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Design Performances of Linked Wire Rope Absorbers in the Chain of Simple Gun Recoil Device[*]

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Abstract. One of the basic requirements for the artillery weapons is the dumping of dynamical impulse loadings by the recoil system. Recoil subsystems, which are displaced during the recoiling process, have to be stopped and returned to the initial position, keeping the stability of weapon's position. Displacement position control as a consequence of the recoiling process is an indirect performance of the recoil system design. These performances are changing vs. time gradients more or less rigid, regarding forces or displacements making recoil characteristics more or less elastic. Guides by forces in hydraulic brakes of recoil system are proportional to the recoil velocities, which are not always applicable for full control displacements profile on the weapon. Paper considered implementation of stiffness proportional linear or nonlinear to the displacement as possible variator of maximum force and their position in time for joined recoil system. As possible solution wire rope absorbers have been considered as redundant links in hydraulic brakes serial junctions. The appropriate position and variable stiffness of redundant have been considered by Bouc-Wen modified approximation redesigned for artillery gun loading profiles. Paper integrates experimental laboratory dynamic testing data with simulation software of gun recoil systems regarding the howitzer 152 mm.
Keywords: mechanics, armament, recoil gun system

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1. INTRODUCTION

Processes of the projectile launching in artillery weapon causes unsteady state behavior of forces and related displacements, velocities, accelerations, etc. reflected on the weapon recoiling masses. As the complex mechanical system relied on the ground, recoil system exposes high intensity and short duration shock loadings. The study of displacement requires well determined models to predict movement and constrain loads during recoil and return recoiling mass, and their affects on the stability and immobility of weapon in the firing process. The important aspects of analysis are structurally connected with one of the most important mechanical performance of any designed structure well-known as the stiffness, which orientated description of system behavior during perturbed motion.

For description of the artillery weapons behavior during the firing process several models can be used. Some of them are:

- simplified mechanical model with one degree of freedom (recoil of barrel with additional masses),
- simplified mechanical model with two degrees of freedom,
- complex mechanical models with more degrees of freedom, etc.

Most of them require complex numerical analyses mainly because of nonlinearities in unsteady state motions, but not because of complex links which are usually formed in one direction of orientated frame. This fact makes analyses of DOF linked serial by simplified algebraic expressions in place of complex systems designed with more than one DOF referred from three axis translations and rotations. Different methods for solutions could be used as it:

- finite element method (concentrated masses and elastic beams) and
- commercial software packages for simulation of complex mechanical systems.

Bearing in mind that the forming of an experimental model for real tests (prototype) is expensive and time consuming, mechanical and mathematical models are the best tools in optimizing the basic performance of a weapon in the initial phases of development. Simulation models are especially interesting in the case of the modification of existing artillery systems, in order to predict their behavior in variable conditions of exploitation, whether it was required for improvements in a ballistic system, in the integration of new and redesigned of existing components or systems in general [1]. Especially simulation models is welcome when some experimental tests or particular components intent to be employed on the system as it recoiling mass. Recognized features of stiffness and shock absorbing as key problem considered for the heavy gun artillery orientated research in this paper to use combination of experimental and simulation tests as the first approach in further research.

For that purposes the simplified model of mechanical oscillator has been taken as an approximation for the recoil system and influences prepared by well experimental tests of particular components have been included in the model to vary stiffness behavior as well as the most important. Schematic view of the classic artillery weapon is shown in Figure 1.

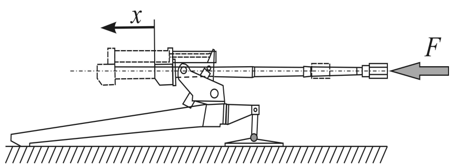

Fig. 1. Schematic view of the artillery weapon

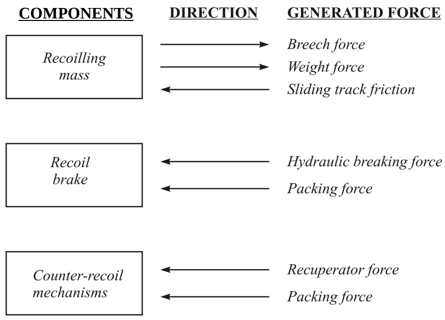

Fig. 2. Components of total resistance force of recoil

During the firing process on the weapon acts short impulse force F (Figure 1) that causes recoil of recoiling mass in the x direction. Motion in the mentioned direction corresponds to appropriate length of recoil, which is opposed by the total resistance force. Designed components, which make mentioned force, are given in Figure 2.

2. THE ANALYTICAL MODEL

Simplified model understood observing the total force of resistance R caused only by one appropriate equivalent resistant component represented by dumping coefficient equivalent b_e, depends on the recoil velocity. In that case the basic model of artillery weapon can be considered as a system with one degree of freedom Figure 3.

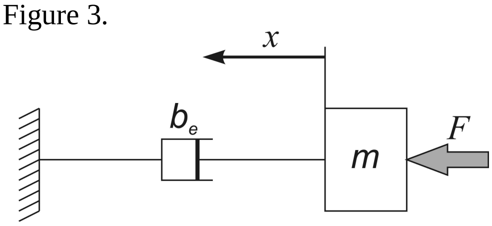

Fig. 3. Mechanical model of the system with one degree of freedom

Adopting the scheme shown in Figure 3, differential equations of barrel motion has the form:

$$m\ddot{x}_A + b_e\dot{x}_A = F \tag{1}$$

After very short strong interior ballistic force of propellant gas pressure, the system can be considered as free damped oscillator, expressed by:

$$m\ddot{x}_A = -b_e\dot{x}_A \tag{2}$$

The main problem of this system is not approximation of the steady state inertial force (2), but its gradient, known as jerk whose value definitely is determined by equivalent stiffness of the system linking influence with flowing velocity. This causes also stability of weapon immediately after firing and during recoil process along design distance on the weapon. This value, respecting (2), is given in the form:

$$\dddot{x}_A = -\frac{1}{m}\left(\dot{b}_e - \frac{b_e^{\,2}}{m}\right)\dot{x}_A \tag{3}$$

Designed features of recoil system on the existing weapon assemblies provide enough method to control b_e by static tests in the maintaining handling and preparing phases of the weapon. These methods can not vary performances of recoil brakes and recuperators adopted for the gun forces out of design interior ballistic. Usually ammunition prevails ages of guns and new interior ballistics requirements change necessities of designed recoiling stiffness. This set up new tasks containing in application of absorption stiffness variators which characteristics may or may not, correspond to a linear model. Combining management of the inertia force gradient – jerk and velocity by suitable choice of variator structural properties problem of the jerk control should be sold by new additional absorbers. Further study will be presented schematic model of additional stiffness integrated as common with available recoil system as new damping element.

Potential locations of damping elements mounting in the simplified designed hydraulic brake recoil system integrated with barrel Figure 4 are:

- between the barrel and the cradle of artillery weapon,
- between barrel and moving parts of recoil system and
- inside the recoil device.

The proposed mechanical model can be represented by the stiffness scheme in Figure 5. In the new concept of improved system stiffnesses old one is presented in the part of Figure 5 linked as redundant subassembly which was not obvious from Figure 4.

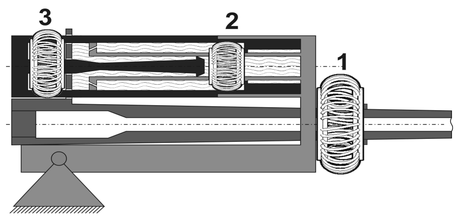

Fig. 4. Potential locations of damping elements

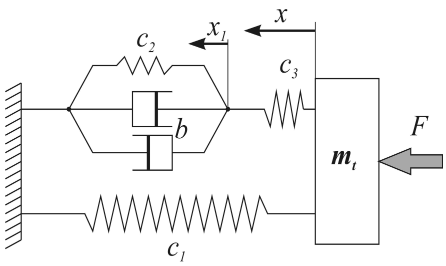

Fig. 5. The proposed mechanical model

Resistance forces will have to follow equations of motion regarding Figure 5 and in that sense of meaning elements 1, 2 and 3 became stiffness variators. Their links became redundant and serial transformers of old stiffness systems.

Absorbers c_1 and c_3 are differently loaded by impulse initial forces in place of fact that are initiated by same displacements. These differences mean additional degree of freedom x_1 expressed as linked by stiffness c_3 with main motion x.

This provides to consider main equation of motion, referred as case B, as a linked form of two displacements as:

$$m\ddot{x}_B - c_3\left(x_B - x_1\right) + c_1 x_B = 0 \tag{4}$$

Balances of resistance forces in recoil system, linking two motions x_1 and x_B by expression:

$$c_3\left(x_B - x_1\right) - b_e\dot{x}_1 - c_2 x_1 = 0\,, \tag{5}$$

which is additional condition for the solution of (4).

Values x_1 and $\dot{x}_1$ are analogues with x_A and $\dot{x}_A$ of old system, but resolved by new stiffnesses linked in model by Figure 5. Expression for jerk as gradient of inertial force within the new modified stiffnesses is derived from (4) in the form:

$$\dddot{x}_B = \frac{c_3}{m}\dot{x}_1\left(1 - \frac{c_1 + c_3}{c_3}\frac{\dot{x}_B}{\dot{x}_1}\right) \tag{6}$$

Corresponding dumping forces for the new and old system with and without additional stiffness variators in the same braking equipment are correlated by velocities $\dot{x}_1$ and $\dot{x}_A$.

These expressions are:

$$F_{bB} = b_e \dot{x}_1 \tag{7}$$

$$F_{bA} = b_e \dot{x}_A \tag{8}$$

The hypothesis about the behavior can be adopted by brake cylinder properties, which operate in different displacements conditions. By ratio of velocities, regarding resistance internal forces correlation over pressures next expressions are given in the form:

$$p_B A = b_e \dot{x}_1 \tag{9}$$

$$p_A A = b_e \dot{x}_A \tag{10}$$

This promotes relations of velocities as proportional by cylinder pressures increasing coefficient *k* as:

$$\dot{x}_1 = k\dot{x}_A = \frac{p_B}{p_A}\dot{x}_A \tag{11}$$

The new jerk can be expressed as relation of stiffness characteristics of external elements to the basic hydraulic value limited by design of the brake system *k* as:

$$\dddot{x}_B = \frac{c_3}{m}k\dot{x}_A - \frac{c_1 + c_3}{m}\dot{x}_B \tag{12}$$

The jerk expression is function of two jerk parts, one modified by old velocities $\dot{x}_A$ with serial stiffness link c_3 and one modified by new velocities $\dot{x}_B$ with redundant stiffness links c_1 and c_3.
General jerk equation is:

$$\dddot{x}_B = \frac{c_3}{m}k\dot{x}_A\left(1 - \frac{c_1 + c_3}{c_3}\frac{\dot{x}_B}{k\dot{x}_A}\right) \tag{13}$$

The relationship between old and new recoil system obviously is dependable of stiffness and equivalent parameters of the hydraulic brake system in the form:

$$\frac{\dddot{x}_B}{\dddot{x}_A} = -\frac{mc_3k}{m\dot{b}_e - b_e^{\,2}}\left(1 - \frac{c_1 + c_3}{c_3}\frac{\dot{x}_B}{k\dot{x}_A}\right) \tag{14}$$

This determines a condition under which the stiffness variator coupled with pressures ratio in brake cylinder determines the value of jerk compared to a system that does not have the mentioned elements. Stiffness conditions related to the brake performances b_e and recoiling mass m by expression:

$$c_3 k - (c_1 + c_3)\frac{\dot{x}_B}{\dot{x}_A} < \frac{b_e^{\,2} - m\dot{b}_e}{m}. \tag{15}$$

3. EXPERIMENTAL TESTING OF NONLINEAR ABSORBERS

The applicant for the stiffness variations as element inserted in the system this paper considers so-called ring wire rope absorber determining referred by properties in the papers [3, 4, 5, 8, 10].

Based on referred papers, as the first consequence of assumed absorber, the behavior of system is expected to be nonlinear. Stiffness referred [4] shows anomalies during shock loading conditions important for this research. These experiments were tested former years in the laboratories conditions and shows well reproducing of results, making tested data high reliable as results to be taken in simulation model.

Tested model accepted as variator in this paper is shown in Figure 6 and curves in Figure 7 represent sample of dynamical response by force of ring wire rope absorbers.

Fig. 6. Ring wire rope absorber

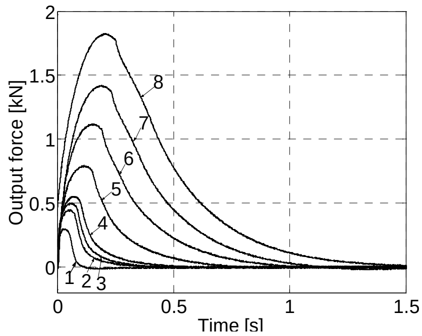

Fig. 7. Dynamical response of ring wire rope absorber

Experimental testing [4] for these special cases of ring form wire rope absorbers also shows nonlinear expression of stiffness representatively shown in Figure 8, related to the absolute x and relative Δx displacements.

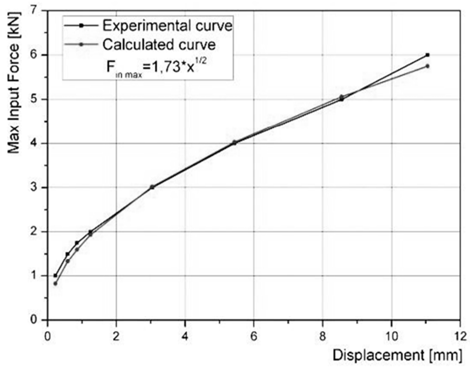

Fig. 8. Nonlinearities of experimental behavior

Corresponding to placing cases of absorbers in Figures 4 and 5, nonlinearities of experimental behavior is given by:

$$c_1 = \frac{k_1\sqrt{x}}{x}$$
$$c_3 = \frac{k_1\sqrt{\Delta x}}{\Delta x} \qquad (16).$$

Equation (16) is valid as experimental proved and could be replaced in the any of former basic or derived equations respecting, for small perturbations Δx, as argument for the stiffness c_3 This lead from expression (13) in the similar directly proportional linear form expression as:

$$\dddot{x}_B\left(c\left(x\right)\right)=\frac{c_3\left(\Delta x\right)}{2m}k\dot{x}_A\left(1-\frac{c_1\left(x\right)+c_3\left(\Delta x\right)}{c_3\left(\Delta x\right)}\frac{\dot{x}_B}{k\dot{x}_A}\right) \qquad (17)$$

Unsteady state (17) is not general jerk but valid for the special cases of ring form tested absorbers initiated by prompt shock forces with determined impulses as initial conditions of free dumped motion. Jerk ratio is also determined by this new quasi-linear form as:

$$\frac{\dddot{x}_B}{\ddot{x}_A}=-\frac{mkc_3\left(\Delta x\right)}{2\left(m\dot{b}\left(x,\dot{x}\right)-b^2\left(x,\dot{x}\right)\right)}\left(1-\frac{c_1\left(x\right)+c_3\left(\Delta x\right)}{c_3\left(\Delta x\right)}\frac{\dot{x}_B}{k\dot{x}_A}\right) \qquad (18)$$

This expression is used as capstone approval of behavior new stiffness variators linked in Figure 5.

To test further properties of absorbers links former tests also offers available reliable data [7, 8, 9, 10] about absorbers behavior. As stated in the literature [6], wire rope absorbers generally used for damping impact loads with small displacement amplitudes.

Greater displacements usually require solutions, which correspond to the other types of absorbers. Design properties of wire rope absorbers orientated their relative displacements on the values less than half of its diameters. Sequentional including of serial linking absorbers solved problem of displacements particularly, but some of the singularities in serial clutching for each of them in time appears as the constrains in the predictions as the continual mathematical treatment. It is clear that absorber due to its size would not be practical for use in larger displacement amplitudes, as it is required for gun barrel recoil systems.

Therefore, goal of the research referred in [8, 10] was to test the characteristics of absorbers stiffnesses combined in series and redundant links during static and control dynamic impulses loadings, regarding expected inclination of their linear composition stiffness behavior in the both cases.

First of all dynamical controlled conditions was tested (Figure 10) for the referencing one absorber by extending forces and displacements in the loading and relieving phases in the continual cycles. This was taken as the referent behavior of absorbers with ring wire ropes composition design regarding expected hysteretic anomaly, which appears in the phases of loading and relieving of continual cycles. Referencing result is given in Figure 10 for the designed frame of absorber in Figure 9.

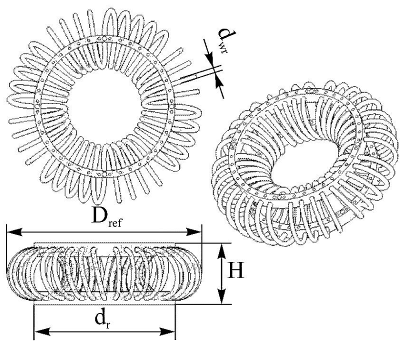

Fig. 9. Dimensions of experimental model

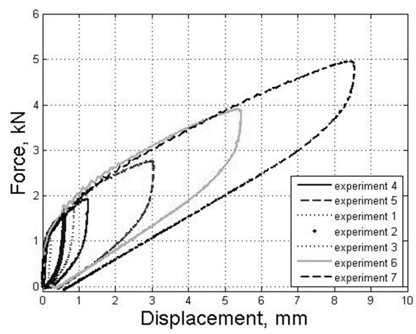

Fig. 10. Phenomenon of hysteresis

Figures show corrected progressive behavior of individual hysteretic closed frame curves for the single absorber loaded by constrained minimum and maximum dynamical forces.

Testing conditions was vary around 10-15 m /s, which was expected to be referent for the values of gun recoiling masses during first phase of motions in the launching conditions. This referencing experimental conditions referred curves accepted for the further experimental variations of stiffness required for the composition of gun recoil system.

Figure 11 shows the experimental research scheme for equivalent stiffness variation tests of linked absorbers. Two serial and redundant absorber links without intermediate switching distances shows referent testing for different linking inclinations regarding toward one absorber as new nonlinearities caused by serial or redundant links Figure 11 shows experimental scheme of laboratories tests.

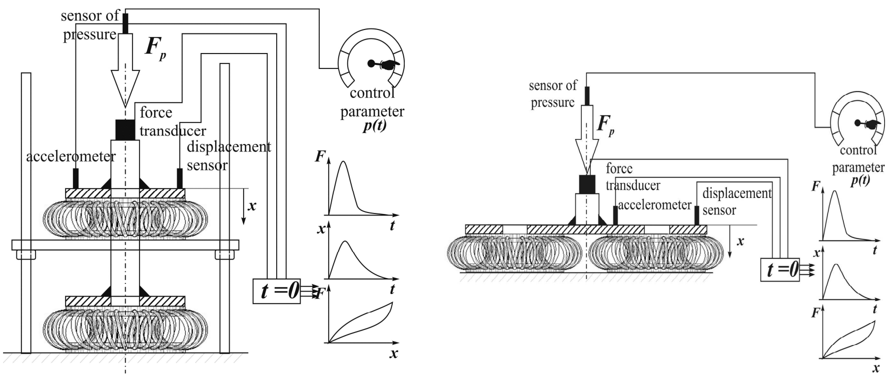

Fig. 11. Testing scheme of absorber serial and parallel connection

Based on the expressed difference in the loading and relaxation processes, from diagrams in Figure 10 indication of the phenomenon known as hysteresis is also obvious, for two serial and parallel linked absorbers Figure 13, as for single one [8].

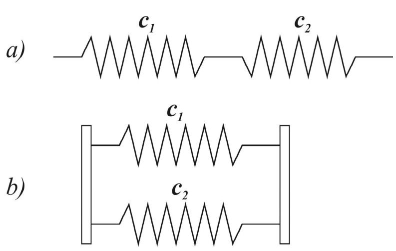

Fig. 12. Serial and parallel combination of springs

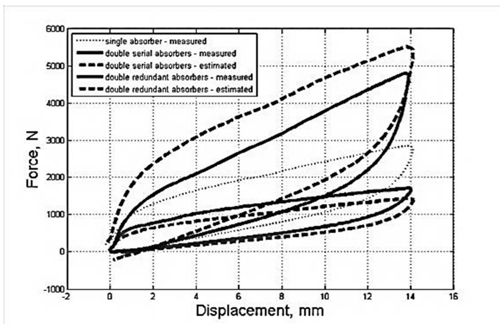

Fig. 13. Experimental obtained behavior for serial and parallel linked absorbers

Elastic force for serial F_{elser} and redundant F_{elpar} links of ring wire rope absorbers are composed according to Figure 12. Ideal cases of both links satisfy well-known relations of redundant stiffnesses sums and serial stiffnesses harmonic ratios, for equivalent stiffness of system. This ideal estimation is reflected also for hysteretic closed loop (dash curves in Figure 13) serial and redundant correlated with dot curve for the single absorber sample. Real behavior of mentioned links is presented in Figure 13 shows well correlation with ideal links proved by measured (solid line in Figure 13). Inclination is fixed determined for absorber design but less the 10 percents around corrected friction factor $n(d_{wr}) \approx 1.2$ equal for both links. Equivalent stiffnesses can be written as:

$$c_{eser} = \frac{1}{n(d_{wr})} \frac{c_1 \cdot c_2}{c_1 + c_2} \quad \text{for serial and} \tag{19}$$

$$c_{e\,par} = n(d_{wr}) \cdot (c_1 + c_2) \quad \text{for redundant link,} \tag{20}$$

Last twenty years one of the most used model, which explains hysteretic behavior is Bouc-Wen [9]. Model explained hysteretic behavior of internal forces regarding assumption taken in the form:

$$\dot{z}(t) = \dot{x}(t)\left\{\alpha - \left[\gamma + \beta sgn\left(\dot{x}(t)\right)\operatorname{sgn}\left(z(t)\right)\right]\left|z(t)\right|^N\right\} \tag{21}$$

as referencing relation between jerk of internal properties and velocity $\dot{x}$ as external disturbance. Coefficients α, β, γ, N referred importance of velocity and inertial force of internal processes.

Application of this model is referred on the assumption that variators linked redundant and serial in recoil system cause undetermined equivalent hysteretic behavior. By this logic, expression (21) becomes equal, and determined, with internal jerk differences caused by system with or without absorbers (3) and (13). This state is expressed by:

$$\dddot{x}_B - \dddot{x}_A = \dot{z}_e(t), \tag{22}$$

or, by replacing:

$$\frac{c_3(\Delta x)}{2m} k \left(1 - \frac{c_1(x) + c_3(\Delta x)}{c_3(\Delta x)} \frac{\dot{x}_B}{k\dot{x}_A}\right) + \frac{1}{m}\left(\dot{b}_e - \frac{b_e^{\,2}}{m}\right) = \alpha - \left[\gamma + \beta sgn\left(\dot{x}(t)\right)\operatorname{sgn}\left(z(t)\right)\right]\left|z(t)\right|^N \tag{23}$$

Assumption for (23) is that velocity, which causes hysteretic behavior, is not actually formed in the case of linked absorbers. This value has to be ideal for undisturbed conditions.

Continual hysteretic process velocity also could be taken in consideration, but requires iterative proceeding to evaluate right and left side of (22) and expressed by analytical solution (23) would not be valid. Ratio of velocities during hysteretic and free dumped motion of recoil system is taken as measured joint with stiffnesses *c* to refer inertial force of internal process *z(t)* expressed in (23).

4. SIMULATION EXAMPLE

- Simulation of recoil system is performed for the howitzer 152 mm for data [1] of equivalent dumping resistant (3) respecting b_e as full nonlinear coefficient. Calculation is realized for the maximal propellant charge and zero elevation angles.
- Stiffness c_2 is taken as variator of recuperator gas pressure and is not exposed in hysteretic manner. The role of his redundant link was only to vary influence of resistance force caused by initial gas pressure.
- Short distances of initial recoil are observed as reference for stiffness variation estimations relevancy for effects of jerk and initial resistance impulse.
- Real forced motion is taken for further free motion of recoil system and barrel for evaluation of unsteady state b_e from simulation experiment.

Two types of simulation tests are realized in paper. One of the so-called case A (Figure 3) and case B preferred as joint variation stiffness model. Results were observed for the initial 0.4 m of recoil. Forced motion simplified observed corresponds to the part of 0.1 m.

Results of system behavior in time for full recoil length in time, in the case A, is shown in Figure 14. Data of dumping coefficient and velocities have taken from this simulation on the observed part of recoil path are shown in Figures 15 and 16.

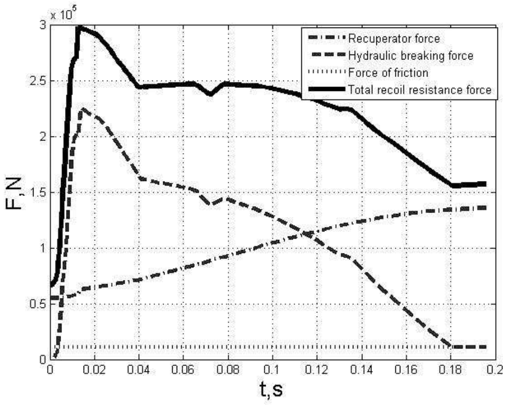

Fig. 14. Time change of total resistance recoil force

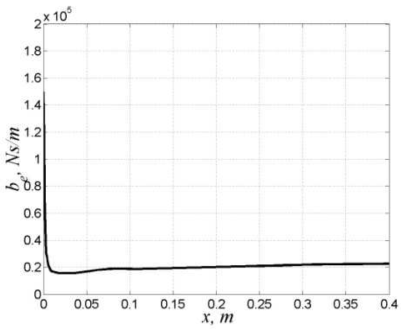

Fig. 15. Equivalent dumping

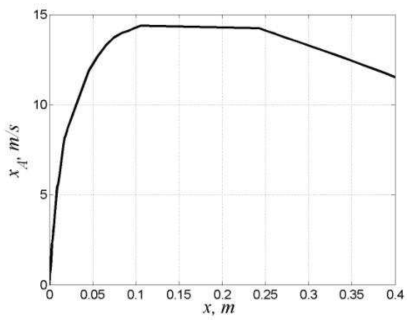

Fig. 16. Recoil velocity $\dot{x}_{\mathrm{A}}$ vs. recoil path

New simulation test is realized based on assumptions for the case B (Figure 5) with next values:

- $c_e = 1000 \cdot \dfrac{\sqrt{3000x}}{x}$ as referencing joint $c_{1\mathrm{e}}$ and $c_{3\mathrm{e}}$ placed absorbers in redundant and serial equivalent links,
- b_{e} = const = 0.2×10^5 Ns/m on the interval 0.1 to 0.4 m,
- c_2 as recuperator stiffness combined with placed absorber 2 (Figure 4) is shown as referencing resistant by dash dot line in Figures 17 and 18.
- pressures ratio $k = 0.9$.

Results are shown in Figure 17 for referencing case A and Figure 18 for referencing case B. It is obvious that differences caused by stiffness variators referred resistance force changes initial behavior of recoil system.

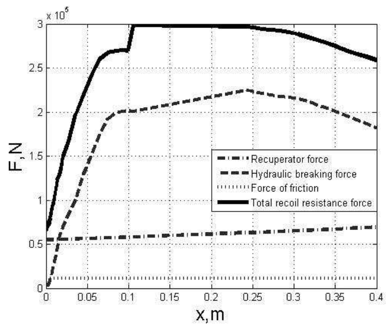

Fig. 17. Case A

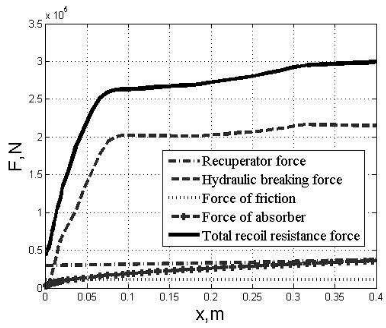

Fig. 18. Case B

Jerk ratio is expressed in Figure 19 as the referencing increasing and decreasing values in the initial phase by follow velocities in both cases.

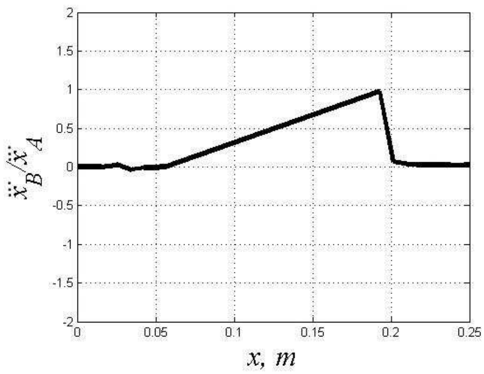

Fig. 19. Ideal ratio of recoil inertial forces

5. CONCLUSIONS

- Displacement position control as consequence of recoiling process is indirectly caused by performances of recoil system design. These performances are changing vs. time gradients more or less rigid, regarding forces or displacements making recoil characteristics more or less stiff.
- Implementation of elements with linear or nonlinear performances of stiffness vs. displacements is variators of maximum resistance force and caused jerks.
- Solution of wire rope absorbers applied as possible variators expresses controlled hysteretic nonlinear behavior employed in the initial shock phase of the artillery weapon recoil seems as applicant solution.
- Control of the absorber links by position and linking manner provides simulation of the best composition for orientated purposes, which can appear for the case of new interior ballistic forces for modernized ammunition. This extends life-cycle of heavy weapons and provides extension of its using.

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