Vibration Source Creation of Unbalance Type with Liquid-Filled Internal Operating Chamber

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Abstract. The creating design features of harmonic vibrations’ submersible source with an internal working chamber, filled with a liquid are considered in the article. The technical requirements for a vibration source operating in a liquid containing mechanical impurities are formulated. Numerical simulation of the unbalance rotation in a liquid are carried. The stationary distribution of fluid pressure on the unbalance blade midsection is calculated under standard conditions and the coefficient of frontal resistance is determined. Calculations were made, taking into account design parameters of source, stresses and deformations of its elements in the process of operation were determined, and the critical frequencies were found. The materials choice for its nodes was substantiated and the advantages in the work of the created source were shown.

1. Introduction
Unbalanced vibration sources are widely used in the construction industry. So, equipment was created on their basis to compact soils, road surfaces, concrete mixtures during the construction of monolithic and reinforced concrete buildings, to loosen and disperse bulk materials. Wells are drilled and water is pumped through them in the technology of dewatering and draining of pits. Increasing water flow to the well will significantly reduce drilling costs, since it will be necessary to discharge the well grid by increasing the grid spacing.

2. Mathematical processing of the research result
A solution to use submersible unbalanced vibration sources, installed in wells at a depth of aquifer is proposed. These types of sources generate seismic vibrations that propagate radially from the well along aquifer, contributing to water inflow acceleration [1-5]. We will consider the possibility of creating a submersible vibration source with an internal working chamber, filled with a liquid in this article.

Previously, we analyzed the main schemes for constructing submersible unbalanced vibration sources, the main parameters were calculated, the features of their work in various environments were considered [6-10].

In Figure 1 a, diagram of an unbalanced vibration source with an offset center of gravity is shown. Centrifugal force acted on the borehole walls through the bearing assemblies and fastening system, as a result of the shaft and bearings experience a load increased.
Unbalanced vibration source with running-in on the trunk (Figure 1 b) creates harmonic vibrations due to running-in with a roller, fixed to the shaft along the inner wall of the trunk. This type of source has high energy performance and low reliability.

Unbalanced vibration source with a free end (Figure 1 c), creates harmonic vibrations due to the running-in of the step with the inner part of a hollow cone, fixed to the shaft through a pliable articulated joint. The source has the smallest overall dimensions of the case and can generate high-frequency radiation at a much lower engine speed. When liquid enters the body, the contact of the step with the cone disappears, which leads to a complete absence of the generated force.

The performed analysis of the existing generators made it possible to determine the structural diagram of the vibration source capable to work with an internal chamber, filled with liquid. The source, shown in Figure 1 a, has the ability to operate in a liquid containing mechanical impurities.

The fluid filling the inner chamber of the vibration source, operating in the well, creates a drag force that prevents the unbalance from rotating and increases the load on the drive.

Let us consider the stationary operating mode of the source. We suppose that the unbalance rotates around an axis $Oz$, directed vertically, upward from the base of the source (Figure 2) with a constant frequency $f = 1 \text{ Hz}$. Let us determine the value of the maximum linear speed that will be observed at the most distant point at a distance $r$ from the axis of rotation $\nu = \omega r = 2 \pi f r = 0.3 \text{ m/s}$ [11].

The unbalance axis of rotation passes through point $O$, $OP$ is the midsection of the unbalance blade. Fluid flow velocities at points $M$ and $N$ are directed along the vectors $\vec{u}_r$ and $\vec{u}$, accordingly.
points O and M are connected by a radius vector $\vec{r}$. The line $\overline{O\text{--}M}$ defines the area border, where the unbalance rotates $\overline{\text{--}}$ and the area located between the chamber’s wall $\overline{\text{--}}$.

The vector of frontal resistance force is directed opposite to the vector of the rotation speed, and its value is determined by the formula [10]:

$$R = \zeta \cdot S \cdot \frac{y \cdot v}{2},$$

where $\zeta = \zeta \left( Re, Fr \right) Re^{y} \cdot Fr^{w}$, $y, w > 0$ – dimensionless drag coefficient; $S$ – area of the unbalance body projection onto a plane perpendicular to the direction of movement (“midsection”), $\rho$ – fluid density, $v$ – linear speed of unbalance movement in liquid on the site dS, $Re$ – Reynolds number, $Fr$ – Froude number.

The system of hydrodynamics equations for the unbalance blade rotates formation:

$$\frac{\partial (\rho u)}{\partial t} + \nabla \cdot (\rho u u) + 2\rho (\alpha \times u) + \rho \omega \times (\alpha + r) = \nabla p + \nabla \cdot (\mu \nabla u),$$

where $F_{k} = 2\rho (\omega \times u)$ – Coriolis force, $F_{c} = \rho \omega \times (\omega + r)$ – centrifugal force, $\omega$ – angular velocity of rotation in a non-inertial frame of reference, $u_{r}$ – relative linear velocity of a material point of a fluid, $p, \rho, \mu$ – pressure, density and dynamic viscosity of the fluid, $r$ – radius vector outgoing from the unbalance axis of rotation to the point in question.

The system of Navier-Stokes equations for the region with liquid located, between the chamber wall and the region where the unbalance rotates:

$$\frac{\partial (\rho u)}{\partial t} + \nabla \cdot (\rho u u) = -\nabla p + \nabla \cdot (\mu \nabla u).$$

The system is closed by the equations of continuity $\rho \nabla \cdot u = 0$ and states $p = f(\rho)$.

On the border of two speed regions $u_{r}$ and $u$ connected with the relationship: $u_{r} = u - \omega \times r$. The no-leakage and adhesion condition is set for normal and tangential velocities on the wall of the unbalance blade and the chamber ($\vec{u} \cdot \vec{n} = 0$, $\vec{u} \cdot \vec{t} = 0$).

Comparison of the results, obtained using the drag formula and Ansys fluent, calculating by the formula for the drag force, a dimensionless coefficient $\zeta = 1.01$ and water density were taken and shown in Figure 3.

Comparison of the obtained results, using the drag formula and Ansys fluent is shown in fig3. Dimensionless coefficient and water density were taken, calculating by the formula for the drag force $\rho = 999.87$ kg/m$^3$.

![Figure 3](image-url)

**Figure 3.** Comparison of the calculating results of water pressure distribution on the blade of the rotating unbalance, made according to the formula for the drag force and in Ansys fluent.
Modal analysis was carried out in order to determine the degree of possible resonance modes danger, during the vibration source design, [13–16]. The first two modes of the unbalance structure’s natural vibrations with resonant frequencies of 86.6 Hz and 103.5 Hz are shown in Figure 4. Campbell’s diagram was also built, Figure 5 and two critical shaft rotation speeds are determined - 5197 rpm and 6211 rpm. The found resonant harmonics did not fall within the operating range of the acting external loads, and therefore the structure of the vibration source can be considered strong and resistant to vibration.

![Figure 4](image)

**Figure 4.** Calculated fields of natural unbalance oscillations. The first two modes of oscillations with resonant frequencies are given: a) - 86.6 Hz, b) - 103.5 Hz.

![Figure 5](image)

**Figure 5.** Campbell diagram. In order to increase the overhaul period of the vibration source, its design safety margin is increased five times, Figure 6.

![Figure 6](image)

**Figure 6.** Calculated field of the shaft safety factor, loaded with the added mass, rotating with an angular speed of 300 rpm.
Numerical modeling of strength characteristics of the unbalanced vibration source showed that the maximum deformation limiting values and shaft stresses did not exceed the yield strength of steel 40X (175 MPa). The vibration source design allows it to develop a force of up to 480 N at a frequency of 5 Hz, and its rotating parts have a fivefold safety margin.

On the basis of the research carried out, a design was developed and a prototype was manufactured. Its operability was checked on the test bench, power consumption, heating of bearing assemblies, change in the slip coefficient of the drive from the pumped liquid viscosity through the internal chamber of the generator and recorded seismic vibrations, emitted by a source with a liquid-filled internal chamber and in the air were also determined (Figure 7) ...

![Figure 7. Seismogram in the during vibration source operation: solid line is source operation on dry, dotted line is source operation with liquid in the inner chamber.](image)

3. Results
The developed submersible borehole vibration source of unbalanced type is capable of continuously working together with a pump in water-saturated and silty soils. There are two modes of its operation: without pumping and with the possibility of pumping a fluid containing fine mechanical impurities through the inner chamber of the generator to the pump.

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