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Research on a Single-Chamber Pneumatic Vibroactuator

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1. Introduction

Single-chamber pneumatic vibroactuators are devices that are used in a variety of applications, including vibration testing, material handling, and precision manufacturing. These devices are designed to produce a controlled amount of vibration, which can be used to achieve specific outcomes in different industries. The research of single-chamber pneumatic vibroactuators is important in understanding their performance, behaviour, and potential for further development.

In recent years, the study of pneumatic actuators has gained increasing attention due to their advantages over other types of actuators. Pneumatic actuators offer several benefits such as low cost, high speed, and clean operation. The dynamic behaviour of a pneumatic actuator is a complex process, which depends on several parameters such as the supply pressure, the mass of the working body, the geometric parameters of the chamber, and the mechanical parameters of the spring.

Many researchers have studied the dynamic behaviour of pneumatic actuators to optimize their performance. For example, researchers in [1] have investigated the effect of supply pressure on the dynamic behaviour of pneumatic actuators. The results showed that increasing the supply pressure resulted in an increase in the amplitude and frequency of the vibrations. Additionally, other researchers have studied the effect of the mass of the working body on the dynamic behaviour of pneumatic actuators [2]. The results showed that increasing the mass of the working body resulted in a decrease in the frequency of the vibrations.

Different authors have analysed the dynamics of various pneumatic actuators in their studies [3-10]. To investigate the dynamics of such a system theoretically, researchers use not only various methods of solving mathematical differential equations, but also computational FEM modelling, simulation with various software and even neuro computing tools such as self-organising map (SOM) visualisation techniques. There are numerous methods available for analysing and solving vibroactuator parameters, which have been implemented in various industries. However, the authors of this paper tend to concentrate on analysing the self-exciting auto-vibration zone in the study of pneumatic vibroexciters and actuators [9]. In all cases, the modelling is complex and the individual mathematical models are limited. Hence, the authors are seeking new approaches to solve the dynamic analysis of pneumatic exciters by employing modern methods.

The theoretical research presented in this paper introduces a hybrid mathematical model that is based on earlier studies conducted by the authors of this manuscript, as well as research on single-action pneumatic actuators as described in [6]. The mathematical model was simulated using the MATLAB Simulink tool. The results obtained from the simulation were compared to the results of an experimental study conducted by the authors.

This study aims to determine the suitability of a pneumatic vibroactuator simulation model to analyse the motion of the vibroexciter working organ in the autovibration mode.

2. Mathematical model

A diagram of a pneumatic single chamber vibroactuator with actuating element is presented in Fig. 1.

The operational principles of such vibroexciters are described in [9].

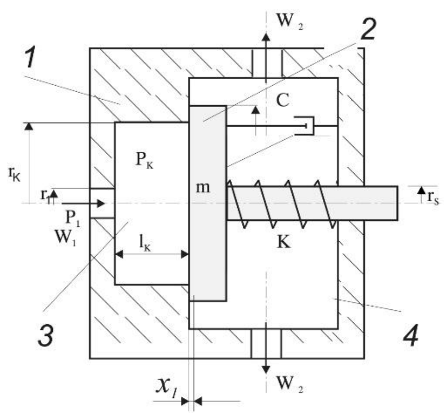

Fig. 1 General view of single chamber pneumatic vibroactuator: *1* – housing; *2* – vibrating mass; *3* – chambers of vibroexciter

Vibroexciter is supplied with compressed air (gas) P_1. Other parameters that influence the movement of actuating element *2* are: m – mass; C – resistance factor – damping coefficient; K – stiffness factor – elasticity coefficient; x – shift of actuating element in chamber. Pressure P_K is generated in chamber *3*.

In their earlier studies, the authors utilized differential equations of the second order that comprised of the equilibrium of the dynamic equation and formulas from De Saint Venant and Vantzel or Prandtl for theoretical research purposes. The mathematical model that is presented in this work includes the equilibrium of the dynamic Eq. (1), which describes the movement of the actuating element, as well as equations that describe the model of air pressure in the actuator's chamber by [6].

$$m\frac{d^2x}{dt^2}+C\frac{dx}{dt}+K\left(x+x_0\right)=p_K\cdot S_K\,, \tag{1}$$

where: p is pressure of pressed air in chamber (overpressure); C is coefficient of resistance; K is coefficient of rigidity; x is amplitude of vibrations of chambers working body. The air mass density of compressible air ρ will be found by the ideal gas equation:

$$\rho=\frac{M}{R\cdot T}\cdot\left(p_K+p_a\right), \tag{2}$$

where: M is the air mol density; R is the gas constant; T is the absolute air temperature, p_K is overpressure; p_a is atmospheric ambient pressure in temperature T.

Mass of the air Q in the chamber equals:

$$Q=\rho\cdot\left(S_K\cdot l_K+S_K'\cdot x\right), \tag{3}$$

where: S_K is the cross-sectional area of the chamber with actuating element. Taking into account that p_a=*const* differentiation of Eq. (3) is as follow:

$$\frac{dQ}{dt}=\frac{M}{R\cdot T}\cdot\left[\left(S_K\cdot l_K+S_K'\cdot x\right)\cdot\frac{dp_K}{dt}+\left(p_K+p_a\right)\cdot\frac{dx}{dt}\right]. \tag{4}$$

From [2], the mass flow of compressible air could be expressed as Eq. (5):

$$\frac{dQ}{dt}=C\cdot a\cdot\sqrt{2\frac{M}{R\cdot T}}\cdot\sqrt{P_1\left(P_1-P_2\right)}\,, \tag{5}$$

where: C is the dimensionless orifice flow coefficient; a is the cross-sectional area of the orifice hole in an pressure supply equipment; P_1 is the absolute upstream air pressure and $P_1=p_s+p_a$; P_2 is absolute downstream pressure and $P_2=p_K+p_a$. According to Eqs. (2)-(5) we achieve Eq. (6).

A system of differential equations, consisting of Eq. (1) and Eq. (6), is utilized to determine the motion of the actuator vibrating mass. The solution was implemented using the MATLAB Simulink tool by creating a simulation model.

$$\frac{dp_K}{dt}=\frac{C\cdot a\cdot\sqrt{2\frac{M}{R\cdot T}}\cdot\sqrt{P_1\left(P_1-P_2\right)}-P_2\cdot S_K'\cdot\frac{dx}{dt}}{S_K\cdot l_K+S_K'\cdot x}\cdot\frac{R\cdot T}{M}. \tag{6}$$

3. Simulation

The simulation of the pneumatic actuator was carried out in the MATLAB Simulink environment by creating two interconnected models. The principles of modelling and design technique is described in [11, 12]. The interconnected structure, as presented in Fig. 2, consists of a model that describes the dynamics of the pneumatic actuator and a model that simulates air pressure fluctuations in the chamber. The air pressure model is based on Eq. (6), and the dynamics of the actuator is modelled by Eq. (1).

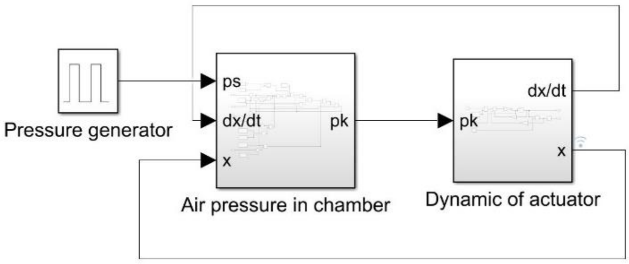

Fig. 2 Connected structure of models for the simulation of pneumatic actuator

The dynamic model of the pneumatic actuator is displayed in Fig. 3, and the air pressure simulation model is presented in Fig. 4.

The value of pressure (p_s) supplied to the chamber and value of mass (m) of actuating element were varied during the simulation research. The geometric parameters (r_k, l_k, r_1) and mechanical parameters (C, K) of the vibroexciter are based on the pneumatic actuator used in the experiment conducted by the authors. The parameters utilized in the simulation of the dynamic behavior of the pneumatic actuator are presented in Table 1.

Also, the initial data x_0 = 0 m, v_0 = 0 m/s were set for the start of simulation.

The parameters used in the simulation of the air pressure fluctuations in the chamber of the pneumatic actuator are presented in Table 2.

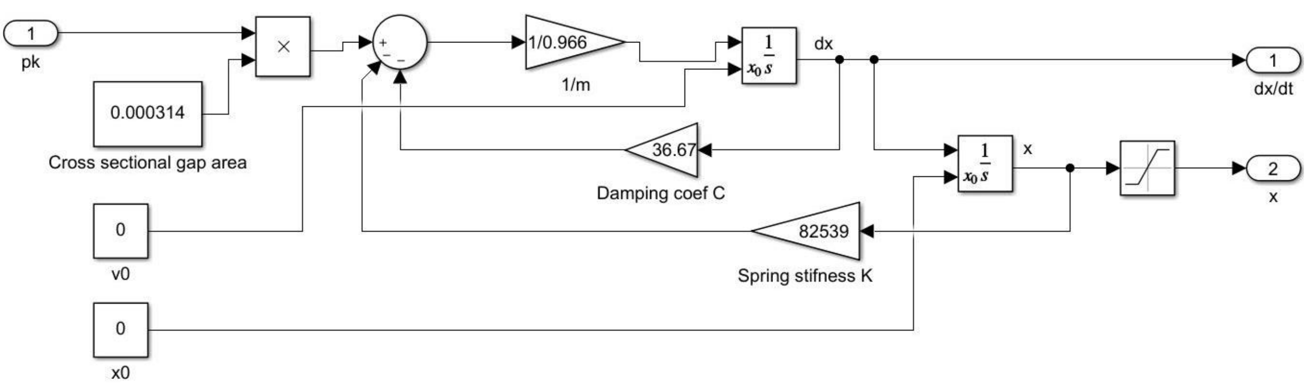

Fig. 3 Model of the spring influence to the dynamic of the pneumatic actuator

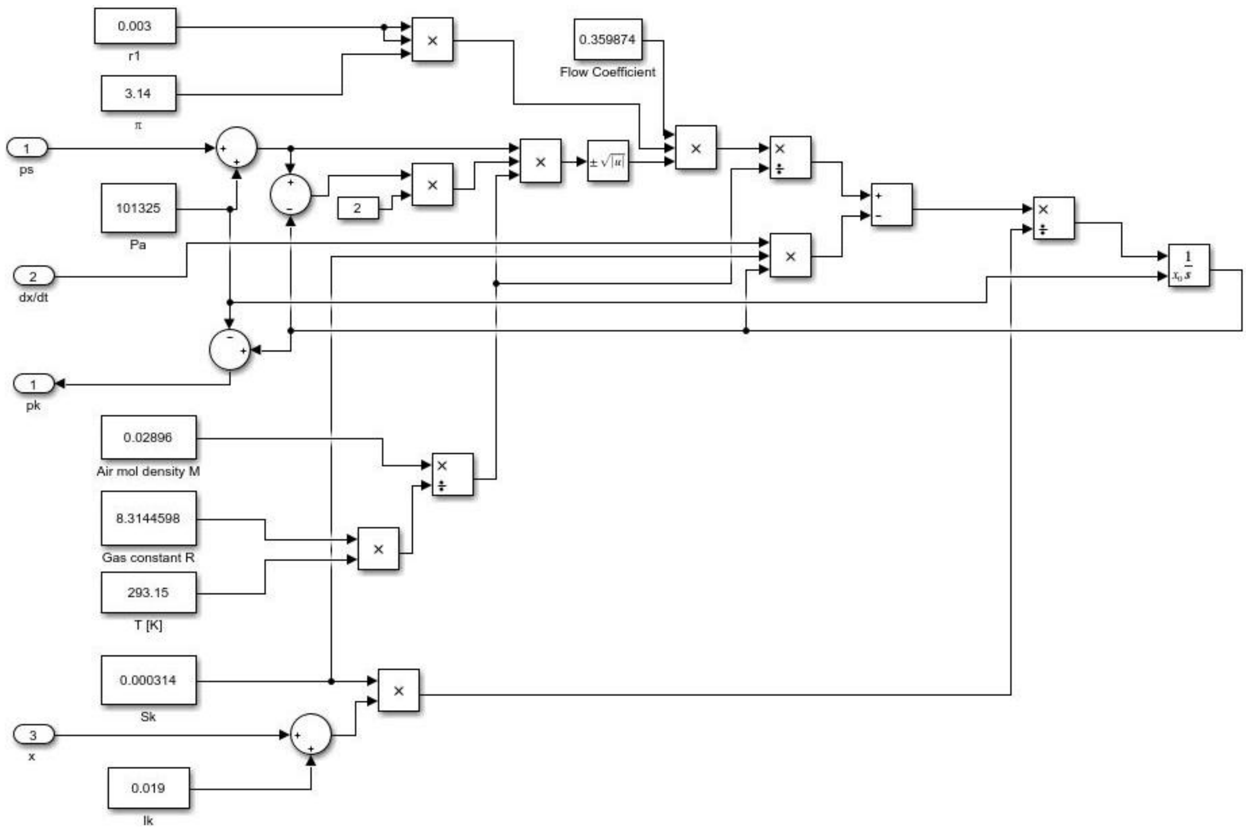

Fig. 4 Model of the air pressure fluctuations in the chamber of pneumatic actuator

Table 1

Initial data for simulating of dynamic of pneumatic actuator

Mass, m_1	0.542	kg
Mass, m_2	0.966	kg
Coefficient of resistance, C	36.67	kg/s
Spring stifness coefficient, K	82539	N/m
Radius of chamber, r_K	10×10^{-3}	m

The simulation control was carried out by varying the vibrating mass, the supply pressure, and the geometrical parameters of the chamber of the vibroexciter. The diagrams from the Simulation Data Inspector windows with different values of mass, pressure, and height of the chamber are presented in Figs. 5-7. The simulation results demonstrate that the simulated system correctly responds to changes in these parameters.

Table 2

Initial data for simulating of the air pressure in the chamber

Diameter of orifice, $2\times r_1$	3×10^{-3}	m
Flow coefficient	0.359874	Kv
Air density, M	0.02896	kg/mol
Gas constant, R	8.3144598	J/(K·mol)
Absolute air temperature, T	293.15	K
Atmospheric ambient pressure, p_a	101325	Pa
Radius of chamber, r_K	10×10^{-3}	m
Hight of chamber, l_K	19×10^{-3}	m

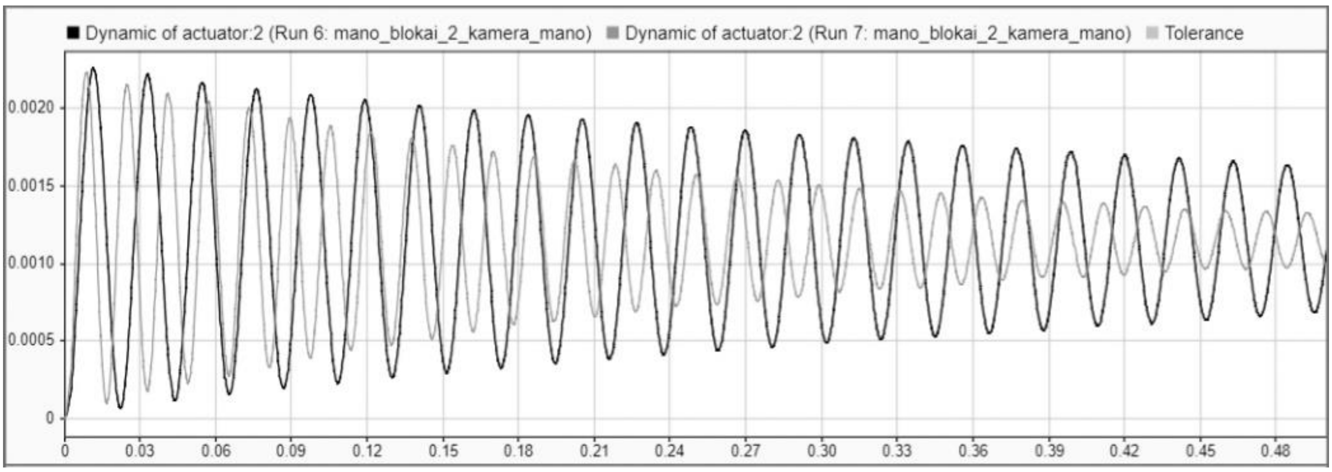

Fig. 5 Comparative diagram of the auto vibrations of the working body of the vibroexciter, when P_1 = 2.0×10^5 Pa; l_k = = 0.019 m (black line – m_2 = 0.966 kg; grey line – m_1 = 0.542 kg)

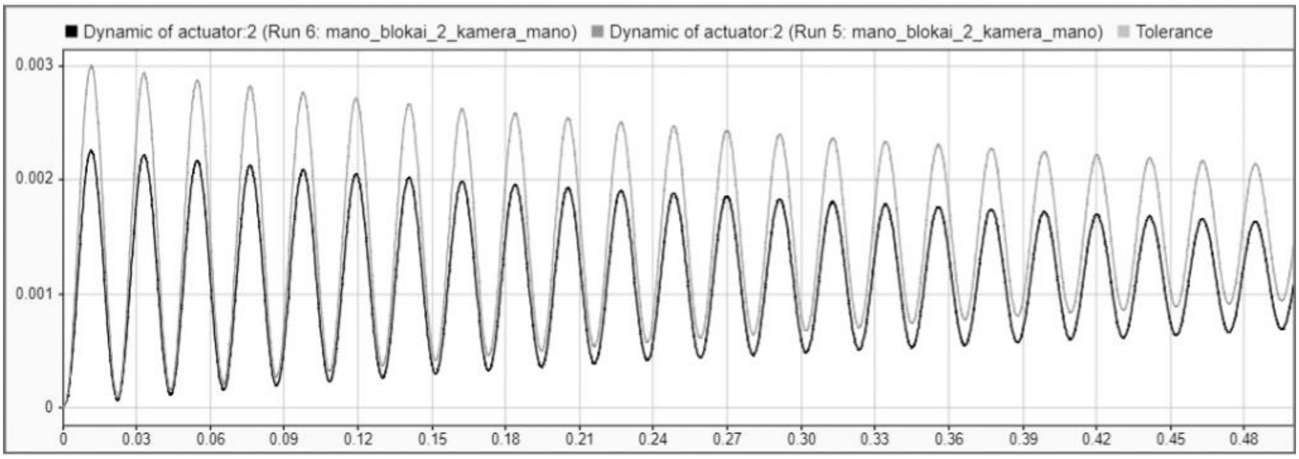

Fig. 6 Comparative diagram of the auto vibrations of the working body of the vibroexciter, when m_2 = 0.966 kg; l_k = 0.019 m (black line – P_1 = 2.0×10^5 Pa; grey line – P_1 = 3.0×10^5 Pa)

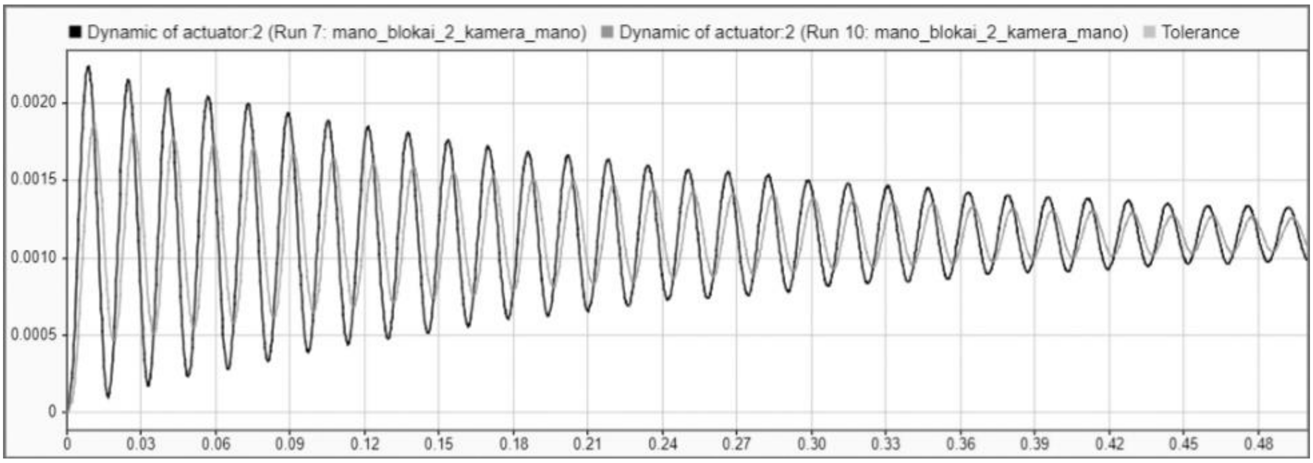

Fig. 7 Comparative diagram of the auto vibrations of the working body of the vibroexciter, when m_2 = 0.966 kg; P_1 = = 2.0×10^5 Pa (black line – l_k = 0.019 m; gray line – l_k = 0.045 m)

4. Comparison of study results

An experimental stand was installed for the research of a one-chamber actuator. The general view of the measuring equipment is presented in Fig. 8. The output data of this system (amplitude and frequency) depend on the value of the vibrating mass, the geometrical parameters of the chamber, mechanical parameters of the spring, and the input of compressed air pressure. The parameters of the vibrations of the pneumatic vibroexciter were measured and recorded using measuring equipment. The data captured by the inertia sensor accelerometer KD35a 70292 is transmitted to the signal via the vibrometer PicoScope 3424.

During the experiments after choosing geometrical parameters r_k, l_k; the values of the mass m of the working body of the vibroexciter, and the characteristics of supply pressure were changed.

At a pressure P_1=2.0×10^5 Pa, the dynamic system has entered to an auto vibration mode with the frequency f = = 144 Hz and amplitude $A \approx$ 0.25×10^{-3} m.

A comparison is made between the results of the theoretical simulation study and the experiment. The results are shown in Fig. 9.

At a pressure P_1=3.0×10^5 Pa, and a mass of the working body m_2=0.966 kg, the vibroexciter entered into an auto-vibration mode with a frequency f of approximately 73 Hz and amplitude $A \approx$ 0.75×10^{-3} m. The results of the theoretical simulation study and the experiment are presented in Fig. 10.

The research has shown that the results obtained by the simulation method are close to the maximum amplitudes of the vibrations obtained in the experiment. However, the oscillation character of the working mass in the auto-vibration mode is more complex. It should be noted that the simulation does not fully account for efficiency losses during energy conversion. The effects of friction and the fact that the pressure in the fixed chamber is equal to the supply pressure were not considered in the mathematical study.

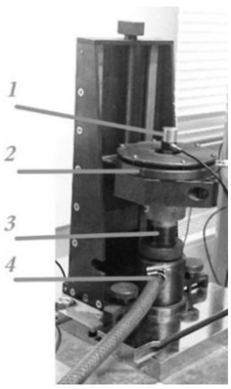

Fig. 8 General view of single chamber pneumatic vibroexciter construction: *1* - accelerometer KD35a 70292; *2* - flat springs; *3* - vibroexciter and chamber; *4* - compressed air system

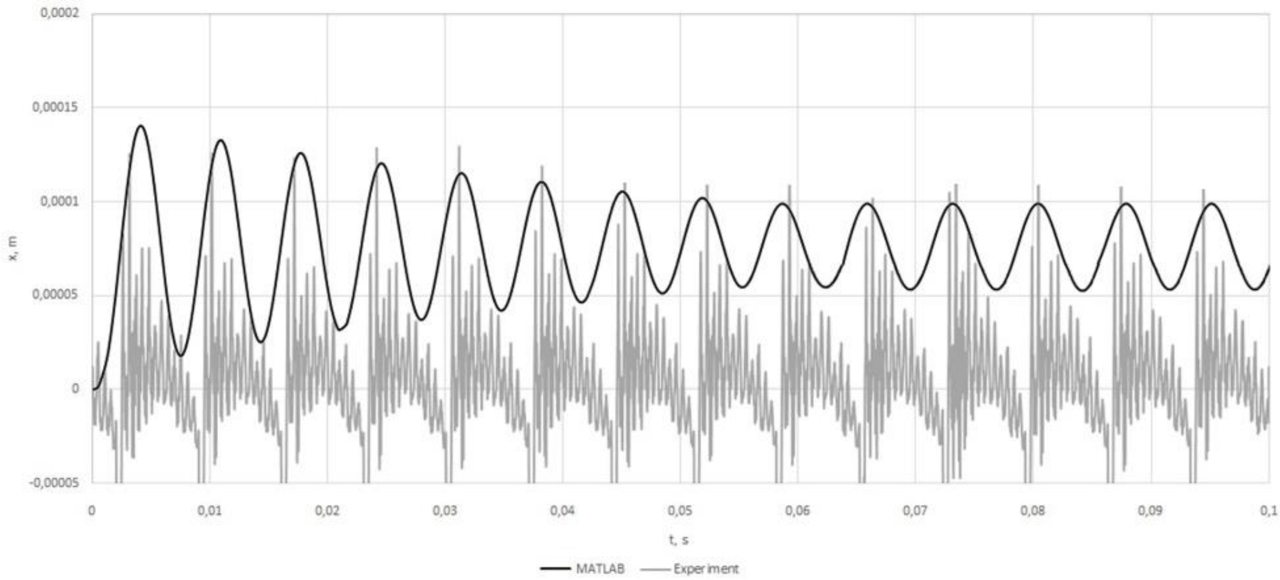

Fig. 9 Diagrams of the auto vibrations of working body of vibroexciter in the range of 0 to 0.1 s diagram, when $P_1 = 2.0\times10^5$ Pa; $m_1 = 0.542$ kg (black line – simulation results, grey line – experiment results)

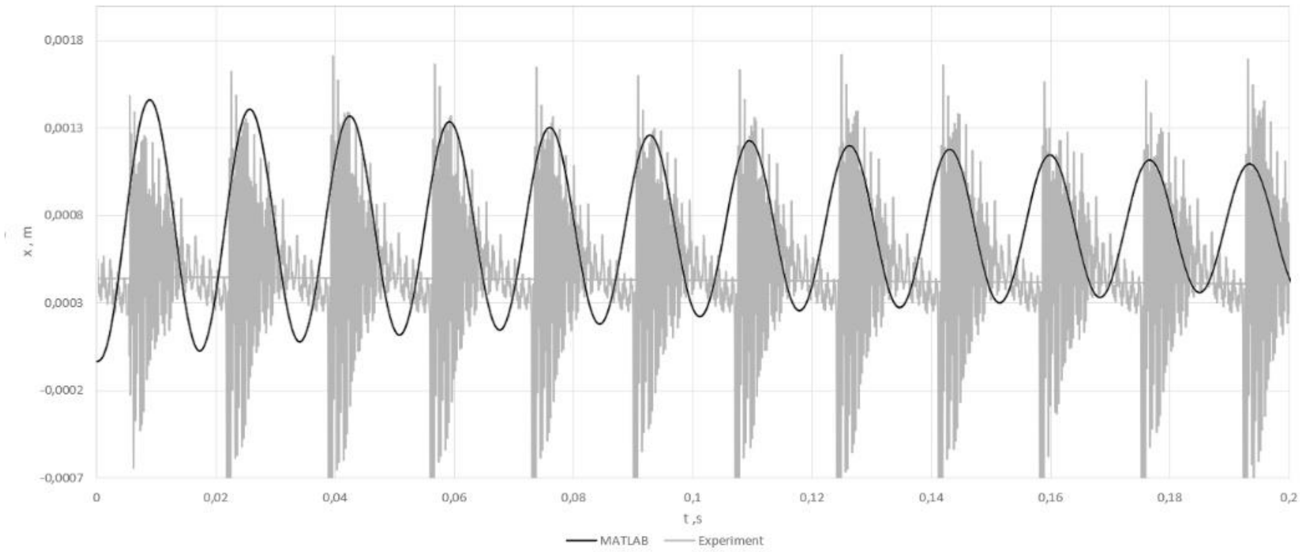

Fig. 10 Diagrams of the auto vibrations of working body of vibroexciter in the range of 0 to 0.2 s diagram, when $P_1 = 3.0\times10^5$ Pa; $m_2 = 0.966$ kg (black line – simulation results, grey line – experiment results)

5. Conclusions

A single-chamber pneumatic vibroexciter was studied, and a comparison was made between the results obtained from the theoretical simulation research and the experiment.

The experiment analysed the dynamic system of a single-chamber vibroactuator for a set of parameters. The supply pressure was varied from 1.5×10^5 Pa to 3×10^5 Pa and the vibrating mass was set to 0.542 kg or 0.966 kg.

The results of both experimental and theoretical studies on certain oscillation characteristics, such as amplitude and frequency, demonstrate a partial qualitative overlap. The results suggest that the authors can use the further improved simulation model for their future research.

The improved mathematical model must produce results with oscillations of a more complex law of motion. The auto vibrations with a complex movement law could be applied in technological processes where it is necessary to increase the intensity of the process in a shorter period of time.

The authors plan to apply the methodology presented in this paper to the study of a dual-chamber actuator in the future. Another research direction they intend to pursue is the modelling of a pneumatic actuator operating in auto-vibration mode using the MATLAB Simscape tool.

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RESEARCH ON A SINGLE-CHAMBER PNEUMATIC VIBROACTUATOR

S u m m a r y

The linear type single-chamber pneumatic vibroactuator is widely used in industrial automation due to its fast response, low energy consumption, and high-power density. Many researchers have conducted studies on the dynamic behaviour of pneumatic actuators to optimize their performance.

A pneumatic vibroexciter with one chamber was investigated in this study. A simulation model was developed in the MATLAB environment to conduct theoretical studies. The simulation results were compared with experimental measurements previously made by the authors of this paper.

The findings of this study can be utilized to optimize the design and control of pneumatic actuators in industrial automation.

Keywords: pneumatic actuator, vibroexciter, vibrations.

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