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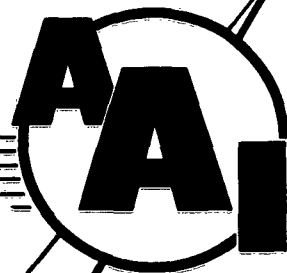
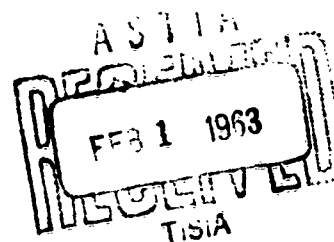
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RAPID EMPLACEMENT AND RETRIEVING DEVICE FOR GROUND STAKES GP-112/G AND GP-113/G

REPORT NO. 2 SECOND QUARTERLY REPORT
1 OCTOBER 1962 TO 31 DECEMBER 1962

CONTRACT NO. DA-36-039 SC-90760

U.S. ARMY ELECTRONICS RESEARCH AND DEVELOPMENT LABORATORY
FORT MONMOUTH, NEW JERSEY



AIRCRAFT ARMAMENTS, Inc.
COCKEYSVILLE, MARYLAND

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Report No: ER-2843A

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I I. PURPOSE The purpose of this contract is to provide a comprehensive feasibility study, evolve a design plan, and develop a ground stake emplacement-retrieving device. Capabilities of the device should include emplacement and retrieving of the GP-112/G and GP-113/G Ground Stakes within one minute in various specified soils (1-1/2 minutes in darkness). Other limitations are weight, 25 pounds, length 42 inches and be non-expendable. The purpose of the work conducted during the second quarter has been to conduct a design study for the approach recommended at the completion of Phase I -- the ballistic impact hammer. The engineering data developed is to be sufficient to permit formulation of a design plan for fabrication, development and test of operating models. 3

II. ABSTRACT

Analysis of detail design parameters has been conducted. Extensive background investigation had indicated that there were no rapid emplacement retrieving devices developed at this time which were capable of satisfying Signal Corps requirements. At the completion of Phase I AAI had recommended that a ballistic impact device be considered in greater detail as it appeared to show the greatest promise of satisfying the performance parameters established. USAELRDL concurred in this recommendation and detail design and experimental work was conducted. A design concept has been developed to permit maximum energy output within the tolerable recoil limits dictated by human engineering factors. The optimum piston weight-velocity relation is being developed as the result of ballistic tests. Design calculations are shown which establish the range of ballistic characteristics as well as stress analysis, kinematic analysis, and life cycle analysis where applicable.

In addition to the primary design objectives related to actual performance of the device, factors of producibility, availability of materials, ease of handling and incorporation of standard components have also been considered.



III. PUBLICATIONS, REPORTS & CONFERENCES

During the period covered by this report, two conferences occurred between representatives of USAELRDL and AAI. On 19 October 1962 USAELRDL was visited by R. G. Strickland to review the changes to the first quarterly report and to review the draft of the Phase I report.

On 5 December 1962 R. Lione and R. Johnson visited AAI to review work to date. The design layout was examined and suggestions received concerning several detail considerations. Four test firings with the ballistic test model were demonstrated at the same time.

During this period formal approval of the First Quarterly Progress Report, AAI Engineering Report 2843, was received and distribution was made per the list supplied by USAELRDL. The Phase I report, AAI Engineering Report 2853, was delivered on schedule. In addition two monthly progress reports have been prepared and delivered. These were AAI Engineering Report 2792B covering the period 21 September through 20 October 1962, and AAI Engineering Report 2792C covering the period 21 October through 20 November 1962.

IV FACTUAL DATA

A. Design Description

Conclusions drawn from the First Quarterly Progress Report have led to the preliminary design of a semi-automatic ground stake emplacement and retrieving device.

Basically, the device is powered by standard grenade cartridges of the caliber 30 carbine fired with the standard carbine action. By cutting off the majority of the barrel of the rifle and threading this section, it is easily adapted to a piston type impact hammer. The carbine action permits semi-automatic operation after merely inserting a clip of cartridges which are self-feeding and self-ejecting hand charging the first round.

Approximately 15,000 psi pressure drives a piston to deliver 500 ft.-lbs. of energy to the stake. This energy level was selected after considering the recoil effects on the operator. A maximum tolerable recoil level of 8 lb.-sec. was assumed with a 2-pound piston to yield 500 ft.-lbs. of energy. After impacting the stake the remaining piston velocity is buffed to absorb most of the high impact forces transmitted to the hammer housing. A light spring returns the piston to the battery after the gases have exhausted through the exhaust ports.

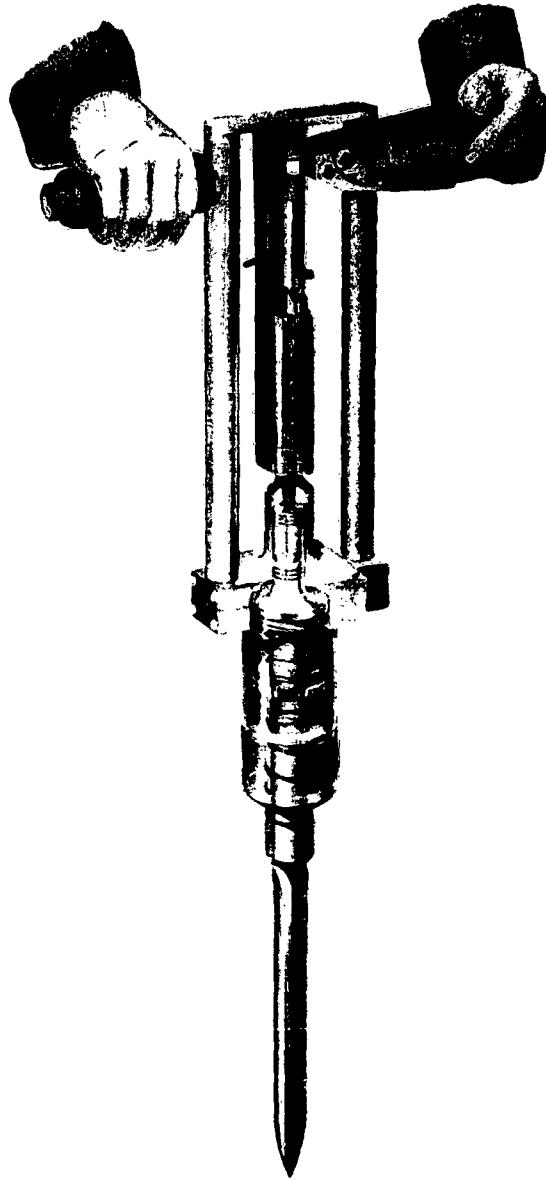
Shortly after the piston completes its power stroke, the spent case is extracted and ejected by the bolt recoil action. At this time the high pressure gases have ported to the large chamber volume and



exhaust passages to be vented through mufflers to the atmosphere. Spent case ejection occurs during the bolt recoil. As the bolt counter recoils it strips the next round and feeds it into the chamber. The next round is fired after depressing the trigger and pushing the hammer to seat fully on the stake. A safety device is incorporated prohibiting firing the next round until the operator exerts sufficient force to seat the hammer on the stake. This is done to assist in minimizing the counterrecoil effects and assure proper hammer-stake orientation.

Shock absorbing hand grips are provided for the comfort of the operator. Trigger mechanisms and safeties can be operated with mittens. Some difficulty may be observed ejecting the spent round clip with mittens, but provisions for replacing the small detent are being made.

No clear solution to the extraction technique has been developed but several possibilities are evident. In soft ground several hammer blows to the side of the stake may compress the ground to widen the emplacement permitting easy extraction. In the case of solid frozen ground, permafrost, etc., a cantilever pivot may be adopted to the side of the hammer to deliver the blow upward.

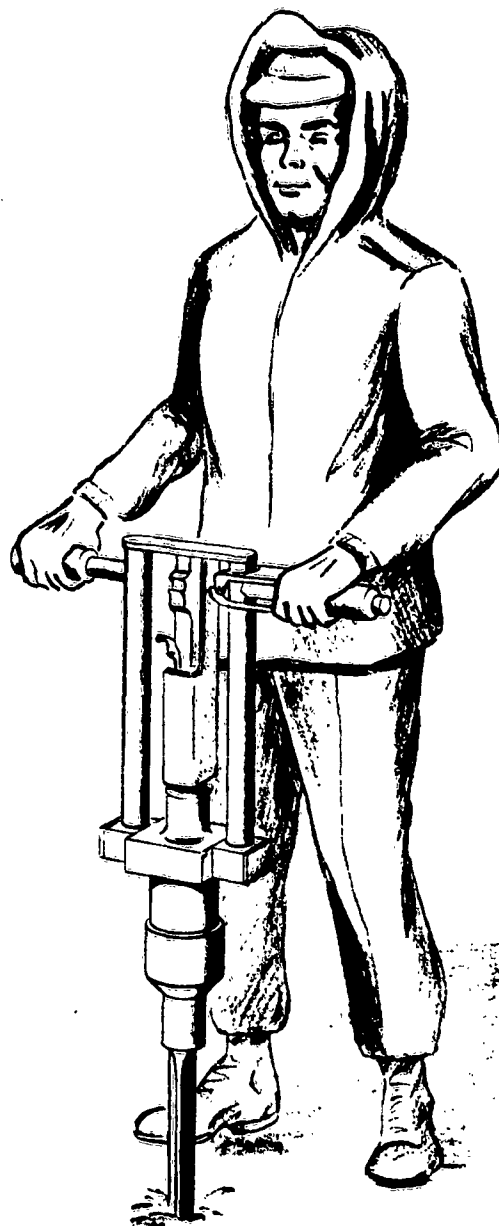


ARTIST'S CONCEPT, BALLISTIC IMPACT HAMMER

Figure 1



AIRCRAFT ARMAMENTS, Inc.



EMPLOYMENT CONCEPT, BALLISTIC IMPACT HAMMER

Figure 1A

B. Operating Parameters

The emplacement-retrieving device is to be designed to emplace Ground Stake GP-112/G in permafrost, ice and hardpan, and emplace Ground Stake GP-113/G in 5 specified normal soils. Emplacement and retrieving times should not exceed 1 minute in daylight or 1-1/2 minutes at night. A maximum weight of 25 pounds and length of 42 inches should not be exceeded assuring a light compact device.

In addition to these design factors the human engineering aspects involved in manipulating the tool with both stakes has been considered. The positioning of the tool with respect to the stakes has been determined with respect to a human operator's reach and strength.

Consideration has been given to reduce noise, eliminate flash and operate safely and simply. Components are not complicated nor delicate or expensive. The bolt action has the most moving parts but is a well proven service accepted item.

With these considerations in mind a preliminary design analysis has been made in an attempt to present a design plan at a later date for fabrication and testing. The following sections describe the areas of design considered and areas remaining to be more fully covered. Certain design changes will become evident as the various components are tested.



C. Interior Ballistics

Certain desired design parameters for the impact hammer will prescribe the associated interior ballistics. These parameters include:

1. Hammer energy required
2. Maximum allowable pressure
3. Maximum desired exhaust pressure
4. Piston area

The piston energy is derived thermodynamically from extremely hot propellant gases. The magnitude of pressure created in a fixed volume by a given propellant is described by the propellant impetus as

$$P_1 = \frac{12 CF}{V_1} \quad (1)$$

where

C = charge weight	lbs.
F = propellant impetus	ft.-lb./lb.
P_1 = pressure	lb./in. ²
V_1 = fixed volume	in. ³

As the gas expands from one state to another, assuming an adiabatic relation, the work done is

$$W = \frac{P_1 V_1 - PV}{12(\gamma - 1)} \quad (2)$$

where

P = pressure at any volume V	lbs/in. ²
V = second state of expansion	in. ³
P_1 = initial pressure	lb./in. ²

V_1 = initial volume

in.³

W = work output

ft.-lbs.

γ = ratio of specific heats of propellant gases

From the adiabatic relation

$$P = P_1 \left(\frac{V_1}{V} \right)^\gamma \quad (3)$$

Combining equations (2) and (3)

$$W = \frac{P_1 V_1 \left[1 - \left(\frac{V_1}{V} \right)^{\gamma-1} \right]}{12(\gamma-1)} \quad (4)$$

Substituting CF for $P_1 V_1$ and describing the work done as the kinetic energy of the hammer,

$$\frac{1}{2} m v^2 = \frac{CF \left[1 - \left(\frac{V_1}{V} \right)^{\gamma-1} \right]}{\gamma-1} \quad (5)$$

where

m = mass of hammer slugs

v = velocity of hammer ft./sec.

One final relation is the volume expansion. Part is due to the piston movement and part due to the hammer itself as it recoils.

$$V = V_1 + A(X + X_H) \quad (6)$$

where

A = piston area in.²

X = piston movement in.²

X_H = hammer movement in.



From the conservation of momentum the velocity of the hammer will be related to the piston by

$$m_H v_H = m_P v_P$$

$$v_H = \frac{m_P}{m_H} v_P \quad (7)$$

Similarly the travel will be

$$x_H = \frac{m_P}{m_H} x \quad (8)$$

Then equation (6) can be written

$$V = V_1 + A \left(X + \frac{m_P}{m_H} X \right)$$
$$V = V_1 + \left(1 + \frac{m_P}{m_H} \right) AX \quad (9)$$

Equations (5) and (9) relate the derived piston energy from a given charge weight at a given piston stroke, x , measured relative to a fixed axis. The following table is prepared assuming a hammer piston with a 1-inch drive diameter and an initial volume $V_1 = .5 \text{ in.}^3$. Using the standard carbine grenade cartridge as the power source provides 20 grains of propellant. The impetus is estimated as $F = 300,000 \text{ ft.-lb./lb.}$ corresponding to a peak pressure of 20,000 psi in the locked shut condition. This pressure seldom occurs as the piston usually moves creating a larger volume when all the propellant is burned. None of the energy is lost by assuming all burning occurs in the initial volume but the peak pressure is

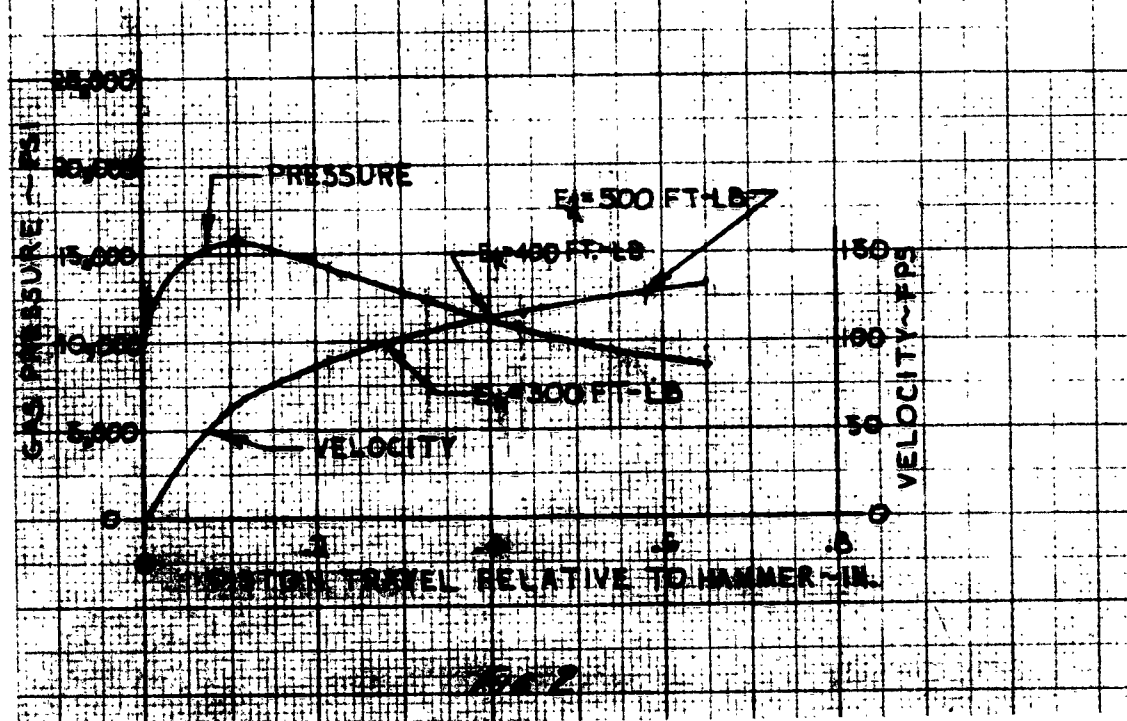
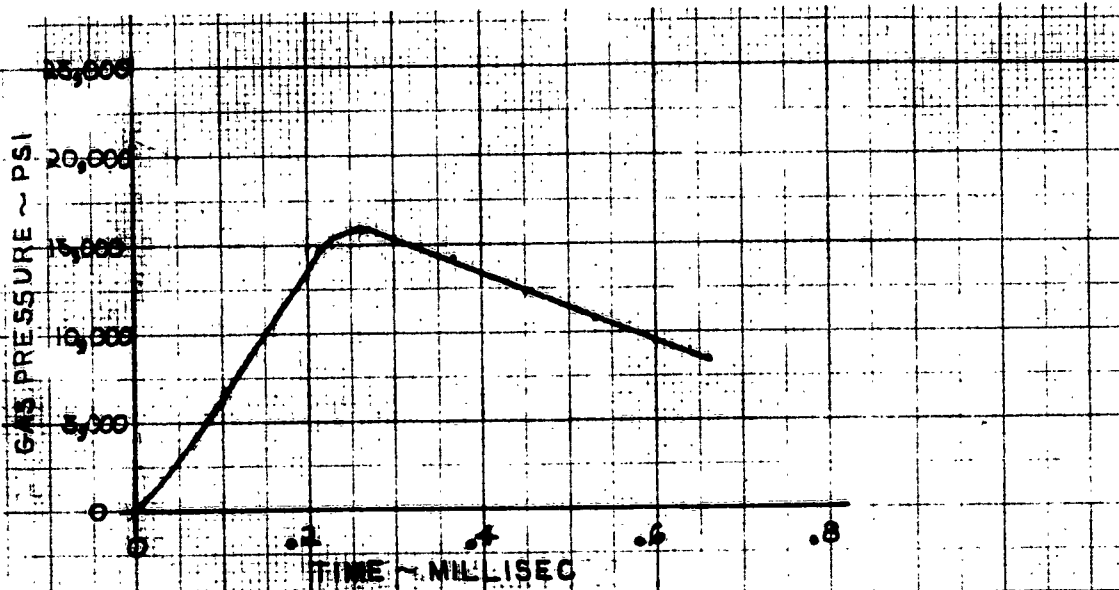
something less than indicated at $x=0$. Explosive ordnance experience shows that this type propellant usually burns in .2 - .3 millisecond and from the pressure-travel curve, a fair estimate of the peak pressure can be obtained on the adiabatic expansion curve.

Increments of time are obtained over small travel increments using the mean velocity over that increment. The hammer weighs 23 pounds and the piston 2 pounds.

Table of Calculations

① x (in.)	② $1.087Ax$ (in ³)	③ V (in ³)	④ V/V_1	⑤ $Q^{.25}$	⑥ $1/Q$	⑦ $1 - ⑥$	⑧ $\frac{25}{V-1}$ (ft.-lbs.)	⑨ V (ft./sec.)	⑩ $Q^{.25}$	⑪ P (Psi)	⑫ Δt (millisec)	⑬ t (millisec)
0										20,600		0
.1	.085	.585	1.17	1.04	.962	.038	130	64.6	1.285	16,000	.258	.258
.2	.171	.671	1.344	1.077	.928	.072	247	89.1	1.448	14,200	.108	.366
.3	.256	.756	1.511	1.109	.902	.098	336	104.	1.675	12,300	.086	.452
.4	.341	.841	1.681	1.139	.878	.122	418	116.	1.915	10,800	.076	.528
.5	.426	.926	1.851	1.166	.861	.139	477	124.	2.160	9,500	.070	.598
.6	.512	1.012	2.021	1.192	.839	.161	552	133.	2.410	8,550	.065	.663

This data is plotted in Figure 2 with the estimated time-pressure trace. The travel is relative to the hammer equal to $(1 + \frac{m_P}{m_H})x$ and determines the actual distance required by the piston.

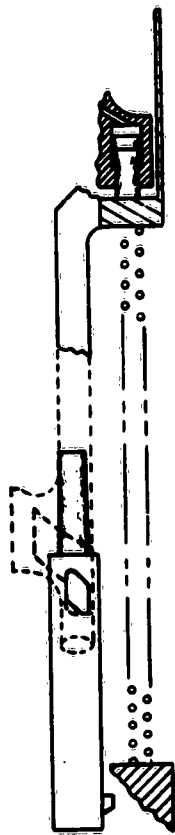


D. Kinematic Analysis

Before extraction of a cartridge case can occur, sufficient time must have elapsed or extraction will occur under excessive pressure. For this reason a kinematic analysis is usually conducted of the cyclic action to insure proper extraction time. In addition, reasonable impact velocities must be maintained to assume stress levels below that for the design life of the various components.

Figure 3 depicts the 9 cyclic events during the semi-automatic bolt action of the U. S. Caliber 30 carbine used in the ground stake device. Following is the related analysis with the numerical evaluation for the preliminary design data. Such parameters as the pressure-time function, spring rates, recoiling masses and functional operations are subject to change after preliminary testing.

The analysis has arbitrarily set the initial time equal to zero and the time relation is just for that event with the total time being the sum of the preceding event times. All symbols refer to the individual event.



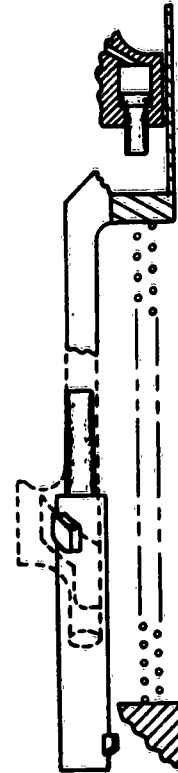
STATION 0



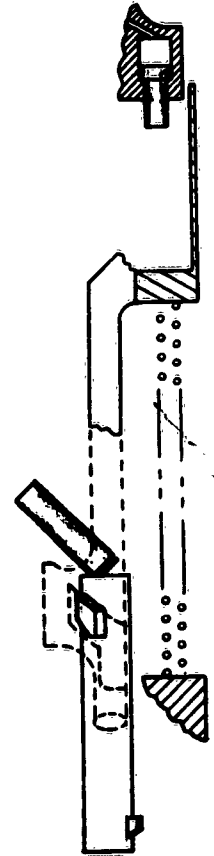
STATION 1



STATION 2



STATION 3

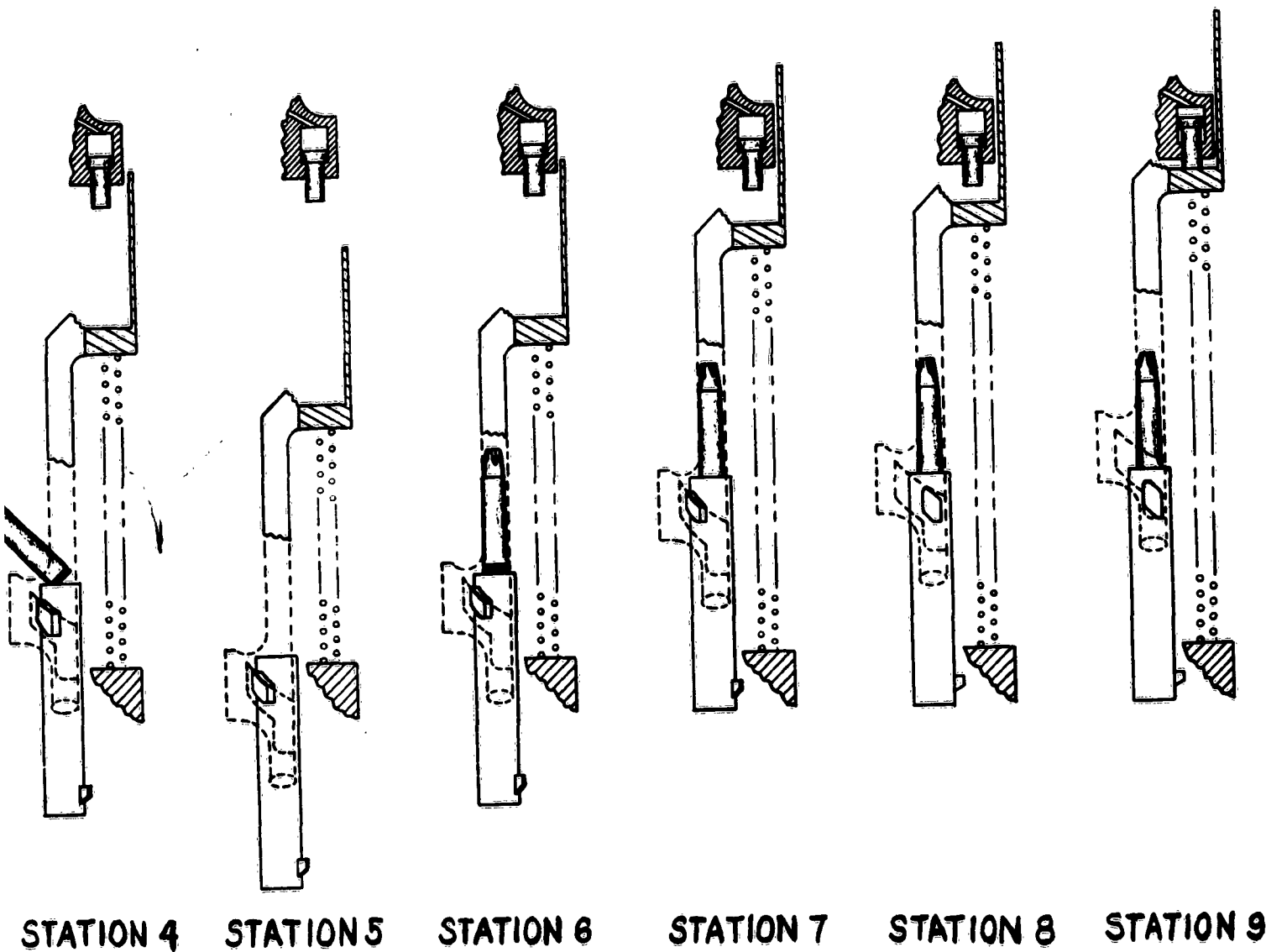


STATION 4

STATION 5



BOLT CYCL
FIG. 3



BOLT CYCLE
FIG. 3



Section 0 - Station 1

As the gas pressure comes in contact with the recoil piston face it forces it rearward against the operating slide and compresses the operating slide spring. The pressure-time curve is written as an exponential equation approximating it mathematically as

$$P = P_0 e^{-bt} \quad (1)$$

Since there is an associated rise time that the exponential - pressure decay cannot account for, the initial time is selected as one half the rise time.

The equation of motion of the small recoil piston is

$$\begin{aligned} m \ddot{x} &= AP - (F_0 + kx) \\ \text{or} \quad \ddot{x} + \frac{k}{m} x &= \frac{AP_0}{m} e^{-bt} - \frac{F_0}{m} \end{aligned} \quad (2)$$

where	A = recoil piston area	in. ²
	b = pressure decay exponent	sec. ⁻¹
	F ₀ = operating spring preload	lb.
	k = operating slide spring rate	lb./in.
	m = recoiling mass	slugs-ft./in.
	P ₀ = pressure decay coefficient	psi
	t = time	sec.
	x = piston recoil travel	in.
	$\dot{x}$ = piston recoil velocity	in./sec.
	$\ddot{x}$ = piston recoil acceleration	in./sec. ²



The characteristic solution of equation (2) is

$$x_c = B \sin \sqrt{\frac{k}{m}} t + C \cos \sqrt{\frac{k}{m}} t \quad (3)$$

The particular solution of equation (2) is

$$x_p = D e^{-bt} + E \quad (4)$$

The general solution becomes

$$x = B \sin \sqrt{\frac{k}{m}} t + C \cos \sqrt{\frac{k}{m}} t + D e^{-bt} + E \quad (5)$$

Arbitrarily for $t=0$, $x=0$, hence,

$$0 = B(0) + C + D + E \quad (6)$$

Differentiating equation (5) and with $t=0$, $\dot{x}=0$,

$$\dot{x} = B \sqrt{\frac{k}{m}} \cos \sqrt{\frac{k}{m}} t - C \sqrt{\frac{k}{m}} \sin \sqrt{\frac{k}{m}} t - D b e^{-bt} \quad (7)$$

$$0 = B \sqrt{\frac{k}{m}} - C \sqrt{\frac{k}{m}} (0) - D b \quad (8)$$

Differentiating equation (7)

$$\ddot{x} = \frac{-Bk}{m} \sin \sqrt{\frac{k}{m}} t - \frac{Ck}{m} \cos \sqrt{\frac{k}{m}} t + D b^2 e^{-bt} \quad (9)$$

Substituting equations (9) and (5) into equation (2) and collecting terms

$$(b^2 + \frac{k}{m}) D e^{-bt} + \frac{k}{m} E = \frac{A P_o}{m} e^{-bt} - \frac{F_o}{m} \quad (10)$$

Equating coefficients of like terms

$$\frac{k}{m} E = \frac{-F_o}{m}$$

$$(b^2 + \frac{k}{m})D = \frac{AP_o}{m}$$

$$E = -\frac{F_o}{k} \quad (11)$$

$$D = \frac{AP_o}{mb^2 + k} \quad (12)$$

From equations (8) and (12)

$$B = \frac{Db}{\sqrt{k/m}}$$

$$B = \frac{AP_o b}{(mb^2 + k)\sqrt{k/m}} \quad (13)$$

From equations (6), (11) and (12)

$$C = -E - D$$

$$C = \frac{F_o}{k} - \frac{AP_o}{mb^2 + k} \quad (14)$$

Hence the solution to equation (2) for the displacement of the piston is

$$x = \frac{AbP_o \sin\sqrt{\frac{k}{m}} t}{(mb^2 + k)\sqrt{k/m}} + \left(\frac{F_o}{k} - \frac{AP_o}{mb^2 + k}\right) \cos\sqrt{\frac{k}{m}} t + \frac{AP_o}{mb^2 + k} e^{-bt} - \frac{F_o}{k} \quad (15)$$

Equation (7) describing the velocity is

$$\dot{x} = \frac{AbP_o}{(mb^2 + k)} \cos\sqrt{\frac{k}{m}} t - \left(\frac{F_o}{k} - \frac{AP_o}{mb^2 + k}\right) \sqrt{\frac{k}{m}} \sin\sqrt{\frac{k}{m}} t - \frac{AbP_o}{mb^2 + k} e^{-bt} \quad (16)$$

Station 1 to Station 2

After the piston thrusts and accelerates the operating slide for a small distance it rams a stop and the inertia of the operating slide carries it to cam open the bolt. This extracts a certain amount of energy in rotating the bolt to unlock it.

Differentially

$$m \ddot{x} = -(F_1 + kx) \quad (17)$$

where

$$F_1 = F_0 + kx_1 \quad (18)$$

x_1 = total travel of piston in inches

x = additional travel of operating from past x_1 in inches

m = recoiling mass (less piston) slug-ft./in.

Equation (17) has the general solution

$$x = B \sin \sqrt{\frac{k}{m}} t + C \cos \sqrt{\frac{k}{m}} t - \frac{F_1}{k} \quad (19)$$

Arbitrarily $t=0$, $x=0$

$$0 = B(0) + C - \frac{F_1}{k}$$

$$C = \frac{F_1}{k} \quad (20)$$

Differentiating equation (19) with $\dot{x} = \dot{x}_1$, $t=0$

$$\dot{x} = B \sqrt{\frac{k}{m}} \cos \sqrt{\frac{k}{m}} t - C \sqrt{\frac{k}{m}} \sin \sqrt{\frac{k}{m}} t \quad (21)$$

$$\dot{x}_1 = B \sqrt{\frac{k}{m}} - C \sqrt{\frac{k}{m}} (0)$$

$$B = \frac{\dot{x}_1}{\sqrt{k/m}} \quad (22)$$

Hence the solutions for $t \geq t_1$ $x \geq x_1$

$$x = \frac{\dot{x}_1}{\sqrt{k/m}} \sin \sqrt{\frac{k}{m}} t - \frac{m_1}{k} (1 - \cos \sqrt{\frac{k}{m}} t) \quad (23)$$

$$\dot{x} = \dot{x}_1 \cos \sqrt{\frac{k}{m}} t - \frac{F_1}{k} \sqrt{k/m} \sin \sqrt{k/m} t \quad (24)$$

Station 2 to Station 3

At station 2 the bolt has a kinetic energy of

$$(\frac{1}{2} m \dot{x}_2^2)$$

At station 3 this energy is decreased by compressing the slide spring, imparting a rotational velocity to the bolt, and cocking the hammer. This energy is equated to the slide energy as,

$$\frac{1}{2} m \dot{x}^2 + \frac{1}{2} I_B \omega_B^2 + (F_2 x + \frac{1}{2} kx^2) + \frac{1}{2} I_H \omega_H^2 + (F_H x' + \frac{1}{2} k_H x'^2) = \frac{1}{2} m \dot{x}_2^2 \quad (25)$$

Equation (25) has a discontinuous value at $x=0$, and must be used for $x>0$ since ω_B and ω_H are zero at $x=0$, but $\dot{x}$ is not zero.

For the bolt rotation, the angular displacement is related to the slide translation by C_1 as

$$\theta_B = C_1 x$$

$$\omega_B = C_1 \dot{x}$$

Similarly for the hammer,

$$\theta_H = C_2 x$$

$$\omega_H = C_2 \dot{x}$$



Approximating the hammer spring motion, which is very small during bolt unlocking by C_3 ,

$$x_H = C_3 x$$

Because of the small compression of the hammer spring during bolt rotation the term $\frac{1}{2} k_H x'^2$ is neglected. Substituting the geometrical relations,

$$\frac{1}{2} m \dot{x}^2 + \frac{1}{2} I_B C_1^2 \dot{x}^2 + (F_2 x + \frac{1}{2} k x^2) + \frac{1}{2} I_H C_2^2 \dot{x}^2 + F_H C_3 x = \frac{1}{2} m \dot{x}_2^2$$

or

$$\dot{x}^2 = \frac{\frac{1}{2} m \dot{x}_2^2 - (F_2 + F_H C_3)x - \frac{1}{2} k x^2}{\frac{1}{2} m + \frac{1}{2} I_B C_1^2 + \frac{1}{2} I_H C_2^2} \quad (x > 0) \quad (26)$$

Let

$$a = \frac{\frac{1}{2} m \dot{x}_2^2}{\frac{1}{2} m + \frac{1}{2} I_B C_1^2 + \frac{1}{2} I_H C_2^2}$$

$$b = \frac{F_2 + F_H C_3}{\frac{1}{2} m + \frac{1}{2} I_B C_1^2 + \frac{1}{2} I_H C_2^2}$$

$$c = \frac{\frac{1}{2} k}{\frac{1}{2} m + \frac{1}{2} I_B C_1^2 + \frac{1}{2} I_H C_2^2}$$

Integrating equation (26) for the time involved:

$$t = \frac{1}{\sqrt{c}} \left[\sin^{-1} \frac{2cx + b}{\sqrt{4ac + b^2}} - \sin^{-1} \frac{b}{\sqrt{4ac + b^2}} \right] \quad (27)$$

Equation (26) can be written for the slide velocity during bolt unlocking as:

$$\dot{x} = \sqrt{a - bx - cx^2} \quad (28)$$

Station 3 to Station 4

At this point, station 3, the slide has fully unlocked the bolt and the recoiling mass is increased by the mass of the bolt and spent cartridge case. Assuming no energy losses during impact the instantaneous change in recoil velocity is obtained from the momentum relation,

$$\dot{x}'_3 = \dot{x}_3 \left[\frac{m}{m + m_b + m_c} \right] \quad (29)$$

Letting $m_3 = m + m_b + m_c$ the slide continues compressing the hammer spring, cocking the hammer, and compressing the slide spring.

From the conservation of energy

$$\begin{aligned} \frac{1}{2} m_3 \dot{x}^2 + (F_3 x + \frac{1}{2} kx^2) + (F_{2H} c_3 x + \frac{1}{2} k_H c_3^2 x^2) + \frac{1}{2} I_H c_2^2 \dot{x}^2 &= \frac{1}{2} m_3 \dot{x}'_3{}^2 \quad (30) \\ \dot{x}^2 &= \frac{\frac{1}{2} m_3 \dot{x}'_3{}^2 - (F_3 + F_{2H} c_3)x - \frac{1}{2} (k + k_H c_3^2) x^2}{\frac{1}{2} m_3 + \frac{1}{2} I_H c_2^2} \end{aligned}$$

Let

$$a = \frac{\frac{1}{2} m_3 \dot{x}'_3{}^2}{\frac{1}{2} m_3 + \frac{1}{2} I_H c_2^2}$$



$$c = \frac{\frac{1}{2}(k + k_H c_3^2)}{\frac{1}{2} m_3 + \frac{1}{2} I_H c_2^2}$$

Integrating equation (30) and solving for t

$$t = \frac{1}{\sqrt{c}} \left[\sin^{-1} \frac{2cx + b}{\sqrt{4ac + b^2}} - \sin^{-1} \frac{b}{\sqrt{4ac + b^2}} \right] \quad (31)$$

For the velocity

$$\dot{x} = \sqrt{a - bx - cx^2} \quad (32)$$

Station 4 to Station 5

At station 4 the hammer is fully cocked and the spent case is ejected. The energy transferred to the spent case is

$$\frac{1}{2} I_c c_4^2 \dot{x}^2$$

where c_4 relates the angular and translational motion of case and slide. Extracting this energy from the total energy with the change in mass to obtain the new slide velocity $\dot{x}'_4$ at station 4,

$$\frac{1}{2} m_3 \dot{x}_4^2 - \frac{1}{2} I_c c_4^2 \dot{x}_4^2 = \frac{1}{2} (m + m_b) \dot{x}'_4^2 + \frac{1}{2} m_c \dot{x}'_4^2$$

where c_4 is the ejector location measured on the case radius. Then,

$$\dot{x}'_4 = \dot{x}_4 \sqrt{\frac{m_3}{I_c c_4^2 + (m_3)}} \quad (33)$$

After the spent case is ejected the bolt and slide continued under inertia to the end of the stroke compressing the slide spring. Therefore calling $m_c = m + m_b$

$$m_4 \ddot{x} = -(F_4 + kx)$$

$$\ddot{x} + \frac{k}{m_4} x = -\frac{F_4}{m_4} \quad (34)$$

The general solution to equation (32) is

$$x = B \sin \sqrt{\frac{k}{m_4}} t + C \cos \sqrt{\frac{k}{m_4}} t - \frac{F_4}{k} \quad (35)$$

Arbitrarily $x = 0, t = 0, \dot{x}_0 = \dot{x}_4'$

Then $0 = B(0) + C - \frac{F_4}{k}$

$$C = \frac{F_4}{k}$$

$$\dot{x} = B \sqrt{\frac{k}{m_4}} \cos \sqrt{\frac{k}{m_4}} t - C \sqrt{\frac{k}{m_4}} \sin \sqrt{\frac{k}{m_4}} t$$

$$\dot{x}_4' = B \sqrt{\frac{k}{m_4}} - C \sqrt{\frac{k}{m_4}} (0)$$

$$B = \frac{\dot{x}_4'}{\sqrt{k/m_4}}$$

Equation (35) is therefore

$$x = \frac{\dot{x}_4'}{\sqrt{k/m_4}} \sin \sqrt{k/m_4} t - \frac{F_4}{k} (1 - \cos \sqrt{k/m_4} t) \quad (36)$$



The velocity at impact is

$$\dot{x}_5 = \dot{x}_4 \cos \sqrt{k/m_4} t - \frac{F_4}{k} \sqrt{k/m_4} \sin \sqrt{k/m_4} t \quad (37)$$

Equation (37) completes the bolt opening cycle at the fully compressed station. It now remains to determine the closing cycle. Only the energy stored in the slide spring is assumed to be returned. Rebound energy usually is kept to minimum resulting in lower impact stresses and longer component cycle life.

Station 5 to Station 6

The equation describing the bolt and slide return to the point of beginning to strip the round is

$$\begin{aligned} m_4 \ddot{x} &= F_5 - kx \\ \ddot{x} + k/m_4 x &= F_5/m_4 \end{aligned} \quad (38)$$

The solution to equation (37) is

$$x = B \sin \sqrt{k/m_4} t + C \cos \sqrt{k/m_4} t + \frac{F_5}{k} \quad (39)$$

For $t=0$, $x=0$, and

$$0 = B(0) + C + \frac{F_5}{k}$$

$$0 = -\frac{F_5}{k}$$

Differentiating equation (38) with the second boundary condition

$$\dot{x} = B \sqrt{k/m_4} \cos \sqrt{k/m_4} t - C \sqrt{k/m_4} \sin \sqrt{k/m_4} t$$

$$0 = B \sqrt{k/m_4} - C \sqrt{k/m_4} (0)$$

$$B = 0$$

Substituting the constant values equation (38) becomes

$$x = \frac{F_5}{k} (1 - \cos \sqrt{k/m_4} t) \quad (40)$$

$$\dot{x} = \frac{F_5}{k} \sqrt{k/m_4} \sin \sqrt{k/m_4} t \quad (41)$$

Station 6 to Station 7

At Station 6 the mass of the next round is added to the bolt and slide reducing the velocity to $\dot{x}_6'$ as

$$\dot{x}_6' = \dot{x}_6 \left[\frac{m + m_b}{m + m_b + m_r} \right] \quad (42)$$

As the round is stripped from the case a constant stripping force must be overcome due to the frictional resistance of the round and clip. Calling

$$m_6 = m + m_b + m_r$$

$$m_6 \ddot{x} = F_6 - kx - F_f$$

$$\ddot{x} + k/m_6 x = (F_6 - F_f)/m_6 \quad (43)$$

The solution to equation (43) is



$$x = B \sin \sqrt{k/m_6} t + C \cos \sqrt{k/m_6} t + \frac{F_6 - F_f}{k} \quad (44)$$

At $t=0$, $x=0$, $\dot{x} = \dot{x}_6'$

$$0 = B(0) + C + \frac{F_6 - F_f}{k}$$

$$C = - \left(\frac{F_6 - F_f}{k} \right)$$

Differentiating equation (44)

$$\dot{x} = B \sqrt{k/m_6} \cos \sqrt{k/m_6} t - C \sqrt{k/m_6} \sin \sqrt{k/m_6} t$$

$$\dot{x}_6' = B \sqrt{k/m_6} - C \sqrt{k/m_6} (0)$$

$$B = \frac{\dot{x}_6'}{\sqrt{k/m_6}}$$

The equations of motion during stripping are

$$x = \frac{\dot{x}_6'}{\sqrt{k/m_6}} \sin \sqrt{k/m_6} t + \frac{F_6 - F_f}{k} (1 - \cos \sqrt{k/m_6} t) \quad (45)$$

$$\dot{x} = \dot{x}_6' \cos \sqrt{k/m_6} t + \frac{F_6 - F_f}{k} \sin \sqrt{k/m_6} t \quad (46)$$

Station 7 to Station 8

After the round is rammed to battery, the bolt is rotated and locked during small slide travel. From the conservation of energy

$$\frac{1}{2} m_6 \dot{x}^2 + \frac{1}{2} I_B \omega_B^2 = \frac{1}{2} m_6 \dot{x}_7^2 + (F_7 x - \frac{1}{2} kx^2) \quad (47)$$

$$\dot{x}^2 = \frac{\frac{1}{2} m_6 \dot{x}_7^2 + F_7 x - \frac{1}{2} kx^2}{\frac{1}{2} m_6 + \frac{1}{2} I_B C_1^2}$$

Let

$$a = \frac{\frac{1}{2} m_6 \dot{x}_7^2}{\frac{1}{2} m_6 + \frac{1}{2} I_B C_1^2}$$

$$b = \frac{F_7}{\frac{1}{2} m_6 + \frac{1}{2} I_B C_1^2}$$

$$c = \frac{\frac{1}{2} k}{\frac{1}{2} m_6 + \frac{1}{2} I_B C_1^2}$$

Then

$$t = \frac{1}{\sqrt{c}} \left[\sin^{-1} \frac{2 cx - b}{\sqrt{4 ac + b^2}} - \sin^{-1} \frac{-b}{\sqrt{4 ac + b^2}} \right] \quad (48)$$

$$\dot{x} = \sqrt{a + bx - cx^2} \quad (49)$$

Station 8 to Station 9

The slide is now accelerated by the drive spring for an additional small stroke, then engages the recoil piston. The equation is derived for the entire stroke neglecting the change in mass as it picks up the small piston.

$$m\ddot{x} = (F_8 - kx) \quad (50)$$



and the solution is

$$x = B \sin \sqrt{k/m} t + C \cos \sqrt{k/m} t + \frac{F_8}{k}$$

$$\dot{x} = B \sqrt{k/m} \cos \sqrt{k/m} t - C \sqrt{k/m} \sin \sqrt{k/m} t$$

For $t=0$, $x=0$, $\dot{x} = \dot{x}_8$, hence

$$0 = B(0) + C + \frac{F_8}{k}$$

$$C = -\frac{F_8}{k}$$

$$\dot{x}_8 = B \sqrt{k/m} - C \sqrt{k/m} (0)$$

$$B = \frac{\dot{x}_8}{\sqrt{k/m}}$$

$$x = \frac{\dot{x}_8}{\sqrt{k/m}} \sin \sqrt{k/m} t + \frac{F_8}{k} (1 - \cos \sqrt{k/m} t) \quad (51)$$

$$\text{and } \dot{x} = \dot{x}_8 \cos \sqrt{k/m} t + \frac{F_8}{k} \sqrt{k/m} \sin \sqrt{k/m} t \quad (52)$$

Equations (1) through (52) describe the kinematic relationships of the caliber 30 carbine rifle semi-automatic action used in the ground stake emplacement-retrieving device. Following is the numerical evaluation of the time to travel from one station to next and summarized to provide a time displacement curve of the operating slide. This data will assure safe case extraction at a time when the gas pressure is sufficiently low.

Numerical Evaluation

Equations (1) through (52) have been used to evaluate the parameters of interest for the cyclic operation of the cal. 30 rifle mechanism. The pertinent information for each station series is listed below:

A. Station 0 to Station 1

1. k = operating spring constant
= 2 lb/in.
2. F_0 = initial operating spring preload
= 2 lb.
3. A = recoil piston area
= .108 in.²
4. m = the mass of the recoiling parts
= the mass of the recoil piston, spring guide, and
operating slide
= 1.35×10^{-3} slugs-ft./in.
5. P_0 = pressure decay coefficient appearing in the exponential pressure-time relation approximation
= 16,000 psi
6. b = a constant appearing in the exponential pressure-time relation approximation
= $.67 \times 10^3$ sec.⁻¹
7. x_1 = distance from Station 0 to Station 1
= 3/16 inch



8. t = time for recoiling parts to travel from Station 0
to Station 1.

9. $\dot{x}_1$ = velocity of recoiling parts at Station 1

B. Station 1 to Station 2

1. $k = 2 \text{ lb./in.}$

2. F_1 = operating spring preload
 $= 2.375 \text{ lb.}$

3. m_1 = the mass of the spring guide and operating slide
 $= 1.30 \times 10^{-3} \text{ slugs-ft/in.}$

4. x_2 = distance from Station 1 to Station 2
 $= 3/16 \text{ inch}$

5. t = time for recoiling parts to travel from Station 1
to Station 2.

6. $\dot{x}_2$ = velocity of recoiling parts at Station 2

C. Station 2 to Station 3

1. $k = 2 \text{ lb/in.}$

2. F_2 = operating spring preload
 $= 2.75 \text{ lb.}$

3. m_2 = the mass of the spring guide and operating slide
 $= 1.30 \times 10^{-3} \text{ slugs-ft/in.}$

4. x_3 = distance from Station 2 to Station 3
 $= 1/4 \text{ inch}$

5. t = time for recoiling parts to travel from Station 2 to Station 3.
6. $\dot{x}_3$ = velocity of recoiling parts at Station 3
7. C_1 = a constant relating the translation of the slide to the rotation of the bolt
 $C_1 = 2.09 \text{ rad/in.}$
8. C_2 = a constant relating the translation of the slide to the rotation of the hammer
 $C_2 = .712 \text{ rad/in.}$
9. C_3 = a constant relating the translation of the slide to the corresponding compression (translation) of the hammer spring.
 $C_3 = .170 \text{ in/in.}$
10. I_B = the moment of inertia of the bolt about its axis of rotation
 $I_B = 2.17 \times 10^{-5} \text{ lb-sec.}^2\text{-in.}$
11. I_H = the moment of inertia of the hammer about its point of rotation
 $= 1.05 \times 10^{-4} \text{ lb-sec}^2\text{-in.}$
12. F_H = the preload in the hammer spring
 $= 16 \text{ lbs.}$
13. k_H = the spring rate of the hammer spring
 $= 13.6 \text{ lb/in.}$



D. Station 3 to Station 4

1. k = operating spring spring constant
= 2 lb/in.
2. m_3 = the mass of the spring guide, operating slide, bolt,
and spent cartridge case
= 1.88×10^{-3} slugs-ft/in.
3. $I_H = 1.05 \times 10^{-4}$ lb-sec²-in.
4. $C_2 = .712$ rad/in.
5. $C_3 = .170$ in/in.
6. F_3 = preload in the slide spring
 $F_3 = 3.25$ lb.
7. F_{2H} = preload in the hammer spring
= 16.58 lb.
8. x_4 = distance from Station 3 to Station 4
= 1-7/32 inches
9. t = time for recoiling parts to travel from Station 3
to Station 4
10. $\dot{x}_4$ = velocity of recoiling parts at Station 4

E. Station 4 to Station 5

1. $k = 2$ lb/in.
2. $\dot{x}'_4$ = the velocity of the recoiling parts at Station 4
after the ejection of the spent case

3. I_C = the moment of inertia of the spent case about its point of ejection
 $= 2 \times 10^{-5} \text{ lb-sec}^2\text{-in.}$
4. m_4 = the mass of the slide, guide, and bolt
 $= 1.85 \times 10^{-3} \text{ slugs-ft/in.}$
5. F_4 = the preload in the slide spring
 $= 5.69 \text{ lb.}$
6. x_5 = the distance from Station 4 to Station 5
 $= 1\text{-}1/16 \text{ inch}$
7. t = the time for the recoiling parts to travel from Station 4 to Station 5.
8. $\dot{x}_5$ = the velocity of the recoiling parts at Station 5.

F. Station 5 to Station 6

1. $k = 2 \text{ lb/in.}$
2. m_5 = the mass of the slide, guide, and bolt
 $= 1.85 \times 10^{-3} \text{ slugs-ft/in.}$
3. F_5 = the preload in the slide spring
 $= 6.82 \text{ lb.}$
4. x_6 = the distance from Station 5 to Station 6
 $= 1\text{-}1/16 \text{ inches}$
5. t = the time for the recoiling parts to travel from Station 5 to Station 6



6. $\dot{x}_6$ = the velocity of the recoiling parts at Station 6

G. Station 6 to Station 7

1. $k = 2 \text{ lb/in.}$

2. $\dot{x}'_6$ = the velocity of the recoiling parts at Station 6
after the addition of the cartridge mass.

3. F_f = force to strip one cartridge from the magazine
= 4 lb.

4. m_6 = the mass of the slide, guide, bolt and cartridge.

5. F_6 = the preload in the slide spring
= 5.69 lb.

6. x_7 = the distance from Station 6 to Station 7
= .25 inch

7. t = the time for the recoiling parts to travel from
Station 6 to Station 7

8. $\dot{x}_7$ = the velocity of the recoiling parts at Station 7

H. Station 7 to Station 8

1. $k = 2 \text{ lb/in}$

2. m_7 = the mass of the spring guide and operating slide
= 1.30×10^{-3} slugs-ft/in.

3. F_7 = the preload in the slide spring
= 3.25 lb.

4. $I_B = 2.17 \times 10^{-5} \text{ lb-sec}^2\text{-in.}$

5. $C_1 = 2.09 \text{ rad/in.}$

6. $x_8 = \text{the distance from Station 7 to Station 8}$
 $= .25 \text{ in.}$

7. $t = \text{the time for the recoiling parts to travel from}$
 $\text{Station 7 to Station 8}$

8. $\dot{x}_8 = \text{the velocity of the recoiling parts at Station 8}$

H. Station 8 to Station 9

1. $k = 2 \text{ lb/in.}$

2. $m_8 = \text{the mass of the spring guide and operating slide}$
 $= 1.30 \times 10^{-3} \text{ slugs-ft/in.}$

3. $F_8 = \text{the preload in the slide spring}$
 $= 2.75 \text{ lb.}$

4. $x_9 = \text{the distance from Station 8 and Station 9}$

5. $t = \text{the time for the recoiling parts to travel from}$
 $\text{Station 8 to Station 9}$

6. $\dot{x}_9 = \text{the velocity of the recoiling parts at Station 9.}$

Summary of Bolt-Slide Recoil and Counterrecoil

<u>Station</u>	<u>Δt</u> <u>(millisec.)</u>	<u>t</u> <u>(millisec.)</u>	<u>x</u> <u>(inches)</u>	<u>$\dot{x}$</u> <u>(ft/sec)</u>
	.1			
0		.1	0	0
	.64			
1		.74	.1875	55.4
	.28			
2		1.02	.375	55.3
	.39			
3		1.41	.625	52.4
	2.80			
4		4.21	1.844	34.2
	2.51			
5		6.73	2.907	33.5
	26.0			
6		32.7	1.844	6.41
	15.4			
7		48.1	.625	5.56
	3.5			
8		51.6	.375	6.10
	4.1			
9		55.7	0	6.70

The summary of the parameters of interest is found on page 39 . A plot of the displacement of the recoiling parts versus time is shown on Fig. 4 . On this plot the location and occurrences at each of the stations are indicated.

It should be pointed out that the time, t , was measured .1 milliseconds after the cal. 30 cartridge was fired. This was done in order that the area under the exponential pressure-time relation approximation most closely approaches that under the calculated pressure-time relation.

From these calculations, it can be seen that the camming open of the bolt is begun (Station 2) .921 milliseconds after the firing of the cartridge. It was calculated that .65 milliseconds after the first cartridge is fired the piston of the impact hammer uncovers the exhaust ports. This allows the exhaust gases to expand into the atmosphere, which drops the chamber pressure to a safe extraction level. The significance of this is that the pressure in the chamber is at such a low value that the exhaust gases will have no possibility of exhausting through the chamber of the rifle mechanism either during or after the uncaming (unlocking) of the bolt.

Note that the initial chamber pressure of 16,000 psi was assumed to act on the recoil piston. In the actual mechanism, however, the gases must travel through a nozzle before acting on this piston, with a corresponding considerable pressure drop. This pressure drop in the actual mechanism will increase the calculated cycling time of the mechanism. Also

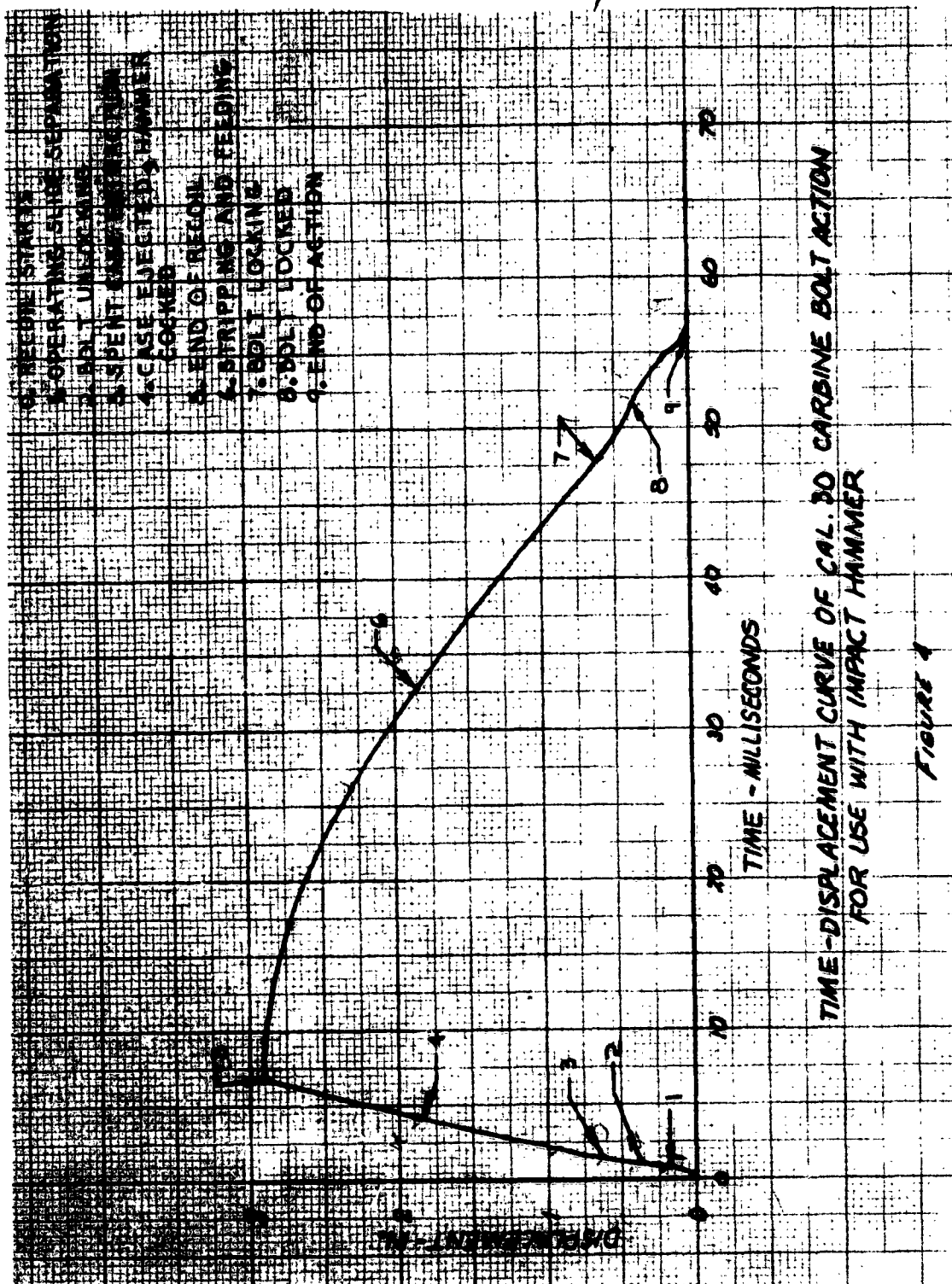


FIGURE 4

friction in the mechanism was neglected (with the exception of that required to strip the cartridge from the magazine). This friction in the actual mechanism will also increase the calculated cycling time of the mechanism. For these reasons it can be stated that the calculated cycle time is quite low compared to the actual cycling time and that, in the actual application, the piston of the impact hammer will uncover the exhaust port well before the uncamming (unlocking) action of the bolt begins.



E. Weight Analysis

The weights of the major components of the impact hammer have been calculated and are tabulated below:

<u>Item</u>	<u>Weight (lb)</u>
Caliber 30 rifle mechanism including full magazine- - - - -	2.66
Ram* - - - - -	1.95
Barrel - - - - -	3.09
Flange at top of Barrel - - - - -	.50
Chamber - - - - -	9.70
Handles - - - - -	1.88
Adaptor for Large Stake - - - - -	2.32
Adaptor for Small Stake - - - - -	1.50

The total of all above = 23.65 pounds.

* NOTE: Throughout the remainder of the report the terms "ram" and "piston" are used interchangeably.

F. Stress Analysis

This section of the report discusses several examples of what are considered the most severe stress problems which might be encountered under any conditions of use. Most of the situations considered are not likely to occur with any frequency but the fact that analysis indicates safe stress levels under these conditions indicates a desirable degree of safety.

A. Barrel

1. The barrel of the impact hammer must first be designed to withstand the most severe condition expected to be encountered in operation. This would occur if the operator actuated the firing mechanism with the impact hammer not positioned on the stake. If this occurred, the barrel of the impact hammer would be subjected to the whole of the available kinetic energy of the piston (after energy losses).

For the purposes of analysis, the following assumptions will be made:

- a. Upon impact with the barrel flanges, some of the kinetic energy of the piston will be dissipated. ⁽²⁾
- b. The effects of the shock absorbers will not be considered. (In other words, the ram-barrel impact will be treated as metal-to-metal.)

(2) "Formulas for Stress & Strain;" Raymond J. Roark; Third Edition; 1954; page 331.



c. The barrel metal absorbs all of the available kinetic energy of the ram as strain energy.

d. The velocity of the barrel is zero at all times.

The kinetic energy of the piston at the instant before impact (KE_r) is:

$$KE_r = 1/2 m_r v_r^2$$

where

m_r = the mass of the ram

= .0621 slugs

v_r = the velocity of the ram at the instant before impact

= 125 ft/sec.

$$\therefore KE_r = 1/2 (.0621)(125)^2 \\ = 485 \text{ ft-lb.}$$

Upon impact some of the kinetic energy of the piston will be dissipated. "This loss is most conveniently taken into account by multiplying the available energy by a factor K --- ." (2)

For the case under consideration:

$$K = \frac{1 + 1/3 m_1/m}{(1 + 1/2 m_1/m)^2} \quad (2)$$

(2) "Formulas for Stress and Strain;" Raymond J. Roark; Third Edition; 1954; pps. 331-332.

where

m_1 = the mass of the barrel

= .096 slugs

m = the mass of the ram

= .0621 slugs

K = the energy loss factor

$$(E_{\text{lost}} = K(KE_r))$$

$$\therefore K = \frac{1 + \frac{.096}{3(.0621)}}{\left[1 + \frac{.096}{2(.0621)}\right]^2}$$

$$= \frac{1.515}{3.14}$$

$$K = .482$$

$$E_{\text{lost}} = K(KE_r)$$

$$= .482(485)$$

$$E_{\text{lost}} = 234 \text{ ft-lb.}$$

The available kinetic energy of impact (KE_A) is therefore:

$$KE_A = KE_r - E_{\text{lost}}$$

$$= 485 - 234$$

$$KE_A = 251 \text{ ft-lbs.}$$

According to assumption 3, all of the available kinetic energy of impact of the ram (KE_A) is converted to strain energy in the barrel (U_b).



$$U_b = KE_A = 1/2 \frac{F_{\max}^2}{K_b} \quad (1)$$

where

$F_{\max}$ = the maximum force in the barrel resulting from the impact.

$$K_b = \text{the stiffness factor for the barrel} \\ = \frac{A_b E_b}{L_b}$$

$$L_b = \text{the length of the barrel} \\ = 4 \text{ inches}$$

$$A_b = \text{the cross-sectional area of the barrel} \\ = 2.36 \text{ in.}^2$$

$$E_b = \text{Youngs' Modulus for the barrel (steel)} \\ = 3 \times 10^7 \text{ lb/in.}^2$$

$$\therefore K_b = \frac{2.36 (3 \times 10^7)}{4}$$

$$K_b = 1.77 \times 10^7 \text{ lb/in.}$$

$$U_b = 1/2 \frac{F_{\max}^2}{K_b}$$

$$F_{\max}^2 = 2 K_b U_b \\ = 2(12) 1.77 \times 10^7 (251)$$

$$F_{\max}^2 = 10.65 \times 10^{10}$$

$$F_{\max} = 326,000 \text{ lbs.}$$

(1) "Design of Machine Element;" M. F. Spotts, 2nd Ed., p. 383

The barrel will be investigated for failure in three areas:

1. Axial stresses in the barrel
2. Shearing stresses at the junction between the flange and barrel
3. Bending stresses at the junction between the flange and barrel.

1. Axial stress in barrel = σ_B

$$\sigma_B = \frac{F_{\max}}{A}$$

where

$$\begin{aligned} A &= \text{the cross-sectional area of the barrel} \\ &= 2.36 \text{ in.}^2 \end{aligned}$$

$$\therefore \sigma_B = \frac{326,000}{2.36}$$

$$\sigma_B = 1.38 \times 10^5 \text{ lb/in.}^2$$

2. Shearing stress at junction = τ_j

$$\tau_j = \frac{F_{\max}}{A_j}$$

where

$$A_j = \text{the shear area at the junction between the barrel and flange}$$

$$= \pi (2.7) .6$$

$$= 5.1 \text{ in.}^2$$

$$\tau_j = \frac{3.26 \times 10^5}{5.1}$$

$$\tau_j = 6.4 \times 10^4 \text{ lb/in.}^2$$

3. Maximum bending stress at junction = S_j .

(Flange treated as a plate with outer edge rigidly fixed and uniform pressure acting over surface.)

$$S_j = \frac{3W}{4t^2} \left[a^2 - 2b^2 + \frac{b^4(m-1) - 4b^4(m+1) \log a/b + a^2 b^2 (m+1)}{a^2 (m-1) + b^2 (m+1)} \right] \quad (2)$$

where

 W = unit applied load

$$= \frac{326 \times 10^3 \times 4}{\pi(2.7^2 - 1.4^2)}$$

$$W = 78 \times 10^3 \text{ lb/in.}^2$$

 t = thickness of flange

$$= .6 \text{ inch}$$

 a = outer radius of flange

$$= 1.35 \text{ inch}$$

 b = inner radius of flange

$$= .7 \text{ inch}$$

 m = 1/Poisson's ratio

$$= 3.33$$

$$S_j = \frac{3(78 \times 10^3)}{4(.36)} \left[1.82 - .98 + \frac{.24(2.33) - .96(4.33) + .89(4.33)}{1.82(2.33) + .49(4.33)} \right]$$

$$S_j = 149 \times 10^3 \text{ lb/in.}^2$$

(2) "Formulas for Stress & Strain;" Raymond J. Roark; 3rd Ed., 1954, pps. 194 and 199

Since 4340 heat-treated steel is used for the barrel material (ultimate strength = 250,000 lb/in²), it can be seen that the barrel will be structurally safe for the condition described. Note that the energy dissipating effect of the shock absorbers was completely neglected. (In other words, metal-to-metal impact was assumed.) Also, it was assumed that the ram transferred all of its available kinetic energy to the barrel flanges. Actually the ram would have a quite considerable rebound velocity and would absorb a considerable amount of its available kinetic energy. For these reasons it can be safely said that the present barrel design will be completely safe for any foreseeable condition which could be encountered in operation.

According to the maximum cyclic stress versus number of stress cycles curve, it can be seen that the barrel would endure nearly 10,000 stress cycles under this design condition.⁽³⁾ Under normal operating circumstances, it is unlikely the barrel would be subjected to this design condition for this many cycles.

2. The barrel design will now be investigated with respect to its expected life under usual operating conditions.

For the purposes of analysis, the following assumptions will be made:

⁽³⁾ "Maximum Stress versus Number of Stress Cycles Curve for 4340 heat-treated steel" (ultimate strength=250,000 lb/in²); International Nickel Co. -- "Product Engineering," October 1953



- a. Upon impact with the stake, some of the kinetic energy of the ram is dissipated.
- b. Half of the available kinetic energy of the ram is used to drive in the stake and half is transferred to the barrel flanges.
(Effect of shock absorbers not considered.)
- c. The velocity of the barrel is zero at all times.
- d. The small hardened steel stake is being used.

The kinetic energy of the ram at the instant before impact is:

$$KE_r = 485 \text{ ft-lb.}$$

Upon impact with the stake some of the kinetic energy of the ram will be dissipated. "This loss is most conveniently taken into account by multiplying the kinetic energy (KE_r) by a factor K ----."

For the case under consideration:

$$K = \frac{1 + 1/3 \frac{m_1}{m}}{(1 + 1/2 \frac{m_1}{m})^2}$$

where

m_1 = the mass of the stake
= .09 slugs

m = the mass of the ram
= .0621 slugs

K = the loss factor; ($E_{\text{lost}} = K(KE_r)$)

Therefore:

$$K = \frac{1 + 1/3 \left(\frac{.09}{.0621} \right)}{\left[1 + 1/2 \left(\frac{.09}{.0621} \right) \right]^2}$$

$$= \frac{1.482}{2.98}$$

$$K = .496$$

$$\begin{aligned} E_{\text{lost}} &= K(KE_r) \\ &= .496(485) \end{aligned}$$

$$E_{\text{lost}} = 241 \text{ ft-lb.}$$

The available kinetic energy is, therefore:

$$\begin{aligned} KE_A &= KE_r - E_{\text{lost}} \\ &= 485 - 241 \end{aligned}$$

$$KE_A = 244 \text{ ft-lb.}$$

According to assumption 2, one-half of KE_A will be absorbed by the barrel as strain energy (U_b).

$$\therefore U_b = 122 \text{ ft-lb.} = 1/2 \frac{F_{2 \text{ max}}^2}{K_b}$$

$$F_{2 \text{ max}}^2 = 2(12) 1.77 \times 10^7 (122)$$

$$F_{2 \text{ max}}^2 = 5,200 \times 10^7$$

$$F_{2 \text{ max}} = 2.27 \times 10^5 \text{ lb.}$$

As was done previously, the barrel will be investigated for the three areas of failure.

$$1. \text{ Axial stress in the barrel} = \sigma_{B_2}$$

$$\sigma_{B_2} = \sigma_B \frac{F_{2 \text{ max}}}{F_{\text{max}}}$$



$$\tau_{B_2} = 1.38 \times 10^5 \left(\frac{2.27 \times 10^5}{3.26 \times 10^5} \right)$$

$$\tau_{B_2} = 9.6 \times 10^4 \text{ lb/in}^2$$

2. Shearing stress at junction = τ_{j_2}

$$\begin{aligned} \tau_{j_2} &= \tau_j \left(\frac{F_{2 \text{ max}}}{F_{\text{max}}} \right) \\ &= 6.4 \times 10^4 \left(\frac{2.27 \times 10^5}{3.26 \times 10^5} \right) \end{aligned}$$

$$\tau_{j_2} = 4.45 \times 10^4$$

3. Maximum bending stress at junction = s_{j_2}

$$\begin{aligned} s_{j_2} &= s_j \left(\frac{F_{2 \text{ max}}}{F_{\text{max}}} \right) \\ &= 149 \times 10^3 \left(\frac{2.27 \times 10^5}{3.26 \times 10^5} \right) \end{aligned}$$

$$s_{j_2} = 103,500 \text{ lb/in.}^2$$

Through reference to the maximum cyclic stress versus number of stress cycles curve, it can be seen that the barrel would endure nearly one million stress cycles under this condition.

Again it must be pointed out that the assumptions used were quite conservative. It is highly likely that in operation practically all of the available kinetic energy of the ram is transferred to the stake, with little or none transferred to the barrel. Also any energy about to be transferred to the barrel is decreased by the effects of the rubber shock absorbers and

recoil springs. In addition to this, it is not transferred in metal-to-metal contact, as was assumed. Taking these factors into consideration it can be safely said that the barrel of the proposed mechanism will safely withstand one million cycles of normal operation.

P. Piston of the Impact Hammer

As with the case of the barrel, the piston will be designed to withstand the most severe condition expected to be encountered in operation. This would occur if the operator continuously actuated the firing mechanism while attempting to drive the 2.9 pound hardened steel stake into an impenetrable medium, such as rock.

The following assumptions will be made for the purpose of analysis:

1. When the ram impacts with the stake, some of the impact kinetic energy of the ram is dissipated.
2. After impact, the ram absorbs all of its available kinetic energy as strain energy, with no energy transferred to the stake.
3. No contact occurs between the ram and barrel flanges.
4. The velocity of the barrel is zero at all times.

The kinetic energy of the ram at the instant before impact is:

$$KE_r = 485 \text{ ft-lb.}$$



Upon impact with the stake some of the kinetic energy of the ram will be dissipated. This loss is most conveniently taken into account by multiplying the available energy (KE_r) by a factor K .

For the case under consideration:

$$K = \frac{1 + 1/3 \ m_1/m}{(1 + 1/2 \ m_1/m)^2}$$

$$K = .496$$

$$E_{\text{lost}} = 241 \text{ ft-lb.}$$

The available kinetic energy of impact (KE_A) is, therefore:

$$KE_A = 244 \text{ ft-lb.}$$

The ram will absorb all of its available kinetic energy of impact (assumption 2) as strain energy.

$$\text{Therefore, } KE_A = U_r = 244 \text{ ft-lbs.}$$

$$U_r = 1/2 \frac{F_{\text{max}}^2}{K_r}$$

where

F_{max} = the maximum force in the ram resulting from the impact

$$K_r = \frac{A_r E_r}{L_r}$$

= stiffness factor for the ram

where L_r = length of the ram
= 4 inches

A_r = cross-sectional area of the ram

$$= \frac{\pi}{4} (1.25)^2$$

$$A_r = 1.23 \text{ in.}^2$$

$$\begin{aligned} E_r &= \text{Young's Modulus for the ram (steel)} \\ &= 3 \times 10^7 \text{ lb/in.}^2 \end{aligned}$$

$$K_r = \frac{A_r E_r}{L_r}$$

$$= \frac{1.23(3 \times 10^7)}{4}$$

$$K_r = 9.22 \times 10^6 \text{ lb/in.}$$

$$\begin{aligned} F_{\max}^2 &= 1/2 K_r U_r \\ &= 1/2(12) 9.22 \times 10^6 (244) \\ &= 13,500 \times 10^6 \end{aligned}$$

$$F_{\max}^2 = 135 \times 10^8$$

$$F_{\max}^2 = 11.6 \times 10^4$$

The maximum axial stress in the ram ($\sigma_{\max}$) is:

$$\begin{aligned} \sigma_{\max} &= \frac{F_{\max}}{A} \\ &= \frac{11.6 \times 10^4}{1.23} \end{aligned}$$

$$\sigma_{\max} = 9.43 \times 10^4 \text{ lb/in.}^2$$

$$\sigma_{\max} = 94,300 \text{ lb/in.}^2$$



The material used for the ram has been designated as 4340 heat-treated steel (ultimate stress = $250,000 \text{ lb/in}^2$). Reference (3) indicates that 4340 heat-treated steel will withstand one million stress cycles at a maximum stress of $102,000 \text{ lb/in}^2$. The calculated maximum stress in the ram is $94,300 \text{ lb/in}^2$. The ram, therefore, will endure over one million stress cycles at a maximum stress of $94,300 \text{ lb/in}^2$.

It should be pointed out that the stress calculation for the ram is quite conservative. This is because the ram was assumed to absorb all of its kinetic energy as strain energy, with no energy (or force) imparted to the stake or hammer frame (barrel).

The fact that the calculations indicate that the ram will endure over one million cycles of attempting to drive the hardened steel stake into rock certainly indicates that it will function satisfactorily for over one million cycles of normal operation.

C. Stress Analysis of Stakes

It should be pointed out that the stresses produced in the stakes by the impact of the piston were not analyzed. The reason for this is that although it is conceivable that the yield stress (or even the ultimate strength) of the stakes' heads could be exceeded by the impact, the most important design consideration is the tendency of the head of each stake to "mushroom" due to the force of the impact. This "mushrooming" could tend to jam the stake in its guide. At this point it is felt that this tendency is not considerable, and, at any rate, it would be better to analyze it experimentally rather than analytically.

G. Experimental Program

A limited experimental program has been conducted and is still in progress. Attempts are being made to measure the maximum tolerable recoil a man can sustain. This has a direct relation on the maximum energy that can be delivered to the stake and on the optimum design of the impact hammer.

Another parameter which can reasonably be evaluated is the depth of penetration at the various high level impacts. From an average penetration per blow an estimate of the expected number of rounds per stake will be obtained. This may vary with the type soil encountered but a maximum number can be established.

Also of importance is the amount of propellant residue from each round. This should have negligible effect on performance due to the abrupt change from a small initial volume to a large exhaust reservoir. Nevertheless an attempt will be made to establish the effects of residue or wadding deposits from each round and the possibilities of exhausting these deleterious effects.

To date the experimental program has been limited to some very basic drop energy tests as described in the First Quarterly Progress Report and to a simple mock-up device weighing 25 pounds to establish recoil jump with a 5 pound piston fired at nearly 100 ft./sec. Approximately four tests were made where the piston was guided to impact the Ground Stake GP-112/G setting on a sand-clay soil. Penetration of 4 to 5 inches



was obtained per shot but the recoil effect was to raise the 20-pound hammer nearly 3 to 4 feet off the stake. By adding 25 pounds additional weight, simulating an operator's effort, the recoil was reduced to raising the hammer 4 to 6 inches off the stake. The current configuration with a 2-pound piston is expected to significantly reduce the recoil effects since the momentum is reduced to about 1/2 that of the 5-pound piston.

Future testing will include the determination of a reasonable force a man can exert with the Ground Stake GP-113/G in the various positions of heights and angles to resist and reduce recoil effects.

V. CONCLUSIONS

During this report period extensive design analysis has been conducted to establish the optimum detail design for the ballistic impact hammer. The general characteristics of a device have been developed although some detail considerations remain to be established. Limited experimental work was conducted with a ballistic test model. Work is continuing upon the design layout, additional tests and analysis should be completed before the design plan is formulated. This work is being performed at this time.

The primary consideration has been the ability of the design approaches considered to satisfy Signal Corps requirements for an emplacement-retrieving device. Parameters not directly related to performance such as flash, noise, human engineering, safety, logistics, maintenance, and simplicity have also been considered.



VI. PROGRAM FOR NEXT INTERVAL

During the next quarterly interval the following items are scheduled to be accomplished:

The design analysis will be completed and a design plan formulated to fully outline the design of a development test model. This design plan will be prepared in report form and submitted to USAKLRDL for review and design approval. Visualization data consisting of artists' concepts of the device will be prepared and delivered.

Upon approval of design plan, work will be started on fabrication of an emplacement-retrieving device for development tests. The scope of these tests will be outlined in the design plan.

Monthly progress reports will be prepared and delivered according to schedule.

Final copies of Second Quarterly Progress Report will be submitted upon draft approval.

Draft of Third Quarterly Progress Report will be submitted according to schedule.

VII. IDENTIFICATION OF KEY TECHNICAL PERSONNEL

Brief resumes of key technical personnel who have contributed to this program are shown on the following pages.



Irwin R. Barr, Vice President - Development

Mr. Barr is an original member of AAI, first serving as Chief Design Engineer, and attaining his present position in 1960. Between 1952 and 1960, he directed the company's ordnance design activities in the capacity of Chief Ordnance Engineer. He has had extensive experience in both ordnance and missile development, being co-inventor of the Viking rocket control system and holding patents on a variety of ordnance items, including a vehicle, turrets, bearings, special fin stabilized ammunition, and several automatic guns. Outstanding examples of ordnance material developed under his direction at AAI include the T175E1 Dual Purpose Machine Gun, a series of turret-type cupolas, and the T92 mm Gun Tank.

Mr. Barr is a graduate in Aeronautical Engineering (1940) from the Casey Jones School of Aeronautics. He worked for two periods, 1940-1944 and 1946-1950, for the Glenn L. Martin Company, primarily in missiles. During the last of these two periods he served in the U. S. Army Air Force, receiving the Army commendation medal for design of aircraft rocket launchers.

Nicholas J. LaCosta, Manager, Explosive Ordnance Department

Education: Casey Jones School of Aeronautics, A.E.

Work History: 1959 to present, Aircraft Armaments, Inc.
Manager, Explosive Ordnance Department. Responsible for directing and coordinating all activities of the Explosive Ordnance Department. These activities include the study, design, development, test and manufacture of systems and components involving propellant actuation, interior and exterior ballistics, terminal ballistics, kinematics, gas dynamics and explosive train design, applications and feasibility studies.

1951-1959, Aircraft Armaments, Inc.
Project Manager responsible for stabilized ammunition development program, Corporal Warhead development, a number of explosively actuated bomb racks, canopy removers, thrusters and initiators and control systems. Recent responsibilities also include new infantry weapon concepts, recoilless rifle program, special weapons tools and an automatic sequencing device.

1939-1951, The Glenn L. Martin Company
Layout Engineer, Group Engineer, Design Specialist.

1938, Bell Aircraft Company. Layout Engineer.

1936-1937, Consolidated Aircraft Company. Draftsman.



Richard G. Strickland, Senior Design Engineer

Education: City College of New York, B.S., 1953
University of Maryland, Graduate Work.

Work History: 1956 to present, Aircraft Armaments, Inc.
Project Manager responsible for design and development of tools and procedures for special weapons handling and disassembly, underwater timing device, and special devices for underwater recovery operations. Design of special tools for remote handling of explosive devices. Survey and study of CAD in all U.S. aircraft and missiles. Experimental study of recoilless rifle internal ballistics; study of recoilless weapon optimization. Spotting round development. Study of missile fuel spark compatibility. Development of timer concepts, applied explosives research. Design and development of underwater guns and ammunition, measurement of effects of underwater explosions. Responsible for development and fabrication of M125 Demolition Firing Device and associated test equipment. Study and development of wide area mine fuze. Design and development of ground stake emplacing device and earth anchor concepts.

1954-1956, U. S. Navy Special Weapons Disposal School. Special Weapons Instructor. Nuclear Physics, Health Physics, Weapon Principles, Weapon Configuration and Weapon Effects.

1953 - 1954, U. S. Navy Special Weapons Disposal Officer. Field test of special weapons. Attended Nuclear Weapons Training Schools at Field Command AFSWP, Picatinny Arsenal and Lowry Air Force Base.

Theodore G. Stastny, Senior Engineer

Education: Georgia Institute of Technology, BME, 1958
Georgia Institute of Technology, MME, 1959

Work History: 1958 to present, Aircraft Armaments, Inc.
Responsible for analytical and experimental evaluations of the internal ballistics of recoilless rifles and both hybrid and solid-fuel rockets. Study and design of various cartridge actuating devices employed in ejection mechanisms, parachute deployment, and separation problems. Trajectory, stability and aerodynamic analysis of projectiles and gun boosted rockets. Stress and cycle analysis, including dynamics and kinematics of automatic weapons. Responsible for compiling theories of detonation and explosive phenomena, and fragmentation of bombs and grenades for handbook usage. Designer of "Lo to Hi Pressure Admission Valve" of hot gases (Pat. Pend.). Heat transfer and stress analysis of large, high pressure flat-end chemical reactor. Investigation of underwater ballistics on Mach 0.1 to 0.5 projectiles covering velocity decay, vaporization bubble, stability and nose configurations. Responsible for extensive literature survey and analysis covering techniques of rapid ground anchor emplacement devices including related studies of soil mechanics and soil properties.

1954-1958, Koppers Company, Inc., concurrent with schooling. Design and study of gas cleaning equipment, design of gear teeth for coupling misalignment, studies of sealing ring and piston ring problems.

Technical Society: Member and past officer of American Rocket Society, Maryland Section.



Arthur C. Powell, Senior Engineer

Education: Massachusetts Institute of Technology,
B.S. in M.E., 1948
U. S. Navy Electronics Technician's Schools,
1944-1945.

Work History: 1958 to present, Aircraft Armaments, Inc. Senior
Engineer. Design and test of a parachute sequencing
device, a hydraulic power supply, recoilless weapons,
cartridge actuated devices, a remote arming device,
escape system components, hydraulic timers, and
production tooling.

1952-1958, Van Zelm Associates, Inc. Design
Engineer on a variety of aircraft arresting gears,
target drone catapults, and bridle arresting
equipment. Field test engineer on aforementioned
equipment.

1949-1952, Friez Instrument Division, Bendix
Aviation Corporation. Design and test engineering
on numerous weather, recording and electrical
instruments.

1948-1949, Industrial Research Laboratories.
Design, construction, and test of a filled-
bottle goods inspection machine.

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