Theory and Development of Vibratory Pile-Driving Equipment

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ABSTRACT

This paper discusses three topics: 1) an overview of the development of vibratory hammers, along with a basic description of the equipment; 2) some basic theory of operation and driving capability; and 3) the development of a new series of vibratory hammers that operate at 40 Hz while retaining the drivability of lower frequency vibratory hammers.

INTRODUCTION

Vibratory hammers have been in the installation of deep foundations for about forty years now. They are especially useful for installing sheet piling walls and coals. As shown by Jonker (1967), this type of equipment is now more and more in use on shore projects, both with platforms and coastal or work.

This paper will discuss three topics: 1) an overview of the development of vibratory hammers, along with a basic description of the equipment; 2) some basic theory of operation and driving capability; and 3) the development of a new series of vibratory hammers that operate at 40 Hz while retaining the drivability of lower frequency vibratory hammers.

DEVELOPMENT OVERVIEW

Basic Equipment Description

A vibratory pile driver is a machine that installs piling into the ground by applying a rapidly alternating force to the pile. This is generally accomplished by rotating eccentric weights about shafts. Each rotating eccentric produces a force acting in a single plane and directed toward the centerline of the shaft. If we separate this force into components along each of the Cartesian axes, this becomes a sinusoidal force. The exact mechanics of this process will be described later in this paper. Although there are many variations in design and construction, the vast majority of vibratory hammers are of the configuration shown in Figure 1. Briefly, there are two main components of the system: the exciter, which produces the actual vibrating force, and the power pack, which provides the usable energy for the motor(s) on the hammer to spin the eccentrics.

Development in the U.S.S.R.

According to Schmidt and Hill (1966), the first vibratory pile driver used was in the Soviet Union, a model BT-5 developed and first used under the direction of D. O. Barkan. This hammer had a dynamic force of 214 KN and the eccentrics rotated at 2500 rpm, powered with 28 kW of power. Used in the construction of the Gorky Hydroelectric development, the hammer drove 3700 sheet piles 9-12 m long in 2-3 minutes each.

Erofeev et al. (1985) describe in detail both current Soviet vibratory equipment and some equipment produced elsewhere. As is the case in most of the world, Soviet made vibratory pile drivers can be divided into two groups:

1) Low frequency machines, with a vibratory frequency of 300-500 rpm, used primarily with piles with high mass and toe resistance, such as concrete and large steel pipe piles.

2) High frequency machines, with a vibratory frequency of 700-1500 rpm, used for piling such as sheet piles, small pipe piles, etc. This is the type of piling generally associated in the U.S. with vibratory pile driving.

With both types of machines, the Soviets have a range of equipment. This is outlined in Table 1.
2) For most conventional vibratory hammers, one can consider the entire system a rigid mass. This is because the relatively low frequency vibrations of most vibratory hammers do not bring the distributed mass and elasticity of the system into play. By definition, with the sonic pile drivers the resonant properties of the system become significant, and the analysis becomes more complicated.

With these assumptions in mind, we present Figure 4, which shows the basic setup for the rotating eccentric weights used in most current vibratory pile driving/extracting equipment. The weight is set off centre of the axis of rotation by the eccentric arm \( r \).

For a rotating body, the force exerted on the centre shaft is given by the equation

\[
F = m \cdot r \cdot \omega^2 \tag{1}
\]

If we define \( K = m \cdot r \), we can substitute to

\[
F = K \cdot \omega^2 \tag{2}
\]

If only one eccentric is used, in one revolution a force will be exerted in all directions. Giving the system a good deal of lateral whip. To avoid this problem, we pair the eccentrics so that the lateral forces cancel each other, leaving us with only axial force for the pile. We can also have several pairs of smaller, identical eccentrics synchronized and obtain the same effect as with one larger pair. Thus, we should interpret the 'm' term to mean the sum of all the eccentric weights, the eccentric arm length for each being equal.

Turning to the whole system, without considering the effects of gravity, the equation of motion is

\[
M \cdot \ddot{x} + C \cdot \dot{x} + F \cdot \sin(\alpha + \phi) = 0 \tag{4}
\]

The solution of this equation is

\[
x = K \cdot \sin(\alpha + \phi) / (M \cdot \sqrt{(1+2\gamma)}) \tag{5}
\]

where the dimensionless quantity \( \gamma \) is

\[
\gamma = C / (M \cdot r) \tan(\alpha) \tag{6}
\]

and \( \phi \) is the phase angle of the amplitude.

Generally speaking, many of the traditionally measured quantities for vibratory hammers such as amplitude, acceleration ratio, etc., are computed for the "free-hanging" case, i.e., with only the mass of the system taken into account and no soil resistance. Under this assumption \( \gamma = 0 \). In any case, from equation (4) the maximum displacement is

\[
x_{\text{max}} = K / (M \cdot \sqrt{(1+2\gamma)}) \tag{7}
\]

Since the acceleration, velocity, and displacement of the system solved from Equation (4) are all sinusoidal with respect to time, this quantity is measured from the zero line of the sine wave. Customarily, the maximum cycle displacement of the vibrator, called the amplitude, is measured from peak to peak and is expressed as

\[
A = 2 \cdot x_{\text{max}} \tag{8}
\]

The instantaneous torque driving the eccentrics is

\[
T_{\text{inst}} = F \cdot r \cdot \omega / (2 \cdot M \cdot \sqrt{(1+2\gamma)}) \tag{9}
\]

The maximum instantaneous torque is

\[
T_{\text{max}} = F \cdot r \cdot H / (2 \cdot M) \tag{10}
\]

where \( H \) is the quantity

\[
H = (1+2\gamma)^2 \tag{11}
\]

and this equation is graphically represented in Figure 5. It is important to note that \( H \) is never greater than approximately 1.293; this maximum value takes place when \( \gamma = 0.5 \) and \( \alpha = 60^\circ \).

Looking forward to the power requirements, normally one would use a root mean square (rms) value to match an application to a motor, so

\[
T_{\text{rms}} = T_{\text{max}} / (2) \tag{12}
\]

From the torque the power is simple to compute, given by

\[
P_{\text{rms}} = T_{\text{rms}} \cdot \omega / 1000 \tag{13}
\]

This power is ideal and does not take into account frictional losses of any kind. From the equations above, the minimum power required for a vibratory hammer for all conditions is

\[
P_{\text{min}} = F^2 / (2 \cdot 177 \cdot \gamma \cdot M) = K^2 \cdot \gamma^2 / (2 \cdot 177 \cdot M) \tag{14}
\]

Similar formulations appear in Schmid and Hill (1968) and Erofeev et al. (1985).

One more important formula is that for the peak acceleration experienced by the vibratory system, which would be

\[
\alpha = F / (N \cdot g \cdot (1+2\gamma)) \tag{15}
\]

Discussion

Now that we have established some basic theory of operation, we must now apply this to actual equipment design.

Looking at the whole process of vibratory pile driving from the thixotropic standpoint, it would seem that the best way to design vibratory pile driving equipment is to design a unit that would induce the maximum property change in the soil to facilitate driving. Traditionally, however, this has not been the primary consideration of most vibratory hammer manufacturers. Instead, the favoured approach taken has been the "amplitude drives piling" approach. This concept is most succinctly stated by Erofeev et al. (1985): "The amplitude of the system's vibrations is decisive for the insertion for insertion of the pile. At a low vibrational amplitude, displacement of the soil with respect to the side surface of the element being inserted does not exceed the limit of its elastic deformation and the pile is not sunk into the ground."
As the amplitude of the vibrations increases, residual deformation of the soil occurs and the pile begins to slip relative to the soil, i.e., it is sunk into the ground. Mathematically put, this means that

\[ x_{\text{max}} = q \]  \hspace{1cm} (16)

and this is shown graphically in Figure 6a. This principle, of course, is also applied in wave equation analysis of impact driving, as no set of the pile is possible without exceeding the elastic limit, or quake, of the soil.

While the concept described above has many merits, we do not feel that it tells the whole story in vibratory driving for the following reasons:

1) Thixotropy is a molecular process. For instance, considering a typical standard such as the one described by Sadler (1988), we cannot see how an amplitude requirement such as

\[ x_{\text{max}} = q \]  \hspace{1cm} (17)

will influence such a process.

2) The concept assumes that the soil quake is more or less a fixed property of the soil. Especially with shaft friction, to some degree this concept has also been current in wave equation practice. In recent years, though, soil models have been developed that show the spring constant of the soil to be a function of both basic physical properties of the soil and pile geometry as well, as shown by Mitwally and Novak (1988). Thus, if the soil resistance is decreased with thixotropic change, with a constant soil spring rate both the quake and the amplitude needed to enter the plastic region is likewise decreased. The effects of both constant soil spring constant and variable quake are shown in Figure 6b.

Based on this, we propose that effective vibratory hammers can be developed that meet, among others, the following two requirements:

1) They must have sufficient power to effect thixotropy and to maintain sinusoidal vibration. On the face of it, this could lead to infinite power requirements. Fortunately, as we see in our theory above, because of the upper limit of \( N \), there exists for any amount of damping -- and by extension static capacity -- an upper limit of power requirement for a machine of a given size. Thus, the resistance the hammer experiences may increase, and the penetration slow down, but adding more power will not help matters. Among these lines, Billet and Siffert (1985) show that increasing the power for a given size of vibratory driver (in their case, a laboratory unit driving through sand) increases the speed of penetration. Additionally, though, their results also show a "diminishing returns" type of result with an increase in power.

2) The power transmitted to the soil must be done in an efficient manner from a high energy source through the pile-soil interface to a low one in the soil. As we have pointed out above, large displacements will not necessarily accomplish this. This leaves us with high velocities or accelerations, and to achieve the former we must have the latter. Thus, the quantity "n" must be sufficiently high. Based on experience, for at least the steel non-displacement piles normally driven with a vibratory hammer, we normally design equipment to drive systems with a ratio of accelerations in the range of 9-18. This also establishes the relationship between the dynamic force and the optimum maximum system mass.

Another factor which has not been given much attention is that of bias weight. As we said before, once thixotropic transformation has taken place, the resistance of the soil decreases to the point where the pile drops by its own weight. Obviously, the more weight applied, the more rapid the drop, especially if the weight can be applied outside of the vibrating mass and thus not drawn down the system acceleration. This is done in the deadweight of the suspension system. Oppositely, for extraction a high crane pull is important, and this is facilitated by a suspension that is both effective in vibration dampening and high in static pulling capacity. The effects of bias weight on driving are not well understood at this time.

Relating all this to the theory presented, and taking into account the theory's limiting assumptions, we conclude that the theory presented takes into account most of the basic requirements of a vibratory hammer. We now proceed to the actual equipment development.

**EQUIPMENT DEVELOPMENT**

**Conventional Vibrators**

Vulcan began manufacturing vibratory pile hammers in 1984. In the beginning, the principal design objective was to produce equipment which could be competitive in performance with existing equipment that was and is on the market. With this in mind, the equipment developed was powered with a minimum ratio of accelerations of approximately 9. The eccentrics were configured to rotate at 1600 rpm, which was typical for most American manufactured vibratory pile driving equipment. This effort resulted in three sizes, the 1150, 2300 and 4600, with maximum dynamic forces of 371.7, 743.4, and 1486.9 kN respectively. These machines have performed quite satisfactorily in the field on land based piling normally driven or pulled, such as sheeting, caissons, and wood piles.

Although the development of these units has produced a very competitive line of vibratory hammers, Vulcan felt some advances were necessary in the equipment to maintain an advantage. The types of advances desired were as follows:

1) Reduce potential transmitted vibrations to adjoining structures. This is not a major consideration for marine structures but for land based units a reduction in transmitted energy can be important in certain applications.

2) Reduce mechanical complexity of equipment.

3) Reduce cost of equipment, related to (2).

**New Units**

As unusual as our thinking was about the nature of vibratory pile driving, the risks involved in producing
new equipment were considerable. The advent of aluminum sheet piling, popular in some areas for marinas and other light work, gave us an opportunity to test our theories with a minimum of investment. The first result of the application of the theories was the Vulcan 400 vibratory pile hammer. This unit produces 151 kN of dynamic force with two eccentrics rotating at 2400 rpm (40 Hz). Using this configuration, we were able to accomplish the following:

1) Produce comparable dynamic force to slower machines with less eccentric moment, thus allowing smaller, more compact eccentrics for the dynamic force. This was important to make a compact machine. It may also be important when large hammers for offshore driving are designed, as it will allow large forces to be generated with relatively small eccentrics.

2) Direct drive the eccentrics with the motor. This eliminates the pinion gear, which both simplifies the construction and adds to the efficiency of the unit. The speed of 2400 rpm was chosen in part because it is a good, general upper bound for the continuous operation of a large number of fixed-displacement hydraulic motors, both vane and piston.

3) Combine the eccentric and gear into one piece. This eliminates difficulties with fasteners holding a gear-eccentric assembly together.

4) Reduce potential vibrations to other structures near vibrodriving.

The first job a 400 was used on was in September 1967, installing aluminum sheet piling at a marina in Ft. Pierce, FL. The hammer performed satisfactorily and the piling were installed without difficulty. Some subsequent performance of this machine is described by Garrett (1988).

Following the success of the 400, the size 1400 was developed. This 437 kN dynamic force machine is designed to comparable to the 1150. It was first used in the field in April 1989, and has been used to drive steel piling and extract wood ones. This hammer Town in Figure 7 (the 400 is very similar to this).

Although no rigorous research has been done on the performance of this hammer, tests in the field have shown that the hammer is in most cases comparable in performance to the 1150 and other hammers similar in size. Other, larger sizes of this type of hammer are both in production and in the planning stages.

One interesting footnote to the development of the 40 Hz machines documented above concerns the research of Mr. E. Jonker and M. G. Jonker (1988). In the process of developing a wave equation program to analyze the driving of vibratory piles, they studied the driving of an ICE 216 hammer, normally a 325 kN hammer operating at 26.5 Hz, sped up to 38.3 Hz with dynamic force consequently increased to 670 kN. Although the calculated amplitude of the machine would not increase due to the constant eccentric moment, while driving a 20m long Armed 8U-12 sheet pile, amplitude magnification was observed with the higher frequency, in some instances almost doubling the computed amplitude. This is probably a pile resonance effect. It also indicates probably magnification of both velocity and acceleration. These resonance effects add both performance to the hammer/pile system and complexity to the analysis.

CONCLUSION

Both the development of the theory and the equipment have shown the following:

1) The driving of vibratory piles requires more of a machine than to just produce large amplitudes. The driver and pile in conjunction with each other must effectively transmit to the soil energy to effect thixotropy. This will lead in turn to the liquefaction of the neighboring soil and then to the reduction of instantaneous static capacity necessary to allow the weight of the hammer/pile system to force the pile into the ground, or the pull of the crane to effect extraction.

2) Once the system is set up to effect energy transmission, sufficient power is required to transmit energy to the soil. This is a finite quantity for a given system.

3) The development of 40 Hz machines results in vibratory pile drivers that are effective, economical, and mechanically simple.

NOMENCLATURE

A = Vibrator amplitude, m
C = Soil Dampening, N-s/cm
F = Dynamic Force, N
G = Soil Spring Constant before thixotropy, N/cm
k = Soil Spring Constant after thixotropy, N/cm
K = Eccentric constant, kg
m = Vibrating mass, kg
n = Eccentric mass, kg
p = Ratio of accelerations
Pmin = Minimum required power, kW
Prms = Root mean square power, kW
Qmax = Maximum torque, N-m
r = Eccentric moment arm, m
R = Soil Static Capacity before thixotropy, N
R' = Soil Static Capacity after thixotropy, N
T = Instantaneous torque, N-m
Tmax = Maximum torque, N-m
Trms = Root mean square torque, N-m
x = Displacement, m
x' = Acceleration, m/sec²
xmax = Maximum displacement, m
φ = Phase angle, radians
ζ = Damping Coefficient
β = Eccentric rotational speed, rad/sec

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REFERENCES


Fig. 2—Soviet vibratory pile driver, Leningrad.

Fig. 3—VPM-170 vibratory pile driver (after Erolev).
Figure 4  Eccentric Diagram for Vibratory Hammers

Figure 5  Values of "H" vs. Phase Angle
Figure 6 Vibration-Soil Response Diagrams

Figure 7 2400 rpm (40Hz) Vibratory Hammer