Experimental Measurement of Hydro-Pneumatic Semi-Active Damper Characteristics and Studying Its Effects on Vehicle Ride Comfort

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Abstract: The hydro-pneumatic semi-active suspension system consists of a damper with variable damping coefficient and parallel coil spring. This paper is concerned with the studying the performance of hydro-pneumatic semi-active suspension system theoretically predicted using two degree of freedom of quarter car model. The theoretical results are generated when the system is excited by random road input. The damping coefficient of the hydro-pneumatic damper is taken from the experimental to predict the results. Finally the results are given and analyzed.

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Key words: quarter car model, hydro-pneumatic suspension system.

1. Introduction

The design of ground vehicle suspension system is influenced by a number of factors. These factors, on the other hand, a good vibration isolation is required to secure the occupants' ride comfort, whereas good road holding is important for vehicle handling and, in general, enhanced safety Key design constraints are represented by maximum allowable relative displacement between the vehicle body and various suspension components, including wheels, trucks and other unsprung masses[1-3]. Additional constraints are imposed by the overall system robustness, reliability and cost requirements. Different suspensions satisfy the above requirements to differing degrees. Although significant improvements can result from a designer's ingenuity, on the average, suspension performance mainly depends on the type or class of suspension used. Here one distinguishes, in an ascending order of improved performances between passive, semi-active, and fully active suspensions. Passive suspensions can be found on most of the conventional vehicles. Roughly they are characterized by the absence of external power sources, whereas the active suspensions require additional energy sources, such as compressors or pumps, to achieve superior ride and/or handling performance[1,4,6]. Suppression of vibration in passive suspensions depends on the spring stiffness, damping Suppression of vibration in passive suspensions depends on the spring stiffness, damping coefficient, and car mass. Due to the fact that they cannot satisfy the comfort requirement under different road conditions, significant interest is being devoted to the control of active and semi- active suspension in both academia and industry. Many analytical and experimental studies on active and semi-active suspensions have been performed to improve ride quality and handling performance. The results of studies show that active and semi-active suspensions can provide substantial performance improvements over passive suspensions in general[1]. The design of controlled suspension systems for road vehicles aims to optimize the performance of the vehicle with regard to comfort and road handling. Vehicle suspensions should serve several conflicting purposes. In addition to counteracting the body forces resulting from cornering, acceleration or braking and changes in payload, suspensions must isolate the passenger compartment from road irregularities. For driving safety, a permanent contact between the tyres and the road should be assured. Passive suspension systems built of springs and dampers have serious limitations. Their parameters have to be chosen to achieve a certain level of compromise between road holding, load carrying and comfort, under wide variety of road conditions. Specific vehicle models need to be used in order to analyse the effectiveness of the active suspension system on vehicle dynamics[2,5,8]. Passenger cars are complex multibody systems consisting of many rigid and deformable components. A full vehicle model needs to present the nonlinear kinematics of wheels and axles, the effects of suspension geometry and has to include the drive train, the steering mechanism and the tyre dynamics, resulting in a high number of degrees of freedom. Since it makes no sense to try to build a universal vehicle model that can be used to solve all dynamic problems, reduced dynamic models for specific investigation purposes are often designed instead [1,8]. In this work a hydro-pneumatic semi-active suspension system is considered. This system in which the hydro-pneumatic damper is mounted in parallel with a conventional coil spring. The theoretical predicted analysis of this system is based on quarter car model subjected to random road input and realistic damping coefficient values is presented.

2. Mathematical Model

2.1 Passive suspension system model

Fig. 1 shows the two degree of freedom of quarter car vehicle model for passive suspension system which is used in this work. Passive suspension system is consists of a hydro-pneumatic damper parallel with a coil spring. The equations of motion are[10,11]:

$$M_b X_b^{\cdot\cdot} = K_s(X_w - X_b) + C_s(X_w^{\cdot} - X_b^{\cdot}) \qquad \text{Eq.1}$$

$$M_w X_w^{\cdot\cdot} = K_t(X_0 - X_w) - K_s(X_w - X_b) + C_s(X_w^{\cdot} - X_b^{\cdot}) \qquad \text{Eq.2}$$

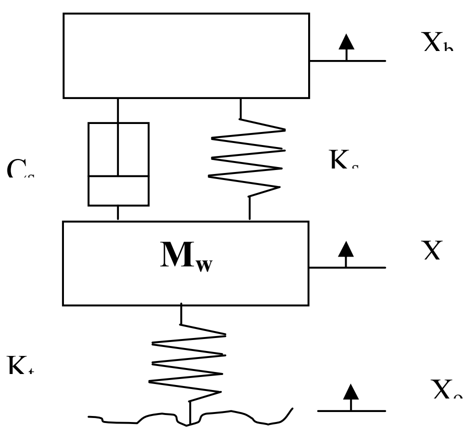
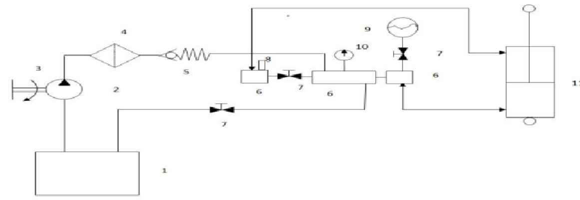


Fig.1 The quarter car vehicle model

2.2 Hydro-pneumatic semi-active damper

The major components of the hydro-pneumatic semi-active damper are a double acting cylinder with a piston diameter 32 mm designed to represent the damper but without drilling holes in the piston. The output and input of the double acting cylinder are controlled by an external manual valve in order to change the orifice area and obtain a variable damping coefficient. A rechargeable accumulator represented hydraulic components, which is a gas under pressure in a bottle contained within a diaphragm effectively a balloon. This allows pressurized fluid to compress the gas and as the pressure drops the gas push the fluid back to keep the system pressure up. The aim of the accumulator is to dissipate the energy in rebound that will change the damping coefficient. Fig. 2 Shows the layout of the hydro-pneumatic active damper[9,12].



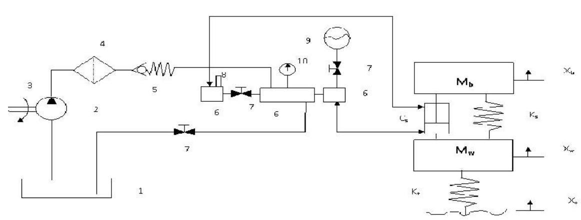


Fig.3 The main components of the hydro-pneumatic semi-active suspension system

1- tank 2- Pump 3-Motor 4-Filter 5- Non-return valve 6- block 7- Manual valve 8- Pressure Transducer 9- Gas accumulator 10 - Pressure Gauge

$$M_b X_b^{\cdot\cdot} = K_s(X_w - X_b) + C_s(X_w^{\cdot} - X_b^{\cdot}) \qquad \text{Eq.3}$$

$$M_w X_w^{\cdot\cdot} = K_t(X_0 - X_w) - K_s(X_w - X_b) + C_s(X_w^{\cdot} - X_b^{\cdot}) \qquad \text{Eq.4}$$

3. Road Profile

The representation of road surface as power spectral density are proposed as random road profile as :

$$D_r(f) = G * V^{p-1}/f^p \qquad \text{Eq.5}$$

This formula, was proposed by Robson [8]. The parameters chosen for this work are vehicle speed 20 m/s, n wave number (cycle /m), road roughness coefficient G=$1.0*10^{-6}$, p=2.5.

The vehicle parameters used in this work are given in Table1.

Table. 1 Suspension system components parameter.

Parameters	Value
Sprung mass, Kg	250
Unsprung mass, Kg	50
Tyre spring stiffness, kN/m	126
Coil spring stiffness, kN/m	16.72

4. The characteristics of the hydro-pneumatic semi-active damper

The hydro-pneumatic semi-active damper characteristics were measured at a test rig is designed and developed in the vehicle dynamic laboratory. It is used to measure the non linear damper characteristics. The characteristics were measured at different cases, after that the force velocity curve was plotted and the characteristics of the hydro-pneumatic semi-active damper were extracted [12]. The cases are showed below in Figs.(4-7) and Tables(2-5).

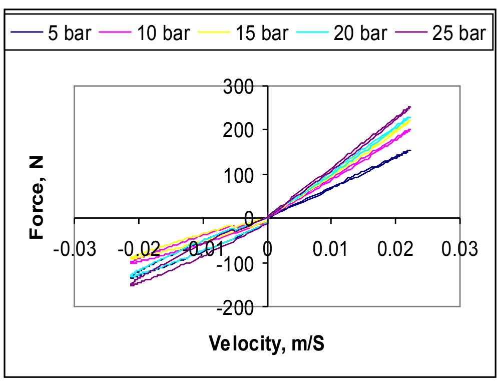


Fig. 4 The effect of oil pressure changing at 0.4Hz

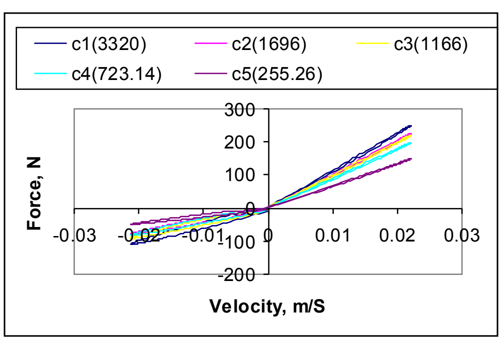


Fig. 5 The effect of changing orifice area at 0.4Hz

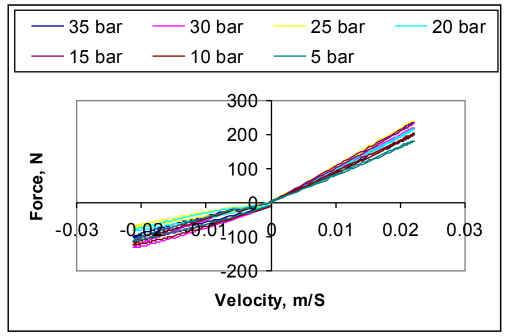


Fig. 6 The effect of changing accumulator gas pressure at 0.4Hz

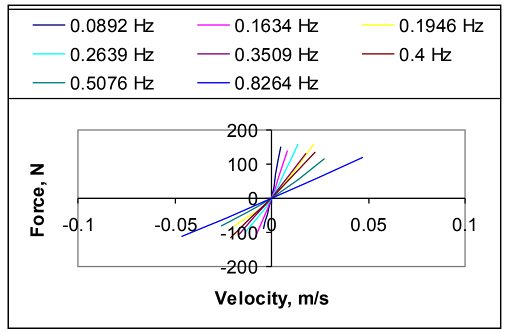


Fig.7 Shows the effect of changing velocity frequency from (0.0892 to 0.8964Hz)

Table 2 The effect of oil pressure on the damping coefficient

Oil pressure(bar)	Damping coefficient(N.s/m)
5	696.
10	905.
15	996.
20	1040.
25	1134.

Table 3 The effect of speed on the damping coefficient

Speed (m/sec)	Damping coefficient(N.s/m)
0.451724117	3320.
0.819387041	1696
1.352673	1166
2.165407647	723
4.699134217	255

Table 4 The effect of orifice area on the damping coefficient

Orifice area	Damping coefficient(N.s/m)
1	1131
2	1018
3	966
4	896
5	672

Table 5 The effect of oil pressure on the damping coefficient

Accumulator pressure(bar)	Damping coefficient(N.s/m)
5	824.
15	1065.
20	977.
25	1085.
30	996.
35	905.

5. Results and Discussion

In this work, the results of hydro-pneumatic semi-active suspension systems are presented in terms of power spectral densities and root mean square values. The road input used is described. The quarter vehicle model is used, and the vehicle parameters are shown in Table 1. The damping coefficients of the hydro-pneumatic semi-active damper obtained from the experimental tests were used in the mathematical model to show their effects on vehicle ride comfort.

5.1. The effect of changing the hydraulic oil pressure

Calculations were carried out with five different values of hydraulic oil pressure. Figures 8 and 12 show the influence of hydraulic oil pressure on body acceleration, suspension working space and dynamic tyre load in terms of power spectral density it can be seen that as the damping coefficient increases the acceleration decreases till 15 bar after the acceleration increases. Dynamic tyre load decrease with higher damping coefficient (higher pressures) around body and wheel resonance peaks. It can be seen that the best ride comfort is obtained at pressure of 25 bar[4,10 and 11].

5.2. The effect of frequency

Calculations were carried out with five different values of frequency. Figures 9 and13 show the effect of frequency on body acceleration, suspension working space and dynamic tyre load in terms of power spectral density it can be seen that as the damping coefficient decreases the acceleration decreases till 1166 N.s/m after that the acceleration increases. Dynamic tyre load decrease as the damping coefficient increases around body and wheel resonance peaks. It can be seen that the best ride comfort is obtained at damping coefficient of 1166 N.s/m[4,10].

5.3 The effect of changing orifice area

Results were carried out with five different orifices area. Figures 10 and 14 show the effect of changing orifice area on body acceleration, suspension working space and dynamic tyre load in terms of power spectral density it can be seen that as the damping coefficient decreases the acceleration decreases till 1018 N.s/m after that the acceleration increases. Dynamic tyre load increases as the damping coefficient decreases around body and wheel resonance peaks. It can be seen that the best ride comfort is obtained at damping coefficient of 1018 N.s/m [4,10 and 11].

5.4. The effect of accumulator gas pressure

Results were carried out with four different accumulator gas pressure. Figures 11 and 15 shows the effect of changing accumulator gas pressure on body acceleration, suspension working space and dynamic tyre load in terms of power spectral density it can be seen that the best damping coefficient is 977 N.s/m (25 bar). Dynamic tyre load increases as the damping coefficient decreases around body and wheel resonance peaks. It can be seen that the best ride comfort is obtained at pressure of 977 N.s/m[9].

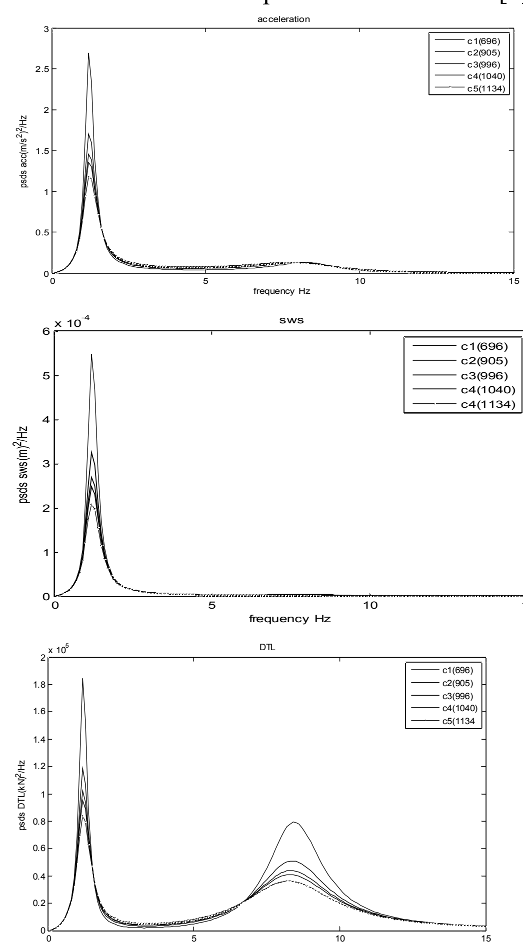


Fig.8 The effect of changing hydraulic oil pressure on the power spectral densities of body acceleration, suspension working space and dynamic tyre load

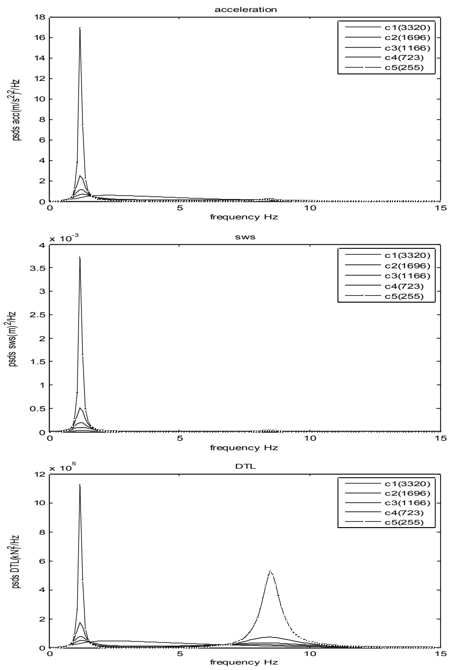


Fig.9 The effect of changing speed on the power spectral densities of body acceleration, suspension working space and dynamic tyre load.

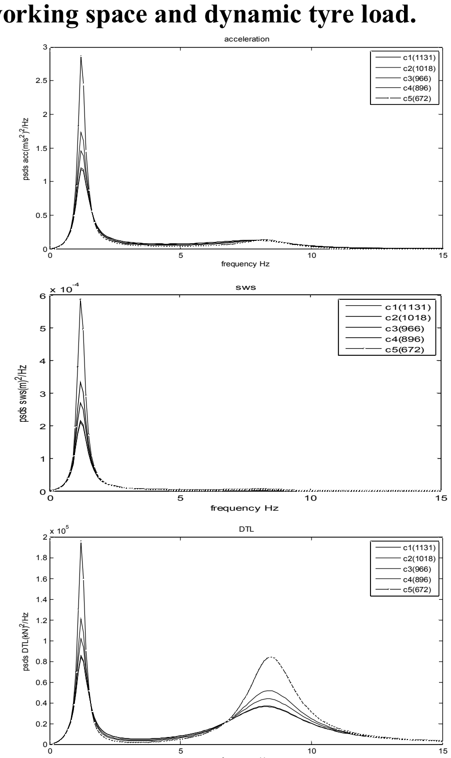


Fig.10 The effect of changing orifice on the power spectral densities of body acceleration, suspension working space and dynamic tyre load

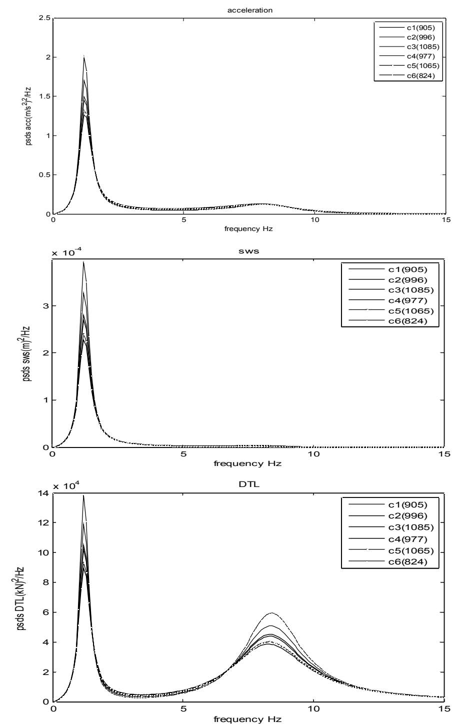


Fig.11 The effect of changing accumulator gas pressure on the power spectral densities of body acceleration, suspension working space and dynamic tyre load

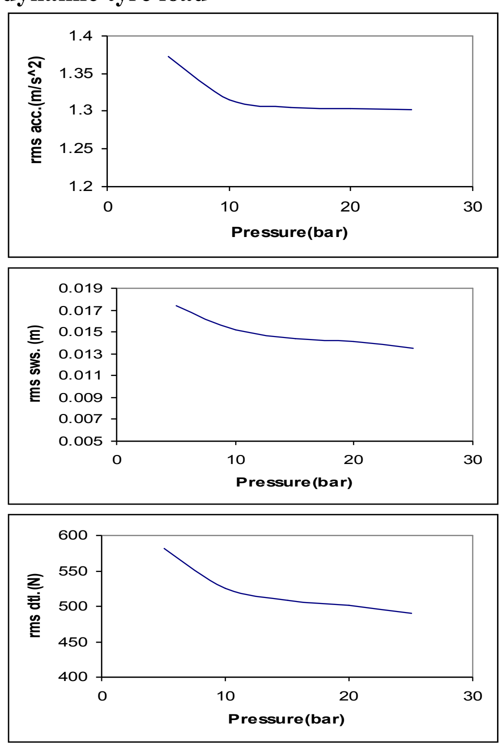


Fig. 12 The effect of changing oil pressure in root mean square values of body acceleration, suspension working space and dynamic tyre load

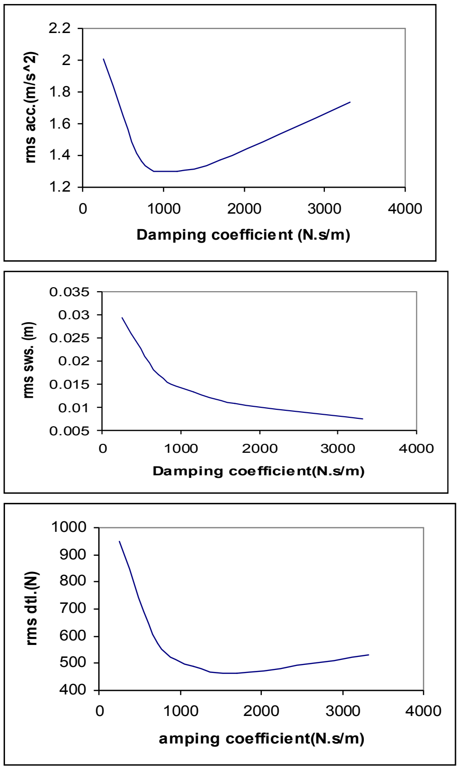


Fig. 13 The effect of changing speed in root mean square values of body acceleration, suspension working space and dynamic tyre load

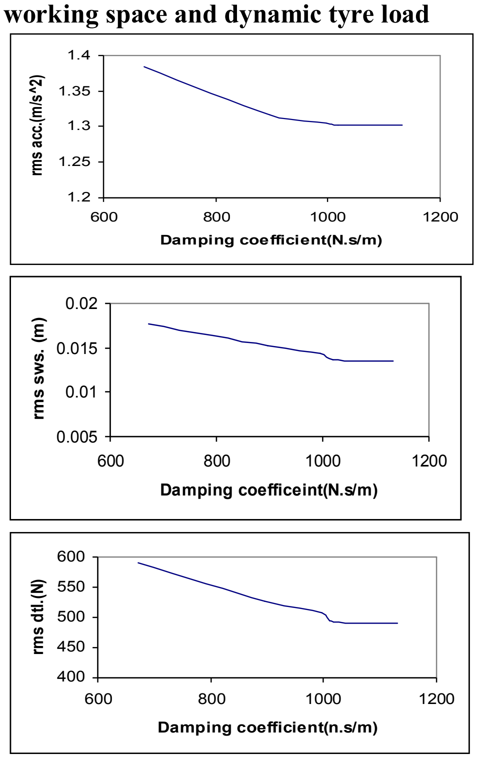


Fig. 14 The effect of changing orifice area in root mean square values of body acceleration, suspension working space and dynamic tyre load.

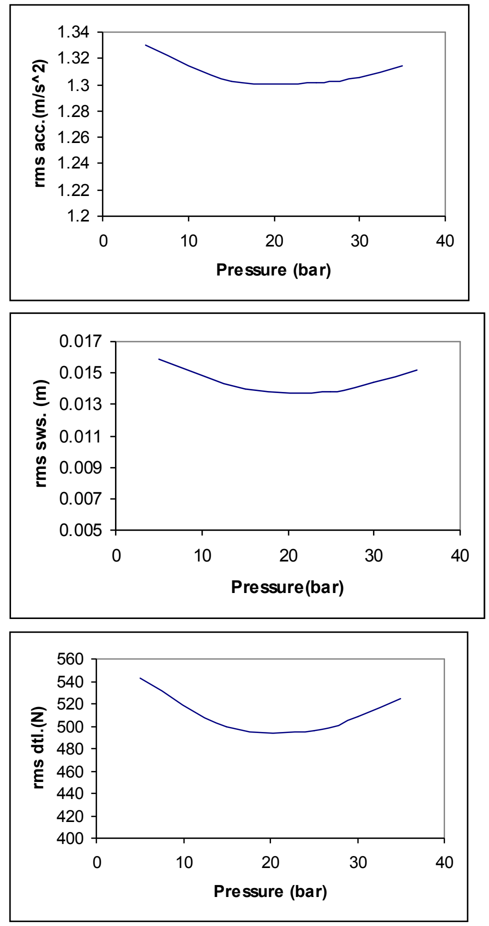


Fig. 15 The effect of changing accumulator gas pressure in root mean square values of body acceleration, suspension working space and dynamic tyre load.

6. Conclusions

1- In this paper, the performances of the hydro-pneumatic semi-active suspension system are studied theoretically. The results, on random road surface and hydro-pneumatic damping coefficient from experimental. Quarter car model was used to calculate the performances.

2- The results showed that the best ride comfort was at the hydraulic oil pressure of 25 bar and the second orifice area also at the accumulator gas pressure of 25 bar. So the best design of the damper is at the hydraulic oil pressure of 25 bar and the second orifice area and the accumulator gas pressure of 25 bar.

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