Parametric design of MEMS fuze safety release insurance mechanism

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Abstract. At present, MEMS safety systems have the disadvantages of complex structure and easy to be affected by the electromagnetic environment of the battlefield. Based on this, this paper proposes a planar structure to realize remote protection of MEMS security systems. First, a theoretical analysis of the MEMS remote release insurance mechanism is carried out. It mainly calculates the centrifugal force of the centrifugal slider, the magnitude of the deflection of the threshold elastic beam caused by the active tooth collision fixed tooth, and the time when the active tooth of the safety mechanism is released by the fixed tooth. Then, the motion characteristics of the mechanical MEMS remote release insurance mechanism are analyzed by simulation. Through theoretical analysis and simulation analysis, the theoretical calculation results of the maximum deformation of the active tooth are in good agreement with the simulation results, and the error is less than 9.7%. The delay time can reach 1.33ms or more, which meets the requirements.

1. Introduction

The long-distance disarming function of the traditional safety system is generally implemented by means of gunpowder extension, clockwork, electric firework delay, ball rotor, belt, fusible alloy and quasi-fluid [1-2]. Due to the limitation of the processing technology of the MEMS safety system, it is required that the long-distance release insurance mechanism has the characteristics of a planar structure, a small size, and is easy to process and assemble. At present, MEMS safety systems often use a flat-type timepiece to remotely release the insurance mechanism or use a micro-sensor to detect the distance of the projectile from the muzzle, combined with the micro-actuator and form a long-distance disengagement mechanism with a key module structure on the MEMS safety system [3-5].

The long-distance release of the flat-type timepiece is more complicated and takes up a lot of space. The rotary pair connected with the escape wheel is a difficult point of processing [6]; An electronic remote disengagement mechanism is formed using a microsensor combined with a micro-actuator and combined with a key module on the MEMS safety system. As shown in Figure 1 [7-8]. The structure is simple, the design form is diverse, the control precision is high, and the principle of high-density integration is more suitable. The disadvantage is that it is susceptible to the harsh electromagnetic...
environment of the battlefield. According to the process characteristics of MEMS and the characteristics of long-distance disarming mechanism, this paper proposes a planar structure to realize long-distance disarming of MEMS safety system.

2. Theoretical Analysis of Mechanical MEMS Remote Discharge Insurance Mechanism

2.1 Conception and parametric design of mechanical MEMS remote release insurance mechanism

Long distance insurance is a basic requirement for the design of MEMS safety systems. In this paper, a mechanical MEMS remote release mechanism is designed based on the silicon-based DRIE process, as shown in Figure 2.

The elastic beam delay mechanism mainly includes a threshold elastic beam on the auxiliary centrifugal slider, the active tooth and the fixed tooth on the substrate, as shown in Figure 2. The action process is as follows: under the action of the centrifugal environment, the secondary centrifugal slider is moved by the centrifugal force. The active teeth on the secondary centrifugal slider are interlaced with the fixed teeth on the substrate, and the movement of the secondary centrifugal slider is blocked. Since the threshold elastic beam and the corresponding cavity are designed on the auxiliary centrifugal slider,
the active teeth on the auxiliary centrifugal slider move passively toward the cavity, thereby moving the auxiliary centrifugal slider to the outside. When entering the next tooth, the active tooth rebounds to its original position by the influence of the threshold elastic beam. The active tooth reciprocates up and down, and the secondary centrifugal slider moves slowly to the outside to achieve the purpose of delaying for a certain time. Its purpose is a planar design to achieve a short delay function of the MEMS security system. Parametric design for the main variables.

Table 1. Parameterization of mechanical MEMS remote disarming mechanism.

| num | 1 | 2 | 3 | 4 | 5 |
|-----|---|---|---|---|---|
| Line width B/mm | 0.1 | 0.08 | 0.1 | 0.1 | 0.1 |
| inclination α /° | 90 | 90 | 70 | 90 | 90 |
| Arm length L/mm | 1.2 | 1.2 | 1.2 | 2.4 | 1.4 |
| Gap E/mm | 0.03 | 0.03 | 0.02 | 0.03 | 0.03 |
| Number of active teeth | 2 | 2 | 2 | 2 | 1 |
| Active tooth spacing L'/mm | 0.4 | | | | |
| Fixed tooth spacing L''/mm | | | | | 0.4 |

2.2 Theoretical calculation of mechanical MEMS remote release insurance mechanism

For the remote release of the insurance mechanism, the theoretical calculations are carried out on two aspects: (1) the centrifugal slider is subjected to centrifugal force, which causes the active tooth to collide with the fixed tooth to cause the deflection of the threshold elastic beam; (2) the long-distance release mechanism of the active tooth through the fixed tooth.

2.2.1 Centrifugal slider deflection calculation. Assume that the model shown in Figure 2 assumes that the model does not move up and down and affects the motion performance of the elastic beam. Establish a force analysis diagram of the mechanical MEMS remote release insurance mechanism, as shown in Figure 3.

![Figure 3. Force diagram of mechanical MEMS remote release mechanism.](image)

As shown in Fig. 3(a), the main force is the centrifugal force $F$ generated by the rotation of the centrifugal slider by the projectile. The centrifugal slider is in motion, and the active tooth is in contact with the passive tooth. The active tooth gives the passive tooth ramp a pressure perpendicular to the passive tooth $F_N(w)$. $F_N(w)$ is the force that changes with the deflection of the elastic beam delay mechanism, while the active teeth are subjected to reaction forces $F_N'$. Where $F_N' = F_N(w)$ is the friction generated by the relative motion of the active and passive teeth. The centrifugal force generated by the projectile rotating centrifugal slider is as follows:

$$F = mr\omega^2 = mr\left(\frac{2\pi n}{60}\right)^2$$  

Where $m$ is the mass of the centrifugal slider, $r$ is the eccentricity of the centrifugal slider, and $n$ is the rotational speed of the projectile.
Applying the engineering mechanics cantilever beam deflection curve equation, the mechanical MEMS remote release insurance mechanism is analyzed, as shown in Figure 4:

**Figure 4.** Schematic diagram of deflection analysis of elastic beam.

Can get the maximum deflection value at node C:

\[ w_{SC_{max}} = w_{SC_{1max}} + w_{SC_{2max}} \]  \hspace{2cm} (2)

In the formula:

1. **Calculation of maximum deflection at node C:**

\[ w_{SC_{1max}} = \frac{4F}{EHB(2L+L')^3} \left( L \cdot \frac{L'}{4} + L \cdot \frac{5L'}{4} \right) \left( \sin(\frac{\alpha}{2}) - \mu \cos(\frac{\alpha}{2}) \right) \]

2. **Calculation of maximum deflection at node C:**

\[ w_{SC_{2max}} = \frac{2}{EHB(2L+L')^3} \left( -3M_A - R_A \left( L \cdot \frac{5L'}{4} \right)^2 \right) \]

Calculated by calculation \( w_{SC_{max}} >> w_{SC_{2max}} \), which is \( w_{SC_{max}} = w_{SC_{1max}} \). According to the geometric relationship shown in Figure 4, the displacement of the slider motion is obtained.

\[ S_x = w_{SC_{max}} \tan(\frac{\alpha}{2}) \]  \hspace{2cm} (4)

In the case of \( F_x = 0 \), the centrifugal slider has a speed and the speed is the maximum. After that, the speed gradually decreases until the speed drops to zero, and the deflection reaches a maximum.

Assuming that the motion of the active tooth contact passive tooth is from uniform acceleration to uniform deceleration, the average acceleration of the process of uniform acceleration is \( a_A = \frac{a_{A_{max}}}{2} = \frac{F}{2m} \), and the average acceleration of the process of uniform deceleration is \( a_D = \frac{a_{D_{max}}}{2} \).

\[ S_D = w_{MC_{max}} \tan(\frac{\alpha}{2}) - S_A = (w_{MC_{max}} - w_{SC_{max}}) \tan(\frac{\alpha}{2}) \]  \hspace{2cm} (5)

Calculated as follows:

\[ w_{MC_{max}} = \frac{8F}{EHB(2L+L')^3} \left( L \cdot \frac{L'}{4} + L \cdot \frac{5L'}{4} \right) \left( \sin(\frac{\alpha}{2}) - \mu \cos(\frac{\alpha}{2}) \right) \]  \hspace{2cm} (6)

2.2.2 **Exercise time estimation.** In this design, the delay time is mainly achieved by the active tooth collision passive tooth reducing the motion speed. The relationship between displacement and velocity, acceleration, and time is as follows:

\[ S = v_0t + \frac{1}{2}at^2 \]  \hspace{2cm} (7)

1) **Active tooth first contact with passive teeth**

According to the previous analysis of the deflection of the dynamics, it is known that from the first contact of the active tooth to the passive tooth, to the escape of the passive tooth, an acceleration and a deceleration process are experienced. It is assumed to be a uniform acceleration process and a uniform deceleration process. We can get:
\[ t_{1A} = \left( \frac{2E \tan \left( \frac{\alpha}{2} \right)}{v_2} \right) \left( \frac{2\pi n}{60} \right) \]

Where \( r_0 \) is the initial eccentricity of the centrifugal slider.

It can be seen from the analysis of the kinetic mid-deceleration process that the uniform deceleration process and the uniform acceleration process calculated by the above formula are inverse processes. Therefore \( t_{1S} = t_{1A} \), i.e. \( t_1 = 2t_{1A} \).

(2) active teeth second contact with passive teeth

After the first contact of the active tooth and the acceleration of the intermediate portion, the second contact with the passive tooth has an initial velocity. Therefore, the initial velocity at the second contact is first requested, and then the time is calculated by the formula (7).

It is assumed that the acceleration of the middle portion of the first contact and the second contact takes the average value of the maximum value and the minimum value. The velocity is 0 when the first contact is passed and the passive tooth passes. The middle part of the exercise time is

\[ t_1' = \frac{v_3 - v_1}{a_1} \]

Where \( a_1' \) is the average acceleration of the middle part and \( v_2 \) is the initial speed of the second contact passive tooth.

When the active tooth touches the passive tooth for the second time, although the two teeth of the active tooth are in contact with the passive tooth at the same time, since the two teeth of the active tooth are connected, only the force of one tooth can be calculated. After analysis, \( t_2 \) satisfies the following relationship.

\[ E \cdot \tan \left( \frac{\alpha}{2} \right) = v_2 t_2 + \frac{1}{2} a_2 t_2^2 \]

\( a_2 \) is the average acceleration of the active tooth in the second contact with the passive tooth process.

The total time the safety slider has moved into place is:

\[ t = t_1 + t_1' + t_2 \]

3. Simulation analysis of mechanical MEMS remote release insurance mechanism

Through the above analysis, theoretical calculation and parametric design of the mechanical MEMS remote release insurance mechanism and optimization mechanism are carried out. This section mainly analyzes the dynamics of the working process of the mechanical MEMS remote release insurance mechanism, and analyzes the motion characteristics of the mechanical MEMS remote release insurance mechanism through simulation.

In the dynamic simulation of mechanical MEMS remote release insurance mechanism, the Pro/E software is used to establish the three-dimensional geometric model of the mechanical MEMS remote release insurance mechanism and the optimization mechanism, and the finite element model of the model is established by HyperMesh software. As is shown in Figure 5.
Figure 5. Finite element model of mechanical MEMS remote release insurance mechanism.

In this mechanism, the centrifugal slider and the substrate are made of silicon material. The material parameters in the model are listed in Table 2:

Table 2. Material mechanics properties used in simulation.

| Material name | Density $\rho$ (kg/m$^3$) | Elastic Modulus $E$ (GPa) | Poisson ratio $\nu$ |
|---------------|-----------------------------|---------------------------|---------------------|
| Si            | $2.3 \times 10^3$           | 180                       | 0.3                 |

3.1 Deflection simulation

Combining the parametric model listed in Table 2 with the structure shown in Figure 2, a finite element model as shown in Figure 5 is created and a constant size of 30 000 g is applied to the centrifugal slider. Adjust the gap $E$ between the active tooth and the passive tooth in Figure 2, and explore the displacement of the active tooth in the working condition to overcome the displacement of the passive tooth. Combined with calculation and simulation, it can be obtained as shown in Table 3. In the theoretical and simulation cases, different parametric models produce maximum deflection values and comparison tables.

Table 3. Maximum deflection value comparison.

| Comparison project | 1       | 2       | 3       | 4       | 5       |
|--------------------|---------|---------|---------|---------|---------|
| quality (m/kg)     | $2.88 \times 10^{-6}$ | $2.836 \times 10^{-6}$ | $2.866 \times 10^{-6}$ | $2.84 \times 10^{-6}$ | $2.868 \times 10^{-6}$ |
| Theoretical maximum deflection (mm) | $2.86 \times 10^{-2}$ | $5.50 \times 10^{-2}$ | $2.0 \times 10^{-2}$ | 0.20 | $3.23 \times 10^{-2}$ |
| Simulation maximum deflection (mm) | $2.92 \times 10^{-2}$ | $5.64 \times 10^{-2}$ | $2.1 \times 10^{-2}$ | 0.213 | $3.37 \times 10^{-2}$ |
| error              | 2.1%    | 2.5%    | 5.0%    | 6.5%    | 4.3%    |

It can be seen from Table 3 that the simulation results about the deflection are in good agreement with the theoretical calculation results. The error of each number of the original organization is less than 6.5%. The reason is that when the simulation, the active tooth and the passive tooth have a certain gap at the initial moment, and there is a velocity at the time of contact, so that the force of the active tooth is increased, and the simulated deflection is greater than the theoretical calculation deflection.

3.2 Exercise time simulation

To simplify the calculation process, a constant 30 000 g acceleration is applied to the centrifugal slide. The theoretical calculation results of the time required for the active tooth to pass through two passive teeth are shown in Table 4.
Table 4. Theoretical calculation results of the time required for the active tooth to pass through 2 passive teeth.

| Time/μs | 1   | 2   | 3   | 4   | 5   |
|--------|-----|-----|-----|-----|-----|
| $t_1$  | 276 | 283 | 231 | 730 | 293 |
| $t_1'$ | 497 | 480 | 503 | 365 | 495 |
| $t_2$  | 19  | 38  | 13  | 183 | 22  |
| $t_2'$ | 32  | 32  | 32  | 320 | 32  |
| Total time/μs | 824 | 833 | 779 | 1330 | 842 |

As can be seen from Table 4, the most important factor affecting the exercise time is the value of the gap E. Under the same conditions, the larger the gap E, the longer the motion is displaced. As the displacement of the active teeth increases, the required time passes through the same distance, and the time required is shorter and faster.

Combining the parametric model listed in Table 1 with the structure shown in Figure 5, the active tooth gap is taken as the simulated value shown in Table 3. The simulation results of the displacement-time and speed-time of the active tooth passing through two passive teeth are shown in Figure 6.

Figure 6. Simulation results of different structural parameters of the active tooth through two passive teeth.
According to the displacement-time and velocity-time simulation results shown in Figure 6, the simulation results are in good agreement with the theoretical calculation trends shown in Table 4. Overall, after the same displacement, the simulation time is shorter than the theoretical calculation. According to Figure 6(b), after the active tooth collides with the passive tooth, the theoretically calculated velocity is 0 m/s, and the speed calculated by the simulation is obtained. More than 0 m/s, due to the existence of the storage speed, the average speed of the simulation results is larger than the theoretical calculation, and the time for the same displacement is shorter. The active tooth has a maximum delay time of 1.33 ms through 2 passive teeth.

4. Conclusions
The parametric design of the MEMS fuze safety insurance mechanism is carried out. Through theoretical analysis and simulation analysis, the maximum deformation amount and the delay time parameterized design law are studied. The theoretical calculation results of the maximum deformation prediction are in good agreement with the simulation results. The error is less than 9.7%; the fourth scheme has the best effect, and its parameters are line width 0.1 mm, inclination angle 90°, tooth arm length 2.4 mm, gap 0.03 mm, number of active teeth 2, active tooth spacing 0.4 mm, fixed The tooth pitch of 0.4 mm can make the delay time reach 1.33 ms or more, which meets the requirements.

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5. References
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