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Advance the stability of the vehicle by using the pneumatic suspension system integrated with the hydraulic actuator

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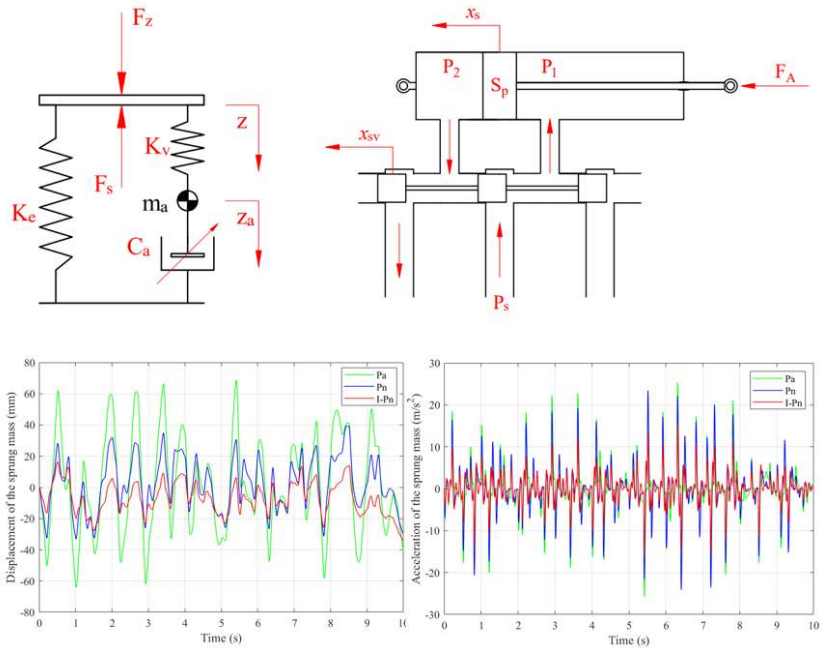
Abstract

The stability and comfort of vehicles depend on the operation of the suspension system. To increase the smoothness and comfort for passengers in the vehicle, the stiffness of the suspension system needs to be changed flexibly. The conventional pneumatic suspension system can partially meet these requirements. However, the change is not much. This paper introduces a model of the pneumatic suspension system integrated with a hydraulic actuator. This is a completely novel and unique method. In the case that the excitation is random, average values of displacement and acceleration of the sprung mass are only 12.56 (mm) and 3.78 (m/s^2) if the vehicle uses the integrated pneumatic suspension system. In contrast, this value is very large, up to 30.70 (mm) and 6.53 (m/s^2) if the passive suspension system is used. Similarly, this change is also very large in the remaining survey situations. Overall, the values of acceleration and displacement of the sprung mass are significantly reduced when the vehicle is equipped with an integrated pneumatic suspension system. The results of the research showed the advantages of this method compared to other methods.

Keywords

Pneumatic suspension system, hydraulic actuator, integrated suspension, comfort, stability

Graphical Abstract



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1 INTRODUCTION

The vibration of the vehicle when moving on the road is one of the extremely important issues. Oscillation usually occurs when there is an external stimulus. To quell these oscillations, the suspension system is fitted on all models today. The syspension system separates the vehicle into two completely separate parts, including the sprung mass and the unsprung mass. All the components above the suspension system (passengers, cargoes, etc.) are called the sprung mass. It takes up most of the vehicle's total mass. In contrast, the components below the suspension system (wheels, axles, brakes, etc.) are assumed to be unsprung mass [1]. The optimal division of these two values is extremely important, it helps to control the oscillations properly.

Normally, the suspension system consists of three parts: spring, damper, and lever arm. For the conventional passive suspension system, the stiffness of the spring and damper is constant. As a result, the stability and comfort of the vehicle can be greatly affected. In order to improve this problem, it is necessary to change the characteristics of the spring and damper, or both. The method of using the semi-active suspension system (changing the damping stiffness) has been introduced and equipped on some models [2, 3]. However, this method can only help the vehicle to partially extinguish the vertical vibration. It is not able to guarantee stability in other cases. The improvement of the suspension system's smoothness through changing the stiffness of the spring is appreciated. The conventional metal spring will be replaced by the pneumatic spring, which can control stiffness through internal pneumatic pressure [4]. The pneumatic suspension system is often equipped on high-end vehicles or large passenger vehicles, its price is very expensive.

The pneumatic suspension system model has been studied in recent years. Characteristics of the pneumatic suspension system were introduced and analyzed in [5]. According to Yin et al., the pneumatic suspension system has a direct effect on the stiffness and height of the vehicle [6]. These problems are again exemplified by Eskandary et al. based on their paper [7]. Various models of the pneumatic suspension system have been used such as the Nishimura model, the VAMPIRE model, the SIMPAC model, and the GENSYS model. In which, GENSYS model is the novel model and it is appreciated [8]. In [9], Gavriloski et al. used this model for their paper. Besides, Moheyeldein et al. also evaluated the effectiveness of the GENSYS model in their research [10]. In addition, a nonlinear dynamic model of pneumatic spring with damper has also been introduced by Zhu et al. [11]. Similarly, the pneumatic suspension model used novel damper has also been proposed by Xiao et al. They have come up with an optimal control algorithm PID for the system [12].

The pneumatic suspension system is not only used in passenger vehicles, but it is also widely used on special vehicles. According to [13], the pneumatic suspension system has been mounted in the hub of the electric vehicle, which uses the motor in-wheel. Therefore, the structure of the vehicle is very compact. However, it still meets the requirements of the vehicle’s stability and safety. On agricultural vehicles, the pneumatic suspension system has also been equipped, which achieves very high performance [14]. In addition, the pneumatic suspension system can be integrated with the energy harvesting system that is found on some special models. It can provide optimum efficiency in terms of energy when the vehicle oscillates [15]. Further, the hydropneumatic suspension system has been equipped on many specialized vehicles such as trains, tanks, soil compactor, etc. to improve the vibration efficiency of the vehicle [16-19]. Overall, this system can meet the vehicle's stability requirements well.

When the vehicle is equipped with a pneumatic suspension system, the vehicle's vibrations are better controlled than with a conventional passive suspension system [20]. This has also been demonstrated experimentally by Kumbhar et al. [21]. The pneumatic suspension system can also be supposed as a form of the active suspension system. Therefore, control methods for the active suspension system are proposed to be used for the pneumatic suspension system. If the system is considered to be linear, PID and LQR control methods can be applied. In [22], Anh compared the effectiveness of these two methods. When the LQR controller is combined with the Gaussian filter, it becomes the LQG controller [23]. Besides, many nonlinear and intelligent control methods for the suspension system have also been proposed. In [24], Zhao et al. introduced a robust control method for the pneumatic suspension system. This method helps the height of the vehicle body to be controlled stably, the input parameters can be changed continuously. Compared with conventional linear control methods, this method helps the vehicle oscillation approach the desired threshold. In addition, robust control methods for the active suspension system can also be applied to pneumatic suspension systems [25, 26]. In [27], Nieto et al. controlled the active suspension system by the adaptive control method. This method was again used in the study of Fu et al. According to [28], the displacement and the acceleration of the sprung mass are almost unchanged even though the sprung mass changes continuously. Besides, intelligent and integrated control methods have also been proposed. In [29], Rui introduced a nonlinear adaptive sliding mode controller for a pneumatic suspension system. This is a combination of two nonlinear control methods. The tracking nonlinear controller was also used in the paper of Zhao et al. [30]. The predictive control model for the pneumatic suspension system has also been used, the effect of which is extremely positive [31]. The sliding mode control method has also been

shown in the studies of Chen et al. and Zhou et al. [32, 33]. Besides, other control methods for the active suspension system that have been highly effective can also be applied to the pneumatic suspension system [34-39].

The pneumatic suspension system helps to improve the vehicle's stability and safety when moving on the road. However, this improvement is not much. In many cases, the stability and comfort of the vehicle are still not guaranteed. Therefore, this paper has proposed the use of the pneumatic suspension system, which is integrated with a hydraulic actuator to improve comfort and smoothness. The actuator used in this research is a hydraulic piston, which is widely used on mechatronics systems. According to [40], the hydraulic piston operates based on the opening and closing of the valve of the servo valve system. The fluid in the system is conveyed by the hydraulic pump. In [41], Lee et al. introduced a new method to control the operation of the hydraulic pump. The opening and closing of the actuator's valves are controlled based on the voltage signal that is sent from the controller. Shafie et al. developed an algorithm to control the hydraulic actuator, they used two PID controllers for each hydraulic piston [42]. According to [43], the efficiency of the actuator can be improved by using the sliding mode controller. Similarly, this method has also been mentioned in the paper of Sam and Osman [44]. In addition, integrated controllers have also been introduced to control the operation of hydraulic actuators [45]. In general, the above control methods bring positive effects to the system.

Previous studies on the pneumatic suspension system often only mentioned the use of the independent pneumatic suspension system. This does not solve the vehicle's vibration problems. Therefore, this paper proposes the use of the pneumatic suspension system that integrates a hydraulic actuator to improve the stability and comfort of the vehicle when moving on the road. The combination of the pneumatic spring and the hydraulic actuator is a novel and unique solution. They can operate independently from each other. Besides, they can also support each other. It enhances the vehicle's safety and comfort more than other conventional solutions. This research analyzes, simulates, and evaluates the efficiency of the integrated pneumatic suspension system. Simulation is performed in the Matlab-Simulink environment. The method and control model are detailed in the content below.

2 MATERIAL AND METHOD

2.1 Pneumatic Suspension Model

The physical model of the pneumatic spring is given as shown in Figure 1. Equation (1) shows the balance of the force acting on the pneumatic spring.

$$F_z = A_e p_b = A_e \left(p_0 - p_{at} \right) \tag{1}$$

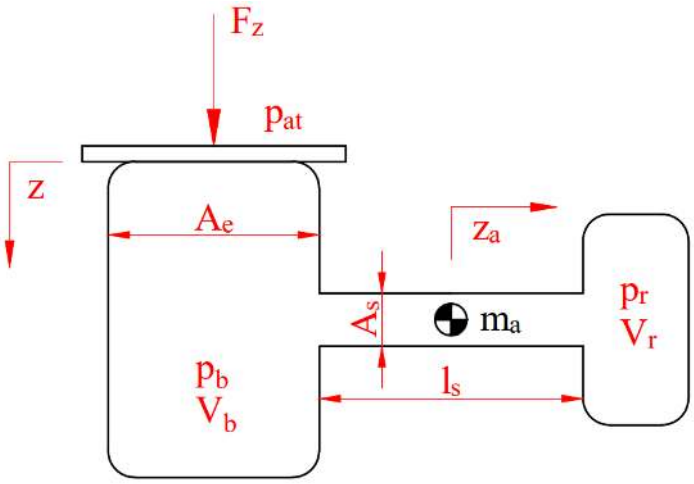

Figure 1 The physical model of the pneumatic spring.

When the spring is subjected to a static load F_z, the pressure between the two chambers changes. Therefore, the force F_p appears, which pushes the gas's mass to move along the pipe. As this mass moves, it generates a frictional force F_f, which is a nonlinear frictional force. This shift is described through equation (2):

$$m_a \ddot{z}_a = F_p - F_f = \left(p_b - p_r \right) A_s - C_a \dot{z}_a^2 \tag{2}$$

Based on this principle, many dynamic models of pneumatic springs have been proposed. In this paper, the GENSYS model is used. This is a novel model, and it has high accuracy and is suitable for the vibration simulation problems of the vehicle [8].

The pneumatic spring is converted equivalently to the oscillating system in Figure 2. This system consists of the main spring K_e, the auxiliary spring K_v, these are two linear springs. The compressed air in the spring is converted to mass ma. As air moves through the pipes, they create friction. This friction process is shown through C_a nonlinear damping.

The stiffness of the main spring K_e depends on the initial absolute pressure p_0, the effective area of the spring A_e, the initial volumes of the balloon and reservoir V_{b0} and V_{r0}, and multivariable coefficient n.

$$K_e = \frac{p_0 n A_e^2}{V_{b0} + V_{r0}} \tag{3}$$

The stiffness of the auxiliary spring K_v is proportional to the stiffness of the main spring K_e through the ratio between the initial volumes.

$$K_v = \frac{p_0 n A_e^2}{V_{r0} + V_{b0}} \frac{V_{r0}}{V_{b0}} = K_e \frac{V_{r0}}{V_{b0}} \tag{4}$$

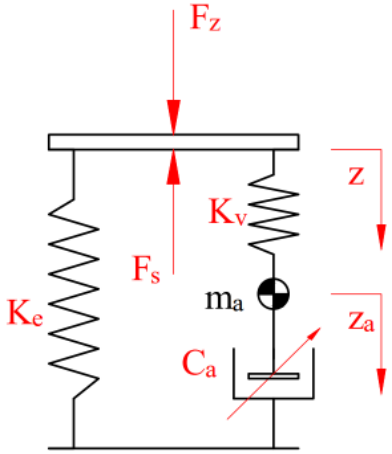

Figure 2 GENSYS model.

To simplify the model, the compressed air is equivalently converted to mass m_a. This is a nonlinear function, and it depends on the characteristics of the air and the size of the pneumatic spring.

$$m_a = \rho A_s l_s \left(\frac{A_e}{A_s} \frac{V_{r0}}{V_{r0} + V_{b0}} \right)^2 \tag{5}$$

The damping coefficient C_a of the pneumatic spring is nonlinear. In order to calculate this value, the parameters of the size of the spring A_e, the size of the pipe A_s, the initial volume of balloon and reservoir V_{r0}, V_{b0}, the density of the air ρ, and the total loss coefficient of the connection pipes k need to be determined before.

$$C_a = \frac{1}{2} A_s \rho k \left(\frac{A_e}{A_s} \frac{V_{r0}}{V_{r0} + V_{b0}} \right)^{1+b} \tag{6}$$

After the necessary coefficients have been determined, the vehicle dynamics model can be established. In the oscillation and control problems of vehicles, the quarter dynamic model is often used (Figure 3). This model consists of the linear damper C, the pneumatic spring (K_e, K_v, C_a), and the hydraulic actuator A. Instead of the usual two degrees of

freedom, this model uses three degrees of freedom (z_1, z_2, and z_a). The vehicle's oscillation is generated from the excitation from the road surface h.

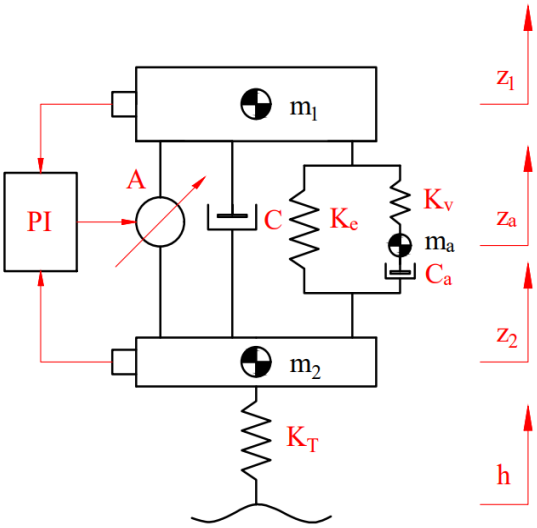

Figure 3 The quarter dynamic model.

The vertical displacement of the sprung mass m_1, the unsprung mass m_2, and the mass of the air m_a are expressed by the following equations:

$$m_1\ddot{z}_1 = F_C + F_{Ke} + F_{Kv} + F_S + F_A \tag{7}$$

$$m_2\ddot{z}_2 = F_{KT} - F_C - F_{Ke} - F_{Ca} - F_A \tag{8}$$

$$m_a\ddot{z}_a = F_{Ca} - F_{Kv} \tag{9}$$

Where:
Pneumatic force:

$$F_S = p_b A_e = p_0\left[\left(\frac{V_{b0}}{V_{b0} - A_e\left(z_2 - z_1\right)}\right)^n - 1\right]A_e \tag{10}$$

Linear damping force:

$$F_C = C\left(\dot{z}_2 - \dot{z}_1\right) \tag{11}$$

Main spring force:

$$F_{Ke} = K_e\left(z_2 - z_1\right) \tag{12}$$

Auxiliary spring force:

$$F_{Kv} = K_v\left(z_a - z_1\right) \tag{13}$$

Nonlinear damping force:

$$F_{Ca} = C_a \left(\dot{z}_2 - \dot{z}_a \right)^2 \tag{14}$$

Tire force:

$$F_{KT} = K_T \left(h - z_2 \right) \tag{15}$$

The value of force F_A that is generated from the controller needs to be calculated through the control model.

2.2 Control model

Different from other common research which only uses pneumatic springs in the suspension system, this research has integrated the hydraulic actuator with the pneumatic suspension system. When the vehicle's body oscillates, the hydraulic actuator will generate a force F_A acting on the sprung mass m_1 and the unsprung mass m_2. The actuator is automatically controlled through the previously established controller (Figure 4).

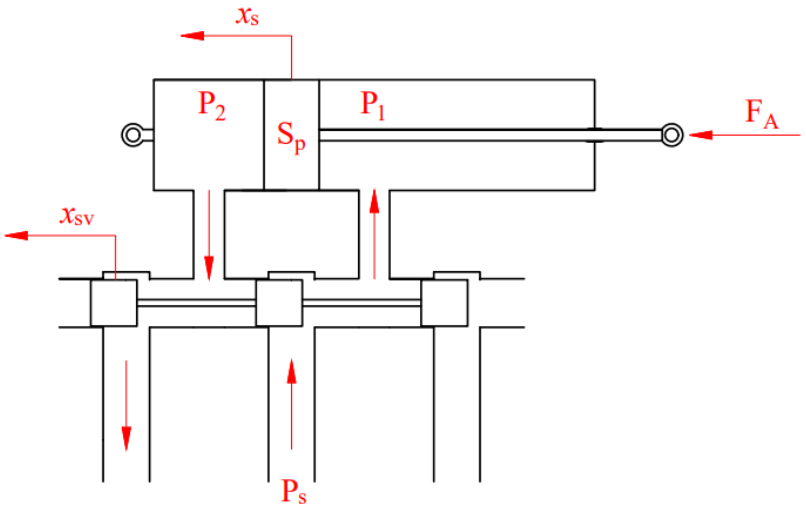

Figure 4 The hydraulic actuator.

The actuator force F_A depends on the change in hydraulic pressure inside the system:

$$F_A = S_p \Delta P \tag{16}$$

The difference in hydraulic pressure inside the actuator is expressed through the piston's displacement and liquid's flow:

$$\Delta P = \gamma_1 \int \left(Q - \frac{\gamma_2}{\gamma_1} P - S_p \dot{x}_s \right) dt \tag{17}$$

$$Q = \frac{\gamma_3}{\gamma_1} x_{sv} \sqrt{P_s - sgn\left(x_{sv} \right) \Delta P} \tag{18}$$

The flow of liquid through the valve depends on the opening and closing of the valve. The valves are opened and closed by the control voltage signal u(t), which is generated from the controller.

$$x_{sv} = \frac{1}{\tau}\int \left(k_{sv}u(t) - x_{sv}\right)dt \tag{19}$$

Where: γ: Actuator coefficient
τ: Time coefficient
S_p: Piston cross section
k_{sv}: Servo valve gain

x_{sv}: Displacement of the servo valve

This paper uses the PI controller to control the operation of the hydraulic actuator. Compared with other controllers, the PI controller has many advantages such as low cost, high reliability, easy control. Besides, this controller works stably with the SISO control object.

Although the established dynamics model has three degrees of freedom, the control object here is only the acceleration of the sprung mass. When the acceleration of the sprung mass is controlled, the value of displacement of the sprung mass can also be improved. Therefore, it is perfectly appropriate to use a PI controller.

The PID controller consists of three stages: Proportional (P), Integral (I), and Derivative (D). The mathematical model of this controller is given as (20). If the Derivative stage (D) is eliminated ($T_D = 0$), it becomes a PI controller, which consists of only two stages.

$$u(t) = k_p\left[e(t) + \frac{1}{T_I}\int_0^t e(\tau)d\tau + T_D\dot{e}(t)\right] \tag{20}$$

Where:
e(t): input signal of the controller.
u(t): output signal of the controller.
k_p: proportional coefficient.
T_I: integral coefficient.
T_D: derivative coefficient.

There are many methods used to determine the parameters of the PI controller. In this paper, the controller's parameters are determined by the second Ziegler-Nichols method [46]. The transfer function of the controller is given as (21).

$$R(s) = k_p\left(1 + \frac{1}{T_I s}\right) = \delta k_{th}\left(1 + \frac{1}{\lambda T_{th} s}\right) \tag{21}$$

3 RESULTS AND DISCUSSIONS

3.1 Simulation conditions

After the vehicle dynamics model has been established, simulation is done. The specifications of the vehicle and hydraulic actuator are given in Tables 1 and 2 respectively [47].

Table 1 The specifications of the vehicle.

Symbol	Description	Value	Unit
m_1	Sprung mass	400	kg
m_2	Unsprung mass	40	kg
C	Damper coefficient	3000	Ns/m
K_T	Tire coefficient	180000	N/m

Table 2 The specifications of the hydraulic actuator.

Symbol	Description	Value	Unit
A_e	Effective area of the pneumatic spring	0.002	m^2
A_s	Connecting pipe cross section	3×10^{-4}	m^2
l_s	Connecting pipe length	2	m
p_{at}	Atmospheric pressure	1×10^5	N/m^2
p_0	Initial absolute pneumatic spring pressure	4×10^5	N/m^2
V_{r0}	Initial volumes of the reservoir	0.004	m^3
V_{b0}	Initial volumes of the balloon	0.008	m^3
ρ	Density of the air at initial condition	1.29	kg/m^3
β	Constant	2	
k	Total loss coefficient of the connection pipes	3.5	
n	Multivariable coefficient	1.4	
γ_1	Actuator coefficient	4.5×10^{13}	N/m^5
γ_2	Actuator coefficient	1	s^{-1}
γ_3	Actuator coefficient	1.5×10^9	$N/kg^{1/2}m^{5/2}$
τ	Time coefficient	2.5×10^{-3}	s
S_p	Piston cross section	3.5×10^{-4}	m^2
P_s	Supply pressure	1056240	N/m^2
k_{sv}	Servo valve gain	1×10^{-3}	m/V

The results of the simulation process are the displacement of the sprung mass, the acceleration of the sprung mass, and the change in pressure of the pneumatic suspension. These are the parameters that characterize the vehicle's oscillation. The maximum value and changing trend of these parameters are interesting objects. Besides, the average value is also calculated to be able to determine the stable oscillation threshold of the system. The average value of the oscillation is calculated by the Root Mean Square method (RMS).

$$RMS = \sqrt{\frac{1}{n}\sum_{i=1}^{n} x_i^2} \tag{22}$$

Stimulation from the road surface is a factor that directly affects the smoothness and comfort of the vehicle. In this paper, three types of the excitation are used, including sine wave, random, and step. The values of displacement of the sprung mass and the acceleration of the sprung mass will be determined corresponding to the simulation cases, including:
+ Vehicles using passive suspension system (Pa)
+ Vehicles using conventional pneumatic suspension system (Pn)
+ Vehicles using integrated pneumatic suspension system (I-Pn)

3.2 Results

Case 1: Sine wave type

In this case, the excitation from the road surface takes the form of a periodic sine wave function. This type is often used in control problems. The graph in Figure 5 shows the change of displacement of the sprung mass over time. If the vehicle uses only the conventional passive suspension system, the value of the displacement of the sprung mass is quite large, reaching about 50.10 (mm). Its trajectory closely matches that of the excitation signal from the road surface. When the vehicle uses the pneumatic suspension system, this value may decrease. However, the change is small. If the vehicle is equipped with a pneumatic suspension system integrated with the hydraulic actuator, the displacement value of the sprung mass is significantly reduced. The maximum amplitude of vibration is only about 27.55 (mm), it is much smaller than the two cases mentioned above. According to equation (22), the mean value of oscillation in this case reaches RMS_{Pa} = 34.69 (mm), RMS_{Pn} = 34.63 (mm), and $RMS_{I\text{-}Pn}$ = 20.79 (mm), respectively.

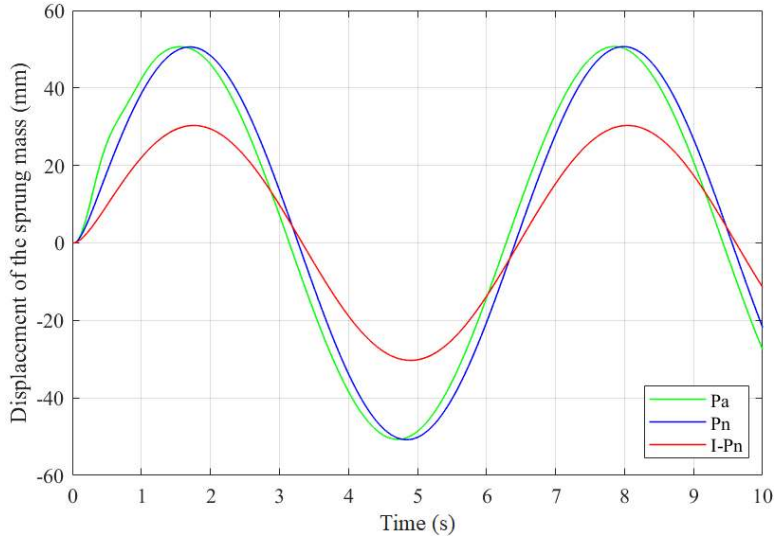

Figure 5 Displacement of the sprung mass.

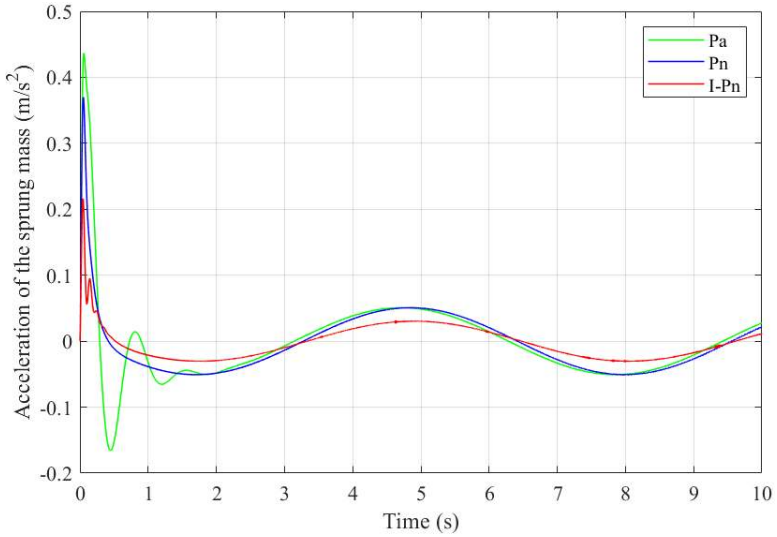

Figure 6 Acceleration of the sprung mass.

Acceleration of the sprung mass is a characteristic parameter for the smoothness and comfort of the vehicle when moving on the road. Figure 6 shows the change of this value in the case of excitation from the road surface having a sine form. If the vehicle uses only the passive suspension system, the maximum value of acceleration is quite large, reaching about 0.44 (m/s^2). This value tends to decrease gradually and fluctuates stably around the threshold of 0.05 (m/s^2). When the pneumatic suspension system is used to replace the passive suspension system, the change of this value is also not much. Therefore, the smoothness and comfort of the vehicle cannot be improved. If the pneumatic suspension system

is integrated with a hydraulic actuator, the vehicle's stability and comfort can be further improved. Its maximum value reaches 0.22 (m/s^2), only half that of the vehicle using the passive suspension system. After reaching the maximum value, its amplitude of oscillation gradually decreases. The value fluctuates steadily between -0.023 (m/s^2) to 0.023 (m/s^2). Besides, the average value of acceleration in this case is RMS_{Pa} = 0.052 (m/s^2), RMS_{Pn} = 0.043 (m/s^2), and $RMS_{I\text{-}Pn}$ = 0.025 (m/s^2), respectively. In general, if the vehicle is equipped with an integrated pneumatic suspension system, the vehicle's stability and comfort can be significantly improved.

The pressure of the pneumatic suspension system changes continuously depending on the stimulus from the road surface. This change is shown graphically in Figure 7. The change in pressure when the vehicle is equipped with an integrated pneumatic suspension system is larger than when the vehicle is used with the conventional pneumatic suspension system, this is a perfect fit. The pressure in the system can change continuously in response to the fluctuations of the stimuli from the road surface.

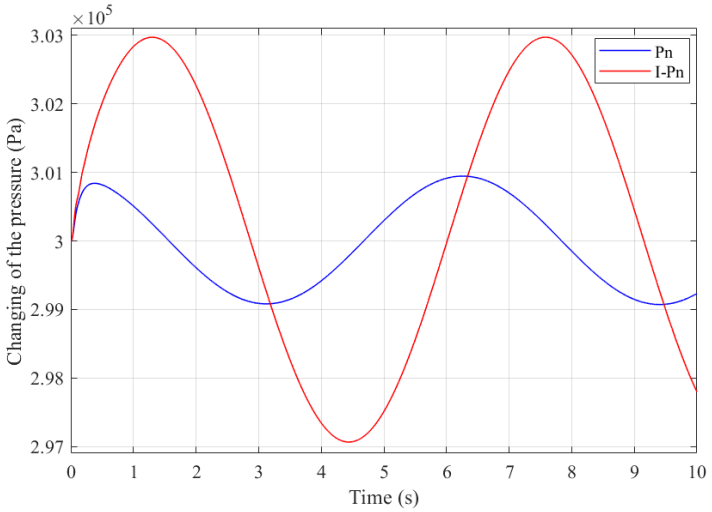

Figure 7 Changing of the pressure of the pneumatic spring.

Case 2: Random type

In this case, random excitation is used. This is the actual pavement type, and it gives more accurate results than the sine form. In Figure 8, the change of displacement of the sprung mass over time is clearly shown. This value changes continuously, and it does not follow any rules because the excitation from the road surface is random. Although the maximum amplitude of the stimulus from the road surface is only 50.00 (mm), the displacement value can reach 70.12 (mm) and 39.70 (mm) respectively in the case of the vehicle using the passive suspension system and the conventional pneumatic suspension system. If the integrated pneumatic suspension system is equipped, this value is further reduced. Besides, the difference in displacement at different times is not large. The average value of vehicle body's displacement in this case, respectively, reaches RMS = {30.70; 17.06; 12.56} (mm).

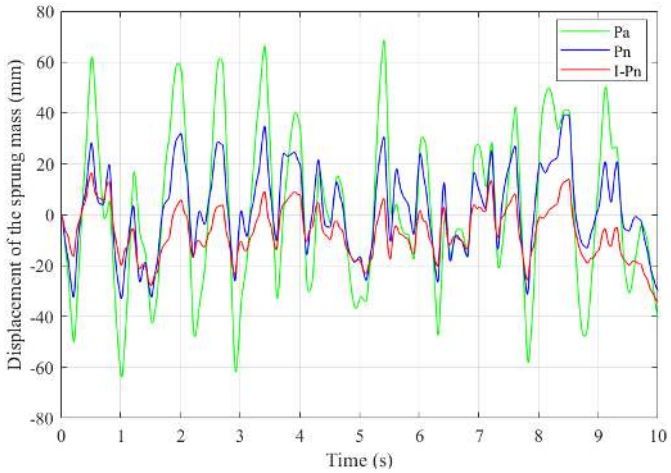

Figure 8 Displacement of the sprung mass.

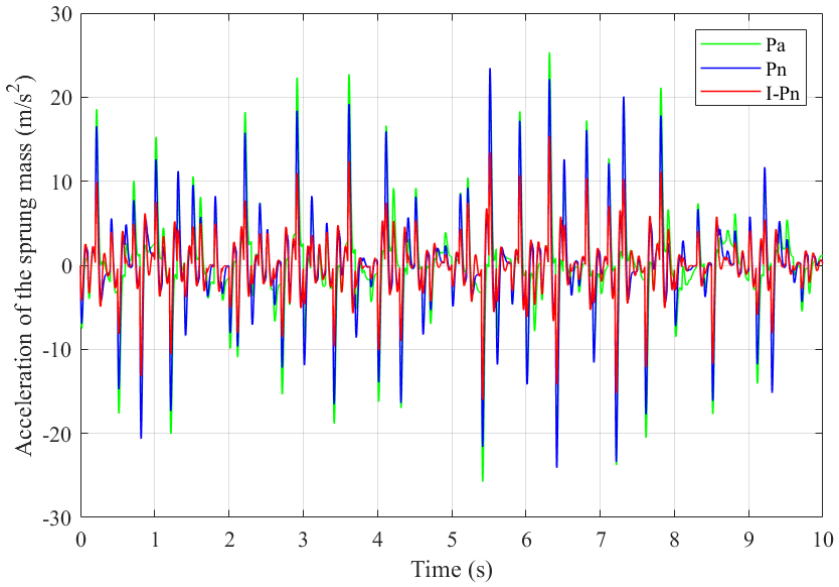

Figure 9 Acceleration of the sprung mass.

The acceleration of the vehicle body in this case fluctuates continuously (Figure 9). Besides, its maximum value is also very large, can reach 25.51 (m/s^2) if the vehicle only uses the passive suspension system. This is a huge value, and it can affect the smoothness and comfort of the vehicle. If the vehicle uses the conventional pneumatic suspension system, the maximum value of acceleration can reach 23.12 (m/s^2), the difference is not too large. Otherwise, if the integrated pneumatic suspension system is equipped to replace the other suspension systems, stability and comfort can be further improved. The maximum value of the acceleration of the sprung mass, in this case, is only 15.72 (m/s^2), which is much smaller than the other two cases. This stability is also shown through the average value of acceleration RMS_{Pa} = 6.53 (m/s^2), RMS_{Pn} = 6.18 (m/s^2), and $RMS_{I\text{-}Pn}$ = 3.78 (m/s^2).

To be able to respond well to changes from road surface stimuli, the pressure of the pneumatic suspension system needs to be constantly changing. The change in pneumatic pressure when the vehicle uses the integrated pneumatic suspension system is better and more consistent than with the conventional pneumatic suspension system (Figure 10). As a result, stability and safety issues of the vehicle can be more assured.

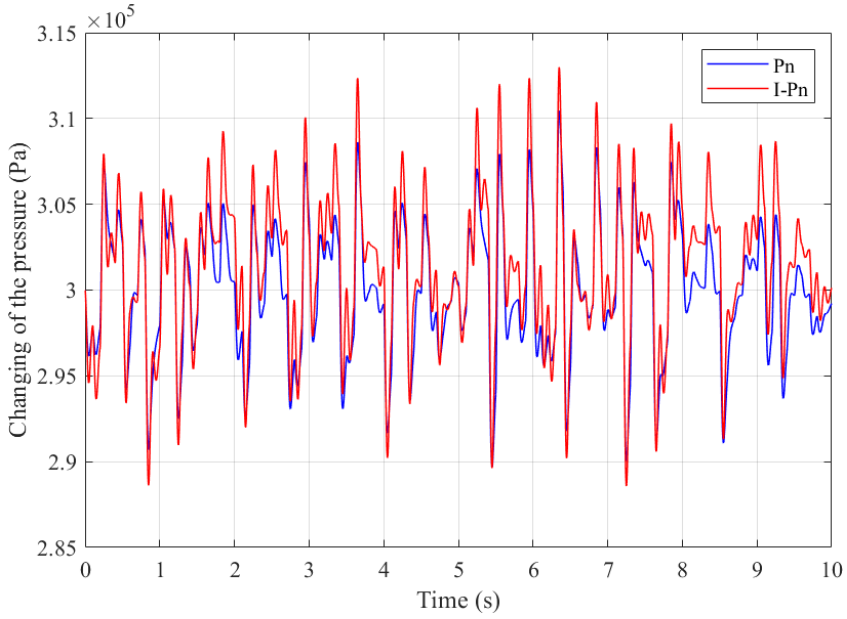

Figure 10 Changing of the pressure of the pneumatic spring.

Case 3: Step type

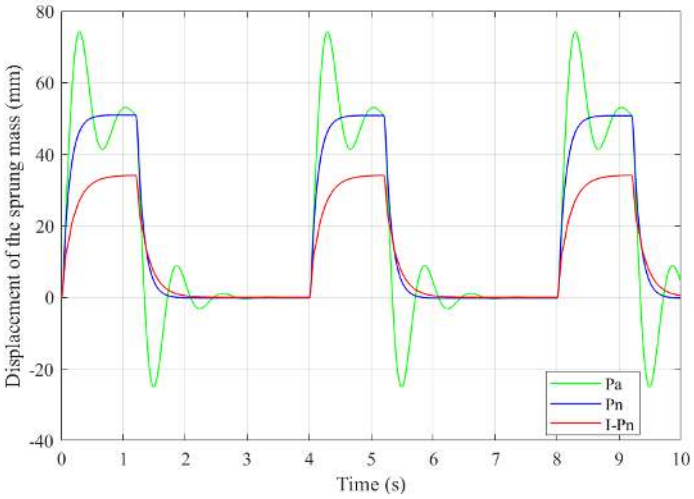

Figure 11 Displacement of the sprung mass.

Step stimulation is used in this case. The displacement of the sprung mass was markedly different in the three simulation conditions (Figure 11). The maximum value of amplitude reached 73.42 (mm), 49.53 (mm), and 32.82 (mm) respectively. The acceleration of the sprung mass is also significantly different when the vehicle uses the integrated pneumatic suspension system (Figure 12). Its maximum value is only about 57.3% compared to the vehicle using the passive suspension system. The pneumatic pressure variation of the suspension system is suitable, which response well to changes from external stimuli (Figure 13).

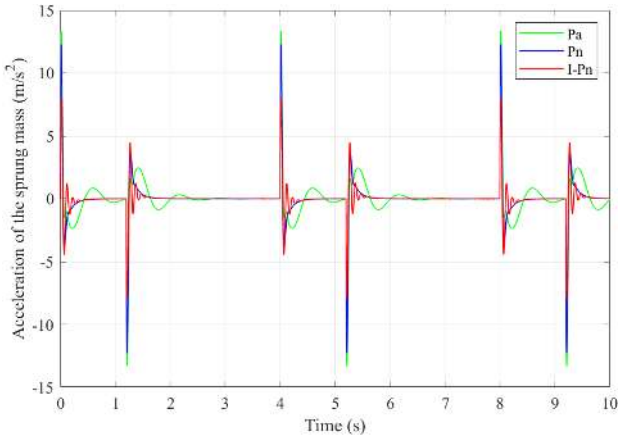

Figure 12 Acceleration of the sprung mass.

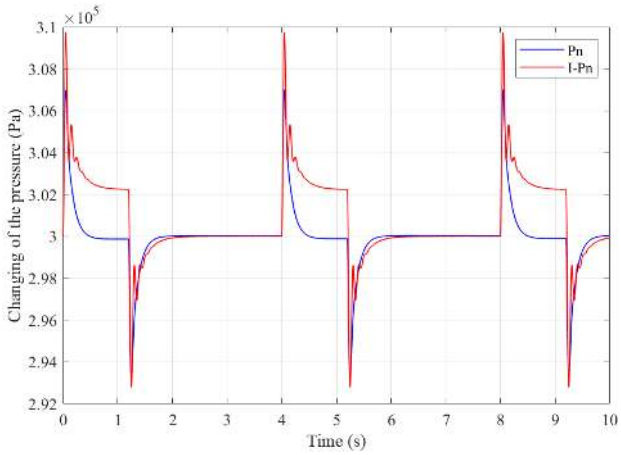

Figure 13 Changing of the pressure of the pneumatic spring.

In general, when the vehicle was equipped with the pneumatic suspension system that integrated the hydraulic actuator, the displacement and acceleration values of the sprung mass were significantly reduced compared with the other two cases. This has been demonstrated through various excitation conditions from the road surface. Therefore, the smoothness and comfort of the vehicle have been greatly improved. The results of the simulation process are shown in Tables 3 and 4.

Table 3 The maximum value of the oscillation.

	Pa		Pn		I-Pn	
	Dis (mm)	Acc (m/s^2)	Dis (mm)	Acc (m/s^2)	Dis (mm)	Acc (m/s^2)
Case 1	50.10	0.44	49.91	0.37	27.55	0.22
Case 2	70.12	25.51	39.70	23.12	26.31	15.72
Case 3	73.42	13.63	49.53	12.52	32.82	7.81

Table 4 The average value of the oscillation.

	Pa		Pn		I-Pn	
	Dis (mm)	Acc (m/s^2)	Dis (mm)	Acc (m/s^2)	Dis (mm)	Acc (m/s^2)
Case 1	34.69	0.052	34.63	0.043	20.79	0.025
Case 2	30.70	6.53	17.06	6.18	12.56	3.78
Case 3	32.30	2.40	28.89	2.15	18.83	1.46

NOMENCLATURE

F_A : Actuator force, N
F_C : Linear damping force, N
F_{Ca} : Nonlinear damping force, N
F_f : Friction force, N
F_{Ke} : Main spring force, N
F_{Kv} : Auxiliary spring force, N
F_p : Push force, N
F_S : Pneumatic force, N
F_z : Static force, N
h : Bump on the road, m
m_1 : Sprung mass, kg
m_2 : Unsprung mass, kg
m_a : Pneumatic mass, kg
z_1 : Displacement of the sprung mass, m
z_2 : Displacement of the unsprung mass, m
z_a : Displacement of the pneumatic mass, m

4 CONCLUSION

Stability and comfort are very important issues of the vehicle when moving on the road. It directly affects passengers, cargoes, and the durability of the vehicle. The comfort and smoothness are expressed through the vibrations of the sprung mass. Characterizing the main oscillations are the values of displacement and acceleration of the sprung mass (maximum amplitude, stability amplitude, oscillation frequency, etc.). The vehicle's suspension system has the role of regulating and extinguishing the oscillations generated from external stimuli.

In order to improve the efficiency of the suspension system, the stiffness of the spring and damper needs to be changed continuously. The method of equipping the conventional pneumatic suspension system to replace the passive suspension system has been proposed and used. However, the results it brings are still not great. This paper has introduced a method of using the pneumatic suspension system integrated with the hydraulic actuator instead of just using the conventional pneumatic suspension system. The hydraulic actuator is controlled through the established linear controller. Based on the basic parameters of the system, the simulation process was performed for different excitation conditions from the road surface. The results of the research show that when the vehicle is equipped with the integrated

pneumatic suspension system, the values of displacement and acceleration of the sprung mass are greatly reduced compared to the other two cases. To be able to do this, the system's compressed pneumatic pressure changes are continuous.

The method of using the integrated pneumatic suspension system is completely novel. Its results are also very positive. However, the investment cost is quite expensive. In the future, nonlinear control and intelligent control methods can be used to control this system to improve its efficiency. Besides, the experimental process is necessary to be able to demonstrate the effectiveness of the integrated pneumatic suspension system.

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