An Algorithm to Suppress the Low Frequency Oscillation of EMS Maglev Vehicles

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ABSTRACT: Compared with other transportation system, smooth, comfortable and noiseless represents the advantages of maglev transportation system. Whereas, in some cases a kind of low frequency oscillation of the levitation gap in small amplitude can be obviously observed, which greatly decreases the total performance of the levitation control. In order to suppress the low frequency oscillation, the principle of the oscillation is firstly analyzed in this paper, and then an adaptive vibration control algorithm using a virtual damper is put forward based on the principle; a nonlinear oscillation observer that used in the vibration control algorithm is also studied. Both the nonlinear oscillation observer and the vibration control algorithm proposed in this paper are small and effective, which can be easily embedded in the digital levitation controllers that are limited in computational resources. Numerical simulation and experiments undertaken on maglev vehicle CMS03A prove the effectiveness of this algorithm.

1 INTRODUCTION

The levitation control system of EMS Maglev vehicles is a complex, nonlinear system. Many control algorithms have been applied in the suspension control. Under some conditions, the levitation gap between the electromagnets and the guideway oscillates in small amplitude periodically. The frequency of the oscillation is quite low, generally less than 10 Hz. Although this kind of oscillation does not lead to an unstable control, it greatly decreases the performance of the levitation system and increases power costs. Detailed analyzes of this low frequency oscillation have been discussed by Zhou & Li (2007). In this paper, the reason of the oscillation is analyzed in a theory of virtual suspension system, and an adaptive virtual damper is proposed to suppress the oscillation. Simulations show that the virtual damper can absorb the oscillation effectively, and the experiments carried out in the middle-low speed Maglev vehicle CMS03A in Tangshan test line proves the validity of the theory and the effectiveness of the algorithm.

2 ANALYSIS OF THE OSCILLATION

We have had a detailed discussion of the low frequency oscillation occurred in EMS Maglev vehicles (Zhou & Li 2007). Here we adopt another method using virtual suspension theory to analyze the causes of the oscillation. Recently, the virtual suspension theory is found to be applied in many fields, especially in active vibration absorbing field (Wu et al. 2007; Sun et al. 2007), and most of them use liner motors as their actuators. The EMS levitation system uses the electromagnetic force as its control force, thus it has a natural born actuator- the electromagnets. We can deem the levitation system as an active virtual suspension system, and some active suspension designing methods and virtual vibration absorbing methods can be used here.

Consider the simplified levitation system of the EMS Maglev vehicles in figure 1, where F = magnetic force of the electromagnet; δ = levitation gap between the electromagnet and the guideway; u and i are the control voltage and current of the coil, respectively.

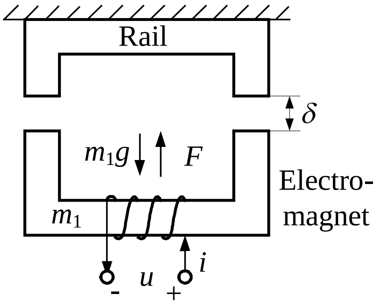

Figure 1. Simplified structure of the levitation system.

Suppose the control law of the levitation system in an ideal condition is

$$F = k_p\delta + k_d\dot{\delta} \tag{1}$$

then the levitation system can be equivalent to a mass-spring-damper system, as shown in figure 2.

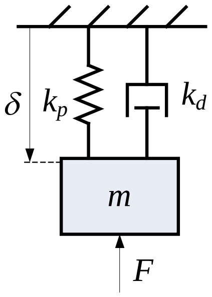

Figure 2. Equivalent structure of the levitation system.

The characteristic polynomial of the mass-spring-damper system is

$$\Delta = ms^2 + k_d s + k_p \tag{2}$$

Whereas nearly all the algorithm applied in magnetic levitation control can only adjust the voltage or the current of the coil instead of the magnetic force. It can be proved that the control delay of the magnetic force caused by the inductance of the coil can be equivalent to a low-pass filter. The voltage equation of the electromagnet is (Zhou & Li 2007)

$$u = iR + L\dot{i} - F_i\dot{\delta} \tag{3}$$

where L = inductance of the coil; R = DC resistance of the coil. When the levitation system is in static state, the current can be simplified in Laplace form as

$$i(s) = \frac{1}{Ls+R}u(s) \tag{4}$$

Obviously, the delay of the current would lead to the delay of the control force. Thus, the mass-spring-damper system shown in figure 2 turns to be

$$\begin{cases} F_0 = k_p\delta + k_d\dot{\delta} \\ F(s) = F_0(s)\dfrac{1}{\tau s+1} \\ m\ddot{\delta} = -F \end{cases} \tag{5}$$

where F_0 is the expected control force, and τ is the equivalent time delay of the control force. In this case, the characteristic polynomial of the third order system is

$$\Delta = ms^3 + \frac{1}{\tau}\left(ms^2 + k_d s + k_p\right) \tag{6}$$

The root locus of the third order system related to τ is shown in figure 3.

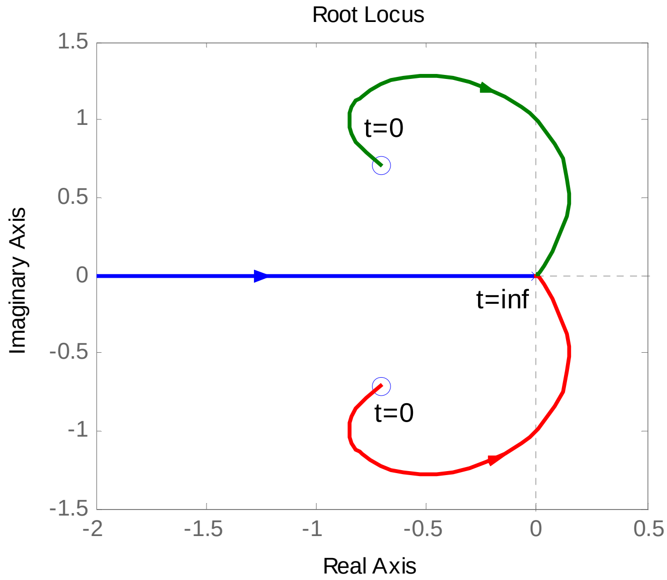

Figure 3. The root locus related to time delay τ.

We can see that, as the time delay τ increases, the damping coefficient of the dominant poles decreases. When the time delay is sufficiently large, the closed loop system turns to be marginal stable, or even unstable.

Practically, various factors may lead to time delays. Some of the time delays are caused by control algorithms. For example, the application of filters. All these delays can lead to the decrease of the damping coefficient. When the damping coefficient approaches critical, the closed loop system would oscillate in a fixed frequency due to the nonlinear character of the system. A much more complicated principle of the nonlinear oscillation is studied by Wang et al, (2007).

Figure 4 shows the waveform of the low frequency oscillation recorded in low speed Maglev vehicle- CMS03A, in Tangshan test line.

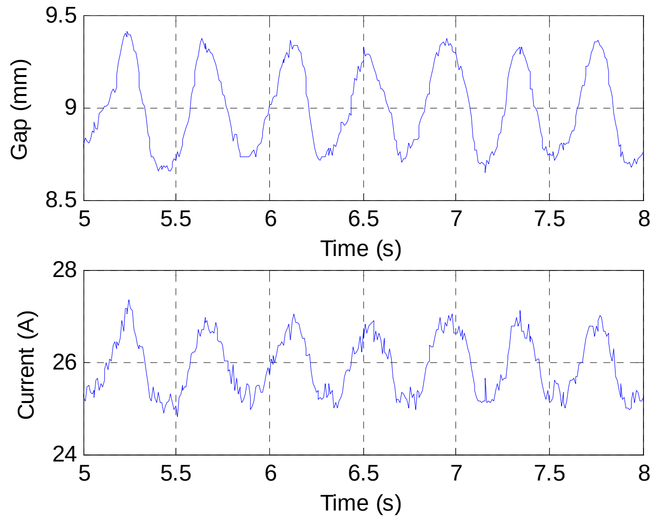

Figure 4. The waveform of the oscillation recorded by CAN bus.

The waveform in figure 4 was recorded by a CAN bus monitor, when the vehicle was parking in the garage.

It seems that the system can be deemed as a mass-spring system without damping, as shown in figure 5.

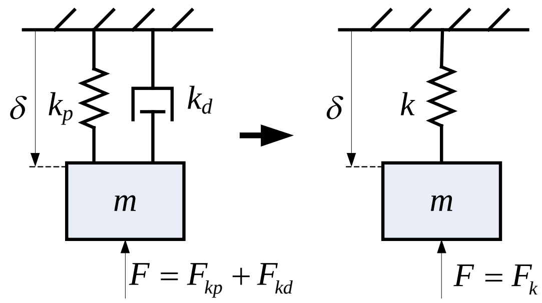

Figure 5. An equivalent model of the levitation system when the low frequency oscillation occurs.

We can imagine that, if we can fix a damper between the electromagnet and the guideway, as shown in figure 6, then the energy of the oscillation would be absorbed by the damper, and the oscillation would then attenuate. Based on this idea, an adaptive virtual damper, which is independent of the original control algorithm, is proposed in this paper.

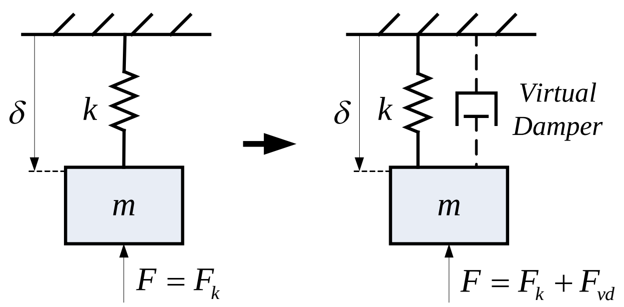

Figure 6. The application of a virtual damper.

3 DESIGN AND REALIZATION OF THE ADAPTIVE VITUAL DAMPER

The levitation force of the EMS Maglev vehicle is controllable, thus we can realize the damping force though adjusting the electromagnetic force.

The damping force of the virtual damper in figure 6 can be simply written as:

$$F_{vd} = k_c \dot{\delta} \tag{7}$$

But this is not proper in practice. Because it greatly changes the character of the original control algorithm, and it may stimulate the high frequency coupled vibration of the guideway easily.

Our task is then to design a special damper that only works in a specific frequency. That is to say, the damping force of the virtual damper should be large enough to absorb the oscillation, while small enough in the other frequency, causing as little affection to the original algorithm as possible. To realize this function, we need to add a band pass filter after the damping force of the virtual damper. The center frequency of the band pass filter should be identical to the frequency of the low frequency oscillation.

Considering the frequency of the oscillation varies according to different conditions, the center frequency of the band pass filter should not be fixed. It should be kept identical to the frequency of the oscillation. This function can be realized by designing a frequency estimator. There are many kinds of frequency estimators applied in many fields, most of the researches are focus on single frequency estimators (Trapero et al. 2007; Sarma 2007; Bodson & Douglas 1997), while some researches aim at multiple frequency estimators (Abeysekera 2008). A detailed overview of these frequency estimators can be found in the articles written by Quinn (2008) and Trapero et al. (2007). Here we choose the adaptive frequency estimator introduced by Bodson & Douglas (1997), as shown below:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = -2\zeta\omega x_2 - \omega^2 x_1 + ku \\ \dot{\omega} = -g\left(ku - 2\zeta\omega x_2\right) x_1 \end{cases} \tag{8}$$

where u = input signal; ω = result of the estimation; ζ = damping coefficient of the damper; and g = gain of the estimator. Choose ζ and g properly, then we can get a quick and stable response of the estimator. It should be mentioned that, to keep the estimator work normally, the DC bias and the high frequency noise of the input signal should be filtered, and the amplitude of the input signal should be limited in a proper range.

The designed block diagram of the closed loop system including the virtual damper is shown in figure 7. In figure 7, P is the control plant including the electromagnet and the secondary suspension system, C1 is the original controller; BPF1 and BPF2 are two band pass filters; and FE is the frequency estimator. The components in the dark colored block compose the virtual damper.

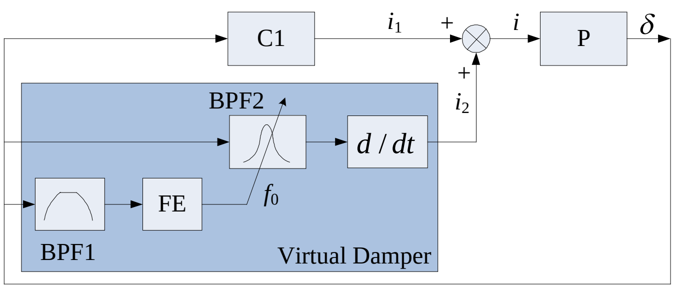

Figure 7. The block diagram of the closed loop system.

According to the frequency of the oscillation that may be encountered in practice, the pass band of BPF1 is chosen to be 1Hz ~ 10Hz, while BPF2 is designed to be a second order LC filter. The transfer function of BPF2 is:

$$BPF2 = \frac{s^2 + 2\zeta_z \omega_0 s + \omega_0^2}{s^2 + 2\zeta_p \omega_0 s + \omega_0^2} \quad (9)$$

where ω_0 is the center frequency of BPF2. ζ_z and ζ_p determine the maximum gain and the bandwidth of the band pass filter. This design can ensure that the shape of BPF2 is kept unaffected when the center frequency of BPF2 is changing. What's more, this form of filters can be easily realized in the adaptive algorithm, especially in the digital controller of the levitation control system. In the test, we chose ζ_z = 1.0, ζ_p = 0.14, then we got 17dB gain in the center frequency. The bandwidth of BPF2 is proportional to the center frequency ω_0.

4 SIMULATION AND EXPERIMENT

4.1 Simulation Results

Figure 8 shows the simulation result of the shape of BPF2. The center frequency of BPF2 is adjustable, while the shape of it is independent to the center frequency. The gain in the center frequency is 17dB. We should select a suitable maximum gain of BPF2, because the higher the maximum gain of BPF2 is, the narrower the bandwidth of it will be, and the worse the phase character will be.

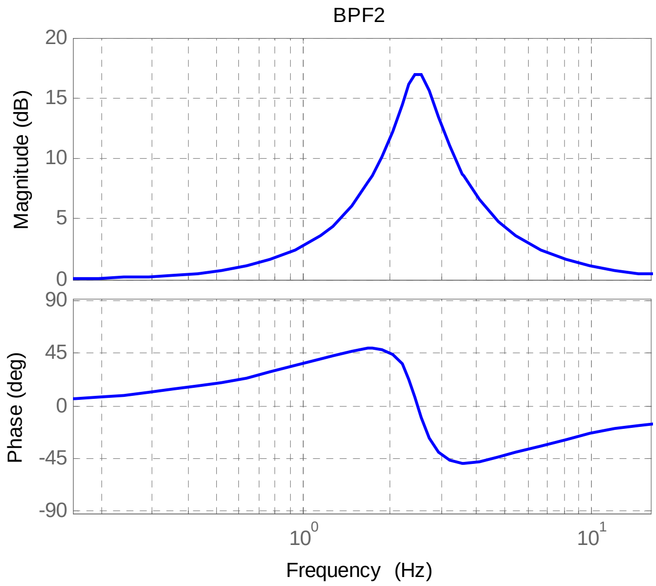

Figure 8. The shape of BPF2.

Figure 9 shows the simulation of the frequency estimator (FE). The input signal of the frequency estimator is a 2.5Hz sinusoid, and the sample frequency of the discrete time model of (8) is 2KHz. We can see that the estimation of the frequency approaches to the stable state in 2 seconds.

Figure 10 shows the simulation result of the closed loop system. To ensure the frequency estimator converges to the stable state, the center frequency of BPF2 is adjusted after 3 seconds, and the damping force is then added to the closed loop system after BPF2 adjusted.

We can see that after the virtual damper is activated, the low frequency oscillation attenuates quickly.

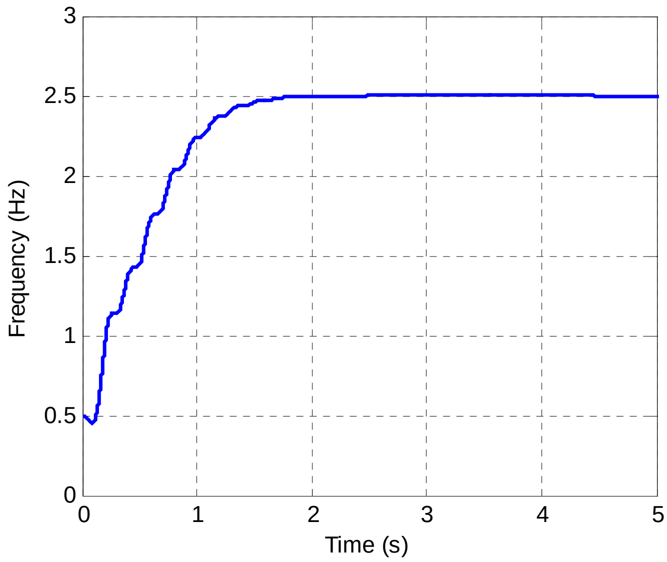

Figure 9. Simulation of the frequency estimator.

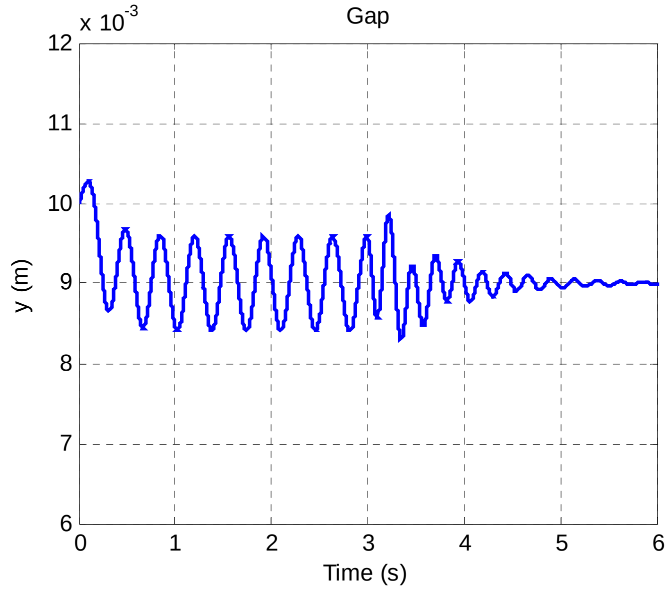

Figure 10. Simulation of the closed loop system. Note that the virtual damper is activated after 3 seconds.

4.2 Result of experiment

Figure 4 shows the oscillation of an electromagnet of CMS03A in Tangshan test line, when the vehicle was parking in the garage. We can see that the frequency of the oscillation is about 2.4Hz, and the current of the coil oscillates obviously. It decreases the comfort of the vehicle, and causes more power consumption.

Then we added the virtual damper into the closed loop, we found that the oscillation disappeared. What's more, there wasn't any obvious affection of the stability found when the vehicle was running fast. It proves that the virtual damper can absorb the oscillation effectively, and it brings little affection to the original control algorithm.

5 CONCLUSIONS

The low frequency oscillation of EMS Maglev vehicles sometimes occurs when the vehicle is standing still. It decreases the ride comfort and increases power consumption, and sometimes it appears that the control algorithm cannot suppress the oscillation itself. In this paper, we had a brief analysis of the oscillation using equivalent suspension system theory, and proposed a virtual damper to absorb the energy of the low frequency oscillation. In order to limit the damping force in a specified frequency identical to the frequency of the oscillation, a frequency estimator and a band pass filter was applied. All these designs were aiming at the application in a digital levitation controller, and they could be easily embedded in the DSP controller. The idea of adaptive virtual damper in suppressing the oscillation was proved to be effective through numerical simulation and experiments carried out in CMS03A.

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