An Effective Evaluation of Hydro-Pneumatic Suspension Strut for an Agricultural Truck in Terms of Vehicle Ride Comfort

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Abstract - *Purpose of this study is to simulate and analyze the effectiveness of hydro-pneumatic suspension (HPS) strut on vehicle ride comfort. To achieve the goals, a quarter-vehicle dynamic model of an agricultural truck under random excitation of road surface is set up to simulate and analyze their performance to vehicle ride comfort. The ride performance of hydro-pneumatic suspension strut compared with leaf spring suspension system (LSSs) is respectively analyzed based on the root-mean-square (RMS) acceleration of the vehicle body according to the international standard ISO 2631-1 (1997). The obtained results indicate that the ride comfort performance of hydro-pneumatic suspension strut is better than leaf spring suspension system. The study results are the theoretical basis for the design and control of the hydro-pneumatic suspension strut.*

Key Words: Agricultural truck, hydro-pneumatic suspension strut, leaf spring suspension, dynamic model, ride comfort.

1. INTRODUCTION

Hydro-pneumatic suspension systems (HPS) have outstanding advantages in comparison with other suspension systems due to nonlinear characteristics of gas and liquid, so HPS are widely used on vehicles, especially on heavy truck and bus. Research on improving the performance of HPS has been studied by many scientists. The basic parameters of the HPS including the area of piston, orifice and check valve were selected through design sensitivity analyses and optimisation, considering ride vibration and roll and yaw-plane stability performance measures to improve performance of the HPS [1]. Lin Yang et al have studied and compared the performance of a passive inerter-based on HPS and semi-active HPS based on skyhook control. The results of this research show that the performance of the passive inerter-based HPS was as good as that of the semi-active HPS based on skyhook control [2]. Magdy N.Awad et al have researched and compared the Hydro-Pneumatic Regenerative Suspension System in comparison with other HPS such as HPS with single and double acting cylinder. This work shows that the performance of the Hydro-Pneumatic Regenerative Suspension System was better than the HPS with single and double acting cylinder [3]. The systematic objective function was used to optimize the parameters of HPS and the Lugre friction model in combination with the HPS model was applied to improve accuracy of the HPS model [4]. The traditional HPS with extra an accumulator through by a controllable throttle valve was studied and the appropriate selection of the HPS model parameters were optimized to improve the performance of the HPS [5]. The nonlinear characteristics of the HPS was considered to improve the precise determination the height of the HPS in the process control and the height of the HPS was controlled by adjusted by a fuzzy controller to improve the ride comfort of vehicle [6-7]. The air suspension systems for the trucks are analyzed and evaluated through the ride comfort indicators in the references [11-17]. Bian Gong et all have proposed the VRPEI-Kriging meta-model to determine the optimal parameters of the HPS and the study results shown that the ride comfort of the multi-axle heavy truck was improved based on VRPEI-Kriging meta-model [8].

The main objective of this paper is to analyze and compare with the performance of hydro-pneumatic suspension strut and leaf spring suspension system in the direction of vehicle ride comfort. Firstly, a 2-DOFs quarter-vehicle dynamic model is set up under random excitation of road surface, and then the vertical forces of both two suspensions are determined which connected to the vehicle dynamics model to analyze the performance of two suspensions.

2. QUARTER -VEHICLE DYNAMIC MODEL

A 2-DOFs quarter-vehicle dynamic model with two suspensions for an agricultural is established based on the reference, as shown in Fig-1, where, m_a, and m_b are vehicle axle and vehicle body masses, respectively; k_t and c_t are the stiffness and damping coefficients of tire, respectively; k and c are the stiffness and damping coefficients of the leaf spring suspension system (LSSs), respectively; k_h and c_h are the stiffness and damping coefficients of the hydro-pneumatic suspension (HPS) strut, respectively; z_a and z_b are the vertical displacements of vehicle axle and vehicle body masses, respectively; q is the excitation of road surface roughness.

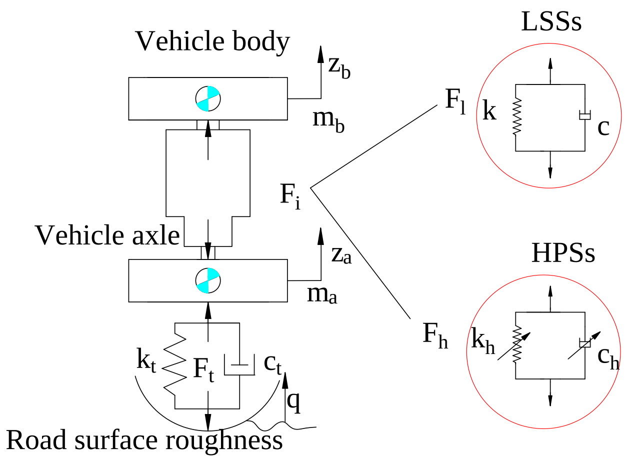

Fig-1: Quarter-vehicle dynamic model

The equations of motion: From the quarter vehicle dynamic model with LSSs in Fig-1, the motion equations of vehicle axle and vehicle body masses using Newton's second law are written as follows

$$m_a\ddot{z}_a = k\left(z_a - z_b\right) + c\left(\dot{z}_a - \dot{z}_b\right) - k_t\left(z_a - q\right) - c_t\left(\dot{z}_a - \dot{q}\right) \tag{1}$$

$$m_b\ddot{z}_b = -k\left(z_a - z_b\right) - c\left(\dot{z}_a - \dot{z}_b\right) \tag{2}$$

From the quarter vehicle dynamic model with HPSs in Fig-1, the motion equations of vehicle axle and vehicle body masses using Newton's second law are written as follows

$$m_a\ddot{z}_a = k_h\left(z_a - z_b\right) + c_h\left(\dot{z}_a - \dot{z}_b\right) - k_t\left(z_a - q\right) - c_t\left(\dot{z}_a - \dot{q}\right) \tag{3}$$

$$m_b\ddot{z}_b = -k_h\left(z_a - z_b\right) - c_h\left(\dot{z}_a - \dot{z}_b\right) \tag{4}$$

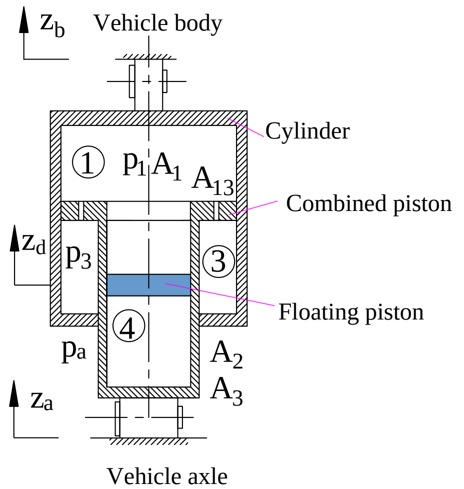

Fig.2: Structural schematic of the hydro-pneumatic cylinder

The vertical dynamic force of HPSs: From Fig-2 with HPSs, the vertical dynamic force is determined based on the laws of thermodynamics method

$$F_h = p_1A_1 - p_2\left(A_1 - A_2\right) - \left(p_0 - \frac{m_d g}{A_3}\right)A_2 \tag{5}$$

Where A_1, A_2, and A_3 are the area of cylinder, rod and floating piston, p_1 and p_2 are pressure in the oil chamber (1) and oil chamber (2).

The pressure in air chamber is defined following the laws of thermodynamics:

$$p_0V_0^n = pV^n \tag{6}$$

Where p_0 and V_0 are the initial absolute pressure and volume in air chamber, p and V are the absolute pressure and volume in air chamber, n is the polytrophic rate ($1 < n < 1.4$).

$$p = p_0\left(\frac{V_0}{V}\right)^n, \quad V = V_0 + A_3(z_b - z_a) \tag{7}$$

The flow rate through the orifice is calculated by:

$$Q = C_dA_{13}\sqrt{2\frac{(p_2 - p_1)}{\rho}sign(\dot{z}_b - \dot{z}_a)} \tag{8}$$

In which C_d is coefficient of discharge, A_{13} is the area of the orifice, cylinder and floating piston and ρ is the density of oil.

On the basic of volume balance, the flow rate is inferred:

$$Q = (A_1 - A_2)(\dot{z}_b - \dot{z}_a) \tag{9}$$

From Eq. (8) and Eq. (9), relationship p_1 and p_2 is determined:

$$p_2 = p_1 + \frac{\rho(A_1 - A_2)^2\left|\dot{z}_b - \dot{z}_a\right|(\dot{z}_b - \dot{z}_a)}{2C_d^2A_{13}^2} \tag{10}$$

The differential motion equation of floating piston can be defined:

$$(p_1 - p)A_3 + m_d.g = m_d.\ddot{z}_d \tag{11}$$

The equation (9) can be rewritten:

$$p_1 = p + \frac{m_d.(\ddot{z}_d - g)}{A_3} \tag{12}$$

From Eq. (5), Eq.(10) and Eq.(11), the vertical dynamic force is infered:

$$F_h = -\frac{\rho(A_1 - A_2)^3\left|\dot{z}_b - \dot{z}_a\right|(\dot{z}_b - \dot{z}_a)}{2C_d^2A_{13}^2} + p_0A_2\left(\left(\frac{V_0}{V_0 + A_3(z_b - z_a)}\right)^n - 1\right) - m_d\ddot{z}_d\frac{A_2}{A_3} \tag{13}$$

Road surface excitation [18], [19]: The random road surface roughness of random white noise is selected as excitation source waveform for vehicle suspension, the random road profile is produced by filtering the white noise using the following mathematical model of the road roughness

$$\dot{q}(t) + 2\pi f_0q(t) = 2\pi n_0\sqrt{G_q\left(n_0\right)v(t)}w(t) \tag{14}$$

where, $G_q(n_0)$ is the road roughness coefficient which is defined for typical road classes from A to F according to ISO 8068(1995) [18], n_0 is a reference spatial frequency which is equal to 0.1 m; v(t) is the speed of vehicle; f_0 is a minimal boundary frequency with a value of 0.0628 Hz; n_0 is a reference spatial frequency which is equal to 0.1 m; w(t) is a white noise signal.

3. RESULTS AND DISCUSSION

The differential equations of motion are solved by using MATLAB/Simulink software with a set of parameters of an agricultural truck by the reference [20] when vehicle moves on the ISO class D road surface at v=40km/h with fully loaded. The comparing results of the time acceleration response of the vertical vehicle body are shown in Fig-3. From Fig-3 we see that the peak amplitude values of the time acceleration response of the vertical vehicle body with HPSs decrease in comparison LSSs which leads to enhance vehicle rider comfort with HPSs.

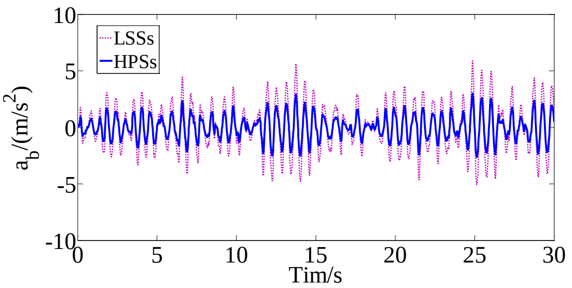

Fig-3: Time acceleration response of the vertical vehicle body with two suspension system.

From the results of Fig- 3, we could determine the value of the root-mean-square (RMS) acceleration response of the vertical vehicle body (a_{wb}) through Eq.(15) as 2.1033 m/s^2 with LSSs and 1.0926 m/s^2 with HPSs. The results show that a_{wb} value with HPSs reduce by 92.5% in comparison LSSs. Vehicle ride comfort with HPSs has been greatly improved in comparison LSSs.

Vehicle ride comfort indicator is based on ISO 2631-1 (1997) [15], vibration evaluation based on the basic evaluation method including measurements of the root-mean-square (RMS) acceleration is defined by:

$$a_w = \left[\frac{1}{T}\int_0^T a^2(t)\,\mathrm{d}t\right]^{\frac{1}{2}} \qquad (15)$$

where a(t) is the weighted acceleration (translational and rotational) as a function of time, m/s^2; T is the duration of the measurements.

The a_{wb} values with HPSs are compared to LSSs when vehicle moves on the different road surface conditions from ISO class B road surface to ISO class C road surface at vehicle speed of 40 km/h and full load, as shown in Fig-4. From the results of Fig- 4, it indicates that the a_{wb} values with HPSs reduce in comparison LSSs when vehicle moves on from ISO class B road surface to ISO class C road surface. Vehicle ride comfort with HPSs has been greatly improved in comparison LSSs under the different road surface conditions

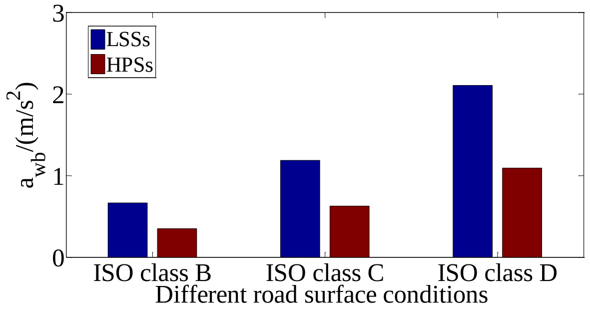

Fig-4: a_{wb} under different road surface conditions

The a_{wb} values with HPSs are compared to LSSs when vehicle moves on ISO class D road surface at the different vehicle speed conditions from 20km/h to 40km/h and full load, as shown in Fig-5. From the results of Fig- 5, it indicates that the a_{wb} values with HPSs respectively reduce by 0.9033 m/s^2, 0.8964 m/s^2 and 0.9250 m/s^2 in comparison LSSs vehicle moves at vehicle from 20 km/h to 40 km/h. Vehicle ride comfort with HPSs has been greatly improved in comparison LSSs at the different vehicle speed conditions

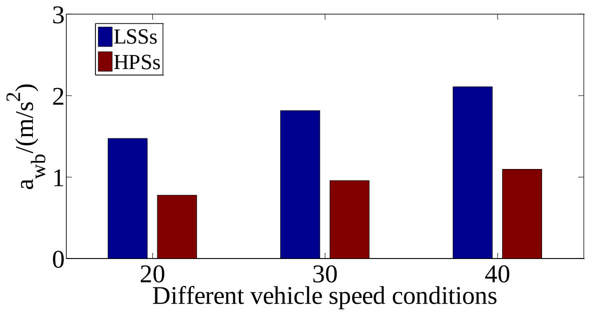

Fig-5: a_{wb} under different vehicle speed conditions

4. CONCLUSION

In this study, a 2-DOFs quarter-vehicle dynamic model was set up under random excitation of road surface, and then the vertical forces of both two suspensions are determined which connected to the vehicle dynamics model to analyze the performance of two suspensions such as HPSs and LSSs. The study results show that a_{wb} value with HPSs reduce by 92.5% in comparison LSSs. Vehicle ride comfort with HPSs has been greatly improved in comparison LSSs. Finally, the ride performance of two suspensions is tested under different operating conditions. The results indicate that vehicle ride comfort with HPSs has been greatly improved in comparison LSSs under the different operating conditions.

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