Dynamics of mounted automatic cannon on track vehicle

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***Abstract*—**The article deals with the vibrations of the main parts of the weapon system where the automatic cannons are mounted. The dynamic model has 8 DOF with three parts: hull, turret and elevation parts. The presented procedure is able to evaluate the possible changes of the elevation angles like the aiming errors during the variation of the dynamic characteristics for example the masses, mass moment of inertia, stiffness, damping coefficients, weapon operation principle. The sensitive analysis was applied on the system. The excitation force obtained from measuring was used for tuning of the system and afterwards was replaced with the simple analytical formula.

***Keywords*—** Hit probability, Hull vibration, Gatling weapon, Clearance, Sensitive analysis.

I. Introduction

One of the main requirement posed on the current anti-aircraft systems and weapons mounted on the infantry fighting vehicles (IFV) is to achieve of the high hit probability of the targets.

The radar, computer and optoelectronics technology development enables to parameterize of targets with accuracy comparable with the weapon technical dispersion, see [2], [4]. For example the bearing and elevation angles determination is possible with 2 mrad errors and the target velocity even with 0.1 %. The aiming of allowance is calculated with 0.5 mrad accuracy. The limiting factor deteriorative the fire accuracy are vibrations of mounting and basic structure of the weapon when the constant errors are the highest in elevation and they can achieve up to 10 mrad, see [1], [4]. The errors vary in course of burst fire for every shot and the fall can get under 50 % in ranges d_t overlapping 500 metres as it is demonstrated in Fig. 1 and Fig. 2. The constant errors (theoretical case) are marked as Θ during every shot. The variable errors Θ_p (real situation) in the Table I are errors varying in every shot as it is known from experience and experiments mentioned in [3]. The calculated time is when the projectile exits the muzzle of the barrel. The next examples have been calculated for data belonging to the combat helicopter, as the target, in the hanging position, the height under earth has been 50 metres. The weapon is twin 30 mm AA cannon burst firing 2x420 rounds/min. The second case of hit probability course when the combat helicopter is in the lateral position is in the Fig. 2. The decreasing of the hit probability is greater than in previous case.

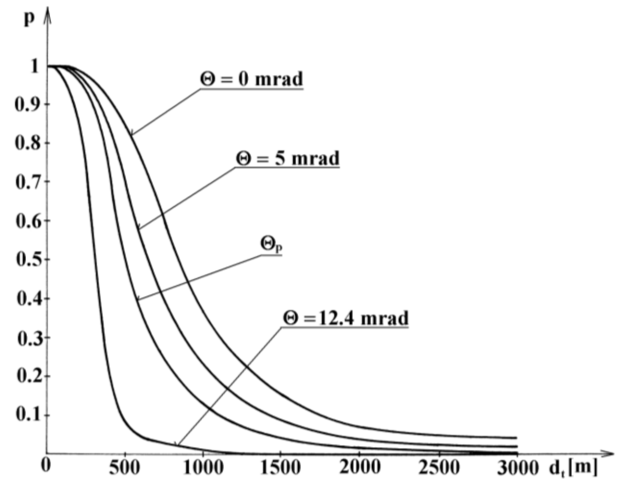

Fig. 1 Influence of elevation errors on hit probability – target in the front position

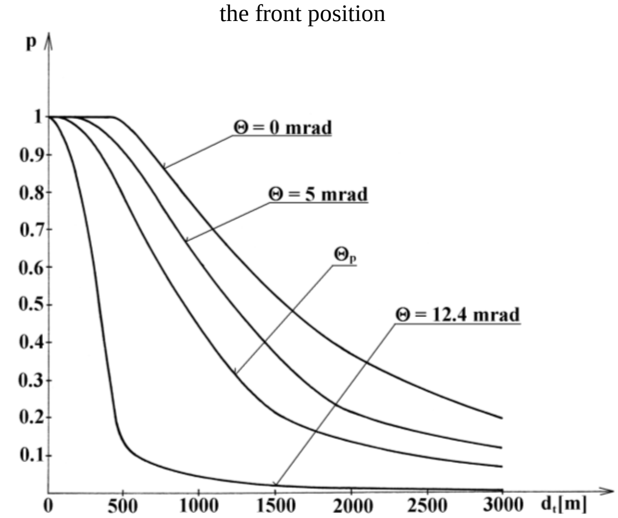

Fig. 2 Influence of elevation errors on hit probability – target in the lateral position

TABLE I
Elevation errors during six shots

i	1	2	3	4	5	6
Θp (mrad)	1.1	5	8	10	11.5	12.4

The decreasing of the hit probability when the target is

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moving is much greater. The schematic analyses show that to ensure the high hit probability is very difficult. One of the ways is to eliminate the unfavourable effects of vibration mounting and the basic structure of the weapon incurred by the burst firing. The determination of these vibrations makes more difficult the nonlinearity being in principle type of backlashes, backstops, and dry frictions. The fundamental problem is the unfamiliarity with nonlinearity real values. In course of practice calculations is trouble with verification on the real system which leads to the effect that only one nonlinear element is studied. The calculations lead into the make up of the discrete models having inertial, elastic and damping properties mainly as lumped parameters and the continuum characteristics are considered less (FEM).

II. Dynamic Model of Weapon Mounted on Vehicle

The scheme of the dynamic model following from the Fig. 1 consists of the oncoming parts: hull, turret, and elevation parts, see [10], [14], [19].

In the Fig. 3 and Fig. 4 (where the dynamic model is) there are studied the track systems as the ones of the most extended version using in the anti-aircraft artillery systems and IFV when on the chassis there is mounted one, two or four automatic machine guns or automatic cannons burst firing.

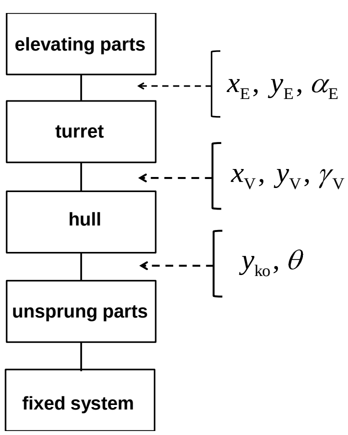

Fig. 3 Structure of weapon

The dynamic model has been made up with the next presumptions:

The weapon system is being at the rest before firing.

The fire plane is coincident with the vertical plane passing through the longitudinal vehicle axis; it means the weapon has zero bearing.

The elevation angle is constant, i.e. the target is not tracked,

The limited rigidity has coupling between the hull and unsprang masses, between hull and turret and between turret and elevation parts.

During whole action the masses are changeless,

Backslashes are expressed by means of the nonlinear reduced rigidities in the couplings.

All considered forces and moments act in one vertical plane and the system is symmetrical, i.e. the planar dynamic system is studied.

The force acting in the weapon and causing the motion of all weapon parts depends on the type of operation. During burst firing they are periodic in nature, see [1], [3]. The exciting force in the case study is indicated as F_E in Fig. 4.

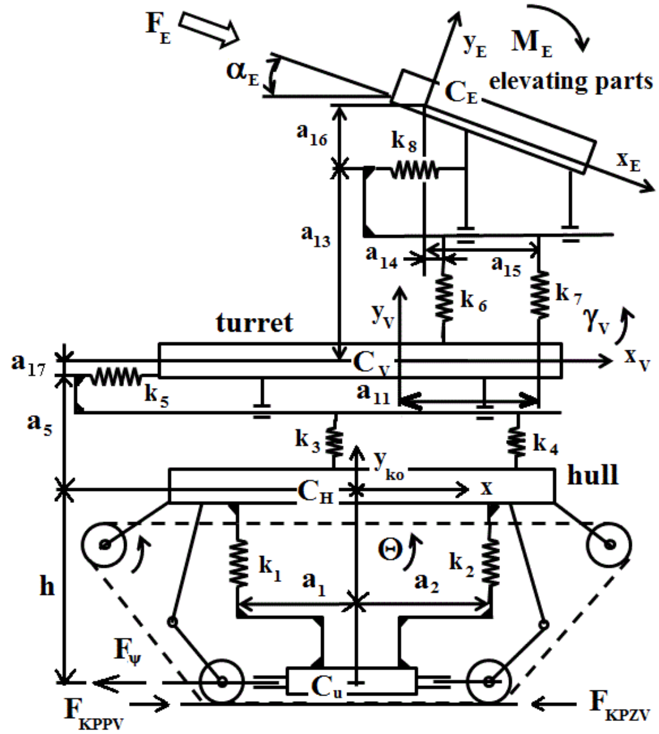

Fig. 4 Dynamic model

The considered system has 8 DOF:

$$[\mathbf{q}] = [y_{ko}, \Theta, x_v, y_v, \gamma_v, x_E, y_E, \alpha_E], \tag{1}$$

where

y_{ko} - vertical displacement of hull,

Θ - angular displacement of hull,

x_V - longitudinal displacement of turret,

y_V - vertical displacement of turret,

γ_V - angular displacement of turrret,

x_E - longitudinal displacement of elevating parts,

y_E - vertical displacement of elevating parts,

α_E - angular displacement of elevating parts.

Then the system can be described by the matrix equation:

$$\mathbf{M}\ \ddot{\mathbf{q}} + \mathbf{K}\ \mathbf{q} = \mathbf{Q}\ , \tag{2}$$

where

$$\mathbf{M} = m_{ko}, I_{ko}, m_V, m_V, I_V, m_E, m_E, I_E\ , \tag{3}$$

is diagonal mass matrix determined from the system kinetic energy where

m_{ko} - mass of hull,

I_{ko} - hull mass moment of inertia with respect to the transverse axis passing through the gravity centre,

m_v – mass of turret,

I_v - turret mass moment of inertia with respect to the transverse axis passing through the gravity centre,

m_E - mass of elevating parts,

I_E – elevating parts mass moment of inertia with respect to the transverse axis passing through the gravity centre.

When the mass matrix is diagonal it is favourable for calculations because the inversion of this matrix is simple. In order to the mass matrix will be diagonal several principles have to be kept such as deviation mass moments of inertia will be removed using of the suitable coordinate system, [15], [16].

The $\mathbf{K}$ stiffness matrix elements are determined from the formulae (4), see [13], [16]:

$$k_{jl} = \left(\frac{\partial^2 E_P}{\partial q_j \partial q_l} \right)_0, \qquad j, l = 1, 2, ..., n \tag{4}$$

where

E_P - system potential energy,

n – number of DOF,

0 subscript – value in the balanced position.

The stiffness matrix is symmetrical due to the interchangeability of the corresponding derivation according to the q_j, q_l coordinates in (4).

For better understanding let us denote every general coordinate like this:

y_{ko} - 1, Θ - 2, x_v - 3, y_v - 4, γ_v - 5, x_E - 6, y_E - 7, α_E - 8.

The potential energy of the system is:

$$\begin{aligned} E_P &= 0.5k_1(y_{ko} - \Theta a_1)^2 + 0.5k_2(y_{ko} + \Theta a_2)^2 + \\ &+ 0.5k_3(\Theta a_3 - y_{ko} + y_V - \gamma_V a_6)^2 + \\ &+ 0.5k_4(y_V + \gamma_V a_7 - \Theta a_4 - y_{ko})^2 + 0.5k_5(\Theta a_5 + x_V + \gamma_V a_{17})^2 + \\ &+ 0.5k_6(y_E \cos\alpha_E + \alpha_E a_{14} - x_V \sin\alpha_E - y_V - \gamma_V a_{10})^2 + \\ &+ 0.5k_7(y_V \cos\alpha_E + \alpha_E a_{15} - x_E \sin\alpha_E - y_V - \gamma_V a_{11})^2 + \\ &+ 0.5k_8(y_E \sin\alpha_E + x_E \cos\alpha_E + \alpha_E a_{16} + \gamma_V a_{13} - x_V)^2. \end{aligned} \tag{5}$$

The distances a_i are evident from the Fig. 4 with the exceptions these:

a_3 - between C_H hull gravity centre and spring k_3,

a_4 - between C_H hull gravity centre and spring k_4,

a_6 - between C_V turret gravity centre and spring k_3,

a_7 - between C_V turret gravity centre and spring k_4,

a_{10} - between C_V turret gravity centre and spring k_6,

h - vertical distance between C_H hull gravity centre and C_u unsprung mass gravity centre.

The meaning of the elastic elements k_i is following:

k_1 - reduced stiffness of the vehicle left part suspension,

k_2 - reduced stiffness of the vehicle right part suspension,

k_3, k_4, k_5 -reduced stiffness of the turret on the hull,

k_6, k_8 - reduced stiffness of the trunnion,

k_7 - this element is used for modelling of the stiffness of the elevating gear.

After derivation of the potential energy from (5) by means of (4) the stiffness matrix is given with the formulae (6).

The main nonlinearities considered in the system are: slackness in the ball raceway between turret and hull modelled by means k_5 stiffness spring, slackness in the trunnion between turret and elevating parts modelled by means k_8 stiffness spring and backlash in the elevating gear is represented by k_7 stiffness spring.

$$\mathbf{K} = \begin{pmatrix} k_{11} & k_{12} & & k_{14} & k_{15} & & & \\ k_{21} & k_{22} & k_{23} & k_{24} & k_{25} & & & \\ & k_{32} & k_{33} & & k_{35} & k_{36} & k_{37} & k_{38} \\ k_{41} & k_{42} & & k_{44} & k_{45} & k_{46} & k_{47} & k_{48} \\ k_{51} & k_{52} & k_{53} & k_{54} & k_{55} & k_{56} & k_{57} & k_{58} \\ & & & k_{64} & k_{65} & k_{66} & k_{67} & k_{68} \\ & & & k_{74} & k_{75} & k_{76} & k_{77} & k_{78} \\ & & & k_{84} & k_{85} & k_{86} & k_{87} & k_{88} \end{pmatrix}. \tag{6}$$

The general forces, represented by $\mathbf{Q}$ matrix, encompass the others forces which are not included into matrix mentioned in parts of $\mathbf{M}$ and $\mathbf{K}$ matrixes. They are damping forces and exciting forces. Due to the experimental research which confirmed that vehicle having braked track wheel usually does not drive backward (against the fire direction) the x coordinate and corresponding movement not have to be considered.

Then the F_ψ, F_{KPZV} and F_{KPPV} forces (tractive resistance force, force straining backside part of track and force straining front side part of track) can be excluded from the solution. Otherwise the number of the degrees of freedom increases to nine.

The damping forces are implicated in forces having form

$$F_{d_{q_i}} = b_{q_i} \dot{q}_i \tag{7}$$

where

b_{q_i} - damping coefficient pertaining to the i^{th} degree of freedom,

$\dot{q}_i$ - the second derivation (velocity) determined from (3).

The damping coefficients for the case study are marked in the format:

$$b_{y_{ko}},\ b_\Theta, b_{x_V}, b_{y_V},\ b_{\gamma_V}, b_{x_E}, b_{y_E}, b_{\alpha_E}. \tag{8}$$

The equation are solved after arrangements in the next form

$$\ddot{\mathbf{q}} = \left[\mathbf{M}^{-1}\right] \mathbf{Q} - \mathbf{K}\ \mathbf{q}\ . \tag{9}$$

The equations describing the hull movement are:

$$m_{ko}\ddot{y}_{ko} + k_{11}y_{ko} + k_{12}\theta + k_{14}y_V + k_{15}\,y_{\gamma_V} = -\,\dot{y}_{ko}b_{yko}, \tag{10}$$

$$I_{ko}\ddot{\theta} + k_{21}y_{ko} + k_{22}\theta + k_{23}x_V + k_{24}y_V + k_{25}\gamma_V = -\,b_\theta\dot{\theta}. \tag{11}$$

The turret movements are expressed with three differential equations:

$$\begin{aligned} &m_V\ddot{x}_V + k_{32}\theta + k_{33}x_V + k_{35}\gamma_V + k_{36}x_E + \\ &+ k_{37}y_E + k_{38}\alpha_E = -\dot{x}_V b_{X_V} \end{aligned}, \tag{12}$$

$$m_{\mathrm{V}}\ddot{y}_{\mathrm{V}} + k_{41}y_{\mathrm{ko}} + k_{42}\,\theta + k_{44}\,y_{\mathrm{V}} + k_{45}\gamma_{\mathrm{V}} + k_{46}\,x_{\mathrm{E}} + k_{47}y_{\mathrm{E}} + k_{48}\,\alpha_{\mathrm{E}} = -\dot{y}_{\mathrm{V}}b_{y_{\mathrm{V}}}, \tag{13}$$

$$I_{\mathrm{V}}\ddot{\gamma}_{\mathrm{V}} + k_{51}y_{\mathrm{ko}} + k_{52}\theta + k_{53}x_{\mathrm{V}} + k_{54}y_{\mathrm{V}} + k_{55}\gamma_{\mathrm{V}} + k_{56}x_{\mathrm{E}} + k_{57}\,y_{\mathrm{E}} + k_{58}\alpha_{\mathrm{E}} = -\dot{\gamma}_{\mathrm{V}}b_{\gamma_{\mathrm{V}}}, \tag{14}$$

$$m_{\mathrm{E}}\ddot{x}_{\mathrm{E}} + k_{63}x_{\mathrm{V}} + k_{64}y_{\mathrm{V}} + k_{65}\gamma_{\mathrm{V}} + k_{66}\,x_{\mathrm{E}} + k_{67}\,y_{\mathrm{E}} + k_{68}\alpha_{\mathrm{E}} = -\dot{x}_{\mathrm{E}}b_{x_{\mathrm{E}}} + F_{\mathrm{E}}\cos\varphi, \tag{15}$$

$$m_{\mathrm{E}}\ddot{y}_{\mathrm{E}} + k_{73}\,x_{\mathrm{V}} + k_{74}\,y_{\mathrm{V}} + k_{75}\,\gamma_{\mathrm{V}} + k_{76}\,x_{\mathrm{E}} + k_{77}\,y_{\mathrm{E}} + k_{78}\alpha_{\mathrm{E}} = -\dot{y}_{\mathrm{E}}b_{y_{\mathrm{E}}} - F_{\mathrm{E}}\sin\varphi, \tag{16}$$

$$I_{\mathrm{E}}\ddot{\alpha}_{\mathrm{E}} + k_{83}\,x_{\mathrm{V}} + k_{84}\,y_{\mathrm{V}} + k_{85}\,\gamma_{\mathrm{V}} + k_{86}\,x_{\mathrm{E}} + k_{87}y_{\mathrm{E}} + k_{88}\,\alpha_{\mathrm{E}} = -\dot{\alpha}_{\mathrm{E}}b_{\alpha_{\mathrm{E}}} - F_{\mathrm{E}}\,r. \tag{17}$$

III. Calculation Results

The system of the differential equations (10) – (17) has been solved using the Runge-Kutta of the 4th order integration method. The suitable integration step has been chosen as 0.0001s for the specific purpose. It corresponds to known condition between the minimal integration step and the maximal considered frequency $f_{\max}$ of the undamped system as it is recommended in [11] or [15] as well

$$\Delta t_{\min} = \frac{1}{\pi f_{\max}}. \tag{18}$$

In case of the spring stiffness change during impacts the integration step has to be cut down several times. It is necessary to get suitable results.

The main parameters of the parts under vibrations are given in the Tables II, III and IV.

TABLE II
HULL PARAMETERS

Symbol	Quantity	Value
m_{ko}	hull mass	12 538 kg
I_{ko}	hull mass moment of inertia	22 544 kg.m^2
k_1	suspension stiffness on the front side of hull	4.56 x 10^5 N/m
k_2	suspension stiffness on the rear side of hull	4.56 x 10^5 N/m
a_1	distance of the front hull spring from to gravity centre	1.174 m
a_2	distance of the rear hull spring from to gravity centre	1.174 m
b_{Θ}	dumping coefficient of angular motion	7.41 x 10^4 N.m.s/rad
b_{yko}	dumping coefficient of linear motion	4.43 x 10^4 N.s/m

The parameters in Table II have been determined from the technical documentation (m_{ko}, k_1, k_2) where suspension stiffnesses consist of the torsion bar springs and their values have been linearized in the operating point. The reduced distances a_1, a_2 have been calculated from the six sprung wheels by the torsion bars on the both sides. The damping coefficients b_{Θ}, b_{yko} have been settled by measurement of the real weapon during the fire and the movement.

The parameters in Table III and Table IV have been set by the same way.

TABLE III
TURRET PARAMETERS

Symbol	Quantity	Value
m_{V}	turret mass	1 920 kg
I_{V}	turret mass moment of inertia	752 kg.m^2
k_3	spring stiffness on the front side of turret	8.5 x 10^6 N/m
k_4	spring stiffness on the rear side of turret	8.5 x 10^6 N/m
k_5	spring stiffness in the horizontal direction of turret	0 ÷ 5 x 10^7 N/m
$b_{\gamma_{\mathrm{V}}}$	dumping coefficient of angular motion	1.63 x 10^4 N.m.s/rad
$b_{x_{\mathrm{V}}}, b_{y_{\mathrm{V}}}$	dumping coefficients of linear motion	5 x 10^4 N.s/m

TABLE IV
ELEVATION PART PARAMETERS

Symbol	Quantity	Value
m_{E}	elevating parts mass	668 kg
I_{E}	elevating parts mass moment of inertia	350 kg.m^2
k_6	spring stiffness of trunnion in vertical direction	0 ÷ 4 x 10^8 N/m
k_7	spring stiffness of elevating gear element	8.5 x 10^6 N/m
k_8	spring stiffness of trunnion in horizontal direction	0 ÷ 5 x 10^7 N/m
$b_{\alpha_{\mathrm{E}}}$	dumping coefficient of angular motion	1.63 x 10^4 N.m.s/rad
$b_{x_{\mathrm{E}}}, b_{y_{\mathrm{E}}}$	dumping coefficients of linear motion	4.43 x 10^4 N.s/m

The exciting force which was determined from the measuring is portrayed in Fig. 5. It is necessary to remind that the resultant force in Fig. 5 is put together from two 30 mm coupled cannons burst firing. They have the advanced primer ignition. It has been determined measuring by means of strain gauges where weapon casings are fixed to the cradle. The sampling rate has been 615 Hz which matches to the time between samples 1.625 ms. The force is being determined according to the operation of the automatic weapon. In [1] there is reminded that the force transmitted to the mount has a maximum value for the first shot fired. For the second shot there is a reduction in the firing force which is still further reduced for the third shot. It can be seen from the functional diagram in [1] that the maximum force applied to the mount occurs at the instant that the barrel is arrested and the breech carrier begins to act on the buffer. The spectral density variation of the firing force acting on the mount has shown that the basic frequency is given by the rate of fire. The force magnitude belonging to the basic frequency is much greater than other higher harmonics components. The impulse of the force acting on the mount is the same as the impulse of the firing force, which allows quick and simple calculations of the force applied to the mount to be made.

In this case study has been considered the gas operated automatic cannon, see [7], [8], having recoiling barrel and triggering when barrel moves into front position before every shot (the system is known as soft recoil system), see [17]. In addition the buffers damping impacts of the breech block carrier and the barrel in the rear and front positions have been used see [9]. The calculation of this weapon tends to be somewhat complicated and then the use of the measured data seems to be more suitable. Afterwards it is possible to proceed to the simplification of the exciting force when the main condition – the whole impulse of the exciting force during one shot equals to the impulse of the shot force causing by the gas pressure in the barrel – will be filled.

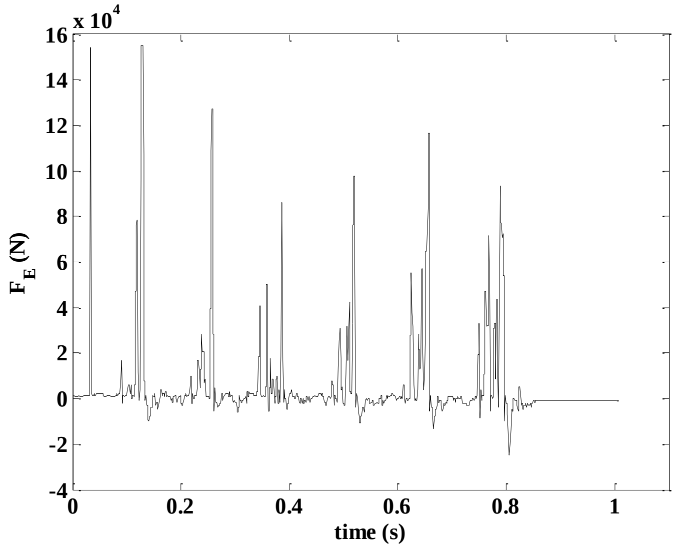

Fig. 5 Measured excitation force

Then the F_{E} force has been approximated by the analytical formula preserving the impulse of the shot force, as it is explained in [4], [5] or [6] and pictured in the Fig. 6 as well:

$$F_{\mathrm{E}} = F_{\mathrm{SS}} \left[1 + \sin\left(\omega_{\mathrm{E}} t + \psi\right)\right], \tag{19}$$

where

F_{SS} - direct component of the exciting force determined from

$$F_{\mathrm{SS}} = \frac{I_{\mathrm{H}}}{t_{\mathrm{FC}}}, \tag{20}$$

where

I_{H} - impulse of shot force depending on the ballistic properties of the round,

t_{FC} - time of the functional cycle.

The impulse of shot force is possible to calculate as the integral from the shot beginning to the t_{K} shot final time:

$$I_{\mathrm{H}} = \int_0^{t_{\mathrm{K}}} F_{\mathrm{H}}\, \mathrm{d}t = (m_{\mathrm{q}} + \beta m_{\omega}) v_0, \tag{21}$$

where

m_{q} - projectile mass,

m_{ω} - charge mass,

v_0 - initial velocity,

β - aftereffect coefficient depending on the relations among parameters describing the gas flow from the barrel when projectile leaves the barrel muzzle.

The main excitation frequency and the angular frequency are given using the following equations

$$f_{\mathrm{E}} = \frac{1}{t_{\mathrm{FC}}}, \text{ and}$$
$$\omega_{\mathrm{E}} = 2\pi f_{\mathrm{E}}. \tag{22}$$

The phase shift ψ in (19) enables to solve tasks when a few weapons are mounted (multiple mounting of weapons) one platform and firing with different initial time and with different characteristics as are the F_{SS} force, the f_{E} excitation frequency, the I_{H} impulse etc. When one weapon fires then $\psi = -90°$.

Then the resultant excitation force is given as follows

$$F_{\mathrm{E}} = \sum_{\mathrm{j}=1}^{\mathrm{n}} F_{\mathrm{E_j}}, \tag{23}$$

where j – number of firing weapons.

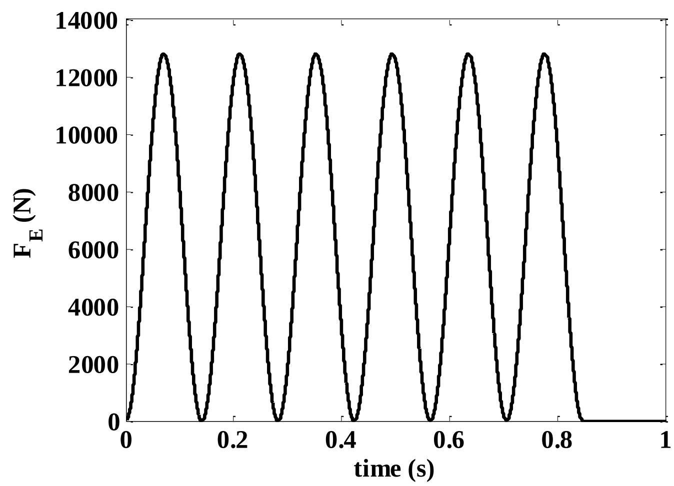

Fig. 6 Approximated excitation force used in calculations

It is possible to draw an inference from checking calculations that the hull behaves as a low frequency filter uniformly transmitting signals up to the specific cut off frequency. The range is known as a band pass where for the cut off frequency is valid the formulae, see [2]:

$$f_{\mathrm{m}} = \frac{1}{2T_{\mathrm{n}}}, \tag{24}$$

where

T_{n} – the rise time given with the tangent line in the point where the output (hull angular displacement) achieves the half of the steady state value.

In this case there is approximately 1.56 Hz (hull eigenfrequency f_{Θ} = 1.1 Hz).

The angular displacement/time history of the hull after 10 shots represents the Fig. 7. After three seconds when vibrations of the weapon system are damped the hull is in the basic position. The influence of the clearance (in our case study 5 mm) in the turret ball path on the hull vibration does not the considerable difference.

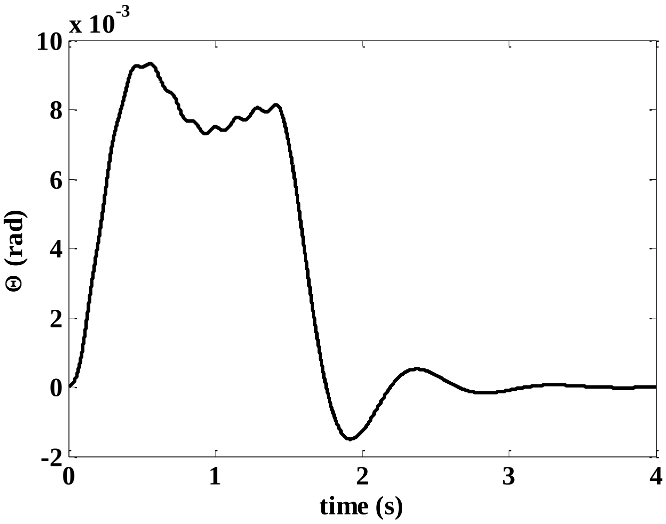

Fig. 7 Hull angular displacement

On the other hand at the beginning of the turret motion several impacts of the hull and the turret occur as it is pictured in Fig. 8 where angular vibrations of the turret with respect to the hull are. After the transient period when the clearance is taken-up the turret moves with regular frequency.

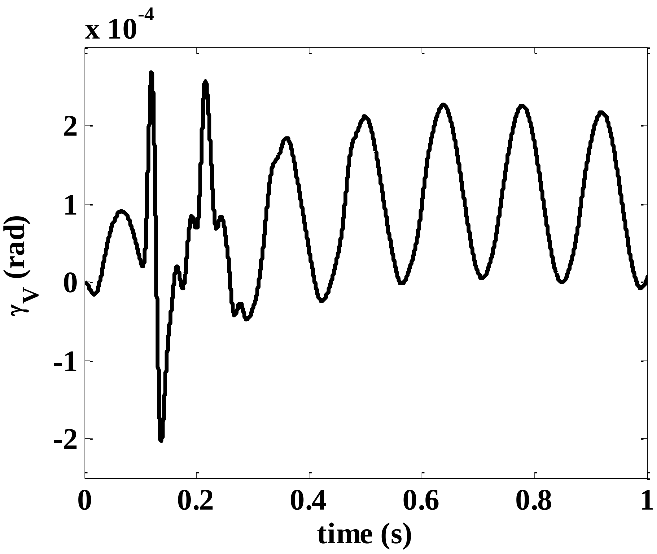

Fig. 8 Turret angular displacement with respect to the hull

The case of the turret vibration in reference to the hull, when the clearance in the turret seating is zero, is displayed in Fig. 9 and it is similar to the previous event with 5 mm clearance.

The elevating parts vibration (absolute motion), see Fig. 10, during ten shots gives very important outcome that the main influence on the aiming errors is cased with the hull as it is possible to observe comparing with the Fig. 7. The relative variances of the elevating parts vibrations and the turret or the elevating parts and the hull are less than 1 mrad.

The influence of the other operation type of weapon mounted on the track vehicle will be explained on the example of the Gatling weapon. The main difference during exciting of weapon and its mounting is increasing of the rate of fire during acceleration time from zero until the nominal value, see [8].

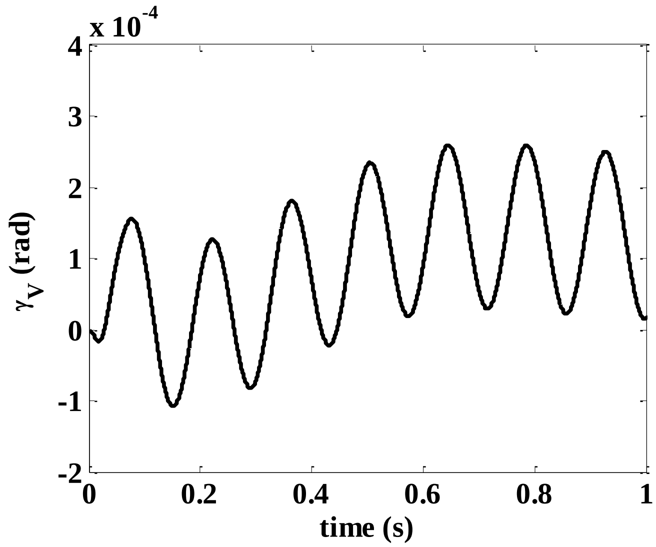

Fig. 9 Turret angular displacement with respect to the hull

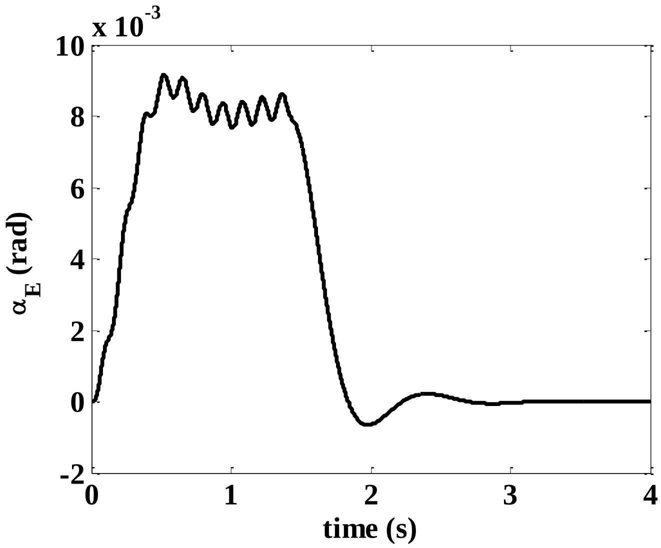

Fig. 10 Elevating parts angular displacement

The principle of his system consists of a joining several barrels in a group rotating about a longitudinal axis in a fixed weapon casing, see Fig.11. Each of the barrels must complete one revolution of the barrel group to complete the functional cycle. The cycles of the different barrels are displaced by one barrel pitch, which is through an angle of $\frac{2\pi}{n}$ radians (n being the number of barrels in the group). The interval between shots corresponds to the time taken for the group to turn through such an angle. After pressing the trigger the gun begins rotating until it reaches the operational rotational velocity.

The breech blocks move axially along a guiding groove in the weapon casing. During one revolution of the barrel group each breech block moves backwards and forwards in turn with delays at the front locked position for firing and at the rear position for case ejection and feeding the next cartridge. Breech block movement is controlled by a cam formed as a groove in the inner surface of the weapon casing. Each of the breech blocks is provided with a guide with a roller which moves along the groove of the cam as shown in Fig.11. When the barrel group rotates the breech casing remains stationary. The shape of the functional groove illustrated in the plane with positions of all breeches for a Gatling weapon with four barrels (and four breeches) is shown in Fig. 11 right.

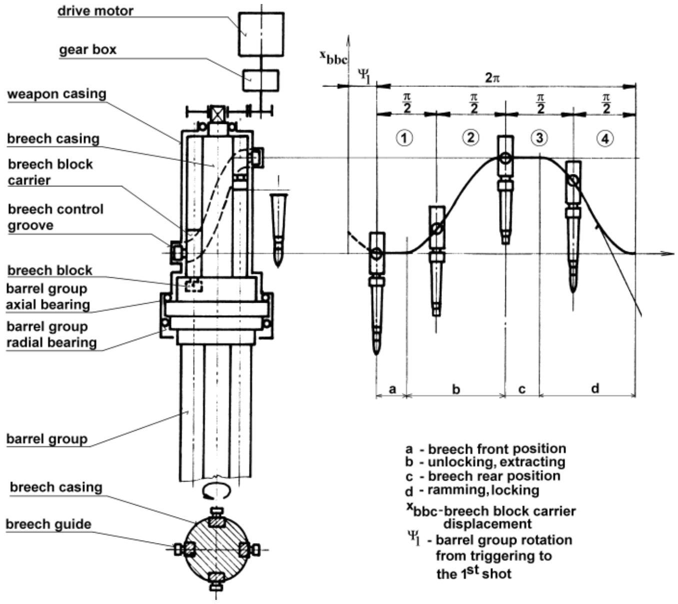

Fig. 11 Principle of Gatling gun

The weapon parts are accelerated depending on the weapon calibre and usually for the 30 mm weapons the time is 0.5 s as it is shown in Fig. 12.

The change of the excitation force frequency in Gatling weapon represented in Fig. 13 is similar to the chirp signal used in the electrical engineering. The F_{SS} direct component increases due to the time shortening of the t_{FC} functional cycle according to the equations (20) and (21). Finally the hull and elevation parts angular displacement are higher, see Fig. 14.

On the other hand, the dynamic characteristics stay same and hull and elevation parts behaviour change only in the magnitude not in the time history where hull preserves the properties of the low frequency filter. When the special springs or gas shock absorbers are used for automatic weapons mounting then the excitation force can be replaced with the resultant force composed of the shock absorber force and friction force between the weapon casing and the cradle as one of the supporting structure of the weapon mounting.

The characteristics of a shock absorber can influence the functioning of the automatic system above all if the shock absorber used with a gas operated weapon. If the shock absorber has a functional time that is shorter than the time for one functional cycle (t_{FC}) then:

- The rearward breech block velocity increases because of the effect of the initial velocity of the whole weapon up to the beginning of the gas take-off.
- The absolute displacement of the breech block carrier increases because of the whole displacement of the weapon.

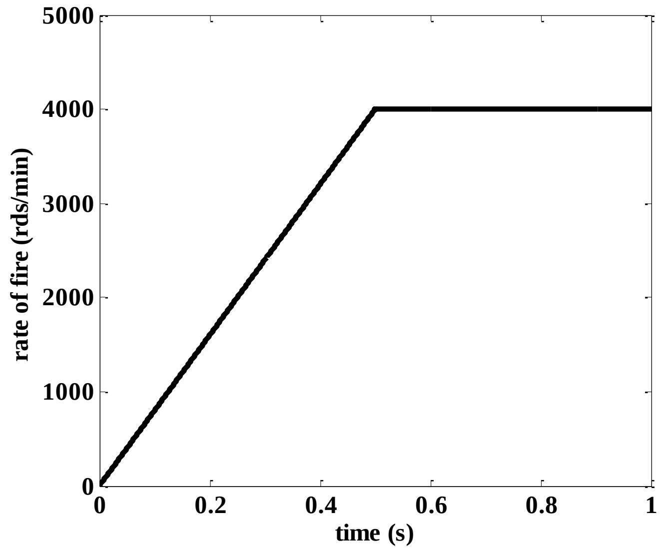

Fig. 12 Acceleration of the Gatling weapon when fires

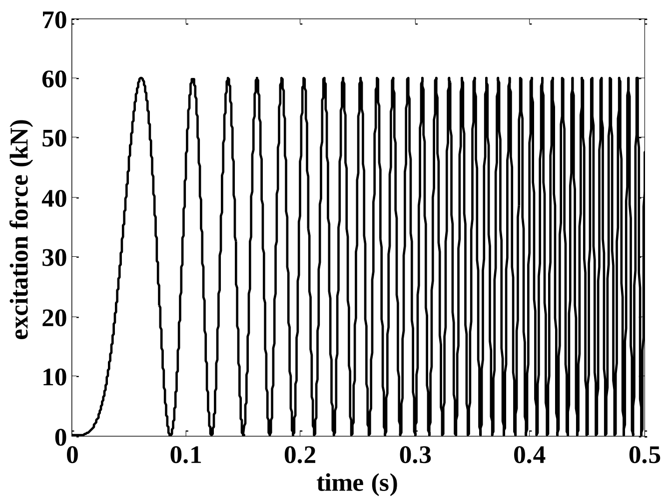

Fig. 13 Excitation force during acceleration of weapon

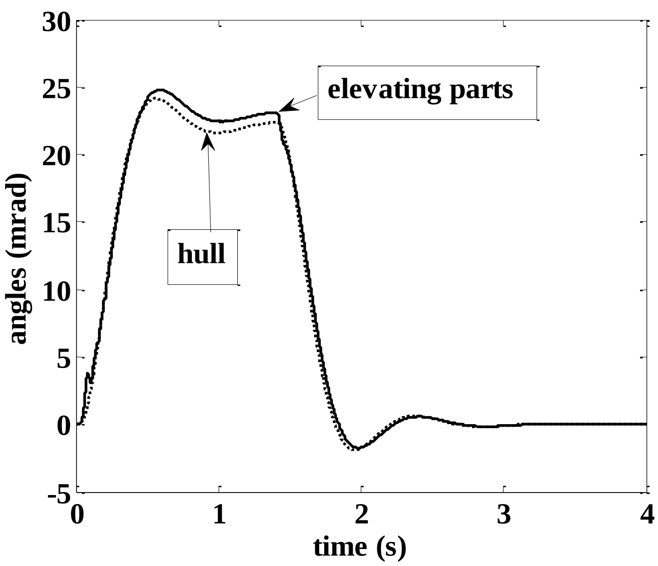

Fig. 14 Vibration of weapon system parts with Gatling weapon

- The ratio of the breech block carrier velocity before and after impact varies.

- The functional cycle time may be shorter and therefore the rate of fire may be higher if the breech block impacts the buffer when the weapon is moving foreword.

Shock absorbers used with recoil-operated weapons reduce the rate of fire because of the reduction in the counter-recoil velocity of the barrel and the breech block.

The shock absorbers parameters influence the force acting on to weapon mounting and therefore the behaviour of the weapon parts when fire.

The sensitive analysis, whose principle is described for example in [4], has been used for the changes evaluation of the parameters including in the dynamic model in Fig. 4 and equations (2). The simplified scheme of the procedure is given in Fig. 15. The change of the j^{et} parameter in the dynamic model causes the change of the absolute angular displacement of the elevation parts at the moment when projectile leaves the muzzle of the barrel in the ith shot. At the moment the projectile obtains the final direction in the vertical plane caused with the vibrations of weapon parts having the influence on the fire precision.

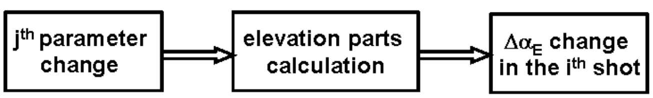

Fig. 15 Scheme of the sensitive analysis

It is necessary to know the change range of all parameters included in analysed system. The first group contains parameters varying significantly in the prolonged period. In case of the fire of the automatic weapon mounting on the vehicle they are: the centre gravity position of the hull, turret and elevation parts, mass of the hull, turret and elevation parts, the mass moments of inertia of the hull, turret and elevation parts, hull suspension stiffness.

The second group has parameters varying considerably and accidentally, in every shot. The following model parameters belong to this section: all damping coefficients of motions included in equation (8), nonlinear stiffnesses k_3, k_4, k_5, k_6, k_7, and k_8 modelling the clearances in the mounting of the turret and elevation parts. The forces acting on the weapon mounting during burst fire can vary considerable due to the rate of fire change, in case of malfunction of some weapon when the multiple mounting is used. When the main ballistic characteristics (due to the mass charge and mass projectile variation) are different in round then the initial velocity of the projectile and then the maximal force and impulse of the force of shot change as it is given according to (20). Only the variation of the mounting characteristics will be discussed now.

The sensitivity of the fluctuation of the I_{ko} hull mass moment of inertia and the k_1, k_2 springs do not have essential effect on the $\Delta\alpha_E$ angle change during operation. On the other hand the very important result is that mounting of automatic weapons on the tank hull improves the firing stability characteristics by reduction of the hull and elevation parts maximal angular displacement which is useful for the hit probability increasing as it is demonstrated in Fig. 1 and Fig. 2. The next profit is decreasing of the maximal accelerations of the turret and elevating parts where the important devices are located (aiming and sigh device, elevating and traversing gear, loading system etc.). Decreasing of $b_{y_{ko}}$ and b_Θ dumping coefficients to halves increases the $\Delta\alpha_E$ angle to 30 %.

The influence of the other turret and elevating parts parameters as stiffnesses k_3, k_4, k_5, k_6, k_7, and k_8 have neglectable effect on the hull behaviour.

One of the most frequent variations which can occur during field operation is the enhancement of the radial clearance in the turret bearing. Since the real value has not been known only the comparative calculation has been made with respect to the instant when the clearance is zero. The results of the calculations for events when clearances are V5 = 0.5 mm, V5 = 2 mm and V5 = 3 mm are demonstrated in Fig. 15 during six shots firing from two coupled automatic cannons at the elevation angle 0°. The results confirmed that at the end of the transient effect also higher clearance has small impact on the disturbance of the elevation angle α_E and its change $\Delta\alpha_E$.

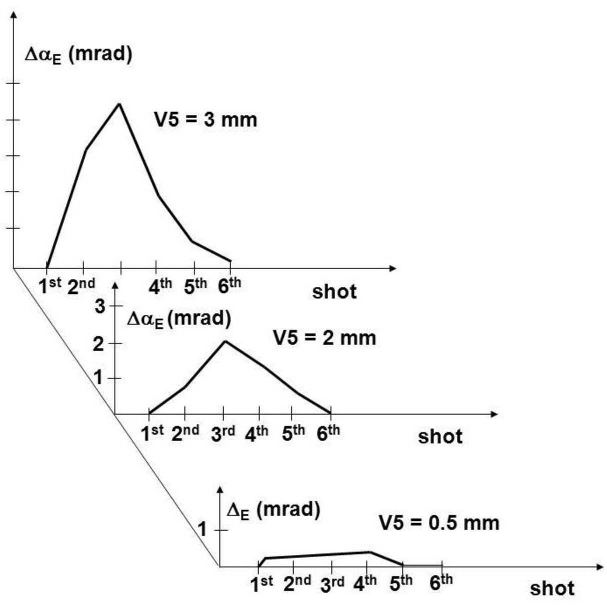

Fig. 16 Influence of the clearance in turret bearing

In case of fire in the high elevation, for example at 75°, the V5 clearance influence is smaller. The variation of the $\Delta\alpha_E$ angle achieves the value to 1.5 mrad.

Similar situation is being in case of the radial clearance in the trunnion. The difference is in the size of the error aiming when weapon fires in the high elevation angles. Then the errors are half as compared with the fire at the zero elevation angles.

The angular clearance in the elevation gear can achieve several mrad. The courses of the aiming errors $\Delta\alpha_E$ for the clearances 1 mrad and 3 mrad represented in Fig. 17 show that the elevation parts oscillate along new position during whole burst fire. It is the main difference between the radial clearance and the angular clearance in weapon mounting. The clearances in the elevation gear cause high dynamic load both trunnion and the end couple of the elevation gear.

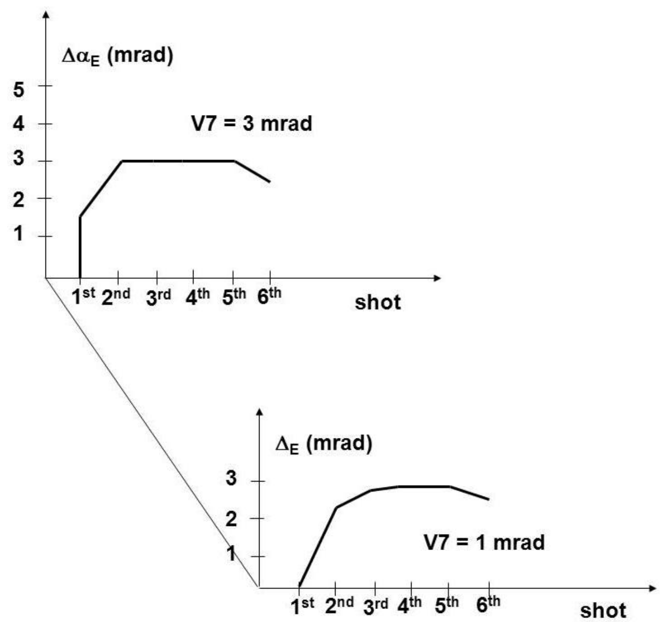

Fig. 17 Influence of the angular clearance in elevation gear

IV. CONCLUSION

The dynamic model in Fig. 4 enables to determine the effect of the parameters used there mainly their changes in course of the operation and in the service.

The results given in the figures reflect a good coincidence with the real piece which was explored according to presented theory. The theory was verified on the two coupled 30 mm automatic cannon. The success of the analysis depends on the experimental verification of the other parameters as are clearances in seating etc., see [12], [23].

Using the equation (19) it is possible to solve the different excitation of the system when the operation system of the automatic weapon is changed.

It has been proved that the main influence on the aiming errors have the hull vibrations. It makes possible to simplify the dynamic model to the two or three DOF, see [21], [22].

In future it is supposed to study the influence of the change operation of the elevation and azimuth drive, the influence of the move on the ground when firing and the changes of the rate of fire. The outputs of this work will be used in application of the theory on the solving of the problems connecting with the loading devices of guns evaluation which are operated in course of the fire and on the move. The dynamic loads from vibration of the weapon parts very embarrass the loading processes as feeding and ramming of the rounds.

The concluding step will work out the three-dimensional dynamic model involving among others the bearing angle and weapon mounting on the wheel combat vehicles, see [20].

The outcomes can be used for the frequency system tuning in order to the main eigenfrequencies of the system will be among the main harmonic frequency of the excitation force as it is explained in Fig. 18. As it is evident from the figure the hull, turret and elevating parts eigenfrequency are set among the main harmonic frequency of the excitation force $F_{E1}, F_{E2}, F_{E3}, F_{E4}$.

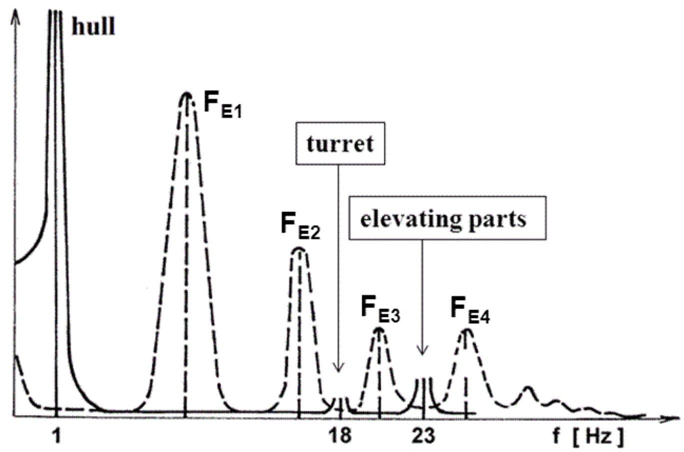

Fig. 18 Distribution of frequencies in main parts of system

The published theory has been applied in the Czech research institutes and in the University of Defence in Brno as an additional study material for students.

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