Multifunctional vibrational source of seismic waves

Yu A Burian¹, V N Sorokin¹ and V V Lizunov²

¹Omsk State Technical University, 11, Mira ave., Omsk, 644050, Russia
²Omsk State University, 55A, Mira ave., Omsk, 644077, Russia

Abstract. The paper deals with a powerful low-frequency surface source of seismic waves with an electrohydraulic drive and a force closure in the “inertial mass-soil system”. A mathematical model of a hydromechanical vibromodule is developed, and according to the computer simulation, an estimate of the seismic wave generation efficiency is given as well as the operating modes are determined for signals reproduction over a wide range of frequencies and shapes.

1. Introduction
The creation of a powerful, low-frequency and transportable source of seismic waves, designed to solve a wide range of problems, is a relevant problem. Among the priority problems are: effective ground sounding, increasing seismic exploration efficiency (non-explosive seismic exploration), active environment monitoring and determining the strain-stress state of media (earthquake prediction), vibroseis effects on oil-gas formations (oil formation productive capacity increase), etc. [1, 2, 3, 4, 5].

A powerful low-frequency surface source of seismic waves with an electrohydraulic drive and a force closure in the inertial mass-soil system capable of radiating signals in a wide range of frequencies and shapes is considered. A mathematical model of a hydromechanical vibromodule is developed. Based on the computer simulation results, the seismic wave generation efficiency is estimated and the operating modes are determined for playing a "sweep" signal and a complex signal.

In [6] the source efficiency is shown whose construction diagram corresponds to the force closure principle in the "inertial mass-soil" system when operating in the of a monochromatic signal generation mode. It is of interest to consider the possibility of obtaining an impulse seismogram analogue during monochromatic signal emission, as well as the possibility of reproducing a linear-frequency modulated signal ("sweep" signal) by such a source, which will significantly expand its area of application.

2. Problem statement
In case of a force closure in the inertial mass-soil system, the inertial mass, as shown in [6], should be hung on spring elements with high load capacity and low stiffness. As such spring elements, rubber-cord shells (RCS) can be used.

The analytical model of such a source is shown in figure 1.

To generate the multimass system motion differential equations, $Z_1$, $Z_2$, $Z_3$ are taken as generalized coordinates, and it is assumed that the line of $F(t)$ force application by a hydraulic cylinder passes through the centre of mass while spring and dissipation forces application lines are at the same distance $\ell$ from the mass centre point C.

It is also assumed that at $F(t) = 0$, the inertial mass $m_1$ is hung on the RCS, and the masses $m_2$ and $m_3$ are also in equilibrium positions, while the generalized coordinates values are measured from the equilibrium positions.
Figure 1. Analytical model: \( m_1 \) - inertial mass; \( m_2 \) - the mass of the supporting plates together with the apparent soil mass; \( m_3 \) – the mass of the base plate, the hydraulic cylinder and the apparent soil mass; \( c_1, b_1 \) - stiffness and damping coefficients for the inertial mass suspension spring elements; \( c_2, c_3, b_2, b_3 \) - stiffness and damping coefficients characterizing the soil properties under the supporting and base plates; \( F(t) \) - the force created by the hydraulic cylinder; \( Z_1, Z_2, Z_3 \) – the displacement of the corresponding inertial mass, the supporting plates and the base plate.

Differential equations based on the above-given assumptions and following [6] can be represented in the form of

\[
\begin{align*}
\dot{Z}_1 + \frac{Z_1}{m_1} f_1(Z) + \frac{2b_1}{m_1} \dot{Z}_1 \frac{2b_1}{m_1} \dot{Z}_2 &= -\frac{F(t)}{m_1}; \\
\ddot{Z}_2 + \frac{c_2}{m_2} Z_2 - \frac{1}{m_2} f_1(Z) - \frac{b_1}{m_2} \dot{Z}_1 + \frac{b_1 + b_2}{m_2} \dot{Z}_2 &= 0; \\
\ddot{Z}_3 + \frac{c_3}{m_3} Z_3 + \frac{b_3}{m_3} \dot{Z}_3 &= \frac{F(t)}{m_3}; \\
F(t) &= S \cdot P(t),
\end{align*}
\]

(1)

where \( f(Z) \) is the RCS nonlinear load characteristic; \( Z = Z_1 - Z_2 \); \( S \) is the hydraulic cylinder piston area; \( P(t) \) is the pressure in the hydraulic cylinder cavities; \( b_1 \) is the viscous friction coefficient in the RCS; \( c_2, c_3 \) are soil stiffness coefficients; \( m_2, m_3 \) is the apparent soil mass together with the plates mass; \( b_2, b_3 \) are soil damping coefficients.

To ensure the base plate displacements within the soil elastic deformations, under the force of \( 10^6 \) N, it is necessary to take the value of \( S_{pl} \) of the base plate as equal to 12 m² [6], while its mass is \( 5 \cdot 10^3 \) kg.

For low frequencies, the apparent soil mass \( m_{ap} \) can be determined from the expression [7,8]

\[
m_{ap} = \frac{8(1-\gamma^2)(2-\gamma^3)}{10-2\cdot\gamma^2} \rho \cdot r^3,
\]

(2)

where \( \gamma = \frac{V_t}{V_p} \); \( V_t \) and \( V_p \) are the velocities of the transverse and longitudinal waves, respectively; \( \rho \) is the soil density; \( r \) is the plate conditional radius.

For the values \( \gamma = 0.25 \) and \( \rho = 2000 \) kg/m³, the value \( m_{ap} = 2.3 \cdot 10^4 \) kg, and the values \( m_2 \) and \( m_3 \) can be assumed as equal, i.e. \( m_2 = m_3 = 2.8 \cdot 10^4 \) kg.
The soil stiffness and damping coefficients can be calculated from the expressions:

\[
\begin{align*}
    c &= 2 \cdot \frac{E \cdot r}{1 - \nu^2} \\
    b &= b \left(1 - \gamma^2\right) V \cdot \rho \cdot r^2
\end{align*}
\]

where \( E = 1.67 \cdot 10^8 \) Pa is elasticity modulus; \( \nu = 0.49 \) is Poisson's ratio.

For the adopted values of \( E, \nu \), and \( r \), these values are \( c = 4.4 \cdot 10^8 \) N/m, \( b = 1.2 \cdot 10^7 \) N/s/m.

If we assume \( m_1 = 10^5 \) kg, then N-578 type RCS with a quasilinear vertical and lateral rigidity of 739.2 kN/m and 116.7 kN/m, respectively, can be used to hang the inertial mass.

Equations system (1) must be supplemented with the hydraulic drive equations. We assume that force \( F(t) \) (figure 1) is created by a double-acting hydraulic motor with a two-stage electrohydraulic distributor [4, 7, 8]. Schematic diagram of the hydraulic system is shown in figure 2.

![Figure 2. Schematic diagram of the hydraulic drive](image)

Figure 2. Schematic diagram of the hydraulic drive: 1 – the hydraulic cylinder; 2 – the piston; 3 – the main slide valve position feedback sensor; 4 – the main (second) cascade slide valve; 5 – the control (first) cascade slide valve; 6 – electromagnetic drive (EMD) slide valve 5; \( P_s, P_d \) – the supplied working fluid and drainage pressure; \( U_{in}(t) \) - the master frequency generator voltage.

A linearized equations system describing the hydraulic system behaviour has the following form:

\[
\begin{align*}
    \frac{V_0 + V_l}{2B_l} \frac{dp}{dt} + K_{Qp} \cdot P &= S + x_1K_{Qx} \\
    T \frac{m}{c} \frac{d^2x_1}{dt^2} + \left(\frac{m}{c} + T \cdot \frac{b}{c}\right) \frac{dx_1}{dt} + \left(T + \frac{b}{c}\right) x_1 &= K_1 u_i \\
    u_i &= K_g \left(U_{in} - K_{fb} x_2\right) \\
    \frac{dx_2}{dt} &= K_{he} \cdot x_1
\end{align*}
\]

where \( V_0 \) is the hydraulic cylinder cavity space; \( V_l \) is the hydraulic line space; \( B_l \) is liquid bulk modulus; \( K_{Qp}, K_{Qx} \) are transmission coefficients; \( x_1 \) is the first cascade slide valve displacement; \( x_2 \) is main cascade slide valve displacement; \( K_1 \) is EMD transmission coefficient; \( T \) is EMD time constant; \( m \) is EMD moving parts mass; \( h \) is viscous resistance coefficient in EMD; \( c \) is EMD anchor suspension stiffness; \( K_g \) is amplifier gain coefficient; \( K_{fb} \) is feedback coefficient; \( K_{he} \) - hydraulic gain coefficient according to slide valves 4 and 5 displacements.
Equations systems (1) and (4) together with the tabulated values of the nonlinear function \( f(z) \) with the above-given assumptions describe the hydromechanical oscillatory system dynamics in figures 1 and 2.

3. Theory

Equations systems (1) and (4) are solved numerically using the Matlab application package with the "Simulink" extension (the license was purchased for educational and scientific purposes by OmSTU). The simulation circuit in "Simulink" is shown in figure 3.

\[
\mu = 1 + \frac{\left| \dot{Z}_3 \right|}{\left| \dot{Z}_2 \right|} \cdot \cos |\Delta \phi|, \quad (5)
\]

where \( \Delta \phi \) is the phase shift between \( \dot{Z}_3 \) and \( \dot{Z}_2 \).
Masses $m_1$ and $m_2$ oscillations analysis showed that the oscillations occur almost in phase in the frequency range (2–5 Hz).

![Figure 4. Phase-frequency characteristics of the masses oscillations for a seismic source SV 100/20.](image)

The values of the seismic waves generation efficiency coefficient in the range of 2–20 Hz are given in figure 5 for different values of the relative damping coefficient $\xi = \frac{1}{2} h \sqrt{\frac{m_1}{c_1}}$.

![Figure 5. Dependence of the coefficient $\eta$ on the frequency: 1 - $\xi = 0.1$; 2 - $\xi = 0.3$.](image)

The emission efficiency of the source with a force closure in the "inertial mass - soil" system is preserved in the entire frequency range of 2–20 Hz. It is done at low frequencies by means of almost in-phase oscillations of the supporting plates and the base plate, and at high frequencies it is carried out thanks to the difference in the oscillations amplitude.

It is of interest to consider the possibility of obtaining an impulse seismogram analogue when a discrete series of monochromatic frequencies is generated by such a source.

It is known [5] that in the traditional “vibroseis” method the impulse seismogram $x_i(t)$ is formed by calculating the mutual correlation function between the seismogram $x_m$ measured at the receiving point and the probing (generated) signal from the vibrator $x_v(t)$.

$$x_i(t) = \int_{-T}^{T} x_m(t - \tau) x_v(\tau) d\tau,$$

where $T$ is the duration of the "sweep" signal.

It is known that for a fixed $T$ the wider the frequency range at sending the linear-frequency modulated signal ("sweep" signal), the closer the transformed signal to the impulse seismogram. For instance, for
seismic exploration purposes, the frequency range of two octaves is sufficient, while the signal/noise ratio increase is achieved by looping the “sweep” signal transmission and summing the result. An impulse seismogram can also be obtained by generating a discrete frequency series of monochromatic signals with duration $T$, summing them and calculating the convolution integral of the resulting sum during computer processing, i.e. calculating the following integral

$$x_i(t) = \sum_{j=1}^{n} x_j^n(t - \tau) \cdot \sum_{j=1}^{n} x_j^j(\tau) d\tau,$$

(7)

where $n$ is the number of discrete frequency changes.

The computer simulation result for expression (7) at different values of $T$ and $n$ and the frequency range shows a good approximation of the calculations result to the impulse seismogram. For instance, the simulation result with relative values of $x_i$ at a frequency range of 2–8 Hz, a discrete of 0.2 Hz and $T = 10$ s is shown in figure 6.

The results of the outlined above approach show that by means of powerful low-frequency vibrators in a monochromatic emissions mode one can obtain analogues of impulse seismograms with the required degree of approximation to a unipolar impulse. Moreover, in case of an unfavorable signal/noise ratio, the signal summation is possible for multiple emission of frequencies range. When the vibrator operates in the monochromatic frequencies series generation mode, it is possible to obtain amplitude-frequency and phase-frequency characteristics for active seismic monitoring of the stressed medium state.

It should be noted that the monochromatic mode was realized [9] in a vibrating module with a "force closure to the soil" SV 100/20 (figure 7), which was designed and manufactured in OmSTU for vibroseismic influence on oil-gas formations and it successfully passed the pilot industrial testing in oil-and-gas production department Buzulukneft. The main technical characteristics of mobile vibromodule SV 100/20 are as follows:

- vibro-pulling force - 1 MN (100 tnf);
- frequency range - 5–20 Hz;
- relative frequency instability - $10^{-5}$.

![Figure 6. Impulse seismogram.](image)

![Figure 7. Seismic vibrator SV 100/20.](image)
The seismic vibrator dynamics in the "sweep" signal mode was estimated from the computer simulation results in accordance with the scheme in figure 3. The speed graphs $\dot{Z}_2$ and $\dot{Z}_3$ at frequency increase and decrease in the "sweep" for the frequency range of 2–26 Hz and duration $\tau = 6 \text{ s}$ are shown in figures 8 and 9.

![Figure 8. Graph $\dot{Z}_2$ and $\dot{Z}_3$ in the "sweep" mode with frequency increase 20 $Z_2$, $Z_3$ (mm).](image1)

The motion dynamics analysis for the supporting and base plates in figures 8 and 9 shows that it is necessary to create a LFM from the maximum to the minimum operating frequency for a vibration module with a force closure on the soil. The vibromodule dynamics modelling for a long "sweep" ($\tau >> 6 \text{ s}$) revealed the same pattern. The phase shift $|\Delta \phi|$ between the speed of the base and supporting plates was analysed during the emission in the "sweep" signal mode for a duration of $\tau = 6 \text{ s}$ and a range of 16–2 Hz, and the result is shown in figure 10.

![Figure 9. Graph $\dot{Z}_2$ and $\dot{Z}_3$ in the "sweep" mode with frequency decrease.](image2)

If account is made of the graph in figure 8, expression (5) and the fact that the ratio is large at high frequencies, one may conclude that the vibrator under consideration will emit a seismic signal in the "sweep" signal mode efficiently.
The behavior of such seismic waves source is of considerable interest if the frequency range is expanded up to 100–150 Hz, corresponding to the frequency range of seismic sources. Also, the studies were conducted on a mathematical model. The modelling results for the source masses oscillations in the frequency range of 5–100 Hz are presented in Table 1.

**Table 1. Modelling results for the source masses oscillations in the extended frequency range**

| f (Hz) | 5  | 10 | 15 | 20 | 25 | 30 | 35 | 40 | 45 | 50 | 70 | 100 |
|-------|----|----|----|----|----|----|----|----|----|----|----|-----|
| \(2a_{\Delta 3}\) (mm) | 6.4 | 6.2 | 5.4 | 3.5 | 2  | 1.4 | 0.8 | 0.6 | 0.4 | 0.35 | 0.1 | 0.038 |
| \(\Delta z_3\) (mm) | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 | 3.2 |
| \(P_{\text{min}} - P_{\text{max}}\) (MPa) | 0.2- | 0.2- | 1- | 2.5- | 4- | 5- | 6.5- | 6.7- | 7- | 7.8- | 8.5- |
| &nbsp; | 20 | 20 | 19 | 17.5 | 16 | 15 | 14.5 | 14 | 13.7 | 13 | 12.8 | 11.7 |

The notations used in the table are:

\(a_{\Delta 3}\) is base plate oscillations amplitude;

\(\Delta z_3\) is the position of the base plate oscillation center in relation to the day surface;

\(P_{\text{min}} - P_{\text{max}}\) are minimum and maximum values of the pressure oscillations in the hydraulic cylinder.

The analysis of the seismic source elements behavior in a wide frequency range of forced oscillations shows that the working frequency increase results in the force interaction amplitude decrease. To keep the vibro-pulling force in the high-frequency range area, it is necessary to increase the discharge characteristics of the pump station and the electro-hydraulic amplifier.

At the same time, when expanding the frequency range of a seismic source with a force closure in the “inertial mass-soil” system to the high-frequency area, one should remember that all its elements represent oscillatory systems having their own frequencies.

It is known that when the natural object frequencies coincide with the forcing one, a resonance effect occurs and the oscillation amplitude of these objects can acquire values that may lead to their destruction.

The base plate and the supporting posts of the seismic source are characterized by high stiffness and are likely have relatively large values of the oscillation frequencies. At high frequencies the oscillation amplitudes caused by the exciting force are not large. In addition, they rely on a soil foundation with dissipative properties that smooth out the resonance peak significantly and, therefore, the danger of their destruction is not high.

The inertial mass of the seismic source is a welded volumetric structure assembled from I-shaped rollings and channels into cells with reinforced concrete blocks. These blocks have a triangular cross-section and, during the source operation, they are pulled together with the steelwork by steel ropes, which leads to their interlocking and the provision of a monolithic structure.
Interlocking of the reinforced concrete blocks, however, prevents their oscillations and indirect source emissions.

The inertial mass metal structure oscillations of various configurations, but with strictly defined geometric dimensions, were analysed by means of the finite element method and data processing with the Ansys software package.

Studies have shown that the load-bearing metal structure in the process of loading by the periodic force generated by the hydraulic cylinder produces not only bending oscillations, but also shear and torsional ones. Moreover, the bending oscillations natural frequency lies within the range of 22–26 Hz, the shear oscillations are within 30–36 Hz, and the torsional oscillations are within 14–16 and 30–36 Hz.

Consequently, when expanding the frequency range of a seismic source with a force closure in the “inertial mass-soil” system to the high-frequency part, it is necessary to take into account the values of the elements natural frequencies and to provide the necessary resistance during the design process.

It should be noted that at the discharge characteristics increase for the pumping station and the electro-hydraulic amplifier a hydromechanical source of the seismic waves with a force closure in the “inertial mass-soil” system can operate effectively not only in the low-frequency region, but also in a rather wide frequency range.

In recent years the oil industry has widely used vibroacoustic reservoir treatment to increase their productive capacity. Moreover, a large effect is achieved when vibration is produced at the layer natural frequency (the dominant frequency) and quasiresonant phenomena take place in it. As a rule, oil fields have several fluid deposit layers (formations) and each has its own dominant frequency [12, 13, 14].

To carry out the vibration treatment of two layers simultaneously, it is necessary to use two seismic sources, each operating at the dominant frequency of a specific layer. This method is very expensive and with the current shortage of powerful low-frequency seismic sources it is extremely difficult to implement.

Another way to perform simultaneous treatment of several oil formations is as follows. The seismic source emits a complex signal, being the sum of monochromatic signals, whose frequencies correspond to the formations dominant frequencies [10].

This complex signal puts each of the formations into resonance oscillations. Such treatment promotes fracture porosity development and internal potential energy release, which contributes to the intensification of the jets-clusters formation process. Thus, the production rates as well as the efficiency of its vibroacoustic treatment are increased.

Sources with a force closure in the “inertial mass-soil” system with an electrohydraulic exciter of impulse-type vibrations can radiate complex seismic signals if their control system is supplemented with the working fluid pressure feedback in the power hydraulic cylinder.

A block scheme of the control system in such a source, intended to operate at several dominant frequencies, has the following form (figure 11).

![Figure 11](image-url)

**Figure 11.** Block scheme of a seismic source for operation at several dominant frequencies: 1 – sin-dominant frequencies generators; 2 – amplifier; 3 – summation unit; 4 – vibrations exciter servo system; 5 – the vibrations generator; 6 – pumping station
Each of the sinus generators at the output produces a signal with a frequency corresponding to a dominant one. After amplification and summation these signals are transmitted to the input of the seismic source servo system. The vibrations generator, powered by the pump station and controlled by the servo system, processes a complex signal from the summation unit output.

When producing a complex combined control signal, according to the formation properties, the values of the amplitudes and the combined signals phases may vary greatly. A sufficiently flexible control system for the electrohydraulic vibrations exciter allows a seismic source with such characteristics to operate in a group of similar sources.

Realizing the source simultaneous operation at several dominant frequencies of the productive formation and thus putting blocks of various sizes into the resonance, one should expect a significant increase in productive capacity (with other identical parameters of vibroacoustic reservoir treatment).

4. Conclusion

Thus, the conducted research shows that a low-frequency hydromechanical vibromodule with a force closure in the “inertial mass-soil” system is a universal and convenient tool to solve the problems of active environment seismic monitoring, seismology and regional seismic exploration, and the manufactured and tested prototype SV 100/20 can become a prototype of commercial low-frequency mobile sources of seismic waves.

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