Research on muzzle response characteristics of automatic gun on launching based on rigid-flexible coupling dynamics

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Abstract. In order to study the important influence for firing dispersion of muzzle response of automatic gun on launching, the rigid-flexible coupling dynamics model of automatic gun was established based on the rigid-flexible coupling system dynamics theory in software RecurDyn. And the flexible body of gun barrel assembly was set up by the modal synthesis method of dynamic sub-structure. The Hertz contact theory and flexible Bushing forces model were made equivalent to these constrained joints. According to the characteristic of automatic gun structure and continuous launching load, the dynamic vibrating response of simulation and analysis was made for the automatic gun on launching. The muzzle vibration response displacements in vertical and horizontal direction were obtained in the 300 rate of fire. Then the muzzle vibration velocity and acceleration with different structure clearance were also calculated by the finite element analysis program of the rigid-flexible coupling dynamics model of automatic gun. Through the comparison and analysis, the numerical solutions are closely matched with the experimental results. The results show that the vertical angle error of firing dispersion is less than 3.8% and the horizontal angle error is less than 10.2%. Thus, the simulation and experimental results verify the correctness and feasibility of this model. Further the structure design and optimization of automatic gun can be provided a reference foundation.

1. Introduction

The firing dispersion of automatic gun is an important technical index of artillery weapon system, which influences the operational effectiveness. For the small caliber automatic gun, the dynamic angular displacement response of the muzzle is an important factor affecting the dispersion. Automatic gun is a typical high-speed impact launcher with complex structural response. When shooting, its recoil characteristics, structural stiffness and turret stiffness have a great impact on the muzzle vibration response. It is difficult to accurately simulate the response rule of automatic gun based on multi-rigid-body dynamics. Multi-flexible body system dynamics technology has been widely used in the design of artillery weapons and the optimization of tactical indicators, considering the influence of flexible deformation of barrel and other components on the dynamic response of the structure [1-3].

At present, the research focus based on the launch of the coupled dynamics mainly concentrated in the anti-aircraft gun system considering the coupled response of tube and shell, self-propelled guns moving is set up based on Kane method multi-rigid-body dynamics model and dynamic characteristic simulation, multi-rigid-body dynamics of small caliber automatic gun and the muzzle vibration response of multi-body system and firing stability [4-7]. The muzzle response study for large caliber gun and small caliber automatic gun generally used the modal synthesis method, the modal information through the analysis of the contact/collision between the soft model, by applying finite element software to each discrete components into fine grid and setting the boundary constraints. Thus,
the flexible body is obtained to calculate the modal information, using the modal synthesis method to realize the flexible sub components and gun multi-body dynamics model of coupling and the joint simulation. The dynamic response of the system structure is obtained for the contact the influence law of clearance on the muzzle disturbance and the barrel body flexibility [8-10].

At present, the dynamic response research of automatic gun firing dynamics only carries out structural simulation for the automatic gun firing without considering the turret, the gap between the contact surface of the structure and the characteristics of light materials, etc. The simulation results are quite different from the actual situation, which cannot effectively guide the design and structural optimization. Therefore, it is of great practical value to establish the rigid-flexible coupling dynamic simulation model and study the dynamic response of the muzzle.

Based on the rigid-flexible coupling multi-body dynamics theory, this paper simplified the barrel into a flexible beam for a small caliber automatic gun, established a rigid-flexible coupling dynamics model based on three-dimensional solid modeling software and RecurDyn multi-body dynamics analysis software, and studied the dynamic response of the muzzle during the shooting process of the automatic gun. Through dynamic simulation analysis of the vibration response of the continuous firing of the automatic gun, the gun muzzle in the vertical direction and in the horizontal direction vibration response displacement of the automatic gun were obtained, and verified through the shooting test data, providing a basis for the structural design and optimization of the automatic gun.

2. The basic theory of rigid-flexible coupling multi-body dynamics modeling

2.1. Theory of multi-rigid body dynamics
Multi-rigid-body Lagrange's equation to establish dynamic system differential equations as follows [11]:

\[ \frac{d}{dt} \left( \frac{\partial L}{\partial \dot{\xi}} \right) - \frac{\partial L}{\partial \xi} + \frac{\partial \Gamma}{\partial \xi} + \left( \frac{\partial \Psi}{\partial \xi} \right)^T \lambda - Q = 0 \]

\[ \Psi = 0 \]

(1)

Where, \( \lambda \) is corresponds to the Lagrange multiplier of the constraint equation; \( \xi \) is the generalized coordinates; \( Q \) is generalized force on the projection to \( \xi \); \( L=T-W \) is Lagrangian term; \( T \) and \( W \) represent kinetic energy and potential energy respectively; \( \Gamma \) is represents the energy loss function; \( \Psi \) is constraint equations.

2.2. Theory of flexible body dynamics
In order to establish the dynamic model of multi-flexible body system, the Lagrangian equation considering the whole system and multi-constraints is [9,12]:

\[ \delta \dot{q}^T \left( M \ddot{q} + \Phi_q \lambda - Q^* \right) = 0 \]

(2)

Where, \( \delta q = (\delta q_1^T, \delta q_2^T, \cdots, \delta q_m^T)^T \) ; \( \ddot{q} = (\ddot{q}_1^T, \ddot{q}_2^T, \cdots, \ddot{q}_m^T)^T \) ; \( q \) is the generalized displacement

Generalized mass matrix of the system: \( M = \text{diag}(M_1, M_2, \cdots, M_m) \)

Generalized force matrix: \( Q^* = (Q_1^{stT}, Q_2^{stT}, \cdots, Q_m^{stT})^T \)

Where, \( \Phi_q \) is the partial derivative of the constraint equation with respect to \( q \), and \( \lambda \) are corresponds to the Lagrange multiplier array.

In the dynamic analysis and calculation of multi-body system, the flexible elastic deformation is described by the modal expansion method. The basic principle of deformation description of flexible body (modal flexible body) based on modal synthesis method is that the flexible body is regarded as a set of finite element model nodes, and then the elastic body is represented by modal. The basic idea of modeling modal flexible body is to give each flexible body in the system with a modal set, and use the modal expansion method to approximate the linear local motion of nodes in the flexible body as the linear superposition of modal shape. The deformation of the flexible body in the system is described by calculating the elastic displacement of the body at each moment. The most typical modal synthesis
method is fixed interface modal synthesis. The Craig-Bampton method is one of the most representative and widely used methods in fixed interface modal synthesis.

The shortcoming for Craig-Bampton substructure modal synthesis method is that the constraint mode of the is obtained based on the static loading method. Therefore, the dynamic frequency response of the flexible body cannot be reflected, and its modal cannot correspond to the frequency, so it is difficult to conduct structural dynamic analysis. However, the modified Craig-Bampton method can effectively solve this problem.

The modified Craig-Bampton method means that each flexible body is regarded as a substructure, and each substructure first fixes all the interfaces to solve the low-order modal, and then releases the freedom of the interface to obtain the constrained modal. The dynamic equation for each substructure is

$$m^\lambda \ddot{\mu}^\lambda + c^\lambda \dot{\mu}^\lambda + k^\lambda \mu^\lambda = F^\lambda$$

(3)

Where, $m^\lambda$ is the mass matrix of substructure; $c^\lambda$ is the damping matrix of substructure; $k^\lambda$ is the stiffness matrix of substructure; $F^\lambda$ is the external load matrix of substructure.

The undamped motion equation of a single substructure is

$$[\tilde{M}]{\ddot{p}} + [\tilde{K}]{p} = \{\tilde{F}\}$$

(4)

Where, $[\tilde{M}]$ and $[\tilde{K}]$ are represent the modal mass matrix and modal stiffness matrix of substructures respectively.

Since the calculation and analysis of each substructure are carried out independently, in order to extend the coordinate transformation to the whole structure, a set of uncoupled but not independent motion equations needs to be established, which requires two coordinate transformations to realize the connection of each substructure.

The uncoupled global motion equation of the system under the modal coordinate $[p]$ is established as

$$[\tilde{M}']{\ddot{p}} + [\tilde{K}']{p} = \{\tilde{F}'\}$$

(5)

Since each substructure is connected by the interface, according to the unique continuity condition of the interface, the dynamic equations of each substructure are synthesized into the whole dynamic equation. Assuming that two substructures have docking surfaces between $\alpha$ and $\beta$, the modal coordinates of $\alpha$ and $\beta$ can be written as follows:

$$\begin{pmatrix} \alpha \\ \beta \end{pmatrix} p_k = \begin{pmatrix} \alpha \\ \beta \end{pmatrix} p_k$$

(6)

The displacement connection condition of the bonding surface is

$$\begin{pmatrix} \alpha \\ \beta \end{pmatrix} p_c = \begin{pmatrix} \alpha \\ \beta \end{pmatrix} p_c$$

(7)

Thus, According to the equations (6) and (7), the second coordinate transformation is obtained, and the system equation connected to each substructure is assembled into $\{q\}$ generalized coordinates:

$$\begin{pmatrix} \alpha \\ \beta \end{pmatrix} p_k = \begin{pmatrix} \alpha \\ \beta \\ \alpha \\ \beta \end{pmatrix} p_k$$

(8)

Where, $\{p_k\}$ is represents the modal coordinates corresponding to the main modes of all substructures; $\{p_c\}$ is represents the modal coordinates corresponding to the independent modal of all substructure interfaces.

Substitute equation (8) into equation (5) to get

$$[\tilde{M}'']{\ddot{p}} + [\tilde{K}'']{p} = \{\tilde{F}''\}$$

(9)

Therefore, the characteristic equation of the whole system can be obtained, expressed as

$$[\tilde{K}'' - \omega^2 \tilde{M}']\{q\} = 0$$

(10)
By solving the equation (10), the eigenvalues and eigenvectors of the whole system are obtained, and the dynamic response of the whole system in physical coordinates can be obtained after two inverse coordinate transformations[9,11,12,14].

2.3. Dynamics of rigid-flexible coupled multi-body systems

The dynamics of rigid-flexible coupled multi-body system is based on the dynamics equation of multi-body system, and the Lagrangian multiplier is introduced to establish the dynamics equation of each rigid body or flexible body. The dynamics equation of the i-th rigid body or flexible body can be obtained as follows:

\[ M_i \ddot{q}_i + \Gamma(q, \dot{q}) + C_{\dot{q}i}(q) \lambda = F_{ei} + F_{vi} \]  

(11)

The constraint equation of the system is:

\[ C(q, t) = 0 \]  

(12)

Where, \( F_{ei} \) is the external force on the i-th rigid or flexible body; \( F_{vi} \) is the velocity binomial. The dynamics equation of rigid-flexible coupled multi-body system [3,11,12,] can be obtained by simultaneous equations (11) and (12).

3. Dynamic modeling of rigid-flexible coupled multi-body system for automatic gun

3.1. The basic assumptions

According to the characteristics of the structure of the automatic gun and the launching physical process, the rigid-flexible coupling dynamic model of the automatic gun is established. The main assumptions are as follows:

- Ignore the inertia of the spring;
- All hinge constraints are regarded as ideal integrity constraints;
- All the deformation of the flexible body is small deformation. The barrel and spring are simplified into a flexible body, and the other parts are simplified into rigid bodies;
- The recoil device of the automatic gun connects the recoil part and the cradle. The recoil part relative to the cradle makes recoil and recoil motion along the axis of the gun chamber.

3.2. The geometric model

The automatic gun system is mainly composed of turret, cradle, elevating mechanism, traverse mechanism, equilibrator and automatic gun. The geometric modeling of the gun system adopts \( UG \) modeling software to establish the three-dimensional models of the system. According to the size of the structural parts of the gun system, the three-dimensional system model of all parts (including barrel assembly components, receiver components, spring-hydraulic device components) are established, and the automatic gun system is assembled in the \( UG \) software environment. It is defined material properties for all parts of the gun system. The turret body and the cradle are made of aluminum alloy, the barrel and main parts are made of high-strength gun steel, the upper rib barrel and muzzle device of the automatic gun are made of titanium alloy, and the remaining parts are made of alloy steel.

3.3. Imposed constraints

According to the structural connection and movement relationship of various components in the automatic gun system, the muzzle device, ribbed barrel, gun breech and barrel are fixed and dealt with flexible bodies, the turret body is hinged with the cradle, the gear is engaged and contacted, including the topological structure of the connection relationship between various components of the gun system.

3.4. Define contact and force connections

The surface-to-surface contact between parts are subject to collision and contact. The surface-to-surface contact adopts the key surface to simplify the complex solid contact, and can calculate the contact problems with complex appearance and arbitrary shape. The calculation of contact force with the dynamics software RecurDyn is based on the Hertz contact theory, and improvements are made on this basis. The formula for the calculation of contact normal contact force \( f_n \) is as follows [13]:

\[ f_n = \frac{K \cdot (r - 0.5 \cdot b)}{a} \]  

(13)

Where, \( r \) is the normal contact force; \( a \) is the contact force radius; \( b \) is the contact force width; \( K \) is the contact stiffness coefficient; \( a \) is the contact force radius, and \( b \) is the contact force width.
Where, \( k \) is the contact stiffness coefficient; \( c \) is the damping coefficient; \( \delta \) is the contact penetration depth; \( \dot{\delta} \) is the derivative of contact penetration depth; \( m_1, m_2 \) and \( m_3 \) are respectively stiffness index, damping index and dent index.

The Bushing is a very important flexible connection form, and connecting two components and applying force on the two components. In general, using the Bushing instead of the Joint is an effective method. The Bushing does not reduce the degree of freedom of the system and can also deal with the constraint problem. The mechanical model of the Bushing is as follows:

\[
\begin{bmatrix}
F_{ax} \\
F_{ay} \\
F_{az} \\
T_{ax} \\
T_{ay} \\
T_{az}
\end{bmatrix} = -
\begin{bmatrix}
K_{11} & 0 & 0 & 0 & 0 & 0 \\
0 & K_{22} & 0 & 0 & 0 & 0 \\
0 & 0 & K_{33} & 0 & 0 & 0 \\
0 & 0 & 0 & K_{44} & 0 & 0 \\
0 & 0 & 0 & 0 & K_{55} & 0 \\
0 & 0 & 0 & 0 & 0 & K_{66}
\end{bmatrix}
\begin{bmatrix}
x^{11} \\
y^{12} \\
z^{13} \\
\theta_{ab}^{11} \\
K_{66}^{0} \\
\theta_{ab}^{00}
\end{bmatrix}
+ 
\begin{bmatrix}
C_{11} & 0 & 0 & 0 & 0 & 0 \\
0 & C_{22} & 0 & 0 & 0 & 0 \\
0 & 0 & C_{33} & 0 & 0 & 0 \\
0 & 0 & 0 & C_{44} & 0 & 0 \\
0 & 0 & 0 & 0 & C_{55} & 0 \\
0 & 0 & 0 & 0 & 0 & C_{66}
\end{bmatrix}
\begin{bmatrix}
x^{m1} \\
y^{m2} \\
z^{m3} \\
\theta_{ab}^{m1} \\
\theta_{ab}^{m2} \\
\theta_{ab}^{m3}
\end{bmatrix}
+ 
\begin{bmatrix}
F_1 \\
F_2 \\
F_3 \\
T_1 \\
T_2 \\
T_3
\end{bmatrix}
\]  \hspace{1cm} (14)

Where, \( K_{ii} (i=1\sim6) \) is the stiffness coefficient; \( C_{ii} (i=1\sim6) \) is the damping coefficient; \( x, y, z, \theta_{ab1}, \theta_{ab2}, \theta_{ab3} \) are relative translational displacement and relative rotational displacement of the two connectors; \( \dot{x}, \dot{y}, \dot{z}, \dot{\theta}_{ab1}, \dot{\theta}_{ab2} \) and \( \dot{\theta}_{ab3} \) are relative translational velocity and relative rotational velocity; \( F_1, F_2, F_3, T_1, T_2 \) and \( T_3 \) are preloads; \( k, l, m, n (i=1,2,3) \) are exponents.

When the modal synthesis method is used in practical engineering application, only the modal parameters with large contribution factor of the first few modals are calculated, while the other modals are ignored. In this way, the calculation accuracy can be guaranteed and the calculation time can be greatly reduced. Considering the complexity of gun system launching, the 6 order modes are selected. figure 1 and table 1 shows the flexible body modals calculated in ANSYS software. According to the vibration modal diagram and frequency of the body, the first 6 bending modals of the flexible body correspond to the two vibration modal diagrams.

(a) The first order modal  
(b) The second order modal  
(c) The third order modal  
(d) The fourth order modal  
(e) The fifth order modal  
(f) The sixth order modal

**Figure 1.** The first 6-order mode vibration diagrams of barrel assembly.
Table 1. The first 6-order mode six frequencies.

| frequency (Hz) | The first order | The second order | The third order | The fourth order | The fifth order | The sixth order |
|---------------|----------------|-----------------|----------------|-----------------|----------------|----------------|
|               | 71.47          | 71.48           | 437.2          | 438.2           | 1136.5         | 1145.4         |

3.5. Applied load
During the firing process of the automatic gun system, the load is caused by the gun breech force, which is related to the gun breech pressure during the interior ballistic period and the gun breech pressure during the after-effect period. The calculation formula of breech pressure in the period of interior ballistic is as follows:

\[ p_i = \left(1 + \frac{\omega}{6\phi_1 m + 2\omega}\right)p \]  \hspace{1cm} (15)

Where, \( \omega \) is the dosage; \( \phi_1 \) is the resistance coefficient. In general, the resistance coefficient \( \phi_1 \) is about 1.02. \( m \) is the mass of the projectile; \( P \) is the average pressure, which can be solved by classical interior ballistic model.

The calculation formula of the gun breech bottom pressure in the after-effect period is as follows:

\[ p_t = \left(1 + \frac{\omega}{6\phi_1 m + 2\omega}\right)p_g e^{\frac{t}{b}} \]  \hspace{1cm} (16)

Where, \( p_g \) is the average pressure of the muzzle; \( t \) is the time from the beginning of the after-effect period; \( b \) is the time constant of pressure attenuation in the after-effect period.

The calculation formula of gun breech force is as follows:

\[ F_i = S \cdot p_i \]  \hspace{1cm} (17)

Where, \( S \) is the section area of the gun chamber.

According to the rigid-flexible coupling dynamics model of the automatic gun system, the applied launch load is the gun breech force when firing 5 rounds at a rate of 300 rounds/min, and its gun chamber force curve is shown in figure 2. In the rigid-flexible coupling dynamics model of the automatic gun system, constraints and loads are applied as shown in figure 3. The gun breech force is applied to the center position of the face of the gun chamber, and the direction is the shooting direction of the barrel axis.

3.6. Mathematical model of vibration response of continuous firing of automatic gun
The vibration response of the automatic gun system model in the continuous firing process is finally reduced to the forced vibration equations of the multi-freedom system after the launch load and boundary constraints are applied. The vibration differential equation can be expressed as:

\[ M \ddot{x}(t) + C\dot{x}(t) + kx(t) = f(t) \]  \hspace{1cm} (18)
Where, $M$ is the mass matrix, and it is the matrix of the mass and moment of inertia of each component of the system. $K$ is the stiffness matrix; $C$ is the damping matrix; the proportional damping can be expressed as: 

$$C = \alpha M + \beta K,$$

$x(t)$, $\dot{x}(t)$ and $\ddot{x}(t)$ are displacement, velocity and acceleration vectors of the system vibration response respectively, and $f(t)$ is the gun chamber force.

Based on modal analysis theory, under zero initial conditions, the dynamic response of the system can be expressed as:

$$x(t) = \sum_{r=1}^{n} X_r^T X_r \int_{0}^{t} f(\tau) e^{-(t-\tau)\zeta r} \sin \eta_r \eta_r (t-\tau) d\tau$$

(19)

Where, $X_r$ is the r-th mode vector, $M_r$ is the r-th mode mass, $p_r$ is the r-th mode frequency, and $\zeta_r$ is the r-th mode damping rate.

$$P_r = \sqrt{1 - \zeta_r^2}$$

(20)

4. Simulation analysis of vibration response of barrel in the process of automatic gun shooting

On the vibration response model of continuous firing of the automatic gun, the recoiling displacement of the automatic gun, the vibration displacement in vertical direction of the muzzle and in horizontal direction of the muzzle were obtained by applying the data of the gun bore force of five consecutive shots. The analysis of muzzle vibration response during continuous firing is carried out under the condition of firing speed of 300 rounds/min and continuous firing of 5 rounds. The simulation results show that the maximum recoil displacement is about 26mm. The time curve of the vibration displacement in vertical direction of the muzzle and in horizontal direction of the muzzle obtained by simulation during continuous firing are shown in figure 4 and figure 5.

![Figure 4](image1.png)

**Figure 4.** The curves diagram of muzzle response displacements in vertical direction during five firing.

![Figure 5](image2.png)

**Figure 5.** The curves diagram of muzzle response displacements in horizontal direction during five firing.

According to the curves in figure 4 and figure 5, the maximum vibration displacement in the vertical direction is 0.5mm, the maximum vibration displacement in the horizontal direction is 0.9mm, the vibration time in the vertical direction is 140.7ms, and the vibration time in the horizontal direction is 161ms. According to the calculation of firing rate, the cycle time of the automatic gun is 200ms. After the vibration process of vertical and horizontal direction is finished, the next shot is carried out after the initial state is restored. Therefore, when firing at a firing rate of 300 rounds/min, the continuous firing process has no direct impact on the muzzle vibration. Because the interior ballistic time of 2.8 ms, shooting process the muzzle vibration did not reach the maximal displacement, the projectile has been flying out of the muzzle, follow-up the muzzle vibration displacement of projectile the muzzle flash attitude cannot produce disturbance, also cannot affect the dispersion, so in the projectile at muzzle, the muzzle vibration displacement is the main factors influencing the dispersion directly. According to the simulation results, the muzzle response displacement in vertical direction is
0.2578mm, and the muzzle response displacement in horizontal direction is -0.4612mm. According to the structural size of the barrel, the angular displacement is calculated and the dispersion obtained after conversion is 0.303 mil in the vertical direction and 0.496 mil in the horizontal direction.

5. Test

In order to verify the validity of the simulation results of the rigid-flexible coupling dynamics model, a shooting test was conducted on a automatic gun system. In the test, a target board was set up in the 110m target path, and the firing line of the automatic gun system was aimed at the target through target calibration. Under the firing rate of 300 rounds/min, three groups of 7 shots and one group of 12 shots were carried out, and the positions of a group of 7 shots and a group of 12 shots were shown in figure 6 and figure 7. By measuring the target position of the projectile, the calculated dispersion in vertical and horizontal direction is shown in table 2. The maximum vertical dispersion is 0.36mil, with an average of 0.315mil. The maximum horizontal dispersion is 0.6mil, with an average of 0.5525mil; Compared with the simulation results, the dispersion error of vertical angle is 3.8% and that of the horizontal angle is 10.2%.

![Figure 6](image1.png)
![Figure 7](image2.png)

**Figure 6.** The diagram of projectile impact during seven firing.  
**Figure 7.** The diagram of projectile impact during twelve firing.

**Table 2.** The calculated result of firing dispersion.

| Shooting Modes | \( X \) | \( Y \) |
|----------------|--------|--------|
|                | Intermediate deviation \( EX \) (mm) | Dispersion (mil) | Intermediate deviation \( EY \) (mm) | Dispersion (mil) |
| twelve firing  | 65.92  | 0.57   | 33.28  | 0.29   |
| even firing    | 60.47  | 0.52   | 40.96  | 0.36   |
| even firing    | 69.27  | 0.60   | 37.93  | 0.33   |
| even firing    | 59.92  | 0.52   | 32.37  | 0.28   |
| average value  | 63.895 | 0.5525 | 36.135 | 0.315  |

6. Conclusion

By establishing the rigid-flexible coupling dynamic model of the automatic gun system, the dynamic muzzle response during the continuous firing process is simulated and analyzed, and the simulation results are verified. The simulation results are consistent with the test results, and the validity and correctness of the model are verified. The following conclusions can be draw:

- Through Hertz contact theory and equivalent constraint pair of flexible Bushing force, the rigid-flexible coupling multi-body system dynamic model was established and the muzzle response was simulated. The muzzle vibration displacement trend in vertical and horizontal direction were consistent with the test results.
- The comparison between the test results and the simulation results shows that the vertical angle error is less than 3.8%, and the horizontal angle error is less than 10.2%, which further
verifies that the established rigid-flexible coupling multi-body system dynamic model can meet the requirements of engineering design accuracy, and provides a reference for the structural design and optimization of automatic gun system.

References
[1] Kui X f, Rong B and Wang G P 2011 Science China Technological Sciences. 54 (5) 1061-71
[2] Feng Y, Ma D W and Xue C. 2006 Acta Armamentarii 27(3) 545-8
[3] Du Z Y, Wang X Z and Cheng Y Q. 2017 Journal of Projectiles, Rockets, Missiles and Guidance 37(1) 9-12
[4] Ke B, Gao Y F and CAO H S. 2013 Journal of Gun Launch & Control 3 24-8
[5] Liu L. 2005 China Self-propelled guns rigid-flexible coupling launch dynamics.(Nanjing: Nanjing University of Science and Technology.)
[6] Qian M W and Wang L M. 2004 Acta Armamentarii 25 (5) 520-4
[7] Luo J H and Xu D. 2015 Computer Simulation 32(6) 23-5
[8] Yang G L and Chen Y S. 2006 Journal of Nanjing University of Science and Technology 30(4) 495-8
[9] Zeng J C. 2010 Dynamics research of the rigid-flexible coupling firing for vehicle-mounted gun. (Nanjing: Nanjing University of Science and Technolog.)
[10] Min J P, Chen Y S and Yang G L. 2000 Journal of Gun Launch & Control 2 28-31
[11] Liu Y Z, Pan Z K and Ge X S. 2018 Dynamics of multi-body systems. (Beijing: Higher Education Press.)
[12] Lu Y F. 1996 China Dynamics of flexible multi-body systems. (Beijing: Higher Education Press.)
[13] Jiao X J, Zhang J W and Peng B B. 2010 RecurDyn optimization and simulation technology of multi-body system. (Beijing: Tsinghua University Press.)
[14] Deng F Y, He X S and Zhang J. 2004 Journal of Machine Design 21(3) 41-3