Comparison Between Simulation- and Test Results of an Observer-Controlled MAGLEV Vehicle on Elastic Guideway

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Abstract

Based on the full state-controller theory, a new observer-controller for MAGLEV vehicles considering the guideway elasticity was proposed in early publications in Germany and now realized in practice in China. It is expected to eliminate the phenomenon of the vehicle-guideway coupling vibration caused by the hypothesis of a rigid guideway, which usually happens in the current high-speed MAGLEV line, and reduce the over dependence of system stability on the guideway properties. A test rig for the elastic guideway levitation system was founded and the effectiveness of the new observer-controller was verified.

The research work was performed in 2010 and 2011 when Prof. Dr.-Ing. Reinhold Meisinger was Advisory Professor in the Urban Mass Transit Railway Institute at Tongji University in Shanghai.

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1. Introduction

As a new kind transportation facility, MAGLEV vehicles are based on the principle of electromagnetic levitation and there is no mechanical contact in running. Compared with the traditional wheel-rail vehicles, the MAGLEV system offers a plenty of advantages, such as energy saving, less risk of derailment, low noise, less maintenance cost, etc. [1,2]. There are broad development prospects for the MAGLEV technology in the fields of the intercity high-speed rail transit and the medium-low-speed urban mass transit. Among the existing MAGLEV levitation systems, the normal conducting EMS with active control technology is a relatively developed technique. And the phenomenon of vehicle-guideway coupling vibration is one of the must-be-resolved problems [3]. Because of the flexibility of the guideway, the interaction happens between the running MAGLEV vehicle and guideway. The vehicle and the guideway may vibrate intensively if the controller is designed less appropriately, which will affect the stable operation of the vehicle. There always happens of the vehicle-guideway coupling vibration on the particular lines for the MAGLEV vehicles of German TR04, KOMET, Japanese HSST04 and American AMT [4,5]. The greater mass and stiffness of the guideway is the common engineering application to avoid the phenomena, but this will result in a substantial increase in the construction cost of the MAGLEV line, and severely limit the development and promotion of the MAGLEV. According to statistics, the completed projects the cost of the guideway beams takes 60%~80% of the total MAGLEV system cost [6].
In the recent decades, a lot of studies on the vehicle-guideway coupling vibration have been carried out and some progresses have been made. In an early computer simulation R. Meisinger [7,8] investigated new time variable observer-controllers for the coupled system of elastic MAGLEV vehicles with continuous magnet distribution running over elastic single- and double-span guideways. In order to improving the system performance, Young Chol Kim, et al. [9] taking the single magnet levitation system as the object, adjusted the feedback gain coefficients according to the disturbance and suspended mass with the gain scheduling approach. J.E. Snyder, et al. [10] obtained the dynamic equations of the vehicle-guideway coupling system by treating the guideway as the simply supported Bernoulli-Euler beam, and the effects on the system dynamic responses of the parameters were investigated, such as the mass radio of vehicle and the beam, levitation stiffness, etc. P.K. Sinha, et al. [11] pointed out that the suitable system damping and the filter in the forward path could restrain the guideway vibration. With the simulation method, Xie Deyun, et al. [12,13] investigated the effects on the system frequency and the stiffness of the elastic beam. Fang Mingxia, et al. [14] developed a controller with the optimal control theory to improve the system stability. The studies mentioned above except [7] do not take directly control for the guideway, therefore they cannot eliminate its vibration fundamentally and the system will be unstable when the stiffness or the damping of the guideway is relatively lower. Liu Hengkun, et al. [15] indicated that the control method with the full states feedback theory might suppress the coupling vibration, but the unavailability of the full system states made it unachievable in practice.
Therefore this paper compares simulation- and test results of a new observer-controller trying to solve the coupling problem. With this control method, it is expected to reduce the over dependence of system stability on the guideway properties and cut the construction cost of the guideway beam in practice.

2. Modeling of the System

The single magnet levitation system is the basic unit of the MAGLEV vehicle and stable levitation of the single magnet is the basis for the system to achieve stable operation [16,17,18,19]. So it is more versatile to study the dynamic characteristics of the single magnet system. Therefore, this paper simplifies the vehicle-guideway coupling system into the elastic guideway single-magnetic levitation model as shown in the Fig.1, and investigates its stability under the different control methods.

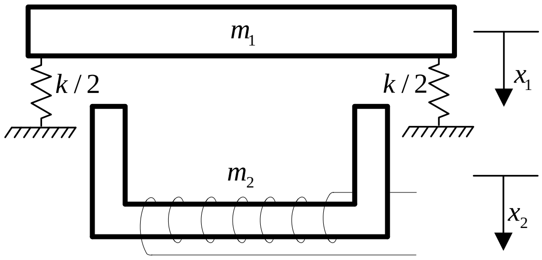

Fig.1 Vehicle-guideway coupling model

With the number of windings N and the pole area A the electromagnetic force F between the magnet and the guideway is [20]

$$F(t) = \frac{\mu_0 N^2 A}{4}\left[\frac{I(t)}{s(t)}\right]^2 . \tag{1}$$

From the Eq.1, the force F is inversely proportional to the levitation gap s, which is the primary cause of the instability for the open-loop MAGLEV system. On the equilibrium levitation point, defining the nominal gap and the nominal current in the magnet coil as $S_{N,}$ I_N respectively, the electromagnetic force F can be linearized and expressed as the following form:

$$F = -P_s\left(x_2 - x_1\right) + P_I I , \tag{2}$$

where P_S, P_I are the gap coefficient and the current coefficient, respectively.
In terms of the Newton's second law, the system dynamics equations can be expressed with guideway mass m_1, magnet mass m_2 and guideway stiffness coefficient k as follows:

$$m_1\ddot{x}_1 = F - kx_1 \tag{3}$$

$$m_2\ddot{x}_2 = -F \tag{4}$$

Referring to the Refs. [7, 19,21], the electrical equation can be derived as

$$\dot{I} = -\frac{R}{L}I + (1-\eta)\frac{P_s}{P_I}(\dot{x}_2 - \dot{x}_1) + \frac{U}{L} \tag{5}$$

where R is the magnet electric resistance, L is the electrical inductance and U is the voltage across the magnet. As introduced in [7] η is the magnetic flux leakage on the equilibrium levitation point.
Substituting the Eq.2 into the Eq.4, the current I can be derived as

$$I = \frac{P_s}{P_I}(x_2 - x_1) - \frac{m_2}{P_I}\ddot{x}_2 \tag{6}$$

The derivation of the Eq.6 is

$$\dddot{x}_2 = \frac{P_s}{m_2}(\dot{x}_2 - \dot{x}_1) - \frac{P_I}{m_2}\dot{I} \tag{7}$$

Taking x_1, $\dot{x}_1$, x_2, $\dot{x}_2$, $\ddot{x}_2$ as the state variables, the magnet vertical acceleration $\ddot{x}_2$ and the levitation gap $x_2 - x_1$ as the outputs of the system, the vehicle-guideway coupling model can be described as the following state-space equation.

$$\begin{aligned}\dot{X} &= AX + BU \\ Y &= CX\end{aligned} \tag{8}$$

$$X = \begin{bmatrix} x_1 & \dot{x}_1 & x_2 & \dot{x}_2 & \ddot{x}_2 \end{bmatrix}^T , \qquad A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ -\frac{\mathrm{k}}{m_1} & 0 & 0 & 0 & -\frac{m_2}{m_1} \\ 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 \\ -\frac{P_s R}{m_2 L} & -\frac{\eta P_s}{m_2} & \frac{P_s R}{m_2 L} & \frac{\eta P_s}{m_2} & -\frac{R}{L} \end{bmatrix},$$

$$B = \begin{bmatrix} 0 & 0 & 0 & 0 & -\frac{P_I}{m_2 L} \end{bmatrix}^T , \qquad C = \begin{bmatrix} -1 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}.$$

3. Test Rig for the Elastic Guideway Single-Magnet Levitation System

A test rig for the elastic guideway single magnetic levitation system is founded in order to verify the effectiveness of the control methods. The test rig is composed of a single magnet levitation table, a power amplifier, dSPACE, a direct current power supply, etc, and the main parameters of the test rig are shown in the Tab.1. For the single magnet levitation table, shown by the Fig.2, the magnet is bolted at one end of the swing arm and the other end of the arm is pined to the foundation, so that the movement of the magnet can be regarded as a linear movement in the vertical direction when arm swiveling within a small angle range. The guideway (the steel plate in the Fig.1) is connected to the foundation via the springs, and it can vibrate vertically. The vibration of the guideway beam in practice is simplified as the guideway vibration. By alternating the different springs, the natural frequency of the guideway can be changed. The gap coefficient P_S, the current coefficient P_I and the magnetic flux leakage η in the Tab.1 are all obtained by the experiments.

Tab. 1 Parameters of the test rig

Physical quantities	values
Mass of the magnet, m_2(equivalent)	1.8*kg*
Mass of the guideway, m_1	0.8*kg*
Electric resistance of the magnet, *R*	12Ω
Electric inductance of the magnet, *L*(gap: 10*mm*)	0.3*H*
Magnetic flux leakage η (gap: 10*mm*)	0.65
Stiffness of the springs *k*	1.0e4*N/m*
Gap coefficient, P_S	3382*N/m*
Current coefficient, P_I	15.75*N/A*
Acceleration sensor	*LC0701-5*
Gap sensor	*LXC-M05P2*
DC power supply	*PS-605D*

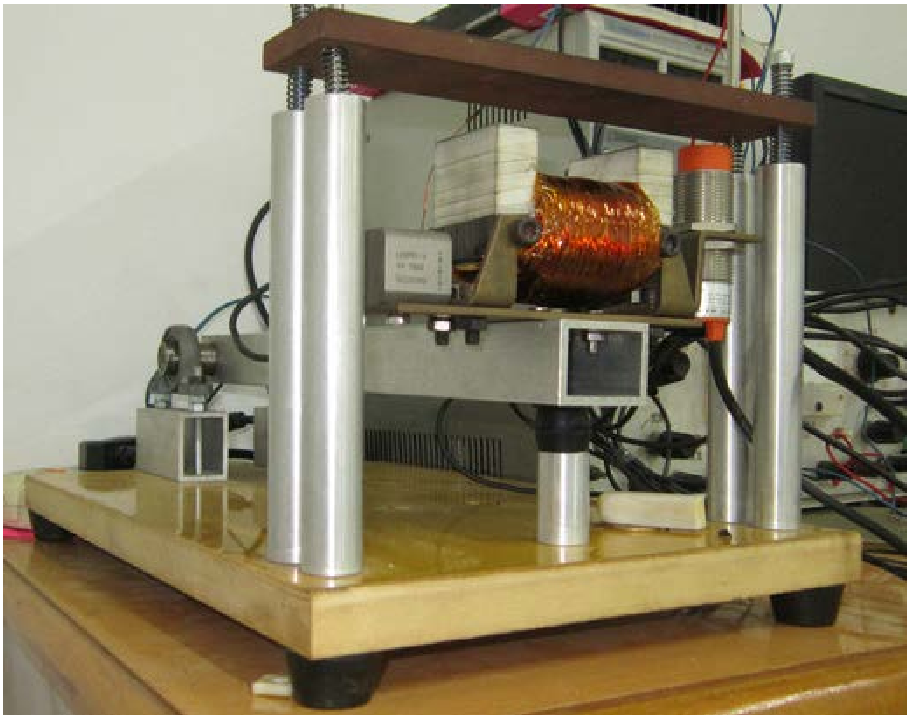

Fig. 2 Test rig for the elastic guideway single-magnet levitation system

4. Controller Design

4.1 State Observer

Feedback control is a general method for most controllers. The feedback information is usually composed of the state variables, but not all the state variables are measurable in physics [22]. However, the system state variables can be obtained by state observer if the system is observable. In the design of the magnetic levitation control system, for the velocity sensor still cannot reach the required measurement accuracy by the present technique, the vertical oscillation velocities of the magnet and the guideway are not available. Therefore, the acceleration sensor and the gap sensor are used to get the magnet acceleration and the air gap as shown in the Tab.1, and the other state variables including the vertical velocity, displacement of the guideway and the magnet are derived by the state observer.
The judgment matrixes of the system controllability and observability can be described respectively as follows:

$$M = \begin{bmatrix} B & AB & A^2B & A^3B & A^4B \end{bmatrix}$$
$$N = \begin{bmatrix} C & CA & CA^2 & CA^3 & CA^4 \end{bmatrix}^T$$

Substituting the related values into the two matrixes M and N, the calculation results show they are all full-ranks matrixes, which means the system described by the Eq.8 is controllable and observable.
The schematic diagram of the observer is shown in the Fig.3, in which *G* is the feedback matrix of the output errors (observer feedback matrix).

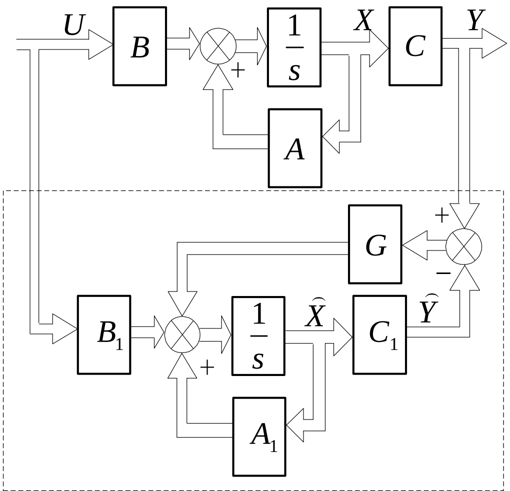

Fig.3 State observer

4.2 Control Method Without Considering the Guideway Vibration

The control method without considering the guideway vibration ignores the vibration of the guideway. So there are only 3 feedback variables x_2, $\dot{x}_2$ and $\ddot{x}_2$ in designing the controller, which stand for the displacement, velocity and acceleration of the magnet. Therefore, the control law can be expressed by the following equation:

$$U = -K\hat{X} = -K_1\hat{x}_2 - K_2\dot{\hat{x}}_2 - K_3\ddot{\hat{x}}_2 \tag{9}$$

where $K = \begin{bmatrix} K_1 & K_2 & K_3 \end{bmatrix}$ and $\hat{X} = \begin{bmatrix} \hat{x}_2 & \dot{\hat{x}}_2 & \ddot{\hat{x}}_2 \end{bmatrix}^T$. *K* is the controller feedback vector and $\hat{X}$ is the state vector obtained by the state observer. So, the matrixes A_1 B_1 C_1 of the state observer shown in the Fig.3 should be written as follows.

$$A_1 = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ \frac{P_s R}{m_2 L} & \eta \frac{P_s}{m_2} & -\frac{R}{L} \end{bmatrix}, \quad B_1 = \begin{bmatrix} 0 \\ 0 \\ -\frac{P_I}{m_2 L} \end{bmatrix}, \quad C_1 = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

The feedback matrix of the output errors G and the controller feedback vector K can be calculated with the method of the Pole Placement or the Linear Quadratic Control method as [23,24,25,26,27]:

$$K = \begin{bmatrix} -4712.2 & -190.0 & -2.43 \end{bmatrix}$$

$$G = \begin{bmatrix} 56.86 & 2286.87 & 91978.6 \\ 0.015 & 0.586 & 23.58 \end{bmatrix}^T$$

Taking x_1, $\dot{x}_1$, x_2, $\dot{x}_2$, $\ddot{x}_2$, $\hat{x}_2$, $\dot{\hat{x}}_2$, $\ddot{\hat{x}}_2$ as the state variables, the closed-loop system matrix is 8-dimensional, and can be written as the following form.

$$A_r = \begin{bmatrix} A & -BK \\ GC & A_1 - GC_1 - B_1 K \end{bmatrix} \tag{10}$$

The matrix A_r is 8-dimensional and the distributions of its characteristic roots have the closed relationships with the dynamic behavior of the system. Substituting the values shown in the Tab.1 into the Eq.10 and making the natural frequency of the guideway changes in the scope of 1*Hz*~100*Hz* by changing the spring stiffness, the characteristic roots of the matrix A_r are calculated. And then, the maximum real parts of the roots in every case are picked up and drawn out in the Fig.4. The figure shows that with the increase of the guideway natural frequency or the stiffness, the maximum real parts decrease and tend to 0, but they are always above 0, which means the system is unstable. Assuming the initial vertical position of the magnet is deviated from the equilibrium point 5*mm* and the natural frequency of the guideway is 17.8*Hz* (same values will be taken in the following time-domain simulation if without special version), the displacement response of the magnet is calculated and the result is shown in the Fig.5. Fig.6 also shows that the control method without considering the guideway vibration cannot make the system stable if the there is damping for the guideway.

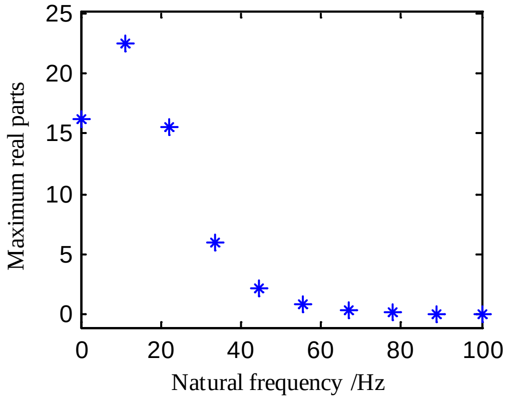

Fig.4 Maximum real parts vs. guideway natural frequency

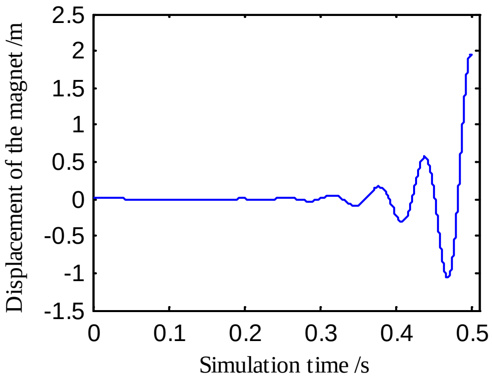

Fig. 5 Displacement of the magnet

The Eq.3 becomes the following form if considering the damping effect of the guideway. Then the matrix A_r should be changed accordingly with the guideway damping coefficient c, but it will not be restarted here for the continuity of the paper:

$$m_1 \ddot{x}_1 = F - kx_1 - c\dot{x}_1 \tag{11}$$

Taking the guideway damping with $c = 10 N \cdot s / m$ into account and assuming guideway natural frequency changes in the scope of 1*Hz*~100*Hz*, the characteristic roots of the matrix A_r are calculated. The maximum real parts of the roots for the different frequencies are shown in the Fig.6. The figure shows that with the increase of the natural frequency, the maximum real parts decrease and become negative when the frequency or the stiffness is above a certain value, which means the system can be stable if the guideway stiffness is hard enough. That is why the guideway beams are built as hard as possible in practice. The time-domain simulation result (for a natural frequency of the guideway of 35Hz) shown in the Fig.7 indicates that it takes about 0.2s for the system to reach the steady state and the maximum overshoot is about 8%.

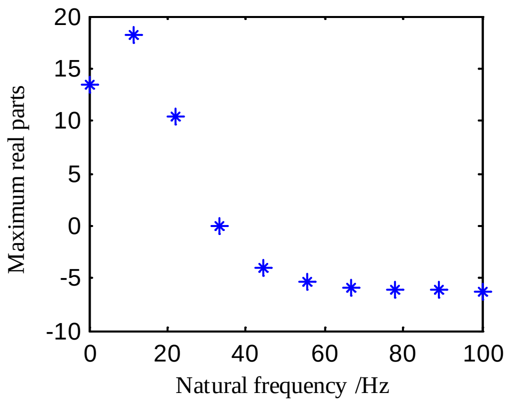

Fig. 6 Maximum real parts vs. guideway natural frequency

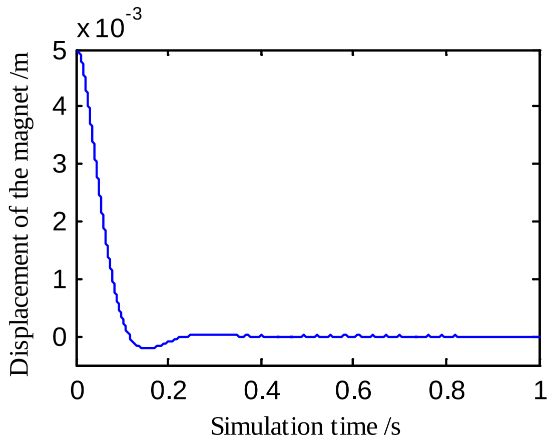

Fig. 7 Displacement of the magnet

4.3 Control Method Considering the Guideway Vibration

From the analysis results above, if there is no guideway damping effect, the control method without considering the guideway vibration cannot maintain the MAGLEV system stable; Although there exists a certain damping for the guideway beam in practice, the stable operation of the MAGLEV vehicles is at the expense of the greater stiffness of the beam and higher construction cost of the MAGLEV line.
The control method above takes the velocity, acceleration of the magnet and the levitation gap as the feedback variables to design the control law. It is easy to see that, among the three variables, only the levitation gap reflects partly the vibration state of the guideway. That is to say, as an important component part of the MAGLEV system, the guideway's vibration information is not fully introduced into the control system. This is why the control method without considering the guideway vibration with the chosen values of *K* and *G* cannot maintain the system stable. For this reason, this paper, using the state observer, introduces the guideway's vibration states (for zero vehicle speed) into the control system, and develops the controller with the full state feedback.
Therefore, there are 5 feedback variables: $\hat{x}_1$, $\dot{\hat{x}}_1$, $\hat{x}_2$, $\dot{\hat{x}}_2$, $\ddot{\hat{x}}_2$, and the control law can be written as follows:

$$U = -K\hat{X} = -K_1\hat{x}_1 - K_2\dot{\hat{x}}_2 - K_3\hat{x}_2 - K_4\dot{\hat{x}}_2 - K_5\ddot{\hat{x}}_2 \tag{12}$$

Where $K = \begin{bmatrix} K_1 & K_2 & K_3 & K_5 & K_5 \end{bmatrix}$ and $\hat{X} = \begin{bmatrix} \hat{x}_1 & \dot{\hat{x}}_2 & \hat{x}_2 & \dot{\hat{x}}_2 & \ddot{\hat{x}}_2 \end{bmatrix}^T$. *K* is the controller feedback vector and $\hat{X}$ is the state vector obtained by the state observer. Taking x_1, $\dot{x}_1$, x_2, $\dot{x}_2$, $\ddot{x}_2$, $\hat{x}_1$, $\dot{\hat{x}}_1$, $\hat{x}_2$, $\dot{\hat{x}}_2$, $\ddot{\hat{x}}_2$ as the state variables, the closed-loop system matrix A_r is 10-dimensional, and can be described as the following form:

$$A_r = \begin{bmatrix} A & -BK \\ GC & A - GC - BK \end{bmatrix} \tag{13}$$

Assuming the natural frequency of the guideway is 17.8 *Hz*, and ignoring the damping of the guideway, the controller feedback vector *K* and the feedback matrix of the output errors *G* are obtained with the Linear Quadratic Control Method as:

$$K = \begin{bmatrix} 1455.9 & 167.5 & -7256.2 & -319.0 & -6.6 \end{bmatrix}$$

$$G = \begin{bmatrix} -42.8 & -3100.5 & 63.0 & 2511.7 & 11218.9 \\ -0.001 & -2.3 & 0.001 & 1.0 & 3122.3 \end{bmatrix}^T$$

And then, the characteristic roots of the matrix A_r are calculated: $\lambda_1 = -3163.86$, $\lambda_2 = -185.92$, $\lambda_{3,4} = -24.94 \pm 111.05i$, $\lambda_{5,6} = -27.57 \pm 110.07i$, $\lambda_{7,8} = -25.39 \pm 42.36i$, $\lambda_{9,10} = -22.37 \pm 21.01i$.

All the real parts of the roots are negative, so the closed-loop system is stable. The time-domain simulation is carried out and the displacement responses of the magnet and the guideway are shown in the Fig.8 and Fig.9. The figures indicate that the system takes about 0.2*s* to reach the steady state and the maximum overshoot is 4%. The simulation results verify the effectiveness of the control method.

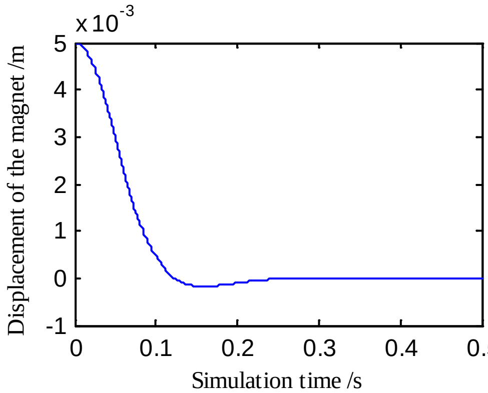

Fig. 8 Displacement of the magnet

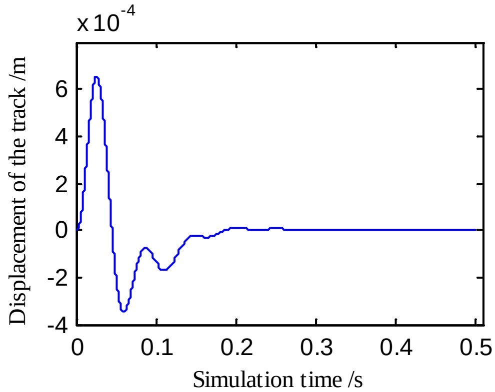

Fig. 9 Displacement of the guideway

4.4 Verification Test of the Control Method on the Test Rig

The verification test of the control method considering the guideway vibration is carried out on the test rig. The spring stiffness is 10000 *N/m*, and the other parameters are shown in the Tab.1. For the limitation of the gap sensor range, the initial gap between the magnet and the guideway is set 13.3*mm.* From the Fig.10, the test result indicates that the system takes about 0.3*s* to reach the steady state and the maximum overshoot is 0.2*mm*. Compared with the test result, the simulation result shows the settling time is about 0.2*s*, and the over-shot is a little bigger. Considering the simplification of the mathematical model and the parameter errors of the test rig, the simulation result and the test result are coincident with each other, which means the simulation model is correct and the control method is effective.

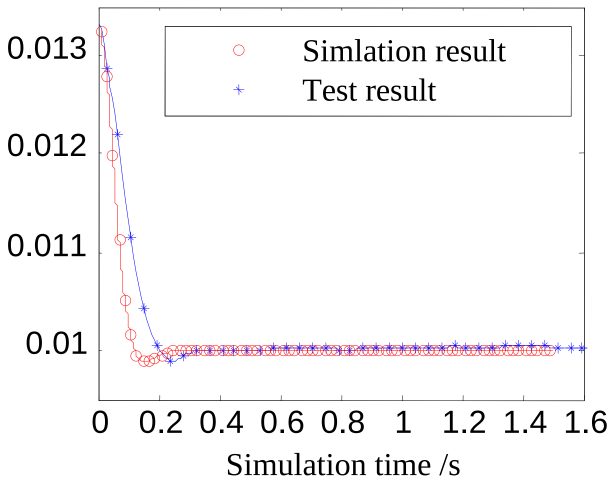

Fig. 10 Displacement of the magnet /m

For the limitation of the gap sensor range, the initial gap between the magnet and the track is set 13.3*mm.* From the Fig.10, the test result indicates that the system takes about 0.3*s* to reach the steady state and the maximum overshoot is 0.2*mm*. Compared with the test result, the simulation result shows the settling time is about 0.2*s*, and the overshot is a little bigger. Considering the simplification of the mathematical model and the parameter errors of the test rig, the simulation result and the test result are coincident with each other, which means the simulation model is correct and the control method is effective.

5. Conclusion

For the solution of the vehicle-guideway coupling vibration in the MAGLEV system and reduction the over dependence of system stability on the guideway properties, a new control strategy based on the full state observer-controller theory was realized. The dynamic characteristics of the MAGLEV system under different control methods were investigated, and the following conclusions can be drawn: (1) If there is no the guideway damping effect, the control method without considering the guideway vibration cannot maintain the MAGLEV system stable; For there exists a certain damping effect for the guideway in practice, the greater guideway stiffness can also make the MAGLEV vehicles operate steadily, but it is at the expense of a higher project cost. (2) The control method considering the guideway vibration introduces the guideway's vibration states into the control system using the state observer. Even if ignoring the damping effect of the guideway, the magnet can levitate steadily on the guideway with soft stiffness, which has been testified by the simulation and the test. With this control method, it is expected to reduce the over requirement of the guideway beam performance and save the construction cost of the MAGLEV line.

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