Dynamic simulation on vibration control of marching tank gun based on adaptive robust control

Yu Chen, Guolai Yang and Quanzhao Sun

Abstract
In order to better understand the dynamic behavior and decrease the muzzle vibration of marching tank, a mechanical–electrical–hydraulic integrated dynamic model of marching tank was established based on a novel dynamic co-simulation method. The hydraulic system model was modeled in Amesim and the dynamic model of marching tank was established in RecurDyn based on multi-body system theory, vehicle terramechanics, and gun launch dynamics. The control system model was modeled in MATLAB/Simulink. Therein, the adaptive robust control algorithm was introduced to design the vertical stabilizer controller and the simulation program of the designed controller was developed by C language. The simulation results show that the muzzle vibration of marching tank can be controlled effectively by the ARC method. Furthermore, the muzzle error compensation signal was added in the designed controller to weaken the detrimental effect of the barrel flexibility on muzzle vibration. This work provides an approach to investigate the dynamic behavior of marching tank considering effects among the mechanical, hydraulic, and control subsystems.

Keywords
Marching tank, vibration, adaptive robust control, co-simulation, error compensation

Introduction
Firing accuracy is a major index related with performance of guns, and muzzle vibration is an important factor that affects the firing accuracy. Stabilizer is a key component of fire control system that can keep the bore axis of marching tank near the sighting angle and improve the first-round-hit accuracy. Therein, the vertical stabilizer is an electrohydraulic position servo system, which is often used as the loading actuator in the simulation experiences. There are inevitably some nonlinear characteristics (the change of fluid direction controlled by servo valves, friction, etc.) and various model uncertainties (disturbance uncertainty, parameter uncertainty, etc.). However, traditional vertical stabilizers are usually designed as linear time-invariant systems based on the classical control theory which cannot compensate the nonlinearity of the systems effectively. With the increasing requirements on mobility and firing accuracy of modern tanks, traditional stabilizers don’t meet the system requirements. Therefore, the modern intelligent control algorithms are introduced to control the vibration of marching tank gun.

At present, the vertical stabilizer controller has been studied based on some modern control theories such as adaptive control, sliding mode control, fuzzy control, and neural network control. Performances of these controllers are generally superior to the traditional Proportion Integration Differentiation (PID) controller. But in these studies, the dynamics model of marching tank was simplified as a linear transfer function and the coupling relationships between the mechanical, hydraulic, and control subsystems were ignored. Obviously, the numerical
results based on this assumption were not accuracy. With the increasing requirements on high mobility, high initial speed, and high firing accuracy, this deviation is gradually important and cannot be ignored any more.12 In fact, a tank is a complex system composed of mechanical, hydraulic, and control subsystems. They work together and affect each other. The traditional sequential design method ignores the mutual coupling among the subsystems. As a result, the design results are often not optimal, the cycle of the products design is longer and the additional cost is higher.13 At present, researches about the property of mechanical–electrical–hydraulic coupling system have been done in the related field.14–17 It is necessary to study the mechanical–electrical–hydraulic integrated dynamic model of marching tanks.

For the controller design, the more accurate system information we know, the better control performance can be obtained. That is, without considering the complexity of the control structure, accurately modeling can obtain better tracking performance. The vertical stabilizer adjusts the piston rod to reduce the maladjustment angle. However, the desired piston rod displacement, velocity, and acceleration are closely associated with the pitch motion of hull. For traditional vertical stabilizer, it is difficult to ensure the accuracy of the controller by only depending on the cradle elevation angular displacement. So the stabilizer control structure needs be improved.

In the past years, Yao et al.18 put forward a modern control method, whose argument is mathematically rigorous. Adaptive robust control has the advantages and overcomes the shortcomings of the adaptive control and robust control method simultaneously. Its effectiveness has been proved by a large number of researches.19 The control method has been applied on horizontal stabilizers5,20 and its feasibility was demonstrated through MATLAB simulations. But more researches are needed. In this paper, the nonlinear characteristics and model uncertainties of the electrohydraulic position servo system of vertical stabilizer were considered. A new structure of the vertical stabilizer controller was designed. The adaptive robust control algorithm was introduced to design the vertical stabilizer controller and the simulation program was compiled by C language. By combining the controller with the established electrohydraulic position servo system model of the vertical stabilizer and the dynamic model of marching tank, a mechanical–electrical–hydraulic integrated dynamic model of marching tank was established. Furthermore, the muzzle error compensation signal was added in the designed controller to weaken the detrimental effect of the barrel flexibility on muzzle vibration.

Basic structure and working principle of the vertical stabilizer

The vertical stabilizer is an electrohydraulic type automatic adjusting system and the simplified control block diagram is shown in Figure 1.11 When tank is on the move, the pitch motion of hull causes the rotation of gun through the trunnion friction torque. Gun deviates from the sighting angle which generating the maladjusted angle. The control signal is calculated based on the maladjusted angle and with the control signal; the piston rod is driven to control the opening size and direction of electrohydraulic servo valve. Therefore, the bore axis of the marching tank can be kept near the sighting angle and the change of the cradle angular displacement becomes zero. Due to the control input of hydraulic servo valve is zero and the valve spool is in middle position, the pressure in the two chambers of hydraulic cylinder is consistent and the bore axis is kept near the sighting angle.

The main structure of the hydraulic system of the vertical stabilizer is shown in Figure 2. $P$, $P_r$ are the supply pressure and return pressure of the system, $A$ is the effective active area of the actuator, $Q_1$ is the supply flow rate of the forward chamber, $Q_2$ is the return flow rate of the return chamber, $u$ is the control input. The dynamics of the hydraulic cylinder moving parts can be described by5,21

$$m\ddot{y} = PA - f_i - B\dot{y} - A_f S_f - d_n$$

(1)

where $m$ and $y$ represent the moving parts mass and the displacement of piston rod, respectively; $P = P_1 - P_2$ is the load pressure; $f_i$ is the external load because of the driving load; $B$ is the viscous friction coefficient of the

![Figure 1. The block diagram of vertical stabilizer.](image-url)
system; \( A_f \) is the amplitude of the approximated nonlinear Coulomb friction; \( S_f \) is the continuous shape function of the approximated nonlinear Coulomb friction; \( d_a \) is the un-modeled disturbances in the force balance.

Considering the linear internal leakage and the compressibility of fluid, the pressure dynamics inside the cylinder chambers can be described by

\[
\dot{P}_1 = \frac{\beta}{V_1}(Q_1 - A_1\dot{y} - C_l P) \\
\dot{P}_2 = \frac{\beta}{V_2}(-Q_2 + A_2\dot{y} + C_l P)
\]

(2)

where \( \beta \) is the effective bulk modulus of fluid; \( C_l \) is the leakage coefficient; \( V_1 = V_{01} + Ay \), \( V_2 = V_{02} - Ay \) are the actuator control volumes, respectively; \( V_{01}, V_{02} \) are the actuator original control volumes of the two chambers, respectively.

The dynamic equation of servo valve can be approximately described by the first-order element

\[
\dot{x}_v = -\frac{1}{\tau_v} x_v + \frac{k_i}{\tau_v} u
\]

(3)

where \( x_v, \tau_v, k_i \) are the spool displacement of servo valve, the time constant and the spool current gain, respectively. When the bandwidth of the servo valve is much higher than that of the system, the servo valve dynamic can be simplified as a proportional element \( x_v = k_i u \). Then, \( Q_1 \) and \( Q_2 \) can be modeled by

\[
Q_1 = g u[s(u)\sqrt{P_s - P_1} + s(-u)\sqrt{P_1 - P_r}] \\
Q_2 = g u[s(u)\sqrt{P_2 - P_r} + s(-u)\sqrt{P_2 - P_s}]
\]

(4)

where

\[
g = \sqrt{2k_q k_i}
\]

(5)

\[
k_q = C_d w \sqrt{\frac{1}{\rho}}
\]

(6)

where \( g \) is the total flow gain with respect to the control input \( u \); \( k_q \) is the flow gain with respect to the spool displacement; \( C_d, w, \rho \) are the orifice flow coefficient of servo valve, the orifice area gradient and the density of fluid, respectively. The sign function is defined by

\[
s(u) = \begin{cases} 
1, & u \geq 0 \\
0, & u < 0
\end{cases}
\]

(7)

\[\text{Figure 2. The main structure of hydraulic system.}\]
Controller design of vertical stabilizer

Adaptive robust control

Adaptive robust control integrates the working mechanisms of the adaptive control and robust control. The structure is shown in Figure 3. $x$, $u$ are the output and control input of the system, respectively; $\varphi^T$ is a known function called basis function; $\theta$ is the unknown weight factor; $\Delta$ is the approximation error; $\hat{\theta}$ is the online estimation parameter; $x_d$ is the desired output; $z$ is the output error.

The controlled object $\dot{x} = \varphi^T \theta + \Delta + u$ subjected to both uncertain nonlinearities $\Delta$ and strong parametric uncertainties $\theta$. The uncertain parameters are estimated online by the adaptive controller and the detrimental effects of uncertainty nonlinearities are eliminated by the robust controller. So the control accuracy and robustness of the system are both improved.

The control structure design of vertical stabilizer

According to the basic structure and working principle of the vertical stabilizer, the control law is designed based on the feedback tracking error. And the error is eliminated by exerting negative feedback control on the vertical stabilizer. The traditional vertical stabilizer is designed only by the cradle elevation angular displacement. Without considering the complexity of the control structure, the controller needs to be accurately modeled to obtain better tracking performance.

The installation location of hydraulic cylinder is shown in Figure 4. So the displacement of piston rod can be expressed by

$$x = \arccos \left( \frac{\omega^2 + f_{th}^2 - l^2}{2al_{th}} \right) + \omega_z - \omega_p + \theta_s$$  \(8\)

$$y = \Delta l = \sqrt{\frac{a^2 + f_{th}^2 - 2al_{th}\cos(x)}{l}} - l$$  \(9\)
where \( z \) is the corresponding vertex angle of the hydraulic cylinder shown in Figure 4; \( \theta \) is the sighting angle; \( l \) is the initial length of hydraulic cylinder; \( a \) is the distance between the center of the trunnion and the installation position of the hydraulic cylinder on the turret; \( l_d \) is the distance between the center of the trunnion and the driving point location of hydraulic cylinder on the cradle; \( \omega_1 \) is the vertical rotation angle of gun around the center of gravity; \( \omega_p \) is the vertical rotation angle of tank around the center of gravity; \( \Delta l \) is the change length of hydraulic cylinder.

The maladjustment angle of marching tank is mainly caused by the pitch motion of hull which should be considered when designing the control system of the vertical stabilizer. The adaptive robust controller present in this paper can monitor the displacement, velocity, acceleration of piston rod, and the pressures in the two chambers of hydraulic cylinder in real time. It can guarantee the actual displacement of the piston rod tracking the desired displacement steadily. Then the gun stability can be ensured and the muzzle vibration can be controlled. The desired piston rod displacement can be calculated according to equations (8) and (9). And the actual value of pitch motion can be measured by the gyroscope installed on the hull.

### Design of adaptive robust controller

Before designing the controller, the mathematical model of the electrohydraulic position servo system of the vertical stabilizer needs to be rewritten into the state space form.\(^{13}\) According to the nonlinear model represented by equations (1)–(3), the state variables are defined as

\[
x = [x_1, x_2, x_3]^T = [y, \dot{y}, PA]^T
\]  

(10)

The hydraulic dynamic system can be expressed in a state-space form as

\[
\begin{align*}
\dot{x}_1 &= x_2 \\
mx_2 &= x_3 - Bx_2 - A_0\gamma_T - d_n - d_l \\
\dot{x}_3 &= g_3u - f_{31} - f_{32}C_t
\end{align*}
\]  

(11)

where

\[
\begin{align*}
d_l &= f_t \\
g_3 &= \left(\frac{R_1}{V_1} + \frac{R_2}{V_2}\right)Ag\beta \\
f_{31} &= \left(\frac{1}{V_1} + \frac{1}{V_2}\right)A^2\beta x_2 \\
f_{32} &= \left(\frac{1}{V_1} + \frac{1}{V_2}\right)\beta x_3
\end{align*}
\]  

(12)

\[
\begin{align*}
R_1 &= s(u)\sqrt{P_s - P_t} + s(-u)\sqrt{P_t - P_r} \\
R_2 &= s(u)\sqrt{P_2 - P_t} + s(-u)\sqrt{P_t - P_2}
\end{align*}
\]  

(13)

The design goal of controller is to get a bounded continuous control input \( u \). It makes sure that the tracking error of the piston rod displacement of marching tank calculated above is either close to zero or in the desired range. There are many uncertain parameters in the system, and the unknown parameters of the system in this paper are defined as

\[
\theta = [\theta_1, \theta_2, \theta_3, \theta_4]^T = [B, A_f, d_n, C_t]^T
\]

\[
\theta \in \Omega_\theta \triangleq \{\theta : \theta_{\text{min}} \leq \theta \leq \theta_{\text{max}}\}
\]  

(14)
The entire system can be expressed as

\begin{align*}
\dot{x}_1 &= x_2 \\
mx_2 &= x_3 - \theta_1x_2 - \theta_2S_f - \theta_3 - d_i \\
x_3 &= g_3u - f_{31} - \theta_{f32}
\end{align*}

(15)

In this paper, \( \hat{\theta} \) is the estimate of \( \theta \). In order to ensure the system stability of the adaptive control rate, the following adaptation law structure will be sought

\[ \hat{\theta} = \text{Proj}_{\hat{\theta}}(\Gamma \tau) \]

(16)

where \( \text{Proj}_{\hat{\theta}}(\cdot) = [\text{Proj}_{\hat{\theta}_i}(\cdot) \ldots \text{Proj}_{\hat{\theta}_i}(\cdot)]^T \); \( \Gamma > 0 \) is a diagonal matrix, which represents adaptive gain; \( \tau = \phi_2z \) is a parameter adaptation function. The concrete form is given in the subsequent design process. Based on the assumption that the parameter uncertainty of the system is bounded, the parameter adaptive discontinuous projection mapping is defined as\(^\text{22}\)

\[ \text{Proj}_{\hat{\theta}_i}(\cdot) = \begin{cases} 0 & \text{if } \hat{\theta} = \theta_{\text{max}} \text{ and } \gamma_i > 0 \\ 0 & \text{if } \hat{\theta} = \theta_{\text{min}} \text{ and } \gamma_i < 0 \\ \gamma_i & \text{otherwise} \end{cases} \]

(17)

A set of switching-function-like quantities are defined as

\[ z_2 = \hat{z}_1 + k_1z_1 = x_2 - x_{2eq} \]

(18)

where \( k_1 \) is a positive feedback gain; \( z_1 = x_1 - x_{1id} \) is the output tracking error. According to the equation (15), the time derivative of \( z_2 \) can be written as

\[ m\dot{z}_2 = m\dot{x}_2 - m\dot{x}_{2eq} = x_3 - \theta_1x_2 - \theta_2S_f - \theta_3 - m\dot{x}_{2eq} - d_i \]

(19)

where \( x_3 \) can be treated as the input.

We can develop a virtual control input \( z_2 \) for \( x_3 \) to ensure that \( z_2 \) is as small as possible.

\begin{align*}
x_2 &= x_{2a} + x_{2t} \\
x_{2a} &= \hat{\theta}_1x_2 + \hat{\theta}_2S_f + \hat{\theta}_3 + m\dot{x}_{2eq} \\
x_{2s} &= x_{2s1} + x_{2s2} \\
x_{2s1} &= -k_{2s1}z_2
\end{align*}

(20)

where \( x_{2a} \) functions as a model-based feed-forward control law used to achieve an improved model compensation. \( x_{2s1} \) is a linear feedback law to stabilize the nominal model of the hydraulic system and \( x_{2s2} \) is a robust controller.

The discrepancy between the control function \( x_2 \) and virtual control input \( x_3 \) is defined as \( z_3 = x_3 - x_2 \). By substituting equation (20) into equation (19), we can get

\[ m\dot{z}_2 = z_3 - k_{2s1}z_2 + x_{2s2} - \phi_2^T\hat{\theta} - d_i \]

(21)

where \( \hat{\theta} = \hat{\theta} - \theta \) represents the parameter estimation error and it is bounded.

\[ d_i(x_1, x_2, t) \leq \delta_d(x_1, x_2, t) \]

(22)

\[ \phi_2^T\hat{\theta} \leq [-x_2, -S_f(x_2), -1, 0] \]

(23)
According to the equation (21), $a_{22}$ need to meet the following calming conditions

$$z_2 \left[ a_{22} - \varphi_2 \dot{\varphi} - d_1 \right] \leq e_2 \quad (24)$$

$$z_2 a_{22} \leq 0 \quad (25)$$

where

$$a_{22} = -k_{22}(x_1, x_2, \theta_M, \delta_d)z_2 \pm \frac{h_2}{2e_2} z_2 \quad (26)$$

where $k_{21}, k_{22}, e_2$ are the design parameters of the controller. $h_2$ is defined by

$$h_2 \geq \|\varphi_2\|^2 \|\theta_M\|^2 + \delta_d^2 \quad (27)$$

where $\theta_M = \theta_{\text{max}} - \theta_{\text{min}}$.

The Lyapunov function $V_2$ is defined as

$$V_2 = \frac{1}{2} m z_2^2 + \frac{1}{2} k_{12}^2 z_1^2 \quad (28)$$

The time derivative of $\dot{V}_2$ can be written as

$$\dot{V}_2 = z_2 m \ddot{z}_2 + k_{12}^2 \dot{z}_1 \dot{z}_1$$

$$= z_2 z_3 - k_{21} z_2^2 + k_{12}^2 z_1 z_2 - k_{11} z_1 \dot{z}_1 + z_2 \left[ a_{22} - \varphi_2 \dot{\varphi} - d_1 \right] \quad (29)$$

According to the equation (24)

$$\dot{V}_2 \leq z_2 z_3 - k_{21} z_2^2 + k_{12}^2 z_1 z_2 - k_{11} z_1^2 + e_2 \quad (30)$$

If $z_3 = 0$ and the matrix $A_2$ as below is a positive definite matrix, the tracking errors $z_1$ and $z_2$ will be bounded. So the next step is to make sure that the tracking error $z_3$ is either close to zero or in the desired range and the transient process of the controller should be ensured.

$$A_2 = \begin{bmatrix}
  k_{11}^3 & \frac{1}{2} k_{11}^3 \\
 -\frac{1}{2} k_{11}^3 & k_{21}
\end{bmatrix} \quad (31)$$

According to the definition of $z_3$ and the equation (15), we can get

$$\dot{z}_3 = \dot{x}_3 - \dot{\dot{x}}_2 = g_3 \dot{u} - f_{31} - \theta_{df_{32}} - \dot{x}_2 \quad (32)$$

According to the definition of $R_1$ and $R_2$, $g_3 > 0$ is right in all time. Similar to equation (20), the following adaptive robust control law $u$ is proposed

$$u = u_a + u_s$$

$$u_a = \frac{1}{g_3} \left( f_{31} + \dot{\theta} f_{32} + \dot{x}_{2c} \right)$$

$$u_s = \frac{1}{g_3} \left( u_{s1} + u_{s2} \right)$$

$$u_{s1} = -k_{31} z_3 \quad (33)$$

where $k_{31}, k_{32}$ are the design parameters of the controller.
By substituting equation (33) into equation (32), we can get

$$
\dot{z}_3 = -k_{31}z_3 + u_{s2} - \varphi_3^T \bar{\theta} + \frac{\partial z_2}{\partial x_2} \frac{d_1}{m}
$$

(34)

where

$$
\varphi_3^T = \begin{bmatrix}
\frac{\partial z_2}{\partial x_2} & \frac{\partial z_2}{\partial x_2} S_M(x_2) & \frac{\partial z_2}{\partial x_2} \frac{1}{m} & -f_{32}
\end{bmatrix}
$$

(35)

According to the equation (34), $u_{s2}$ needs to meet the following calming conditions

$$
z_3 \left[ u_{s2} - \varphi_3^T \bar{\theta} + \frac{\partial z_2}{\partial x_2} \frac{d_1}{m} \right] \leq \varepsilon_3
$$

(36)

$$
z_3 u_{s2} \leq 0
$$

(37)

where

$$
u_{s2} = -k_{32} \left( x, \theta_M, \delta_d \right) z_3 \triangleq \frac{-h_3}{2\varepsilon_3} z_3
$$

(38)

where $\varepsilon_3$ is the design parameter of the controller. $h_3$ is defined by

$$
h_3 \geq \| \varphi_3 \|^2 \| \theta_M \|^2 + \left( \frac{\partial z_2}{\partial x_2} \frac{d_1}{m} \right)^2
$$

(39)

Similar to equations (28)–(30), the stability also can be identified. Therein the matrix $A_3$ as below is also a positive definite matrix

$$
A_3 = \begin{bmatrix}
  k_1^3 & -\frac{1}{2} k_1^3 & 0 \\
  -\frac{1}{2} k_1^3 & k_2 & -\frac{1}{2} \\
  0 & -\frac{1}{2} & k_3
\end{bmatrix}
$$

(40)

By assigning the parameters $k_{2,2}$ and $k_{3,2}$ as sufficiently large constants, the robust controller $z_{2,2}$ and $u_{s2}$ can be simplified and this method is usually effective. Based on the above design, the simulation program of the controller was developed by C language with the S function form. The control system model of the vertical stabilizer was established in the Simulink module of MATLAB.

**Mechanical–electrical–hydraulic integrated dynamic model of marching tank**

According to the control principle of the vertical stabilizer, the electrohydraulic position servo system model was established in Amesim software with standard hydraulic, mechanical, and signal libraries to calculate the output force of hydraulic cylinder. It mainly consists of hydraulic cylinder, electrohydraulic servo valve, hydraulic pump, motor, accumulator, and fuel tank. The main parameter values of hydraulic servo position system are shown in Table 1.

Combining the controller with the established electrohydraulic position servo system model of vertical stabilizer and the dynamic model of tank, a mechanical–electrical–hydraulic integrated dynamic model of marching tank was established. In the dynamic model, the hydraulic cylinder force was simulated by creating an axial force
between the piston rod and cylinder and it was calculated by the electrohydraulic position servo system model. The output variables called by the controller program include the elevation angular displacement, velocity and acceleration of hull, the displacement and velocity of piston rod, the pressures in the two chambers of hydraulic cylinder. During the numerical calculation, the sub models are connected through the equation of state, and the data exchange is carried out in each fixed sampling time step. When the sampling time small enough, the system is approximately considered unchanged in a single sampling time. That is, the data exchange is approximately considered to be real time.

In order to verify the established dynamic model of marching tank, the test results in literature\textsuperscript{23} were adopted. The vibration response of the hull is verified by real vehicle tests on the typical pavement. The road roughness is similar to the B level road. The vertical vibration acceleration was measured by the acceleration sensor on the first road wheel. During the test, three different gears were chosen, and the speeds corresponded to 14 km/h, 21 km/h, and 31 km/h, respectively. It can be found by comparing the test and simulation results, the peak frequencies of test and simulation date of the second gear were 27.8 Hz and 24.8 Hz, respectively and the error was 10.79%. The peak frequencies of test and simulation date of the third gear were 41.3 Hz and 38.8 Hz, respectively and the error was 6.05%. The peak frequencies of test and simulation date of the fourth gear were 61.6 Hz and 56.0 Hz, respectively and the error was 9.01%. The peak frequencies of the simulation and test were basically the same. With the increasing of speed, the peak frequency increases and the tendency is similar. It verified the rationality and credibility of the established mechanical–electrical–hydraulic integrated dynamic model of marching tank.

### Simulation and analysis

Stabilization accuracy\textsuperscript{1} is a major index related with the performance of stabilizer. It refers to the arithmetic mean of the swinging amplitude of marching tank gun and can be expressed as

\[
\sigma = \frac{1}{N} \sum_{i=1}^{N} |\theta_i| \tag{41}
\]

where \(N\) is the sampling quantity and \(\theta_i\) is the elevation angular displacement.

In the dynamic model, road roughness is the main factor causing hull vibration of marching tank. In order to analysis the performance of the designed vertical stabilizer controller, the three-dimensional road roughness model of D level road was reconstructed by using the harmonic superposition method.\textsuperscript{24,25} The random process of the three-dimensional road can be represented as

\[
q(X, Y) = \sum_{i=1}^{N} \sqrt{2} A_i \sin[2\pi(n_i X + \tau_i)] \tag{42}
\]

\[
\tau_i = \frac{e^{-2\pi Y_{nl}^{1.5}} \tau_1 + \sqrt{1 - e^{-2\pi Y_{nl}^{1.5}}}}{\sqrt{1 - e^{-2\pi Y_{nl}^{1.5}}} + e^{-2\pi Y_{nl}^{1.5}}} \tag{43}
\]

### Table 1. Parameters value of hydraulic position servo system.

| Parameter                        | Value   |
|----------------------------------|---------|
| Bulk modulus of fluid/MPa       | 1700    |
| Fluid density/kg\(\cdot m^{-3}\) | 850     |
| Piston diameter/mm              | 80      |
| Rod diameter/mm                 | 50      |
| Viscous friction coefficient/(N\(\cdot s/m\)) | 80      |
| Leakage coefficient/m\(^3)/(N\(\cdot s\)) | 9e-12   |
| Length of stroke/mm             | 400     |
| Relief valve cracking pressure/MPa | 20      |
| Total flow gain/(m\(^3\)/s\(\cdot mA\)\(\cdot \sqrt{N}\)) | 4e-8    |
| Valve rated current/mA          | 40      |
where $X$ is the length in the road direction; $Y$ is the length vertical to the road direction; $A_i$ is the amplitude of the harmonic fluctuation corresponding to the center frequency; $\alpha_Y$ is the random phase of road roughness excitation in the direction of $X$; $\alpha_1, \alpha_n$ are the evenly distributed random number in $[0, 1]$. The three-dimensional road roughness model accords with the classification standard of China and has been verified. Referencing the actual driving condition of tank, the vibration characteristics at the speed of 20 km/h on the D level road are calculated and analyzed. The sighting angle is $0^\circ$. Because the driving speed is not stable at the initial phase of simulation and only the vertical stabilizer is considered, the vertical vibration characteristic of gun after 5 s is analyzed in this paper.

**Stability analysis of controller**

The transient process of the system is concerned with the controller parameters. From the performance analysis of the adaptive robust controller in the literature, better tracking performance can be easily obtained by increasing the controller gain. However, large controller parameters may cause the higher gain feedback of system. Some of the assumptions when designing the controller may be destroyed. Obviously, the conclusion is not always correct at this time. In the process of dynamics calculation, the high controller gain caused a high frequency vibration of the gun system and the vertical stabilizer cannot control the vertical vibration of gun effectively. Therefore, the tracking accuracy of the actual vertical stabilizer cannot be simply improved by increasing the gain of the controller. It indirectly indicates that it is not accurate to design the vertical stabilizer controller based on an approximate transfer function of the tank mechanical system. After several trial calculations, the final control parameters are shown in Table 2.

As a position servo system, the vertical stabilizer controls the gun through the movement of piston rod. The tracking error of the adaptive robust controller is shown in Figure 5. It can be seen from the figure that, due to the

Table 2. Controller typical parameters.

| $k_1$ | $k_2$ | $s_1$ | $k_3$ | $s_2$ |
|------|------|------|------|------|
| 500  | 400  | 250  | 500  | 400  |

**Figure 5.** The tracking error of ARC.
effect of the adaptive rate, the actual displacement of the hydraulic rod can effectively track the expected displacement during the whole process of tank moving. The tracking error is small and the extreme value is only 1.03 mm. It meets the design requirement of vertical stabilizer.

In order to verify the performance of the designed adaptive robust controller, the traditional PID controller was introduced. To ensure the fairness of comparison, the parameters of the PID controller were set to make sure that the controller works on the optimal control parameters as far as possible. The control parameters of PID controller selected in the paper are shown in Table 3. The tracking error of PID is shown in Figure 6, and the extreme value is about 3.79 mm. By comparing the steady-state errors of the two controllers, the tracking error of the adaptive robust controller is smaller than the value of PID controller, and the extreme value reduces 56.99%. Furthermore, according to Figure 7, the piston rod velocity can also track the desired value effectively with the adaptive robust controller. It further demonstrates the robust performance of the adaptive robust controller.

The vertical vibration of the gun can be characterized by the cradle elevation angular displacement when the nonlinear factors of the gun are not taken into account. The comparison between the cradle elevation angular displacements with two different controllers is shown in Table 4 and Figure 8. The cradle elevation angular displacements of marching tank are controlled effectively by these two controllers. From the literature, the target distance deviation caused by the fire angle deviation because of the vertical angular vibration of hull is 3–5 times greater than that caused by the projectile horizontal velocity. Table 4 shows that the cradle vertical amplitude of marching tank without control is large, and the extreme value is 15.38 mrad. Under such conditions, the firing accuracy is difficult to meet the requirement.

Under the control of the adaptive robust controller, the extreme value of cradle elevation angular displacement decreases to 1.41 mrad and the stabilization accuracy is 0.67 mrad. Under the control of the PID controller, the extreme value of cradle elevation angular displacement decreases to 2.92 mrad and the stabilization accuracy is

| Table 3. Controller typical parameters of PID. |
|-----------------------------------------------|
| $k_p$  | $k_i$  | $k_d$  |
| 300    | 60     | 1      |

Figure 6. The tracking error of PID.
Figure 7. The velocity of the piston rod of ARC.

Table 4. Cradle vertical angular displacement.

| Vertical angular displacement (mrad) | Extreme value | Stabilization accuracy | RMS   |
|--------------------------------------|--------------|------------------------|-------|
| ARC                                  | 1.41         | 0.67                   | 0.77  |
| PID                                  | 2.92         | 0.76                   | 0.95  |
| No control                           | 15.38        | 5.26                   | 6.35  |

Figure 8. The cradle elevation angular displacement. (a) ARC and (b) PID control.
0.76 mrad. In literature, the vertical stabilization accuracy of tanks driving on a standard medium rolling road with a moderate speed (20–25 km/h) is about 0.5–1.0 mili (about 0.52–1.05 mrad). The two controllers can both satisfy the requirement of actual vertical stabilization accuracy. The control method is feasible and the established mechanical–electrical–hydraulic integrated dynamic model is correct and credible.

Compared with the PID controller, the stability accuracy of the marching tank is improved by 11.84% under the action of the adaptive robust controller. The extreme value of the elevation angular displacement decreases by 51.72% and the Root Mean Square (RMS) decreases by 18.94%. In addition, from Figure 8, the elevation angular displacement curve under the control of the adaptive robust controller is smoother. Obviously, the adaptive robust controller has better stabilizing effect.

Nonlinear factors analysis

In the actual driving process of tank, the muzzle vibration is different from that at the cradle due to the nonlinear factors such as the flexibility of the rear-seat components and the collision between components. In this paper, the following nonlinear factors were considered: the barrel flexibility, the contact between the barrel and the bushing, the contact between the trunnion and the bearing.28,29 The finite element model of the barrel was established in Hypermesh and it was discretized by isoparametric hexahedral elements. The modal neutral file of the barrel was obtained by the modal analysis based on the modal reduction method. The flexible barrel model was established using the modal neutral file, which was connected to the breech ring with the interface node in the model. The contact forces were calculated by the user subroutines, which were inserted with application programming interface. The normal contact force $f$ between the barrel and the bushing was calculated by the nonlinear spring damping model and it can be written as

$$f = \begin{cases} k \cdot \delta^n + c(\delta) \cdot \dot{\delta} & \delta > 0 \\ 0 & \delta < 0 \end{cases}$$

(44)

where $k$ is the contact stiffness coefficient; $\delta$ is the penetration depth, and $\dot{\delta}$ is the velocity vector at the contact point; $n$ is the nonlinear index; $c(\delta)$ is the nonlinear damping coefficient that can be expressed as

$$c(\delta) = \text{step}(\delta, 0, 0, d_{\text{max}}, c_{\text{max}})$$

(45)

where $c_{\text{max}}$ is the normal maximum damping coefficient, $d_{\text{max}}$ is the maximum permissible penetration depth.

The contact force between the trunnion and bearing was calculated by the revolution clearance joint model, and it can be written as

$$f = K_n \delta^n + D_n \dot{\delta}$$

(46)

The first term on the right side of equation (46) represents the elastic deformation force during the collision, and the second term represents the damping force during the collision, in which

$$K_n = \frac{1}{8} \pi E' \sqrt{\frac{2\delta (3(r_B - r_T) + 2\delta)^2}{(r_B - r_T + \delta)^3}}$$

(47)

$$D_n = \frac{3K_n (1 - c_e^2) e^{2(1-c_e)\delta^n}}{4\delta^{(\gamma)}}$$

(48)

where $r_B$, $r_T$ are the radiuses of the shaft and bearing; $c_e$ is the recovery coefficient; $\dot{\delta}^{(\gamma)}$ is the initial relative velocity at the contact point; $E'$ is the composite elastic modulus, which can be calculated by

$$\frac{1}{E'} = \frac{1 - v_1^2}{E_1} + \frac{1 - v_2^2}{E_2}$$

(49)
The muzzle elevation angular displacements under the control of two kinds of controllers are shown in Figure 9. The barrel bends downward when tank moving with a low speed on a flat road because of the barrel flexibility. It causes the muzzle angular displacement smaller than the value of cradle. But the waveforms of the two different elevation angular displacement curves are almost accordant. It demonstrates that the excitation receiving from the random road is the main factor influencing the vibrations of muzzle and cradle when the tank drives at a low speed. However, the vibration frequency of muzzle is higher than that of cradle because of the nonlinear factors of gun. In addition, we can also find that the muzzle vertical vibration amplitude is greater under the control of PID controller and it has higher frequency vibration components. Then, the frequency of the gun passing the shooting gate under the control of PID controller is higher compared with that under the control of adaptive robust controller. As a result, the gun passes the shooting gate at a high speed. Obviously, it is not conducive to ensure the firing accuracy of tank firing on the move.

Figure 9. The muzzle elevation angular displacement.

Figure 10. The power spectrum density curve of the muzzle elevation angular acceleration. (a) ARC and (b) PID control.
Figure 10 is the power spectrum density curve of the muzzle elevation angular acceleration obtained by fast Fourier transform. Due to the filtering of the track for road excitation, the upper frequency of road input is limited. The dynamic response of the muzzle is mainly distributed below 100 Hz, and the vibration energy of the part greater than 100 Hz is small. So frequency range between 0 and 100 Hz is analyzed.

As shown in Figure 10, the muzzle vibration energy of marching tank is mainly distributed around 20 Hz, 35 Hz, 52 Hz, and 80 Hz and the vibration energy at the low-frequency component around 20 Hz is the largest. Comparing with the value under the control of PID controller, the vibration energy at the low-frequency component around 20 Hz under the control of adaptive robust controller is bigger. The vibration energy at the high-frequency component around 52 Hz and 80 Hz is smaller obviously.

Through the above analysis, we can found that the vertical stabilizer controller designed by the adaptive robust control method had better comprehensive stability when comparing with the traditional PID control. However, some nonlinear factors of the gun system will seriously influence the desired muzzle stabilization accuracy of the vertical stabilizer. It is not conducive to improve the firing accuracy of marching tank, and more in-depth studies are needed.

In addition, these nonlinear factors have great influence on the tracking performance of the control system. The tracking error of ARC without considering the collisions between components is shown in Figure 11. The contact between the barrel and the bushing, the contact between the trunnion and the bearing were replaced by the translational joint and revolute joint. The control law and the controller parameters were the same as above. From Figure 11, the tracking error is greatly reduced without considering the collisions between components, and the extreme value is only 0.54 mm. The cradle stabilization accuracy is 0.31 mrad and the extreme value of elevation angular displacement is only 0.85 mrad. Through the comparison in Figure 5, the curve of piston rod displacement tracking error is smoother. The collisions between components have a detrimental effect on the tracking performance of control system. The gun system and the vertical stabilizer control system are mutually coupled. It is not accurate to simplify the tank dynamics model as a linear transfer function. The high stabilization accuracy based on that assumption is difficult to get in practice due to the limitation of mechanical system.

**Modeling and analysis of muzzle error compensation signal**

From the above analysis, though the vertical stabilizer worked efficiently, the muzzle vertical vibration was still overall greater than the value of cradle. It was because the desired displacement of piston rod calculated according
to equations (8) and (9) ignored the nonlinear factors of gun system such as the barrel flexibility. In fact the cradle and muzzle vibrations were thought as the same during designing the vertical stabilizer. The angle and angular velocity gyroscope providing control input to the control system are both installed on the cradle, and the stabilization goal is cradle. In order to decrease the muzzle vibration of marching tank, the muzzle error compensation signal was added into the designed controller, thus, the calculation formula of the desired displacement of the piston rod can be expressed as

![Figure 12. The tracking error of adaptive robust controller with muzzle error compensation signal.](image)

![Figure 13. The muzzle elevation angular displacement with muzzle error compensation signal.](image)
\[ x = \arccos\left(\frac{a^2 + l_{th}^2 - l^2}{2al_{th}}\right) + \varphi - \omega_p + \theta_i \quad (50) \]

\[ y = \Delta l = \sqrt{a^2 + l_{th}^2 - 2al_{th}\cos(x)} - l \quad (51) \]

where, \( \varphi \) is the difference between the cradle elevation angular displacement and muzzle. It can be read directly from the model by displacement function in calculating or measured by the eddy current displacement sensor installed on the muzzle. The control law and control parameters of the controller were the same as above. In order to analyze the stabilizing effect of the vertical stabilizer after adding the muzzle error compensation signal, the muzzle vibration at the speed of 20 km/h on the D level road was calculated and analyzed. Figure 12 is the tracking error of the adaptive robust controller with muzzle error compensation signal. The tracking error is small with an extreme value of 1.03 mm. It is similar to the tracking error in Figure 8, so the change of adding the muzzle error compensation signal has little impact on the stability of control system. The tracking performance of the designed controller changes within an acceptable range.

The muzzle elevation angular displacement with muzzle error compensation signal is shown in Figure 13. The muzzle vibrates near the sighting angle. As shown in Table 5, the muzzle elevation angular displacement is small with an extreme value of 1.64 mrad and the muzzle stabilization accuracy is 0.59 mrad. Comparing with Figure 9, the muzzle vertical vibration is smaller than the value before adding the muzzle error compensation signal. The detrimental effect of the nonlinear factors of gun system on muzzle vibration is controlled. So the method of adding the muzzle error compensation signal in the designed adaptive robust controller is feasible and effective.

As show in Figure 13, the muzzle elevation angular displacement has more high-frequency component than that of cradle because of the muzzle error compensation signal. This is also reflected in the tracking error in Figure 12. Obviously, it also affects the first-round-hit accuracy of marching tank and it will be the focus of future work.

**Conclusions**

Achieving high firing accuracy of marching tank is challenging. In order to better understand the dynamic behavior and decrease the muzzle vibration of marching tank, dynamic simulation on vibration control of marching tank gun was conducted. The main contributions and conclusions are as follows:

1. By comparing with the traditional PID control method, the adaptive robust controller present in the paper had better tracking performance and control effect. The low-frequency components in the muzzle vibration energy were greater.

2. The mechanical, hydraulic, and control subsystems of tank system affect each other. It is not accurate to simplify the tank dynamics model as a linear transfer function in the existing work, and the high stability accuracy obtained based on this is difficult to achieve in practice due to the limitations of the mechanical system. Research about the mechanical–electrical–hydraulic integrated dynamic model of marching tank is valuable and very helpful for the calculation accuracy.

3. By adding the muzzle error compensation signal in the designed controller, the detrimental effect of the nonlinear factors of gun system on muzzle vibration was controlled and the muzzle elevation angular displacement was reduced. The compensation signal had little influence on the tracking error and stability of the controller.

This study provides an approach to decrease the muzzle vibration of marching tank. However, the established model needs to be verified with more test values. In addition, the controller designed in the paper ignored the
flexibility, clearance, and some other nonlinear factors of gun system, and it is the focus of the follow-up study. The projectile-barrel coupling problem of marching tank and the calculation of impact point based on the exterior ballistics can be carried out on this basis.

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ORCID iD
Yu Chen https://orcid.org/0000-0002-1773-4612

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