ISSN 1453 – 7303 **"HIDRAULICA" (No. 1/2024)**
Magazine *of Hydraulics, Pneumatics, Tribology, Ecology, Sensorics, Mechatronics*

Modeling and Simulating the Operation of the Pneumatic Cylinders

Prof. PhD Eng. **Anca BUCUREȘTEANU**[1,*]

[1] University POLITEHNICA of Bucharest

* ancabucuresteanu@gmail.com

Abstract: *This paper presents the simulation and mathematical models necessary for studying the behavior of the pneumatic cylinders in static and dynamic conditions. The mathematical models are simplified, accessible and adaptable to different applications; they can be used in the design phases of new pneumatic units in the industrial fields. The models for cylinders with or without braking at travel end are presented. Depending on each system, these models can be supplemented with models of equipment for regulation and distribution, in order to study the respective systems. The proposed models try to take into consideration the particularities of the pneumatic systems: compressible working environment, maximum working pressure limited to 10 bar at the most, specific damping effects, etc.*

Keywords: *Mathematical models, static and dynamic mode, pneumatic cylinders, simulation*

1. Introduction

Pneumatic cylinders are motors that transform the pneumatic energy, characterized by the pressure and flow received in a certain time, into mechanical energy provided in the form of a linear displacement of a force at the same time [1].
There is a large typology of pneumatic cylinders that are manufactured by specialized companies and are available for industrial applications in different fields: production equipment and machines (machine-tools, presses, injection machines), mobile equipment, light industry, robotics etc. [2, 3, 4, 5, 6]. Cylinders can operate in open or closed systems [1, 5].
These motors are powered by means of specific equipment [1, 2, 3] from a source defined, among other things, by a maximum working pressure, usually expressed in [bar], and a maximum flow rate expressed in [Nm^3/s]. Due to the compressibility of the air, the pressure and flow rate can change depending on load, the DN of the unit and the length of the pipes. The characteristics of the equipment influence each other.
In these conditions, a series of approximations and linearizations of the proposed mathematical models are needed to easily create applicable models. Moreover, the operation of each pneumatic cylinder is also influenced by a series of constructive parameters: the type of the seals used, the friction coefficients of the live elements compared to the fixed ones, the existence of accessories such as the locking and/or braking systems on one or both ends of the cylinder stroke, the operating method, the length of strokes etc. For these reasons, the author tries to include also these types of parameters in the mathematical models.

2. Simplified Mathematical Model of the Pressure Source

The maximum working pressure p_{Max}, the maximum velocity v_{Max} allowed by the cylinder and the active surface S_1 of the piston (the surface on which the pressure actuates) are considered to be given values. The second surface will be denoted by S_2 and is the one that removes the existing air left after the previous actuation.
When the maximum pressure is reached, the cylinder is blocked because of a resistive force higher than the one that can be overcome or because the cylinder reached the end of the stroke. In these cases, the flow provided by the source becomes null.
Regarding the proposed model, it is assumed that at the supply moment there is no air in the piston chamber, nor in the piston supply pipes, because the air had been evacuated during the previous actuation. It is also considered that this total volume will be filled at the initial p_0 atmospheric pressure, after which the piston stroke can be achieved, depending on the load.
Under these conditions, the flow rate supplied as a function of pressure will be:

$$Q_{(p)} = \begin{cases} Q_M; p \leq p_0 \\ Q_M \frac{p_M - p}{p_M - p_0} \;\; ; p_0 < p \leq p_{Max} \\ 0; p > p_{Max} \end{cases} \tag{1}$$

$$Q_M = S_1 v_{Max} \tag{2}$$

As it is shown in Figure 1 also, the mathematical model above is characterized by the linearization of the dependence between flow and pressure.

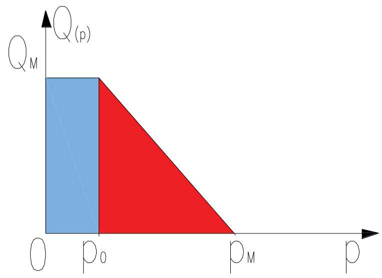

Fig. 1. Characteristic of the pressure – flow source

The pressure source will ensure a flow rate in the range [0, Q_M] at an instantaneous pressure that will not exceed the p_M value.

3. Simplified Mathematical Model of the Pneumatic Cylinder

Let consider the cylinder in Figure 2.

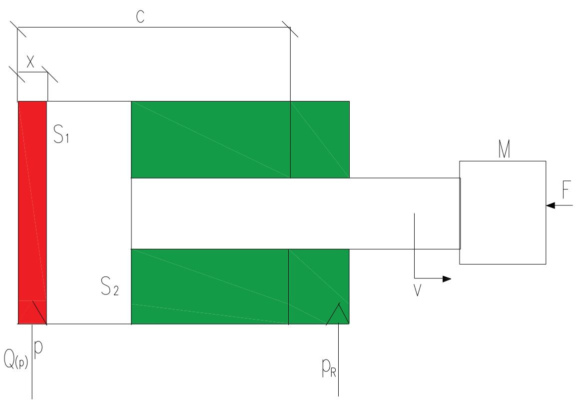

Fig. 2. Pneumatic cylinder to be actuated

The piston of the cylinder goes on stroke c against the resistive force F, moving the mass M. The instantaneous velocity is v and appears as a function of the pressure of the source p, which actuates on the surface S_1 of the piston and of the return pressure p_R that actuates on the surface S_2. The instantaneous position of the piston is given by the size x.
If the flow rate $Q_{(p)}$ and the force F are considered as inlet quantities, the following mathematical model can be taken into consideration:

$$M\frac{d^2x}{dt^2} + b\frac{dx}{dy} + F_F + F = pS_1 - p_R S_2 \tag{3}$$

$$Q_{(p)} = Q_1 + Q_2 + Q_3 \tag{4}$$

In the relations above it was also denoted: t - time, b - linearized damping coefficient, F_F – friction force in the mobile seals, Q_1 – useful flow rate, which entails the movement of the piston, Q_2 – flow rate lost through the mobile seals of the piston, Q_3 - flow rate lost due to the air compressibility.
The useful flow rate Q_1 will be:

$$Q_1 = S_1 \frac{dx}{dt} \tag{5}$$

The lost flow, if any, can be considered as:

$$Q_2 = ap \tag{6}$$

The coefficient a is the one that ensures the proportionality between the lost flow rate and the instantaneous pressure.
The flow rate Q_3, the one lost due to the compressibility of the air, can be expressed considering that the transformations undergone by the air in the cylinder body are of isothermal or adiabatic type.
In the case of the isothermal transformation (for longer durations, of the order of minutes), it is considered:

$$pV = ct \tag{7}$$

In this relation, V is the instantaneous volume in the active chamber of the cylinder.
By deriving the relation above as a function of time, it shall be obtained:

$$Q_3 = \frac{dV}{dt} = \frac{V}{p}\frac{dp}{dt} \sim \frac{V_{Med}}{p_{Med}}\frac{dp}{dt} \tag{8}$$

In this latest relation, there were denoted also: V_{Med} – the average volume of air, which can be considered as $cS_1/2$ and p_{Med} – the average pressure during operation equal to $p_M/2$.
In the case of the adiabatic transformation (for shorter durations, of the order of a few seconds) it will be considered:

$$pV^{\gamma} = ct. \tag{9}$$

After derivation, the following expression will be obtained for the flow rate:

$$Q_3 = \frac{dV}{dt} = \frac{V}{p\gamma}\frac{dp}{dt} \sim \frac{V_{Med}}{\gamma p_{Med}}\frac{dp}{dt} \tag{10}$$

Besides the sizes already defined, in the relation above there is also γ – the adiabatic coefficient considered to be γ = 1.4.
The linearized damping coefficient b in the relation (3) is more difficult to determine, as it depends on several constructive factors, out of which some are specific to the assembly in question. Thus, the longer the return pipes and the more numerous the local resistances, the higher the value.
If there is no backpressure on the return and the length of the hose is negligible, it will be considered $p_R = p_0$.
The mathematical model above applies to the cylinders that do not have braking at the end of the stroke. If the braking is available, the return pressure p_R will be defined as follows:

$$p_R = \begin{cases} p_{R1}; & x \le c - c_F \\ p_{R2} = p_{R1} + \Delta p; & c - c_F < x \le c \end{cases} \tag{11}$$

In the relation above, it was also denoted: p_{R1} – back pressure before braking, p_{R2} - back pressure during braking, Δp – pressure drop due to braking (assumed to be constant), c_F – stroke made under braking.

4. Simulation of the Pneumatic Cylinder Operation in Dynamic Mode

Based on the presented mathematical models, simulations were carried out for different variants, by means of Matlab - Simulink programs packages [7].
The behavior of a pneumatic cylinder with the following characteristics was studied: D – piston diameter 40 mm, d – rod diameter d = 20 mm, total stroke c = 1000 mm, braking stroke c_F = 10% c, maximum imposed velocity v_{Max} = 0.5 m/s. The reduced mass that can be moved is M = 50 Kg and the resistive force is F = 600 N.
The friction forces were neglected and the damping coefficient was considered to be b = 100 N/m/s, including the damping provided by the return devices and pipes.
The pneumatic source can operate at the maximum pressure p_{Max} = 6 bar.
For the first simulation, it was considered that the supply pressure is of step - type and the compressibility of the air in the start-up phase was not taken into account.
In this case, the characteristic shown in Figure 3 was obtained for the stroke velocity.

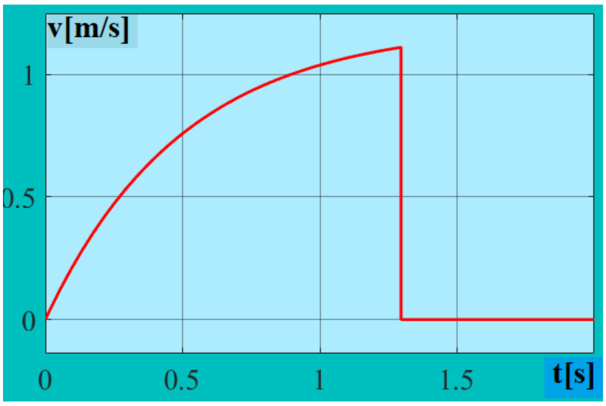

Fig. 3. Characteristic of the velocity for supply with constant pressure (step)

It can be noticed that the velocity increases even above the imposed value, and the stroke is completed in approximately 1.3 seconds as in Figure 4.

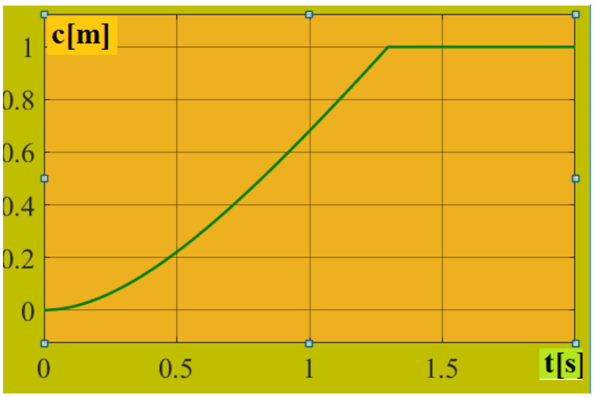

Fig. 4. Characteristic of the travel for supply with constant pressure (step)

This characteristic shows, as the previous one, that the lack of braking at the end of the stroke leads to the sudden stop of the cylinder.
Next, it was considered that the supply is made with the pressure-dependent flow rate, according to the relations (1) and (2) and the characteristic in Figure 1. In this case, the velocity v has the characteristic shown in Figure 5.
The maximum velocity in this case is below 0.2 m/s, therefore below the allowed maximum. The time necessary for the entire stroke is approximately 14 s.

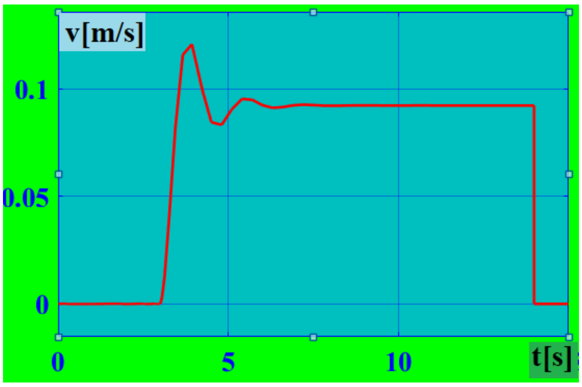

Fig. 5. Characteristic of the velocity for supply with the flow rate Q(p), a function of pressure

As it can be seen from the stroke characteristic presented in Figure 6, the absence of braking at the end of the stroke entails the sudden stop.

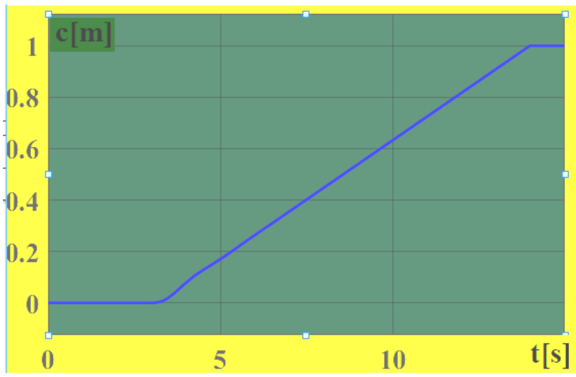

Fig. 6. Characteristic of the stroke for supply with the flow rate Q(p), a function of pressure

At the moment of making the stroke c, the stop is sudden and the pressure p will reach the maximum value p_{Max} and the supplied flow will become zero. The evolution of the pressure over the entire stroke is shown in Figure 7.

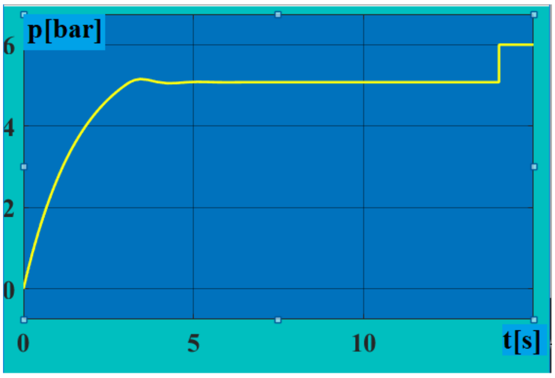

Fig. 7. Characteristic of pressure for supply with the flow rate Q(p)

The characteristic of the source and the dependence of the flow rate on the pressure lead to the increase of the time necessary for completing the stroke.
The lack of braking at the end of the stroke can entail the destruction of the unit because of the sudden stops over time.
Then it was considered that the source is of the type above (Q(p)) and the braking is performed during the last part of the stroke, on the length $c_{F,}$ [1]. The diagram in Figure 8 was used for the simulation.

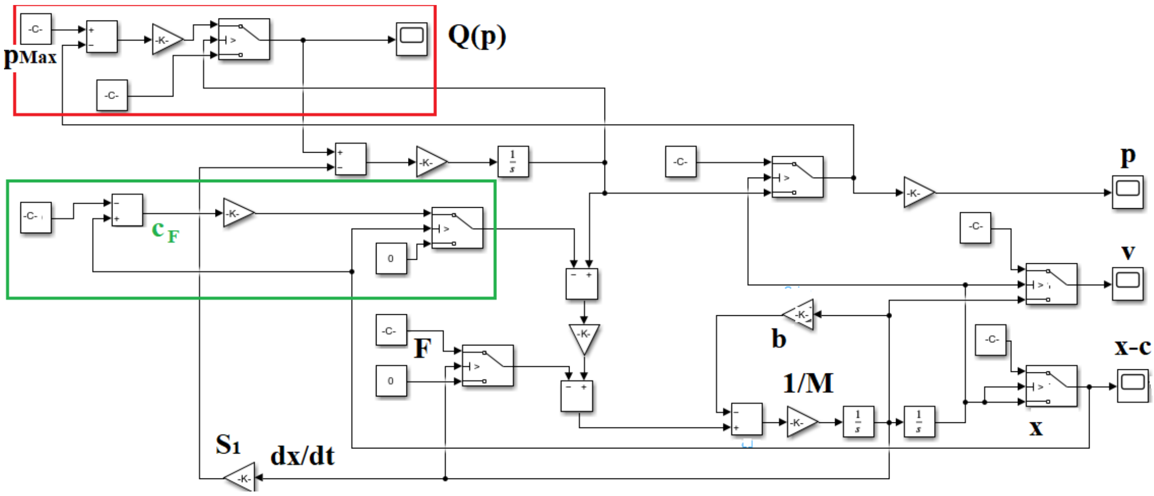

Fig. 8. Simulation diagram for the model with variable pressure and braking at the end of the stroke

The flow rate generation module as a function of the pressure Q(p) can be seen in the upper part of the diagram and immediately below is presented the module that simulates the braking on the segment of stroke c_F.
The characteristic of the flow rate Q(p) on the entire stroke is presented in Figure 9.

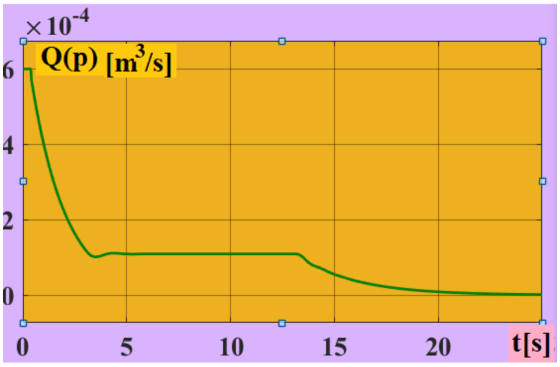

Fig. 9. Characteristic of the source

Initially, the maximum flow is the one programmed for the maximum allowed velocity, after which, increasing the pressure, the flow decreases along the stroke without braking. At the start of the braking, the pressure increases even more, which leads to the decrease of the flow that becomes zero at the end of the stroke.
The evolution of the pressure p is shown in Figure 10.

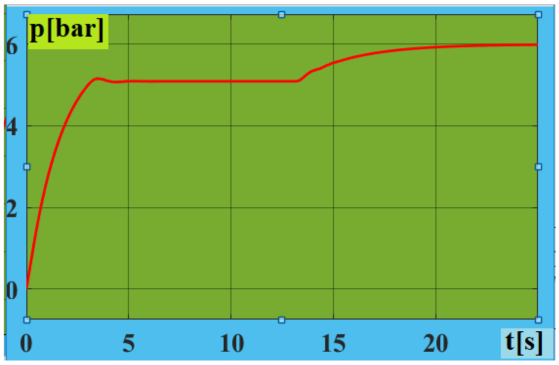

Fig. 10. Evolution of the pressure from the active chamber of surface S_1

The pressure increases until the stroke starts, then – at the beginning of the braking (approximately after 14 s) – the pressure increases until the end of the stroke when it will reach the value $p_{Max} = 6$ bar.
The pressure must overcome the resistive force; afterwards, the stroke starts with the velocity v, according to the characteristic in Figure 11.

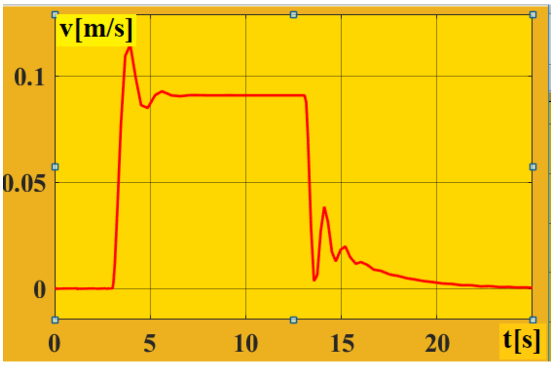

Fig. 11. Characteristic of velocity

The maximum velocity is approximately 0.13 m/s, lower than the maximum allowed. The movement is made with constant velocity up to second 14, after which the braking intervenes and the velocity decreases; the stopping is performed after 20 s from the moment of the supply.
The moment of stopping and the development of the stroke over time is viewed in the form of the instantaneous displacement x and/or of the stroke c as shown in Figure 12.

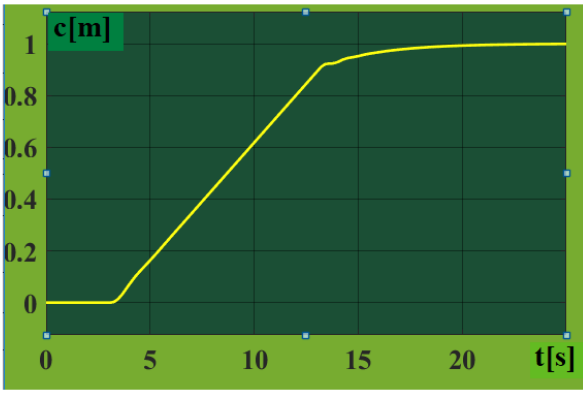

Fig. 12. Characteristic of displacement

From the moment when the braking at the end of the stroke becomes active, the velocity decreases and the effect of damping at the end of the stroke is reduced. In the case of the cylinders with braking at the end of the stroke, the damping (coefficient b) too is improved.

5. Conclusions

Due to the operation with a very compressible environment, the modeling and simulation of the pneumatic cylinders raises much more problems than in the case of the hydraulic cylinders.
The development of models which allow the determination of theoretical characteristics similar to the real ones requires the acceptance of some approximations and linearization.
As for the hydraulic drives, due to the high modulus of elasticity of the oil (about 1.5×10^4 daN/cm^2), the flow is slightly influenced by the pressure. In the case of the pneumatic cylinders, it can be considered that the pressure and the flow rate influence each other isothermally or adiabatically.
The damping coefficient b is variable, depending on the cylinder, but also on the elements of the pneumatic diagram, on the length and DN of the pipes, much more in the case of the pneumatic drive than in the case of the hydraulic drive.

Due to the high velocities developed by the pneumatic cylinders, if big strokes and high masses are involved, the braking at the end of the stroke becomes mandatory.
In order to create correct models, experimental determinations and measurements are necessary for each individual case.
The Matlab - Simulink program package ensures the possibility of creating useful models.

References

[1] Bucuresteanu, Anca, and Daniela Isar. *Pneumatic Elements and Systems. Industrial Applications/Elemente si sisteme pneumatice. Aplicatii industriale.* Bucharest, Certex Publishing House, 2007.
[2] ***. Catalogues and leaflets AIRON, FESTO, PARKER, BOSCH REXROTH.
[3] Moreno, S., and E. Peulot. *Pneumatics in Automation Production Systems/La pneumatique dans les systèmes automatisés de production*. Paris, Éditions Casteilla, 2001.
[4] Johnson, James L. *Introduction to Fluid Power*. U.S.A., Publisher: Delmar Cengage Learning, 2001.
[5] Ali, H.I., S.M. Noor, S.M. Bashi, and M.H. Marhaban. "A review of pneumatic actuators (modeling and control)." *Australian Journal of Basic and Applied Sciences* 3, no. 2 (2009): 440-454.
[6] Rahmat, M.F., N.H. Sunar, S.S. Salim, M.Z. Abidin, A.M. Fauzi, and Z.H. Ismail. "Review on modeling and controller design in pneumatic actuator control system." *International Journal on Smart Sensing and Intelligent Systems* 4, no. 4 (December 2011): 630-661.
[7] ***. MATLAB - SIMULINK Software package.