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Research on Truck Active Suspension Systems Effectiveness with Auxiliary Hydraulic Cylinders

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Abstract. The vehicle's body oscillation when traversing bumps not only causes driver discomfort but also negatively impacts the quality of the transported cargo. Consequently, numerous research studies have been conducted to improve suspension system characteristics with the goal of enhancing the vehicle’s ride comfort and body stability. The majority of them have shown that equipping trucks with an active suspension is the most effective technical solution for ensuring comfortable working conditions for driver and cargo safety. This paper focuses on modeling and controlling an active suspension system for a truck in specialized simulation software, followed by evaluating its effectiveness compared to traditional passive suspension. In particular, the paper presents a quarter-vehicle suspension model integrated with an auxiliary hydraulic cylinder submodel controlled by a PID controller. The input parameters of the simulated suspension system were determined during full-scale experiments with real vehicles in laboratory and road conditions. To confirm the adequacy of the proposed mathematical model an experimental scenario was designed for measuring the vehicle body's oscillation parameters when traversing a step bump in the “passive” suspension control mode. After validating the model, the study proceeded to investigate the effectiveness of the active suspension system with the auxiliary hydraulic cylinder pressure PID control and compare the obtained results with the “passive” control option of the suspension system. The survey results indicate that the active suspension system, in conjunction with the PID control algorithm, significantly improves key performance metrics of the system. Specifically, the study found a reduction in oscillation damping time from 1.61 sec to 0.92 sec, a 16.7 % decrease in maximum amplitude of vehicle body oscillation and a substantial 61.5 % average reduction in vehicle body oscillation acceleration. On the other hand, in the active suspension system, the damping ratio also improved by about 5.8 % (from 0.260 to 0.245). These findings underscore the effectiveness of the active suspension system, as developed in this research, in enhancing the overall performance of the vehicle in terms of stability, safety, and ride comfort.

Keywords: truck, active suspension, auxiliary hydraulic cylinder, PID controller, 1/4 suspension model, mathematical modeling, semi-natural experiment, damping ratio, damping characteristics

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Исследование эффективности активной подвески грузового автомобиля со вспомогательными гидравлическими цилиндрами

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Реферат. Колебания кузова автомобиля при преодолении неровностей дороги не только вызывают дискомфорт у водителя, но и негативно сказываются на качестве перевозимого груза. Многочисленные проведенные исследования,

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направленные на улучшение характеристик систем подрессоривания с целью повышения комфорта езды и устойчивости кузова автомобиля, показали, что оснащение грузовых транспортных средств активной подвеской является наиболее эффективным техническим решением для обеспечения комфортных условий труда водителя и сохранности транспортируемого груза. В данной статье акцент сфокусирован на моделировании и управлении активной подвеской грузового автомобиля в специализированном программном обеспечении с последующей оценкой ее эффективности по сравнению с традиционной пассивной подвеской. В частности, в работе представлена четвертная модель подвески автомобиля с интегрированной субмоделью вспомогательного гидравлического цилиндра, управляемого посредством программного ПИД-регулятора. Входные параметры моделируемой системы подрессоривания определялись в ходе натурных экспериментов с реальными транспортными средствами в лабораторных и дорожных условиях. Для подтверждения адекватности предложенной математической модели разработан сценарий и проведен эксперимент по измерению параметров колебаний кузова при преодолении грузовым автомобилем неровностей в режиме «пассивного» управления подвеской. Валидация модели позволила в дальнейшем сфокусироваться на исследовании эффективности активной подвески с ПИД-регулированием давления в рабочей полости вспомогательного гидравлического цилиндра и сопоставлении полученных результатов с вариантом «пассивного» управления системой подрессоривания. Результаты исследования показали, что система активной подвески в сочетании с ПИД-алгоритмом управления давлением вспомогательного гидроцилиндра значительно улучшает оценочные показатели процесса затухания колебания кузова автомобиля. Так, время затухания колебаний кузова сокращается с 1,61 до 0,92 с, а максимальная амплитуда и среднее ускорение колебаний уменьшаются соответственно на 16,7 и 61,5 %. При этом коэффициент демпфирования системы уменьшается с 0,260 до 0,245 (приблизительно на 5,8 %), что в совокупности подтверждает эффективность разработанной в рамках данного исследования активной системы подрессоривания, а также способствует повышению общей производительности автомобиля с точки зрения устойчивости, безопасности и комфортабельности езды.

Ключевые слова: грузовой автомобиль, активная подвеска, вспомогательный гидроцилиндр, ПИД-регулятор, 1/4 модель подвески, математическое моделирование, полунатурный эксперимент, коэффициент демпфирования, характеристики затухания колебаний

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1. Introduction

The suspension system is a soft connection between the vehicle body and the wheels, responsible for ensuring a smooth ride for the driver, passengers, and cargo, transmitting vertical and horizontal forces from the wheels to the body and vice versa. Currently, with increasing demands for the smoothness of transported goods, active and passive suspension systems are being researched and developed into practical products [1, 2]. Among them, the active suspension system stands out in terms of both research trends and developments [3, 4].

An active suspension system can alter the forces exerted from the wheels to the vehicle body, thereby controlling the body's movement by adjusting the forces of the elastic and damping elements. Presently, several trends in active suspension systems include electronically controlled air suspension systems and electronic regenerative suspension systems.

Current research on active suspension focuses on optimizing suspension parameters to enhance driver, passenger, and cargo comfort. In Kumar's investigation [5] passive and active suspension systems based on criteria such as ride comfort, suspension travel, and traction ability were compared. Additionally, research ideas for improving slip control have also been proposed by several publications [6–10]. In 2017, a study on limiting vertical acceleration to improve passenger and cargo comfort was published by Fabian León-Vargas and colleagues [11].

To achieve the forces acting on the vehicle body, various actuators have been studied for their effectiveness in active suspension systems. Several papers analyzing the performance of magnetorheological (MR) damper actuators indicate that MR dampers effectively absorb body vibrations and require low input power [12–14]. Besides MR fluid dampers, hydraulic actuators have also been extensively researched [15–18]. In one study, Y. M. Sam [19] simulated an active suspension system using hydraulic cylinders, demonstrating that this system significantly reduces body acceleration and displacement compared to passive suspension systems.

In recent years, many control methods for active suspension systems have been developed to optimize actuator performance. Studies on using Fuzzy Logic controllers to improve active suspension stability and flexibility were published in 2018 by Tiechao Wang and colleagues [20]. This method has also been used by researchers Hongyi Li and M. V. C. Rao in their studies [21, 22]. Additionally, PID controllers are widely used in active suspension control due to their simplicity and quick response to road surface irregularities [23–25]. Some researchers have combined two or more control methods to achieve the best results. In one of the articles [26] the authors combined PID and Fuzzy algorithms to demonstrate the active suspension effectiveness based on two criteria: body acceleration and wheel-body displacement.

This paper focuses on the evaluation of an active suspension system using a servo valve and hydraulic cylinders by simulation method [27–29]. The advantage of this actuator is its rapid response and individual installation for each wheel, ensuring continuous smoothness for the vehicle. After developing the mathematical equations, the passive suspension system is simulated using specialized simulation software. Real vehicle experiments were conducted to obtain input parameters for the simulation and verify the model. The hydraulic actuator was added to the verified model to develop the active suspension simulation. Subsequently, the research series under various scenarios was conducted to compare and evaluate the effectiveness of active and passive suspension systems according to three criteria: oscillation amplitude, oscillation acceleration, and damping time.

2. Research methodology and developing an active suspension system model

This paper analyzes the application of an active suspension system on a truck through simulation and investigation of a 1/4 suspension system's responses. The active suspension system model is built based on a passive suspension system model with the addition of a hydraulic cylinder to alter the force applied to the vehicle frame. Each control valve of the hydraulic cylinder is managed by a PID controller. The physical model of such 1/4 active suspension system is shown in Fig. 1. The operation of the system is represented and simulated by the set of mathematical equations (1) below. The active suspension system is modeled based on the passive suspension system with the addition of force F_a generated by the hydraulic cylinder actuator. In the case of studying the passive suspension system for comparison, the force F_a is assigned a value of zero.

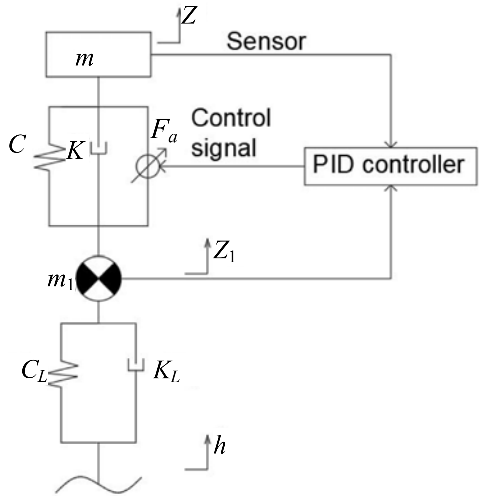

Fig. 1. Dynamic Diagram of the active suspension system

The active suspension model is described by the following mathematical equations:

$$
\begin{aligned}
m\ddot{Z} &= F_C + F_K + F_a;\\
m_1\ddot{Z}_1 &= (F_{CL} + F_{KL}) - (F_C + F_K + F_a);\\
F_C &= C\left(Z_1 - Z\right);\\
F_K &= K\left(\dot{Z}_1 - \dot{Z}\right);\\
F_{CL} &= C_L\left(h - Z_1\right);\\
F_{KL} &= K_L\left(\dot{h} - \dot{Z}_1\right),
\end{aligned}
\qquad (1)
$$

where m, m_1 – respectively sprung and unsprung mass, kg; F_a – hydraulic cylinder force of the hydraulic control system, N; F_C and C – force and stiffness of the elastic element, N and N/m; F_K and K – damping force and damping coefficient of shock absorber, N and Ns/m^2; F_{CL} and C_L – force and stiffness of tire, N and N/m; F_{KL} and K_L – force and damping coefficient of tire, N and Ns/m^2; h – road profile, m; $\dot{h}$ – speed change of road profile, m/s; Z and Z_1 – displacement of the sprung and unsprung masses, m; $\dot{Z}$ and $\dot{Z}_1$ – vertical speed of the sprung and unsprung masses, m/s; $\ddot{Z}$ and $\ddot{Z}_1$ – acceleration of the sprung and unsprung masses, m/s^2. In this study, we utilize the parameters of a truck with the following set of values: m = 1300 kg; m_1 = 200 kg; C = 220000 N/m; C_L = 29000 N/m; K = 3500 Ns/m^2; K_L = 3500 Ns/m^2.

The mathematical model of the actuator system, that generates the force F_a has a structure as shown in Figure 2, which includes a force cylinder and a spool valve. The paper uses a pressure source assumed to be stable, generated by a high-pressure hydraulic pump P_v, supplied to the distribution valve. During operation, the distribution valve directs high-pressure hydraulic oil from the pump to the hydraulic cylinder chambers A and B. Distance sensors installed on the vehicle send signals regarding the displacement of the sprung mass and body roll angle to the controller. The controller then generates signals to control the servo valve displacement, thereby producing a force on the sprung mass due to the pressure difference between chambers A and B of the hydraulic cylinder.

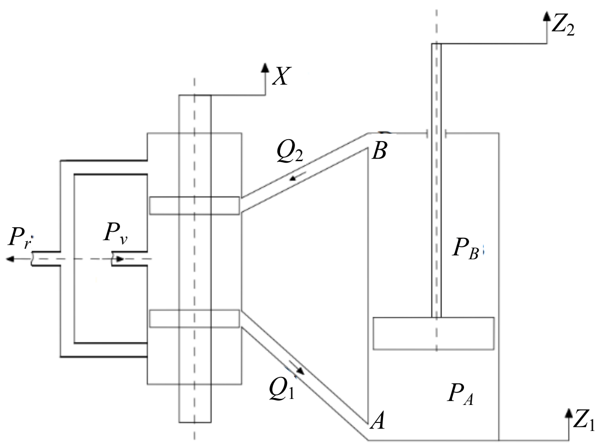

Fig. 2. Hydraulic control system model

When calculating and simulating the hydraulic system, this paper makes several assumptions, such as neglecting the fluid flow from chamber A to chamber B due to the gap between the piston and the cylinder (internal leakage), and the fluid flow from chambers A and B to the outside (external leakage) due to the non-absolute sealing of the cylinder structure. The initial pressure in chambers A and B is considered to be zero. With these assumptions, the equations for the pressures in chambers A and B of the cylinder are obtained as follows:

$$P_A = \frac{K_e}{V_A}\int\left(-\left(\dot{Z}_1 - \dot{Z}\right)S_A + Q_1\right)dt; \qquad (2)$$

$$P_B = \frac{K_e}{V_B}\int\left(-\left(\dot{Z}_1 - \dot{Z}\right)S_B + Q_2\right)dt, \qquad (3)$$

where V_A and V_B – A and B chamber's cylinder volume, m^3; Q_1 and Q_2 – fluid flow rates into and out respectively A and B chambers, m^3/s; $\dot{Z}_1$ and $\dot{Z}$ – displacement's velocity of the unsprung and sprung masses, m/s; K_e – bulk modulus, Pa.

The equation for the fluid flow rate entering the force cylinder chambers is calculated by the formula:

$$Q = k_t X\sqrt{\frac{2}{\rho}(P_v - sign(X)P_L)}, \qquad (4)$$

where Q – flow rate into the chamber, m^3/s; k_t – adjustable flow coefficient of the valve; P_v – pump pressure into the distribution valve, Pa; P_L – is the pressure in the piston chambers, Pa; X – the displacement of the distribution valve, m; ρ – liquid density, kg/m^3.

However, the flow coefficient k_t is an adjustable variable and challenging to determine precisely. Therefore, the fluid flow rate is calculated using a proportional formula, where a certain displacement X_{dn} of the servo valve and the pressure difference ΔP_{dh} between chambers A and B will result in a flow rate Q_{dn}

$$Q_1 = Q_{dn}\frac{X}{X_{dn}}\sqrt{\frac{\Delta P}{\Delta P_{dn}}}. \qquad (5)$$

Therefore, the force F_a generated by the hydraulic cylinder is determined as follows

$$F_a = P_B S_B - P_A S_A. \qquad (6)$$

where S_A, S_B – piston's area on the side respectively without and with the piston rod, m^2 (Fig. 2).

As for the displacement X < 0 of the servo valve, the calculations follow similar steps as described above. In this study, the hydraulic parameters are calculated and selected as follows: P_v = 10342500 Pa; S_a= 0.01 m^2; ρ = 9000000 kg/m^3; X_{dn} = 1 mm; V_A = V_B = 0.002 m^3.

Above presenting mathematical models of the hydraulic actuator and suspension system kinematics are simulated using MATLAB Simulink, which structures are depicted in Fig. 3.

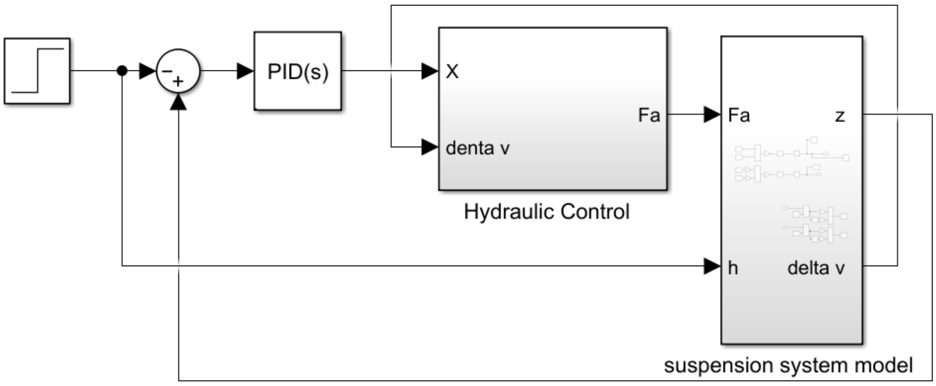

Fig. 3. Simulation model structures of an active suspension system

After simulating the mechanical system, we need to select a controller to control the displacement of the servo valve, providing input signals to the hydraulic control model. For the active suspension model in this study, achieving high precision in controlling the hydraulic actuator is essential. The PID controller is considered suitable and effective for the active suspension system due to its widespread use in closed-loop control systems (with feedback signals). The PID controller calculates the error value as the difference between the measured value of the variable parameter and the desired setpoint. It minimizes the error by adjusting the control input value. The feedback signal of the controller used in this study is the deviation between the vehicle body displacement and road irregularities. In practice, this parameter can be approximately interpolated from the distance between the vehicle body and the axle. The PID controller is designed with the following parameters: k_P = 0.002, k_D = 0.002, and k_I = 0.026. The controller output is then utilized to calculate the displacement of the hydraulic control valve.

3. Experimental procedure and simulation model validation

The purpose of the experiments is to determine the input parameters for the simulation model, specifically the damping coefficient, and to compare the simulated system model when the force exerted by the hydraulic cylinder is set to zero with the suspension system of a real vehicle under the same bumpy road conditions.

The characteristic of the real vehicle damper is determined by the test stand following Fig. 4 below.

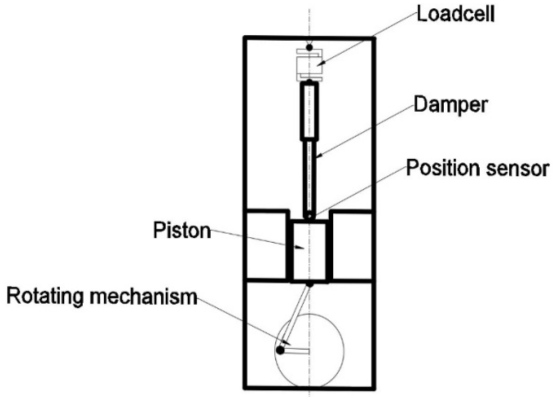

Fig. 4. Damper test stand

The system in Fig. 4 includes a drive mechanism that generates translational motion for the damper and sensors that measure damping force and the velocity of the damper's head. The damping characteristics obtained from these measurements are incorporated into the simulation model as input parameters. To validate the model, the paper utilized the experimental setup shown in Fig. 5. In this study, two HF sensors and one S-motion sensor from Kistler were used to measure the vehicle body displacement and acceleration as the vehicle moves over a 17.5 cm high step bump.

Due to the power limitation of the electric motor on the damper test stand, the experiment to determine the damping characteristics was only able to cover a small working area, as shown in Figure 6, where $v_k = \dot{Z}_1 - \dot{Z}$ is damper speed, m/s. The remaining working area of the damper (Fig. 7) was determined by interpolation based on the damping theory. The equivalent damping characteristics in Fig. 7 served as input for the previously developed quarter-car suspension simulation model.

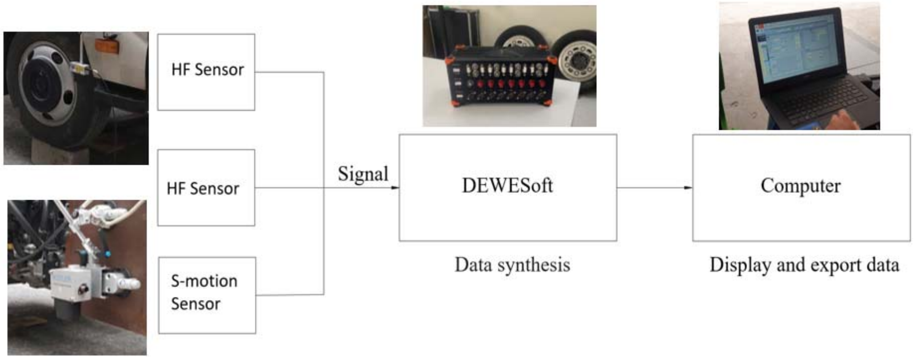

Fig. 5. Experimental configuration for determining the vehicle vibration

The paper validated the simulation model by comparing the experimental results of the passive suspension system on a real vehicle with the simulated passive suspension system model using the parameters obtained from the experiments (with no force applied from the hydraulic actuator). The results in Fig. 8 showed the amplitude values between the simulation and experiment differed by less than 10 % in general. Particularly, at the maximum oscillation amplitude, the deviation was only 0.61 %. The timing of peak values also showed close similarity, with the largest deviation being 7.7 %.

The simulated vehicle body acceleration in Fig. 9 closely matched the experimental results. Especially at the maximum acceleration (a_{max}), the deviation was only 0.82 %, indicating a high level of accuracy in the simulation. Through these comparisons, the simulation model closely approximated reality across three evaluation aspects: vehicle body displacement, body acceleration, and oscillation damping time, demonstrating the reliability of the constructed simulation model. Therefore, this simulati on model can be effectively used in conjunction with the hydraulic control model to assess and evaluate the effectiveness of the active suspension system.

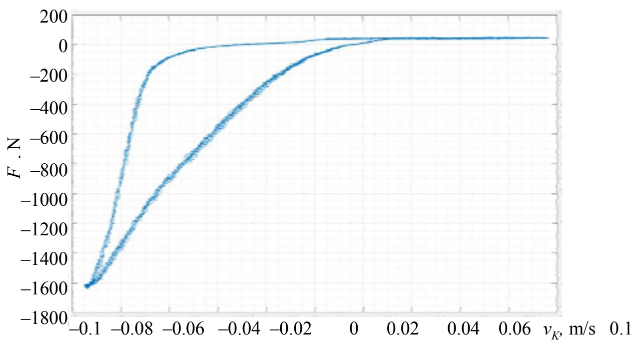

Fig. 6. Damping characteristics

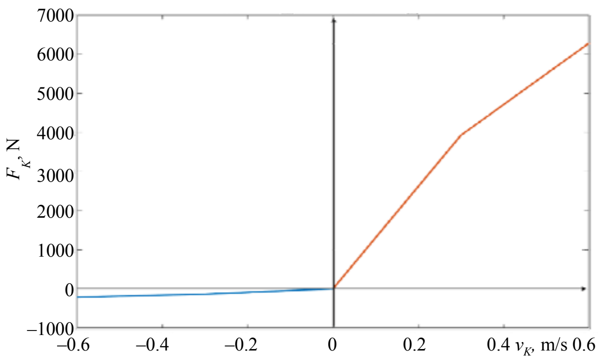

Fig. 7. Equivalent damping characteristics as input for simulation

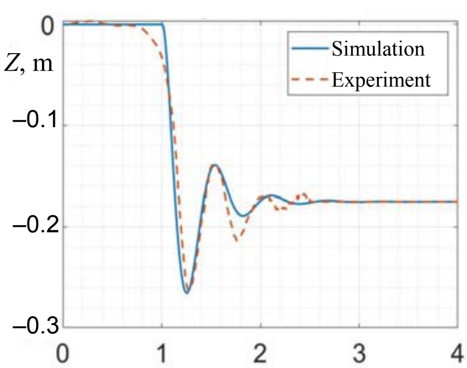

Fig. 8. Comparison of simulated and experimental displacement of the sprung mass in the passive suspension system

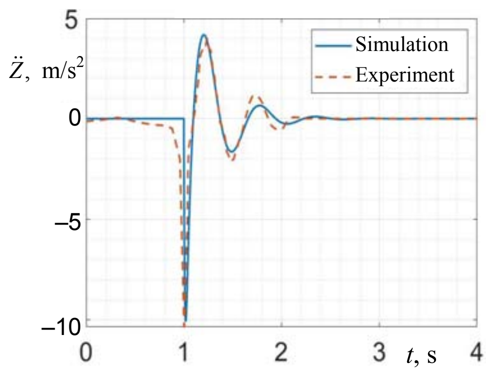

Fig. 9. Comparison of simulated and experimental vehicle body acceleration in the passive suspension system

4. Comparison and evaluation of active and passive suspension systems

After conducting comparisons to validate the simulation model, the paper proceeded to input experimental parameters into the active suspension system model. Simultaneously, the study determined the parameters for the PID controller. The results have indicated that the constructed active suspension system significantly optimizes vehicle body motion. Fig. 10 depicts the displacement of the vehicle body using active and passive suspension systems when passing through a 17.5 cm high step input. A_i represents the i^{th} peak of the vehicle body displacement, corresponding to time t_i. Amplitude values of vehicle body oscillation are shown in Table 1.

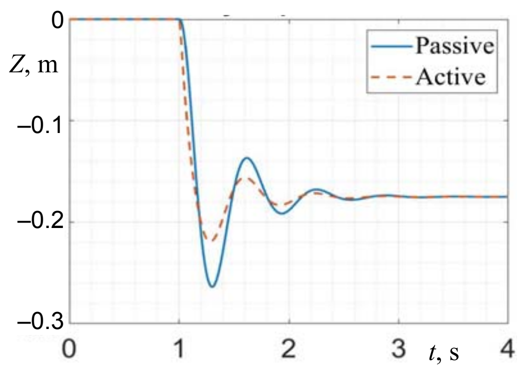

Fig. 10. Vehicle body displacement

Through simulation, it is observed that the active suspension system reduces oscillation amplitude by up to 16.7 %, particularly in the two largest oscillations. The damping time of oscillations also significantly improves, achieving 0.92 seconds compared to 1.63 seconds for the passive suspension system. From the above results, it is also possible to calculate the damping ratios, which are determined using the logarithmic decrement method, for the passive and active suspension systems. The received values – 0.260 and 0.245, respectively show the damping ratio reduction compared to the passive suspension system 5.8 %. This indicates that the active suspension system also enhances the vehicle body's oscillation-damping capability. Therefore, this demonstrates that the active suspension system restores the vehicle to a stable operating state more quickly compared to the passive suspension system.

Table 1

Amplitude values of vehicle body oscillation

i	Az_{i_active} (mm)	$A_{i\_passive}$ (mm)	Reduction ratio compared to passive suspension system (%)
1	–219.1	–263.1	16.7
2	–155.9	–136.7	14.0
3	–183.1	–193.0	5.1

Fig. 11 shows the vehicle body acceleration when passing through a step input, comparing active and passive suspension systems. From the comparison Table 2, it is evident that the simulation model of the active suspension system greatly reduces vehicle body acceleration compared to the passive system. Particularly, for the first two maximum accelerations, vehicle body acceleration decreases by 45.2 and 77.8 %, respectively, compared to the passive suspension system. In conclusion, the received results show that the average reduction ratio of vehicle body acceleration is 61.5 % in comparison to the passive suspension system.

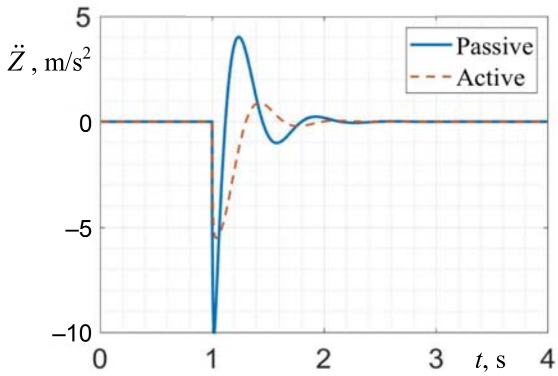

Fig. 11. Vehicle body acceleration

Table 2

Acceleration values of vehicle body oscillation

i	a_{i_active} (m/s^2)	$a_{i\_passive}$ (m/s^2)	Reduction ratio compared to passive suspension system (%)	Average reduction ratio compared to passive suspension system (%)
1	–5.49	–10.02	45.2	61.5
2	0.89	4.01	77.8	

CONCLUSION

In summary, the paper has successfully constructed dynamic models of both semi-passive and semi-active suspension systems. Using computer simulation software, the paper also simulated semi-passive with semi-active suspension systems, and validated the models against experimental results, thereby demonstrating the effectiveness of the active suspension system on trucks. In general, simulation results showed an error of less than 10 % across all three criteria: oscillation amplitude, oscillation acceleration, and damping time. Specifically, the high precision of the developed suspension model was confirmed at the pick points of the vehicle oscillation graph. This study has substantiated and evaluated the specific effectiveness of the active suspension system with high precision due to actual experimentation. The PID controller exhibited rapid computation and control capabilities, meeting the real-time demands of the suspension system. Moreover, the PID controller significantly improved the damping ratio (5.8 %), average acceleration (61.5 %), and maximum amplitude (16.7 %) compared to the passive suspension system. From this research, further developments could involve whole-vehicle simulations, replacement of PID controllers with other control systems, and standard stimulation for wheel response.

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