STATEMENT OF THE PROBLEM

A significant increase in the effectiveness of the artillery system can be achieved by improving the stability of the gun when firing. The solution to this problem seems possible by improving the course of work processes in recoil devices. As a result, based on the well-known models of the artillery gun working processes during firing, the authors consider it possible to build a refined mathematical model of these processes using the method of small deviations. In this case, the following subtasks should be solved:

– a comprehensive study of the course of the hydrodynamic working process of the brake of the retractable parts was carried out, which will make it possible to simultaneously observe changes in the dynamic and kinematic characteristics of the center of mass of the retractable parts, eliminating the need to solve the direct and inverse problems of the recoil;

– the characteristics of the brake of the sliding parts have been formalized by the method of small deviations, which will take into account the changing operating conditions and the technical condition of the brake, which have a significant effect on the immobility and stability of the tool;

– the links between the working process of the brake of the sliding parts and the stability of the tool have been formalized, which will make it possible to find and justify the maximum permissible deviations of the brake characteristics;

– scientifically substantiated a multi-parameter relationship between stability and the characteristics of the brake, which will make it possible to clarify the methods of restoring the characteristics of the brake of the retractable parts, and, therefore, to maintain the stability of the tool within the acceptable limits.

The indicated approach to the construction of a mathematical model will allow avoiding the use of experimental coefficients, unreasonable assumptions and restrictions, unauthorized linearization of dependencies. Provided a positive solution to the above subtasks, the mathematical model can be used as the basis for the study of the working process of the brake of the sliding parts, its calculation and design.

MODELING OF HYDRODYNAMIC WORKING PROCESS AND HEAT TRANSFER OF BRAKE RELAY PARTS OF ARTILLERY GUN

According to [1, 2], an artillery gun as an irreversible thermohydraulic machine consists of four main mechanisms: recoil masses, recoil, recoil brakes and muzzle brake.

In order to build mathematical models of the hydraulic workflow and heat transfer, we schematically represent the process of interaction of the main units in the form of Fig. 1, a, directly, the dynamics of the brake of the tool of the spindle type – Fig. 2.
Let’s compose the equation of conservation of energy of the recoil process using the differential equation of motion of the center of gravity of the tool:

\[ M_0 \frac{d^2 x_0}{dt^2} = P_{sw} - \sum R, \]

whence

\[ P_{sw} = M_0 \frac{d^2 x_0}{dt^2} + \sum R, \tag{2} \]

where \( M_0 \) - gun mass; \( \frac{d^2 x_0}{dt^2} \) - acceleration of the movement of guns; \( R \) - the reduced force of resistance to rollback or is the resultant of the resistance forces in the oil seals and on the collars, the friction force on the cradle guides and the axial component of the gravity of the rollback parts; \( R_{roll} \) - resultant forces acting on the barrel in the direction of the axis; the components of this resultant are: during the period of movement of the projectile along the channel – the pressure force of the powder gases against the channel walls, the resistance force of the rifling, and after the projectile leaves the channel, the reactive force of the gases flowing out of the barrel bore, the pressure force of the powder gases on the muzzle, in the presence of a muzzle the brakes are also the force developed by the last one.

In this case, the work of forces (2) at small displacements will be:

\[ P_{\text{roll}} \cdot \Delta x = M_0 \left( \frac{dx_0}{dt} \right)^2 \cdot \Delta t + M_0 \left( \frac{dx_0}{dt} \right) \Delta t - \sum Q_\text{roll} (\Delta t) + P_{\text{roll}} \cdot \Delta x_0, \tag{3} \]

hence the force acting on the brake rod \( P_{\text{roll}} \) on rollback will be equal to:

\[ P_{\text{roll}} = \frac{1}{\Delta x_0} \left[ \left( P_{\text{roll}} \sigma \Delta x_0 \right) + M_0 \left( \frac{dx_0}{dt} \right)^2 \cdot (\Delta x_0) + M \left( \frac{dx_0}{dt} \right) (\Delta x_0) \right] \tag{4} \]

where \( \sum Q_\text{roll} \) and \( \sum Q_\text{roll} \) - warmth corresponding to work.

This equation is solvable, if \( P_{\text{roll}} \) is determined through the forces acting on the tool according to the d’Alembert principle, and when calculating the speeds of movement of the centers of mass of the rollback parts and the center of gravity of the tool, the hydraulic workflow is considered taking into account the heat released into the environment.

We accept the force \( P_{\text{roll}} \) as outrageous. Its value determines the rate of fluid overflow through variable calibration sections and the rate of pressure change \( \left( P_1, P_2, P_3, P_4, P_5 \right) \) in variable volumes \( \left( V_1, V_2, V_3, V_4, V_5 \right) \) of the spindle brake (Fig. 2).

To draw up differential equations for the displacement of the center of mass of the recoil parts, we will use the hydrodynamic parameter – the acoustic resistance of the fluid during overflow [3]:

\[ Z = c \cdot \rho = \frac{dV}{d\tau} \cdot \frac{1}{\left( \mu f \right)}, \tag{5} \]

where \( z \) - remote impulse, \( c \) - fluid flow rate in the section; \( \rho \) - density; \( \mu f \) - effective values of the calibration flow areas, \( m^2 \).

Leaving the dimension of expression (5), this is the specific impulse:

\[ Z = \frac{\rho}{dx/d\tau}, (kg/s/m^3). \tag{6} \]

Consequently, the product of the specific impulse \( z \) and the fluid flow rate is the force:

\[ Z \cdot \rho = \sqrt{2 \cdot \Delta P} = F(H). \tag{7} \]
Under the action of the latter, the speed of fluid flow changes, and, consequently, the speed of movement of the recoil parts.

Taking into account dependencies (5 – 7), the differential equation of the piston rod displacement, i.e. the center of mass of the brake of the retractable parts will take the form (Fig. 3):

\[
\frac{d^2 x_0}{d\tau^2} = \frac{1}{m_K} \left[ F_K + \delta_K x_K \right] - f_x P_3
\]

(9)

\[
\frac{d^2 x_K}{d\tau^2} = \frac{1}{m_K} \left[ P_4 f_{k,4} - (F_6 + \delta_{k,4}) \right]
\]

(10)

When solving the system of equations (1, 8 – 10) at the first stage, the values \( P_1 \ldots P_3 \) are calculated through the rate of their change at the moment \( \tau + \Delta \tau \). Let’s carry out the derivation of the parameters \( P_1 \ldots P_5 \), using the example of one volume. The rest will be written by analogy.

For an arbitrary brake volume \( V_i \), you can write (volume \( V_i \) (Fig. 3))

\[
\frac{dV_i}{d\tau} = \frac{1}{\beta} \frac{dV_i}{d\tau} = \frac{1}{\beta} \frac{dV_i}{d\tau} = \frac{1}{\beta} \frac{dV_i}{d\tau}
\]

where \( dV_i \frac{dV_i}{d\tau} = \left[ \frac{dx_0}{d\tau} f_x - \left( \sum \frac{dV_i}{d\tau} \right) \right] \) – rate of volume change \( V_i \); \( \beta \) – fluid compressibility factor, \( [m^3/l] \); \( \sum \frac{dV_i}{d\tau} \) – change amount \( \Delta V_i \); \( f_x \) – piston area.

In this case, equation (12) can be written in expanded form:

\[
\frac{dP_i}{d\tau} = \frac{1}{\beta V_i} \left[ \frac{dx_0}{d\tau} \left( f_x - \left( \sum \frac{dV_i}{d\tau} \right) \right) \right] + \mu \rho \left( \frac{2}{\beta} \sqrt{P_4 - P_3} + f_x \frac{dx_0}{d\tau} \right)
\]

(13)

For volumes \( V_2, V_3 \ldots V_6 \) and \( V_5 \) dependencies will have a similar appearance. Differential equation for the rate of change of pressure in the volume \( V_j \):

\[
\frac{dP_j}{d\tau} = \frac{1}{\beta V_j} \left[ \frac{dx_0}{d\tau} f_j - \left( \sum \frac{dV_j}{d\tau} \right) \right] + \mu \rho \left( \frac{2}{\beta} \sqrt{P_4 - P_3} + f_j \frac{dx_0}{d\tau} \right)
\]

(14)

Differential equation of pressure change in volume \( V_i \):

\[
\frac{dP_j}{d\tau} = \frac{1}{\beta V_j} \left[ \frac{dx_0}{d\tau} f_j - \left( \sum \frac{dV_j}{d\tau} \right) \right] + \mu \rho \left( \frac{2}{\beta} \sqrt{P_4 - P_3} + f_j \frac{dx_0}{d\tau} \right)
\]

(15)

Differential equation of pressure change in volume \( V_4 \):

\[
\frac{dP_4}{d\tau} = \frac{1}{\beta V_4} \left[ \frac{dx_0}{d\tau} f_4 - \left( \sum \frac{dV_4}{d\tau} \right) \right] + \mu \rho \left( \frac{2}{\beta} \sqrt{P_4 - P_3} + f_4 \frac{dx_0}{d\tau} \right)
\]

(16)

Differential equation of pressure change in volume \( V_5 \):

\[
\frac{dP_5}{d\tau} = \frac{1}{\beta V_5} \left[ \frac{dx_0}{d\tau} f_5 - \left( \sum \frac{dV_5}{d\tau} \right) \right] + \mu \rho \left( \frac{2}{\beta} \sqrt{P_4 - P_3} + f_5 \frac{dx_0}{d\tau} \right)
\]

(17)

To close the solution of equation (4), it remains to add two equations of heat transfer from the roller and the brake cylinder, as for unsteady heat transfer in one cycle or for \( n \) cycles.

Considering that in equations (4 – 16) of the rollback process, the heat transfer components take into account only the energy losses during rollback, we introduce a number of restrictions. Determination of the components \( \sum Q_1 \) and \( \sum Q_2 \) is a separate problem of unsteady heat transfer, the simplification of which can lead to large errors. Therefore, its solution will be considered for the process of internal gasdynamic heat release, in the brake this heat will be equated to
the work of absorption of kinetic energy at steady-state heat exchange for one cycle:

\[
\frac{dP_{kin}}{dt} - \int dQ_h = \frac{dQ_h}{d\tau} \tag{18}
\]

For one cycle of the roll forward (during rollback, the compression process is assumed polytropic) [5]:

\[T_2 = T_1 \cdot \lambda^{n-1},\]

where \(\lambda\) - compression ratio in the knurling; \(n\) - polytropic index (average) during rollback and roll forward [4].

Then

\[dQ_h = \alpha \cdot f_h(T_2 - T_0) \cdot d\tau, \tag{19}\]

where \(\alpha = \alpha_k + \alpha_f\) - average heat transfer coefficient; \(f_h\) - cooled surface area; \(T\) - hour mizh in steps, \(a_k\) - convective heat transfer coefficient; \(a_f\) - radiant heat transfer coefficient.

Let us dwell on the heat exchange of the brakes of the recoil parts with the environment - the phase of transformation of the kinetic energy of the liquid flowing through the gauge sections into the heat energy released through the heat exchange surfaces.

Then, taking into account free convection and radiant heat transfer, the total heat transfer coefficient, as for systems with internal heat release, will be written [6]:

\[\alpha = \frac{a_k}{t_f - t_p} \cdot 5 \times \left[Bm / \Lambda \cdot C\right],\]

where \(t_f\) - surface temperature of the brake of the retractable parts of the artillery gun; \(t_p\) - temperature liquid; \(P_{kin}\) - Pomerantsev criterion, defined as recommended in the work [6].

Thus, equations (3, 4, 8−10, 13−19) represent a complete mathematical model of the brake of the recoil parts during recoil, taking into account the compressibility of the fluid, heat transfer and hydrodynamic processes in the brake of the rollback parts of the tool.

Differential equations describing the run-up process have a similar form (Fig. 4). The only difference is that the disturbing force, acting on the center of gravity of the tool, will be the force transmitted from the expansion of gases in the roll forward

Taking the process of gas expansion as polytropic, we can write (Fig. 1) [7]:

\[P_2 \delta V = P_0 \delta V; \quad F_2 = f_1P_2 = P_0 \left( \frac{\alpha_f}{\alpha_k} \right)^{n_f} \cdot f_1, \tag{20}\]

where \(f_1, f_2\) - corresponding areas of the roll forward piston; \(n_f\) - expansion polytropic exponent.

Then the equation of displacement of the center of gravity of the tool can be written:

\[\frac{d^2x}{dt^2} = \frac{1}{M_0} \left[ F_1 - \left( P_{kin} + M \cdot \frac{d^2x_h}{dt^2} \right) \right], \tag{21}\]

Differential equations describing the parameters \(P_f, P_k\) for the rolling process will have the form (Fig. 4):

\[\frac{d^2x_h}{dt^2} = \frac{1}{m_{T_h}} \left[ \left( P_3 - P_1 \right) \cdot \left( f_k + f_{K3} \right) \right] - \left( \delta_k x_{K3} + F_{K3} \right), \tag{22}\]

\[\frac{d^2x_k}{dt^2} = \frac{1}{m_{T_k}} \left[ \left( P_3 - P_1 \right) \cdot \left( f_k + f_{K3} \right) \right] - \left( \delta_k x_{K3} + F_{K3} \right), \tag{23}\]

Heat flows are defined similarly to the rollback process (equations 18 and 19) [7, 8].

Thus, we have a complete mathematical model of the hydraulic workflow taking into account the change in the coordinates of the center of gravity, i.e. a model linked to the immobility of the tool. In the given elements of the complete mathematical model of rollback and rollback processes, the defining parameters (in the first parts of the equations) are the...
products $\mu f_i$. This is nothing more than the boundary values of the brake characteristics, during rollback and during roll-off, varying within: 0-max-0, max-0-max or const on the rollback path, and during roll-off. Let’s note that they are unstable, and also depend on the conditions of temperature change, compressibility of the working fluid, time and coordinates of the energy source, which requires additional clarification.

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МАТЕМАТИЧНА МОДЕЛЬ РОБОЧИХ ПРОЦЕСІВ ГІДРАВЛІЧНОГО ГАЛЬМА ВІДКОТНИХ ЧАСТЕНІВ АРТИЛЕРІЙСЬКОЇ ГАРМАТИ

Аналіз сучасних підходів щодо моделювання внутрішньо камерних процесів гідравлічних гальм артилерійських гармат показує достатньо повне врахування авторами більшості аспектів постійні марматі, а саме: стисливості робочої рідини при гідравлічному виконанні гальм, наявності додаткових пристроїв – клапанів, модераторів, голов. Основою сучасних методик для визначення активних сил і моментів є рівняння Даламбера. Период післядії газів розглядається як нестабільна відповідь на вплив сил, які за законом Дурема, а вільний відхід – на основі законо збереження кількості руху на всіх періодах відкоту й накату. Однак, помітимо, що набути при виконанні усіх розрахункових умов даних методик за стабільність артилерійської гармати залежить повного гасіння кількості руху принципового рішення не зважаючи, що підходи допоможуть шляхом проведення комплексного дослідження цієї галузі досліджень гідравлічного робочого процесу гальма відкотних частин з одночасним спостереженням за змінами динамічних і кінематичних характеристик центру мас відкотних частин; формуванню характеристики дужнього гальма відкотних частин за методом маліх відхилень. Запропонований в статті підхід дозволяє авторам провести комплексне дослідження гідравлічного робочого процесу гальма відкотних частин з одночасним спостереженням змін динамічних і кінематичних характеристик центру мас відкотних частин. У цьому випадку усувається необхідність розв'язання прямого і зворотного задач накату. На основі методу маліх відхилень авторами запропонований теоретичний підхід з обґрунтуванням гранично допустимих відхилень характеристик гальма, що дозволяє формувати зв’язок робочого процесу гальма відкотних частин зі стабільністю гармати, а також уточнити методи відновлення характеристик гальма відкотних частин, що спираються на обґрунтований багатопараметричний зв’язок стійкості гармати з вказаною характеристикою.

Ключові слова: внутрішньо камери процеси артилерійських систем, гідравлічна гальма, відкатні пристрої.

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Стаття надійшла до редколегії 30.07.2020.