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# Selection of settings of the PID controller by automatic tuning at the control system of the hydraulic fatigue stand

Leszek Kasprzyczak*, Ewald Macha

*Department of Mechanics and Machine Design, Opole University of Technology, Mikołajczyka St. 5, 45-271 Opole, Poland*

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**Abstract**

The paper presents a method of automatic tuning of PID controllers for the controlled stress, strain and energy parameter during structural material tests at the hydraulic fatigue stand. Settings of the controllers are determined for the given amplitude and phase margins of the system stability, for some values of the coefficient joining the integral and derivative times. Changes of the controller settings depending on amplitude and frequency of the reference signal have been determined.


*Keywords:* Mechatronic system; Hydraulic strength machine

## 1. Introduction

The mechatronic system for fatigue tests described in [1], using Matlab/Simulink software, requires suitable settings of the PID controller. The aim of this paper is to check possibilities of determination of the PID controller parameters in the fatigue test stand UFP 400 with the auto-tuning method and to test behaviour of the control system against the sinusoidal reference signal.

## 2. The strength test stand

### 2.1. Description of the UFP 400 machine

The UFP 400 machine is equipped with a hydraulic actuator; owing to that a specimen can be subjected to tension and compression with force up to $\pm 320\,\mathrm{kN}$ and frequency up to $5\,\mathrm{Hz}$. It is possible to load the specimen statically with force to $\pm 400\,\mathrm{kN}$. The test stand includes three sensors for measurements of force $F$, strain $\varepsilon$ and displacement $S$ (Fig. 1). Fatigue tests can be performed under controlled stress $\sigma(t) = F(t)/A$ (i.e. force $F(t)$ related to the specimen section area $A$), strain $\varepsilon(t)$, energy parameter $W(t)$, or other derivative signals resulting from mathematical operations on signals from the sensors of force and strain [1].

*Corresponding author. Tel.: +48 77 400 63 80; fax: +48 77 400 63 43.
*E-mail address:* l.kasprzyczak@po.opole.pl (L. Kasprzyczak).

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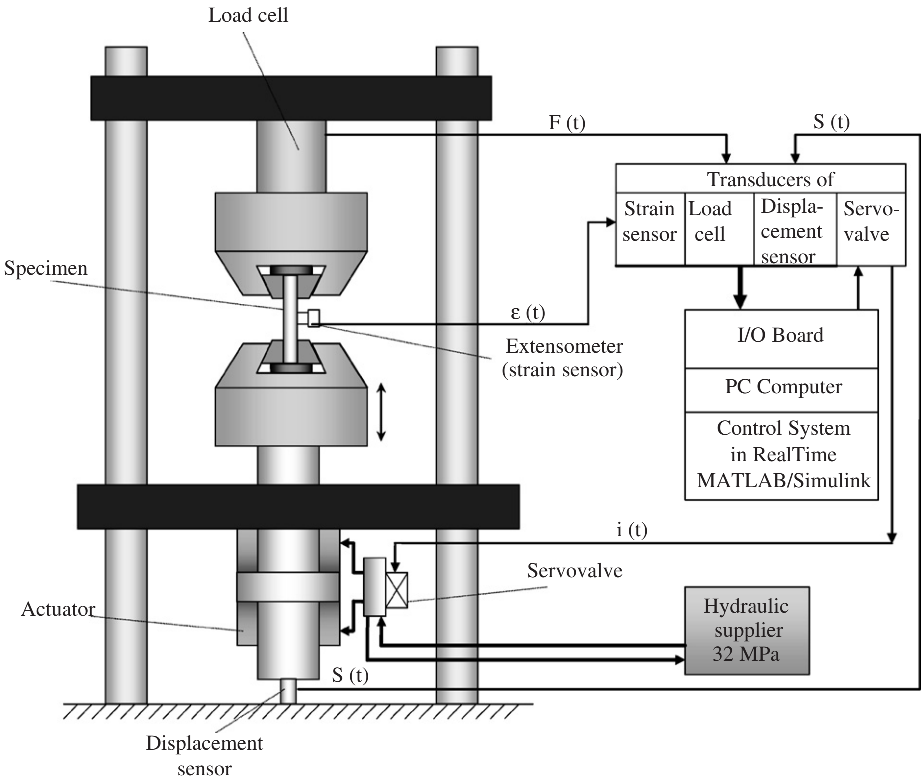


Fig. 1. Hydraulic test stand UFP 400 with a digital control system in Matlab/Simulink.

### 2.2. Mathematical model of the electrohydraulic servo drive

The electrohydraulic servo drive, equipped with an actuator and a servovalve, is an integrating plant with oscillations [2]. Its transfer function between the actuator piston rod displacement and the servovalve command signal is defined as

$$K_o(s) = K_{SV}(s)K_A(s) = \frac{k_{SV}\omega_{SV}^2}{s^2 + 2\xi_{SV}\omega_{SV}s + \omega_{SV}^2} \times \frac{k_A\omega_A^2}{s(s^2 + 2\xi_A\omega_A s + \omega_A^2)}, \tag{1}$$

where $K_{SV}(s)$ is the transfer function of the servovalve; $K_A(s)$ the transfer function of the actuator; $k_{SV}$ the servovalve gain coefficient; $k_A$ the actuator gain coefficient; $\omega_{SV}$ the servovalve natural frequency; $\omega_A$ the actuator natural frequency; $\xi_{SV}$ the servovalve damping ratio; and $\xi_A$ the actuator damping ratio.

The above linear model can be assumed in the case of negligence of structural servo drive nonlinearities. The transfer function coefficients for the servovalve are not constant and they depend, among others, on a pressure drop on the servovalve, the flow intensity, a value and a sign of the command signal and oil temperature. In the case of all the servo drive, the following factors should be also taken into account: a value of loading force or displaced mass, supplied pressure, a piston position, resistance to motion (friction) [2]. Fig. 2 shows the $s$-plane Nyquist contour characteristics of the servovalve, the actuator and all the servo drive, assuming constant parameters.

It is assumed that the plant under control of stress or strain in elastic range is linear. Then, the feedback loop is closed from the sensor of force or strain, respectively (Fig. 3). If the applied load is outside the range of