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Advance Movement Modelling and Simulation of the Electrodischarge Equipment

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Abstract: This paper presents the analytical modelling of the electro-hydraulic advance system of the electrodischarge processing machine. The input-state-output mathematical model allows numerical simulation in Matlab-Simulink, which is used to determine the system's behaviour in various working regimes, and especially in the transient regime. The study enables the determination of the system's characteristic parameters. These parameters are used in the optimal design of the advance servomechanism and in the control strategy.

Keywords: Electrodischarge, mathematical modelling, numerical simulation, control system.

1. INTRODUCTION

Electrodischarge is a processing method based on the discontinued and localized erosive effect of amorsed electrical discharges between the working electrode (+) and the work-piece electrode (Fig.1).

An erosive agent intermediates the contact between the two electrodes, and the advance system enables both necessary conditions for the local and spatial control of the process. Industrial practice indicates the use of various advance mechanisms which are based on the piezoelectric, electrothermic, electromagnetic, or the electromechanic effects. Golovanov et al (1999, 2000), Yahya et al (2003).

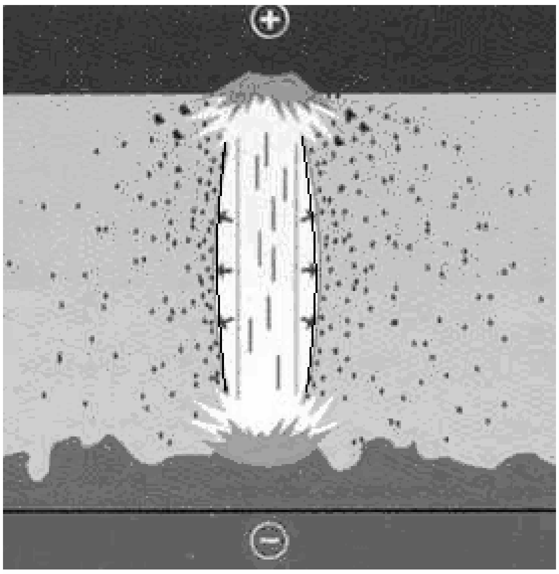

Fig. 1. Amorsed electrical discharges

In order to keep an optimal distance during operation it is required to have a relative closing motion between the two electrodes and an advance speed correlated to the erosion speed. The usual values of the working interstice are (10…100) μm for advance speeds of (0.01…1) mm/min.

2. MATHEMATICAL MODEL OF THE ELCTROHYDRAULIC ADVANCE SYSTEM

In electrodischarge processing practice, electrohydraulic systems are used in equipments which have a massive electrode, where change of the advance speed and the electrode's direction of motion are required. In these conditions, knowing the mathematical model of the advance system is essential for determining the behaviour in various working regimes, especially in the transient regime. In the process of designing the model, the comparison between the physical and the analytical models is essential, for the purpose of avoiding an imprecise model or one which is to simply structured. Dulau et al (2008), Voicu (2002).

There are harsh requirements for the mechanical assembly in order for it to provide small displacements without vibrations. In medium and large scale machines (e.g. Romanian type ELER machines) the drive is performed electrohydraulicaly, using a hydraulic engine with double effect piston.

According to Soaita et al (2006) and Technical Documentation 1, the components of the electrohydraulic servomechanism are: (Fig. 2)
- the electromechanical converter, having as output the angular displacement $\theta(t)$ of the choke, which is proportional to the value of the control current $i(t)$;
- a hydraulic module, which provides as output the difference in pressure $p(t) = p_1(t) - p_2(t)$ depending on the angular displacement of the choke;
- the hydraulic engine, having as input the control pressure and as output the displacement $z(t)$ of the piston's rod.

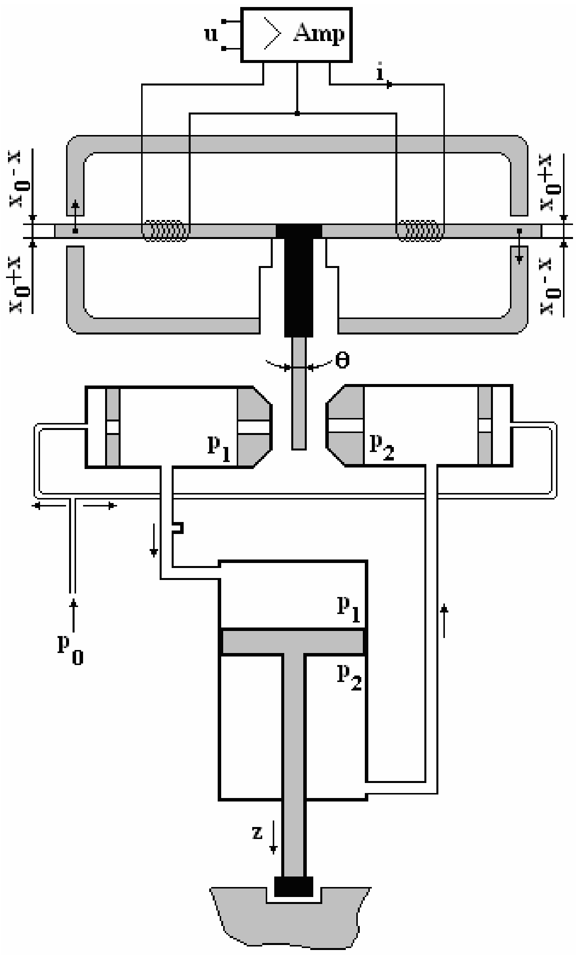

Fig. 2. The electrohydraulic advance system of the ELER-01 machine. (TechDoc)

In order to determine the differential equations that describe the dynamics of the electrohydraulic mechanism, the dynamics of each block is analyzed as well as the nature of the transformations that take place during operation.

Thus, for small displacements of the electromechanical converter's armature ($x/x_0 << 1$), the dynamic of the electrical control circuit is given by the differential equation:

$$2L_b \frac{di}{dt} + (R_b + R_i)i(t) + K_{em}\frac{d\theta}{dt} = u(t) \tag{1}$$

where: L_b, R_b represents the inductivity and resistance of each coil; R_i is the internal resistance of the amplifier Amp; K_{em} is the magnetic constant.

For small displacements around the position zero, the dependence between the control current $i(t)$ and the angular displacement $\theta(t)$ is obtained using the equation of the equilibrium of moments ∑M=0:

$$J_a \frac{d^2\theta}{dt^2} + C_a \frac{d\theta}{dt} + K_e\theta(t) = 4K_{em}i(t) - K_r p(t) \tag{2}$$

where: J_a is the inertia momentum of the mobile assembly; C_a is the viscous friction coefficient; K_r is a constant depending on the length of the choke; $p(t)$ is the pressure.

By considering the hydraulic module supplied with a constant pressure p_0 and the flow and continuity equations on a quasi-linear domain $\theta < 1/3\theta_0$, the following differential equation results:

$$T_p \frac{dp}{dt} + p(t) = K_{\theta p}\theta(t) - T_{dp}\frac{dz}{dt} + \frac{1}{6}p_0 \tag{3}$$

where: T_p is the time constant dependent on the size of the hydraulic system; $K_{\theta p}$ is the gain in pressure; T_{dp} is the time constant dependent on the supply pressure and the displacement of the choke.

In the case of the hydraulic engine operating vertically, the force of friction during load is considered and the phenomenon of air being dissolved by the oil at the change in pressure, is neglected. Under these assumptions, from the dynamic equilibrium at the piston's level, we get the displacement $z(t)$. For this, the following differential equation was considered:

$$M_r \frac{d^2z}{dt^2} + C_{fv}\frac{dz}{dt} = A_m(\alpha \cdot p_1 - p_2) - M_r g \tag{4}$$

where: M_r is the weight of the mobile subassembly; C_{fv} is the viscous friction coefficient; A_m represents the active surface of the piston.

The dynamic of the electro-hydraulic system is described using the general form of a linear input-state-output (ISO) model:

$$\begin{cases} \dot{\mathbf{x}}(t) = \mathbf{A}\mathbf{x}(t) + \mathbf{B}\mathbf{u}(t) \\ \mathbf{y}(t) = \mathbf{C}\mathbf{x}(t) + \mathbf{D}\mathbf{u}(t) \end{cases} \tag{5}$$

where: $\mathbf{A}, \mathbf{B}, \mathbf{C}, \mathbf{D}$ are the system matrixes; $\mathbf{x}(t)$ is the state vector; $\mathbf{y}(t)$ is the output vector; $\mathbf{u}(t)$ is the input vector.

The differential equations (1), (2), (3), (4) with $d\theta/dt = \omega$ and $dz/dt = v$, lead to (6), (7), (8), which show: the variation of the control current di/dt; the angular acceleration of the choke $d\omega/dt$; the angular velocity of the choke $d\theta/dt$; the variation of the control pressure dp/dt; the acceleration of the piston dv/dt; the speed of the piston dz/dt. The output $z(t)$, for $t > t_0$, is determined by the initial state $x(t_0)$ and by the control signal $u(t)$, for $t > t_0$. The initial state and the control signal determines all states $x(t)$, for $t > t_0$.

The ISO model (6), in the form given in (5) highlights the coefficient matrix **A**, the control matrix **B**, the disturbance matrix **P**, caused especially by the supply pressure, and the output matrix **C.**

$$\begin{cases} \dfrac{di}{dt} = -\dfrac{1}{T_c} i(t) - \dfrac{T_{dc}}{T_c}\dfrac{d\theta}{dt} + \dfrac{K_c}{T_c} u(t) \\ \dfrac{d\omega}{dt} = -\dfrac{2\beta}{T_m}\omega(t) - \dfrac{1}{T_m^{\ 2}}\theta(t) + \dfrac{K_m}{T_m^{\ 2}} i(t) - \dfrac{1}{T_m^{\ 2}}\cdot\dfrac{K_r}{K_e} p(t) \\ \dfrac{d\theta}{dt} = \omega(t) \\ \dfrac{dp}{dt} = -\dfrac{1}{T_p} p(t) + \dfrac{K_{\theta p}}{T_p}\theta(t) - \dfrac{T_{dp}}{T_p}\dfrac{dz}{dt} + \dfrac{1}{6T_p} p_0 \\ \dfrac{dv}{dt} = -\dfrac{1}{T_h} v(t) + \dfrac{K_h}{T_h} p(t) - \dfrac{K_g}{T_h} \\ \dfrac{dz}{dt} = v(t) \end{cases} \tag{6}$$

$$\mathbf{A} = \begin{bmatrix} \dfrac{-1}{T_c} & \dfrac{-T_{dc}}{T_c} & 0 & 0 & 0 & 0 \\ \dfrac{K_m}{T_m^2} & \dfrac{-2\beta}{T_m} & \dfrac{-1}{T_m^2} & \dfrac{-K_r}{T_m^2 K_e} & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & \dfrac{K_{\theta p}}{T_p} & \dfrac{-1}{T_p} & \dfrac{-T_{dp}}{T_p} & 0 \\ 0 & 0 & 0 & \dfrac{K_h}{T_h} & \dfrac{-1}{T_h} & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \tag{7}$$

$$\mathbf{B} = \begin{bmatrix} \dfrac{K_c}{T_c} \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix};\ \mathbf{P} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \dfrac{p_0}{6T_p} \\ \dfrac{-K_g}{T_h} \\ 0 \end{bmatrix};\ \mathbf{C}^T = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 1 \end{bmatrix} \tag{8}$$

In system (7), (8), all the coefficients have physical meaning and were determined either by analytic means or through experimentation (for the ELER-01 type machine): $T_c = 0.0223\,[s]$; $K_c = 0.0019\,[\Omega^{-1}]$; $T_{dc} = 0.0053\,[Nm/A\Omega]$; $T_m = 7.2482\cdot 10^{-5}\,[s]$; $K_m = 0.4085\,[rad/A]$; $\beta = 9.8502\cdot 10^{-7}$; $K_r = 3.2195\cdot 10^{-9}\,[m^3]$; $K_e = 26.5731\,[Nm/rad]$; $T_p = 0.1282\,[s]$; $T_{dp} = 4.5373\cdot 10^9\,[Ns/m^2]$; $T_h = 8.9853\,[s]$; $K_{\theta p} = 1.407\cdot 10^9\,[N/m^2 rad]$; $K_h = 4.7209\cdot 10^{-4}\,[m^2/Ns]$; $K_g = 442.1281\,[m/s]$.

In this way the evolution of the system for a given control input can be determined.

3. SIMULATION OF THE ADVANCE SYSTEM. EXPERIMENTAL RESULTS

The Matlab-Simulink environment provides libraries of component blocks, which support the parameterization needed for each mathematical model from the differential equations (6).

The analysis of the model is performed using various simulation algorithms. The linearization and determination of the equilibrium points was done using numerical integration methods (e.g. Runge-Kutta methods, *ode* functions) for which the beginning and end times were set. Mat1, Mat2.

The input-state-output causal transitions described in (6), allow a representation using a combination of graphical elements of proportional, integrative, derivative, summing and multiplying type, which offer an intuitive support for understanding the processing and the interactions of the signals.

Fig. 3 shows the Simulink model which was used to perform the simulations. Fig. 4 shows the simulation results for a step input with an amplitude of 1V and Fig. 5 shows the results for an input voltage profile defined by a sequence of step signals.

The outputs of the integrator elements represent the state variables: current, angular position, angular velocity, pressure, linear speed, position of the electrode.

The plots in the response figures show oscillations of the state variables and of the output variable, the position of the electrode. For the simulations an input voltage range of (1…5)V was considered. For these values, the control current varied up to a maximal limit of 10mA, which corresponds to a real admissible situation for the ELER-01 machine.

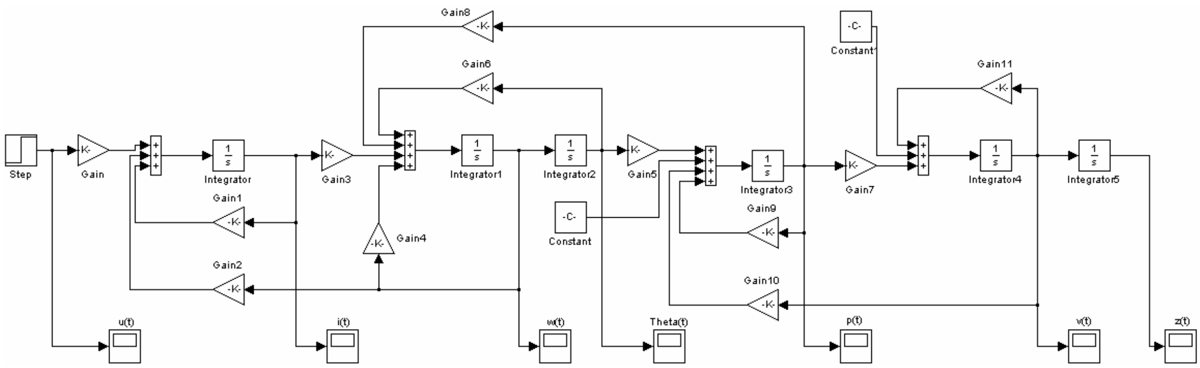

Fig. 3. The Matlab-Simulink model used for the simulation

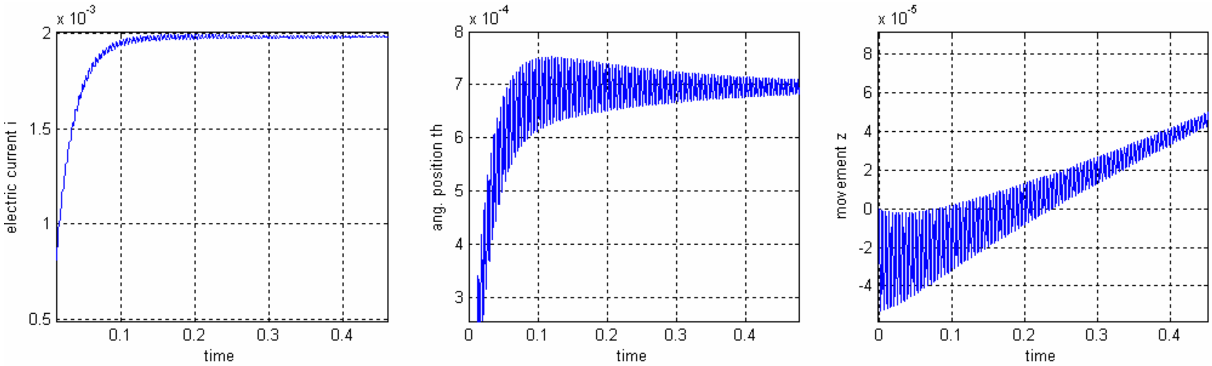

Fig. 4. Simulation response for a step input: current; angular position; position of the electrode

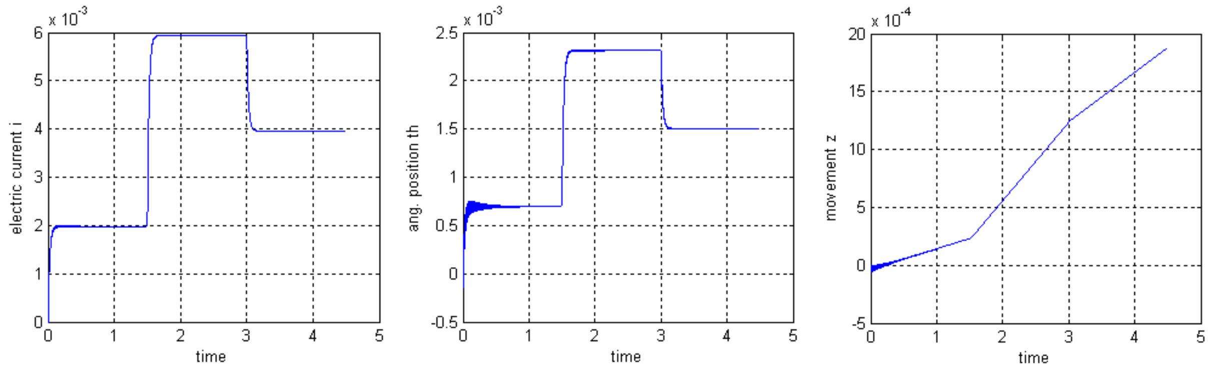

Fig. 5. Simulation response for an input profile: current; angular position; position of the electrode

To validate the results shown by the simulation, several experimental measurements of the dynamic advance system were performed under the following technical conditions:

- pressure supply: $6 \cdot 10^{6}\,[N/m^{2}]$;
- maximum displacement of the hydraulic engine: $1{,}7 \cdot 10^{-1}\,[m]$;
- coefficient of viscous friction: $9.1\,[Ns/m]$;
- measured voltage balance: $0.35 \div 0.38\,[V]$;
- control voltage: $4 \div 5\,[V]$;
- cylindrical electrodes made of copper;
- working interstice washed by a dielectric mixture.

Measurement of the electrode's position, during the processing, was conducted using the measuring equipment Numerom 303, with an analogical transducer (Olivetti inductosin type, normal linear metric: step 2 *mm*, cursor length 101.6 *mm*; accuracy class $\pm 1\mu m$). Data from the Numerom 303, was acquired using a PCL-812 universal interface, at 1ms sampling time. Experimental data collection is coordinated by PC, running a C language program.

During the experimental investigations, the technological conditions regarding the surface geometry, washing conditions of interstitial and electrical regime were maintained, as much as possible.

Fig. 6 shows the servomechanism's response, to a signal corresponding to the rapid advance of the electrode.

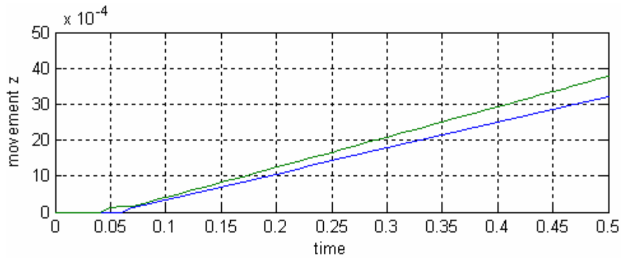

Fig.6. Servomechanism response to the rapid advance of the electrode

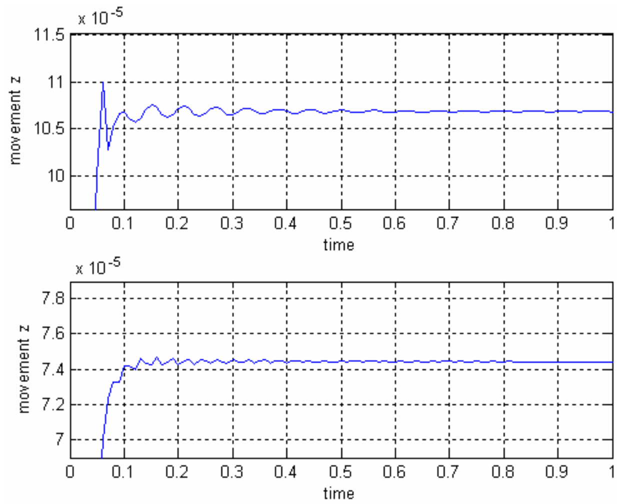

Fig.7. Servomechanism's response to a step input for sliding stop

Fig. 7 presents the servomechanism's dynamic using a step control input which is necessary to stop the electrode's sliding motion, due to the inertial load.

Fig. 8 shows the electrode's dynamic for a time of 0.72s (720 samples) during the processing.

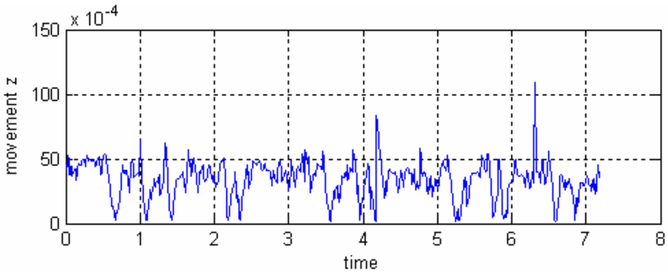

Fig.8. On-line electrode dynamic during the processing (24 μs work time, 4 μs pause time)

The linear displacement of the piston presents some oscillations, but these are bounded to the work interstice. In order to reduce the amplitude of the oscillations and to improve performance during operation, the implementation of control algorithms is required.

4. CONCLUSION

The stable evolution of the manufacturing process by electrical erosion depends on the capacity of the advance system to maintain the work interstice at values close to optimal during the whole operation. This way, the frequency of the electrical discharges in impulse, through the plasma column is maximum, the discharges by electrical arch are diminished and short circuits are avoided.

The capacity of the advance servomechanism to (re)establish rapidly and precisely the optimum interstice determines its quality and the technological performances of the machine.

The requirements for the advance mechanism are: speed, precision and stability. Unfortunately there are two contradictory conditions, great positioning accuracy, demanded by the productivity level, implies a reduced positioning speed and higher control rate.

Practically, the analysis of the simulation results and the direct tests using the ELER-01 type machine, allow a way to determine the optimal values for the gain, the viscous friction coefficient and the system's time constant.

The experiments and the simulations which were performed led to several observations and indications:

- the most elastic elements in the system, that accumulate potential energy, are the oil columns, that are subject to compression and relaxation;
- in the absence of power supply, the piston (the mobile electrode) should have a fixed position. In reality this doesn't happen and it is necessary to calculate an equilibrium voltage which fixes the piston at $v = 0$;
- it is required to oversize the execution element in the command chain as well as the driving element of the power circuit. This means a higher pressure falling on the servo-valve and a greater surface of the hydraulic engine, accomplished in the detriment of the transmission's energetic efficiency;
- the length of the working distance is narrowed down to the minimum necessary;
- it is necessary to reduce the dry friction, which for small speeds produces a discontinued motion.

For the next developing stage the use of the Simulink environment is suggested. It would allow the use of S-Functions, the inclusion of the simulation scheme in a closed control loop with negative feedback, testing the performances of the system with various control structures and the determination of means for reducing oscillations and for increasing the stability.

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