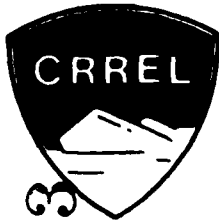


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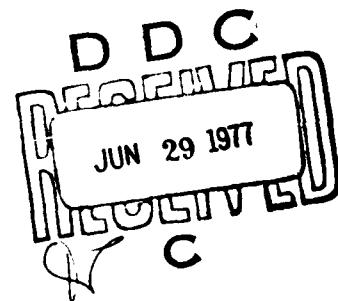
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Special Report 77-13

STAKE DRIVING TOOLS A Preliminary Survey

Austin Kovacs and Ronald T. Atkins

May 1977



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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) The military has frequently encountered difficulty in installing and removing anchors in the field where hard frozen ground exists. This difficulty has often precluded rapid deployment of military hardware requiring anchors for their stability or support. This report gives results of a study of four commercial breaker-rock drills, a prototype hydraulic stake driver-receiver and a prototype propellant-actuated hammer which were evaluated for the driving of anchors into hard frozen ground. The tests found that commercial		

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20. Abstract (cont'd)

breaker-rock drills can be used without modification to drive standard military GP-112/C and GP-113/G stakes into frozen ground. The study revealed that, while the hydraulic stake driver would require further development to increase its reliability, it could drive the above stakes into frozen ground. The propellant-actuated stake driver was found incapable of driving stakes into hard frozen ground and was not considered worthy of further development as a stake driver.

PREFACE

This report was prepared by Austin Kovacs, Research Civil Engineer, Foundations and Materials Research Branch, Experimental Engineering Division, and Ronald T. Atkins, Supervisory Electronics Engineer, Technical Services Division, U.S. Army Cold Regions Research and Engineering Laboratory (USA CRREL).

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CONVERSION TABLE

<u>Multiply</u>	<u>By</u>	<u>To Obtain</u>
foot	30.48*	centimeter
foot/second	30.48*	centimeter/second
inch	2.54*	centimeter
horsepower	1.014	horsepower (metric)
pound	0.4536	kilogram
foot-pound	0.1383	kilogram-meter
pound/foot ³	16.02	kilogram/meter ³

*Exact

INTRODUCTION

Rapid deployment of military hardware such as artillery base plates, communication towers, and a wide variety of structures requires the provision for expedient anchorage. This provision is often difficult, if not nearly impossible, for troops to achieve when operating over frozen ground and has made necessary the investigation of numerous devices which would be capable of driving anchors into a wide range of soil types including frozen ground.

In the early 1960's, considerable effort was made to evaluate commercial tools and to develop new concepts for the driving into, and retrieval of anchors from, frozen and unfrozen ground. Studies were conducted by, or performed under contract to, such U.S. Army agencies as Rock Island Arsenal⁴, Natick Laboratories¹, Frankford Arsenal⁶, and the Electronics Research and Development Laboratory^{7,8,9}. In the late 1960's, additional work² was undertaken by the U.S. Army Engineer Waterways Experiment Station² and laboratory evaluation of a resonant stake driver concept was funded by Natick Laboratories³. Overall, the drivers considered ranged from the simple to the exotic, as seen from the following list:

1. Sledge hammers
2. Impact hammers
 - a. gasoline drivers
 - b. diesel powered
 - c. pneumatic
 - d. electric
 - e. propellant actuated
3. Augers
 - a. hand turned
 - b. gasoline driven
 - c. pneumatic
 - d. electric
 - e. rocket propelled
4. Resonant drivers
 - a. eccentric weight oscillator
 - b. pneumatic piston actuator
 - c. hydraulic piston actuator
 - d. electroviscous fluid actuator
 - e. magnetic fluid actuator
 - f. piezoelectric exciter
 - g. electrostatic exciter
 - h. electromagnetic exciter
 - i. magnetostrictive exciter

In addition to the above, the shaped charge was evaluated for producing in frozen ground holes in which stakes could be inserted or easily driven. The vacuum plate concept, adhesive bonding, and heated stake (for thawing frozen ground) were mentioned by some investigators but not seriously considered.

None of the concepts studied have, to date, become line items except for the sledge which, although long a tool of the military, has proved to be inadequate for driving anchors in frozen ground.

In 1969, CRREL initiated a study titled, "Military Anchorages in Frozen Materials." It quickly became apparent that sustained interest by user agencies still existed as evidenced by continuing requests for information on methods for driving anchors in frozen ground. It was decided to make another attempt to fill this need with a man-portable tool. From a review of the findings of the previous studies, it was decided to test newer, more powerful versions of commercial equipment previously evaluated and rejected to determine if these improved tools could satisfactorily drive the military GP-112/G and GP-113/G stakes and the 3, 4, 6 and 8-in. arrowhead anchor into frozen ground. In addition, it was decided to try the MXG321()/G (Ballistic Hammer) developed for the Electronics Research and Development Laboratory as this device appeared to be capable, not only of driving, but even more importantly, of retrieving GP-112/G stakes from frozen ground. The findings of a literature survey by the investigators also indicated that a high-frequency hydraulic piston-actuated stake driver-retriever would have a very high chance of success. A contract was therefore let covering the development of a first-generation, hydraulic stake driver-retriever.

This report covers the ability of the Ballistic Hammer, several commercial breaker-rock drills, and the first-generation hydraulic stake driver-retriever to drive anchors into frozen and unfrozen ground. The tools evaluated may be classed as impact drivers. They differ from one another in the way the ram velocity is developed. The hydraulic stake driver-retriever's ram is oscillated by hydraulic pressure. The self-contained internal-combustion-engine breaker-rock drills and the electric rock breaker develop blow energy from the velocity of a floating ram propelled by an internal piston. The Ballistic Hammer's ram is accelerated from the ignition of powder in a 30-caliber grenade launcher cartridge blank.

BALLISTIC HAMMER

The MXG321()/G Ballistic Hammer was developed under contract to the U.S. Army Electronics Research and Development Laboratory, now the

U.S. Army Electronics Command, by Aircraft Armaments Inc.^{7, 8, 9} An assembly drawing of the hammer is shown in Figure 1. The upper body of the hammer incorporates a standard military 30-caliber carbine action without the stock and the majority of the barrel. In the Ballistic Hammer, high pressure gas, resulting from the firing of grenade launcher cartridge blanks, propels the 4-lb hammer to a velocity v of 116 ft/sec against the head of a stake. The corresponding kinetic energy K per blow is:

$$\begin{aligned} K &= \frac{1}{2} m v^2 \\ &= 0.5 (4/32.2) 116^2 \\ &= 835 \text{ ft-lb} \end{aligned}$$

where: m = hammer mass.

Repetitive dynamic loadings are achieved by successive firings until the stake has been fully driven or the cartridge clip has been emptied. Extracting a stake requires a retriever adapter, which must be locked onto both the stake head and the base of the driver. Figure 2 shows several views of the Ballistic Hammer as it is used to drive and retrieve GP-112/G and GP-113/G stakes (Fig. 3, 4). Note the adapter used during stake retrieval and the awkward position the operator must assume during its use as shown in Figure 2b. A closeup of the hammer on top of a partially driven GP-112/G stake is shown in Figure 5.

The Ballistic Hammer was first tried in unfrozen silt behind the CREEL facility in Hanover, New Hampshire, on 10 December 1969. The silt had an average bulk density of 110 lb/ft³ and a moisture content of 22%.

The hammer was used to drive the GP-113/G stake vertically, as illustrated in Figure 2a. Four stakes were driven, each to a depth of 3 ft. The average driving time was 3 min. An average of 6 clips or 90 rounds of blank cartridges was used to drive each stake. Jamming of spent cartridges occurred approximately twice during the driving of each stake. Had this not occurred, it is believed that the stakes could have been driven in about 2-1/2 min.

The stakes were extracted with the Ballistic Hammer in an average of 1 min 40 sec using an average of 50 rounds. While the driving of each stake was relatively effortless, the retrieval of the stakes was very fatiguing because the driver had to be held against the stake lip by the operator during the exercise. This undesirable aspect, together with the dynamic forces transferred to the operator and the large number of blanks needed to both drive and retrieve GP-113/G stakes from unfrozen ground, resulted in a lack of enthusiasm for the tool. However, one aspect found valuable was the hammer's ability to extract the GP-113/G stakes. The stakes could not be removed with a sledge hammer slamming against the side of the stake as is often the practice of troops in the field.

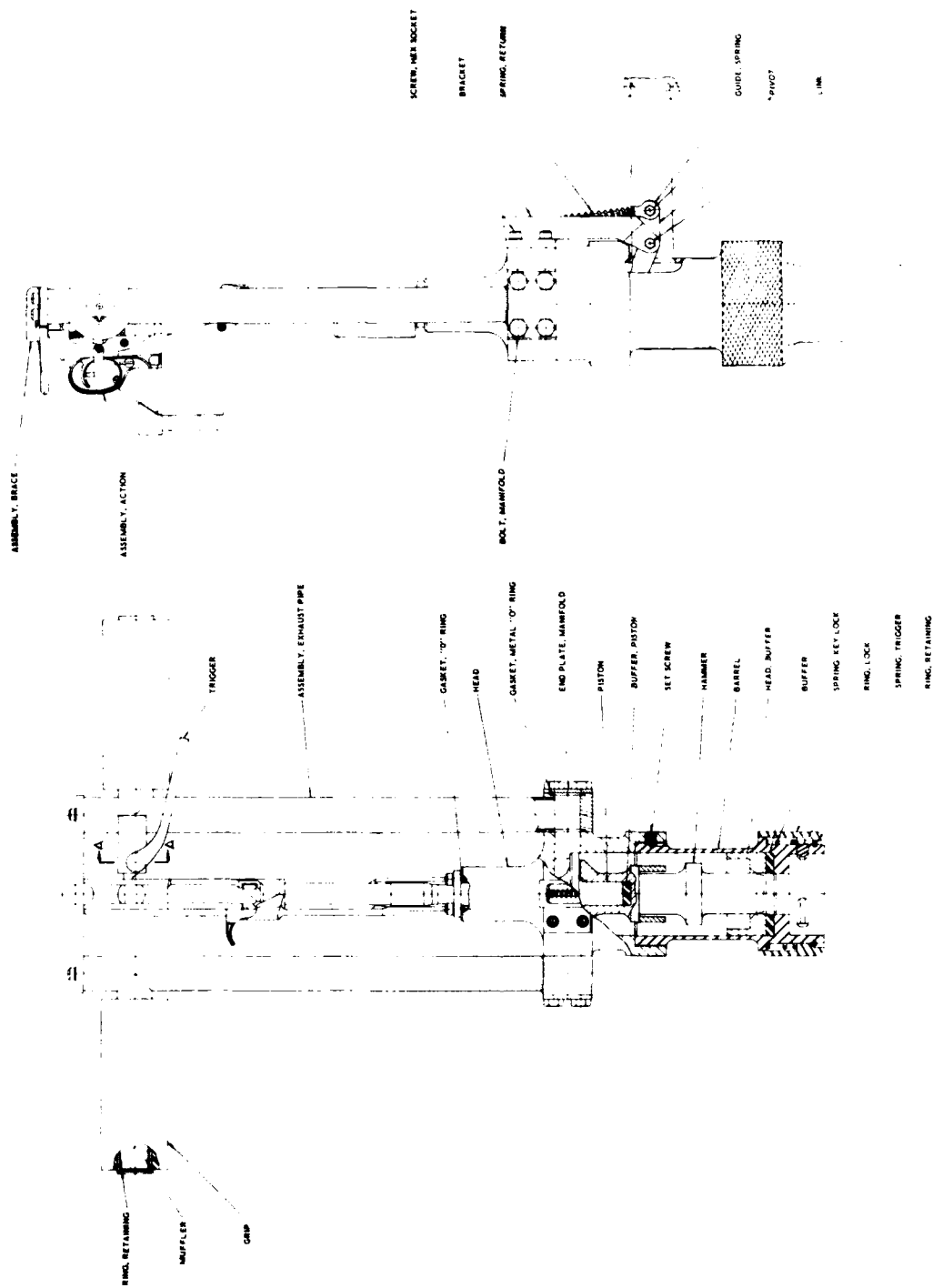
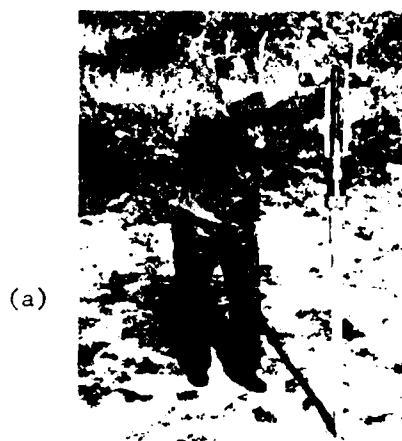


FIGURE 1. Assembly drawing of the Ballistic Hammer (emplacement mode)



(a)



(b)



(c)



(d)

- a. Emplace Stake GP-113/G
- b. Retrieve Stake GP-113/G
- c. Emplace Stake GP-112/G
- d. Retrieve Stake GP-112/G
- e. Emplace Stake GP-113/G at approximately 30°



(e)

FIGURE 2. Typical employment modes for the Ballistic Hammer



FIGURE 3. GP-112/G ground stake

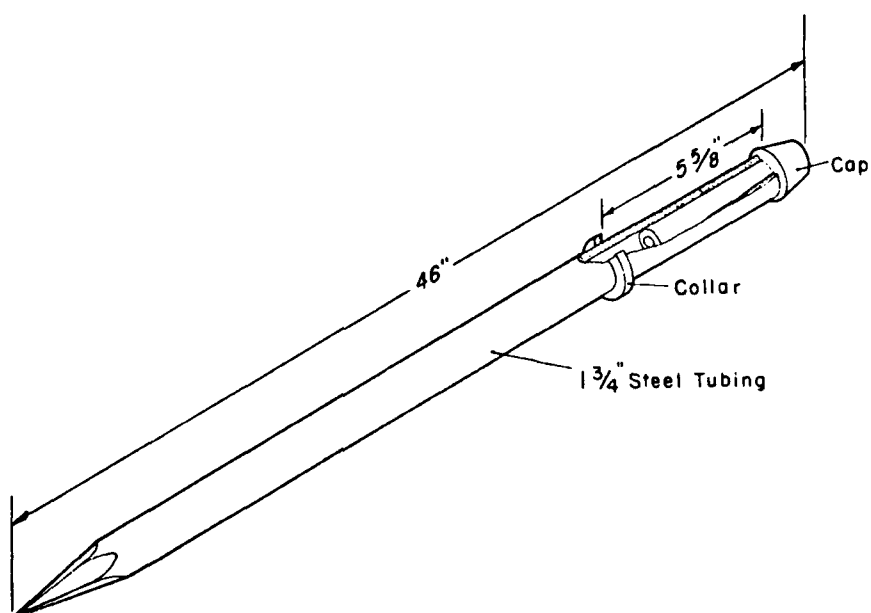


FIGURE 4. GP-113/G ground stake

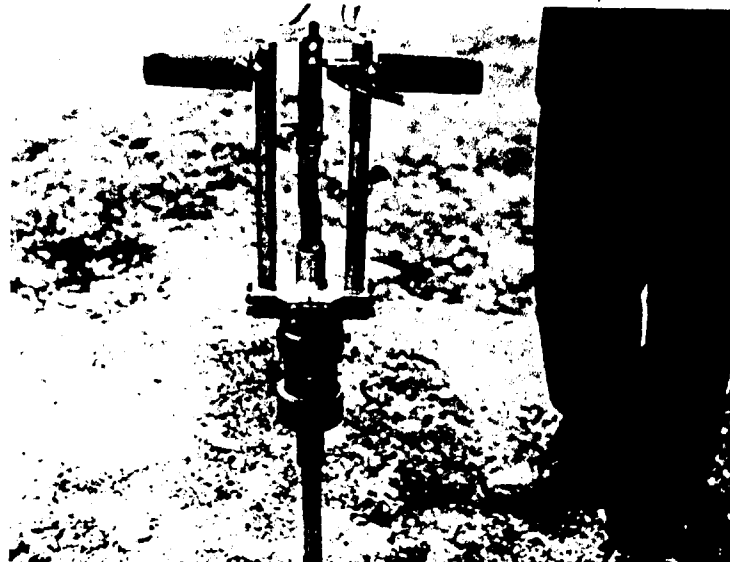


FIGURE 5. Ballistic Hammer set for driving on top of a GP-112/G stake

In April 1970, the hammer was field-tested in Fairbanks, Alaska. The hammer was used to drive GP-112/G stakes into -9°C silt (bulk density 88 lb/ft^3 , dry density 60 lb/ft^3 and moisture content 47%). When the stake reached a depth of 9 in., the hammer when fired would simply bounce off the stakehead, imparting painful forces on the operator without any additional stake penetration occurring. Reaching this depth took 2-1/2 min of driving time and the expenditure of 55 rounds. As a comparison, the same size stake was driven its full 11-in. embedment depth in less than 1 min. using an 8-lb sledgehammer.

In short, because of its inability to drive the GP-112/G stake more than 9 in., the fatiguing forces placed on the operator, and the large logistic propellant requirements, the Ballistic Hammer is not recommended for further consideration as a stake driver in frozen ground. However, for a stake retriever, the propellant extraction principle seems worth pursuing. The stakes driven by the Ballistic Hammer and the 8-lb sledge could not be broken loose and removed after 5 min of abusive lateral impact from the 8-lb sledge. The Ballistic Hammer, however, was capable of removing a GP-112/G stake, embedded 11 in. in the frozen silt, in 9 sec, using eleven 30-caliber carbine blanks. Where stakes embedded in frozen ground must be rapidly removed from artillery base plates, to allow quick redeployment, a lightweight propellant-actuated device should be considered. First generation propellant-actuated stake retrievers have been fabricated by Frankford Arsenal⁶ and by Aircraft Armaments Inc.^{7,8} Further work in this area could lead to a GP-112/G frozen-ground stake extractor weighing about 10 lb.

GASOLINE-DRIVEN PAVEMENT BREAKERS

Gasoline-driven pavement breakers are self-contained 2-cycle internal-combustion-engine machines, which typically weigh between 29 and 65 lb. Built of lightweight, high-strength materials, they are manually portable and are now used worldwide for such tasks as performing demolition work, exploring soil, compacting soil and ballast, driving small piles, and breaking frozen ground. Many of the machines now come with a switch-over lever which permits the unit to be quickly changed from an impact-breaking mode of operation to an impact-rotation mode for rock drilling. The tools can be typically operated at 45° above the horizontal.

Pionjar BR-80

The first machine tried was the Swedish-built Pionjar BR-80. A cut-away view of the tool is shown in Figure 6. By studying this drawing one can quickly grasp the overall assembly and mechanical operation of the tool. The unit has a dry weight of 62 lb and is rated at 26.9 ft-lb per blow maximum.

This machine was demonstrated at CRREL on 19 Nov 1969. It was first used to drive 1-3/4-in.-diam GP-113/G stakes into unfrozen Hanover silt to a depth of 2-1/2 ft. Driving time varied from 60 to 70 sec.

At the same location a GP-113/G stake was driven with an 8-lb sledge to the same depth. The stake was driven by an experienced man in 2 min 20 sec at a constant rate of 1 blow each 1.5 sec. A total of 93 blows were required. At the end of the sledge driving test, the head of the stake was found to be deformed, whereas the stake driven with the Pionjar hammer was undamaged. While driving the GP-113/G stake with the sledgehammer took only twice as long as it did with the Pionjar, it was quite obvious that the sledgehammer operator was tired and could not be expected to repeat the performance without a rest. The Pionjar could be expected to undertake repetitive stake driving without delay. It should also be pointed out that an inexperienced man could not be expected to wield the sledge as fast as an experienced one and therefore could not drive the GP-113/G stake as quickly.

Once driven, the stakes could not be loosened for removal by hitting them on their sides with the 8-lb sledge. A pick and shovel were required to remove the earth around each stake in order to free it. This points out that while stakes can often be driven into hard ground with a sledgehammer or by a mechanical device in a short time, the task of removing them is not as easy.

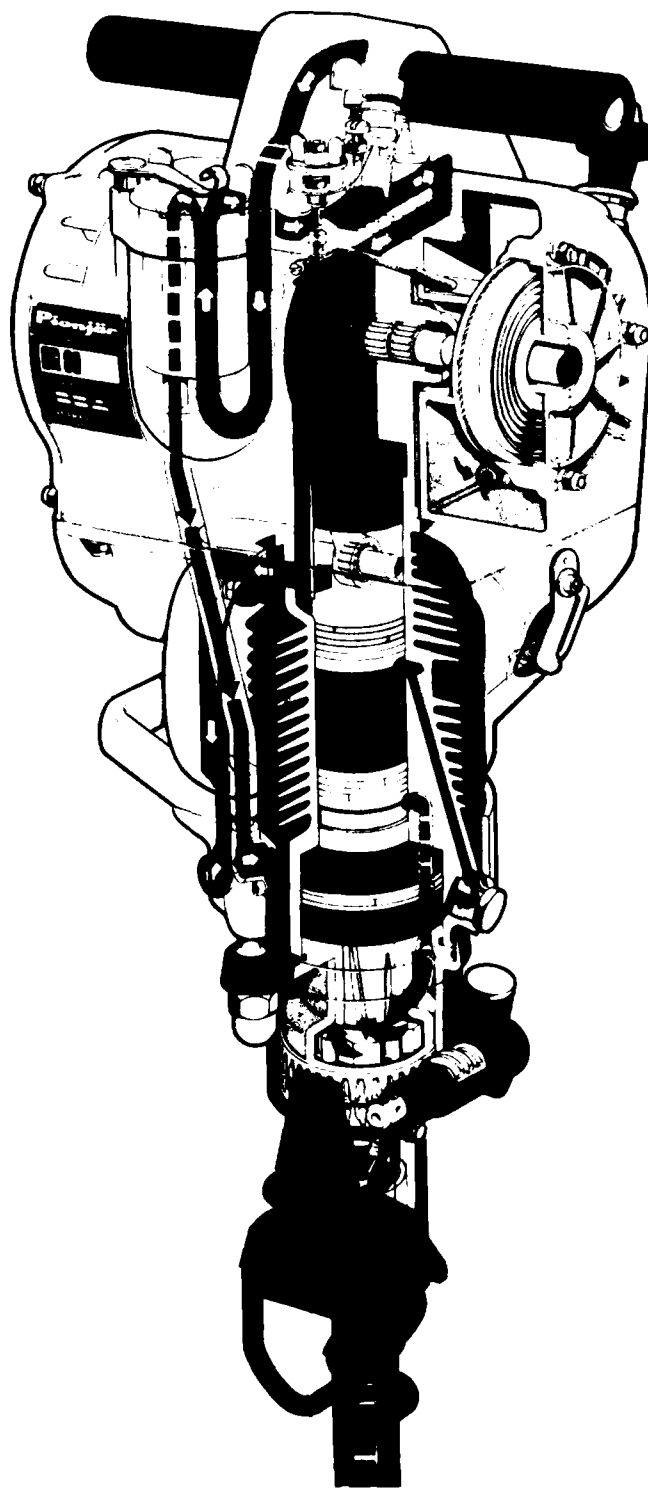


FIGURE 6. Cutaway view of the Pioneer BP-80 breaker-rock drill.
(Manufacturer, Bergman Borr, Solna, Sweden)

The Pionjar BR-80 was used next to drive GP-112/G and GP-113/G stakes into Manchester silt (bulk density 113 lb/ft³, water content 16%, frozen to 20°F). The GP-112/G stakes were driven 12 in. in 10 to 15 sec. During this driving, the soil around the stakes was thawed as a result of friction developed between the soil and the dynamically loaded stake. The thawed soil was gradually forced up to the surface as the stake was driven forward to occupy the space left by the displaced soil. When the soil refroze around the stakes, they became firmly locked into the soil and could only be removed by thawing of the soil.

The Pionjar BR-80 was also used to drive GP-112/G stakes into containers of saturated Ottawa sand (bulk density 123 lb/ft³) frozen at -20°F. Samples of saturated Ottawa sand were selected for the test because it was believed that if GP-112/G stakes could be driven into this highly resistant material at -20°F, they could be driven into virtually all frozen nonrocky soils that troops might encounter in the field. The stakes were driven in less than 1 min. Again, thawed material oozed up along the stake onto the surface while the stake was being driven.

Another anchor driving performed with the Pionjar BR-80 was as follows. At CRREL, 1-5/8-in.-OD, 1/8-in.-wall open-end pipe was driven through a layer of 1-in. crushed stone fill (1 to 4 ft thick), then through ≈2 ft of frozen silt into unfrozen silt. Pile lengths varied from 3 to 7 ft depending upon the depth of crushed stone to be penetrated to allow ≈2 ft of pipe embedment in the silt. The pipes were used to provide lateral stability to a 4x6-in. timber sill upon which a large portable ATCO steel building was erected. Most of the pipe anchors were driven without difficulty in less than 2 min. Some pipes, however, encountered a rock or other obstacle which increased driving time two to three times and on occasion required a second or third man's weight pulling down on the Pionjar hammer.

Several 3/4-in.-diam, 4-ft-long reinforcing bars were also driven through the material described above. In each case, the bar was easily driven in about 1 min.

Also, at the CRREL site, 6-in.-wide arrowhead anchors were vertically driven into the frozen silt. A sketch showing the general configuration of these anchors and how they are normally installed and often used is presented in Figure 7. The anchors quickly penetrated the stone fill but slowed abruptly when contact was made with the frozen silt. In fact, 5 min of hard driving was often necessary to drive these anchors to a depth of 18 in. into the frozen silt. Because of anchor cable limitations, the anchors were not driven through the frozen layer. As a result, when the anchors were preloaded the frozen soil would not allow them to turn to a horizontal plane as required. Instead, the anchors simply slid upward in

the soil cavity formed during their penetration. Therefore, these anchors could not be preloaded to fully utilize their holding capacity until the soil thawed in the spring.

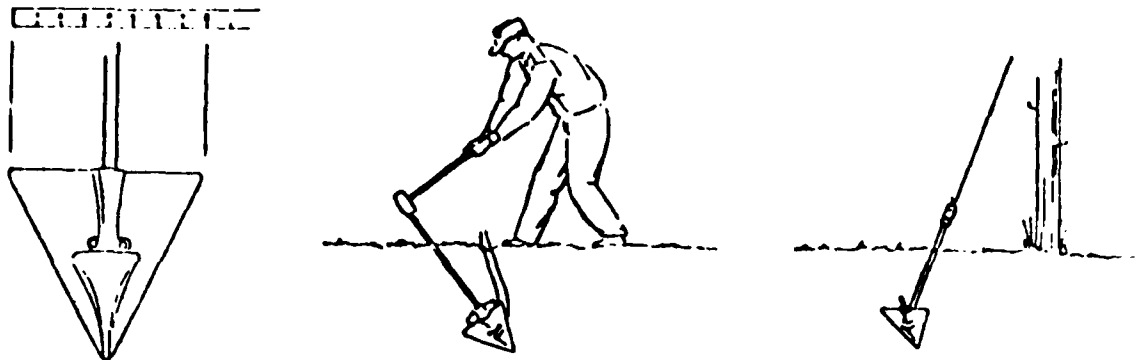


FIGURE 7. The arrowhead anchor

The tests using the Pioneer BR-80 as a rock drill were minimal. It was found that the tool could drill a 1-1/4-in.-diam. hole in Barre granite at the rate of 6 in./min. The unit was found unsuitable for drilling holes in frozen ground because the soil at the tip of the drill rod would turn to mud and could not be blown out of the hole by the air leaving the tip of the drill rod.

The Pioneer BR-80 has the capability of drilling holes in rock and this is important to the military. Troops can drive stakes in hard ground with these tools and can drill holes in ledge for the insertion of stakes or rock bolts to tie down artillery base plates, towers, etc.

The last anchor driving trials using the Pioneer were made in 25°F silt (bulk density 110 lb/ft³) at a Fairbanks, Alaska, test site. Both pipe and arrowhead anchors were driven. One pipe type was a 1/4-in.-wall, 2-in.-diam. casing open on each end; the second pipe type was a 1-1/2-in.-diam. open-end electrical conduit. The 2-in.-diam. pipe was consistently driven to a depth of 3 ft in approximately 5 min, while the 1-1/2-in.-diam. conduit was driven 7 ft at a rate of 1 ft/min.

Three-, four-, six- and eight-in.-wide arrowhead anchors were driven to a depth of 2 ft in average times of 0.75, 1.05, 1.4 and 3.9 min respectively. It took 3 min to drive a 6-in. arrowhead anchor to a depth of 3 ft. It is clear from these elapsed times that a considerable amount of energy was expended in driving these anchors. The energy may be calculated as follows: the engine runs at 2500 rpm, which is also the internal hammer

blow rate. The tool is rated at a maximum of 26.9 ft-lb/blow. In 1 min. the tool can put out a maximum of 67,250 ft-lb of energy under hard driving conditions. To drive the 3-, 4-, 6- and 8-in. arrowhead stakes to a depth of 2 ft may have taken respectively 50,400, 70,600, 94,200 and 262,300 ft-lb of energy. However, these values are undoubtedly high when energy losses are taken into consideration. The actual values may be as much as 50% less. These values clearly show that a large expenditure of energy is necessary to drive arrowhead anchors in frozen ground and help to explain why troops find the driving of stakes in frozen ground with sledge hammers both difficult and very tiring.

Atlas Copco Cobra

The Cobra is another Swedish-made breaker-rock drill. It weighs 56 lb, or 6 lb less than the Pionjar BR-80, and is more compact; however, it does produce several foot-pounds less impact energy. A cutaway view of this tool is shown in Figure 8.

The Cobra tried was not operating well on the day it was used. Nevertheless, it is fair to say that the tool was not capable of matching the Pionjar blow for blow in driving GP-112/G stakes into saturated Ottawa sand (bulk density 123 lb/ft³) frozen at -5°F or saturated Manchester silt (bulk density 116 lb/ft³) frozen at 25°F. Indeed, this would not be expected because, as pointed out above, the Cobra is a lighter, less powerful tool than the Pionjar.

The Cobra was able to drive GP-112/G stakes into the frozen Ottawa sand in about 1 min and into the frozen Manchester silt in under 20 sec. These rates are considered quite acceptable.

Maruzen Mini-75

The Maruzen Mini-75 is made in Japan and is the lightest breaker-rock drill tried. The unit has a dry weight of 29 lb and can be operated at any angle. A cutaway view of this tool is presented in Figure 9. The Mini-75 was used at CRREL to drive GP-112/G stakes into saturated silt and Ottawa sand frozen at 15°F. The tool was found very ineffective in driving these stakes. As a result, the Mini-75 is not recommended for driving military stakes in frozen ground.

Maruzen also makes the Maxi VL-100, a larger (56-lb) unit which was not tried. However, judging from the specifications on this unit one can expect that its performance would place it in the same capability class as the Pionjar and the Cobra.

1. **Fuel tank** of light alloy.
2. **Motor-compressor piston** designed for double action.
3. **Suction valve** (right) and **delivery valve** (left) of the built-in compressor.
4. **Torsion bar** imparts rotary movement from the crankshaft to the drill steel.
5. **Impact piston** moves freely in the cylinder.
6. **Selector handle** for changeover to drilling, breaking or idling.
7. **Pawl feed.**
8. **Spark plug.** The spark plug cover seals perfectly to prevent escape of cooling air.
9. **Turbo type fan** forces cooling air to all vital points.
10. **Floatless carburettor** ensures foolproof running - even at 45° above the horizontal.
11. **Top-mounted throttle control** gives full power in working position; returns to idling automatically when hand pressure is released.
12. **Recoil starter.**
13. **Air filter** ensures clean air to both engine and compressor.

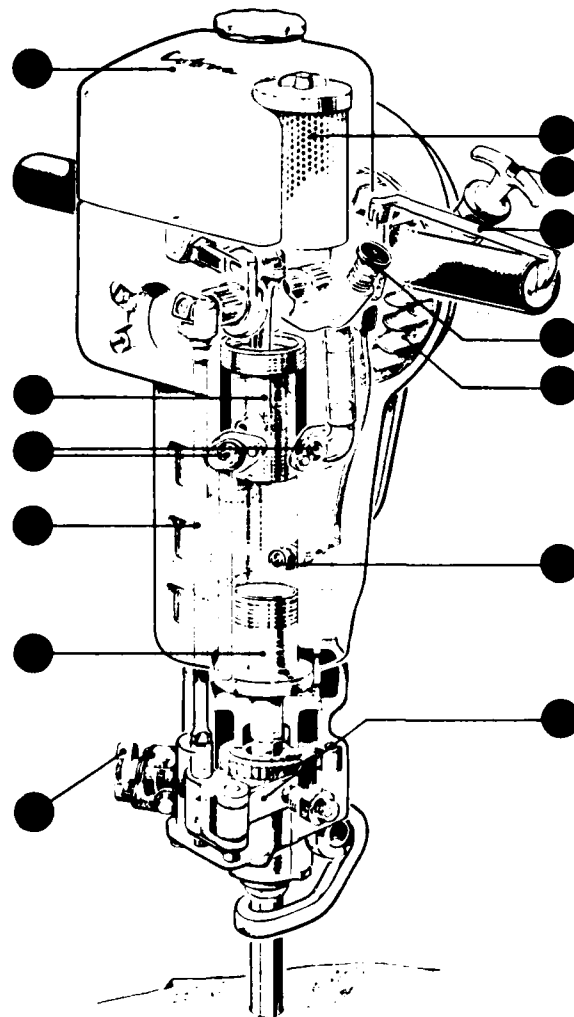


FIGURE 1. Drawing view of the air breaker drill.
(Manufacturer, Atlas Copco, Inc., Hackensack, N.J.)

WEIGHT — Appox 13kg(29lbs)

ENGINE — Cylinder Capacity—54c.c.

Ignition — Flywheel Magneto

Carburetor—Floatless, Suction Valve Type

Dust blow-off— Automatic

Starter— Automatic Recoil

R.P.M. — 2,700 ~ 3,000

COMPRESSOR — Direct driven, Single Cylinder

Cylinder displacement 21c.c.

CONTROL — Fuel knob

OVERALL LENGTH — 545mm × 242mm × 224mm
(21½" × 9½" × 8¾")

DRILLING SPEED — 12cm/min. (5"/min) — 26 φ
15cm/min. (6"/min) — 24 φ

DRILLING DEPTH — 2 meters (7 feet)

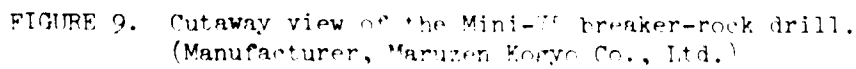
CHUCK SIZE (hex) — 19mm × 108mm (¾" × 4¼")

FUEL — 12:1 Gasoline/oil

FUEL TANK CAPACITY — 0.8 litres (0.21 U.S. Gal.)

FUEL CONSUMPTION—1.3 litres/h (0.34 U.S. Gal/h)

DRILLING ANGLE — 360 Degrees



Wacker BHF30

The Wacker BHF30 is an American-made tool weighing 65 lb. It is reported to be the most powerful tool on the market at a maximum energy output of 30 ft-lb/blow. However, its blow rate is only 1350/min, or nearly one half that of the Pioneer, Cobra or Maxim ML-100. There is no reason to believe that this unit cannot drive, for example, the GP-112/G stake in the same soils as did the Pioneer or the Cobra. The only expectation is that the driving may take slightly longer, again because less energy is being delivered per unit time. The unit is shown in Figure 10.

ELECTRIC DEMOLITION HAMMER

The Power Sledge Electric Breaker (No. 5026) is manufactured by Black and Decker. It weighs 63 lb, runs on 120-V current, and produces 60 ft-lb/blow at 610 blows/min. A photograph and breakaway view of the tool are shown in Figure 11.

Time did not permit an evaluation of the ability of this tool to drive GP-112/G stakes into frozen soil. The tool was used to drive only the 1-1/8-in. chisel-point shank shown in Figure 11 into Hanover silt (bulk density 114 lb/ft³) frozen at -0.5°C in less than 15 sec. This driving was performed only to verify that the tool was operational when received in the the spring of 1973. Further testing is needed; **however, from the limited use of the tool, it is believed that it will be more than capable of repeatedly driving GP-112/G stakes into hard frozen ground.**

HYDRAULIC STAKE DRIVER-RETRIEVER

During this review of the various methods that have been tried or could be used to drive stakes into frozen ground, it became apparent that more than just a stake driver was needed. Indeed, it was learned that, although troops could often drive stakes into hard or frozen ground (with considerable effort), they could not extract them. From Mr. Lione of the Mechanical Structures Branch at the U.S. Army Electronics Command, it was learned that Signal Corps troops are often unable to retrieve guy stakes from hard or frozen ground. Thus, when troops are forced to abandon one site for another, they are at times unable to set up their communication towers at the new site because they were unable to retrieve their guy stakes at the first site. This problem also places an additional burden upon the support system that must bring a new supply of stakes to the front line.

**UTILITY BREAKER
MODEL
BHF 30**

- Medium Duty Breaker
- 55 # Air Equivalent
- Breaks up to 8" of Concrete
- 30 ft. lbs./blow
- Steel Cased Percussion System
- Shock Mounted Operator Handle

FIGURE 10. The Wacker BHF-30 breaker-rock drill
(Manufacturer, Wacker Corp., Milwaukee, Wisconsin.)

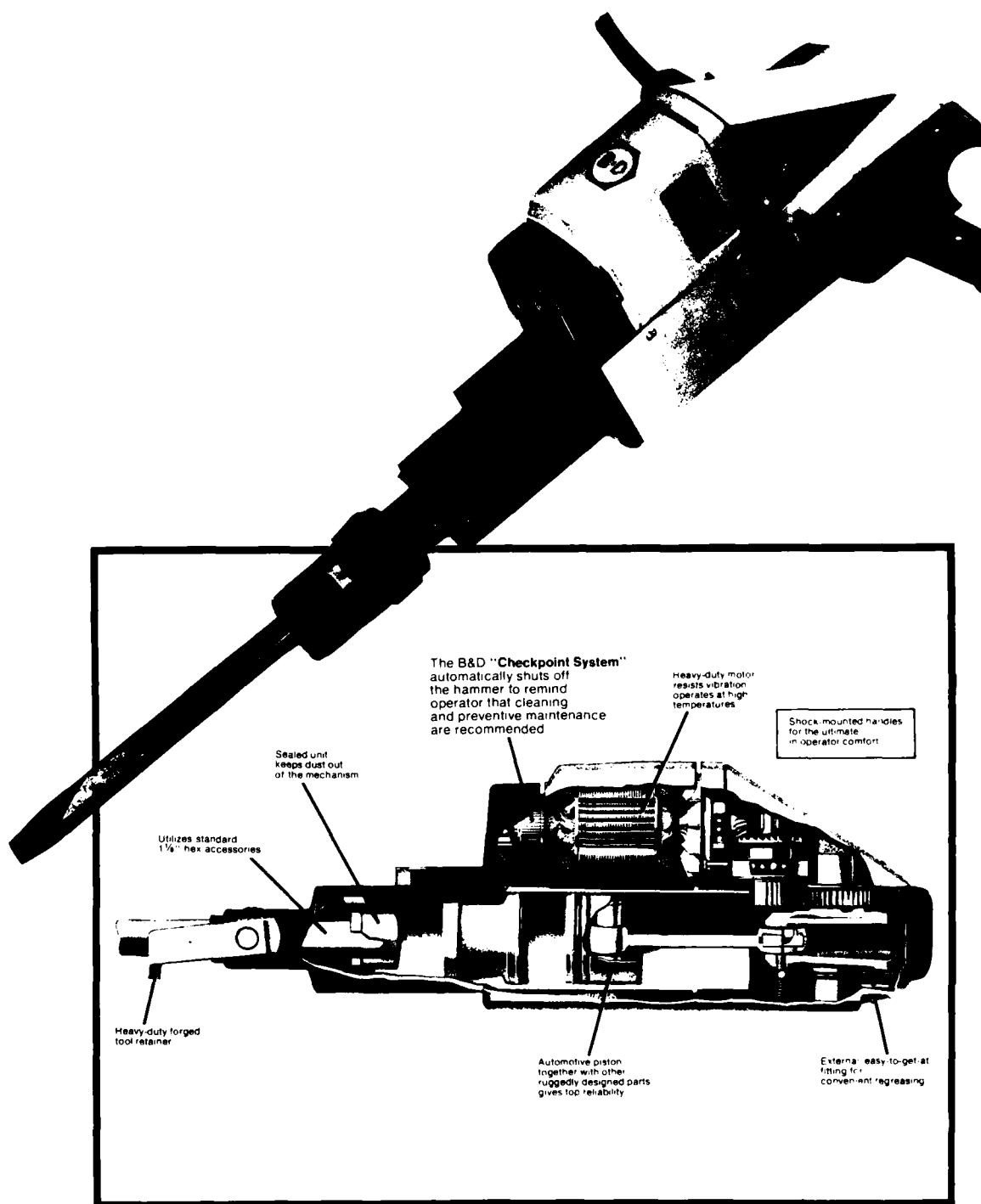


FIGURE 11. The Power Sledge Electric Breaker (Manufacturer, Black and Decker Mfg. Co., Towson, Maryland).

A similar problem exists when 105-mm and XM102 howitzer base plates are staked to frozen ground. Here, the driving of the stakes has been found very difficult, but the retrieval has often been impossible on short notice. As a result, the base plates have had to be left behind when the howitzers were moved. This restricts the ability of the military to redeploy this equipment rapidly if new base plates and stakes are not in immediate supply.

As a result of these findings, it was decided in 1970 not only to try commercial tools for driving stakes into frozen ground, but to explore the possibility of developing a lightweight hydraulic stake driver-retriever. It was also envisioned that such a tool could be used to punch holes in frozen ground for inserting a wide assortment of military devices as well as for obtaining cores from frozen and unfrozen ground and submarine soils.

Funding limitations precluded a major design and development program. Therefore, our specifications to industry were minimal in order to allow the flexibility in design which each firm might find necessary to achieve the overall objective. The statement of work to industry was as follows:

"Provide labor, materials and engineering for the design, fabrication and field test of a hand-held, hydraulically-operated stake driver-retriever as per the following specifications.

"The tool shall consist of a hydraulic vibrator with appropriate mandrel to accommodate both rigid and loose (1/32-in. play between stake and mandrel) attachment of a standard Military GP-112/G ground stake and shall include a portable hydraulic power supply driven by an internal-combustion engine. The hydraulic vibrator shall be a hand-held unit not weighing over 80 lb. The power supply shall not exceed 100 lb in weight.

"The frequency range of the vibrator shall be variable up to 300 Hz and have a dynamic load range from 500 to 3000 lb. Operation of the tool shall be controlled at the handles and the controls shall include an on-off lever and a frequency-range selector. The handle shall be isolated from vibration to avoid operator discomfort. The overall system shall be completely operable over the temperature range of -40° to +30°C. The vibrator shall be capable of underwater (salt or fresh) operation to a depth of 300 ft and to an altitude of 10,000 ft above sea level. The vibrator shall be sealed for operation under severe wind, rain, mud and dust conditions. The hydraulic lines shall be functional and flexible at -40°C, have quick disconnect couplings and be 25 ft in length.

"Prior to the acceptance of the system, the hydraulic tool shall be tested and evaluated predicated upon stake driving trials held at the manufacturer's facilities. The trials shall be witnessed by A. Kovacs of USACRREL and shall demonstrate the unit's capability of driving the GP-112/G stake 12 in. into saturated ASTM-C-109 Ottawa sand at -10°C and into

80 to 90% saturated silt at -10° and -20°C . Driving time for each stake shall not exceed 45 sec.

"Once driven, the stakes shall be allowed to "freeze back" for 15 to 20 min. The tool must then be capable of extracting each stake in 30 sec or less. The hydraulic system shall also be cold soaked at -40°C in a cold box and then started to verify its operational capability at this temperature.

"Upon successful completion of test and evaluation, the tool will be accepted and used by USACRREL for further feasibility studies and evaluation."

Clarification discussions were held with potential contractors, and in November 1971 a contract was given to the Team Corporation for the design and fabrication of a first-generation hydraulic stake driver-retriever system. What may be considered a breadboard model of the first generation tool is shown in Figure 12. This tool weighed approximately 65 lb and was used to evaluate the unique valving system for activating and controlling the frequency of the floating piston-ram. This model was also used to determine if the force output of the tool was adequate for driving GP-112/G stakes in 45 sec into saturated ASTM C-109 Ottawa sand frozen to -20°C . A view of this driving is shown in Figure 13. Note that the thawed sand-water slurry was displaced by the stake during driving.

The breadboard model tests revealed that certain deficiencies existed in the tool's valving motor and that the tool's driving force needed to be increased if the GP-112/G stake was to be driven within the time frame specified by the contract. As a result of lessons learned from the breadboard model, the Team Corporation decided to make a number of changes which resulted in a lighter, more powerful tool. To achieve these advantages, it was necessary to reduce the frequency range of the tool from 20 to 350 Hz to 20 to 300 Hz and increase the piston-ram weight. The piston-ram velocity, however, remained high. This guaranteed that a high energy per blow (maximum velocity) would be maintained at the highest possible frequency rate.

A cutaway view of the first-generation Hydraulic Stake Driver-Retriever (HSD-R II) is shown in Figure 14. The body of the driver is approximately 14 in. long and 5 in. square and has a wet weight of 47.5 lb. A porting and filtering manifold is shown attached to one side. Details of the valve shaft are not shown as this unique system is quasi-proprietary to the Team Corporation.

The HSD-R II works on the principle of alternating pressure and flow. The flow out of the valve is constant because of the unique rotary valving. When the valve rotates, the pressurized hydraulic fluid operates alternately,



FIGURE 12. Breadboard model of the Hydraulic Stake Driver-Retriever (manufacturer, Team Corp, South El Monte, Calif).

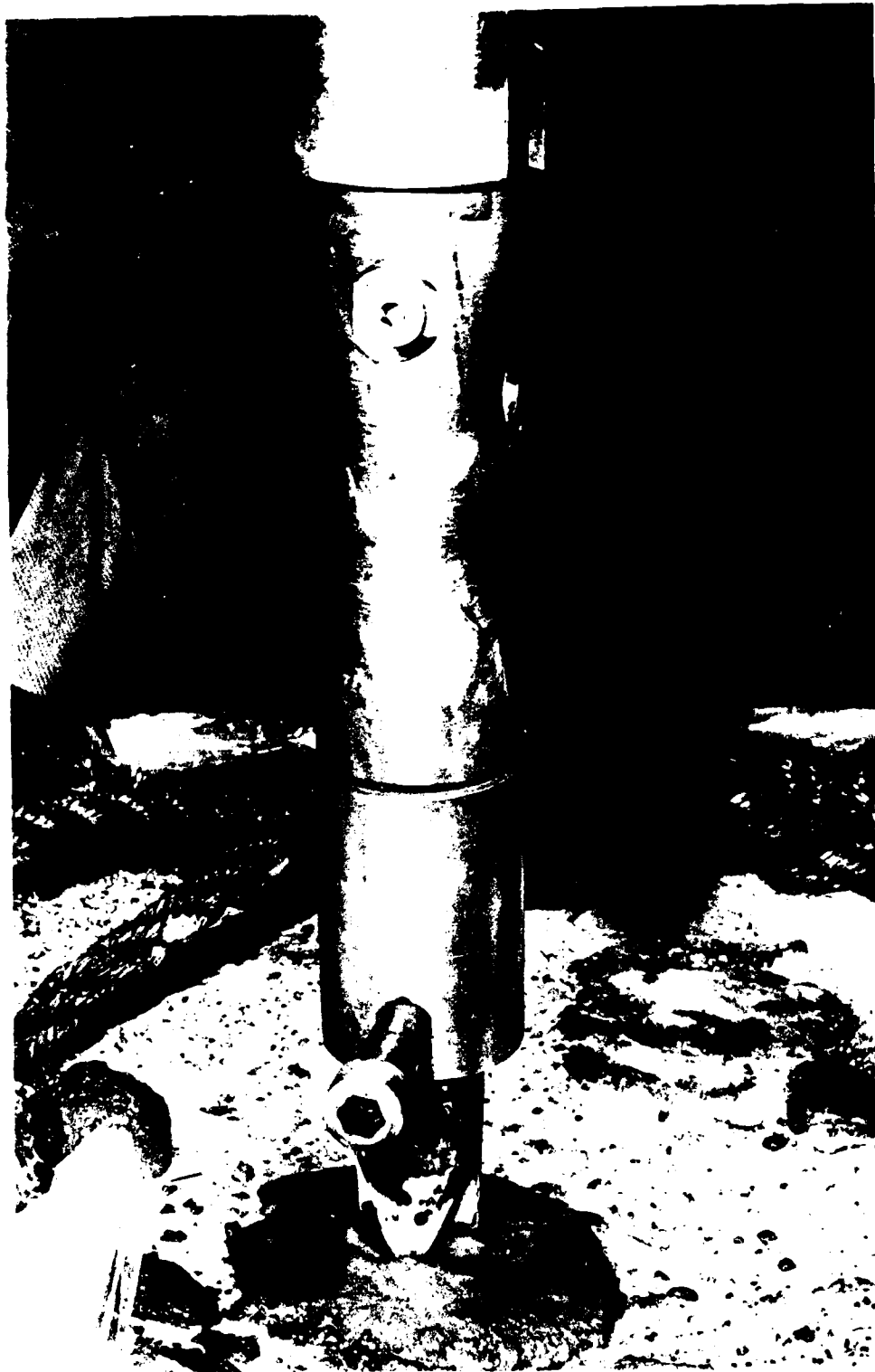
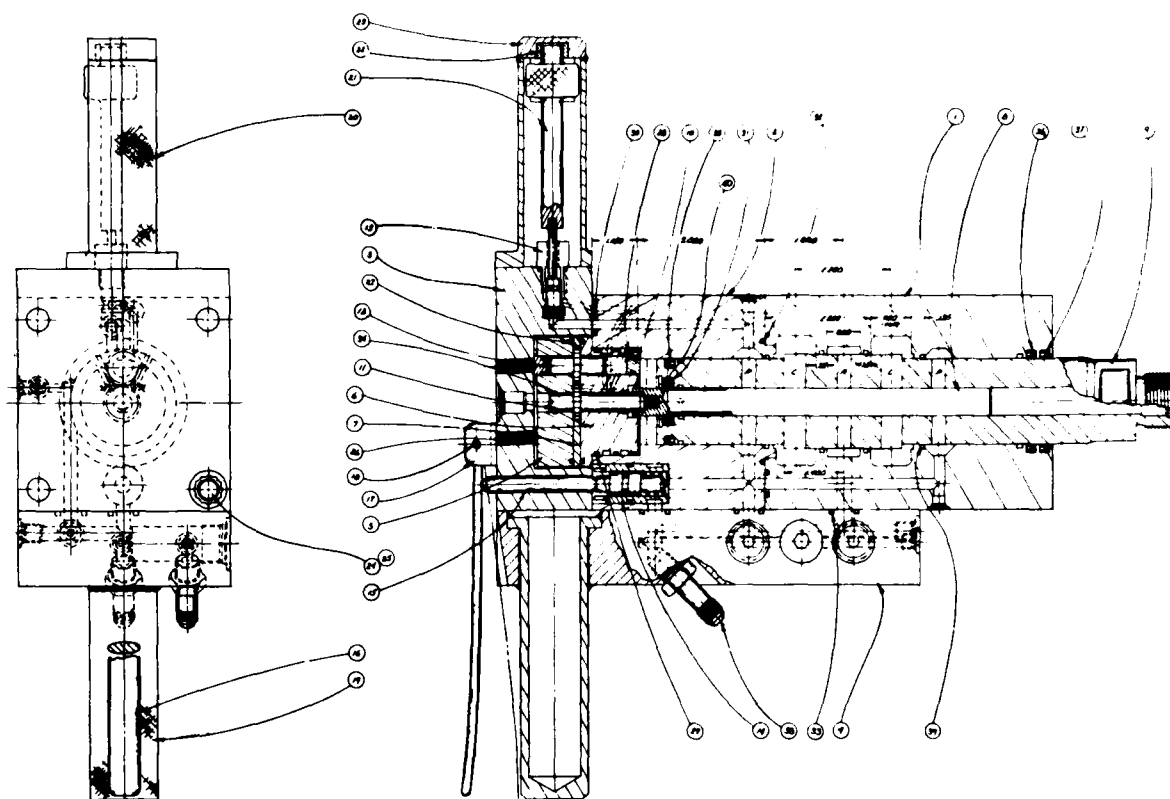


FIGURE 13. Driving GP-112/G stake into saturated frozen (-20°C) Ottawa sand.



- | | | | |
|----|----------------------|----|-------------------------------|
| 20 | Left Handle | 40 | Bal Seal |
| 19 | Right Handle | 39 | Teflon "O" Ring |
| 18 | Roll Pin | 38 | Straight Thd. Long Connectors |
| 17 | Lever Bracket | 37 | Rod Wirer |
| 16 | Lever | 36 | Front Piston Seal |
| 15 | Cartridge Rod | 35 | Rear Piston Seal |
| 14 | Cartridge Valve | 34 | Bal Seals |
| 13 | Dragon Valve | 33 | " |
| 12 | Idler Gear | 32 | " |
| 11 | Driving Gear | 31 | " |
| 10 | Motor Plugs | 30 | " |
| 9 | Piston | 29 | " |
| 8 | Valve Shaft | 28 | " |
| 7 | Center Plate | 27 | Parker "O" Ring |
| 6 | Motor Plate (Bottom) | 26 | Set Screws |
| 5 | Motor Plate (top) | 25 | ESNA Nuts |
| 4 | Manifold | 24 | Studs |
| 3 | Top Plate | 23 | Handle Cap |
| 2 | Upper Housing | 22 | Knob Bushing |
| 1 | Lower Housing | 21 | Valve Adj. Knob |

FIGURE 14. Cutaway view of the Hydraulic Stake Driver-Retriever II.

first on one side then on the other side of a collar extending from the piston side in Figure 14. This is accomplished through the slotted rotating valve shaft alternately aligning with the control ports on each side of the piston. Since the area of the valve slots is fixed, the flow per unit time remains constant. If the velocity of the valve shaft is increased, the slots spend less time aligned with the control ports. Because more slots are passing per unit time, the average flow to the piston remains constant. The decrease in alignment time, however, does reduce the volume of fluid necessary to achieve maximum peak-to-peak displacement. Thus, as frequency increases, the double amplitude of the piston-ram decreases. The frequency of the piston-ram is adjusted by a thumbscrew on the handle of the hammer. The advantage of this control is that during stake driving or retrieval the frequency can be changed to suit exactly the best tool response for the particular soil conditions.

Originally, it was planned to have the stake rigidly attached to the piston. It is obviously important for this coupled mode of operation that the weight of the accelerated parts, i.e., the piston (3.2 lb), adapter (4.15 lb, that part which is threaded onto the base of the piston and to which the stake is attached), and the stake (3 lb) be kept as low as possible relative to the reactive mass, i.e., the body of the driver. For the HSD-R II, the ratio of the accelerated to reactive parts is approximately 1 to 5, which was considered acceptable. The reactive force is increased to an unknown amount by the bias force applied by the operator. Therefore, the overall force available to drive a stake in the coupled mode includes the bias weight applied by the operator, the weight of the tool, and the dynamic force produced by the ram.

The reason for driving with a rigid piston-stake coupling was predicated upon the belief that high-frequency axial excitation of the stake would reduce viscous friction between the stake and the soil by keeping the soil in a quasi-fluidized state. This would enhance energy transfer to the stake at the time where the energy is necessary to displace the soil. It was also known that conventional vibratory pile drivers (piles are rigidly fixed to the driver) are at their best in granular materials and that, by frequency variation, spectacular driving rates are possible. Even in the stiffest clay, good driving results are possible if the driver is operated at a frequency best suited for that particular site. Therefore, by direct coupling, and given the ability to vary the frequency, the investigators hoped to lock into a state of soil-stake excitation, resulting in the highest possible stake penetration for any soil condition encountered. Another reason for rigidly attaching the stake to the ram was to eliminate damage to the stake head, however hard the driving, and to reduce the noise associated with all impact systems.

It may be summarized here that the HSD-R II did not drive the 30-110 stake satisfactorily when the stake was rigidly attached to the piston. The

driving time was far above that specified by the contract, i.e., 12-in. embedment in 45 sec or less. This may be explained by the fact that in resonant pile driving the double amplitude of the pile tip plays a very important role. All other factors being equal, the double amplitude of the pile tip not only controls the rate of penetration, but also whether or not penetration will occur at all. If these findings are applicable to HSD-R II rigid coupled stake driving, for satisfactory penetration rates it is necessary that the stake double-tip amplitude be at least 0.15 in. and preferably higher than 0.20 in. As will be shown later, this amplitude was not achieved at frequencies above 100 Hz, i.e., at the higher frequencies where it was anticipated higher penetration rates would occur as a result of more energy output per unit time.

Poor driving results in the rigid coupled mode may have also occurred as a result of energy losses between the stake and the "fluidized" soil. The driving friction between the stake and soil is in the form of viscous friction⁷ and is on the order of 40 to 95 lb sec/ft³ for steel against saturated sand. The units are damping per unit of embedded area. The effect of this friction is as though the accelerated mass had increased, thus reducing the ratio of the accelerated parts to the reactive parts. As a result, more energy is diverted from doing useful work with the stake to moving the reactive mass. The effect is also to cause the system to go out of resonance, a subject to be covered in greater detail later.

The loosely coupled mode of operation allows the piston to impact the stake head repetitively at up to 300 Hz, the frequency being easily adjusted by a thumbscrew on one handle of the HSD-R II. In the impact mode, the ram has a given energy available for transfer to the stake. This energy E is of course a function of piston velocity V and weight W and may be determined by $E = WV^2/2g$ where g is the gravitational acceleration. The total amount of energy available is not necessarily transferred to the stake and certainly not to the tip where work must be done to displace the soil. This is due to a loss of energy occurring during energy transfer from the piston to the stake head as a result of imperfect contact and friction in the coupling head and to the friction developed along the stake/soil interface. Therefore, the force available to drive the stake is a direct function of the overall dynamic response of the driver-stake-soil system. While no effort was made to determine the through force to the stake tip, the force produced at one level of no load bench operation was determined and the results were compared with those of the manufacturer. This evaluation is presented in a following section.

The hydraulic power supply for the HSD-R II is shown in Figures 15 and 16. It has a wet weight of 99 lb without the muffler, which is not shown. The frame, oil tank, and gas tank are all fabricated out of aluminum and hard coat anodized. The gas tank is mounted above the reservoir tank for the

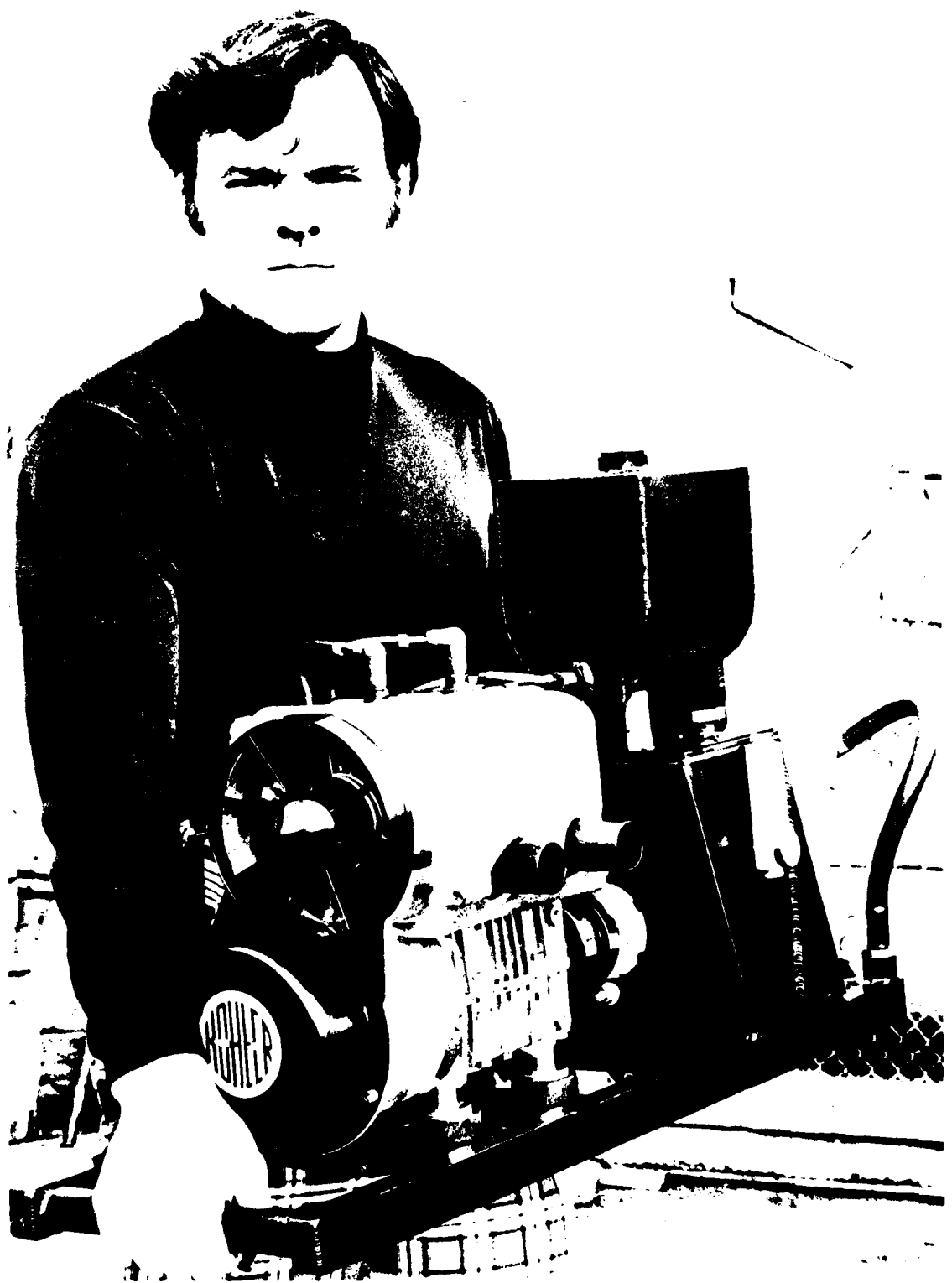


FIGURE 15. Hydraulic power supply for the Hydraulic Stake Driver-Retriever II.

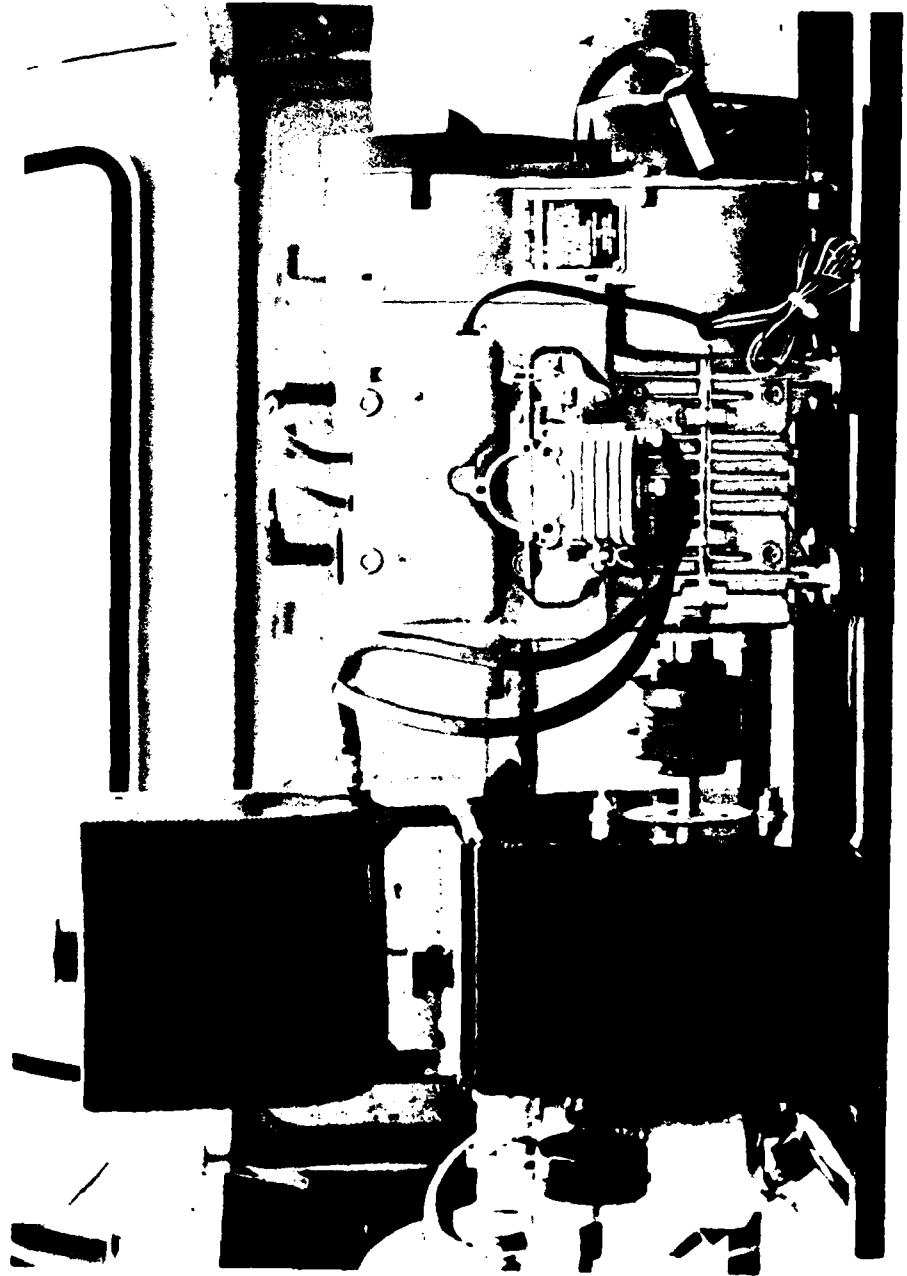


FIGURE 16. Front view of the Hydraulic Stake Driver-Retriever II power supply.

hydraulic oil. The reservoir tank is ported to the hollow framework. This extra porting adds to the total oil volume and can provide appreciable extra cooling when the device is set upon ice or frozen ground. Return oil first flows through the lower framework and then back into the reservoir tank where an internally mounted pump again moves it along at up to 2500 psi and 8 gal/min to the HSD-R II.

The hydraulic unit is designed for an intermittent stake driving and retrieval duty cycle and cannot, without the addition of a heat exchanger, be used for continuous duty operation at temperatures above 30°F.

The major problem with the power supply has been with muffling the exhaust from the 26-hp internal combustion engine. Several muffler types have been tried without a satisfactory reduction in noise. Other mufflers will be investigated at a later date in an effort to achieve what is considered an acceptable noise level.

The HSD-R II was received at CRREL in March 1973 after it was inspected and tested in January at the Team Corporation facilities in Los Angeles. At CRREL, the HSD-R II was retested to determine its ability to drive 1-1/4 in. stakes into both saturated Ottawa sand (bulk density 124 lb/ft³) and Manchester silt (bulk density 114 lb/ft³) frozen at -20°C. Six stakes were driven 12 in. into each of the two soil types. The average driving time was 47 sec in Ottawa sand and 40 sec in Manchester silt. All stakes were retrieved in under 30 sec after having been allowed to freeze back for 20 min. These driving-retrieving results satisfied the driving requirements of the contract. Further stake driving tests in various types of frozen ground are desirable.

In addition to stake driving, the HSD-R II was used to drive a 1-1/4 in. electrical conduit into in-situ Hanover silt (-0.5°C), freshwater ice (-11°C), and lake mud and clay (-2°C). The objective of this driving was to determine the feasibility of using the HSD-R II for coring. The conduit was rigidly attached to the piston because it was believed that this mode of operation would be less likely to disturb the soil sample captured within the conduit; this has not yet been verified. In the Hanover silt, it was possible to consistently drive the conduit to a depth of 2 ft in 7 to 8 sec. The rate in the hard freshwater ice was very rapid, 10 sec/ft. Driving in the stiff, overconsolidated clay was slow in comparison, at approximately 3-4 min/ft. The soft lake muds were penetrated to a depth of 7 ft in less than 10 sec.

Samples of the cores obtained from the Hanover silt are shown in Figure 17. The outer surface of the core was thawed by the heat generated between the conduit and the soil during driving. However, when the core was snapped in two it was found that the disturbed zone around it was about 1/8 in. thick and that the center of the sample was still at the temperature of the ground from which it was removed. A properly designed core barrel



FIGURE 17. Core samples of frozen Hanover silt. The outer 1/8 in. of these cores was thawed during driving of the core barrel.

may eliminate much of this melt zone around the soil and improve the overall quality of the core.

To sample the lake muds, the rapidly deteriorating ice cover of a nearby lake was used as a platform. The drilling operation is shown in Figure 18. The conduit was vibrated through the ice cover and then into the muds below. The warm, deteriorated spring lake ice cover offered very little resistance to penetration. Samples of both the ice and sediment penetrated are shown in Figure 19. The entire depth of sediment penetrated was not retrieved when the conduit was vibrated back up out of the mud. It is believed that suction at the base of the conduit effectively prevented complete core recovery. A properly designed core barrel should overcome this difficulty.

The ice samples obtained as a result of our driving tests in the hard freshwater ice were of poor quality. In general, they consisted of shattered ice pieces. This was attributed to the fact that freshwater ice at -11°C is quite brittle and easily shattered. It is not difficult to visualize how the vibrating conduit could have fractured the ice at the tip of the conduit and, once the ice was captured with the conduit, how continued tube wall vibration may have caused additional ice shattering. It is believed that much of the shattering experience could have been eliminated by using an electrical conduit 3 or more in. in diameter.



FIGURE 18. Driving a 10-ft-long, 1-1/4-in.-OD electrical conduit through the lake ice cover into the bottom sediments using the Hydraulic Stake Driver-Retriever II.



FIGURE 19. Lake ice and bottom sediment core samples retrieved using the arrangement shown in Figure 18.

The stiff clay was the most difficult of the four materials to penetrate. Heat generated during the driving thawed the clay, turning it into a thick, viscous goo which was difficult to vibrate out of the conduit when the conduit was removed from the ground. The slow driving experienced in the clay was not unexpected because resonant and vibro pile drivers are often slowed to unproductive penetration rates in stiff clays and clay silts. Again, a properly designed core barrel might have improved the quality of the core obtained.

During our preliminary driving trials, it became apparent that best stake and conduit penetration was being achieved at what was "sensed" to be in the range of 125 to 150 Hz. In other words, the HSD-R II was more efficient at the lower frequencies than at the higher frequencies. To verify this observation and to learn more about the force and double amplitude response of the HSD-R II versus frequency, it was decided to perform a laboratory evaluation of the tool. A statement of the results of displacement measurements on the HSD-R II and an analysis of those results follows.

Measurement System

Since displacement was the only measurement to be made, a linear motion potentiometer (LMP) and a linear variable differential transformer (LVDT) were considered as transducer elements. The LMP was rejected, since the higher frequencies of operation would have caused overheating of the resistance element, possibly leading to erroneous reading and even damage to the LMP; the "whipping action" of the knife edge on the resistance element at the higher frequencies would have led to an excessive noise level on the signal output, especially when small displacements were being observed; and the small displacements expected would have required an LMP with a relatively small stroke range and hence no allowance to prevent damage from overshoot when the HSD-R II was starting or stopping. An LVDT has none of these problems and in addition is a more sensitive system; therefore, an LVDT with a measurement range of ± 0.3 in. and a drive frequency of 2500 Hz was chosen as the primary sensing element.

A static calibration of the LVDT, using a calibrated micrometer and a digital voltmeter readout, established that 1.0-V change in the output of the LVDT corresponded to a change of 81.08 mils on the input and that the linearity was well within 1% of full-scale output. Unfortunately, there was no guarantee that a static calibration could be used for measurements up to 300 Hz. Therefore, it was decided to run a check channel initially in order to establish that the LVDT was giving valid measurements at the higher frequencies. The check channel was a capacitive sensor (sensing change in capacitance as the piston extension moved in and out of a metal collar around it). This capacitor probe had a much smaller sensitivity than the LVDT (1.0 mV for a 1.38-mil displacement), but had a drive frequency of

800,000 Hz and therefore a much better frequency response. The sensitivity of the capacitive probe was also established by a static calibration.

Initial tests were run, using the sensors described above, a dual beam oscilloscope with 1.0 mV/cm sensitivity as a readout device, and the knowmobile engine as a driver. These tests established that the static calibration of the LVDT was valid at frequencies up to at least 320 Hz. Displacement measurements made with the LVDT and the capacitive sensor agreed within 10% of each other; however, the LVDT was much more sensitive and therefore was considered to be the more accurate, the capacitive sensor serving merely to corroborate high-frequency response.

No other conclusive results were obtained from these tests and they proved to be generally unsatisfactory since:

1. The hydraulic pump heated up very quickly, necessitating short runs with long waiting periods in between.
2. Changing drive frequency on the HSD-R II changed the loading conditions of the drive motor, causing it to change speed with a resultant change in operating pressure.
3. Both of the above conditions resulted in extremely unstable oscilloscope traces, which were difficult to read and nearly impossible to photograph.

Because of the problems noted, a 5-hp DC motor with speed control circuits was used in place of the internal-combustion engine as a driver for the hydraulic pump. Although only a 1000-psi line pressure was possible with this system, it proved to be much more stable and was therefore used to take the data shown in Figure 20a. Both frequency and amplitude were from photographs of the oscilloscope traces.

Analysis of Results

The differential equation for the motion of the piston is

$$\frac{W}{g} \frac{d^2 y}{dt^2} + c \frac{dy}{dt} + ky = F_{\max} \sin wt$$

- where
- W = weight of the piston
 - g = gravitational acceleration, 32.2 ft/sec²
 - c = coefficient of friction
 - k = spring constant associated with the hydraulic fluid
 - y = displacement of the piston from a center position
 - F_{max} = peak force applied to the piston, either forward or reverse
 - w = angular velocity of the piston in radians
 - t = time, sec.

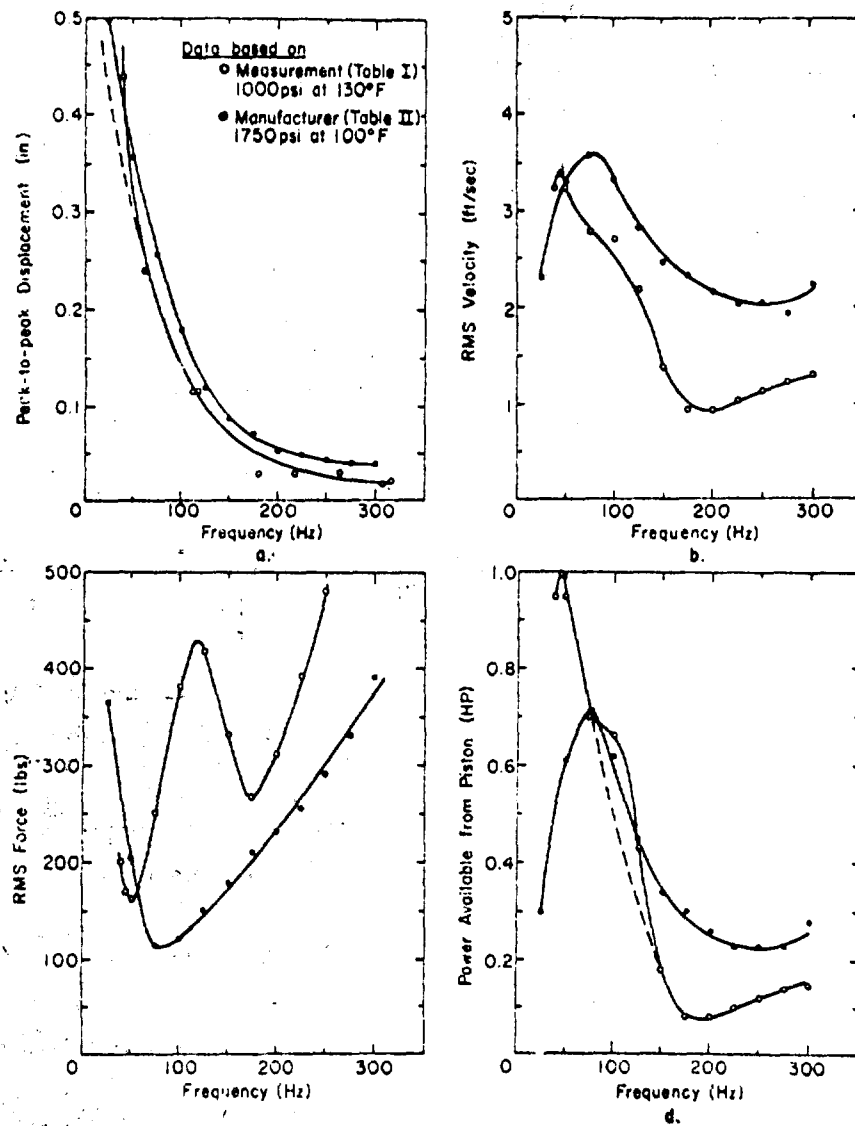


FIGURE 20. Measured data and manufacturer's data from HSD-R II bench test.

This differential equation may be solved in the usual manner by finding a general solution and a particular solution (sometimes called a transient and a steady-state solution). However, in this case only the steady-state response is of interest, and therefore the solution for the transient response can be omitted.

Following the usual method for solving differential equations of this type, the solution is assumed to be of the form:

$$Y = A \cos wt + B \sin wt$$

where y = displacement amplitude
 A and B = constants of integration

hence $Y' = -Aw \sin wt + Bw \cos wt$

and $Y'' = -Aw^2 \cos wt - Bw^2 \sin wt.$

Substituting these values in the original equation:

$$\begin{aligned} \frac{W}{C} (-Aw^2 \cos wt - Bw^2 \sin wt) + c(-Aw \sin wt + Bw \cos wt) \\ + k(A \cos wt + B \sin wt) = F_{\max} \sin wt \\ - \frac{W}{C} Aw^2 \cos wt - \frac{W}{C} Bw^2 \sin wt - Acw \sin wt + Bcw \cos wt \\ + Ak \cos wt + Bk \sin wt = F_{\max} \sin wt. \end{aligned}$$

A and B may now be solved by equating sine and cosine terms to form two equations which may be solved simultaneously:

Cosine terms:

$$-\frac{W}{C} Aw^2 + Bcw + Ak = 0 \text{ or } (k - \frac{W}{C} w^2)A + cwB = 0.$$

Sine terms:

$$-\frac{W}{C} Bw^2 - Acw + Bk = F_{\max} \text{ or } -cwA + (k - \frac{W}{C} w^2)B = F_{\max}.$$

Solving for A and B simultaneously:

$$B = \frac{-(k - \frac{W}{C} w^2)A}{cw}$$

$$-c\omega A - \frac{(k - \frac{W}{g}\omega^2)^2}{c\omega} A = F_{\max}$$

$$A \left(-c\omega - \frac{(k - \frac{W}{g}\omega^2)^2}{c\omega} \right) = F_{\max}$$

$$A = \frac{F_{\max}}{\left(-c\omega - \frac{(k - \frac{W}{g}\omega^2)^2}{c\omega} \right)} = \frac{F_{\max} \omega c}{\left(-\omega^2 c^2 - (k - \frac{W}{g}\omega^2)^2 \right)}$$

$$B = \frac{-(k - \frac{W}{g}\omega^2)}{c\omega} \times \frac{F_{\max} \omega c}{-\omega^2 c^2 - (k - \frac{W}{g}\omega^2)^2} = \frac{-(k - \frac{W}{g}\omega^2) F_{\max}}{-\omega^2 c^2 - (k - \frac{W}{g}\omega^2)^2} = \frac{(k - \frac{W}{g}\omega^2) F_{\max}}{\omega^2 c^2 + (k - \frac{W}{g}\omega^2)^2}$$

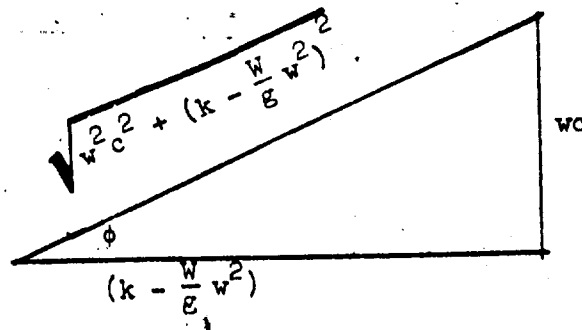
Therefore, the steady-state solution is:

$$\begin{aligned} Y(t) &= \frac{F_{\max} \omega c}{-\omega^2 c^2 - (k - \frac{W}{g}\omega^2)^2} \cos \omega t - \frac{(k - \frac{W}{g}\omega^2) F_{\max}}{-\omega^2 c^2 - (k - \frac{W}{g}\omega^2)^2} \sin \omega t \\ &= \frac{F_{\max} (k - \frac{W}{g}\omega^2)}{\omega^2 c^2 + (k - \frac{W}{g}\omega^2)^2} \sin \omega t - \frac{F_{\max} \omega c}{\omega^2 c^2 + (k - \frac{W}{g}\omega^2)^2} \cos \omega t. \end{aligned}$$

This equation may be written in the form:

$$Y(t) = \frac{F_{\max}}{\sqrt{\omega^2 c^2 + (k - \frac{W}{g}\omega^2)^2}} \left[\frac{(k - \frac{W}{g}\omega^2) \sin \omega t}{\sqrt{\omega^2 c^2 + (k - \frac{W}{g}\omega^2)^2}} - \frac{\omega c \cos \omega t}{\sqrt{\omega^2 c^2 + (k - \frac{W}{g}\omega^2)^2}} \right]$$

Looking at the expressions inside the brackets, the algebraic terms can be represented as sides of a right triangle as follows:



Utilizing this triangle, $Y(t)$ may now be written as:

$$Y(t) = \frac{F_{\max}}{\sqrt{w^2 c^2 + (k - \frac{W}{g} w^2)^2}} (\sin wt \cos \phi - \cos wt \sin \phi)$$

and by taking advantage of the trigonometric identity: $\sin(x - y) = \sin x \cos y - \cos x \sin y$

$$Y(t) = \frac{F_{\max}}{\sqrt{w^2 c^2 + (k - \frac{W}{g} w^2)^2}} \sin(wt - \phi) \quad (1)$$

where $\phi = \arcsin \frac{wc}{\sqrt{w^2 c^2 + (k - \frac{W}{g} w^2)^2}}$

Since $Y(t)$ has been measured (resulting in the lower curve of Fig. 20a), the equation may be used to determine the force applied to the piston as a function of frequency f (since $w = 2\pi f$). However, values for k , W and c must be established. The value of k is available from the manufacturer of the hydraulic fluid (26,800 lb/ft), the weight of the piston is available from the manufacturer (3.2 lb), with 5.8 lb added to make a total of 9 lb for the measured data shown in Figure 20a. The value of c (the coefficient of friction plus other energy losses related to velocity) is not directly obtainable.

A value for c may be determined as follows:

1. Displacement data are available from Figure 20a as a function of frequency and for each point they may be written in the form

$$d(t) = Y_{\max} \sin wt$$

where $Y_{\max} = \frac{\text{peak to peak}}{2}$

and $w = 2\pi f$.

Since the derivative of displacement vs time is velocity:

$$\text{velocity} = \frac{d(Y_{\max} \sin wt)}{dt} = w Y_{\max} \cos wt$$

or, velocity at any frequency is w times the displacement. Moreover, if the displacement is in root mean square (rms) ft, then w times it will be rms velocity in foot/second. A plot of rms velocity versus frequency is shown in Figure 20b. The curve shows a maximum or resonant condition at a specific frequency (45 Hz for curve of measured data and 75 Hz for curve of manufacturer's data).

2. Equation 1 may be differentiated with respect to time in order to obtain a velocity equation. This velocity equation may then be differentiated with respect to w , and set equal to zero in order to determine at what frequency the maximum velocity will occur. But since the frequency is already known (Fig. 20b), this equation may be used to solve for c , the coefficient of friction plus other velocity-related losses.

Proceedings with the differentiation of the displacement equation:

$$Y_{\max} \sin wt = \frac{F_{\max}}{\sqrt{w^2 c^2 + (k - \frac{W}{g} w^2)^2}} \sin (wt - \phi)$$

$$\text{velocity} = \frac{d(Y_{\max} \sin wt)}{dt} = \frac{w F_{\max}}{\sqrt{w^2 c^2 + (k - \frac{W}{g} w^2)^2}} \cos (wt - \phi)$$

$$\text{rms velocity} = v_{\text{rms}} = \frac{F_{\text{rms}}}{\frac{1}{w} \sqrt{w^2 c^2 + (k - \frac{W}{g} w^2)^2}}$$

Looking at this equation, the velocity will be maximum when the denominator is minimum. The denominator will be minimum when the term under the radical is minimum. This minimum will occur at the frequency where the derivative of the expression with respect to w is set equal to zero. Therefore,

$$\frac{d \left[w^2 c^2 + k^2 - 2 \frac{Wk}{g} w^2 + \frac{W^2}{g^2} w^4 \right]}{dw} = 0$$

$$2c^2 w - 4 \frac{Wk}{g} w + 4 \frac{W^2}{g^2} w^3 = 0$$

$$w(2c^2 - 4 \frac{Wk}{g} + 4 \frac{W^2}{g^2} w^2) = 0$$

$$\therefore 2c^2 = 4 \frac{Wk}{g} - 4 \frac{W^2}{g^2} w^2$$

$$c = \sqrt{2 \frac{Wk}{g} - 2 \frac{W^2}{g^2} w^2}$$

but since $w = 2\pi f$ and f is known to be 45 Hz,

$$c = \sqrt{2 \times \frac{9 \times 26,800}{32.2} - 2 \frac{9^2 \times (2\pi f)^2}{32.2^2}}$$

$$= 49.9 \approx 50 \text{ lb/ft-sec.}$$

Performing the same calculation for data supplied by the manufacturer, where f is 75 Hz and $W = 3.2 \text{ lb}$,

$$c = 30.7 \text{ lb/ft-sec.}$$

With the coefficient of friction determined, eq. 1 may now be used to determine the force applied to the piston as a function of frequency.

These calculations were performed and resulted in the curves shown in Figure 20c.

A much more important parameter than force is energy, since this is actually what is required to drive the stake. The general equation for horsepower is $\frac{F \times v}{550}$. In the case of reciprocating motion, the instantaneous horsepower may be integrated over one time period T in order to find average (or rms) horsepower. Therefore,

$$HP = \frac{1}{T} \int_0^T \frac{F_{\max} \sin(wt - \phi) v_{\max} \cos wt}{550} dt$$

$$= \frac{F_{\max} v_{\max}}{550} \frac{1}{T} \int_0^T \sin(wt - \phi) \cos wt dt$$

When using the trigonometric identity, $\sin \alpha \cos \beta = \frac{1}{2} \sin(\alpha + \beta) + \frac{1}{2} \sin(\alpha - \beta)$, then:

$$HP = \frac{F_{\max} v_{\max}}{2 \times 550} \times \frac{1}{T} \int_0^T \sin(wt - \phi + wt) + \sin(-\phi) dt$$

$$= \frac{F_{\max} v_{\max}}{2 \times 550} \left[\frac{1}{T} \int_0^T \sin(2wt - \phi) + \frac{1}{T} \sin(-\phi) \int dt \right]$$

$$= \frac{F_{\max} v_{\max}}{2 \times 550} \left[\frac{1}{T} \left(\frac{-\cos(2\omega t - \phi)}{2\omega} \right) \Big|_0^T + \frac{1}{T} \sin(-\phi) (t) \Big|_0^T \right].$$

Or, by the trigonometric identity $\cos(x - y) = \cos x \cos y + \sin x \sin y$,

$$HP = \frac{F_{\max} v_{\max}}{2 \times 550} \left[-\frac{1}{2\omega t} (\cos 2\omega t \cos \phi + \sin 2\omega t \sin \phi) \Big|_0^T - \frac{1}{T} \sin \phi (t) \Big|_0^T \right]$$

and, putting in limits,

$$HP = \frac{F_{\max} v_{\max}}{2 \times 550} \left[-\frac{1}{2\omega T} (\cos \phi - \cos 2\omega T \cos \phi - \sin 2\omega T \sin \phi) - \frac{1}{T} \sin \phi (0 - T) \right].$$

But, $T = \frac{2\pi}{\omega}$ (i.e., $2\pi f = \omega$ or $f = \frac{\omega}{2\pi} = \frac{1}{T} \therefore T = \frac{2\pi}{\omega}$); therefore,

$$HP = \frac{F_{\max} v_{\max}}{2 \times 550} \left[-\frac{1}{2\omega T} (\cos \phi - \cos 2\omega \frac{2\pi}{\omega} \cos \phi - \sin 2\omega \frac{2\pi}{\omega} \sin \phi) + \sin \phi \right].$$

Since $\cos 4\pi = 1$ and $\sin 4\pi = 0$,

$$\begin{aligned} HP &= \frac{F_{\max} v_{\max}}{2 \times 550} \left(-\frac{1}{2\omega T} (\cos \phi - \cos \phi) + \sin \phi \right) \\ &= \frac{F_{\max} v_{\max}}{\sqrt{2} \times \sqrt{2} \times 550} (\sin \phi) = \frac{F_{\text{rms}} v_{\text{rms}}}{550} \sin \phi. \end{aligned}$$

This equation shows that, not only are the force and velocity of the piston important, but the sine of the phase angle between them is equally important. If the energy available to drive the stake is plotted as a function of frequency, the curves of Figure 20d result. The curves show that the most energy is available at the resonant frequency, and therefore this is the point where the stake driver should be operated.

All the data used to construct the curves are shown in Tables I and II.

Other points which can be made from this analysis of the data are:

1. The highest resonant frequency for HSD-R II is a function of hydraulic pressure, piston and load, etc., and was found to be under 100 Hz. Therefore,

TABLE 1. Measured Data from HSD-R II Bench Test

$K = 26,800$

$c = 50 \text{ lb/ft-sec}$

$W = 9 \text{ lb}$

$f_r = 45$

<u>f</u>	<u>Peak-to-Peak Displacement (in.)</u>	<u>Displ rms (ft)</u>	<u>F_{rms} (lb)</u>	<u>ϕ (°)</u>	<u>v_{rms} ft/sec</u>	<u>HP</u>
40	0.438	12.9×10^{-3}	200.5	53.9	3.24	0.95
45	0.395	11.6×10^{-3}	171.9	72.5	3.39	1.01
50	0.350	10.3×10^{-3}	162.0	87.1	3.24	0.95
75	0.201	5.92×10^{-3}	250.1	33.7	2.78	0.70
100	0.145	4.27×10^{-3}	381.1	20.6	2.70	0.66
125	0.094	2.77×10^{-3}	417.7	15.1	2.18	0.43
150	0.050	1.47×10^{-3}	332.8	12.0	1.39	0.18
175	0.029	0.85×10^{-3}	268.5	10.0	0.94	0.08
200	0.025	0.74×10^{-3}	310.3	8.6	0.93	0.08
225	0.0248	0.73×10^{-3}	391.6	7.6	1.03	0.10
250	0.0245	0.72×10^{-3}	480.6	6.7	1.13	0.12
275	0.024	0.71×10^{-3}	574.6	6.1	1.22	0.14
300	0.0235	0.69×10^{-3}	670.0	5.6	1.30	0.15

TABLE II. Manufacturer's Data from HSD-R II Bench Test

$K = 26,800$

$c = 30.7 \text{ lb/ft-sec}$

$W = 3.2 \text{ lb}$

$f_r = 75$

<u>f</u>	Peak-to-Peak Displacement (in.)	Displ rms (ft)	F_{rms} (lb)	ϕ (°)	v_{rms} ft/sec	HP
25	0.500	14.7×10^{-3}	364.9	11.2	2.31	0.30
50	0.357	10.52×10^{-3}	205.5	29.6	3.30	0.61
75	0.257	7.57×10^{-3}	115.2	71.9	3.57	0.71
100	0.180	5.30×10^{-3}	121.6	57.2	3.33	0.62
125	0.122	3.60×10^{-3}	151.5	34.9	2.83	0.45
150	0.089	2.62×10^{-3}	178.0	25.2	2.47	0.34
175	0.072	2.12×10^{-3}	210.4	19.9	2.33	0.30
200	0.058	1.71×10^{-3}	232.1	16.5	2.15	0.26
225	0.049	1.44×10^{-3}	255.2	14.2	2.03	0.23
250	0.044	1.30×10^{-3}	290.8	12.4	2.04	0.23
275	0.041	1.21×10^{-3}	332.8	11.1	1.93	0.23
300	0.040	1.18×10^{-3}	391.0	10.0	2.22	0.28

the valve to control frequency should have its best resolution and control at frequencies from about 125 Hz down to around 25 Hz in order to provide best performance.

2. The best method of driving a stake is to establish and hold a resonant condition. This is most readily accomplished by loosely coupling the piston to the stake so that the piston is allowed to vibrate at its own natural frequency and couple energy into the stake by impact. If the stake is rigidly coupled to the piston, then the friction between the stake and the ground becomes part of the coefficient of friction in the equations and tends to force the system out of resonance as the stake starts to penetrate. Also, once the friction forces between stake and ground become high, the effect is the same as if the mass had been increased by the weight of the soil, which again will cause the system to go out of resonance.

3. The "load" determines the energy drawn from the system. Therefore, increasing the energy available (such as increasing line pressure or using a larger horsepower engine) does not increase or improve performance. If greater energy out is desired, the "load" must be increased by increasing the mass of the piston. This increased mass lowers the resonant frequency until ultimately it is so low as to be uncontrollable (a hydraulic fluid with a larger "spring constant" k would partially compensate for this effect). The foregoing statements are demonstrated by the curves in Figure 20c where more energy is available when the system is operated at 1000 psi from a 5-hp electric motor and with a 9-lb piston than when it is operated at 1750 psi with a 16-hp gasoline engine but with only a 3.2-lb piston.

4. The measured data curves show "dips" in output at about 175 Hz. These low points are assumed to be caused by a hydraulic resonance in the system. These "dips" do not show up in the manufacturer's curves. However, no displacement data were taken by the manufacturer at around 175 Hz; therefore, his curve would be expected to be much smoother. In any case, 175 Hz is well above the resonant frequency and is of little practical interest.

DISCUSSION

From this limited test and evaluation of the tools, it may be concluded that the MXG 321()G Ballistic Hammer and the lightweight Maruzen Mini-75 are not adequate for the driving of GP-112/G or larger stakes into hard frozen ground. It was found that the Pioneer BR-80 and the Atlas Copco Cobra breaker-rock drills were effective in driving GP-112/G stakes into saturated silt and Ottawa sand frozen at -20°C . The Power Sledge Electric Breaker also appeared to be more than capable of driving GP-112/G stakes into frozen ground, but further evaluation is necessary in order to verify this.

The hydraulic stake driver-retriever was found capable of not only driving but retrieving GP-112/G stakes from frozen soil. The evaluation of this first-generation tool revealed that certain design changes should be incorporated into a second-generation tool. These changes would specify an even lighter tool weight of 40 lb versus 48 lb as at present, a lower frequency range of 20 to 150 Hz versus 20 to 300 Hz as at present, a double tip amplitude of 0.2 in. at 100 Hz and 0.15 in. at 150 Hz, a piston weight of 5 lb versus 3.2 lb now and a more positive control system for varying the frequency of the tool. These changes should markedly improve stake driving performance. These design changes are only of a preliminary nature and are subject to further review and state-of-the-art considerations.

In short, as a result of this exploratory program, the military can be made aware of commercial tools that are readily available and that can be used to drive GP-112/G stakes into hard frozen ground. Further testing would determine the size range of military stakes that these tools could be expected to drive. In addition, a first-generation hydraulic stake driver-retriever has been developed that can both drive and retrieve GP-112/G stakes from frozen ground. Further development of this tool will result in a lighter, more compact and more powerful unit.

The first-generation hydraulic stake driver-retriever was also found capable of extracting core from frozen soil, lake bottom sediments and ice. However, properly designed core barrels are needed to improve the quality of the core obtained.

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