Structure analysis of levitation chassis of medium-to-low speed maglev vehicles based on left-right decoupling

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Abstract: Levitation chassis, as an extremely important component of maglev vehicles, provides functions of transmitting levitation force and steering force, and directly affects the safety performance of the vehicle. Based on the vertical dynamics model of the levitation chassis, kinetic equations of the model are established, and a simulation program is designed to analyze the structural decoupling function of the chassis, especially under the influence of elastic constraints between the left and right modules, which are exclusively owned by maglev vehicles. A finite element model of the levitation chassis based on left-right decoupling is constructed. Analysis results of the model show that the mechanical properties of the chassis tailored for the vehicle meet the design requirements, and the stiffness and strength is adequate to bear the weight of the whole vehicle.

Key words: maglev vehicle; levitation chassis; dynamics; left-right decoupling; finite element

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1. Introduction

Long-stator medium-to-low speed maglev vehicles are a new kind of rail transit system, which features low noise, small vibration, moderate transport volume, safety, and economy, and is more suitable to be the city's dominant means of rail transportation over trams, light rails, and subways [1].

Levitation chassis, as well as the chassis of the whole maglev vehicle, is an extremely important component of the vehicle that provides functions of transmitting levitation force and steering force [2], and also has a direct impact on safety performance of the vehicle. Therefore, it is quite necessary to analyze the structural strength and stiffness of the levitation chassis. In addition, because of the comparatively complicated structure of the maglev, finite element method (FEM) is an effective method for calculation of structural strength and stiffness of the levitation chassis. Current research on structure strength analysis of the levitation chassis only considers one side of the module instead of both sides in left-right decoupling. Jiang et al. [3] presented some suggestions for structure design and improvements of

anti-rolling sills after analyzing of the relationship between anti-rolling sills and curve negotiation of maglev vehicles. Luo and Zhang [4] offered credible data for optimized design of anti-rolling sills of maglev bogie after electromagnetisms calculation of U-shaped levitation electromagnet. Simulation of structural decoupling functions of right and left levitation modules is rarely reported. Therefore, it has great theoretical values to conduct an FEM analysis on the levitation chassis based on the left-right decoupling.

This paper establishes a vertical dynamics model of a levitation chassis, and analyzes the influence of elastic constraints between the left and right modules of the maglev vehicle. An FEM model of the levitation chassis is built with consideration of the left-right decoupling. Stress and deformation data under different conditions are obtained from the model by ANSYS to test whether the vehicle meets the requirements. They may provide a reference for the structural improvements of levitation chassis.

2. Vertical dynamics of levitation chassis

2.1. Vertical dynamics model

This paper only considers a common levitation chassis unit running on a single-span flexible track with a simple support. The elastic structure deformation of

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levitation chassis itself is ignored, and then levitation chassis can be simplified to a vibration system consisting of two rigid bodies interconnected by elastic damping elements (K_b, C_b). Each rigid body has three degrees of freedom: plunging, nodding, and rolling motion. The vehicle body is installed above the levitation chassis through an air spring and the force acted on the air spring is F .

When the maglev vehicle is in steady-state suspen-

sion, the magnetic gap has a small-scale change only near the steady-state point. Thus, the magnet/rail relationship can be regarded as linear [5-7]; that is, the interaction effect between levitation chassis and track (magnet/rail relationship) is equivalent to the spring-damper suspension (K_p, C_p). The vertical dynamics model of levitation chassis is established, as shown in Fig. 1.

Fig. 1 Vertical dynamics model of levitation chassis

2.2. Kinetic equation and solution

2.2.1. External disturbances

The vertical irregularity of the track, caused by processing, installation error, thermal stress, and static deflection under the influence of vehicles and other loads, can be described by the cosine function excitation model [8]:

$$W(t) = \frac{1}{2} a(1 - \cos \omega t) \quad (0 \leq t \leq \frac{L}{v}), \quad (1)$$

where $\omega = 2\pi v / L$, L is the irregularity wavelength; a is the irregularity wave depth; and v is the vehicle speed.

2.2.2. Vibration equation of levitation chassis

The general forms of vibration equations of left and right levitation chassis modules are given below.

Equation of plunging motion:

$$M_t Z_t'' + C_p(2Z_t' - W_1' - W_2') + K_p(2Z_t - W_1 - W_2) = F_1 + F_2 + M_t g. \quad (2)$$

Equation of nodding motion:

$$J_{at} \alpha_t'' + 2C_p l_1^2 \alpha_t' + l_1 C_p (W_1' - W_2') + 2K_p l_1^2 \alpha_t + l_1 K_p (W_1 - W_2) = (F_2 - F_1) l_2. \quad (3)$$

Equation of rolling motion:

$$J_{\beta t} \beta_t'' + C_b(l_a^2 + l_b^2) \beta_t' + 2C_b l_a l_b \beta_t' \pm (l_a + l_b) C_b (Z_{tl}' - Z_{tr}') + 2C_\beta \beta_t' + K_b(l_a^2 + l_b^2) \beta_t + 2K_b l_a l_b \beta_t \pm (l_a + l_b) K_b (Z_{tl} - Z_{tr}) + 2K_\beta \beta_t = 0. \quad (4)$$

In Eqs. (2)–(4), Z_t , α_t , and β_t represent the plunging displacement, nodding angle, and rolling angle, respectively; K_p , K_β , C_p , and C_β represent the vertical stiffness, rolling angle stiffness, vertical damping, and rolling angle damping, respectively; K_b and C_b are the spring stiffness and damping of anti-rolling sills, respectively. The rest are shown in Fig. 1.

In actual calculation for the left and right modules, the subscript t should be replaced by the corresponding tl or tr . Besides, subscripts 1 should be replaced by 1l or 1r, and 2 by 2l or 2r. Symbol “ $\pm$ ” in Eq. (3) is decided by the rule: “ $-$ ” for the left module, “ $+$ ” for the right module.

According to Eqs. (2)–(4), the vibration dynamics equation of levitation chassis can be written in the following unified form:

$$M\ddot{X} + C\dot{X} + KX = P, \quad (5)$$

where, X , $\dot{X}$, and $\ddot{X}$ are the generalized displacement, velocity, and acceleration, respectively; M , C , and K are the system mass, damping, and stiffness matrices, respectively; P is the generalized load matrix. Expansion of Eq. (5) is shown as follows:

$$\begin{aligned}
& \begin{bmatrix} M_t \\ J_{\alpha t} \\ J_{\beta t} \\ M_t \\ J_{\alpha t} \\ J_{\beta t} \end{bmatrix} \begin{bmatrix} Z_{tl}'' \\ \alpha_{tl}'' \\ \beta_{tl}'' \\ Z_{tr}'' \\ \alpha_{tr}'' \\ \beta_{tr}'' \end{bmatrix} + \begin{bmatrix} 2C_p & 0 & 0 & 0 & 0 & 0 \\ 0 & 2C_p l_1^2 & 0 & 0 & 0 & 0 \\ C_b(l_a+l_b) & 0 & C_b(l_a+l_b)^2+2C_\beta & -C_b(l_a+l_b) & 0 & 0 \\ 0 & 0 & 0 & 2C_p & 0 & 0 \\ 0 & 0 & 0 & 0 & 2C_p l_1^2 & 0 \\ -C_b(l_a+l_b) & 0 & 0 & C_b(l_a+l_b) & 0 & C_b(l_a+l_b)^2+2C_\beta \end{bmatrix} \begin{bmatrix} Z_{tl}' \\ \alpha_{tl}' \\ \beta_{tl}' \\ Z_{tr}' \\ \alpha_{tr}' \\ \beta_{tr}' \end{bmatrix} + \\
& \begin{bmatrix} 2K_p & 0 & 0 & 0 & 0 & 0 \\ 0 & 2K_p l_1^2 & 0 & 0 & 0 & 0 \\ K_b(l_a+l_b) & 0 & K_b(l_a+l_b)^2+2K_\beta & -K_b(l_a+l_b) & 0 & 0 \\ 0 & 0 & 0 & 2K_p & 0 & 0 \\ 0 & 0 & 0 & 0 & 2K_p l_1^2 & 0 \\ -K_b(l_a+l_b) & 0 & 0 & K_b(l_a+l_b) & 0 & K_b(l_a+l_b)^2+2K_\beta \end{bmatrix} \begin{bmatrix} Z_{tl} \\ \alpha_{tl} \\ \beta_{tl} \\ Z_{tr} \\ \alpha_{tr} \\ \beta_{tr} \end{bmatrix} = \quad (6) \\
& \begin{bmatrix} M_t g + F_{1l} + F_{2l} + C_p W_{1l}' + C_p W_{2l}' + K_p W_{1l} + K_p W_{2l} \\ (F_{2l} - F_{1l})l_2 - C_p l_1 (W_{1l}' - W_{2l}') - K_p l_1 (W_{1l} - W_{2l}) \\ 0 \\ M_t g + F_{1r} + F_{2r} + C_p W_{1r}' + C_p W_{2r}' + K_p W_{1r} + K_p W_{2r} \\ (F_{2r} - F_{1r})l_2 - C_p l_1 (W_{1r}' - W_{2r}') - K_p l_1 (W_{1r} - W_{2r}) \\ 0 \end{bmatrix}
\end{aligned}$$

We can solve Eq. (6) using a new rapid display integration method introduced by Zhai [9] without solving the high-order algebraic equations in the integration process. The method proves to be fast, and accurate, and is especially suitable for solving large-scale engineering dynamics problems.

3. Left-right decoupling simulation of levitation chassis

Anti-roll constraint parameters of levitation chassis (K_b , C_b) are unique and important suspension parameters of maglev systems, the value of which is crucial in the right-left decoupling of the chassis because too large or small parameters will cause an unstable rolling motion. Left-right decoupling requires that anti-roll constraints have damping effects when the levitation chassis module on one side is subjected to an external disturbance, and that the vibration response of the module on the other side converges quickly to maintain a relative independence.

In the given simulation conditions, the right module loses its balance from a sudden power failure when the levitation gap reaches 8 mm and then descends to the track in uniform acceleration under the action of the body force, and then ascends in uniform acceleration to the suspension gap under the action of levitation force

when power is restored again. The trajectory of the right module is a saw-tooth shape, as shown in Fig. 2. Then the vibration response of the left module is examined.

Fig. 2 Right module input excitation in stimulation

Eq. (4) shows that the interaction between left and right modules relates only to the rolling motion. Vibration response of the left module is examined with the response of the right module as an input, as shown in Fig. 3. Fig. 3 reveals that the left module enters a state of steady vibration with smaller amplitude, the maximum of which is 0.003 25 rad, and tends to converge after 2.0 s attenuation vibration. It demonstrates an independent stability between left and right modules that the left module converges when the right module is unstable; and also shows that anti-roll constraint design parameters are reasonable and meet the decoupling requirement of levitation chassis.

Fig. 3 Left module output response of left-right decoupling

4. Structure analysis of levitation chassis

4.1. Finite element model of levitation chassis based on left-right decoupling

Prior to the modeling, some simplifications of the real model are required. Those features having no or minimal influence on the actual problems are inhibited or deleted in this paper to simplify the model [10].

Levitation chassis is always in bilateral symmetry, so only half of the model is needed in modeling. Emergency braking devices, walking wheel devices, suspension sensor devices, and traction base which have a limited impact on the overall structure can be ignored in modeling. In addition, the levitation chassis is of a welded structure, and the whole structure can be viewed as a rigid connection structure. The contact relationships of the various parts of the assembly can be set as “bonded” in the ANSYS Workbench and the bolts connecting the assembly can be simplified, which reduces the complexity in ANSYS.

Using Solidworks and a bottom-up approach in design, we built the three-dimensional solid models of different parts on the levitation chassis and then the three-dimensional solid model of the whole system according to the assembly relationship, as shown in Fig. 4.

The principle of finite element modeling is to simplify the model to the full extent while accurately conforming to the mechanical properties of the simulation structure. The Solidworks model is imported into ANSYS Workbench by utilizing seamless connection technology, and the pre-designed anti-roll constraint parameters (suspender elastic damping unit) are added to ANSYS, which are $K_b=30$ kN/m, and $C_b=200$ N·s/m. The whole model is discretized into 183 547 units in 370 185 nodes using the algorithm of SOLID 187 element in free meshing. The finite element model is shown in Fig. 5.

Fig. 4 Solidworks model of levitation chassis

Fig. 5 Finite element model of levitation chassis

4.2. Material parameters

Corbel and walking wheel is made of ZL101A aluminum alloy and the lower longitudinal beam is made of 6061(T6) aluminum alloy while the upper longitudinal beam is made of carbon steel Q235A. The property parameters of these materials are shown in Table 1.

Table 1 Material parameters

Material	Elastic module (MPa)	Poisson's ratio	Density (kg/m ³)	Yield limit (MPa)
ZL101A	0.7×10^5	0.33	2 680	274
6061(T6)	0.7×10^5	0.33	2 550	355
Q235A	2.1×10^5	0.30	7 800	235

4.3. Working conditions and load calculation

Calculation specifications for levitation chassis of the maglev vehicles have not been established. Therefore, by analyzing the actual working conditions of magnetic levitation trains, and referring to the calculation of rail bogies, we consider the strength calculation conditions of the levitation chassis of maglev train as follows.

Condition 1: When the levitation chassis is under the condition of a static suspension, the loads on it are as follows: levitation chassis and vehicle body weigh

940.5 kg and 3 605.3 kg, respectively, and vehicle carrying load is 2 000 kg.

Condition 2: When the levitation chassis is in the normal driving state, the loads are as follows: thrust acceleration is 1.1 m/s^2 , vertical impact load is 1.5 m/s^2 , and other loading conditions are the same as condition 1.

Condition 3: When the levitation chassis is parking, the loads are as follows: levitation chassis weighs 940.5 kg, weight on the air spring is 2 639.3 kg, vehicle carrying load is 2 000 kg, and electromagnet weight is 966 kg.

Condition 4: When the levitation chassis is in the state of breaking, the loads are as follows: breaking acceleration is $0.25g$, vertical impact load is 1.5 m/s^2 , and other loading conditions are the same as condition 3.

4.4. Result analysis

4.4.1. Static stiffness analysis

The contour maps of the vertical deformation under four conditions are shown in Fig. 6.

(a) Condition 1

(b) Condition 2

(c) Condition 3

(d) Condition 4

Fig. 6 Vertical deformation contour map under four conditions

From the deformation maps mentioned above, the FEM computation results of static stiffness are obtained as shown in Table 2.

From Table 2, the following can be observed: the tiny vertical deformations of levitation chassis have a limited effect on the levitation gap between electromagnet and track when the gap is less than 8 mm, and the stiffness meets the requirement; the maximum deformation appears primarily in the suspender connecting two anti-rolling sills, indicating that the elastic damping unit has left and right decoupling function.

Table 2 Results of maximum vertical deformation

Conditions	Maximum deformation position	Maximum deformation value (mm)
1	Anti-rolling sills	0.974 3
2	Anti-rolling sills	1.305 7
3	Anti-rolling sills	0.333 5
4	Lower longitudinal beam	0.074 8

4.4.2. Strength analysis

Generally, von Mises equivalent stress of the fourth strength theory can be employed in analyzing the strength of steel, copper and aluminium and other plastic materials. The von Mises stress contour maps under the four conditions are shown in Fig. 7. The maximum stress and its position of the von Mises stress maps above are summarized in Table 3.

Table 3 reveals that under the four conditions the

Table 3 Results of maximum von Mises stress

Conditions	Maximum stress position	Maximum stress value (MPa)
1	Reinforcing plate of outrigger	15.318
2	Reinforcing plate of outrigger	19.459
3	Corbel	21.97
4	Corbel	10.888

(a) Condition 1

(b) Condition 2

(c) Condition 3

(d) Condition 4

Fig. 7 The von Mises stress contour maps under four conditions

stress in the outrigger or corbel is maximum, while comparatively smaller and uniform in other positions. The material of the outriggers and corbels is ZL101A aluminum alloy with a yield limit of 274 MPa. The maximum stress is much smaller than the yield limit and meets requirements of static strength with a surplus, which provides a basis for further optimization calculation.

5. Conclusions

A vertical dynamics model for levitation chassis is established with consideration of the factors of plunging, nodding, and rolling directions of vibration. Analysis of left-right decoupling through simulation confirms that the constraint parameters are reasonable and meet the decoupling requirements of levitation chassis, and can provide a reference for setting anti-roll suspension parameters for maglev vehicles. Based on the left-right decoupling, the finite element model of the levitation chassis is established. The FEM analysis results show that the levitation chassis can meet the design requirements in mechanical properties and the levitation chassis has enough rigidity and strength to carry the whole train. The results also provide a reference for reasonably determining the location of the stress test patch.

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References

- [1] Q. Han, Y.P. Yang, X.H. Yao, et al., Dynamics of urban-rail magnetic levitation vehicle with five bogies, *Journal of South China University of Technology (Natural Science Edition)*, 2010, **38**(1): 102-107 (in Chinese).
- [2] P.Z. Zhang, Research on the technology for the self-innovation of low /medium-speed maglev traffic engineering, *Journal of Railway Engineering Society*, 2009(10): 90-94 (in Chinese).
- [3] H.B. Jiang, S.H. Luo, Z.M. Dong, Analysis of ant-rolling sill on middle or low speed magnetic levitation trains, *Rolling Stock*, 2006, **44**(11): 8-10 (in Chinese).
- [4] F. Luo, K.L. Zhang, Numerical calculation and analysis of electromagnetic forces of U-shape suspension electromagnet for maglev vehicle, *Electric Drive for Locomotive*, 2002(3): 32-39 (in Chinese).
- [5] K. Popp, A. Kraus, T. Heiss, Dynamics analysis of a simple vehicle on a periodic guideway, *Vehicle System Dynamics*, 1982, **11**: 107-120.
- [6] J. Shi, Q.C. Wei, Studies on the magnet/rail relationship of electromagnetic suspension transport system, *Journal of Northern Jiaotong University*, 2004, **28**(4): 41-44 (in Chinese).
- [7] C.F. Zhao, W.M. Zhai, Dynamic Characteristics of Electromagnetic Levitation Systems, *Journal of Southwest Jiaotong University*, 2004, **39**(4): 464-468 (in Chinese).
- [8] W.M. Zhai, *Vehicle-Track Coupling Dynamics (Third Edition)*, Beijing: Science Press, 2007 (in Chinese).
- [9] W.M. Zhai, Two simple fast integration methods for large-scale dynamic problems in engineering, *International Journal for Numerical Methods in Engineering*, 1996, **24**(39): 4199-4214.
- [10] H.L. Yu, C.W. Xu, Finite element analysis of drillwell hook using Pro/E and ANSYS, *Manufacturing Automation*, 2009, **31**(10): 119-121 (in Chinese). (Editor: Dongju CHEN)