Dynamics of shell conveyer with Maltese cross

Jiri Balla, Van Yen Duong, Zbynek Krist

Abstract**—**The paper deals with one weapon application of the Maltese cross in the shell or cartridge conveyer using a hydraulic drive. Kinematics and dynamics of this mechanism is discussed. The first model with one DOF and the second model having two DOF are worked out. Both models use mechanism with varying transmission function. The equations describing the mechanical and hydraulic part are explained.

Keywords**—** Maltese cross, Shell conveyer, Hydraulic motor, Transmission function, Mechanism with varying transmission function.

I. INTRODUCTION

BEFORE the shot it is necessary to load the cartridge into the cartridge chamber of the barrel. This loading influences the rapidity of fire of guns and it is also one of the most difficult operations, see [2], [3], [4], and [7]. This difficulty results mainly from:

- great weight and length of the artillery cartridges,
- great length of the displacement at the cartridge ramming,
- necessity to ram the cartridge into the barrel by the velocity ensuring the pressing of the shell ring into the forcing cone,
- compression of ejectors spring,
- storing of the ammunition far from the elevating parts,
- necessity of the short time of ramming,
- complicated working condition of the crew at towed guns and especially for self-propelled guns.

Therefore it is very difficult to ram the artillery cartridges into the barrel by hands of members of the gun crew and its mechanization is useful.

Let us explain main parts of the loading system for the separated ammunition, because in comparison with the fixed ammunition it is more complicated. Such a heavy gun loading system in Fig. 1, see [4], [5], [11], [12], and [22], consists of:

- shell storage system,
- case storage system,
- shell feeding device,
- case feeding device,
- ramming device,
- device for removing of cartridge cases from the loading system,
- control system.

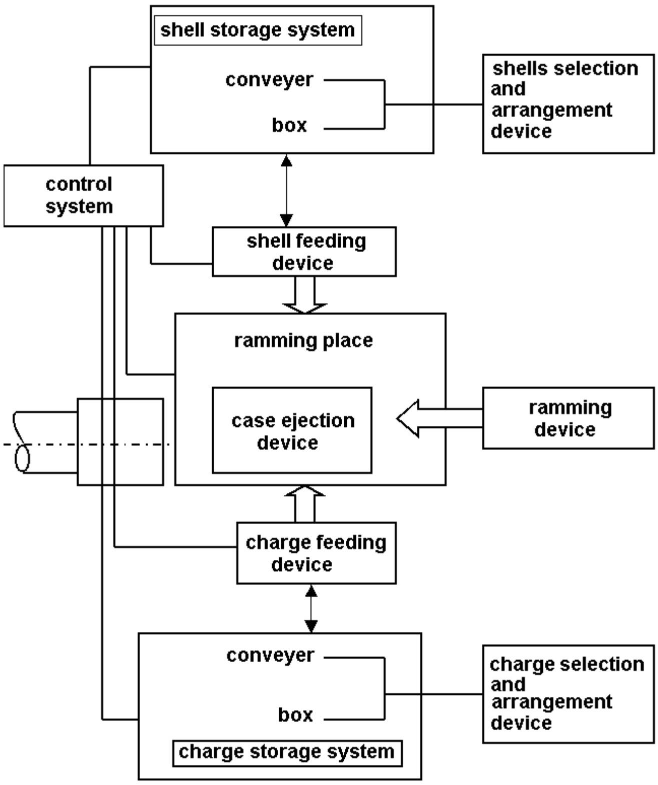

Fig. 1 Loading system of heavy gun scheme

Two storage systems are used in the loading system for separate ammunition: for shells and for cases. Each system includes the magazines (the box for shells and the box for cases with propellant charge) and the conveyers (also for two components of the cartridge). Modern self-propelled guns use the placing of the ammunition in the rotating traverse parts mostly.

In the Czech 152mm self-propelled cannon-howitzer M77 there are both the shells and the cases placed vertically in four rows conveyers in the left and right cabin mounted on rotating parts of the weapon, see Fig. 2. The capacity of conveyers satisfies the firing of 30 rounds without completing of ammunition. Each component of the cartridge has a bed and a

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fixation at its disposal. Individual beds are connected into the belt by means of chains. Their guiding is provided with rollers to decrease the resistance against the motion, see [7], and [22].

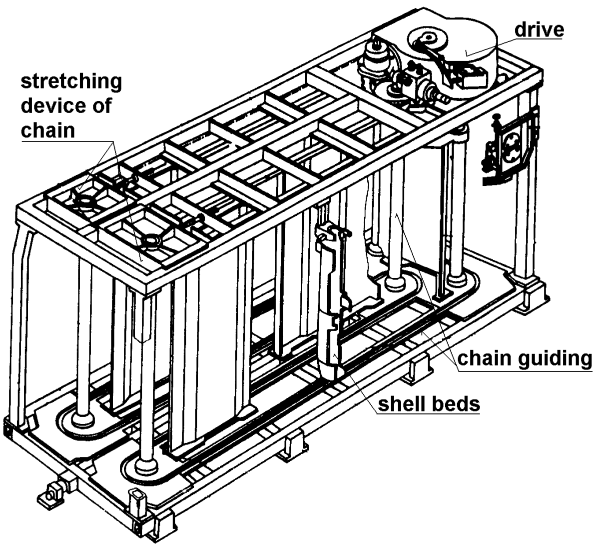

Fig. 2 Shell conveyer

II. Problem Formulation

The drive consists of the MA-2 rotary hydraulic motor and the gearbox connected with driven chains via chain wheels. The kinematical scheme is represented in Fig. 3, see [7], and [17]. An emergency drive by hand (with angular velocity ω_6) is possible when the hydraulic drive does not operate. This shell conveyer is interesting with an application of the Maltese cross in the gearbox that represents a typical mechanism with varying transmission ratio i_3, variable reduced mass moment of inertia and variable static workload. It makes different behavior of the whole system with respect to the conveyers having the constant transmission. Varying transmission rations are used between driving and driven elements in the other mechanisms as are the crank mechanisms, slotted-link mechanisms and the mechanisms in the automatic weapons. The varying transmissions reduce the accelerating and decelerating forces. It is not possible to employ the commonly used equations of motion of the form “force = mass x acceleration”, and it is necessary to perform a reduction of the forces and masses. Displacement, velocity, and acceleration of a point on the element of a mechanism can be determined by the equations of motion if the forces acting on the different elements are known. The reverse of this can be carried out whereby the forces acting on the different points of the element can be determined from given values of velocity and acceleration. Kinematic and dynamic analysis of the operating mechanism of a weapon part is not only used in the design of the weapon part, but also as a method of understanding the function of the mechanism and to find the critical points in it. The basic of design consists, especially when computer-aided design is used, of the development and solution of the equations of motion of the main elements of the mechanism. Many mechanisms in weapons rely on lever and cam mechanisms to control the relative movement of their components. The shape of the control curve is of crucial importance for the correct operation of the weapon, especially between the main functional element (the driving element) and the secondary element (the driven element). The high velocities and accelerations of the driving element make it necessary to avoid any sudden changes of motion of the driven element. There is an instantaneous ratio of velocity between the driving element and the driven element, which is called the transmission ratio. The expression of dependence of the change in transmission ratio with respect to the displacement of the driving element is known as the transmission function. The shape of this function describes the quality of the mechanism. For continuous motion of the driven element and to achieve a high rapidity or rate of fire it is necessary to ensure that the transmission function is continuous and without any sudden changes. The shape of the acceleration and deceleration curves is particularly important. To avoid impacts, high stresses and energy losses in the mechanism, it is necessary to pay particular attention to transitions from an acceleration phase to a deceleration phase or into a phase or into a phase of uniform motion. The Maltese cross is a typical example of these mechanisms. It ensures an intermittent motion in two phases.

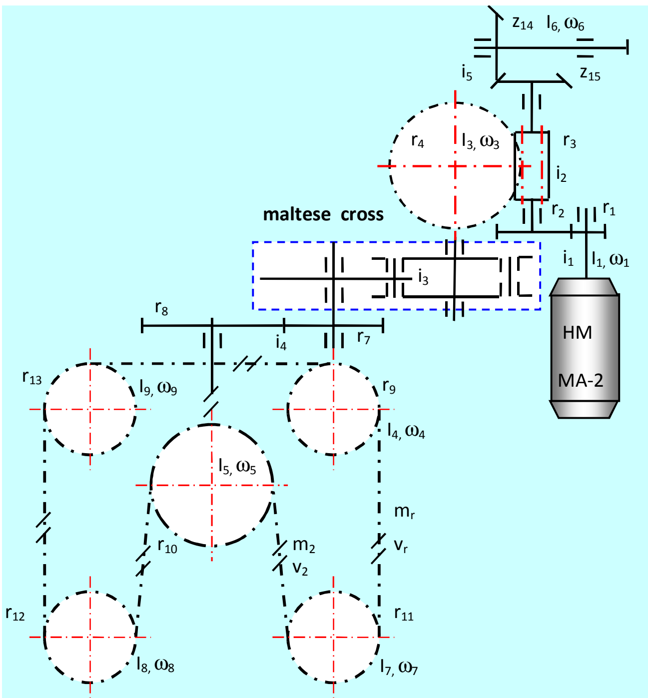

Fig. 3 Kinematic scheme of shell conveyer

The Fig. 4 shows two positions of the Maltese cross - during meshing and no meshing. The six-arm cross is connected with the driver having two rollers meshing in the Maltese cross slots.

The kinematical relations in the Maltese cross are apparent from scheme in Fig. 5.

The Maltese cross is being replaced to a slotted-link mechanism, see [9], [18], and [20] in course of cross meshing with a driver. The distance between the carrier rotation axis and the Maltese cross rotation axis is constant value and is the length from the center roller meshing with the cross to the driver axis rotation.

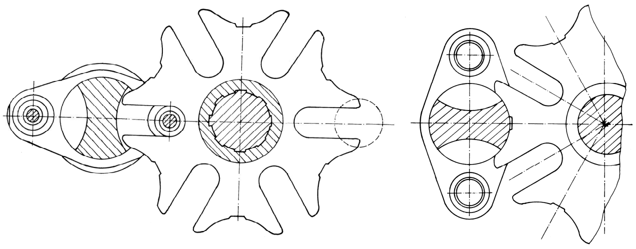

Fig. 4 Maltese cross in and without meshing

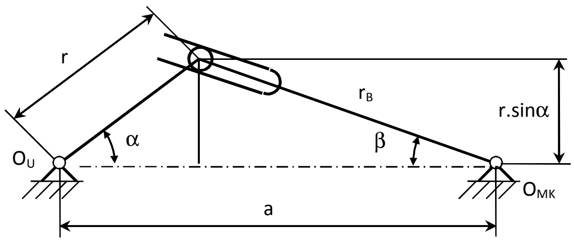

Fig. 5 Kinematics of Maltese cross scheme

Then we can write the relation between angles α (the driver turning) and β (the Maltese cross steering angle) there is known as the control curve, Fig. 6 as well,

$$tg\beta = \frac{r\sin\alpha}{a - r\cos\alpha}. \tag{1}$$

To achieve correct evaluation of the control curve and the quality of the transmission function it is not only the continuity and acceptability of the control curve in terms of its first order derivative which is important, but also the second order derivative, with respect to time and displacement, because β is function of the α and of the time t.

The transmission ratio of Maltese cross, see Fig. 7, is

$$i_3 = \frac{\mu\cos\alpha - 1}{\mu^2 - 2\mu\cos\alpha + 1}, \tag{2}$$

where

$$\mu = \frac{a}{r}.$$

Therefore the angular velocity of the Maltese cross is

$$\dot{\beta} = \dot{\alpha}\, i_3, \tag{3}$$

where $\dot{\alpha}$ is driver angular velocity.

The first derivation of the transmission ratio (4) is necessary to determine because the acceleration of the Maltese cross depends on it as well, see next formulae (5) and Fig. 8,

$$i_3' = \frac{\left[\mu\sin\alpha\left(1-\mu^2\right)\right]}{\left(\mu^2 - 2\mu\cos\alpha + 1\right)^2}. \tag{4}$$

The angular acceleration of the Maltese cross is hence

$$\ddot{\beta} = i_3\ddot{\alpha} + \dot{\alpha}^2 i_3', \tag{5}$$

where $\ddot{\alpha}$ is driver angular acceleration.

Then the velocity and acceleration of the shell conveyer may be determined with respect to the total constant transmission ratio i_{const}

$$v_{conv} = i_3 i_{const}\dot{\varphi}_1, \tag{6}$$

and

$$a_{conv} = i_3'\left(i_{const}\dot{\varphi}_1\right)^2 + i_3 i_{const}\ddot{\varphi}_1, \tag{7}$$

where

$\dot{\varphi}_1$, $\ddot{\varphi}_1$ – angular velocity and angular acceleration of the hydraulic motor.

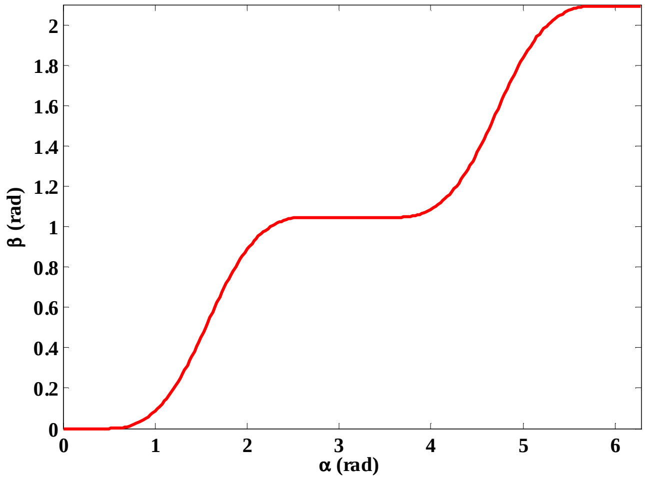

Fig. 6 Maltese cross angular displacement

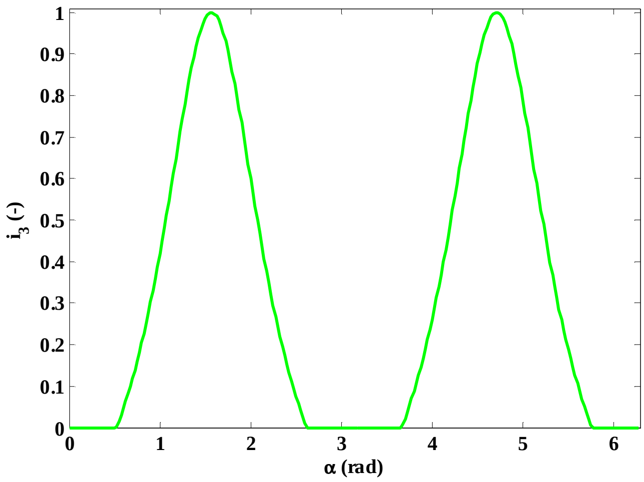

Fig. 7 Transmission ratio function

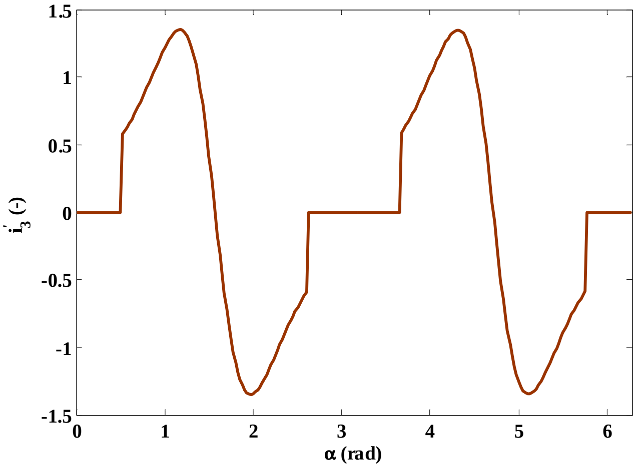

Fig. 8 The first derivative of transmission ratio function

After the kinematical analysis we can approach to the dynamic solution of the drive. Rigidity and damping effects are small, so the equation of motion is described with the following formulae, see [19], [21], [25], [26], and [27]:

$$I_M \ddot{\varphi}_1 + 0.5\dot{\varphi}_1^2 \frac{dI_M}{d\varphi_1} = M_M - M_Z - M_D, \tag{8}$$

where

I_M – reduced mass moment of inertia of the whole system,

φ_1 – angular displacement of the hydraulic motor shaft,

M_M – driving torque,

M_Z – reduced moment of workload,

M_D – damping moment.

The mass moment of inertia of the whole system is determined from the system kinetic energy, see [6], [8], [23], and [24]:

$$E_K = 0.5\omega_1^2 I_M, \tag{9}$$

$$I_M = \left(\begin{array}{l} I_1 + I_2 \dfrac{\omega_2^2}{\omega_1^2} + I_3 \dfrac{\omega_3^2}{\omega_1^2} + I_4 \dfrac{\omega_4^2}{\omega_1^2} + I_5 \dfrac{\omega_5^2}{\omega_1^2} + \\ + I_6 \dfrac{\omega_6^2}{\omega_1^2} + I_7 \dfrac{\omega_7^2}{\omega_1^2} + I_8 \dfrac{\omega_8^2}{\omega_1^2} + I_9 \dfrac{\omega_9^2}{\omega_1^2} + m_r \dfrac{v_r^2}{\omega_1^2} \end{array} \right) \tag{10}$$

Where

I_i – shaft mass moment of inertia with its gear wheel or wheels,

m_r – mass of chain with shells and beds for shells,

v_r – chain velocity (i.e. conveyer velocity).

The Maltese cross transmission ratio to the angular displacement of the hydraulic motor shaft is written with formulae $i_3 = \dfrac{\omega_4}{\omega_1}$. It means that the mass moment of inertia I_M can be divided onto the constant part I_{M1} and the variable part I_{M2}. The variability of the I_{M2} mass moment of inertia is caused with the variable transmission ratio i_3.

Then after arrangements and putting outside bracket i_3^2 we can write (11) and plot in Fig. 9

$$I_M = I_{M1} + i_3^2 I_{M2}. \tag{11}$$

The mass moment of inertia is depending on it if the conveyer is full or empty, see Fig. 9. Its value for full conveyer is four times greater than the conveyer without the shells, for example.

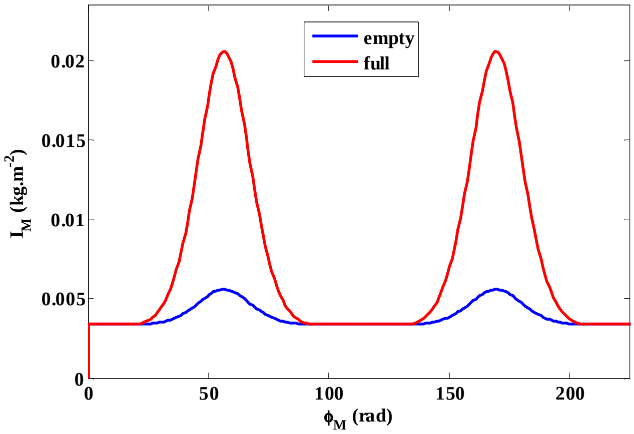

Fig. 9 Inertia mass moment of system I_M

After the first derivative with respect to φ_1 we get

$$\frac{dI_M}{d\varphi_1} = 2I_{M2} i_3 \frac{di_3}{d\varphi_3} i_{13}, \tag{12}$$

where

i_{13} – constant transmission ratio between the Maltese cross and the hydraulic motor shaft.

The hydraulic motor driving torque is given, see [4], and [24],

$$M_M = \frac{V_G}{2\pi}(p_1 - p_2), \tag{13}$$

where

V_G – the geometrical volume of the hydraulic motor.

The hydraulic equations for determining input and output pressures can be introduced in a way following from the hydraulic circuit in Fig. 10, as well, see [7]:

$$\frac{dp_1}{dt} = \frac{(Q_1 - \frac{V_g}{2\pi}\dot{\varphi}_M - Z_1 p_1)}{C_1}, \tag{14}$$

$$\frac{dp_2}{dt} = \frac{(\frac{V_g}{2\pi}\dot{\varphi}_M - Q_2 - Z_2 p_2)}{C_2}, \tag{15}$$

$$Q_1 = \mathrm{sgn}(p_G - p_1)\sqrt{\frac{|p_G - p_1|}{R_1}}, \tag{16}$$

$$Q_2 = \mathrm{sgn}(p_2 - p_0)\sqrt{\frac{|p_2 - p_0|}{R_2}}\,. \qquad (17)$$

The parameters used in equations above are:

Q_1 – input flow of hydraulic motor,

Q_2 – output flow of hydraulic motor,

$C_1 = \beta_L V_{01}$ – input hydraulic capacity,

$C_2 = \beta_L V_{02}$ – output hydraulic capacity,

V_{01} – input liquid volume in the pipe, leading from distributor to hydraulic motor,

V_{02} – output liquid volume in the pipe, leading from hydraulic motor to distributor,

β_L – liquid volume compressibility factor set $6.8\cdot10^{-10}$ Pa^{-1},

R_1 – input hydraulic resistance,

R_2 – output hydraulic resistance,

p_0 – waste pressure taken 0.6 MPa,

p_G – source pressure, given as

$$p_G = 4.6 + 0.005\sin(30\pi t)\ \text{[MPa]}. \qquad (18)$$

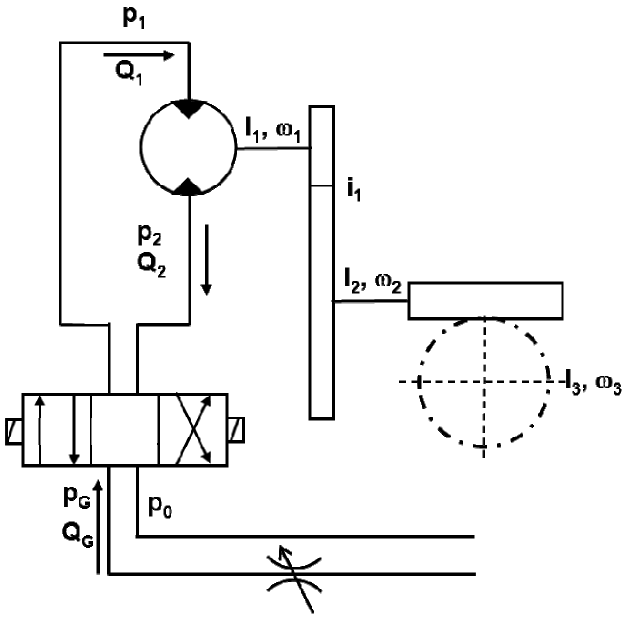

Fig. 10 Hydraulic drive of shell conveyer

The source pressure has been considered stable on the 4.6 MPa level, how it is supposed for the new hydraulic pump enabling to change both the pressure and the flow. It is an opposite case than it was calculated in [7]. Damping coefficient $b_D = 0.036$ N·m·s·rad^1 has been determined by the measurement in the course of steady-state motion of the system under off-load conditions, see [7]. The damping moment of the system (reduced value) is

$$M_D = b_D\omega_1\,. \qquad (19)$$

The hydraulic resistances R_1 ($2.8\cdot10^{13}$ Pa·s^2·m^{-6}) and R_2 ($17.35\cdot10^{12}$ Pa·s^2·m^{-6}) have been chosen from measured pressures and angular velocity of the hydraulic motor output shaft. These values of hydraulic resistances have been considered when the electro-hydraulic distributor is fully closed. The opening time has been considered 40ms. Afterwards the hydraulic resistances have been $R_1 = 9.3\cdot10^{12}$ Pa·s^2·m^{-6} and $R_2 = 2.45\cdot10^{12}$ Pa·s^2·m^{-6}. The linear drop of the hydraulic resistances has been chosen with respect to the opening time of the electrohydraulic distributor.

Both hydraulic losses Z_1, Z_2 have been used in the same values $1\cdot10^{-12}$ m^3·s^{-1}·Pa^{-1} from experiences and comparing with similar hydraulic motors.

The comparing calculations have been carried out using the following input values of the hydraulic capacities, by way published for example in [4], and [7]:

$C_1 = 8\cdot10^{-12}$ m^3·Pa^{-1}, and $C_2 = 6\cdot10^{-12}$ m^3·Pa^{-1}.

The varying reduced moment of workload has been determined from the equation (20) which is plotted in Fig. 11

$$M_Z = F_N f r_{10} i_\Sigma\,, \qquad (20)$$

where

F_N – the normal force in guiding from the weight or from the all combat vehicle inclination [1],

f – friction coefficient in the conveyer guiding,

r_{10} – pitch diameter of the driving chain wheel,

i_Σ – total transfer ratio from a conveyer to the hydraulic motor.

The friction force fF_N varies depending on the conveyer loading from 100 N to 4250 N. The workload for the empty and full conveyer depend not only on the normal force, see (20), but on the friction coefficient depending on real working conditions – if the guiding parts are clear or dusty and dirty, see Fig. 11.

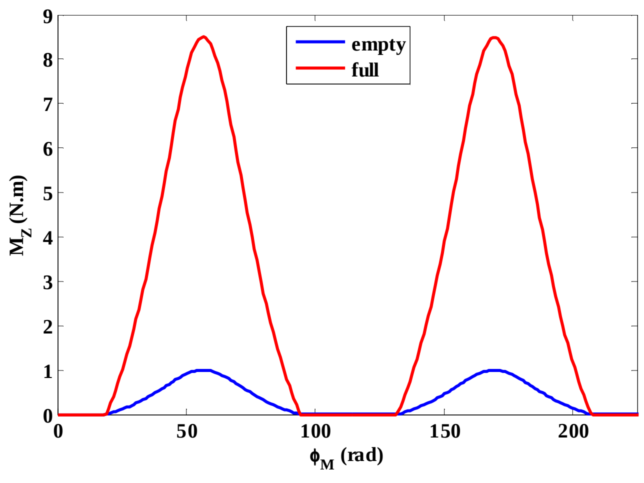

Fig. 11 Workload of system

III. Results of Calculation

The time depending graphs have been obtained from the equations (8), and (12) – (20). The motion equation, the pressures equations and other algebraic equations have been solved by Runge-Kutta integration method. The initial conditions have been zero for input and output pressures p_1, p_2 and for the initial angular velocity $\dot{\varphi}_1$ and the initial angular displacement φ_1. The initial integration step has been used $\Delta t = 0.0001$ s.

The kinematical values, the conveyer displacement x_{conv}, the conveyer velocity v_{conv} and the conveyer acceleration a_{conv}, are displayed in Fig. 12. From this figure follows that the Maltese cross enables the smooth rising of the velocity from zero to the maximum value and back. It means that the velocity is controlled by the mechanical transmission.

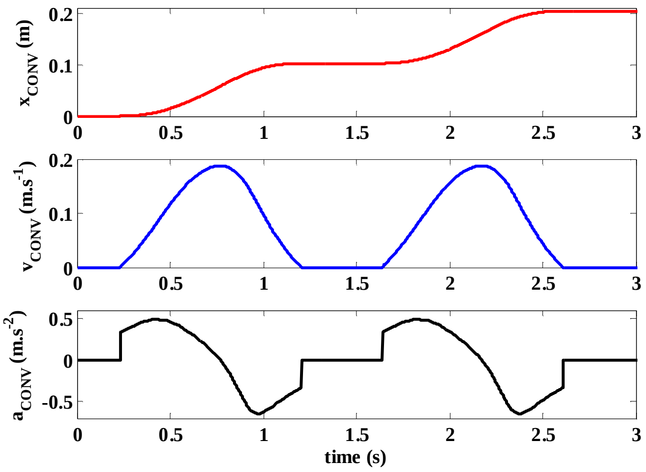

Fig. 12 Kinematic values of shell conveyer

The angular velocity of the hydraulic motor shaft represents the Fig. 13 and it corresponds to the input flow of the hydraulic motor, see Fig. 14.

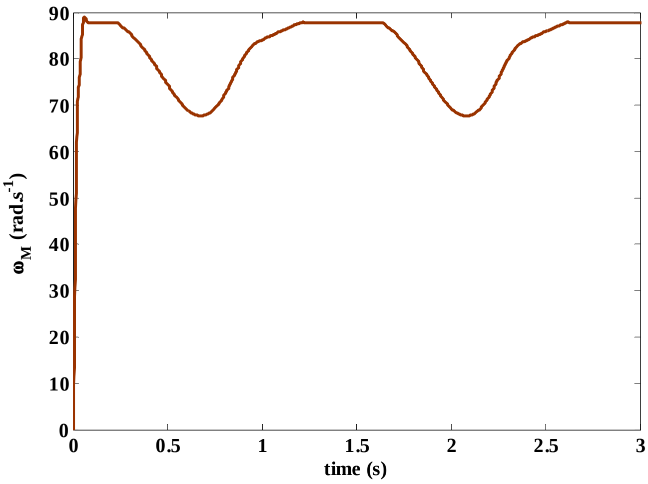

Fig. 13 Angular velocity of hydraulic motor

Its course depends on the system workload and it varies in the other way around than the input pressure and in the same direction as the output pressure. Their curves are plotted in Fig. 15 and the torque of motor in Fig. 16. When the conveyer is empty the pressures are smaller, the maximal value of the input pressure is approximately 2.0 MPa. Both pressures depend on the instantaneous values of the source pressure p_G and the waste pressure p_0 as it has been presented on the real system.

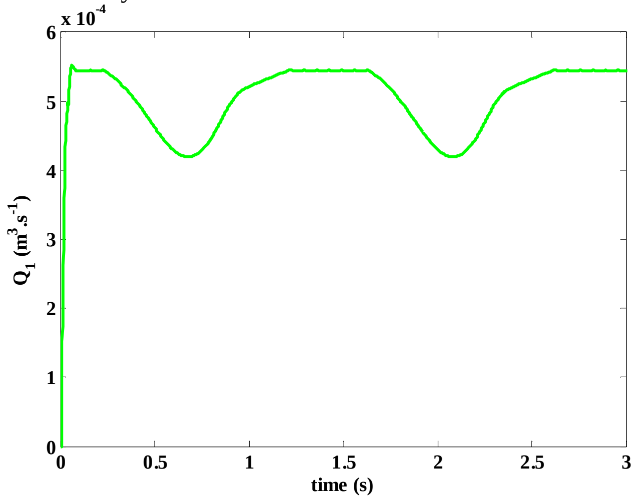

Fig. 14 Input flow in hydraulic motor

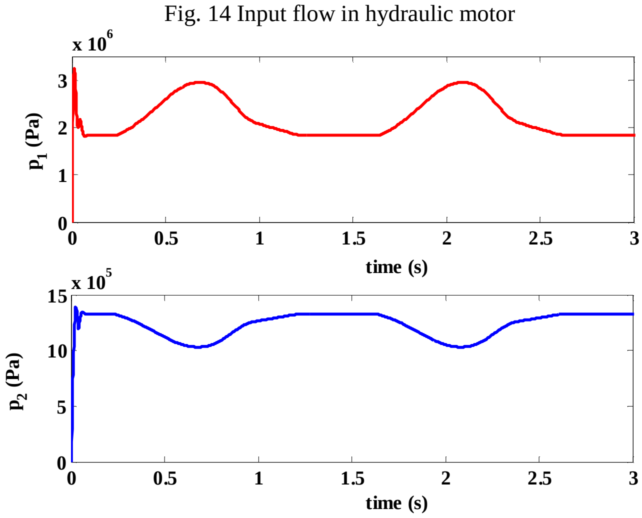

Fig. 15 Input and output pressures in hydraulic motor

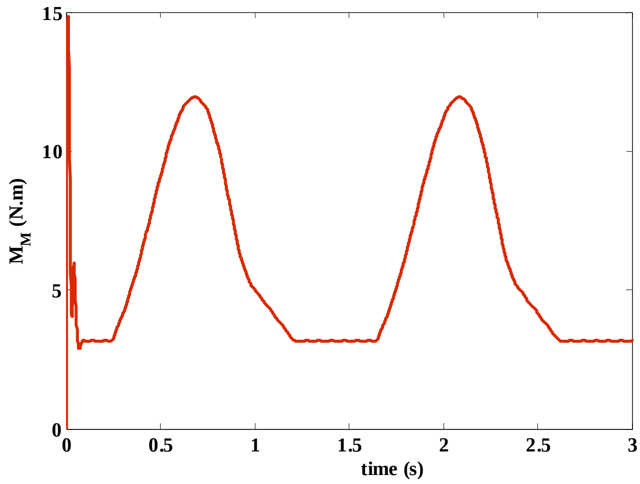

Fig. 16 Torque of hydraulic motor

The drive with the MA-2 hydraulic motor is able to operate in a satisfactory manner with very variable workload without the special control elements, the PID controller for example. The conventional throttle valve and electro-hydraulic distributor ensure the drive functioning as it is shown in Fig. 11.

The second member on the left part in the motion equation (8) influences the behavior of the system mainly in the part where it has negative value. Then it accelerates the system and gives the additional energy for the conveyer. This member signed as

$$dir = 0.5\dot{\varphi}_1^2 \frac{\mathrm{d}I_{\mathrm{M}}}{\mathrm{d}\varphi_1} \qquad (21)$$

represents the Fig. 17 and it has quantity as torque.

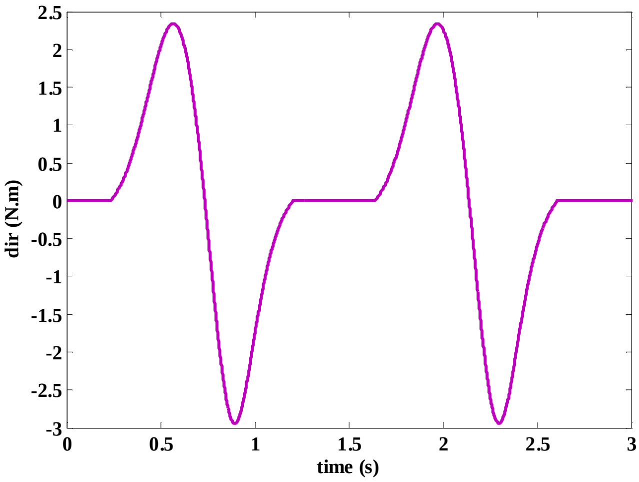

Fig. 17 Member with derivative of inertia mass moment

The increase of the hydraulic capacities C_1, C_2 and the hydraulic losses Z_1, Z_2 generate the higher oscillates of the pressures p_1 and p_2 at the beginning of the operation. Nevertheless the variable workload and reduced mass moment of inertia achieve quite high values, the used hydraulic motor MA-2 has sufficient the power reserve for the potential increase of the workload.

IV. Model with Two degrees of freedom

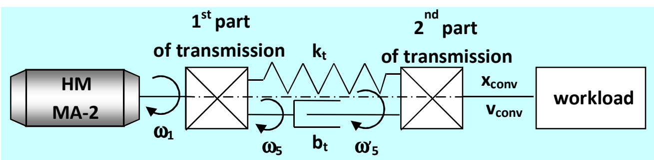

Fig. 18 Flexible binding in dynamic model of shell conveyer

The measuring on the real system have shown that the pressures and the angular velocities of the hydraulic motor have oscillating courses with respect to the calculated results of the system with 1 DOF. These characteristics are possible to approach to the measuring values by an implementation of the other degrees of freedom and flexible bindings. The main problem for using of more degrees of freedom for mechanical and hydraulic systems are to find the modeled values as are stiffness, damping coefficients and sometimes mass moments of inertia for more complicated bodies. If not the new applying can be used, see [10], [19], [24], and [28]. In the weapon applications it was published in [2]. The first part solves the system from the hydraulic motor to the shaft having the I_5 mass moment of inertia and the second part is from this to the rest of the conveyer as it is explained in Fig. 3. The shaft I_5 has been chosen due to the stiffness of torsion is the least.

The equation for the first degree of freedom is

$$I_{\mathrm{red_{1V}}} \frac{\mathrm{d}^2\varphi_1}{\mathrm{d}t^2} + \frac{1}{2}\left(\frac{\mathrm{d}\varphi_1}{\mathrm{d}t}\right)^2 \frac{\mathrm{d}I_{\mathrm{red_{1V}}}}{\mathrm{d}\varphi_1} = M_{\mathrm{M}} - M_{\mathrm{D_1}} - M_{\mathrm{TORZ}}, \qquad (22)$$

and for the second one is

$$I_{\mathrm{red_{2V}}} \frac{\mathrm{d}^2\varphi_2}{\mathrm{d}t^2} = M_{\mathrm{TORZ}} - M_{\mathrm{Z}} - M_{\mathrm{D_2}}, \qquad (23)$$

where

$I_{\mathrm{red_{1V}}}$ – reduced mass moment of inertia of the 1st degree of freedom,

$I_{\mathrm{red_{2V}}}$ – reduced mass moment of inertia of the 2nd degree of freedom,

φ_1, ω_1 – angular displacement and angular velocity of the motor shaft,

φ_2, ω_2 – angular displacement and angular velocity of the flexible shaft,

$\varphi_5 = i_1 i_2 i_3 i_4 \varphi_1$, $\omega_5 = i_1 i_2 i_3 i_4 \omega_1$,

$\varphi_2 = \varphi_5'$, $\omega_2 = \omega_5'$,

k_{t} – rigidity of shaft (N·m·rad^{-1}),

b_{t} – damping factor in coupling (N·m·s^{-1}),

$M_{\mathrm{TORZ}} = k_{\mathrm{t}}(\varphi_5 - \varphi_5')$,

$M_{\mathrm{D_2}} = b_{\mathrm{t}}(\omega_5 - \omega_5')$,

$M_{\mathrm{D_1}}$ – same value as in (19).

In the (22) and (23) there were given the main input data according to the Table I.

TABLE I
MAIN INPUT DATA FOR TWO DOF SYSTEM

Symbol	Quantity	Value
$I_{\mathrm{red_{1V}}}$	mass moment of inertia of the 1st DOF	0.00327 kg·m2
$I_{\mathrm{red_{2V}}}$	mass moment of inertia of the 1st DOF	0.000187 kg·m^2
k_{t}	stiffness of shaft	58 000 N·m·rad^{-1}
b_{t}	damping coefficient in system	96 N·m·s-1

The results of the main dynamic characteristics are presented in Fig. 19, Fig. 20, Fig. 21, and Fig. 22 where the input flow, input and output pressure, torque and power are in hydraulic motor successively. The main differences are in the oscillating courses of calculated characteristics but the values after the transient action are same as for the system with one DOF. The equations (22) and (23) give better results corresponding to the measurements on the real weapon during research project POV DELO No. OVUOFVT200901 as it was published in [1], [2], [13], [14], [15], and [16] as well.

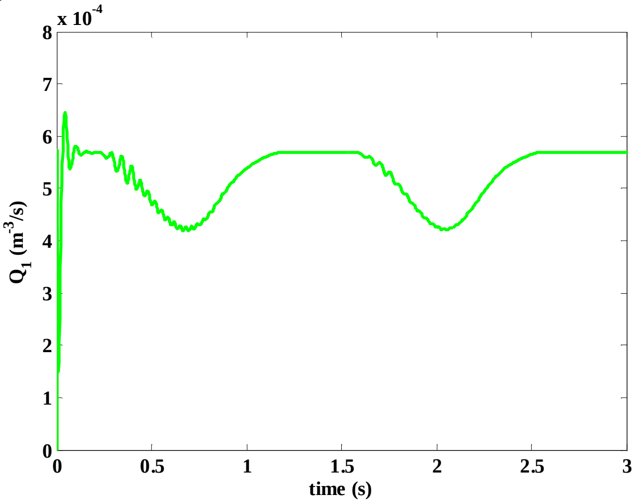

Fig. 19 Input flow in hydraulic motor

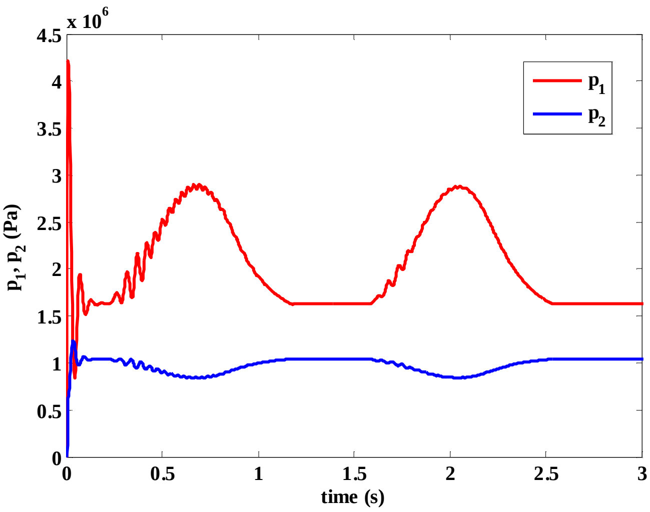

Fig. 20 Input and output pressures in hydraulic motor

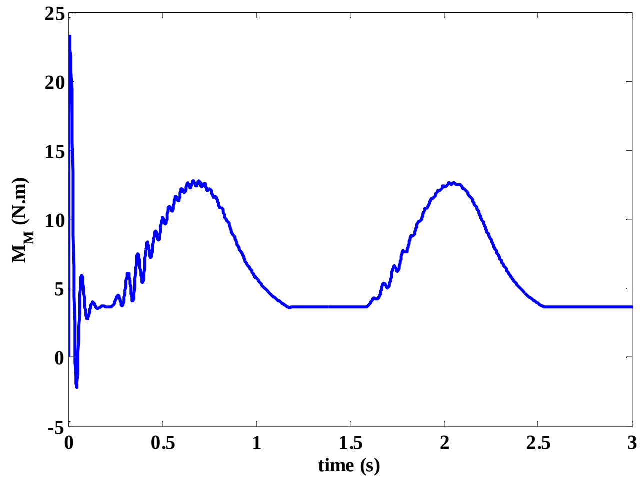

Fig. 21 Torque of hydraulic motor

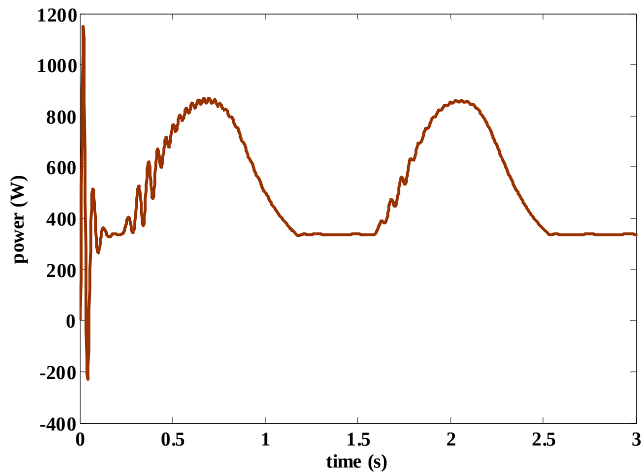

Fig. 22 Power of hydraulic motor

On the other hand the solution describes results which are satisfied comparing with the technical experiments in [13]. Mainly the beginning and the end of periods have to be improved to obtain the more accurate results.

V. Conclusion

The results given in the figures reflect a good coincidence with the real piece which was explored according to presented theory. The theory has been verified on the Czech 152mm self-propelled cannon-howitzer M77. The procedure used in this article has been applied in the Czech research institutes and in the University of Defense in Brno as additional teaching material for students of weapons and ammunition branch. For the change of input hydraulic parameters has been introduced the new laboratory hydraulic source with possibility to vary output pressure and output flow, see Fig. 23. This hydraulic source has the sense loading system enabling to hold set parameters before technical experiments.

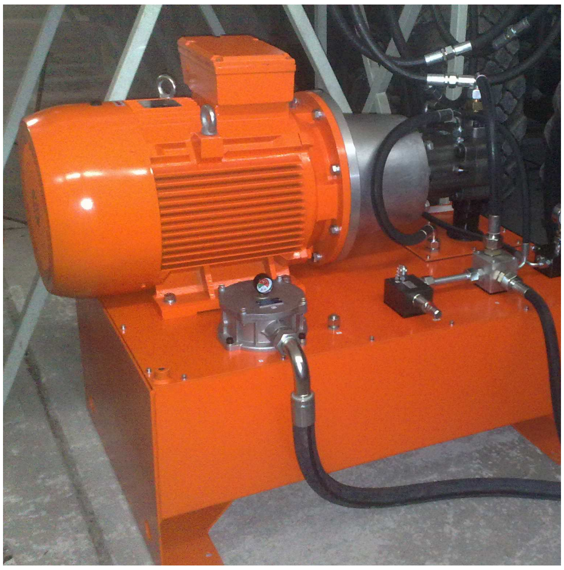

Fig. 23 New hydraulic source

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