THE IMPACT OF THE CANNON ON THE COMBAT VEHICLE CHASIS DURING FIRING

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Abstract

In this paper the mathematical model of combat vehicle as well as mathematical models of input function generated by the cannon during firing and terrain unevenness was presented. This model enables the analysis of dynamic loads of the equipment mounted inside combat vehicle as well as analysis of dynamic loads of the crew. Parameters of the impulse forces acting on the vehicle during firing were determined on the theoretical basics of internal ballistic of cannon. The results of calculation were presented for hypothetical basic combat vehicle with a 125 mm and 73 mm calibre smoothbore gun. Firing during short stops and during combat vehicle motion at different speeds on terrain unevenness as well as angle of elevation of the gun barrel and rotation angle of the turret on the level of dynamic load were considered.

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Keywords: *combat vehicle, cannon, dynamic loads, mathematical model, numerical simulations*

1. Introduction

Despite various opinions, experiences in past and present armed conflicts confirm the assumption that combat vehicles are still the elementary means of combat of Land Forces. The effectiveness of the combat vehicle is characterized by its three principle properties:

- firepower,
- armour,
- mobility.

During maintenance combat vehicles are subjected to constant modernization. The aim of modernization results from the necessity of upgrading the equipment or even the necessity of adapting them to the standard of allied forces' equipment. There is a need to possess suitable methods as well as testing tools, which would allow evaluating effects of conducted changes and properties of the new (modernized) structure at the stage of designing or modernization of combat vehicles. At the Institute of Motor Vehicle and Transportation MUT, research studies have been carried out for many years in this area. The presented study is closely connected with directions of these tests.

Its main aim is to conduct a method of testing the dynamic load of the combat vehicle during firing from the mounted weapon and especially from the cannon being the main weapon. The gun barrel is connected to the fitted cradle in the turret by the recoil and counterrecoil mechanism. During firing force impulses are generated (with an intensity of a few hundred kN), which by the pivot cradle are transferred to the turret and then on to the support structure (frame) of the combat vehicle. The magnitude of these forces as well as the effect of their interaction depends on the type and structure of the armaments (cannons), the type of ammunition used as well as firing conditions, i.e. the angle, in which the cannon is lifted; the rotation angle of the turret as well as driving conditions of the combat vehicle (terrain, speed).

2. Mathematical model of the interaction of the cannon with the combat vehicle

2.1. Introductory notes

The gun barrel is positioned in the cradle, which through the pivot rests in the combat vehicle turret. The barrel is connected with the cradle by the recoil and counterrecoil mechanism. After firing, in the period of recoiling and returning to the output position, the barrel moves on rails of the cradle. The resultant force acting on the cannon recoiling assembly during firing transfers by the motionless parts of the recoiling and counterrecoiling mechanism on to the cradle and then through its pivot to the turret and the combat vehicle chassis. During recoiling the squeezing of the springs or the compression of air in the recoil and counterrecoil mechanism takes place, in which energy is accumulated. After the recoil finishes the recoiling assembly is shifted back to the position before firing. During further considerations the force causing the return of the recoiling assembly will be omitted. It comprises of 10-20% of the force resistance value and its interaction with the combat vehicle is inconsiderable.

2.2. Recoil force resistance

The recoil force resistance R is a variable in time. The precise determination of its course requires the consideration of complex physical processes taking place during firing. For practical aims a simplified theory of internal ballistics is used that consists of determining the substitute force in the form of a rectangular pulse with a value and duration, so that its interaction with the combat vehicle would have an exact effect as the action of the real force.
With the above-mentioned assumptions in mind the value of the recoil R force is determined from the Vallier formula [4].

$$R = \frac{0.5 \cdot M_o \cdot v_{max}^2}{L_{max} \cdot L_k + v_{max} \cdot t_p}, \tag{1}$$

$$v_{max} = \frac{M_p + \beta \cdot m_m}{M_o} \cdot v_o, \tag{2}$$

$$\beta = \left(\frac{700 + v_o}{v_o} \right)^{1,1}, \tag{3}$$

where:
R – recoil force resistance,
v_{max} – maximum velocity of the free recoil (at R = 0),
L_{max} – maximum recoil path,
L_k – free recoil path (at R = 0),
t_p – duration of gunpowder gases exhaust activity,
M_p – projectile mass,
v_o – initial velocity of the projectile,
m_m – mass of propellant charge,
M_o – mass of recoiling assembly,
β – activity of gunpowder gases factor (according to Rheinmetall [3]).

Magnitudes required to determine the R force from the equation (1) are determined using the theoretical basics of internal ballistics. Based on this theory the duration of the impulse is also determined.

3. Combat vehicle model

3.1. Basic assumptions

The crawler vehicle, as a combat vehicle is, is a complex dynamic system with a high degree of freedom of movement. The mathematical analysis of such a complex system is very intricate. That is why the analysis of the model has been limited to the following vehicle vibrations:

- vertical vibrations of the body,
- angular vibrations of the body in a longitudinal plane,
- angular vibrations of the body in a diagonal plane,
- angular vibrations of armament (cannon),
- vertical and angular vibrations of seats for the crew.

Working out the vehicle model, the following simplifying assumptions were established:

- self-supporting vehicle body along with the turret is a rigid body with a known mass and mass moments of inertia,
- the interaction of caterpillar treads on the frame were omitted,
- characteristics of elasticity of components and damping of the suspension can be accepted as linear or non-linear,
- kinematics input function is specified for the axles of wheels; after the exhaustion of the suspension dynamic bend additional components start to operate (wheel jump limiters),
- the diagonal and longitudinal displacement of the frame were omitted as well as its rotational motion in relation to the vertical axle,
- the seat for the crew are treated as rigid bodies supported by elastic and damping components,
- the constraints applied on the system are holonomic, independent of time and bilateral.

3.2. Combat vehicle mathematical model

The discrete combat vehicle model is presented in fig.1. The rigid body of combat vehicle (frame with turret and cannon) supported by elastic and damping components mapping turning shaft and the shock absorber as well as the rigid bodies modelling the seats for the crew are distinguishable in the model, each supported in relation to the frame on for elastic and damping components.

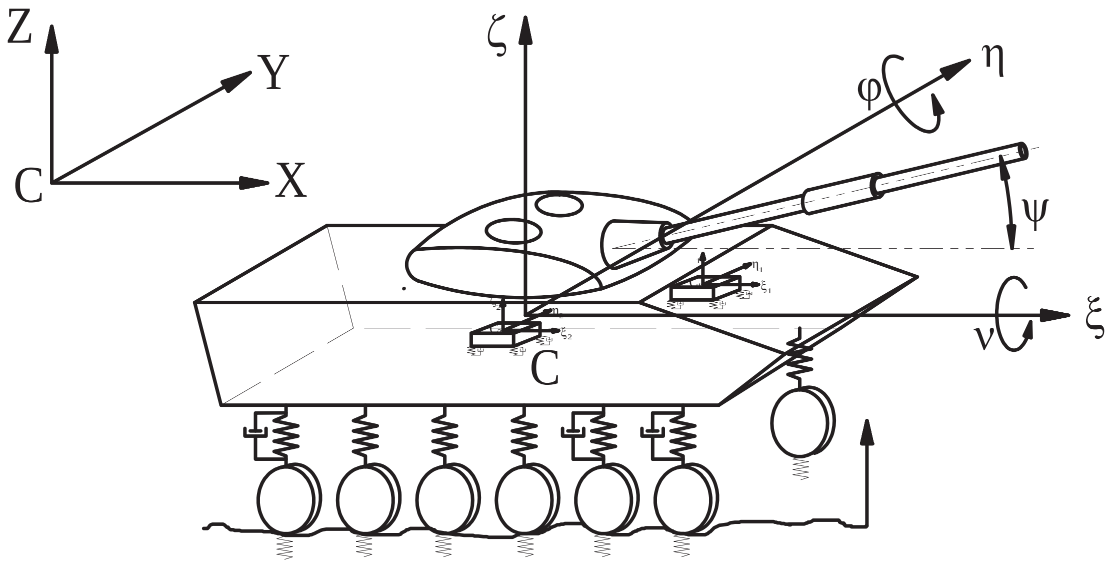

Fig. 1. Discrete combat vehicle model scheme

The base stiffness was taken into consideration in the model, which allowed more precise mapping of occurrences taking place during the exhaustion of the dynamic jump of the wheels. Accepting linear characteristics of suspension components in the combat vehicle model is a big simplification. This results, among others, from the fact that at a linear characteristic of the turning shaft elasticity when using the equalizer spring causes an occurrence of the so-called geometric non-linearity. That is why the suspension stiffness factor *k* was determined by the following dependency:

$$k = \left(\frac{G \cdot I_o}{l \cdot r^2}\right) \cdot \left(\frac{cos(\alpha_o - (\alpha_{st} + \delta)) - (\alpha_{st} + \delta) \cdot sin(\alpha_o - (\alpha_{st} + \delta))}{cos^3(\alpha_o - (\alpha_{st} + \delta))}\right), \tag{4}$$

where:

G – the Kirchhoff module of the turn shaft material,
I_o – polar moment of inertia of the turn shaft diagonal intersection,
l – active length of the turn shaft,
α_{st} – static bend angle of the equalizer spring,
δ – current bend angle from the static equilibrium position,
α_o – equalizer spring assembly angle,
r – turn shaft radius.

The equation (4) is apt until the moment of exhaustion of the dynamic jump of the wheels. Then an impact occurs of the equalizer spring on the limiter of high stiffness, which means, that the elastic and damping components do not undergo further deformation. It is assumed in the proposed calculation model, that at the moment of impact of the suspension against the propulsion system, they form one rigid body supported by elastic components, which are rubber bands for the wheels and a deformable base [1].

The substitute stiffness factor of the elastic component connection series modelling the wheels rubber band and the elastic components modelling the plane (ground) were determined from the following dependency:

$$k' = \frac{k_1 \cdot k_2}{\left(k_1^{\frac{2}{3}} \cdot k_2^{\frac{2}{3}}\right)^{\frac{2}{3}}} \,, \tag{5}$$

where:

k_1 – ground stiffness,
k_2 – rubber band stiffness.

Ground stiffness was determined by the following dependency:

$$k_1 = \frac{E_o}{\omega \cdot B \cdot \left(1 - v^2\right)} \,, \tag{6}$$

whereas the rubber band stiffness with the following expression:

$$k_2 = \frac{\sqrt{32} \cdot E \cdot b \cdot \sqrt{r_1}}{3 \cdot h} \,, \tag{7}$$

where:

E_o – Young module for the ground,
ω – caterpillar tread shape factor,
B – caterpillar tread width,
v – ground side expansion factor,
E – Young module for rubber,
h – rubber band height,
b – rubber band width,
r_1 – wheel radius along with band.

The adopted ground model includes its elastic properties in both directions (vertical and horizontal) and its values depend on the type of ground (coherence modulus and internal friction modulus), its density and moisture. The factor describing the ground parameters were accepted for static conditions, because its change in case of dynamic interaction has a slight impact on the test results.
The equation of non-linear motion was formed in an increment linear version at an incrementation step used in non-linear mechanics. They take the following form [2]:

$$\boldsymbol{M}\Delta\ddot{\boldsymbol{q}} + \boldsymbol{C}\Delta\dot{\boldsymbol{q}} + \boldsymbol{K}\Delta\boldsymbol{q} = \Delta\boldsymbol{Q} \,, \tag{8}$$

where:

$\boldsymbol{M}$ – inertia matrix,
$\boldsymbol{C}$ – damping matrix,

$\boldsymbol{K}$ – stiffness matrix,

$\boldsymbol{\Delta Q}$ – vector increment of generalized external forces,

$\boldsymbol{\Delta \ddot{q}}, \boldsymbol{\Delta \dot{q}}, \boldsymbol{\Delta q}$ – accordingly: vector increment of generalized acceleration, speed and relocation in the given integration step.

The occurrence of the stiffness and damping factors in the matrix are calculated for each incrementation step depending on the bending of the wheel as well as wheel speed in relation to the frame.

A computer program was designed for the analysis of vibrations of the crawler vehicle. This program enables:

– determining the frequency and the form of free vibration,
– analysis of forced vibrations,
– elaborating amplitude and frequency characteristics.

Model testing can be carried out at a specified driving speed and defined forced parameters for the following driving conditions:

– driving over a single V-block,
– driving on a sinusoidal track,
– driving on a track of random characteristics.

In each case, an additional force input function may act on the vehicle from the cannon recoil force applied at the place where it is fixed (cannon pivot).

For integrating the equation of motion, specifying the combat vehicle model vibration, the direct integration Newmark method was used.

3.3. Force model

3.3.1. Kinematics force of random distribution

Generating kinematics force of random distribution acting on the wheels of the vehicle carry out based on assigned spectral concentration characteristics. The execution of force is based on the assumption that the stationary, centralized process of normal distribution can be presented in the form of an infinite sum of harmonics [9].

$$z(t) = \sum_{k=1}^{\infty} A_k \cdot cos(\omega_k t - \varphi_k), \tag{9}$$

where:

A_k – k harmonic amplitude,

φ_k – k harmonic phase angle (random variable of uniform distribution in an interval of <0,2π>).

The path profile, on which the vehicle moves, can be interpreted as a sum of waves of different lengths and of random variable phases [5]. Waves of lengths exceeding more than 100 m create a so-called macro-profile. It has an essential impact on the vehicle dynamics, however its impact on vehicle vibration is minimal, because of a very low frequency. Because the caterpillar system is not capable of projecting irregularity of small lengths, waves of lengths smaller than 0.1 m (so-called surface roughness) also is not taken into consideration in the vehicle vibration analysis. So the subject of numerical testing was only the profile of the path, consisting of irregular waves of lengths in an interval of 0.1 – 100 m.

Based on this as well as introducing the idea of spectral density $G_w(\omega)$ the following can be written:

$$z(t)=\sum_{k=1}^{N}\sqrt{G_w(\omega)\cdot\Delta\omega}\cdot cos(k\Delta\omega t-\varphi_k). \tag{9}$$

In numerical usage, usually the estimated specification of the spectral density of path irregularity is used. In study [7] the following dependency is suggested:

$$G_w(\omega)=\frac{1}{v}\cdot G_w(\Omega_o)\cdot\left[\frac{\Omega}{\Omega_o}\right]^{-w}, \tag{10}$$

where:

Ω_o - wheeled frequency relation,

$G_w(\Omega_o)$ - path irregularity indicator, from which it can be concluded whether the path is generally good or poor,

w - waviness indicator, informing whether short waves or long waves have a dominating impact in the spectrum.

The final equation (9) after considering the above dependencies will look like the following:

$$z(t)=\sum_{k=1}^{N}\sqrt{\frac{G_w(\Omega_o)\cdot\Delta\omega\cdot v^{w-1}}{(k\Delta\omega)^w}}\cdot cos(k\Delta\omega t-\varphi_k). \tag{3.9}$$

Presented in table 1 are the average accepted values for specifying the spectral concentration power of path irregularity depending on the different conditions of the path surface of dirt roads for the frequency relation of Ω_o= 1 m^{-1} [7].

Tab. 1. Average values for specifying the power spectral density of path irregularity for different conditions of dirt road pavement

Path type	**Pavement condition (subjective evaluation)**	**Average values**	
		w	$G_w(\Omega_o)$ [cm³]
Dirt road	Good	2.25	32
	Average	2.25	155
	Poor	2.14	602
	Very poor	2.14	16300

Presented in fig. 2 is an example of the execution of the kinematics force applied to the wheels of a vehicle moving at a speed of 30 kmph on a dirt road of an average surface condition.

3.3.2. Force input function

The crawler vehicle model can be also loaded with force from the recoil of the cannon resulting from a shot. Based on considerations included in chapter 2, input function was accepted as a rectangular course presented in fig. 3. The characteristic magnitudes needed to conduct calculations are:

– recoil force R determined from dependence of (1),
– initial time t_o (shot fired),
– terminal time t_k.

The difference $t_k - t_o$ is the duration of force impulse. The R recoil force is applied at the place were the cannon pivots are fixed. The duration of recoil force impulse and its value for different kind of cannon are presented in table 2.

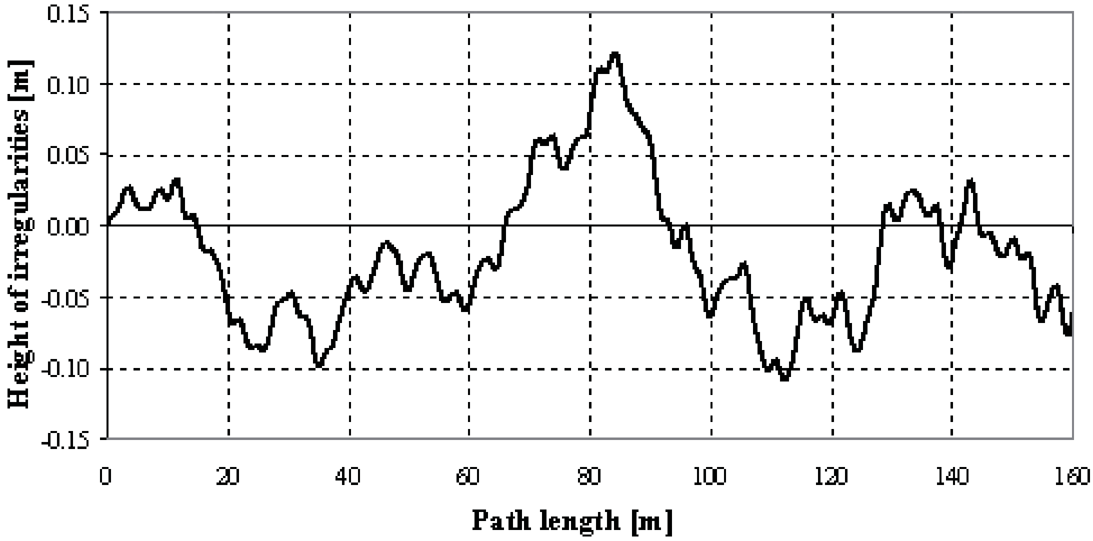

Fig. 2: Example of executing kinematics force

Tab. 2. The values of recoil force impulse

Type of cannon	Combat vehicle	Recoil force [kN]	Duration of recoil force [ms]
125 mm cannon 2A46	T-72, PT-91, T-80	524	56
100 mm cannon D10-TG	T-55, T-54	280	78
73 mm cannon 2A28	BMP-1	126	123

A change in direction is possible as well as the return recoil force activity by:

– changing the angle of the cannon α_A,
– changing the rotation angle of the turret α_W.

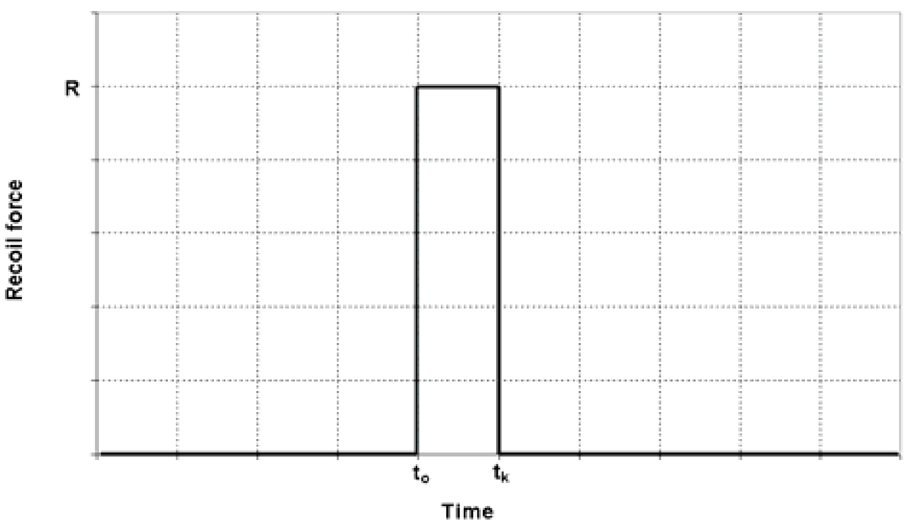

Fig. 3. Course of the recoil force applied on the vehicle during firing

4. Simulation tests

4.1. Range of testing

The dynamic load of the combat vehicle was examined in the study, being the result of the input function from the recoil force of the combat vehicle cannon, as well as the kinematics input function caused by motion over irregular terrain. Calculations were conducted for two type of vehicle – the first one with 125 mm cannon (tank) and the second one with 73 mm cannon (combat infantry carrier). The simulation was carried out during vehicle standstill as well as motion over a dirt road of an average condition at a speed of 7 and 30 kmph. The selection of speed values resulted from the characteristics of basic types of activity carried out on the battlefield (average attack speed of infantry "on foot" as well as average combat vehicle motion speed in the terrain).

4.2. Test results of the model

In the model tests, the vertical acceleration acting on the driver and the vehicle centre of mass as well as the frame angular acceleration in relation to transverse axis were stated during a simulated fire from the combat vehicle cannon preformed moving on a irregular terrain with parameters: $G_w(\Omega_o)$ = 155 cm³, w = 2.25.

On figures 4 – 6 exemplary time courses of vertical acceleration of the driver's seat and the centre of mass of the vehicle are presented as well as angular acceleration in relation to its transverse axis. A sudden impulsive rise in acceleration at the moment of firing can be observed on these charts (4 seconds). Through analysis of these courses it can also be ascertained that the driver is submitted to approximately twice greater dynamic load in comparison to the rest of the crew members found in the inner part of the vehicle, this is caused by the considerable distance of the driver from the combat vehicle's centre of mass.

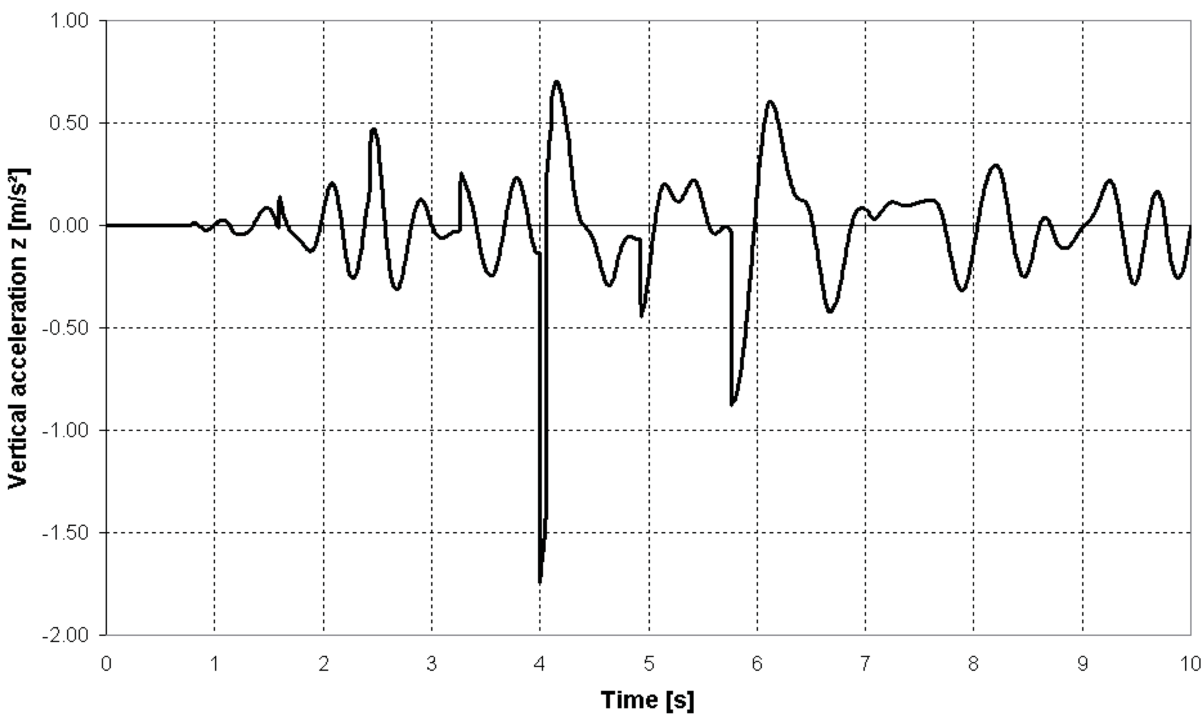

Fig. 4. Vehicle centre of mass vertical acceleration during firing and driving at a speed of v=7 kmph over terrain irregularity of parameters: $G_w(\Omega_o)$ = 155 cm³, w = 2.25

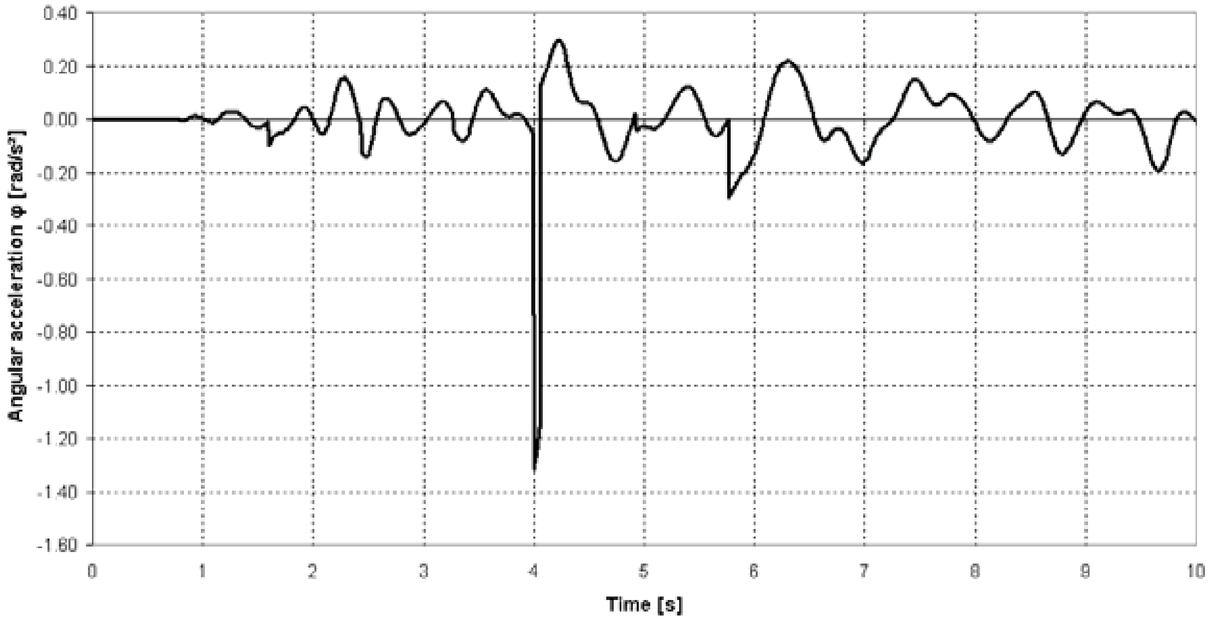

Fig. 5. Vehicle angular acceleration during firing and driving at a speed of v=7 kmph over terrain irregularity of parameters: $G_w(\Omega_o)$ = 155 cm³, w = 2.25

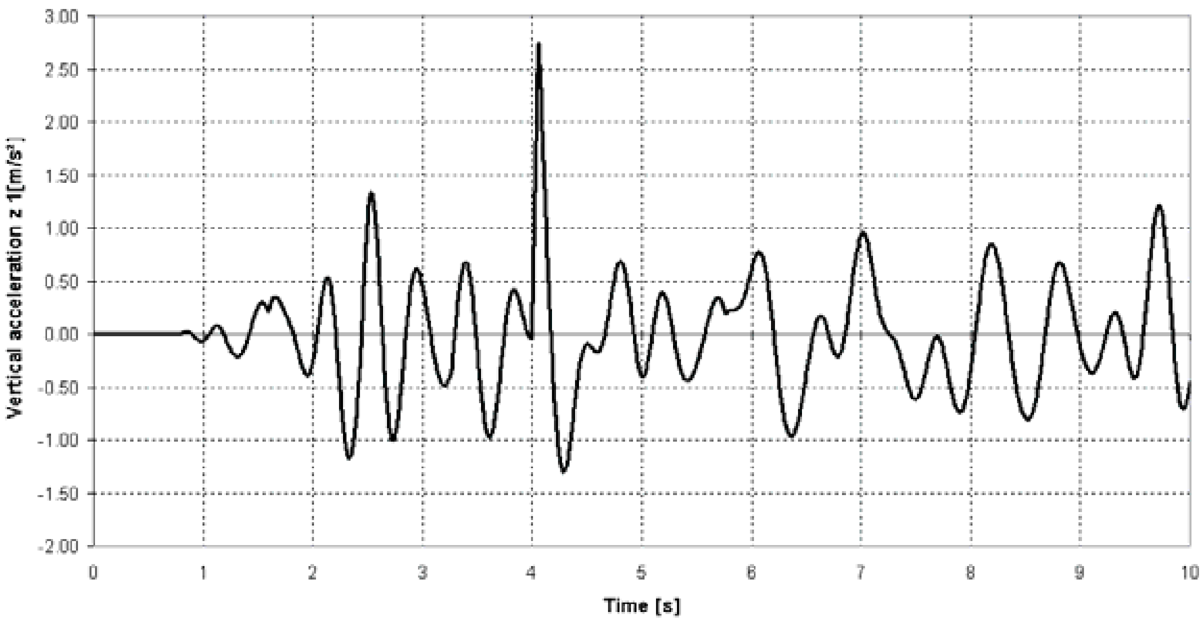

Fig. 6. Driver's seat vertical acceleration at a speed of v=7 kmph over terrain irregularity of parameters: $G_w(\Omega_o)$ = 155 cm³, w = 2.25

Maximum values of vertical acceleration of the driver's seat as well as angular acceleration of the combat vehicle frame in relation to the transverse axis are presented in fig. 7 – 10. These accelerations a presented in the function of turret rotation angle for three cannon elevation angles: 0º, 7.5º and 15º (for a tank) and 0º, 15º and 30º (for a combat infantry carrier).

From the curves on the figure, a dependency of acceleration values from both the turret rotation angle and the cannon barrel elevation angle can be observed. The centre of mass vertical acceleration at a considerably small degree depends on the position of the turret with the cannon during firing. This results from a slight difference of the position of the centre of mass and the cannon pivots, on which the recoil force directly acts. In the range of the conducted tests the changes in centre of mass vertical acceleration values did not exceed the value of 1m/s² and therefore these values are not presented in the chart. Acceleration values generated during firing are not very great and do not influence in an essential way on the operation conditions of the combat vehicle crew.

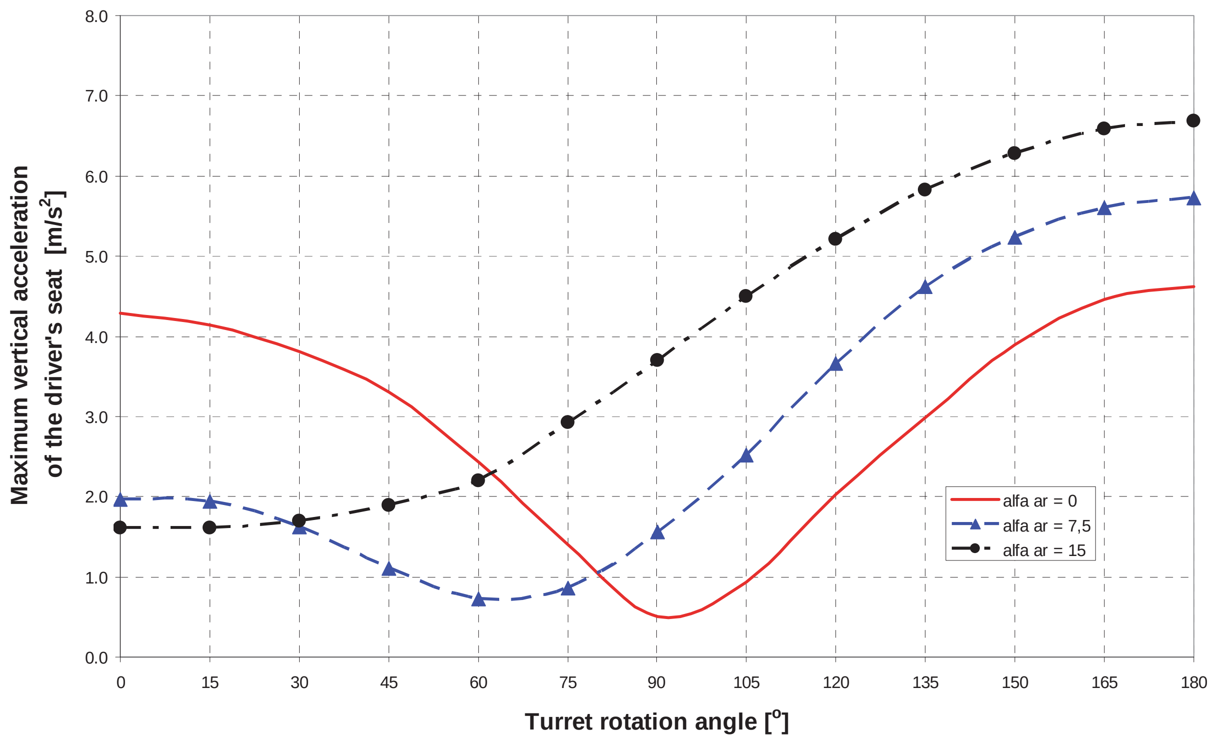

Fig. 7. Maximum values of vertical acceleration of the driver's seat – tank

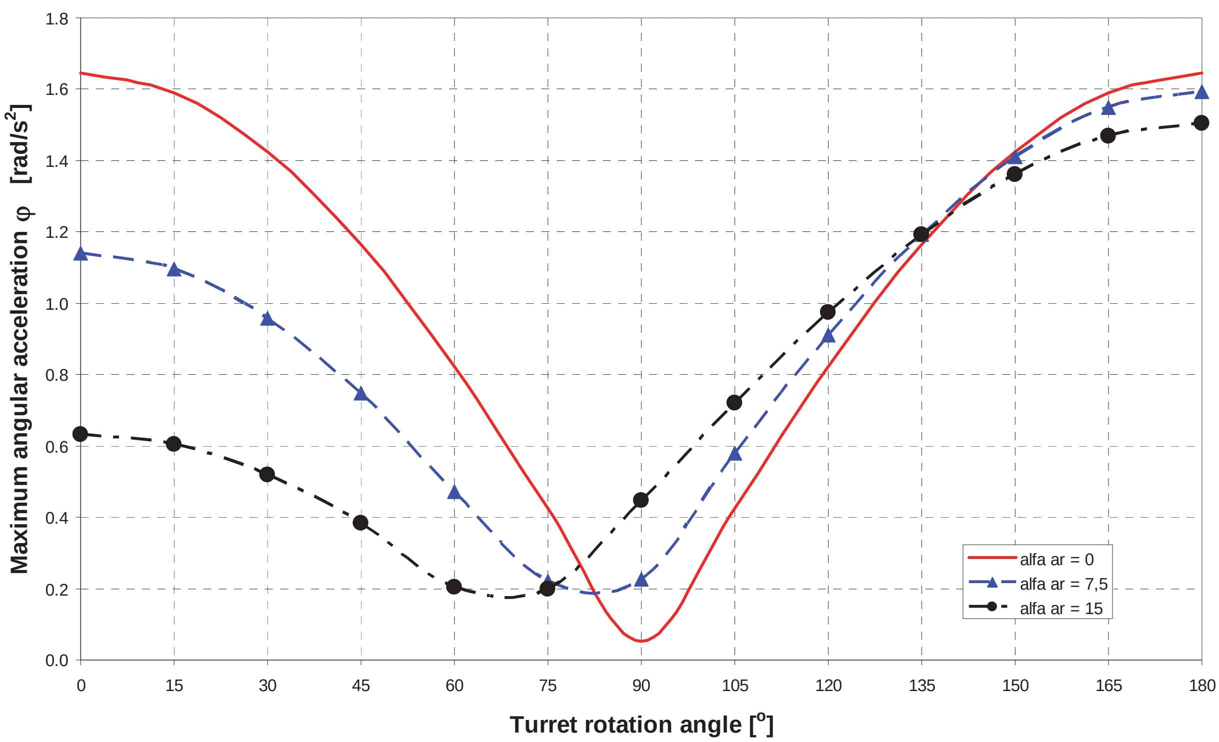

Fig. 8. Maximum values of angular acceleration in relation to the transverse axis – tank

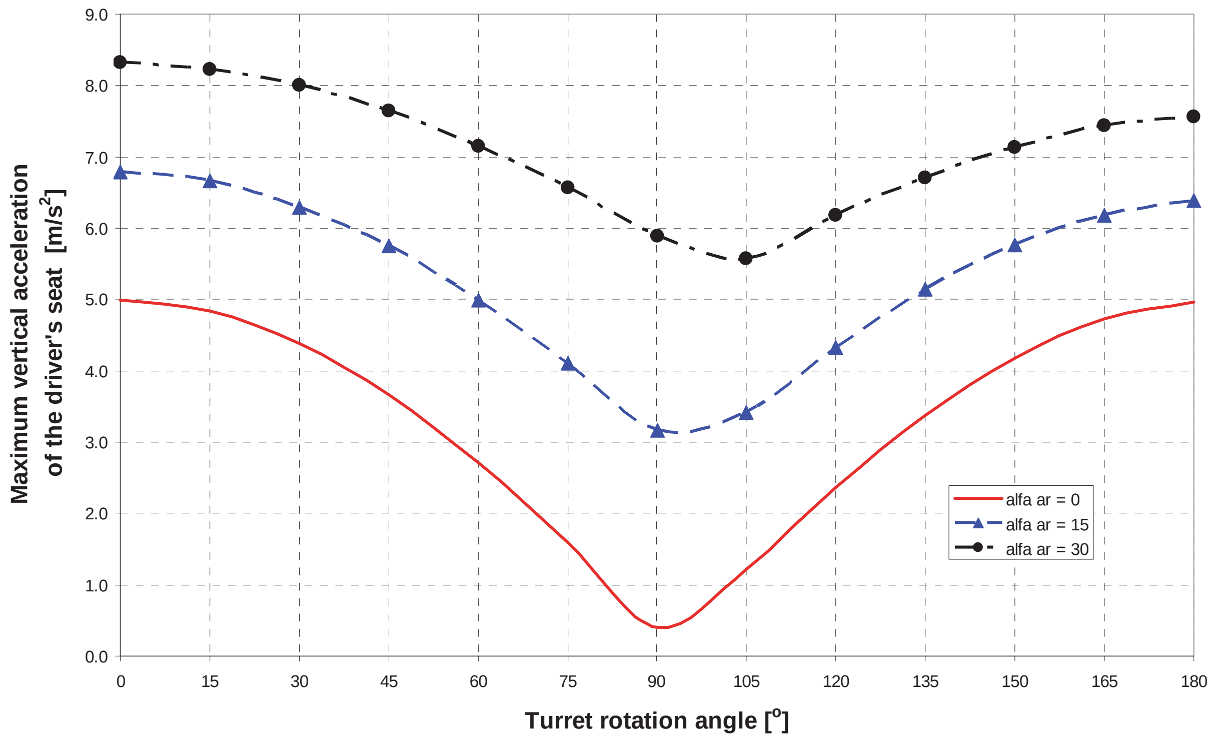

Fig. 9. Maximum values of vertical acceleration of the driver's seat – combat infantry carrier

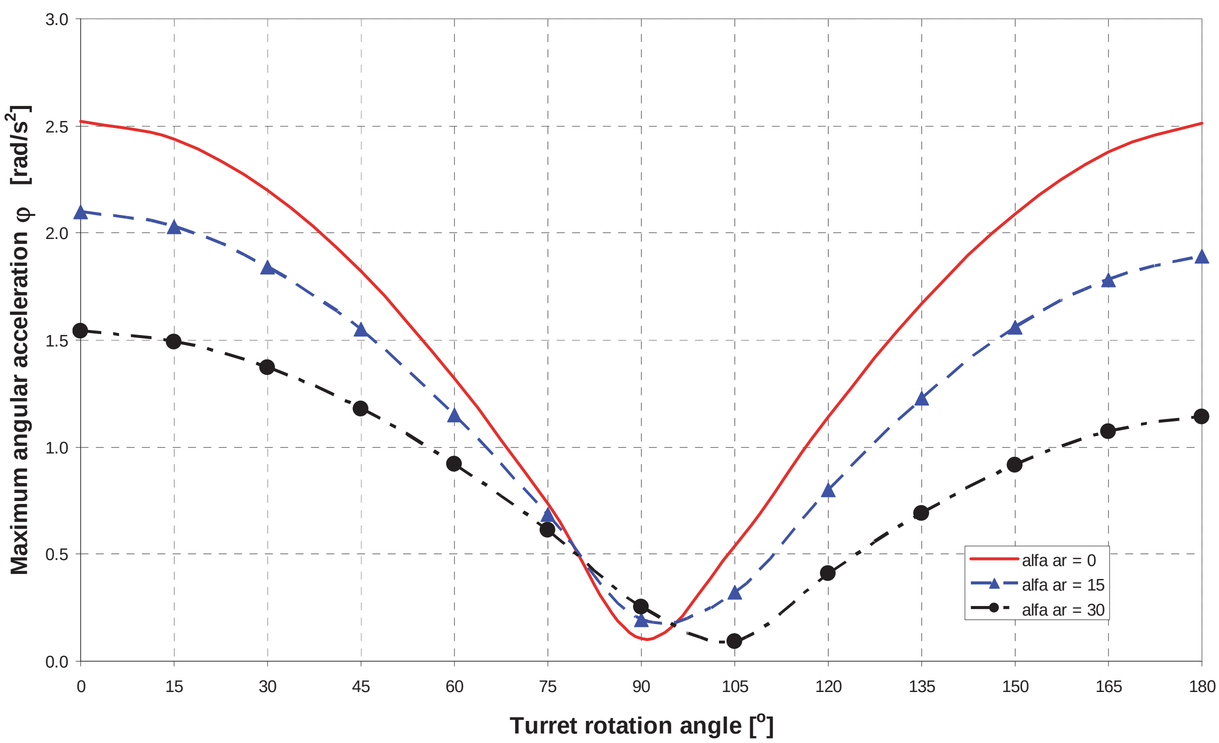

Fig. 10. Maximum values of angular acceleration in relation to the transverse axis – combat infantry carrier

Though forces generated during firing have an essential impact on the dynamic load of the combat vehicle support structure as well as its assemblies, this fact should be in mind during modernisation of combat vehicles especially during changing of armaments.

5. Conclusion

The achieved test results have a cognitive aspect as well as a practical meaning. No results of this type of test are published.

The knowledge of dynamic interaction created during motion and firing enables the evaluation of their impact on equipping the combat vehicle, the accuracy of conducting fire as well as working conditions of the crew. The analysis of these interactions can also be the basis for stating the direction of modernisation.

The presented testing methodology in this study as well as the created specialised programs for simulation testing at the Institute can be used by the technical back-up in charge of designing or modernising combat vehicles or constructing specialised vehicles on their chassis.

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