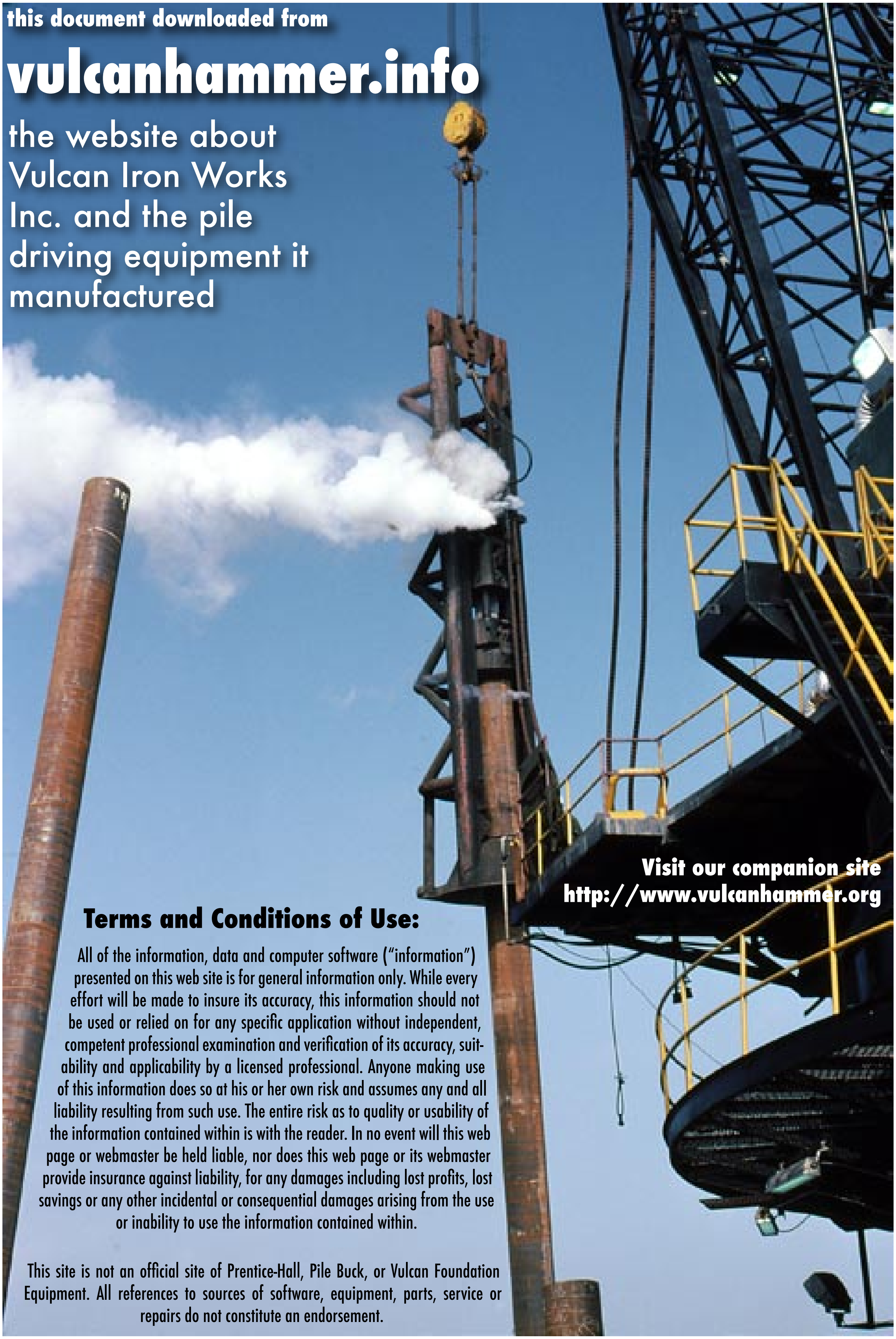

# IMPROVEMENT OF THE CAPACITY OF VIBRATING HAMMERS FOR DRIVING PIPES INTO SOILS

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UDC 624.155.15-868

It is known that vibrating hammers are highly effective for driving pipes. However, their wide use is prevented by their insufficient durability and high energy-consumption, which are determined mainly by the high velocity of the hammer at the moment of impact against the driven element.

The results of the investigations described in this article indicated that the driving capacity of vibrating hammers in soils of medium density can be improved without increasing the impact velocity, by increasing the length of the path traveled jointly by the hammer and the pipe after each blow. When the vibrating hammer operates under longitudinal action, the above-mentioned effect can be obtained by adjusting the force in the springs, in case of large negative gaps. High driving capacity is possessed also by vibrating hammers operating under longitudinal-rotary action, in which the path traveled jointly by the hammer and the pipe is increased as a result of torsional vibrations of the driven pipe.

The experimental investigations conducted in order to justify the analytical model indicated that during vibration-impact driving of pipes open at the bottom, the blow can be regarded as inelastic and instantaneous, while the penetration resistance can be regarded as plastic.

A longitudinal-action free spring vibrating hammer and a longitudinal-rotary action vibrating hammer were investigated.

In the first case (Fig. 1), the springs, which were not connected to the driving element, were compressed by applying an external force. For the study of this analytical model, the following assumptions were made:

1. The vibrating hammer and the pipe are absolutely rigid bodies.
2. The disturbing force created during rotation of the eccentric weights acts according to the law $P = P_0 \sin \omega t$.
3. The impact of the vibrating hammer against the pipe is instantaneous and inelastic.
4. The forces which resist the motion are constant.

The investigators considered a steady motion with a period $2\pi/\omega$, in which driving of the pipe starts at the moment of the impact caused by the ram of the vibrating hammer and ends before the following impact.

The motion of the vibrating hammer for $0 < t < t_n$ is described by the equation

$$M_1 \ddot{z}_1 = P_0 \sin(\omega t + \alpha) - Q_1 - c(z_1 + z_p) + M_1 \ddot{z}_2, \tag{1}$$

in which $M_1$ and $Q_1$ are the mass and weight of the vibrating hammer;

$z_1$ is the coordinate of the vibrating hammer with respect to the driven element;

$z_2$ is the coordinate of the driven element;

$z_p$ is the initial force in the springs which compress the vibrating hammer against the driven pipe (considered to be constant);

c is the coefficient of rigidity of the springs;

All-Union Scientific-Research Institute of Hydro and Sanitary Engineering. Leningrad. Translated from Osnovaniya, Fundamenty i Mekhanika Gruntov, No. 3, pp. 6-9, May-June, 1973.

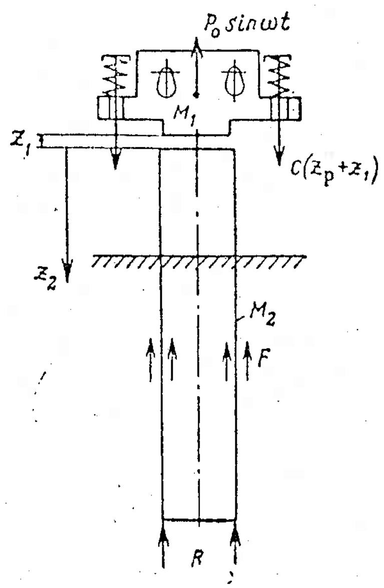


Fig. 1

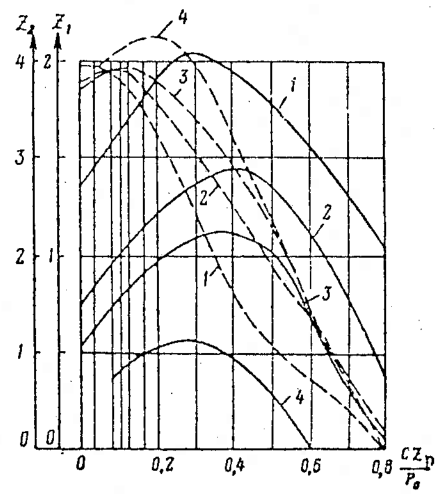


Fig. 2

Fig. 1. Analytical scheme of free spring vibrating hammer.

Fig. 2. Graphs representing the variation of the impact velocities (dashed lines) of the vibrating hammer and the penetration (Full lines) after each blow for different spring compressions and penetration resistances.

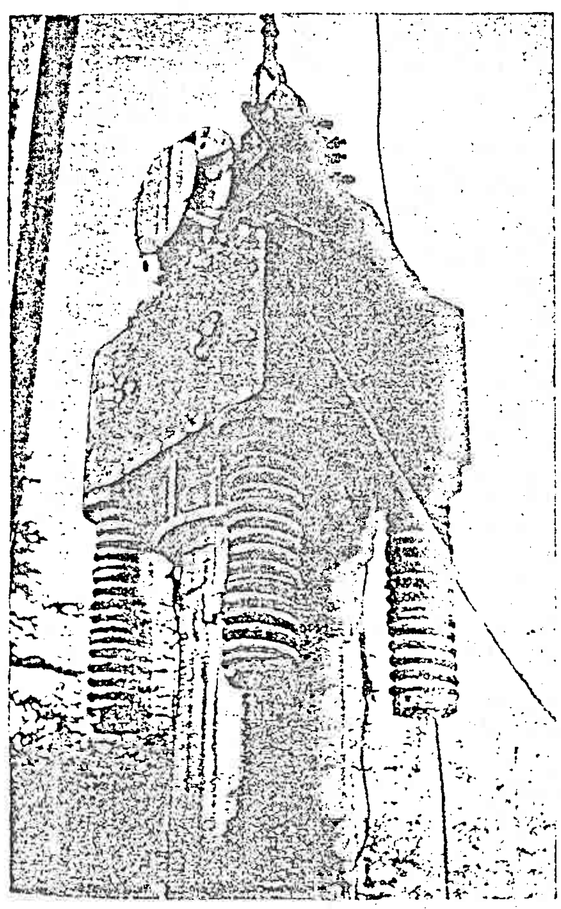

Fig. 3. General view of BVS-1 free spring vibrating hammer.

$t$ is the time reckoned from the moment at which the vibrating hammer is separated from the driven element;

$t_s$ is the time at which the pipe stops moving;

$P_0$ is the amplitude of the disturbing force;

$\omega$ is the angular frequency of the disturbing force;

$\alpha$ is the initial phase angle.

Equation (1) describes the general case of travel of the vibrating hammer, which can start when the pipe is already moving.

For a fixed pipe, the travel of the vibrating hammer for $t_s < t < t_c$ is expressed by the equation

$$M_1 \ddot{z}_1 = P_0 \sin(\omega t + \alpha) - Q_1 - c(z_1 + z_p), \qquad (1, a)$$

in which $t_c$ is the time of completion of the travel of the vibrating hammer.

When the motion of the vibrating hammer takes a time $0 < t < t_s$, the driving of the pipe is described by the equation

$$M_2 \ddot{z}_2 = Q_2 - (R + F), \qquad (2)$$

for joint motion of the vibrating hammer and the pipe after the blow, and for $t_c < t < 0$

$$(M_1 + M_2)\ddot{z}_2 = P_0 \sin(\omega t + \alpha) + (Q_1 + Q_2 + cz_p) - (R + F), \qquad (3)$$

in which $M_2$ and $Q_2$ are the mass and weight of the driven element; and R and F are the frontal and lateral penetration resistances.

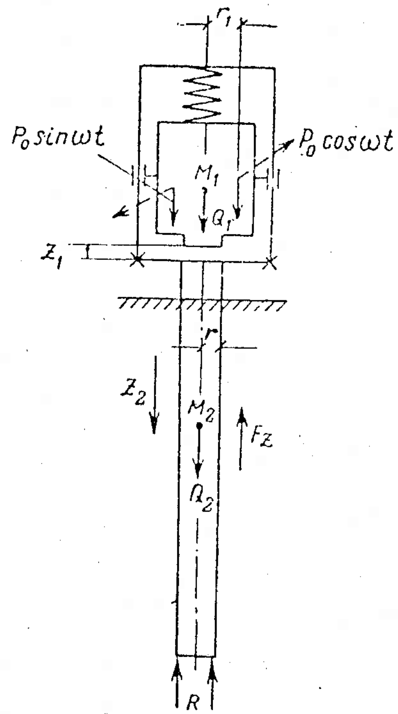


Fig. 4. Analytical scheme of longitudinal-rotary vibrating hammer.

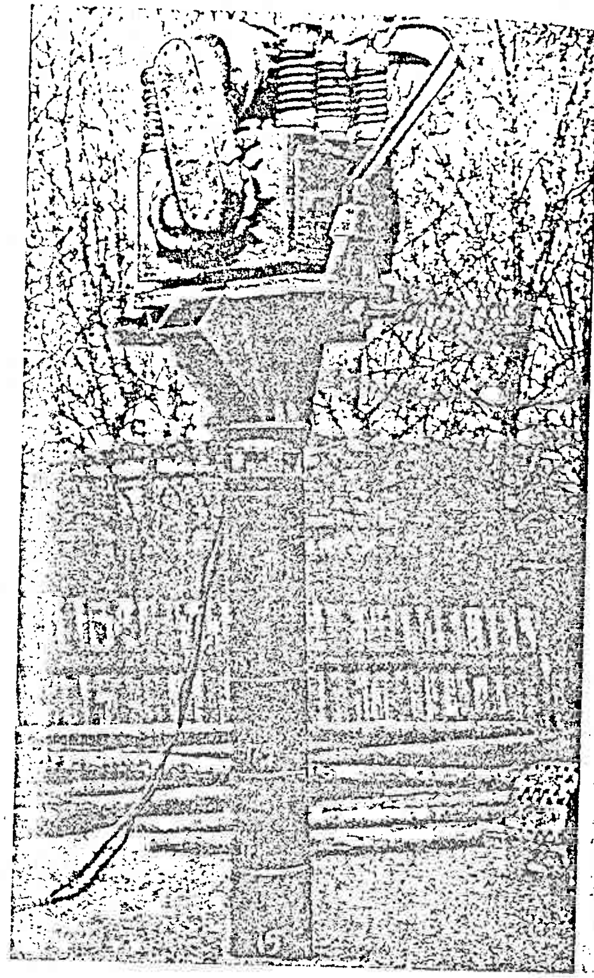

Fig. 5. General view of Type-PVN-1 longitudinal-rotary vibrating hammer.

According to the adopted assumptions, the velocity of the pipe after the impact is equal to

$$z'_{2y} = \frac{M_1}{M_1 + M_2} z'_{1y}, \tag{4}$$

in which $z'_{1y}$ is the impact velocity of the vibrating hammer.

The solutions of Eqs. (1)-(3), using the condition (4), by means of an analog computer, permitted constructing graphs representing the variation of the impact velocities and the penetrations after the blows, for different spring pressures $(cz_p)/P_0$ and different soil resistances $[\Delta = (R + F)/P_0]$.

An examination of the graphs in Fig. 2 indicates that the maximum values of the penetration after each blow are displaced with respect to the maximum impact velocities toward the side of increasing spring compression.

As the penetration resistance increases, the maximum values of the penetration become closer to the maximum values of the impact velocity.

For $\lambda/\omega$ = 0.1-0.3 and $Q_1/Q_2$ = 0.25, the optimum values of the spring compression lie in the range 0.3-0.5 for $\Delta$ = 8-1.5, respectively. An analysis of the oscillograms indicated that, under these conditions, the position of the eccentric weights at the time of the impact, corresponding to the maximum subsequent displacement of the vibrating hammer and the pipe, is characterized by an angle of 160-220°.

The above results were confirmed in a practical manner by constructing an industrial model of a BVS-1 free spring vibrating hammer [2, 3], which operated with negative gaps and exhibited good results when used for driving casing pipes at the Promburvod Trust (Fig. 3).

The vibrating hammer can be used for driving and extracting casing pipes in 100-m deep wells by means of UKS-22 and UKS-30 rigs. It has the following indices:

| | |
|---|---|
| Torque of eccentric weights of vibrator, kg-cm | 2500 |
| Speed of shafts of eccentric weights, rpm | 600 |
| Ratio of period of blows to period of rotation of shafts of eccentric weights | 1 |
| Capacity of electric motor, kW | 22 |
| Compressive force of operating springs, kg | 9600 |
| Rigidity of operating springs, kg/cm | 800 |
| Mass of vibrating hammer, kg | 2400 |

The vibrating hammer was used for driving pipes 325, 426, 539, and 630 mm in diameter, and 20 to 90 m long, through saturated and dense soils, at a rate of 0.5 to 2.0 m/min. When the winch for tightening the operating springs was connected, it became possible to drive the casing pipes 50-60 m into the soil.

Depending upon the nature of the soil and the degree of filling of the internal cavity with earth during the process of driving of a pipe, the operating springs were adjusted so as to obtain a force between 4 and 8 tons; this ensured a maximum driving rate for stable operation of the vibrating hammer in the regimen of "blow after return."

The use of the BVS-1 vibrating hammer made it possible to increase the rate of driving of the pipes by a factor of 2 and even more, to increase the penetration of the columns into the soil by 30%, and to reduce the number of changes from sludge pumping to driving by a factor of about 3.

The tightening of the springs by a hauling rope wound around a winch installed separately or on a drilling rig (movable unit) is not always technically convenient; it should correspond to the specific characteristics of the given type of work.

Another method considered for increasing the driving capacity of the vibrating hammers without increasing the impact velocity was the use of longitudinal-rotary vibrating hammers, the analytical scheme of which is shown in Fig. 4. In this case an additional factor taken into account is the rotary motion of the hammer–pipe system under the action of the torsional moment $P_0 r_1 \cos \omega t$.

During vibration-impact driving, the following resistance is overcome on the lateral surfaces:

$$F_z = F \frac{z_2'}{\sqrt{(z_2')^2 + (\varphi' r)^2}} \text{ along z axis,}$$

$$F_r = \frac{\varphi' r}{\sqrt{(z_2')^2 + (\varphi' r)^2}} \text{ along } \varphi\text{-axis}$$

in which r is the pipe radius, $r_1$ is the radius of application of the disturbing force, and $\varphi$ is the angular coordinate of the hammer-pipe system.

The equations of motion for this scheme are presented in [1].

An analysis of several recorded vibration-impact driving regimens indicated that in the presence of a rotary component, the velocity increases on the average by 10-15% after the blow.

For experimental work carried out on an area covered by clay soils having different consistencies, use was made of an experimental vibrating hammer, which had been set for longitudinal and longitudinal-rotary impact regimens, for driving pipes 168 mm in diameter and 10 m long [4].

A comparison of the effectiveness of the longitudinal and longitudinal-rotary vibration hammers indicated that for the same vibration parameters the limiting driving depth of the latter type of vibrating hammer was greater in all the tests. The value of the capacity per unit depth of driving for longitudinal blows ($W_2 = N_2/L$) is 1.5-4.5 times greater than the value of the capacity for longitudinal-rotary blow ($W_{2r} = N_{2r}/L$).

The substantial advantages of the longitudinal-rotary vibrating hammers were the bases for the development of an experimental model of a Type-PVN-1 vibrating hammer, for construction of bored piles up to 450 mm in diameter and 18 m long (Fig. 5), which has the following indices:

| | |
|---|---|
| Rated capacity, kW | 60 |
| Torque of eccentric weights, kg-cm | 7000 |

| | |
|---|---|
| Amplitude of disturbing force, tons | 32 |
| Amplitude of torsional moment with respect to the maximum force, ton-cm | 9.6 |
| Frequency of vibrations, min | 420, 520, 600, 645 |
| Total rigidity of springs, kg/cm | 59, 154, 250, 345 |
| Mass of ram, tons | 3.6 |
| Mass of entire installation, tons | 7.6 |

The tests on a PVN-1 vibrating hammer used for driving 426-mm diameter pipes in different soils ranging from soft-plastic clays to hard sandy loams, indicated its high driving capacity. (The mean rate of driving to a depth of 18 m was 0.8-1.5 m/min.)

Thus, the investigations indicated the possibility of a substantial increase of the driving capacity of vibrating hammers when used for pipes, by applying regulating tensions on the hammer springs, for large negative gaps and rotary vibrations of the driven pipes.

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