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MODEL AND CALCULATION ALGORITHMS FOR GEAR PERCUSSION

Prof.Dr.Eng.Gavril AXINTI
„Dunarea de Jos" University of Galati

ABSTRACT

The paper highlights some aspects of modeling the dynamic processes taking place in the percussion equipments, known as hydraulic hammers. Theoretical and experimental aspects of the behaviour of equipments on test stand are highlighted. The methodology allows dimensioning of the models starting from percussive energy and frequency. It is presented a case study approach as an example.

KEYWORDS: percution, algorithm, calculation.

1. Introduction

As shown in [1], percussion equipment is that equipment wich operates in a continuous dynamic process of accelerating and slowing down the components that intervene in the process (striker, distributor, tool, pressure valve, etc.).. The dynamic process of operating the percussion equipment [8] is very much influenced in the work environment where the equipment works, (rock, concrete, masonry structures, etc.) by modifying the parameters of percussion.(plastic, elastic, real).

This makes calculating the size of such equipment difficult and creates problems on the in the dimensioning process and in the diversification of the equipments, in particular for those designed for high energy facilities. (About 1 kJ), as those that operates on hydraulic excavator. Dimensioning the equipment is difficult because the system must be very flexible in operation but, in the same time, this allows very easy the transition from astable operation mode to an unstable operation mode, induced by a fast modification of all functional parameters.

2. Standard cycle percussion equipment

To make possible the dimensioning of such equipments is necessary to impose a particular scheme of the equipment and certain dynamic process phases and to establish a standard operating cycle.

For the standard cycle of such equipments are proposed the following values for the lenght of the functional phases:

τ_1-phase represents the time of striker lifting

$$\tau_1 \in (0,30 - 0,32)T \;;$$

τ_2-phase represents the time of striker slowing down

$$\tau_2 \in (0,03 - 0,027)T \;;$$

τ_3-mean-time postfilling

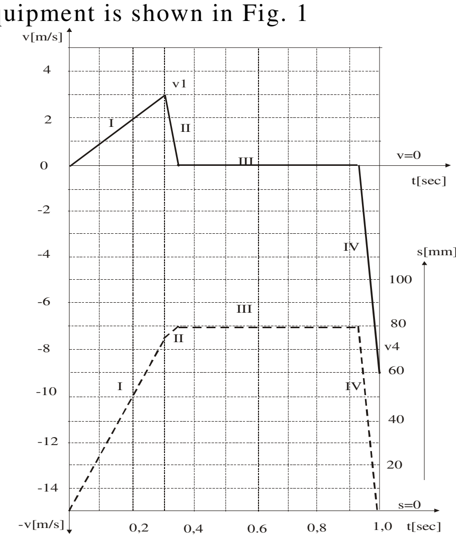
$$\tau_3 \in (0,61 - 0,563)T \;;$$

τ_4-mean-time striker swoop

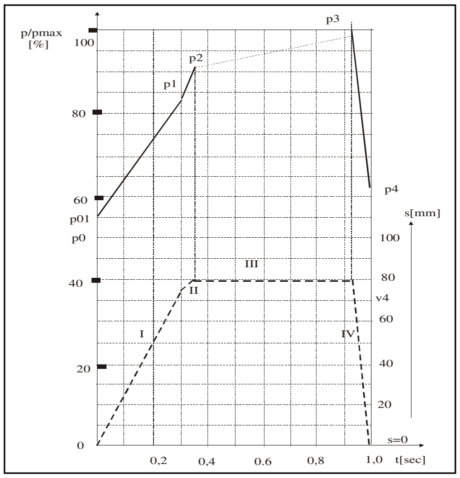
$$\tau_4 \in (0,06 - 0,09)T \;.$$

where T is the cycle length of thefunctional equipment.

Simplified standard cycle of operation of the equipment is shown in Fig. 1



a) change in speed and stroke a momentary striker



b) change in pressure in the system

Fig.1. Dynamic parameters variation of the cycle.

I-striker lifting, II-striker slowing down, III-postfilling, IV-striker lowering.

v [m/s] striker speed, s[mm]-striker stroke, p_0-original hydraulic accumulator pressure, p_{01}-initial lifting striker pressure, p_1-final lifting striker pressure, p_2-final slowing down pressure, p_3-initial lowering pressure, P_4-percussion pressure.

Since the parameters that characterize such equipment are percussion frequency (f) and energy percussion, (E), when dimensioning the percussion equipment, we use these two main parameters. From the frequency parameter is resulting the movement period and the values of specific working cycle, according the relations:

$$T = \frac{60}{f} \qquad (1)$$

$$\tau_1 + \tau_2 + \tau_3 + \tau_4 = T \qquad (2)$$

For the standard cycle of fig.1., kinetic parameters of the striker are calculated with the relations presented in Table 1.

Changes of the momentary stroke depending on time and phase, and considering the current time τ are:

-for lifting the striker:

$$s_1 = \frac{a_1\tau^2}{2};$$

-for slowing down:

$$s_2 = v_1\tau - \frac{a_2\tau^2}{2};$$

- for lifting-down the striker:

$$s_4 = s = \frac{a_4\tau^2}{2};$$

Table 1

no.	phase	time [sec]	kinetic parameters			obs.
			acceleration [m/s^2]	speed [m/s]	stroke[m]	
0	1	2	3	4	5	6
1	Striker lifting up	τ_1	$a_1 = \frac{v_1}{\tau_1}$	$v = \frac{v_1}{\tau_1}\tau$	$s_1 = \frac{v_1\tau_1}{2}$	v_1- adopted speed
2	Striker slowing down	τ_2	$a_2 = \frac{v_1}{\tau_2}$	$v = v_1 - \frac{v_1}{\tau_2}\tau$	$s_2 = \frac{v_1\tau_2}{2}$	$s = s_1 + s_2$
3	Post filling	τ_3	$a_3 = 0$	$v = 0$	s = 0	
4	Striker lifting-down	τ_4	$a_4 = \frac{v_4}{\tau_4}$	$v = -\frac{v_4}{\tau_4}\tau$	$s = \frac{v_4\tau_4}{2}$	$v_4 = \frac{2(s_1 + s_2)}{\tau_4}$

3. Simplified dynamics percussion equipment

In order to determine the mass of the striker, denoted (m), is required the amount of kinetic energy E of the equipment, also taking into account the speed of the percussion (v_4) according the relationship:

$$m = \frac{2.E}{v_4^2} \qquad (3)$$

Depending on the adopted values, the required striker mass is:

$$m = 2\frac{E}{v_1^2}\left(\frac{\tau_4}{\tau_1 + \tau_2}\right)^2; \qquad (4)$$

The initial force for lifting the striker, taking into account the force of inertia and resistant force of the evacuation of agent in hydraulic tank, is:

$$F_1 = m.a_1 + A_2.p_t = A_1.p_1 \qquad (5)$$

where A_2 is the area of the the upper part of the striker, p_t is the tank pressure (cca. 3% of the required nominal pressure).

Also, the viscous friction forces between the striker and the body have been neglected

beeing small compared to the active considered forces).
The initial force for lowering the striker, taking into account the force of inertia is:

$$F_4 = m.a_4 = (A_2 - A_1).p_4 \tag{6}$$

Slowing down force of the striker is:

$$F_2 = m.a_2 = A_2.p_2 \tag{7}$$

In relations (5), (6) (7), p_1, p_2 and P_4 are the minimum pressures on the phases of lifting, slowing down and lowering. Between these pressures we may establish the following relationships of dependence:

$$\frac{A_1}{A_2}.\frac{p_1}{p_2} - \frac{p_t}{p_2} = \frac{\tau_2}{\tau_1} \tag{8}$$

$$\frac{A_2}{A_2 - A_1}.\frac{p_2}{p_4} = \frac{\tau_4}{\tau_2}.\frac{v_1}{v_4} \tag{9}$$

$$\frac{(A_2 - A_1).p_4}{A_1 p_1 - A_2 p_t} = \frac{\tau_1}{\tau_4}.\frac{v_4}{v_1} \tag{10}$$

If we consider that the striker areas are:

$$A_1 = \frac{\pi}{4}(D^2 - d^2) \text{ and } A_2 = \frac{\pi}{4}D^2 \tag{11}$$

where: D is the diameter of the bore of the equipment, and d is the diameter of the impact surface and the velocity ratio is:
and reports are

$$\frac{v_4}{v_1} = \frac{\tau_1 + \tau_2}{\tau_4} \tag{12}$$

$$\frac{A_1}{A_2} = 1 - \left(\frac{d}{D}\right)^2,$$

and areas ratio are:

$$\frac{A_2}{A_2 - A_1} = \left(\frac{D}{d}\right)^2 \tag{13}$$

from the relations (8), (9), (10), follows the interdependence of the operating pressures, in order to have a smooth or generally continuous operation:

$$\frac{p_1}{p_2} = \frac{\frac{\tau_2}{\tau_1} + \frac{p_t}{p_2}}{1 - \left(\frac{d}{D}\right)^2} \tag{14}$$

$$\frac{p_2}{p_4} = \left(\frac{d}{D}\right)^2 \frac{\tau_4^2}{\tau_2(\tau_1 + \tau_2)} \tag{15}$$

$$\left[\left(\frac{D}{d}\right)^2 - 1\right]\frac{p_1}{p_4} - \left(\frac{D}{d}\right)^2 \frac{p_t}{p_4} = \frac{\tau_4^2}{\tau_1(\tau_1 + \tau_2)} \tag{16}$$

Given that the pressure on the tank circuit, because it is negligible, then relations (14) and (16) become:

$$\frac{p_1}{p_2} = \frac{\frac{\tau_2}{\tau_1}}{1 - \left(\frac{d}{D}\right)^2} \tag{17}$$

$$\frac{p_1}{p_4} = \frac{\tau_4^2}{\tau_1(\tau_1 + \tau_2)} \frac{1}{\left[\left(\frac{D}{d}\right)^2 - 1\right]} \tag{18}$$

4. Debits and circulating volume

- striker lifting phase:

$$Q_1 = \frac{\pi.v_1}{4.\tau_1}\left(D^2 - d^2\right).\tau$$

and

$$V_1 = \frac{\pi.v_1.\tau_1}{8}\left(D^2 - d^2\right); V_{AC1} = Q.\tau_1 - V_1 \tag{19}$$

- slowing down phase:

$$Q_2 = \frac{\pi.D^2}{4}\left(v_1 - \frac{v_1}{\tau_2}\tau\right)$$

and

$$V_2 = \frac{\pi.v_1.\tau_2}{8}.D^2; V_{AC2} = Q.\tau_2 + V_2 \tag{20}$$

- phase postfilling:

$$Q_3 = Q$$

and

$$V_3 = Q.\tau_3 = V_{AC3} \tag{21}$$

- phase exit:

$$Q_4 = \frac{\pi.d^2.v_4}{4\tau_4}.\tau$$

And

$$V_4 = \frac{\pi.d^2.v_1}{8}(\tau_1 + \tau_2); V_{AC4} = V_4 - Q.\tau_4 \tag{22}$$

In the schemes of Fig.2 it is shown the circulation of hydraulic oil flow between striker chambers, hydropneumatic accumulator and system pump

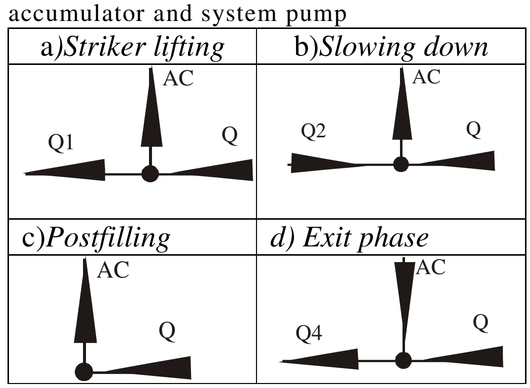


Fig.2.Sistem flow circulation

The volume of hydraulic oil pumped during a whole cycle between the hidropneumatic accumulator and the pump-equipment system comes from the ecuation:

$$V_{AC} = \frac{\pi . v_1}{8}\Big[d^2\left(\tau_1 + \lambda_u \tau_2\right) - D^2(1-\lambda_u)(\tau_1 - \tau_2)\Big] \text{ [m3];} \quad (23)$$

for phases in which the hydraulic oil is accumulating, or

$$V_{AC} = \frac{\pi . v_1}{8}\Big[d^2\left(\tau_1 + (1-\lambda_c)\tau_2\right) + \lambda_c . D^2(\tau_1 - \tau_2)\Big] \text{ [m3];} \quad (24)$$

for the exhaust phase of the hydraulic oil from the accumulator. (see Fig.2).

where:

$$\lambda_u = \frac{\tau_1 + \tau_2 + \tau_3}{T}$$

or

$$\lambda_c = \frac{\tau_4}{T} \quad (25)$$

or

$$\lambda_u = (1 - \lambda_c)$$

• required flow of pump system is obtained from the condition that the pump ensures only the differences in circulating volumes along the working cycle, or the pump ensures a flow necessary to compensate the hydraulic losses from the system.

Relationship to calculate the minimum flow required for pump equipment is:

$$Q = \frac{\pi . v_1}{8T}\left[d^2 \tau_2 + D^2(\tau_1 - \tau_2)\right] \quad (26)$$

provided to ensure continuity in the flow between the phases of movement and structural components of the system (equipment-hydraulic accumulator-pump).

5. Pressures and dynamic interdependencies between process stages

To ensure continuity in the dynamic process of of the active part of the equipment and to overhelme functional resistances (newtonian viscous-friction, volumetric leakiness od hydraulic oil, losses by uncorellated stages, impact energy losses, structural losses, etc.), it is necessary to impose limits for the system pressures.

Pressure values are obtained from the loading and unloading adiabatic hidropneumatic accumulator. Hydraulic oil volumes of acquired in or disposed from the hydropneumatic accumulator are those fromm relations (19)...(22.)

Pressure relations are:

$$p_{01} = p_0\left(\frac{V_0}{V_0 - V_{01}}\right)^k \quad (27)$$

where v_{01} is the remanent-in hydraulic accumulator volume, provided in the initial phase of operating the equipment, (about 0.2 V_0); p_{01}- initial pressure of the dynamic process, k-adiabatic exponent;-V_0 the initial volume of the hydraulic accumulator; P_0- initial pressure for loading the hidropneumatic accumulator with nitrogen.

$$p_1 = p_{01}\left(\frac{V_0 - V_{01}}{V_0 - V_{01} - V_{AC1}}\right)^k;$$

$$p_2 = p_1\left(\frac{V_0 - V_{01} - V_{AC1}}{V_0 - V_{01} - V_{AC1} - V_{AC2}}\right)^k;$$

$$p_3 = p_2\left(\frac{V_0 - V_{01} - V_{AC1} - V_{AC2}}{V_0 - V_{01} - V_{AC1} - V_{AC2} - V_{AC3}}\right)^k \quad (28)$$

$$p_4 = p_3\left(\frac{V_0 - V_{01} - V_{AC1} - V_{AC2} - V_{AC3}}{V_0 - V_{01} - V_{AC1} - V_{AC2} - V_{AC3} + V_{AC4}}\right)^k$$

Pressure values obtained from relations (28), must exceed the minimum arising from the relations (14), (15), (16).

Mechanical power of the equipment shall be determined by the relationship:

$$N = \frac{E.f}{60.000} \text{ [KW]} \quad (29)$$

Hydraulic power of the equipment is determined by the relationship:

$$N_h = \frac{Q.p_3}{603} \text{ [KW]} \quad (30)$$

Equipment efficiency is:

$$\eta = \frac{N}{N_h}.100 \text{ [\%]} \quad (31)$$

6. Case study

Let's calculate a percussion equipment with the following parameters: percussion energy E = 1400 J and frequency f = 300 strikes/min. Equipment should be designed for a small excavator capacity. (0.4 m^3).

Resolution:

Following the calculation presented algorithm is derived: the movement-T = 0.2 sec., Time-phases $\tau_1 = 0.064$ s, $\tau_2 = 0.0054$ s, $\tau_3 = 0.1126$ s, $\tau_4 = 0.018$ s, Accelerations: $a_1 = 39.1$ ms^{-2}, $a_2 = 463$ ms^{-2}, $a_3 = 0$,

$a_4 = 535.5$ ms^{-2}, Speeds: $v_1 = 39,1.\tau$ ms^{-1}, $v_2 = 2,5 - 463.\tau$ ms^{-1}, $v_3 = 0$, $v_4 = -535,5\tau$ ms^{-1}, Strokes: $s_1 = 80$ mm, $s_2 = 6,75$ mm, $s = 86,75$ mm, percussion speed $v_4 = 9,64$ ms^{-1}; striker mass $m = 30,135$ kg, striker size: $A_1 = 2$ cm^2, $A_2 = 20$ cm^2, $D = 50.8$ mm, $d = 48.3$ mm, the minimum hydraulic oil flow of $Q = 48,3$ dm^3/min., the hydraulic accumulator $V_0 = 0.5$ dm^3, $p_0 = 80$ bar, $V_{01} = 0.1$ dm^3, $p_{01} = 108.4$ bar; process pressure: $p_1 = 110$ bar; $p_2 = 111.3$ bar, $p_3 = 157.2$ bar (take -160 bar), $p_4 = 108$ bar; power-mechanical N = 7 kW, hydraulic power-N_h = 12.88 kW; $\eta = 54.4\%$.

Functional diagram of the equipment is shown in fig.3.

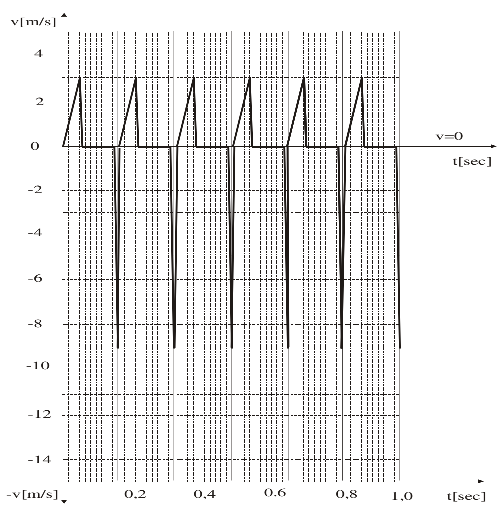


Fig.3 Changes in pressure and velocity model

7. Conclusions

From the case study developed according the methodology of calculation proposed in the paper it can be seen that the methodology is useful for dimensioning percussion equipments.

The methodology and the algorithm was based on reconsidering the dynamic phenomena occurring in such equipment and, in particular, the striker movement as a uniformly accelerated motion.

The calculation methodology was based on careful observation of actual behavior patterns of real or experimental models and of non-linear mathematical models. [1], [7].

The paper can help the designers of technological percussive equipments for dimensioning new ones new or diversifying models from an existing game.

The methodology is according the scheme used by the author. Methodology for other schemes should be adapted accordingly functional phases of the models.

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