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Kinematics and Dynamics of Ramming Devices

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Abstract:

The paper deals with basic kinematics and dynamic characteristics of ramming devices with hydraulic drive.

Keywords:

Ramming device, ramming force, hydraulic drive, motion equations.

1. Introduction

Ramming of gun cartridges is a very difficult operation mainly when loading by hand. Cartridge ramming is an operation which influences especially the rate of fire and it belongs to hardest activities before a shot. Mechanization of cartridge ramming was the first step destined to ease the loading process.

Easiest to ram is the fixed ammunition since only one action step is needed. The projectile and cartridge case are mated (or-mated for loading after the charge has been adjusted).

Separate ammunition is a slower option requiring ramming of the projectile and then the loading of the propellant. Power ramming is very desirable, especially for heavier projectiles.

The main characteristics of ramming devices which are checked during technical inspections are the ramming velocity and the retention force. Their values are recommended to evaluation of the ramming device, see [9].

In [1], there are given recommended values of ramming velocities for unitary and separated cartridges, (see Table 1).

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Table 1 Ramming velocities

ramming velocity	v_{min} (m/s)	v_{max} (m/s)
unitary cartridge	0.6 – 1	4 – 6.5
separated cartridge	more than 0.3	1 – 1.4 (charges) 3 (projectiles)

These velocities have been verified by technical experiments in [4], for example.

But in this table there are not mentioned the values for flick rammers used nowadays in most western countries. The paper [6] shows that these velocities at the departure of ramming are from 6m/s up to 8m/s and after that the projectiles move due to inertia.

The following chapter explains as well as demonstrates the possibility of the dynamic modelling using an example of hydraulic ramming device of the 152mm SPH Mod. 77.

2. Dynamic Model of Ramming Device

The mechanical-hydraulic scheme of loading device for the 152mm SPH Mod. 77 is represented in Fig. 1, see [11]. The loading device for the separate loaded ammunition consists of: shell storage system, propelling charge with cartridge case storage system, propelling charge with cartridge case feeding device, shell feeding device, ramming device, cartridge case ejecting device and control system. All parts are driven by hydraulic motors with the operation pressure 4MPa. Loading is possible at every elevation angle. First loading takes 15s. The rate of fire is 4 rounds per minute and 5 rounds per minute when first round is loaded in the barrel. Loading by hand is possible as well but the rate of fire goes down to two rounds a minute.

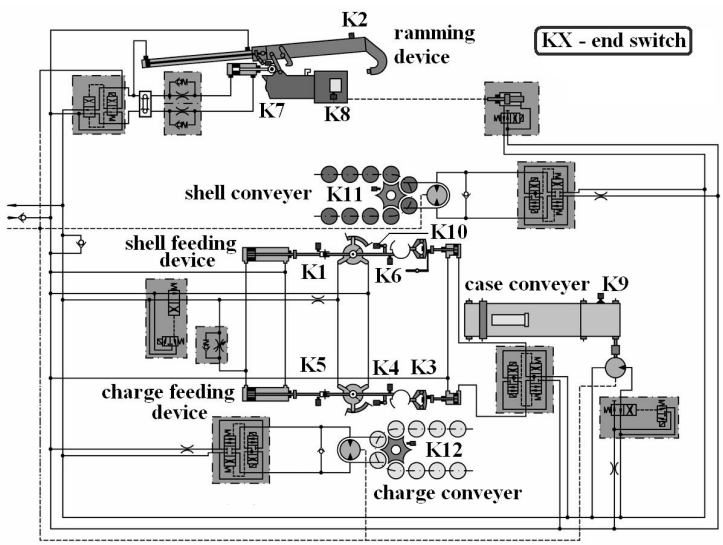

Fig. 1 Self propelled howitzer loading device scheme

The model outgoing from the scheme in Fig. 1 is represented in Fig. 2 and describes the motion of two bodies namely projectile and rammer. It has two degrees of freedom.

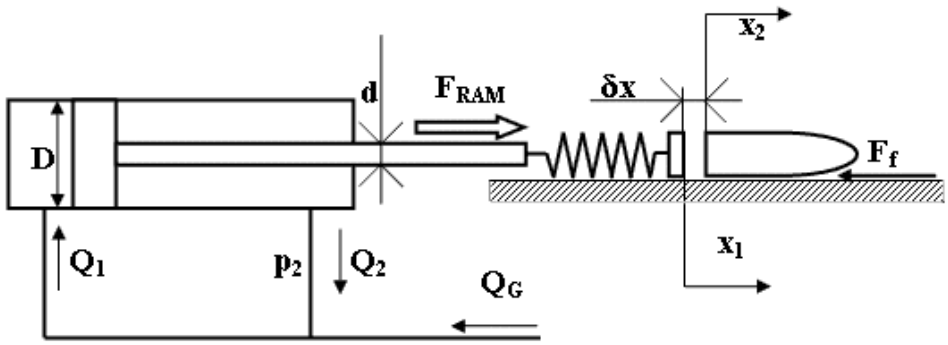

Fig. 2 Dynamic model of ramming device

The system motion is described with two absolute coordinates: x_1 - rammer displacement and x_2 - projectile or charge displacement.

The flexible linkage is modelled by a spring with spring constant k_{RAM} between both bodies. A small clearance $d\,x$ of some tens of millimetres is included at the beginning of the projectile motion. The hydraulic part is characterized by input and output pressures p_1, p_2 and input, output and generator flows Q_1, Q_2 and Q_{G}. The ramming force F_{RAM} is created with a hydraulic motor. Against ramming motion acts the F_{f} force.

The equations of motion of the mechanical system are obtained from the Lagrange equation form:

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial E_{\mathrm{K}}}{\partial \dot{q}_{\mathrm{j}}}\right) - \frac{\partial E_{\mathrm{K}}}{\partial q_{\mathrm{j}}} = -\frac{\partial E_{\mathrm{P}}}{\partial q_{\mathrm{j}}} + F_{\mathrm{j}} \tag{1}$$

where is

E_{K} – system kinetic energy,

E_{P} – system potential energy,

F_{j} – generalized force.

The kinetic energy is given as

$$E_{\mathrm{K}} = \frac{1}{2} m_1 \dot{x}_1^2 + \frac{1}{2} m_2 \dot{x}_2^2 \tag{2}$$

where is

m_1 – rammer mass,

m_2 – projectile mass,

$\dot{x}_1$ – rammer velocity,

$\dot{x}_2$ – projectile velocity.

The potential energy

$$E_{\mathrm{P}} = \frac{1}{2} k_{\mathrm{RAM}} (x_2 - x_1)^2 . \tag{3}$$

The derivatives of the potential energy represent components of elastic force for every degree of freedom:

$$F_1^{\mathrm{P}} = -\frac{\partial E_{\mathrm{P}}}{\partial x_1} = k_{\mathrm{RAM}} (x_2 - x_1) , \tag{4}$$

$$F_2^{\mathrm{P}} = -\frac{\partial E_{\mathrm{P}}}{\partial x_2} = - k_{\mathrm{RAM}} (x_2 - x_1) , \tag{5}$$

$$\frac{\partial E_{\mathrm{K}}}{\partial q_{\mathrm{j}}} = 0 .$$

The generalized forces are obtained using principle of virtual work. The force for the first coordinate (rammer) is the damping force of the hydraulic motor.
The damping force depends on the piston velocity and damping coefficient.
Its expression in first approximation is

$$F_{\mathrm{D}} = b_{\mathrm{D}} \dot{x}_1 , \tag{6}$$

where b_{D} - damping coefficient (N.s.m^{-1}).

The second force acting against the projectile motion is the resultant of the projectile weight and friction force, see Fig. 3,

$$F_{\mathrm{f}} = G(\sin j + f \cos j) , \tag{7}$$

where is

G - weight of projectile,

f - friction coefficient between projectile and loading tray,

j - elevation angle.

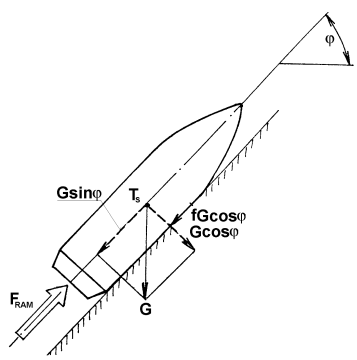

Fig. 3 Forces against projectile motion

These forces are drawn for 122mm and 152mm projectiles in Fig. 4.

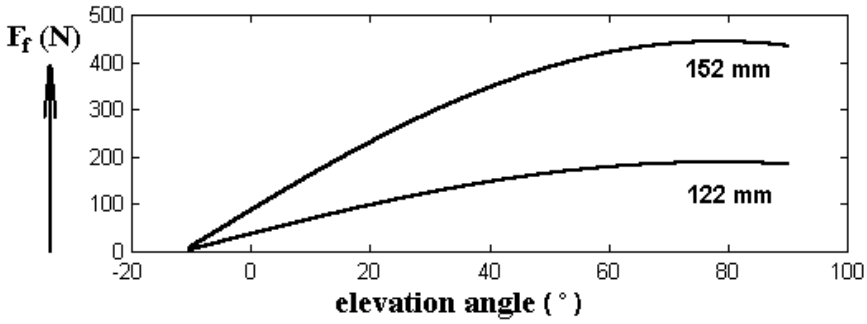

Fig. 4 Static force

The equations of motion are written in the following way suitable for computation:

$$\begin{aligned} m_1\ddot{x}_1 &= F_{\mathrm{RAM}} + k_{\mathrm{RAM}}(x_2 - x_1) - F_{\mathrm{D}} \\ m_2\ddot{x}_2 &= -k_{\mathrm{RAM}}(x_2 - x_1) - F_{\mathrm{f}} \end{aligned} . \tag{8}$$

The ramming force of the hydraulic motor is

$$F_{\mathrm{RAM}} = p_1 S_1 - p_2 S_2 , \tag{9}$$

where $S_1 = \frac{p}{4} D^2$, $S_2 = \frac{p}{4}(D^2 - d^2)$, see Fig. 2.

The pressures p_1, p_2 are derived from the following differential equations of 1st order with hydraulic losses denoted by Z_1, Z_2, (see e.g. [7]):

$$\frac{\mathrm{d}p_1}{\mathrm{d}t} = \frac{(Q_1 - S_1\dot{x}_1 - Z_1 p_1)}{C_1} , \tag{10}$$

$$\frac{\mathrm{d}p_2}{\mathrm{d}t} = \frac{(S_1\dot{x}_1 - Q_2 - Z_2 p_2)}{C_2} . \tag{11}$$

The input and output flows Q_1 and Q_2, respectively are given by the algebraic equation

$$Q_1 = Q_{\mathrm{G}} + Q_2 , \tag{12}$$

where

$$Q_1 = \mathrm{sgn}(p_{\mathrm{G}} - p_1)\sqrt{\frac{|p_{\mathrm{G}} - p_1|}{R_1}} , \tag{13}$$

$$Q_2 = \mathrm{sgn}(p_2 - p_1)\sqrt{\frac{|p_2 - p_1|}{R_2}} , \tag{14}$$

Q_{G} - source flow.

The other variables in equations above are:

$C_1 = C_{01} + b_L S_1 x_1$ - input hydraulic capacity,

$C_{01} = b_L V_{01} + b_H V_{H1}$,

$C_2 = C_{02} + b_L S_2 (h - x_1)$ - output hydraulic capacity,

$C_{02} = b_L V_{02} + b_H V_{H2}$,

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

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

h – whole piston displacement, in this case is equal to 1.747 m,

V_{H1} – input liquid volume in the hose, leading from input pipe to hydraulic motor,

V_{H2} – output liquid volume in the hose, leading from hydraulic motor to output pipe,

b_L – liquid volume compressibility factor, in this case is equal to 6.8x10^{-10} Pa^{-1},

b_H – hose volume compressibility factor,

x_1 – piston displacement,

R_1 – input hydraulic resistance,

R_2 – output hydraulic resistance,

p_G – source pressure, given as

$$p_G = 3.9 + 0.2 \sin(30 p t) \text{ [MPa]}. \tag{15}$$

Values C_{01} (2x10^{-11} m^3.Pa^{-1}) and C_{02} (2x10^{-12} m^3.Pa^{-1}) have been obtained from measurement, published in [4], for example. Damping coefficient b_D (1380 N.s.m^{-1}) was determined by measurement in the course of steady-state motion of the ramming system under off-load conditions. In that case $m_1 \ddot{x}_1 = 0$, $k_{RAM} = 0$ and $F_{RAM} = F_D$, see first of equations (8).

The hydraulic resistances R_1 (2.2x10^{11} Pa.s^2.m^{-6}) and R_2 (1.14x10^{12} Pa.s^2.m^{-6}) were calculated from measured flows and pressures, see [4].

The hydraulic losses Z_1, Z_2 have not been considered in the calculations.

The clearance $d\,x$ has been considered 20 mm as it is at the gun.

3. Results of The Solution

The system equations (8), (10), (11), (13) and (14) have been solved by Runge-Kutta integration method of 4th order. The suitable integration step has been chosen as 0.0001s. It corresponds to known condition between the minimal integration step and the maximal considered frequency f_{max} of the undamped system

$$\Delta t_{min} = \frac{1}{p\, f_{max}}. \tag{16}$$

The basic calculations have been carried out for the 152mm self propelled howitzer ramming device. The projectile mass (m_1) is 43.5kg and rammer mass (m_2) is 15kg.

The main results – the velocities of both shell and rammer piston, and pressures in the hydraulic motor – are shown in Figs. 4 and 5.

The beginning of the shell motion comes after piston taking up the clearance dx. The velocity course shape depends mainly on spring constant k_{RAM}. The special kinematic measuring of shells and rammers corresponds with it. In this connection would be convenient to measure entire process of ramming with engraving of driving bands into the forcing cone.

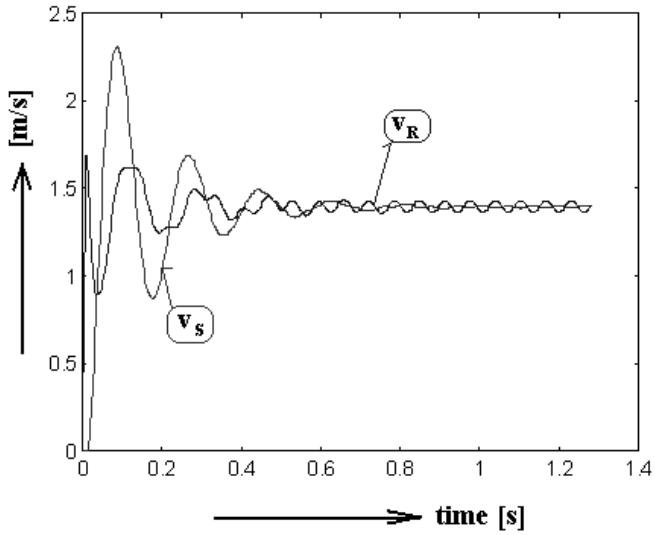

Fig. 5 Shell v_S and rammer v_R velocity with linear hydraulic motor

The rammer velocity was compared, see [4], [4], by measuring the input flow with a turbine flow meter. The used flow meter was not too suitable due to small dynamic range. The beginnings of ramming cannot be visualized quite exactly and the ramming start cannot be taken into consideration for the evaluation of dynamic processes in the hydraulic system.

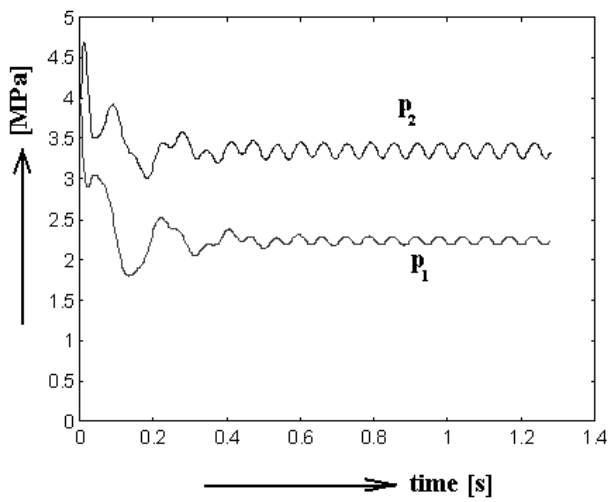

Fig. 6 Input and output pressures in linear hydraulic motor

Regardless the pressure calculations results are in accordance with measuring primarily at the course of steady-state motion.

The pressure pulsations are likely caused by source pulsations (axial piston hydrogenerator), vibrations of the hydropneumatic accumulator and regulating slide valve. But we need to know the stabilized state, in the first instance velocities before shell engraving. The developed system of equations permits to determine it.

It is very interesting that the input pressure is smaller than the output one. It follows from the properties of the differential piston which is used in this ramming device. The reason for this application has been the acquirement of the desired ramming velocity (Table 1). This velocity has to ensure that the shell does not fall-back from the barrel when it is rammed. The linear hydraulic motor applied in the original ramming davice design has failed to achieve the ramming velocity at given pressures and flows before finishing the ramming.

Better solution would be to use a rotary hydraulic motor with a transmission to rack on the rammer, see Fig. 7. This solution requires a lower liquid amount than the design based on a linear hydraulic motor.

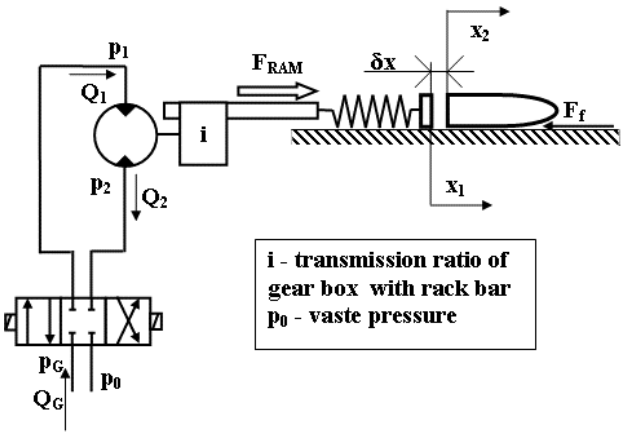

Fig. 7 ramming device driven by rotary hydraulic motor

The hydraulic motor is controlled by a three-position electrohydraulic distributor instead of the contemporary used two-position distributor.

The mathematical description uses the same equation (1) where the absolute coordinate x_1 is equalled to $x_1 = ij_M$. The transmission ratio i (it is not dimensionless but its dimension is m) relates the rammer linear displacement and the hydraulic motor j_M angular displacement.

Then the kinetic energy and potential energy of the system are given by equations similar to (2) and (3):

$$E_K = \frac{1}{2} I_M \dot{j}_M^2 + \frac{1}{2} m_2 \dot{x}_2^2 , \tag{17}$$

$$E_P = \frac{1}{2} k_{RAM} (x_2 - ij_M)^2 . \tag{18}$$

Further

$$F_1^{\mathrm{P}} = -\frac{\partial E_{\mathrm{P}}}{\partial j_{\mathrm{M}}} = ik_{\mathrm{RAM}}(x_2 - ij_{\mathrm{M}}) \tag{19}$$

$$F_2^{\mathrm{P}} = -\frac{\partial E_{\mathrm{P}}}{\partial x_2} = -k_{\mathrm{RAM}}(x_2 - ij_{\mathrm{M}}). \tag{20}$$

The generalized forces are obtained by the same way as before. The generalized force for first coordinate (rotor angular displacement) is the damping force of the hydraulic motor determined by measuring. Its expression is

$$M_{\mathrm{D}} = b_{\mathrm{D_M}} \dot{j}_{\mathrm{M}}, \tag{21}$$

where $b_{\mathrm{D_M}}$ is the damping coefficient.

The value for the given case is 0.04 (N.m.s.rad^{-1}).

The friction force F_{f} is replaced by the torque

$$M_{\mathrm{f}} = iF_{\mathrm{f}}. \tag{22}$$

The driving torque follows from the next relation:

$$M_{\mathrm{T}} = \frac{V_{\mathrm{G}}}{2p}(p_1 - p_2) \tag{23}$$

where V_{G} is the geometrical volume of the hydraulic motor (this value has been selected as 3.9x10^{-5} m^3.

The equations of motion of the mechanical system are written as

$$\begin{aligned} I_{\mathrm{M}}\ddot{j}_{\mathrm{M}} &= M_{\mathrm{T}} - M_{\mathrm{D}} + ik_{\mathrm{RAM}}(x_2 - ij_{\mathrm{M}}) \\ m_2\ddot{x}_2 &= -k_{\mathrm{RAM}}(x_2 - ij_{\mathrm{M}}) - M_{\mathrm{f}} \end{aligned}. \tag{24}$$

The hydraulic equations for determining input and output pressures can be introduced in a similar way as for the linear hydraulic motor, see [4], [7]:

$$\frac{\mathrm{d}p_1}{\mathrm{d}t} = \frac{(Q_1 - \frac{V_g}{2p}\dot{j}_{\mathrm{M}} - Z_1 p_1)}{C_1}, \tag{25}$$

$$\frac{\mathrm{d}p_2}{\mathrm{d}t} = \frac{(\frac{V_g}{2p}\dot{j}_{\mathrm{M}} - Q_2 - Z_2 p_2)}{C_2}, \tag{26}$$

$$Q_1 = \mathrm{sgn}(p_{\mathrm{G}} - p_1)\sqrt{\frac{|p_{\mathrm{G}} - p_1|}{R_1}}, \tag{27}$$

$$Q_2 = \mathrm{sgn}(p_2 - p_3)\sqrt{\frac{|p_2 - p_1|}{R_2}}. \tag{28}$$

The notation is the same as for the linear hydraulic motor case.

The comparing calculations have been carried out using the following input values:

C_1 - input hydraulic capacity ($8x10^{12}$ $m^3.Pa^{-1}$),

C_2 - output hydraulic capacity ($6x10^{12}$ $m^3.Pa^{-1}$),

R_1 - input hydraulic resistance ($9.33x10^{12}$ $Pa.s^2.m^{-6}$),

R_2 - input hydraulic resistance ($1.23x10^{12}$ $Pa.s^2.m^{-6}$).

The values of hydraulic resistances have been considered at the distributor full opening. At the beginning these values have been chosen $2.8x10^{13}$ $Pa.s^2.m^{-6}$ (R_1) and $7.35x10^{12}$ $Pa.s^2.m^{-6}$ (R_2). The opening time has been 40ms.

Main characteristics of ramming, i.e. shell and rammer velocities are shown in Fig. 7.

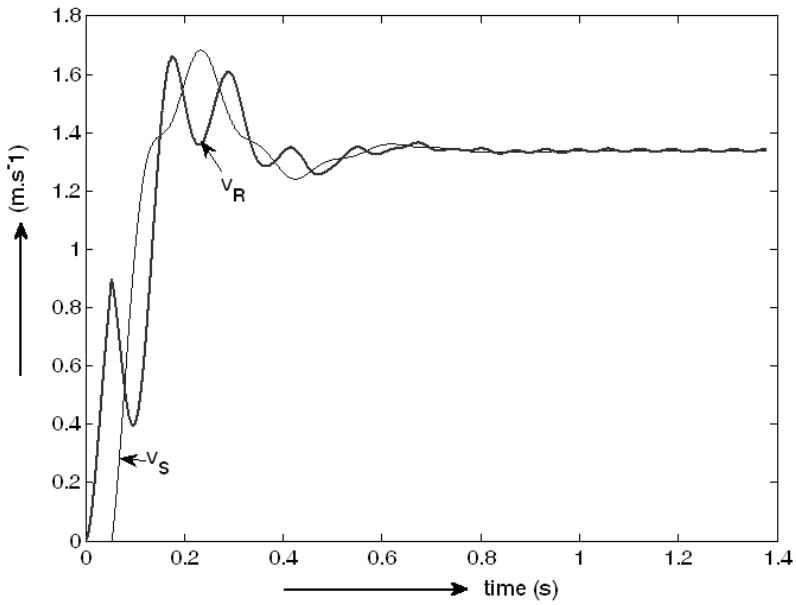

Fig. 7 Shell v_S and rammer v_R velocity with rotary hydraulic motor

After a similar time interval as in the case with the linear hydraulic motor (Fig. 5) the shell achieves the end position before engraving. The velocities are same.

The input and output pressures in the hydraulic motor represented in Fig. 8 are lower than those in the linear hydraulic motor (Fig. 6).

The best advantage of this design is saving of the necessary hydraulic liquid supply from $3x10^{-3}$ $m^3.s^{-1}$ to $0.5x10^{-3}$ m^3 s^{-1}.

The other advantage is a prevention of the liquid escaping from linear hydraulic motor (the piston rod has got very long displacement - 1.7m) in course of storing when obturators are impaired and a soakage occurs. The rotary hydraulic motors do not have this leakage.

The future work must be focused on optimizing the rammer stiffness and the investigation of driving band cutting into the forcing cone as well.

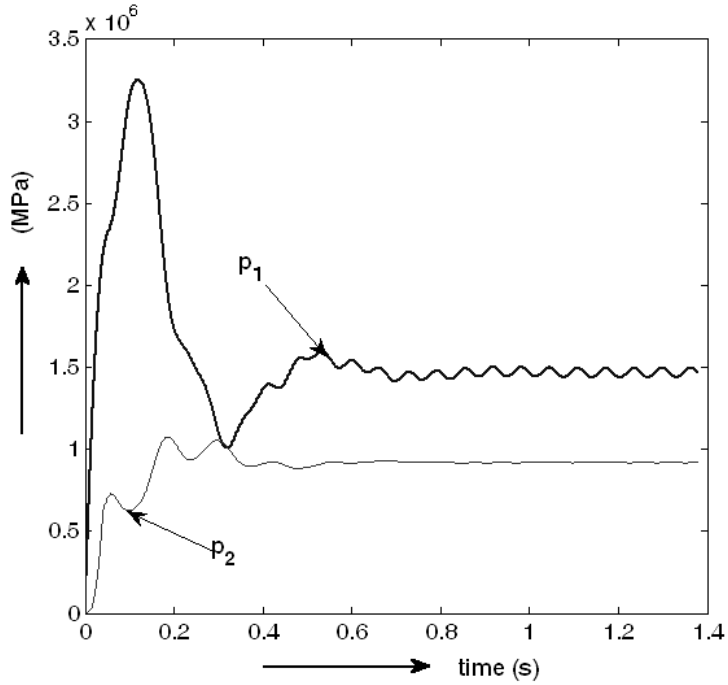

Fig. 8 Input and output pressures in the rotary hydraulic motor

4. Conclusions

The mathematical model published here can be considered as a correct one. The method used in the self propelled howitzer analysis has shown the possibility of dynamic modelling in the loading device domain and it will be applied to other systems, existing nowadays or in future.

The procedures published here are applicable at technical inspections of guns. They will be taken into the consideration during a new Czech defence standards preparation dealing with the loading devices. The theoretical base is unavoidable for professional application of modern measurement methods.

As published in [11] methods for direct measurement of shell motion can not be easily applied. The indirect measurement of shell motion is carried out by comparing the theoretical and measured acceleration time history of the fired projectile. But the initial condition (in internal ballistics) comes out from a state when the projectile is already engraved in the forcing cone. As mentioned before in [9] a better evaluation of the ramming process is to measure the retention force necessary to pull the projectile from the barrel after the loading. In the present gun system (equipped with hydraulic or electric drives) it is possible to measure these forces via increasing pressures or currents in ramming actuators.

This article wants to help weapon experts to prepare good condition for their diagnostics, selection of equipment during acquisitions and the professional preparation.

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