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Design of Automotive Active Suspension System and Simulation for Intelligent Control Strategy

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Abstract. This article focuses on the main suspension systems of current automobiles and the problems existing in active suspension systems. Firstly, the working principle of electromagnetic active suspension systems is explained, and the system design of active suspension is completed. In order to facilitate analysis, a mathematical model of active suspension is built. Then, under the same road parameters, the model is simulated and verified to verify the improvement of structural performance. Secondly, for the control strategy of vehicle stability, a PID controller is adopted, with vehicle acceleration, suspension travel, and tire dynamic load as the input objectives of the controller. The input parameters are automatically tuned and optimized using a bee colony optimization algorithm, and a mathematical expression for the optimization objective is designed. Finally, the optimal solution of the optimization objective is obtained through simulation. Through the method presented in this article, a better design and control scheme for active suspension systems can be obtained, and the practical results have strong guiding significance.

Keywords: active suspension, PID control, random road surface model, controller

1 Introduction

The suspension system is a mechanical system that connects the vehicle body and tires, and its main function is to provide support, cushioning, and stability during driving. According to the different structures and working principles, car suspensions can be divided into MacPherson suspension, double A-arm suspension, multi link suspension, rigid beam suspension, torsion bar suspension, air suspension, magneto rheological suspension, etc. From the perspective of ground adaptation of suspension, it can be divided into passive suspension, semi-active suspension, and active suspension. Passive suspension is widely used in the household sedan market due to its mature technology and low cost. With the development of technology and the continuous improvement of comfort requirements for passenger cars, as well as the continuous improvement of technology, active suspension is increasingly mentioned and paid attention to. With the continuous improvement of chip technology and the continuous improvement of algorithm computing speed, active control suspension is expected to become the main force in the market. Therefore, this article takes automotive active suspension as the research object, elaborates on a new suspension design scheme, and proposes targeted active control strategies. Therefore, the work done in this article is as follows:

1) By analyzing the principles of existing active suspension, the system design of active suspension was completed, and a simplified model of the system design was provided. At the same time, for the convenience of analysis, a mathematical model of active suspension was built.

2) An optimization objective model for active suspension was constructed to prepare for further optimization, and reasonable weighting coefficients were used in the optimization model.

3) Obtain the optimal values of each parameter through simulation experiments.

In order to clearly describe the work done in this article, the main chapters of this chapter are as follows:

Chapter 2 presents the research results of relevant scholars, Chapter 3 mainly introduces the modeling process of active suspension, Chapter 4 discusses the process of establishing the objective function, Chapter 5 presents the simulation experiment process to find the optimal solution for the structure and parameters of active suspension, and Chapter 6 is the conclusion section.

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2 Related Work

China is a major country in the automotive industry, especially in recent years when national policies have been tilted. Various automobile manufacturers have made significant progress in key automotive technologies, and research on active suspension is also increasing.

Huijie Zhang established a semi-active suspension dynamic model and road input model for the entire vehicle, and used a feedforward control strategy to adjust the damping force of the semi-active suspension shock absorber. Through comparative analysis with the passive suspension system, the results showed that the feedforward control method can effectively reduce vehicle acceleration, improve the performance of the suspension system in adapting to the road, and improve the smoothness of the vehicle's driving [1].

Yufei Liu, using hybrid greenhouse control as a reference model, combined sliding film variable structure control and adaptive control, created a self adaptive inverse sliding film controller using the design concept of inversion, and obtained gradually stable sliding film motion using Lyapunov function. Simulation experiments have shown that the suspension system has a more obvious damping effect [2].

Changshu Zhan designed a particle swarm optimization algorithm to optimize the PID control parameters for active suspension in automobiles. After optimization, the PID control improved the smoothness and handling stability of the vehicle, while also solving the problem of PID controller parameter tuning [3].

Lin Wang utilized the low energy consumption and fast response characteristics of electromagnetic drive to design a different type of active suspension system, and then used finite element analysis software to verify key components, ultimately improving the practicality and safety of the suspension [4].

Farong Kou, the research object is the working parameters of the electromagnetic linear suspension actuator. Using multi-objective particle swarm optimization algorithm, the correctness of the finite element model was verified by comparing the theoretical calculation of radial magnetic flux density with finite element simulation values. The simulation results show that when the structural parameters of the electromagnetic actuator are optimized, the effective electromagnetic force, total harmonic distortion, and electromagnetic wave force are all improved, and the damping effect is significantly enhanced [5].

Ke Ding proposed an active suspension system using a cylindrical transverse flux linear motor actuator. To verify the performance of the suspension, a road excitation model was built, and Matlab/Simulink was used for simulation to obtain the active force design goal of the active suspension actuator. The results showed that the design goal was basically consistent with the electromagnetic force values obtained from Maxwell simulation, and the performance of the active suspension was verified [6].

ChongChong Li discussed a semi-active suspension system with an air chamber, which adjusts the suspension stiffness by controlling the amount of air in the air chamber. Based on the establishment of a suspension mathematical model, a vehicle smoothness random road input driving test was conducted. The test and simulation data were obtained through root mean square values, and the controller simulation was solved using PID control in the MATLAB/Simulink toolbox. The experimental results effectively improved the stability of the vehicle's center of mass and seat in the mid to low frequency range (0.5-12.5) Hz [7].

Haohan Zhang from Yantai University's research focuses on active suspension and proposes an optimal control strategy based on online iterative algorithms. He addresses the difficulties in solving the HJB equation and constructs a novel strategy for online learning of approximate solutions to the HJB equation using the Actor Critic framework of reinforcement learning, while improving the Robustness of the system [8].

Changbo Lin from Dongfeng Motor Group has designed a new type of semi-active suspension system to analyze its stability,Using the acceleration damping control strategy in Matlab, the acceleration damping control not only reduces the acceleration of the spring mass, but also causes deterioration of the suspension dynamic stroke. Based on the acceleration damping control, a continuously variable damping acceleration damping control is proposed to avoid this deterioration result [9].

The above scholars have provided good ideas for the structural design, control, and simulation experiments of semi-active suspension and active suspension. Among the research results of many scholars, this article has designed a novel active suspension, and establishing an accurate mathematical model is crucial, which determines the optimization strategy of subsequent control optimization algorithms and the reliability of simulation results.

3 Active Suspension System Design

The electromagnetic active suspension system is a new type of electromagnetic suspension system that integrates the electromagnetic active suspension actuator into the traditional shock absorber on the basis of the traditional suspension structure. It can not only suppress vehicle vibration, improve driving smoothness and ride comfort, but also recover and utilize the energy dissipated by the spring. This chapter will explain the working principle of the electromagnetic active suspension system, complete the system design of the active suspension, and build a mathematical model of the system suspension [10].

3.1 Active Suspension Model

The driving mode of active suspension is closer to wheel edge drive because each individual wheel is equivalent to an independent system. Due to the addition of control system and control structure, the vertical negative effect problem of the car is caused. By re matching the suspension parameters of the vehicle, some improvement can be achieved. However, parameter adjustment and matching involve issues related to the overall vehicle handling stability, and cannot fundamentally balance the requirements of handling stability and comfort. Therefore, the premise of designing the suspension configuration in this article is to improve the smoothness of the vehicle by designing the wheel damping mechanism while retaining the passive suspension system.

Based on the principle of “dynamic absorber”, the connection between the wheel drive and the axle (wheel) is designed as an elastic damping system, which allows the wheel drive and the vehicle body to have vertical motion relative to the wheel. By adjusting the parameters of the elastic damping system, the wheel drive reduces the vibration transmitted to the vehicle body. At the same time, to ensure the effective transmission of motor power to the wheels, a flexible transmission mechanism is designed between the rotor of the hub motor and the wheels, so that the motor rotates at the same angular speed as the wheels with relative vertical displacement. In addition, a new type of linear electromagnetic active suspension actuator inside the wheel has been designed, which can provide electromagnetic damping force for the suspension system and convert vibration energy into electrical energy [11].

Therefore, the specific design concept of the system structure of the electromagnetic active suspension is based on the transmission MacPherson suspension. On the basis of the MacPherson suspension, an electromagnetic actuator is added to the design, which is parallel to the spring and damper. When the car is driving on the road, the actuator can provide active force to control the vehicle’s driving smoothness and body posture. The active control mode is the mode of controlling the vehicle’s vibration by feedback control of various physical quantities based on the measured signals such as body acceleration and suspension travel during the car’s driving process. When a car is driving on the road, the signals generated by displacement sensors, acceleration sensors, gyroscopes, and other sensors fixed on the body position are converted by A/D and sent to DSP [12]. After being controlled, the signals are converted by D/A and amplified into current signals and input into the electromagnetic active suspension actuator. The energized coil of this actuator generates active force, which can suppress body vibration and attenuate discomfort caused by uneven road surfaces. The working principle diagram is shown in Fig. 1.

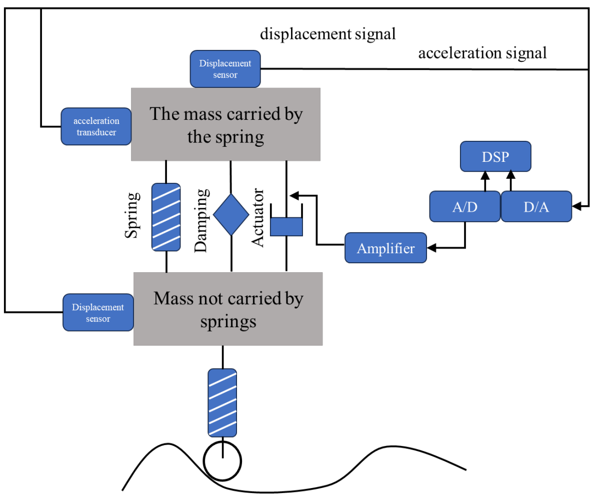


Fig. 1. Schematic diagram of working principle

3.2 Modeling of Quarter Electromagnetic Active Suspension System

When only considering the vertical vibration intensity of the vehicle, the quarter suspension of the car is simplified as a two degree of freedom linear model for analysis. The mechanical analysis model diagram of the quarter active linear suspension of the car is shown in Fig. 2, where the tires and lower support below the suspension are simplified as mass blocks with springs and dampers [13]. The reference direction for the forces generated by springs, dampers, and actuators is also chosen to be upward as the positive direction. For the simplified vehicle suspension system model, we need to propose the following four assumptions:

(1) The unevenness of the road surface when the left and right wheels pass through is completely the same;
(2) Simplify the human body into a rigid body with a fixed mass;
(3) The filtering effect of the human body is negligible;
(4) Neglecting tire damping;
(5) The conversion coefficient for rotational mass is close to "1."

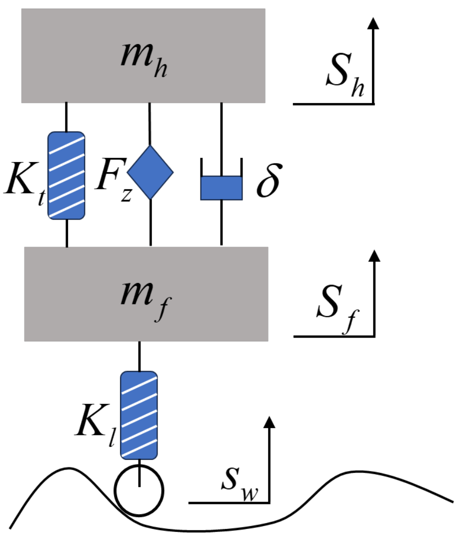


Fig. 2. Mechanical model

Table 1. Parameter list

Parameter	Parameter symbols
The mass carried by the spring	m_h
Mass not supported by springs	m_f
The displacement of the mass carried by the spring	S_h
The displacement of the Mass not supported by springs	S_f
Spring stiffness	K_t
Tire stiffness	K_l
Active control force	F_z

The differential equation of a 2-degree-of-freedom 1/4 vehicle vibration model established using Newton's law is shown in Equation 1:

$$\begin{cases} m_h\ddot{S}_h + \delta\left(\ddot{S}_h - \ddot{S}_f\right) + K_t\left(S_h - S_f\right) - F_z = 0 \\ m_f\ddot{S}_f - \delta\left(\ddot{S}_h - \ddot{S}_f\right) - K_t\left(S_h - S_f\right) + K_l\left(S_f - s_w\right) + F_z = 0 \end{cases}. \tag{1}$$

Represent each parameter using a state vector, and the results are as follows:

$$X=\left[\dot{S}_h,\dot{S}_f,S_h,S_f,s_w\right]. \tag{2}$$

The input of the system is the active suspension control force F_z, and the center of mass vertical acceleration, suspension dynamic stroke, and tire dynamic displacement are selected as the system output vectors, represented as follows:

$$Y=\left[\dot{S}_h,S_h-S_f,S_f-s_w\right]. \tag{3}$$

The vector relationship of the entire system is represented as:

$$\begin{cases}\dot{X}=\alpha(t)X+\beta(t)\omega(t)+\rho(t)F_z\\ Y=\phi(t)X+\varphi(t)F_z\end{cases}. \tag{4}$$

$$\alpha(t)=\begin{bmatrix}-\frac{c_k}{M_h} & \frac{c_k}{M_h} & -\frac{K_t}{M_h} & \frac{K_t}{M_h} & 0\\ \frac{c_k}{M_f} & -\frac{c_k}{M_f} & \frac{K_t}{M_f} & -\frac{K_t+K_l}{M_h} & \frac{K_l}{M_h}\\ 1 & 0 & 0 & 0 & 0\\ 0 & 1 & 0 & 0 & 0\\ 0 & 0 & 0 & 0 & -2\pi f_0\end{bmatrix}. \tag{5}$$

$$\beta(t)=\begin{bmatrix}0\\0\\0\\0\\2\pi\sqrt{f_q(n_0)v}\end{bmatrix}. \tag{6}$$

$$\rho(t)=\begin{bmatrix}\frac{1}{M_h}\\-\frac{1}{M_f}\\0\\0\\0\end{bmatrix}. \tag{7}$$

$$\phi(t)=\begin{bmatrix}-\frac{c_k}{M_h} & \frac{c_k}{M_h} & -\frac{K_t}{M_h} & \frac{K_t}{M_h} & 0\\ 0 & 0 & 1 & -1 & 0\\ 0 & 0 & 0 & 1 & -1\\ 0 & 0 & 0 & 0 & 0\end{bmatrix}. \tag{8}$$

In this model, a random road signal $\omega(t)$ is used as the external interference input for the suspension system, and the step signal can be replaced with other types of road interference input signals according to research needs. The active suspension model includes two speed sensors installed at the vehicle body and wheel positions to detect the vertical runout speed of the vehicle body and wheels. During simulation, the sensor output data can be adjusted.

3.3 Establishment of Random Road Input Model

Road surface excitation causes vibration in the suspension system, which may lead to a series of issues in comfort, handling stability, and safety [14]. This article uses the filtered white noise method to obtain road surface time-domain signals. The road signal model is shown in Fig. 3.

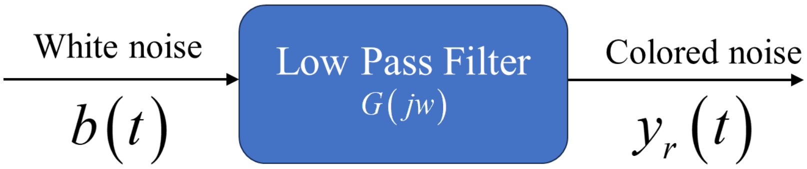


Fig. 3. Road model

The relationship between the road surface spatial frequency domain model A and the road surface temporal frequency domain model B is expressed as:

$$f_q(n) = \frac{f_q(t)}{v}. \tag{9}$$

Among them, A is the driving speed of the car. When the frequency is 0, the road power spectrum is infinite, which will cause distortion of the output result. Therefore, the road cutoff frequency B is set, and C. Therefore, the expression is expressed as:

$$f_q(t) = f_q(n_0)\frac{v}{t^2 + f_0{}^2}. \tag{10}$$

The time threshold model of the road surface, i.e. the input signal of the road surface, is obtained by inverse Laplace transform of the above equation, represented by $\omega(t)$.

$$\omega(t) = -2\pi f_0 y_r(t) + 2\pi b(t)\sqrt{f_0(n_0)v}. \tag{11}$$

3.4 Analysis of the Rationality of Suspension System Structure

In order to verify the rationality of the suspension system and actuator design in this article, the suspension configuration dynamics model is written in the form of a state equation and simulated in MATLAB/Simulink [15]. Select B-level road surface, with random road surface and convex hull road surface as excitation, and set the random road speed to 100 km/h; The vehicle speed on convex roads is set to 30 km/h. The convex hull road input model selects a simplified model that approximates speed bumps, which can be expressed by trigonometric functions:

$$q_t = h_t\left[1-\cos\left(2\pi\frac{v}{l_t}\right)\right]/2. \tag{12}$$

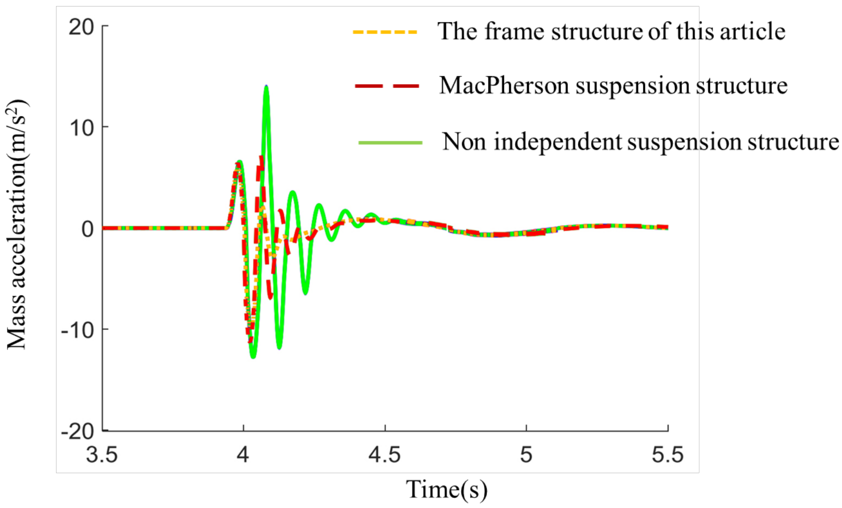
q_t -Input of convex road surface;

h_t -The height of the convex road surface;

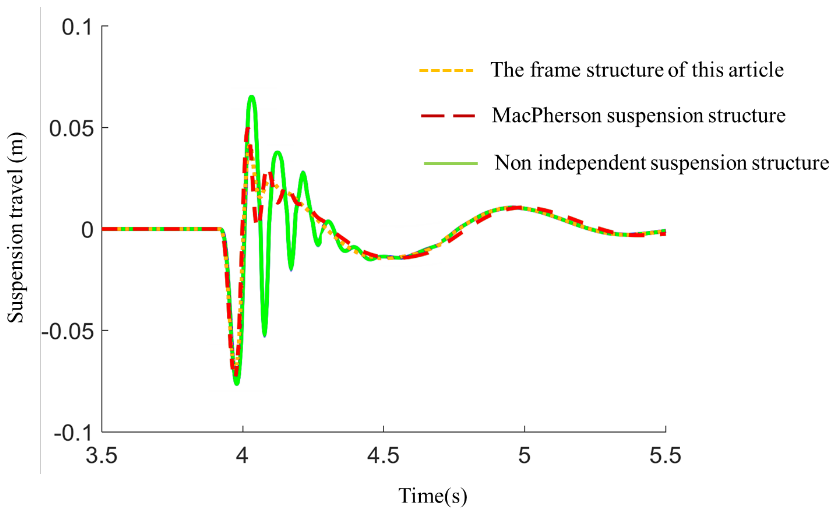
v -Car driving speed;

l_t -The width of the convex road surface.

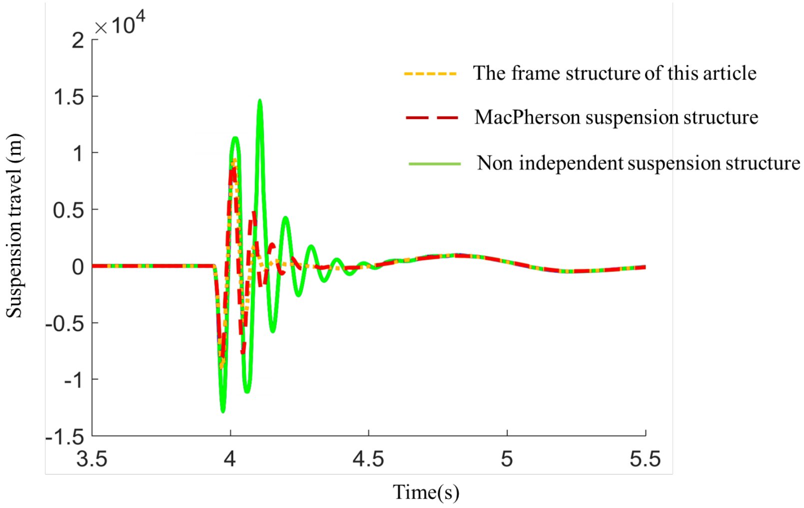
This article sets $L_t = 1.73m$ and $h = 0.27m$, and the simulation results are shown in Fig. 4. In order to demonstrate the advantages of the active suspension designed in this article, the simulation results of non independent suspension, MacPherson suspension, and the improved active suspension in this article will be compared.



(a) Spring loaded mass acceleration under convex road surface



(b) Suspension travel under convex road surface



(c) Dynamic load of vehicle wheels on convex road surface

Fig. 4. Simulation results of different frame structures

From the graph, it can be seen that the active suspension structure designed in this paper has a better damping effect on the acceleration response parameters of the vehicle when driving on random road surfaces. In terms of suspension travel, the three are not significantly different. In terms of wheel dynamic load, the MacPherson suspension is optimal, but it is not much different from the suspension designed in this article. A comparative analysis of the three parameters of vehicle dynamics performance under convex hull road excitation reveals that active suspension is more effective in suppressing the negative effects of wheel drive vehicle vertical vibration caused by an increase in non spring loaded mass.

4 Design of Active Suspension Controller

The body acceleration, suspension travel, and tire dynamic load are the most important performance evaluation indicators for suspension, therefore this section serves as the output of the suspension system. The performance between body acceleration, suspension travel, and tire dynamic load cannot be improved simultaneously. For example, high body acceleration naturally leads to a decrease in comfort, while excessive suspension travel can frequently hit limit blocks, resulting in a decrease in comfort. Therefore, the main research content of this section is how to select the optimal results for each output quantity [16]. Weighting is applied to each parameter, and the weighting function is selected based on the following indicators:

1) Highlight the importance of each suspension evaluation indicator according to the requirements;
2) Normalize the system;
3) Balanced control of the relative importance of various evaluation indicators;
4) To avoid exceeding the limit range of a certain evaluation indicator and not achieving reasonable output.

4.1 Analysis of PID Control Mechanism

The control principle diagram of the PID [17] controller is shown in Fig. 5. The comprehensive output quantity is used to represent that the task of designing a PID controller is to ensure that each object outputted through the input feedback controller has stability and meets certain practical conditions or determined performance indicators. The comprehensive output is represented by z, and the task of designing a PID controller is to ensure that each object outputted through the input feedback controller is stable and meets certain practical conditions or performance indicators. To achieve optimal performance of electromagnetic active suspension, it is necessary to weight the controlled output A and the control input q, using a performance weighting matrix $t(s)$. Considering the output saturation of the active force, it is necessary to prevent the actuator from generating too much active force and control actions that cannot be achieved by the suspension damper. The weighting coefficient $t(s)$ of the selected actuator's control current is:

$$t_1(s) = \frac{60s + 490}{s^2 + 60s + 1200}. \quad \textbf{(13)}$$

The most sensitive frequency range for human body acceleration is 4-8Hz. Therefore, the weighting function of vehicle acceleration can be selected from the "fatigue reduction efficiency limit" curve given by this international standard for the human body. The weighting function $t_2(s)$ of vehicle acceleration is selected as:

$$t_2(s) = \frac{s}{s^2 + 50s + 2286}. \quad \textbf{(14)}$$

The other weighting coefficients mainly depend on normalization and the relative weights of each control output. The suspension dynamic stroke weighting coefficients are expressed as:

$$t\ (s) = \frac{3.2s + 16}{s + 80}. \quad \textbf{(15)}$$

Therefore, the output expression of the entire active suspension is:

$$z(s) = \left[t_1(s) \cdot I + t_2(s) \cdot \ddot{S}_h + t_3(s)(S_h - s_w) + k_l(s_w - S_f)\right]. \quad \textbf{(16)}$$

The control algorithm of PID controller is represented as:

$$u(t) = k_p\left[e(t) + \frac{1}{T_i}\int_0^t e(t)\,dt + T_d\frac{de(t)}{dt}\right]. \quad \textbf{(17)}$$

In the formula, $e\ (t)$ and $u\ (t)$ represent the system deviation input and control output, respectively; k_p represents the proportional constant; T_i represents the integration time constant; T_d represents the differential time constant. Its transfer function form is as follows:

$$G(s) = \frac{U(s)}{E(s)} = k_p\left(1 + \frac{1}{T_I s} + T_d s\right). \quad \textbf{(18)}$$

Automatic control systems are implemented through computers, where sampling control is commonly used. Unlike continuous control systems, the characteristic of sampling control systems is discrete pulses or digital signals. In this article, the signal needs to be discretized, which is a digital PID controller. The mathematical expression obtained is:

$$u(t) = k_p\left\{e(t) + \frac{T}{T_I}\sum_{j=0}^{K} e(j) + \frac{T_d}{T}\left[e(k) - e(k-1)\right]\right\}. \quad \textbf{(19)}$$

In the formula, k represents the sampling number (k = 0, 1, 2, 3...); T represents the sampling period; $e\ (k-1)$ and $e\ (k)$ represent the deviation signals at time k and time $k-1$, respectively.

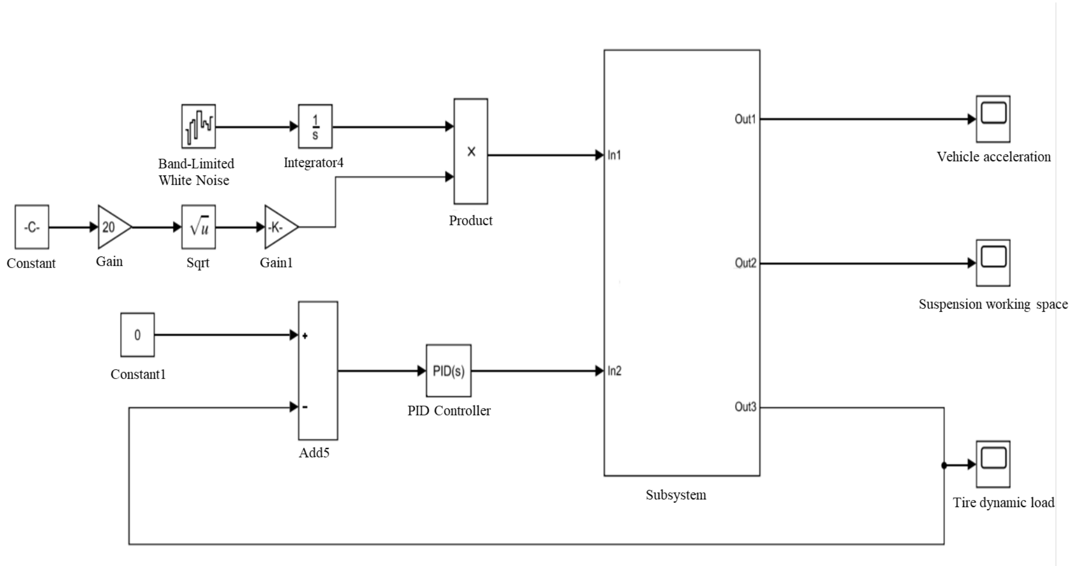


Fig. 5. Schematic diagram of PID controller principle

4.2 Improving the Use and Simulation of PID Controllers

The control object of this article is a two degree of freedom 1/4 suspension system, with body acceleration, suspension dynamic stroke, and tire dynamic load as output responses. In the PID control algorithm, P, I, and D correspond to lead correction, lag correction, and lag lead correction respectively during the signal conversion process. The selection of the three parameters k_p, k_i, k_d is crucial. The function of the proportional term is to accelerate the system response, but there may be steady-state errors. The meaning of the integral term is to eliminate steady-state errors, and in the case of discretization, the differential term is the difference between this moment and the previous moment, which plays a role in reducing system oscillations.

Therefore, when tuning the three parameters of the PID controller, empirical values or table lookup can be used, as well as system reality [18].

Realize online adjustment of changes in the output response curve. In order to demonstrate the scientificity of the intelligent tuning method proposed in this article, a trial and error method was used to preliminarily tune the three PID parameters, and the output results were simulated and displayed in Matlab.

The improved artificial bee colony algorithm performs better in avoiding local optima, solving accuracy, and computing speed. Applying it to the parameter tuning of the active suspension PID controller can output the active control force faster and more accurately. The optimization variables of the bee colony algorithm are k_p, k_i, k_d. The vehicle acceleration, suspension travel, and tire dynamic load are selected as evaluation indicators for suspension performance. Therefore, the objective function must include these three variables. The units and magnitudes of these three indicators are different. Therefore, by dividing the root mean square value of the active suspension corresponding to these three indicators by the root mean square value of the passive suspension, and then summing them up, the objective function and fitness function can be obtained, represented as follows:

$$\min Q(x) = \frac{A}{ASS} + \frac{B}{BWB} + \frac{C}{CTL}. \quad \textbf{(20)}$$

$$f = \frac{1}{Q(x)}. \quad \textbf{(21)}$$

A, *B*, *C* respectively represent the root mean square values of the vertical acceleration of the active suspension body, suspension travel, and tire dynamic load, while the corresponding denominator values represent the root mean square values of the passive suspension indicators. The simulation code is as follows:

Algorithm.

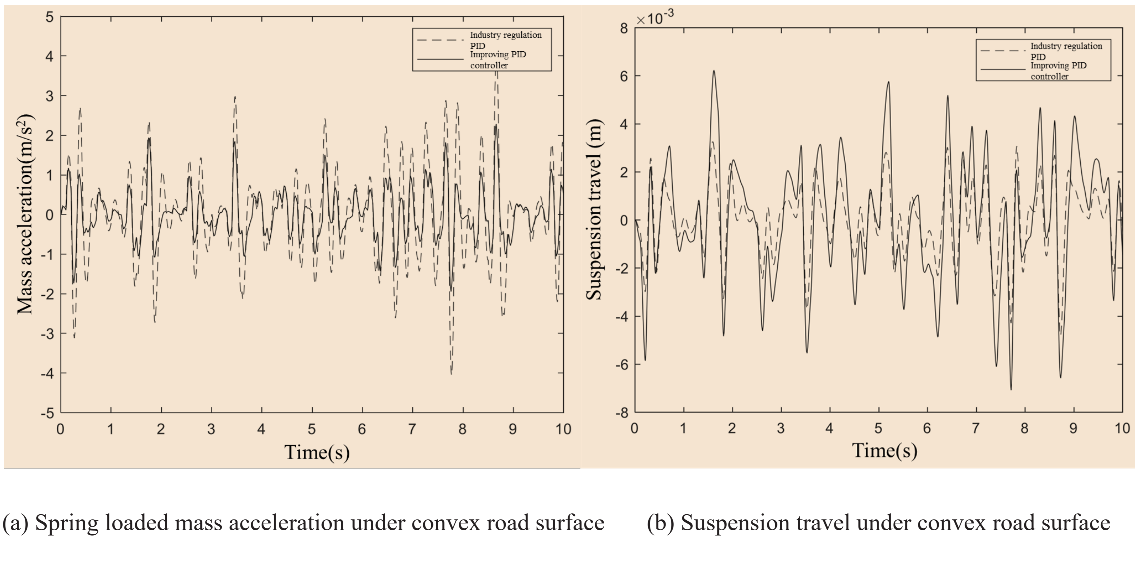
```
Start
    function f = objectiveF(x)
    assignin("base","kp",x(1));assignin("base","ki",x(2));assign-
in("base","kd",x(3));
    sim("PassiveSuspension",[0,11]);sim("PIDACSuspen-
sion",[0,11]);
    A = sqrt(mean(acc.signals.values.^2));B = sqrt(mean(sws.signals.
values.^2)); …
    ASS = sqrt(mean(pacc.signals.values.^2));
    SWS = sqrt(mean(psws.signals.values.^2)); …
    f = A/ASS+B/BWB+C/CTL;
    end.
```

The improved bee colony algorithm optimizes the parameters of the active suspension PID controller in the following steps:

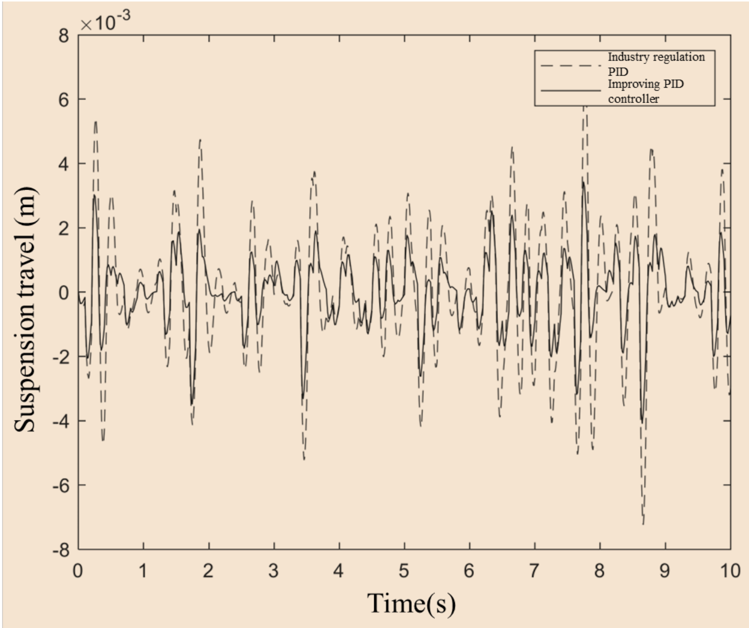
(1) Algorithm begins. Set the dimension of the variable to 5 and the value range to [0500]; Initialization parameters: maximum iteration count is 200, number of honey sources=number of leading bees=100, reconnaissance bee threshold is determined by the upper and lower limits of variable values, and acceleration coefficient is set to 1; (2) Start population iteration. In the leading bee stage, assign the randomly generated honey source to A, run a 1/4 suspension model, obtain the objective function value corresponding to each possible solution, perform greedy comparison, and choose the best over the worst; Observe bees selecting honey sources with higher returns based on improved selection strategies, perform the same operation, and perform certain boundary treatments; When the reconnaissance bee threshold is reached, the reconnaissance bee search begins;

(3) According to the established fitness value function, calculate the fitness value of each honey source. If it meets the requirements or reaches the maximum number of iterations, the algorithm ends; Otherwise, return to the second step.

The output result is shown in Fig. 6.



(a) Spring loaded mass acceleration under convex road surface (b) Suspension travel under convex road surface

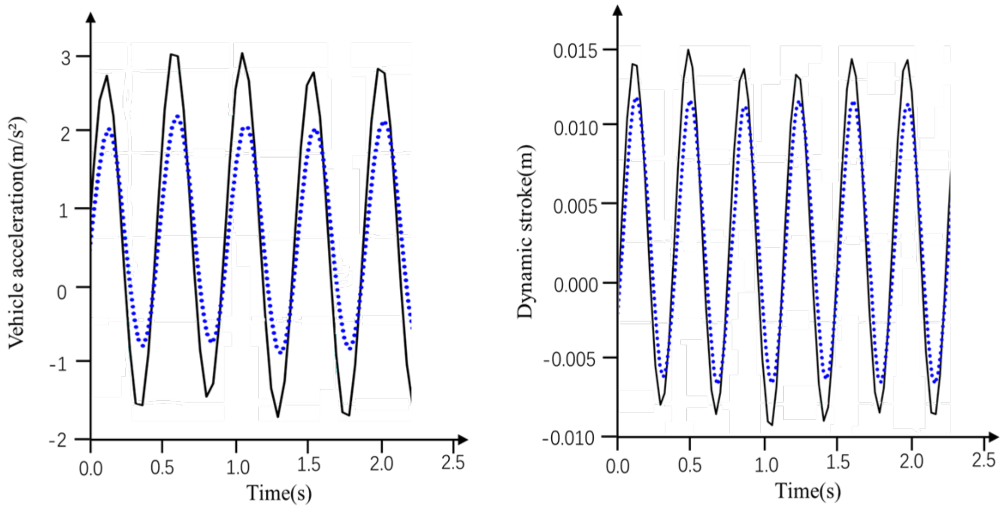


(c) Dynamic load of vehicle wheels on convex road surface

Fig. 6. Improved PID controller simulation results

5 Simulation of Active Suspension Control Strategy

The amplitude of the random road input signal is set to 0.01m and 6Hz, and the dynamic simulation and analysis of PID control of the magnetic active suspension under random road signals are obtained. The simulation results are shown in Fig. 7. The blue dotted line represents the state curve before PID control, and the black solid line represents the state curve after PID control.



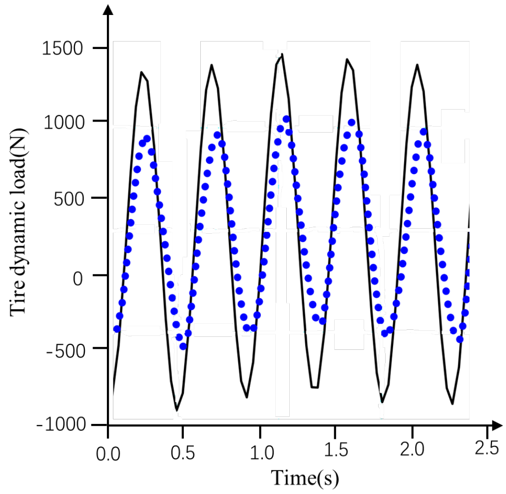


Fig. 7. PID control dynamics simulation results

The comparison object is the passive suspension system, and the analysis of the comparison results shows that the body acceleration of the active suspension is reduced by 0.90m/s2 compared to the value before active control. The dynamic stroke of the active suspension has decreased by 0.0031m compared to the peak value before active control. The peak value of tire dynamic load decreased by about 293.426N compared to before active control. The comparison of body acceleration, suspension travel, and tire dynamic load between before PID control and electromagnetic active suspension based on PID controller is shown in Fig. 8. The electromagnetic active suspension system based on a PID controller significantly improves the acceleration of the vehicle under the excitation of a sinusoidal input signal, effectively improving the driving smoothness of the vehicle and enhancing the PIDness of the suspension system.

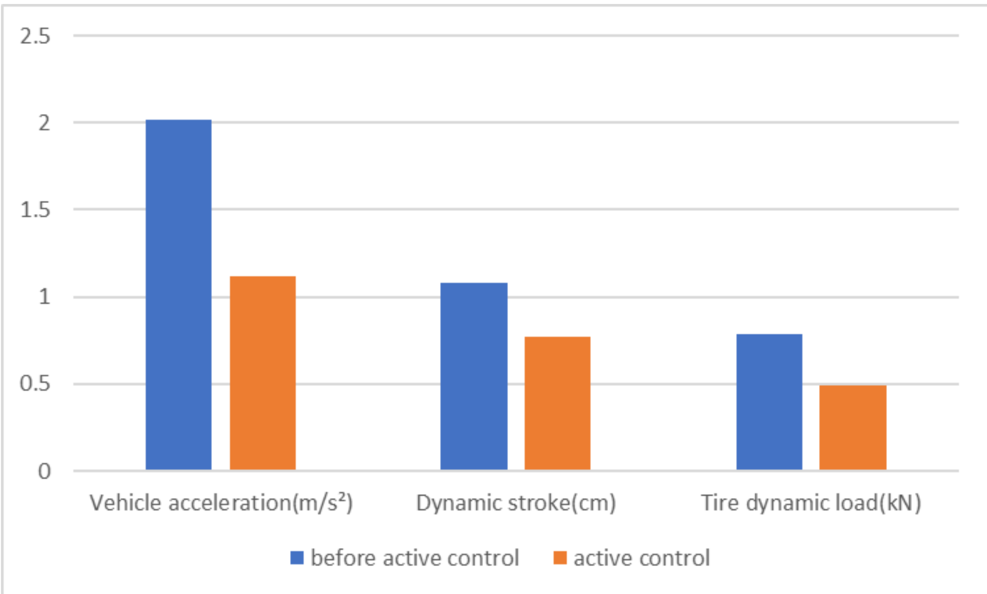


Fig. 8. Comparison of main parameter results after active control

6 Conclusion

This article analyzes the principles of existing active suspension and completes the system design of active suspension. A simplified model of the system design is provided. In order to facilitate analysis, a mathematical model of active suspension is constructed. Based on the mathematical model, an optimization target model of active suspension is constructed to prepare for further optimization. Reasonable weighting coefficients are used in the optimization model. Then, the optimal values for each objective were found through simulation, and through comparison, all parameters were optimized compared to before active control.

1) In the design phase of active suspension, the innovation of active suspension and the design of actuators were described in detail, and a suspension system model and road input model were established. The model was used as input and simulation was used to verify the rationality of the new active suspension design.

2) This article uses a PID controller, but as the parameters of the controller, there is an optimal coordination between the vertical acceleration of the vehicle body, suspension dynamic stroke, and tire dynamic load. The conventional method uses trial and error to obtain the parameters. This article uses a bee colony algorithm to solve for the optimal parameters and the weight of the optimal solution.

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