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Theoretical Analysis of the Impact of Deviations in the Magneto-Rheological Damper Axis from the Runner Axis in the Butt of a Firearm During a Shot

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Abstract. This paper presents a method of modelling the impact of a magneto-rheological (MR) butt-mounted damper on both the weapon and the shooter during the use of a firearm. A modification of the model presented in [1] is suggested. The study focused on determining the impact of deviations of the MR damper axis from the runner axis in the butt during recoil, especially on the energy dissipated. Sample results are presented for the analysis of a smoothbore hunting weapon, calibre 12/70. The results demonstrate that the size of this angle has no significant impact on the course of the recoil nor on dissipation of the recoil energy. Therefore, in weapon design practice, there is no justification for implementing constructional solutions in the butt enabling adjustment of the MR damper deviation axis from the runner axis.
Keywords: mechanics, firearms, recoil, magneto-rheological damper

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1. INTRODUCTION

The phenomenon of recoil in firearms has been extensively covered in the reference literature. Theoretical analyses of the phenomenon can, for example, be found in monographs by Rausenberger [2], Sieriebriakov [3], Orlov (ed.) [4], Germershausen (ed.) [5], Longdon (ed.) [6], as well as Wilniewczyc [7] or Carlucci and Jacobson [8].

In recent years, many works have been published on the issue of the interactions between firearm and shooter, such as those by Zakharenkov et al. [9], Burns [10], Lee and Choi [11], as well as Suchocki and Ewertowski [12]. Noteworthy are also works related to the application of magneto-rheological (MR) dampers for dissipating the recoil energy in firearms. One of the leading centres in this field of research is the Institute of Mechanics and Printing Technology at the Warsaw University of Technology [1, 13, 14, 15].

The problem of dissipating the recoil energy is particularly significant in the case of firearms characterised by large recoil energy, such as those exceeding 20 J, which includes grenade launchers, high calibre sniper rifles, anti-tank rifles, and smoothbore hunting weapons. The unfavourable impact of the firearm on the shooter occurring at that time fundamentally reduces the functionality of the firearm-shooter system, including accuracy and rate of fire. The main reasons include the unfavourable physical reactions and the psychological sensations of the shooter. In order to limit or eliminate the impact of these factors on the shooter, various types of recoil absorbers and muzzle brakes are installed, and particular rules and limitations are introduced in the firearm operation manuals.

One suggestion to improve the process of dissipation of part of the recoil energy was to introduce an option enabling adjustment of the MR damper deviation axis from the runner axis in the butt of a smoothbore hunting weapon, calibre 12/70, as illustrated in Fig. 1.

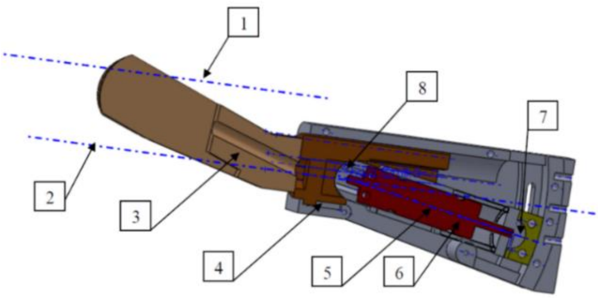

Fig. 1. Constructional solution enabling adjustment of the MR damper deviation axis from the runner axis in the butt of a 12/70 shotgun; 1 – barrel axis; 2 – runner axis; 3 – butt part with boxlock action; 4 – butt part with plate; 5 – MR damper axis; 6 – MR damper; 7, 8 – mounting points for MR damper

2. DAMPING SYSTEM MODEL

The mechanical system involving a butt with a damper, as shown in Fig. 1, can be analysed in the first approximation as an absorber, the kinematic diagram of which is presented in Fig. 2.

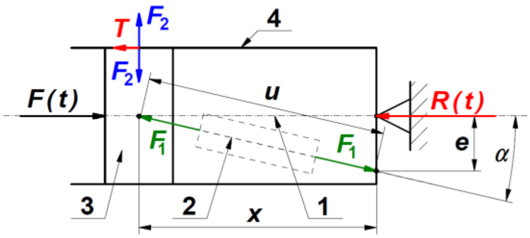

Fig. 2. Recoil absorber kinematic diagram, MR damper axis deviated; 1 – runner axis; 2 – MR damper axis; 3 – recoil absorber piston; 4 – recoil absorber casing; $F(t)$, $R(t)$ – applied force and reaction, acting on the absorber; F_1, F_2 – internal forces of interacting absorber parts; T – friction force; x – absorber piston displacement; u – MR damper piston rod displacement; e – distance of MR damper mounting point from absorber axis; α – MR damper axis deviation angle from the runners axis in the butt

Force $F(t)$ is an active force acting on the absorber, the module of which is equal to the response of the absorber to this input. Forces F_1 and F_2 are internal forces, where force F_2 generates friction force T. Furthermore, the reaction force R(t) appears in the support. A static analysis of this mechanical system based on the equation of equilibrium of forces allows the following relations to be determined:

$$F(t) - T = F_1 \cos\alpha \tag{1}$$

$$F_2 = F_1 \sin\alpha \tag{2}$$

$$T = \mu F_2 = \mu F_1 \sin\alpha \tag{3}$$

$$F(t) = F_1(\cos\alpha + \mu \sin\alpha) \tag{4}$$

with:

α – MR damper axis deviation angle from the runner axis in the butt,
μ – coefficient of friction.

It can be assumed that an MR damper is a Voigt element. Therefore, force F_1 will be a function of transfer u and velocity of displacement $\dot{u}$ of the MR damper piston rod:

$$F_1 = k_a u + c_a \dot{u} \tag{5}$$

with:

k_a – spring stiffness coefficient,
c_a – viscous damping coefficient.

Displacements of absorber piston x and damper piston rod u are dependent values, and result from the equation of the geometric system constraints:

$$x^2 + e^2 - u^2 = 0 \tag{6}$$

or

$$u = \sqrt{x^2 + e^2} = \frac{x}{\cos\alpha} \tag{6a}$$

with e – distance between MR damper mounting point and recoil absorber axis. Differentiation of this equation with respect to time t gives:

$$\dot{u} = \dot{x}\frac{x}{u} = \dot{x}\cos\alpha \tag{7}$$

After substituting (5), (6a) and (7) in (4), this gives:

$$F(t) = \left(k_a \frac{x}{\cos\alpha} + c_a \dot{x}\cos\alpha\right) \cdot (\cos\alpha + \mu\sin\alpha) \tag{8}$$

In practice, angle α does not exceed 20° (sin 20° = 0.3420), whereas the friction coefficient amounts to approximately 0.1. The product of these values is only 0.0342, or 3.6% of the value of cos 20° = 0.9397. Omission of the second component in the second bracket of (8) gives:

$$F(t) \cong k_a x + c_a \dot{x}\cos^2\alpha \tag{9}$$

Analysis of relations (8) and (9) allows the conclusion that the deviation of the MR damper axis from the runner axis in the butt by angle α exerts a minor influence on the value of force $F(t)$, and therefore on the response of the absorber to this input. The average changes in angle α vs. recoil absorber displacement x found in practice are presented in Fig. 3.

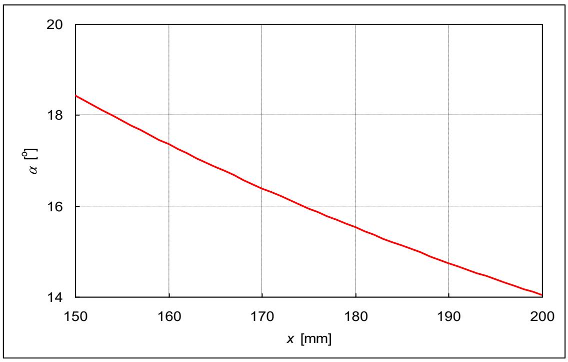

Fig. 3. Angle α vs. recoil absorber displacement x, for the distance between MR damper mounting point and shock absorber axis e = 50 mm

This trend was observed for the distance between the MR mounting point and the recoil absorber axis e = 50 mm and the initial distance of the mounting points for the damper calculated along the axis of the recoil absorber, which amounted to 200 mm (Fig. 2).

Assuming a maximum angle α = 20°, we can expect a decrease in the value of the recoil absorber reaction force to force F(t), which will only be observed in the damping element (a decrease by approximately 12%). At the same time, angle α will have a negligible influence on the operation of the elastic element of the absorber.

3. MODEL OF THE SHOOTER-FIREARM SYSTEM

To describe the shooter-firearm mechanical system, the modified model described in [1] was used, based on the model proposed by Hutchings and Rahe [16]. This model omits the influence of the conscious force reactions of the shooter to the inputs and assumes that this is sufficient for the initial evaluation of the operation of a recoil absorber for shooting from a high standing position [17] (Fig. 4).

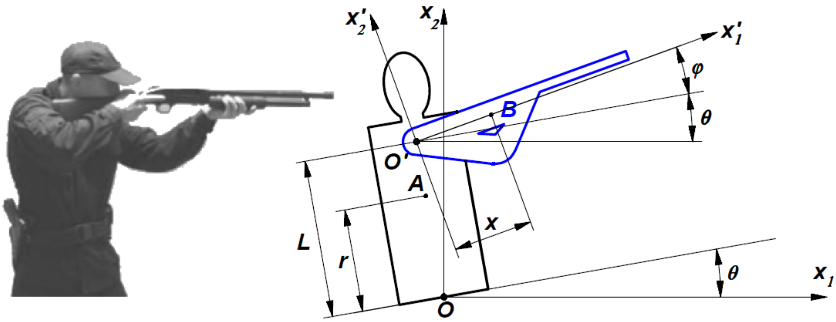

Fig. 4. Shooter-firearm mechanical system diagram; A, B – centres of mass of rigid bodies A and B (shooter's picture taken from [17]); O – fixed joint, connecting body A with the base; O' – movable joint, connecting body A with body B; θ – rotation angle of body A, in the inertial reference system $Ox_1x_2x_3$; φ – rotation angle of body B with respect to body A; $O'x'_1x'_2x'_3$ – movable reference system, connected to body B; r, L – distances of the centre of mass of body A and of joint O_1 from joint O

This mechanical system involves two rigid bodies. Body A includes the trunk, neck and head of the shooter, and this can only rotate around the stationary joint at point O, which is the origin of the inertial reference system $Ox_1x_2x_3$. The generalised coordinate fully describing the movement of this body is angle θ.

Body *B* consists of both arms of the shooter (arms, forearms and hands) plus the firearm. This body is connected to body *A* by means of a joint at *O'*, which is the origin of the moving reference system $O'x'_1x'_2x'_3$. In this system, body *B* can only move along axis Ox'_1, which is described by generalised coordinate *x*. The reference system $O'x'_1x'_2x'_3$ can only rotate with respect to body *A*. This movement is described by generalised coefficient φ. The interaction of bodies *A* and *B* at joint *O'* and the interaction of body *A* with the lower part of the body of the shooter at joint *O* were included by means of introducing Voigt elements. Furthermore, another Voigt element was included at joint *O'*, mapping the interaction of the shooter's shoulder and the weapon in the translatory motion described by generalised coefficient *x*.

The modification to the described model consisted of the serial association of an additional Voigt element, which described the influence of the recoil absorber in the direction of axis $O'x'_1$, as presented in Fig. 5. At the same time, the MR damper axis deviation from the recoil absorber axis was taken into account, as presented in Fig. 2.

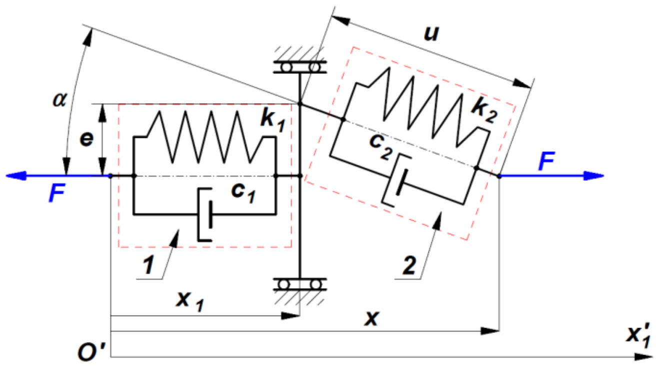

Fig. 5. Diagram of the modified model of the shooter's shoulder and weapon interaction in translatory motion, described by generalized coordinate *x*. 1 – Voigt element existing in original model; 2 – additional Voigt element taking into account the effect of the recoil absorber; k_1, k_2 – spring stiffness coefficients; c_1, c_2 – viscous damping coefficients; *e* – distance between MR damper mounting point and recoil absorber axis; *u* – MR damper piston rod displacement; α – MR damper axis deviation angle from the runners axis in the butt; *F* – force acting upon the system

Due to the low mass of the recoil absorber in relation to the masses of bodies *A* and *B*, its influence on the movement of the system was taken into account by adding it to body *B*.

The serial connection of the Voigt elements is characterised by the fact that, for each element, an equal force *F* is present, and the sum of the arrows of deflection for a particular element is equal to the arrow of deflection of the entire set.

Neglecting friction force T (Fig. 2) and taking into account relations (6) and (9), we can express:

$$F = k_1(x_1 - x_{10}) + c_1\dot{x}_1 = k_2\left[1 - \sqrt{\frac{(x_0 - x_{10})^2 + e^2}{(x - x_1)^2 + e^2}}\right](x - x_1) + c_2\frac{(x - x_1)^2}{(x - x_1)^2 + e^2}(\dot{x} - \dot{x}_1) \tag{10}$$

where:

k_1, k_2 – spring stiffness coefficients,
c_1, c_2 – viscous damping coefficients,
x_0, x_{10} – initial locations of Voigt elements.

Relation (10) is in fact an equation of the kinematic constraints of the analysed mechanical system. Taking (10) into account in the model by Hutchings and Rahe leads to the following system of ordinary differential equations describing the dynamics of the mechanical system: shooter – recoil absorber – firearm:

$$\frac{dx_1}{dt} = \frac{1}{A}\left[Bk_2(x - x_1) - k_1(x_1 - x_{10}) + c_2C\dot{x}\right] \tag{11}$$

$$\frac{d\dot{x}}{dt} = \frac{1}{D}\left(b_{11}f_x + b_{12}f_\theta + b_{13}f_\varphi\right) \tag{12}$$

$$\frac{d\dot{\theta}}{dt} = \frac{1}{D}\left(b_{12}f_x + b_{22}f_\theta + b_{23}f_\varphi\right) \tag{13}$$

$$\frac{d\dot{\varphi}}{dt} = \frac{1}{D}\left(b_{13}f_x + b_{23}f_\theta + b_{33}f_\varphi\right) \tag{14}$$

$$\frac{dx}{dt} = \dot{x} \tag{15}$$

$$\frac{d\theta}{dt} = \dot{\theta} \tag{16}$$

$$\frac{d\varphi}{dt} = \dot{\varphi} \tag{17}$$

with:

$$k_1 = \begin{cases} k_{xw} & for \quad x_1 \geq x_s \\ k_{xs} & for \quad x_1 < x_s \end{cases} \tag{18}$$

$$A = (c_1 + c_2C)^{-1} \tag{19}$$

$$B = 1 - \sqrt{\frac{(x_0 - x_{10})^2 + e^2}{(x - x_1)^2 + e^2}} \tag{20}$$

$$C = \frac{(x - x_1)^2}{(x - x_1)^2 + e^2} \tag{21}$$

$$D = m_B \left(I_A + m_A r^2\right)\left(I_B + m_B x^2\right) + I_B m_B^2 L^2 \sin^2 \varphi \tag{22}$$

$$b_{11} = \left(I_A + m_A r^2 + m_B L^2\right) I_B + m_B x^2 \left(I_A + m_A r^2 + m_B L^2 \cos^2 \varphi\right) \tag{23}$$

$$b_{12} = \left(I_B + m_B x^2\right) m_B L \cos\varphi \tag{24}$$

$$b_{13} = -\left[I_B + m_B (x + L \sin\varphi) x\right] m_B L \cos\varphi \tag{25}$$

$$b_{22} = m_B \left(I_B + m_B x^2\right) \tag{26}$$

$$b_{23} = -m_B \left[I_B + m_B x (x + L \sin\varphi)\right] \tag{27}$$

$$b_{33} = m_B \left[I_A + m_A r^2 + I_B m_B (x + L \sin\varphi)^2\right] \tag{28}$$

$$\begin{aligned} f_x = & -F_z(t) - R_p + m_B x(\dot{\theta} + \dot{\varphi})^2 + m_B \dot{\theta}^2 L \sin\varphi + \\ & - m_B g\left[\sin(\theta + \varphi) - \sin(\theta_0 + \varphi_0)\right] \end{aligned} \tag{29}$$

$$\begin{aligned} f_\theta = & \, L R_p \cos\varphi + M_p - 2 m_B \dot{x}(\dot{\theta} + \dot{\varphi})(x + L \sin\varphi) + \\ & - m_B L x \dot{\varphi}(2\dot{\theta} + \dot{\varphi}) \cos\varphi - c_\theta \dot{\theta} - k_\theta (\theta - \theta_0) + \\ & + (m_A r + m_B L) g (\sin\theta - \sin\theta_0) + \\ & - m_B g x\left[\cos(\theta + \varphi) - \cos(\theta_0 - \varphi_0)\right] \end{aligned} \tag{30}$$

$$\begin{aligned} f_\varphi = & \, \delta F(t) - 2 m_B x \dot{x}(\dot{\theta} + \dot{\varphi}) + m_B L x \dot{\theta}^2 \cos\varphi + \\ & - M_p - m_B g x\left[\cos(\theta + \varphi) - \cos(\theta_0 + \varphi_0)\right] \end{aligned} \tag{31}$$

$$F_z(t) = \begin{cases} 0 & \textit{for} \quad t > t_b \\ \dfrac{F_A}{2}\left[1 - \cos(\omega t)\right] & \textit{for} \quad t \le t_b \end{cases} \tag{32}$$

$$F_A = p_{mz} s; \quad t_b = \frac{2 J_F}{F_A}; \quad \omega = \frac{2\pi}{t_b} \tag{33}$$

$$R_p = \begin{cases} k_{xw}(x_1 - x_{10}) + c_1\dot{x}_1 & \text{for} \quad x_1 \geq x_s \\ k_{xs}(x_1 - x_s) + k_{xw}(x_s - x_{10}) & \text{for} \quad x_1 < x_s \end{cases} \tag{34}$$

$$M_p = k_\varphi(\varphi - \varphi_0) + c_\varphi\dot{\varphi} \tag{35}$$

m_A, m_B, I_A, I_B – masses and main central moments of inertia of bodies *A* and *B,*
k_{xw}, k_{xs}, c_1 – spring stiffness coefficients and viscous damping coefficients of the Voigt element, describing the interaction of the shooter's shoulder in the direction of coordinate x_1,
x_s – coordinate characterising the change in spring stiffness coefficient, which results from the characteristics of the shooter's shoulder and clothes,
k_θ, k_φ, c_θ, c_φ – spring stiffness coefficients and viscous damping coefficients of the Voigt elements, describing the interaction of the shooter at joints *O* and *O'* (Fig. 1),
k_2, c_2 – spring stiffness coefficient and viscous damping coefficient of the Voigt element describing the interaction of the recoil absorber,
r, *L* – distances from the centre of mass of body *A* and of joint O_1 from joint *O,*
g – gravitational acceleration,
p_{mz} – maximum pressure acting on the breech block,
s – cross-sectional barrel surface,
δ – distance from the barrel axis to axis $O'x'_1$ (Fig. 4),
J_F – impulse of input force during free recoil of the firearm,
F_A – maximum exciting force,
t_b – ballistic time,
x_{10}, x_0, θ_0, φ_0 – particular initial values, which also ensure the static equilibrium of the system for $t = 0$ and:
$F(t) = 0$ $\dot{x}_1 = 0; \quad \dot{x} = 0; \quad \dot{\theta} = 0; \quad \dot{\varphi} = 0.$

For the presented model it was additionally assumed that the exciting force $F_z(t)$ could be approximated by means of the harmonic function proposed by Boutteville [5]. Constants F_A and t_b in formula (32) can be calculated from formulas (33), as proposed in [13].

4. NUMERICAL CALCULATION RESULTS

Computer simulation of the movement of the mechanical system: shooter – recoil absorber – firearm was carried out with the MATLAB system by MathWorks, Inc. for IBM PC microcomputers. Library function ode45 was also used for this purpose. The simulation was run for cases of a single shot fired by a shooter having a total body mass of 78 kg and height of 185 cm, from a smoothbore hunting weapon, calibre 12/70.

In each case, two system variants were analysed, i.e. with an absorber: without deviation of the MR damper axis (e = 0) and with deviation of the MR damper axis (e = 50 mm). The numerical data characterising the shooter were taken from [16], whereas those for the hunting weapon, calibre 12/70, from [18, 19, 10].

At the same time, it was assumed that bodies A and B were characterised by the same masses, moments of inertia and distances: r, L, x_0. The numerical data used in the calculations are listed in Table 1.

Table 1. Numerical data used during the computer simulation

Value	Designation	Units	Value
Mass of body A	m_A	kg	44.15
Moment of inertia	I_A	kg·m^2	2.3304
Mass of body B	m_B	kg	14.97
Moment of inertia	I_B	kg·m^2	1.2428
Distance	L	mm	533.4
Distance	r	mm	355.6
Distance	x_0	mm	241.3
Distance	$x_0 - x_s$	mm	1.0
Distance	$x_0 - x_{10}$	mm	50
Distance	$x_{10} - x_s$	mm	1.0
Distance	δ	mm	32.0
Maximum exciting force	F_A	N	13452.46
Ballistic time	t_b	ms	2.426147
Gravitational acceleration	g	m/s^2	9.81
Spring stiffness coefficient	k_θ	N·m/rd	23.5
Spring stiffness coefficient	k_φ	N·m/rd	23.5
Spring stiffness coefficient	k_{xw}	N/m	14610.0
Spring stiffness coefficient	k_{xs}	N/m	43830.0
Spring stiffness coefficient	k_2	N/m	14000.0
Damping coeff.	c_θ	N·m·s/rd	100.0
Damping coeff.	c_φ	N·m·s/rd	30.0
Damping coeff.	c_1	N·s/m	1000.0
Damping coeff.	c_2	N·s/m	1600.0
Displacement	e	mm	50

The simulations were run in the time interval from 0 to 0.45 s, in accordance with the data provided in the references [20], namely that the delay in the appearance of a conscious reaction in humans to external stimuli ranges from 0.1 s to 0.45 s. Furthermore, the following initial conditions were assumed, for

$$t = \dot{x}_1 = \dot{x} = \dot{\theta} = \dot{\varphi} = \theta = \varphi = 0; \quad x_1 = x_{10}; \quad x = x_0 .$$

The following figures present the results of the computer simulation, for the listed numerical data. These are the trends in firearm displacement differences described by coordinate x and its tilt angle ($\beta = \theta + \varphi$), as well as the impact force of the firearm on the shooter's shoulder $R(t)$, for the analysed cases. Differences $\Delta x(t)$, $\Delta\beta(t)$, $\Delta R(t)$ were calculated by means of subtracting the corresponding results for the case e = 50 mm, from the results for the case e = 0 mm.

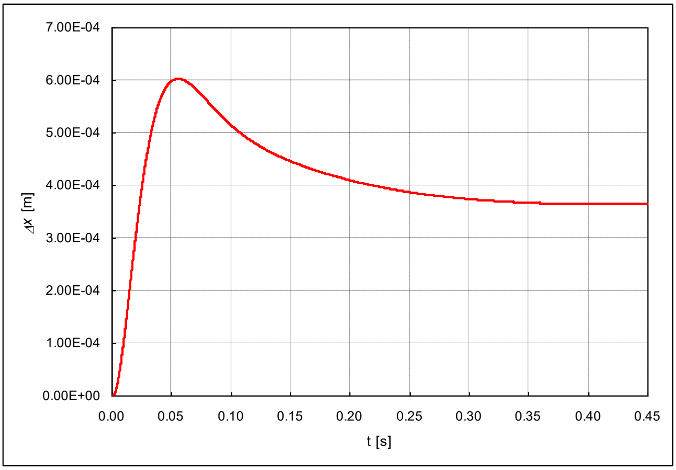

Fig. 6. Displacement differences $\Delta x(t)$ for body B

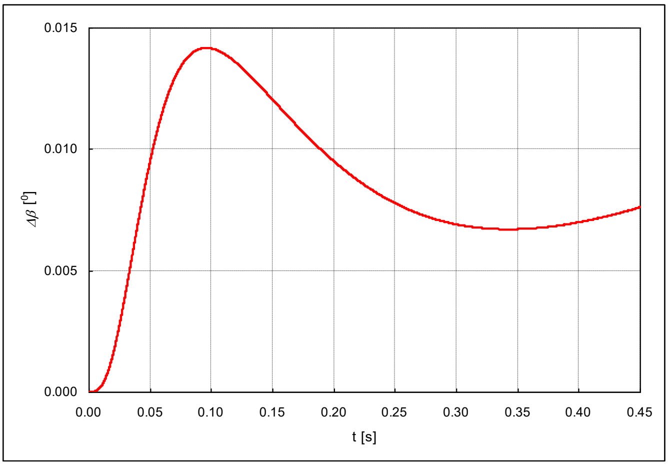

Fig. 7. Firearm pitch differences $\Delta\beta\,(t)$

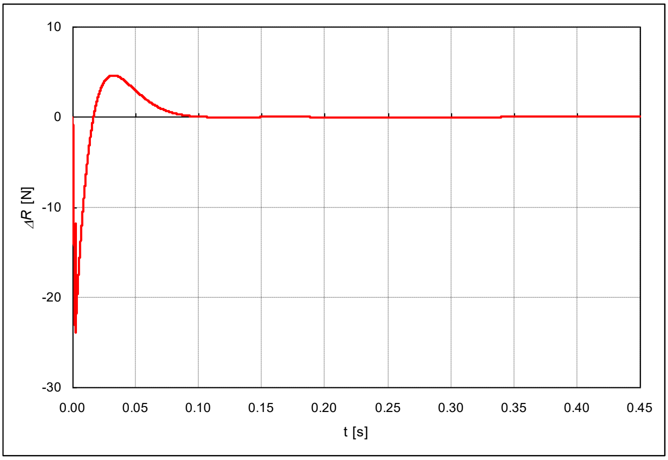

Fig. 8. Shooter's arm reaction force differences $\Delta R(t)$

5. SUMMARY AND CONCLUSIONS

The presented method of modelling the influence of a butt-mounted magneto-rheological (MR) damper on the shooter while firing a shot enabled the quantitative and qualitative evaluation of the deviation axis from the runner axis in the butt on the trend of the recoil, especially the movement of the mechanical system and the reaction force of the shooter's shoulder during a single shot. The results indicate that the size of this angle does not have a significant impact on the course of the recoil or the dissipation of the recoil energy. For both analysed cases the observed differences in the firearm position did not exceed the order of a tenth of a millimetre or a hundredth of a degree, whereas the difference in the shooter's shoulder reaction force did not exceed approximately 35 N.

It can therefore be concluded that in weapon design practice there is no justification for implementing constructional solutions in the butt enabling adjustment of the MR damper axis from the runner axis.

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Analiza teoretyczna wpływu kąta odchylenia osi tłumika magnetoreologicznego od osi prowadnic w kolbie na jego działanie podczas strzału

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Streszczenie. W pracy przedstawiono sposób modelowania oddziaływania na broń i strzelca tłumika magnetoreologicznego (MR) zamocowanego w kolbie, podczas strzału. W tym celu zaproponowano modyfikację modelu przedstawionego w pracy [1]. Badania skoncentrowano na określeniu wpływu kąta odchylenia osi tłumika MR od osi prowadnic w kolbie na przebieg zjawiska odrzutu, a szczególnie na energię dyssypowaną podczas strzału. Zamieszczono przykładowe wyniki analizy dla gładkolufowej broni myśliwskiej kal. 12/70. Otrzymane wyniki pozwoliły stwierdzić, że wielkość tego kąta nie ma istotnego wpływu na przebieg zjawiska odrzutu i rozpraszanie energii odrzutu. W praktyce projektowania broni palnej nie ma zatem uzasadnienia wprowadzanie rozwiązań konstrukcyjnych umożliwiających regulowanie kąta odchylenia osi tłumika MR od osi prowadnic w kolbie.

Słowa kluczowe: mechanika, broń palna, odrzut broni, tłumik magnetoreologiczny