FACTORS AFFECTING THE ACCURACY OF OPERATION OF THE ELECTROHYDRAULIC ACTUATION POSITION SYSTEM WITH THROTTLING CONTROL

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The paper analyses most of the factors that influence the positioning error in the stationary mode of the electrohydraulic positional actuation systems and several ways to calculate the positioning accuracy in relation to the considered factors that influence the accuracy. Criteria are defined so that it is possible to check whether the desired accuracy of positioning can be achieved in the early design phase, but also the repeatability of accuracy over the full range of positioning. This is important to know, i.e. to define in the early design phase in relation to the basic limit performance of the actuation system, above all in relation to the bandwidth of the actuation system and to see if there are possibilities for algorithmic and/or structural correction of the positioning accuracy of the actuation system in the later stages of design. It starts from the basic 3-order linearized model, but in the analysis, factors of unmodeled dynamics are added in the 3-order linear model.

Keywords:
hydroelectric power plant,
ship lock,
hydraulic actuating system,
positioning error,
accuracy

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1 Introduction

One of the important tasks at hydroelectric power plants is to ensure the synchronous motion of two cylinders that operate the flat plate valve of the upper head of the ship lock system. The issue concerns the hydroelectric power plant Djerdap 1.

The electrohydraulic actuation system with throttling control is speed-driven in nature, i. e. the control distributor controls the direction of fluid movement, flow intensity and rate of flow change. Depending on which type of feedback loop we close, we get a positional, speed, force- and acceleration-controlled actuation system. The requirements for a positional actuation system are accuracy with repeatability, speed of response and stability. Accuracy is the difference between the set and desired position, the positioning error, and if we can maintain that value after repeated positioning with a certain precision and speed, then the repeatability feature is ensured. That speed must be such that it provides a sufficient reserve of stability, that is, the gain in the actuation system that ensures the speed of response during positioning is chosen in such a way that it does not impair the stability of the actuation system.

When designing an electrohydraulic actuation system, it is useful to know already in the first stages of designing whether the desired positioning accuracy can be ensured. Basically, every linear actuator (executive hydraulic cylinder) as a positional electrohydraulic system can be loaded inertially, positionally and with friction, individually or in combination with these three types of loads. The character of the load primarily affects the project solutions in order to ensure the required positioning accuracy.

2 Factors affecting positioning accuracy

In addition to the character of the load, the configuration of the electrohydraulic actuation system and the algorithmic solution for position control are the three basic groups of factors that affect positioning accuracy. When it comes to load, inertial and positional are dictated by the character of the drive mechanism whose drive member is the hydraulic cylinder as the executive organ of the electrohydraulic actuation system. Friction exists internally in the hydraulic cylinder and externally in the loading mechanism. Friction as a type of load is the most difficult to compensate with the control algorithm. There are some solutions with sliding surface mode variable structure algorithms, the application of which is limited. In terms of reduced impact on positioning accuracy for friction compensation, emphasis is placed on the quality of the hydraulic cylinder and on ensuring minimal friction in the external elements of the drive mechanism.

The configuration of the electrohydraulic actuation system, as another factor, is important in terms of the correct dimensioning of the hydraulic cylinder and the choice of the distributor with the required dynamic properties, which will be considered separately in the next chapter. The correct choice of the initial control algorithm is particularly important because it directly affects the shortest path to the optimal algorithmic solution for positioning accuracy, i.e. to compensate for the influence of the load and certain phenomena of the non-linear nature of the configuration of the electrohydraulic system, the consideration of which also follows. Within the nonlinear nature of the configuration of the electrohydraulic system, gap-type nonlinearity (hysteresis-type memory nonlinearity) must also be considered, which includes the influence of mechanical clearances and partial friction on the positioning accuracy of the electrohydraulic system and is most often the cause of the oscillatory behaviour of the positional response. The speed of the computer as well as the level of discretization also affects the positioning accuracy, which is also analysed by simulation.

3 Selection of distributor and hydraulic cylinder dimensions

The bandwidth of the actuation system is the best representation of the accuracy of the electrohydraulic actuation system in the initial stages of design, while respecting the compromise for the stability of the actuation system. There are two important elements for the performance of the actuation system that need to be reconciled in terms of achieving a sufficiently large bandwidth. It is the maximum speed of the hydraulic cylinder, dictated by the dynamics of the drive mechanism and the dimensions of the hydraulic cylinder dictated by the speed, but also the working pressure, which defines the elements that determine the bandwidth of the cylinder with the load that should be modeled as precisely as possible. In order to choose the distributor properly, we need to define the bandwidth of the actuation system. In this way, one component of accuracy is defined. The basic principle is that the dynamics of the actuation system must not affect the object where the actuator is the driving element of some mechanism of the object. There are experimentally verified recommendations for most facilities. Then we can simply arrive at the necessary dynamics of the distributor valve, through whose performance we can further consider the correctness of its selection and the impact on positioning accuracy, primarily through flow amplification and according to static stiffness conditions. It should be emphasized that the computationally-simulated bandwidths in practice are lower by about 27 % than actually possible [1].

In addition, here we have an open question whether to choose a distributor with a linear flow change or with a gap, which is again related to static hydraulic stiffness and its influence on positioning accuracy. Here also remains the dilemma of whether to "consume" a larger part of the algorithmic amplification through the amplification by the flow of the distributor. Both amplifications are contained in the amplification of the open circuit of the actuation system, Figure 1. This then leaves us with fewer opportunities to compensate for some elements of nonlinear phenomena of electrohydraulic positioning systems through greater algorithmic amplification.

With that in mind, there is also the application of a distributor with two amplifications, Figure 2, one for example for +/- 30 % of the displacement of the piston, the other, larger for the remaining part of the range, in order to have more opportunities for algorithmic amplification in case of smaller displacements for positioning, in order to have more accurate positioning.

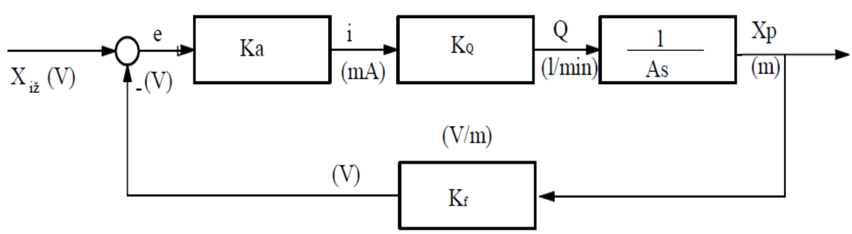

Figure 1: Basic structure diagram of electrohydraulic actuation position system.

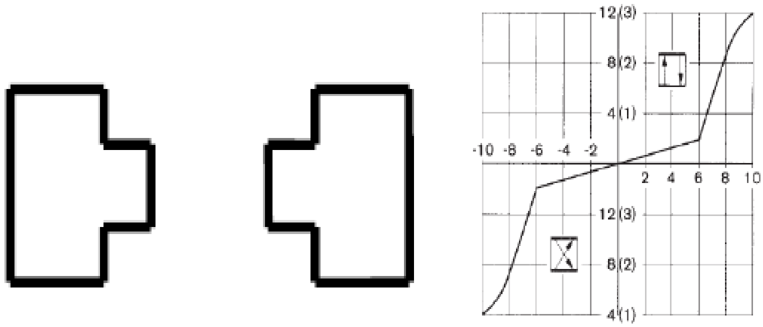

Figure 2: Two gain static characteristic of distributor.

$$K_{open} = \frac{K_a K_q K_f}{A}(1/s) \tag{1}$$

$$K_a = \frac{A K_{open}}{K_q K_f}(amp/volt) \tag{2}$$

Based on equation (1) and on the recommendations that exist in practice it follows:

$$K_{open1} = 0.1\omega_{cil\_opt\_nat\_freq} \tag{2.1}$$

$$K_{open2} = 0.4\omega_{servorazvodnik\_nat\_freq} \tag{2.2}$$

As can be seen from the above, the idea is to determine the algorithmic gain Ka. Simple transformations lead to basic expressions for analysis:

$$\frac{X_i(s)}{X_{iž}(s)} = \frac{G(s)}{1+G(s)H(s)} \tag{3}$$

$$G(s) = \frac{K_a K_q}{As} \qquad H(s) = K_f \tag{4}$$

$$\frac{X_i(s)}{X_{i\check{z}}(s)} = \frac{\frac{1}{K_f}}{\frac{A}{K_a K_q K_f}s + 1} = \frac{\frac{1}{K_f}}{\frac{1}{K_{open}}s + 1} = \frac{\frac{1}{K_f}}{Ts + 1} \tag{5}$$

$$T = \frac{1}{K_{open}} \sec \tag{6}$$

$$\frac{X_i(s)}{X_{i\check{z}}(s)} = \frac{1}{Ts + 1} \tag{7}$$

$$X_i = \frac{1}{K_f} X_{i\check{z}} \tag{8}$$

Previous equation represents the simplest mathematical model of the actuation system, which only allows us to check the maximum gain, ignoring the cylinder load. Based on the block diagram, Figure 1, we can write:

$$I = K_a K_f x_{err} \tag{9}$$

$$X_{i\check{z}} = 0 \tag{10}$$

$$I = K_a K_f x_{doz} \tag{11}$$

$$x_{doz\_\max} = \frac{0.05 I_N}{K_a K_f} \tag{12}$$

Based on the previous expression (12), we get an estimate of the positioning error by assuming that 5 % of the control signal is the minimum value sufficient to compensate for the asymmetry of the distributor and the negative impact of friction during positioning, i.e. that 5 % is the maximum hysteresis of the distributor, based on the previously calculated algorithmic gain.

In the following approach to positioning error estimation, we start from the assumption that we have external force data. Here we have taken 2 % as the control signal because for most servo and proportional distributors this is the gain value per pressure. This means that the position error caused by an external force can be compensated for by changing the pressure.

$$I_{ser\_raz} = 0.02 * I_N * \frac{F_{opter.}}{A * P_N} \tag{13}$$

$$x_{doz\_\max\_opter.} = \frac{I_{ser\_raz}}{K_a K_f} \tag{14}$$

$$x_{doz\_\max\_opter.} = 0.02 * \left[\frac{I_N}{K_a K_f}\right] * \left[\frac{F_{opter.}}{A * P_N}\right] \tag{15}$$

As can be seen from expression (15), already after 2 % of the control signal, the position error becomes a function of the external load. Figure 3 shows the experimental diagram of the pressure gain of the proportional-servo manifold, MOOG 671, which shows the asymmetry of the distributor when it comes to the pressure gain.

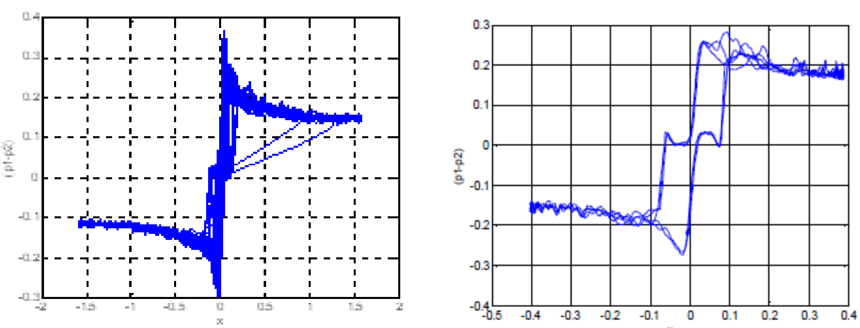

Figure 3: Experimental diagram for pressure gain asymmetry for MOOG 671 distributor.

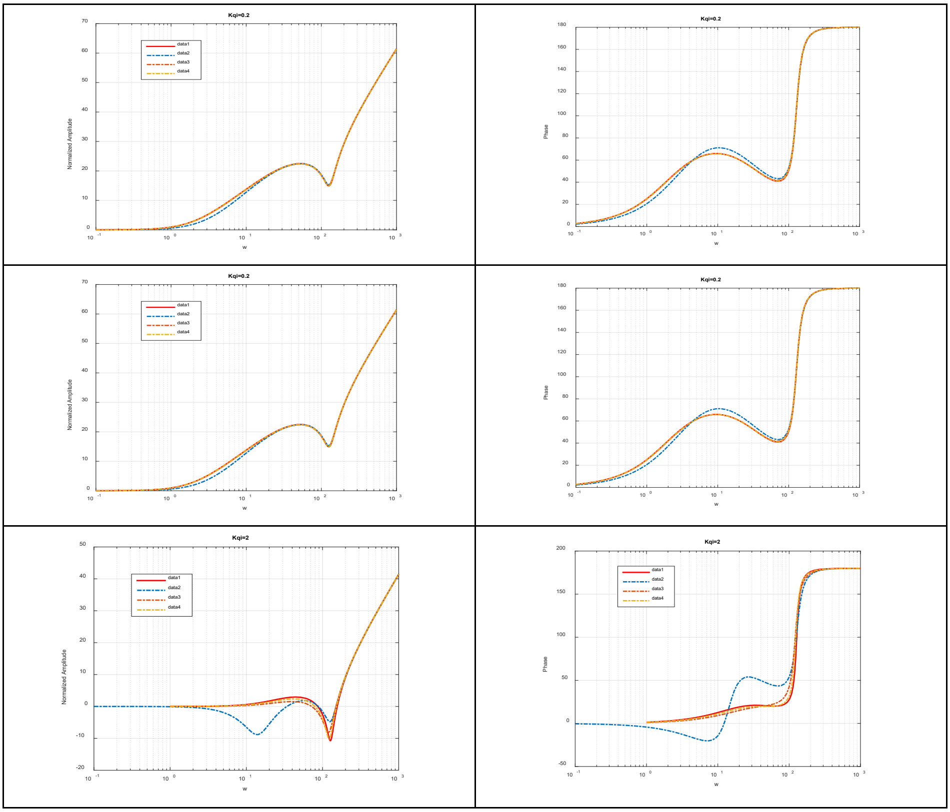

Figure 4: Results of simulation for static stiffness of distributor.

The static stiffness of the distributor was analysed by considering its dynamics. At this point we do not consider the influence of the overlap of the distributor. For the dead zone type nonlinearity, there are hardware and software solutions to compensate. The simulation check was done for three values of gain per flow and three values of the distributor bandwidth, as well as for the variant when the distributor bandwidth is not considered. The results are presented through frequency analysis. It can be seen from Figure 4 that high bandwidths and high gains per flow are critical.

Data 1: the distributor dynamics is neglected

Data 2: $\omega_v = 10$,data 3: $\omega_v = 130$, data 4: $\omega_v = 300$

4 Acceleration and deceleration time of the cylinder - influence on the accuracy of reaching the set position

The time of acceleration and deceleration also affects the positioning accuracy. There are several approaches to define this time. One is approximate, based on well-known relations:

$$\omega_n = \sqrt{\frac{40 \times E \times A_1}{h \times m}} \times \frac{1 + \sqrt{\frac{A_2}{A_1}}}{2} \tag{16}$$

$$t_{\min} = \frac{35}{\omega_n} \tag{17}$$

Basically, this time is also a criterion to determine the maximum bandwidth of the actuation system. Since this criterion is not unique, the values 18 or 35 are adopted, it is much more useful to take the following approach. Ramp time is based on [2]:

$$x = v_0 \left[1 - \frac{t}{t_r} \right], t \le t_r \tag{18}$$

$$y = C_0 + C_1 t + C_2 t^2 - C_0 e^{-\alpha t} \left[\cos(\beta t) + \frac{\alpha}{\beta} \sin(\beta t) \right] \tag{19}$$

$$e = \left| x_{t=t_r} - y_{t=t_r} \right| = \frac{v_0}{\omega_n^2 t_r} \left\{ 1 - e^{-\alpha t_r} \left[\cos(\beta t_r) + \frac{\alpha}{\beta} \sin(\beta t_r) \right] \right\} \tag{20}$$

$$e_{rel} = \frac{e}{x_{t=t_r}} = \frac{2}{\omega_n^2 t^2{}_r} \left\{ 1 - e^{-\alpha t_r} \left[\cos(\beta t_r) + \frac{\alpha}{\beta} \sin(\beta t_r) \right] \right\} \tag{21}$$

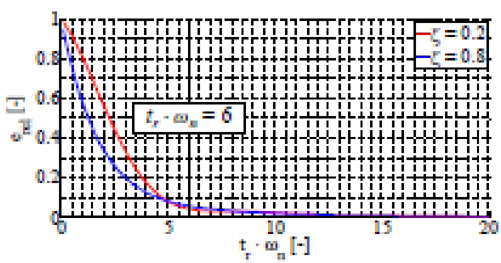

Figure 5: Graphic representation of the error for two limiting degrees of damping [2].

The point of intersection of these two curves defines the minimum time of deceleration:

$$t_r \geq \frac{6}{\omega_n} \quad . \tag{22}$$

Expression (17) is a very loose approximation. Expression (22) was obtained based on the previous settings for two dampings in the electrohydraulic actuation system. Although the two constants 35 and 18 are widely used, (22) is a more accurate limiting value.

In any case, the acceleration limits are important for the overall quality when designing an electrohydraulic actuation system. It should also be considered that in reality, the natural frequencies are lower than those obtained by calculation

5 Simulation verification of positioning accuracy

Simulation verification of positioning accuracy is suitable for certain fixed parameters (gains) to check the impact on positioning accuracy of the basic features of computer control, selection time and discretization resolution. This is done through the models shown in Figure 6, and the results are shown in Figure 7. In this case, only the computer component of the control and its influence on the positioning accuracy is checked.

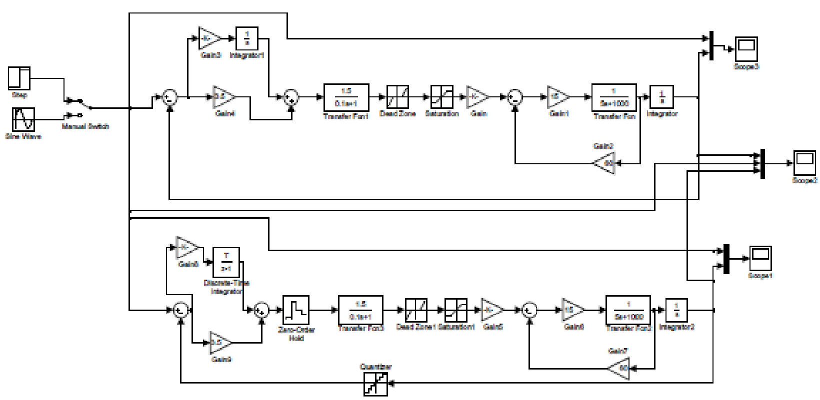

Figure 6: Basic MatLab simulation model.

The model shown in Figure 6 allows for the variation of a large number of simulation parameters, which are used to check the accuracy, sampling time, resolution of the discretization, gains, proportional and integral, the bandwidth of the directional valve as well as its gains. In addition, it is easy to switch from the constant reference option to the variable sinusoidal one.

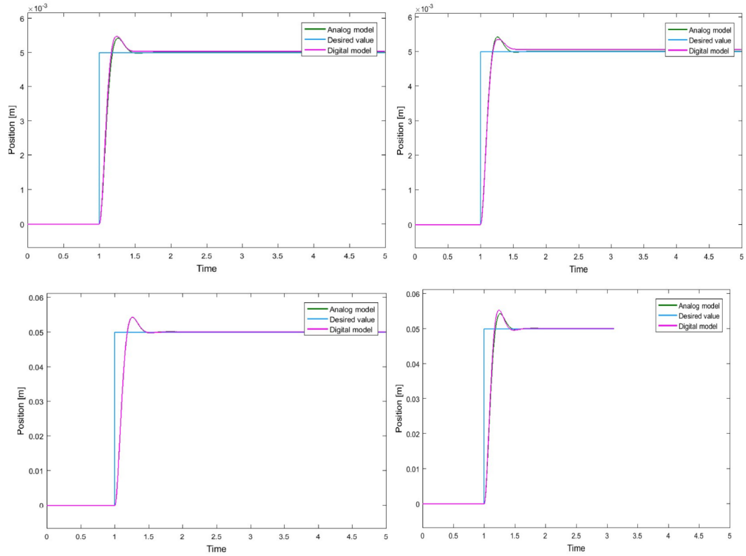

Figure 7a: Simulation results

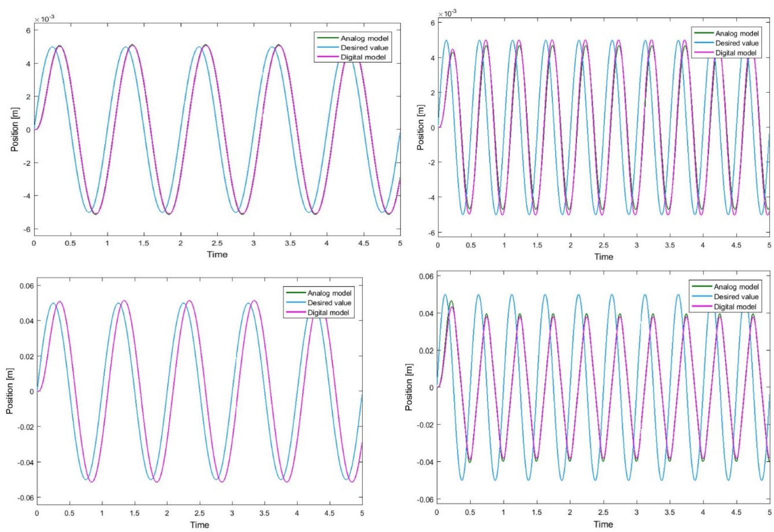

Figure 7b: Simulation results.

Simulation results depicted in Figures 7a and 7b, show the changes when varying the desired position value, 0.005 to 0.05 m (set point and amplitude). In the case of a sinusoidal change, we have two frequencies, 1 Hz and 2 Hz, (2 Hz need different gain). At steps in Figure 7a we have an average gain of 8.5 to 9.5. We see that for the selected computer parameters, the sampling time of 0.001 s (1 Hz) and the discretization of 0.0005 we have almost a coincidence of the „analog“ and „digital“ curves, small deviations are only at the desired value of 0.005 m there is a small difference and at 0.05 m we have no difference.

6 The initial optimal structure of the actuation system

The Figure 8 shows the structure that is shown in the literature [3] as optimal for the performance of the electrohydraulic actuation system, and thus for the issue of accuracy. Regarding the application of different control algorithms, there are many works that compare two or at most three algorithmic solutions, but there are few works that compare multiple algorithmic solutions with full experimental verification. One of these is [4], which graphically shows, Figure 9, the quality of algorithmic solutions through statistics for position control.

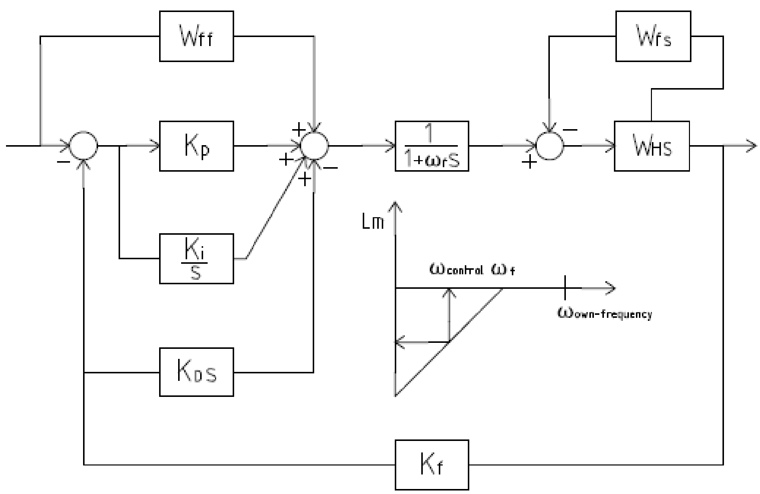

Figure 8: „Ideal“ structural diagram of electrohydraulic actuation system.

In the Figure 8, we clearly see that the structural PID algorithm is "broken", by putting the differential action in the feedback loop in order to avoid the negative influence of the differential action when we have the desired value of the set point type, when there is an impulse component in the response that generally generates an oscillatory positional response. In addition, the differential gain increases with frequency, so that certain components of the signal caused by disturbances can be severely amplified, so a filter block is necessary. Stabilizing feedback block shows that it is useful to use the resources of computer control and available sensors, in order to improve the response of the control system, increase the damping, by adding feedbacks primarily by pressure. A block with feed forward gains is necessary to reduce the effect of control change on positioning error. In addition to this structural solution, to improve the algorithmic solutions, it is useful to apply a cascade configuration for the actuation structure, so that we have an inner loop for control by force and an outer loop for control by position.

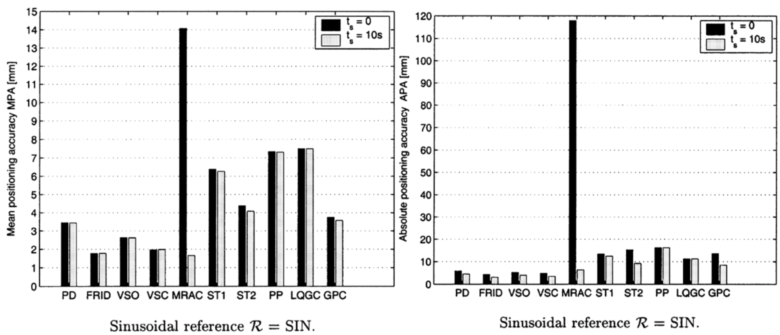

Figure 9: Mean positioning accuracy MPA of various controllers, for nominal plant P = NOM [4].

Designations of the considered control algorithms on the diagrams: proportional derivative (PD), acceleration feedback using an experimentally identified friction model (FRID), acceleration feedback using a variable structure friction observer (VSO),variable structure with sliding mode (VSC), model reference adaptive control (MRAC), self-tuning using a recursive least square parameter estimator (ST1), self-tuning using a Kalman filter for parameter estimation (ST2), pole placement (PP), linear quadratic Gaussian control (LQGC), self-tuning generalized predictive control (GPC). The diagrams (Figure 9) show the tracking error according to the criterion being defined. The MPA of a controller is defined as the root mean squared position error obtained for a reference signal, a plant condition, and averaged over a defined time interval. The APA of a controller is defined as the maximum absolute position error obtained for a reference signal, a plant condition, over a defined time interval. The time interval for tracking positioning error is 10 seconds.

7 Friction and mechanical backlash, influence on positioning accuracy

Friction as a mechanical phenomenon and the non-linear nature of friction have the greatest impact on positioning accuracy. The main problem is that friction cannot be measured directly, but only certain algorithmic compensations are possible, but with limited possibilities. Friction is a function of speed and is greatest at low speeds and when changing direction. That is why one of the tasks is to design the actuator system in such a way that the positioning speeds are as high as possible. Figure 10 shows an experimental recording for two frequencies, 02 Hz and 4 Hz, (friction force, red colour, blue colour position) for a hydraulic cylinder of size Ø200/125x650, where the difference in the effect of friction for two movement speeds of the hydraulic cylinder, for a time-varying position, is clearly visible. The basic question is what can be done in the initial design stages. The literature, [5] suggests that it is useful to test the model of the actuation system with some of the description of the friction nonlinearity with the PD control algorithm, and if we get satisfactory results, then we can expect better results with the variable structure algorithm or the observer with the friction model at a later stage.

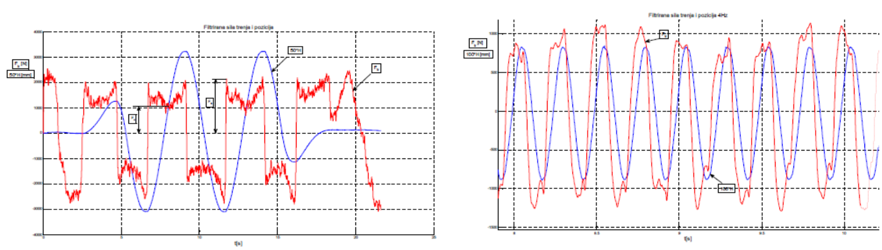

Figure 10: Experimental result for cylinder friction for two frequencies [6].

Backlash (gap) control, the clearance between the cylinder connecting rod and the drive mechanism, the positional electrohydraulic actuation system and the effect on the positioning accuracy is the least investigated in the large number of works that consider electrohydraulic actuation systems. There is no simple methodology that can be applied in the initial stages of designing electrohydraulic actuation systems for this analysis. There is experience that generally requires one to try the inverse nonlinearity at the level of simple linearized models and then add the nonlinearity to compensate for the gap nonlinearity in the control algorithm. It means a simulation check is required, while there are no general computational approaches. One of the main advantages of an electromechanical actuation system over an electrohydraulic one is that it is possible to generate a torque several times higher than the nominal torque for a short period of time, which compensates for the effect of friction. [7]

8 Error of synchronous motion of two hydraulic cylinders in relation to position accuracy – the example of slide gate of the upper head of the HPP "Djerdap 1" ship lock

The position accuracy affects another control function of electric hydraulic drives – the accuracy of synchronous motion of two cylinders. Here, the accuracy of synchronous motion of two cylinders for a slide gate will be analysed as a specific practical example. Slide gate is the most common steel construction within hydromechanical equipment of every hydroelectric power plant, be it a pre-turbine gate, a spillway gate or a ship lock gate. Only the initial error of synchronous motion is analysed which sometimes happens to be the maximum error, because the greatest disbalance of load is at the beginning of manipulation (one cylinder starts moving before the other, so a significant error is accumulated in the acceleration phase, Figure 11).

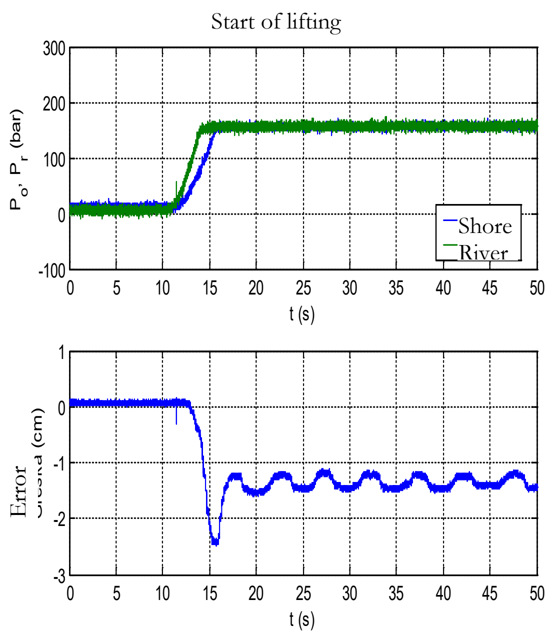

Figure 11: Pressure and synchronous motion error - slide gate of the upper head of HPP „Djerdap 1" ship lock.

It is important to be able to estimate the error in the initial phase of design of an electric hydraulic actuation drive using a mathematical model whose outline is given below [8]. The idea is to determine the necessary acceleration time for slide gate lifting based on initial error of synchronous motion, since it is important for the definition of basic parameters of electric hydraulic drive.

Cylinder motion time:

$$t = \frac{\Delta V}{Q} \tag{23}$$

Cylinder motion time delay (the difference between the beginning of motion of two cylinders due to the difference in load each of them is subjected to):

$$\Delta t = \frac{\Delta V_1}{Q_1} - \frac{\Delta V_2}{Q_2} \tag{24}$$

$$\Delta t = \frac{V\beta\left(p_1 - p_2\right)}{Q} \tag{25}$$

Force equation is given as follows

$$A_1 p_1 - A_2 p_2 = M_{red}\frac{d^2x}{dt^2} + f_{v\_fr}\frac{dx}{dt} + F_{fr} + F \tag{26}$$

Where:

$$p_2 = 81,6\frac{lv\rho A_2}{d^4}\frac{dx}{dt} \tag{27}$$

Flow equation discarding leakage is:

$$K_q X_v = A_1\frac{dx}{dt} + V_0\beta\frac{dp_1}{dt} \tag{28}$$

Combining equations (26), (27) and (28) and solving the equation using Laplace transformation yields the following:

$$x = \frac{K_q X_v}{A_1} - \frac{2K_q X_v M_{red} V_0 \beta}{A_1^3 T} + \frac{2K_q X_v M_{red} V_0 \beta}{A_1^3 T} e^{-\frac{t}{T}} \cos \omega t - \\ - \frac{K_q X_v \sqrt{M_{red} V_0 \beta}}{A_1^2} e^{-\frac{t}{T}} \sin \omega t \tag{29}$$

where

$$T = \frac{2M_{red}}{f_{fr} + 81{,}6 \dfrac{l \nu \rho A_2^2}{d^4}} \tag{30}$$

$$\omega = \frac{A_1 \sqrt{M_{red} V_0 \beta}}{M_{red} V_0 \beta} \tag{31}$$

or

$$x = \frac{K_q X_v}{A_1} - \frac{2K_q X_v M_{red} V_0 \beta}{A_1^3 T} - \frac{K_q X_v \sqrt{M_{red} V_0 \beta}}{A_1^2} e^{-\frac{t}{T}} \sin\left(\omega t - \varphi\right) \tag{32}$$

where:

$$\varphi = arctg \frac{2\sqrt{M_{red} V_0 \beta}}{A_1 T} \tag{33}$$

Substituting t with Δt (cylinder motion delay time for the cylinder subjected to greater load) in (32) yields the initial position error – distance already covered by the cylinder subjected to lesser load before the motion of the one subjected to greater load starts

$$\delta_{x_{init}} = \frac{K_q X_v}{A_1}\Delta t - \frac{2K_q X_v M_{red} V_0 \beta}{A_1^3 T} - \frac{K_q X_v \sqrt{M_{red} V_0 \beta}}{A_1^2} e^{-\frac{\Delta t}{T}} \sin\left(\omega \Delta t - \varphi\right) \tag{34}$$

Differentiating (29) and discarding the higher order terms, the piston velocity is defined as follows

$$v = \frac{K_q X_v}{A_1}\left(1 - e^{-\frac{t}{T}} \cos \omega t\right) \tag{35}$$

From (31) and (35) and assuming $t = 3T$ and $v \approx v_{\max}$, piston acceleration time to nominal velocity is defined as follows:

$$t_{acc} \approx \frac{6M_{red}}{f_{fr} + 81{,}6\dfrac{l\nu\rho A_2^2}{d^4}} \tag{36}$$

9 Conclusion

The design of an electro-hydraulic actuation position system, including the synthesis of the control algorithm, is a complex process and consists of several phases. In order to verify certain phases, it is necessary to check the basic parameters of the actuation system. One of the main basic parameters for an actuation positioning system is positioning accuracy. Here are several ways of checking the accuracy of positioning in the first phase of designing an electrohydraulic positioning system, in order to define the actuation system as precisely as possible at the beginning of the design, which in any case is an important condition for the rational and efficient design of an electrohydraulic actuation positioning system. The question of precisely defining the actuation system is also connected with the choice of the initial structure of the electrohydraulic actuation system.

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