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Hydraulic Control of Speed and Position of a Pneumatic Actuator

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Abstract. The paper presents an original pneumo-hydraulic system that controls the speed and the position of the actuated load by using a hydraulic circuit containing a hydraulic proportional throttle. A mathematical model of the solution is developed and numerical simulations are performed. The experimental results obtained on the built model confirm the theoretical ones.

Introduction

In the domain of pneumatic actuation systems, accurate control of the speed and position of the actuated load are still actual problems. There are many researches aiming to increase the accuracy of these functions. The difficulties are generated by the properties of the working fluid (compressed air): high compressibility and very low viscosity [1].

Speed and position are strongly related: an accurate positioning can be achieved only if a rigorous control of the actuated load speed exists.

Nowadays more and more applications demand accurate positioning, some only in specific points, and others in any point of the working stroke. Once the number of stop points rises, the positioning becomes more and more difficult. There are several special motors that can deal with this problem, but they are rarely used [2]. The problem that has not been solved yet properly is the accurate positioning in any point of the working stroke of the actuator.

Usually the control of the speed is achieved by integrating a proportional pneumatic flow regulator (proportional throttle or a proportional directional control valve) within the structure of the system. The last version simplifies the structure of the system, stopping the load in the programmed points by blocking the pressure supply circuits of the active chambers of the pneumatic motor. This way a limited accuracy of the speed control and a positioning accuracy of tenths of millimeters are obtained [3]. A way of overtaking this deadlock is the use of a hydraulic control circuit. In this situation, the system becomes a pneumo-hydraulic one [4, 5].

The paper presents an original structure of a pneumo-hydraulic system developed by the authors. The principle scheme, the mathematical model, the numerical simulation and the results of the tests performed on the built experimental model are presented.

The functional scheme of the pneumo-hydraulic system

The principle of the proposed system is to control the speed and the position of the actuated load by using a hydraulic circuit containing a hydraulic proportional throttle. This way the liquid flow is accurately controlled. The load speed is rigorously controlled and full stop is achieved by blocking the hydraulic circuit.

The connection diagram of the system is shown in figure 1. The used notations are the following: *AP* – rodless linear pneumatic actuator, with integrated braking device *DB* and position sensor T_p; *AH* –linear hydraulic actuator; DP_a– 5/3 pneumatic directional control valve with preferential position, used for actuating; DP_b– 5/2 pneumatic directional control valve with preferential position, used for braking; *DrP* – hydraulic proportional throttle.

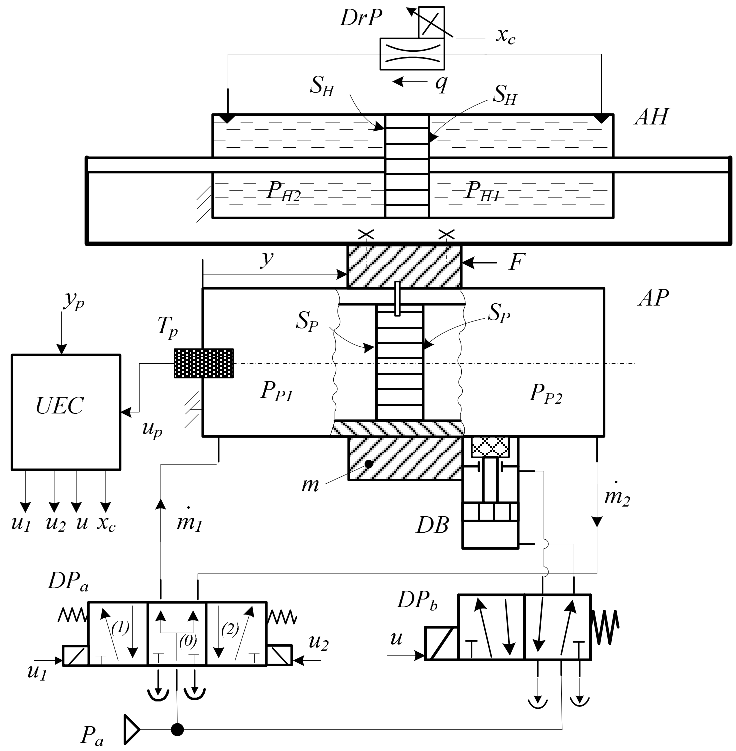


Fig. 1 The connection diagram

Mathematical model

In order to study the dynamic behavior of the pneumo-hydraulic unit, the authors developed its mathematical model starting from the scheme presented in figure 1. Table 1 presents the used notations.

Due to the complexity of the model and its description by differential equations, numerical simulation was chosen in order to theoretically study the behavior of the system [6].

The movement of the mobile assembly will start if a flow section is established through the proportional throttle.

The model is valid in both the case of the speed control of the actuated load and the case when rigorous positioning with an imposed error is required.

The mathematical model of the system is consists of the equation of the movement of the mobile subassembly and the differential equations of the pressures from the active chambers of the hydraulic and pneumatic actuator (Eq. 1). The used notations are presented in Table 1.

Table 1. Notations used in the model of the pneumo-hydraulic unit

Used not.	Significance	Used not.	Significance	Used not.	Significance
y	Displacement of the two cylinder rods mounted in continuation	P_{P1}, P_{P2}	Pressures in the active chambers of the pneumatic actuator	V_{P0}	The dead volume of the chambers of the pneumatic actuator
$\dot{m}_1$, $\dot{m}_2$	Mass flow rates in the chambers of the pneumatic actuator	P_{H1}, P_{H2}	Pressures in the active chambers of the hydraulic actuator	V_{H0}	The dead volume of the chambers of the hydraulic actuator
m_r	Mass of the mobile assembly piston – rod - load	V_{P1}, V_{P2}	Volumes of the chambers of the pneumatic actuator	S_P	Active sections of the pneumatic actuator
D_s	Diameter of the throttle slide valve	V_{H1}, V_{H2}	Volumes of the chambers of the hydraulic actuator	S_H	Active sections of the hydraulic actuator
x	Displacement of the throttle slide valve	q	Oil flow rate through the throttle	S_c	Flow section through the throttle

P_a	Supply pressure	T_a	Absolute temperature	x_n	Nominal opening of the throttle
P_0	Atmospheric pressure	R	Universal gas constant	x_p	The programmed travel
c_η	Damping factor	E	Oil elasticity module	v	Speed of the mobile assembly
c	Maximum travel of the rods	F_r	External force	χ	Adiabatic factor
d_n	Nominal diameter	K	Constant factor	ρ	Oil density

$$\begin{cases} \dfrac{dv}{dt} = \dfrac{1}{m_r}\cdot\left[(P_{P1}-P_{P2})\cdot S_P-(P_{H1}-P_{H2})\cdot S_H-c_\eta\cdot v-F\right] \\ \dfrac{dy}{dt}=v \\ \dfrac{dP_{H1}}{dt}=\dfrac{E}{V_{H0}+(c-y)\cdot S_H}\cdot\left(q-S_H\cdot\dfrac{dy}{dt}\right) \\ \dfrac{dP_{H2}}{dt}=\dfrac{E}{V_{H0}-y\cdot S_H}\cdot\left(-q+S_H\cdot\dfrac{dy}{dt}\right) \\ \dfrac{dP_{P1}}{dt}=\dfrac{\chi}{V_{P0}+y\cdot S_P}\cdot\left(\dot{m}_1\cdot R\cdot T_a-S_P\cdot P_1\cdot\dfrac{dy}{dt}\right) \\ \dfrac{dP_{P2}}{dt}=\dfrac{\chi}{V_{P0}+(c-y\cdot)S_P}\cdot\left(-\dot{m}_2\cdot R\cdot T_a-S_P\cdot P_2\cdot\dfrac{dy}{dt}\right) \end{cases} \tag{1}$$

The flow rates involved in the system of first order nonlinear differential equations presented above can be written as following:

- flow rate q of the oil transported between the two active chambers of the hydraulic actuator:

$$q = sign\,(P_{H1}-P_{H2})\cdot S_c\cdot\sqrt{\frac{2}{\rho}\cdot\left|P_{H1}-P_{H2}\right|} \tag{2}$$

where: $S_c(x)=S_{nH}\cdot x/x_n$, $S_{nH}=\pi\cdot D_n^2\cdot/4$ and $x_n=D_{nH}^2/(4\cdot D_s)$ if the proportional throttle controls the flow section through a translating cylindrical spool in translation.

For such a throttle, the variation of the position of the spool can be approximated by Eq. 3, as can be noticed in Fig. 2:

$$\frac{x}{x_n}(t)=\begin{cases}\dfrac{x_p}{x_n}\cdot\dfrac{t}{t_a} & \text{if} \quad 0\le t\le t_a \\ \dfrac{x_p}{x_n} & \text{if} \quad t>t_a\end{cases} \tag{3}$$

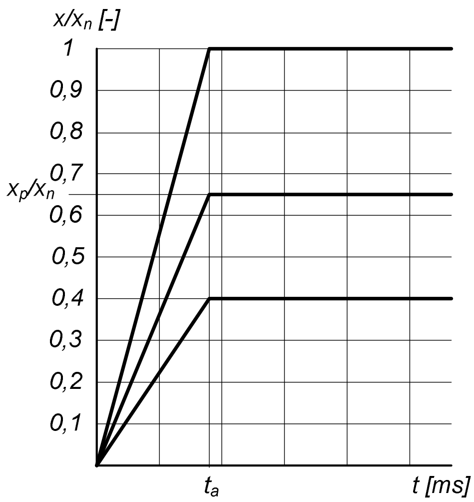


Fig. 2 Characteristic diagram of the throttle in the case when the system is used for the control of the speed

- the flow rates controlled by the proportional throttle DrP are equal to:

$$\dot{m}_1 = sign\,(P_a-P_{P1})\cdot\frac{K\cdot S_{nP}}{\sqrt{T_a}}\cdot\max\{P_a,P_{P1}\}\cdot N\left[\min\left\{\frac{P_{P1}}{P_a},\frac{P_a}{P_{P1}}\right\}\right] \tag{4}$$

$$\dot{m}_2 = sign\,(P_{P2}-P_0)\cdot\frac{K\cdot S_{nP}}{\sqrt{T_a}}\cdot\max\{P_{P2},P_0\}\cdot N\left[\min\left\{\frac{P_0}{P_{P2}},\frac{P_{P2}}{P_0}\right\}\right] \tag{5}$$

where:

$$N(z)=\begin{cases}1 & \text{if } 0\le z\le 0.528\\ 3.8\cdot\left[x^{\frac{2}{\chi}}-x^{\frac{\chi+1}{\chi}}\right]^{\frac{1}{2}} & \text{if } 0.528< z\le 1\end{cases} \tag{6}$$

The model can be simplified if it is supposed that there is a delay between the actuation of the directional control valve *DPa* and of the proportional hydraulic throttle *DrP* (Fig. 1).

The variation law of the spool position x (the input) is established depending on the function that the system must accomplish. There are two cases:

- a law described by Eq. 3, when a certain speed of the mobile assembly when the stabilized conditions are met is needed (Fig. 2);
- a law described by Eq. 7 when the actuated load has to reach the imposed position y_{op} with an imposed error; at the moment t_{op}, when the position sensor confirms the positioning in the imposed error domain, the proportional throttle closes the flow section and blocks the hydraulic circuit, leading to the stop of the mobile assembly (Fig. 3).

$$\frac{x}{x_n}(t)=\begin{cases}\dfrac{x_p}{x_n}\cdot\dfrac{t}{t_a} & \text{if} \quad 0\le t\le t_a\\ \dfrac{x_p}{x_n} & \text{if} \quad t_a< t\le t_r\\ \dfrac{x_f}{x_n}+\left(\dfrac{x_p}{x_n}-\dfrac{x_f}{x_n}\right)\cdot\dfrac{t_f-t}{t_f-t_r} & \text{if} \quad t_r< t\le t_f\\ \dfrac{x_f}{x_n} & \text{if} \quad t_f\le t\le t_{op}\end{cases} \tag{7}$$

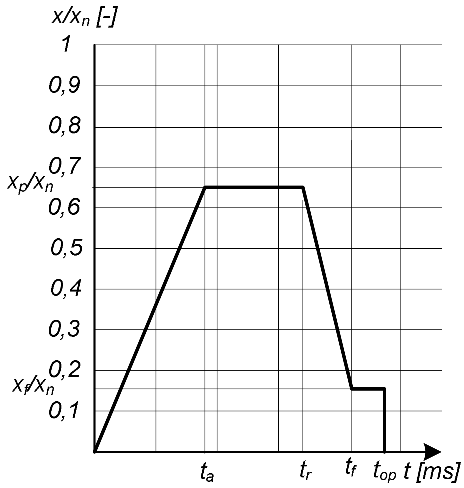


Fig. 3 Characteristic diagram of the throttle in the case when the system is used for the control of the position

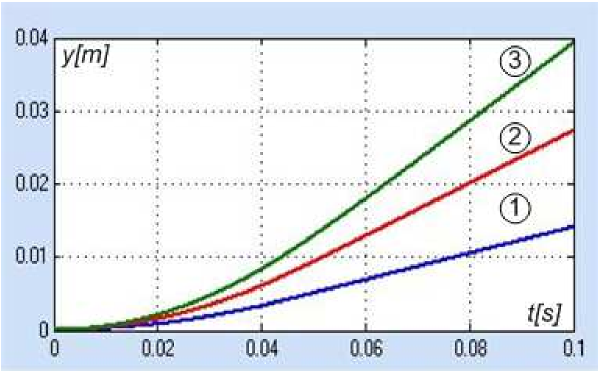
Starting from this mathematical model, the dynamic behavior of the unit was simulated using SIMULINK. Relevant diagrams obtained from the simulation are presented in Fig. 4 and Fig. 5.

The experimental model

Starting from the functional scheme presented in Fig. 1 and from the results of the numerical simulation, an experimental model was built.

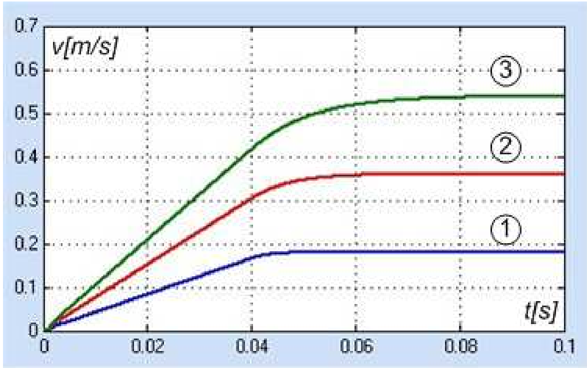
A series of general design considerations were taken into account. Among the imposed requirements were: to obtain a modular structure of the system, to use actuators with integrated positioning sensors; to place the distribution equipment on the motor stator; to integrate the electronic control circuitry; to achieve an appropriate placement of various sensors in order to compensate the perturbation factors.

Figure 6 shows a view of the built physical model. The slide of the pneumatic linear motor with integrated position sensor DGPIL-25-500-PPV-B-KF-AIF-GK-SV-AV (FESTO) and the rod of the hydraulic cylinder CHD2FWB40B-700A are coupled through a rigid system *SC*. Classic pneumatic directional control valves from SMC were used: SY5440-5LOU-02 type for DP_a and SY5140-5LOU-Q type for DP_b. The hydraulic control subassembly consists of the hydraulic motor with bilateral rod and the hydraulic proportional throttle 4 WREE 6 EA08-2X/G24K31/A1V, mounted on the connection way between the two motor chambers; the hydraulic proportional throttle has the orifices connected so that to control only one flow area.



1: x_p/x_n=0.25; 2: x_p/x_n=0.5; 3: x_p/x_n=0.75

Fig. 4 Displacement of the actuated load for various programmed travels



1: x_p/x_n=0.25; 2: x_p/x_n=0.5; 3: x_p/x_n=0.75

Fig. 5 Speed of the actuated load for various programmed travels

The first experimental tests showed functioning characteristics that are pretty close to the results of the numerical simulation.

Conclusions

The proposed solution allows the rigorous control of speed as well as of the driven load position. The experimental model could be configured used equipment available in the catalogues of the producers in the field. The experimental results obtained till present confirm the theoretical ones. A series of control algorithms will be developed in a further research stage.

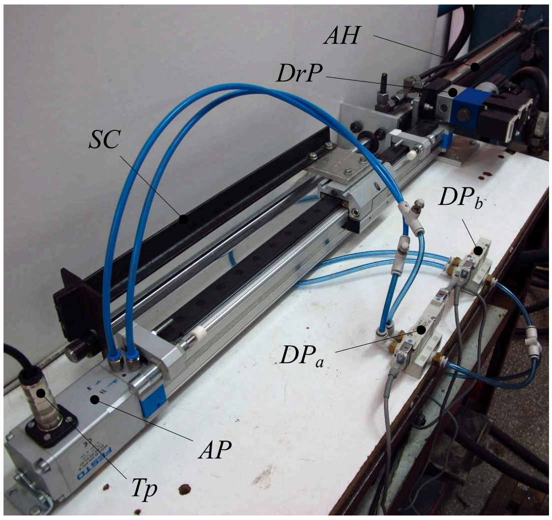


Fig. 6 A view of the built experimental model

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