Matching of energetic, mechanic and control characteristics of positioning actuator

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Abstract. The problem of preliminary choice of parameters of the automated drive power channel is discussed. The drive of the mechatronic complex divides into two main units – power and control. The first determines the energy capabilities and, as a rule, the overall dimensions of the complex. The sufficient capacity of the power unit is a necessary condition for successful solution of control tasks without excessive complication of the control system structure. Preliminary selection of parameters is carried out based on the condition of providing the necessary drive power. The proposed approach is based on: a research of a sufficiently developed but not excessive dynamic model of the power block with the help of a conditional test control system; a transition to a normalized model with the formation of similarity criteria; constructing the synthesis procedure.

1. Introduction
Positioning actuator consists of two main parts, i.e. power and control units (Fig.1). Its dynamics and specified tracking performance depend on coordination of characteristics both units. The complexity of the control system depends directly on energetics and mechanics of the power unit. In one’s turn the later determines the energetic activity, overall dimensions of an actuator as a whole device and their capability in generating of the modes desired.

![Figure 1. Positioning actuator.](image)

There are a lot of publications, where design procedure of the actuator is reduced to only control system design without optimization of the power unit. Hence it follows the great variety of relatively complicate control systems with the neurocontrollers, self-tuning fuzzy PID controllers etc., intended
for devices of the same type [1 -3]. But, on the other hand, there is very difficult to design both units taken as a whole because of the considerable complexity of actuator dynamics.

Some authors proposed a compromised methodology based on the separate but the interconnected designing procedure of the power and control units [4-7]. The reference mode of the output link motion, typical for the actuator designed, is taken as a basis of the methodology considered. As a reference mode for positioning actuator some kind point-to-point movement of the output link can be selected. The vibration mode can be used as a reference mode for the actuator operated in tracking mode, etc. In such cases the structure and parameters of the power unit may be determined on account for the degree of two modes (reference and actual mode) correlation on the base of an analysis of the actuator dynamic normalized model simulation in the specified mode. Information obtained is used further for the revealing of the basic relations between the similarity criteria of the system for the tradeoff solution within specified constraints.

2. The choice of parameters.

The parameters selection on account for two mentioned modes correlation, means that the actuator performance is tuned namely on the reference mode given. It gives the opportunity to use the reverence mode signal as a signal controlling the output link movement on the main part of the travel up to the end position. In some point near the end of the stroke, the reference mode signal is substituted automatically for any type specified positioning signal.

As stated before, the design procedure is based on the simulation results, in general case, of nonlinear dynamic model of the actuator. The model of the positioning actuator considered here, is presented in normalized dimensionless form. As a general requirements for point-to-point law of movement can be, e.g., specified travel time \( t_s \), near to zero velocity at the end of the process, gradually acceleration mode over the first half of the stroke and the gradually deceleration mode over the second half of the stroke. The law of movement describing as \( \dot{\gamma} = 0,5(1 - \cos \hat{\omega} \hat{\tau}) \) meets the requirements mentioned; \( \hat{\omega} = \pi / \hat{\tau}_f \) is normalized, conditional frequency of the process, \( \hat{\tau}_f \) is normalized travel time. The designing procedure is illustrated further by designing of the point-to-point positioning actuator with electric motor, linear mechanical characteristic, inertial and power loads.

The dynamic model \( \dot{\gamma} = \chi - \hat{\gamma} / \Lambda - \lambda_L \dot{\gamma} + \chi_L \) of the power unit is expressed in the output coordinates, where \( \chi = \gamma - \hat{\gamma} / \Lambda \) represents the reduced motor moving force. The terms \( \chi_L \) and \( \lambda_L \dot{\gamma} \) represent constant load and fluid friction load, where \( \lambda_L = 0,2 \Lambda \). The index value is in consequence of an fluid friction force makes \( \dot{\gamma} \) 1/10 of the actuator starting force when \( \dot{\gamma} = 0,5 \Lambda \). The variable \( \gamma \) characterizes the active part of the reduced motor moving force realized as a response to the input control signal \( \mathcal{F} \) with time delay \( \hat{\gamma} = (1 / \tau_e)(\mathcal{F} - \gamma) \), where \( \tau_e \) is time constant of the control system.

Using the Painlev'e test one can study the analytical properties as was suggested in reference [8-10].

The main purpose of the current study is to present the methodology of an application of the reference signal as a control signal in point-to-point positioning actuator. As a reference mode, mentioned above the law \( \dot{\gamma} = 0,5(1 - \cos \hat{\omega} \hat{\tau}) \) was selected. On the main part of the stroke (the region of active reference mode) the control signal \( \mathcal{F} = k_1(\dot{\gamma} - \gamma) + k_2(\dot{\gamma} - \gamma) \) is activated. Near the end point the reference signal is substituted for special positioning signal, for example, for \( \mathcal{F} = k_1(V_e - \gamma) - k_2\dot{\gamma}, \) where \( V_e \) is the end coordinate of positioning object. The dynamic model simulation showed, that the distance between the reference and actual modes depends on two main factors - travel time \( \tau_e \) (the main factor) and moving force \( \chi \) (that is a loop gain \( k_1 \)); decreasing of \( \tau_e \) leads to decreasing of the distance between curves mentioned and, as will be shown further, decreasing of the motor power needed. The
process performance depends, as well, on the similarity criteria of the system: normalized no-load velocity of the output link $\Lambda$ and load factors $\chi_L$ and $\lambda_L$.

Figure 3 illustrates a correlation between the reference and actual modes. It can be seen that curves are disposed fairly close each other since the magnitude of $\tau_s$ is close to its lower limit. Some diagrams showing total control process performance are represented in Figure 2. Here both parts of the control mode (reference and final control) are shown as integrated process for diverse $\chi_L$. Figure 4 gives a notion of the system performance on condition that the load is changed while the output link is moving. In that case, the load is changed sharply from zero to .4 and back to zero in the middle of the stroke. It can be seen that this action has prompted only a rather small violation in velocity curve while the displacement curve has no signs of the disturbance. This way can be used to study any other environmental effects on a system performance.

In conclusion the procedure of parameters estimation based on the simulation results is to be considered. The required power of the motor $W$ is determined in general case as [6 – 7]

$$W = P_x \dot{x}_h = \frac{m_x s_y^2}{t_x^3} \left( \Lambda \tau_s^3 + \frac{s_y^2 \dot{\tau}_x^2}{s^2} (\alpha_{xy} \tau_s / \Lambda) \right)$$

where $m_x, m_y$ are input and output inertial load, $s_y = y_e - y_i$, $t_x$ is a stroke time desired, $\dot{x}_h$ is a no-load velocity of an motor, $\alpha_{xy} = m_x / m_y$, $P_x$ is a starting moving force of a motor. If $\alpha_{xy} \approx 0$, then $W_{\min}$ is to be determined using a relationship $\Lambda \tau_s^3 = (\Lambda \tau_s^3)_{\min} \approx 23 + 400 \chi_L^2$, ($\chi_L = 0 - 0.4$). Mechanism gear ratio is defined according to $i = \dot{x}_h t_x / 3s_y$. The more the value of $\dot{x}_h$, the more noticeable is an effect of $\alpha_{xy}$. The $W \approx W_{\min}$ is defined in this case using the relation given above,
assuming \( \tau_s = 3 + 10 \chi_s^2 \) and \( \Lambda = 1 \). Mechanism gear ratio is defined using relationship \( i = \dot{x}_p t_s / s \tau_s \), where \( \tau_s = 3 + 10 \chi_s^2 \). The deigning procedure is considered in details in [6-7].

Methodology of the matching performance characteristics of the actuator is based on the assumption that the totality its performance modes can be represented by some only mode of general type. So the point-to-point positioning mode can be represented by any mode having desired time of the motion and monotonic profile of the transient processes on the symmetrical acceleration and deceleration parts. In continuous position (or other variable) tracking systems. The moving target following mode can be considered as some general mode as well.

3. Conclusion
In this paper is considered the point-to-point positioning actuator characterized by the general mode \( y = 0.5(1 - \cos \Omega t) \), where \( \Omega = \pi / t_s \) is the conditional frequency of the process, \( t_s \) is the stroke time, \( y \) is the displacement coordinate of the output link. The proposed mode proved to be very suitable since it contains only parameter \( \Omega \) and do not change its state when the initial and end positions are varied. It gives the possibility to employ the selected mode in cases when the initial conditions are varied. For reasons mentioned this mode was assigned as a basic mode and is called the reference mode.

The same reference mode was employed for the power unit parameters estimation of the point-to-point actuator with hydraulic motor, but not used for the fulfillment of the control functions [6,7]. In this paper the usage of the proposed reverence mode as control mode acting up to coming near the end position is considered. The graphs representing the positioning process performance shows the fairly accurate reproduction of the motion desired on account that normalized stroke time is over its lower limit. The reference mode reproduction is supplemented by relatively simple control mode, acting on the end part of the stroke. But it is separate problem to be solved.

It is to be noted that in the parameters estimation the dynamic model (linear or nonlinear) of the system was used intensively. The transition to the dimensionless form facilitates essentially the designing procedure and gives an additional chance for the matching of the dynamic and control performance of the actuator.

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References
[1] K. Nikzad, J. Chahoussi, L. Paul. Actuators dynamics and delay compensation using neurocontroller. Journ. of Engineering Mechanics Vol.122, N10, 1996.
[2] Zulfatman and M.F. Rahmat. Application of self-tuning fuzzy PID controller on industrial hydraulic actuator. International Journ. on smart sensing and intelligent systems, Vol.2, N2, June, 2009.
[3] Z. Wang, J. Shao, J. Lin, G, Han. Research on controller design and simulation of electro-hydraulic servo-system. Proc. of the 2009 IEEE International Conference on Mechatronics and Automation. August 9-12, Changchun, Chine.
[4] B.I. Petrov, B.I. Polkovnikov, L.B. Pobinovich. Dynamics of servo drives. Moscow, 1982, 496p.
[5] G.V. Kreinin and S.Yu. Misyurin. Dynamics and Synthesis of the parameters of a Positioning Drive. ISSN 1052-6188, Journal of Machinery Manufacture and Reliability, 2009, Vol. 38, No. 6, pp.523-530
[6] S. Yu. Misyurin and G. V. Kreinin. Power Optimization Criteria of a Mechanical Unit of an Automated Actuator. ISSN 1028_3358, Doklady Physics, 2015, Vol. 60, No. 1, pp. 15–18.
[7] S. Yu. Misyurin and G. V. Kreinin. Dynamics and design of a power unit with a hydraulic piston actuator. ISSN 1028_3358, Doklady Physics, 2016, Vol. 61, No. 7, pp. 354–359.
[8] N. A. Kudryashov. JOURNAL OF PHYSICS A-MATHEMATICAL AND GENERAL Volume:
35 Issue: 21 Pages: 4617-4632 Article Number: PII S0305-4470(02)33994-5 Published: MAY 31 2002.

[9] N. A. Kudryashov; A. S. Zakharchenko. APPLIED MATHEMATICS LETTERS Volume: 32 Pages: 53-56 Published: JUN 2014

[10] N. A. Kudryashov, M. V. Demina. APPLIED MATHEMATICS AND COMPUTATION Volume: 210 Issue: 2 Pages: 551-557 Published: APR 15 2009