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Экспериментальное исследование динамических свойств канала нагружения стенда для прочностных испытаний авиационных конструкций*

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Проведение прочностных испытаний необходимо для повышения надежности летательных аппаратов, разработки и обоснования рекомендаций по эксплуатационному ресурсу в авиационной технике. С целью проведения ресурсных испытаний разрабатываются специальные стенды натурных испытаний, которые включают десятки гидроприводов, осуществляющих согласованное управление режимом силового нагружения элементов авиационных конструкций. Необходимость ускорения процесса проведения прочностных испытаний и повышения достоверности оценок для эксплуатационного ресурса элементов конструкций приводит к высоким требованиям на точность реализации заданных циклограмм силового нагружения конструкций. Рассматриваемая в статье задача связана с проблемой разработки аппаратных и программных средств для автоматизации процесса проведения прочностных испытаний в авиационной технике. Построение адекватной процессу испытаний математической модели стенда необходимо для правильного выбора структуры и параметров алгоритма управления стендом, что в конечном итоге позволяет обеспечить повышение точности реализации заданных циклограмм силового нагружения. В частности, в данной работе ставится и решается задача проведения предварительных экспериментальных исследований динамических свойств специально разработанного одноканального стенда для прочностных испытаний. В работе приведено описание стенда электрогидравлического привода. Представлены описание методик проведения экспериментальных исследований характеристик стенда и полученные результаты экспериментов. Рассмотрена упрощенная математическая модель стенда в виде интегратора со звеном транспортного запаздывания в канале управления. На основе проведенных экспериментов получены оценки для параметров математической модели, которые могут быть использованы в дальнейшем при решении задачи синтеза регуляторов для стенда силового нагружения.

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Ключевые слова: авиационные конструкции, испытания на прочность, испытательный стенд, эксперимент, статические испытания, система управления, сервопривод, математическая модель

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INTRODUCTION

Accelerated strength testing of aircraft structural elements is an important part of improving the aircraft reliability and guaranteeing of the specified service life time in aeronautical engineering [1–4]. Special testing benches are used in order to investigate an endurance capability of aircraft structural elements. Such testing benches can include dozens of hydraulic drives [5–8].

Preliminary studies of dynamical properties of hydraulic drive in order to its mathematical model derivation and refinement of drive parameters allow improving efficacy of control systems which are used in loading force test operations. Construction of such drive mathematical model, which should be adequate to testing process, is necessary for the correct choice of a control algorithm structure and selection of controller parameters. Finally, improvement in control system performance gives a possibility to accelerate dynamic mechanical strength testing, raise an accuracy of mission loading profile implementation, and enhance the reliability of estimates for the service life time of structural elements in aeronautical engineering.

In order to do a study of dynamical properties of hydraulic drive, the special loading force single channel stand was designed where electro-hydraulic control based on the servomechanism and software is provided [5]. The stand allows the experimental study of different versions of control algorithm and conduct comparative analysis of the closed-loop system properties.

In this paper the hardware configuration of the developed strength testing stand is described, the results of stand based experimental studies are presented, the structure and parameters of the mathematical model for the stand are derived based on experimental results, variation ranges of model parameters are estimated.

1. TESTING STAND CONFIGURATION

The designed strength testing bench consists of the following subsystems:

– Mechanical loading system
– Hydraulic loading system
– Loading control system
– Control system of loading force
– Emergency protection control system

Mechanical part of the loading force system generates and distributes the efforts of the executive hydraulic drive to the testing object. The hydraulic system and control system provide the desired force loading of the testing object. Single cylinder shaft unilateral or bilateral action is used as the executive mechanism. Loading control for testing object in accordance with the assigned mission loading profile is carried by system loading control system. General view of the discussed stand for strength testing is shown in Fig. 1.

Fig. 1. Single channel testing stand

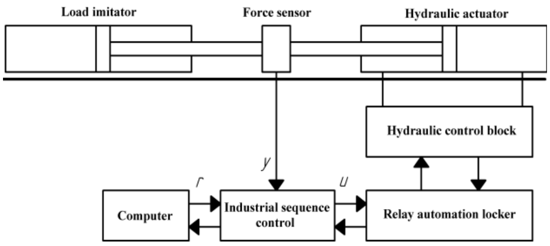

Fig. 2. Block diagram of the testing stand

The block diagram of the strength testing stand, which includes hydraulic, mechanical stress associated with imitator, is showed in Fig. 2 where magnitude of loaded force is measured by the dynamometer and the output signal $y(t)$ is sent to controller. Control algorithm is implemented by industrial controller and control signal $u(t)$ is sent to hydraulic servomechanism. The control unit provides spool valve management in order to regulate the flow of high pressure oil in the hydraulic cylinder.

2. INVESTIGATION OF STAND WITHOUT FEEDBACK

The hydraulic cylinder and the control block are included into hydraulic drive. The dynamic characteristics of the hydraulic drive depend on the constructive execution and spool valve [5,9]. The degree of the detailed elaboration for dynamic characteristics of the hydraulic drive is essentially dependent on the requirements for speed and precision of the hydraulic drive system. To evaluate the structure of the mathematical model for the hydraulic servo stand, series of open-loop tests were performed, where the control action is generated in the form of a stepped signal as shown in Fig.3. A digital to analog converter controller generates a signal in the range from –10V to + 10V where the level of 10V is treated as 100 %. So, the range of control action in Fig. 3 is presented in %. The output of the testing bench is provided by the power hydraulic actuator, where the loading force $y(t)$ is measured by a dynamometer. The calibrating force measuring is provided by the dynamometer where such unit as “kgf” is used. The range of efforts could be developed at this testing bench is from -10^4kgf to +10^4kgf.

Experimental results on the testing stand during the open loop test are shown in Fig.3 where the control input has the form of a meander with signal amplitude given by $u_0 = \pm 5\%$. A separate part of the plots from Fig.3 is selected and presented in Fig.4.

From test results in case of the open-loop system shown in Figs. 3, 4 it follows that the output response (loading force) of the stand to a step input is of the form of the sawtooth signal that corresponds to the step response of an integrator, where the gain of the integrator can be estimated from the slope of the ramp signal. At the same time a more detailed examination of the output signal $y(t)$ shows that there are both nonlinearity and additional inertia in the dynamic characteristics of the hydraulic drive. By taking into account the relative smallness of the speed for the generated loading profile, assume that the simplified mathematical model of the hydraulic drive can be treated as the integrator with the gain g and time delay τ in control variable, that is

$$\dot{y}(t) = g\,u(t-\tau)\,. \tag{1}$$

Assume that the change in the slope of the graph $y(t)$ in Fig. 4 during the constant value of the input signal $u(t)$ is caused by the change of the gain g. Then, from graphs in Fig. 4, we obtain an estimate for the value of the gain g based on the following relationship:

$$g = \frac{y(t_2) - y(t_1) - \Delta\cdot(t_2 - t_1)}{u_0\cdot(t_2 - t_1)}\,. \tag{2}$$

Take, for example, $t_2 = 7.5$ sec. and $t_1 = 6$ sec. The value of Δ is defined as the constant output signal trend on the graph $y(t)$ in Fig. 3.

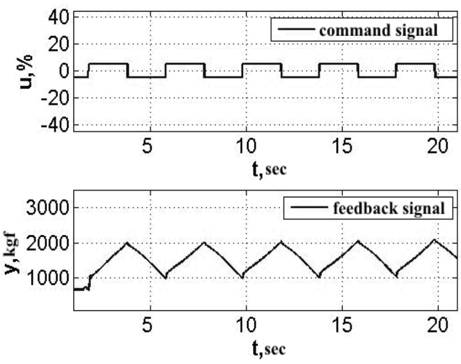

Fig. 3. The experimental results on the testing stand with the open loop and the control signal amplitude given by $u_0 = \pm 5$ %

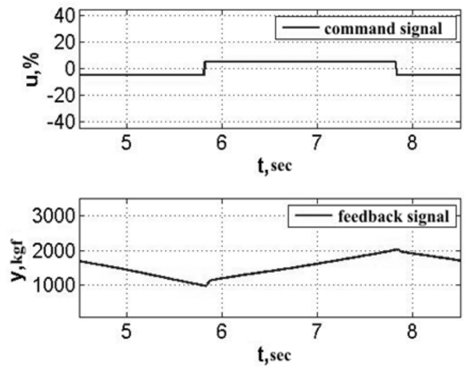

Fig. 4. The experimental results on the testing stand with the open loop and the control signal amplitude given by $u_0 = \pm 5$ % in the time range from 4 to 9 sec

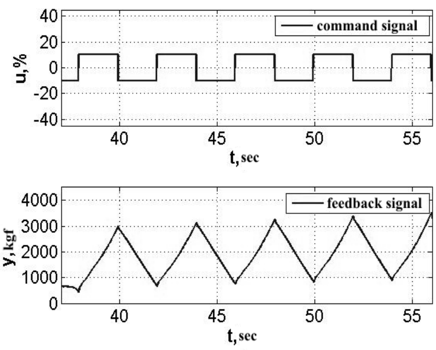

Fig. 5. The experimental results on the testing stand with the open loop and the control signal amplitude given by $u_0 = \pm 10$ %

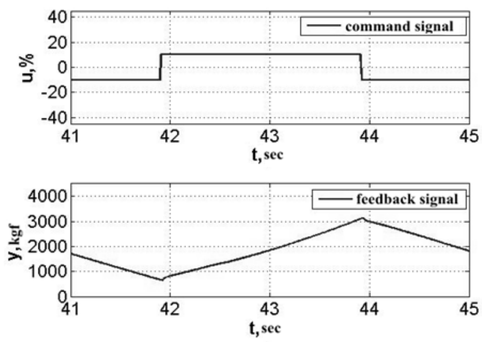

Fig. 6. The experimental results on the testing stand with the open loop and the control signal amplitude given by $u_0 = \pm 10$ % in the time range from 41 to 45 sec

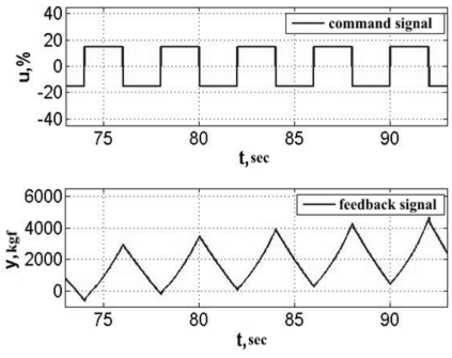

Fig. 7. The experimental results on the testing stand with the open loop and the control signal amplitude given by $u_0 = \pm 15$ %

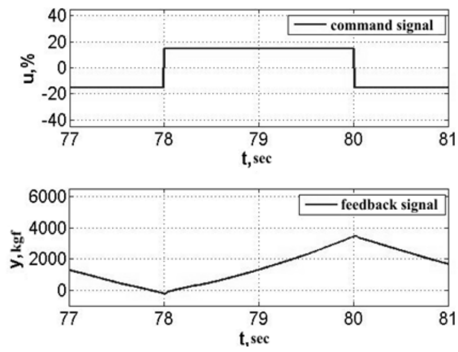

Fig. 8. The experimental results on the testing stand with the open loop and the control signal amplitude given by $u_0 = \pm 15$ % in the time range from 77 to 81 sec

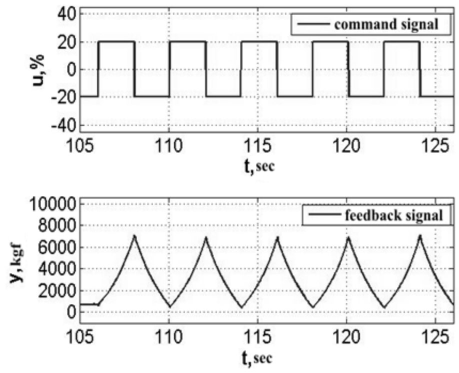

Fig. 9. The experimental results on the testing stand with the open loop and the control signal amplitude given by $u_0 = \pm 20$ %

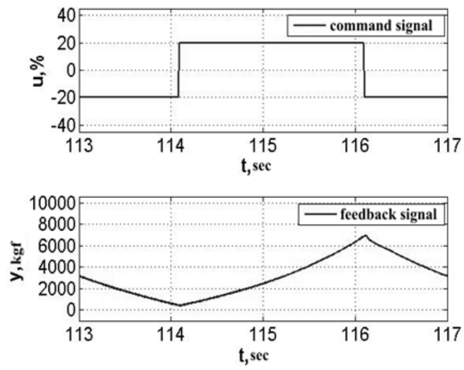

Fig. 10. The experimental results on the testing stand with the open loop and the control signal amplitude given by $u_0 = \pm 20$ % in the time range from 113 to 117 sec

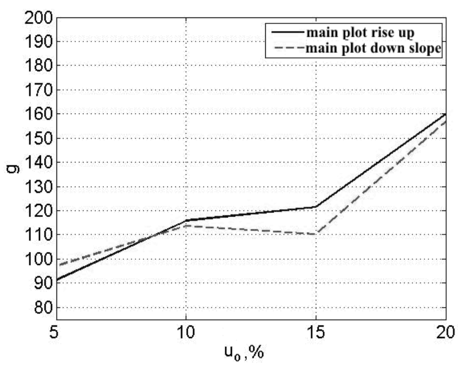

Fig. 11. Magnitude estimates for the gain g by (2) during the main portion for different values of u_0

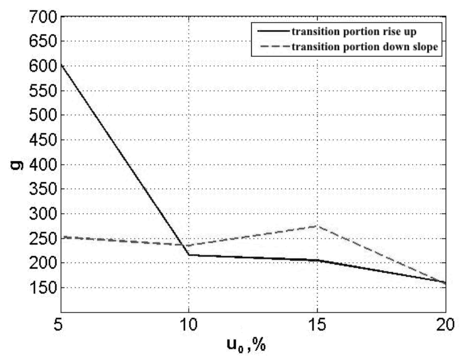

Fig. 12. Magnitude estimates for the gain g by (2) during the transition portion for different values of u_0

From the graphs in Fig.4 it is clear to see that the estimate of the gain g by (2) essentially depends on the selected time instances t_2 and t_1. In particular, four separated portions can be identified on the graph $y(t)$ such as the main portion "rise up", the main portion "down slope", transitional portion "rise up", transitional portion "down slope". The estimates of the gain g by the relationship (2) for each portion were evaluated based on the simulation results in Figs.3, 4.

In order to determine the influence of u_0 on the derived estimates of the gain g by the relationship (2), additional experiments were performed when $u_0 = 10\%$, $u_0 = 15\%$, and $u_0 = 20\%$. These experimental results on the testing stand with the open loop are presented on Figs. 5–10.

Based on the experimental results shown on Fig.3-10 and the expression (2) final relations of the estimates g from u_0 were derived which are presented in Fig.11 and Fig.12. From conducted experiments it can be concluded that the value of the gain g during the transitional portion (Fig.12) is substantially greater than one is during the main portion (Fig.11). The effect of the transitional portion is decreased with the increase of the input signal amplitude u_0.

3. INVESTIGATION OF STAND WITHOUT FEEDBACK

Consider the other method of estimating the value of the gain g in the model (1) based on analysis of amplitude frequency responses of the testing stand. The experimental scheme is shown in Fig. 13 where we assume that $r(t) = A_r \sin(\omega t)$ and the gain k is chosen experimentally such that the system on Fig.13 is stable.

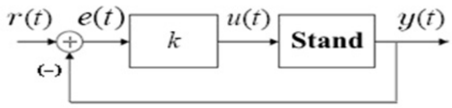

Fig. 13. The scheme of the experiment for the stand with a proportional feedback

At stationary regime, by neglecting the effects of nonlinearity characteristics of the stand, control action and output value of the stand are considered as harmonic signals $u(t) = A_u \sin(\omega t + \varphi_u)$, $y(t) = A_y \sin(\omega t + \varphi_y)$.

If the mathematical model (1) of the stand is valid, then the relation $A_y = g\omega^{-1} A_u$ for stationary regime is satisfied. Hence, from test results according to the scheme in Fig.13, the following estimate for the gain g can be obtained:

$$g = A_y \omega / A_u . \tag{3}$$

As an example of test results from the scheme on Fig. 13, graphs for $u(t)$ and $y(t)$ are shown on Fig.14 which have been obtained when $k = 0.005$, $A_r = 2000$ [kgf], $\omega = 0.6283$ [rad/sec].

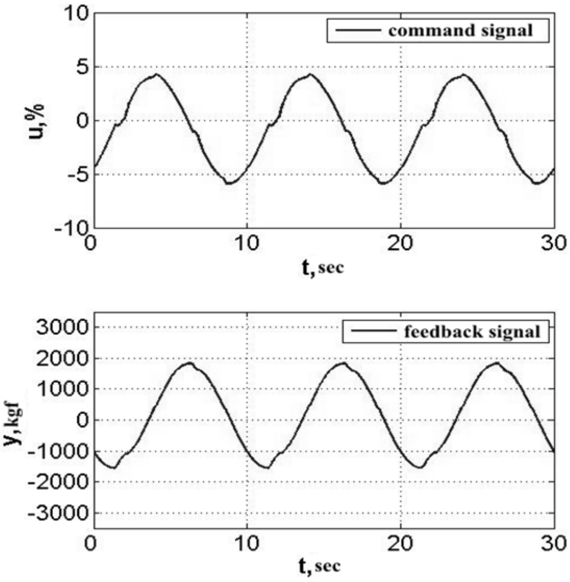

Fig. 14. Graphs $u(t)$ and $y(t)$ obtained as a result of an experiment to stand with proportional feedback and harmonic reference input $r(t)$

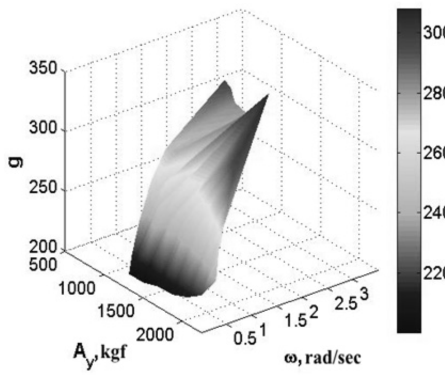

Fig. 15. The results of the experiment to the stand with a proportional feedback and harmonic reference input $r(t)$, main view

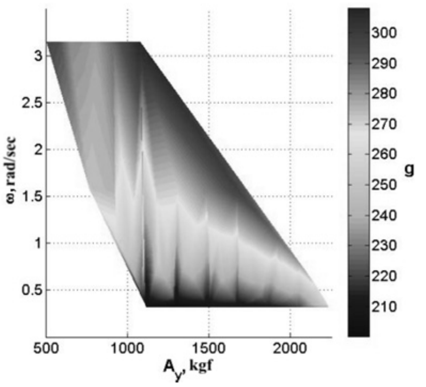

Fig. 16. The results of the experiment to the stand with a proportional feedback and harmonic reference input $r(t)$, view from the axis g

A set of experiments for different values of frequency ω and amplitude A_r is executed and then the estimates for the gain g were obtained based on relationship (3). Final results are represented by graphics that are shown in Figs. 15–18. In particular, the results on Fig. 15 clearly shows that with an increase in the frequency ω of the harmonic reference input $r(t)$ there is a tendency to increase the value of the gain g, which can be regarded as an increase in the role of the transitional portion which was discussed in the previous section.

4. RESEARCH OF STAND WITH RELAY FEEDBACK

In order to obtain estimates for the value of the time delay τ in the mathematical model (1) of the testing stand, additional experiments on the stand with the relay in the feedback loop were conducted in accordance with the scheme shown in Fig. 19 where $u = u_0 \text{sign}(e)$.

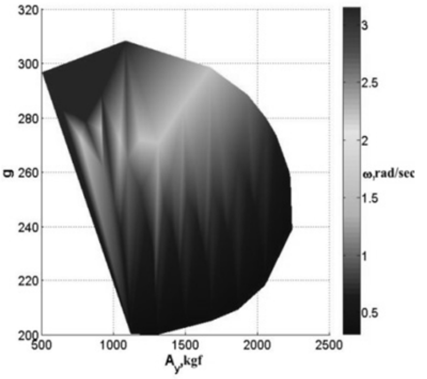

Fig. 17. The results of the experiment to the stand with a proportional feedback and harmonic reference input $r(t)$, view from the axis ω

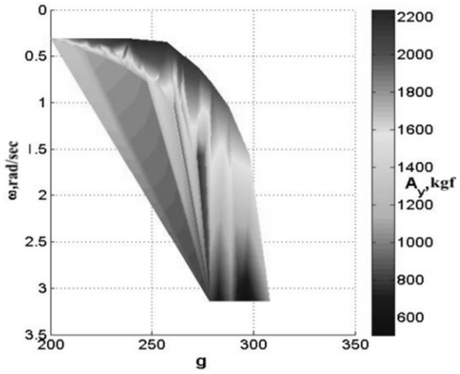

Fig. 18. The results of the experiment to the stand with a proportional feedback and harmonic reference input $r(t)$, view from the axis A_y

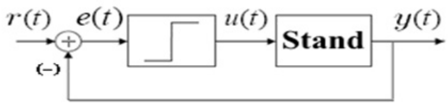

Fig. 19. The scheme of the experiment to the stand with relay in feedback loop

Introduction of the relay in control loop leads to a self-oscillating mode in the system on Fig. 19. If the mathematical model (1) of the stand is valid, then the parameters of the self-oscillation mode can be estimated based on the describing function method [10] under assumption that the stand passes vibrations of the fundamental frequency and suppresses the higher harmonics.

Assume that $r(t) = 0 \ \forall t$, and consider the first harmonic in the waveform at the output of the stand, that is

$$y(t) = A_y \sin(\omega t).$$

So, in accordance with the model (1), the 1st order harmonic balance equation yields:

$$g = \frac{A_y \pi \omega}{4u_0}, \quad \tau = \frac{\pi}{2\omega}. \tag{4}$$

As an example of test results from the scheme on Fig.19, graphs of experimental results $u(t)$ and $y(t)$ are shown in Fig.20 which were obtained when $r(t)=0\ \forall t$, $u_0=10\%$.

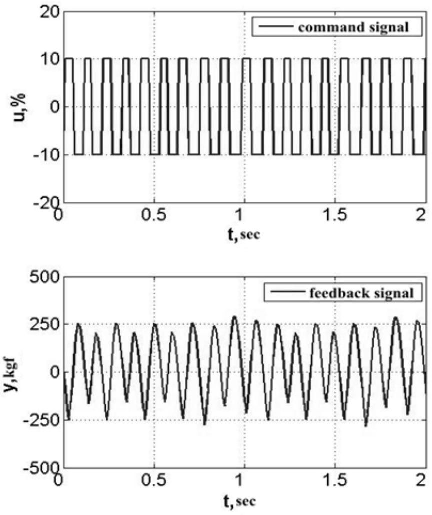

Fig. 20. Graphs $u(t)$ and $y(t)$ obtained as a result of an experiment to stand with relay in the feedback loop

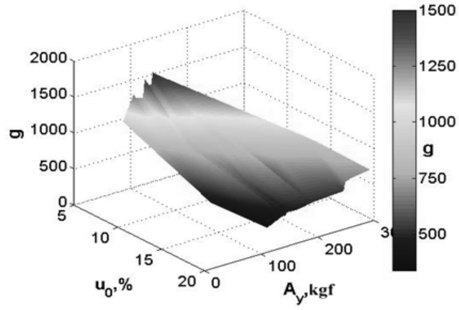

Fig. 21. The results of the experiment to stand with relay in the feedback loop, main view

A set of experiments for the stand with relay in the feedback loop is executed when $u_0=5\%$, $u_0=10\%$, $u_0=15\%$, and $u_0=20\%$, as well as in case of the different constant reference input signal r. Range for variations of r is selected from -2500 to 2500 in step of 500. Final estimates for g and τ based on relationships (4) and from experimental results in accordance with the scheme on Fig.19 are shown in Figs.21-23. Note that the width of the strip in vertical direction in Fig.22 and Fig.23 is caused by the influence of r.

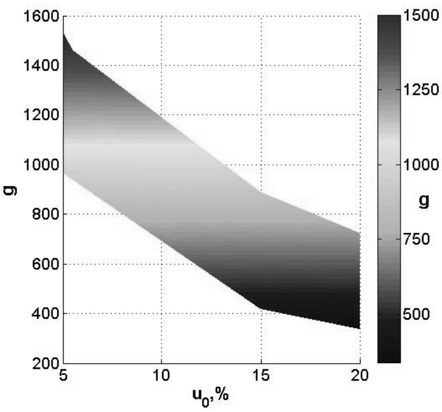

Fig. 22. The values on the estimates g based on the relationships (4) for different values u_0, view from the axis A_y

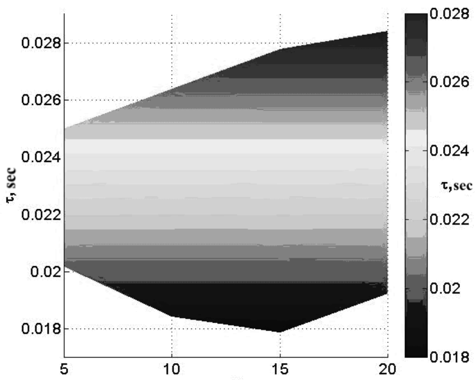

Fig. 23. The values on the estimates τ based on the relationships (4) for different values u_0

DISCUSSIONS

From the results of experiments on Figs.3-10 it follows that the change of the direction in the movement of the hydraulic cylinder rod brings to appearance of the short transitional portion in the output response where the transitional portion corresponds to a significant increase of the gain g in the model (1) of the stand. Research of stand amplitude frequency response in accordance with the experimental scheme in Fig.13 gives the average estimate for the gain g and does not allow identifying the change of g in the transitional portion. However, the experiments based on scheme in Fig.13 reveal a general trend to increase g with an increase in the reference input frequency. In particular, the dependence on Fig.15 can be used to adjust the gain of the controller while changing the reference input frequency. The emergence of self-oscillation mode during the experiment on the stand with relay in the feedback loop (Fig.19) confirms the need for the introduction to the mathematical model of the stand such additional dynamics as, for example, time delay component. Test results for the scheme of Fig.19 allow estimating the amount of equivalent delay for the model given by (1). Plot in Fig.23 shows that the time delay τ is slightly dependent on the experimental conditions. The increasing of g (see Fig.21) when the relay amplitude u_0 is decreased in the experiment on Fig.19 is also consistent with the results in Fig.12 and can be explained by an increase in the transitional portion contribution to the processes in the closed-loop system.

These studies reveal the cause of the high self-oscillating processes in the case when the direction in the movement of the hydraulic cylinder rod is changed, which took place in the testing stand with PI-controller [11]. In accordance with the presented results, follow-up research will be aimed to develop a method of controller design that allow to provide high accuracy realization of the mission loading profile and at the same time to eliminate the occurrence of high-frequency self-oscillating processes in the case when the direction in the movement of the hydraulic cylinder rod is changed. In particular, an approach based on adaptive gain tuning and time scaling [12,13] can be applied in order to resolve this problem.

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*Experimental study of dynamical properties of a stand loading channel for strength tests of aircraft structures**

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Strength tests are necessary to increase aircraft reliability, to develop and validate recommendations on the aeronautical equipment service life. To carry out fatigue tests special full-scale test stands which include tens of hydraulic actuators exercising a coordinated control of the power loading mode of aircraft structural elements are developed. The need to accelerate the process of strength tests and to increase the reliability of estimates for structural element service life leads to high requirements to the implementation accuracy of structure power loading cyclograms. The task discussed in the paper is related to the problem of hardware and software development for the automation of the strength test process in the aeronautical equipment. It is necessary to develop a stand mathematical model that is adequate to the test process to make the right choice of the structure and parameters of the stand control algorithm, which finally increases the implementation accuracy of power loading cyclograms. In particular, the problem of preliminary experimental studies of dynamic properties of a specially developed single-channel stand for strength tests is set and solved in this paper. The description of an electrohydraulic drive stand is provided in the work. The description of techniques of stand characteristics experimental studies and the test results obtained are presented. A simplified mathematical model of the stand in the form of an integrator with a link of transport delay in the control path is considered. Estimates of the mathematical model parameters which can be used further to solve the problem of synthesizing power loading stand regulators are obtained on the basis of the tests.

Keywords: Aircraft design, strength testing, test stand, experimental results, fatigue properties, control system, servo drive, mathematical model

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