INVESTIGATE THE EFFECT OF THE THROTTLING RING WEAR ON THE MOVEMENT OF THE ARTILLERY ON THE TANK WHEN FIRING

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ABSTRACT

Artillery weapons in general and the artillery types on the tank in particular, after long-term use, some parts on the cannon will be worn out due to friction, and impact, especially the details that need high precision such as the throttling ring and the throttling rod. In this paper, a mathematical model is established to evaluate the influence of the change in diameter of the throttling ring on the reliability of artillery when firing. Numerical calculation applied to the 100mm artillery on the T55 tank. The calculation results show that: The law of recoil braking force is changed when the diameter of the throttling ring dR is greater than 38.5mm, then the maximum pressure in the cavity 1 decreases suddenly by 16.6%. This is undesirable to maintain the stability of the cannon on the tank when firing. The results of the article contribute to re-evaluating the tactical and technical features of the cannon to propose a plan to preserve, maintain, replace, and improve the service life of the weapon in combat.

Keywords: *Recoil mechanism, wear of throttling ring, stability, cradle.*

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1. INTRODUCTION

The recoil mechanism is an intermediate link between the recoiling parts and the carriage. It has the purpose of consuming most of the recoil brake energy, consuming the remanent energy of the recoiling parts when counter-recoil motion. Especially with the use of adjusting hydraulic braking force according to a defined rule, this rule is predetermined in the design process.

The recoil brake during operation is affected by many factors such as air mixed in the oil with the phenomenon of the "diesel effect". At the same time, increasing the pressure above the recommended value will affect the sealing ability of the sealing element. One of the important factors affecting the stopping force of the artillery when firing is the clearance area between the throttling ring and the throttling rod, this rule is determined by the diameter of the throttling ring and the profile of the throttling rod whether when the artillery was pushed back and pushed up. This change in the gap area will change the law of movement of the cannon on the rack when firing. In addition, the working of related parts and structures such as the shell removal mechanism, the shell ejection mechanism, the loading mechanism, etc. will be seriously affected. Thực tế làm việc của pháo, dòng dầu chuyển động với vận tốc cao qua khe hở giữa vòng điều tiết và cán điều tiết. In practice, the oil flow moves at high velocity through the gap between the throttling ring and the throttling rod. This movement causes wear on the parts, especially the throttling ring, which has a thin thickness. The wear of the throttling ring increases the diameter of the throttling ring, and the clearance area between the throttling ring and the throttling rod also increases. Therefore, it is necessary to investigate the effect of the change of the regulation ring on the stability of the cannon when firing.

In the world, there have been many studies on the recoil mechanism of tank artillery. However, studies have only focused on showing the influence of the motion characteristics of the oil flow in the chambers of the recoil mechanism [1-3], Some studies have shown the influence of oil temperature on cannon's reverse resistance [4-7]. In the framework of the article, a mathematical model was set up to investigate the influence of the change of throttling ring diameter on the movement of the artillery on the tank when firing.

2. THEORETICAL BASIS

2.1. Set up a mathematical model to determine the resistance of the recoil mechanism

The theoretical and computational basis will be conducted on tank artillery, with a penetrating round. Figure 1 shows the preliminary structure of the recoil brake mechanism with 4 spaces and illustrates the flow of oil through the orifice area (the main component that creates the braking resistance when recoil).

The principle of energy consumption of a hydraulic recoil brake is: when the piston and piston rod move backward, compressing oil from space 1 (working space) through the orifice area to space 2 (non-working cavity), first the kinetic energy of the recoiling parts is converted into the motion of the oil (liquid) with strong jet streams when passing through the orifice area to space 2, see Fig. 1. Because the working area of the piston is much larger than the area of the orifice area, the speed of the jet streams is extremely large (can reach several hundred m/s), while the velocity of the recoiling parts is in the range of 8 - 15m/s. Then the jet streams chafe against each other causing the oil to heat up. Thus, the recoil energy is completely converted into heat, then transmitted to the outside and dissipated.

To simplify the process of setting up the mathematical model, we make some assumptions as follows:

- Oil is considered incompressible;
- Oil is considered to fill the chambers when working;
- The braking force created by the pressure difference between chambers is very small and it is difficult to determine the pressure pulsation;
- In the calculation process, we consider space 5 and space 1 to be a working space with the working area $S_{15} = S_1 + S_5$.

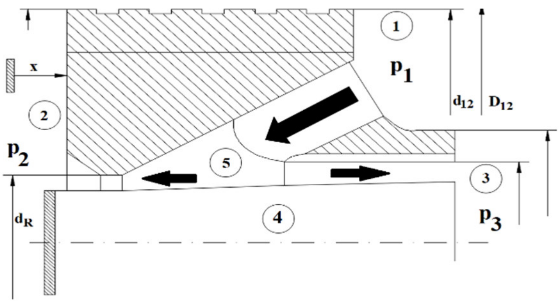

Fig. 1. Illustrate the flow of oil in the recoil mechanism

1, 2, 3, and 5 are working spaces of the recoil brake, 4 is the control rod; p_1, p_2, and p_3 are pressure of oil in spaces 1, 2, 3

The value of pressure p_2 is approximated by the formula, see [7, 8]:

$$p_2 \doteq \begin{bmatrix} \hat{p}_2 \text{ for } \hat{p}_2 > p_w \\ p_w \text{ for } \hat{p}_2 < p_w \end{bmatrix} \quad (1)$$

Where:

$$p_2 = p_{oo}\left(\frac{\rho_{v2}}{\rho_{v0}}\right)^n$$

p_w is the saturated vapor pressure of the liquid, n is the polytropic index;

p_{00} is the initial pressure in the recoil brake;

ρ_{v0} is the initial density of air and oil vapor at pressure p_{00};

$\rho_{v2} = \frac{m_{v2}}{V_{v2}} > 0$ is the instantaneous density of the mixture of air and oil:

$m_{v2} > 0$ is the instantaneous mass of the air in space 2;

V_{v2} is the instantaneous volume of the air in space 2 and determined by the formula:

$$\begin{cases} m_{v2} = m_{vz0} + \Delta m_v \\ V_{v2} = V_2 - V_{k2} \\ V_2 = V_{20} + S_2 \cdot x \end{cases} \quad (2)$$

Where: $m_{v20} > 0$ s the initial mass of the air in space 2; $V_{k2} = V_{k20} + \frac{\Delta m_{k12}}{\rho_{k0}}$ is the instantaneous volume of the oil in space 2; V_{k20} is its initial volume; Δm_{k12} is the mass of oil flowing from space 1 to space 2 during recoil motion; ρ_{k0} is the density of the oil at low pressure.

Concerning a very rapid recoil (between 100 and 150ms on the tank gun and from 250 to 300ms on the heavy towed gun) the vaporization of the oil in space 2 through can be neglected with the size of m_{v2} [11].

From the foregoing, it can be concluded that the pressure p_2 in space 2 of the recoil brake during recoil motion can be neglected compared with the pressure p_1, and p_3 in space 1 and space 3. This means that it does not affect the operation of the brake.

Thus, the generated braking resistance is mainly caused by the pressure p_1, p_3 and is determined by the equation, see [11, 12]:

$$F_b = F_1 - F_3 = S_1 p_1 - S_3(p_1 - p_3) \quad (3)$$

Where: S_1 is the working area of the piston when recoiling, and S_3 is the working area of space 3.

The gradient $\Delta p_{13} = p_1 - p_3$ can be expressed as quasi-stationary:

$$\Delta p_{13} = k_{13} \cdot \dot{x} = \frac{\rho_k}{2\mu_{13}^2}\left(\frac{S_3}{S_0}\right)^2 \cdot \dot{x} \quad (4)$$

Where S_0 is the flow area into cavity 3, S_3 is the cross-sectional area of cavity 3, μ_{13} is the relative flow coefficient from cavity 1 to cavity 3 of the liquid, and k is the density of the liquid.

The value of the discharge coefficient μ_{13} can be approximated to be a constant during the recoil, which corresponds to the value when the Reynolds number is small. Therefore, Δp_{13} does not have a great influence on the braking force F_b during recoil motion. Thus, to evaluate the reliability of the operation of the brake, we can evaluate it through the oil pressure in space 1, as well as the velocity and displacement of the recoiling parts versus time.

Applying the constant flow equation to the compressible liquid in the cavity 1 of the recoil mechanism is shown as:

$$S_1 dx - dV_c = \varepsilon_1 S_r(x) V_1 dt + S_\delta dx \quad (5)$$

Where:

$S_1 dx - dV_c$ is the flow of oil that is forced out of cavity 1; dV_c is the compression of the oil column due to pressure p_1; $\varepsilon_1 S_r(x) V_1 dt$ is the flow of oil pushed into the cavity 2 through the gap between the throttling ring and the throttling rod; V_1 is the actual velocity of the oil flow from cavity 1 to cavity 2; ε_1 is the compressibility of the oil flow; $S_r(x) = \frac{\pi dR^2}{4} - S_\delta$ is the area of the gap between the throttling ring and control rod of the recoil brake, dR is throttling ring diameter, S_δ is the regulatory rolling cross-sectional area at the time of calculation.

The compressibility of the oil is determined by the following expression:

$$\frac{1}{K} = \frac{1}{V_c}\frac{dV_c}{dp_1} \tag{6}$$

Where V_c is the volume of oil in cavity 1.

Substitute dV_c into (4). After the transformation, the pressure in cavity 1 is expressed in differential form as follows:

$$\frac{dp_1}{dt} = K\frac{\rho_k}{m_k}\left[\left(S_1 - S_\delta\right)\cdot\dot{x} - \varepsilon_1\cdot S_r(x)\cdot V\right] \tag{7}$$

Where: K is the bulk modulus of elasticity of liquid, rok is the density of the liquid, S_δ is the area of the control rod, ε_1 is the contraction coefficient of the stream, $S_r(x) = \frac{\pi dR^2}{4} - S_\delta$ is the area of the gap between the throttling ring and control rod of the recoil brake, dR is throttling ring diameter.

Substitute $S_r(x)$ into (6) equation:

$$\frac{dp_1}{dt} = K\frac{\rho_k}{m_k}\left[\left(S_1 - S_\delta\right)\cdot\dot{x} - \varepsilon_1\cdot\left(\frac{\pi dR^2}{4} - S_\delta\right)\cdot V\right] \tag{8}$$

2.2. Setting up the problem of recoiling part dynamics

All types of recoil brakes work based on the principles of accumulating part of the recoil energy in an elastic medium and dissipating a larger part of the energy of the recoiling parts into the liquid in the recoil brake. In this section, we will focus on the process of throttling the force of shot (impulse force of shot) acting on the carriage through the forces of the recuperator and the recoil brake. The dynamic model is generally based on the construction of the brake, see in Fig. 2.

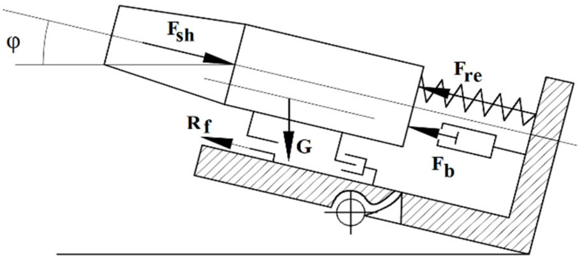

Fig. 2. Forces acting on the recoiling part

After the shot, the recoiling parts are braked by the following forces: the force of the brake (brake force) F_b, the force of the recuperator F_{re}, and the total friction resistances R_f. These forces are opposite to the driving force of the shot F_{sh} and the gravitational component G_r in the direction of the recoil motion.

Differential equation of recoil movement [8]:

$$\begin{cases} \frac{dV}{dt} = \frac{F_{sh} - R_{total}}{m_r} \\ \frac{dX}{dt} = V \end{cases} \tag{9}$$

The total braking resistance is given by the relation:

$$R_{total} = F_b + F_{rp} + R_f - G_r \sin\varphi \tag{10}$$

Where:

$$\begin{cases} F_b = F_{B1} - F_{B3} \\ F_{B1} = p_1 \cdot S_1 \end{cases} \tag{11}$$

p_1 is the liquid pressure in volume 1 of the brake, S_1 is the reduced working area of the brake piston during recoil, and F_{B1} is the braking force caused by the flow of liquid from volume 1 to volume 2. To simplify the calculation process, we assume that the liquid in volume 3 of the brake is incompressible, the volume has no air and has rigid walls. The deformation of the inner wall of the brake cylinder caused by liquid pressure creates a gap with a time-dependent cross-section. However, the braking resistance generated by this gap is not significant, so in this mathematical model, the deformation of the cylinder will not be taken into calculation.

The braking force caused by the flow of liquid from volume 1 to volume 3 [10]:

$$F_{B3} = p_{13}\cdot S_3 = \frac{\rho_K}{2}\frac{1}{\mu_3^2}S_3^3\left(\frac{\dot{X}}{S_0}\right)^2 \tag{12}$$

Where μ_3 is the discharge coefficient, S_3 is the area of the counter-recoil break, and S_0 is the area of the counter-recoil valve.

The force of the recuperator during recoil is given by the relation [10]:

$$F_{rp} = F_{rp0}\left(\frac{H_0}{H_0 - X}\right)^n,\ n = 1.4 \tag{13}$$

Where: H_0 is the reduced height of the gas column in the recuperator, $H_0 = \frac{V_{rp}}{S_{rp}}$; n is a polytropic exponent, the value of which is between 1.2 and 1.4 for the gas's nitrogen and air. In this case n = 1.4 [11].

F_{rp0} is the initial recuperator force and is given by:

$$F_{rp0} \doteq 1{,}1\cdot G_r\left(\sin\varphi_{max} + f\cdot\cos\varphi_{max} + \upsilon\right) \tag{14}$$

φ_{max} is the maximum angle of elevation and $\varphi_{max} = 30^o$

The total frictional resistance is given by the relation, see [10]:

$$R_f = G_r\left(f\cos\varphi + \upsilon\right) \tag{15}$$

Where: G_r is the recoiling part weight; f is the friction coefficient between the recoiling parts and the guiding in the cradle; υ is the friction coefficient in the seals, in the packing of the brake, the value of which depends on the pressure in the brake and recuperator, φ is the limit angle of elevation (most often 0°). We choose values for the calculation f = 0.16, and υ = 0.4..

Solving the system of differential equations (8), and (9) using the Runge-Kutta method of type 4 in Matlab software, the obtained results are the motion law of the recoiling part and the parameters working of the recoil mechanism.

3. RESULTS AND DISCUSSION

3.1. Input parameters

The input parameters of the cannon's recoiling part motion problem include: Input parameters are determined based on design documents, and some parameters such as the mass, center of gravity, and moment of inertia of objects are determined based on Solidworks software; Some other parameters are measured directly on the artillery; The equation for determining the force of the shot acting on the cannon and the system of interior ballistic equations are detailed in the document [10], the results are shown in Figs. 3, and 4.

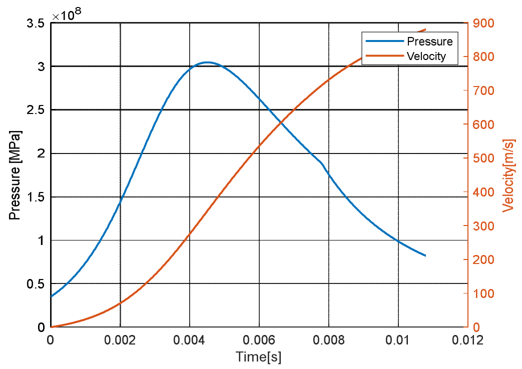

Fig. 3. Pressure in the barrel versus time

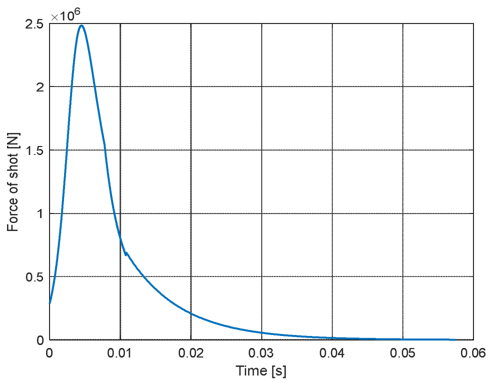

Fig. 4. The force of shot versus time

3.2. Investigate the influence of the change in the diameter of the throttling ring on the movement of the artillery when firing

To investigate the effect of the change in diameter of the throttling ring on the movement of the cannon's recoil part when firing, the diameter of the throttling ring was changed to 38.01, 38.1, 38.3, 38.5, 39.5, and 40mm, respectively. These values are substituted into the equation for determining cavity pressure 1, solving the system of equations (8), and (9), and the motion law of the recoil part is determined. The results are shown in Figs. 5, 6 and 7.

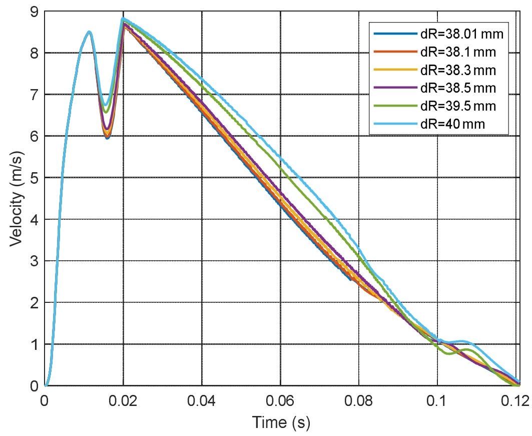

Fig. 5. Velocity of the recoiling parts versus time

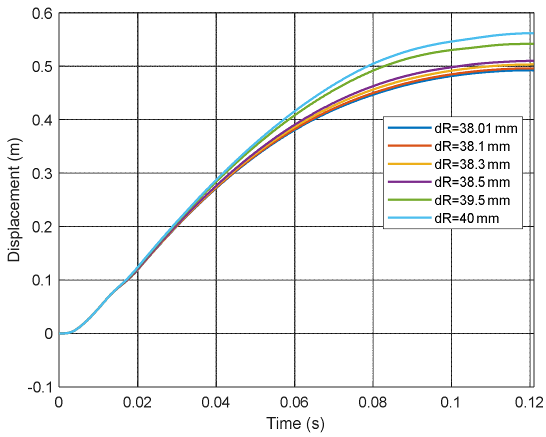

Fig. 6. Displacement of the recoiling parts versus time

It can be seen in Fig. 5 that the velocity of the recoiling parts does not change much as the diameter dR increases. However, when the wear of dR is greater than 2mm (dR = 40mm), in the second phase of the recoil motion, the velocity is destabilized leading to the oscillation of the recoiling parts. Similar to Fig. 6 the displacement of the recoiling parts will increase gradually as the diameter dR increases, in the special case when dR = 40mm (corrosion is close to 2mm), the recoil length exceeds the allowable limit (560mm). This caused a mechanical impact at the end of the recoil motion and led to instability of the gun as well as damage to some parts of the brake mounted on the gun mount.

Comment:

In Fig. 7 it is clear that the maximum value of the pressure in space 1 gradually decreases as the diameter dR increases, and when dR > 38.5 mm, the maximum pressure value achieved by the oil in space 1 will move at the end of the period of recoil motion. For the tank's tactical performance requirements, changing the braking resistance graph (pressure graph) leads to errors during direct aiming as well as adjustment for the next shot.

To limit the wear of the dR diameter, it is required to regularly check the quality of the oil in the recoil brake such as pH, lubricating ability, and kinematic viscosity,... During the design and machining process need to pay attention to checking and measuring accurately; During the repair and maintenance process, there must be a strict procedure for removing and assembling the details and assemblies of the recoil brake to avoid impact and abrasion.

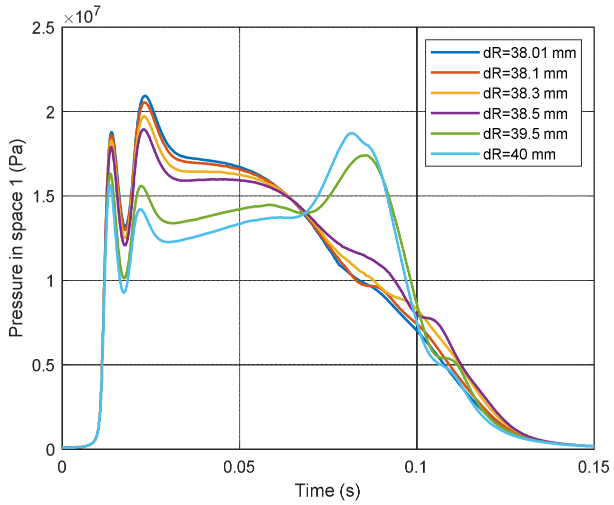

Fig. 7. Pressure in space 1 of the brake versus time

4. CONCLUSIONS

In this paper, a mathematical model that determines the motion of the recoil part on the artillery is established. Research content has applied modern means with the help of calculation software for high-accuracy results. With the research results obtained, some conclusions have been made as follows:

- The establishment of a system of differential equations of the motion of the recoil part on the artillery is done carefully and completely, bringing the theoretical problem closer to the actual model;

- The survey results show the influence of the change in the diameter of the throttling ring on the movement of the artillery when firing. With just a small change of this value, the artillery can operate unreliably or damage some parts when fired;

- The article also shows the working limit of the artillery when the diameter of the throttling ring is changed excessively (> 38.5mm). In this case, the movement law of the artillery is abnormally changed, seriously affecting the working cycle of the artillery.

- The results are applied to research on the reliability of weapons systems as well as in the design, manufacture, and repair of artillery guns and equipment using hydraulic recoil mechanisms with similar structures.

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