692. Verification of active controlled air spring model

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Abstract. In previous works we have described the model of actively controlled bellow-type air spring for compensation system of the vibration-isolation platform with gyroscopic stabilizer. Air pressure inside the air spring is controlled by the electro-magnetic valve, which is controlled by a signal from the displacement sensor. The experimental stand was designed and used for verification of the model. The paper is aimed on description of the model and experiments conducted for its verification.

Keywords: active control, air spring, electro-pneumatic valve model, PID controller.

Introduction

The new concept of the compensation system (described in [1]) drive for vibration-isolation platform with gyroscopic stabilizer, introduced in [2], consists of a pair of actively controlled bellow-type air springs. The separate compensation torque motor was substituted by this way and system of the vibration-isolation platform with gyroscopic stabilizer was simplified. The soundness of this concept was proven by numeric simulations with the mathematical model (see [3]). New pneumatic drive model contains the model of air spring and the model of the control valve. Air spring model has been used on our department for a long time and we can consider it to be reliable. The model of the control valve is a new one and must be validated. Experimental stand model, model of control valve and experiments for its verification will be described bellow in this paper. Results of simulations and experiments will be compared.

The experimental stand and its mathematical model

As was mentioned above the experimental stand was designed for purposes of verification of mathematical model of compensation system drive. The simple one-mass model was chosen – the mass is mounted on the end of arm, which is supported by bellow-type air spring similar type as is considered for vibration-isolation platform. The arm is connected by joint to the frame (see Fig. 1. and 2.). Design of experimental stand with rotary mounted mass was chosen, instead of linearly guided mass directly supported by the air spring, because its construction is simpler. Air spring is connected to the control valve. The SMC electro-pneumatic pressure regulator VY1A00 (electronically adjustable requested pressure) was chosen for this purpose. Actually it is an electro-pneumatic 3-port solenoid valve, which is controlled by integrated controller. Position of the arm is measured by contactless laser optical sensor of displacement optoNCDT 1302. The position PID controller is fed by signal from displacement sensor and provides value, which corresponds with requested pressure p_r for the input of the pressure regulator.

Motion of mechanical part of the experimental stand system is described by equation:

$$me \cdot \ddot{\varphi}(t) + b_{pas}\dot{\varphi}(t) = -meg + a\left(p(t) - p_a\right)S_{ef}\left(\varphi(t)\right) \tag{1}$$

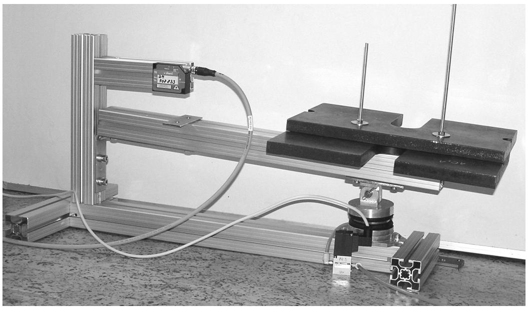

Fig. 1. Experimental stand

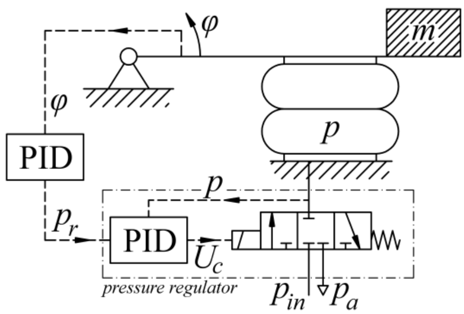

Fig. 2. Scheme of experimental stand

where m, b_{pas}, p_a and g stand for the mass, damping of passive resistances, atmospheric pressure outside the air spring and the gravitational acceleration; $\varphi(t)$ is the angular displacement of the arm; $p(t)$ stands for the pressure inside the air spring; e and a are dimensions of the mechanism; S_{ef} is the effective area of the air spring dependent on the air spring length. Pressure inside the spring is described by the equation derived from the ideal gas law by time differentiating when isotherm process is considered:

$$\dot{p}(t)\cdot V\left(x(t)\right)+p(t)\cdot S_{ef}\left(x(t)\right)\cdot\dot{x}(t)=GRT \tag{2}$$

where V is for volume of air inside the air spring; $x(t)$ stands for the length of the air spring; R stands for gas constant; T is for thermodynamic temperature. For the bellow type air spring is considered the simple relation between the effective area and the volume inside the spring:

$$S_{ef}(x)=\frac{dV(x)}{dx} \tag{3}$$

The quantity marked G on the right side of the equation (2) stands for the mass flow of air in direction into or from the air spring and is driven by the control valve.

Air mass flow through the valve from the source of pressured air to the air spring can be expressed (according to ISO6358 standard):

$$G_{PA}=\begin{cases} p_{in}\cdot C\left(U_c\right)\cdot\rho_a\sqrt{1-\dfrac{\left(\dfrac{p(t)}{p_{in}}-b_{PA}\right)^2}{\left(1-b_{PA}\right)^2}} & \dfrac{p(t)}{p_{in}}>b_{PA} \\ p_{in}\cdot C\left(U_c\right)\cdot\rho_a & \dfrac{p(t)}{p_{in}}\le b_{PA} \end{cases} \tag{4}$$

similarly the air mass flow from the air spring to the atmosphere can be expressed:

$$G_{AR}=\begin{cases} -p(t)\cdot C\left(U_c\right)\cdot\rho_a\sqrt{1-\dfrac{\left(\dfrac{p_a}{p(t)}-b_{AR}\right)^2}{\left(1-b_{AR}\right)^2}} & \dfrac{p_a}{p(t)}>b_{AR} \\ -p(t)\cdot C\left(U_c\right)\cdot\rho_a & \dfrac{p_a}{p(t)}\le b_{AR} \end{cases} \tag{5}$$

where the first part of both expressions (4) and (5) stands for subsonic flow and the second part is for choked flow; p_{in} is the pressure of the pressured air source; p_a is the atmospheric pressure;

ρ_a is air density at atmospheric pressure; $p(t)$ is pressure of air inside the pneumatic spring; C is pneumatic conductivity of valve dependent on valve opening (must be measured for real valve complying with ISO6358 standard); b_{PA} and b_{AR} are critical pressure ratios in appropriate directions (must be measured for real valve – ISO6358).

Valve opening is actually dependent on the current pressures and the electric current in the controlling electromagnet. Another mathematic model of dynamics of valve control component would be required for the correct determination of valve opening, but certain description is not necessary for this model. For this reasons was chosen a simple dependency of pneumatic conductivity on the control quantity, which is the output of the integrated controller of pressure regulator. This quantity was marked by $\underline{U}_c$ and it is an abstract value. The pneumatic conductivity dependency was derived from the manufacturer documentation.

Only proportional regulation was chosen for the first draft. The PID controller was considered for the next steps, but as will be shown below the proportional term is sufficient for good description of valve behavior. The control quantity of pressure regulator controller can be expressed:

$$U_c = k_p \cdot \left(p(t) - p_r \right) \quad \left(+ k_d \cdot \frac{d}{dt}\left(p(t) - p_r \right) + k_i \cdot \int_0^t \left(p(\tau) - p_r \right) d\tau \right) \tag{6}$$

where the first member stands for proportional term and next two members (in brackets) are for derivative and integral term; k_p, k_d and k_i are coefficients of proportional, derivative and integral term; p_r is for requested pressure, which is determined by PID feedback from the sensor of displacement of experimental stand arm. For first draft were coefficients set $k_d = k_i = 0$. Requested pressure p_r is determined by position controller, can be expressed:

$$p_r = p_0 + k \cdot \left(\varphi(t) - \varphi_r \right) \tag{7}$$

where p_0 is initial pressure in the air spring necessary for reach the initial position (length of the spring in the middle of working interval – horizontal arm position, zero angular displacement); k is the coefficient of the proportional term; φ_r is the angle of requested position of the experimental stand arm.

Finally it is necessary to remark that above described position control of the load supported by the air spring is not able of correct function. System is not able to reach the requested arm position. It is impossible to control the air spring length by proportionally controlled pressure. The direct regulation of mass flow of air is necessary for these purposes. Described simple position control was chosen only for the verification of the pressure regulator model.

Simulations and experiments

Two experiments were designed for verification of the mathematical model. Free motion was simulated in the first experiment – system was loaded by additional load and quickly relieved. By this experiment was verified the correctness of mechanical part of the mathematical model (especially the parameters of the air spring and mass properties of the arm and the load) also the damping coefficient of passive resistances was derived from the time response of the arm angular displacement. The model of pressure regulator was verified by second experiment. System was experimented and simulated with step change of the arm requested angular displacement. Series of these experiments was considered for determination of pressure regulator integrated controller coefficients (k_p, k_d, k_i). By comparing the first experimental data with model simulations it was determined that the proportional term of pressure regulator controller is satisfactory for good description of its behavior. Experiments were preformed with two different loads and for several intensities of required arm displacement step change.

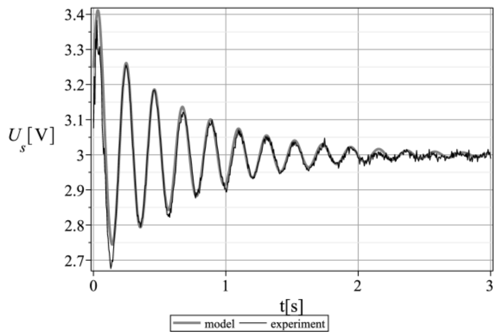

Fig. 3. First experiment – free motion, 20 kg load – signal from sensor of displacement of arm

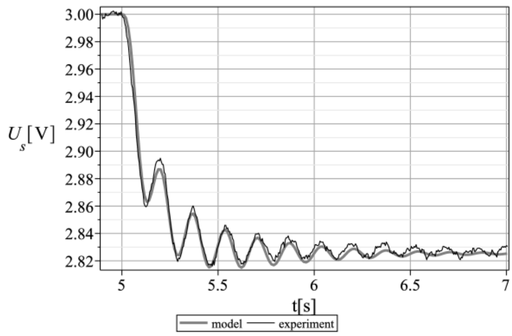

Fig. 4. Second experiment – step change of required displacement, 10 kg load – signal from sensor of displacement of arm

Conclusions

The correctness of mathematical model of the correction system control valve was proven by above described experiments. No need of derivative and integral term of controller in the pressure regulator model is very satisfying because of its application in the more complicated models. The examples of time response comparisons are viewed in Figs. 3-4 and a good agreement is observed. Comparison of the experimental data and simulations of the second experiment enabled determination of the proportional term coefficient of mathematical model of the pressure regulator controller. Finally, its model is prepared for simulations in the model of vibration-isolation platform with gyroscopic stabilizer.

Acknowledgement

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References

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