The vibration reduction optimization of the body tube fastening of EM rail launcher

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Abstract. In order to further optimize the vibration reduction effect of the body tube fastening of the electromagnetic rail launcher, the vibration reduction optimization scheme of the body tube fastening is determined by the modal superposition method. The electromagnetic rail launcher is simplified to Bernoulli-Euler beam, the vibration equation of the rail is established, and the inherent characteristics of the rail are analyzed. The multi-field coupled finite element model of the electromagnetic rail launcher is established and the complete transient launching process of the launcher is simulated. Based on the modal analysis of the system stiffness at the critical velocity, the optimal position of the tube fastening is determined, and the influence of different fastening positions on the damping effect is analyzed. The results show that the fastening of the body tube can effectively improve the structural stiffness of the electromagnetic rail launcher, and the damping effect of the fastening is affected by the vibration characteristics of the fastening position. The addition of fastening leads to the stress concentration in the adjacent part of the rail, which increases the requirement of rail strength. When the fastening is set in the front part of the rail, the vibration reduction effect is obvious. When it is set in the back part, the resonance range of the critical velocity should be avoided and the priority should be given to the position before the critical velocity.

1. Introduction
Electromagnetic launch technology is a new concept weapon technology that uses electromagnetic force to propel the load to high speed [1,2]. During the launching process of the electromagnetic rail launcher, resonance will occur when the armature speed reaches the critical speed, resulting in severe vibration and deflection deformation of the rail [3]. Due to the existence of electromagnetic impact force, the severe vibration and deflection of the rail will inevitably lead to the planing damage of the rail, which will affect the stability and service life of the rail launch [4]. Therefore, in practical applications, a pre-tightening force is applied to the launcher body-management to increase the structural rigidity and achieve the purpose of vibration reduction [5]. Many scholars have carried out in-depth research on the vibration characteristics and damping of electromagnetic rail launchers. WU L Z et al. conducted a coupling simulation on the bolt fastening package, and calculated the diameter deformation of the launcher under different bolt preloads, but they ignored a variety of influencing factors, such as pivot-rail interference fit, speed skin effect, etc. [6]; WANG Z C et al. proposed to balance the rail by applying hydraulic servo preload, however, the impact of the rail transient vibration response was not taken into account when the pre-tension position was set [7]; the literature [8] used
fiber grating strain sensors to measure the rail strain, pointing out that the propagation of stress waves at critical speeds would cause the rail to produce large stress mutation; the literature [9] pointed out that the damping effect between rails can alleviate the deflection deformation of the rail at critical speed; the literature [10] analyzed the vibration characteristics of the rail under different structural stiffness and preload. These works are not based on the analysis of the spatial distribution characteristics of the stress load during the transient launch of the complete rail, which will inevitably reduce the calculation accuracy of the study.

In this paper, a multi-field coupled finite element model of the electromagnetic rail launcher is established, and the complete transient launch process of the launcher is simulated, and the dynamic response of the pivot rail during the transient launch process is obtained. The modal is performed based on the system stiffness at the critical speed moment. The superposition analysis determines the optimal position of the body-management fastening. Finally, the optimization effects of different fastening positions are compared and analyzed, and further optimization directions for body-management fastening and vibration reduction are proposed, which have guiding significance for the optimization of vibration reduction of electromagnetic rail launchers.

2. Materials and Methods

2.1. Dynamics analysis

The basic structure of the electromagnetic rail launcher is shown in Figure 1, where the package and elastic support are used to resist deformation of the rail.

![Fig.1 The schematic diagram of electromagnetic rail launcher section](image)

In order to facilitate the analysis of dynamics theory, the rail of the launcher can be simplified as a Bernoulli-Euler beam, as shown in Figure 2. Then the dynamic equation of the rail is:

\[
EI \frac{\partial^4 y}{\partial x^4} + m \frac{\partial^2 y}{\partial t^2} + ky = F(x, t)
\]  

(1)
In the formula, $E$ is the elastic modulus of the rail material, $I$ is the moment of inertia of the rail section, $m$ is the rail linear density, $m = \rho A$, $\rho$ is the rail material density, $A$ is the rail cross-sectional area, and $k$ is the elastic support stiffness. The moving load $F(x,t)$ can be expressed as:

$$F(x,t) = q(x,t)(1 - H(x - vt)) + f(x,t)\delta(x - vt)$$

In the formula, $q(x,t)$ is the electromagnetic repulsive force between the rails, $f(x,t)$ is the pressing force of the armature against the rails, $H$ is the Heaviside step function, and $\delta$ is the Dirac pulse function.

Based on elastic mechanics analysis, it can be obtained that when the armature speed is close to the critical speed, it will cause the pivot rail resonance, resulting in a sharp increase in the stress concentration of the rail. The critical speed expression is:

$$v_{cr} = \sqrt[4]{\frac{4EIk}{m^2}}$$

### 2.2. Finite element model

In the basic finite element model established, the geometric parameters of the launcher rail are $1000 \text{mm} \times 30 \text{mm} \times 10 \text{mm}$, the caliber of the transmitter is $20 \text{mm} \times 45 \text{mm}$, and the material parameters of the armature and rail are shown in Table 1. According to formula (3), the critical velocity of the rail is $1448.66 \text{m/s}$.

| Tab.1 The material parameters of armature and rail |
|---------------------------------------------------|
| material parameters | armature | rail | elastic support | unit           |
| material density $\rho$ | aluminum | copper |                | Kg / m$^3$     |
| elastic modulus $E$ | 2820     | 8900   |                | GPa           |
| foundation stiffness $k_0$ | 70       | 124    |                | $8.44 \times 10^{11}$ N / m$^3$ |

According to the above-mentioned parameter settings, the electromagnetic rail launcher is simulated by multi-field coupling based on LS-DYNA, and the current, armature speed and displacement changes with time during the transient launch process are obtained, as shown in Figure 3. It can be seen from Figure 3 that the armature reaches the critical speed of the rail at 525mm from the starting end at $t=0.69$ms.

![Fig.3 The Changes of current, velocity and displacement over time](image_url)

### 2.3. Vibration reduction optimization based on critical speed moment

The optimization process in this section is: first perform a nonlinear static analysis of the electromagnetic rail launcher, and then perform a modal superposition analysis based on the system stiffness at the critical speed, and finally define constraint points based on modal nodes. Figure 4 is a
schematic diagram of fastening, where a, b, and c are measuring points, respectively 250mm, 375mm, and 625mm from the starting end.

![schematic diagram of fastening](image)

Fig.4 The schematic diagram of fastening mode

Table 2 shows the participation coefficients and cumulative quality factors of the first 10 modes of the rail axial mode. Up to the 10th order, the sum of the effective masses of each order is approximately equal to the total mass of the structure, and the modal order is sufficient. Normalize the participation coefficient of each mode to obtain the mode superposition coefficient.

| order | frequency/Hz | participation coefficient | superposition factor | cumulative quality factor |
|-------|--------------|---------------------------|----------------------|--------------------------|
| 1     | 1879.25      | 4.65E-02                  | 0.308968             | 0.441410                 |
| 2     | 3440.13      | 4.02E-02                  | 0.266992             | 0.771039                 |
| 3     | 3757.87      | 0                         | 0.771039             |                          |
| 4     | 3931.18      | 2.51E-02                  | 0.167089             | 0.900137                 |
| 5     | 5635.15      | 1.55E-02                  | 0.102947             | 0.949145                 |
| 6     | 6881.29      | 7.82E-04                  | 0.005196             | 0.949270                 |
| 7     | 7510.23      | 0                         | 0.949270             |                          |
| 8     | 7848.73      | 3.77E-04                  | 0.002504             | 0.949299                 |
| 9     | 9381.76      | 9.28E-03                  | 0.061698             | 0.966902                 |
| 10    | 10321.2      | 1.27E-02                  | 0.084601             | 1.00000                  |

The Design Assessment module is used for modal superposition analysis, and the deformation of the upper and lower surfaces of the rail is shown in Figure 5.

![deformation cloud map](image)

Fig.5 The middle line deformation cloud map of upper and lower surface of rail

The deformation value in the modal analysis is the normalized deformation, which is used to qualitatively observe the resonance area and vibration amplitude of the natural vibration mode of the structure. Create a selection set of all nodes based on the centerline of the upper and lower outer surfaces of the rail, and the derived node numbers and corresponding deformations are sorted. Find the node combination with the minimum total deformation in the node set: point d, the total deformation is 8.2874mm, 270mm from the starting end; point e, the total deformation is 8.4269mm, 660mm from the starting end.
3. Results & Discussion

3.1. Comparison of optimization results

According to the optimization plan of the tightening position, the d-point tightening and the e-point tightening are simulated and compared. Figure 6 shows the changes in deflection of the rail at various points before and after optimization with time, and Figure 7 shows the changes in the maximum shear stress at each position of the rail before and after optimization.

From Figure 6, it can be seen that the three fastening methods all have a good damping effect. Compared with the mid-point fastening, the d-point fastening has a further improved damping effect on the front of the rail. When the armature moves to the rear of the rail, there is a deflection fluctuation, which means that the d-point fastening has a distance attenuation. Similarly, the e-point fastening has a stronger damping effect on the rear of the rail.

![Fig.6 Deflection at point a, b, c before and after optimization](image)

It can be seen from Figure 7 that all three types of fastening methods have stress concentration near the fastening point. Among them, the stress mutation at point Q near the fastening point e is the most obvious, and the maximum shear stress is 100.79Mpa. This is because with the movement of the armature, the rail deflection fluctuations gradually extend to the muzzle, and the fastening point e is in the severe resonance range of the rail. At the same time, the point P near the fastening point d also produced a sudden change in stress, and the maximum shear stress was 96.69 MPa. This is due to the large acceleration of armature motion in the initial stage of launch, which leads to large acceleration of rail vibration. Although the rail midpoint is closer to the critical speed position than the e point, the concentrated stress when using the midpoint fastening method only reaches 44MPa. It can be seen that the resonance caused by the critical velocity has less influence on the rail behind the armature movement than the front.
3.2. Evaluation index analysis

The vibration transmission rate $\eta$ is introduced in the project to facilitate the comparison of the effects before and after vibration reduction:

$$\eta = \frac{a_h}{a_q}$$

(4)

In the formula, $a_h$ is the rail vibration acceleration value after vibration reduction, and $a_q$ is the rail vibration acceleration value before vibration reduction.

In the same way, the stress change rate $\gamma$ is introduced to facilitate the comparison of the rail strength changes before and after vibration reduction:

$$\gamma = \frac{\tau_h - \tau_q}{\tau_q}$$

(5)

In the formula, $\tau_h$ is the maximum shear stress value of the rail after vibration reduction, and $\tau_q$ is the maximum shear stress value of the rail before vibration reduction.

**Tab.3 The rail vibration acceleration value and transfer rate before and after optimization**

| Fastening method | Position /m | Before damping /$10^{-6}$m/s$^2$ | After damping /$10^{-6}$m/s$^2$ | Transmission rate /% |
|------------------|-------------|----------------------------------|---------------------------------|---------------------|
| Mid-point fastening | 0.25 | 9.31 | 2.19 | 23.54 |
| | 0.375 | 14.43 | 1.08 | 7.47 |
| | 0.625 | 13.29 | 0.10 | 0.76 |
| D point fastening | 0.25 | 9.31 | 0.03 | 0.28 |
| | 0.375 | 14.43 | 0.48 | 3.36 |
| | 0.625 | 13.29 | 1.01 | 7.62 |
| E point fastening | 0.25 | 13.29 | 7.94 | 85.27 |
| | 0.375 | 14.43 | 7.25 | 50.23 |
| | 0.625 | 13.29 | 0.25 | 1.88 |

**Tab.4 The maximum shear stress value and the rate of stress change before and after optimization**

| Fastening method | Position /m | Before damping /MPa | After damping /MPa | Rate of change /% |
|------------------|-------------|---------------------|-------------------|------------------|
| Mid-point fastening | 0.52 | 26.70 | 44.04 | 64.90 |
| D point fastening | 0.29 | 63.87 | 96.69 | 51.39 |
| E point fastening | 0.64 | 15.70 | 100.79 | 541.85 |

It can be seen from Table 3 that the transmission rate of each measuring point of the rail is less than 10% when the d-point is fastened, and its vibration damping effect is the best; the mid-point fastening has a slight damping effect on the front of the rail compared to the d-point fastening Poor; and the vibration reduction effect of e-point fastening is the worst. It can be seen from Table 4 that the stress change rate of fastening at point d is also the smallest, and it has the lowest requirement for rail...
strength. Therefore, considering the damping effect of fastening and the negative effect of stress concentration, the d-point fastening method based on the simulation model can be considered as the optimal damping solution.

4. Conclusions
Concluded as follow:

1) The vibration damping effect of tightening is affected by the vibration characteristics of the tightening position, and tightening constraints will cause large shear stress, which in turn may cause damage. Therefore, when optimizing vibration reduction, comprehensive consideration should be given to rail strength and rigidity requirements, and appropriate locations should be selected to add fastenings to appropriately increase rail strength requirements and reduce vibration effects.

2) The single-point fastening has achieved a good vibration reduction effect. For the problem of the weakening of the remote end of the vibration damping effect, try to perform secondary modal superposition on the basis of the existing fastening, and determine the constraints of multiple modal nodes to achieve a better damping effect.

3) During the launch process, the vibration amplitude of the front section of the rail is large, and the vibration reduction effect of the front section of the fastened rail is more obvious; when the rear section of the rail is fastened, the critical velocity effect should be avoided, and the deflection fluctuation of the rail gradually moves towards the muzzle. It is better to extend and tighten before the critical speed position.

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