

The Study of the Feasibility of Generating a Combined Sweep Signal

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Abstract. This research is a continuation of a series of studies aimed at developing the shock vibration source of seismic signals with superior parameters compared to existing signal sources. The seismic signal source currently under development will feature compact dimensions and a high capability for extracting useful information against a background of noise. This is achieved through amplitude modulation of the sweep signal and by pressing the base plate down while it is being excited by a force pulse. In this article a schematic diagram, as well as mathematical and physical models of a shock vibration source have been developed. It was conducted a research of the parameters of the amplitude-modulated sweep signal of the shock vibration source of seismic signals and dynamic characteristics of the hydro-volume vibration mechanism. In the article: the results of computer simulations based on a mathematical model are presented; the feasibility of generating an amplitude-modulated sweep signal with a modulation law approximating a harmonic function is proved; the information regarding the experimental design is given.

Keywords: source of seismic signals, sweep signal, amplitude modulation, mathematical model, computer simulations, experimental studies.

Introduction

It's always a current interest to study the vibration parameters during the implementation of various technological processes [1–5].

One of the areas within this field is vibration seismic exploration sweep signal excitation.

One area of focus within this field is seismic exploration involving the generation of sweep signals – vibrational loads produced according to a specific law [6–11].

It was considered in the research works [12–14] a design scheme for a hydro-volume source of seismic oscillations. In the design of a hydrostatic seismic vibrator, where an actuating vibration mechanism based on elastic shells is combined with an impact-based system for pressing the base plate against the ground surface.

A combined signal containing two frequency components will be transmitted to Earth:

- a low-frequency component – the carrier force load;
- a high-frequency component – the informative sweep signal.

In the research [14] it is noted that the oscillatory load generated by a seismic vibration source has a complex, combined form.

The assumption that the force pulse has a nearly sinusoidal shape made it possible to:

- replace the impact assembly with a low-frequency resonant circuit excited by a second generator;
- put forward a hypothesis about the nature of the oscillatory load generated by the source and its physical model (the vibration device) is identical [14].

Thus, the article reviews the operation of a vibration mechanism featuring two oscillation generators:

- a high-frequency generator with oscillation amplitude X_1 ;
- a low-frequency generator with oscillation amplitude X_2 .

The research [14] reviewed the dynamic system of a hydrostatic vibration mechanism, focusing on the influence of the disturbing action of each generator on the displacement amplitude of the moving element.

It was determined in the research the operating resonant frequency of the low-frequency generator and the operating range of the high-frequency generator.

Two generators are involved in the signal generation process, therefore, the low-frequency power-carrying component will be distorted by high frequencies. The first task solved in the research is the theoretical and experimental determination – across various loading conditions – of the high-frequency sweep signal's sweep limit frequencies at which this effect is negligible.

The second task is to determine the excitation parameters at which the useful high-frequency component of the oscillatory load (signal) is maximized relative to the low-frequency force component. Due to the fact that the synchronizing the low- and high-frequency components under laboratory conditions is difficult, the system's operation was analyzed with the generators running at constant frequencies.

The second research part presents the results of computer simulation on the mathematical model of the vibration mechanism (2) [14] with the introduction of nonlinear elements considered in [15]. It is considered the possibility of obtaining a combined oscillatory load with a smooth sinusoidal amplitude envelope [16].

The third part describes the information-measuring system designed for conducting experimental studies.

The results of experimental studies are presented below, and the following issues are considered:

- the influence of the high-frequency generator frequency on the lower resonant frequency to determine the limiting frequencies at which this influence is minimal;
- determining the operating modes at which the ratio of the high-frequency component F_{hf} to the low-frequency component F_{lf} (to the force impulse F_{im}) is the highest (Fig. 1) [13];
-

$$\left(\frac{F_{hf}}{F_{lf}} \right) \rightarrow \max \text{ or } \left(\frac{F_{hf}}{F_{im}} \right) \rightarrow \max.$$

It is shown in the research [15] that the generators load the oscillatory system almost identically at the input values of $X_2 = 0.603$ mm and $X_1 = 0.701$ mm. Thus, after rounding to $X_2 \approx 0.6$ mm and $X_1 \approx 0.7$ mm, nominal loading modes were adopted (Table 1).

Table 1. Adopted loading modes

No	Low-frequency generator	High-frequency generator
	Piston stroke, mm	Pistons strokes, mm
1	1.2	1.4
2	1.2	2.8
3	1.2	4.2

The conclusion provides a comparative analysis of computational and experimental data.

1 Theoretical Research

Theoretical studies were conducted using the mathematical model (2) [14] in order to determine the feasibility of obtaining a combined vibration signal.

When modelling the following uniform conditions were adopted:

- an average pressure in the high-pressure hoses: $p_0 = 0.5$ MPa;
- an excitation frequency of the low-frequency generator: $f_2 = 4.2$ Hz (close to resonant, as its exact value is not determined (3);
- the loading modes №3 – $X_2 = 1.2$ mm and $X_1 = 4.2$ mm

The results were obtained from the mathematical model study (Fig. 1 and 2).

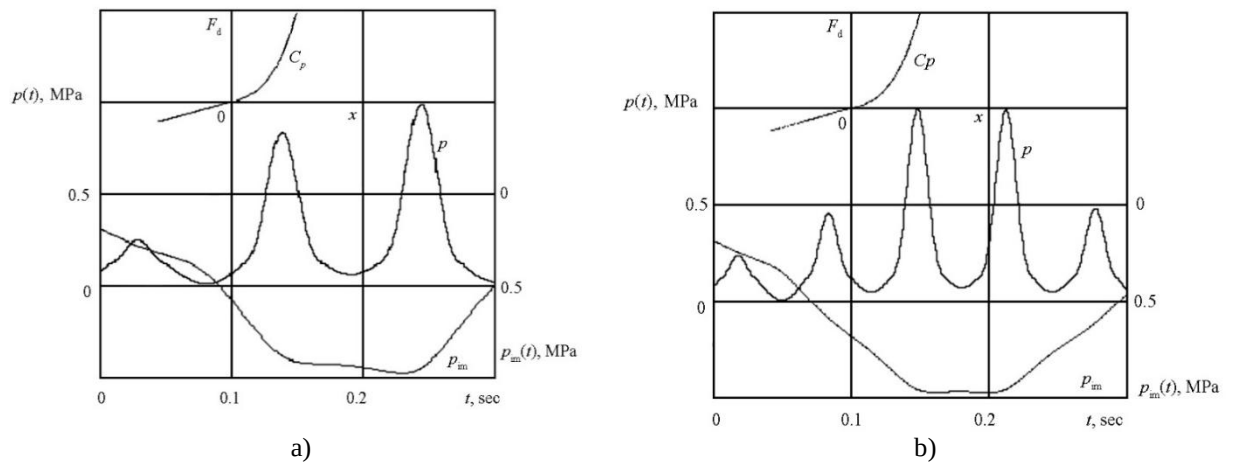


Fig. 1. – Computer simulation results at various frequencies: a – $f_1 = 10$ Hz; b – $f_1 = 15$ Hz

F_d are the force transmitted through high pressure hoses (HPH) to the «ground», the driving force;

C_p is the hydraulic stiffness due to volumetric deformation; p_{im} – impulse of force;

p – pressure in HPH when two generators are running (p_2); t – impulse duration

An analysis of the diagrams shows that the influence of the lower (initial) frequencies of the high-frequency generator (f_{1L}) on the impulse decreases as generator operating frequency increases (Fig. 1, 2):

- at initial frequencies of $f_{1L} = 28$ Hz – insignificantly,
- at $f_{1L} = 36$ Hz – there is no significant influence.

The high frequency limit of the sweep signal for existing generators is in the range of 130÷150 Hz.

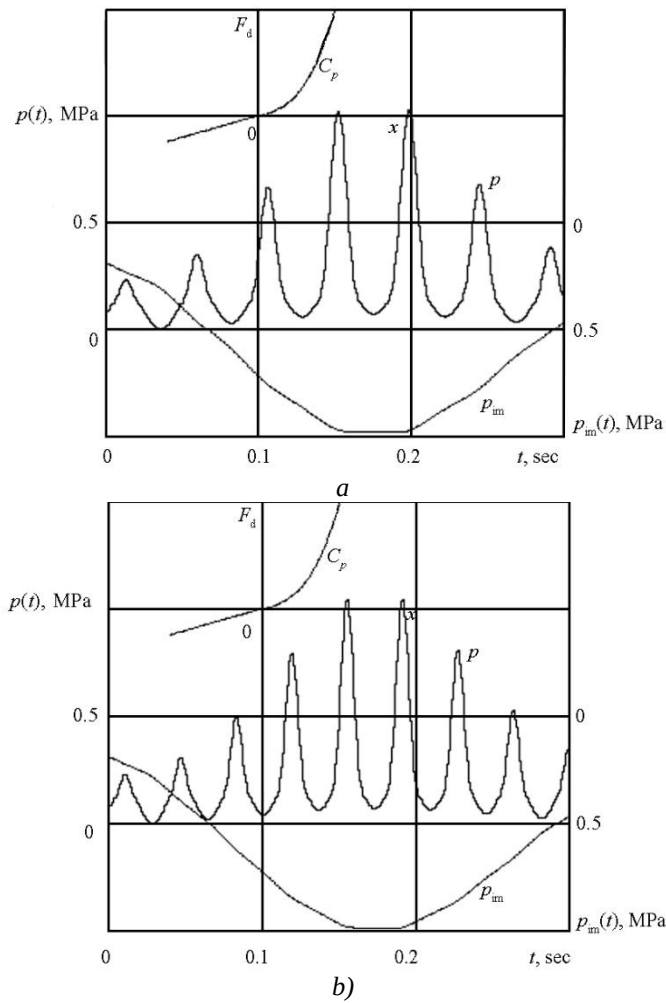


Fig. 2. – Computer modeling results at frequencies: a – $f_1=25$ Hz; b – $f_1=34$ Hz

Given that the determined lower frequency limit under No 3 loading (tabl. 1) [15] is approximately 30–36 Hz (Fig. 1), then for a signal duration of 0.3 sec (Fig. 1), we obtain a signal base of

$$TBWP = \Delta f_0 \cdot T = [(130...150) - (30...36)] \cdot 0.3 = 28.2...36 \text{ Hz} \cdot \text{sec}.$$

This is discussed in more detail in article [16].

Thus, it is recommended to start the sweep signal at a frequency of 30 Hz.

2 Information and measuring system

Let us consider the schematic diagram of the information-measuring system (Fig. 3), which consists of the actuator [14] and the measuring system.

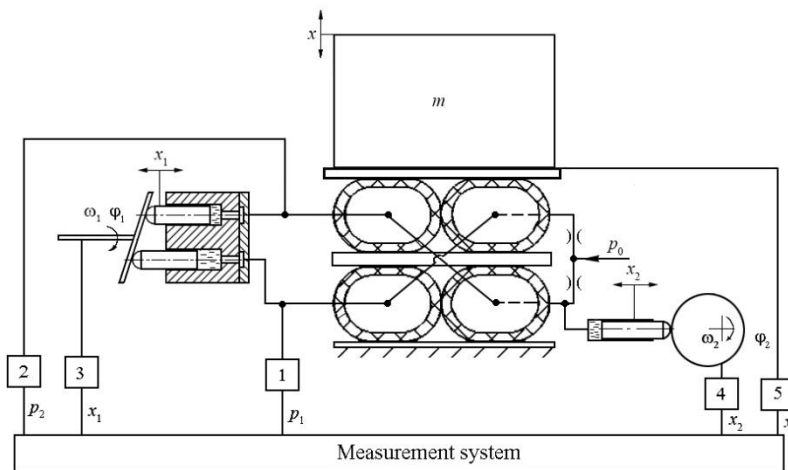
The measurement system is equipped with five sensors (1–5):

- 1, 2 – pressure sensors in the HPH;
- 3, 4 – displacement sensors for the plungers of the oscillation generators;
- 5 – load displacement sensor.

Sensors transmit information about the actuator parameters to the measurement system. The design of sensors 1–5 is based on using:

- photodiodes as the sensing element;
- LEDs as the luminous flux source.

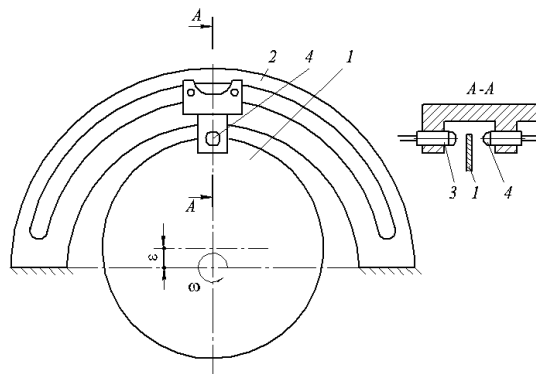
The measurement principle was based on changing the potential difference across the photodiode by altering the luminous flux incident on it.



m – load mass; p_0 – average pressure in the high-pressure hose; p_1 and p_2 – pressure in the high-pressure hose generated by the oscillators; x_1 , ω_1 , φ_1 и x_2 , ω_2 , φ_2 – parameters of the high-frequency and low-frequency vibration generators (displacement coordinates, angular frequencies, angles of rotation)

Fig. 3. – Information and measurement system

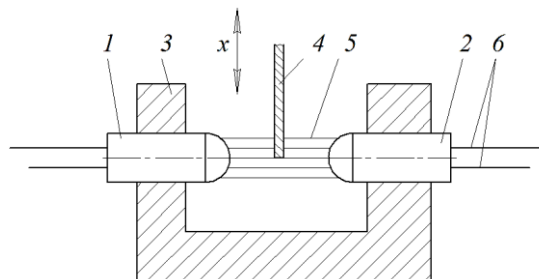
The output signals of generators x_1 and x_2 were measured from the corresponding eccentric 1 rotating on the generators shaft. The phase of signals x and dp was determined by moving the elements of sensor 3 along the curved guide 2 (Fig. 3).



1 – eccentric; 2 – curved guide; 3, 4 – sensor LED and photodiode

Fig. 4. – Sensor installation diagram

The pressures p_1 и p_2 were recorded by blocking the light flux with a shutter attached to the Bourdon tube. Pressure changes in the cavities caused deformation of the tube, resulting in varying degrees of light flux obstruction (Fig. 5).



1 – LED; 2 – housing; 3 – light flux; 4 – shutter; 5 – photodiode

Fig. 5. – Sensor operating diagram

The measurement system is shown in Figure 6. For simplicity and cost-effectiveness, it is built using analog instruments:

- C1-91 and C1-93 oscilloscopes;
- Г6-26 signal generator.

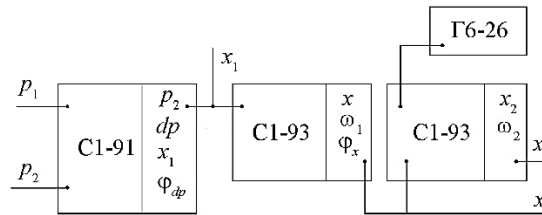


Fig. 6. – The measurement system

The data were digitized after the measurements.

3 Experimental Research

The experimental research methodology in [13] involved separate excitation of the vibromechanism's dynamic system by the generators:

- first, by the high-frequency oscillator;
- then, by the low-frequency oscillator.

As a result, experimental logarithmic amplitude-frequency characteristics (LAFC) were obtained for the following quantities:

- output signal x ;
- pressure drop dp ;
- phase shift angles $\varphi_{x1/x2}$ and phase shift angle $\varphi_{dp1/dp2}$.

Signal amplitude values were read from the measurement scale on the screens of the C1-91 and C1-93 oscilloscopes, while the phase shift angles were determined by the position of the Lissajous figure on the C1-93 oscilloscope screen. The vibration frequency was determined by comparing the mechanism frequencies with the reference frequency of the Г6-27 signal generator.

An analysis of the LAFC [13] indicates that the dynamic system of the vibromechanism is virtually linear at the average pressure in the high-pressure hose of $p_0=2\dots5$ MPa used during the experiments. This is due to the fact that, outside the second resonance zone – which should not occur in an actual signal source, the LAFC exhibits an asymptotic slope of -40 dB/dec, typical for linear systems.

It is known [17–19] that if a linear system is subjected to two disturbing forces, the total effect can be determined by superimposing the effects caused by each force individually. Thus, the disturbing force caused by the operation of both generators will induce complex vibrations

$$x(t) = X_1 \sin(\omega_1 t + \varphi_{01}) + X_2 \sin(\omega_2 t + \varphi_{02}).$$

If the frequency values ω_1 and ω_2 (f_1 and f_2) are close to each other, the resultant motion takes on a character known as amplitude modulation (Fig. 7).

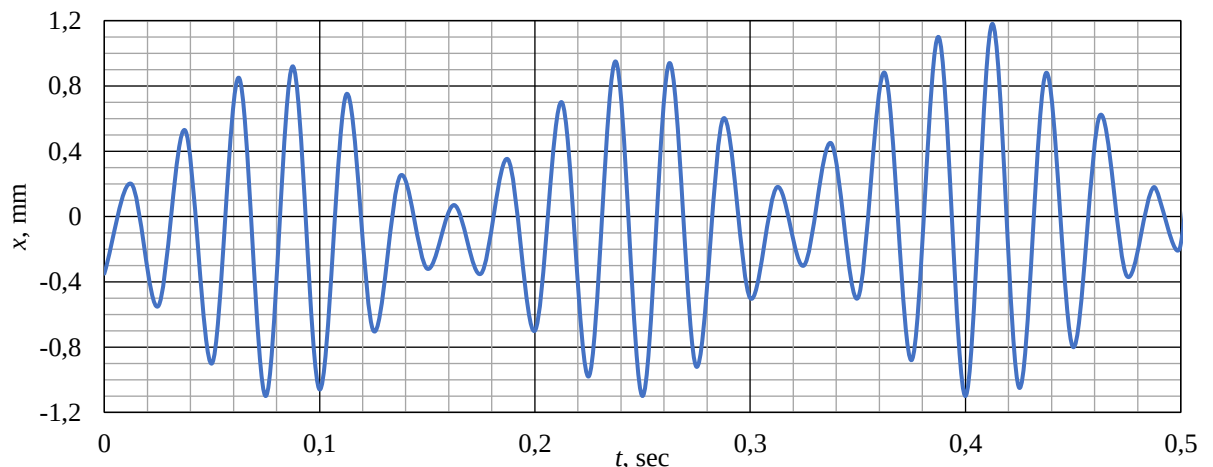


Fig. 7. – Oscillogram of displacement amplitude beats at $p_0=2$ MPa, $f_1=40$ Hz and $f_2=6,5$ Hz

Experimental studies, as well as theoretical ones, demonstrated that as the lower cutoff frequencies of the high-frequency generator increase, the degree of influence of one generator on the other decreases (Fig. 8). There is a threshold frequency at which this influence is virtually unobservable (Fig. 8, b).

Experimental studies conducted under various loading conditions of the vibration mechanism confirmed that an increase in the limiting frequency as the degree of loading increased.

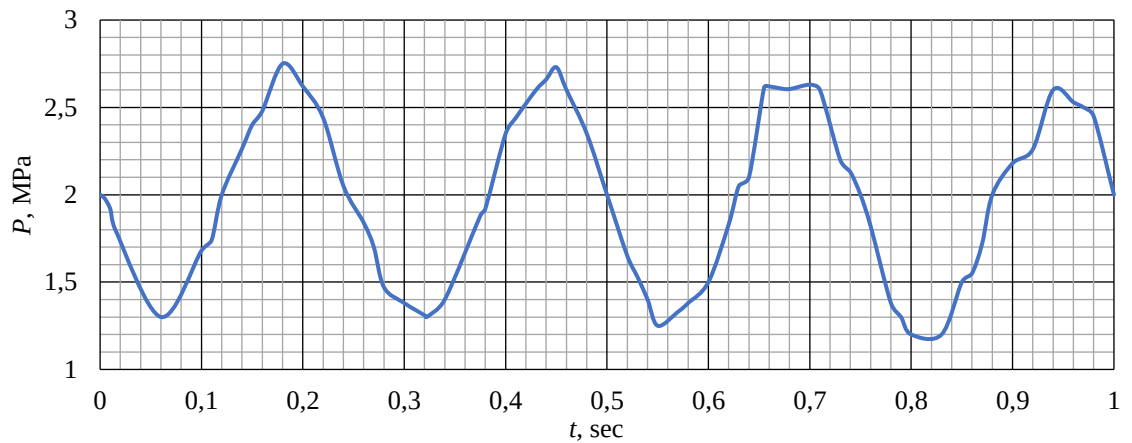
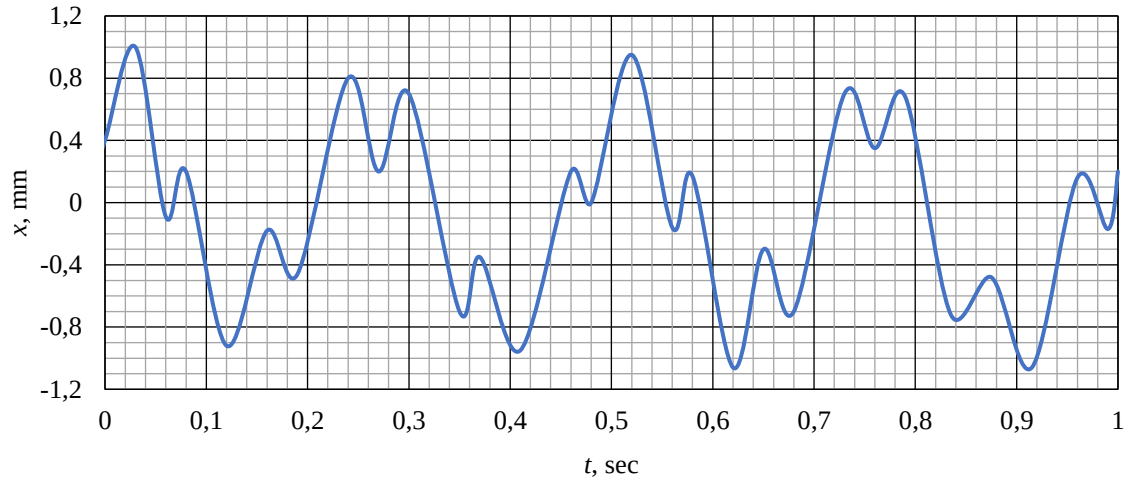


Fig. 8. – Oscillogram of displacement amplitude at $p_0=2$ MPa at frequencies: a – $f_1=12$ Hz, $f_2=4.2$ Hz, b – $f_1=22$ Hz, $f_2=4.2$ Hz

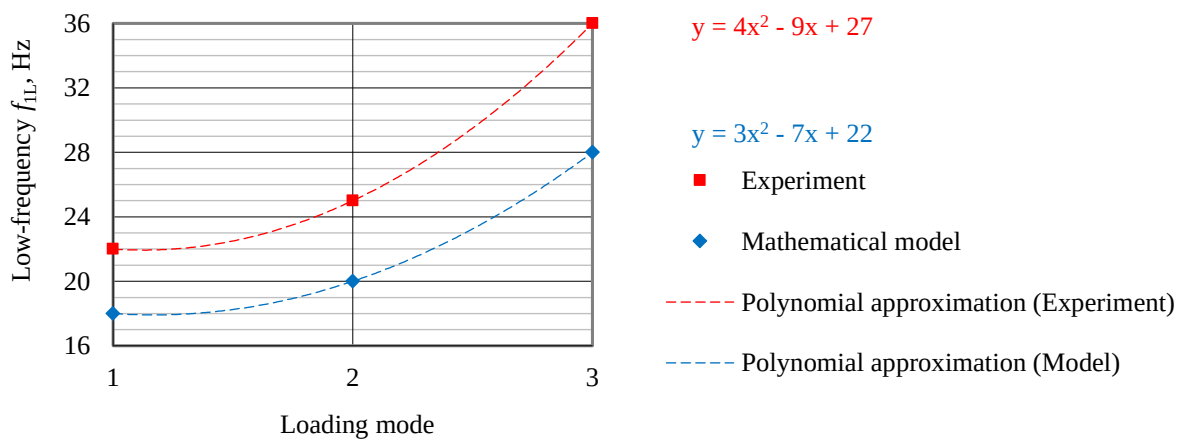


Fig. 9. – Dependence of the limiting frequency on the loading mode: a comparison of the theoretical and experimental research results

The difference between the experimental and theoretical data is due to the presence of an additional resonance [13]. Therefore, for the laboratory prototype of the vibration mechanism, the limiting frequency values were increased during experimental studies.

During the experimental studies, the pressure p (p_2) generated in a single line of the shells by both generators was recorded (Figs. 10–12) under various average pressures p_0 and loading conditions.

The conducted experiments confirmed an increase in the stiffness of the elastic shells with an increase in the average pressure. At the following values:

- at $p_0=2$ MPa, the system is practically linear (Fig. 10, b);
- at $p_0=0.5$ MPa, the system exhibits significant nonlinearity (Fig. 10, a).

Accordingly, the results of studies at an average pressure of $p_0=0.5$ MPa are presented below, which allows for an increase in the fraction of the high-frequency (informative) component in the sweep signal.

The figures illustrate the change in the ratio of the high-frequency component to the low-frequency component $\frac{F_{hf}}{F_{lf}}$ (Figs. 10–12). The best results were obtained with loading mode No. 3 and an average pressure of $p_0=0.5$ MPa (Fig. 12)

$$\frac{F_{hf}}{F_{lf}} \approx 1.$$

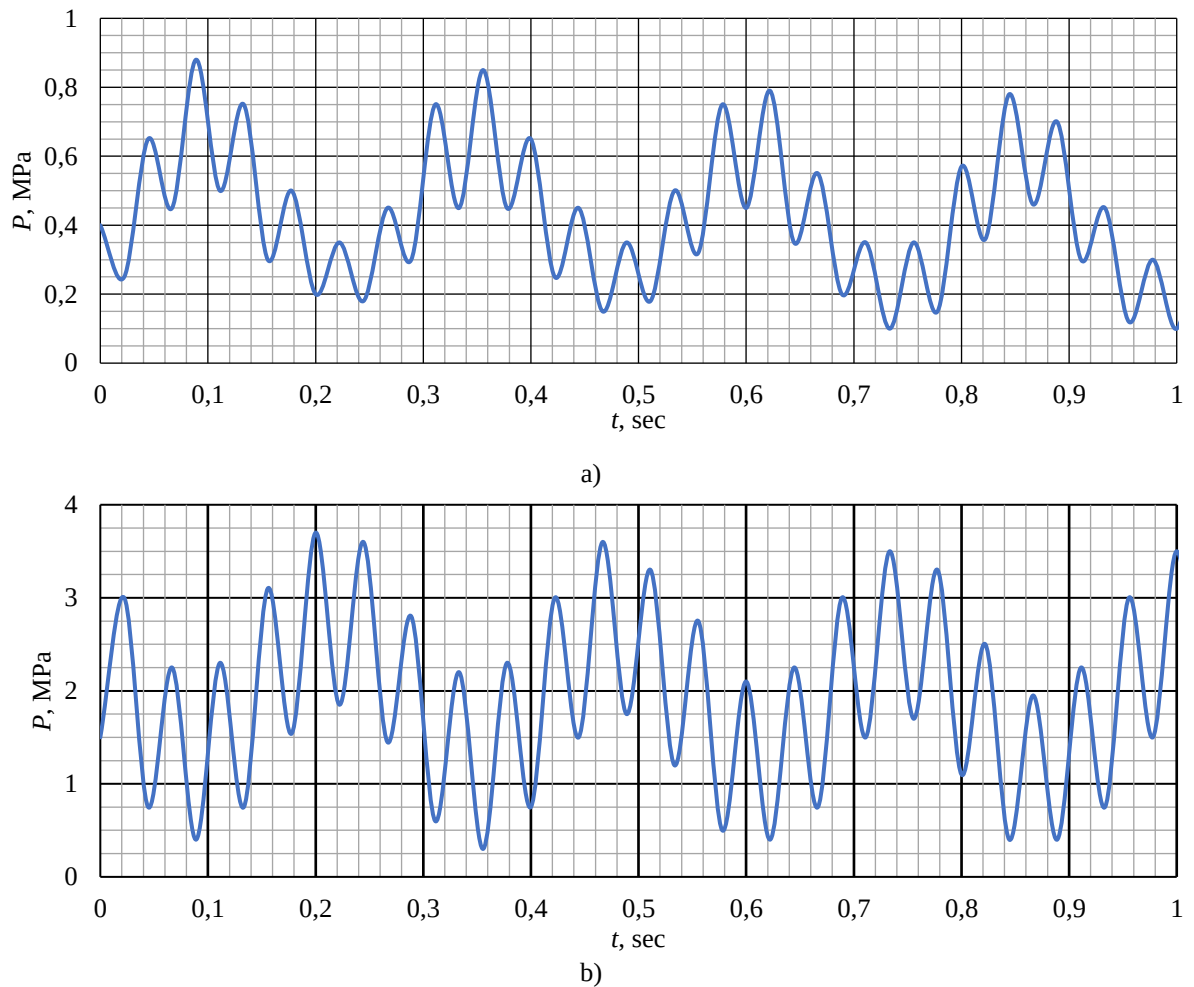


Fig. 10. – Oscillograms of pressure p_2 under loading mode No. 1 at $f_1=24$ Hz; $n f_2=4.2$ Hz: a – $p_0 = 0.5$ MPa; b – $p_0 = 2$ MPa

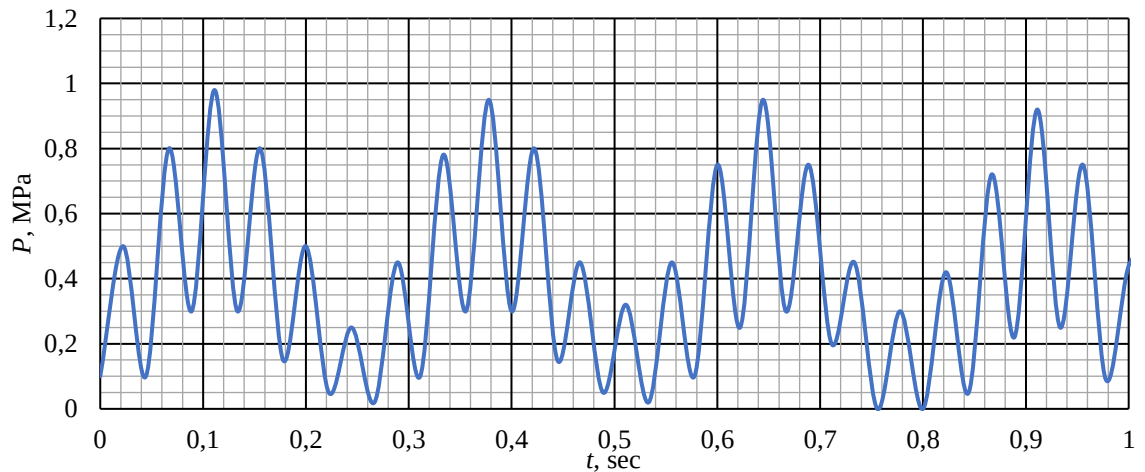


Fig. 11. – Oscillogram of pressure p_2 under loading mode No. 2 and $p_0 = 0.5$ MPa at $f_1=24$ Hz; ω $f_2=4$ Hz

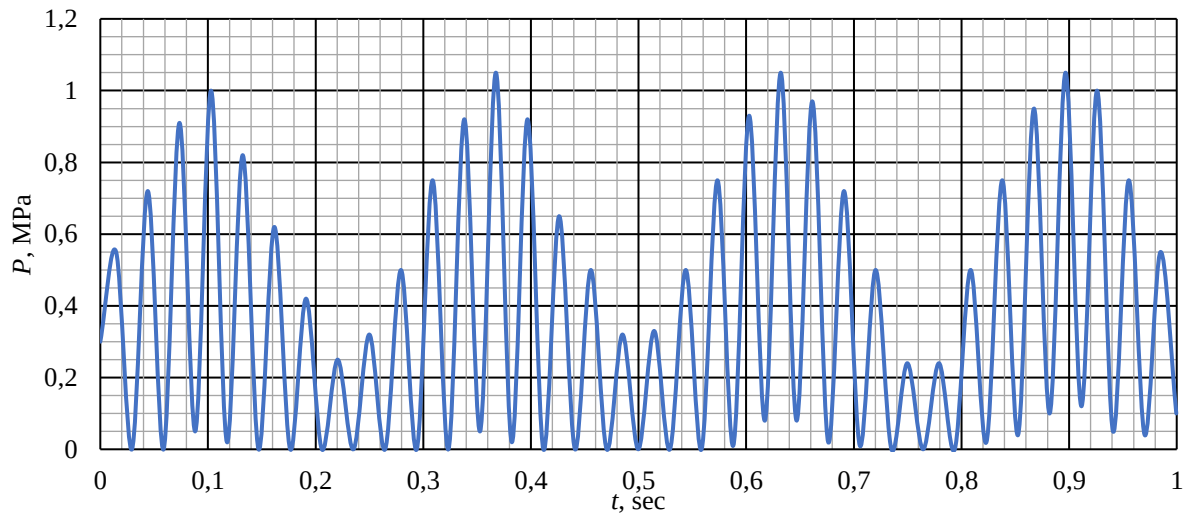


Fig. 12. – Oscillogram of pressure p_2 under loading mode No. 3 and $p_0 = 0.5$ MPa at $f_1=34$ Hz; ω $f_2=4$ Hz

Comparing the waveform and amplitude values of the computer-simulated combined signal with those excited by the vibro-mechanism, both qualitative and quantitative agreement of the obtained results can be noted.

Conclusion

This research is a continuation of a series of studies [13–16] addressing to the possibility of exciting an amplitude-modulated sweep signal with a smooth envelope of oscillation peaks.

In the introduction, the authors explain the relevance of the research, highlight the results of previous studies, outline the objectives of the current work, and provide a brief overview of the article's sections.

The research methodology involved a combination of computer modeling and physical experimentation based on the developed information-measuring system.

The chosen approach made it possible to construct and analyze the logarithmic amplitude-frequency characteristics of the system [15] and to plot pressure variations under various loading conditions. At the same time:

- approximate limiting frequencies for the loading modes have been determined (Table 1);
- a decrease in limiting frequency values was recorded as the loading degree of the vibromechanism decreased (Fig. 9).

Experimental studies confirm the conclusion made in [13]: as the mean pressure increases, the system tends toward linear behavior.

By increasing the loading level, it is possible to obtain the maximum possible ratio of the high-frequency sweep component to the dynamic clamping force.

A comparison between the simulated and experimentally formed combined signals shows both quantitative and qualitative agreement (Fig. 1, b and Fig. 12).

As a result of the research, it can be concluded that the objectives set in this work have been achieved and that it is physically possible to implement an amplitude-modulated sweep signal with a smooth envelope of oscillation peaks.

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