Research on armature vibration characteristics of electromagnetic gun's extreme sliding electrical conta

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Abstract. The electromagnetic railgun is a new launch method that uses electromagnetic force to drive payloads and accelerate objects to super high speeds. In the electromagnetic railgun system, the armature is a key component that converts electrical energy into kinetic energy of projectile. It will vibrate when it moves between rails, and the vibration characteristic is the behavior when the armature slides along the rail. Armature vibration will affect firing accuracy, damage integrity of rail, and affect life of rail. Armature vibration may also cause arc discharge, which will damage the armature, consume energy, and reduce the launching efficiency of the electromagnetic railgun. In order to provide a more reasonable design for the armature, based on the finite element simulation software ANSYS, a modal analysis was performed on the armature during the launch process, and the possible modes of its free vibration were simulated. The analysis found that the vibration modes are mainly divided into translational movements along the X, Y, and Z directions and torsional movements along the X, Y, and Z axes. Whether or not the prestress is loaded on the armature and the magnitude of the prestress does not significantly affect its vibration mode. Changing the armature structure and materials will affect the vibration mode of the armature accordingly.

1. Introduction
In the electromagnetic railgun system, the energy conversion is reflected in the armature as the conversion of electrical energy to kinetic energy. The armature is a key component of energy conversion [1]. The first requirement for an armature is the ability to transfer current from one rail to another. Therefore, the performance of the armature is directly related to the performance of the railgun, which determines the launch stability of the entire system.

The vibration characteristic is the behavior when the armature slides along the rail. The armature vibration will cause the projectile to vibrate, because the projectile is at the front of the armature. In this case, when the armature and the projectile fly away from the muzzle, the outer ballistic trajectory will deflect and affect the accuracy of the shot. In addition, the vibration of the armature will cause an impact on the rail. Due to the extremely short launch time of the railgun, the impact will produce a local high-intensity lateral load in an instant. This process can easily cause planing pits on the rail surface and damage the integrity of the track, thus, affecting the orbital life [2]. The armature vibration will also bring deformation and wear of itself, which may change the contact state of the armature and the rail, resulting in intermittent contact at the interface, which means that arc discharge may occur. The arc will
damage the armature, and will also consume a lot of energy, reducing the launching efficiency of the railgun [3]. In view of the complexity and variety of possible vibrations during the sliding of the armature, a preliminary understanding of the vibration of the armature is started from the most basic mode, that is, the free vibration, in order to propose methods to effectively suppress its vibration, and help to optimize the design of the armature structure and materials.

2. Extreme sliding electrical contact and Electromagnetic force in electromagnetic launch
In current-carrying sliding electrical contact, electrical, thermal, and mechanical effects will occur on the contact surface, and these effects will change the state of the contact surface [4]. In the process of relative sliding under load, most of the work done by friction will be converted into thermal energy. At the same time, the contact resistance will also generate a lot of Joule heat, which will increase the temperature of contact surface. The high temperature will cause the armature and metal materials of rail to change state, forming a liquid metal film, which will cause great damage to the armature and rails [5].

The limits of the extreme sliding electrical contact are reflected in its large working current, which can reach the order of MA, and the maximum current density on the electrical contact surface is close to the current bearing limit of the conductor material, the contact pressure is strong, the action time is extremely short, and the thermal effect is significant. The sliding electrical contact under extreme conditions is more extreme than the case under small and medium current loads. It is accompanied by more severe mechanical effects, thermal effects, and arcs and electric sparks caused by intermittent contact during sliding. The abrasion and wear of the contact surface, the deformation of the material, and the change in the surface state of the material all affect the operation of the system.

The propulsive force related to electromagnetic launch is an electromagnetic force generated by a high current exhibited by the Lorentz force. The propulsive force is generated from a megaampere-level current conducted through the rail and the armature. Due to the existence of large currents, electric and magnetic fields are locally generated in the rail, and the armature is accelerated according to the principle of Lorentz force [6]. Figure 1 shows a simple diagram of electromagnetic railgun circuit.

![Figure 1. Diagram of electromagnetic railgun circuit.](image)

Position 1 in figure 1 is the starting position of the armature, and it reaches position 2 after elapse of time \( dt \). Assuming the power electromotive force is \( V \), the current flowing into the railgun is \( I \), and the armature displacement is \( dx \), the inductance increment of this distance from the rail is \( dL \). Assuming that the magnitude of the current \( I \) is constant, the inductance value of the unit length of the track is \( L' \), and \( F \) is the propulsive force acting on the armature, then the mechanical work done by \( F \) is shown as follows.

\[
W_{m} = F \, dx \quad \text{\textsuperscript{* MERGEFORMAT (1)}}
\]

The increase in the induced magnetic energy of the railgun during this distance is as follows.

\[
W_{i} = \frac{dLI^{2}}{2} = \frac{L' I^{2} dx}{2} \quad \text{\textsuperscript{* MERGEFORMAT (2)}}
\]

According to Faraday’s law \( V = d\Phi/dt \), the voltage \( V \) in the circuit is equal to the rate of change of its magnetic flux \( d\Phi/dt \), that is,

\[
V = d(\Phi I)/dt = L' I \, dx/dt = L' I v \quad \text{\textsuperscript{* MERGEFORMAT (3)}}
\]
The work $W_g$ transferred to the circuit during this time is shown as follows.

$$W_g = \int t' \, dv = L' \int L' \, d\alpha$$  \hspace{1cm} ^{\text{MERGEFORMAT} (4)}$$

According to the Law of conservation of energy $W_g = W_m + W_1$, it is concluded as follows.

$$F = \frac{1}{2} L' \, I^2$$  \hspace{1cm} ^{\text{MERGEFORMAT} (5)}$$

3. Finite element simulation of armature

The natural frequency and modal shape of the free vibration of the armature with or without prestress are determined by analysis. The material selection and structural design of the armature are analyzed. For prestressed cases, the armature was analyzed using a maximum contact pressure standard that did not exceed the structural analysis. The importance of modal characteristics is reflected in its behavior as the armature slides inside the barrel along the rail. During this entire launch period, a good constant contact between the armature arm and the rail is required to maintain a constant current flow. However, the mode and size of the vibration modal may cause intermittent contact at the pivot-rail interface, which will affect the normal operation of the entire electromagnetic railgun system.

3.1. Establishment of Armature Geometric Model

Figure 2 shows the plane structure of armature used in this modal analysis [7].

![Figure 2. Plane structure of armature.](image)

3.2. Material Properties

Three metals (aluminum, copper and titanium) are selected for comparison in the armature material analysis. The parameter attributes are shown in table 1.

| Attribute Parameter | aluminum | copper | titanium |
|---------------------|----------|--------|----------|
| Elastic Modulus /GPa| 70       | 119    | 102.04   |
| Poisson's ratio     | 0.33     | 0.3    | 0.326    |
| density / g/ cm³    | 2.7      | 8.9    | 4.51     |

3.3. Element types and meshing

As shown in figure 3, the mesh division adopts the method of free mesh division. The element type used is Solid186 element, which is defined by 20 nodes, each node has three degrees of freedom: translation in the X, Y, and Z directions of the node coordinate system.
3.4. Boundary conditions and loads

In order to simulate the impact of the current, the electromagnetic force generated by the current is loaded on the armature. Due to the speed skin effect, the current share of the armature tail is large, so an equivalent electromagnetic force is loaded on the armature tail. The magnitude of the electromagnetic force is calculated according to the formula (5), taking the inductance gradient $L'$ as 0.42$\mu$H/m, and current $I$ as 1800kA, so the electromagnetic force $F$ is $6.804\times10^5$ N [8].

As shown in figure 4, the prestress is loaded along the edge of the pivot tail, and its directions on both sides are away from the track. The magnitude of prestress does not exceed the maximum contact pressure obtained from the structural analysis, which is 0.215GPa.

4. Analysis of Numerical Simulation Results
4.1. Modal Analysis

The results of the modal analysis show that the first-order modal mode is in-phase bending in the y-direction, the second-order modal mode is different-direction bending in the y-direction, and the third-order modal mode is bending in the z-direction, the fourth-order modal mode is bending in the x-direction, the fifth-order modal mode is torsion along the x-axis, the sixth-order modal mode is torsion along the z-axis, the seventh-order mode mode is the superposition of the torsion along the z-axis and the in-phase bending of the pivot tail in the y-direction, the eighth-order modal mode is torsion along the y-axis. Based on the comprehensive analysis results, the modal modes can be divided into six types of motion, namely translation in the X, Y, and Z directions, and torsion in the X, Y, and Z axes, or a superposition of these six types of motion. The six types of motion are shown below.

![Outline figure](image1.png)  ![Vector figure](image2.png)  
**Figure 6.** Translation in x-direction.

![Outline figure](image3.png)  ![Vector figure](image4.png)  
**Figure 7.** Translation in y-direction.

![Outline figure](image5.png)  ![Vector figure](image6.png)  
**Figure 8.** Translation in z-direction.
From the perspective of the natural modes of each order, the impact of different modes is also different. For example, from the perspective of the type of translation in the y direction, when the pivot arms are bent in opposite directions toward the two sides of the rail, as shown in figure 12, it will generate greater pressure on the rail, but still maintain the pivot track. However, when the pivot arm is bent in the same direction, as shown in figure 7 (a), one side of the pivot arm will inevitably lose contact with the rail, which will affect the normal flow of current and cause a series of damage, resulting in the normal operation of the entire electromagnetic railgun system is affected. Similarly, in the case of torsion along the z axis, there may also be a phenomenon that the pivot arm loses contact with the rail. And according to the results, it can be seen that the armature's displacement due to vibration is more serious at the tail of armature, so the possibility of loss of contact at the armature tail is higher. Considering that as the speed increases, the current is mainly concentrated at the tail of armature. Therefore, the loss of contact has a huge impact on the normal operation of the electromagnetic railgun system.
4.2. Comparison of Armature Material

Table 2. Natural frequency and natural period of armature vibration of different materials.

| Order   | Natural Frequency (kHz) | Natural Period (μs) | Copper (μs) | Titanium (μs) |
|---------|-------------------------|---------------------|-------------|---------------|
| first   | 15.677                  | 14.595              | 11.253      | 63.78771449   | 68.51601528 | 68.8651915 |
| second  | 16.938                  | 15.778              | 12.159      | 59.03884756   | 63.37938902 | 82.2436056 |
| third   | 20.166                  | 18.78               | 14.475      | 49.58841615   | 53.24813632 | 69.0846287 |
| fourth  | 20.552                  | 19.151              | 14.754      | 48.65706501   | 52.21659443 | 67.7782296 |
| fifth   | 22.293                  | 20.809              | 16.007      | 44.8571304    | 48.05612956 | 62.4726082 |
| sixth   | 22.586                  | 21.091              | 16.218      | 44.27521473   | 47.41358873 | 61.6598841 |
| seventh | 25.885                  | 24.143              | 18.585      | 38.63241259   | 41.41987236 | 53.8068335 |
| eighth  | 27.276                  | 25.454              | 19.584      | 36.6626719    | 39.28655614 | 51.0620195 |
| ninth   | 27.591                  | 25.789              | 19.815      | 36.24270266   | 38.77622242 | 50.4668181 |
| tenth   | 29.428                  | 27.528              | 21.126      | 33.98124225   | 36.32664923 | 47.3126419 |
| eleventh| 34.641                  | 32.427              | 24.884      | 28.86752602   | 30.83849878 | 40.1864052 |
| twelfth | 37.128                  | 34.767              | 26.676      | 26.93385046   | 28.76290735 | 37.4053313 |
| thirteenth | 41.311             | 38.755              | 29.683      | 24.20662777   | 25.80312218 | 33.6893171 |
| fourteenth | 50.192            | 46.863              | 36.041      | 19.92349378   | 21.33870607 | 27.746178  |
| fifteenth | 51.473             | 48.217              | 36.977      | 19.4276611    | 20.73957318 | 27.0438381 |

Table 3. Maximum displacement of armature vibration of different materials.

| Order   | Copper (mm) | Titanium (mm) | Aluminum (mm) |
|---------|-------------|---------------|---------------|
| first   | 0.34        | 0.26          | 0.187         |
| second  | 0.409       | 0.313         | 0.225         |
| third   | 0.219       | 0.169         | 0.121         |
| fourth  | 0.174       | 0.134         | 0.096         |
| fifth   | 0.252       | 0.195         | 0.139         |
| sixth   | 0.222       | 0.171         | 0.122         |
| seventh | 0.403       | 0.309         | 0.221         |
| eighth  | 0.271       | 0.201         | 0.148         |
| ninth   | 0.368       | 0.288         | 0.203         |
| tenth   | 0.439       | 0.337         | 0.241         |

It can be seen from the results of Table 2 and Table 3 that the largest natural period of vibration is 88.87 μs in the first-order case of the copper armature, which is much shorter than the movement time of the armature in the barrel. Therefore, intermittent contact between the armature and the rail may occur when the armature moves in the barrel.

From the perspective of the natural frequency of each order and the maximum displacement, as the
elastic modulus of the material increases, its natural frequency of vibration decreases and its amplitude decreases. It shows that the method of increasing stiffness to suppress vibration is feasible.

4.3. Comparison of armature structure

For the armature structure, a comparison was made from the length $L$ of contact surface, the radial depth $d$ at the tail, the maximum depth $D$ of armature, and the size of the caliber. As shown below.

![Diagram of armature size](image)

**Figure 13.** Diagram of armature size.

The length of contact surface was compared for armatures of 25mm, 30mm, 35mm, and 40mm respectively, the radial depth at the tail was compared for armatures of 3mm, 5mm, and 7mm respectively. The maximum depth of armature is compared with the armatures of 8mm and 9mm respectively, and armature calibers are compared for three caliber sizes, 30mm * 30mm, 40mm * 40mm, and 50mm * 50mm. The results are shown below.

**Table 4.** Natural frequency of vibration at each order with different length of contact surface.

|                 | 25mm(kHz) | 30mm(kHz) | 35mm(kHz) | 40mm(kHz) |
|-----------------|-----------|-----------|-----------|-----------|
| first-order     | 15.677    | 13.428    | 12.373    | 10.875    |
| second-order    | 16.938    | 14.204    | 12.837    | 11.154    |
| third-order     | 20.166    | 17.973    | 17.657    | 16.422    |
| fourth-order    | 20.552    | 19.127    | 18.654    | 17.21     |
| fifth-order     | 22.293    | 19.287    | 19.443    | 18.34     |
| sixth-order     | 22.586    | 21.731    | 20.812    | 18.559    |
| seventh-order   | 25.885    | 22.53     | 21.17     | 18.614    |
| eighth-order    | 27.276    | 22.962    | 21.38     | 19.69     |
| ninth-order     | 27.591    | 23.858    | 22.741    | 22.052    |
| tenth-order     | 29.428    | 25.59     | 24.681    | 22.245    |

**Table 5.** Maximum displacement of vibration at each order with different length of contact surface.

|                 | 25mm(mm) | 30mm(mm) | 35mm(mm) | 40mm(mm) |
|-----------------|----------|----------|----------|----------|
| first-order     | 0.34     | 0.286    | 0.249    | 0.205    |
| second-order    | 0.409    | 0.332    | 0.261    | 0.211    |
| third-order     | 0.219    | 0.214    | 0.209    | 0.191    |
| fourth-order    | 0.174    | 0.162    | 0.152    | 0.108    |
| fifth-order     | 0.252    | 0.248    | 0.167    | 0.191    |
| sixth-order     | 0.222    | 0.208    | 0.166    | 0.161    |
| seventh-order   | 0.403    | 0.392    | 0.376    | 0.354    |
| eighth-order    | 0.271    | 0.259    | 0.247    | 0.232    |
| ninth-order     | 0.368    | 0.352    | 0.332    | 0.317    |
| tenth-order     | 0.439    | 0.427    | 0.408    | 0.393    |
Table 6. Natural frequency of vibration at each order with different depth of tail radial.

|          | 3mm(kHz) | 5mm(kHz) | 7mm(kHz) |
|----------|----------|----------|----------|
| first-order | 14.169   | 15.677   | 16.668   |
| second-order | 15.286   | 16.938   | 17.943   |
| third-order  | 19.802   | 20.166   | 20.538   |
| fourth-order | 19.892   | 20.552   | 20.972   |
| fifth-order  | 21.673   | 22.293   | 22.661   |
| sixth-order  | 22.316   | 22.586   | 23.197   |
| seventh-order| 25.346   | 25.885   | 26.207   |
| eighth-order | 26.325   | 27.276   | 27.672   |
| ninth-order  | 26.576   | 27.591   | 28.513   |
| tenth-order  | 27.966   | 29.428   | 30.201   |

Table 7. Maximum displacement of vibration at each order with different depth of tail radial.

|          | 3mm(mm) | 5mm(mm) | 7mm(mm) |
|----------|---------|---------|---------|
| first-order | 0.407   | 0.34    | 0.31    |
| second-order | 0.483   | 0.409   | 0.368   |
| third-order  | 0.23    | 0.219   | 0.205   |
| fourth-order | 0.177   | 0.174   | 0.168   |
| fifth-order  | 0.259   | 0.252   | 0.248   |
| sixth-order  | 0.264   | 0.222   | 0.212   |
| seventh-order| 0.525   | 0.403   | 0.323   |
| eighth-order | 0.392   | 0.271   | 0.247   |
| ninth-order  | 0.372   | 0.368   | 0.365   |
| tenth-order  | 0.465   | 0.439   | 0.405   |

Table 8. Natural frequency of vibration at each order with different maximum armature depth.

|          | 8mm(kHz) | 9mm(kHz) |
|----------|----------|----------|
| first-order | 15.506   | 15.677   |
| second-order | 16.441   | 16.938   |
| third-order  | 20.608   | 20.166   |
| fourth-order | 20.74    | 20.552   |
| fifth-order  | 22.465   | 22.293   |
| sixth-order  | 22.576   | 22.586   |
| seventh-order| 24.589   | 25.885   |
| eighth-order | 27.013   | 27.276   |
| ninth-order  | 27.569   | 27.591   |
| tenth-order  | 28.845   | 29.428   |

Table 9. Maximum displacement of vibration at each order with different maximum armature depth.

|          | 8mm(mm) | 9mm(mm) |
|----------|---------|---------|
| first-order | 0.394   | 0.34    |
| second-order | 0.439   | 0.409   |
| third-order  | 0.205   | 0.219   |
| fourth-order | 0.234   | 0.174   |
| fifth-order  | 0.258   | 0.252   |
| sixth-order  | 0.26    | 0.222   |
| seventh-order| 0.427   | 0.403   |
| eighth-order | 0.302   | 0.271   |
| ninth-order  | 0.374   | 0.368   |
| tenth-order  | 0.462   | 0.439   |
Table 10. Natural frequency of vibration at each order with different calibers.

| Order   | 30mm*30 mm(kHz) | 40mm*40 mm(kHz) | 50mm*50 mm(kHz) |
|---------|-----------------|-----------------|-----------------|
| first-order | 12.373          | 9.8312          | 9.7164          |
| second-order | 12.837          | 10.071          | 10.313          |
| third-order | 17.657          | 15.679          | 13.079          |
| fourth-order | 18.654          | 15.727          | 13.3            |
| fifth-order | 19.443          | 16.806          | 14.048          |
| sixth-order | 20.812          | 16.825          | 14.733          |
| seventh-order | 21.17           | 17.814          | 16.085          |
| eighth-order | 21.38           | 18.301          | 17.48           |
| ninth-order | 22.741          | 18.509          | 18.064          |
| tenth-order | 24.681          | 20.628          | 18.907          |

Table 11. Maximum displacement of vibration at each order with different calibers.

| Order   | 30mm*30 mm(mm) | 40mm*40 mm(mm) | 50mm*50 mm(mm) |
|---------|----------------|----------------|----------------|
| first-order | 0.349          | 0.299          | 0.199          |
| second-order | 0.361          | 0.317          | 0.228          |
| third-order | 0.239          | 0.15           | 0.085          |
| fourth-order | 0.252          | 0.115          | 0.114          |
| fifth-order | 0.247          | 0.234          | 0.114          |
| sixth-order | 0.466          | 0.229          | 0.132          |
| seventh-order | 0.376          | 0.243          | 0.188          |
| eighth-order | 0.347          | 0.326          | 0.311          |
| ninth-order | 0.362          | 0.33           | 0.315          |
| tenth-order | 0.243          | 0.182          | 0.157          |

Based on the analysis of the results shown above, it can be known that increasing the length of the contact surface of the armature can reduce the natural frequencies and amplitudes of the armature vibration at each order, and weaken the vibration of the armature to a certain extent. However, when the length of the contact surface is increased to a certain value, the attenuation of the vibration is also reduced. As the radial depth of the armature tail is increased, the natural frequencies of the armature vibration at each order are slightly increased, and the amplitudes at each order are reduced, but as the radial depth continues to increase, the amplitude start to decrease slightly. It can be seen that increasing the radial depth of the armature tail has a certain effect on reducing the vibration of the armature. Increasing the maximum armature depth has little effect on the low-order natural frequency, and the high-order natural frequency increases slightly, at the same time, the maximum displacement of the vibration will decrease, so increasing the maximum armature depth can weaken the vibration of the armature. And increasing the caliber of the armature can reduce the vibration frequency and the maximum displacement of the armature vibration at each order. However, increasing the caliber of the armature will increase the volume of the railgun, so it is not necessarily that the larger the caliber, the better.

Based on the comparison of the above structures, increasing the length of contact surface of the armature, increasing the radial depth of the armature tail, increasing the maximum depth of the armature, and increasing the caliber can reduce the vibration of the armature.

4.4. Modal vibration under damage

Calculating the natural frequency of the mode also helps to judge the damage of the armature to a certain extent. Leave a crack in the middle of the upper arm of the armature to simulate the damage of the armature. As shown in the figure 14.
Figure 14. Model of damaged armature.

A total of six different damage conditions are selected, as shown in the table below.

| Table 12. Six different sizes of damage. |
|----------------------------------------|
|      | damage1     | damage2     | damage3     |
| size | 0.1mm*0.05mm*30mm | 0.2mm*0.1mm*30mm | 0.4mm*0.2mm*30mm |
|      | damage4     | damage5     | damage6     |
| size | 0.6mm*0.3mm*30mm | 0.8mm*0.4mm*30mm | 1mm*0.5mm*30mm |

Table 13. Natural frequencies in six different sizes of damage.

| order | damage 1(kHz) | damage 2(kHz) | damage 3(kHz) | damage 4(kHz) | damage 5(kHz) | damage 6(kHz) |
|-------|---------------|---------------|---------------|---------------|---------------|---------------|
| first-order | 15.389        | 15.453        | 15.488        | 15.545        | 15.594        | 15.651        |
| second-order | 16.886        | 17.008        | 17.078        | 17.417        | 17.643        | 17.997        |
| third-order | 19.658        | 19.676        | 19.703        | 19.741        | 19.787        | 19.805        |
| fourth-order | 19.765        | 19.826        | 19.792        | 19.87         | 19.887        | 19.928        |
| fifth-order | 21.812        | 21.924        | 21.949        | 21.988        | 21.972        | 22.073        |
| sixth-order | 22.228        | 22.352        | 22.461        | 22.619        | 22.862        | 23.277        |
| seventh-order | 25.531        | 25.519        | 25.607        | 25.641        | 25.731        | 25.738        |
| eighth-order | 26.314        | 26.327        | 26.15         | 26.293        | 26.364        | 26.454        |
| ninth-order | 27.057        | 27.227        | 27.325        | 27.385        | 27.469        | 27.754        |
| tenth-order | 29.285        | 29.667        | 30.067        | 30.983        | 31.565        | 32.462        |

It can be seen from Table 13 that the damage of the contact surface of the pivot arm and the rail has an influence on the natural frequency, and the more severe the damage, the higher the natural frequency. Taking damage 1 and damage 6 as examples, when the size of damage is changed from size 1 to size 6, the natural frequency of vibration increases by 1.7% at first-order, 6.6% at second-order, 0.7% at third-order, 0.8% at fourth-order, 1.2% at fifth-order, and 4.7% at sixth-order.

Since the natural frequency is a unique attribute of the armature vibration, the damage of the armature can be judged to a certain extent according to the change of the natural frequency.

5. Summary
As one of the key components of the electromagnetic rail gun, the armature's performance is directly related to the performance of the electromagnetic railgun. In this paper, the vibration modal analysis of the armature of the electromagnetic railgun is realized, and the corresponding vibration suppression methods are proposed based on the analysis results through the simulation of ANSYS. When the armature slides in the barrel, its vibration mode is mainly divided into translation in the X, Y, and Z directions and torsion in the X, Y, and Z axes, and the vibration period is much shorter than the movement time in the barrel, so the vibration mode may cause the armature to lose contact with the rail. From the perspective of the armature material, increasing the elastic modulus of the material can suppress the vibration of the armature. And the structure of armature also affects the vibration. The vibration of the armature can be reduced by increasing the length of contact surface of the armature,
the radial depth of the armature tail, the maximum depth of armature, and the caliber. In addition, the modal analysis of the armature can also provide a reference for judging the damage of the armature.

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