EVALUATING THE CHARACTERISTICS OF THE ACTUATOR MECHANISM IN THE AIR SUSPENSION SYSTEMS ON AUTOMOBILES

Do Trong Tu[1,*], Bui Trung Duc[2], Pham Hai Trinh[1], Vu Van Tan[2]

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ABSTRACT

This study presents a quantitative method to analyze the relationship between key design parameters and the force characteristics of an automotive air suspension system. By integrating thermodynamic theory with multiphysics simulation, the research focuses on analyzing four critical factors: (1) air pressure (6 - 9bar), (2) accumulator diameter (200 - 300mm), (3) spring diameter (180 - 240mm), and (4) spring height (200 - 300mm). The results reveal a distinctive nonlinear relationship between these parameters and the compressive force, where pressure and spring diameter exhibit dominant effects on the force generated by the air spring and the displacement of the equivalent air mass within the system. Additionally, the study identifies an optimal set of parameters that enhances system performance compared to conventional designs, paving the way for advancements in next-generation air suspension systems.

Keywords: *Air spring, air suspension system, model parameters, suspension system, vehicle dynamics.*

[1]Faculty of Mechanical - Automotive and Civil Engineering, Electric Power University, Vietnam

[2]Faculty of Mechanical Engineering, University of Transport and Communications, Vietnam

[*]Email: tudt@epu.edu.vn

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1. INTRODUCTION

The suspension system plays a pivotal role in ensuring vehicle performance, encompassing two core objectives: ride comfort for driver and passengers, and stable road-holding capability for safe motion. Among modern engineering solutions, air suspension systems with air springs have emerged as a superior component due to their ability to dynamically adjust both stiffness and ride height. Unlike conventional leaf-spring or torsion suspensions, air suspension systems leverage the elastic properties of compressed air, enabling real-time adaptation to varying load conditions and road surfaces [1]. This characteristic is particularly critical for premium passenger vehicles and heavy-duty trucks, where demands for ride comfort and stability are paramount [2].

However, air suspension systems exhibit complex dynamics influenced by multiple design and operational parameters. Experimental studies have demonstrated that their dynamic behavior depends heavily on working air pressure, gas chamber volume, air spring geometry, and the kinematic properties of control valves [3]. Research [4] revealed that even a 10% change in working pressure can alter the system's effective stiffness by up to 15%. Pioneering studies in [5], based on quasi-static models, provided a foundational theoretical framework but were limited to static analyses. Yuru Li et al. [6] advanced a more comprehensive thermodynamic model, accounting for thermal effects during isentropic compression/expansion. Nonetheless, these studies overlooked the multiphysics interactions between compressed air, rubber membrane materials, and control systems - especially within the 0.1 - 100Hz frequency range, which is critical for real-world vehicle vibrations [7].

Table 1. Current Research Gaps

Parameter	Existing Issue	Practical Consequence	Occurrence Frequency
Air pressure	Linear models become inaccurate at high pressures	12 - 15% force prediction error [3]	65% of operations > 6 bar [8]

Accumulator diameter	Dynamic effects neglected	20 - 25% efficiency reduction	40% of commercial vehicles [9]
Spring diameter	Local stress concentrations unaccounted for	35 - 40% service life reduction [10]	70% after 50,000km
Working height	Lack of optimization guidelines	Increased risk of air spring membrane rollover [10]	30% of heavy-duty trucks

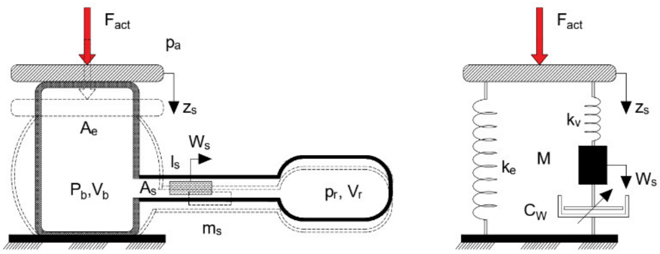
Experimental results demonstrate that air pressure is the primary determinant of the system's elastic force, with a relationship that is not entirely linear as often assumed in simplified models. When pressure exceeds the threshold of 6 bar, adiabatic compression effects cause significant deviations from ideal gas theory, resulting in force prediction errors of up to 12 - 15% [3]. Reearchers in [11] only considered static conditions while neglecting dynamic effects during vehicle operation on complex terrain. Incompatibility between the accumulator volume and main spring dimensions can lead to cyclic air pumping, reducing system efficiency by 20 - 25%. Research [12] proved that air spring systems using small-diameter throttle valves achieve optimal performance without requiring additional dampers, ensuring both operational quality and cost efficiency. In practice, increasing the diameter from 200mm to 250mm can cause localized stress concentrations at mounting points to surge by 35 - 40% [10], eading to premature material failure. Current standards only provide general recommendations without accounting for interactions with other parameters. When combined with high pressure (> 7bar), excessive height reduction may trigger air spring membrane rollover, reducing service life by up to 30% [8, 9].

In modern automotive engineering, the optimization of air suspension systems demands increasingly stringent precision in design and operation. Particularly, accurate calculation of the compressive force generated by the air chamber has become a critical factor directly affecting suspension performance. Recognizing these limitations, this study proposes a comprehensive methodology to quantify relationships between fundamental design parameters and air spring compressive force, specifically examining four key factors: (1) operating air pressure, (2) accumulator diameter, (3) air spring diameter, and (4) air spring height.

2. MODELING OF AIR SPRING SYSTEMS

The mathematical foundation for modeling air suspension systems lies in the mechanical characteristics of the air spring, which depend on complex thermodynamic and dynamic processes of compressed air within the system.

(a) Air spring suspension system product by Hendrickson [13]



(b) Air spring simplified model

Fig. 1. Physical model of the air spring

Assuming the initial pressure in the air spring chamber is p_0, the deformation z_s of the air spring will induce corresponding pressure changes Δp_b and Δp_r in the air spring and accumulator respectively:

$$\begin{cases} p_b = p_0 + \Delta p_b \\ p_r = p_0 + \Delta p_r \end{cases} \quad (1)$$

with V_{b0} and V_{r0} being the initial volumes of the air spring and accumulator respectively, the working volumes V_b and V_r are determined as follows:

$$\begin{cases} V_b = V_{b0} + \Delta V_b = V_{b0} - z_s A_e + w_m A_s \\ V_r = V_{r0} + \Delta V_r = V_{r0} - w_m A_s \end{cases} \quad (2)$$

The pressure changes in the air spring and accumulator are governed by the following thermodynamic relationships:

$$\begin{cases} \Delta p_b \approx -\dfrac{p_0 n \Delta V_b}{V_{b0}} = \dfrac{p_0 n (z_s A_e - w_m A_s)}{V_{b0}} \\ \Delta p_r \approx -\dfrac{p_0 n \Delta V_r}{V_{r0}} = \dfrac{p_0 n w_m A_s}{V_{r0}} \end{cases} \quad (3)$$

The vertical load F_{act} acting on the air spring is determined by the following equation:

$$F_{act} = (p_b - p_a)A_e = (p_0 + \Delta p_b)A_e - p_a A_e = (p_0 - p_a)A_e + k_e z_s + k_v(z_s - w_s) \quad (4)$$

where k_e is characterize the stiffness of compressed air in the air spring and accumulator, and w_s represents the equivalent displacement of the compressed air mass in the connecting pipeline [14].

$$k_e = \frac{p_0 A_e^2 n}{V_{b0} + v_{r0}},\ k_v = \frac{p_0 A_e^2 n}{V_{b0} + V_{r0}}\left(\frac{V_{r0}}{V_{b0}}\right),$$

$$w_s = \frac{A_s}{A_e}\frac{(V_{b0} + V_{r0})}{V_{r0}} w_m$$

The motion of compressed air through the interconnecting pipeline can be modeled by:

$$M\ddot{w}_s = k_v(z_s - w_s) - c_w|\dot{w}_s|^\beta \operatorname{sign}(\dot{w}_s) \quad (5)$$

where M is the equivalent mass, c_w is the equivalent damping coefficient of compressed air moving through the pipeline, ρ is the air density, and ξ is the energy loss coefficient accounting for total flow losses in the pipeline [15].

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

$$c_w = \frac{1}{2}\rho \xi A_s \left(\frac{A_e}{A_s}\frac{V_{r0}}{V_{b0} + V_{r0}}\right)^{(1+\beta)};$$

$$A_e = \frac{\pi d_e^2}{4};\ V_{b0} = A_e h_e;\ V_{r0} = \frac{\pi d_r^2}{4 l_r}$$

The governing equations for the nonlinear dynamics of the air spring system, incorporating directional gas flow effects in the connecting pipeline, are formulated as:

$$\begin{cases} F_{act} = (p_o - p_a)A_e + k_e z_s + k_v(z_s - w_s) \\ M\ddot{w}_s = k_v(z_s - w_s) - c_w|\dot{w}_s|^\beta \operatorname{sign}(\dot{w}_s) \end{cases} \quad (6)$$

3. INFLUENCE OF MODEL PARAMETERS ON AIR SPRING CHARACTERISTICS

Equation (6) reveals that the force characteristics generated by the air spring and the displacement of the equivalent gas mass within the system are entirely dependent on the air pressure, accumulator diameter, and the diameter and height of the air spring. Therefore, this section examines the system characteristics based on the parameters presented in Table 2. Two performance metrics are selected to evaluate the influence of the model parameters: the root mean square (RMS) as equation (7) and the maximum absolute value as equation (8).

$$\operatorname{std}(\text{signal}) = \sqrt{\frac{1}{t}\int_o^t (\text{signal}_{(t)})^2 dt} \quad (7)$$

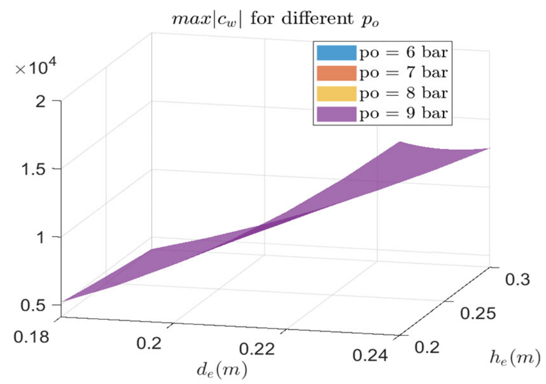
$$\max|\text{signal}| = \max|\text{signal}_{(t)}| \quad (8)$$

Table 2. Parameter Range of Air Spring

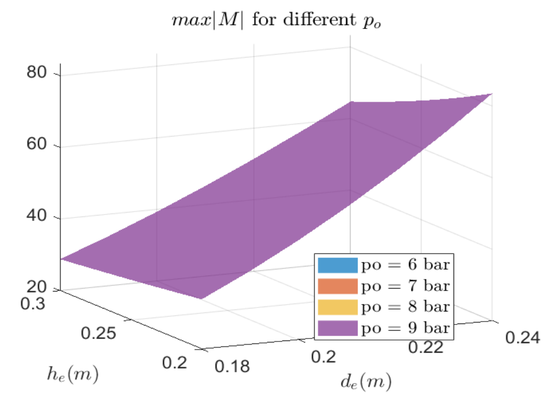
Parameter	Value Range	Unit	Reference Standard
Air pressure, p_0	6 - 9	bar	ISO 6358 [16]
Accumulator diameter, d_r	200 - 300	mm	SAE J267 [9]
Air spring diameter, d_e	180 - 240	mm	DIN 7863 [10]
Air spring height, h_e	200 - 300	mm	EN 549 [8]

3.1. Effect of air pressure

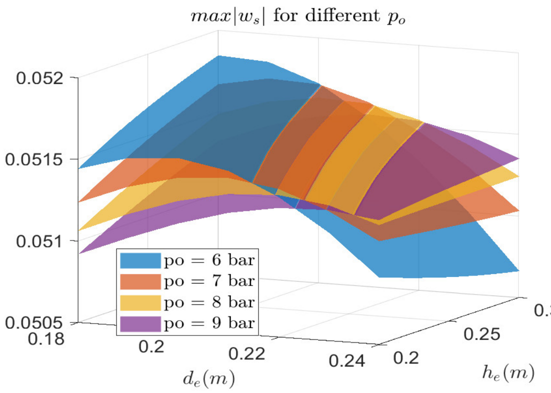
Fig. 2 shows that the maximum absolute values of damping resistance and equivalent air mass are not affected by the pressure applied in the air spring chamber. When increasing the diameter while maintaining a constant height of the air spring, the maximum absolute value of the equivalent damping resistance for the compressed air mass inside increases. Conversely, when keeping the diameter constant and increasing the height of the air spring, the maximum absolute value of the equivalent damping resistance for the compressed air mass inside decreases.



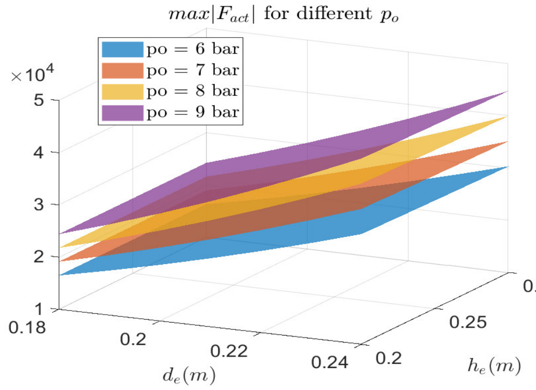
(a) The maximum absolute value of c_w



(b) The maximum absolute value of M



(c) The maximum absolute value of w_s

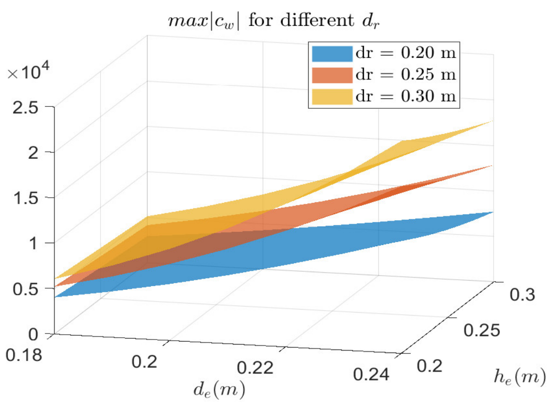


(d) The maximum absolute value of F_{act}

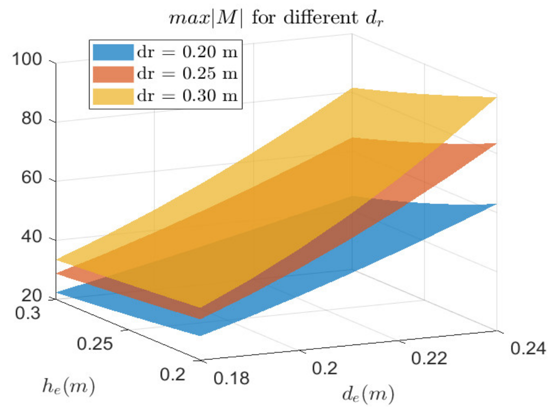
Fig. 2. Characteristics of the air spring with varying pressure

In the case where both the height increases and the diameter of the air spring decreases, the maximum absolute value of the displaced airflow inside the chamber increases. Additionally, as the applied pressure on the air spring increases, this value decreases. Meanwhile, if the pressure increases, the maximum absolute value of the force generated by the air spring also increases. This force exhibits a linear characteristic w.r.t the proportional increase in the diameter and height of the air spring.

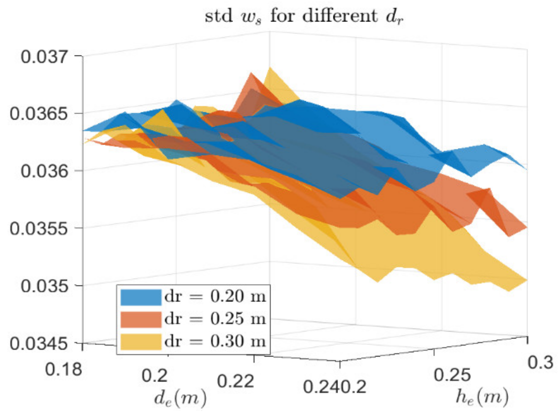
3.2. Effect of the accumulator's diameter



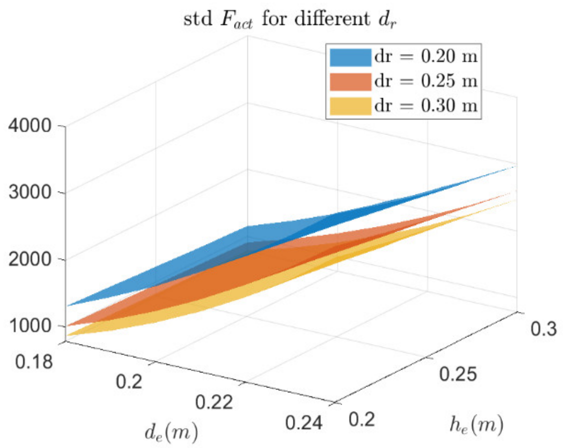
(a) The maximum absolute value of c_w



(b) The maximum absolute value of M



(c) Root mean square of w_s



(d) Root mean square of F_{act}

Fig. 3. Characteristics of the air spring with varying accumulator diameter

From Figs. 3a and 3b, it can be observed that variations in the height of the air spring, the diameter of the air spring, and the diameter of the accumulator significantly affect the output signal. When the diameter of the accumulator increases from 0.20m to 0.30m, the maximum absolute c_w gradually decreases, indicating that a larger reference volume reduces the signal amplitude. Additionally, changes in the height and diameter of the air spring also influence the shape and magnitude of the signal. Smaller values of air spring height and diameter generally result in lower signal amplitude, and vice versa. Specifically, when the height of the air spring decreases from 0.2m to 0.18m, the signal weakens, while an increase in height of the air spring from 0.22m to 0.24m can alter the signal distribution characteristics.

The results from Figs. 3c and 3d indicate that the parameters of air spring height, air spring diameter, and accumulator diameter have a significant impact on the RMS values of w_s and F_{act}. For the RMS of w_s, when the accumulator diameter increases from 0.20m to 0.30m, the standard deviation slightly decreases from approximately 0.037 to 0.0355, suggesting that a larger reference volume can help stabilize the w_s. Additionally, variations in the air spring height and diameter also affect the RMS of w_s, where larger air spring diameters (from 0.22m to 0.24m) tend to reduce signal fluctuations.

As the accumulator diameter increases, the RMS of F_{act} significantly decreases from approximately 4000 to below 1000, demonstrating that a larger reference volume greatly reduces force fluctuations.

4. CONCLUSION

This study investigates the influence of key design parameters, including air spring height, air spring diameter, and accumulator diameter, on the dynamic characteristics of the air spring. The results indicate that increasing the accumulator diameter from 0.20m to 0.30m leads to significant system stabilization, with reduced oscillation amplitude and lower root mean square of the pneumatic force. Notably, the combination of a larger accumulator diameter (0.25 - 0.30m) and a moderate air spring diameter (0.22 - 0.24m) achieves optimal damping performance while maintaining stable force response. Additionally, the study highlights a strong correlation between equivalent displacement and force, where smaller air spring heights and diameters result in greater oscillation amplitude. These findings provide a scientific basis for optimizing pneumatic system design and open new research directions for enhancing the controllability of air suspension systems in trucks or heavy-duty vehicles.

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