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IDENTIFICATION OF THE PNEUMATIC SERVO SYSTEM USING THE SELF-EXCITED OSCILLATIONS

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ABSTRACT

The paper describes the identification of the variable critical gain and natural frequency of the pneumatic cylinder using the self-excited oscillations method. The pneumatic drives are characterized by the low stiffness of the pneumatic cylinder which depends on the piston position and determines the natural frequency of the cylinder. The variety of the stiffness in dependence on the piston position can be obtained using the curve describing the course of the critical gain and natural frequency in dependence on the piston position. This feature is a disadvantage by the closed loop control of the piston position. The low stiffness should be taken into the account by the design of the pneumatic servo system operated in the closed loop, because the natural frequency of the cylinder depends on the stiffness and limits the closed loop gain and quality of the step response of the controlled system. The identified curve can be used for the design of the adaptive controller with variable on the piston position dependent gain. The described approach is presented by the measurements and results realized on the test rig.

KEY WORDS

Pneumatic drive, Identification, Natural frequency

NOMENCLATURE

h stroke of the pneumatic cylinder
K_M gain of the pneumatic cylinder
K_{SV} gain of the servo valve
K_{sn} gain of the transducer
M output of the relay
m mass
n polytropic exponent of air
p system pressure
r_{ocrit} critical gain
S piston thrust section
T_{0M} time constant of the pneumatic cylinder
T_{0SV} time constant of the servo valve
T_{crit} period of the oscillations
V_{01} volume – inlet 1 of the cylinder
V_{02} volume – inlet 2 of the cylinder
ξ_M damping ratio of the pneumatic cylinder
ξ_{SV} damping ratio of the servo valve
ω_M angular frequency of the pneumatic cylinder
ω_{SV} angular frequency of the servo valve
ω_{0M} natural frequency of the pneumatic cylinder
ω_{0SV} natural frequency of the servo valve

INTRODUCTION

The pneumatic servo actuators are very important drives with a wide range of applications in mechatronics. The design of the high dynamic pneumatic drives operated in the closed loop as a position servo drives is not simple task. The dynamic properties of the pneumatic servo drive – the pneumatic servo valve, pneumatic cylinder and tubes must be taken into the account by the controller design [2, 3]. We can use the mathematical modeling to obtain the dynamic model and use the simulation program to analyze the dynamic behavior of the system in different working points. This approach needs the parameterization of the mathematical model and their verification. Some parameters are given from the drive specification, some must be estimated or identified by the experiment [3]. The pneumatic drive is characterized as a low damped system with lower stiffness and natural frequency as the hydraulic drive. The reason is the higher compressibility of the air as oil. The natural frequency of the pneumatic cylinder varies with the piston position and influences the controller tuning. Their course in dependence on the piston position can be useful for the design of the controller of the position closed loop controlled system.

The paper describes the identification of the natural frequency of the pneumatic cylinder using the self-excited oscillations by the arranging a non linear element in the feedback [1], Figure 1. It is recommended to set the amplitude of the relay characteristic normally to 10%, max 20% of the command value. The oscillations occur in the output and should be similar to the sinus signal. It is also possible to use the saturation element with the same amplitude and with the proportional gain little bit higher as the critical gain. This approach allows to identify the natural frequency and the damping ratio too after evaluation of the results using the simplified linear model of the pneumatic drive given by the transfer function of the second order system and using the stability analysis of the closed loop system.

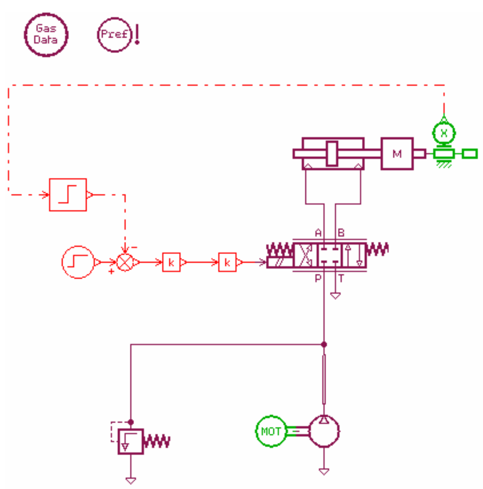

Figure 1 Pneumatic drive with non linear element in the feedback

ANALYSIS OF THE PNEUMATIC DRIVE

The typical structure of the pneumatic servo drive consists of the pneumatic cylinder controlled by the servo valve that allows continuously to control the mass flow through the variable resistances into the chambers of the pneumatic cylinder. The Figure 2 shows the block diagram of the closed loop position control of the pneumatic cylinder. The dynamic behavior of the main elements of the pneumatic drive can be described by the linear model after neglecting the nonlinearities. The transfer function

$$G_1(s) = \frac{K_M}{T_{0M}^2 s^2 + 2\xi_M T_{0M} s + 1} \tag{1}$$

describes the dynamics of the pneumatic cylinder, the transfer function

$$G_2(s) = \frac{K_{SV}}{T_{0SV}^2 s^2 + 2\xi_{SV} T_{0SV} s + 1} \tag{2}$$

describes the dynamic properties of the servo valve. The transfer function G_C in Figure 2 describes the controller.

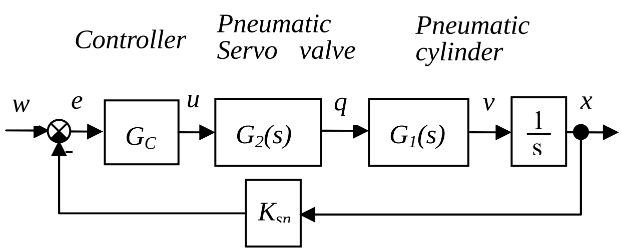

Figure 2 Block diagram of the closed loop position control

Because the bandwidth of the servo valve is higher than the natural frequency of the pneumatic cylinder, the dynamics of the servo valve can be neglected and in the following analysis the servo valve can be described only by the gain K_{SV}. After this simplification and for the proportional controller described by the gain K_R the transfer function of the closed loop system has the form

$$G(s) = \frac{K_0}{T_{0M}^2 s^3 + 2\xi_M T_{0M} s^2 + s + K_0}, \tag{3}$$

where K_0 is the gain of the open loop system

$$K_0 = K_R K_{SV} K_{sn} K_M \,. \tag{4}$$

The stability analysis of this linear system allows to express the critical gain K_{0crit} in marginal stability

$$K_{0crit} = \frac{2\xi_M}{T_{0M}} = 2\xi_M \omega_{0M} \,. \qquad (5)$$

In this formula ω_{0M} is the natural frequency of the pneumatic cylinder and ξ_M is the damping ratio of the pneumatic cylinder. These parameters limited the critical gain of the closed loop system, the controller gain and in this way the step response and quality of the control of the positioning system.

Using the results of the mathematical modeling of the pneumatic drive and under assumption that the thermodynamic processes can be good enough described as the polytropic or adiabatic (for $n=\kappa$) processes the natural frequency of the pneumatic cylinder can be calculated from the formula

$$\omega_{0M} = \sqrt{\frac{S^2 np}{m}\left(\frac{1}{Sx+V_{01}} + \frac{1}{S(h-x)+V_{02}}\right)} \,. \qquad (6)$$

The natural frequency of the pneumatic drive which limits the controller tuning is variable in dependence on the piston position. The Figure 3 shows the typical course of the natural frequency.

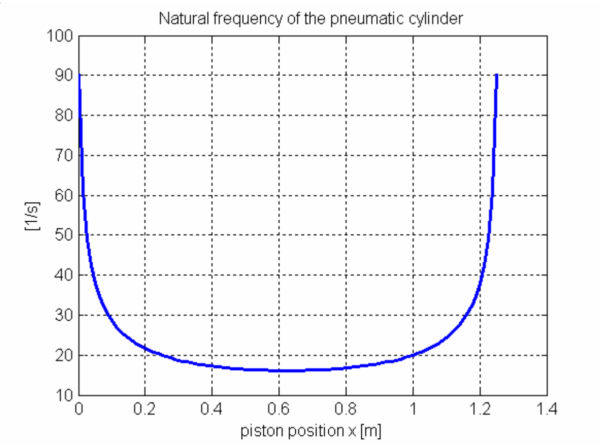

Figure 3 Natural frequency of the pneumatic cylinder

This natural frequency is the angular frequency of the non damped oscillations. Relations between the natural frequency ω_{0M} and the angular frequency ω_M of the observed oscillations of the system on the margin stability is depending on the damping ratio ξ_M of the pneumatic cylinder and is given by

$$\omega_M = \omega_{0M}\sqrt{1-\xi_M^2} \,. \qquad (7)$$

The pneumatic cylinder is characterized by the very low damping. The damping ratio is typically less than 0.2. For $\xi_M = 0.2$ becomes the expression (7) the form

$$\omega_M \doteq 0.98\omega_{0M} \,. \qquad (8)$$

For ξ_M less than 0.2 the difference between ω_M and ω_{0M} is less than 2%. The difference is smaller than the standard occurring final error by the experimental identification. For this reason it is possible in some cases to identify the angular frequency ω_M as a natural frequency ω_{0M}.

IDENTIFICATION USING THE SELF-EXCITED OSCILLATIONS

The identification method based on the self-excited oscillations is very simple and allows in very simple way to identify the angular frequency of the oscillations on the marginal stability.

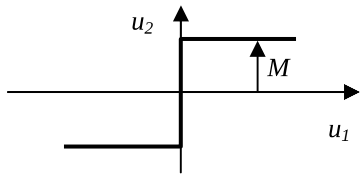

Figure 4 Relay characteristic of the non linear element

The oscillations occur after arranging the nonlinear element in the feedback. It is possible to receive the same result after placing the non linear element in the forward side of the closed loop system. It is suitable as a non linear element to use element with the relay characteristic, see Figure 4.

The closed loop pneumatic positioning system with the non linear element in the feedback is shown in Figure 1. After the oscillations occur the typical form of the output signal is shown in Figure 5.

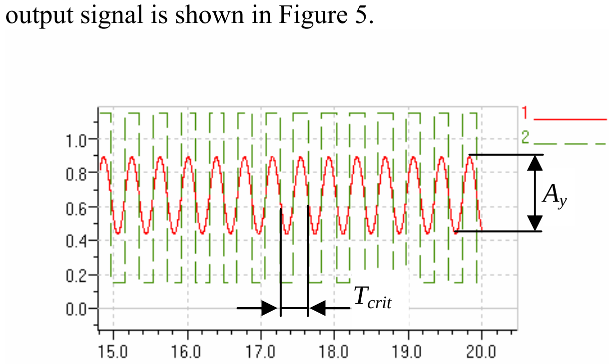

Figure 5 Self-excited oscillations of the piston of the pneumatic cylinder

Solid line – piston position

Dashed line - output from the non linear element

Using the theory of this identification method described for example in [1] the critical gain of the closed loop system is given by

$$r_{0crit} = \frac{4M}{\pi A_y} . \qquad (9)$$

The values of the variables A_y and M can be obtain from the course of the self-excited oscillations and from the output of the relay characteristic, see Figure 5 and 4. The angular frequency of the critical oscillations can be expressed as

$$\omega_{crit} = \frac{2\pi}{T_{crit}} . \qquad (10)$$

In accordance with the expression (5) for the critical gain of the pneumatic positioning system the values of the formulas (9) and (5) should be the same that means

$$K_{0crit} = r_{0crit} , \qquad (11)$$

resp.

$$2\xi_M \omega_{0M} = r_{0crit} \qquad (12)$$

and the critical frequency or period determines the angular frequency of the piston oscillations ω_M, which can be calculated from

$$\omega_M = \frac{2\pi}{T_{crit}} . \qquad (13)$$

Using the expression (7) the natural frequency of the pneumatic cylinder is given by

$$\omega_{0M} = \frac{2\pi}{T_{crit}\sqrt{1-\xi_M^2}} . \qquad (14)$$

From the expression (12) can be calculated the damping ratio of the pneumatic cylinder

$$\xi_M = \frac{r_{Ocrit}}{2\omega_{0M}} . \qquad (15)$$

After substitution of the expression (14) in the expression (15) we obtain the expression for the damping ratio which respects the difference between the natural frequency ω_{0M} and angular frequency ω_M of the pneumatic cylinder

$$\xi_M = \frac{r_{0crit} T_{crit}}{\sqrt{16\pi^2 + T_{crit}^2 r_{0crit}^2}} . \qquad (16)$$

Provided that the described difference (8) between the ω_{0M} and ω_M for lower damping is accepted the simplified expressions can be used for the direct calculation of the natural frequency and damping ratio of the pneumatic cylinder from the measured values

$$\omega_{0M} \cong \omega_M = \frac{2\pi}{T_{crit}} \qquad (17)$$

and

$$\xi_M = \frac{r_{0crit} T_{crit}}{4\pi} . \qquad (18)$$

APPLICATION OF THE METHOD AND EXPERIMENTAL RESULTS

The described method was applied for the experimental identification of the natural frequency of the pneumatic cylinder of the test rig installed in the laboratory.

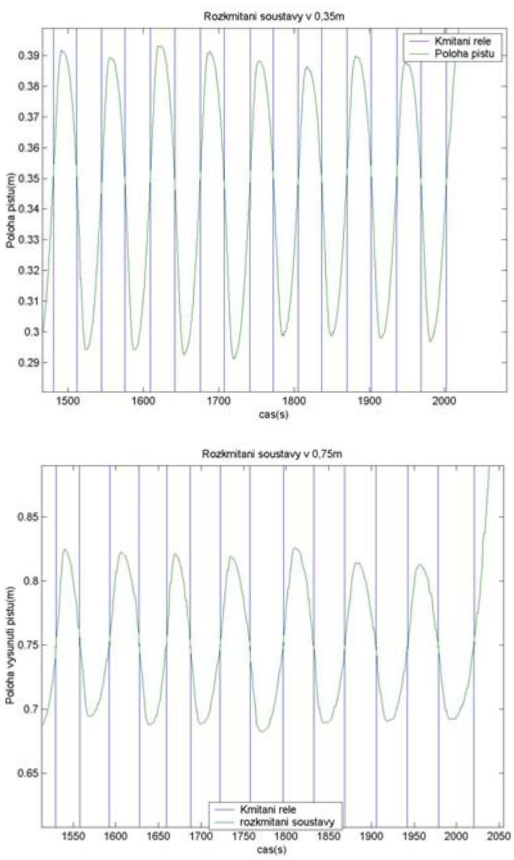

Figure 6 Measured self-excited oscillations of the piston of the pneumatic cylinder at the position x = 0.35m (up), at the position x = 0.75m (down)

The same drive and identification experiment was simulated in MATLAB – Simulink using the created mathematical model and in the simulation program AMESim, Figure 1, 5. The following drawings in Figure 6 show the measured signals at two different positions of the piston [4]. The evaluated values and approximated course of the natural frequency of the pneumatic cylinder compared with the analytical values

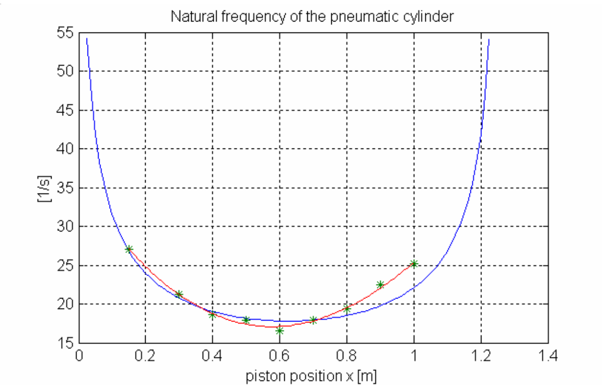

Figure 7 Natural frequency of the pneumatic cylinder – calculated analytically (6) and evaluated from the measurement "*"

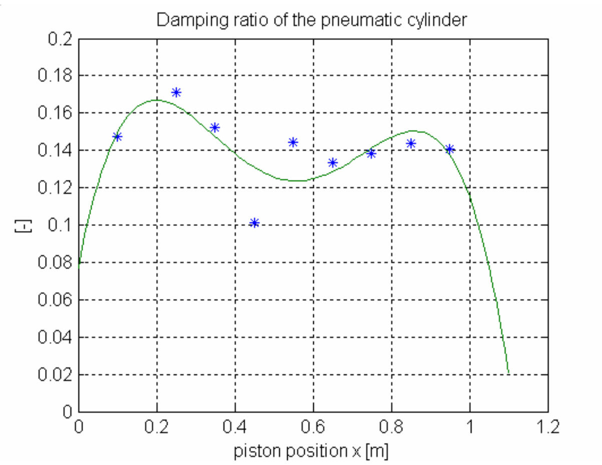

Figure 8 Damping ratio of the pneumatic cylinder – evaluated from the measurement "*" and polynomial approximation

calculated from the expression (6) are depicted in Figure 7. The values of the damping ratio calculated from the expression (16) and approximated by the polynomial function in dependence on the piston position are shown in Figure 8.

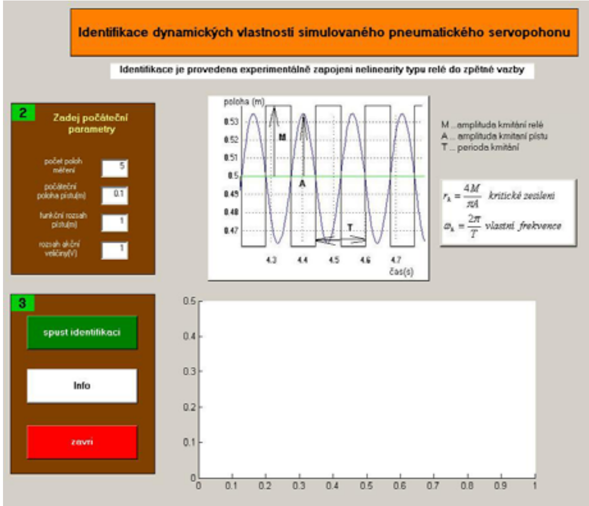

Figure 9 Control panel of the identification using the self-excited oscillations

The created program in MATLAB-Simulink with the Real Time Toolbox controls the identification process and realizes the calculation of the identified parameters of the pneumatic drive. The program controls the motion of the piston, the achieving of the desired positions and in each position starts the identification procedure. After measuring the excited oscillations the calculation of the critical gain and natural frequency runs automatically. The control panel of this program is shown in Figure 9.

CONCLUSIONS

The simple method for identification of the dynamic properties of the pneumatic drive based on the self-excited oscillations was presented in this paper. This method allows to calculate the natural frequency of the pneumatic cylinder, the critical gain and damping ratio at different positions of the piston. These values can be used for the controller tuning, for the design of the adaptive controller with variable gain. Simple realization is the advantage of this approach. The application of this method was presented by the identification of the pneumatic cylinder installed on the test rig.

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