MAGNETIC DAMPING FOR MAGLEV*

S. S. Chen, S. Zhu, Y. Cai, and D. M. Rote

Energy Technology Division
Argonne National Laboratory
Argonne, Illinois 60439

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

The submitted manuscript has been authored by a contractor of the U.S. Government under contract No. W-31-109-ENG-38. Accordingly, the U.S. Government retains a nonexclusive, royalty-free license to publish or reproduce the published form of this contribution, or allow others to do so, for U. S. Government purposes.

To be submitted for presentation at ASME Winter Annual Meeting, Chicago, IL, November 13-18, 1994.

This work was performed under the sponsorship of the U.S. Army Corps of Engineers and the Federal Railroad Administration through interagency agreements with the U.S. Department of Energy.
DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.
MAGNETIC DAMPING FOR MAGLEV

S. S. Chen, S. Zhu, Y. Cai, and D. M. Rote

Energy Technology Division
Argonne National Laboratory
Argonne, Illinois 60439

Magnetic damping is one of the important parameters to control the response and stability of maglev systems. An experimental study is presented to measure the magnetic damping using a direct method. A plate attached to a permanent magnet levitated on a rotating drum was tested to investigate the effect of various parameters on magnetic damping such as conductivity, gap, excitation frequency, and oscillation amplitude. The experimental technique is capable of measuring all magnetic damping coefficients, some of which can not be measured by an indirect method.
1. INTRODUCTION

One of the key elements in controlling the dynamic characteristics of maglev systems is magnetic damping. Two types of magnetic damping can be introduced, active and passive magnetic damping. Several aspects of magnetic damping have been studied: (1) magnetic damping as a function of speed (Moon 1977, Iwamoto et al. 1974, Yamada et al. 1974); (2) damping constant as a function of frequency (Saitoh et al. 1992); (3) passive damping (Iwamoto et al. 1974, Fujiwara 1980); and (4) active secondary suspension (Nagai and Tanaka 1992). It is fair to say that damping characteristics under various conditions are still not well characterized.

Different methods can be used to analyze or measure magnetic damping: direct method and indirect method. It appears that most of the past studies are based on the indirect method. A direct method was introduced recently to maglev system by Chen et al. (1993). The direct method is capable of measuring the self-induced and mutual magnetic damping. In the study of maglev response, all magnetic damping must be quantified; without those damping values, it is difficult to predict the response of maglev systems.

This paper presents two series of tests to quantify the passive magnetic damping. The purpose is to determine the effect of various parameters on magnetic damping such as conductivity, gap, excitation amplitude, and oscillation frequency. Once magnetic damping and stiffness are known, it can be applied to maglev systems.
2. EXPERIMENTAL SETUP

The general experimental setup is shown in Fig. 1. It includes a rotating drum, shaker, force transducer, and displacement transducer. The drum is covered with aluminum sheet with 26.99 cm diameter, 14.61 cm wide, and 0.635 cm thick. The rotating speed can vary from 0 to 3500 rpm with its speed from 1 m/s to 50 m/s. The shaker provides proper excitation force at given frequencies and the impedance transducer measures the displacement of the supporting bar.

The force transducer to measure the magnetic force is shown in Fig. 2a. The magnet, 2.54 x 5.08 x 0.318 cm, is connected to aluminum plates, 5.0 x 7.6 x 0.8 cm, with copper brackets, and then supported by an aluminum bar, 2.64 cm wide, 22.86 cm long, and 1.27 cm thick (Fig. 2b). The aluminum bar is attached to the shaker at one end and to the magnet and aluminum plate at the other. One set of strain gauges is placed on the smaller section of the aluminum bar to measure the force due to the excitations at other end.

Without magnetic field, the force transducer is calibrated by dynamic method. The supporting bar is fairly rigid with a natural frequency of larger than 100 Hz. For a given excitation frequency and amplitude given to the supporting bar by the shaker, the inertia force of the magnet, aluminum plate, and support structure can be calculated. Typical results are shown in Fig. 3. Theoretically, the inertia force should be proportional to the square of excitation frequency. The curve given in Fig. 3 shows that the power is very close to 2. From the inertia force, displacement, and strain gauge, the calibration constant of the force transducer can be calculated. The force transducer measures the lift force with a sensitivity of about 1 volt for 120 g of force acting on the middle of the magnet and damping plate.
3. TEST CASES AND DATA ANALYSIS

The detailed arrangement of the magnet and aluminum plate is shown in Fig. 2b. Two cases are tested.

Case A: Magnet and Pure Aluminum Plate: An aluminum plate with its purity equal to 99.999%, 5.0 x 7.6 x 0.8 cm, is attached to the magnet. The gaps between aluminum plate and magnet, a, and aluminum plate and rotating drum, b, can be set at different values. Five tests are performed:

A.1   a = 1 mm, b = 3 mm
A.2   a = 1 mm, b = 5.5 mm
A.3   a = 6 mm, b = 3 mm
A.4   a = 3.5 mm, b = 5.5 mm
A.5   a = 1 mm, b = 8 mm

Case B: Magnet and Ordinary Aluminum Plate: An aluminum plate (6061-T6), 5.0 x 7.6 x 0.8 cm, is attached to the magnet. Three tests are performed.

B.1   a = 1 mm, b = 3 mm
B.2   a = 1 mm, b = 5.5 mm
B.3 \( a = 6 \text{ mm}, \ b = 3 \text{ mm} \)

In each test, the shaker provides an excitation at a given frequency with a specific amplitude. The displacement of the aluminum support measured by the displacement transducer and the forces consisting of the inertia force of the active element (magnet, aluminum plate, copper bracket, and aluminum support below the strain gauges) and magnetic force measured by the strain gauges are measured simultaneously. The RMS magnitude of the displacement and force as well as the phase angle between them at the excitation frequency are obtained. The magnetic stiffness and damping can be calculated from those data. Each test was performed at several excitation frequencies for the whole range of rotating speed.

The measured dynamic force is given as follows:

\[
F = -m \frac{d^2u}{dt^2} - C \frac{du}{dt} - Ku,
\]

where \( u \) is the displacement, \( m \) is the mass of the active element, and \( C \) and \( K \) are damping coefficient and stiffness to be determined, respectively. Note that \( m \) should also include magnetic mass. However, because the excitation frequency of maglev is fairly low, the magnetic mass will be very small and is negligible (Iwamoto et al. 1974).

Let the RMS values of the displacement and force be \( d_o \) and \( f_o \), respectively, and the phase angle between the two be \( \phi \). \( C \) and \( K \) are given by

\[
C = F_o \sin (\phi)/\omega u_o,
\]
Magnetic damping and stiffness are calculated from Eq. (2)

\[ K = m\omega^2 - \frac{F_0}{u_0} \cos(\phi). \]  

4. EXPERIMENTAL RESULTS

Typical results are given in Figs. 4 and 5 for Test A.5 with the excitation frequency equal to 6 Hz. In Fig. 4, the time histories of displacement, measured force, and magnetic force at three different speeds, 0, 4.8, and 16.5 m/s are given. The measured force includes inertia force and magnetic force. The magnetic force is determined by subtracting the inertia force from the measured force. The inertia force is out of phase with the displacement, while the magnetic force is approximately in phase with displacement. In this case, inertia force and magnetic force are approximately the same magnitude. Therefore, the resultant force becomes fairly small. It is noted that the resultant force is approximately in phase with displacement at 0 and 4.8 m/s, while they are out of phase at 16.5 m/s. The transition occurs in the neighborhood of the characteristics speed of the aluminum sheet on the rotating drum.

Figure 5 shows the RMS displacement of the support bar, RMS force measured by the strain gauge, and the phase between the two as a function of speed. Those data are obtained from the time histories of the displacement of the support bar and the measured force shown in Fig. 4. Magnetic damping and stiffness are calculated from Eqs. 2. The RMS displacement decrease slightly with increasing speed for a fixed input to the shaker. The measured RMS force changes significantly with speed. The phase angle changes drastically at a speed corresponding to the
characteristic speed. The magnetic stiffness increases with speed while magnetic damping decreases with speed. In this case, the magnetic damping is positive.

Tests at three frequencies, 2, 4, and 6 Hz, are conducted for the whole range of rotating speed for all test cases. Typical results are shown in Fig. 6 for Test A.4. This figure contains the same information as those in Fig. 4 except that three excitation frequencies are included. The inertia force is proportional to the square of the excitation frequency. At 2 and 4 Hz, the inertia force is smaller, the measured force will be mainly attributed to magnetic force. For different excitation frequencies, the drastic change of phase angle between displacement and measured force also changes with excitation frequencies, but all of them are close to the characteristic speed.

Figures 7 and 8 show the magnetic damping and stiffness for different tests as a function of speed for three excitation frequencies. Some general characteristics are noted:

- Magnetic damping decrease with speed. At zero speed, it is always positive. Once the speed exceed the characteristic speed, its value change very little with the speed.

- Magnetic stiffness increases with speed. At high speed, its values are almost independent of speed.

- Magnetic stiffness is independent of the excitation frequency, while magnetic damping depend on the excitation frequency with it values increasing slightly with excitation frequency.
By comparing magnetic damping and stiffness for different cases, we can understand the effects of various parameters.

4.1 Conductivity

Tests A.1, A.2, and A.3 are corresponding to Test B.1, B.2, and B.3; the difference is the aluminum plate. In the series A the plate has the purity of 99.999% and in the series B, it is 6061-T6. From Figs. 7 and 8, the following are noted:

- Magnetic Stiffness: The magnetic stiffness are about the same for different excitation frequencies in test series A and B. The purity of the aluminum plate does not affect the magnetic stiffness.

- Magnetic Damping: The magnetic damping for Test A.2 and B.2 are about the same. In this case, the aluminum plate is 5.5 mm from the drum. However, when the gap is 3 mm for Tests A.1, A.3, B.1, and B.3, magnetic damping values for the series A are larger than those of the series B. This means that the aluminum plate with higher purity will provide higher damping.

4.2 Gap

- In the tests A.3, A.4, and A.5, the gap between the magnet and drum is kept at a constant, while the aluminum is placed at different gaps with respect to the drum, 3, 5.5, and 8 mm. Magnetic stiffnesses for different gaps are approximately the same. However, magnetic damping values depend on the gap.
As the aluminum plate moves closer to the drum, magnetic damping increases. The magnetic damping values for 3 and 5.5 mm are larger than those for 8 mm.

- Tests A.1 and A.3, or Tests A.2 and A.4, the gap between the aluminum and drum is fixed while the gap between the magnetic and aluminum is varied. We can also compare the magnetic damping values for the two sets of tests. When the magnet is further away from the drum, magnetic stiffness decrease and magnetic damping increases.

4.3 Frequency

The effect of excitation frequency is presented in Figs. 9 and 10 for four speeds, 0, 4.8, 8.3, and 37.6 m/s. The magnetic stiffness is practically independent of the frequency while magnetic damping increase slightly with frequency.

4.4 Excitation Amplitude

Figure 11 shows the magnetic stiffness and damping for test B.3 as a function of excitation amplitude at 33.8 m/s. Similar tests have also been performed at other speeds. Regardless of the speed, it is noted that excitation amplitude does not affect both magnetic stiffness and damping in the parameter range tested. This means the linear theory can be used if the displacement is small.
5. APPLICATIONS TO MAGLEV

Once the maglev damping and stiffness are known, it can be applied to vehicle dynamics. As an example, consider a maglev vehicle, with the mass \( m \), moving on a guideway. The equation of motion is

\[
m \frac{d^2u}{dt^2} + (C_s + C) \frac{du}{dt} + Ku = q(t)
\]  

where \( m \) is the total mass of the vehicle; \( C_s \) is structural damping; \( C \) is magnetic damping including aerodynamic damping; \( K \) is magnetic stiffness; and \( q(t) \) is external excitation.

The natural frequency and modal damping ratio are

\[
f = \frac{1}{2\pi} \sqrt{\frac{K}{m}},
\]

\[
\zeta = \frac{C_s + C}{2\sqrt{Km}}.
\]

For a given vehicle, \( C \) and \( K \) depend on clearance and speed. Using the experimental data measured in this experiment, we can analyze the system characteristics. For example, magnetic damping and stiffness obtained in Tests A.1, A.2, and A.5 can be used to simulate a vehicle with a passive damping plate levitated on a guideway. Magnetic damping and stiffness depend on the clearance \( b \) (see Fig. 2b) and oscillation frequency \( f \). From Figs. 7, \( C \) and \( K \) can be obtained as follows:
From Eqs. (4) and (5) as well as static magnetic force, the natural frequencies and modal damping ratio can be calculated as a function of $b$. Furthermore, the response of a maglev system can be predicted from Eq. (3) for a given excitation $q(t)$.

6. CLOSING REMARKS

A direct method is used to measure magnetic damping and stiffness for two series of tests. The effect of various parameters are investigated: conductivity, gap, excitation amplitude and excitation frequency. The direct method of measurement is useful in determining magnetic damping and stiffness.

Once magnetic damping and stiffness are know, the dynamic response of maglev systems can be predicted. In addition, passive damping plate can be introduced to improve ride quality and to control stability. Other control techniques using the characteristics of magnetic forces can be applied to maglev.
This technique will be very useful to measure magnetic damping and stiffness coefficients for prototype. This will provide the necessary elements for the prediction of maglev response as well as the control of maglev systems.

ACKNOWLEDGMENTS

This work was performed under the sponsorship of the U.S. Army Corps of Engineers and the Federal Railroad Administration through interagency agreements with the U.S. Department of Energy.

REFERENCES

Chen, S. S., Zhu, S., and Cai, Y., 1993, *On the Unsteady-Motion Theory of Magnetic Forces for Maglev*, Argonne National Laboratory Technical Report ANL-93/39.

Fujiwara, S., 1980, *Damping Characteristics of the Repulsive Magnetic Levitation Vehicle*, Japanese Railway Technical Research Institute, Quarterly Reports, Vol. 21, No. 1, pp. 49-52.

Iwamoto, M., Yamada, T., and Ohno, E., 1974, *Magnetic Damping Force in Electrically Suspended Trains*, IEEE Trans. Magn., Vol. Mag-10, No. 3, pp. 458-461.

Moon, F. C., 1977, *Vibration Problems in Magnetic Levitation and Propulsion*, Chapter 6 of *Transportation without Wheels* by E. R. Laithwaite, Paul Elek (Scientific Books) Ltd., London, pp. 123-161.
Nagai, M., and Tanaka, S., 1992, *Study on the Dynamic Stability of Repulsive Magnetic Levitation Systems*, JSME International Journal, Series III, Vol. 35, No. 1, pp. 102-108

Saitoh, T., Maki, N., Kobayashi, T., Shibata, M., and Takizawa, T., 1992, *Electromagnetic Force and Eddy Current Loss in Dynamic Behavior of a Superconducting Magnetically Levitated Vehicle*, Presented at the Applied Superconductivity Conference, Chicago, August 26, 1992, Paper LKC-1.

Yamada, T., Iwamoto, M., and Ito, T., 1974, *Magnetic Damping Force in Inductive Magnetic Levitation System for High-Speed Trains*, Elect. Eng. Jpn, Vol. 94, No. 1, pp. 49-54.
Figure Captions

1. Schematic of test setup

2. Force transducer and magnet support: (a) Overview (b) Active element

3. Force-displacement ratio as a function of excitation frequency

4. Displacement, total force, and magnetic force as a function of time to an excitation at 6 Hz, Test A.5

5. Displacement, force, phase angle, and magnetic damping and stiffness excited at 6 Hz, Test A.5

6. Displacement, force, phase angle, and magnetic damping and stiffness excited at 2, 4, and 6 Hz, Test A.4

7. Magnetic damping and stiffness for series A

8. Magnetic damping and stiffness for series B

9. Magnetic damping as a function of excitation frequency. Tests A.3, A.4, and A.5

10. Magnetic stiffness as a function of excitation frequency, Tests A.3, A.4, and A.5

11. Magnetic damping and stiffness as a function of excitation amplitude at 33.8 m/s, Test B.3
Fig. 2b
Fig. 5
Fig. 6
Fig. 7
Fig. 7 (cont.)
