ORIGINAL ARTICLE

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Performance Analysis of Giant Magnetostrictive Actuator and Its Application on Active Suspension of Train

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https://doi.org/10.1590/1679-78256688

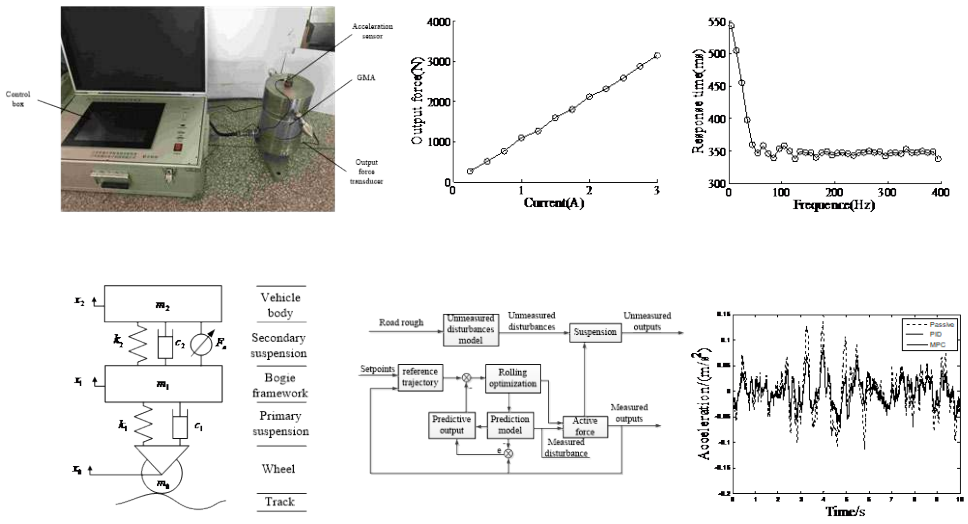
Abstract

Active suspension is considered to be a good way to improve the ride comfort of high-speed trains. According to the output index requirements of train's active suspension, a giant magnetostrictive actuator(GMA) is proposed. This is mainly because giant magnetostrictive materials(GMM) has the characteristics of fast response, large output force and high energy conversion rate. It is verified by experiments that the output force is proportional to the excitation current. It is found in the experiment that the excitation frequency should be greater than 120Hz to obtain a stable output force, and it is also found that preload and excitation frequency will affect response time. On the basis of experiments, a 2-DOF physical and mathematical model of the vertical quarter train is built. An MPC algorithm is designed to control GMA active suspension. Through simulation analysis, the proposed control algorithm is compared with passive suspension and active suspension based on PID control algorithm. Both theory and practice show that the proposed control algorithm is effective.

Keywords

train, active suspension, GMA, response time, MPC algorithm

Graphical Abstract



Received August 04, 2021. In revised form August 11, 2021. Accepted August 11, 2021. Available online August 19, 2021.
https://doi.org/10.1590/1679-78256688

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1 INTRODUCTION

Active suspension system is an active control system which can reduce the vibration by using dynamic force. Firstly, the vibration state information of the vehicle in the running process is obtained in real time by various sensors installed on the bogie and car body. The controller uses the measured vibration information, obtains the optimal control amount of vibration reduction through the corresponding control algorithm, and controls the output force of the actuator to reduce the vehicle vibration (Alessandro et al. 2010). Due to the obvious advantages of active suspension system in bogie safety performance, vehicle vibration damping effect and train's running stability performance, more attention should be paid to the design of high-speed train bogie. However, the active suspension system consumes a lot of energy when the output force is used for vibration reduction (Liu et al. 2021), and the general actuator is difficult to meet the application occasions of the train's suspension system, which greatly limits the development of active suspension for trains. As the core actuator of active suspension system, actuator plays a very important role in the development of active suspension. Due to the continuous development of technology in the material field, the application of actuators using new functional materials such as GMM has been expanding in other industrial fields (Kim and Lee, 2020). GMA have some advantages that other actuators cannot match, such as high electric-mechanical energy conversion efficiency, fast response speed, large output power, etc., which can meet the performance requirements of the suspension system. The combination of GMA and active suspension system to train's bogie is a new direction of research on active suspension system, which is an effective method to reduce vehicle vibration and improve the safety, stability and comfort of high-speed train's operation.

GMA is a displacement actuator designed by using GMM to transform electromagnetic energy into mechanical energy. In 2001, Bartlett P.A. et al. (2001) made a GMA with amplification mechanism by using Terfenol-D material, whose output displacement can reach 4mm and can be used for low-frequency vibration control. In 2017, Wang and Thomas (2017) used GMM to design a ultra-thin control module. They also adopted a multi-channel active control which can achieve significant noise reduction performance in a short time. In 2018, Mori K. et al. (2018) designed an energy collecting device made of GMM, and developed two kinds of portable vibration isolation mass blocks using thin Terfenol-D layer. Yafang et al. (2021) designed a new GMT, and determined through experiments that the transducer had high magnetic and mechanical energy conversion efficiency at high frequency. Zhang et al. (2019) designed an active vibration isolation method using GMM. Combined with GMA, a lot of feedback is used in this control method. Wang and Zhu (2019) built an experimental platform and analyzed the effects of different skeleton materials, different material processing methods and different prestresses on GMA output performance. Tian et al. (2021) designed a GMA with a macro displacement of 30mm and a response time of less than 0.2s. Guoping et al. (2020) designed a stacked GMA to study the heat dissipation, output response and output accuracy of the actuator. To Kato et al. (2018) verified the noise control effect of GMA on small cars through experiments. The researches of these scholars have made great progress in GMA. However, there are few studies on the combination of GMA and active suspension of high speed trains.

At present, the active suspension technology based on GMA is still in the theoretical research stage. In this paper, the specific performance requirements of the active suspension for the actuator and some practical problems to be considered when the actuator is used in the active suspension are comprehensively considered. A GMA is designed for active suspension of train. According to the designed actuator model, the output characteristics and response characteristics of the actuator are analyzed through experiments. Finally, the active suspension's dynamic model is built, and the controller of suspension system is studied and simulated. The research work in this paper can provide reference for the design of new active suspension system, and has important practical significance for the promotion and application of GMA in the field of railway vehicles. In view of the wide application prospect of GMA, and at present, the research of this kind of actuators in the active suspension of vehicles is few. Therefore, it is very important to study the theory and application of GMA applied to the vibration control of train's active suspension.

2 DESIGN AND PERFORMANCE ANALYSIS OF GMA

To design the GMA used in the vibration damping system of train's active suspension, it is necessary to make clear the requirements of the output parameters of the damper of active suspension system. The design of the actuator is based on the actual needs of the train so that it can be used in active suspension of the train. The requirements of the active train's suspension on the output parameters of the actuator are shown in Table 1. Firstly, the selection and geometric parameters of the GMM bar are determined according to the output displacement and operating force requirements, and then the subsequent design of other components is completed according to the selected GMM bar parameters.

Table 1. Output parameter requirements of train's actuator.

Parameter names	Values
Output displacement	2000μm
Output force	20kN
Maximum vibration frequency	400Hz
Frequency control accuracy	±3%
Energy conversion efficiency	>50%

The actuator was also designed with a special displacement amplification mechanism, which is not introduced here. According to the obtained parameters, the product specifications of manufacturers were compared, and a certain margin was reserved. Finally, the geometric parameters of GMM bar were determined to be 200mm long and 40mm diameter. According to the experimental conditions and current processing conditions, we produced a reduced version of the GMA for theoretical research. The specific size is shown in Table 2.

Table 2. Parameters of the actuator.

Parameter names	Values
Length of GMM rod(mm)	100
Diameter of GMM rod(mm)	20
GMM density(Kg/m^3)	9.2×10^3
GMM rod mass(kg)	0.29
Turns number of drive coil	1600

The energy of the giant magnetostrictive actuator comes from the alternating magnetic field generated by the drive coil. The design of the magnetic circuit directly affects the efficiency of the actuator. In addition, efficient and reasonable magnetic circuit design can reduce the volume of the actuator and improve the performance of the actuator. Therefore, the magnetic circuit design is the core part of the actuator design. Combined with the characteristics of the actuator, the specific parameters of the internal magnetic circuit of the actuator were determined through theoretical analysis, and then the design parameters of the drive coil were calculated.

Due to the relatively low permeability of GMM, the magnetic circuit of the actuator is usually designed as a closed magnetic circuit in order to reduce the magnetic leakage of the actuator and to improve the efficiency of the drive coil. The magnetic circuit consists of a magnetic yoke composed of the upper and lower guide blocks at both ends of the GMM and the upper and lower guide magnets, a drive coil, a permanent magnet, an air gap and a GMM bar. There is an air gap of 2mm between the upper guide block and the upper magnet to transfer the output displacement of the GMM bar, while the lower guide block is directly connected with the lower magnet with a transition fit. According to the characteristics of magnetic materials, the yoke is made of metal soft magnetic material DT3 with relative permeability up to about 20000. The permanent magnet uses NdFeb magnet, which has the strongest magnetic force at room temperature so far, and its relative permeability is about 1.05, which belongs to the same order of magnitude as GMM. Figure 1 shows the internal magnetic circuit structure of the actuator. The magnetic field passing through the GMM bar consists of two parts, one part is the bias magnetic field provided by the permanent magnet, the other part is the driving magnetic field provided by the drive coil.

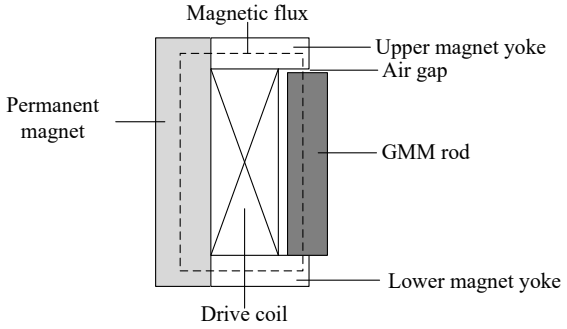


Figure 1. Actuator internal magnetic circuit structure.

Figure 2 shows the physical picture of GMA and measurement and control system, including GMA, acceleration sensor, output force sensor and control box. In the process of testing the output force, the excitation frequency varies in the range of 1~400Hz, which can be fixed frequency or sweep frequency. The excitation current ranges from 1 to 30A. In order for GMA to have a good giant magnetostrictive effect, it is necessary to provide appropriate preloading force. The preload force is generated by the elastic deformation of the preload spring, and the deformation is regulated by the bolt. Adjusting bolt adopts fine thread to improve accuracy, and the pitch is 1mm. Given that the actual stiffness of the preload spring is 3.6×10^5N/m, the preload force corresponding to each turn of the thread is 360N.



Figure 2. Physical picture of GMA test system.

The application of GMA in train's active suspension mainly needs to consider three performances. The first is the magnitude and linearity of the output force. The second is the response time of the actuator. The third is the output displacement. This paper mainly discusses the output force and response time. The output displacement is achieved by a special displacement amplification mechanism which is not described here.

When GMA is applied to the active vibration reduction system of trains, the first consideration is the stability of the output force. Figure 3 shows the relationship between excitation current and output force. At this time, the excitation frequency is 150Hz. With the increase of current, the output force gradually increases, and maintains good linearity, which is consistent with the designed simulation results.

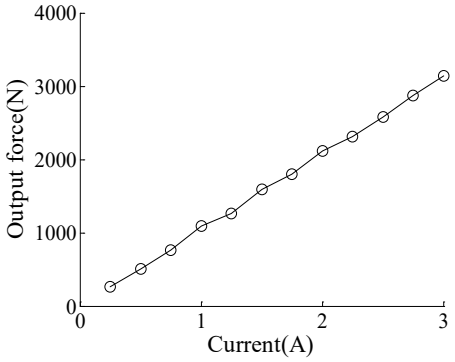


Figure 3. Relationship between excitation current and output force.

Because of the compressive stress characteristics of giant magnetostrictive materials, it is necessary to set a precompressive stress application mechanism in the design of actuators. Applying prepressure to the GMM rod can improve the magnetostrictive effect and the mechanical efficiency of the actuator. Prepressure mechanism has many forms, common hydraulic, pneumatic, prepressure spring and so on. The important parameters of prepressure mechanism are stiffness constant and adjustable pressure range. In order to reduce the influence of the preloading mechanism on the output of the actuator, the preloading spring with small stiffness constant is generally selected. Considering the influence of various factors comprehensively, this paper adopts spiral spring as preloading spring, which not only meets the condition of small stiffness constant, but also can ensure that the prepressure will not change when the actuator output displacement. At the same time, the size of preloading pressure can be adjusted with adjusting nut.

Figure 4 shows the influence of GMA's excitation frequency on output force, and the excitation current is 0.5A. In our previous theoretical research work, it could be known that the GMA's output force was not related to the magnitude of the frequency (Wang et al. 2021), but in the process of our experiments, we found that this was not the case. When the excitation frequency was less than 120Hz, the GMA's output force would decrease with the increase of the excitation frequency. At the same time, the output force fluctuates greatly. When the excitation frequency was greater than 120Hz, the output force tended to a stable value of about 518N. Similar characteristics were observed when the excitation current was selected for other values.

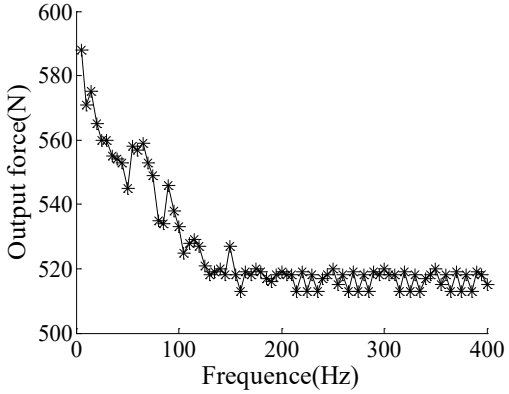


Figure 4. Influence of excitation frequency on output force.

When GMA is applied to active suspension, in addition to the stability of the output force, another important factor to be considered is the response time. The response time of the actuator will affect the time delay of the controller, which will affect the overall performance of the train's active suspension. According to our previous theoretical study (Wang et al. 2021), the theoretical response time was only 1.25ms, but it was almost impossible to achieve such a short response time in the actual experimental verification process, since the response time was also related to some external factors. Figure 5 shows the preload's influence on response time of the GMA. The excitation current is 1A and the excitation frequency is 200Hz. It is not difficult to see from the figure that when the preload is 1000N, the response time is about 550ms, when the preload increases to 2800N, the response time will gradually decrease to 350ms. However, when the preload further increases, the response time will actually increase.

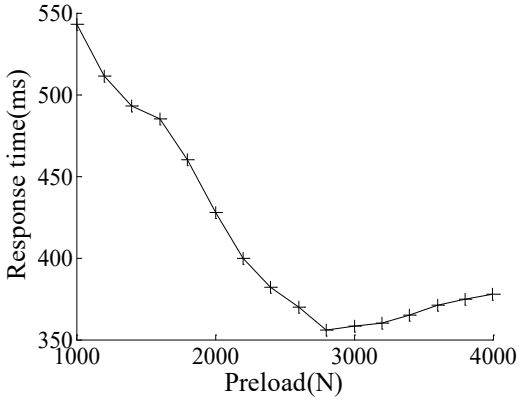


Figure 5. Relationship between preload and response time.

In the course of our experiment, we also found that the response time was related to the excitation frequency in addition to the preload. Figure 6 shows the relationship between excitation frequency and response time, and the excitation current is 1A. When the excitation frequency was less than 50Hz, the response time increases. However, when the frequency is greater than 50HZ, the excitation frequency has almost no effect on the response time.

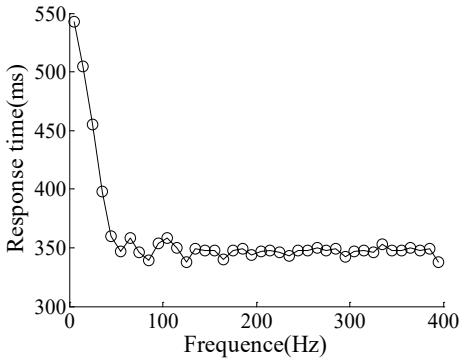


Figure 6. Relationship between excitation frequency and response time.

3 APPLICATION OF GMA IN ACTIVE SUSPENSION

The train's active suspension is mainly composed of external energy input, actuator, measurement sensor system and feedback control system, which is a closed-loop feedback control system. The real-time running state of the train can be obtained by measuring the sensor system, and the controller can get the appropriate suspension control parameters through algorithm calculation. The actuator can output the control force under the designed parameters of the controller, so that the train can achieve the best vibration suppression effect under different running conditions. At present, the models used to study the dynamic performance of the train's suspension system mainly include 1/4 vehicle 2-DOF model, 1/2 vehicle 4-DOF model and the whole vehicle 7-DOF model. Referring to the most commonly used classical model of vehicle suspension control system and based on train's system dynamics, this paper simplified the active suspension model of GMA system to a 1/4 vehicle two-degree-of-freedom dynamics model, which has fewer parameters and can well reflect the basic characteristics of vehicle operation. The main body of the model is composed of wheelset, bogie frame and car body.

A parallel active suspension with an actuator on the basis of passive suspension is adopted in this paper. Its characteristic is that when the actuator fails, it can still work in the passive suspension mode, which strengthens the safety of the train's running. The vertical two-degree-of-freedom physical model of the train is shown in Figure 7. The system is mainly composed of mechanical and GMA control system. Specifically, there are primary suspension, secondary suspension, wheel, track, bogie frame, GMA, vehicle body and various kinds of sensors. Sensors measure body acceleration, bogie frame dynamic displacement and other signals and feed back to the controller. The calculated control current is input to the drive coil, and then the output force of GMA is controlled to slow down the vibration of the vehicle body. The specific parameter values are shown in Table 3.

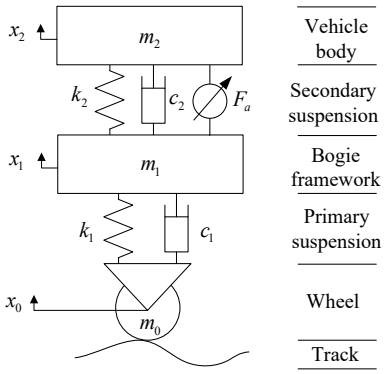


Figure 7. Vertical active vibration reduction model of 1/4 train's car.

There are many nonlinear factors in the train's vertical active vibration reduction system. It is almost impossible to analyze and model every nonlinear factor. Therefore, for the sake of research and calculation, most components of the suspension system are considered to be linear. The 2-DOF 1/4 vehicle model is built, and Newton's second law was applied to the dynamic analysis of the suspension, and the differential equation of its motion was obtained:

$$m_2\ddot{x}_2 = k_2(x_1 - x_2) + c_2(\dot{x}_1 - \dot{x}_2) + F_a \tag{1}$$

$$m_1\ddot{x}_1 = -k_2(x_1 - x_2) - c_2(\dot{x}_1 - \dot{x}_2) + k_1(x_0 - x_1) + c_1(\dot{x}_0 - \dot{x}_1) - F_a \tag{2}$$

Where, m_2 is the mass of a quarter of the train's body. m_1 is the bogie framework mass; c_2 is the damping coefficient of the secondary suspension. c_1 is the damping coefficient of the primary suspension. k_2 is the elastic stiffness coefficient of the secondary suspension. k_1 is the elastic stiffness coefficient of the primary suspension. x_2 is the vertical displacement of the vehicle body. x_1 is the vertical displacement of the frame. x_0 is the excitation of orbital irregularity input. F_a is the active control force generated by GMA.

White noise is selected as the track irregularity input. If the wheels are in contact with the track at all times, then the road input can be expressed as:

$$\dot{x}_0(t) = -2\pi f_0 x_0(t) + 2\pi\sqrt{S_q(n_0)v}\omega(t) \tag{3}$$

Where, $x_0(t)$ is the track irregularity, f_0 is the cutoff frequency, $S_q(n_0)$ is the track irregularity coefficient, v is the speed, and $\omega(t)$ is white noise.

The dynamic displacement of the frame $x_{21} = x_2 - x_1$, the vertical velocity of the car body, the dynamic displacement of the wheel $x_{01} = x_0 - x_1$, the vertical velocity of the frame are selected as the input variable, that is $X = [x_{21}, \dot{x}_2, x_{01}, \dot{x}_1]^T$. Output variables are vehicle body's acceleration, frame dynamic displacement and wheel dynamic load, that is $Y = [\ddot{x}_2, x_2 - x_1, k_1(x_0 - x_1) + c_1(\dot{x}_0 - \dot{x}_1)]^T$. Then the state equation matrix of the system is:

$$\begin{aligned} \dot{X} &= AX + BU \\ Y &= CX + DU \end{aligned} \tag{4}$$

Where, $A = \begin{bmatrix} 0 & 1 & 0 & -1 \\ -k_2/m_2 & -c_2/m_2 & 0 & c_2/m_2 \\ 0 & 0 & 0 & -1 \\ k_2/m_1 & c_2/m_1 & k_1/m_1 & -(c_2+c_1)/m_1 \end{bmatrix}$, $B = \begin{bmatrix} 0 & 0 \\ 1/m_2 & 0 \\ 0 & 1 \\ -1/m_1 & c_1/m_1 \end{bmatrix}$, $U = \begin{bmatrix} F_a \\ \dot{x}_0 \end{bmatrix}$, $D = \begin{bmatrix} 1/m_2 & 0 \\ 0 & 0 \\ 0 & c_1 \end{bmatrix}$, $C = \begin{bmatrix} -k_2/m_2 & -c_2/m_2 & 0 & c_2/m_2 \\ 1 & 0 & 0 & 0 \\ 0 & 0 & k_1 & -c_1 \end{bmatrix}$.

Before designing the controller, the control process of GMA train active suspension system should be clarified. According to the working principle of the actuator and the structure of the active suspension system, the GMA train active suspension system relies on the GMA output to achieve vibration reduction. As the center of the whole system, the controller receives the vehicle status information collected by sensors and generates expected power control signals. GMA drive unit is the drive power supply according to the control signal output corresponding drive current, drive GMA work. At the same time, the actual GMA output acts as dynamic feedback to the controller to adjust the control signal of the controller. If the GMA output dynamic is closer to the expected value, it indicates that the design of the system controller is more accurate and the vibration reduction effect of the active suspension system is better.

The appropriate inputs and outputs should be selected to the controller. In this paper, the vertical acceleration of the car body and the displacement difference between the car body and the bogie frame, which can reflect the ride comfort, are selected as the input variables of the controller. The power of GMA output changes with the adjustment of

drive current, and the drive current is determined by the control current output by the controller, so the control current is selected as the output variable of the controller.

Since the train runs on a fixed track, track irregularity is known to some extent. In addition, the train's active suspension system is also a multiple input multiple output (MIMO) system, so the model predictive control (MPC) method is suitable. In the theory of predictive control, it is necessary to have a basic model describing the dynamic behavior of the system, which is called the predictive model. It has the function of prediction, that is, it can predict the future output value of the system according to the historical data and future input of the system (Nikesh et al. 2021, Peng et al. 2021). The components of GMA active suspension predictive control system mainly include predictive model, rolling optimization, online correction, reference trajectory and controlled object, as shown in Figure 8.

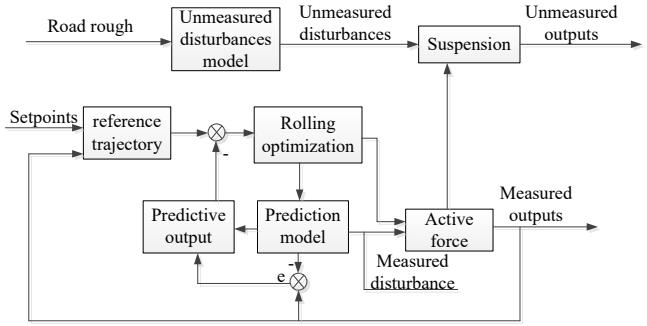


Figure 8. GMA active suspension's control block diagram.

Model predictive control (MPC) algorithm is produced directly from the practical application of industrial process control and is continuously improved in the close combination with industrial application. Since the predictive control adopts rolling optimal control, the predictive control model must be discrete. Therefore, the equation of state in Equation (4) should be discretized. The difference equation can be obtained by combining the characteristics of MPC theory with the actual running condition of the vehicle:

$$x(k+1) = Ax(k) + B_u u(k) + B_d d(k) \tag{5}$$

$$y_u(k) = C_u x(k) + D_{uu} u(k) + D_{du} d(k) \tag{6}$$

$$y_m(k) = C_m x(k) + D_{dm} d(k) \tag{7}$$

Where, $y_u(k) = [k_1(x_1 - x_0) + c_1(\dot{x}_1 - \dot{x}_0)]^{\mathrm{T}}$ is the unmeasurable output of the system. $y_m(k) = [\ddot{x}_2, x_2 - x_1]^{\mathrm{T}}$ is the measurable output. $u(k) = [F_a]^{\mathrm{T}}$ is the control variable. $d(k) = [\dot{x}_0]^{\mathrm{T}}$ is unmeasurable interference.

The purpose of rolling optimization is to determine the future control effect by minimizing a performance index. The optimized performance indicators are:

$$J = \sum_{i=1}^{p} q_i (y(k+i) - \omega(k+i))^2 + \sum_{j=1}^{m} r_j \Delta u^2 (k+j-1) \tag{8}$$

$Q_y = diag(q_1, q_2, \cdots, q_p)$, $R_u = diag(r_1, r_2, \cdots, r_m)$ are the weighted matrix of output variable and control variable respectively. The optimization objective is concerned only with predicting the system's dynamic performance in the time domain. Moreover, only the first term of the optimal control sequence is applied to the controlled process, that is, the control increment at time k is $\Delta u(k) = d^T (W - Y_0)$, where the variable d can be solved offline in advance. After the new measurement value is obtained at the next sampling time, the above optimization problem is solved again to obtain a new control sequence, so as to realize rolling optimization. The rolling optimization constraints are as follows: (1) The space between the body and the frame limits the dynamic displacement $|x_2 - x_1| \le l_{\max}$. (2) In order to ensure the

grounding of the wheel, the dynamic load cannot exceed the static load $\left|k_1(x_1-x_0)+c_1(\dot{x}_1-\dot{x}_0)\right| \le m_1 g + m_2 g$. (3) Considering the limitation of GMA output force, there are constraints on the action force of the suspension $\left|F_a\right| \le F_{\max}$.

Due to the uncertainty of objects and environment, the actual output $y(k+1)$ at time $k+1$ is not equal to the predicted output after the control effect is implemented at time k, which needs to constitute the prediction error:

$$e(k+1) = y(k+1) - \hat{y}(k+1) \tag{9}$$

The prediction of other future moments is corrected by this error weighting, that is:

$$\hat{Y}_p(k+1) = \hat{Y}_p(k) + he(k+1) \tag{10}$$

Where, $\hat{Y}_p = [\tilde{y}(k+1), \tilde{y}(k+2), \cdots, \tilde{y}(k+p)]^T$ is the system output at time $t=(k+1)T$ predicted at time $t=(k+1)T$ after error correction. $h=[h_1, h_2, \cdots, h_p]^T$ is the error correction matrix, $h_1 = 1$.

After correction, $\hat{Y}_p$ is taken as the initial forecast value at the next moment. Since the initial forecast value at time $t=(k+1)T$ is used to predict the output value of $t=(k+2)T, \cdots, (k+p+1)T$ at the future moment, let:

$$y_0(k+i) = \tilde{y}(k+i+1) \quad (i=1,2,\cdots,p-1) \tag{11}$$

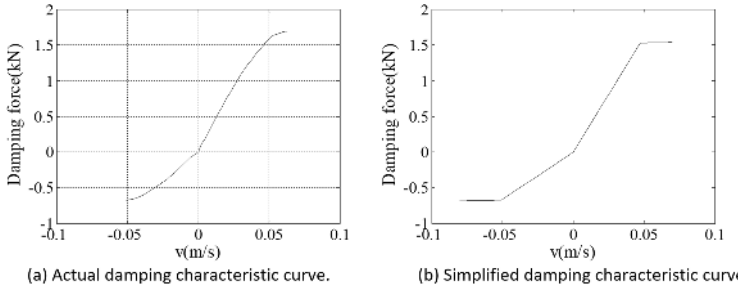
Thus, the initial forecast value at the next moment is:

$$\begin{cases} y_0(k+i) = \hat{y}(k+i+1) + h_{i+1}e(k+1) \\ y_0(k+p) = \hat{y}(k+p) + h_p e(k+1) \end{cases} \tag{12}$$

The introduction of the above modifications makes the system become a closed-loop negative feedback system and improves the system performance.

4 SIMULATION AND ANALYSIS

The suspension system of the train has obvious nonlinear characteristics, and the most important nonlinear components are the oil shock absorbers of primary suspension, whose nonlinear damping characteristics are shown in Figure 9. The damping characteristics of actual primary suspension's oil shock absorber are shown in Figure 9(a). It can also be seen from the figure that there is a big difference between the damping coefficient of the oil pressure shock absorber in the tensile stage and the compression stage. Therefore, we can approximate the damping characteristics of the oil shock absorber to piecewise linear, which is convenient for us to study the dynamic performance of the whole suspension simulation, as is shown in Figure 9(b). In the programming process, if $x1 - x0 \ge 0$, the oil damper damping of the primary suspension adopts tensile damping. If $x1 - x0 < 0$, the oil damper damping of the primary suspensions adopts compression damping.



(a) Actual damping characteristic curve. (b) Simplified damping characteristic curve.

Figure 9. Velocity damping force characteristic curve of oil shock absorber.

We choose a high-speed train in China as the simulation object, and the specific simulation parameters are shown in Table 3. The variable names in the table are shown in Figure 7.

Table 3. Physical parameters of suspension system.

Parameter names	Values
Vehicle body mass/10^4kg	4.2
Bogie mass/10^3kg	2.8
Primary suspension stiffness/(10^5N·m^{-1})	7.5
Primary suspension tensile damping/[10^4(N·s)·m^{-1}]	3.1
Primary suspension compression damping/[10^4(N·s)·m^{-1}]	1.2
Secondary suspension stiffness/(10^5N·m^{-1})	3.2
Secondary suspension damping/[10^4(N·s)·m^{-1}]	5.5

Before constructing the mathematical model of the train suspension system for vibration analysis, it is necessary to build the corresponding road excitation model. As one of the inputs of suspension system, the road excitation model is the content that must be researched and realized in the simulation analysis and bench test system of train's suspension. The vibration characteristics of the suspension system can be obtained by applying certain road excitation input to the suspension system to make it run normally. The establishment of a reasonable and accurate road surface excitation model is of great significance to the dynamics analysis of train's suspension system, ride comfort and road surface adhesion. Track irregularity generated by filtering white noise method, sinusoidal excitation, impact excitation and measured track irregularity were compared. Measured track irregularity was selected as track excitation in this paper, as shown in Figure 10. The simulation speed is set as 55.6m/s.

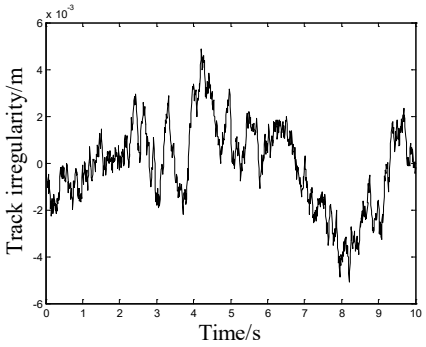


Figure 10. Measured track irregularity of a section of Beijing-Shanghai line.

PID control algorithm is used as the comparison algorithm to verify the effectiveness of the model predictive controller. To facilitate the observation of the output results, the passive suspension, PID control and predictive control of the active suspension output together for analysis. According to the input data provided in Table 3, various parameters of the simulation system are set. The simulation results are shown in Figure 11. Compared with passive suspension, the improvement of vertical acceleration of active suspension is particularly obvious, as is shown in Figure 11(a). As can be seen from Figure 11(b), MPC significantly improves the dynamic displacement of the frame. Because the response curves of wheel dynamic load of the three types of suspension are relatively close, the graph is not obvious, so it is not listed.

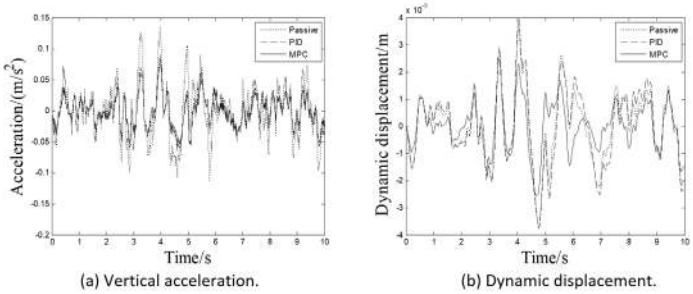


Figure 11. Response diagram of train's suspension with track irregularity input.

The specific improvement value of train's vertical dynamic performance by different suspensions is shown in Table 4. Compared with passive suspension, the active suspension's body acceleration controlled by PID is improved by 40.5%, which is very impressive. The active suspension's acceleration performance controlled by MPC is improved by 9.8% compared with the active suspension controlled by PID. Under actual conditions, the two active suspension control methods can significantly improve the ride comfort of the vehicle. It shows the effectiveness of active suspension in improving ride comfort. MPC suspensions deliver the best ride comfort because the MPC provides real-time, online optimization.

By comparing Figure 11(b) and Table 4, it can be seen that compared with Passive, PID has a deterioration in the dynamic travel of suspension, and the dynamic travel increases by 16.7%. Compared with Passive and PID, MPC reduces the suspension dynamic travel. This is because PID only takes acceleration as feedback information, while MPC takes acceleration and dynamic travel into consideration. Finally, compared with the dynamic load of the wheels, the dynamic load of the three suspensions does not change very much. This is because the actuator is mounted on the secondary suspension and is filtered through the primary suspension so that the load applied to the wheels is already small.

Table 4. Performance comparison of suspension under different controls.

Performance index	Passive suspension	PID control	Improvement over passive suspension	MPC	Improvement compared with PID
Acceleration/($m \cdot s^{-2}$)	0.0427	0.0254	40.5%	0.0229	9.8%
Dynamic displacement/mm	1.2	1.4	-16.7%	0.88	37.1%
Dynamic load/N	869.3	837.2	3.7%	831.5	0.68%

5 CONCLUSIONS

Experiments shown that the output force of the GMA designed by us had a good linear relationship with the excitation current, and the GMA could be applied to the active suspension of the train. However, we found that the output force fluctuated greatly under the influence of small excitation frequency, and the actuator's response time was also large when the excitation frequency was small. To sum up, when GMA was applied to active suspension of train, the excitation frequency should be greater than 120Hz. During the experiment, we also found that the GMA's response time was related to the preload. With the increase of the preload, the response time of the output force firstly decreased and then increased, and the response time was the shortest when the preload was about 2800N. After determining some control performance of GMA, we built a vertical dynamic model of a quarter vehicle's body. A model predictive control (MPC) algorithm was proposed to solve the problem that the oil damper has different equivalent damping coefficients under tension and compression. PID controlled suspension and passive suspension are used as a reference. The research shows that the GMA active suspension with MPC algorithm can improve the ride comfort and suspension dynamic displacement better. At the same time, the dynamic load of the wheel is also slightly improved.

Funding

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This research was funded by the National Natural Science Foundation of China (No. 62063013 and No. 72061021) and Science Research Funding Project of Nanjing Institute of Technology (No. QKJ201713).

Author's Contributions: Conceptualization, SC Dong and JJ Meng; Methodology, SC Dong; Investigation, DC Li and H Song; Writing - original draft, SC Dong, JJ Meng and DC Li; Writing - review & editing, SC Dong; Funding acquisition, JJ Meng; Resources, DC Li and H Song; Supervision, DC Li.

Editor: Marcílio Alves.

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