SIMULATION OF PNEUMATIC DIRECTIONAL VALVE

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Abstract. There is well known the fact that to design and simulate a system there may be used two methods: mathematical modelling, using dynamics equation of the system, or using oriented software that includes predefined elements associated with different type of systems (electric/hydraulic or pneumatic components, controllers, mechanical components etc.). The present papers present the second method applied to a pneumatic directional valve. Simulation is done using a toolbox of Matlab software, Simscape.

Keywords: function-generating mechanism, synthesis, pneumatic

1. Introduction

Modelling and simulation of systems are the first two steps to create a real system, to adapt an existing system to new conditions, to create flexible systems.

These two steps are very difficult to be done, in real conditions, if the system is a very complex one, because any mathematical model is developed based on some design assumptions that, in most cases, simplifies to conditions and the functionality. For a designer to be able to simulate systems by writing mathematical model, implies to know all the laws that govern the system, independently of the type of system. This assumption is not realistic. It is true that specialised papers contain mathematical models for almost everything, from neurons to ships, from music to aerospace engineering, but applying such models is a real challenge.

The authors of the present paper focused on simulation of a pneumatic directional valve. There are many mathematical models for this pneumatic component [1] but the authors of this paper decided to use predefined elements implemented in Matlab's toolbox Simscape.

This toolbox has a demo where a 5/3 pneumatic directional valve is already implemented [2], but in that demo there is no command applied to this directional valve. Thus, the authors of the present paper recreated this pneumatic directional valve and implemented the electromagnetic command.

2. Pneumatic directional valve subsystems

Simulation of dynamic behaviour of pneumatic directional valve was done by decomposing the system. Thus, the subsystems of such a valve are presented in figure 1 and implemented in the simulation individually.

Theoretical simulation was done using the data sheets of MPYE-5/3-1/8 pneumatic directional valve, offered by Festo. This valve is a very complex one and has high level of applicability, being one of the most used valves.

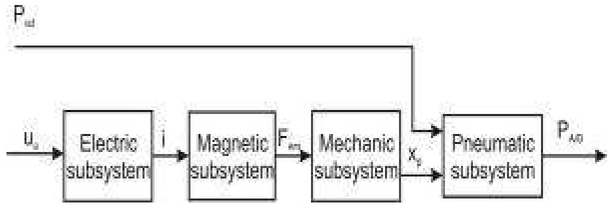

Figure 1. Pneumatic directional valve subsystems

2.1. Electric and magnetic subsystem

Input of this subsystem is the input voltage and the output is the electromagnetic force. In the following are presented the components of this subsystem.

1. Controlled voltage source - maintain the specified voltage at its output, regardless of the current passing thought it; input voltage is 24 V.

2. Translational electromechanical converter – defines the command electromagnet and provides the interface between the electrical part and mechanical one. The subsystem is defined by the equations [3]:

$$F = k_{em} \cdot I\,, \quad (1)$$

$$U = k_{em} \cdot v\,, \quad (2)$$

where: F [N] – mechanical force; I [A] – current; U [V] – voltage; v [m/s] – mechanical speed; and k_{em} [N/A] or [V/(m/s)] – proportionality constant. For the chosen directional valve k_{em} = 1 N/A.

3. Resistor – input resistor for command circuit. For the analysed directional valve, the value for this resistor, R_{em} = 22 Ω, was adopted from data sheets [4] where there are given input voltage (24 V) and maximum current consumption (1100 mA).

2.2. Mechanical subsystem

Input of this subsystem is the electromagnetic force and output is displacement of valve spool. Simulation defines the dynamic behaviour of the pneumatic proportional directional valve. Thus, in the simulation it was considered that the valve – spool – air system may be associated with a mass – spring – damper system. In the following there are presented the components and the assumptions considered.

1. *coulomb friction* is considered included in perturbation;
2. *spool mass*, m_p = 10 g [4], without initial velocity;
3. *spring* – zero initial deformation (no pretension), spring rate k_e = 100 N/m;
4. *damper* – defines the viscous friction, damping coefficient: c_v = 0.01 N/(m/s);
5. *translational hard stop* – based on data sheets, the maximum spool displacement is 5 mm and therefore it is necessary to be implemented a hard stop. The stop is implemented as a spring that comes into contact with the slider as the gap is cleared.
6. *command bloc* – command the valve spool displacements in both directions, based on data sheets.

2.2. Pneumatic subsystem

Pneumatic model input is the position of valve spool and as output the pressure through the two chambers of the motor. In the following are presented the main components of this subsystem together with the assumptions made:

1. *pneumatic pressure source* – ideal compressor, which maintains a specified pressure difference regardless of the flow rate. Compressor has an output of 6 bar = $6 \cdot 10^5$ Pa;
2. atmospheric pressure 101325 Pa and temperature 293.15 K; specific heat for constant pressure $1.005 \cdot 10^3$ J/kg·K; specific heat for constant volume 717.95 J/kg·K; dynamic viscosity $1.821 \cdot 10^{-5}$ s·Pa;
3. *proportional valve* has three ways, and the driven orifices were modelled using specific elements of Simscape software. These elements are similar to variable section throttles with discharge coefficient C_d = 0.82 and minimum area 10^{-10} m^2.

This throttles block is defined by mass flow law [5]:

$$Q_m = k \cdot C_d \cdot A \cdot T_i^{-0.5} \cdot (p_i - p_0) \tag{3}$$

where: Q_m – mass flow rate; k – is a constant such that the flow predicted for p_0/p_i is the same as that predicted by the original flow equation for p_0/p_i = 0.999; C_d – discharge coefficient; A – orifice cross-sectional area; p_i, p_0 – absolute pressures at the orifice inlet and outlet. There is no heat loss between inlet and outlet of throttle because the shape of the throttle section determines zero dynamic pressure and energy loss is transformed into internal loss. This internal loss increases the throttle internal temperature, thus the input and output temperatures become equal.

4. each *variable section throttles* has a linear characteristic of area, dependent on valve spool displacement (between – 5 and 5 mm);
5. *initial position* of proportional directional valve is extreme one, when pressure feeds A chamber and determines the displacement of the cylinder. Next, the directional valve is positioned in central way (no feeding pressure), maintaining the cylinder's position. The third position determines a negative displacement of the cylinder;
6. *outputs* of the proportional directional valve have two pneumatic accumulators that simulates the cylinder's chambers. The volume variation of the two accumulators is different for different types of cylinders.

3. Pneumatic directional valve simulation

Simulations for the proportional directional valve were done for a 6 bar input and different values for the volume of the accumulators.

Simulation diagram of the proportional directional valve is presented in figure 2.

First simulation focused on valve spool displacement. To determine this movement it was implemented a linear transducer and was done a 2 s simulation. The result is presented in figure 3.

The response time of the proportional directional valve is given in data sheet [5]. This time is specified to be determined between 20% of maximum value and 80% of maximum value. Considering that, data sheet specifies a 5 ms response time. Analysing the simulation (figure 3) there was obtained: t_{feed} = 5 ms – feed stroke displacement and t_{return} = 7.5 ms – return stroke displacement.

Comparative with data sheet the values from simulation are very closed, thus the pneumatic proportional directional valve may be considered well done modelled.

The very small response times allows the MPYE-5/3-1/8 pneumatic proportional directional valve to be used in serial material handling systems, with very small operational times.

For the first simulations the volume of the accumulators was considered 1 ml (associated with the volume of the cylinder's chambers). The input signal is shown in figure 4 and the simulation results are presented in figures 5, 6 and 7.

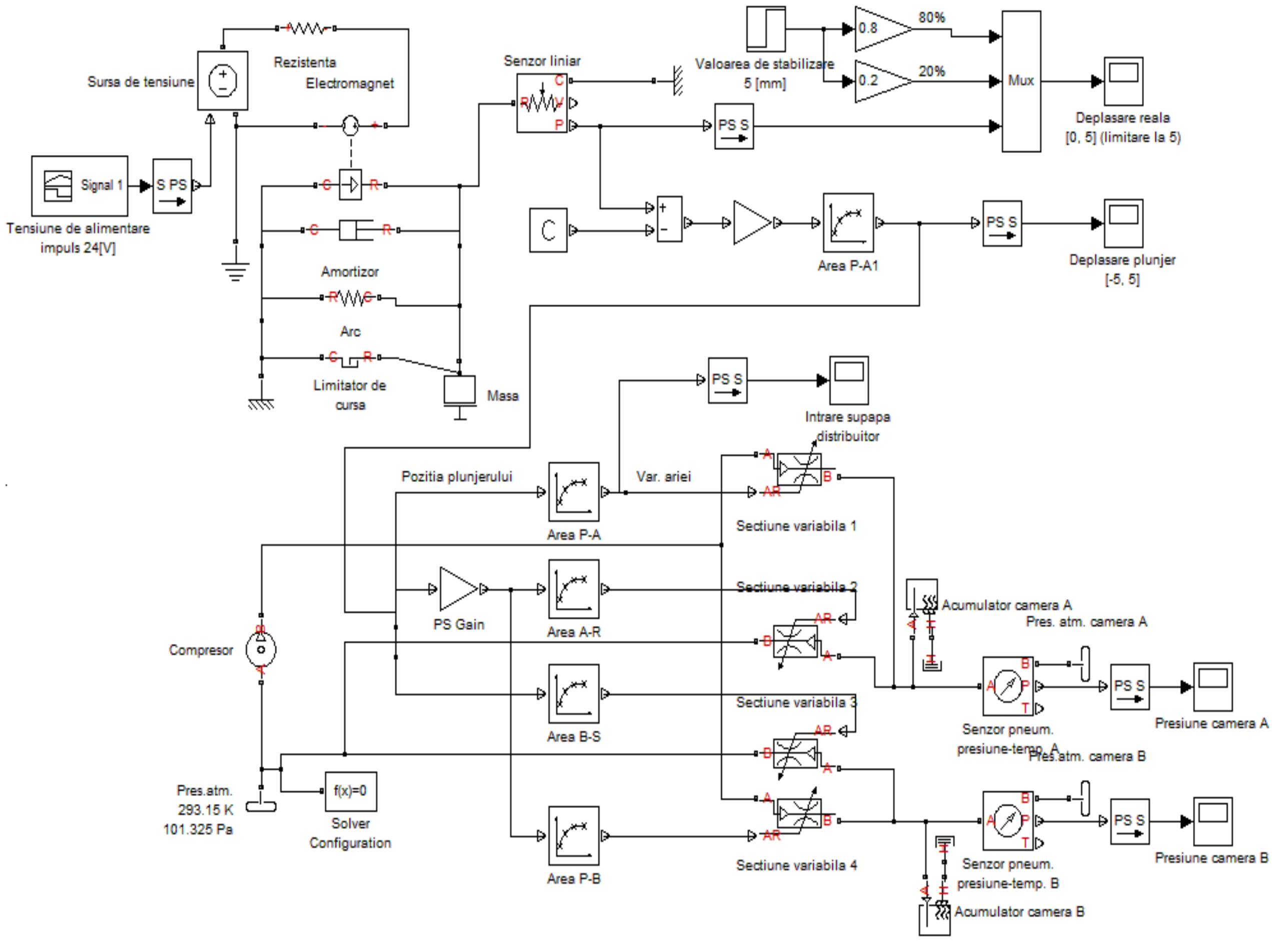

Figure 2. Simulation diagram for proportional directional valve

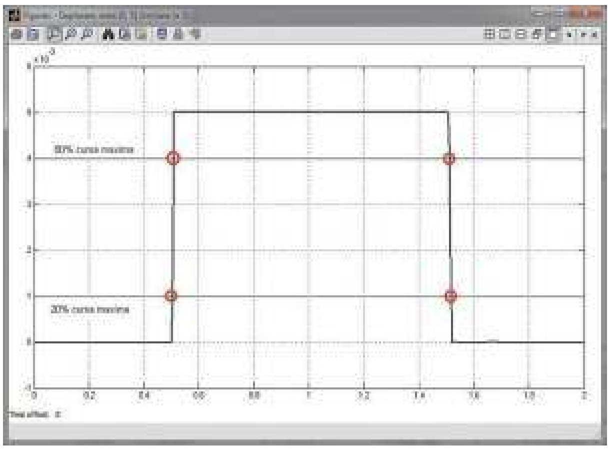

Figure 3. Valve spool displacement

Analysing the simulation (figure 5) there can be concluded: starting from a 6 bar input pressure, output pressure has the same value and the response time (increase time – between 10% and 90% from steady state value) is 1.5 ms (almost instantaneous).

Comparative result for pressures through the two chambers (both resulted pressures were determined for the 6 bar input – figure 4) of the cylinder is presented in figure 6.

Output signal (pressure through chamber A or B) has a significant dependence on accumulator volume. That implies a significance dependence on the type of cylinder used.

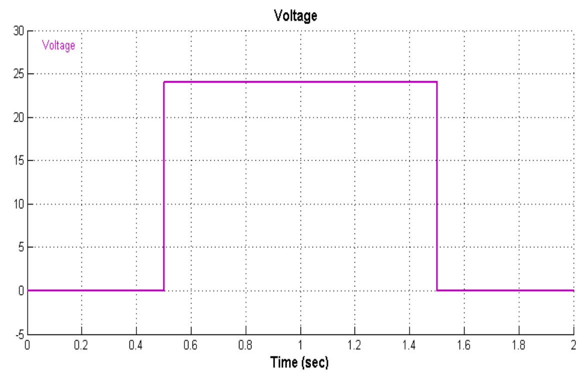

Figure 4. Input signal, voltage 24 V

There were done simulations for different values for the volume of the accumulators. Thus, if the volumes increase from 1 ml to 20 ml (equivalent to a cylinder with 20 mm diameter and 62 mm stroke) the response time is significantly increasing (figure 7).

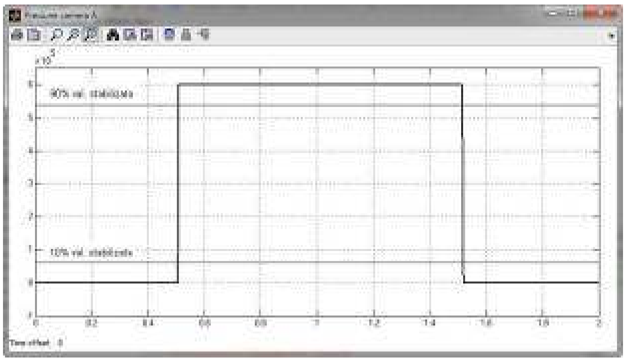

Figure 5. Output pressure
(through A chamber of cylinder)

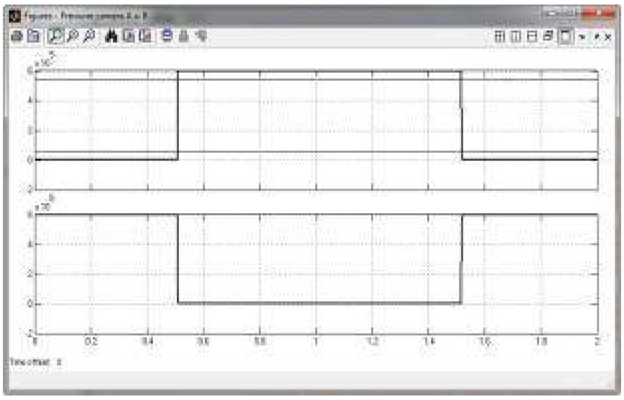

Figure 6. Pressures through the two chambers of the cylinder (for a 1 ml accumulator volume)

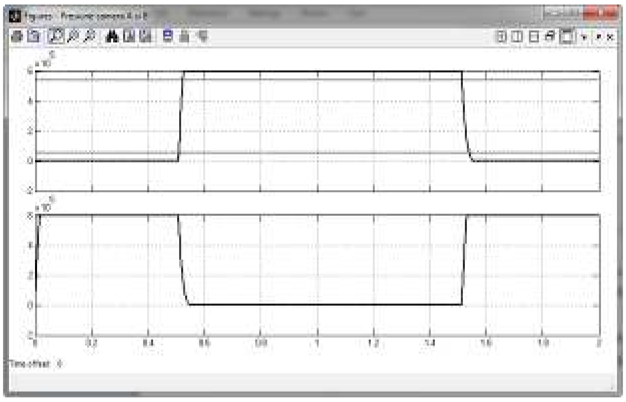

Figure 7. Pressures through the two chambers of the cylinder (for a 20 ml accumulator volume)

In the case of a cylinder with 25 mm diameter and 500 mm stroke (figure 8), simulation shows a delayed response (response time is 16 ms – ten times higher than the first simulation). It also can be underlined that the steady state interval is much shorter than the initial simulation interval.

4. Conclusions

Modelling and simulation of a 5/3 way pneumatic directional valve is very important because it is part of any pneumatic circuit with high level of importance and frequency. Independent of type of modelled element it is very important to be accurate modelled because the error of one element directly influences all the others and thus, the real behaviour of the system will be compromised.

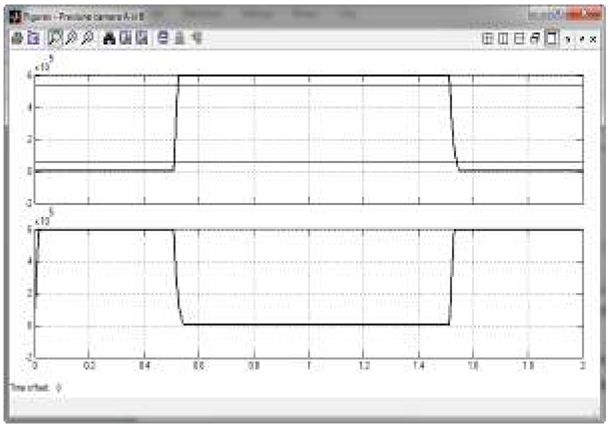

Figure 8. Pressures through the two chambers of the cylinder (for a 25 mm diameter and 500 mm stroke cylinder)

The authors of the present paper succeed to obtain a high accuracy model for the 5/3 way directional valve. This model was included, by the authors of the present paper, in three different circuits: double acting pneumatic cylinder – directional valve, single acting pneumatic cylinder – directional valve, pneumatic muscle – directional valve. These circuits were created to compare the dynamic behaviour of different pneumatic actuators and their applicability in material handling system.

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