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doi:10.15199/48.2019.04.07

Vibroabrasive machining of large-size products on hydropulse drive machines

Abstract. *The paper considers the issue of complex shape items surface depuration together with the most appropriate hydropulse drive future oriented equipment, used in order to address this challenge adequately. There had been considered the mathematical model of the vibration machine with complex space load, developed on the base of the structural chart and behavior pattern. There had been presented the results of the research of the test model plant with the hydro pulse drive for vibroabrasive depuration of the inner surface of the tubular items.*

Streszczenie. *W artykule rozważono zagadnienie kompleksowej obróbki powierzchni przedmiotów kształtowych wraz z najbardziej odpowiednim urządzeniem do sterowania przyszłościowym napędem hydropulsacyjnym. Rozważono matematyczny model maszyny wibracyjnej o złożonym obciążeniu przestrzeni, opracowanej na podstawie wykresu strukturalnego i wzoru zachowania. Przedstawiono wyniki badań modelu testowego z napędem hydropulsacyjnym do oczyszczania wibroabrazyjnego powierzchni wewnętrznej elementów rurowych.* ***(Obróbka wibroakustyczna produktów wielkogabarytowych z wykorzystaniem napędu hydropulsacyjnego)***

Keywords: hydropulse drive, vibration, pressure pulse generator, dynamic model.
Słowa kluczowe: napęd hydropulsacyjny, wibracja, generator impulsów ciśnienia, model dynamiczny.

Introduction

Depuration of outer as well as inner surfaces of complex shape items from forge scales, corrosion, scum as well as processing these surfaces prior to coating by physical and chemical method or paint has been a pressing issue for the enterprises. The present day production faces such difficulties when preparing gas and oil pipes for installation, manufacturing hydraulic power cylinders and turntable-type bearing using low quality pipes, further mechanical processing of which requires expensive tools or ecologically hazardous methods of chemical refinement [2, 3].

Vibroabrasive machining is the most efficient method for depuration of long-length (> 2 meter length) large diameter tubes (> 120 mm). The known plane vibration item load, which is used by the most types of vibratory equipment is not efficient enough when processing large machine parts with complex surfaces. The most technologically advanced equipment for vibroabrasive machining of large machine parts is hydro-pulse drive machine (HPD), which ensures a complex space mode of vibration load. Vibration machines with hydro-pulse drive have a simple system for adjusting frequency and amplitude of oscillation of movable objects as well as energy of one working stroke in each direction of vibration load.

Vibration machines hydro-pulse drive of complex space load may be built according to the control circuit for the actuating hydrocylnders by one or more pressure pulse generator (PPG). PPG may be connected to the actuating hydrocylnders “to the input”, “to the output”, or “combined” [5].

From the point of view of design simplicity, the interest attracts the hydro pulse drive vibromachine of a complex space load with one PPG, switched to the actuating hydrocylnders “to the output”.

The dynamic model applied

When developing the hydropulse drive for the vibromachine of the complex space load, with an aim to receive the highest probable frequency of pressure pulse advancing and improve reliability, the high performance of hydrosystem is required.

Figure 1. presents the simplified block diagram of hydropulse vibromachine with a complex space load.

The operating link 1 with the processing unit (big diameter tubes) shall be set in motions by horizontal 2 and angle 3 hydro cylinders. Hydrocylinder maneuvering is done by PPG 5, switched to the discharged chambers of hydro cylinders switched “to the output”. The operating link 1 and the bed 4 are firmly tided to each other. Movements of the operating link in horizontal and angular directions are restricted by the bump stops.

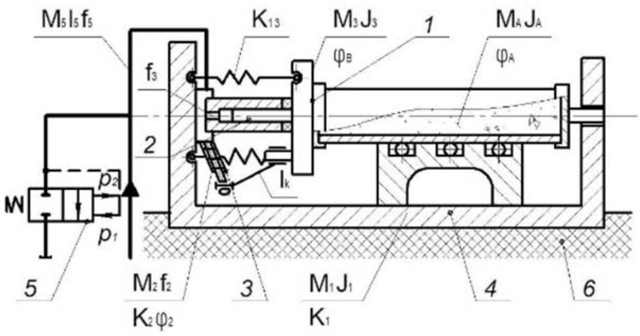

Fig. 1. Structural scheme of hydropulse vibromachine

In this case, on the base of general theory for hydropulse drives calculation [1], and the general structural diagram (Fig.1) of the vibration machine with HPD it is possible to compose a complete multi-mass dynamic model of the vibration machine (Fig.2).

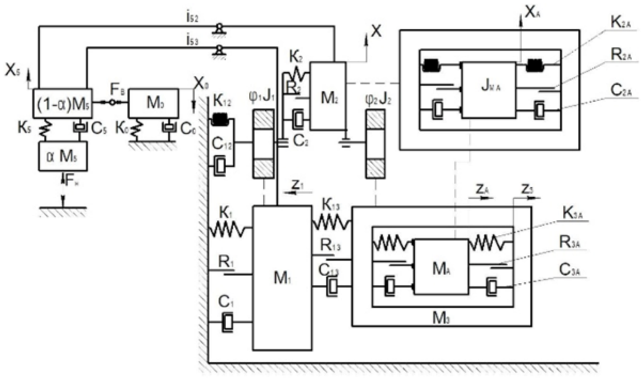

Fig. 2 Complete dynamic model of vibration machine

When composing the complete dynamic model, the following assumptions had been made: hydraulic linkage shall be considered as "resilient-concentrated model" [1] and with the help of resilient and dissipative connections interacts with the other linkages of the machine; masses of the resilient elements of vibration machine linkages backtracking are small in comparison with masses that travel; channel borders and pipelines are rigid; wave processes in hydraulic circuits, due to their short length, do not influence the dynamics of the system; impact interactions of linkages masses in the motion output equations shall not be accounted for; fluid leaks through the clearances between the joint elements is insignificant in comparison with the hydraulic pump output and shall not be considered in the output equations; hydraulic forces which influence the stop valves of the vibration exciter are small in comparison with shifting force, force of the elastic retrieval and resistance; mass of the executive chain, reduced to the angle hydro cylinder plunger, is concentrated in the piston rod articulation pivot of the crank gear with plunger; a reduced volume elasticity modulus of the hydro system is accepted as constant along the whole operating cycle of the vibration machine; hydraulic resistance of the drain line is insignificant therefore not considered in the output dependences; the behavior of the abrasive mass is described on the base of the assumption, that the low part of the layer which directly contacts with the work surface, behaves as a solid body with the constant mass center which is on the top, and the parts of the layer which lie on the top, influence this body by statistic pressure which equals their weight [8-10].

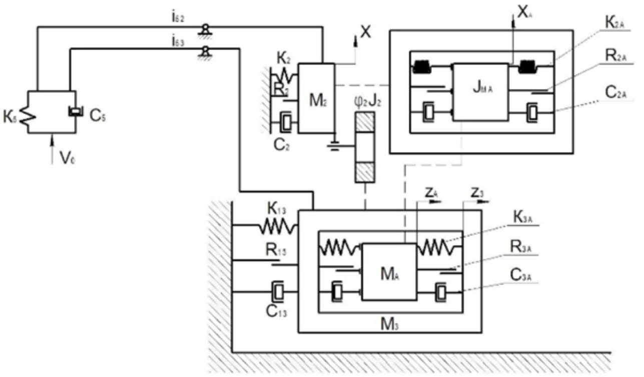

Fig. 3. Simplified dynamic model of vibration machine

General dynamic model of the vibration machine, pretty complicated for mathematical analysis, may be simplified by comparing the relative values of the reduced masses and their travel. Several authors state that the mass of the bed M_1, with an aim of reducing vibratory transfer influence upon the structural units, must be 8...10 times bigger than the maximal inertial mass (M_2, M_3) [4, 10]. This allows to ignore the bed mass movement. To receive the maximum hardness of the hydrosystem, the volumes of the pressure pipe line and working chambers shall be reduced to minimum, which allows to neglect the mass of the liquid in the pipe lines having exchanged it for the massless coil spring with the spring tension of K_5 and a damper with the viscosity of C_5, which shall be deformed with constant speed of $V_o = Q_H / f_5$ (Q_H = const – hydraulic pump output);

- $f_5 = \sum_1^n (l_i \cdot f_i) \Big/ \sum_1^n l_i$ averaged surface area of the pump line; l_i and f_i correspondingly the length and the square of the cross section of some channels of the hydro system discharge chambers of the vibration machine (i = 1...n). In the result we receive the three-mass dynamic model (Fig. 3), which describes the machine system (Fig.1) in general terms.

The composed dynamic model uses the following designations: C_1 – damping factor in expressions for the forces of viscous friction during the movement of the bed; C_{13} – damping complex factor in expressions for the forces of viscose friction during the movement of the hydraulic piston of the linear hydro cylinder C_3, during the movement of the supporting balls C_{bl}, during the interwork of the back lid of the pipe with the friction bearing C_b; C_{21} - damping complex factor in the expressions for the forces of viscose friction during the movement of angle hydro cylinder C_2, during the movement of the supporting balls C_{bl}, during the interwork of the back lid of the pipe with the friction bearing C_n; C_{12} – damping angle factor of the bed relating to the shock damper; C_{3A} – damping linear factor in dependences for the forces of the viscose frictions between the particles of the abrasive materials; C_{2A} - damping angle factor in the dependences for the forces of viscose friction between the particles of the abrasive materials; R_1; R_2; R_{2A}; R_{3A}; R_{13} – constant components of the reduced forces of unlubricated friction; ξ - factor of the turbulent damping; α - factor which accounts for the value of the mass of liquid in the pipe line 15; C_5; C_6 – damping factor in corresponding hydro line; K_5; K_6 – rigidity of the hydro line; F_H – function of the energy source; F_B – function of the vibration exciter; $i_м$ – transfer function of the slide-crank mechanism; gear rations: i_{65} – from hydraulic line l_6 to l_5; i_{52} - from hydraulic line l_5 to the angle piston chamber; i_{53} - from hydraulic line l_5 to the linear piston chamber;

On the initial phase of the first stage of the operation cycle under stationary sections of the unit, mass of the liquid $\alpha M5$ moves in the positive direction, causing the increase in pressure in hydraulic system to the value po, sufficient to overcome the forces of stationary resistance. The rigidity of the hydraulic system, following the Hook's law for the liquids shall be determined by the dependence:

(1) $$K_5 = \frac{f_5^2 \cdot \chi}{\left(W_o + f_3 \cdot z_3 + f_2 \cdot x\right)},$$

where χ - reduced volume module of hydro system elasticity; W_o – initial volume of the discharge chamber of the hydro system; f_2, f_3 – area of cross section of the operation hydrocylinders; z_3, x – axis of motion of the function element of the vibration machine in the angle and horizontal dirctions. As a rule, $W_o \geq f_3 \cdot z_3 + f_2 \cdot x$, then $K_5 \approx f_5^2 \cdot \chi / W_o = const$.

(2) $$\frac{K_5 z_5}{f_5} = \frac{F_2}{f_2} = \frac{F_3}{f_3},$$

where $F_2 = K_2 x_{02}$, $F_3 = K_{13} z_{03}$ – strains of the stationary resistance to the movement of the angle and the linear hydro cylinders.

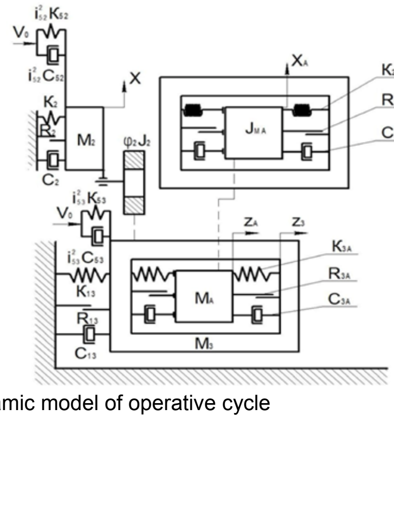

Fig. 4 Dynamic model of operative cycle

The second phase of the first stage of the operation cycle, characterized by movement of masses M_2, M_3, M_A. Dynamic model, which describes this phase is presented in Fig. 4.

Generalized system of the differential equations, which describes the second phase of the first stage of the operation cycle, shall be written as follows:

$$(3)\qquad \begin{cases} M_2\dot{x} = i_{52}^2 K_5\left(z_5 - x\right) - i_{52}^2 C_5\left(\dot{z}_5 - \dot{x}\right) - C_2\dot{x} - \\ -K_2\left(x_{02} + x\right) - \dfrac{J_2\phi_{OB}}{r_2} - R_{2A} - R_2; \\ M_3\dot{z}_3 = i_{53}^2 K_5\left(z_5 - z_3\right) - i_{53}^2 C_5\left(\dot{z}_5 - \dot{z}_3\right) - \\ -C_{13}\dot{z}_3 - K_{13}\left(z_{03} + z_3\right) - R_{3A} - R_{13}; \\ M_A z_A = M_3 z_3 - C_{3A} z_A - K_{3A} z_A - R_{3A} sign\dot{z}_A; \\ J_{MA}\phi_{OB} = J_{M2}\phi_{OB} - C_{2A}\phi_{OB} - K_{2A}\phi_{OB} - R_{2A} sign\dot{\phi}_{OB}. \end{cases}$$

Gear ration relations i_{53} and i_{52} may be calculated by the condition of equality in dynamic pressure of the utility product, which influences the hydraulic chain and the cross-section area of the corresponding element of the drive (converging and the reduced chains):

$$(4)\qquad \begin{cases} \dfrac{K_5 \cdot z_5}{f_5} = \dfrac{i_{53} \cdot K_5 \cdot \left(z_5 - z_3\right)}{f_3}; \\ \dfrac{K_5 \cdot z_5}{f_5} = \dfrac{i_{52} \cdot K_5 \cdot \left(z_5 - x\right)}{f_2}, \end{cases}$$

from which:

$$(5)\qquad \begin{cases} i_{53} = \left(\dfrac{f_3}{f_5}\right) \cdot \left[\dfrac{z_5}{\left(z_5 - z_3\right)}\right]; \\ i_{52} = \left(\dfrac{f_2}{f_5}\right) \cdot \left[\dfrac{z_5}{\left(z_5 - x\right)}\right], \end{cases}$$

here z5 – deformation of the hydraulic chain. Derivatives $i_{52}K_5 = K_{52}$ and $i_{53}K_5 = K_{53}$ are stiffness factors of the hydraulic system, reduced to the corresponding chain of the vibration machine drive.

During the reduction of the hydraulic chain to the cross-section areas of the operating hydrocylinders f_2 and f_3 its potential energy shall be redistributed in accordance with the equation:

$$(6)\qquad 0.5K_5 z_5^2 = 0.5K_5\left[i_{53}\left(z_5 - z_3\right)^2 + i_{52}\left(z_5 - x\right)^2\right].$$

With the purpose of equity in vibration influence upon the processing object in the angular and horizontal directions, the following congruence becomes relevant.

$$(7)\qquad 0.5K_5 i_{53}\left(z_5 - z_3\right)^2 = 0.5K_5 i_{52}\left(z_5 - x\right)^2,$$

from which

$$(8)\qquad \frac{i_{53}}{i_{52}} = \frac{\left(z_5 - x\right)^2}{\left(z_5 - z_3\right)^2},$$

or

$$(9)\qquad i_{23} = \frac{f_3}{f_2} = \frac{\left(z_5 - x\right)}{\left(z_5 - z_3\right)},$$

where i_{23} – gearing ratio, which characterizes the correlations of the vibration influence on the object in the vertical and horizontal directions.

As the analysis of the equation (6) shows, the value of the part of potential energy of the hydraulic chain, spent for the controlling over vibration excitatory does not exceed 10% (for powerful machines 0,5...1%). Considering the remarks and assumptions (7), equation (6) may be written as a system:

$$(10)\qquad \begin{cases} 0.45 \cdot 0.5K_5 z_5^2 = 0.5K_5 i_{53}\left(z_5 - z_3\right)^2 \\ 0.45 \cdot 0.5K_5 z_5^2 = 0.5K_5 i_{52}\left(z_5 - x\right)^2, \end{cases}$$

from which, considering (5) we receive:

$$(11)\qquad \begin{cases} i_{53} = 0.45\dfrac{z_5^2}{\left(z_5 - z_3\right)^2} = \dfrac{1}{0.45} \cdot \left(\dfrac{f_3}{f_5}\right)^2 \\ i_{52} = 0.45\dfrac{z_5^2}{\left(z_5 - x\right)^2} = \dfrac{1}{0.45} \cdot \left(\dfrac{f_2}{f_5}\right)^2, \end{cases}$$

To receive the single-valued solution for the system (3), it is necessary to add to it the equation of flow continuity:

$$(12)\qquad \begin{aligned} &Q_H + \dot{p}\left[W_0 + f_3 \cdot z_3 + f_2 \cdot x\right] \cdot \chi^{-1} = \\ &= f_3 \cdot \dot{z}_3 + f_2 \cdot \dot{x} + Q_B, \end{aligned}$$

where:

$$\dot{p} = \frac{d\left(\dfrac{K_5 \cdot z_5}{f_5}\right)}{dt} = f_5 \cdot \chi \cdot \left[W \cdot \dot{z}_5 - z_5 \cdot \left(f_3 \cdot \dot{z}_3 + f_2 \cdot \dot{x}\right)\right]W^{-2}$$

pressure change rate in the discharge chamber of vibration machine during the direct route of the slave cylinder; $W = W_0 + f_1 z_1 f_2 x$ – current volume; Q_B – discharge of fluid, which travels through the open vibration exciter.

Equation of the mass movement M_2 is of significant importance since it forms the law for movement of the executive chain. Interrelation between the line coordinate x and angle φ_2 may be compared, using the kinematic analysis of interrelation between the chains of the slide-crank mechanisms of the executive chain (Fig. 5).

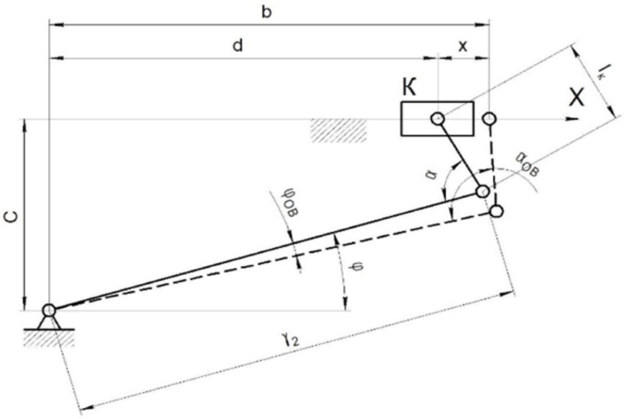

Fig. 5. Calculation diagram of the slide-crank mechanism of steering device

Having projected the lengths of the chains r_2 and l_K on the line of movement of the rotary joint center K and normal to it allows to write the equation of closed-loop system, made by the chains in the still condition:

$$(13)\qquad d = r_2 \cdot \cos\varphi - l_k \cdot \cos\left(\alpha - \varphi\right),$$

$$(14)\qquad c = r_2 \cdot \sin\varphi + l_k \cdot \sin\left(\alpha - \varphi\right),$$

and in the operational cycle

$$(15)\qquad b = r_2 \cdot \cos(\varphi - \varphi_{OB}) - l_k \cdot \cos[\alpha_{OB} - \left(\varphi - \varphi_{OB}\right)],$$

$$(16)\qquad c = r_2 \cdot \sin(\varphi - \varphi_{OB}) + l_k \cdot \sin[\alpha_{OB} - \left(\varphi - \varphi_{OB}\right)].$$

Deducting from (15) equation (16) we receive linear displacement $x = b - d$ of the slide block of the slide crank mechanism during the operation cycle. Dividing the received expression by l_k and designating $\theta = X/l_k$; $X = r_2/l_k$ we obtain:

(17) $$\theta = X \cdot \left[\cos\left(\varphi - \varphi_{OB}\right) - \cos\varphi\right] - -\cos\left[\alpha_{OB} - \left(\varphi - \varphi_{OB}\right)\right] + \cos\left(\alpha - \varphi\right).$$

Having divided the equation (16) by l_k, assuming that we obtain equation (16) in the dimensionless form:

(18) $$\omega = X \cdot \sin\left(\varphi - \varphi_{OB}\right) + \sin\left[\alpha_{OB} - \left(\varphi - \varphi_{OB}\right)\right].$$

Differentiating (17) and (18) we find:

(19) $$\theta = X \cdot \varphi_{OB} \cdot \sin\left(\varphi - \varphi_{OB}\right) + +\sin\left[\alpha_{OB} - \left(\varphi - \varphi_{OB}\right)\right] \cdot \left(\dot{\alpha} - \dot{\varphi}_{OB}\right),$$

(20) $$\dot{\alpha}_{OB} + \dot{\varphi}_{OB} = \frac{X \cdot \phi_{OB} \cdot \cos\left(\varphi - \varphi_{OB}\right)}{\left[-\cos\left[\alpha_{OB} - \left(\varphi - \varphi_{OB}\right)\right]\right]},$$

(21) $$\dot{\alpha}_{OB} = \left\{\frac{X \cdot \dot{\varphi}_{OB} \cdot \cos\left(\varphi - \varphi_{OB}\right)}{\left[\cos\left[\alpha_{OB} - \left(\varphi - \varphi_{OB}\right)\right]\right]}\right\} - 1.$$

Substituting (20) in (19) after transformation we receive:

(22) $$\dot{\varphi}_{OB} = \frac{\theta \cdot \cos\gamma}{X \cdot \sin\alpha_{OB}},$$

or

(23) $$\varphi_{OB} = i_M \cdot \theta,$$

where $i_M = \cos\gamma \cdot \left(X \cdot \sin\left(\beta - \gamma\right)\right)^{-1}$ - transfer function of the slide-crank mechanism; $\gamma = \alpha_{OB} - \left(\varphi - \varphi_{OB}\right)$

Differentiating (23) we determine the relations between the linear and angular acceleration:

(24) $$\varphi_{OB} = \left(i_M\right) \cdot \theta + i_M \cdot \theta,$$

where $\left(\dot{i}_M\right) = \dfrac{i_M \cdot \theta\left[\cos^2\gamma \cdot \cos\alpha_{OB} - \cos^2\left(\varphi - \varphi_{OB}\right)\right]}{X \cdot \sin^2\alpha_{OB} \cdot \cos\gamma}$ - derivative of the transfer function i_M on angles φ_{OB} and α_{OB}. With the consideration of the derivative value the expression (24) looks as follows

(25) $$\dot{\varphi}_{OB} = i_M \left\{\begin{array}{l}\dot{\theta} + \dfrac{\theta^2}{X \cdot \sin^2\alpha_{OB} \cdot \cos\gamma} \times \\ \times\left[\cos^2\gamma\cos\alpha_{OB} - \cos^2\left(\varphi - \varphi_{OB}\right)\right]\end{array}\right\}.$$

Due to the designing features of the rotary mechanism, the boundary changes of angles φ_{OB} and α_{OB} are insignificant, and it is possible to assume $\cos\gamma \approx \cos\left(\alpha - \varphi\right)$, and $\sin\alpha_{OB} \approx \sin\alpha$, then $i = \mathrm{const}$ and the dependence (25) looks as follows

(26) $$\varphi_{OB} = i_M \cdot \theta,$$

whence it follows that

(27) $$\varphi_{OB} = i_M \cdot \theta,$$

Reduced to the sliding block – piston plunger of the hydrocylinder of angular displacement the mass of the operation loop M_2, we find the equation of kinetic energy, which the operating link receives during the rotational motion:

(28) $$\frac{M_2 \cdot \dot{x}^2}{2} = \frac{M_3 \cdot \left(\dot{\varphi} \cdot r_2\right)^2}{2} + \frac{J_2 \cdot \dot{\varphi}_{OB}^2}{2} + +\frac{M_{BC} \cdot V_{BC}^2}{2} + \frac{J_{BC} \cdot w_{BC}^2}{2},$$

from which

(29) $$M_2 = \frac{M_3 \cdot \left(\dot{\varphi}_{OB} \cdot r_2\right)^2 + J_2 \cdot \dot{\varphi}_{OB}^2}{x^2} + +\frac{M_{BC} \cdot V_{BC}^2 + J_{BC} \cdot \omega_{BC}^2}{x^2},$$

where M_{BC}, J_{BC} – mass and mass moment of inertia of the crank rod (l_k); V_{BC}, ω_{BC} – linear and rotating speed of the crank rod.

With the consideration that $M_3 >> M_{BC}$ i $J_2 >> J_{BC}$, we receive more simplified formula of the expression for the determination of M_2.

(30) $$M_2 = \frac{\left(M_3 \cdot r_2^2 + J_2\right)^2 \cdot \dot{\varphi}_{OB}^2}{\dot{x}^2},$$

or

(31) $$M_2 = \frac{\left(M_3 \cdot r_2^2 + J_2\right)^2 \cdot i_M^2}{l_k^2}.$$

Considering equations (21...31) the motion M_2 can be written as follows:

(32) $$\frac{i_M^2}{l_k^2} \cdot \left(M_3 \cdot r_2^2 + J_2\right)^2 \cdot \dot{x} = i_{52}^2 K_5\left(z_5 - x\right) - -i_{52}^2 C_5\left(\dot{z}_5 - \dot{x}\right) - C_2\dot{x} - K_2\left(x_{02} + x\right) - -\frac{J_2 i_M \theta}{r_2^2} - R_{2a} sign\dot{x} - R_2 sign\dot{x}.$$

With the consideration of the expressions (26) i (27) the equation of motion of abrasive mass angularly will look as follows:

(33) $$J_{ma} \cdot i_M \cdot \theta_a = J_{m2} \cdot i_M \cdot \theta_a - C_{2a} i_M \theta_a - -K_{2a} i_M \theta_a - i_M R_{2a} sign\theta_a.$$

In the result the mathematical model the working path may be written as:

(34) $$\begin{cases} \dfrac{i_M^2}{l_k^2} \cdot \left(M_3 \cdot r_2^2 + J_2\right)^2 \cdot \dot{x} = i_{52}^2 K_5\left(z_5 - x\right) - \\ -i_{52}^2 C_5\left(\dot{z}_5 - \dot{x}\right) - C_2\dot{x} - K_2\left(x_{02} + x\right) - \\ -\dfrac{J_2 i_M \theta}{r_2^2} - R_{2a} sign\dot{x} - R_2 sign\dot{x}; \\ M_3\dot{z}_3 = i_{53}^2 K_5\left(z_5 - z_3\right) - i_{53}^2 C_5\left(\dot{z}_5 - \dot{z}_3\right) - C_{13}\dot{z}_3 - \\ -K_{13}\left(z_{03} + z_3\right) - R_{3a} sign z_3 - R_{13} sign z_3; \\ M_a z_a = M_3 z_3 - C_{3a} z_a - K_{3a} z_a - R_{3a} sign z_a; \\ J_{ma}\theta_a = J_{m2} \cdot \theta_a - C_{2a}\theta_a - K_{2a}\theta_a - R_{2a} sign\theta_a. \end{cases}$$

By the end of the stage, the pressure in the pipe line increases up to the value p_1 (valve opening pressure of the main cascade of vibration exciter)

(35) $$p_1 = \frac{K_5 \cdot z_{5\max}}{f_5},$$

where z_{5MAX} – peak strain of liquid, which corresponds to the end of the stage.

From the expression (35) we receive

(36) $$z_{5\max} = \frac{p_1 \cdot f_5}{K_5}.$$

Association between the elasticity of the abrasive material in the linear and angular directions may be accepted with the approximation as follows:

(37) $K_{2a} \cdot \phi_{OB} = K_{3a} \cdot z_a \cdot \mu_a,$

with the consideration $\phi_{OB} = \varphi_O = (i_m \cdot x)/l_k$,

(38) $$K_{2a} = \frac{K_{3a} \cdot z_a \cdot \mu_a \cdot l_k}{i_m \cdot x}.$$

The reverse motion of the operation loop and the lock valve of the vibration exciter, assuming that the motion of the elements starts simultaneously from the initial position, shall be described by analogical (29) system of differential equations, in which deformation of hydraulic loop changes from z_{5MAX} (deformation of the hydraulic loop at the end of the direct movement of the vibration machine loops) up to the value, which corresponds to the draining pressure in the hydraulic system, and then a new cycle of operating loop movement starts.

The aggregate of the system of differential equations which describe the movement of the operation loops of the vibromachine and PPG during the direct and the reverse movements is the mathematical model of hydro pulse drive of the vibromachine with the reverse-screw motion of the operation loop. Engineering methodic for design calculation for this drive had been developed on the base of the research of its mathematical model by the numerical method using computer. The adequacy of the mathematical model to the real dynamic system had been experimentally checked on the vibration machine of such type. The difference between the calculative and research values of parameters of vibration load (pressure, frequency, vibration amplitude) do not exceed (10...15)% [6-11].

Conlusions

Research of the experimental prototype of the unit with hydro pulse drive for vibroabrasive processing of pipe line inner surface, which took place on machine factory in the city of Kalynivka, proved the efficiency of this loading diagram when using hydro pulse drive. Technological results, received during the experimental researches of the experimental prototype of the unit are presented below. High silica sand and cast iron pellets were used as abrasive material. There had been depurated two kinds of surfaces, with and without scale. The research of vibroprocessing allowed determining that metal skimming takes place evenly during the whole period of processing with insignificant increase on the initial stage that is during the burnishing. In the results of the experiments there had been received the dependences between the speed of metal skimming, amplitude and vibration frequency. The experiment shows that the most efficient processing takes place when abrasive material fills up to 50% of the inner volume of the pipe under processing under following regimes: amplitude 4 mm, and frequency of 20 Hz [7, 9].

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