming that heat exchange is not possible, an adiabatic state change occurs described by the subsequent mathematical equations. Parameters for suspension systemare given in table 1. Static suspension force will act when the mass M is suspended and it will compress the suspension to P1 and V1

$F_1=Mg$ (1)

$P_1=F_1/A$ (2)

When isothermal change take place then

$P_1V_1 = P_0V_O \rightarrow V_1= \underline{P_0V_0}\, P_1$

When the vehicle move on the uneven road the piston got displaces by x than (3)

$V_2= V_1-Ax$ (4)

Where A is piston area

When adiabatic change of take place than $P_2V_2^n=P_1V_1^n$ (5)

‘n’ is polytropic constant

Put the value of V2 from Eq. 4 to Eq. 5

$P_2= P_1V_1^n /(V_1-Ax)^n$ (6)

Putting the value of V1 from Eq.3 Force on the piston will bePutting the value of P2

$P_2= P_1V_1^n /(P_oV_o/P_1-Ax)^n$ (7)

$F_k=P_2A$ (8)

$F_k= P_1V_1^n /(PoVo/P_1-Ax)^n * A_k$ (9)

Differentiating the Eq.7 w.r.t. x to get the stiffness,

$K= {dFk}/$

dx

After Differentiating Eq.9 and putting the value of P1 and V1

$K= F_1n\{P_0V_0/F_1\}^n / \{ P_0V_0/F_1\text{-}x\}^{n+1}$ (10)

After putting all the values of variable in Eq.10 we get the value of

K=56792 N/m

Table 1: Parameters of Suspension System

Parameter	Value
Mass of Vehicle(M)	1300Kg
Static Suspension Force(F1)	797.25 N
Pressure(Po)	3 Bar
Volume(Vo)	0.25L
Polytropic Constant(n)	1.3
Stiffness	56792.316N/m

D. Calculation of Damping Coefficient:

The suspension system's hydraulic fluid is used as a source to transfer the surface pressure from the piston to the accumulator. This occurs in the opposite direction of how the pressure is normally transferred. The energy that is transmitted to the suspension as a result of external stimulation needs to be decreased in order to achieve the desired outcome of lower vibration amplitude and to avoid the undesirable result of increased amplitude. Most of the time, when a braking force is applied during the motion of the hanging components, kinetic energy is transformed into heat.

In most cases, the principle of friction serves as the foundation for this delayed damping force (figure 6). Fluid friction, also known as viscous friction or hydrodynamic friction. A flow limiter is placed in the fluid path to create friction inside the fluid. As a result, the upstream pressure of the limiter increases. This additional pressure acts on the cylinder's operating area and produces a deceleration, or damping force. To achieve this Butterfly valve is used.

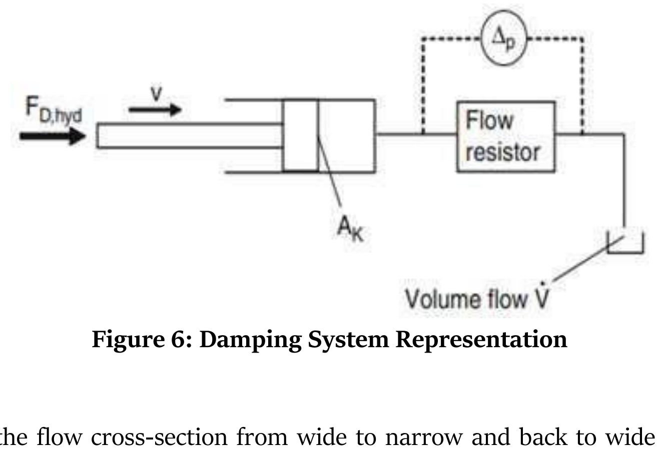

Figure 6: Damping System Representation

E. Throttle:

The gradual change in the flow cross-section from wide to narrow and back to wide slows the flow. A purposefully tiny hole is included into the fluid channel between the cylinder and the accumulator to give the cross-section of a specific throttle valve for specified extra damping, which typically has a circular shape. The hydraulic fluid flows at fast rates due to the narrow cross-section. Due to the significant flow velocity differential between the borehole's inner wall and the flow centre, high shear forces and therefore high-pressure loss. Parameters for orifice are given in table 2.

Flow through the valve is considered as laminar and pressure loss can be calculated as:

$\Delta p = V\, u\rho KD$ (11)

Δp is pressure difference across the flow through valve V is volume flow rateKD is constant
ρ is density of flowing fluid
u is viscosity

KD is calculated using the geometry and dimensions of the valve used

$$K_D = 128 l_D / \pi d^4_D \quad (12)$$

Table 2: Orifice Parameters

Parameter	Value
LD	4cm
dD	0.2cm
Density(ρ)	840Kg/m3
Kinematic Viscosity	0.0000179m2/s
Mass flow rate	21.69551kg/s
Velocity of flow	10m/s

$FD{,}hyd = \Delta p AK$ (13)

FD is damping force AK is piston Area
Δp is pressure difference across the flow through valveComparing damping force with its formula i.e.
FD=c*(velocity)
Putting all the values to get the value of C as C=35609 Ns/m

IV. DYNAMIC BEHAVIORS ANALYSIS

Suspension system analysis is performed using quarter car model to predict the behavior of the system. As the name indicates quarter car model consider one suspension unit in a 4-wheel vehicle. It is two DOF model (figure 7). So there are 2 masses, entire mass of vehicle is supported on the suspension system is sprung mass (M) and unsprung mass is basically a mass of wheel.

Now we will drive the governing equation for the system. As the system 2 DOF hence 2nd degree differential equation will form. We will divide the above quarter car model into two systemi.e. sprung mass and unsprung mass. We will use FBD for depicting the forces acting on the suspension (figure 8). X1 is the input given by the road which will make oscillate the suspension system. X3 movement of the vehicle due to the impact of X1.

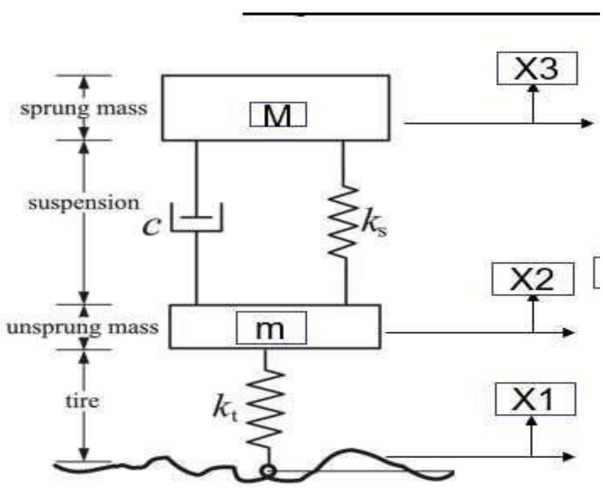

Figure 7: Modeling of Suspension System

A. Sprung Mass:

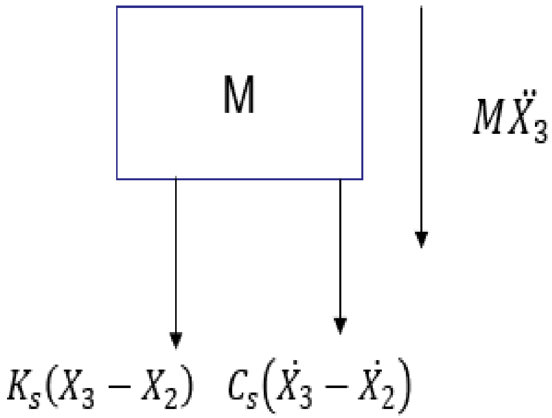

Figure 8: FBD of Sprung Mass

There will be 3 types of force acting on this sprung mass i.e. Spring force, damped force and inertia force which acts in opposite direction to the motion. All these forces will act in downward direction.

Using 2nd law

$M\ddot{X}_3 + (X_3 - X_2) + C_s(\dot{X}_3 - \dot{X}_2) = 0$

$$\ddot{X} = \frac{1}{\ } (K\ (X - \ ) + C\ (\dot{X} - \dot{X}))$$

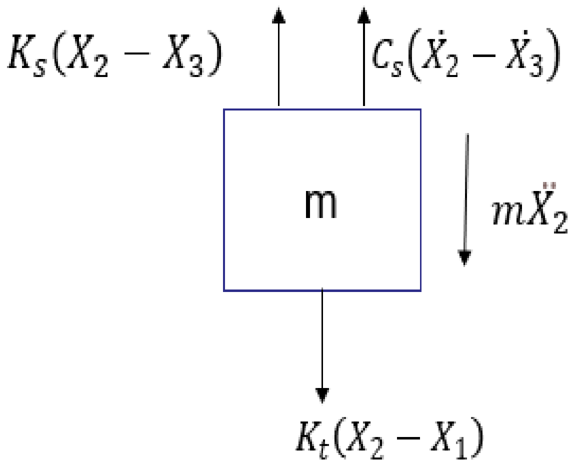

Figure 9: FBD of Unsprung Mass

Fromabove FBD (figure 9) we can see there will 4 different forces act and in this mass there will be influence of both of the spring.

Using 2nd law:

$$m\ddot{X}_2 + (X_2 - X_1) = K_s(X_1 - X_2) + C_s(\dot{X}_1 - \dot{X}_2)$$

$$m\ddot{X}_2 + (X_2 - X_1) - K_s(X_2 - X_3) - C_s(\dot{X}_2 - \dot{X}_3) = 0$$

$$\ddot{X}_2 = \frac{1}{m} (K_t\ (X_1 - \ ) + K_s\ (X_2 - X_3\ ) + C_s\ (\dot{X}_2 - \dot{X}_3))$$

To solve this differential equation we will use the MATLAB/Simulink where we give X1 as input in the form of STEP function of speed bump of 10cm.

Table 3: Simulink Parameters

Parameter	Values
Sprung mass(M)	325 Kg
Unsprung mass(m)	80Kg
Suspension Spring Stiffness(Ks)	227168 N/m
Suspension Damping Coefficient(CS)	35609 N-s/m
Tire Stiffness(Kt)	200000 N/m
Speed Bump Height(X3)	10 cm

As an input signal a Bump of height 10cm is consider. It assumed to be step function which will remain constant with time. Simulink will show the result in the form of displacement of sprung and unsprung masses. It will also show the settling time along with this we can also calculate the peak overshoot which will be helpful in predicting the stability.

Modelling on Simulink model for suspension system is presented in figure 10

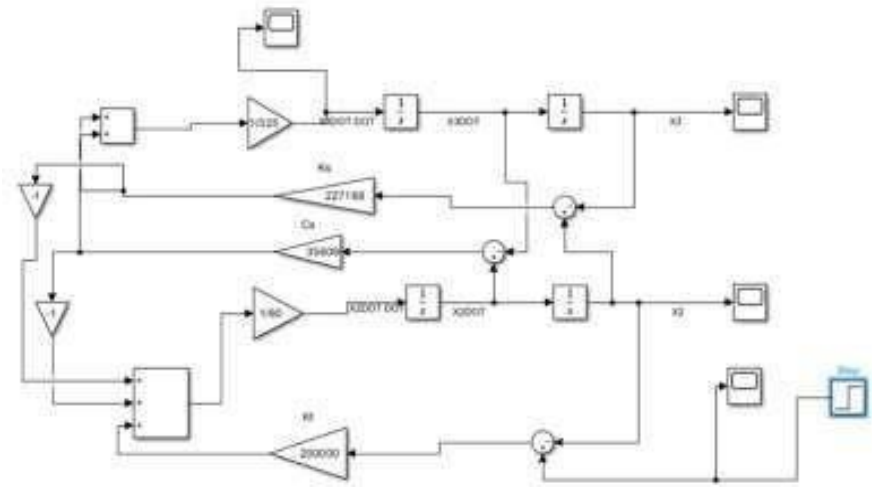

Figure 10: Simulink Model

V. RESULT AND DISCUSSION

Input(X1) is shown in figure 11. Step Input value is taken as 10cm

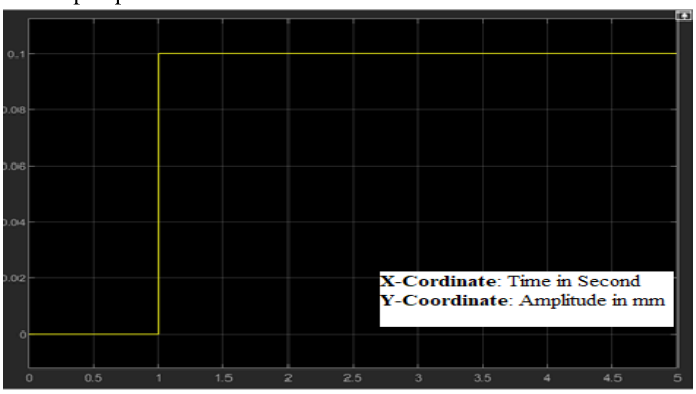

Figure 11: Plot for Step Input Value (X1)

Displacement(X2)

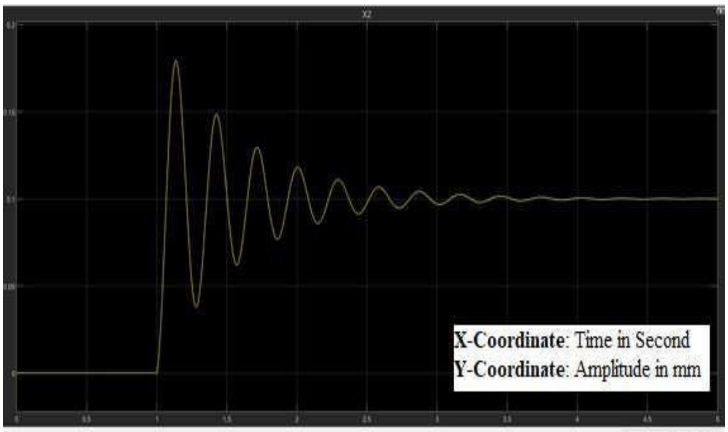

Figure 12: Plot for Displacement (X2)

The displacement for the unsprung mass is plotted w.r.t time (Figure 12). The displacement of wheel at t=1 sec is touching the ground. The settling time is found to be at t=3.737 sec. The highest displacement is found to be 0.1787cm at t=1.41 sec.

Displacement(X3)

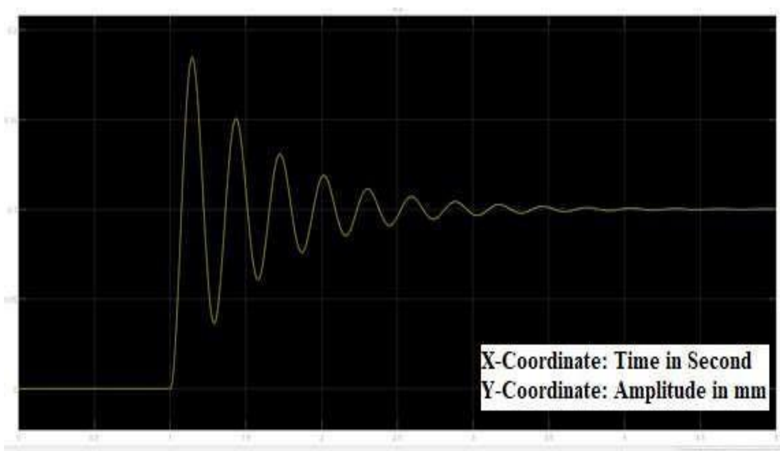

Figure 13: Plot for Displacement (X2)

The displacement for thesprung mass is plotted w.r.t time (figure 13). The displacement of wheel at t=1 sec is touching the ground. The settling time is found to be at t=3.75 sec. The highest displacement is found to be 0.1839cm at t=1.152 sec.

VI. CONCLUSION

The design has been developed in such a manner that the isolation is possible to maximum extent as 4 chambers of hydro pneumatic systems have been incorporated for each wheel. The dimensions have been fixed with respect to the already existing accumulator. The spacing of 4 chambers is done keeping in mind the same dimension so that more space is not required. Chambers of hydro pneumatic systems have been incorporated for each wheel for which the piston rod has been designed such that the impact of uneven road will be shared among the 4 chambers equally for individual wheel and increasing the smoothness of the suspension.

The spring & damping characteristics of the suspension system have been determined successfully.The suspension systemhas been modelled with the free body diagramand the subsequent differential equations have been obtained. The obtained differential equations have modelled in Simulink and have been helpful in the determination of displacement of sprung and unsprung masses.

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