Method and application of extending seismic vibrator bandwidth toward low frequency

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Abstract
Low-frequency content of the output signal is very helpful in seismic exploration. In this study, method and application are conducted to extend the bandwidth of a seismic vibrator toward low frequency. Factors limiting vibrator’s low-frequency extension are studied, and stroke load capacity and pump-displacement load capacity are proposed and act as vibrator’s low-frequency performance evaluation indexes. Low-frequency performances of five seismic vibrators available in the market are analyzed. Then, multi-objective optimization is used to extend low frequency, and a newly designed seismic vibrator EV56 is developed based on this method. Results show that stroke load capacities of these vibrators are not much different, and all are beyond 5 Hz at 100% level. Moreover, none of the vibrators reaches a frequency lower than 8 Hz at a full pump-displacement load capacity. Through the optimization, the ground force increases by 8.55%, and the standard deviation of the ground force is reduced by 32.99%. Compared to other five vibrators, the EV56 vibrator generates more than twice the output force at 70% drive force level of 3 Hz and improves more than one full-load capacity octave. Field test shows that compared to a conventional seismic vibrator, the EV56 vibrator is more helpful to discover and reflect the underground structure.

Keywords
Seismic vibrator, low frequency, stroke load capacity, pump-displacement load capacity, multi-objective optimization, ground force

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Introduction
With the advantages of environmental protection, safety, and high efficiency, a seismic vibrator (as shown in Figure 1(a)) has been widely used in oil and gas exploration. More than half of the land seismic explorations are operated through seismic vibrators.1 Bandwidth of the output signal is one of the major performance indexes for a seismic vibrator. Low-frequency content of the output signal is helpful for the inversion of seismic trace data, improving the resolution of seismic data and obtaining structural information of deeper reservoirs, especially the shale gas reservoirs.2–4 Extending the seismic vibrator bandwidth toward low frequency attracts lots of attention.
Because of the mechanical and hydraulic limitations, it is very challenging for the seismic vibrator to acquire low frequency. Most conventional seismic vibrators cannot produce sufficient low-frequency force below 5 Hz. In order to obtain the low-frequency output signal, engineers and scholars have done a lot of approaches and research studies. Many sweep design techniques have been developed to enhance the seismic vibrator low-frequency content. In 2003, Jeffryes and Martin developed a composite sweep to improve the low-frequency content. Later, Bagaini et al. proposed a new method called maximum displacement sweep to enhance the signal content of low frequency by optimally designing the drive force and the variable sweep rate, and the vibrator mechanical and hydraulic specifications were taken into account. Baeten also proposed a method and system to generate a sweep signal with enhanced low-frequency content by generating a combination of a linear and nonlinear sweep and wherein the nonlinear sweep as a function of time is calculated by a predetermined algorithm. Maxwell et al. and Sallas et al. employed pseudorandom sweeps to extend the low-frequency bandwidth. Their field tests indicated that well-designed pseudorandom sweeps had the potential to provide a bit more output than maximum displacement sweep with lower start frequencies. Such sweep methods require a lower drive level and a slower sweep rate, so the sweep length will be longer. Therefore, extending bandwidth toward a low frequency by sweep methods usually decreases the productivity.

To obtain the low frequency practically, some approaches on structure improvement or design have been conducted. In 2015, Meier et al. presented a counter-rotating eccentric-mass vibrator which can generate large forces (266,880 N) at frequencies between 1 and 5 Hz. In the same year, Reust et al. introduced a counter-rotating seismic vibrator called very low-frequency source. It was designed to reach 266,880 N at 3 Hz. The advantage of these two kinds of counter-rotating seismic vibrator is that significant ground force at low frequencies is achieved. Unfortunately, because of the use of the counter-rotating mechanism, the high frequency of these vibrators is limited to around 20 Hz. And in oil and gas exploration, high frequencies are equally important, which are most useful for enhancing spatial and temporal resolution, notably in shallow seismic or vertical seismic profiling (VSP) surveys. For a conventional seismic vibrator equipped with a hydraulic vibrator, the high frequency can be extended to 120 Hz. To acquire broadband data, extending the low frequency on a conventional seismic vibrator will be a better choice. Wei has done some constructive approaches on this field and proposed a prototype low-frequency seismic vibrator. Tests showed that this vibrator improved the ground force at the low-frequency range (<10 Hz) as well as the normal frequency range (10–100 Hz).

Although some progress on extending the low frequency for a seismic vibrator has been achieved, what factors limit extending the low frequency and how to reach low frequency are still unclear. In this article, the factors limiting the seismic vibrator toward low frequency are studied, and a new low-frequency seismic vibrator is developed according to a multi-objective optimization method. Field test shows that compared to the traditional vibrator, the new seismic vibrator can generate enough low-frequency signal, which is helpful to improve the resolution. This research provides theoretical reference for the design and application of the low-frequency seismic vibrator.

### Seismic vibrator

A vibrator is the key component of the seismic vibrator, which is the source of the output signal and directly determines the bandwidth and quality of the output signal. The vibrator is mounted on the truck and consists of a top plate, reaction mass, piston, baseplate, supporting columns, and other accessory structures, as shown in Figure 1(b). The top plate rigidly connects with the baseplate by four supporting columns and one piston, and the piston is mounted on the center of the baseplate. Reaction mass ringing on the piston are supported by two air suspensions on the baseplate. The upper chamber and lower chamber are between reaction mass and piston, where the hydraulic force drives reaction mass to move. During working process, the seismic vibrator is lifted up by the lifting hydraulic cylinder, and most of its weight applies on the top plate and baseplate to maintain good contact between the baseplate and the ground. Then high-pressure hydraulic fluid flows into the upper chamber and lower chamber alternately, driving reaction mass to move up and down. At the same time, the reactive force impacts on the piston face, and then, the force and output signal are transmitted into the ground through the baseplate.

The force acting on the ground is called ground force and is calculated by the weighted sum method as shown in equation (1). The ground force is defined as the weighted sum of the reaction mass and the baseplate accelerations multiplied by their respective mass

\[
-F = m_r\ddot{a}_r + m_b\ddot{a}_b
\]  

where \(F\) is the ground force, \(m_r\) is the mass of reaction mass, \(\ddot{a}_r\) is the acceleration of reaction mass, \(m_b\) is the mass of the baseplate, and \(\ddot{a}_b\) is the acceleration of the baseplate. Equation (1) shows that reaction mass makes a positive contribution to the vibrator ground force if its acceleration is in-phase with the acceleration of the
baseplate. However, it cannot be obtained from equation (1) that how to reach the low frequency with desirable ground force. The approach of extending the low frequency should be conducted from the source of the ground force.

\[ Q(t) = A \times W(t) \times \sin\left[2\pi \left(f_s + \frac{f_e - f_s}{T}t\right)\right] \quad (0 \leq t \leq T) \]

\[ W(t) = \begin{cases} 
\frac{1}{2} \left\{ 1 + \cos\left[\pi \left(\frac{f_s}{f_c} + 1\right)\right] \right\} & (0 \leq t \leq T_c) \\
1 & (T_c < t < T - T_c) \\
\frac{1}{2} \left\{ 1 - \cos\left[\pi \left(\frac{f_e}{f_c}\right)\right] \right\} & (T - T_c \leq t \leq T) 
\end{cases} \quad (2) \]

where \( A \) is the amplitude of the hydraulic force, \( f_s \) is the start frequency, \( f_e \) is the end frequency, \( t \) is the time, \( T \) is the sweep length or recording time, and \( T_c \) is the taper time. \( W(t) \) is the cosine window function or called cosine taper, which is used to ensure a smooth process for starting and ending the vibration. The cosine taper has two benefits. One is protecting the vibrator system to have a buffering process for starting and stopping the vibrator, and the other is to reduce the influence of the Gibbs effect on the signal. Figure 2 shows a hydraulic force which is a linear sweep from 5 to 100 Hz with a sweep length of 20 s at a force level of 275,000 N, and 0.5-s cosine tapers are applied to the start and the end of the sweep.

**Analysis of low-frequency extension**

**Factors limiting the extension of vibrator toward low frequency**

If the start frequency is lower, the movement period of reaction mass will be longer, resulting in a larger displacement of reaction mass. Specifically, for the structure of the vibrator, the peak displacement of the reaction mass is its stroke. The longer the stroke, the larger the volume of the upper and lower chambers, so more hydraulic oil that pushes the reaction mass is needed, leading to require bigger pump displacement. The reaction mass stroke and pump displacement are limited subjecting to the structure of the truck, which cannot be...
freely increased. These two factors limit the extension of the vibrator toward low frequency. How these two factors limit low-frequency extension, how to extend the low frequency, and other questions are still unclear. Therefore, the study of extending the low frequency will be carried out from two aspects: the reaction mass stroke and the pump displacement.

**Limit of reaction mass stroke on low-frequency extension**

The reaction mass stroke is the distance between the extreme points of the upper and lower movement of reaction mass. When calculating the stroke at a certain frequency \( f \), the hydraulic oil and other losses are ignored, assuming that the reaction mass moves ideally under the excitation of sweep signal. According to equation (2), when the reaction mass moves at a single frequency \( f \), equation (2) can be written as

\[
Q(t) = A \sin(\omega t), \quad \omega = \frac{2\pi}{T_s} = 2\pi f \tag{3}
\]

where \( \omega \) is the angular frequency and \( T_s \) is the period of the frequency \( f \). So, the reaction mass’ acceleration \( a_r \) is

\[
a_r = \frac{Q(t)}{m_r} = \frac{A}{m_r} \sin(\omega t) \tag{4}
\]

Then, the displacement of reaction mass \( S_d \) should be

\[
S_d = \int_0^{T_s} A \sin(\omega t) dt \tag{5}
\]

\[
S_d = \frac{A}{m_r} t \sin(\omega t) \tag{6}
\]

It can be seen from equation (6) that the mass of the reaction mass and the square of the frequency are inversely proportional to the stroke. According to equation (6), the limiting of reaction mass stroke on low frequency can be drawn and analyzed, and the KZ28 seismic vibrator is taken as an example.

For the KZ28 seismic vibrator, the rated output amplitude \( A \) is 275,000 N, and the mass of reaction mass \( m_r \) is 4626 kg. Defining the start frequency as 1.5 Hz and sweep signal bandwidth as 1.5–96 Hz, the relationship between the stroke and frequency for the KZ28 seismic vibrator is as shown in Figure 3. The stroke is determined by the frequency when the mass and rated output are fixed. Figure 3 shows that in the bandwidth below 20 Hz, the stroke increases rapidly with the decrease in frequency. When the frequency is lower, the stroke grows faster. If the full load of 1.5 Hz is to be achieved without changing the mass of the reaction mass, the stroke needs to be astonishing 1.338 m, which is almost impossible to achieve for the vibrator structure. As the red box in Figure 3 shows, when the frequency is above 20 Hz, the stroke is very small and close to zero. The stroke greatly limits the extension of the vibrator toward the low frequency. The stroke of the KZ28 seismic vibrator is 0.076 m. With such stroke,
the KZ28 vibrator can only achieve 6.29 Hz at the rated output, as the pink box shows.

For different model of vibrators with the same rated output level, the mass of reaction mass and the stroke are different, so their low frequency performances are different. In this situation, the stroke load capacity \( j \) is proposed to evaluate how the stroke limits the low-frequency extension of the vibrator. The stroke load capacity \( j \) is a percentage of the ratio of \( S_{\text{vibrator}} \) to \( S_r \). \( S_{\text{vibrator}} \) is the stroke of a vibrator equipped, and \( S_r \) is the stroke required to achieve a certain frequency \( f \). When \( S_{\text{vibrator}} \) is greater than \( S_r \), the stroke of the vibrator can reach the requirement of the rated output, and \( j \) is 100%. When \( S_{\text{vibrator}} \) is less than \( S_r \), the stroke of the vibrator cannot meet the requirement, and \( j \) is equal to \( S_{\text{vibrator}} \) divided by \( S_r \). So \( j \) is calculated as follows:

\[
 j = \begin{cases} 
 \frac{S_{\text{vibrator}}}{S_r} & S_{\text{vibrator}} \leq S_r, \\
 \frac{S_{\text{vibrator}}}{S_r} \times 100\% & S_{\text{vibrator}} > S_r.
\end{cases}
\]

The stroke load capacities of different seismic vibrators can be compared according to equation (7). Table 1 shows technical specifications of five seismic vibrators available in the market. In order to facilitate the comparison, the rated output is uniformly set to 275,000 N, and the start frequency is still 1.5 Hz. According to equation (7), the stroke load capacities of the vibrators at a low frequency can be obtained, as shown in Figure 4.

It is can be seen from Figure 4 that the stroke load capacities of these five vibrators at 1.5 Hz are very low at a level of less than 10%. Before reaching 100%, the stroke load capacity grows faster as the frequency increases. The start frequencies of 100% stroke load capacity for KZ28, HEMI 60, AHV-IV 364, ATS 60, and Nomad 65 Neo are 6.29, 6.47, 5.33, 6.83, and 5.42 Hz, respectively. The AHI-IV 364 vibrator has the best performance of 5.33 Hz. However, the stroke load capacities of these vibrators are not much different, and all are beyond 5 Hz at 100% level.

**Limit of pump displacement on low-frequency extension**

The previous analysis shows that the reaction mass stroke plays a key role in the low-frequency extension of the vibrator. Since the reaction mass is driven by

| Model       | KZ28 | HEMI 60 | AHV-IV 364 | ATS 60 | Nomad 65 Neo |
|-------------|------|---------|------------|--------|--------------|
| Rated output (A, N) | 275,000 | 274,000 | 275,000 | 276,000 | 278,000 |
| Mass of reaction mass (\( m_r \), kg) | 4626 | 3742 | 4998 | 3924 | 4700 |
| Stroke of reaction mass (\( S_{\text{vibrator}} \), m) | 0.0760 | 0.0890 | 0.0983 | 0.0762 | 0.1012 |
| Mass of baseplate (\( m_b \), kg) | 1450 | 2132 | 2027 | 1860 | 1580 |
| Piston area (\( P_{\text{area}} \), mm²) | 13,787 | 13,250 | 13,290 | 12,900 | 11,258 |
| Pump displacement (\( D_{\text{vibrator}} \), L/min) | 483 | 530 | 504 | 530 | 458 |
| Hydraulic system pressure (\( P \), MPa) | 20.477 | 20.684 | 21.093 | 21.374 | 24.683 |

*Figure 4. The stroke load capacities of five vibrators available in the market at low frequency.*
hydraulic force, whether the pump can provide sufficient hydraulic flow to drive the reaction mass is a real problem in extending the low frequency. Therefore, the limit of pump displacement on low-frequency extension is analyzed. \( D \) is the pump displacement required to achieve a certain frequency. The pump displacement required for one cycle (two strokes) at a frequency \( f \) (period is \( T_s \)) is

\[
D = \frac{2S \times P_{\text{area}}}{T_s} = \frac{AP_{\text{area}}}{\pi m_r f} \tag{8}
\]

Equation (8) shows that at a force level, variables related to the pump displacement \( D \) include piston area \( P_{\text{area}} \), mass of reaction mass \( m_r \), and the frequency \( f \). The piston area is proportional to the pump displacement. The reaction mass weight and frequency are inversely proportional to the pump displacement.

According to equation (8) and the parameters of the KZ28 seismic vibrator in Table 1, the relationship between pump displacement and frequency from 1.5 to 96 Hz can be obtained, as shown in Figure 5. It is can be seen that as the pump displacement is inversely related to the frequency, the full-load output (275,000 N) in the low frequency band has a huge demand for the pump displacement. When the frequency is 1.5 Hz, the required pump displacement is as high as 3321.67 L/min. As the frequency increases, the requirement for pump displacement at full-load output decreases rapidly, as the red box shows. From the pink box in Figure 5, it can be seen that the KZ28 vibrator's pump displacement (483 L/min) can only meet the requirement of 10.32 Hz at the rated output of 27,500 N.

To compare the low-frequency performances of different seismic vibrators, pump-displacement load capacity \( \zeta \) is proposed to evaluate how the pump displacement limits the low-frequency extension of the vibrator. The pump-displacement load capacity \( \zeta \) is a percentage of the ratio of \( D_{\text{vibrator}} \) to \( D \). \( D_{\text{vibrator}} \) is the pump displacement of a vibrator equipped, and \( D \) is the pump displacement required to achieve a certain frequency \( f \). When \( D_{\text{vibrator}} \) is greater than \( D \), the pump displacement of the vibrator can reach the requirement of the rated output, and \( \zeta \) is 100%. When \( D_{\text{vibrator}} \) is smaller than \( D \), the pump displacement of the vibrator cannot meet the requirement, and \( \zeta \) is equal to \( D_{\text{vibrator}} \) divided by \( D \). And \( \zeta \) is developed as follows

\[
\zeta = \begin{cases} 
D_{\text{vibrator}} = \frac{D_{\text{vibrator}} \pi m_r f}{AP_{\text{area}} 100\%}, & D_{\text{vibrator}} < D \\
D_{\text{vibrator}} \geq D & 
\end{cases} \tag{9}
\]

According to Table 1 and equation (9), the load capacities of pump displacement of the five vibrators available in the market at a low frequency can be obtained, as shown in Figure 6. The rated output is 275,000 N, and the start frequency is 1.5 Hz. Figure 6 shows that the load capacities of pump displacement of five vibrators at 1.5 Hz are very low, and none of the load capacities are greater than 20%. Before reaching 100%, the pump-displacement load capacity is linear with frequency and grows with the increase of frequency. The pump-displacement load capacities of different vibrators vary greatly. The start frequencies of 100% pump-displacement load capacity for KZ28, HEMI 60, AHV-IV 364, ATS 60, and Nomad 65 Neo are 10.32, 11.17, 8.82, 10.37, and 8.75 Hz, respectively. The pump displacement has a great influence on the low-frequency extension, and none of the vibrators reach a frequency lower than 8 Hz at 100% pump-displacement load capacity.

**Figure 5.** The relationship between pump displacement and frequency for KZ28 seismic vibrator.
Extending low frequency for newly designed vibrator

Improvement of vibrator for extending low frequency

Method for extending low frequency. For a seismic vibrator, the ground force is a key indicator to evaluate the vibration output. The essence of expanding the low frequency is to improve the ground force of the low frequency. However, the vibration output is related to the entire sweep frequency, including low frequencies and high frequencies. Expanding the low frequency should also pay attention to the high-frequency output, and there are multiple parameters involved. So, a reasonable method is needed to achieve a balance between low-frequency output and high-frequency output when we extend the low frequency. Multi-objective optimization is an effective way to solve this problem. Multi-objective optimization involves maximizing or minimizing multiple objective functions' subject to a set of constraints, which has been applied in many fields of science, including engineering, economics, and logistics where optimal decisions need to be taken in the presence of trade-offs between two or more conflicting objectives.

Multiple objective functions. Expanding the low frequency may affect two aspects, one is low-frequency performance and the other is high-frequency response. The objective function of bandwidth expansion can be expressed as follows:

1. From the previous analysis, the stroke is negatively correlated with the start frequency, so the objective function of the stroke can be expressed as when the frequency is 1.5 Hz, the required stroke is the smallest, as equation (10) shows

\[
\min S_r = \frac{A}{2\pi^2 m_r f_L} \tag{10}
\]

where \( S_r \) is the stroke required to satisfy \( f_L \), \( A \) is the output amplitude, \( m_r \) is the weight of reaction mass, and \( f_L \) is the start frequency, which is 1.5 Hz.

2. Similar to the stroke load capacity, the pump displacement is also negatively correlated with the frequency. The objective function of the pump displacement is when the frequency is 1.5 Hz, the required pump displacement is the smallest. It can be written as follows

\[
\min D_{vibrator} = \frac{60,000 A P_{area}}{\pi^2 m_r f_L} \tag{11}
\]

where \( D_{vibrator} \) is the pump displacement required to satisfy \( f_L \) and \( A _{p} \) is the piston area. Theoretically speaking, as a structural parameter, the piston area can be optimized as a variable. In fact, the piston area is limited by the rated output and the bearing pressure of the hydraulic system, which should be treated as a constant.

From equations (10) and (11), it can be seen that \( S_r \) and \( D_{vibrator} \) are inversely proportional to the reaction mass weight. In order to avoid double counting, equation (10) acts as the optimization objective function instead of taking both equations (10) and (11).

3. The ground force is not necessarily as large as possible, but the output should be as equal as possible. 

Figure 6. The load capacity of pump displacement of the five vibrators available in the market at low frequency.
possible to the input, so the volatility of the ground force cannot be too large. Based on this, minimizing the standard deviation of the ground force throughout the sweep frequency is one of the objective functions, which is as equation (12) shows

$$
\text{min } \sigma_{F_g} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (F_{gi} - \bar{F}_g)^2}
$$

where $\sigma_{F_g}$ is the standard deviation of the ground force throughout the sweep frequency, $N$ is the number of all positive amplitude throughout the sweep frequency, $F_{gi}$ is the ith positive amplitude, and $\bar{F}_g$ is the average of the amplitudes.

The ground force can be calculated as follows

$$
F_g = C_z \dot{X}_b + K_z X_b
$$

where $C_z$ is the equivalent dynamic damping, $K_z$ is the equivalent dynamic stiffness, $\dot{X}_b$ and $X_b$ are the velocity and displacement of the baseplate, respectively.

**Constraints.** According to the objective functions, the constraints are determined as follows:

1. Mass of reaction mass $m_r$. The mass of reaction mass is the key factor affecting the low frequency output of the vibrator, and must be optimized. The reaction is limited by the structure of the vibrator truck and the space that can be installed. The reaction mass weight cannot be greater than 10 tons and cannot be zero. So, the mass of reaction mass is in the range of (0, 10,000), in units of kg.

2. Mass of baseplate $m_b$. The mass of the baseplate affects the high-frequency output of the vibrator and needs to be optimized. The mass of the baseplate is also limited by the structure of the vibrator truck and the space that can be installed. It cannot be greater than 5 tons or be zero. Therefore, the mass of reaction has a value range of (0, 5000) in units of kg.

3. Baseplate area $S_b$. The baseplate area has an important influence on the response of the structure. The baseplate is used as a mounting platform for columns, pistons, air springs, and so on, and its area cannot be less than 1 m². Due to the installation space of the vibrator, the baseplate area cannot be greater than 5 m². In this situation, the baseplate area is in the range of (1, 5), and the unit is m².

4. Thickness of baseplate $H$. The baseplate mass and area are not independent of each other but are mutually constrained. The baseplate is made of I-beam. When the baseplate area is selected, the mass of the plate can be determined substantially according to the size of the I-beam. For a seismic vibrator, the height of the I-beam cannot be less than 0.16 m based on the strength requirements. Therefore, the value of $H$ can only be greater than or equal to 0.16 m and can only be taken from the standard heights which are 0.16, 0.18, 0.2, 0.22, 0.25, 0.28, and 0.32, in units of m.

**Multi-objective optimization model.** Based on the above analysis, the multi-objective optimization model for vibrator low-frequency expanding is as follows

$$
\begin{align}
\text{Objective} \\
\begin{cases}
\text{min } S_r = \frac{d}{2\pi m_f \omega} \\
\text{min } S_{Fg} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (F_{gi} - \bar{F}_g)^2}
\end{cases}
\end{align}
$$

subject to

$$
\begin{align}
&\begin{cases}
m_r \in (0, 10000) \\
m_b \in (0, 5000) \\
S_b \in (1, 5) \\
H \in [0.16, 0.18, 0.20, 0.22, 0.25, 0.28, 0.32]
\end{cases}
\end{align}
$$

**Solution of the multi-objective optimization model.** There are many methods to solve multi-objective optimization problems, and particle swarm optimization (PSO) is one of them. PSO has many advantages, such as strong algorithm versatility, group search with memory, easy programming, and parallel collaborative search. The algorithm has been widely used in structural design, image processing, and neural network training. In this article, PSO with Pareto optimality is applied to the multi-objective optimization model of extending the low frequency. In this solution, the objective function comparison takes Pareto dominance into account when moving the PSO particles, and non-dominated solutions are stored so as to approximate the Pareto front.

The detailed solution steps of the PSO method are detailed in Lei and Song et al. For the multi-objective optimization of extending the low frequency, the solution parameters are set as follows: cognitive learning factor $C_1 = 2$, social learning factor $C_2 = 2$, population scale $N = 300$, maximum iterations $t_{max} = 100$, non-dominated solution volume $T = 50$, the archive grid number $T_{grid} = 7$, the archive expansion factor $T_{alpha} = 0.1$, the archive elimination factor $T_{gamma} = 2$, the maximum inertia weight $\lambda_{max} = 0.9$, and the minimum inertia weight $\lambda_{min} = 0.4$. The optimization was conducted with Matlab.
According to the Matlab simulation, Pareto optimal solutions are obtained as shown in asterisks in Figure 7.

It can be seen that the best solution is one of solution 1, solution 2, and solution 3. To determine the best solution, these three solutions are compared as shown in Table 2. In the comparison, $S_{Fg}$ and $S_r$ of the solution 2 are used as references. Compared to solution 2, the $S_r$ of solution 1 reduces 0.0285% and its $S_{Fg}$ increases 2.803%, which means that solution 1 only reduces a little stroke but brings a lot of ground force fluctuations. Comparing with solution 2, solution 3’s $S_{Fg}$ is 0.467% lower than that of solution 2 and its $S_r$ is 0.475% higher than that of solution 2, indicating that solution 3 reduces a little ground force fluctuations but brings longer stroke. Therefore, solution 2 is the best solution of the solutions. The parameters to obtain solution 2 are $m_r = 5904.983$ kg, $m_b = 930.102$ kg, $S_b = 1.660$ m$^2$, and $H = 0.16$ m. According to the manufacturing specifications of the seismic vibrator, the mass of the reaction mass and the mass of the baseplate must be integers. So, the final optimization results are as follows: $m_r = 5905$ kg, $m_b = 930$ kg, $S_b = 1.66$m$^2$, and $H = 0.16$ m.

| Solution | $S_{Fg}$ (N) | $S_r$ (m) | Compared to solution 2 (%) |
|----------|---------------|-----------|----------------------------|
| 1        | $3.301 \times 10^4$ | 1.0521    | +2.803 -0.0285             |
| 2        | $3.211 \times 10^4$ | 1.0524    | 100 100                    |
| 3        | $3.196 \times 10^4$ | 1.0529    | -0.467 +0.475              |

Figure 7. Pareto optimal solutions of the multi-objective optimization model.

Comparison of ground forces before optimization and after optimization. The ground force of 3 Hz is close to full load, which is fit for comparison of low-frequency ground forces before optimization and after optimization. Figure 8 shows the comparison of the ground forces at a stable sweep frequency of 3 Hz with 5-s sweep time. It can been seen that the ground force amplitude before optimization is $2.151 \times 10^5$ N, and the amplitude of the optimized output is $2.335 \times 10^5$ N. Through the optimization, the ground force increases by 8.55%.

The ground forces of full sweep are shown in Figure 9, wherein Figure 9(b) is the envelope curves of the ground forces in Figure 9(a). The amplitudes of the start taper are not taken into account in the calculation of the standard deviation.

Compared with the ground force before optimization, the ground force after optimization is significantly improved at a high frequency band (60–120 Hz). Especially, the ground force at 120 Hz increases from $1.042 \times 10^5$ N to $1.397 \times 10^5$ N, which is increased by 34.07%. At the same time, the uniformity of the output signal is further improved. The standard deviation of the sweep output amplitude is reduced from $4.792 \times 10^4$ N before optimization to $3.211 \times 10^4$ N, which is reduced by 32.99%. The output of the middle frequency band slightly reduces. The maximum output of the middle frequency band decreases from $2.324 \times 10^5$ N to $2.172 \times 10^5$ N, which is reduced by 6.54%. Although the middle frequency output is sacrificed a little, the uniformity of the entire sweep frequency is greatly improved, which has a great effect on improving the signal-to-noise ratio of the output signal.

Results and discussion

Load capacity of newly designed vibrator. According to the optimization and engineering practice, the new low-frequency seismic vibrator EV56 was developed. Technical specifications of the EV56 seismic vibrator is as shown in Table 3.
To fully understand the improvement of the EV56 vibrator compared with other vibrators, the stroke load capacities and the pump-displacement load capacities of these vibrators are plotted in Figure 10. The solid lines represent the stroke load capacity and the dotted lines represent the pump-displacement load capacity.

It is can be seen from Figure 10 that the stroke and pump displacement have phased limitations on the low-frequency load capacity. For each vibrator, there is an intersection between the stroke load capacity curve and the pump-displacement load capacity curve. Taking the EV56 vibrator as an example, the solid line and the dotted line have an intersection point A. At the frequencies below the intersection point, the low-frequency load capacity is limited by the stroke. And when the frequencies are above the intersection point,
the low-frequency load capacity is limited by the pump displacement. The stroke load capacity determines the load capacity at 1.5 Hz, and the pump displacement determines the start frequency of the full rated output. As shown in Table 4, the load capacity of the EV56 vibrator at 1.5 Hz is 19.38%, and the start frequency of full-load capacity is 3.82 Hz. From the point of view of the start frequency of full-load capacity, the performances of discussed vibrators are as follows: EV56 > Nomad 65 Neo > AHV-IV 364 > KZ28 > ATS 60 > HEMI 60. The EV56 vibrator has made a big progress, but no vibrator reaches 100% load capacity below 3 Hz. The start frequency to achieve a full drive force for the EV56 vibrator is 3.82 Hz, while it is 10.32 Hz for the KZ28 vibrator. In field application, the drive force level is generally below 70% of the rated output. At this 70% drive force level, the EV56 vibrator reaches as low as 2.85 Hz and generates more than twice the output force compared to other vibrators.

**Full-load capacity octave of newly designed vibrator.** In seismic exploration, broadband signal improves resolution and is also beneficial for full waveform inversion (FWI) to image deep target of underground. FWI requires more octaves containing more low-frequency content. So the full-load capacity octave $R_{oct}$ is proposed to evaluate the low-frequency performances of the vibrators. In a sweep signal, assuming the end frequency is $f_{end}$ and start frequency of full-load capacity is $f_{start}$, $R_{oct}$ can be calculated as follows

$$R_{oct} = \log_2 \frac{f_{end}}{f_{start}}$$

The start frequencies of full-load capacity of the six vibrators can be obtained from Figure 10. The start frequencies of load capacity for EV56, KZ28, HEMI 60, AHV-IV 364, ATS 60, and Nomad 65 Neo are 3.82, 10.32, 11.17, 8.82, 10.37, and 8.75 Hz, respectively. If the end frequency is 96 Hz, the full-load capacity octaves of EV56, KZ28, HEMI 60, AHV-IV 364, ATS 60, and Nomad 65 Neo are obtained as shown in Figure 11. Figure 11 demonstrates that the EV56 vibrator reaches 4.65 full-load capacity octaves and has the best performance in the vibrators involved in the comparison. The full-load capacity octaves of conventional vibrators are not much different, which are between 3.22 and 3.46. Compared to them, the EV56 vibrator has improved more than one octave, which can provide better output signal for exploration.

**Field test of newly designed vibrator**

To investigate the performance of the newly designed vibrator, a field test was conducted in Xinjiang, northwest China. The surface of the test field was outcrop. In the test, an EV56 seismic vibrator and a conventional seismic vibrator were shot at the same location, respectively. For the EV56 seismic vibrator, the hydraulic force was a linear sweep from 1.5 to 96 Hz with a sweep length of 20 s at a force level of 192,500 N, and 0.5-s cosine tapers were applied to the start and the end of the sweep. The conventional seismic vibrator performed a linear sweep from 5 to 96 Hz in 20 s with cosine tapers of 0.5 s at 192,500 N force. Figure 12 shows the comparison between a conventional seismic vibrator and EV56 seismic vibrator on seismic sections obtained from the test. It is can be seen from Figure 12 that the seismic signal excited by the EV56 vibrator can provide a higher resolution seismic profile than the conventional seismic vibrator. The sections in the black boxes indicate that the EV56 vibrator can explore detailed underground structures, but the traditional vibrator does not have such ability. Comparing the seismic sections in the red boxes, it can be demonstrated that for the same structure, the seismic signal from the EV56 vibrator can provide higher resolution and reflect structural details. This test shows that the newly designed vibrator EV56

| Vibrator          | Load capacity of 1.5 Hz (%) | Start frequency of full-load capacity (Hz) |
|------------------|-----------------------------|-----------------------------------------|
| EV56             | 19.38                       | 3.82                                    |
| KZ28             | 14.54                       | 10.32                                   |
| HEMI 60          | 5.38                        | 11.17                                   |
| AHV-IV 364       | 7.94                        | 8.82                                    |
| ATS 60           | 4.83                        | 10.37                                   |
| Nomad 65 Neo     | 7.58                        | 8.75                                    |

**Figure 11.** Comparison of full-load capacity octaves of the vibrators.

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has successfully provided more low-frequency energy, which is helpful for oil and gas exploration.

Conclusion

In this article, the method and application for a seismic vibrator to extend the bandwidth toward a low frequency are developed. Low-frequency performances of different vibrators are evaluated, as well as the newly designed vibrator. According to this study, the conclusions can be drawn as follows:

1. According to the sweep signal, the method of calculating the reaction mass stroke is developed. The stroke load capacity and the pump-displacement load capacity are proposed to evaluate the low-frequency performance of the seismic vibrator.
2. The low-frequency performances of five seismic vibrators available in the market are analyzed. The stroke load capacities of these vibrators are not much different, and all are beyond 5 Hz at 100% level. In addition, none of the vibrators reach a frequency lower than 8 Hz at 100% pump-displacement load capacity.
3. Multi-objective optimization is used to extend the low frequency. Through the optimization, the ground force increases by 8.55%, and the standard deviation of the ground force is reduced by 32.99%.
4. The frequency of full-load capacity of the EV56 vibrator reaches as low as 3.82 Hz. Compared to other five vibrators, it generates more than twice the output force at 70% drive force level of 3 Hz and has improved more than one full-load capacity octave.
5. The comparison between the newly designed vibrator EV56 and conventional vibrator is conducted based on a field test. Through the comparative analysis of the seismic sections, it shows that the EV56 vibrator is more conducive to discover and reflect the underground structure.

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