DESIGN AND FABRICATION OF HYDRO-PNEUMATIC SUSPENSION SYSTEM

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Abstract: The aim of this paper is to develop model containing the hydro-pneumatic suspension system for four wheelers and heavy vehicles. The hydro-pneumatic system uses an accumulator to generate spring force (similar to the Hydro Active suspension system of Citroën) 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".

Keywords: Hydraulic accumulator, Hydraulic cylinder, helical spring

1. 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.

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.

The most important item of this system is the gas chamber, therefore the stiffness will be defined, basically, by the volume and pressure of the gas. In   some vehicles that use hydro-pneumatic suspensions, the chamber is built from two parts, that joined, take the shape of a sphere, and the gas is separated from the hydraulic fluid by a flexible diaphragm, as Yohsuke (1999). This diaphragm avoids gas leaks even having hydraulic fluid losses, if the system is not sealed perfectly. The hydraulic fluid drained of the system flows to a reservoir and return to the system through the use of a hydraulic pump, keeping the vehicle height constant. A semi-active control can be done through an adjustable valve that increases or decreases the damping, achieving an optimized system. An active suspension may be achieved by the control of the hydraulic fluid volume, adjusting the vehicle height. This configuration flexibility constitutes one of the most outstanding advantages of hydro pneumatic suspension systems
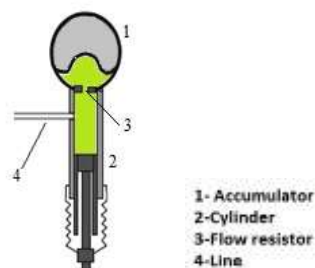

1- Accumulator
2-Cylinder
3-Flow resistor
4-Line

Figure 1.1 Elements of hydro-pneumatic suspension with Accumulator

The model is validated by measurement data on both component level as well as full vehicle level. The model can be used for further research with respect to alive damping control or implementing the system in other vehicles. As the vehicle is designed without wheel suspension, wheel loader drivers are exposed to high levels of whole body vibration which influences ride comfort negatively. The work

presented in this thesis has the aim to investigate the potential in adding an axle suspension to a wheel loader in order to reduce vibrations and increase handling quality.

It is of great significance that developed a suspension which is not only satisfied the stability and passenger comfort but also realized compact space constraints. The hydro-pneumatic suspension system with its superior non-linear characteristics

and excellent vibration reduction performance satisfies the requirement of the engineering large-tonnage vehicle ride comfort, and improves the riding comfort of vehicles to the maximum

2. Theoretical determination of hydro-pneumatic spring characteristics

In the following development the gas is considered inert in order to avoid changes of the spring characteristics while it is working. The ideal gas model is considered as well as an isothermal process, which means, the process of compression and expansion of the gas occurs with constant temperature. This assumption does not represent exactly the reality, due to the gas heating. If the system is running in high frequencies, there is no enough time to allow the heat exchange with the environment, and a hysteresis phenomenon will occur, as showed by Els and Grobbelaar (1993). This effect will be not considered once it contributes to dissipate energy of the system.
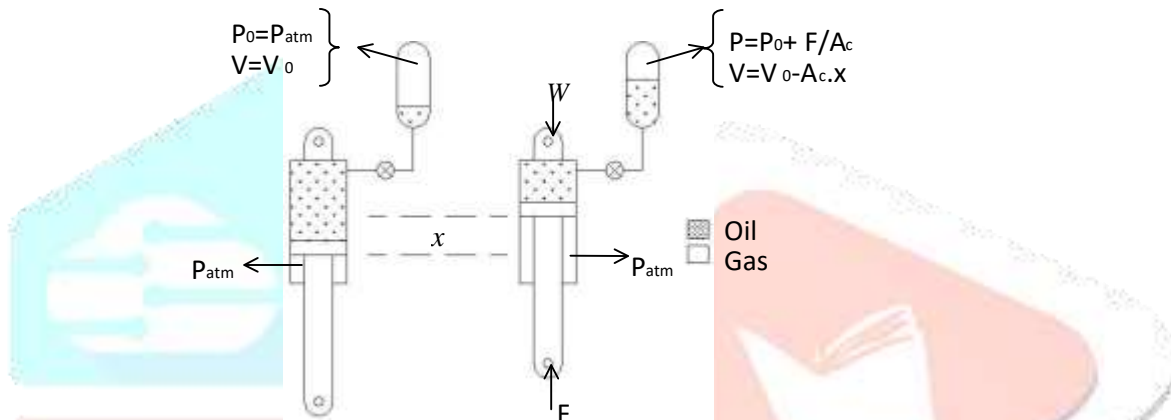

Figure 2.1 Forces and pressures acting on the hydro-pneumatic suspension

According to the hypotheses mentioned above,

$$PV = nRT \qquad (1)$$

$$PV = \text{constant} \qquad (2)$$

Where P is the pressure, V is the volume, T is temperature, n is number of moles and R is the universal gas constant.

To find the gas spring stiffness, a function that represents the force (F) done by the hydraulic cylinder as a function of the wheel displacement (x) must be developed. The initial state considered is where force is zero for null displacement. This condition occurs when the gas pressure inside of the chamber is atmospheric, which means, the force in both sides of the cylinder piston are the same. The Fig. 2(a) and (b) shows the forces and pressures for the initial condition and for generic condition, respectively.

To each piston displacement, due to force acting on the vehicle wheel, an oil volume will be displaced to inside or outside of the chamber, which will decrease

Increase the gas volume since their initial value V_0. As the oil volume displaced (V_d) is

$$V_d = A_c x \qquad (3)$$

where A_c is the piston area, the final gas volume V inside of the chamber is

$$V = V_0 - x A_c \qquad (4)$$

Considering Eq.(2) and Eq.(4) it is possible to see that

$$P_0V_0 = P(V_0 - x A_c) \tag{5}$$

The force acting on the wheel changes the gas pressure inside the chamber to

$$P=(F/A_C)+P_0 \tag{6}$$

Replacing Eq.(6) in Eq.(5)

$$(F/A_C)+P_0=P_0V_0/(V_0-XA_C) \tag{7}$$

so the force will be obtained multiplying Eq.(7) by the cylinder area A_c.

$$F=A_C ((P_0V_0)/(V_0-A_Cx))-P_0 =((P_0A_C^2x)/(V_O-A_Cx)) \tag{8}$$

The Eq. (8) shows that the force as function of piston displacement has a non-linear behaviour. Therefore, the spring stiffness coefficient is not constant, as in the conventional systems.

For each displacement x, with origin in the condition that F(0)=0, we will define the stiffness K_s as:

$$K_S= (F/x) \tag{9}$$

Replacing Eq. (8) in Eq.(9), the hydro pneumatic spring is given by

$$K_S(X)= ((P_0A_C^2)/(V_0-A_CX)) \tag{10}$$

The Eq.(10) shows that the spring stiffness does not depend on the chamber shape where the gas is confined, but depends on the initial pressure, the initial gas volume and the variation of the gas volume. This is an important condition to achieve the best economic project optimization, once a chamber with a simple shape can be used, allowing project costs reduction. As the gas pressure depends on the piston area and the vehicle load, the easiest way to range the spring stiffness is changing the initial gas volume, therefore changes on the hydraulic cylinder can mean high costs. An important issue that must be considered at the spring project conception is when the vehicle load changes, the pressure and the volume of gas change together, resulting in variations on the gas spring stiffness.

The local stiffness K_l is usually intended, for non-linear systems, as the derivative of the force function with respect to the displacement x. Deriving Eq. (8) with respect to x is obtained K_l as:

$$K_l(x)=((P_0V_0A_C^2)/(V_0-A_Cx)^2) \tag{11}$$

Therefore, it is interesting to see that this stiffness is suitable for local incremental analysis only. The product of $K_l(x).x$ does not give the value of F(x), due to non-linearity. Then, this definition is not adequate for suspension analysis, because the hypothesis of small displacements is not acceptable, especially for high non-linearity.

For modelling and simulation aims, it is convenient to work with a new displacement variable z with origin at the static equilibrium position x_s and define a stiffness variable coefficient K(z) that leads the difference between the force at a given displacement z and the force at z=0. Then,

$$z=x-x_s \tag{12}$$

$$K(z)=(AF/z)=((F(x)-F(x_s))/(x-x_s) \tag{13}$$

Where F(x) and F(x_s) can be calculated using Eq. (8).

With this definition is satisfied that $\Delta F(z)=K(z).z$ for all z.In numerical simulations the value of K(0) is set by interpolation, avoiding divide by zero.

3. Definition of Initial Gas Volume and Chamber Dimensions

The geometric configuration of all suspension components, the material selection, the dampers valves adjustments and the dimensions of all components are the keys to the suspension system design. These parameters, including other like tire stiffness, suspension stroke, mass of wheel and tire set, define the vehicle response to external stimulation. This way, the optimization process of all components will define the quality of the suspension system. Frequently the behaviour of the components are unknown analytically, but the main interest is on the vehicle response and not on the mathematical model of the system. To optimize the components, the manufactures usually confront the responses from systems empirically adjusted with the experience of trial pilots, and modify the characteristics of dampers, springs and other components to get a satisfactory system response, as Picado (1998).

However, some parameters may be set to initial values that are near to the adequate value for the application. This is the case of the spring stiffness in conventional suspensions.

For hydro-pneumatic suspensions, the non-linear stiffness is function of initial pressure and volume of the gas, and the area of the cylinder. Assuming that the cylinder area is already specified, the gas volume at the atmospheric pressure will define the stiffness of the system. Obviously, the working volume and pressure depend of the cylinder area and vehicle weight, since

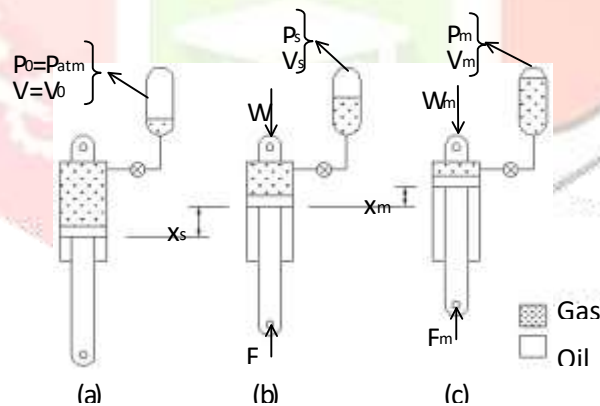$$P_S = W/A_C \tag{14}$$

And

$$V_S = (V_0 P_0)/P_S \tag{15}$$

Where W is the load in each wheel, P_s is the pressure in the static equilibrium, and V_s is the corresponding volume.

Then, additional criteria may be specified to determine initial volume V_0, and consequently the non-linear stiffness function.

One important advantage in relation to conventional suspension system is that any required change of the stiffness from the initial project can be obtained through changing the initial volume of gas only.

The criteria used in stiffness and the initial displacement (x_m), this study to define the hydro pneumatic spring gas volume assumes a maximum axle relatively to the vehicle frame, in the

Maximum attended stroke capability and thus can lead to deterioration damage, as Herovat considered happens when kind of vehicle can have a around five times) and the load. This criteria reflects a limited suspension prevent excessive suspension bottoming, which of ride comfort and possible premature structural (1997). The most severe load condition the vehicle has its maximum static load (f_{em}) (this weight difference between empty and full loaded maximal dynamic load factor (f_d) is achieved.

The Fig. 3.1 shows a scheme of the pressure, volume and load for each condition, used for the initial gas volume setup.

Figure 3.1 Criteria to define the initial volume of gas

The condition (a) means force equal zero acting at the suspension. In other words, the pressure inside of the chamber is atmospheric and the displacement of the cylinder is zero. Starting from this position, any load change will result in an alteration of the static position.

At condition (b), the maximum static load that the vehicle will transport is acting, resulting in a pressure (P_s) and a volume of gas (V_s) inside of the chamber. The static displacement of the piston in this condition is x_s. At the condition (c), the dynamic load is acting, which means, the bigger load static condition times the dynamic load factor, that results in a pressure (P_m) and a volume of gas (V_m) inside of the chamber. As described before, the criteria used to define the initial volume of gas allows a maximum displacement for the axle, therefore the displacement x_m is assumed as the maximum allowed displacement, happened in the maximum load condition acting on the wheel.

Considering conditions (a) and (b), in Fig. 3, and Eq.(2), is obtained:

$$P_0V_0=P_5V_5=P_5(V_0-A_Cx_5) \tag{16}$$

$$V_0=(A_Cx_5P_5)/(P_5-P_0) \tag{17}$$

To determine the x_s, the condition (a) and (c) must be considered, since the x_m is known, doing:

$$P_0V_0=P_mV_m=P_M(V_0-A_C(x_5+x_m) \tag{18}$$

Isolating x_s in Eq.(18) and replacing it in Eq.(17), the expression to calculate the initial volume of gas inside of the chamber at atmospheric pressure, following the criteria above, is given by

$$V_0=A_CP_mP_5x_m/P_0(P_m-P_5) \tag{19}$$

Note that P_0 is the atmospheric pressure, as mentioned, and P_s and P_m can be easily calculated by

$$P_S=W/A_C \tag{20}$$

And

$$P_m=W_m/A_c \tag{21}$$

4. Dynamic Behaviour Analysis

The purpose of this part of this work is to assess the hydro-pneumatic suspension system performance, compare with a conventional suspension system and identify what parameters have significantly influences in its performance. The adopted mathematical model to perform the dynamic analysis was a one-quarter of vehicle, showed in Fig.(4.1). This model is the same for both kinds of suspension, being the suspension spring stiffness K_s the only difference, which means, K_s constant for conventional suspension and given by Eq.(11) for hydro-pneumatic suspension.
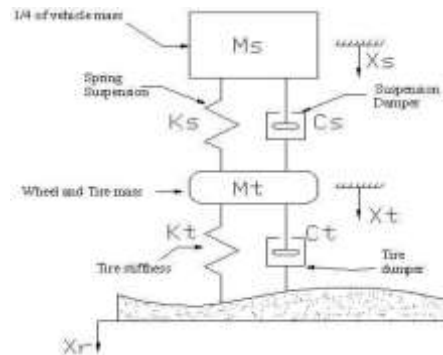
Figure 4.1. Model of on-quarter of vehicle.

Table 4.1. Variables used in the ¼ vehicle model.

Variable	Unit	Description
k_s	N/m	Spring stiffness of the
k_t	N/m	suspension Tire stiffness.
c_s	Ns/m	Suspension damper.
c_t	Ns/m	Tire damper.
x_s	m	Vertical displacement of the vehicle mass.
x_t	m	Vertical displacement of the tire and wheel mass.
x_r	m	Profile of the ground.
m_s	kg	Mass of ¼ of vehicle.
m_t	kg	Mass of wheel and tire.

5.   Influences of the damper

The damping choose of a system is extremely important to assure the vehicle designed characteristics of comfort and safe. It is not a easy task to be solved mathematically due to several existent variables must be taken in account in the math model. The ideal damping is chooses by practical trials with experienced pilots responsible to optimize the suspension, as Picado (1998).

To determine the damper in this work, the Eq.(22), defined in Thomson (1978) for a one-degree of freedom system, is used as a good approach. With this calculated value, the curve time x vehicle mass displacement for an initial pulse is obtained and, if necessary, the damper is changed to adjust this response as desired. The damping coefficient ξ adopted as ideal in this work is 0,7.

$$\xi = C_S/(2\sqrt{k_S m_S}) \tag{22}$$

In the Eq.(22), can be seen that the damping coefficient can be changed by three variables: the vehicle mass, the damper and the suspension stiffness. Considering the hydro-pneumatic suspension system, a load change has more influence on the damping coefficient, when compared with conventional systems, because for the first, a load change means a stiffness change as well, as Eq.(11).

To perform the dynamic analysis a trailer used to spray crop protection is considered due to its load can range from 2746,8N up to 17461,8N and brings large changes on the damping coefficient for the hydro-pneumatic suspension. Three cases are analysed to determine how much the load change affects the damping coefficient and what possible suggestions to minimize this problem. The ideal damping coefficient considered is 0,7 and the stiffness is achieved by Eq.(11) considering the static condition, once during cyclic excitation, the stiffness is around this value.

5.1 1ˢᵗ Case: Damping is defined by the medium vehicle load.

The damper is calculated by Eq.(23), considering a medium load condition, and kept constant for all other load condition. The Tab.(5.1) shows the values used for this analysis. Can be seen that the damper coefficient has largest changes in the hydro-pneumatic suspension system.

The responses for a initial displacement are showed in Fig.(6a) and (b) for the hydro-pneumatic and conventional system, respectively.

Table 5.1 Parameters used in the first case.

m_s [N]	Hydro-pneumatic Suspension			Conventional Suspension		
	k_s [N/m]	c_s [Ns/m]	Ξ	k_s [N/m]	c_s [Ns/m]	Ξ
8829	119475	5245	0.25	529740	21615	0.49
5886	54276	5245	0.46	529740	21615	0.61
4414	31199	5245	0.7	529740	21615	0.7
2943	14471	5245	1.26	529740	21615	0.86
1471	4090	5245	3.35	529740	21615	1.21

5.2 2ⁿᵈ Case: Damping is modified for each static load condition to keep the same ξ for all load condition.

In this case, for each static load, the damper is changed to achieve the desired damping coefficient kept constant during the dynamic analysis. The damping coefficient is kept constant 0,7 for all static load condition. The Tab.(5.2) shows the values used for this case.

The responses for initial displacement for both type of suspension system are showed in fig.(5.1) below.
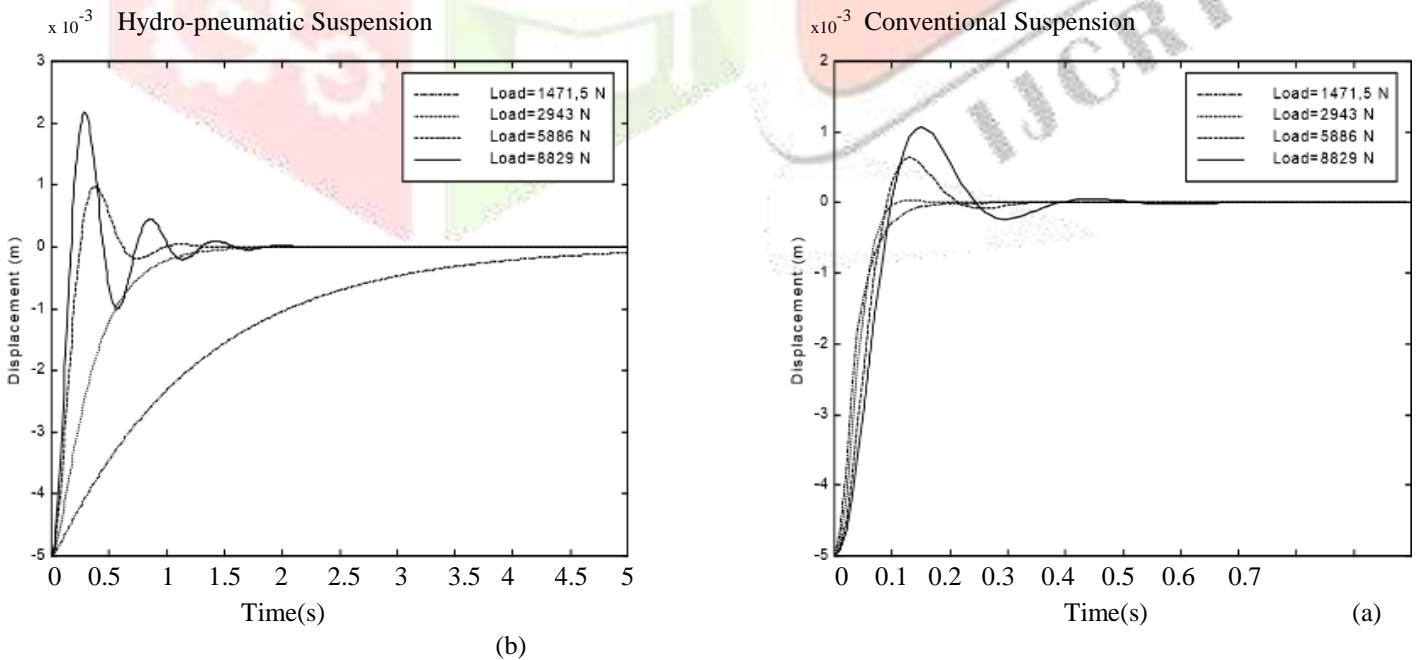

(b)

(a)

Figure 5.1. Response for initial displacement for the first case.

Table 5.2 Parameters used in the second case.

m_s [N]	Hydro-pneumatic Suspension			Conventional Suspension		
	k_s [N/m]	c_s [Ns/m]	ξ	k_s [N/m]	c_s [Ns/m]	ξ
8829	119475	14517	0.70	529740	30568	0.70
5886	54276	7989	0.70	529740	24959	0.70
2943	14471	2917	0.70	529740	17649	0.70
1471	4090	1096	0.70	529740	12479	0.70

From the above results it is observed that the operating frequencies are very less than the yield strength of the material. The yield strength of the material used for Hydro pneumatic suspension is 250 MPa.

According to the VonMises Stress Theory, the VonMises stresses of Hydro pneumatic suspension operating frequencies are less than the yield strength of the material.
Hence Hydro pneumatic suspension is safe under the operating loads, and the design of Hydro pneumatic suspension is safe and having more

5.3   3rd Case: Damping is modified the whole time to always maintain the damping factor constant.
    The behaviour of the suspension is analysed being the damper value changed constantly to keep the damping factor constantly equal 0.7, which means, whenever it happens an increase of the force of the spring, the force of damping will be reduced and vice-versa. This analysis will not be performed for a conventional suspension, because once the damping was adjusted in the static condition, it will not be changed due to the spring stiffness is constant. The Tab.(5.3) shows the values for this case, considering the range from the lower wheel position up to the higher wheel position (from -0.05m up to 0.05m).
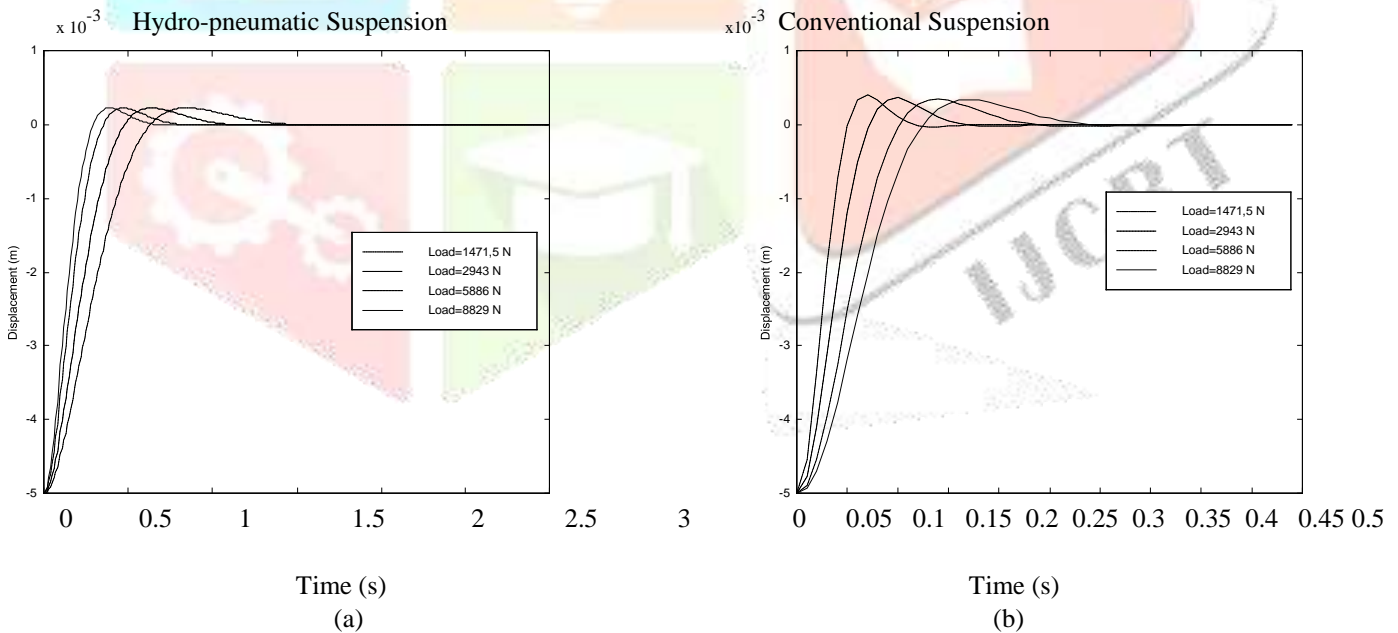

Figure 5.2. Response for initial displacement for the second case

Table 5.3 Parameters used in the third case.

m_s [N]	k_s [N/m]		c_s [Ns/m]		Ξ
	k_s(-0,05)	k_s(0,05)	c_s(-0,05)	c_s(0,05)	

8829	71900	351160	11261	24888	0.70
5886	37536	97971	6643	10733	0.70
2943	11762	18801	2629	3324	0.70
1471	3644	4661	1035	1170	0.70

Only the semi-active control can be avoid high damping, when the vehicle is empty, or damping exaggeratedly small when the vehicle is completely charged, in cases where the rate between the vehicle load capacity and the vehicle net weight is very high.

6. Analysis and Result

We need to some amount of pressure to activate a hydraulic actuator. The flow velocity of the fluid from the hydraulic cylinder gives some pressure in the outlet of the hydraulic actuator. So we analyse the outlet pressure with the help of inlet fluid velocity. We used the ANSYS CFD Analysis method to find the outlet pressure occurred as per the inlet velocity given. Fig (6.1) represents the analysis diagram of inlet velocity during the fluid flow (initial velocity 50 m/s). And the. Fig (6.2) represents the analysis diagram of outlet pressure occurred In the three cases above, the results suggest that the gain to be obtained with the use of a control of damping.
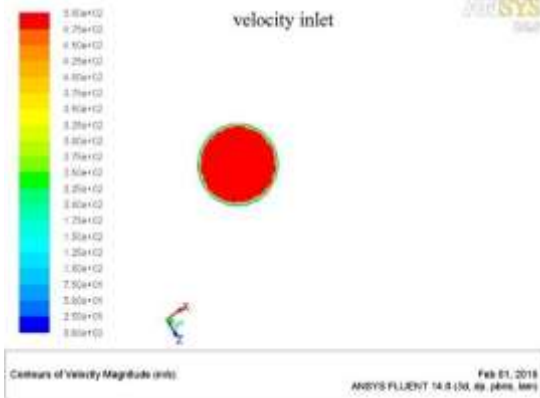

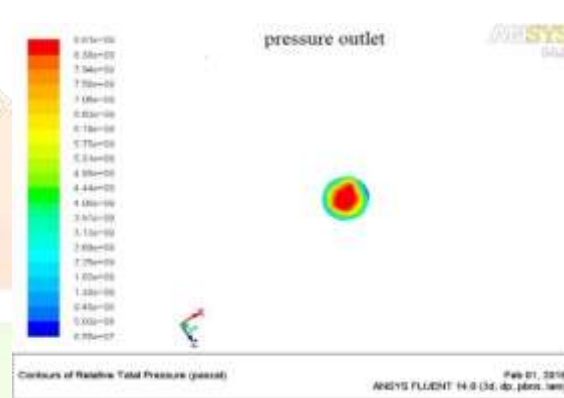Figure 6.1 Analysis of inlet velocity   Figure 6.2 Analysis of outlet pressure

7. Conclusions

This work discusses with the characteristics of hydro-pneumatic suspensions. A model for the nonlinear stiffness is presented, and also a procedure for the setup and design of the chambers and critical components. A study of the damping behaviour is developed, and a experimental verification of the predicted behaviour is presented. Some experiments and complementary studies are being done currently.

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