TECHNICAL UNIVERSITY

Faculty of Mechanical and Precision Engineering

M.Eng. Docho Svetlozarov Dimitrov

„ RESEARCH OF DYNAMIC PROCESSES IN THE CONTROL OF PNEUMATIC MOTORS USING PULSE-WIDTH MODULATION“

AUTOREFERENCE

of a dissertation

for the award of the academic degree of "Doctor"

Field of higher education: 5 „Technical sciences“

Professional field: 5.1 „Mechanical engineering“

PhD program: „Hydraulic and pneumatic drive systems“

Gabrovo, 2026.

The dissertation was reviewed and approved for its official defense at a meeting of the Extended Departmental Council of the Department of Energy Engineering at the Faculty of Mechanical and Instrument Engineering of the Technical University of Gabrovo, held on April 7, 2026.

The dissertation consists of 164 pages. The scientific content is presented in an introduction, five chapters, and a conclusion, and includes 103 figures and 6 tables.

A total of 140 national and international references are cited.

The research for the dissertation was conducted at the Department of Energy Engineering, Faculty of Mechanical and Instrument Engineering, Technical University of Gabrovo.

The official defense of the dissertation will take place on July 9, 2026, at 1:00 p.m. in Room 1300 at the Technical University of Gabrovo.

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Supervisor: Assoc. Prof. Hristo Nedev Hristov, Ph.D., Eng.

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Gabrovo, 2026.

Used symbols and abbreviations

A	area
A_2	throttle cross-sectional area;
A_r	plunger orifice area;
A_t	pipe cross-sectional area (pneumatic line);
A_b	pneumatic cylinder piston area;
a	pressure coefficient;
a_s	speed of sound
b	critical pressure ratio;
C_d	flow coefficient;
C_V	conductivity coefficient;
C_a	acoustic conductivity coefficient;
c_p	piston speed of the cylinder;
d	inner diameter;
D	outer diameter;
F_{mp}	friction force;
H	Heaviside unit function;
I	Electric current;
k	adiabatic index;
k_k	controller gain;
$k_{OB}^{'}$	feedback coefficient;
L	length;
L_t	pipe length;

M	mass flow rate;
M_t	theoretical mass flow rate
M_{b1}	mass flow rate - left side of the pneumatic cylinder;
M_{b2}	mass flow rate - right side of the pneumatic cylinder;
m	fluid mass;
p	pressure;
p_{at}	atmospheric pressure;
p_{in}	system pressure (inlet to the manifold);
p_{out}	outlet pressure from the valve;
p_r	pressure in the receiver;
p_c	compressor pressure;
Q	volume flow rate;
Q_{kor}	corrected volume flow rate;
Q_{SCFM}	volume flow rate according to the NFPA model;
R	gas constant of air;
s	Laplace operator
T_I; T_D	integration and differentiation time constants of the controller;
T_u; k_u	time constant and gain of the electromechanical transducer;
T_v; ξ_v	time constant, damping ratio;
T_{air}	air temperature;
t	time;
t_v	compressor screw pitch;

U	output voltage of the controller;
U_3	setpoint voltage;
U_{OB}	feedback voltage;
ΔU	input voltage;
U_{pc1}	voltage from the left chamber pressure transducer;
U_{pc2}	voltage from the right chamber pressure transducer;
U_a	voltage from the acceleration sensor;
U_{q1}	voltage from the left chamber flow transducer;
U_{q2}	voltage from the right chamber flow transducer;
V	fluid volume;
W_b	cylinder volume, excluding the piston;
x_x	spatial variable;
x_s	plunger displacement;
X	pressure ratios;
X_T	critical pressure drop;
y	piston rod displacement;
y_1, y_2	piston position;
Δy	difference between the initial and final positions of the piston;
β^*	critical pressure ratio;
δ	subsonic coefficient;
μ	throttle flow coefficient;
ρ	gas density;
ϑ	filling coefficient;

$\wp$	integral operator;
ω	angular velocity;
Ψ	flow function;
Ψ_{max}	maximum of the flow function;
ADC	analog-to-digital converter;
EC	electronic controller;
EA	electronic amplifier;
SD	setpoint device;
CO	comparator;
CE	comparator element;
TB	terminal board;
DAC	digital-to-analog converter
PWM	pulse width modulation

RELEVANCE OF THE ISSUE

Pneumatic drive systems are widely used in modern industry due to their simple design, reliability, and ability to achieve fast operating cycles. They are used in automated production lines, robotic systems, transport mechanisms, and various positioning devices.

As industrial automation advances, so do the demands on the precision, energy efficiency, and dynamic characteristics of these systems. A particularly pressing issue arises with electro-pneumatic actuators that use high-speed solenoid valves, where control is often implemented by pulse-width modulation (PWM).

With this type of control, a number of specific dynamic processes arise, related to the characteristics of the valves, air compression, and the nonlinear properties of the pneumatic elements. These processes have a significant impact on positioning accuracy, system stability, and energy efficiency.

Although the significant number of studies in the field of pneumatic actuators, a number of aspects related to the dynamics of PWM-controlled systems and the use of high-speed 2/2-way valves remain under-researched.

This determines the relevance of the present dissertation, which is aimed at investigating the dynamic processes in the pulse-width modulation control of pneumatic actuators. The motivation for developing this dissertation stems from the need for a more in-depth study of the processes occurring in electro-pneumatic systems that use high-speed 2/2 solenoid valves. Conducting such research creates the conditions for improving the mathematical models and control methods for such systems, as well as for the development of more efficient and energy-optimized pneumatic actuators..

CHAPTER I

STATUS OF THE PROBLEM

1.1. Analysis of existing research on this issue.

Electropneumatic control systems are widely used and play an increasingly important role in modern industry. For example, in the field of robotics, in automated machine tools, in the positioning of astronomical and military antennas, and in space technology. In electro-pneumatic systems, the control valve is used to position the cylinder piston. To achieve precise positioning, the correct configuration of the control valves is extremely important. Two types of control valves are most commonly used in research: servo valves and solenoid valves. Solenoid valves include fast-acting two-position two-way (2/2) valves [3], [4], [5], [6]. Servo valves are frequently used due to their operational advantages—stable linear characteristics and high precision. Although high-speed solenoid 2/2 valves exhibit nonlinear deviations in their operating characteristics, they are widely used due to their low cost, simple structure, and ease of maintenance.

1.2. Conclusions and findings from the analysis:

Based on the analysis of existing electro-pneumatic systems and control methods, the following main conclusions can be drawn::

1. Pneumatic drive systems are widely used in modern industry due to their simple design, reliability, and ability to achieve fast operating cycles.

2. Various types of valves are used to control pneumatic actuators—proportional valves, servo valves, and fast-acting solenoid valves.

3. The use of proportional valves allows for smooth control, but these solutions are characterized by higher cost and more complex design.

4. For this reason, 2/2-way high-speed pneumatic valvesolenoid valves controlled by pulse-width modulation (PWM) are used in a number of practical applications.

5. PWM control enables effective regulation of flow rate and pressure, but leads to the occurrence of specific dynamic processes in the pneumatic system.

6. Existing mathematical models of electro-pneumatic systems often do not sufficiently account for the dynamic characteristics of high-speed valves and their influence on system operation.

7. This necessitates a more in-depth study of the dynamic processes in pneumatic systems using high-speed 2/2 solenoid valves and PWM control.

The analysis conducted shows that, despite the widespread use of electro-pneumatic systems employing high-speed solenoid valves, the dynamic processes involved in pulse-width modulation control have not been sufficiently studied. This leads to difficulties in creating adequate mathematical models and in developing effective control algorithms for such systems.

PURPOSE AND MAIN OBJECTIVES OF THE DISSERTATION

Purpose of the dissertation:

The purpose of this dissertation is to investigate the dynamic processes in electro-pneumatic systems using high-speed 2/2-way solenoid valves controlled by pulse-width modulation, and to develop mathematical models that allow for a more accurate description and analysis of the operating characteristics of such systems.

Main objectives of the dissertation:

Based on the literature review and with a view to achieving the dissertation’s objective, the following main tasks have been formulated:

1. Analysis of existing methods for controlling electro-pneumatic actuator systems.
2. Investigation of the structural and dynamic characteristics of 2/2-way high-speed solenoid valves.
3. Development of a mathematical model of the actual flow characteristic of a high-speed solenoid valve.
4. Establishment of a mathematical model of an electro-pneumatic positioning system.
5. Conducting experimental studies to determine the model parameters.

6. Performing simulation studies and verifying the developed mathematical models.

CHAPTER II

DEVELOPMENT OF A FLOW CHARACTERISTIC MODEL FOR A HIGH-SPEED 2/2-WAY PNEUMATIC VALVE

2.1. Theoretical and experimental models of the static characteristics of high-speed 2/2 valves.

To experimentally determine the flow characteristics of the valves used, the ISA model for valve flow was employed, which accounts not only for the valve's flow coefficient but also for the critical pressure drop factor, which is determined experimentally.

This model was developed to account for the fact that two valves with identical flow coefficients can have different flow rates at identical pressures. For every valve, the complexity of the flow velocity profile geometry is an important factor, with more complex geometry leading to higher values.

For volumetric air flow, the ISA flow equation is:

$$Q = \begin{cases} 22.67\, C_V\, p_{in} \left(1 - \dfrac{X}{3X_T}\right)\sqrt{\dfrac{X}{T_{air}}} & ако X < X_T \\ 15.11\, C_V\, p_{in} \sqrt{\dfrac{X_T}{T_{air}}} & ако X \geq X_T \end{cases}, \qquad (1.37)$$

where:

$$X = \frac{p_{in} - p_{out}}{p_{in}} = 1 - \left(\frac{p_{out}}{p_{in}}\right) \qquad (1.38)$$

2.1.1. Development of a mathematical model of the actual flow characteristic of a high-speed 2/2-way (On/Off) pneumatic valve.

To accomplish the set tasks, it is necessary to define an accurate mathematical model that describes the actual static flow rate through the pneumatic high-speed pneumatic valvevalves (2-port) as a function of changes in supply pressure, inlet and outlet pressure, and the geometry of the fluid path. This can be achieved by experimentally determining the flow function and then selecting the closest of the presented mathematical models or creating a new one.

For the experiments, pneumatic high-speed pneumatic valvevalves are used—models SX12F-AH, SX12F-EH, and SX12F-JH from SMC Japan.

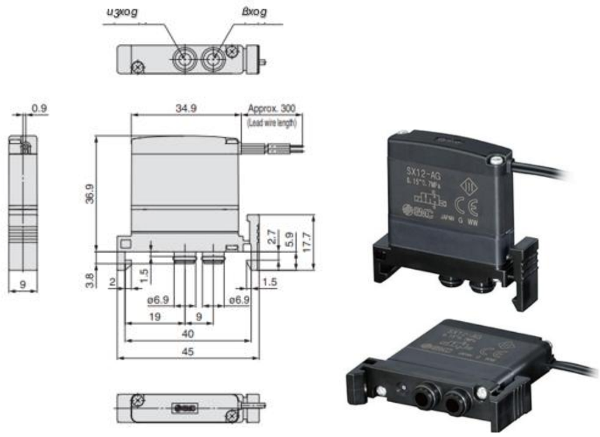


Fig. 2.1. *Schematics and illustrative photo of the SX 12 F high-speed 2/2 valve.*

The closing element of this type of valve consists of a plate made of magnetic material with a specific geometry. During operation, the electromagnet attracts the plate, and when deactivated, it returns to its original position under the action of pressure (no spring is involved in the valve's operation). The closing element of this type of valve consists of a plate made of magnetic material with a specific geometry. During operation, the electromagnet attracts the plate, and when deactivated, it returns to its original position under the action of pressure (no spring is involved in the valve's operation).

This type of electro-pneumatic valves is designed to achieve a shorter response time. Their parameters reach times of: on-time – 0.45 ms, off-time – 0.4 ms. Valves of this type have low power consumption (4 W) and a high operating frequency of up to 120 Hz.

The mathematical model of flow rate is a complex function of many variables, such as supply pressure, inlet pressure, temperature, and others. Various mathematical models are known. When developing mathematical models of the flow rate characteristics of a high-speed pneumatic two-port valve, atmospheric conditions must be assumed as described in ISO 6358: 1989 (E). The model assumes that the air temperature does not differ from the atmospheric temperature while the high-speed 2/2 valve is operating.

2.1.2. Determination of the parameters of the mathematical model of the flow characteristic of a high-speed 2/2 valve through experimental investigation.

An experimental setup (test bench), shown in Fig. 2.2, has been developed for the experimental determination of the flow rate of a 2/2-way high-speed pneumatic valvevalve and the critical pressure drop coefficient, which define the specific flow characteristics of 2-port pneumatic high-speed pneumatic valvevalves.

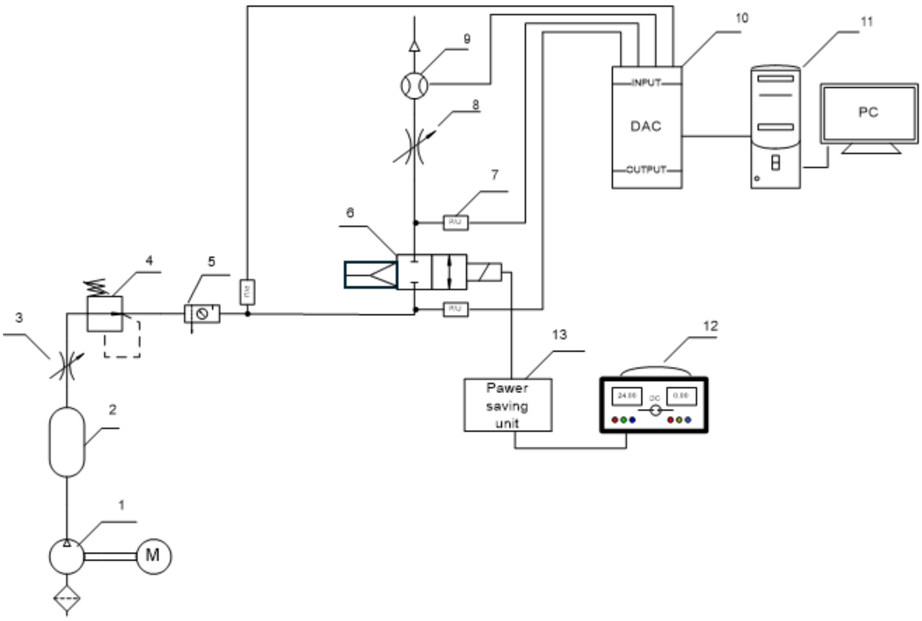


Fig.2.2. *Schematic representation of a pneumatic actuator with a measuring system.*

1 - screw compressor; 2 - receiver; 3 - valve (shut-off valve); 4 - safety valve; 5 – air treatment unit; 6 - high-speed pneumatic valve2/2-way on/off valve; 7 – pressure transducer; 8 – valve; 9 – flow meter; 10 – terminal board; 11 – computer; 12 – power supply unit; 13 – energy-saving unit;

The required pressure in the compressor receiver (1) must reach 8 bar. The inlet pressure from the air preparation unit is measured using the pressure sensor (p/u). The pressure at the inlet of the high-speed pneumatic valvevalve and the pressure at the outlet of the valve are measured. The flow rate passing through the high-speed pneumatic valvevalve is measured using flow meters (9). The experiment is conducted with a control voltage of 24 V from the power supply unit (12) and operating pressures of 2, 3, 4, 5, 6, and 7 bar. The valve (8) is opened from zero flow rate to full opening in steps of ~10 l/min. The measurement data is recorded.

The instruments have been calibrated. The next step is to convert each of the measured channels to natural units.

The measured gauge pressures are converted to absolute pressure, and the ratio

is used to plot the static characteristic curve.:

$$\frac{p_{out}}{p_{in}} = \beta \qquad (2.5)$$

Since all mathematical models describing the flow rate through the valve are proportional to the pressure drop, the corrected flow rate must be determined by multiplying the ratio of the pressure at zero flow to the measured supply pressure at the actual flow rate:

$$Q_{kor} = Q\,\frac{p_{noflow}}{p_{in}} \qquad (2.6)$$

Experimental graphs illustrating the relationship for the three models used are being constructed:

$$\frac{p_{out}}{p_{in}} \text{ to } Q_{kor} \qquad (2.7)$$

p_{noflow} - pressure at zero flow;

p_{in} - valve inlet pressure;

p_{out} - valve outlet pressure.

The graph (Fig. 2.7) shows the static flow characteristics of a pneumatic high-speed pneumatic valve2/2-way valve, model AH.

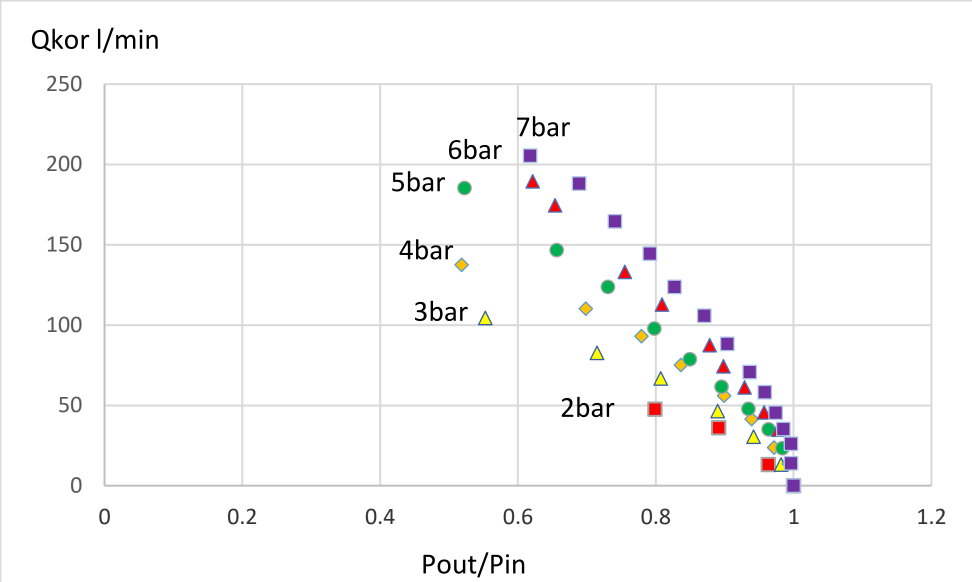


Fig. 2.3. *Experimental data on the static flow characteristics of the SMC Japan SX12F-AH pneumatic high-speed pneumatic valve2/2-way valve from 2 to 7 bar.*

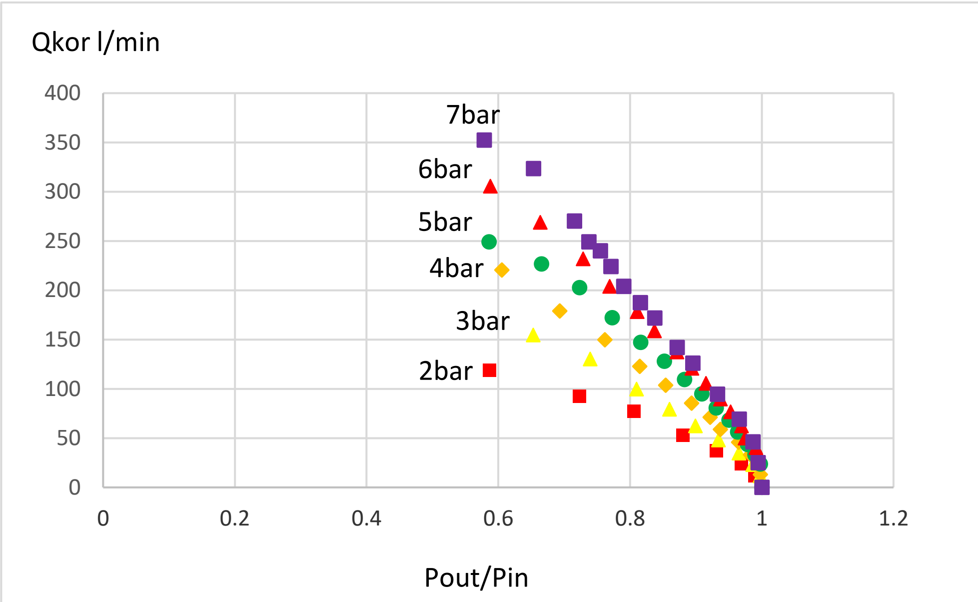


Fig.2.4. *Experimental data on the static flow characteristics of the SMC Japan SX12F-EH pneumatic high-speed pneumatic valve2/2-way valve from 2 to 7 bar.*

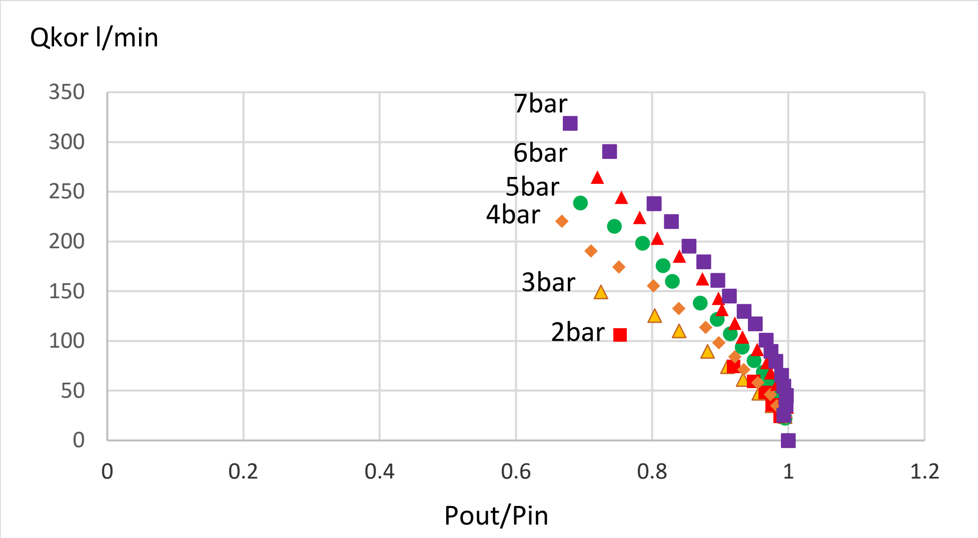


Fig.2.5. *Experimental data on the static flow characteristics of the SMC Japan SX12F-JH 2/2-way high-speed pneumatic valve pneumatic valve from 2 to 7 bar.*

The graphs in Figs. 2.3–2.5 present the experimentally obtained static flow characteristics of a pneumatic high-speed pneumatic valve2/2 valve for the respective JH models. These characteristics determine the flow rate and pressure losses at a given inlet pressure; the closer the ratio between the inlet pressure and the outlet pressure is to 1, the lower the losses. The highest flow rate of approximately 210 liters per minute is achieved at a pressure of 7 bar.

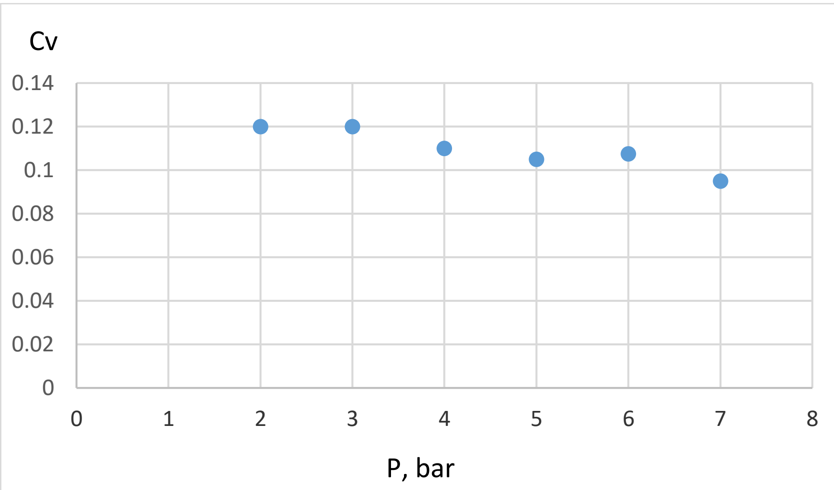


Fig. 2.6. *Conductivity coefficient* C_v *for a high-speed pneumatic valve SX12F-AH.*

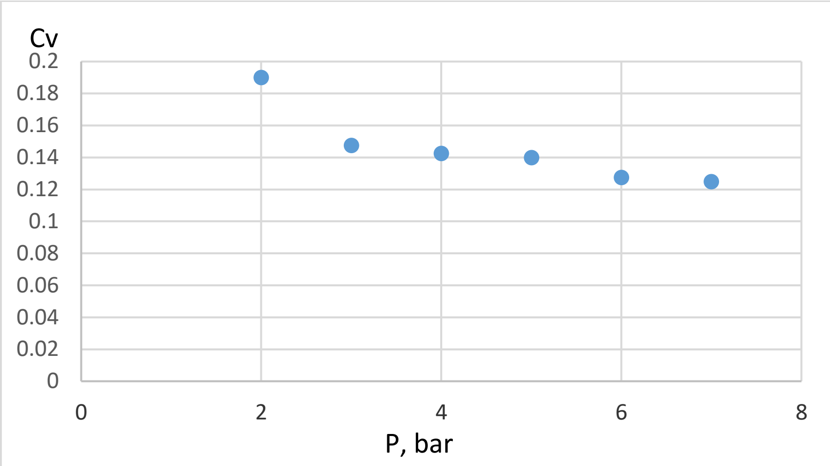


Fig. 2.7. *Conductivity coefficient* C_v *for a high-speed pneumatic valve SX12F-EH.*

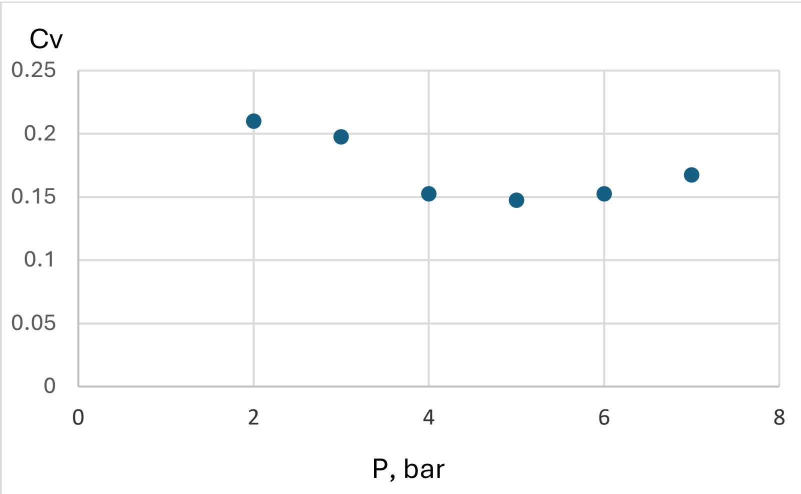


Fig. 2.8*. Conductivity coefficient* C_v *for a high-speed pneumatic valve SX12F-JH.*

Figures 2.6–2.8 show the change in the conductivity coefficient for the various valve models as the inlet operating pressure increases from 2 to 7 bar.

By minimizing the RMS error (root mean square deviation) between the data and the model, we obtain values for C_v:

Table 2.1. Average values for C_v.

Valve model	C_v	X_T
SX12F-AH	0.115	1
SX12F-EH	0.150	1
SX12F-JH	0.175	1

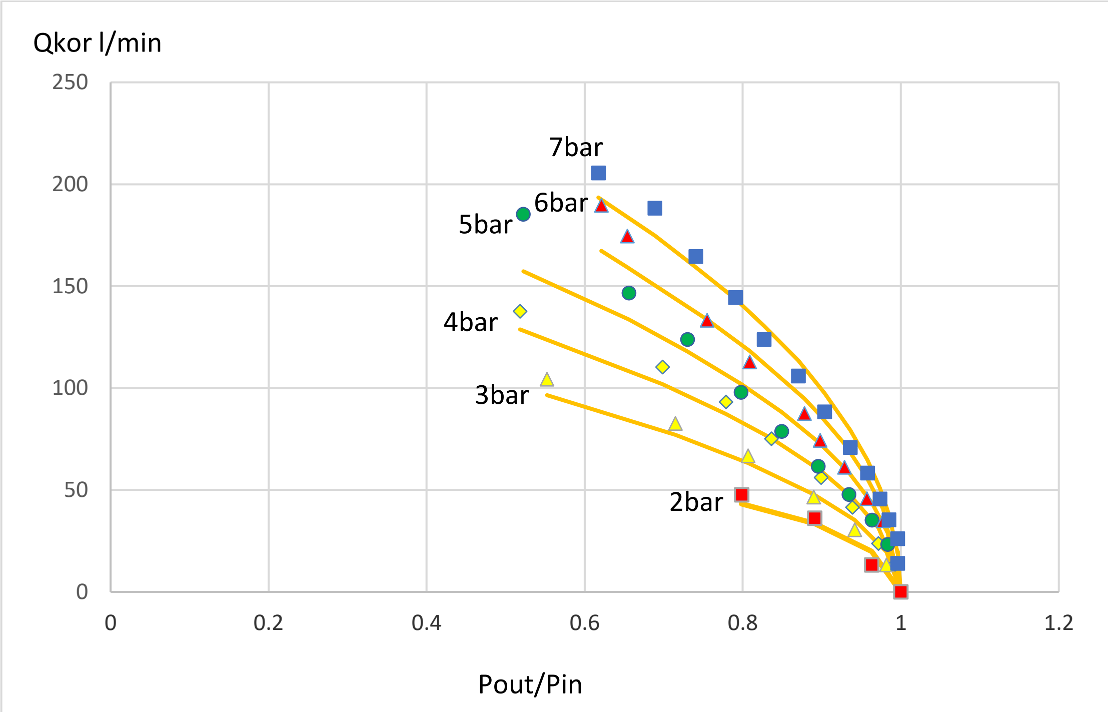


Fig. 2.9. *Static flow-pressure curve 2,3,4,5,6 u 7 bar, voltage from 24V, model ISA -* C_v *=0,115;* X_T *=1 for a high-speed pneumatic valve SX12F-AH.*

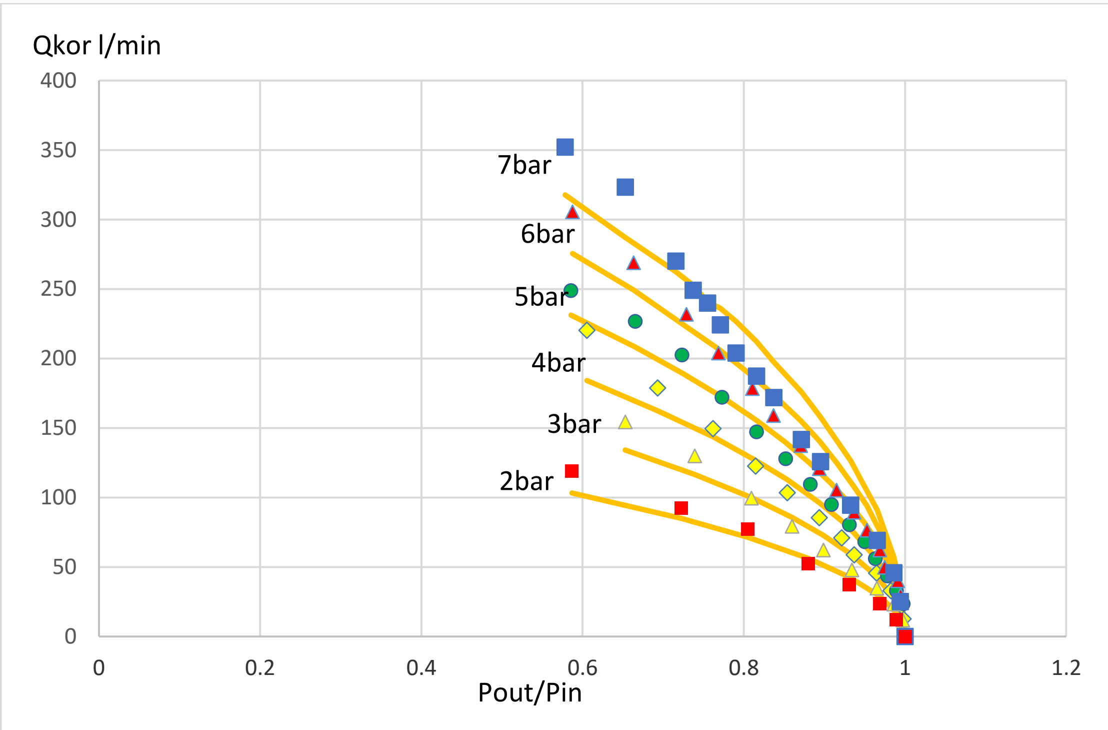


Fig.2.10. *Static flow-pressure curve 2, 3, 4, 5, 6 u 7 bar, voltage from 24V, model ISA -* C_v *=0,150;* X_T *=1 for a high-speed pneumatic valve SX12F-EH.*

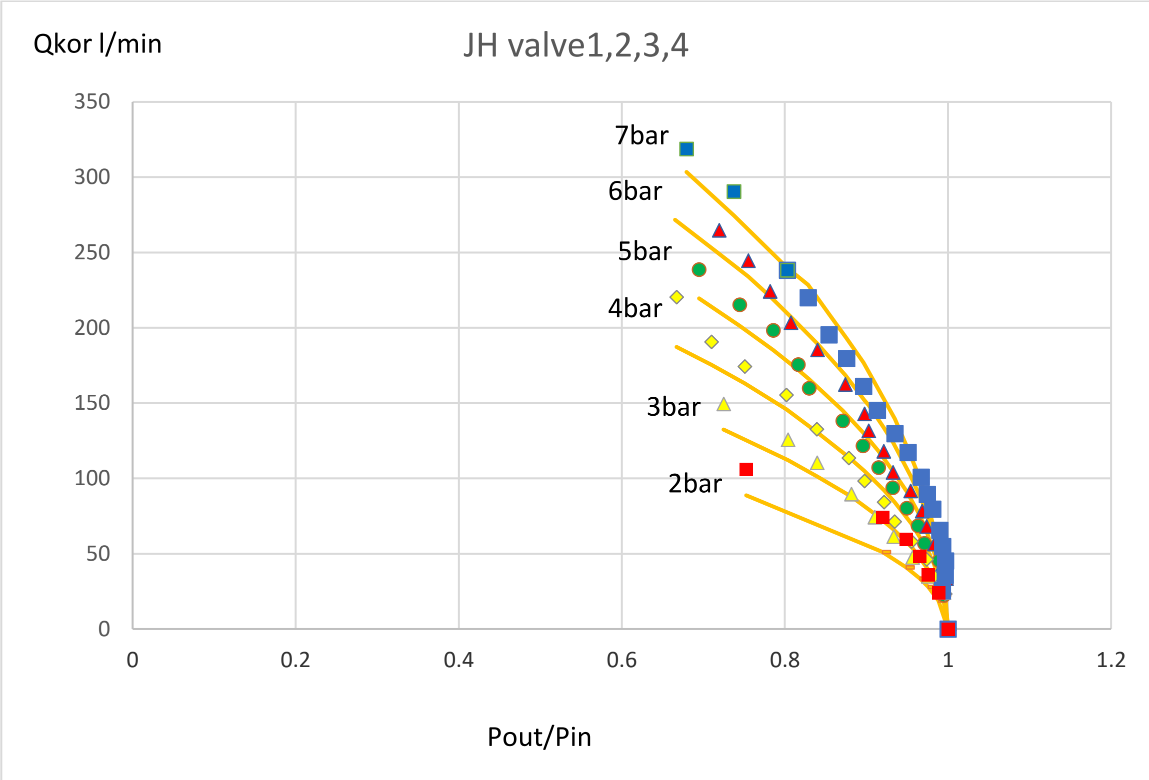


Fig.2.11. *Static flow-pressure curve 2, 3, 4, 5, 6 u 7 bar, voltage from 24V, model ISA -* C_v *=0,175;* X_T *=1 for a high-speed pneumatic valve SX12F-JH.*

The graphs presented in Figs. 2.9–2.11 show the extent to which the experimental data obtained overlap with or approximate the theoretical data from the mathematical models for the respective valve models. Here, the data from the mathematical models are shown as a solid line, while the experimental data are represented by different symbols for the various operating pressures.

CONCLUSIONS AND ANALYSIS OF THE RESULTS:

Based on the theoretical and experimental studies conducted in this chapter, the following conclusions can be drawn:

1. An analysis of existing mathematical models for describing the flow characteristics of pneumatic valves shows that widely used models (St. Venant–Wantzel, NFPA, and ISA) allow for the description of gas flow processes but require adaptation when applied to specific design solutions for high-speed pneumatic valves.

2. To define the flow characteristics of the investigated 2/2-way high-speed pneumatic valves, the ISA model was selected and adapted to the unit system used and to the specific features of the valves.

3. An experimental test bench was developed to determine the static flow characteristics of SMC SX12F high-speed pneumatic valves, allowing for the measurement of pressure and flow rate under various operating conditions.

4. Through the experimental studies conducted, sets of static flow characteristics were obtained at operating pressures ranging from 2 to 7 bar for various valve models.

5. By minimizing the RMS error between the experimental data and the mathematical model, the model parameters were determined:

C_v =0.115 for valve SX12F-AH

C_v =0.150 for valve SX12F-EH

C_v =0.175 for valve SX12F-JH

6. The resulting mathematical model accurately describes the experimentally determined static flow characteristics and can be used to develop a mathematical model of the electro-pneumatic system in the subsequent chapters of the dissertation.

CHAPTER III

DEVELOPMENT OF A MATHEMATICAL MODEL OF AN ELECTRO-PNEUMATIC POSITIONING SYSTEM WITH PWM

3.1. Mathematical model of the electrical and electromagnetic characteristics of a high-speed 2/2-way valve.

The mathematical model of a high-speed 2/2 pneumatic valve can be analyzed in terms of three main characteristics: electrical, electromagnetic, and mechanical characteristics.

The electrical characteristic can be expressed as follows:

$$U = RI + L\frac{dI}{dt} + I\frac{dL}{dt} \tag{3.1}$$

where:

U - operating voltage;

R - equivalent resistance;

I - coil current;

L - equivalent inductance.

It is noted that the air gap is, in fact, negligible. Therefore, the electromagnetic characteristic can be expressed as:

$$IN = H_c l_c + H_g l_g = H_c\left(l_c \frac{H_g}{H_c} l_g\right) \tag{3.2}$$

$$L = \frac{N\phi}{I} = \frac{N^2}{Rm} \tag{3.3}$$

$$F_m = \frac{\lambda\phi^2}{2\mu_0 S} \tag{3.4}$$

where:

N - number of turns on the coil;

H_c, H_g - the equivalent magnetic field intensities in the core and in the air gap;

l_c, l_g - the equivalent lengths of the magnetic circuit in the core and in the air gap;

ϕ - magnetic stream;

R_m - equivalent resistance;

λ - a constant related to the magnetic flux;

μ_0 - air permeability;

S - the cross-sectional area of the plunger.

Combining the following equations:

$$H_C = \frac{\phi}{S_e \mu_c} \tag{3.5}$$

$$l_g = \chi_i - \chi \tag{3.6}$$

where:

S_e - the effective cross-sectional area of the flow path in the core;

μ_c - core permeability;

χ - the movement of the plunger;

χ_i - the initial movement of the plunger;

$\chi = \chi_{\max}$ - The variable inductance of the magnetic circuit can be described as a function of the plunger's position:

$$L = \frac{N^2 S_e \mu_c}{l_c + \frac{H_g}{H_c} l_g} \tag{3.7}$$

Since the air gaps are small enough that the periphery of the magnetic circuit can be neglected:

$$L = \frac{N^2 S_e \mu_c}{l_c + \frac{\mu_c}{\mu_0}(\chi_i - \chi)} \tag{3.8}$$

The dynamic response of a high-speed pneumatic valvepneumatic valve can be recorded as follows:

$$m\frac{d^2 x}{dt^2} = F_m - (F_t - F_s) - k\frac{dx}{dt} P_s A \tag{3.9}$$

$$\int_0^{t\,mon} \frac{dx}{dt} = d \tag{3.10}$$

$$\int_0^{t\,moff} \frac{dx}{dt} = -d \tag{3.11}$$

where:

F_t - peak flow rate;

F_s - the power of constant flow;

k - coefficient of friction;

P_s - inlet pressure;

A - the cross-sectional area of the inlet opening;

t_{mon} - valve opening time;

t_{moff} - valve closing time;

d - stroke of the plunger.

The power of the transient flow rate and the power of the steady-state flow rate can be expressed as:

Under non-steady-state flow conditions:

$$F_t = C_d \omega L_d \sqrt{2\rho \Delta P} \frac{dx}{dt} \tag{3.12}$$

In steady-state flow:

$$F_s = 2C_v C_d A_0 \Delta P \cos\Theta \tag{3.13}$$

where:

C_v - the fluid conductivity coefficient;

C_d - the flow coefficient;

A_o - the cross-sectional area of the valve;

ΔP - the pressure difference across the valve;

Θ - the flow angle,

ω - the angular velocity;

ρ - fluid density.

When the critical leakage condition is reached, the members $\frac{dx^2}{dt}$ and $\frac{dx}{dt}$ are equal to zero. Therefore, the critical electromagnetic force F_{cm} can be written as follows:

$$F_{cm} = P_s A \pm F_s \quad (3.14)$$

When the high-speed pneumatic valve operates in the critical flow regime from the off state to the on state, the flow force acts in the opposite direction to the pressure, so that equation (3.14) can be written as:

$$F_{cm} = P_s A - 2C_v C_d A_0 P_{AB} \cos\Theta \quad (3.15)$$

P_{AB} - pressure difference between the inlet and outlet of the high-speed pneumatic valve.

Conversely, when the high-speed pneumatic valvepneumatic valve operates in the critical flow-off mode from the on to the off state, the flow and pressure forces act in the same direction, so that equation (3.15) can be written as:

$$F_{cm} = P_s A + 2C_v C_d A_0 P_{BA} \cos\Theta \quad (3.38)$$

P_{BA}- pressure difference between the outlet and inlet of the high-speed pneumatic valvepneumatic valve.

By combining equations (3.2–3.5), (3.15), and (3.16), the critical current at the moment of turn-on I_{on} and exclusion I_{off} can be written as follows:

$$I_{on} = \frac{N}{L_{off}} \sqrt{\frac{2\mu_0 S (P_s A - 2C_v C_d A_0 P_{AB} \cos\Theta)}{\lambda}} \quad (3.17)$$

$$I_{off} = \frac{N}{L_{on}} \sqrt{\frac{2\mu_0 S (P_s A + 2C_v C_d A_0 P_{BA} \cos\Theta)}{\lambda}} \quad (3.18)$$

L_{on} и L_{off} - equivalent inductances during switching.

when the high-speed pneumatic valve operates in the off and on states, respectively, these are constants that can be determined using equation (3.30). In both equations (3.17) and (3.18), pressure is a key factor that has a significant influence on the critical current. In a steady state, on-state, or off-state, the equivalent inductance can be treated as a constant, so that the transient process of the winding current can be written as follows:

$$I = I_i + \left(\frac{U}{R} - I_i\right)\left(1 - e^{-t\frac{R}{L}}\right) \tag{3.19}$$

where

I_i the starting current of the winding.

In this way, the delay time can be recorded as:

$$t_d = \frac{L}{R}\ln\frac{U - I_i R}{U - IR} \tag{3.20}$$

where t_d is the delay time.

Therefore, the delay times at the moment of opening $t_{d\,on}$ and closing $t_{d\,off}$ The valve settings can be saved as follows:

$$t_{d\,on} = \frac{L_{off}}{R}\ln\frac{U - I_i R}{U - \frac{NR}{L_{off}}}\sqrt{\frac{2\mu_0 S\left(P_s A - 2C_v C_d A_0 P_{AB}\cos\Theta\right)}{\lambda}} \tag{3.21}$$

$$t_{d\,off} = \frac{L_{on}}{R}\ln\frac{U - I_i R}{U - \frac{NR}{L_{on}}}\sqrt{\frac{2\mu_0 S\left(P_s A + 2C_v C_d A_0 P_{BA}\cos\Theta\right)}{\lambda}} \tag{3.22}$$

According to equations (3.20), (3.21), and (3.22), it is easy to find:

A) During the opening phase, the control voltage has a positive value to increase the current, so that the opening delay time increases when the voltage is higher.

B) The control voltage is zero or even negative during the closing process, which means that the higher pressure helps reduce the closing delay time.

C) A high inrush current is useful for reducing during the turn-on phase, but a low inrush current is useful for reducing during the turn-off phase.

D) A high supply voltage is effective for reducing $t_{d\,on}$, but even a small amount, or even a negative amount, is effective in reducing $t_{d\,off}$ in the process of closing.

Therefore, pressure has a significant impact on the dynamic characteristics of the high-speed pneumatic valvepneumatic valve, but when combined with an effective control algorithm, the flow rate, which determines the operating state of the high-speed pneumatic valve, can be controlled to the desired value to reduce the effect of pressure on the dynamic characteristics. For these reasons, an energy-saving unit, described in Chapter 4 (Fig. 4.4.), is used to control the operation of the high-speed pneumatic valve.

$$F_m = \frac{\lambda \phi^2}{2\mu_0 A_v} \tag{3.23}$$

where: F_m is the electromagnetic force, λ is a constant related to the decaying magnetic flux, μ_0 is the air permeability, and A_v is the cross-sectional area of the magnetic flux through the armature ϕ .[65],[66]

The mechanical characteristics of the moving parts can be described as follows:

$$F_{p2} = (p_{v2} - p_{v1})A_v \tag{3.24}$$

$$p_{v1} = k_{d1} p_{in} \tag{3.25}$$

$$p_{v2} = k_{d2} p_{in} \tag{3.26}$$

$$m\frac{d^2x(t)}{dt^2} + \frac{bdt(t)}{dt} + F_{p2} + mg = F_m \tag{3.27}$$

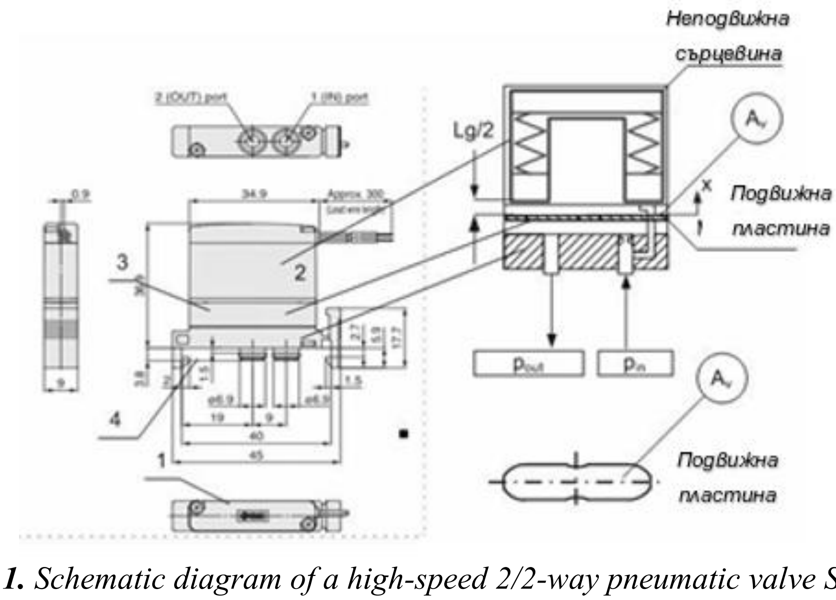


Fig.3.1. *Schematic diagram of a high-speed 2/2-way pneumatic valve SMC SX.*

3.2. Conventional PWM control method for two pairs of high-speed 2/2-way valves:

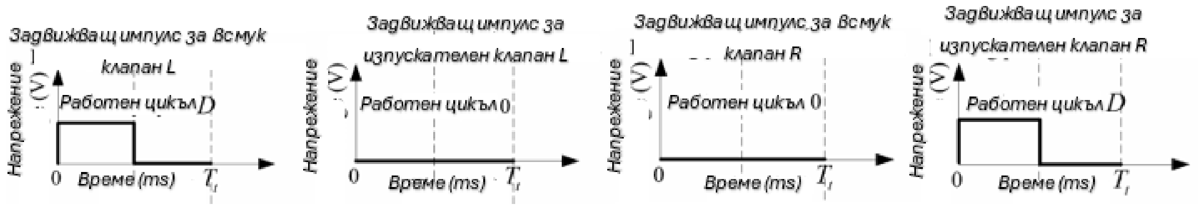


Fig.3.2. *Conventional PWM control method using a high-speed 2/2-way valve SMC SX.*

The operating method for controlling four high-speed 2/2-way valves in an electro-pneumatic servo system with PWM is as follows: After a signal is sent from the signal generator, the first and fourth valves initially activate, while the second and third remain inactive. In the next stage, they switch, with the second and third valves operating.

CONCLUSIONS AND ANALYSIS OF THE RESULTS:

Based on the developed mathematical model of the electro-pneumatic positioning system, the following conclusions can be drawn:

1. An extended mathematical model of an electro-pneumatic positioning system with PWM control has been developed, which integrates the mathematical descriptions of the main elements of the actual system - compressor, receiver, pneumatic lines, pneumatic cylinder, and fast-acting 2/2 solenoid valves.

2. The developed mathematical model accounts for the main physical processes in the system, including the dynamics of the mass flow rate of compressed air, the pressure changes in the receiver and the pneumatic cylinder, as well as the influence of resistances in the pneumatic lines.

3. A mathematical model of the electromagnetic and mechanical dynamics of a 2/2-way high-speed pneumatic valve has been developed, allowing for the analysis of the valve’s opening and closing processes and the determination of switching delay times.

4. Analysis of the resulting equations shows that the pressure at the valve inlet has a significant effect on the valve's dynamic characteristics and on the on and off times.

5. The resulting mathematical model allows for a description of the dynamics of the electro-pneumatic system and can be used in conducting simulation studies and developing control algorithms for the system.

CHAPTER IV

CONTROL OF THE SPEED OF A PNEUMATIC CYLINDER USING HIGH-SPEED 2/2-WAY VALVES WITH A PWM AND DETERMINATION OF THE STATIC FLOW CHARACTERISTICS

In order to experimentally verify and validate the developed mathematical model of the electro-pneumatic positioning system, experimental studies were conducted on a real system.

To control the speed of various types of pneumatic cylinders using a controller that employs pulse-width modulation (PWM) and to obtain the static flow characteristics, an electronic unit with a PWM generator and an energy-saving amplifier was used to control high-speed 2/2 (ON/OFF) valves.

Digitally generated PWM signals, obtained using specialized LabVIEW software and hardware from NI, are used for the study. An output driver implemented with specialized electronic components is used to control the high-speed valves. The use of CMOS for direct control of MOSFET transistors is appropriate due to a number of simplifications in the selection of the operating circuit and the power supply.

4.1. Development of an experimental test bench.

A special test bench, shown in Figs. 4.1 and 4.2, has been developed for experimental studies of dynamic processes in pneumatic actuator systems.

The aim of the study is to determine the speed of a pneumatic cylinder equipped with a width-modulated controller. The regulator consists of high-speed 2/2-way pneumatic valves from SMC Japan (models SX12F-AH, EH, and JH), an electronic control unit, and an energy-saving unit.

The tests should be conducted in three stages, each involving four valves, alternating between models, as the presented models have different characteristics.

Control of the piston position in the double-acting pneumatic cylinder model "AIGNEP MJ0250125" within the electro-pneumatic system is achieved using four high-speed 2/2 pneumatic valves.

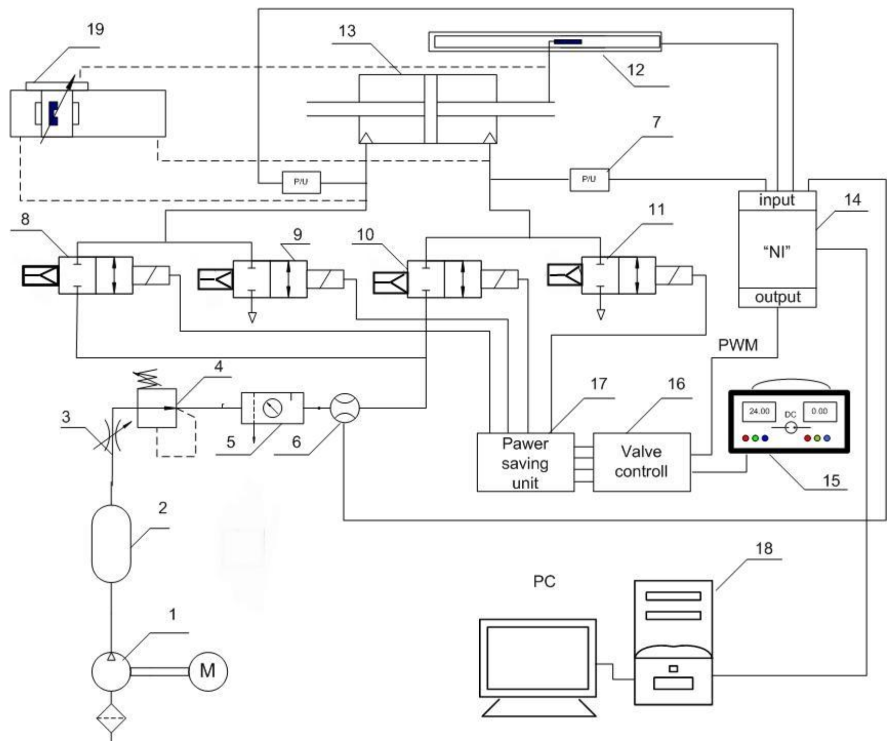


Fig. 4.1. *Schematic of an experimental test rig using a cylinder with a double-ended rod. 1 – screw compressor; 2 – receiver; 3 – shut-off valve; 4 – pressure-reducing valve; 5 – air treatment unit; 6 – flow meter; 7 – pressure transducer; 8, 9, 10, 11 – high-speed pneumatic valve2/2-way on/off valves; 12 - potentiometric sensor, 13 - pneumatic cylinder with double-acting rod; 14 – terminal board; 15 – power supply unit; 16 – controller; 17 – energy-saving unit; 18 – computer; 19 – rodless pneumatic cylinder.*

The operating principle of the pneumatic system shown in Fig. 4.1 is as follows: When all four valves are closed, the piston remains in the desired position. When valve 8 (as numbered of the schematic) and valve 11 are open, the piston moves forward.

Similarly, when valve 9 and valve 10 are open, the piston moves backward. The solenoid is activated via pulse-width modulation (PWM) control, implemented using analog circuits.

To achieve high-speed valve operation and coil displacement, a high-power control coil and high power consumption are required. The technical solution for the high-speed pneumatic valve involves a powerful 80 W, 24 VDC coil, which enables rapid control.

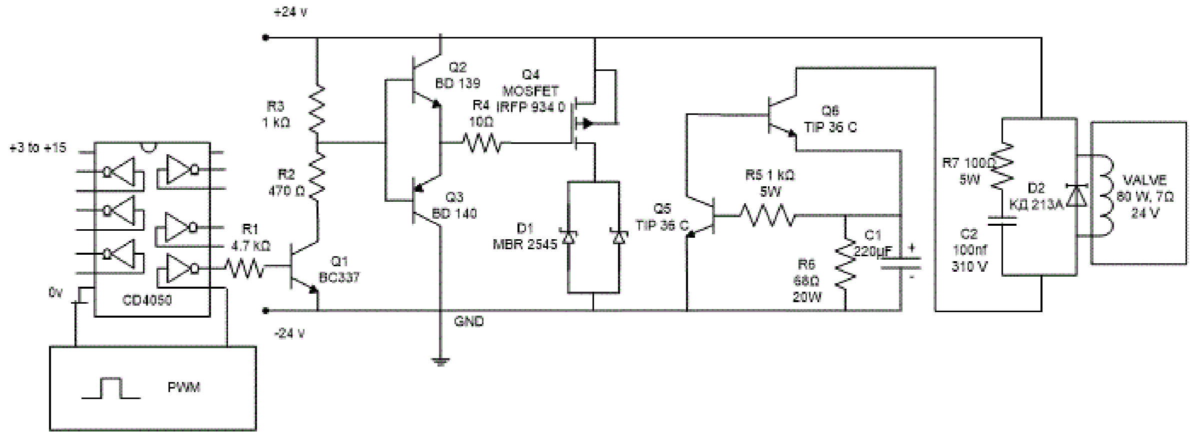


Fig.4.2. Combined control circuit with a PWM signal and MOSFET transistors, and an energy-saving unit for a single valve.

Once the pneumatic valve is powered by 24 VDC, its coil moves to switch from the "OFF" position to the "ON" position, but holding it in this position usually does not require full power, only a partial amount to keep the coil in the operating position; this is typically 70 to 75% less than the rated power (Fig. 2.3).

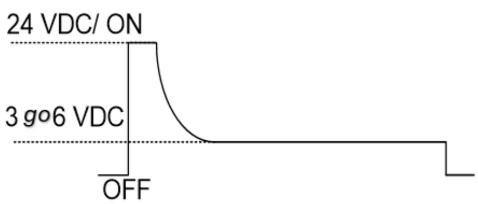


***Fig.4.3.** Start-up and operating voltage of the high-speed pneumatic valve.*

4.2. A virtual tool for managing, collecting, and processing data in real time.

The virtual instrument developed in the LabVIEW environment is a control and measurement system for an electro-pneumatic positioning system, implemented using pulse-width modulation of high-speed electro-pneumatic valves (Fig. 4.6). The tool is designed for experimental research and precise position control of pneumatic actuators, utilizing a closed-loop control system with position feedback and a PID controller to generate the control signal [139].

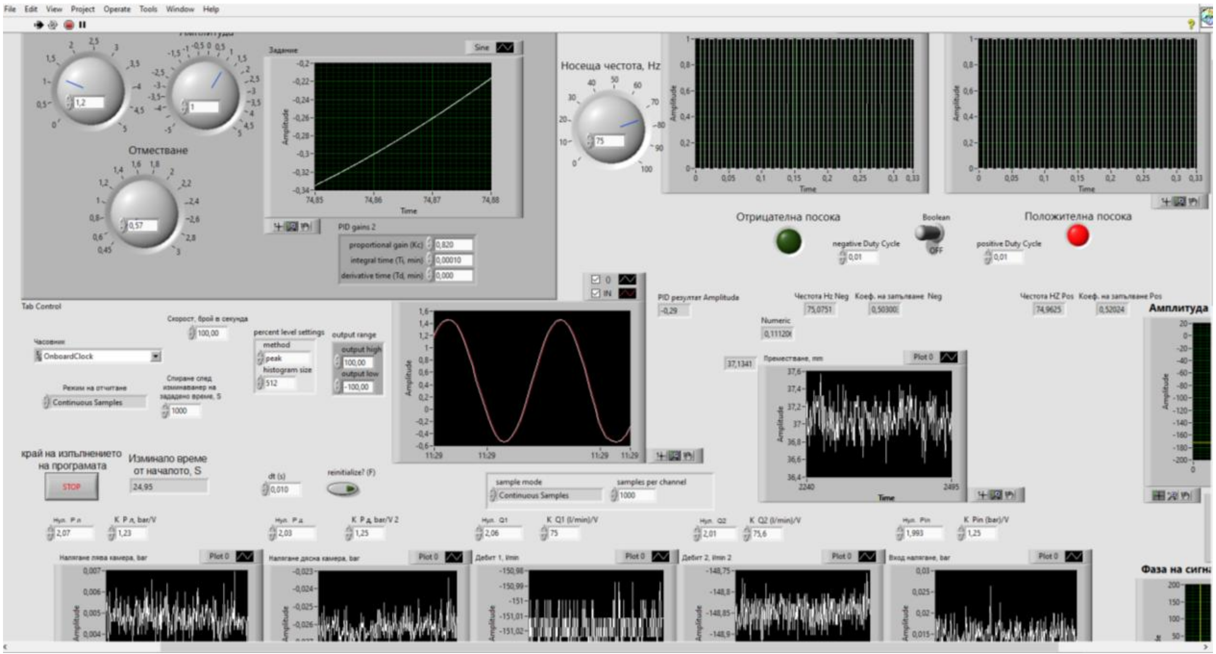


Fig.4.4. *Overview of the virtual instrument interface.*

The virtual instrument consists of a user panel and a block diagram, which together enable the real-time measurement, processing, control, and visualization of measured and controlled variables. The user panel also provides an interface for entering setpoints (Fig. 4.7), selecting the control mode, and configuring the controller parameters, as well as graphical visualization of the cylinder displacement, pressures, flow rate, and control signals (Fig. 4.9). The block diagram implements the control logic, the algorithms for calibrating the measurement channels, and the generation of PWM signals for the electro-pneumatic valves (Fig. 4.8).

4.3. Experiments conducted and presentation of their results.

4.3.1. An Experimental Study of Speed Control Using 2/2-Way Pneumatic High-Speed Valves, an Energy-Saving Unit, and PWM Control.

In the experiments conducted, a single-step input signal is applied; the piston rod of the pneumatic cylinder moves a precisely defined distance and stops at the specified position. The pneumatic cylinder is controlled at different PWM frequencies and different duty cycles.

The objective is to illustrate the displacement of the pneumatic cylinder's piston, the pressures in the left (inlet) chamber of the cylinder, and the flow rate entering the electro-pneumatic positioning system during operation. The results must be presented graphically. The selected operating pressure for the experiments on the electro-pneumatic positioning system is 5 bar..

The results of the experiments, presented graphically:

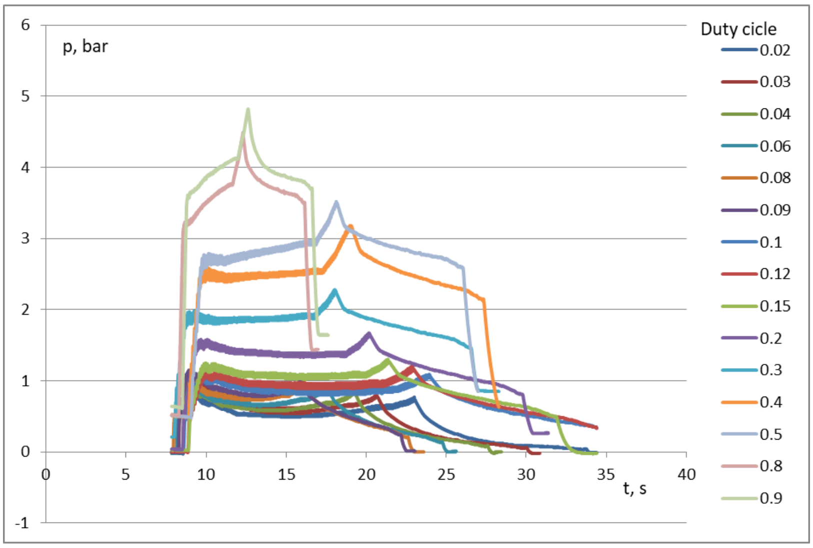


Fig.4.5. *A change in pressure in the left chamber of the pneumatic cylinder with a PWM amplitude of 20 Hz and a change in the duty cycle from 0,02 до 0,9 (2-90%).*

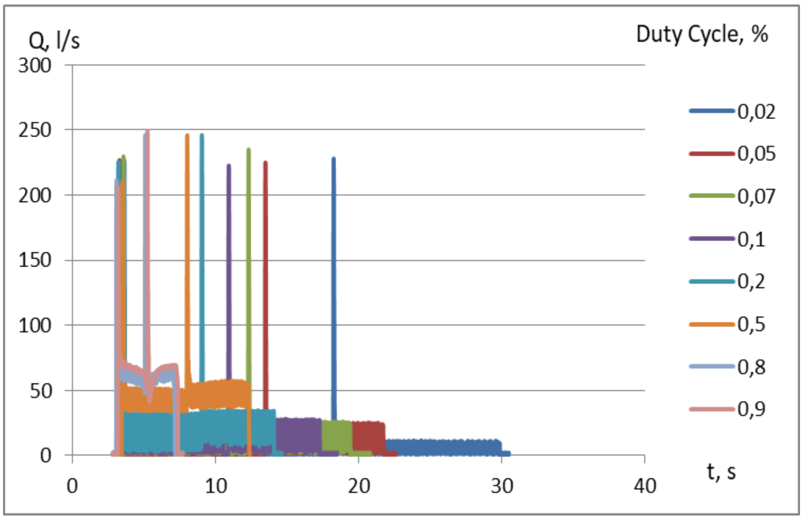


Fig.4.6. *Change in flow rate at a PWM amplitude of 20 Hz and a change in the duty cycle from 0,02 до 0,9(2-90%).*

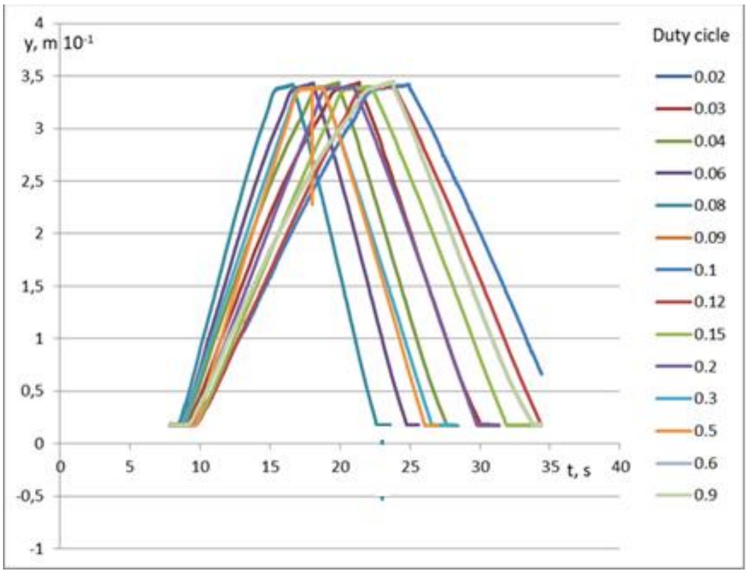


Fig.4.7. *Moving the piston of the pneumatic cylinder with a PWM amplitude of 20 Hz and changing the duty cycle from 0,02 до 0,9(2-90%).*

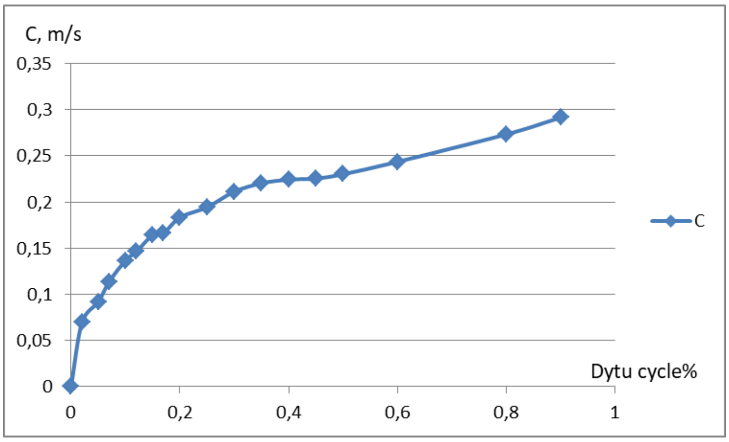


Fig.4.8. *Adjustment of the piston speed of a pneumatic cylinder with a PWM amplitude of 20 Hz and a change in the duty cycle from 0,02 до 0,9(2-90%).*

The graphs in Figs. 4.5–4.8 present data on the pressure change in the left chamber of the pneumatic cylinder, the change in flow rate, the displacement of the pneumatic cylinder piston, and the change in velocity from the experiments conducted on the displacement of the pneumatic cylinder piston over time at a frequency of 20 Hz and various duty cycles in the range from 2 to 90 %.

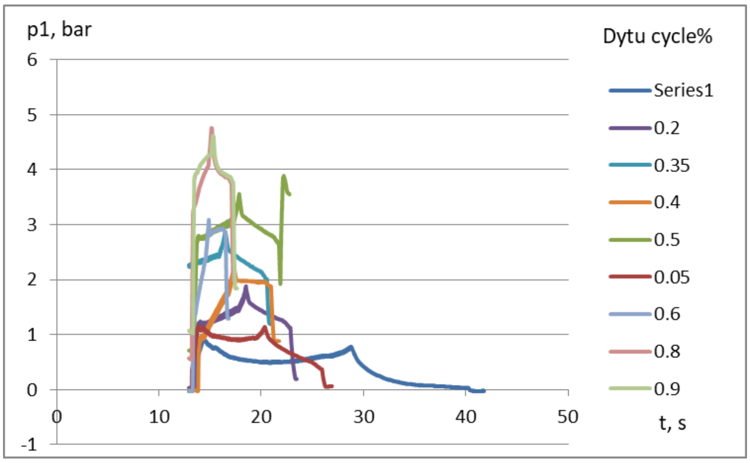


Fig.4.9. *A change in pressure in the left chamber of the pneumatic cylinder with a PWM amplitude of 50 Hz and a change in the duty cycle from 0,02 до 0,9(2-90%).*

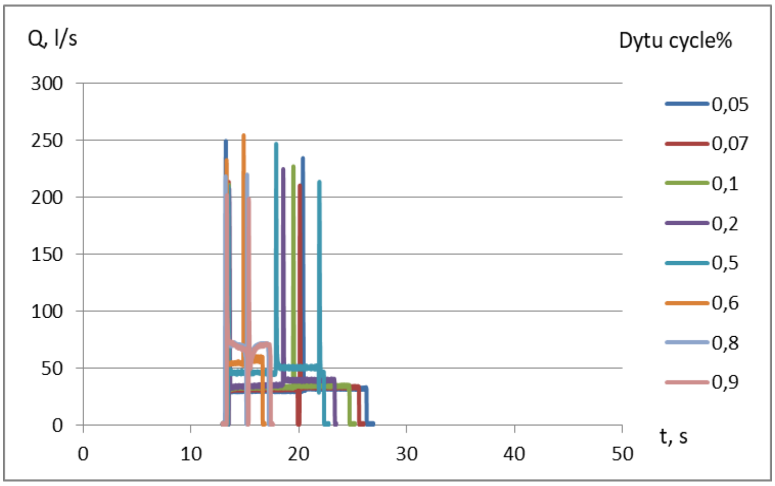


Fig.4.10. *Change in flow rate at a PWM amplitude of 50 Hz and a duty cycle ranging from 0.02 to 0,9 %(2-90%).*

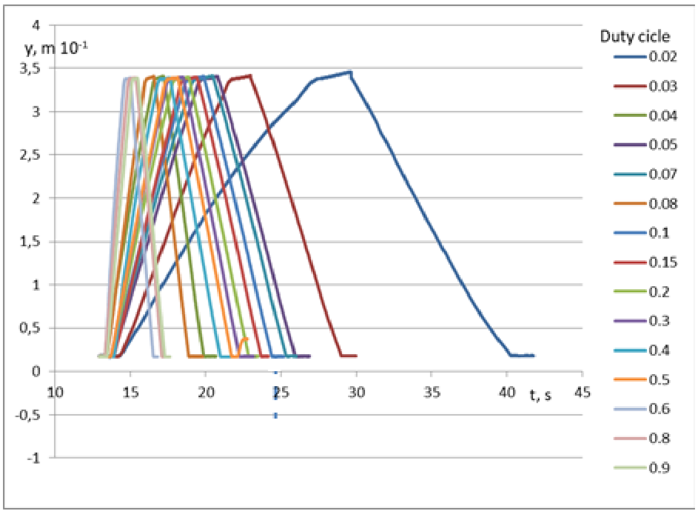


Fig.4.11. *Moving the piston of the pneumatic cylinder with a PWM amplitude of 50 Hz and changing the duty cycle from 0,02 до 0,9 (2-90%).*

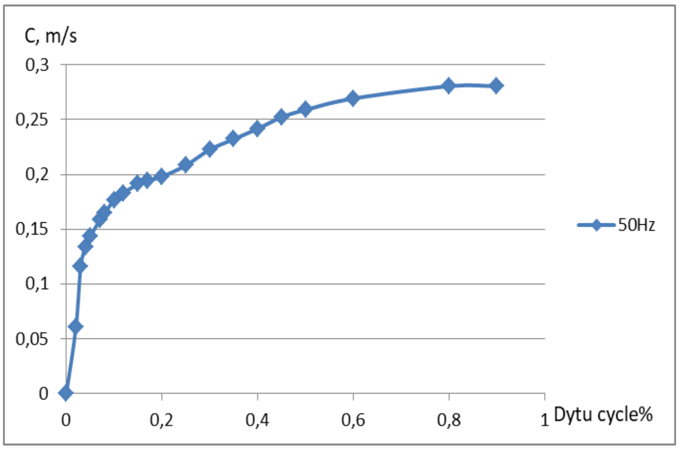


Fig.4.12. *Adjustment of the piston speed of a pneumatic cylinder with a PWM amplitude of 50 Hz and a change in the duty cycle from 0,02 до 0,9 (2-90%).*

The graphs in Figs. 4.9–4.12 show the data on the change in pressure in the left chamber of the pneumatic cylinder, the change in flow rate, the displacement of the pneumatic cylinder piston, and the change in velocity from the experiments conducted on the displacement of the pneumatic cylinder piston over time at a frequency of 50 Hz and various duty cycles in the range from 2 to 90 %.

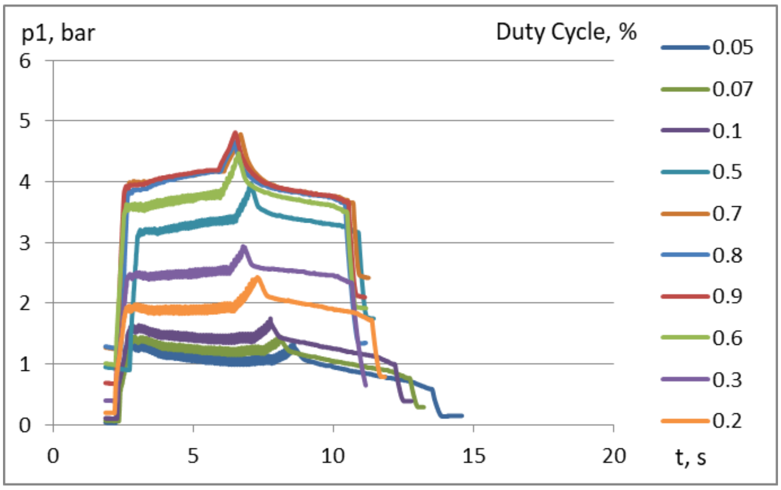


Fig.4.13. *A change in pressure in the left chamber of the pneumatic cylinder with a PWM amplitude of 70 Hz and a change in the duty cycle from 0,02 до 0,9 (2-90%).*

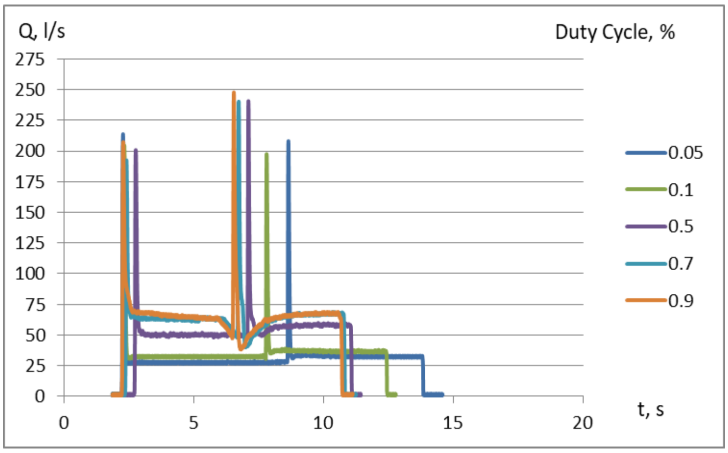


Fig.4.14. *Change in flow rate at a PWM amplitude of 70 Hz and a change in the duty cycle from 0,02 до 0,9 (2-90%).*

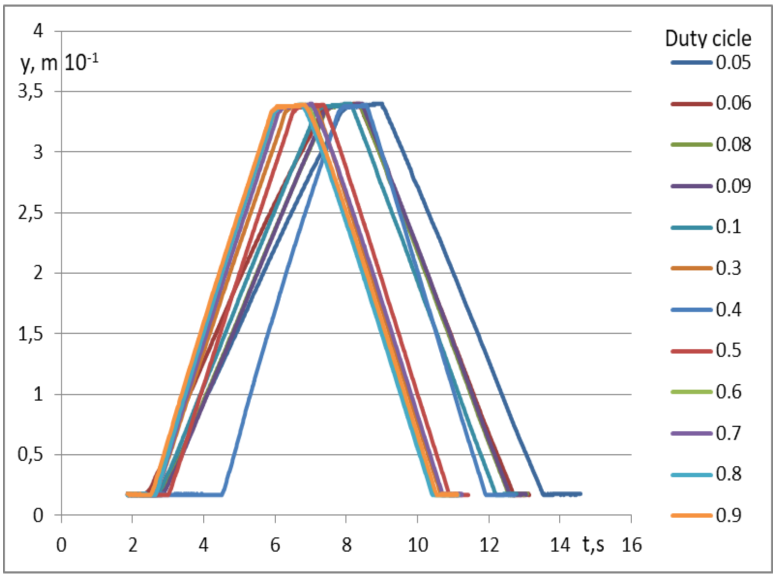


Fig.4.15. *Displacement of the piston in the rodless pneumatic cylinder with a PWM amplitude of 70 Hz and a change in the duty cycle from 0,02 до 0,9 (2-90%).*

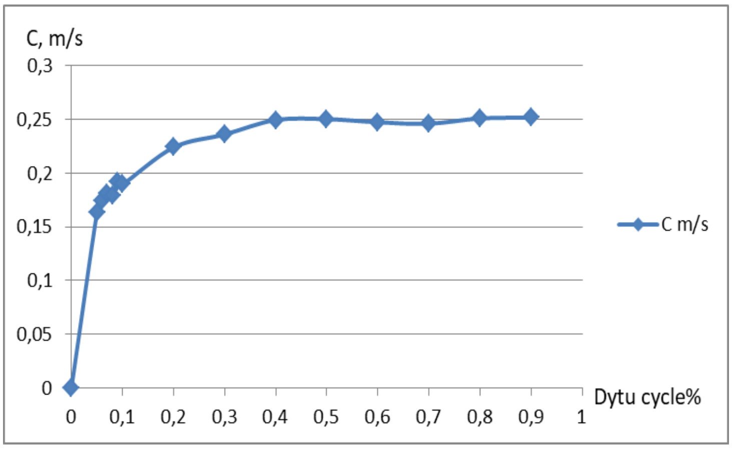


Fig.4.16. *Variation of the piston speed of a rodless pneumatic cylinder with a PWM amplitude of 70 Hz and a change in the duty cycle from 0,02 до 0,9 (2-90%).*

The graphs in Figs. 4.13–4.16 present data on the pressure change in the left chamber of the pneumatic cylinder, the change in flow rate, the displacement of the pneumatic cylinder piston, and the change in velocity from the experiments conducted on the displacement of the pneumatic cylinder piston over time at a frequency of 70 Hz and various duty cycles in the range from 2 to 90 %.

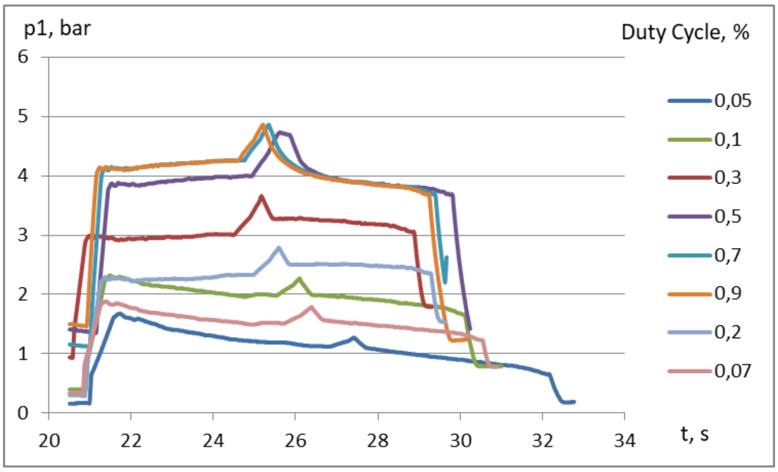


Fig. 4.17. *A change in pressure in the left chamber of the rodless pneumatic cylinder with a PWM amplitude of 100 Hz and a change in the duty cycle from 0,02 до 0,9 (2-90%).*

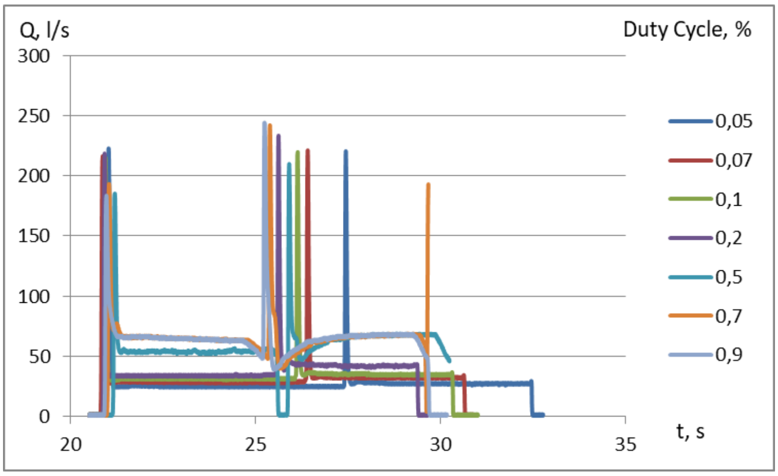


Fig.4.18. *Change in flow rate at a PWM amplitude of 100 Hz and a duty cycle of 0,02 до 0,9 (2-90%).*

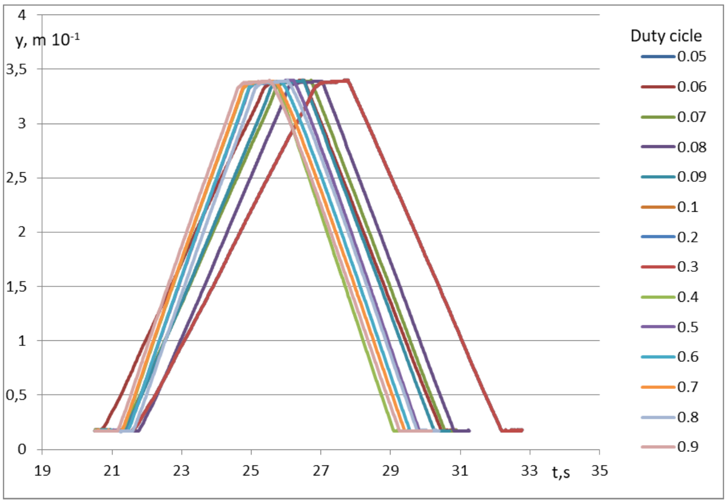


Fig.4.19. *Displacement of the piston in the rodless pneumatic cylinder with a PWM amplitude of 100 Hz and a change in the duty cycle from 0,02 до 0,9 (2-90%).*

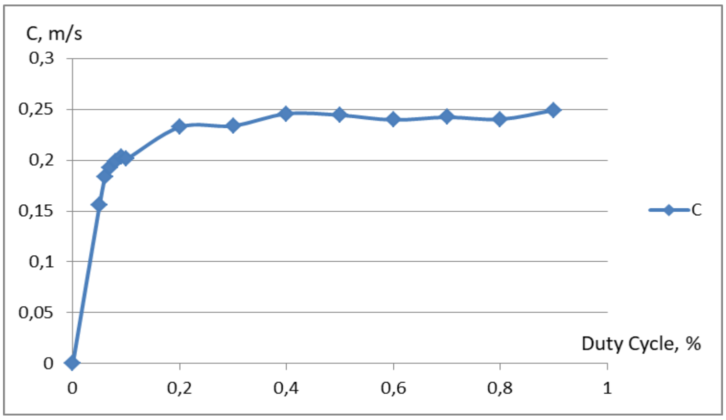


Fig. 4.20. *Variation of the piston speed of a rodless pneumatic cylinder with a PWM amplitude of 100 Hz and a change in the duty cycle from 0,02 до 0,9 %.*

The graphs in Figs. 4.14–4.20 show the data on the change in pressure in the left chamber of the pneumatic cylinder, the change in flow rate, the displacement of the pneumatic cylinder piston, and the change in velocity from the experiments conducted on the displacement of the pneumatic cylinder piston over time at a frequency of 100 Hz and various duty cycles in the range from 2 to 90 %.

4.3.2. An Experimental Study of Dynamic Processes in an Electropneumatic Positioning System with High-Speed 2/2-Way Valves and PWM:

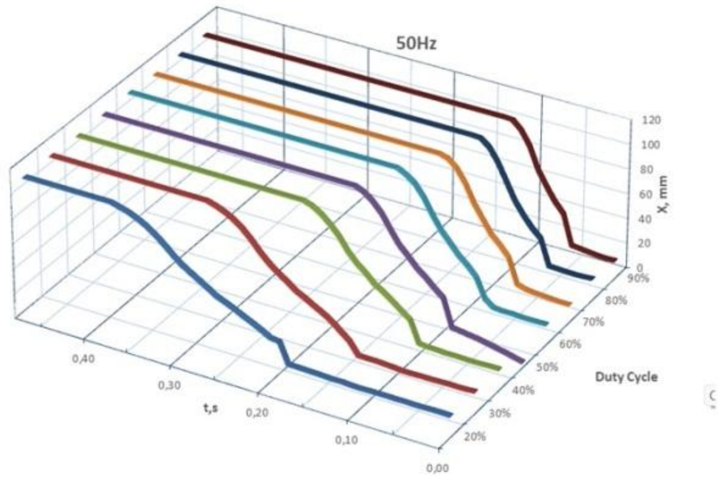


Fig.4.21. *Displacement of the pneumatic cylinder rod at a frequency of 50 Hz and different duty cycles.*

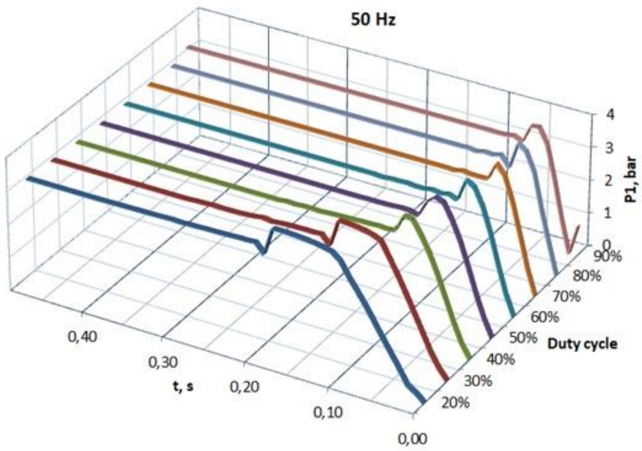


Fig. 4.22. *Pressure in the left chamber of the pneumatic cylinder at a frequency of 50 Hz and different duty cycle percentages.*

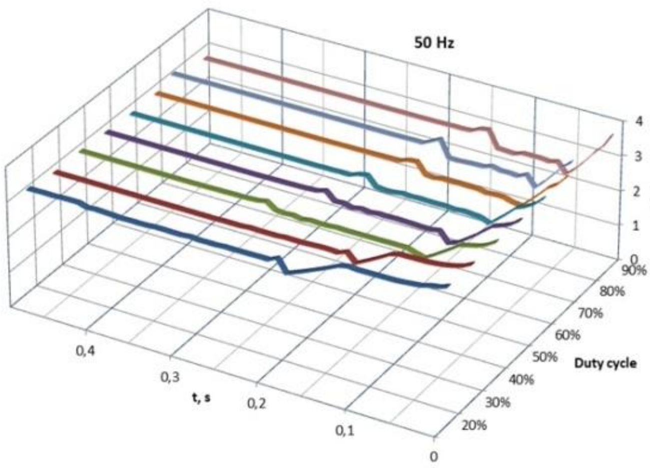


Fig.4.23. *Pressure in the right chamber of the pneumatic cylinder at a frequency of 50 Hz and with varying duty cycle percentages.*

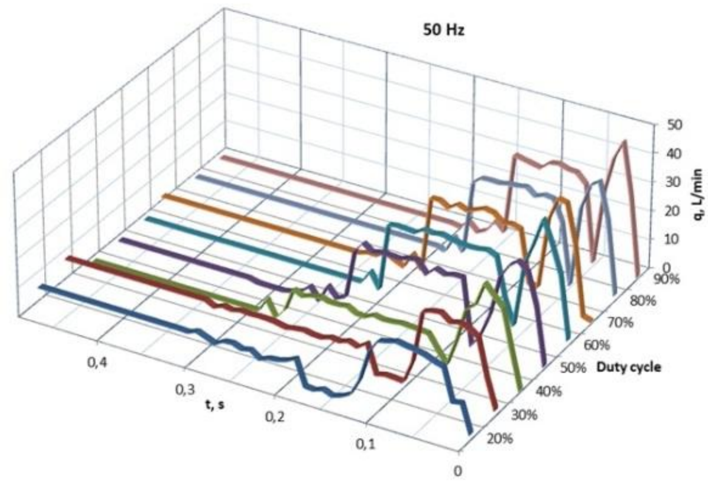


Fig.4.24. *Change in the flow rate entering the electro-pneumatic positioning system at a frequency of 50 Hz.*

The linear three-dimensional graphs in Figs. 4.21–4.24 show the changes over time in the displacement of the pneumatic cylinder piston, the pressures in the left and right chambers, and the inflow rates in the electro-pneumatic positioning system, with the values arranged sequentially for different percentage fillings of the duty cycle, ranging from 20% to 90%. These were experimentally investigated at carrier frequencies of 30, 50, and 90 Hz. At the beginning of the operating processes, an intense increase in flow rates is observed, followed by a decrease with some fluctuations caused by the PID-controlled PWM's tendency to reach the desired position in the shortest possible time, but without deviations. The highest peak flow rate was achieved at 90% duty cycle.

Experimental studies were conducted on the time-dependent changes in the piston displacement of the pneumatic cylinder, the pressures in the left and right chambers, and the inflow rates in the electro-pneumatic positioning system at frequencies of 30, 50, and 90 Hz. Here, the data from the other studies are presented in tabular form; the rest can be found in the dissertation.

Table.4.1. *Experimental results for 30 Hz.*

Dutty cycle, %	**30 Hz**			
	Delay, s	***Time to move, s***	***Distance, mm***	***C, m/s***
20%	0.08	0.08-0.22	100	0.500
30%	0.07	0.07-0.19	100	0.583
40 %	0.06	0.06-0.17	100	0.636
50%	0.05	0.05-0.15	100	0.700
60%	0.04	0.04-0.13	100	0.707
70%	0.03	0.03-0.11	100	0.795
80%	0.03	0.03-0.10	100	1.000
90%	0.03	0.03-0.10	100	1.000

Табл.4.2. *Experimental results for 50 Hz.*

Dutty cycle, %	**50 Hz**			
	Delay, s	***Time to move, s***	***Distance, mm***	***C, m/s***
20%	0.18	0.18-0.37	100	0.368
30%	0.14	0.14-0.29	100	0.467
40 %	0.10	0.13-0.24	100	0.636
50%	0.08	0.08-0.19	100	0.636
60%	0.07	0.07-0.17	100	0.700
70%	0.06	0.06-0.15	100	0.778
80%	0.05	0.05-0.13	100	0.875
90%	0.05	0.05-0.12	100	0.966

Табл.4.3. *Experimental results for 90 Hz.*

Dutty cycle, %	**90 Hz**			
	Delay, s	***Time to move, s***	***Distance, mm***	***C, m/s***
20%	0.24	0.24-0.45	100	0.333
30%	0.17	0.17-0.36	100	0.368
40 %	0.13	0.13-0.28	100	0.467
50%	0.12	0.12-0.26	100	0.500
60%	0.10	0.10-0.23	100	0.538
70%	0.09	0.09-0.21	100	0.538
80%	0.08	0.08-0.18	100	0.700
90%	0.07	0.07-0.17	100	0.700

CONCLUSIONS AND ANALYSIS OF THE RESULTS:

Based on the experimental studies conducted on the electro-pneumatic positioning system, the following conclusions can be drawn:

1. An experimental test bench has been developed to study the dynamics of an electro-pneumatic positioning system controlled by pulse-width modulation.

2. Experimental studies of the actual system were conducted under various operating modes, and the main parameters of the system—pressure, position, and motion dynamics of the actuator—were recorded.

3. The experimental results obtained allow for an analysis of the dynamic characteristics of the electro-pneumatic system when controlled by fast-acting 2/2 solenoid valves.

4. A comparison between the results of the mathematical modeling and the experimental data shows a high degree of agreement, which confirms the appropriateness of the developed mathematical model.

5. The developed mathematical model can be used for simulation studies, analysis of dynamic processes, and development of control algorithms for electro-pneumatic positioning systems.

CHAPTER V

SIMULATION AND VERIFICATION OF THE MATHEMATICAL MODEL OF AN ELECTRO-PNEUMATIC POSITIONING SYSTEM USING PWM CONTROL AND FOUR HIGH-SPEED 2/ 2 VALVES

Based on the developed mathematical model of the electro-pneumatic positioning system and the experimental studies conducted, a control system has been implemented that allows for the investigation of the dynamic characteristics and operational stability of the system when controlling high-speed 2/2 solenoid valves using pulse-width modulation.

A simulation model (Fig. 5.1) of a pneumatic system with four 2/2 valves, a double-acting pneumatic cylinder, a signal generator, a PWM generator, and a PID controller was designed and implemented in the MatLab Simulink software environment..

5.1. Selection, design, and presentation of the elements of the simulation model in Simulink / Mat Lab.

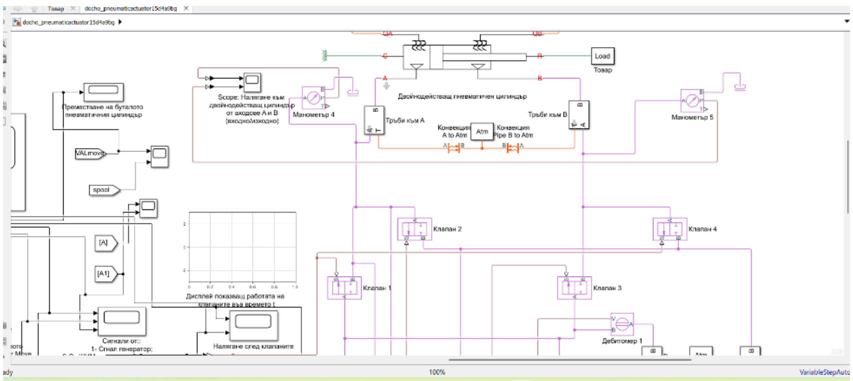
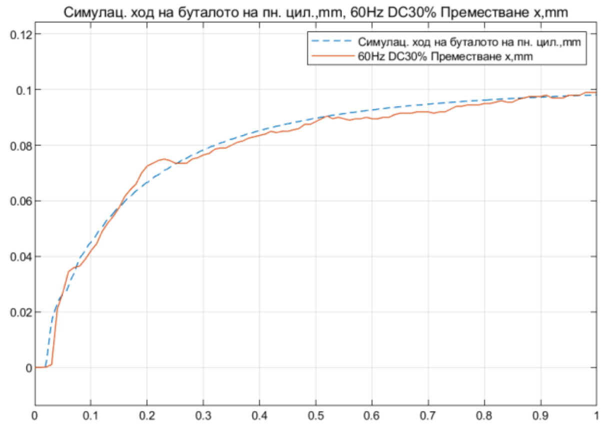
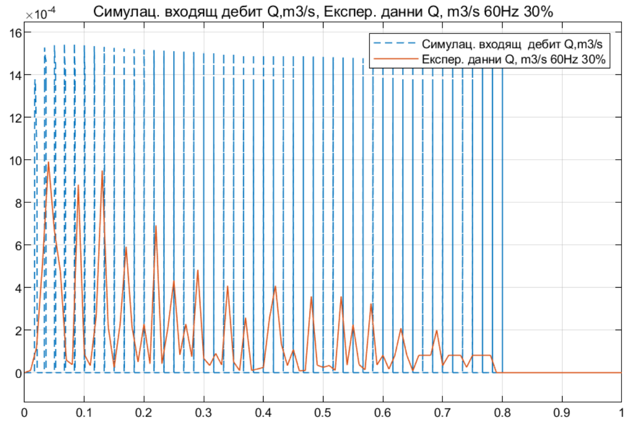


Fig. 5.1. Diagram of a simulation model in Simulink Matlab.

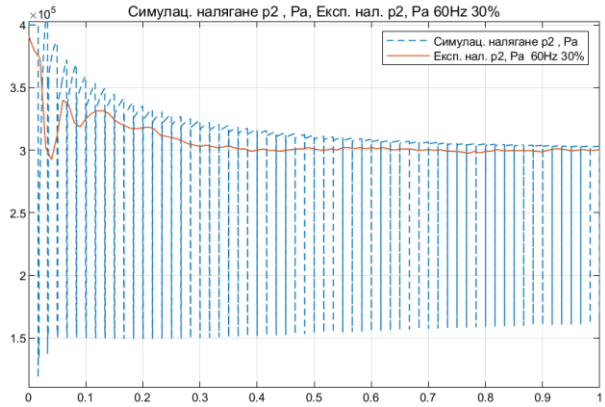
5.2. Data from simulations of a system with four pneumatic valves, a PID controller, and a PWM generator performed in the Mat Lab environment.



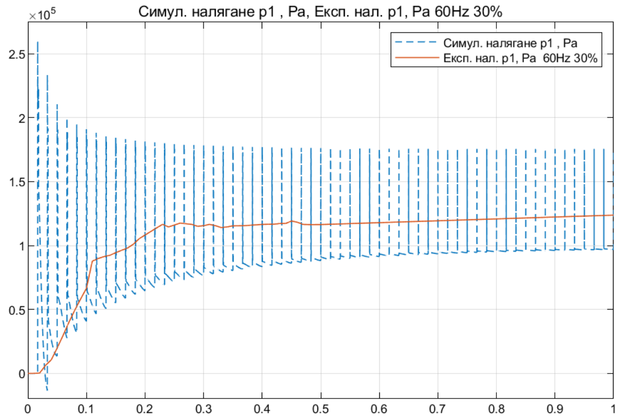
а) movement of the piston in the pneumatic cylinder (x,mm)



б) inflow $(Q, \mathrm{m}^3 / s)$



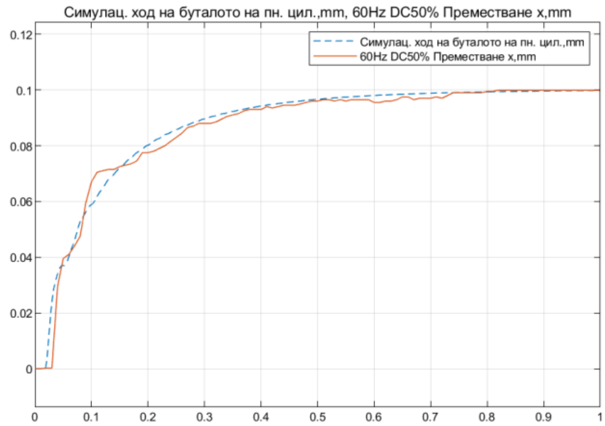
в) pressure p_2 *(Pa)*



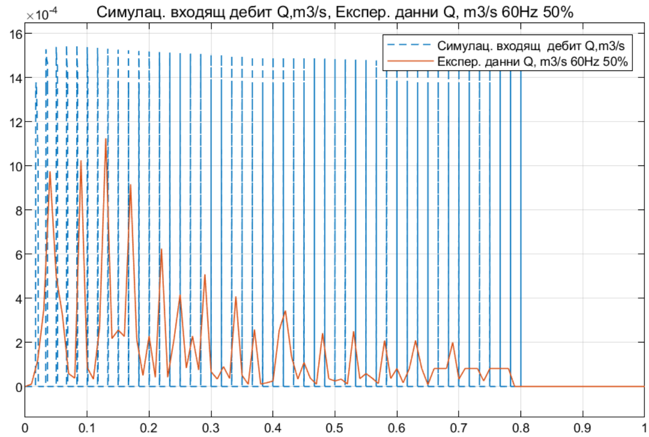
г) pressure p_1 *(Pa)*

Fig.5.2. *Experimental and simulation results of PWM control at 60 Hz and filling from 30% DC.*

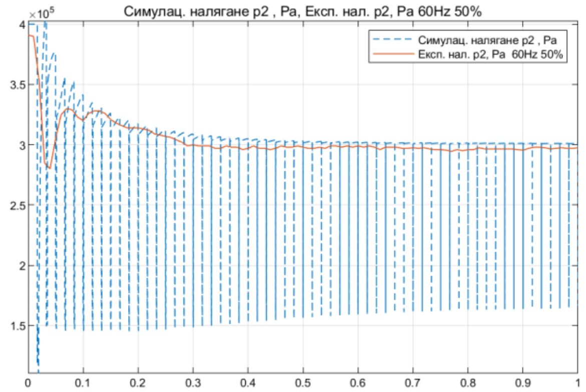
Simulations were conducted to compare experimental data with data from the simulation model of an electro-pneumatic positioning system at carrier frequencies of 30, 60, and 90 Hz; the remaining results can be found in the dissertation.



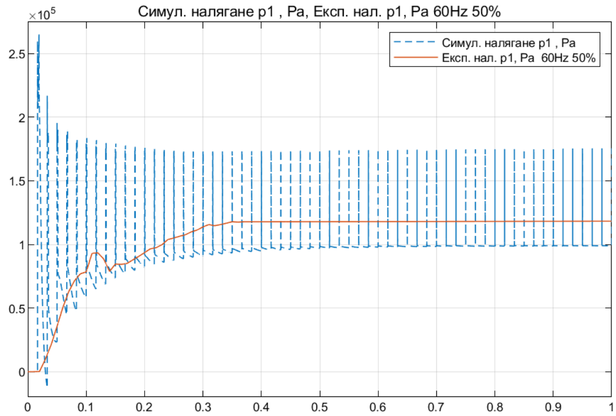
a) movement of the piston in the pneumatic cylinder (x,mm)



б) inflow $(Q, \mathrm{m}^3 / s)$

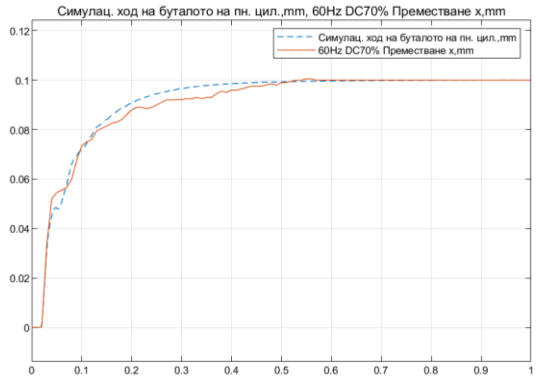


в) pressure p_2 *(Pa)*

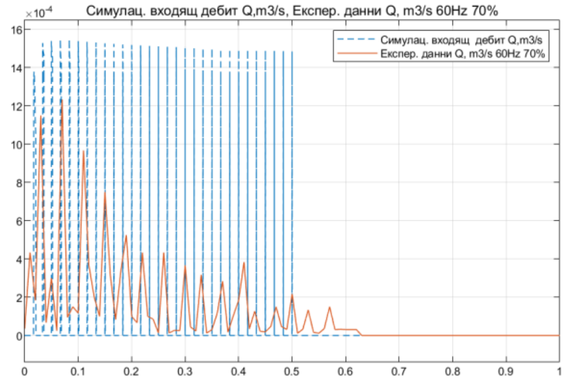


г) pressure p_1 *(Pa)*

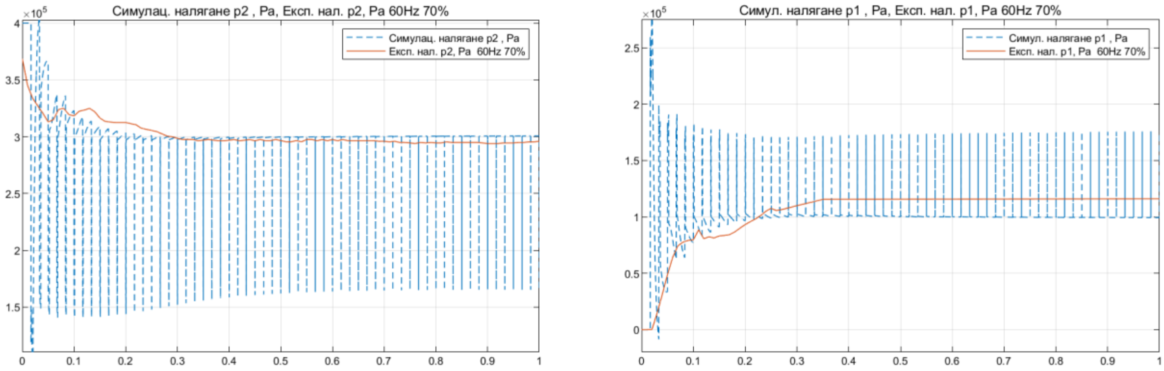
Fig.5.3. *Experimental and simulation results of PWM control at 60 Hz and filling from 50% DC.*



a) movement of the piston in the pneumatic cylinder (x,mm)

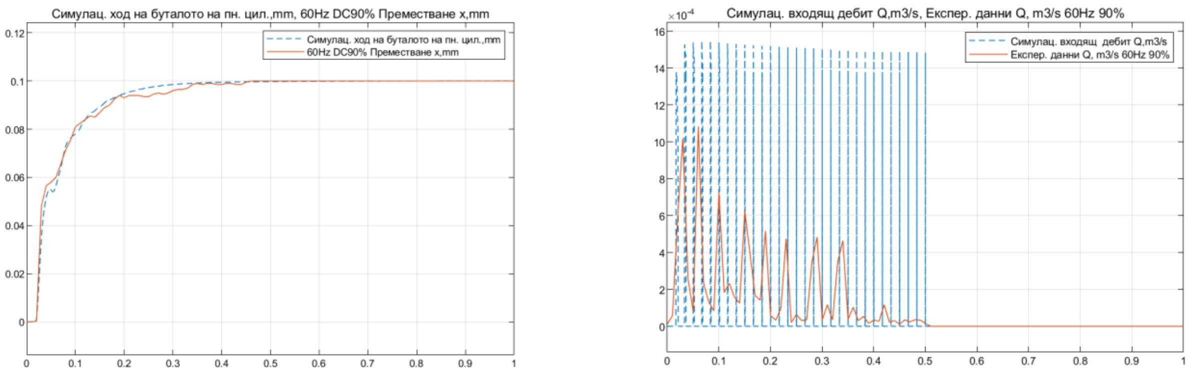


б) inflow $(Q, \mathrm{m}^3 / s)$

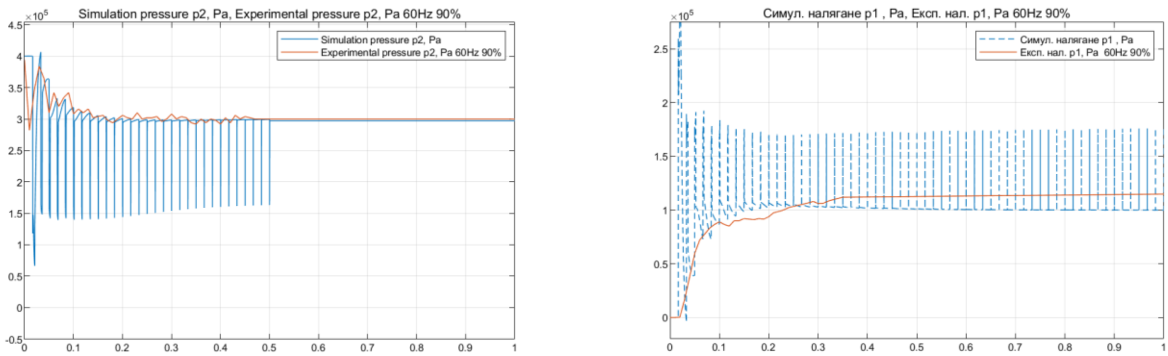


в) pressure p_2 *(Pa)* *г) pressure* p_1 *(Pa)*

Fig.5.4. *Experimental and simulation results of PWM control at 60 Hz and filling from 70% DC.*



a) movement of the piston in the pneumatic cylinder (x,mm) *б) inflow (*Q*,* m^3/s*)*



в) pressure p_2 *(Pa)* *г) pressure* p_1 *(Pa)*

Fig.5.5. *Experimental and simulation results of PWM control at 60 Hz and filling from 90% DC.*

Figures 5.2–5.5 present, in graphical form, the comparative results of the experiments and simulations regarding the displacement of the pneumatic cylinder, inlet flow rates, and corresponding operating pressures under PWM control with a frequency of 60 Hz and a duty cycle (Duty Cycle - DC) ranging from 30% to 90%. The experiments and simulations were conducted in a pneumatic system utilizing PID-controlled PWM with a specified stroke length of the pneumatic cylinder's piston 100 mm.

At a 90Hz frequency, the greatest reduction in ripple is observed in the graphs of the pneumatic cylinder piston displacement.

In the graphs obtained, at a control signal frequency of 90 Hz and a 70% duty cycle, less pronounced pulsations are observed in the displacement curve, both in the simulation curve—the dashed blue line—and in the experimental curve—the solid red line. The process of the pneumatic cylinder piston's stroke is completed slightly before 500 milliseconds, which is more clearly visible in the graphs presented with a process time cutoff of 600 milliseconds.

CONCLUSIONS AND ANALYSIS OF THE RESULTS:

Based on the development and study of the control system for the electro-pneumatic positioning system, the following conclusions can be drawn:

1. Based on the developed mathematical model, a control system for an electro-pneumatic positioning system has been implemented using pulse-width modulation of high-speed 2/2 solenoid valves.

2. A control algorithm for the system has been developed, which enables the implementation of position control of the pneumatic actuator.

3. The experimental studies conducted show that the control system ensures stable operation and improved dynamic characteristics of the electro-pneumatic positioning system.

4. The results confirm the possibility of effectively using pulse-width modulation for controlling electro-pneumatic positioning systems.

5. The developed control system shows the potential for application in positioning and automation systems using electro-pneumatic actuators.

MAIN RESULTS OF THE DISSERTATION

RESEARCH AND APPLICATION CONTRIBUTIONS

1. A mathematical model of the actual flow characteristic of high-speed 2/2-way electromagnetic pneumatic valves has been developed, and the model has been verified using experimentally determined static characteristics.

2. A mathematical model of the electromagnetic and mechanical dynamics of high-speed 2/2 pneumatic valves has been developed, enabling analysis of the valve opening and closing processes and determination of its dynamic characteristics.

3. An extended mathematical model of an electro-pneumatic positioning system controlled by pulse-width modulation has been developed, which integrates the models of the system's main components: the pneumatic power unit, pneumatic lines, pneumatic cylinder, and high-speed solenoid valves.

4. Simulation models have been created in the Matlab/Simulink environment to study the transient processes and dynamic characteristics of an electro-pneumatic positioning system with high-speed valves and PWM control.

PRACTICAL CONTRIBUTIONS

1. An automated measurement system with virtual instruments in the Lab View environment has been developed for the acquisition, processing, and visualization of experimental data in static and dynamic studies of electro-pneumatic systems.

2. An experimental test bench has been implemented to study the dynamics of an electro-pneumatic positioning system controlled by pulse-width modulation.

3. The influence of the control signal frequency and the duty cycle of the PWM signal on the dynamic characteristics of an electro-pneumatic positioning system has been experimentally investigated.

4. An energy-efficient electronic unit for PWM control of high-speed pneumatic valves has been developed, designed for use in electro-pneumatic control and positioning systems.

List of publications related to the dissertation.

1. H. Hristov, G. Iliev, D. Dimitrov, “Speed control of pneumatic power transmisson systems using ON-OFF valves with pulse width modulation”, 62nd Science Conference of Ruse University - SSS, Bulgaria, 2023, volume 62, book 1.1;
2. H. Hristov, G. Iliev, D. Dimitrov, „Experimental static flow characteristics of high speed ON/OFF pneumatic valves. “, "Mechanics of Machines" Days of Mechanics in Varna, September 8 - 10, 2023;
3. 3. D. Dimitrov, K. Varbanov, “Pneumatic System with PWM Control,” Student Research Conference 2024, Technical University of Gabrovo;
4. 4. D. Dimitrov, “Intelligent Control of an Energy-Efficient Pneumatic System,” Student Scientific Session 2024, Technical University of Gabrovo;
5. Iliev, G.; Hristov, H., D. Dimitrov” Experimental Study of the Frequency Characteristics of an Electropneumatic Tracking System with High-Speed Pneumatic Valves and PWM Control”, Environment. Technology. Resources. Rezekne, Latvia,Proceedings of the 16th International Scientific and Practical Conference, 2025, Volume IV, 111-115. DOI: 10.17770/etr2025vol4.8427;