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ENGINEERING DESIGN
HANDBOOK

GUNS SERIES

AUTOMATIC WEAPONS

HEADQUARTERS, U.S. ARMY MATERIEL COMMAND FEBRUARY 1970

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ENGINEERING DESIGN HANDBOOK
AUTOMATIC WEAPONS

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LIST OF SYMBOLS

A = coefficient in equation defining the tube recoil travel

A_b = bore area

A_c = peripheral surface contact area between case and chamber; operating cylinder piston area

A_i = differential area under pressure-time curve

A_o = orifice area

A_{pt} = area under pressure-time curve

$A_1, A_2, \ldots A,$ = coefficients of x in a series

a = general expression for linear acceleration; major axis of elliptical cam; length of short segment of rectangular coil spring

a_a = average linear acceleration

a_c = acceleration of chutes

= counterrecoil acceleration

a_{crc} = major axis of counterrecoil portion of elliptical cam

a_d = tangential acceleration of cam roller on cam path

e = entrance unit acceleration; exit unit acceleration

a_n = nor: al acceleration of cam roller on cam path

= recoil acceleration; retainer acceleration

a_{rec} = major axis of recoil portion of elliptical cam; slide travel during slide deceleration

a_s = slide acceleration

at = acceleration of transfer unit

B = coefficient in equation defining recoil travel; collective term in defining time during polytropic expansion of gas

b = minor axis of an elliptical cam; length of long segment of rectangular coil spring; spring width

b_{cr} = minor axis of counterrecoil section of elliptical cam

b_r = minor axis of recoil section of elliptical cam

C = orifice coefficient

C_r = end clearance of round

D = mean coil diameter

D_b = bore diameter

D_r = horizontal distance between trunnion and rear support

D_c = diameter of cartridge case base

D_d = drum diameter

d = wire diameter

d_c = gas cylinder diameter

d_p = gas port diameter; piston diameter of operating cylinder

dt = differential time

dx = differential distance

E = modulus of elasticity; energy

E_a = energy of ammunition belt

E_b = bolt energy; combined energy of buffer and driving springs

E_{bc} = counterrecoil energy of buffer spring

LIST OF SYMBOLS (Con't.)

E_c = modulus of elasticity of case; energy at gas cutoff

= counterrecoil energy

E_{cs} = energy needed to bring slide up to speed

E_{crb} = counterrecoil energy of barrel at end of buffer action

E_{cri} = counterrecoil energy at beginning of increment

E_{crt} = maximum counterrecoil energy of barrel

E_d = energy of rotating parts, drum energy

E_{dcr} = counterrecoil energy of drum

E_{ds} = energy of drum-slide system

E_e = ejection energy of case

E_i = input energy of each increment; total energy at any given increment

E_o = energy of operating rod

E_{oi} = energy of operating rod at gas cutoff

E_r = energy of recoiling parts; energy to be absorbed by mount

E_{rb} = recoil energy of bolt

E_s = energy transferred from slide to driving spring; driving spring energy

E_{sc} = total work done by all springs until start of unlocking of bolt

E_{scr} = slide counterrecoil energy

E_{sr} = slide recoil energy

E_{ss} = energy transferred from driving spring to **slide**

E_t = barrel energy

E_{to} = barrel energy when bolt is unlocked

E_ϵ = energy loss attributed to spring system

E_μ = total energy loss caused by friction

$E_{\mu d}$ = energy loss in drum

$E_{\mu s}$ = energy loss in slide

e = base of natural logarithms

F = general expression for force; driving spring force; spring force at beginning of recoil

F_a = general expression for average force; average driving spring force; axial inertial force

F_{ab} = average force of spring-buffer system

F_{as} = average force of spring system

F_b = buffer spring force

F_c = operating cylinder force

F_{ca} = average operating cylinder force

F_{cr} = counterrecoil force

F_e = general expression for effective force, exit force, entrance force, maximum extractor load to clear cartridge case

F_{eb} = effective force on barrel

F_{es} = load when cartridge is seated

F_g = residual propellant gas force; propellant gas force

F_i = initial spring force

F_L = transverse force on locking lug; reaction on bolt

F_m = maximum spring force; spring force at end of recoil

LIST OF SYMBOLS (Con't.)

F_{mb} = maximum force of spring-buffer system

F_{ms} = maximum force of barrel and driving spring

F_{mt} = maximum force of barrel spring; of adapter

F_o = minimum spring force; minimum operating load

F_{ob} = initial buffer spring system force

F_{obs} = initial buffer spring force

F_{ot} = initial force of barrel spring; of adapter

F_p = cam roller pin load

F_r = average force during recoil

F_s = sear spring force; centrifugal force of cartridge case or round

F_{sh} = horizontal component of safety spring force

F_{sv} = safety spring force

F_t = force of barrel spring; of adapter; vertical reaction of trigger spring pin

$\bar{F}_t$ = average adapter force for time interval dt

F_{tb} = barrel spring force at end of propellant gas period

F_x = resultant force of x-axis

F_y = resultant force of y-axis

$F_{\mu s}$ = frictional force

Fdt, $F\Delta t$ = general expressions for differential impulse

f_r = rate of fire

$f(T_c/T)$ = function of the ratio of compression time to surge time

G = torsional modulus; shear modulus

g = acceleration due to gravity

H = command height

HP = horsepower

H_s = solid height of spring

h = depth of magazine storage space

h_t = distance between trunnion and pintle-leg intersection

I = area moment of inertia; general term for mass moment of inertia

I_b = mass moment of inertia of bolt

I_d = mass moment of inertia of drum; mass moment of inertia of all rotating parts

I_{de} = effective mass moment of inertia of drum

J = area polar moment of inertia

K = spring constant, general; driving spring constant

= coefficient in gas flow equation

K_b = combined spring constant during buffing

K_{bs} = buffer spring constant

K_s = combined constant of barrel and driving springs

K_t = spring constant of barrel spring, of adapter

K_w = coefficient in the rate of gas flow equation

K_x = directional coefficient in F_x equation

K_y = directional coefficient in F_y equation

k = ratio of specific heats; radius of gyration; bolt polar radius of gyration

L = general expression for lengths; length of recoil; bolt travel; length of flat spring

LIST OF SYMBOLS (Con't.)

L_b = length of buffer spring travel

L_{bt} = length of total bullet travel in barrel

L_c = axial length of cam; length of slide travel; total peripheral length of cam

L_d = decelerating distance prior to buffer contact; operating distance of operating rod spring; length of drum; driving spring travel

L_e = extractor length

L_f = length of front pintle leg

L_p = location of gas port along barrel

L_r = length of round; length of rear pintle leg

L_t = tappet travel; barrel spring operating deflection

M = mass, general; mass of accelerating parts; bending moment

M_a = mass of round; mass of ammunition unit

M_{ae} = effective mass of ammunition

M_b = mass of bolt

M_{cc} = mass of cartridge case

M_d = mass of drum

M_{de} = effective mass of drum and ammunition belt; effective mass of rotating drum

M_e = effective mass, general; of extractor unit

M_{ee} = effective mass of extractor unit

M_{ej} = mass of ejector

M_g = mass of propellant gas

M_n = momentum of recoiling or counterrecoiling parts

M_o = mass of operating rod; bending moment at first bend of flat tape spring

M_p = mass of projectile

M_r = mass of recoiling parts

M_{re} = effective mass of recoiling parts

M_s = mass of slide; mass of spring

M_t = mass of barrel

N = number of coils; normal reaction on cam curve; normal force on roller; number of rounds; number of active segments in flat spring

N_a = axial component of normal force; number of links of ammunition

N_c = number of chambers in drum

N_r = number of retainer partitions

N_t = transverse component of normal force

N_μ = cam tangential friction force

P = general term for power

P_f = power required to drive feed system

P_t = trigger pull

p = pressure, general; pressure in reservoir; general term for space between rounds (pitch)

p_a = average pressure; average bore pressure

p_b = bore pressure

p_c = average gas cylinder pressure

p_{cr} =.critical pressure

p_d = pitch of double helix drive

LIST OF SYMBOLS (Con't.)

p_i	= interface pressure
p_m	=propellant gas pressure as bullet leaves muzzle
p_u	= component of pressure that dilates cartridge case
p ,	= initial pressure
p_2	= final pressure
R	= radius of bolt outer surface
R_a	= reaction of rear support; radius to CG of round
$R,$	= reaction of front support
R_c	=cam radius
R_{ch}	= radius of chamber centers of drum
R_d	= distance from cam contact point to drum axis
R_L	= radius of locking lug pressure center
R_r	= roller radius; track reactions due to rotational forces
RT	= specific impetus
R_t	= track reactions due to tipping forces; trigger reaction on sear
R_y	= horizontal reaction on drum shaft
r	= mean radius of case
$\bar{r}$	= distance from tipping point on rim to CG .of case
r_b	= cam radius to contact point on bolt
r_e	=extractor radius
r_s	= striker radius
r_t	= cam radius to contact point on barrel

s	= travel distance
s_a	= accelerating distance
s_b	=bore travel; distance of bolt retraction during recoil
	= cutoff distance
s_{cr}	= cam follower travel during counterrecoil
s_d	= dwell distance
s_n	= travel distance of operating unit at given time
	= initial distance
s_{ocr}	=straight length of cam during counterrecoil
'or	= straight length of cam during recoil; position where slide contacts gas operating unit
s_r	=cam follower travel during recoil; operating rod travel before bolt pickup
s_1	= travel component due to change in velocity
s_2	= travel component due to velocity
T	= surge time of spring; absolute temperature; torque about gun axis; applied torque of trigger spring
T_c	= compression time of spring
T_d	= required drum torque
T_g	=torque due to friction on drum bearing and case
T_L	=locking lug torque
T_r	= required retainer torque
$T,, T_2$	= applied torques

LIST OF SYMBOLS (Con't.)

T_α = accelerating torque

T_μ = resisting torque

T_θ = applied torque

t = time

t'' = time to complete counterrecoil of slide

t_b = buffer time

t_{bc} = buffer time during counterrecoil

= time of firing cycle

t_{cr} = counterrecoil time

t_{crb} = time of buffer counterrecoil

t_{crt} = counterrecod time of barrel after buffer action

t_{ct} = counterrecoil time of barrel

t_d = decelerating time of bolt before buffer is contacted

t_e = time of gas expansion

t_g = duration of propellant gas period

t_i = time interval of dwell between counter-recoil and recoil

t_r = recoil time of bolt

t_{rb} = bolt decelerating time during recoil

t_{rs} = counterrecod time after buffer action

t_{rt} = recoil time of barrel during pressure decay after bolt unlocking

t_s = thickness of spring

t_{scr} = counterrecoil time of slide

t_t = barrel spring compression time

t_α = accelerating time of rotor

V_b = bore volume

V_c = gas volume in operating cylinder

V_{ch} = chamber volume

V_{co} = operating cylinder displacement

V_e = equivalent gas volume

V_{ec} = equivalent bore volume

V_m = chamber volume plus total bore volume

V_o = initial volume of gas operating cylinder

V_s = vertical component of spring load

V_1 = initial volume in gas equations

V_2 = final volume in gas equations

v = velocity, general

v_a = average velocity; axial velocity

v_b = buffer velocity during recoil

v_{bc} = bolt velocity during counterrecoil

v_c = linear velocity of cam follower along cam; velocity of chutes; linear ejected velocity of cartridge case

v_{cr} = counterrecoil velocity

v_{crb} = counterrecoil velocity of buffer

v_d = peripheral velocity of drum

v_{dm} = maximum peripheral velocity of drum

v_e = extractor velocity; maximum ejection velocity

v_f = velocity of free recoil

v_i = impact velocity

v_m = muzzle velocity of projectile

LIST OF SYMBOLS (Con't.)

v_o = initial velocity, general term

v_{ot} = initial velocity of barrel

v_r = recoil velocity

v_s = slide velocity

v_{sa} = velocity of recoiling parts at end of accelerating travel

v_{scr} = counterrecoil velocity of slide

v'_{scr} = slide velocity before impact

v_{sm} = maximum slide velocity

v_t = barrel velocity; tangential velocity of cartridge case at ejection, velocity of transfer unit

W = general term for weight; wall ratio; work

W_a = weight of round

W_b = bolt weight

W_c = wall ratio of case; weight of gas in cylinder; weight of propellant charge

W_{cc} = weight of cartridge case; weight of empty case

W_{ce} = weight of gas at critical pressure

W_d = weight of drum

W_e = equivalent weight of moving parts

W_g = total weight of propellant or propellant gas

W_o = weight of moving operating cylinder components

W_{os} = combined weight of components and slide

W_p = weight of projectile

W_r = weight of recoiling parts

W_s = work needed to compress spring; slide weight; weight of spring

W_{se} = equivalent weight of spring in motion

W_{sr} = weight of slide with 2 rounds

W_{srp} = weight of slide, 2 rounds, and gas operating unit

W_t = barrel weight

w = rate of gas flow; width of magazine; width of flat spring

w_c = width of cam

x = recoil travel, general; case travel; distance in x-direction

x = axial acceleration of bolt

x_b = bolt travel

x_{bo} = bolt travel at end of propellant gas period

x_m = axial length of parabola

x_r = recoil distance during propellant gas period; recoil travel of drum and barrel assembly

x_{rn} = recoil travel during impulse period

x_{ro} = counterrecoiling travel during impulse period

x_s = slide travel; relative axial travel between cam follower and drum

x_{rd} = travel of recoiling parts during cam dwell period

x_t = barrel travel with respect to gun frame

x_{tf} = barrel travel during free recoil

x_{to} = barrel travel during propellant gas period; after buffer engagement; recoil travel during cam dwell period

LIST OF SYMBOLS (Con't.)

y = distance in y-direction, spring deflection

y_a = peripheral length of constant slope of cam

y_e = shear deflection

y_m = peripheral width of parabola; moment deflection

GREEK LETTERS

α = angular acceleration, general; angular acceleration of bolt; angular acceleration of rotor

α_d = angular acceleration of drum

β = cam angle

β_L = cam locking angle

β_o = slope of cam helix

γ = correction factor

Δt = time differential

Δv = velocity differential

Δx = distance differential

Δy = differential deflection; relative deflection of one spring segment

ϵ = efficiency of spring system

ϵ_b = efficiency of buffer system

ϵ_t = efficiency of recoil adapter

θ = angular displacement, general; angular displacement of rotor; angle of elevation; angular deflection in rectangular coil spring

θ_a = rotor travel for constant cam slope

θ_e = angular shear deflection

θ_m = angular moment deflection

λ = angle of bolt locking cam; slope of lug helix

μ = coefficient of friction

μ_i = index of friction

μ_r = coefficient of rolling friction

μ_s = coefficient of sliding friction

μ_t = coefficient of friction of track

ν = Poisson's ratio

ρ = ratio of spring energy to drum energy

Σ = summation

σ_t = tensile stress

σ_w = working stress of spring

τ = static stress **of** spring

τ_d = dynamic stress of spring

ϕ = angle of double helix drive

ω = angular velocity

ω_d = angular velocity of drum

PREFACE

This handbook is one of a series on Guns. It is part of a group of handbooks covering the engineering principles and fundamental data needed in the development of Army materiel, which (as a group) constitutes the Engineering Design Handbook Series. This handbook presents information on the fundamental operating principles and design of automatic weapons and applies specifically to automatic weapons of all types such as blowback, recoil-operated, gas-operated, and externally powered. These include single, double, multibarrel, and revolver-type machine guns and range from the simple blowback to the intricate M61A1 Vulcan and Navy 20 mm Aircraft Gun Mark II Mod 5 Machine Guns. Methods are advanced for preparing engineering design data on firing cycle, spring design, gas dynamics, magazines, loaders, firing pins, etc. All components are considered except tube design which appears in another handbook, AMCP 706-252, ***Gun Tubes.***

This handbook was prepared by The Franklin Institute, Philadelphia, Pennsylvania, for the Engineering Handbook Office of Duke University, prime contractor to the U.S. Army, and was under the technical guidance and coordination of a special subcommittee with representation from Watervliet Arsenal, Rock Island Arsenal, and Springfield Armory.

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Comments and suggestions on this handbook are welcome and should be addressed to Army Research Office—Durham, Box CM, Duke Station, Durham, N. C. 27706.

CHAPTER 1

INTRODUCTION*

1–1 SCOPE AND PURPOSE

This handbook presents and discusses procedures normally practiced for the design of automatic weapons, and explores the problems stemming from the functions of each weapon and its components. It is intended to assist and guide the designer of automatic weapons of the gun type, and to contain pertinent design information and references.

1–2 GENERAL

The purpose of the handbook is (1) to acquaint new personnel with the many phases of automatic weapon design, and (2) to serve as a useful reference for the experienced engineer. It does not duplicate material available in other handbooks of the weapon series. Those topics which are presented in detail in other handbooks are discussed here only in a general sense; consequently, the reader must depend on the referenced handbook for the details. Unless repetitive, the text – for cyclic analyses, time-displacement (T-D) curves, chamber design, strength requirements, springs, cams, and drive systems – includes mathematical analyses embodying sketches, curves, and illustrative problems. Topics such as ammunition characteristics, lubrication, handling and operating features, and advantages and disadvantages are generally described more qualitatively than quantitatively.

Appendix B is included to merely introduce the idea of the automatic control of a burst of rounds for weapon effectiveness in the point fire mode – a facet which the gun designer may wish to consider.

1–3 DEFINITIONS

An automatic weapon is a self-firing gun. To be fully automatic, the weapon must load, fire, extract, and eject continuously after the first round is loaded and fired – provided that the firing mechanism is held unlocked. Furthermore, the automatic weapon derives all its operating energy from the propellant. Some weapons have external power units attached and, although not automatic in the strictest sense, are still classified as such.

There are three general classes of automatic weapons, all defined according to their system of operation, namely: blowback, gas-operated, and recoil-operated[1]**

a. Blowback is the system of operating the gun mechanism that uses propellant gas pressure to force the bolt to the rear; barrel and receiver remaining relatively fiied. The pressure force is transmitted directly by the cartridge case base to the bolt.

b. Gas-operated is the system that uses the propellant gases that have been vented from the bore to drive a piston linked to the bolt. The moving piston first unlocks the bolt, then drives it rearward.

c. Recoil-operated is the system that uses the energy of the recoiling parts to operate the gun.

Each system has variations that may borrow one or more operational features from the others. These variations, as well as the basic systems, are discussed thoroughly in later chapters.

1–4 DESIGN PRINCIPLES FOR AUTOMATIC WEAPONS

The automatic weapon, in the process of firing a round of ammunition, is essentially the same as any other gun. Its basic difference is having the ability to continue firing many rounds rapidly and automatically. An outer stimulus is needed only to start or stop firing, unless the latter occurs when ammunition supply is exhausted. The automatic features require major effort in design and development. The design philosophy has been established, then the gun is to fire as fast as required without stressing any component to the extent where damage and therefore malfunction is imminent.

An extremely short firing cycle being basic, the designer must exploit to the fullest the inherent properties of each type of automatic weapon. Generally, each type must meet certain requirements in addition to

*Prepared by Martin Regina, Franklin Institute Research Laboratories, Philadelphia, Pennsylvania.

**References are identified by a superscript number and are listed at the end of this handbook.

being capable of operating automatically. These requirements or design features are:

1. Use part of the available energy of the propellant gases without materially affecting the ballistics.
2. Fire accurately at a sustained rate compatible with the required tactics.
3. Use standard ammunition.
4. Be light for easy handling.
5. Have a mechanism that is:
   a. simple to operate
   b. safe
   c. easy to maintain
   d. economical with respect to manufacturing.
6. Have positive action for feeding, extracting, ejecting.
7. Insure effective breech closure until the propellant gas pressure has dropped to safe limits.

All successful automatic weapons meet these requirements but to a degree normally limited by type of weapon. Conflicting requirements are resolved by compromise.

CHAPTER 2

BLOWBACK WEAPONS

2–1 GENERAL

Controlling the response of the cartridge case to the propellant gas pressure is the basic design criterion of blowback weapons. The case responds by tending to move rearward under the influence of the axial force generated by the gas pressure on its base. Meanwhile, because of this same pressure, the case dilates to press on the inner wall of the chamber. The axial force tends to push the bolt rearward, opposed only by the resistance offered by the bolt inertia and the frictional resistance between case and chamber wall. The question now arises as to which response predominates, the impending axial motion or the frictional resistance inhibiting this motion.

Time studies resolve the problem. Fig. 2–1 is a typical pressure-time curve of a round of ammunition.

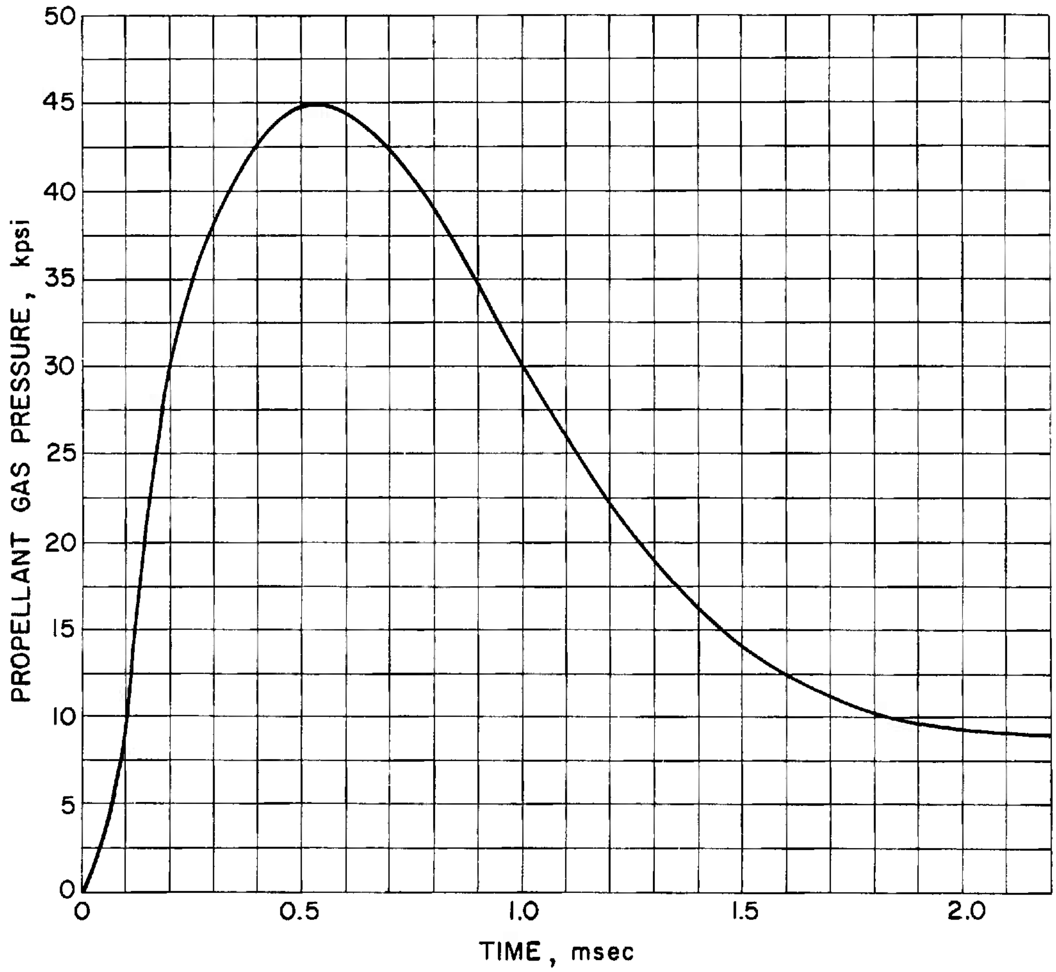


Figure 2–1. Typical Pressure-time Curve

For simplicity, assume unity for bore area and bolt weight. According to Fig. 2–1, the maximum pressure of 45,000 psi develops in 0.0005 sec. Again for simplicity, assume that the pressure varies linearly from t = 0 to t = 0.0005 sec. The pressure p at any time during the interval

$$p = \left(\frac{45000}{0.0005}\right) t = 9 \times 10^7 t \ \text{lb/in.}^2 \qquad (2\text{–}1)$$

The corresponding force F driving the cartridge case and bolt rearward is

$$F = A_b\, p = 9 \times 10^7\, tA \ \text{lb} \qquad (2\text{–}2)$$

where A_b = bore area in square inches

but, by assumption, A_b = 1.0 in? , therefore

$$F = 9 \times 10^7 t = \text{Kt}. \qquad (2\text{–}3)$$

From mechanics

$$F = M_b a \qquad (2\text{–}4)$$

where a = bolt acceleration

M_b = mass of bolt.

According to an earlier assumption $M_b = \frac{W}{g} = \frac{1.0}{g}$

Solve for a in Eq. 2–4

$$a = \frac{F}{M_b} = \text{Kgt} \qquad (2\text{–}5)$$

but

$$a = \frac{d^2 s}{dt^2} = \text{Kgt}. \qquad (2\text{–}6)$$

Integration of Eq. 2–6 yields

$$v = \int \frac{d^2 s}{dt^2} = \frac{ds}{dt} + C = \frac{1}{2} Kgt^2 + C_1 \qquad (2\text{–}7)$$

when t = 0, v = 0, therefore C_1 = 0.

Integration of Eq. 2–7 yields

$$s = \int \frac{ds}{dt} = \frac{1}{6} Kgt^3 + C_2 \qquad (2\text{–}8)$$

when t = 0, s = 0, therefore, C_2 = 0.

Assume that the limiting clearance between case and chamber is equal to the case dilation as it reaches the ultimate strength, and assume further that the cartridge case has a nominal outside diameter of 1.5 in., a wall thickness of 0.05 in., and an ultimate strength of 50,000 psi. Then, according to the thin-walled pressure vessel formula, the pressure at which failure impends and which presses the case firmly against the chamber wall is p_u.

$$p_u = \frac{\sigma_t\, t_c}{r} = \frac{50{,}000 \times 0.05}{0.725} = 3440 \text{ psi} \qquad (2\text{–}9)$$

where

r = 0.725 in., mean radius of case

t_c = 0.05 in., wall thickness

σ_t = 50,000 lb/in.^{2}, tensile stress

From Eq. 2–1, t is the time elapsed to reach this pressure.

$$t = \frac{p_u}{9 \times 10^7} = 3.83 \times 10^{-5} \text{ sec} \qquad (2\text{–}10)$$

From Eq. 2–8, s is the distance that the case and bolt travel during this time, i.e., when only the inertia of the system is considered.

$$s = \frac{1}{6} Kgt^3$$

$$= \frac{1}{6} \times 9 \times 10^7 \times 386 \times 56 \times 10^{-15} \ll 0.001 \text{ in.}$$

This analysis indicates that when optimum conditions prevail, the cartridge case scarcely moves before frictional resistance begins to take effect. Motion will continue until Eq. 2–11 is satisfied.

$$A_b p = \mu A_c p_i \qquad (2\text{–}11)$$

where

A, = bore area

A, = peripheral surface contact area between case and chamber

p = propellant gas pressure

p_i = interface pressure of case and chamber

μ = coefficient of friction

With no initial clearance between the case and chamber, an approximate interface pressure p_i may be determined by equating the inside deflection of the chamber, due to this pressure, to the outside deflection of the cartridge case, due to both interface and propellant gas pressure, when both case and chamber are considered cylindrical. Solve for the interface pressure.

$$p_i = \frac{\dfrac{2p}{W_c^2 - 1}}{\dfrac{E_c}{E}\left(\dfrac{W^2+1}{W^2-1} + \nu\right) + \left(\dfrac{W_c^2+1}{W_c^2-1} - \nu\right)} \qquad (2-12)$$

where

E = modulus of elasticity of chamber

E, = modulus of elasticity of case

W = wall ratio of chamber

W_c = wall ratio of case

ν = Poisson's ratio (assumed to be equal for both materials)

Spot checks indicate that those pressures which dilate unsupported cartridge cases to the limit of their strength are reasonably close to the difference in propellant gas pressure and computed interface pressure. Thus

$$p_u \approx p - p_i \qquad (2-13)$$

Ample clearance between case and chamber is always provided but is never so large that barrel recovery exceeds case recovery after gas pressures subside; otherwise, interference develops, i.e., clamping the case to the chamber wall and rendering extraction difficult*.

2-2 SIMPLE BLOWBACK

Simple blowback is the system wherein all the operating energy is derived from blowback with the inertia of the bolt alone restraining the rearward movement of the cartridge case.

2-2.1 SPECIFIC REQUIREMENTS

Being restricted to low rates of fire because massive bolts are needed for their inertial properties, simple blowback systems are suitable only for low impulse, relatively low rate of fire weapons[3].

The restraining components of a simple blowback mechanism are the bolt and driving spring. Fig. 2-2 is a schematic of an assembled unit. Immediate resistance to case movement offered by the return spring is usually negligible. This burden falls almost totally on the bolt. It begins to move as soon as the projectile starts but at a much lower acceleration so that the cartridge case is still supported by the chamber until propellant gas pressure becomes too low to rupture the case. To realize a low acceleration, the bolt must be considerably heavier than needed as a load-supporting component. In high impulse guns, bolt sizes can be ridiculously large. The large mass, being subjected to the same impulse as that applied to propellant gas and projectile, will develop the same momentum; consequently, its velocity and corresponding kinetic energy will be comparatively low. The slowly moving bolt confines the gun to a low rate of fire.

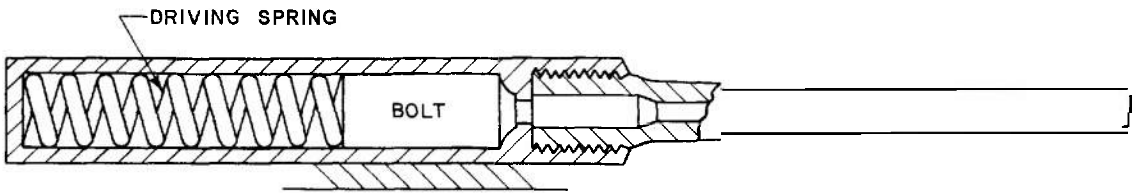


Figure 2-2. Schematic of Simple Blowback Mechanism

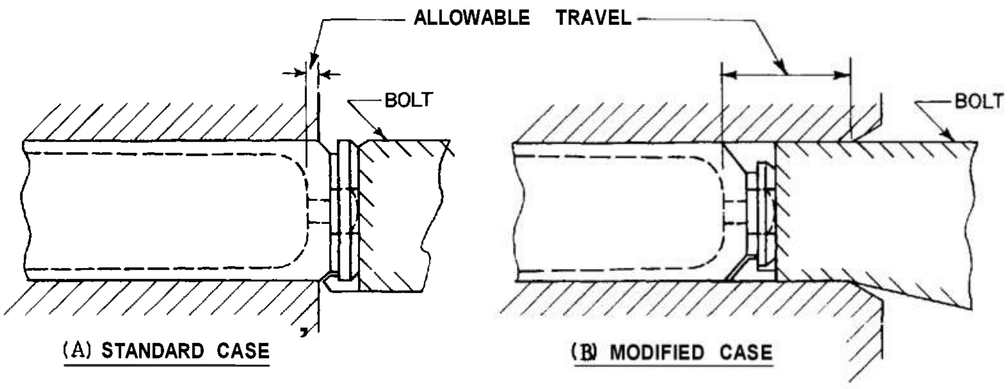


Figure 2–3. Allowable Case Travel

Although the bolt moves slowly, it still permits the case to move. The permissible travel while gas pressures are still high enough to rupture an unsupported case is indicated by Fig. 2–3(A) for a standard cartridge case. Fig. 2–3(B) illustrates how a modified case can increase the permissible travel. The geometry of chamber and cartridge case are also involved. A slight taper or no taper at all presents no problem but, for a large taper, an axial displacement creates an appreciable gap between case and chamber, thereby, exposing the case to deflections verging on rupture. Therefore, for weapons adaptable to simple blowback operation, chamber and case design takes on special significance if bolt travel is reasonable while propellant gases are active. For high-powered guns, exploiting this same advantage gains little. How little effect an increase in travel has on reducing bolts to acceptable sizes is demonstrated later.

The driving spring has one basic function. It stores some of the energy of the recoiling bolt, later using this energy to slam the bolt back into firing position and in the process, cocks the firing mechanism, reloads, and trips the trigger to repeat the firing cycle. That the driving spring stores only some of the energy of the recoiling bolt when firing semiautomatic shotguns, rifles, and pistols is indicated by the forward momentum not being perceptible during reloading whereas the kick during firing is pronounced.

2–2.2 TIME OF CYCLE

The time of the firing cycle is determined by the impulse created by the propellant gases, and by the bolt and driving spring characteristics. The impulse $\int Fdt$ is computed from the area beneath the force-time curve. It is equated to the momentum of the bolt assembly, i.e.,

$$\int_{t_1}^{t_2} F_g dt = M_b v_f \tag{2–14}$$

where

F_g = propellant gas force

M_b = mass of bolt assembly

v_f = velocity of free recoil

dt = time differential

The mass of the bolt assembly includes about one-third the spring as the equivalent mass of the spring in motion. However, the effect of the equivalent spring mass is usually very small and, for all practical purposes, may be neglected. After the energy of free recoil is known, the recoil energy E_r and the average driving spring force become available

$$E_r = \frac{1}{2} M_b v_f^2 \tag{2–15}$$

The average force F_a depends on the efficiency of the mechanical system

$$F_a = \frac{\epsilon E_r}{L} \tag{2–16}$$

where L = length of recoil or bolt travel

e = efficiency of system

2–2.2.1 Recoil Time

The bolt travel must be sufficient to permit ready cartridge loading and case extraction. The initial spring force F_o is based on experience and, when feasible, is selected as four times the weight of the recoiling mass. The maximum spring force F_m, when the bolt is fully recoiled, is

$$F_m = 2F_a - F_o \tag{2–17}$$

The spring force at any time of recoil is

$$F = F_o + Kx \tag{2–18}$$

where K = spring constant

x = recoil distance at time t

At time t the energy remaining in the recoiling mass is

$$\frac{1}{2} M_b v_r^2 = \frac{1}{2} M_b v_f^2 - \frac{1}{\epsilon}\left(F_o x + \frac{1}{2} Kx^2\right) \tag{2–19}$$

where ϵ is the efficiency of the spring system. An inefficient system helps to resist recoil by absorbing energy.

But $v_r = \dfrac{dx}{dt_r}$, therefore

$$\frac{dx}{dt_r} = \sqrt{\frac{2}{M_b}} \sqrt{\frac{M_b}{2} v_f^2 - \frac{1}{\epsilon} F_o x - \frac{1}{2\epsilon} Kx^2} \tag{2–20}$$

Solve for dt_r.

$$dt_r = \frac{\sqrt{\dfrac{M_b}{2}}\, dx}{\sqrt{\dfrac{M_b}{2} v_f^2 - \dfrac{1}{\epsilon} F_o x - \dfrac{1}{2\epsilon} Kx^2}} \tag{2–21}$$

Set $v_r = v_f$, the initial velocity at time zero, and integrate.

$$t_r = \left(\sqrt{\frac{\epsilon M_b}{K}}\ \mathrm{Sin}^{-1} \frac{F_o + Kx}{\sqrt{F_o^2 + \epsilon K M_b v_o^2}}\right)\Bigg]_{x=0}^{x=L} \tag{2–22}$$

This computed time does not include the time while propellant gases are acting. The exclusion provides a simple solution without serious error. Since $Mv_o^2 = \frac{L}{\epsilon}(F_m + F_o)$ and, by definition,

$$K = \frac{F_m - F_o}{L} \text{ and } \sqrt{F_o^2 + \epsilon K M v_o^2} = F_m.$$

Therefore, the time elapsed during recoil t, from $x = 0$ to $x = L$ is

$$t_r = \sqrt{\frac{\epsilon M_b}{K}}\left(\frac{\pi}{2} - \mathrm{Sin}^{-1}\frac{F_o}{F_m}\right) = \sqrt{\frac{\epsilon M_b}{K}}\ \mathrm{Cos}^{-1}\frac{F_o}{F_m}. \tag{2–23}$$

2–2.2.2 Counterrecoil Time

The counterrecoil time is determined by the same procedure as that for recoil, except that the low efficiency of springs deters rapid counterrecoil. The energy of the counterrecoiling mass of the bolt assembly at any time t_{cr} is

$$E_{cr} = \frac{1}{2}M_b v_{cr}^2 = \frac{1}{2}M_b v_o^2 + \epsilon\left(F_m x - \frac{1}{2}Kx^2\right) \tag{2–24}$$

where v_o = initial velocity

v_{cr} = counterrecoil velocity at any time

Since $v_{,,} = \frac{dx}{dt_{cr}}$

$$dt, = \frac{\sqrt{\frac{M_b}{2}}\, dx}{\sqrt{\frac{M_b}{2}v_o^2 + \epsilon F_m x - \frac{\epsilon}{2}Kx^2}} \tag{2–25}$$

Integrating

$$t_{cr} = \sqrt{\frac{M_b}{\epsilon K}}\left(\mathrm{Sin}^{-1}\frac{Kx - F_m}{\sqrt{F_m^2 + \frac{K}{\epsilon}M_b v_o^2}} - \mathrm{Sin}^{-1}\frac{-F_m}{\sqrt{F_m^2 + \frac{K}{\epsilon}M_b v_o^2}}\right) \tag{2–26}$$

When the initial velocity is zero, the time t_{cr} to counterrecoil the total distance is

$$t_{cr} = \sqrt{\frac{M_b}{\epsilon K}}\left(\mathrm{Sin}^{-1}\frac{-F_o}{F_m} - \frac{3\pi}{2}\right) = \sqrt{\frac{M_b}{\epsilon K}}\ \mathrm{Cos}^{-1}\frac{F_o}{F_m} \tag{2–27}$$

2–2.2.3 Total Cycle Time

The mass of the bolt assembly and the bolt travel are the controlling elements of a simple blowback system. Large values will decrease firing rate whereas the converse is true for small values. The driving spring characteristics are determined after mass and travel are established. The total weapon weight limits, to a great extent, the weight and travel of the bolt.

Because of the efficiency of the spring system, counterrecoil of the bolt will always take longer than recoil. The time t_c for the firing cycle is

$$t_c = t_r + t_{cr} + t_i \qquad (2-28)$$

where t_i is time elapsed at the end of counterrecoil until the bolt mechanism begins to move in recoil. Since the firing rate is specified, t_c is

$$t_c = \frac{60}{f_r} \text{ ,sec/round} \qquad (2-29)$$

where f_r = firing rate in rounds/min.

Initial approximations of blowback parameters may be computed by relating average spring forces and acceleration to the recoil energy. The average spring force F_a needed to stop the recoiling mass is

$$F_a = \frac{\epsilon E_r}{L} = \frac{\epsilon M_b v_f^2}{2L} \qquad (2-30)$$

where, according to **Eq.** *2–15*, $E_r = \frac{1}{2} M_b v_f^2$. Since $F_a = \epsilon M_b a_r$

$$a_r = \frac{F_a}{\epsilon M_b} = \frac{v_f^2}{2L} \qquad (2-31)$$

From the general expression for computing distance in terms of time and acceleration, $L = \frac{1}{2} a_r t_r^2$, the recoil time becomes

$$t_r = \sqrt{\frac{2L}{a_r}} = \sqrt{\frac{4L^2}{v_f^2}} = \frac{2L}{v_f}. \qquad (2-32)$$

During counterrecoil, the effectiveness of the spring force is reduced by the inefficiency of the system. This force is

$$F_{cr} = \epsilon F_a = M_b a_{cr} \qquad (2-33)$$

where a_{cr} is the counterrecoil acceleration. According to Eq. *2–30*

$$\epsilon F_a = \frac{\epsilon^2 M_b v_f^2}{2L} = M_b a_{cr}. \qquad (2-34)$$

Proceed similarly as for recoil

$$t_{cr} = \sqrt{\frac{2L}{a_{cr}}} = \sqrt{\frac{4L^2}{\epsilon^2 v_f^2}} = \frac{2L}{\epsilon v_f}. \qquad (2-35)$$

The approximate time of the firing cycle becomes

$$t_c = t_r + t_{cr} = \frac{2L}{v_f}\left(1 + \frac{1}{\epsilon}\right). \qquad (2-36)$$

By knowing the required cycle time and the computed velocity of free recoil, the distance of bolt travel can be determined from **Eq.** *2–36*. This computed distance will be less than the actual because the accelerations are not constant thereby having the effect of needing less time to negotiate the distance in **Eq.** *2–36*. In order to compensate for the shorter time, the bolt travel is increased until the sum of t_r and t_{cr} from Eqs. *2–23* and *2–27* equals the cycle time.

$$t_c = t_r + t_{cr}$$

$$= \left(\sqrt{\frac{\epsilon M_b}{K}} + \sqrt{\frac{M_b}{\epsilon K}}\right) \cos^{-1} \frac{F_o}{F_m}. \qquad (2-37)$$

Substitute $2F_a - F_m$ for F_o and rewrite **Eq.** *2–37*

$$\cos \frac{t_c}{\sqrt{\epsilon M_b} + \sqrt{\frac{M_b}{\epsilon}}} \sqrt{K} = \frac{2F_a - F_m}{F_m}$$

F_a is computed from **Eq.** *2–30*. Note that

$$t_c / \left(\sqrt{\epsilon M_b} + \sqrt{\frac{M_b}{\epsilon}}\right)$$

is a constant for any given problem. Now by the judicious selection of L (using *Eq. 2–36* for guidance) and K, the spring forces may be computed by iterative procedures so that (1) when substituted into Eq. *2–37* the specified time is matched, and (2) then into Eq. *2–17* to check whether F_a corresponds with the computed value obtained earlier from **Eq.** *2–30*.

The actual firing rate is determined from the final computed cycle time.

$$f_r = \frac{60}{t_c} \text{ rounds/min} \quad (2-39)$$

2-2.3 EXAMPLE OF SIMPLE BLOWBACK GUN

2-2.3.1 Specifications

Gun: 11.42 mm (Cal .45) machine gun

Firing Rate: 400 rounds per minute

Interior Ballistics: Pressure vs Time (Fig. 2-4)
Velocity vs Time (Fig. 2-4)

Weight of moving bolt assembly: 3 lb

2-2.3.2 Computed Design Data

The area beneath the pressure-time curve of Fig. 2-4 represents an impulse of

$$\int F_g dt = 0.935 \text{ lb-sec.}$$

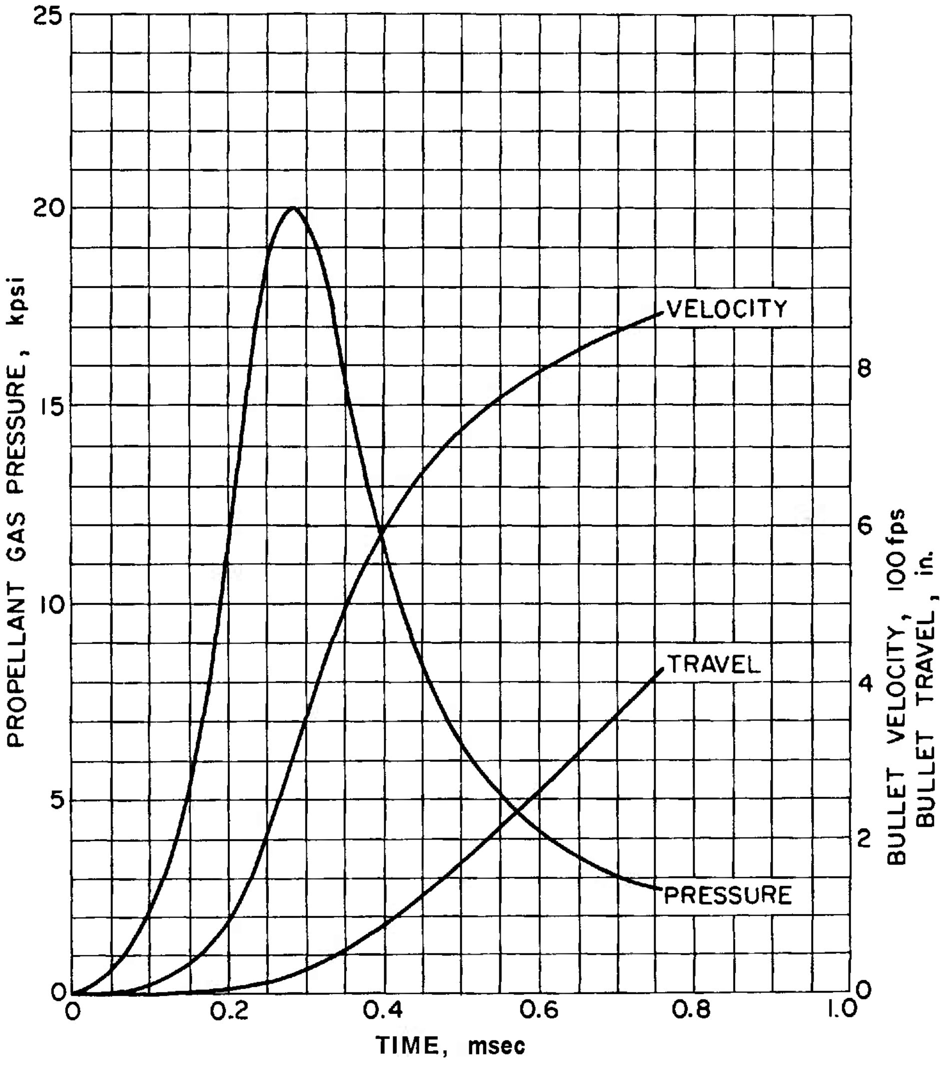


Figure 2-4. Pressure-time Curve of Cal.45 (11.42mm) Round

The velocity of free recoil according to **Eq.** 2–14 is

$$v_f = \frac{\int F_g dt}{M_b} = \frac{0.935 \times 386.4}{3} = 120.4 \text{ in./sec.}$$

The recoil energy from **Eq.** 2–15 is

$$E_r = \frac{1}{2} M_b v_f^2 = \frac{1}{2} \times \frac{3}{386.4} \times 14500 = 56.3 \text{ in.-lb.}$$

The time of the firing cycle for 400 rpm is

$$t_c = \frac{60}{400} = 0.15 \text{ sec.}$$

From **Eq.** 2–36, the approximate bolt travel is

$$L = \frac{1}{2} t_c v_f \left(\frac{\epsilon}{\epsilon + 1}\right) = \frac{1}{2} \times 0.15 \times 120.4 \left(\frac{0.40}{0.40 + 1.0}\right) = 2.58 \text{ in.}$$

where $\epsilon = 0.40$, the efficiency of system.

$K = 1.0$ lb/in. is selected as practical for the first trials. This value may be revised if the bolt travel becomes excessive or other specifications cannot be met. From **Eq.** 2–30 the average spring force

$$F_a = \frac{\epsilon E_r}{L} = \frac{0.40 \times 56.3}{2.58} = 8.72 \text{ lb.}$$

From **Eqs.** 2–17 and 2–18 the minimum and maximum spring forces are

$$F_o = F_a - \frac{1}{2} KL = 8.72 - 1.29 = 7.43 \text{ lb}$$

$$F_m = F_o + KL = 7.43 + 2.58 = 10.01 \text{ lb.}$$

Compute the characteristics of **Eq.** 2–37

$$\sqrt{\epsilon M_b} = \sqrt{\frac{0.4 \times 3}{386.4}} = \sqrt{0.003106} = 0.0557 \text{ lb-sec}^2\text{/in.}$$

$$\sqrt{\frac{M_b}{\epsilon}} = \sqrt{\frac{3}{0.4 \times 386.4}} = \sqrt{0.01941} = 0.1393 \text{ lb-sec}^2\text{/in.}$$

The time of the firing cycle for $K = 1$ lb/in. is

$$t_c = \left(\sqrt{\frac{\epsilon M_b}{K}} + \sqrt{\frac{M_b}{\epsilon K}}\right) \cos^{-1} \frac{F_o}{F_m} = (0.0557 + 0.1393) \frac{1}{\sqrt{1}} \cos^{-1} \frac{7.43}{10.01}$$

$$= 0.195 \cos^{-1} 0.74226 = 0.195 \left(\frac{42.08}{57.3}\right) = 0.143 \text{ sec.}$$

therefore, to have $t_c = 0.15$ sec, $\text{Cos}^{-1} \dfrac{\quad}{F_m} = 0.76923$ rad and cos 0.76923 rad = 0.7185 so that

$$\frac{F_o}{F_m} = \frac{F_o}{F_o + KL} = 0.7185.$$

Since $K = 1$, $\dfrac{F_o}{F_o + L} = 0.7185$ and $F_o = 2.552L$.

Also, since $\frac{1}{\epsilon} F_a L = \dfrac{(F_o + F_m)L}{2E} = E_r = 56.3$ in.-lb

$$2F_o + L = 2F_a = \frac{2(0.40 \times 56.3)}{L} = \frac{45.04}{L}.$$

Therefore

$L^2 = 7.379$ in^2 $\qquad$ $L = 2.72$ in.

$F_o = 6.94$ lb $\qquad$ $F_m = 9.62$ lb

The spring work $W_s = \frac{1}{2}(F_o + F_m)L = 22.52$ in.-lb, which matches the input. The time t_c of the firing cycle for $K = 1$

$$t_c = \left(\sqrt{\frac{\epsilon M_b}{K}} + \sqrt{\frac{M_b}{\epsilon K}}\right) \text{Cos}^{-1} \frac{F_o}{F_m}$$

$$= (0.557 + 0.1393)\ \text{Cos}^{-1} \frac{6.94}{9.62}$$

$$= 0.195\ \text{Cos}^{-1}\ 0.7214 = 0.195 \left(\frac{43.83}{57.3}\right)$$

$$= 0.15 \text{ sec.}$$

2-2.3.3 Case Travel During Propellant Gas Period

Case travel while propellant gas pressures are active is found by numerically integrating the interior ballistics pressure-time curve and the velocity-time curve of the

t	Δt	A_i	$F_g \Delta t$	Δv	v	v_a	Δx

t = time, abscissa of pressure-time curve

$\Delta t = t_n - t_{n-1}$, differential time

A_i = differential area under pressure-time curve

$F_g \Delta t = A_b A_i$, differential impulse

A_b = bore area

$\Delta v = F_g \Delta t / M_b$, differential velocity

M_b = mass of bolt

$v = \Sigma \Delta v$ velocity at end of each time increment

$v_a = \frac{1}{2}(v_n + v_{n-1})$ average velocity for each time increment

$\Delta x = v_a \Delta t$, differential distance of case travel

$x = \Sigma \Delta x$, case travel during propellant gas period

2-2.3.4 Sample Problem of Case Travel

The distance that the case is extracted as the projectile leaves the bore is determined by numerically integrating the pressure-time curve of Fig. 2-4.

$$A_b = \frac{\pi}{4} D_b = \frac{\pi}{4} \times 0.45^2 = 0.159 \text{ in}^2, \text{bore area}$$

$$\Delta v = F_g \Delta / M_b = \frac{32.2 \times 12 F_g \Delta t}{3} = \frac{386.4 F_g \Delta t}{3}$$

$$= 128.8\ F_g \Delta t$$

Δx = 0.053 in., the case travel distance when the projectile leaves the muzzle. This unsupported distance of the case is still within the allowable travel illustrated in Fig. 2-3.

TABLE 2–1. CASE TRAVEL OF CAL .45 (11.42 mm) GUN

t, msec	Δt, msec	A_i, lb-sec/in.^{2}	$F_g \Delta t$, lb-sec	Δv, in./sec	v, in./sec	v_a, in./sec	Δx, in.
0.1	0.1	0.07	0.011	1.4	1.4	0.70	0.00007
0.2	0.1	0.56	0.089	11.4	12.8	7.10	0.00071
0.3	0.1	1.73	0.275	35.4	48.2	30.50	0.00305
0.4	0.1	1.60	0.255	32.8	81.0	64.60	0.00646
0.5	0.1	0.88	0.140	18.0	99.0	90.00	0.00900
0.6	0.1	0.52	0.083	10.7	109.7	104.35	0.01043
0.7	0.1	0.36	0.057	7.3	117.0	113.35	0.01134
0.76	0.06	0.16	0.025	3.2	120.2	118.60	0.01186
Σ	0.76	5.88	0.935	120.2			0.05292

2–2.3.5 Driving Spring Design

Driving springs must be compatible with operation and with the space available for their assembly, two factors that limit their outside diameter, and assembled and solid heights. The driving springs must also be designed to meet the time and energy requirement of the firing cycle and still have the characteristics that are essential for maintaining low dynamic stresses. The criteria for dynamic stresses have been established by Springfield Armory[4]. The procedures in the subsequent analyses follow these criteria.

The spring design data developed for the firing cycle calculations are

K = 1.0 lb/in., spring constant

F_o = 6.94 lb, spring force at assembled height

F_m = 9.62 lb, static spring force at end of recoil

L = 2.72 in., bolt travel

t_c = 0.15 sec, time of firing cycle

$t_r = \left(\dfrac{0.0557}{0.195}\right) t_c$ = 0.0428 sec, time of recoil (see par. 2–2.3.2)

v_f = 120.4 in./sec, spring velocity of free recoil

According to the theory of surge waves in springs, the dynamic stress increases only slightly over the static stress if the following conditions exist:

$$1.67 < \frac{T_c}{T} < 2.0 \text{ when } 25 < v_i < 50 \text{ fps}$$

(Ref. 4)

$$3.33 < \frac{T_c}{T} < 4.0 \text{ when } 20 < v_i < 25 \text{ fps}$$

(Ref. 4)

$$5.0 < \frac{T_c}{T} < 6.0 \text{ when } v_i < 20 \text{ fps}$$

(Ref. 4)

where T = surge time

T_c = compression time of spring

v_i = impact velocity, ft/sec

The impact velocity of 50 ft/sec should not be exceeded, neither should the velocity be less than the lower limit of each range, however, the limits of the ratio $\frac{T_c}{T}$ need not necessarily be restricted to the two lower ranges. For instance, if speeds are less than 20 ft/sec, the limits of $\frac{T_c}{T}$ may be shifted to the upper range which varies between 3.33 and 4.0, or even to the first range of limits 1.67 to 2.0. For speeds between 20 and 25 ft/sec, the limits of the ratio may be shifted to the upper range that varies between 1.67 and 2.0.

The surge time, in terms of spring characteristics is[5]

$$T = 35.5 \times 10^{-6} \left(\frac{D^2}{d}\right) N \qquad (2\text{–}40)$$

where d = wire diameter

D = mean coil diameter

N = number of coils

Refer to the spring design data.

$$T_c = t_r = 0.0428 \text{ sec, compression time of spring}$$

Select $\frac{T_c}{T} = 3.8$, or

$$T = \frac{T_c}{3.8} = \frac{0.0428}{3.8} = 0.01125 \text{ sec}$$

$$K = \frac{Gd^4}{8D^3N} \quad \text{or} \quad N = \frac{Gd^4}{8D^3K} \quad \text{(Ref. 6)} \quad (2\text{–}41)$$

where G = torsional modulus (11.5 x 10^6 lb/in.^{2} for steel)

K = spring constant

Substitute the expression for N of Eq. 2–41 into Eq. 2–40, insert known values, and solve for d

$$d = 0.27 \sqrt[3]{DKT} \qquad (2\text{–}42)$$

When D = 0.5 in., and K = 1.0 (from spring data)

$$d = 0.27 \sqrt[3]{0.5 \times 1.0 \times 0.01126} = 0.048 \text{ in.}$$

From Eq. 2–41

$$N = \frac{Gd^4}{8D^3K} = \frac{11.5 \times 10^6 \times 530 \times 10^{-8}}{8 \times 0.125 \times 1.0} = 61 \text{ coils}$$

$$H_s = Nd = 61 \times 0.048 = 2.93 \text{ in., solid height.}$$

The static torsional stress τ is[6]

$$\tau = \frac{8F_mD}{\pi d^3} = \frac{8 \times 9.62 \times 0.5}{111 \times 10^{-6}\pi} = 110{,}000 \text{ lb/in.}^2 \qquad (2\text{–}43)$$

According to Eq. 34 in Ref. 4, the dynamic torsional stress is

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right] \qquad (2\text{–}44)$$

$$\tau_d = 110{,}000\left(\frac{1}{3.8}\right)4 = 116{,}000 \text{ lb/in.}^2$$

This stress is acceptable since the recommended maximum stress for music wire is 150,000 lb/in.^{2} In Eq. 2–44, $f\left(\frac{T_c}{T}\right)$ is the next largest even whole number larger than the value of $\frac{T_c}{T}$ if this ratio is not an even whole number.

2–3 ADVANCED PRIMER IGNITION BLOWBACK

Timing the ignition so that the new round is fired just before the bolt seats gives the first part of the impulse created by the propellant gas force opportunity to act as a buffer for the returning bolt. The rest of the impulse provides the effort for recoiling the bolt. The system that absorbs a portion of the impulse in this manner is called Advanced Primer Ignition Blowback. This system has its artillery counterpart in the out-of-battery firing system, i.e., the firing of the artillery weapon being initiated during counterrecoil but with the breechblock closed.

2–3.1 SPECIFIC REQUIREMENTS

By virtue of its ability to dispose of the early influence of propellant gas force on recoil, the advanced primer ignition system is much more adaptable to high rates of fire than the simple blowback system. Reducing the effectiveness of the impulse by fifty percent alone reduces the bolt weight by a factor of two with a substantial increase in firing rate.

The restraining components may be considered as real and virtual; the real being the bolt and driving spring; the virtual, the momentum of the returning bolt. Fig. 2–5 is a schematic of the advanced primer ignition system. The firing cycle starts with the bolt latched open by a sear and the driving spring compressed. Releasing the sear, frees the bolt for the spring to drive it forward. The

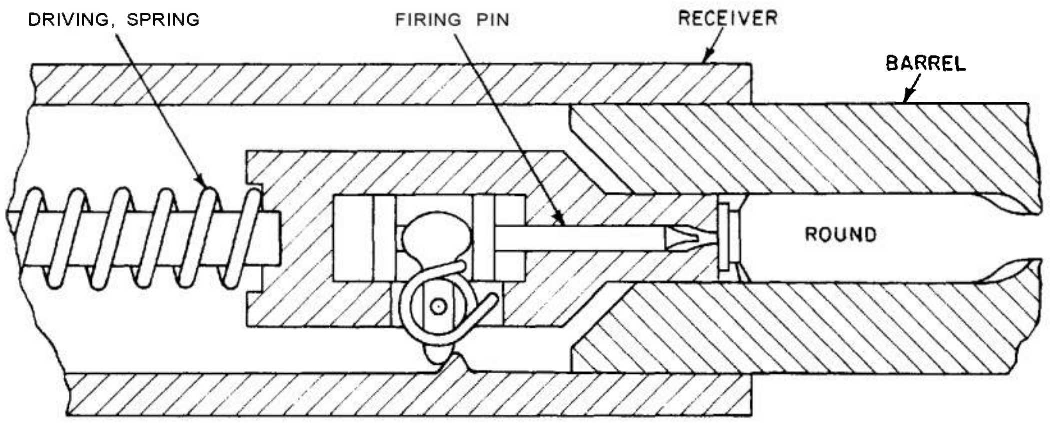


Figure 2–5. Schematic of Advanced Primer Ignition System

moving bolt picks up a round from the feed mechanisms and pushes it into the chamber. Shortly before the round is seated, the firing mechanism activitates the primer. The firing mechanism is so positioned and timed that the case is adequately supported when propellant gas pressures reach case-damaging proportions. The case and bolt become fully seated just as the impulse of the propellant gas force equals the momentum of the returning bolt. This part of the impulse is usually approximately half the total, thus establishing the driving spring characteristics.

As soon as forward motion stops, the continuously applied propellant gas force drives the bolt rearward in recoil. During recoil, the case is extracted and the driving spring compressed until all the recoil energy is absorbed to stop the recoiling parts. If the sear is held in the released position, the cycle is repeated and firing continues automatically. Firing ceases when the sear moves to the latched position.

2–3.2 SAMPLE CALCULATIONS OF ADVANCED PRIMER IGNITION

2–3.2.1 Firing Rate

To illustrate the effectiveness of advanced primer type performance, start with the same initial conditions as for the simple blowback problem with the added provision that half the impulse of the propellant gas is used *to* **stop** the returning bolt just as the cartridge seats. Thus

$$\int F_g dt = \frac{0.935}{2} = 0.4675 \text{ lb-sec}$$

Eqs. 2–14 through 2–39 are again used. Since only half the impulse is available to drive the bolt in recoil, its mass must be reduced by half in order to retain the 120.4 in./sec velocity of free recoil. Thus the weight of this bolt assembly is specified as 1.5 lb and

$$v_f = \frac{\int F_g dt}{M_b} = \frac{0.4675 \times 386.4}{1.5} = 120.4 \text{ in./sec}$$

$$E_r = \frac{1}{2} M_b v_f^2 = \frac{1}{2} \times \frac{1.5}{386.4} \times 14500$$

$$= 28.2 \text{ in.-lb.}$$

According to Eq. 2–36, the approximate bolt travel is the same (2.58 in.) as that for the simple blowback gun in the preceding problem. Again, as in the earlier problem, the 2.58 in. bolt travel docs not yield totally compatible results and must be modified to meet the rate of fire of 400 rounds per minute or the cycle time of t_c = 0.15 sec.

Since the initial dynamic conditions, impulse and energy of recoil are half as much as those of the preceding problem, the spring constant must also be half in order to have the same bolt travel. Eq. 2–37 shows the firing cycle time to be

$$t_c = \left(\sqrt{\frac{\epsilon M_b}{K}} + \sqrt{\frac{M_b}{\epsilon K}}\right) \cos^{-1} \frac{F_o}{F_m} .$$

Since K = 0.5 lb/in., ϵ = 0.40, $M_b = \frac{1.5}{386.4}\ \frac{\text{lb-sec}^2}{\text{in.}}$,

and t_c = 0.15 sec,

$$\frac{F_o}{F_m} = \frac{F_o}{F_o + 0.5L} = \cos\left[\left(\frac{0.15}{0.195}\right)57.3^\circ\right]$$

$$= \cos 44^\circ 04' = 0.7185$$

Solve for F_o

$$F_o = 1.2762L.$$

Also

$$2F_o + KL = 2F_o + 0.5L = 2F_a = \frac{2\epsilon E_r}{L}$$

$$= \frac{2 \times 0.4 \times 28.2}{L}.$$

Substitute for F_o and collect terms

$$L^2 = \frac{22.56}{3.0524} = 7.39 \text{ in.}^2$$

$$L = 2.72 \text{ in.}$$

$$F_o = 3.47 \text{ lb}$$

$$F_m = 4.83 \text{ lb}$$

Recompute the time for the firing cycle

$$t_c = \left(\sqrt{\frac{\epsilon M_b}{K}} + \sqrt{\frac{M_b}{\epsilon K}}\right) \text{Cos}^{-1}\frac{F_o}{F_m}$$

$$= 0.195 \text{ Cos}^{-1}\left(\frac{3.47}{4.83}\right) = 0.195 \times 0.769 = 0.15 \text{ sec}$$

Another approach illustrates the advantage of increasing the firing rate by incorporating the advanced primer technique. The length of recoil in the preceding problems was selected to balance the dynamics of the problem and is not necessarily the ideal minimum distance. Suppose that the ideal bolt travel is 1.5 in. and that the recoil force of the simple blowback gun is acceptable. The mass of the bolt is adjusted to suit the requirements.

$\int F_g dt$ = 0.4675 lb-sec

F_o = 7.49 lb, minimum spring load, simple blowback

F_m = 10.08 lb, maximum

$$K = \frac{F_m - F_o}{L} = \frac{2.59}{1.5} = 1.727 \text{ lb/in.}$$

ϵ = 0.40, efficiency of spring system

The work W_s done to compress the springs is

$$W_s = \frac{1}{2}(F_o + F_m)L = 8.785 \times 1.5 = 13.18 \text{ in.-lb.}$$

The velocity v_f of free recoil is

$$v_f = \frac{\int F_g dt}{M_b} = \frac{g\int F_g dt}{W_b}.$$

$$\text{The recoil energy } E_r = \frac{1}{2}M_b v_f^2 = \frac{1}{2}\left(\frac{W_b}{g}\right)v_f^2.$$

Substitute for v_f

$$E_r = \frac{1}{2}\left(\int F_g dt\right)^2 \frac{g}{W}.$$

When the efficiency of the system is considered, the spring work is

$$W_s = 0.4E_r \quad \text{or} \quad E_r = 2.5W_s.$$

Substitute for E_r and solve for W_b, the weight of the bolt

$$2.5W_s = \frac{1}{2}\left(\int F_g dt\right)^2 \frac{g}{W_b}$$

$$32.95 = \frac{1}{2} \times 0.2185\left(\frac{386.4}{W_b}\right)$$

$$W_b = \frac{42.214}{32.95} = 1.281 \text{ lb.}$$

The velocity of free recoil becomes

$$v_f = \frac{g\int F_g dt}{W_b} = \frac{386.4 \times 0.4675}{1.281} = 141 \text{ in./sec}$$

The recoil energy $E_r = \frac{1}{2}\left(Mv_f^2\right) = \frac{1}{2}\left(\frac{1.281}{386.4}\right) 19880 = 32.95$ in.-lb.

The time of a firing cycle is

$$t_c = \left(\sqrt{\frac{\epsilon M_b}{K}} + \sqrt{\frac{M_b}{\epsilon K}}\right) \cos^{-1} \frac{F_o}{F_m} = 0.0970 \times 0.733 = 0.071 \text{ sec}$$

where

$$\sqrt{\frac{\epsilon M_b}{K}} = \sqrt{\frac{0.40 \times 1.281}{1.727 \times 386.4}} = \sqrt{0.000768} = 0.0277$$

$$\sqrt{\frac{M_b}{\epsilon K}} = \sqrt{\frac{1.281}{0.40 \times 1.727 \times 386.4}} = \sqrt{0.00480} = 0.0693$$

$$\cos^{-1} \frac{F_o}{F_m} = \text{Cos}^{-1}\left(\frac{7.49}{10.08}\right) = 42^\circ = 0.733 \text{ rad}$$

The firing rate is

$$f_r = \frac{60}{t_c} = \frac{60}{0.071} = 845 \text{ rounds/min}.$$

2-3.2.2 Driving Spring Design

The driving spring for the advanced primer ignition blowback gun has been assigned the following characteristics to comply with the requirements of the firing cycle for the simple blowback gun:

$K = 0.5$ lb/in., spring constant

$F_o = 3.48$ lb, spring force at assembled height

$F_m = 4.85$ lb, spring force at end of recoil

$L = 2.73$ in., bolt travel

$t_r = T_c = 0.0428$ sec, compression time of spring

$v_f = v_i = 120.4$ in./sec, velocity of free recoil, spring impact velocity

Select $\frac{T_c}{T}$ = 3.8. Therefore, T = $\frac{0.0428}{3.8}$ = 0.1 126 sec.

When D = 0.5 in., according to Eq. 2–42

$$d = 0.27\sqrt[3]{DKT} = 0.27\sqrt[3]{0.5 \times 0.5 \times 0.01125} = 0.038 \text{ in.}$$

From Eq. 2–41

$$N = \frac{Gd^4}{8D^3K} = \frac{11.5 \times 10^6 \times 208 \times 10^{-8}}{8 \times 0.125 \times 0.5} = 48 \text{ coils}$$

$$H_s = Nd = 48 \times 0.038 = 1.83 \text{ in., solid height.}$$

The static torsional stress, Eq. 2–43, is

$$\tau = \frac{8F_mD}{\pi d^3} = \frac{8 \times 4.85 \times 0.5}{54.8 \times 10^{-6}\pi} = 113{,}000 \text{ lb/in.}^2$$

Eq. 2–44 has the dynamic stress of

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right] = 113000\left(\frac{1}{3.8}\right)4 = 119{,}000 \text{ lb/in.}^2$$

The driving spring for the advanced primer ignition when the recoil force is equal to that of the simple blowback gun has the following characteristics:

K = 1.727 lb/in., spring constant

F_o = 7.49 lb, spring force at assembled height

F_m = 10.08 lb, spring force at end of recoil

L = 1.5 in., bolt travel

$t_r = T_c$ = 0.0203 sec, compression time of spring

$v_f = v_i$ = 141 in./sec, velocity of free recoil

Select $\frac{T_c}{T} = 3.8$. Therefore, $T = \frac{0.0203}{3.8} = 0.00535$ sec.

When $D = 0.5$, according to **Eq.** 2–42

$$d = 0.27 \sqrt[3]{DKT} = 0.27 \sqrt[3]{0.5 \times 1.727 \times 0.00535} = 0.045 \text{ in.}$$

From **Eq.** 2–41

$$N = \frac{Gd^4}{8D^3K} = \frac{11.5 \times 10^6 \times 41 \times 10^{-7}}{8 \times 0.125 \times 1.727} = 27.3 \text{ coils}$$

$$H_s = Nd = 27.3 \times 0.045 = 1.23 \text{ in., solid height.}$$

The static torsional stress, **Eq.** 2–43, **is**

$$\tau = \frac{8F_mD}{\pi d^3} = \frac{8 \times 10.08 \times 0.5}{91.1 \times 10^{-6}\pi} = 141{,}000 \text{ lb/in}^2$$

The dynamic stress, **Eq.** 2–44, is

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right] = 141{,}000\left(\frac{1}{3.8}\right)4 = 148{,}500 \text{ lb/in}^2$$

2–4 DELAYED BLOWBACK

Delayed blowback is the system that keeps the bolt locked until the projectile leaves the muzzle. At this instant an unlocking mechanism, responding to some influence such as recoil or propellant gas pressure, releases the bolt thereby permitting blowback to take effect.

2–4.1 SPECIFIC REQUIREMENTS

Since the tremendous impulse developed by the propellant gases while the projectile is in the bore is not available for operating the bolt, the recoiling **mass** – including driving, buffing, and barrel springs – need not be nearly so heavy as the two types of blowback discussed earlier. The smaller recoiling mass moves relatively faster and the rate of fire increases correspondingly. Delayed blowback guns may borrow operating principles from other types of action, e.g., the piston action of the gas operating gun or the moving recoiling parts of the recoil operating gun. In either case, only unlocking activity is associated with these two types, the primary activity involving bolt action still functions according to the blowback principle. Fig. *2–6* shows **a** simple locking system.

Like any other automatic gun, bolt action is congruous with timing particularly with respect to unlocking time. If recoil operated, distance also becomes an important factor. For this type gun, the barrel must recoil a short distance before the moving parts force open the bolt lock. Sufficient time should elapse to permit the propellant gas pressure to drop to levels below the bursting pressure of the cartridge case but retain enough intensity to blow back the bolt.

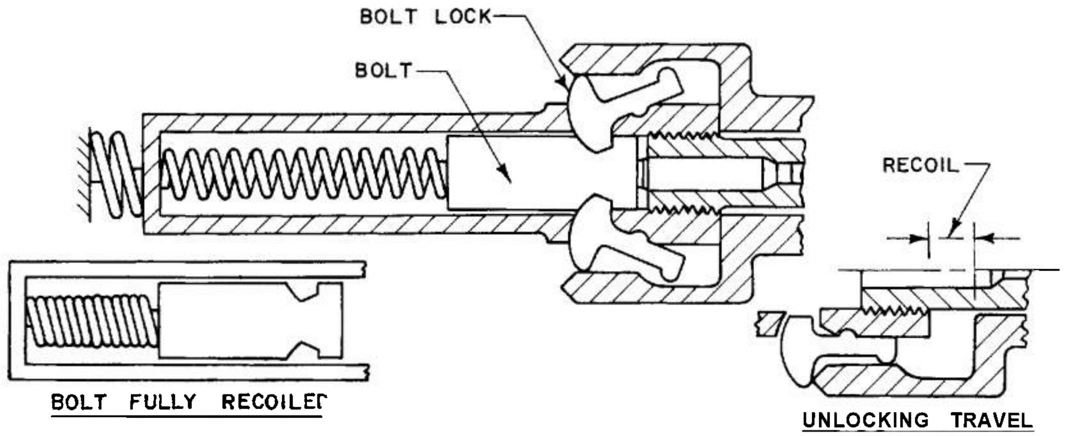


Figure 2–6. Locking System for Delayed Blowback

The stiffness of the springs should not be so great as to interfere unduly with early recoil. Therefore, a system consisting of three springs is customarily used: (1) a barrel spring having an initial load slightly larger than the recoiling weight to insure almost free recoil and still have the capacity to hold the barrel in battery, (2) a buffer spring to stop the recoiling parts and return them, and (3) a bolt driving spring to control bolt activity. Before the bolt is unlocked, all moving parts recoil as one mass with only the barrel spring resisting recoil but this spring force is negligible compared to the propellant gas force and may be neglected during recoil. After the bolt becomes unlocked, the barrel spring combines with the buffer spring to arrest the recoiling barrel unit.

The unlocked bolt continues to be accelerated to the rear by the impetus of the decaying propellant gas pressure whose only resistance now is the force of the driving spring, a negligible resistance until the propellant gas pressure becomes almost zero. Thereafter, the spring stops the bolt and later closes it. Normally the barrel unit has completed counterrecoil long before the bolt has fully recoiled to provide the time and relative distance needed for extracting, ejecting, and loading. After the barrel unit is in battery, the bolt unit functions as a single spring unit.

2–4.2 DYNAMICS OF DELAYED BLOWBACK

While the complete unit is recoiling freely and later while all springs are operating effectively, the dynamics of the system are readily computed by an iterative process. Given the pressure-time curve, by knowing the size of the masses in motion, the dynamics at any given time are determined by the summations of computed values for all preceding increments of time. The impulse during each increment is

$$F\Delta t = A_b A_i \qquad (2\text{–}45)$$

where A_b = bore area

A_i = area under At of pressure-time curve

At = time increment

When the impulse is being determined during low pressure periods due consideration should be given to the resistance offered by the driving spring. $F_g\Delta t$ should be adjusted after the driving spring and gas pressure forces become relatively significant. During each increment, the differential velocity is

$$Av = \frac{F_g\Delta t}{M_r} \qquad (2\text{–}46)$$

where M_r = mass **of** the **recoiling parts influenced** by FAt.

The velocity of recoil at the end of each increment becomes

$$v = v_{(n-1)} + Av \qquad (2\text{–}47)$$

where $v_{(n-1)}$ = velocity at the preceding increment

The distance traveled by the bolt with respect to the gun frame during the increment is

$$\Delta x = v_a \Delta t = \left(v_{(n-1)} + \frac{1}{2} \Delta v\right) \Delta t \qquad (2\text{–}48)$$

where v_a = average velocity for the increment.

The total distance at the end of each increment is

$$x = \Sigma \Delta x. \qquad (2\text{–}49)$$

When the propellant gas pressures cease to be effective, the behavior of the barrel and bolt units depend entirely on springs. One such instance involving the buffer spring occurs when the bolt is unlocked. Although the gas pressure continues its effective action on the bolt, it can no longer influence the barrel unit except secondarily through the driving spring. Rewrite **Eq. 2–19** for the isolated barrel unit and include the influence of the driving spring. Thus, the energy remaining in the recoiling mass is

$$\frac{1}{2}\left(M_t v_t^2\right) = \frac{1}{2}\left(M_t v_{ot}^2\right) - \frac{1}{\epsilon_b}\left[F_{ob} x_t + \frac{1}{2}\left(K_b x_t^2\right)\right] + \frac{1}{\epsilon}\left(F x_t\right) \qquad (2\text{–}50)$$

where F = drive spring force

F_{ob} = initial buffer spring force

K_b = spring constant of combined buffer and barrel springs

M_t = mass of barrel unit

v_{ot} = initial velocity of barrel unit

v_t = final velocity of barrel unit

x_t = travel of barrel with respect to gun frame

ϵ = efficiency of drive spring unit

ϵ_b = efficiency of buffer spring unit which includes the barrel spring

The buffer spring performs in unison with the barrel spring. During counterrecoil, at the end of buffer spring travel, the barrel spring continues to accelerate the barrel unit in its return to battery. During recoil, the propellant gas force is so much larger than the barrel spring force as to render the latter practically ineffective. Except for the last millisecond or two, the bolt driving spring also offers a negligible resistance to the propellant gas force. However, it does contribute a small force opposing the buffer spring and is represented in Eq. 2–50 by the expression $\frac{1}{e}\left(Fx_t\right)$, the effective force of the driving spring. The actual spring force F may be assumed to be the driving spring force at the time when the bolt is unlocked. Preliminary estimates should provide reasonable approximations at this stage of the design study.

An equation can be derived for the recoil time of the barrel unit by developing **Eq. 2–50** by the same procedure used for Eq. **2–19.** The recoil time for the barrel after bolt unlocking until pressure becomes zero is

$$t_{rt} = \sqrt{\frac{\epsilon_b M_t}{K_b}}\left[\sin^{-1}\frac{K_b x_{to} + F_{ob} - \left(\frac{\epsilon_b}{\epsilon}\right)F}{\sqrt{F_{ob} - \left(\frac{\epsilon_b}{\epsilon}\right)^2 + \epsilon_b K_b M_t v_{ot}^2}} - \sin^{-1}\frac{F_{ob} - \frac{\epsilon_b}{\epsilon}F}{\sqrt{F_{ob} - \left(\frac{\epsilon_b}{\epsilon}\right)F^2 + \epsilon_b K_b M_t v_{ot}^2}}\right] \tag{2–51}$$

where x_{to} = barrel travel from time of buffer engagement to end of propellant gas period

All values are known except x_{to}. Since t_{rt} is the time elapsed from bolt unlocking to pressure effectiveness reaching zero, this distance may be computed from Eq. **2–51.** It represents the buffer spring deflection. The total barrel travel with respect to the frame is

$$x_t = x_{to} + x_{tf} \tag{2–52}$$

where x_{tf} = barrel travel during free recoil.

The amount that the driving spring is compressed, while the barrel traverses x_t, is the relative travel distance between barrel and bolt, thus

$$x_b = x - x_t \tag{2–53a}$$

In terms of differential values, the equation becomes

$$\Delta x_b = \Delta X - \Delta x_t \tag{2–53b}$$

On the assumption that the recoil velocity of the bolt has been computed at the time corresponding to x_{to}, the energy of the bolt can be computed and converted to the potential energy of the driving spring from which the spring forces may be determined. The average driving spring force over the remaining distance, Eq. **2–16,** is

$$F_a = \frac{\epsilon E_b}{L_b - x_b} \tag{2–54}$$

where E_b is calculated according to Eq. **2–15** and $L_b - x_b$ is the spring deflection remaining at the end of free recoil of the bolt.

For the remainder of the buffer stroke and for the time that the barrel unit is counterrecoiling, the dynamics of the system may be computed by dividing the time into convenient intervals, and by use of the relationship existing between impulse and momentum, computing the dynamics for each corresponding increment of travel. Both recoil and counterrecoil of the barrel take place while the bolt is recoiling which changes the effective buffer spring forces for the two directions. The expression of the driving spring effective force does not change since the bolt travel direction does not change. The force of the driving spring at the end of each increment of travel is

$$F = F_{(n-1)} + K\Delta x_b \tag{2–55}$$

where

$F_{(n-1)}$ = driving spring force at beginning of increment

K = driving spring constant

Δx_b = incremental driving spring deflection

The buffer spring force at the end of its increment of travel is

$$F_b = F_{b(n-1)} + K_b \Delta x_t \tag{2–56}$$

where

$F_{b(n-1)}$ = buffer spring force at beginning of increment

K_b = buffer spring constant which includes the barrel spring

Δx_t = incremental buffer spring deflection

The effective spring force on the bolt while it is recoiling is

$$F_e = \frac{F + F_{(n-1)}}{2\epsilon} \tag{2-57}$$

The effective spring force on the barrel while it is recoiling is

$$F_{eb} = \frac{F_b + F_{b(n-1)}}{2\epsilon_b} - F_e . \tag{2-58}$$

According to the relationship of impulse and momentum, $Fdt = Mv$, the general expression for differential velocity is

$$\Delta v = \frac{F\Delta t}{M} . \tag{2-59}$$

Based on this expression, the bolt travel during each increment is derived in a sequence of algebraic expressions. Thus

$$\Delta x = v_{(n-1)}\Delta t - \frac{1}{2}\left(\Delta v \Delta t\right) = v_{(n-1)}\Delta t - \frac{F_e \Delta t^2}{2M_b} \tag{2-60a}$$

$$\Delta x = v_{(n-1)}\Delta t - \frac{F_{(n-1)}\Delta t^2}{2\epsilon M_b} - \left(\frac{K\Delta t^2}{4\epsilon M_b}\right)\Delta x_b . \tag{2-60b}$$

But $\Delta x_b = \Delta x - \Delta x_t$ (see Eq. 2-53b). Substituting this expression and collecting terms, the incremental travel of the bolt becomes

$$\Delta x = \frac{4\epsilon M_b}{4\epsilon M_b + K\Delta t^2}\left[v_{(n-1)}\Delta t - \frac{F_{(n-1)}\Delta t^2}{2\epsilon M_b} + \left(\frac{K\Delta t^2}{4\epsilon M_b}\right)\Delta x_t\right] . \tag{2-60c}$$

The incremental travel for the barrel is a similar expression

$$\Delta x_t = v_{t(n-1)}\Delta t - \frac{1}{2}\left(\Delta v_t\,\Delta t\right) = v_{t(n-1)}\Delta t - \frac{F_{eb}\Delta t^2}{2M_t} \qquad (2\text{–}61a)$$

$$\Delta x_t = v_{t(n-1)}\Delta t - \frac{\Delta t^2}{2M_t}\left[\frac{F_{b(n-1)}}{\epsilon} + \frac{K_b\Delta x_t}{\epsilon} - \frac{F_{(n-1)}}{\epsilon} - \frac{K\Delta x_b}{2\epsilon}\right]. \qquad (2\text{–}61b)$$

Again substituting $\Delta x - \Delta x_t$ for Δx_b and collecting terms, the incremental travel of the barrel in recoil is

$$\Delta x_t = \frac{4\epsilon\epsilon_b M_t}{4\epsilon\epsilon_b M_t + \epsilon K_b\Delta t^2 + \epsilon_b K\Delta t^2}\left[v_{t(n-1)}\Delta t - \frac{F_{b(n-1)}\Delta t^2}{2\epsilon_b M_t} + \frac{F_{(n-1)}\Delta t^2}{2\epsilon M_t} + \left(\frac{K\Delta t^2}{4\epsilon M_t}\right)\Delta x\right]. \qquad (2\text{–}61c)$$

While the barrel is counterrecoiling, and the bolt recoiling, the effective spring force on the barrel is

$$F_{eb} = \epsilon_b F_b - F_e = \epsilon_b F_{b(n-1)} - \left(\frac{\epsilon_b}{2}\right)K_b\Delta x_t - \frac{F_{(n-1)}}{\epsilon} - \frac{K\Delta x_b}{2\epsilon}. \qquad (2\text{–}62)$$

The incremental travel of the barrel now becomes

$$\Delta x_t = v_{t(n-1)}\Delta t + \frac{1}{2}\left(\Delta v_t\Delta t\right) = v_{t(n-1)}\Delta t + \frac{F_{eb}\Delta t^2}{2M_t} \qquad (2\text{–}63a)$$

$$\Delta x_t = v_{t(n-1)}\Delta t + \frac{\epsilon_b F_{b(n-1)}\Delta t^2}{2M_t} - \left(\frac{\epsilon_b K_b\Delta t^2}{4M_t}\right)\Delta x_t - \frac{F_{(n-1)}\Delta t^2}{2\epsilon M_t} - \left(\frac{K\Delta t^2}{4\epsilon M_t}\right)\Delta x_b. \qquad (2\text{–}63b)$$

Substitute $\Delta x - \Delta x_t$ for Δx_b, collect terms and solve for Δx_t

$$\Delta x_t = \frac{4\epsilon M_t}{4\epsilon M_t + \epsilon\epsilon_b K_b\Delta t^2 - K\Delta t^2}\left[v_{t(n-1)}\Delta t + \frac{\epsilon_b F_{b(n-1)}\Delta t^2}{2M_t} - \frac{F_{(n-1)}\Delta t^2}{2\epsilon M_t} - \left(\frac{K\Delta t^2}{4\epsilon M_t}\right)\Delta x\right]$$

(2–63c)

While both bolt and barrel are counterrecoiling, the effective spring force on the bolt is

$$F_e = \epsilon(F + F_{n-1})/2 = \epsilon F_{(n-1)} - \epsilon K \Delta x_b/2. \qquad (2\text{–}64a)$$

Now, substitute for F_e and expand **Eq.** 2–60a so that

$$\Delta x = v_{(n-1)}\Delta t - \left[\frac{\epsilon F_{(n-1)}}{2M_b} - \left(\frac{\epsilon K}{4M_b}\right)\Delta x_b\right]\Delta t^2. \qquad (2\text{–}64b)$$

But, according to **Eq.** 2–53b, Δx_b = ***Ax*** - Δx_t, therefore

$$\Delta x = v_{(n-1)}\Delta t - \left[\frac{\epsilon F_{(n-1)}}{2M_b}\right]\Delta t^2 + \left(\frac{\epsilon K}{4M_b}\right)\Delta t^2\,\Delta x - \left(\frac{\epsilon K}{4M_b}\right)\Delta t^2\,\Delta x_t. \qquad (2\text{–}64c)$$

Collect terms and solve for ***Ax***

$$\Delta x = \frac{4M_b}{4M_b - \epsilon K \Delta t^2}\left[v_{(n-1)}\Delta t - \frac{\epsilon F_{(n-1)}\Delta t^2}{2M_b} - \left(\frac{\epsilon K \Delta t^2}{4M_b}\right)\Delta x_t\right]. \qquad (2\text{–}64d)$$

The effective force on the barrel during this period is

$$F_{eb} = \epsilon_b F_b - F_e = \epsilon_b\left[F_{b(n-1)} - \frac{K_b \Delta x_t}{2}\right] - \epsilon\left[F_{(n-1)} - \frac{K\Delta x_b}{2}\right]. \qquad (2\text{–}65a)$$

The incremental barrel travel is, according to **Eq.** 2–63a

$$\Delta x_t = V_{t(n-1)}\Delta t + \frac{F_{eb}\Delta t^2}{2M_t}. \qquad (2\text{–}65b)$$

Substitute the expression for F_{eb} of **Eq.** 2–65b, Δx - Δx_t for Δx_b, and collect terms. The incremental barrel travel now becomes

$$Ax, = \frac{4M_t}{4M_t + (\epsilon K + \epsilon_b K_b)\Delta t^2}\left\{v_{t(n-1)}\Delta t + \left[\frac{\epsilon_b F_{b(n-1)}}{2M_t}\right]\Delta t^2 - \left[\frac{\epsilon F_{(n-1)}}{2M_t}\right]\Delta t^2 + \frac{\epsilon K}{4M_t}\,\Delta t^2 \Delta x\right\}. \qquad (2\text{–}65c)$$

To avoid repetition, general expressions are used to complete the analysis. The distinction between bolt and barrel activities will be demonstrated later in the sample problem. The spring force at the end of each increment is computed by **Eqs.** 2–55 and 2–56. The total energy absorbed by a spring system over a distance Δx is

$$\Delta E = \left[\frac{F_{(n-1)} + F}{2\epsilon} \right] \Delta x. \qquad (2\text{–}66a)$$

The energy released by a spring system over a distance Δx is

$$\Delta E = \epsilon \left[\frac{F_{(n-1)} + F}{2} \right] \Delta x. \qquad (2\text{–}66b)$$

The total energy at the end of the increment is found by adding the incremental energy to the total at the end of the preceding increment when energy is released, or subtracting when it is being absorbed.

$$E = E_{(n-1)} \pm \Delta E \qquad (2\text{–}66c)$$

$$v = \sqrt{\frac{2E}{M}} \qquad (2\text{–}66d)$$

2–4.3 SAMPLE PROBLEM FOR DELAYED BLOWBACK ACTION

2–4.3.1 Specifications

Gun: 20 mm machine gun

Firing rate: corresponding to minimum bolt travel

Interior ballistics: Pressure vs Time, Fig. 2–7

A_b = 0.515 in.^{2}, bore area

2–4.3.2 Design Data

L = 10 in., minimum bolt travel

W_b = 10 lb, weight of bolt unit

W_t = 50 lb, weight of barrel unit

x_{to} = 0.5 in., recoil distance to unlock bolt

ϵ = 0.5, efficiency of driving spring unit

ϵ_b = 0.3, efficiency of combined buffer and barrel spring system

ϵ_t = 0.5, efficiency of barrel spring unit

Table 2–2 has the numerical integration for a recoiling weight of 60 lb. In Table 2–2, the area A_i is measured under the pressure-time curve, Fig. 2–7

$$F_g \Delta t = A_b A_i = 0.515 A_i \text{ lb-sec}$$

$$\Delta v = \frac{F_g \Delta t}{M_r}$$

$$= 6.44\ F\Delta t \text{ in./sec when } t \leq 0.003252 \text{ sec}$$

where

$$M_r = \frac{W_r}{g} = \frac{60}{386.4}$$

$$\Delta v = \frac{F_g \Delta t}{M_b}$$

$$= 38.64\ F_g \Delta t \text{ in./sec when t} > 0.003252 \text{ sec}$$

where

$$M_b = \frac{W_b}{g} = \frac{10}{386.4}$$

$$\Delta x = v_a \Delta t = \left[\frac{v_{(n-1)} + v}{2} \right] \Delta t, \text{ in.}$$

When the bolt is unlocked at t = 3.252 msec, the velocity attained by the recoiling parts is 232.3 in./sec (see Table 2–2).

The energy of the barrel unit at this velocity is

$$E_{to} = \frac{1}{2}\left(M_t v_t^2 \right) = \frac{1}{2}\left(\frac{50}{386.4} \right) 53960$$

$$= 3491 \text{ in.-lb.}$$

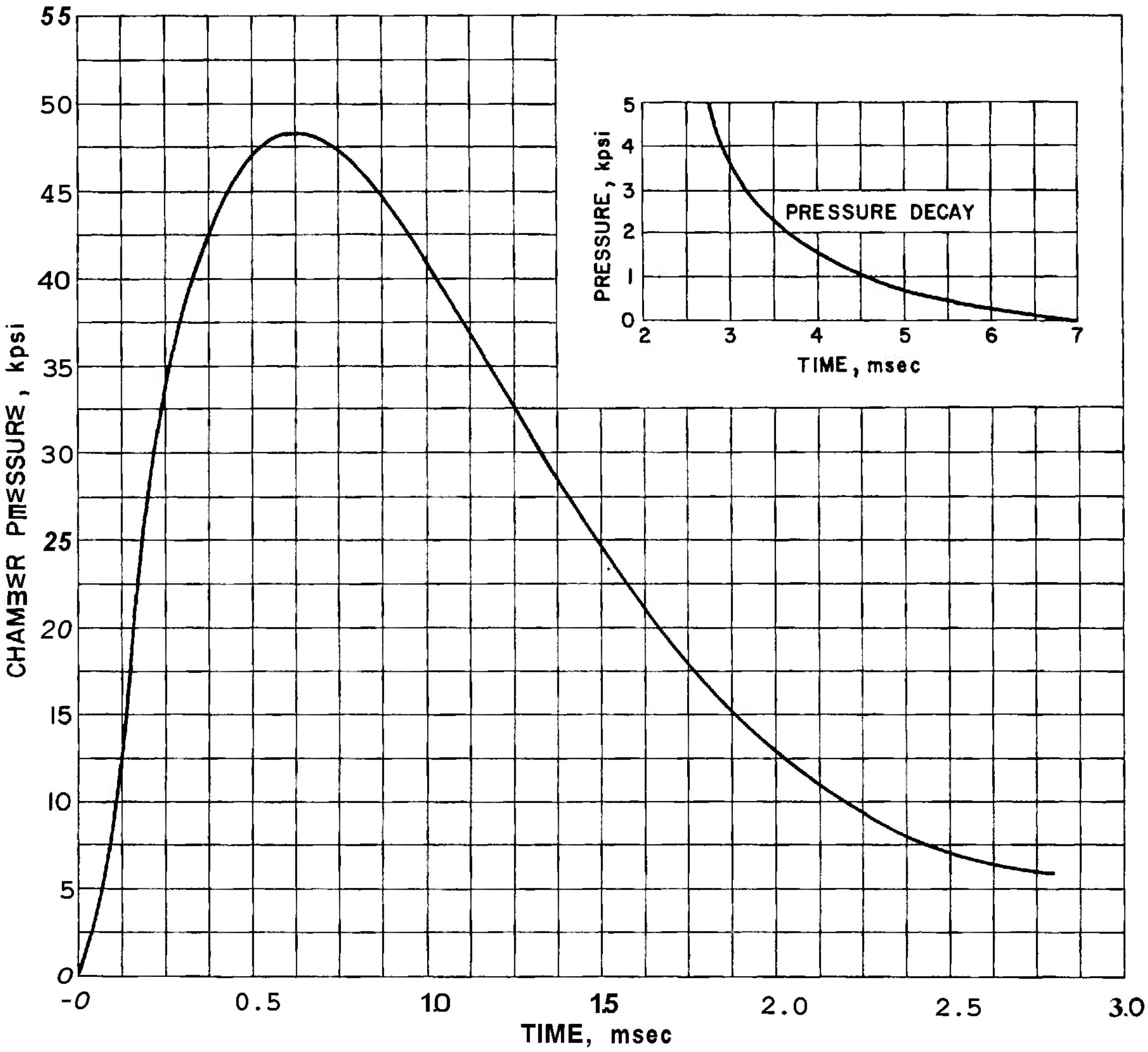


*Figure 2–7. Pressure-time Curve **of** 20 mm **Round***

The velocity attained by the bolt at the end of the pressure period, ν = 285 in./sec (see Table 2–2). The energy of the bolt is

$$E_b = \frac{1}{2}\left(M_b \nu^2\right) = \frac{1}{2}\left(\frac{10}{386.4}\right) 81200$$

$$= 1051 \text{ in.-lb.}$$

On the assumption that the driving spring system absorbs this energy over its total deflection of 10 in., the average force is

$$F_a = \frac{\epsilon_b E_b}{L} = \frac{0.5 \times 1051}{10} = 52.6 \text{ lb.}$$

Earlier, a spring constant of $K=3$ appeared practical, but the resulting stress was too large. Also an earlier attempt at having a buffer stroke of half an inch proved impractical from the dynamic stress point of view. Increasing the stroke to one inch and applying recommended dynamic spring behavior, meanwhile retaining an acceptable firing rate, led to a feasible spring constant for all three springs – driving, buffer,

and barrel. The allowable static shear stress of the spring is

$$\tau = \frac{8F_mD}{\pi d^3} \quad \text{(see Eq. 2-43)}$$

Since $T = 3.55 \times 10^{-5}\left(\frac{D^2}{d}\right)N$ (Ref. 1) and

$$N = \frac{Gd^4}{8KD^3} \quad \text{(see Eqs. 2-40 and 2-41)}$$

substitute for T and N and solve for $\frac{d^3}{D}$ in the equation for τ and T.

Thus

$$\frac{d^3}{D} = \frac{8F_m}{\pi\tau} = \frac{8KT}{3.55 \times 10^{-5}G} \qquad (2\text{-}67a)$$

where $G = 11.5 \times 10^6$ lb/in.^{2}, torsional modulus

$\tau = 135{,}000$ lb/in.^{2}, allowable static shear stress

$$KT = \frac{F_m}{1037} \qquad (2\text{-}67b)$$

Based on constant acceleration, an approximate time of spring compression will, in most applications, determine a spring constant compatible with both spring dynamics and allowable stresses. The approximate time of bolt recoil is

$$t_r = \frac{2x_b}{v} = \frac{2(10)}{285} = 0.070 \text{ sec.}$$

TABLE 2-2. RECOIL TRAVEL OF 20 mm GUN

t, msec	Δt, msec	A_i, lb-sec/in.^{2}	$F_g\Delta t$, lb-sec	Δv, in./sec	v, in./sec	v_a', in./sec	Δx, in.	x, in.
0.25	0.25	3.44	1.77	11.4	11.4	5.7	0.0014	0.0014
0.50	0.25	10.15	5.24	33.7	45.1	28.2	0.0071	0.0085
0.75	0.25	11.89	6.13	39.5	84.6	64.8	0.0162	0.0247
1.00	0.25	11.12	5.74	37.0	121.6	103.1	0.0258	0.0505
1.25	0.25	9.25	4.76	30.6	152.2	136.9	0.0342	0.0847
1.50	0.25	7.30	3.76	24.2	176.4	164.3	0.0411	0.1258
1.75	0.25	5.23	2.69	17.3	193.7	185.0	0.0462	0.1720
2.00	0.25	3.71	1.91	12.6	206.3	200.0	0.0500	0.2220
2.25	0.25	2.58	1.34	8.6	214.9	210.6	0.0526	0.2746
2.50	0.25	1.82	0.94	6.1	221.0	218.0	0.0545	0.3291
2.75	0.25	1.39	0.72	4.6	225.6	223.3	0.0588	0.3849
3.00	0.25	1.06	0.55	3.5	229.1	227.4	0.0569	0.4418
3.252	0.252	0.97	0.50	3.2	232.3	230.7	0.0582	0.5000
4.00	0.748	1.37	0.70	27.0	258.3	245.3	0.1838	0.6838
5.00	1.00	0.97	0.50	19.3	278.6	269.0	0.2690	0.9528
6.00	1.00	0.32	0.165	6.4	285.0	281.8	0.2818	1.2346

Since $v < 25$ ft/sec, (see par. 2–2.3.5), $\frac{Tc}{T} = 3.8$, $T_c = t_r$ and $F_m = F_a + \frac{1}{2}(KL)$ (Eqs. 2–17 and 2–18), Eq. 2–67b, after $T_c/3.8$ is substituted for T, becomes

$$KT_c = \frac{F_a + 0.5KL}{273}$$

$$0.070K = \frac{52.6 + 5K}{273}.$$

Solve for K, thus

$$K = \frac{52.6}{14.11} = 3.72 \text{ lb/in.}$$

$$F_m = F_a + \frac{1}{2}\left(KL\right) = 52.6 + 5 \times 3.72 = 71.2 \text{ lb}$$

$$F_o = F_m - KL = 71.2 - 37.2 = 34.0 \text{ lb.}$$

Compute t from Eq. 2–23

$$t = \sqrt{\frac{\epsilon M_b}{K}}\ \mathrm{Cos}^{-1}\frac{F_o}{F_m}$$

$$= \sqrt{\frac{0.5 \times 10}{3.72 \times 386.4}}\ \mathrm{Cos}^{-1}\left(\frac{34.0}{71.2}\right)$$

$$= 0.059\left(\frac{61.5}{57.3}\right) = 0.063 \text{ sec.}$$

Recomputing by inserting the newly calculated time t for T_c, the data converge to $K = 4.4$ lb/in. and $t = 0.062$ sec.

The first set of detailed calculations (not shown) yielded a bolt recoil time of $t = T_c = 51.5$ msec. For this time, the computed spring constant of $K = 5.8$ lb/in. becomes the final value for computing the dynamics of Table 2–3.

The spring constant of the buffer system is found by assuming that the energy $E_{to} = 3491$ in.-lb will be absorbed over the 1-inch buffer stroke. Later, adjustments will be made to compensate for the small discrepancies involving the spring forces. The time t_b of buffer action during recoil is

$$t_b = \frac{2L_b}{v_b} = \frac{2 \times 1.0}{232.3} = 0.0086 \text{ sec}$$

where L_b = buffer stroke

v_b = recoil velocity of barrel when buffer is contacted

The average buffer force F_{ab} is

$$F_{ab} = \frac{\epsilon_b E_{to}}{L_b} = \frac{0.3 \times 3491}{1.0} = 1047 \text{ lb.}$$

Follow the same procedure for the buffer as for the driving spring

$$K_b T_c = \frac{F_{ab} + 0.5K_b L_b}{273}$$

$$273 \times 0.0086 K_b = 1047 + 0.5K_b.$$

Solve for K_b, thus

$K_b = \frac{1047}{1.85} = 566$ lb/in., buffer spring constant

$F_{mb} = 1047 + 283 = 1330$ lb, max buffer force

$F_{ob} = 1047 - 283 = 764$ lb, min buffer force

Compute t_b from Eq. 2–23

$$t_b = \sqrt{\frac{\epsilon_b M_t}{K_b}}\ \mathrm{Cos}^{-1}\frac{F_{ob}}{F_{mb}}$$

$$= \sqrt{\frac{0.3 \times 50}{566 \times 386.4}}\ \mathrm{Cos}^{-1}\left(\frac{764}{1330}\right) = 0.0079 \text{ sec.}$$

Recompute K_b by inserting the time 0.0079 sec for T_c. The new value of the spring constant K_b is 630 lb/in., but the new time remains $t_b = 0.0079$ sec.

The spring constant for the barrel spring is computed similarly but in this instance the spring force at the assembled height is set at F_o = 70 lb, a minimal value so that recoil distance and velocity are appropriate for bolt action. The time of barrel spring compression is the same as that for the buffer plus the propellant gas period.

$$t_t = t_b + t = 0.0079 + 0.006 = 0.0139 \text{ sec}$$

The barrel spring has an operating deflection of L_t = 2.0 in.

$$F_{mt} = F_{ot} + K_t L_t = 70 + 2K_t \text{, max barrel spring force}$$

With $T = \frac{T_c}{3.8} = \frac{0.0139}{3.8}$ sec,

Eq. 2–67b may be written as

$$0.0139 K_t = \frac{70 + 2K_t}{273}.$$

Thus

$$K_t = \frac{70}{1.8} = 39 \text{ lb/in., barrel spring constant}$$

$$F_{mt} = 70 + 78 = 148 \text{ lb, max barrel spring force}$$

Before recomputing time t_t, according to Eq. 2–23, some allowance must be made for the effective barrel mass. Since both barrel and buffer springs are active over the buffer stroke, a logical distribution can be arranged according to the average spring forces. The effective mass for the barrel spring is

$$M_e = \frac{W_t}{g}\left(\frac{F_{ot} + F_{mt}}{F_{ob} + F_{mb}}\right) = \frac{50}{g}\left(\frac{218}{2096}\right) = \frac{5.2}{g}$$

$$t_t = \sqrt{\frac{\epsilon_t M_e}{K_t}} \cos^{-1}\frac{F_{ot}}{F_{mt}}$$

$$= \sqrt{\frac{0.5 \times 5.2}{39 \times 386.4}} \cos^{-1}\left(\frac{70}{148}\right)$$

$$= 0.0132\left(\frac{61.8}{57.3}\right) = 0.0142 \text{ sec.}$$

Substitute the new value of $T_c = t_t$ and compute. The time and spring constant coverge to t_t = 0.0143 sec and K_t = 37 lb/in., and F_m = 144 lb. The buffer spring constant is

$$K_{bs} = K_b - K_t = 630 - 37 = 593 \text{ lb/in.}$$

Now that the constants of all three springs are firmly established, more exact values of the minimum and maximum spring forces of the buffer spring system are computed. Since the driving spring does deflect while the propellant gas pressure is still effective, less than the full spring deflection is available to absorb the bolt energy. For this reason, increasing the initial load to F_o = 25 lb for early estimates seems advisable. This spring force became effective and therefore is included in Table 2–2 after the 4 msec interval. In the meantime, the driving spring transfers some of the bolt energy to the moving barrel. The average driving spring force after bolt unlocking until, the barrel stops recoiling is approximately 27 lb.

$$E_t = E_{to} + \left(\frac{F_a}{\epsilon}\right)L_t'$$

$$= 3491 + \left(\frac{27}{0.5}\right)1.5 = 3572 \text{ in.-lb}$$

where L_t' is the barrel travel after the bolt is unlocked. From Table 2–2, when t = 3.252, x = 0.5, then $L_t' = L_t - x = 2.0 - 0.5 = 1.5$ in.

The energy absorbed by the barrel spring over the half-inch travel before the buffer is contacted is

$$\Delta E_t = \frac{1}{2}\left[\frac{(F_t' + F_t'')\Delta L}{\epsilon_t}\right] = \frac{(88.5 + 107)0.5}{2 \times 0.5}$$

$$= 98 \text{ in.-lb}$$

where F_t' and F_t'' are the barrel spring forces at half-inch and one-inch travel positions, respectively.

The average force of the buffer spring system found according to Eq. 2–30 is

$$F_{ab} = \frac{\epsilon_b(E_t - \Delta E_t)}{L_b} = \frac{0.3 \times 3474}{1.0} = 1042 \text{ lb.}$$

The initial force of the buffer spring system is

$$F_{ob} = F_{ab} - \frac{1}{2}\left(K_b L_b \right) = 1042 - \frac{1}{2}\left(630 \right)1.0 = 727 \text{ lb}$$

$$F_{obs} = F_{ob} - \left(F_{ot} + 1.0K_t \right) = 620 \text{ lb}$$

where K_b = **630** lb/in., the buffer spring system constant.

The maximum buffer spring system force is

$$F_{mb} = F_{ob} + K_b L_b = 727 + 630 \times 1.0 = 1357 \text{ lb}$$

$$F_{mbs} = F_{mb} - F_{mt} = 1213 \text{ lb}$$

The impact velocity of the barrel on the buffer spring is

$$v_i = \sqrt{\frac{2(E_t - \Delta E_t)}{M_t}} = \sqrt{\frac{2(3572 - 98)386.4}{50}} = 231.7 \text{ in./sec.}$$

The bolt is unlocked at **3.252** msec and continues to be accelerated until **6** msec. During most of the remaining time of **2.748** msec the barrel spring is the sole resistance to barrel recoil. After the barrel recoils a half-inch farther, the barrel spring is joined by the buffer spring. Based on **Eq. 2–51,** the time increment for this half-inch travel is

$$\Delta t_{rt} = \sqrt{\frac{\epsilon_t M_t}{K_t}} \left[\text{Sin}^{-1} \frac{F_t'' - \left(\frac{\epsilon_t}{\epsilon}\right)F}{Z} - \text{Sin}^{-1} \frac{F_t' - \left(\frac{\epsilon_t}{\epsilon}\right)F}{Z} \right]$$

$$= \sqrt{\frac{0.5 \times 50}{37 \times 386.4}} \left[\text{Sin}^{-1}\left(\frac{82}{369} \right) - \text{Sin}^{-1}\left(\frac{63.5}{369} \right) \right]$$

$$\Delta t_{rt} = \mathbf{0.0418\,(12.84 - 9.91)}/57.3 = \mathbf{0.002137} \text{ sec}$$

where F = 25 lb, estimated driving spring force during period

F_t' = 88.5 lb, barrel spring force when bolt is unlocked

F_t'' = 107 lb, barrel spring force when buffer is contacted

ϵ = 0.50, efficiency of driving spring

ϵ_t = 0.50, efficiency of barrel spring

$$\left[F_t' - \left(\frac{\epsilon_t}{\epsilon}\right)F\right]^2 = 63.5^2 = 4032 \text{ lb}^2$$

$$\epsilon_t K_t M_t v_t^2 = 0.5 \times 37 \times 2E_t = 132{,}164 \text{ lb}^2$$

$$Z = \sqrt{\left[F_t' - \left(\frac{\epsilon_t}{\epsilon}\right)F\right]^2 + \epsilon_t K_t M_t v_t^2} = \sqrt{136307} = 369 \text{ lb.}$$

The time still remaining during propellant gas activity is

$$t_{rt} = 0.006 - 0.003252 - 0.002137 = 0.000611 \text{ sec.}$$

During this time the barrel contacts the buffer and continues rearward over a distance that is computed according to Eq. 2–51.

$$t_{rt} = \sqrt{\frac{\epsilon_b M_t}{K_b}}\left(\text{Sin}^{-1}\frac{F_{me}}{Z} - \text{Sin}^{-1}\frac{F_{oe}}{Z}\right)$$

$$0.000611 = \sqrt{\frac{0.3 \times 50}{630 \times 386.4}}\left[\text{Sin}^{-1}\left(\frac{630x_{to} + 711}{1348}\right) - \text{Sin}^{-1}\left(\frac{711}{1348}\right)\right]$$

where

ϵ_b = 0.3, efficiency of buffer spring system

$$F_{oe} = F_{ob} - \left(\frac{\epsilon_b}{\epsilon}\right)F_a = 727 - \left(\frac{0.3}{0.5}\right)27 = 711 \text{ lb}$$

$$F_{me} = K_b x_{to} + F_{oe} = 630\,x_{to} + 711 \text{ lb}$$

$$\epsilon_b K_b M_t v_{ot}^2 = 2\epsilon_b K_b (E_t - \Delta E_t) = 0.3 \times 630 \times 6948 = 1{,}313{,}172 \text{ lb}^2$$

$$Z = \sqrt{F_{oe}^2 + \epsilon_b K_b M_t v_{ot}^2} = \sqrt{1{,}818{,}693} = 1348 \text{ lb.}$$

Continuing

$$0.000611 = 0.00785 \left[\text{Sin-'} \left(\frac{630x_{to} + 711}{1348} \right) - \text{Sin}^{-1} \left(\frac{711}{1348} \right) \right]$$

$$0.0778 = \text{Sin-'} (0.4674x_{to} + 0.5274) - \text{Sin-'}\ 0.5274$$

$$\text{Sin-'} (0.4674x_{to} + 0.5274) = 31'' \ 50' + 4^\circ \ 28' = 36'' \ 18'$$

$$0.4674x_{to} = 0.5920 - 0.5274 = 0.0646$$

$$x_{to} = 0.138 \text{ in.}$$

The barrel has 0.862 in. to go to complete its buffer stroke. The time needed to tranverse this distance is obtained from **Eqs.** 2–60c and 2–61c. Calculate the constants in **Eq.** 2–60c.

$$4\epsilon M_b = 4\left(0.5\right)\frac{10}{386.4} = 0.0518 \text{ lb-sec}^2/\text{in.}$$

$$\frac{1}{2\epsilon M_b} = \frac{386.4}{2 \times 0.5 \times 10} = 38.64 \text{ in./(lb-sec}^2)$$

$$\frac{K}{4\epsilon M_b} = \frac{5.8 \times 386.4}{4 \times 0.5 \times 10} = 112/\text{sec}^2$$

Substitute these values into Eq. 2–60c.

$$\text{Ax} = \text{B} \left[v_{(n-1)} \text{ At} - 38.64\, F_{(n-1)} \text{ At'} + 112 \Delta t^2 \Delta x_t \right] \qquad (2\text{–}68a)$$

where

$$B = \frac{0.0518}{0.0518 + 5.8\ \Delta t^2}$$

Calculate the constants in **Eq.** 2–61c.

$$4\epsilon\epsilon_b M_t = \frac{4 \times 0.5 \times 0.3 \times 50}{386.4} = 0.0776 \text{ lb-sec}^2/\text{in.}$$

$$\epsilon K_b + \epsilon_b K = 0.5 \times 630 + 0.3 \times 5.8 = 316.7 \text{ lb/in.}$$

$$\frac{1}{2\epsilon_b M_t} = \frac{386.4}{2 \times 0.3 \times 50} = 12.88 \text{ in./(lb-sec}^2)$$

$$\frac{1}{2\epsilon M_t} = \frac{386.4}{2 \times 0.5 \times 50} = 7.73 \text{ in./(lb-sec}^2)$$

$$\frac{K}{4\epsilon M_t} - \frac{5.8 \times 386.4}{4 \times 0.5 \times 50} = 22.41/\text{sec}^2$$

Substitute in **Eq.** 2–61c.

$$\Delta x_t = \text{A} \left[v_{t(n-1)} \text{ At} - 12.88 F_{b(n-1)} \Delta t^2 + 7.73 F_{(n-1)} \Delta t^2 + 22.41 \Delta t^2 \Delta x \right] \qquad (2\text{–}68b)$$

$$\text{where A} = \frac{0.0776}{0.0776 + 316.7 \Delta t^2}$$

Solve for the various parameters and then for At.

$$\Delta x_t = L_b - x_{to} = 1.00 - 0.138 = 0.862 \text{ in.}$$

The driving spring force is estimated as the average during this period

$$F_{(n-1)} = 27 \text{ lb (assumed constant)}$$

$$F_{b(n-1)} = F_{ob} + K_b x_{to} = 727 + 630 \times 0.138 = 814 \text{ lb}$$

$$\Delta E'_t = \frac{1}{2\epsilon_b}\left[F_{ob} + F_{b(n-1)} \right] x_{to} = 354 \text{ in.-lb}$$

$$E_{t(n-1)} = E_t - \Delta E_t - \Delta E'_t = 3572 - 98 - 354 = 3120 \text{ in.-lb}$$

$$\text{Ax} = 1.928 \text{ (assumed and then verified below)}$$

$$v_{t(n-1)} = \sqrt{\frac{2E_{t(n-1)}}{M_t}} = \sqrt{\frac{6240 \times 386.4}{50}} = \sqrt{48223} = 219.6 \text{ in./sec}$$

$$0.862 = \left(\frac{0.0776}{0.0776 + 316.7\,\Delta t^2} \right) (219.6 \text{ At} - 10484\,\Delta t^2 + 209\,\Delta t^2 + 43\Delta t^2)$$

$$0.862 + 3518\,\Delta t^2 = 219.6\,\Delta t - 10232\,\Delta t^2$$

$$\Delta t^2 - 15.97 \times 10^{-3} \text{ At} + 62.70 \times 10^{-6} = 0$$

$$\text{At} = 6.96 \times 10^{-3}$$

Since $\Delta t^2 = 48.4 \times 10^{-6}$, the fraction $\dfrac{0.0518}{0.0518 + 5.8\Delta t^2} = 0.9946$

$$\Delta x = 0.9946 \left[v_{(n-1)}\Delta t - 38.64\, F_{(n-1)}\Delta t^2 + 112\Delta t^2 \Delta x_t \right]$$

$$= 0.9946 \left[285 \times 6.96 \times 10^{-3} - (38.64 \times 27 - 112 \times 0.862)\ 48.4 \times \qquad \right]$$

$$\text{Ax} = 0.9946\,(1.984 - 0.046) = 1.928 \text{ in.}$$

The absolute distance traveled by the barrel at this time is 2.0 in., and x_b the distance that the bolt traveled with respect to the barrel is

$$x_b = \Sigma\Delta x - \text{L}, + \text{Ax} = 1.235 - 2.0 + 1.928 = 1.163 \text{ in.}$$

The total time of buffer spring action during recoil is

$$t_{br} = t_{rt} + \text{At} = 0.00061 + 0.00696 = 0.00757 \text{ sec.}$$

The prevailing conditions at the end of the propellant gas period are now computed. The barrel travel is

$$x_t = \Sigma\Delta x_t + x_{to} = 1.0 + 0.138 = 1.138 \text{ in.}$$

buffer at 5.389 in. The bolt travel when driving spring deflection is

$$x_{bo} = \Sigma\Delta x - x_t = 1.235 - 1.138 = 0.097 \text{ in.}$$

where $\Sigma\Delta x$ = 1.235 in., absolute bolt travel at end of propellant gas period (Table 2–2). The average driving spring force for the remaining deflection is

$$F_a = \frac{\epsilon E_b}{L - x_{bo}} = \frac{0.5 \times 1051}{10.0 - 0.097} = 53.1 \text{ lb.}$$

For K = 5.8, the force at 0.10 in. deflection is

$$F'_o = F_a - \frac{1}{2} K(L - x_{bo}) = 53.1 - 28.7 = 24.4 \text{ lb.}$$

The driving spring force when the bolt is fully retracted is

$$F_m = F'_o + K(L - x_{bo}) = 24.4 + 57.4 = 81.8 \text{ lb.}$$

The driving spring force at zero bolt travel is

$$F_o = F_m - KL = 81.8 - 58 = 23.8 \text{ lb.}$$

The bolt travel from the time that the propellant gas becomes ineffective until the barrel is fully recoiled is

$$x'_b = x_b - x_{bo} = 1.163 - 0.097 = 1.066 \text{ in.}$$

The driving spring force F is

$$F = F'_o + K x'_b = 24.4 + 5.8 \times 1.066 = 30.6 \text{ lb.}$$

The absorbed energy expressed as the differential energy ΔE is

$$\Delta E = \left[\frac{(F'_o + F)}{2\epsilon}\right] x'_b = \left(\frac{55}{2 \times 0.5}\right) 1.066 = 58.6 \text{ in.-lb.}$$

The energy remaining becomes

$$E = E_b - \Delta E = 1051 - 58.6 = 992.4 \text{ in.-lb.}$$

The bolt velocity at this time becomes

$$v = \sqrt{\frac{2E}{M_b}} = \sqrt{\frac{2 \times 992.4 \times 386.4}{10}}$$

$$= \sqrt{76693} = 276.9 \text{ in./sec.}$$

The time t at full recoil of the barrel or when barrel begins to counterrecoil is

$$t = t_g + \Delta t = 12.96 \text{ msec}$$

where t_g = 0.006 sec, duration of propellant gas period

Δt = 0.00696 sec, time for barrel to complete recoil after t_g.

2–4.4 COMPUTER ROUTINE FOR COUNTER-RECOILING BARREL DYNAMICS

A digital computer routine is programmed in FORTRAN IV language for computing the dynamics of the barrel during counterrecoil. Since time and distance are the most pertinent parameters, the program is generated about these data. During a given differential time, Eqs. 2–60c and 2–64d give the differential travel distance of the bolt while Eqs. 2–63c and 2–65c give the differential travel distance of the barrel. Eq. 2–53a provides the relative travel which is equivalent to the driving spring compression. The differential relative travel between barrel and bolt is computed from Eq. 2–53b.

After substituting the various known constants, Eqs. 2–60c and 2–64d are rewritten as Eqs. 2–68a and 2–70a to define the action of the bolt during recoil and counterrecoil, respectively. The substitution of constants into Eq. 2–63c yields equations for the barrel travel while the bolt is recoiling; Eq. 2–69a when the buffer is active; and Eq. 2–69b when the barrel spring acts alone. Eq. 2–65c, after the substitution of numerical constants, becomes Eq. 2–70b which defines the differential barrel travel if the buffer is still active. When the buffer is inactive, Eq. 2–70c defines the differential barrel travel.

Since the equation for solving ***Ax*** includes an expression that contains ***Ax,,*** and the equation for solving Δx_t includes an expression that contains ***Ax,*** the computer program contains an interative routine that approximates these two differential distances. The approximations for ***Ax*** and ***Ax,*** eventually approach the true values close enough to render any error negligible.

The computed values of the constants in Eq. 2–63c for the barrel counterrecoiling under the influence of the buffer while the bolt is still recoiling are

$$4\epsilon M_t = \frac{4 \times 0.5 \times 50}{386.4} = 0.2588 \text{ lb-sec}^2/\text{in.}$$

$$\epsilon\epsilon_b K_b - \text{K} = 0.5 \times 0.3 \times 630 - 5.8 = 88.7 \text{ lb/in.}$$

$$\frac{\epsilon_b}{2M_t} - \frac{0.3 \times 386.4}{2 \times 50} = 1.159 \text{ in./(lb-sec}^2)$$

$$\frac{1}{2\epsilon M_t} = \frac{386.4}{2 \times 0.5 \times 50} = 7.73 \text{ in./(lb-sec}^2)$$

$$\frac{K}{4\epsilon M_t} = \frac{5.8 \times 386.4}{4 \times 0.5 \times 50} = 22.41 \text{ / sec}^2$$

Substitute these constants in Eq. 2–63c

$$\text{Ax,} = \text{A}\Big[v_{t(n-1)}\Delta t + \Big(1.159F_{b(n-1)} - 7.73F_{(n-1)}\Big)\Delta t^2 - 22.41\,\Delta t^2\,\text{Ax}\Big] \qquad (2\text{–}69\text{a})$$

where

$$A = \frac{0.2558}{0.2558 + 88.7\Delta t^2}$$

At the end of buffer return, only barrel and driving springs remain effective. These design data for the barrel spring are

$K_t = 37$ lb/in., $F_{ot} = 107$ lb, $\epsilon_t = 0.5$

The constants in Eq. 2–63c now become

$$\epsilon\epsilon_t K_t - K = 0.5 \times 0.5 \times 37 - 5.8 = 3.45 \text{ lb/in.}$$

$$\frac{\epsilon_t}{2M_t} = \frac{0.5 \times 386.4}{2 \times 50} = 1.932 \text{ in./(lb-sec}^2).$$

Substitute the revised constants into Eq. 2–63c.

$$\Delta x_t = A\left\{ v_{t(n-1)}\Delta t + \Big[1.932F_{b(n-1)} - 7.73F_{(n-1)}\Big]\Delta t^2 - 22.41\,\Delta t^2\,\text{Ax}\right] \qquad (2\text{–}69\text{b})$$

where

$$A = \frac{0.2558}{0.2558 + 3.45\Delta t^2}$$

The constants of Eq. 2–64d, when both bolt and barrel counterrecoil, are computed to be

$4M_b = 0.1035$ lb-sec²/in.; $\epsilon K = 2.9$ lb/in.

$\epsilon/2M_b = 9.66$ in./(lb-sec²); $\epsilon K/4M_b = 28/\text{sec}^2$

$$B = \frac{0.1035}{0.1035 - 2.9\Delta t^2}$$

Eq. 2–64d now becomes

$$\text{Ax} = B\Big[V_{(n-1)}\Delta t - 9.66F_{(n-1)}\Delta t^2 - 28\Delta t^2\Delta x_t\Big]. \qquad (2\text{–}70\text{a})$$

The constants of Eq. 2–65c also change.

$4M_t = 0.518$ lb-sec²/in. $\epsilon K/4M_t = 5.6/\text{sec}^2$

$\epsilon/2M_t = 1.932$ in./(lb-sec²)

If the buffer is still active, the computed constants are

$$\epsilon K + \epsilon_b K_b = 0.5 \times 5.8 + 0.3 \times 630 = 191.9 \text{ lb/in.}$$

$$\epsilon_b/2M_t = 1.159 \text{ in./(lb-sec}^2\text{)}; \quad A = \frac{0.518}{+191.9\Delta t^2}.$$

After substituting these constants into Eq. 2–65c, the differential barrel travel becomes

$$\Delta x_t = A\left\{ v_{t(n-1)}\,\Delta t + \left[1.159F_{b(n-1)} - 1.932F_{(n-1)} + 5.6\Delta x\right]\Delta t^2\right\}. \quad (2\text{–}70b)$$

With the buffer no longer active, the constants become

$$\epsilon K + \epsilon_t K_t = 0.5 \times 5.8 + 0.5 \times 37 = 21.4 \text{ lb/in.}$$

$$\epsilon_t/2M_t = 1.932 \text{ in./(lb-sec}^2\text{)}; \quad A = \frac{0.518}{+21.4\Delta t^2}$$

After being assigned these constants, Eq. 2–65c becomes

$$\Delta x_t = A\left\{ v_{t(n-1)}\,\Delta t + \left[1.932F_{b(n-1)} - 1.932F_{(n-1)} + 5.6\Delta x\right]\Delta t^2\right\}. \quad (2\text{–}70c)$$

Although functions of the spring forces appear in the equations for computing the distance traveled by bolt and barrel, the spring forces must be computed for each increment of time since those values are projected into the next increment. At the end of each increment, the spring forces on bolt and barrel are computed, respectively, from Eqs. 2–55 and 2–56. The driving spring force is

$$F = F_{(n-1)} + 5.8\Delta x_b. \quad (2\text{–}71a)$$

While the buffer is operating, the counterrecoil force on the barrel becomes

$$F_b = F_{b(n-1)} - 630\Delta x_t. \quad (2\text{–}71b)$$

When the barrel spring operates alone, this force is

$$F_b = F_{b(n-1)} - 37\Delta x_t. \quad (2\text{–}71c)$$

After the spring forces are computed, the respective energies and velocities of the bolt and barrel are computed from formulas based on Eqs. 2–66a, 2–66b, and 2–66d.

For convenience, the computer program is divided into four periods: (1) during buffer action, (2) during bolt recoil, (3) after buffer action, and (4) during bolt counterrecoil. Bolt action occurs simultaneously with barrel action (buffer action being a part of barrel action), but usually continues after the barrel is fully counterrecoiled. When the bolt is fully recoiled at t = 50.63 msec (Table 2–5), it immediately starts counterrecoiling with the barrel and these two now counterrecoil as one mass. The velocity at this instant is computed from the law of the conservation of momentum.

$$M_t v_t = (M_t + M_b)v_{t(n-1)} \quad (2\text{–}71d)$$

Table 2–3 lists the variables and corresponding **FORTRAN** code in alphabetical order. Table 2–4 lists the input and Table 2–5 lists the output or results of the computer. The program is found in Appendix A–1 and its flow chart in Appendix A–2.

After the barrel has fully counterrecoiled, all remaining activity is confined to the bolt. It still has to negotiate its complete counterrecoil stroke. The time elapsed for the bolt to complete the remaining counterrecoil stroke is computed from Eq. 2–26.

$$t_{crb} = \sqrt{\frac{M_b}{\epsilon K}}\left(\mathrm{Sin}^{-1}\frac{-F_o}{\sqrt{F^2 + \left(\frac{K}{\epsilon}\right)M_b v^2}} - \mathrm{Sin}^{-1}\frac{-F}{\sqrt{F^2 + \left(\frac{K}{\epsilon}\right)M_t v^2}}\right) \quad (2\text{–}72)$$

$$t_{crb} = 0.0945\left[\mathrm{Sin}^{-1}\left(\longrightarrow\right) - \mathrm{Sin}^{-1}\left(\frac{-81.77}{88.38}\right)\right]$$

$$= 0.0945\ [\mathrm{Sin}^{-1}\ (-)0.2693 - \mathrm{Sin}^{-1}\ (-)0.92521$$

$$= 0.0945\ (344.37 - 292.3)/57.3 = 0.0859 \text{ sec}$$

TABLE 2–3. SYMBOL-CODE CORRELATION FOR DELAYED BLOWBACK PROGRAM

Symbol	Code	Symbol	Code
E	E	n	I
ΔE	DE	t	T
E_t	ET	Δt	DT
ΔE_t	DET	v	V
F	F	v_t	VT
F_b	FB	w_b	WB
F_o	FO	w_{bbl}	WBBL
g	G	Δx	DX
K	SK	x_b	XB
K_b	SKB	Δx_b	DXB
K_t	SKT	x_t	XT
M_b	EMB	Δx_t	DXT
M_t	EMT	ϵ	EPS
$M_b v$	BMV	ϵ_b	EPSB
$M_t v_t$	TMV	ϵ_t	EPST

where $M_b = \dfrac{10}{386.4}$ lb-sec²/in., mass of bolt

K = 5.8 lb/in., driving spring rate

F_o = 23.8 lb, spring force when bolt is closed

F = 81.77 lb, last driving spring force in Table 2–5

$M_b v^2 = 2E = 97$ in.-lb, (E is last value of bolt energy in Table 2–5)

ϵ = 0.5, efficiency of driving spring

Time elapsed from the complete cycle of barrel action including free recoil, buffing, and counterrecoil is

t_{tc} = 54.73 msec, elapsed time of barrel cycle (Table 2–5)

$$t_c = t_{tc} + t_{crb} = 0.0547 + 0.0859 = 0.1406 \text{ sec.}$$

The firing rate is

$$f_r = \frac{60}{t_c} = \frac{60}{0.1406} = 426 \text{ rounds/min.}$$

The velocity of the bolt just as counterrecoil is completed is

$$v_{crb} = \sqrt{v_o^2 + \frac{2E}{M_b}} = \sqrt{v_o^2 + \frac{\epsilon(F_o + F)x_b}{M_b}}$$

$$= \sqrt{3745 + \frac{0.5 \times 105.57 \times 386.4 \times 9.995}{10}}$$

$$= 155.3 \text{ in./sec.}$$

TABLE 2–4. INPUT FOR DELAYED BLOWBACK PROGRAM

Code	Input	Code	Input
AI	0.2558	WB	10.0
A2	88.7	WBBL	50.0
A3	3.45	FO	23.8
A4	0.518	FST	107.0
A5	191.9	FKCR	9.66
A6	21.4	DKXTCR	28.0
B1	0.0518	DKXCR	5.603
B2	5.8	F(1)	30.6
B3	0.1035	FB(1)	1357.0
B4	2.9	V(1)	276.9
EPS	0.5	VT(1)	0.0
EPSB	0.3	XB(1)	1.163
EPST	0.5	XT(1)	0.0
FK	38.64	T(1)	12.96
DXKT	112.0	DX(1)	0.0
BUFK	1.159	DXB(1)	1.066
BBLK	1.932	DXT(1)	0.0
DKX	22.41	DE(1)	0.0
FBK	7.73	DET(1)	0.0
SK	5.8	E(1)	992.4
SKB	630.0	ET(1)	0.0
SKT	37.0	G	386.4

2–4.5 SPRINGS

The driving, barrel, and buffer springs have been assigned characteristics in the dynamic analysis to meet the firing cycle requirements. The analyses which follow of the three springs determine their remaining characteristics that are congruous with the operational and strength requirements.

2–4.5.1 Driving Spring

Known Data:

K = 5.8 lb/in., spring constant

F_o = 23.8 lb, spring force with bolt closed

F_m = 81.8 lb, spring force at full recoil

L = 10.0 in., bolt travel

$T_c = t_{rb}$ = 0.0474 sec, compression time of spring

$v_i = v_f$ = 285 in./sec, bolt velocity of free recoil, impact velocity

The compression time of the spring is measured from the time (3.252 msec, Table 2–2) that the bolt is unlocked until it has fully recoiled (t = 50.6 msec, when x_b = 10.0 in., Table 2–5).

Since $v_i < 25$ fps, select $\frac{T_c}{T} = 3.8$, or

$$T = \frac{0.0474}{3.8} = 0.0125 \text{ sec.}$$

TABLE 2–5. COUNTERRECOIL DYNAMICS OF DELAYED BLOWBACK GUN

INCREMENT I	RELATIVE DELTA TRAVEL INCH	DELTA BOLT TRAVEL INCH	DELTA BARREL TRAVEL INCH	TOTAL BOLT TRAVEL INCH	TOTAL BARREL TRAVEL INCH	DRIVING SPRING FORCE POUND	BARREL SPRING FORCE POUND
1	.000	1.066	.000	1.163	.000	30.60	1357.0
2	.549	.554	.005	1.717	.005	33.76	1353.7
3	.538	.555	.017	2.272	.022	36.98	1343.2
4	.527	.555	.028	2.827	.050	40.20	1325.2
5	.514	.554	.040	3.381	.091	43.41	1299.9
6	.499	.551	.052	3.932	.142	46.61	1267.2
7	.483	.546	.063	4.478	.206	49.77	1227.4
8	.466	.540	.074	5.018	.280	52.91	1180.7
9	.447	.532	.085	5.550	.365	55.99	1127.2
10	.426	.521	.095	6.072	.460	59.02	1067.4
11	.404	.509	.105	6.580	.564	61.97	1001.5
12	.330	.494	.114	7.074	.678	64.83	929.9
13	.354	.476	.122	7.550	.800	67.59	853.0
14	.326	.455	.130	8.005	.930	70.23	771.3
15	.155	.225	.070	8.230	1.000	71.53	107.0
16	.278	.416	.138	8.646	1.138	73.95	101.9
17	.244	.383	.139	9.029	1.277	76.16	96.8
18	.206	.346	.140	9.375	1.417	78.17	91.6
19	.163	.305	.142	9.680	1.559	79.94	86.3
20	.114	.257	.143	9.937	1.702	81.43	81.0
21	.017	.064	.047	10.000	1.749	81.80	79.3
22	.000	.000	.000	10.000	1.749	81.80	79.3
23	-.125	-.003	.122	9.997	1.871	81.78	74.8
24	-.125	-.002	.123	9.995	1.994	81.77	70.2
25	-.006	.000	.006	9.995	2.000	81.77	70.0

INCREMENT I	TIME MSEC	DELTA BOLT ENERGY IN-LB	DELTA BARREL ENERGY IN-LB	BOLT ENERGY IN-LB	BARREL ENERGY IN-LB	BOLT VELOCITY IN/SEC	BARREL VELOCITY IN/SEC
1	12.96	.0	.0	992.4	.0	276.9	.0
2	14.96	35.7	2.2	956.7	2.2	271.9	5.8
3	16.96	39.3	6.8	917.5	8.9	266.3	11.7
4	18.96	42.8	11.4	874.6	20.3	260.0	17.7
3	20.96	46.3	15.8	828.3	36.1	253.0	23.6
6	22.96	49.6	20.0	778.7	56.1	245.3	29.4
7	24.96	52.7	23.6	726.1	79.7	236.9	35.1
8	26.96	55.5	26.A	670.6	106.5	227.7	40.6
9	28.96	57.9	29.4	612.7	135.9	217.6	45.8
10	30.96	60.0	31.3	552.7	167.2	206.7	50.d
11	32.96	61.5	32.5	491.2	199.6	194.8	55.5
12	34.96	62.6	32.9	428.6	232.6	182.0	60.0
15	36.96	63.0	32.6	365.6	265.2	168.1	64.0
14	38.96	62.8	31.6	302.8	296.8	153.0	67.7
15	39.99	31.9	15.7	271.0	312.5	144.7	69.5
16	41.99	60.5	7.2	210.5	319.7	127.5	70.3
17	43.99	57.4	6.9	153.0	326.6	108.8	71.1
18	45.99	53.4	6.6	99.6	333.2	87.7	71.8
19	47.99	46.3	6.3	51.4	339.6	63.0	72.4
20	49.99	41.4	6.0	9.9	345.5	27.7	73.1
21	50.63	10.4	1.9	.0	347.4	.0	73.3
22	50.63	48.3	106.2	48.3	241.3	-61.1	61.1
23	52.63	.1	4.7	48.4	246.0	-61.1	61.7
24	54.63	.1	4.5	48.5	250.4	-61.2	62.2
25	54.73	.0	.2	48.5	250.6	-61.2	62.2

Select a mean coil diameter of 1.0 in. Then according to Eq. 2–42, the wire diameter is

$$d = 0.27\sqrt[3]{DKT} = 0.27\sqrt[3]{0.0725} = 0.113 \text{ in.}$$

From Eq. 2–41

$$N = \frac{Gd^4}{8KD^3} = \frac{11.5 \times 10^6 \times 1.63 \times 10^{-4}}{8 \times 5.8 \times 1.0} = 40.5 \text{ coils.}$$

Static shear stress is

$$\tau = \frac{8F_mD}{\pi d^3} = \frac{8 \times 81.8 \times 1.0 \times 10^3}{3.14 \times 1.442} = 144{,}500 \text{ lb/in.}^2$$

Dynamic shear stress is

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right] = \left(144{,}500\right)\frac{4.0}{3.8}$$

$$= 152{,}100 \text{ lb/in.}^2$$

2–4.5.2 Barrel Spring

Known Data:

K_t = 37 lb/in., spring constant

F_{ot} = 70 lb, spring force with barrel in battery

F_{mt} = **144** lb, spring force with barrel fully recoiled

L_r = 2.0 in., barrel travel

T_c = t_r = 0.013 sec, compression time of spring

v_i = v_f = 232.3 in./sec, barrel velocity of free recoil, impact velocity

The compression time of the spring includes the time of free recoil and that for the rest of the recoil distance.

Select $\frac{T_c}{T} = 3.8$, or

$$T = \frac{T_c}{3.8} = \frac{0.013}{3.8} = 0.0034 \text{ sec.}$$

Select a mean coil diameter of 1.0 in. According to Eq. 2–42, the wire diameter is

$$d = 0.27\sqrt[3]{DK_tT} = 0.27\sqrt[3]{0.1258} = 0.136 \text{ in.}$$

From Eq. 2–41

$$N = \frac{Gd^4}{8K_tD^3} = \frac{11.5 \times 10^6 \times 3.42 \times 10^{-4}}{8 \times 37 \times 1.0} = 13 \text{ coils.}$$

The static shear stress is

$$\tau = \frac{8F_{mt}D}{\pi d^3} = \frac{8 \times 144 \times 1.0}{3.14 \times 2.52 \times 10^{-3}} = 145{,}500 \text{ lb/in.}^2$$

The dynamic shear stress is

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right] = \left(144{,}500\right)\frac{4.0}{3.8}$$

$$= 153{,}000 \text{ lb/in.}^2$$

2–4.5.3 Buffer Spring

Known Data:

F_{obs} = 620 lb, spring force when first contacted

F_{mbs} = 1213 lb, spring force at end of buffer stroke

K_{bs} = 593 lb/in., spring constant

L = 1.0 in., length of buffer stroke

T_c = t_{br} = 0.00757 sec, compression time of spring (see p. 2–32)

v_i = 231.7 in./sec, impact velocity of buffer (see p. 2–29)

Select $\frac{T_c}{T} = 3.8$, or

$$T = \frac{T_c}{3.8} = \frac{0.00757}{3.8} = 0.00199 \text{ sec.}$$

Select a mean coil diameter of 1.75 in. The wire diameter, from Eq. 2-42, is

$$d = 0.27\sqrt[3]{DK_{bs}T} = 0.27\sqrt[3]{2.07} = 0.344 \text{ in.}$$

From Eq. 2–41

$$N = \frac{Gd^4}{8K_{bs}D^3} = \frac{11.5 \times 10^6 \times 0.014}{8 \times 593 \times 5.36} = 6.3 \text{ coils.}$$

The static shear stress is

$$\tau = \frac{8F_{mbs}D}{\pi d^3} = \frac{8 \times 1213 \times 1.75}{3.14 \times 0.0407} = 133{,}000 \text{ lb/in.}^2$$

The dynamic stress is

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right] = 133{,}000\left(\frac{4.0}{3.8}\right)$$

$$= 140{,}000 \text{ lb/in.}^2$$

2–5 RETARDED BLOWBACK

The retarded blowback is similar to the simple blowback except that a linkage supplements the massiveness of the bolt as the primary resistance to the early rearward movement of the cartridge case.

2–5.1 SPECIFIC REQUIREMENTS

To develop the same resistance as the inertia of a large mass, a linkage must have a large mechanical disadvantage during the period of high propellant gas pressure and then gradually relax this resistance as the pressure subsides. A linkage showing these features is illustrated in Fig. 2–8. When the force is greatest, the largest component resisting that force is in line with the bolt; thus only a small component is available to accelerate the bolt and linkage. Later, as the link closes, a larger share of the propellant gas force becomes useful for bolt retraction. Although the gas force has degenerated substantially, the accelerating component has grown to the proportions needed for a short firing cycle and, hence, a high rate of fire.

2–5.2 DYNAMICS OF RETARDED BLOWBACK

Fig. 2–9 illustrates graphically the kinematics of a retarded blowback linkage. Point A represents the position of the bolt as it moves linearly on line AC. Point B represents the position of the common joint between links AB and BC as BC rotates about the fixed point C. The equations of dynamic equilibrium are developed from the graphic illustration of Fig. 2–10. The kinematics are found by writing the two equations that define the geometric constraints of the linkage and then differentiating twice; then writing all variables in terms of **x** and its derivatives.

2–5.2.1 Kinematics of the Linkage

The two equations defining the geometric constraint are obtained from the geometry of the linkage, (see Fig. 2–9)

$$AB \cos\phi + BC \cos\theta = AC - x \qquad (2\text{–}73a)$$

$$AB \sin\phi - BC \sin\theta = 0 \qquad (2\text{–}73b)$$

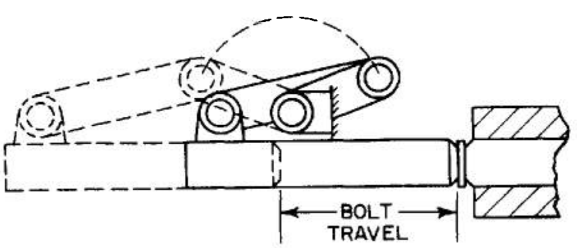
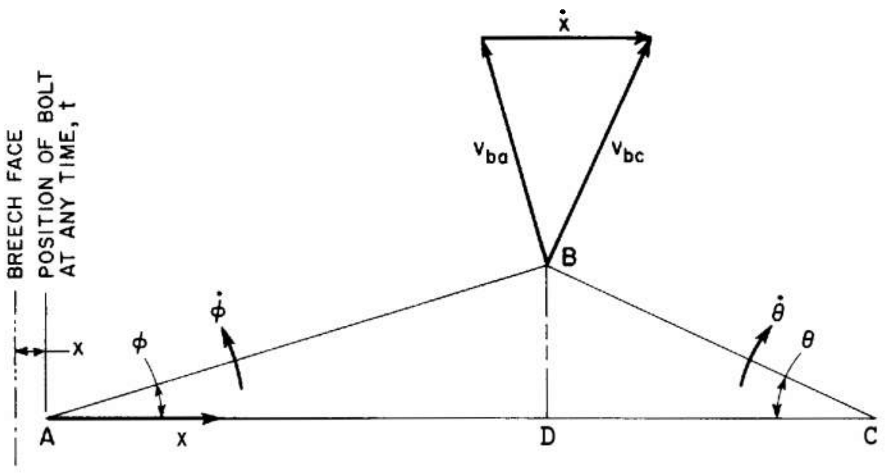
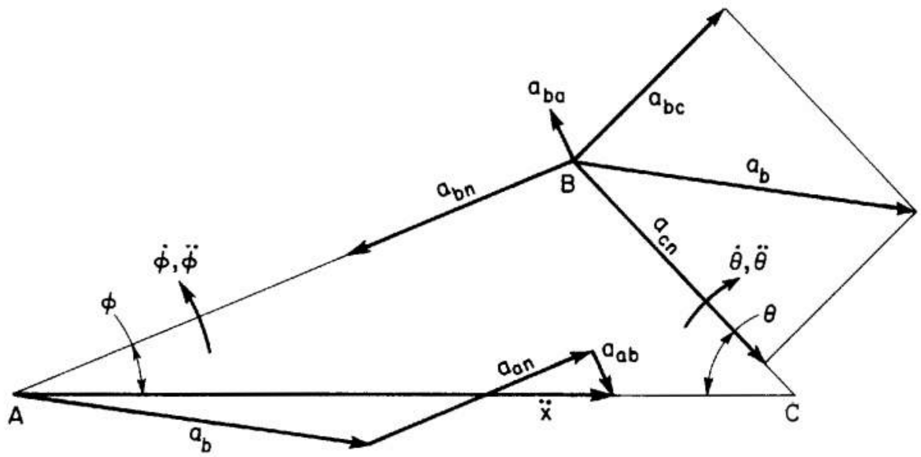


Figure 2–8. Schematic of Retarded Blowback Linkage



(A) VELOCITY DIAGRAM



(B) ACCELERATION DIAGRAM

Figure 2–9. Kinematics of Retarded Blowback Linkage

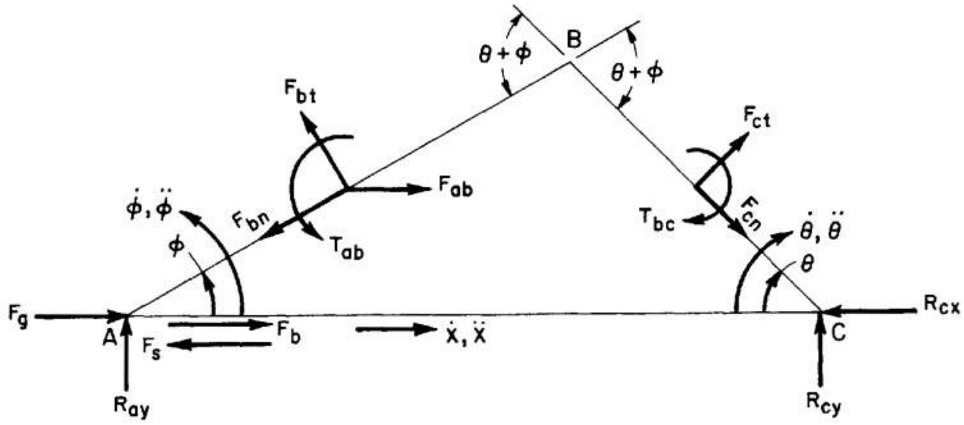


Figure 2–10. Dynamics of Bolt and Linkage

where x is the distance between the breech face and the bolt at any given time t. Differentiate the above equations twice with respect to t.

$$AB\ (\sin\phi)\,\dot\phi \ \text{t}\ BC\ (\sin \text{e})\,\dot\theta = \dot x \qquad (2\text{–}74\text{a})$$

$$AB\ [(\sin\phi)\,\ddot\phi + (\cos\phi)\,\dot\phi^2] + BC\ [(\sin \text{e})\,\ddot\theta \ \text{t}\ (\cos\theta)\,\dot\theta^2] = \ddot x \qquad (2\text{–}74\text{b})$$

$$AB\ (\cos\phi)\,\dot\phi - BC\ (\cos\theta)\,\dot\theta = 0 \qquad (2\text{–}75\text{a})$$

$$AB\ [(\cos\phi)\,\ddot\phi - (\sin\phi)\,\dot\phi^2] - BC\ [(\cos\theta)\,\ddot\theta - (\text{sine})\,\dot\theta^2] = 0 \qquad (2\text{–}75\text{b})$$

Multiply Eq. 2–74b by $\cos\phi$ and Eq. 2–75b by $\sin\phi$ and subtract.

$$AB\ \dot\phi^2 \ \text{t}\ BC\ [(\cos\phi\ \sin \text{e} + \cos \text{e}\ \sin\phi)\,\ddot\theta + (\cos\phi\ \cos \text{e} - \sin\phi\ \sin \text{e})\,\dot\theta^2] = \text{x}\ \cos\phi \qquad (2\text{–}76\text{a})$$

or

$$AB\ \dot\phi^2 \ \text{t}\ BC\ [\ddot\theta\ \sin(\theta+\phi) + \dot\theta^2\ \cos(\theta+\phi)] = \ddot x\ \cos\phi \qquad (2\text{–}76\text{b})$$

Now multiply Eqs. 2–74b and 2–75b by $\cos\theta$ and sin 8, respectively, and add.

$$AB\ [(\cos \text{e}\ \sin\phi + \cos\phi\ \sin \text{e})\,\ddot\phi + (\cos \text{e}\ \cos\phi - \sin\phi\ \sin\theta)\,\dot\phi^2] \ \text{t}\ BC\,\dot\theta^2 = \ddot x\ \cos\theta \qquad (2\text{–}77\text{a})$$

or

$$AB\,[\ddot\phi\ \sin(\theta+\phi) + \dot\phi^2\ \cos(\text{e}+\phi)] + BC\ \dot\theta^2 = X\ \cos\theta \qquad (2\text{–}77\text{b})$$

Multiply Eq. 2–74a by $\cos\theta$ and Eq. 2–75a by $\sin\theta$ and add.

$$AB\ \dot\phi\,(\sin\phi\ \cos\theta + \cos\phi\ \sin\theta) = \dot x\ \cos\theta \qquad (2\text{–}78\text{a})$$

or

$$\text{AB}\ \dot\phi\ \sin(\theta \ \text{t}\ \phi) = \dot x\ \cos\theta \qquad (2\text{–}78\text{b})$$

Multiply the same equations by $\cos\phi$ and $\sin\phi$, respectively, and subtract.

$$\text{BC}\ \dot\theta\,(\sin\theta\ \cos\phi \ \text{t}\ \cos\theta\ \sin\phi) = \dot x\ \cos\phi \qquad (2\text{–}79\text{a})$$

$$BC\ \dot\theta\ \sin(\text{e} \ \text{t}\ \phi) = \dot x\ \cos\phi \qquad (2\text{–}79\text{b})$$

Solve for $\dot{\phi}$ and $\dot{\theta}$

$$\dot{\phi} = \frac{\dot{x}\cos\theta}{AB\sin(\theta+\phi)} \tag{2-80}$$

$$\dot{\theta} = \frac{x\cos\phi}{BC\sin(\theta+\phi)}\,. \tag{2-81}$$

Solve **Eqs.** 2–76b and 2–77b for $\ddot{\theta}$ and $\ddot{\phi}$, respectively

$$\ddot{\theta} = \frac{\ddot{x}\cos\phi - AB\,\dot{\phi}^2}{BC\sin(\theta+\phi)} - \frac{\dot{\theta}^2\cos(\theta+\phi)}{\sin(\theta+\phi)} \tag{2-82}$$

$$\ddot{\phi} = \frac{\ddot{x}\cos\theta - BC\,\dot{\theta}^2}{AB\sin(\theta+\phi)} - \frac{\dot{\phi}^2\cos(\theta+\phi)}{\sin(\theta+\phi)}\,. \tag{2-83}$$

2–5.2.2 Equations of Dynamic Equilibrium

Fig. 2–10 shows the applied and inertial forces of the bolt and linkage. The inertial forces are functions of the kinematics of Fig. 2–9.

Nomenclature of symbols in Figs. 2–9 and 2–10, and in the dynamic analysis follow*.

a_{ab} = acceleration of A with respect to B

a_{an} = normal acceleration of A with respect to B

a_b = acceleration of B

a_{bn} = normal acceleration of B with respect to A

a_{ba} = tangential acceleration of B with respect to A

a_{bc} = tangential acceleration of B with respect to C

a_{cn} = normal acceleration of B with respect to C

AB = length of link AB

BC = length of link BC

F_a = applied force on recoiling parts

F_{ab} = linear inertial force of link AB

F_b = bolt inertial force

F_{bn} = normal force of link AB

F_{bt} = tangential inertial force of link AB

F_{cn} = normal force of link BC

F_{ct} = tangential inertial force of link BC

F_g = propellant gas force

F_s = driving spring force

F_{so} = initial spring force

I_{ab} = mass moment of inertia of link AB

I_{bc} = mass moment of inertia of link BC

K_s = spring constant

M_{ab} = mass of link AB

M_b = mass of bolt

M_r = mass of recoiling parts

R_r = vertical reaction at A

R_{cx} = horizontal reaction at C

R_{cy} = vertical reaction at C

T_{ab} = inertial torque of link AB

T_{bc} = inertial torque of link BC

v_{ba} = velocity of B with respect to A

v_{bc} = velocity of B with respect to C

x = velocity of bolt at A

$\ddot{x}$ = acceleration of bolt at A

ϵ = efficiency of the spring system

E = $1/\epsilon$ during recoil; $E = \epsilon$ during counterrecoil

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***Since these symbols are unique for this par., they are not repeated in the general List of Symbols.**

θ = angle of BC with horizontal

$\dot{\theta}$ = angular velocity of BC, shown positive

$\ddot{\theta}$ = angular acceleration of BC, shown positive

ϕ = angle of AB with horizontal

$\dot{\phi}$ = angular velocity of AB, shown positive

$\ddot{\phi}$ = angular acceleration of AB, shown positive

To achieve equilibrium in the dynamic system, the applied forces and reactions of Fig. 2–10 are equated to the inertial forces. The reactions of the linkage ABC are computed by balancing the moments and forces of the complete system or of any individual link. The inertia forces and moments of each component are expressed in terms of the respective accelerations.

$$F_{ab} = M_{ab}\ddot{x} \tag{2–84a}$$

$$F_b = M_b\ddot{x} \tag{2–84b}$$

$$F_{bn} = M_{ab}\left(\frac{AB}{2}\right)\dot{\phi}^2 \tag{2–84c}$$

$$F_{bt} = M_{ab}\left(\frac{AB}{2}\right)\ddot{\phi} \tag{2–84d}$$

$$F_{cn} = M_{bc}\left(\frac{BC}{2}\right)\dot{\theta}^2 \tag{2–84e}$$

$$F_{ct} = M_{bc}\left(\frac{BC}{2}\right)\ddot{\theta} \tag{2–84f}$$

$$T_{ab} = I_{ab}\ddot{\phi} \tag{2–84g}$$

$$T_{bc} = I_{bc}\ddot{\phi} \tag{2–84h}$$

R_{cy}, the vertical reaction at C (Fig. 2–10) is found by computing the moments about A and dividing by length AC.

$$R_{cy} = \left\{ T_{ab} - T_{bc} - F_{ab}\left(\frac{AB}{2}\right)\sin\phi + \left(\frac{AB}{2}\right)F_{bt} - F_{cn}\,AB\sin(\theta + \phi) + F_{ct}\left[AB\cos(\theta + \phi) + \frac{BC}{2}\right]\right\} / AC \tag{2–85}$$

R_{cx}, the horizontal reaction at C, is found by isolating link BC and equating the applied moment to the inertial moment.

$$R_{cx}\,BC\sin\theta - R_{cy}\,BC\cos\theta = T_{bc} - F_{ct}\left(\frac{BC}{2}\right) \tag{2–86}$$

$$R_{cx} = R_{cy}\frac{\cos\theta}{\sin\theta} + \frac{T_{bc}}{BC\sin\theta} - \frac{F_{ct}}{2\sin\theta} \tag{2–87}$$

The general equation for determining the dynamics of the system consists of the applied horizontal forces and reactions, and the horizontal components of the inertial forces. The inertial moments and vertical force components are not directly involved although they are needed to establish the general equation.

$$F_g - \Sigma F_s - R_{cx} = M_r\ddot{x} - F_{bn}\cos\phi - F_{bt}\sin\phi + F_{cn}\cos\theta + F_{cn}\sin\theta \tag{2–88}$$

where

$$F_s = \Sigma(F_{so} + K_s x)$$

$$M_r = M_{ab} + M_b$$

Eq. 2–88 may be solved by numerical integration after the variables $\dot{\phi}, \dot{\theta}, \ddot{\phi}, \ddot{\theta}$ are written in terms of $\dot{x}$ and x.

2–5.2.3 Digital Computer Program for the Dynamic Analysis

A digital computer program is compiled in FORTRAN IV language for the UNIVAC 1107 computer. The various parameters are solved for each one of many small increments of time into which the recoil and counterrecoil periods are divided. The

solution follows the procedure of the Runge-Kutta-Gill Method of numerical integration. The program listing is in Appendix A–3; the corresponding Flow Chart in Appendix A–4.

Because Eq. 2–88 becomes extremely unwieldly when the appropriate expressions are substituted for $\dot{\phi}$, $\dot{\theta}$, $\ddot{\phi}$, $\ddot{\theta}$, simple coefficients are introduced in sequence to represent the cumbersome expressions. The continued substitution eventually leads to an equation of simple terms. The list of coefficients that follow are determined from **Eqs.** 2–80 to 2–83.

$$C1 = \cos\phi/BC \sin(\theta + \phi)$$

$$C2 = -AB/BC \sin(6 \text{ t } \phi)$$

$$C3 = -\cos(6+\phi)/\sin(6+\phi)$$

$$C4 = \cos\theta/AB \sin(6+\phi)$$

$$C5 = -BC/AB \sin(6 \text{ t } \phi)$$

$$C6 = C2 \cdot C4^2 + C3 \cdot C1^2$$

$$C7 = C5 \cdot C1^2 + c3 \cdot C4^2$$

Rewrite **Eqs.** 2–80 to 2–83 by inserting the proper coefficient.

$$\dot{\phi} = C4\dot{x} \qquad (2\text{–}89a)$$

$$\text{e} = C1\dot{x} \qquad (2\text{–}89b)$$

$$\text{e} = C1\ddot{x} + C6\dot{x}^2 \qquad (2\text{–}89c)$$

$$\ddot{\phi} = C4\ddot{x} + C7\dot{x}^2 \qquad (2\text{–}89d)$$

Rewrite **Eq.** 2–85

$$R_{cy} = E3(C4\ddot{x} + C7\dot{x}^2) - E4X$$
$$- E5 \cdot C1^2\dot{x}^2 + E6(C1\ddot{x} + C6\dot{x}^2) \qquad (2\text{–}90a)$$

where

$$E1 = M_{ab} \cdot AB/2$$

$$E2 = M_{bc} \cdot BC/2$$

$$E3 = (I_{ab} + E1 \cdot AB/2)/AC$$

$$E4 = E1 \sin\phi/AC$$

$$E5 = E2 \cdot AB \sin(6+\phi)/AC$$

$$E6 = \left\{ E2[AB \cos(6+\phi) + BC/2] - I_{bc} \right\}/AC$$

Collect terms and assign new coefficients

$$R_{cy} = C8\ddot{x} + C9\dot{x}^2 \qquad (2\text{–}90b)$$

where

$$C8 = E3 \cdot C4 - E4 + E6 \cdot C1$$

$$C9 = E3 \cdot C7 - E5 \cdot C1^2 \text{ t } E6 \cdot C6$$

Rewrite Eq. 2–87

$$R_{cx} = C8\left(\frac{\cos\theta}{\sin\theta}\right)\ddot{x} + C9\left(\frac{\cos\theta}{\sin\theta}\right)$$
$$+ E7(C1x + C6\dot{x}^2)/\sin 8 \qquad (2\text{–}91a)$$

$$R_{cx} = C10\ddot{x} + C12\dot{x}^2 \qquad (2\text{–}91b)$$

where

$$E7 = \frac{I_{bc}}{BC} - \frac{E2}{2}$$

$$C10 = (C8\cos\theta + E7 \cdot C1)/\sin\theta$$

$$C12 = (C9\cos\theta + E7 \cdot C6)/\sin\theta$$

Recall Eq. 2–88, solve for F and insert appropriate coefficients.

$$F_g = M_r x - E1(C4x + C7\dot{x}^2)\sin\phi$$
$$- E1 \cdot C4^2\dot{x}^2 \cos\phi$$
$$+ E2(C1\ddot{x} + C6\dot{x}^2)\sin 8 + E2 \cdot C1\dot{x}^2 \cos\theta$$
$$+ C10\ddot{x} + C12\dot{x}^2 - E(F_o - K_s x) \qquad (2\text{–}92)$$

Collect terms and solve for $\ddot{x}$

$$C11\ddot{x} = F_g + C13\dot{x}^2 + C14 + C15x \qquad (2-93)$$

$$\ddot{x} = (F_g + C13\dot{x}^2 + C15x + C14)/C11 \qquad (2-94)$$

where

$$C11 = M_r - E1 \cdot C4 \sin\phi + E2 \cdot C1 \sin\theta + C10$$

$$C13 = E1 \cdot C7 \sin\phi + C1 \cdot C4^2 \cos\phi$$
$$- E2 \cdot C6 \sin\theta - E2 \cdot C1^2 \cos\theta - C12$$

$$C14 = -EF_{so}$$

$$C15 = -EK_s$$

The computer solves for $\ddot{x}$ and then all the other variables for each increment of time. The program is also arranged for the interpolation of the gas force F_g when the time, and therefore force, for any particular computations fall between two data points selected from the force-time curve of Fig. 2–7.

Initial spring characteristics are usually based on those of a similar gun. After trial computations, the values are altered to be more compatible with specifications. For instance, in the sample problem, the initial values of initial buffer force and spring constant were F_{so} = 200 lb and K_s = 388 lb/in. This resulted in a buffer travel of almost twice the specified distance. After changing the spring constant to K_s = 760 lb/in. and F_{so} = 800 lb, the computed buffer stroke equalled that specified.

Table 2–6 lists the code for each symbol, Table 2–7 lists the input data for the computer program, and Table 2–8 lists the computed dynamics. Four series of computations are made for each increment I and, since there are almost 2000 increments, only the results of the fourth series of every 15th increment is printed. This procedure is followed except at the ends of the recoil and counterrecoil strokes where the results of each terminal increment are printed. By eliminating most of the output from the record, Table 2–8 is held to reasonable size but still contains enough data to show clearly, the trend in the dynamic behavior of the bolt mechanism during the firing cycle.

The final time (at increment I = 1889) of t = 0.067 sec shows a firing rate of

$$f_r = \frac{60}{t} = 895 \text{ rounds/min.}$$

TABLE 2–6. SYMBOL-CODE RELATIONSHIP FOR RETARDED BLOWBACK

Symbol	Code	Symbol	Code
F_a	FA	W_{ab}	WAB
F_g	FG	W_b	WB
F_{so}	FSO	W_{bc}	WBC
g	G	x	X
I_{ab}	EYEB	ϵ	EPS
I_{bc}	EYEC	θ	THETA
K_S	SK	ϕ	PHI
M_{ab}	EMAB	$\sin\theta$	STHETA
M_b	EMB	$\cos\theta$	CTHETA
M_r	EMR	$\sin\phi$	SPHI
t	T	$\cos\phi$	CPHI
Δt	DT	$\sin(\theta + \phi)$	SUMSIN
v	VEL	$\cos(\theta + \phi)$	SUMCOS

The preferred method of increasing this rate is to increase the moment arm of the linkage, i. e., by decreasing the initial value of *AC*. A lower firing rate may be attained by decreasing the moment arm, i. e., by increasing the initial length of *AC*.

2.6 RATING OF BLOWBACK WEAPONS

The simple blowback machine gun, because of its simplicity, outranks all other types with respect to maintenance and relative cost. Moving parts are few, and normal care exercised in manufacture produces a gun whose reliability is considered good, i. e., ordinary malfunctions can be corrected in the field within 30 sec. Take-down, cleaning, lubricating, and reassembly requires little time and practically no tools. Although these attributes are encouraging, the simple blowback has its limitations. It is restricted to small caliber guns, low rates of fire, low muzzle velocities, and, therefore, short range. However, the gun is light enough to be carried by the foot soldier and is accurate enough at short ranges to make it a good antipersonnel weapon.

The delayed blowback machine gun is almost as easily maintained as the simple blowback but its relative cost is higher. It has a low to medium rate of fire and a medium to high muzzle velocity. The delayed blowback is not confined to small calibers. It outranges and has better accuracy than the simple blowback and, because of its greater fire power, is more versatile, being capable of destroying both materiel and personnel. The delayed blowback gun is durable and reliable, seldom becoming inoperative because of breakdown except after long usage, and can quickly be restored to operation after ordinary malfunction.

When compared with simple and delayed blowback guns, the advanced primer ignition and retarded blowback types are relegated to second position. The retarded blowback type, because it depends on a linkage system to control bolt recoil that is extremely sensitive to geometric proportions, does not have the reliability of the delayed type either in theory during design, or in practice during development and usage. The large loads applied to the linkage while in motion adversely affects the gun's durability. From these aspects alone the delayed blowback is preferred over the retarded type.

The advanced primer ignition gun is superior to the simple blowback because of its higher firing rate and lower recoil momentum. However, favorable performance depends on timing that must be precise. A slight delay in primer function, and the gun reverts to a simple blowback without the benefit of a massive bolt and stiffer driving spring to soften the recoil impact. Delayed primer ignition creates the hazard of extracting the cartridge case while subjected to pressures high enough to blow up the case. Although advanced primer ignition guns have been made, one by Becker, the exacting requirements in design and construction of gun and ammunition reduce this type almost to the point of academic interest only.

TABLE 2–7. INPUT DATA FOR RETARDED BLOWBACK

Code	Data	Code	Data
AB	7.0	NHEAD	630
AZ	12.9985	NPO	15
BC	6.0	N9	96
DT	0.000025	SK1	3.8
DTFG	0.0000625	SK2	760.0
DTNEW	0.00026	TCHANG	0.045
EPS	0.50	WAB	0.85
FS1	63.0	WB	8.0
FS2	800.0	WBC	1.5
G	386.4	XLIM	9.0
N	3000	XREC	9.95859
		XBATY	0.010

TABLE 2–8. RETARDED BLOWBACK DYNAMICS

I	TIME SECOND	APPLIEO FORCE POUND	DISTANCE FROM BREECH INCH	VELOCITY IN/SEC	ACCELERATION IN/SEC/SEC
1	.0000250	398.0	.000000	.0	180.5
15	.0003750	18774.0	.000165	1.6	10185.6
30	.0007500	23834.0	.001883	9.2	36242.9
45	.0011250	18663.9	.009210	34.6	109085.5
60	.0015000	12703.0	.032415	96.7	220112.1
75	.0018750	7733.4	.085583	190.0	260303.3
90	.0022500	4922.7	.174304	281.2	224086.7
105	.0026250	3801.8	.294690	350.9	187483.0
120	.0030000	2670.7	.441364	420.8	140860.9
135	.0033750	1794.4	.608063	466.0	100246.5
150	.0037500	1103.0	.788991	497.0	66704.3
165	.0041250	651.6	.979433	517.3	43452.7
180	.0045000	425.1	1.176159	531.1	30363.6
195	.0048750	218.5	1.377183	540.3	19058.6
210	.0052500	107.0	1.580966	546.2	12324.8
225	.0056250	-9.6	1.786483	549.6	5869.6
240	.0060000	-141.1	1.992836	550.6	-900.6
255	.0063750	-142.7	2.199224	550.1	-1938.7
270	.0067500	-144.3	2.405343	549.2	-2753.7
205	.0071250	-145.0	2.611077	548.0	-3410.0
300	.0075000	-147.4	2.816333	546.6	-3950.3
315	.0078750	-149.0	3.021035	545.1	-4404.2
330	.0082500	-150.5	3.225119	543.3	-4792.3
345	.0086250	-152.1	3.428530	541.5	-5129.6
360	.0090000	-153.6	3.631220	539.5	-5427.2
375	.0093750	-155.1	3.833148	537.4	-5693.4
390	.0097500	-156.7	4.034276	535.2	-5934.5
405	.0101250	-158.2	4.234569	533.0	-6155.4
420	.0105000	-159.7	4.433997	530.6	-6360.1
435	.0108750	-161.2	4.632531	520.2	-6551.7
450	.0112500	-162.7	4.830144	525.7	-6732.7
465	.0116250	-164.2	5.026810	523.2	-6905.4
480	.0120000	-165.7	5.222506	520.5	-7071.6
495	.0123750	-167.2	5.417207	517.9	-7232.8
510	.0127500	-168.6	5.610891	515.1	-7390.5
525	.0131250	-170.1	5.803537	512.3	-7546.0
540	.0135000	-171.6	5.995121	509.5	-7700.6
555	.0138750	-173.0	6.185623	506.5	-7855.5
570	.0142500	-174.5	6.375019	503.6	-8012.0
585	.0146250	-175.9	6.563290	500.5	-8171.3
600	.0150000	-177.3	6.750411	497.4	-8334.9
615	.0153750	-178.7	6.936361	494.3	-8504.2
630	.0157500	-180.1	7.121114	491.1	-8681.2

TABLE 2–8. RETARDED BLOWBACK DYNAMICS (Con't.)

I	TIME SECOND	APPLIED FORCE POUND	DISTANCE FROM BREECH INCH	VELOCITY IN/SEC	ACCELERATION IN/SEC/SEC
645	.0161250	-181.5	7.304647	487.8	-8867.6
660	.0165000	-182.9	7.486933	484.4	-9066.0
675	.0168750	-184.3	7.667944	481.0	-9279.0
690	.0172499	-185.6	7.847650	477.4	-9509.9
705	.0176249	-187.0	8.026019	473.8	-9762.8
720	.0179999	-188.3	8.203013	470.1	-10042.5
735	.0103749	-189.7	8.378596	466.3	-10355.1
750	.0187499	-191.0	a. 552722	462.3	-10708.1
765	.0191249	-192.3	8.725342	458.3	-11110.8
700	.0194999	-193.6	8.896399	454.0	-11575.0
795	.0198749	-1893.8	9.065089	439.4	-83221.1
810	.0202499	-2136.3	9.223000	406.3	-92816.8
825	.0206249	-2358.8	9.369479	369.9	-101264.3
840	.0209999	-2559.6	9.500939	330.5	-108497.8
855	.0213749	-2737.2	9.617160	288.7	-114496.0
870	.0217499	-2890.1	9.717299	244.8	-119280.4
685	.0221249	-3017.5	9.800679	199.4	-122954.1
900	.0224999	-3118.5	9.866784	152.8	-125611.6
915	.0228749	-3192.5	9,915237	105.3	-127399.2
930	.0232499	-3239.2	9.945783	57.3	-128449.3
945	.0236249	-3258.3	9.958274	9.0	-128860.8
948	.0236999	-814.7	9.958599	-.2	-32217.9
960	.0239999	-814.1	9.957093	-9.9	-32205.7
975	.0243749	-811.8	9.951135	-22.0	-32157.0
990	.0247499	-807.8	9.940655	-34.0	-32069.7
1005	.0251249	-802.1	9.925667	-46.0	-31941.7
1020	.0254999	-794.7	9.906187	-58.0	-31769.7
1035	.0256749	-785.5	9.882240	-69.8	-31550.1
1050	.0262498	-774.7	9.053856	-81.6	-31278.5
1065	.0266248	-762.2	9.821074	-93.3	-30950.7
1080	.0269998	-748.0	9.783940	-104.8	-30562.1
1095	.0273748	-732.2	9.742508	-116.2	-30108.6
1110	.0277498	-714.7	9.696843	-127.4	-29586.5
1125	.0281248	-695.7	9,647018	-138.4	-28992.3
1140	.0284998	-675.1	9.593116	-149.1	-28323.7
1155	.0288748	-653.0	9.535232	-159.6	-27578.8
1170	.0292498	-629.4	9.473469	-169.8	-26756.7
1185	.0296248	-604.4	9.407945	-179.7	-25857.1
1200	.0299998	-578.0	9.338783	-189.2	-24880.7
1215	.0303748	-550.2	9.266123	-198.3	-23828.9
1230	.0307498	-521.2	9.190113	-207.1	-22703.7
1245	.0311248	-491.0	9.110909	-215.3	-21507.8
1260	.0314998	-459.6	9.028681	-223.2	-20244.4

TABLE 2–8. RETARDED BLOWBACK DYNAMICS (Con't.)

I	TIME SECOND	APPLIED FORCE POUND	DISTANCE FROM BREECH INCH	VELOCITY IN/SEC	ACCELERATION IN/SEC/SEC
1275	.0318748	-48.5	8.944110	-226.5	-2930.1
1290	.0322498	-48.3	8.858986	-227.5	-2867.3
1305	.0326248	-48.2	8.773458	-228.6	-2809.0
1320	.0329998	-48.0	8.687536	-229.7	-2754.8
1335	.0333748	-47.8	8.601226	-230.7	-2704.3
1350	.0337497	-47.7	8.514535	-231.7	-2657.0
1365	.0341247	-47.5	8.427471	-232.7	-2612.7
1380	.0344997	-47.3	8.340039	-233.6	-2571.1
1395	.0348747	-47.2	8.252246	-234.6	-2531.9
1410	.0352497	-47.0	8.164096	-235.5	-2494.8
1425	.0356247	-46.8	8.075596	-236.5	-2459.8
1440	.0359997	-46.7	7.986750	-237.4	-2426.5
1455	.0363747	-46.5	7.897561	-238.3	-2394.7
1470	.0367497	-46.3	7.808036	-239.2	-2364.5
1485	.0371297	-46.2	7.718179	-240.1	-2335.5
1500	.0374997	-46.0	7.627992	-240.9	-2307.8
1515	.0378747	-45.8	7.537482	-241.8	-2281.1
1530	.0582497	-45.6	7.446650	-242.6	-2255.4
1545	.0386246	-45.5	7.355502	-243.5	-2230.5
1560	.0389996	-45.3	7.264040	-244.3	-2206.5
1575	.0393746	-45.1	7.172267	-245.1	-2183.1
1590	.0397496	-45.0	7.080187	-246.0	-2160.4
1605	.0401246	-44.8	6.987804	-246.8	-2138.2
1620	.0404996	-44.6	6.895120	-247.6	-2116.6
1635	.0408746	-44.4	6.802138	-248.4	-2095.4
1650	.0412496	-44.2	6.708861	-249.1	-2074.6
1665	.0416246	-44.1	6.615293	-249.9	-2054.1
1680	.0419996	-43.9	6.521436	-250.7	-2033.9
1695	.0423746	-43.7	6.427293	-251.4	-2014.0
1710	.0427496	-43.5	6.332066	-252.2	-1994.3
1725	.0431246	-43.4	6.238159	-252.9	-1974.7
1740	.0434995	-43.2	6.143175	-253.7	-1955.2
1755	.0438745	-43.0	6.047915	-254.4	-1935.9
1770	.0442495	-42.8	5.952383	-255.1	-1916.5
1785	.0446245	-42.6	5.856582	-255.8	-1897.2
1800	.0449995	-42.4	5.760514	-256.5	-1877.8
1815	.0485245	-40.7	4.844994	-262.8	-1686.5
1830	.0522745	-38.8	3.848134	-268.7	-1427.9
1045	.0560245	-36.9	2.831350	-273.3	-996.2
1860	.0597745	-34.9	1.801193	-275.4	88.0
1875	.0635245	-33.0	.776233	-267.8	6102.3
1889	.0670245	-31.5	.002236	-53.0	367185.9

CHAPTER 3

RECOIL-OPERATED WEAPONS

3–1 GENERAL

Recoil-operated weapons are those weapons that rely on recoil activity to operate the bolt and related parts. The bolt, locked to the barrel during firing, is released during recoil after the chamber pressure has become safe. Action is confined to two general types; long recoil and short recoil.

Long recoil has the barrel and bolt recoiling as a unit for the entire distance (Fig. 3–1). This recoil distance must be greater than the length of the complete round to provide space for loading. At the end of the recoil stroke, the bolt is held while the barrel counterrecoils alone. When sufficient space develops between bolt and breech, the spent case is ejected. Later, as the barrel reaches the in-battery position, the bolt is released to reload the gun.

Short recoil has the barrel and bolt recoiling as a unit for a distance shorter than the length of the complete round (Fig. 3–2). The bolt is unlocked shortly before the barrel negotiates its full stroke. As the barrel stops, the momentum of the bolt carries it farther rearward opening a space – between it and barrel – large enough for extracting the spent case and reloading. The returning bolt, while reloading, may push the barrel into battery or the barrel may counterrecoil independently of the bolt.

3–2 LONG RECOIL DYNAMICS

The dynamics of the long recoil-operated gun are similar to those of the blowback types except that the barrel and bolt units recoil together. Time of recoil may be decreased by delaying energy of recoil absorption until near the end of the recoil stroke, which can be done with a heavy buffer spring operating over a short stroke. The barrel spring should be stiff enough to hold the recoiling parts in battery while the bolt is returning whereas the bolt driving spring should be capable of closing the bolt in minimal time. The stiffer the spring, the less time needed for the return. However, since the converse is not true, some compromise must be arranged to achieve an acceptable firing rate. For initial estimates, the driving spring should have properties that are approximate to those needed to absorb the recoil energy of the bolt. Later adjustments can be made in the properties of all the springs in the system to achieve appropriate time and velocity criteria.

The buffer characteristics should be so arranged that its useful potential energy, when fully compressed, approximately equals that of the barrel spring, yet still is compatible with other design requirements. This arrangement gives the barrel sufficient momentum at the beginning of the counterrecoil stroke for a quick return without inducing excessive impact when stopping the returning barrel.

3–3 SAMPLE PROBLEM – LONG RECOIL MACHINE GUN

3–3.1 SPECIFICATIONS

Gun: 20 mm machine gun

Firing Rate: corresponding to minimum bolt travel

Interior ballistics: Pressure vs Time, Fig. 2–7

A_b = 0.515 in.^{2} bore area

3–3.2 DESIGN DATA

L = 10 in, recoil distance

W_b = 10 lb, weight of bolt unit

W_t = 50 lb, weight of barrel unit

E = 0.5, efficiency of spring system

Table 3–1 has the numerical integration for a recoiling weight of 60 lb. The column A_i represents the area under the pressure-time curve, Fig. 3–1, for each interval of time.

$$F_g \Delta_t = 0.515\ A_i, \text{ lb-sec.}$$

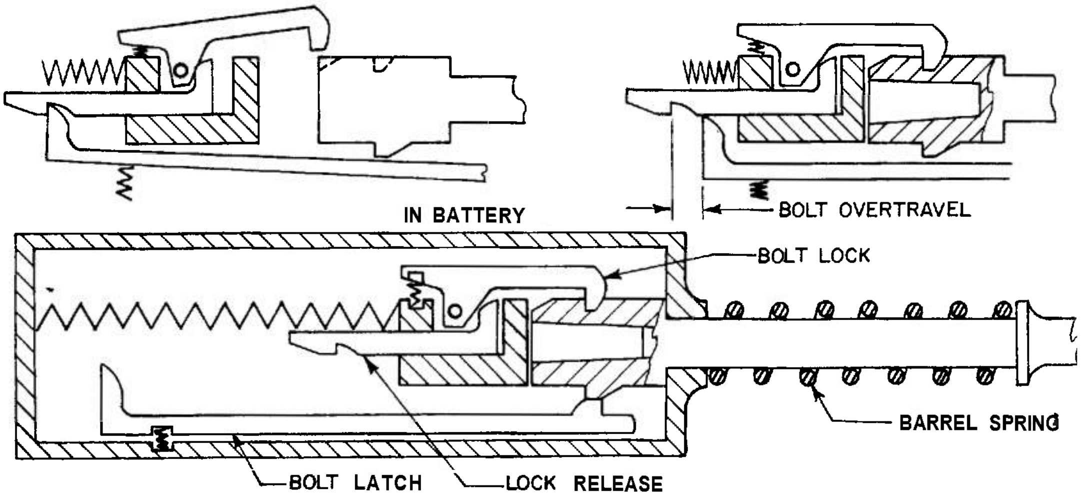


Figure 3–1. Schematic of Long Recoil System

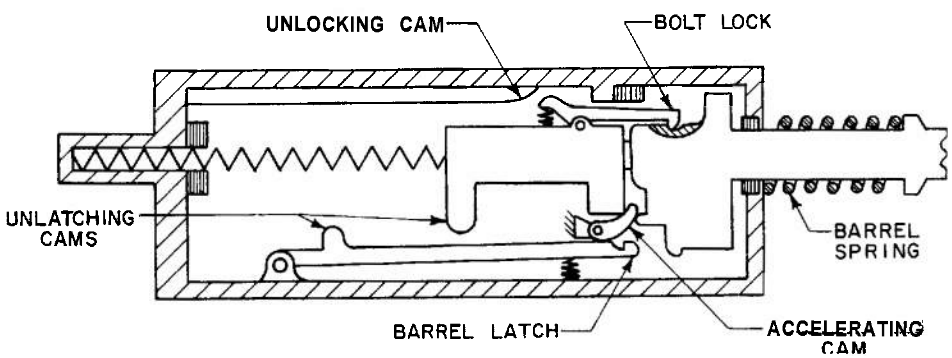


Figure 3–2. Schematic of Short Recoil System

TABLE 3–1. RECOIL TRAVEL OF 20 mm GUN

t, msec	Δt, msec	A_i, lb-sec/in.^{2}	$F_g\Delta t$, lb-sec	Δv, in./sec	v, in./sec	v_a, in./sec	Δx, in.
0.25	0.25	3.44	1.77	11.4	11.4	5.7	0.0014
0.50	0.25	10.15	5.24	33.7	45.1	28.2	0.007 1
0.75	0.25	11.89	6.13	39.5	84.6	64.8	0.0 162
1.00	0.25	11.12	5.74	37.0	121.6	103.1	0.0258
1.25	0.25	9.25	4.76	30.6	152.0	131.9	0.0330
1.50	0.25	7.10	3.76	24.2	176.4	159.3	0.0398
1.75	0.25	5.23	2.69	17.3	193.7	185.0	0.0462
2.00	0.25	3.71	1.91	12.6	206.3	200.0	0.0500
2.25	0.25	2.58	1.34	8.6	214.9	210.6	0.0526
2.50	0.25	1.82	0.94	6.1	221.0	218.0	0.0545
2.75	0.25	1.39	0.72	4.6	225.6	223.3	0.0558
3.00	0.25	1.06	0.55	3.5	229.1	227.4	0.0569
4.00	1.00	2.34	0.97	6.2	235.3	232.2	0.2322
5.00	1.00	1.04	0.40	2.6	237.9	236.6	0.2366
6.00	1.00	0.40	0.09	0.6	238.5	238.2	0.2382

$$Av = F_g\left(\frac{\Delta t}{M_r}\right) = \frac{g}{W_r}\left(F_g\Delta t\right) = \left(\frac{386.4}{60}\right)F_g\Delta t$$
$$= 6.44\ F_g\Delta t \text{ in./sec}$$

where W_r = weight of recoiling parts

$$Ax = v_a\Delta t = \left(\frac{v_{(n-1)}+v}{2}\right)At, \text{ in.}$$

The distance recoiled during the effective propellant gas pressure period

$$x_r = \Sigma\Delta x = 1.15 \text{ in.}$$

The recoil velocity at this time is v = 238.5 in/sec (Table 3–1).

Three springs are in the system (Fig. 3–1). The bolt driving spring and barrel spring work in unison during recoil until the buffer spring is contacted; then all three work as a unit until the barrel and bolt come to a stop whereupon the bolt is latched, permitting the barrel spring and buffer spring to force the barrel to counterrecoil. These two springs function as one until the buffer spring completes its short travel, thereafter the barrel spring alone continues to counterrecoil the barrel. Just as the barrel stops counterrecoiling, the bolt becomes unlatched and the driving spring closes it.

The energy of recoil is

$$E_r = \frac{1}{2}\left(\frac{W_r}{g}\right)v^2 = \frac{1}{2}\left(\frac{60}{386.4}\right)56882$$
$$= 4416 \text{ in.-lb}$$

where

g = 386.4 in./sec^2

W_r = 60 lb, recoiling weight

v = 238.5 in./sec, velocity of recoil

Preliminary estimates of recoil time must be available to determine the spring characteristics. After an approximate recoil time has been established, some of

the data used in early calculations may be altered for greater accuracy. A reasonable approach is achieved by absorbing 75% of the recoil energy before the buffer is reached thereby reducing the recoil velocity by 50% during the same period. Assigning more energy within limits to the buffer will increase the firing rate and conversely, less energy absorbed by the buffer will decrease the firing rate. The energy to be absorbed by the buffer is

$$E_b = 0.25E_r = 0.25 \times 4416 = 1104 \text{ in.-lb}$$

The recoil velocity as the buffer is contacted becomes

$$v_b = \sqrt{\frac{2E_b}{M_r}} = \sqrt{\frac{2208 \times 386.4}{60}} = \sqrt{14220}$$
$$= 119.25 \text{ in./sec}$$

The average force of the system which includes the driving, barrel, and buffer springs is

$$F_{as} = \frac{\epsilon E_b}{L_b} = \frac{0.5 \times 1104}{0.5} = 1104 \text{ lb}$$

where L_b = length of buffer stroke.

For constant acceleration, the buffing time and therefore the compression time of the springs is

$$T_c = t_b = \frac{2L_b}{v_b} = \frac{2 \times 0.5}{119.25} = 0.0084 \text{ sec.}$$

The corresponding surge time is computed to be

$$T = \frac{T_c}{3.8} = \frac{0.0084}{3.8} = 0.0022 \text{ sec.}$$

From Eq. 2–67

$$K_bT = \frac{F_{as} + \frac{1}{2}\left(K_bL_b\right)}{1037}$$

$$1037 \times 0.0022\, K_b = 1104 + 0.25\, K_b$$

$$K_b = \frac{1104}{2.031} = 543.6 \text{ lb/in., combined spring constant}$$

$$F_{mb} = F_{as} + 0.25K_b = 1104 + 136 = 1240 \text{ lb}$$

$$F_{ob} = F_{as} - 0.25\, K_b = 1104 - 136 = 968 \text{ lb.}$$

The new compression time of the springs, Eq. 2–23, becomes

$$T_c = \sqrt{\frac{\epsilon M_r}{K_b}}\ \text{Cos}^{-1} \frac{F_{ob}}{F_{mb}} = 0.0119 \times 0.675$$
$$= 0.0080 \text{ sec.}$$

By repeating the above process, T_c remains at ***0.0080*** sec and K_b changes to 572 lb/in.

$$F_{mb} = F_{as} + \frac{1}{2}\left(K_bL_b\right) = 1104 + 143 = 1247 \text{ lb}$$

$$F_{ob} = F_{as} - \frac{1}{2}\left(K_bL_b\right) = 1104 - 143 = 961 \text{ lb}$$

Assume constant deceleration, then the recoil time from the end of the accelerating period to buffer contact will be

$$t_r = \frac{2L_d}{v + v_b} = \frac{16.7}{238.5 + 119.25} = 0.0467 \text{ sec}$$

where $L_d = L - L_b - x = 10.0 - 0.5 - 1.15 = 8.35$ in.

v = 238.5 in./sec, recoil velocity at end of acceleration

v_b = 119.25 in./sec recoil velocity at start of buffing

The compression time of the springs includes the accelerating time and the buffing time.

$$T_c = t_a + t_r + t_b = \boldsymbol{0.006} + 0.0467 + 0.008$$
$$= 0.0607 \text{ sec.}$$

The corresponding surge time is

$$T = \frac{T_c}{3.8} = \frac{0.0607}{3.8} = 0.0160 \text{ sec.}$$

The average combined force of the driving and barrel springs, based on 75% recoil energy absorption, becomes

$$F_a = \frac{0.75\epsilon E_r}{L_d} = \frac{0.75 \times 0.5 \times 4416}{8.35} = 198.3 \text{ lb.}$$

According to Eq. 2–67b,

$$K_s T = \frac{F_m}{1037} = \frac{F_a + \frac{1}{2}\left(K_s L_d\right)}{1037}$$

$$1037 \times 0.016\ K_s = 198.3 + 4.175\ K_s$$

$$K_s = \frac{198.3}{12.417} = 16.0 \text{ lb/in.}$$

$$F_{ms} = F_a + 4.175 K_s = 198.3 + 66.8 = 265.1 \text{ lb}$$

$$F_{os} = F_a - 4.175 K_s = 198.3 - 66.8 = 131.5 \text{ lb}$$

From Eq. 2–22 the time span between accelerating and buffing is

$$t_r = \sqrt{\frac{\epsilon M_r}{K_s}} \left(\text{Sin}^{-1} \frac{F_{ms}}{Z} - \text{Sin}^{-1} \frac{F_{os}}{Z}\right)$$

$$= 0.0697\ (\text{Sin}^{-1}\ 0.8938 - \text{Sin}^{-1}\ 0.4434)$$

$$= 0.0697 \times 0.647 = 0.0451 \text{ sec}$$

where $Z = \sqrt{F_{os}^2 + \epsilon K_s M_r v_o^2} = \sqrt{87948} = 296.6.$

The new compression time becomes

$$T_c = t_a + t_r + t_b = 0.006 + 0.0451 + 0.008$$
$$= 0.0591 \text{ sec.}$$

Repeating the above series of calculations has the time converging to $t_r = 0.044$ sec, or $T_c = 0.058$ sec and $K_s = 16.7$ lb/in. Before buffing, the driving and barrel springs function as one spring. The combined minimum and maximum forces are

$$F_{os} = F_a - K_s\left(\frac{1}{2} L_d + x_r\right)$$

$$= 198.3 - 16.7(4.175 + 1.15) = 198.3 - 88.9$$
$$= 109.4 \text{ lb}$$

$$F_{ms} = F_{os} + K_s L = 109.4 + 16.7 \times 10 = 276.4 \text{ lb}$$

The combined spring forces at end of acceleration period and at the beginning of buffing are

$$F'_{os} = F_{os} + K_s x_r = 109.4 + 16.7 \times 1.15 = 128.6 \text{ lb}$$

$$F'_{ms} = F_{ms} - K_s L_b = 276.4 - 16.7 \times 0.5 = 268 \text{ lb}$$

By setting the minimum driving spring force at $F_o = 25$ lb, the minimum barrel spring force becomes

$$F_{ot} = F_{os} - F_o = 109.4 - 25.0 = 84.4 \text{ lb.}$$

Maintain the same ratio between spring constants as for the initial forces. The driving and barrel spring constants become, respectively,

$$K = \left(\frac{F_o}{F_{os}}\right) K_s = \left(\frac{25}{109.4}\right) 16.7 = 3.8 \text{ lb/in.}$$

$$K_t = \left(\frac{F_{ot}}{F_{os}}\right) K_s = \left(\frac{84.4}{109.4}\right) 16.7 = 12.9 \text{ lb/in.}$$

The corresponding maximum forces are, respectively,

$$F_m = F_o + KL = 25 + 3.8 \times 10 = 63 \text{ lb}$$

$$F_{mt} = F_{ot} + K_t L = 84.4 + 12.9 \times 10 = 213.4 \text{ lb.}$$

The spring constant of the buffer spring is

$$K_{bs} = K_b - K_s = 572 - 16.7 = 555.3 \text{ lb/in.}$$

The buffer spring force at initial contact with recoiling parts is

$$F_{obs} = F_{ob} - F'_{ms} = 961 - 268 = 693 \text{ lb.}$$

At end of buffing, the maximum spring force is

$$F_{mbs} = F_{mb} - F_{ms} = 1247 - 276.4 = 970.6 \text{ lb.}$$

Table **3–2** lists design data and computed stresses for these three springs as well as for the springs of the three types of action employed in the short recoil gun. The calculations are based on the following formulas.

$$d = 0.27\sqrt[3]{DKT}, \text{ (Based on Eq. 2–67a)}$$

$$N = \frac{Gd^4}{8KD^3}, \text{ number of coils}$$

$$\tau = 2.55\left(\frac{F_m D}{d^3}\right), \text{ static shear stress}$$

$$\tau_d = T_c\left[f\left(\frac{T_c}{T}\right)\right], \text{ dynamic shear stress}$$

$$H_s = dN, \text{ solid height}$$

The available potential energy in the buffer spring for counterrecoil is

$$E_{bc} = \frac{e}{2}(F_{mbs} + F_{obs})L_b = \frac{0.5}{2}(970.6 + 693)0.5$$

$$= 208 \text{ in.-lb.}$$

The available potential energy in the barrel spring for counterrecoil is

$$E_t = \frac{\epsilon}{2}(F_{mt} + F_{ot})L = \frac{0.5}{2}(213.4 + 84.4)\,10$$

$$= 744.5 \text{ in.-lb.}$$

The potential energy of the barrel spring that augments the buffer spring is

$$\Delta E_t = \frac{\epsilon}{2}(F_{mt} + F'_{mt})L_b = \frac{0.5}{2}(213.4 + 207)\,0.5$$

$$= 52.6 \text{ in.-lb.}$$

where

$$F'_{mt} = F_{mt} - K_t L_b = 213.4 - 12.9 \times 0.5 = 207 \text{ lb.}$$

The total energy of the counterrecoiling barrel at the end of buffer action becomes

$$E_{crb} = E_{bc} + \Delta E_t = 260.6 \text{ in.-lb.}$$

The corresponding velocity is

$$v_{crb} = \sqrt{\frac{2E_{crb}}{M_t}} = \sqrt{\frac{521.2 \times 386.4}{50}} = \sqrt{4028}$$

$$= 63.5 \text{ in./sec.}$$

The maximum energy of the counterrecoiling barrel is

$$E_{crt} = E_b + E_t = 208 + 744.5 = 952.5 \text{ in.-lb.}$$

The maximum velocity attained by the bolt in counterrecoil is

$$v_{crt} = \sqrt{\frac{2E_{crt}}{M_t}} = \sqrt{\frac{1905 \times 386.4}{50}} = \sqrt{14722}$$

$$= 121.3 \text{ in./sec.}$$

The maximum energy of the counterrecoiling bolt is

$$E_{crd} = \frac{\epsilon}{2}(F_m + F_o)L = \frac{0.5}{2}(63 + 25)\,10$$

$$= 220 \text{ in.-lb.}$$

The maximum velocity attained by the bolt in counterrecoil is

$$v_{cr} = \sqrt{\frac{2E_{crd}}{M_b}} = \sqrt{\frac{440 \times 386.4}{10}} = \sqrt{17002}$$

$$= 130.4 \text{ in./sec}$$

The time elapsed from the propellant gas period until the buffer is reached, obtained from Eq. **2–51**, will be

$$t_r = \sqrt{\frac{\epsilon M_r}{K_s}}\left(\sin^{-1}\frac{F'_{ms}}{Z} - \sin^{-1}\frac{F'_{os}}{Z}\right)$$

$$t_r = 0.0682(\sin^{-1} 0.8904 - \sin^{-1} 0.4280)$$

$$= 0.0682(63.10 - 25.33)/57.3 = 0.045$$

where

$$F'_{os} = 128.6 \text{ lb} \; ; \; F'_{ms} = 268 \text{ lb}$$

$$\sqrt{\frac{\epsilon M_r}{K_s}} = \sqrt{\frac{0.5 \times 60}{16.7 \times 386.4}} = 0.068 \text{ sec}$$

$$Z = \sqrt{(F'_{os})^2 + \epsilon K_s M_r v_o^2} = \sqrt{16538 + 73747}$$
$$= 301 \text{ lb}$$

$K_s = 16.7$ lb/in.

$\epsilon = 0.5$

$M_r v_o^2 = 2E_r = 8832$ in.-lb

The time elapsed during buffing, **Eq.** 2–23, becomes

$$t_b = \sqrt{\frac{\epsilon M_r}{K_b}} \cos^{-1} \frac{F_{ob}}{F_{mb}}$$

$$= \sqrt{\frac{0.5 \times 60}{572 \times 386.4}} \operatorname{Cos}^{-1} \frac{961}{1247}$$

$$= \sqrt{0.0001357} \operatorname{Cos}^{-1} 0.7706 = 0.0116\left(\frac{39.6}{57.3}\right)$$

$$= 0.008 \text{ sec.}$$

The time elapsed for counterrecoil at the end of buffer activity is obtained from **Eq.** 2–27.

$$t_{crb} = \sqrt{\frac{M_t}{\epsilon(K_{bs} + K_t)}} \cos^{-1} \frac{F_{obs} + F'_{mt}}{F_{mbs} + F_{mt}}$$

$$= \sqrt{\frac{50}{0.5 \times 568.2 \times 386.4}} \operatorname{Cos}^{-1} \frac{900}{1184}$$

$$= 0.0214 \operatorname{Cos}^{-1} 0.7601 = 0.0214\left(\frac{40.53}{57.3}\right)$$

$$= 0.0151 \text{ sec.}$$

Compute the time elapsed for the barrel to negotiate the remaining distance in counterrecoil according to **Eq.** 2–26.

$$t_{crt} = \sqrt{\frac{M_t}{\epsilon K_t}} \left(\operatorname{Sin}^{-1} \frac{-F_{ot}}{Z} - \operatorname{Sin}^{-1} \frac{-F'_{mt}}{Z} \right)$$

$$= 0.1418 \left[\operatorname{Sin}^{-1}(-0.3557) - \operatorname{Sin}^{-1}(-0.8723)\right]$$

$$= 0.1418\,(339.17 - 299.27)/57.3 = 0.0987 \text{ sec}$$

where $$\sqrt{\frac{M_t}{\epsilon K_t}} = \sqrt{\frac{50}{0.5 \times 12.9 \times 386.4}}$$

$$= \sqrt{0.0201} = 0.1418 \text{ sec}$$

$$Z = \sqrt{\left(F'_{mt}\right)^2 + \left(\frac{K_t}{\epsilon}\right) M v^2_{crb}}$$

$$= \sqrt{207^2 + \left(\frac{12.9}{0.5}\right) 521.6} = \sqrt{56306} = 237.3 \text{ lb}$$

The time elapsed for counterrecoil of the bolt, **Eq.** 2–27, is

$$t_{cr} = \sqrt{\frac{M_b}{\epsilon K}} \operatorname{Cos}^{-1} \frac{F_o}{F_m}$$

$$= \sqrt{\frac{10}{0.5 \times 3.8 \times 386.4}} \cos^{-1} \frac{25}{63}$$

$$= \sqrt{0.01362} \operatorname{Cos}^{-1} 0.3968$$

$$= 0.1167 \left(\frac{66.12}{57.3}\right) = 0.1347 \text{ sec.}$$

Time of cycle will be

$$t_c = t_a + t_r + t_b + t_{crb} + t_{crt} + t_{cr}$$

$$= 0.006 + 0.045 + 0.008 + 0.0151 + 0.0987 + 0.1347$$

$$= 0.3075 \text{ sec}$$

The rate of fire becomes

$$f_r = \frac{60}{t_c} = \frac{60}{0.3075} = 195 \text{ rounds/min.}$$

TABLE 3—2. SPRING DESIGN DATA OF RECOIL-OPERATED GUNS

Type / Spring / Data	Long Recoil Driving	Long Recoil Barrel	Long Recoil Buffer	Short Recoil Driving	Short Recoil Barrel	Short Recoil Buffer	Short Recoil, Bolt Buffer Driving	Short Recoil, Bolt Buffer Buffer	Short Recoil, Bolt Accelerator Driving	Short Recoil, Bolt Accelerator Barrel
K, lb/in.	3.8	12.9	555.3	2.6	35.4	579.6	2.4	229.6	20	200
F_m, lb	63	213.4	970.6	53.1	159.4	1645.6	36	344	320	284
T_c, msec	0.058	0.058	0.0080	0.0743	0.0165	0.0105	0.0529	0.0060	0.0347	0.0076
(T_c/T)	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	1.8	3.8
T, msec	0.0153	0.0153	0.0021	0.0196	0.0043	0.0028	0.0139	0.0016	0.0193	0.0020
D, in.	0.5	2.0	1.5	0.5	2.0	1.875	0.5	0.875	1.0	0.5
D^3, in.^{3}	0.125	8.0	3.375	0.125	8.0	6.592	0.125	0.766	1.0	0.125
DKT	0.0291	0.395	1.749	0.0255	0.304	3.043	0.0167	0.321	0.386	0.200
$\sqrt[3]{DKT}$	0.307	0.734	1.205	0.294	0.672	1.449	0.256	0.685	0.728	0.585
d, in.	0.083	0.199	0.325	0.079	0.181	0.391	0.069	0.185	0.196	0.158
$d^3 \times 10^3$, in3	0.572	7.880	34.33	0.493	5.930	59.78	0.329	6.331	7.530	3.944
$d^4 \times 10^4$, in.^{4}	0.475	15.68	111.6	0.390	10.73	233.7	0.227	11.71	14.76	6.232
G, kpsi	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5
N	144	21.8	8.6	173	5.5	8.8	109	9.4	106	34
τ, kpsi	140	138	108	137	137	132	139	118	109	92
$f(T_c/T)$	4	4	4	4	4	4	4	4	2	4
τ_d, kpsi	148	145	114	144	144	139	146	125	115	97
H_s, in.	12.0	4.4	2.8	13.7	1.0	3.5	7.6	1.7	20.8	5.4

3–4 SHORT RECOIL DYNAMICS

The dynamics of the short recoil-operated gun approach those of the retarded blowback types more nearly than the long recoil. To eliminate all blowback tendencies, the bolt latch is not released until the propellant gas becomes ineffective. After unlatching, bolt and barrel continue recoiling, but as separate units. The barrel is arrested by the combined effort of the barrel spring and buffer. Having the same velocity of free recoil, but because it travels a much shorter distance than the bolt, the barrel will stop recoiling before the bolt. Both the bolt driving spring and buffer spring characteristics are determined from the recoil energy of the respective masses. The characteristics of the barrel spring are selected more arbitrarily but still must conform to the same initial load requirement as that for the long recoil barrel spring, i.e., sufficient to hold the barrel in battery.

3–5 SAMPLE PROBLEM — SHORT RECOIL MACHINE GUN

3–5.1 SPECIFICATIONS: Identical to long recoil problem (see par. 3–3.1)

3–5.2 DESIGN DATA

L = 10 in., minimum bolt travel distance

W_b = 10 lb, weight of bolt unit

W_t = 50 lb, weight of barrel unit

ϵ = 0.5, efficiency of spring system

The numerical integration of Table 3–1 also applies to this problem, therefore, the distance recoiled during the effective pressure period of this propellant gas, x_r = 1.15 in. and the corresponding recoil velocity v = 238.5 in./sec. The recoil energy of the bolt is

$$E_{rb} = \frac{1}{2}\left(M_b v^2\right) = \frac{1}{2}\left(\frac{10}{386.4}\right)56882 = 736 \text{ in.-lb.}$$

The average force of the driving spring becomes

$$F_a = \frac{\epsilon E_{rb}}{L - x_r} = \frac{0.5 \times 736}{10.0 - 1.15} = 41.6 \text{ lb.}$$

To be compatible with allowable stresses, the spring characteristics must conform to computed data obtained from Eqs. 2–23 and 2–67b. When based on constant deceleration, the time required to stop the bolt in recoil is

$$t = \frac{2(L - x_r)}{v_r} = \frac{2 \times 8.85}{238.5} = 0.0743 \text{ sec.}$$

Including the time of the effective gas period, the compression time of the driving spring is

$$T_c = t + t_a = 0.0743 + 0.006 = 0.0803 \text{ sec.}$$

The corresponding surge time will be

$$T = \frac{T_c}{3.8} = \frac{0.0803}{3.8} = 0.0211 \text{ sec.}$$

Apply Eq. 2–67b to compute the spring constant K.

$$KT = \frac{F_m}{1037} = \frac{F_a + \frac{1}{2}K(L - x_r)}{1037}$$

$$21.881K = 41.6 + 4.425K$$

$$K = \frac{41.6}{17.456} = 2.4 \text{ lb/in.}$$

$$F_m = F_a + 4.425K = 41.6 + 10.6 = 52.2 \text{ lb}$$

$$F_o' = F_a - 4.425K = 41.6 - 10.6 = 31.0 \text{ lb}$$

The decelerating time, Eq. 2–23, is

$$t = \sqrt{\frac{\epsilon M_b}{K}}\cos^{-1}\frac{F_o'}{F_m} = \sqrt{\frac{0.5 \times 10}{2.4 \times 386.4}}\cos^{-1}\frac{31.0}{52.2}$$

$$= 0.0735 \times 0.935 = 0.0687 \text{ sec.}$$

The total compression time of the spring is

$$T_c = t + t_a = 0.0687 + 0.006 = 0.0747 \text{ sec.}$$

Adjust the time and recompute Eqs. 2–67b and 2–23, the time and spring constant coverge to t = 0.075 sec and K = 2.6 lb/in., respectively.

The maximum driving spring force F_m is

$$F_m = F_a + \frac{1}{2}K(L - x_r) = 41.6 + 11.5 = 53.1 \text{ lb.}$$

The driving spring force at x = 1.15 in. is

$$F'_o = F_m - K(L - x_r) = 53.1 - 23.0 = 30.1 \text{ lb.}$$

The initial driving spring force is

$$F_o = F_m - KL = 53.1 - 26.0 = 27.1 \text{ lb.}$$

According to Eq. 2–22 the time of bolt recoil is

$$t_r = \sqrt{\frac{\epsilon M_b}{K}} \operatorname{Cos}^{-1} \frac{F'_o}{F_m}$$

$$= \sqrt{\frac{0.5 \times 10}{2.6 \times 386.4}} \cos^{-1} \frac{30.1}{53.1}$$

$$= \sqrt{0.004976} \operatorname{Cos}^{-1} 0.5669 = 0.0706\left(\frac{55.47}{57.3}\right)$$

$$= 0.0683 \text{ sec.}$$

According to Eq. 2–27, the time of bolt counterrecoil is

$$t_{cr} = \sqrt{\frac{M_b}{\epsilon K}} \operatorname{Cos}^{-1} \frac{F_o}{F_m}$$

$$= \sqrt{\frac{10}{0.5 \times 2.6 \times 386.4}} \operatorname{Cos}^{-1} \frac{27.1}{53.1}$$

$$= \sqrt{0.0199} \operatorname{Cos}^{-1} 0.5104 = 0.1411\left(\frac{59.3}{57.3}\right)$$

$$= 0.1459 \text{ sec.}$$

The time elapsed during bolt action may determine the firing rate, provided that the barrel returns to battery before the bolt recoils fully. The recoil energy of the barrel is

$$E_{rt} = \frac{1}{2}\left(M_t v^2\right) = \frac{1}{2}\left(\frac{50}{386.4}\right) 56882$$

$$= 3680 \text{ in.-lb.}$$

The average buffing force, to be approximately the same as for the long recoil, should have a buffer travel of L_b = 1.375 in.

According to Eq. 2–16 the average spring force during buffing is

$$F_{ab} = \frac{\epsilon E_{rt}}{L_b} = \frac{0.5 \times 3680}{1.375} = 1338 \text{ lb.}$$

Assume constant deceleration so that the time needed to stop the barrel during recoil becomes

$$t_{rt} = \frac{2L_b}{v} = \frac{2.75}{238.5} = 0.0115 \text{ sec.}$$

This time is also the compression time T_c for the combined buffer and barrel springs. The surge time is

$$T = \frac{T_c}{3.8} = \frac{0.0115}{3.8} = 0.00302 \text{ sec.}$$

Apply Eq. 2–67b to solve for the spring constant and corresponding forces.

$$K_b T = \frac{F_{mb}}{1037} = \frac{F_{ab} + \frac{1}{2}\left(K_b L_b\right)}{1037}$$

$$3.132 K_b = 1338 + 0.688 K_b$$

$$K_b = \frac{1338}{2.444} = 547 \text{ lb/in.}$$

$$F_{mb} = 1338 + 0.688 K_b = 1338 + 376 = 1714 \text{ lb}$$

$$F_{ob} = 1338 - 0.688 K_b = 1338 - 376 = 962 \text{ lb}$$

The decelerating time, Eq. 2–23, will be

$$t = \sqrt{\frac{\epsilon M_t}{K_b}}\ \mathrm{Cos}^{-1}\ \frac{F_{ob}}{F_{mb}}$$

$$= \sqrt{\frac{0.5 \times 50}{547 \times 386.4}}\ \mathrm{Cos}^{-1}\ \frac{962}{1714}$$

$$= 0.0109 \times 0.975 = 0.0106 \text{ sec.}$$

By repeated computation, the time and spring constant quickly converge.

$t_{rt} = 0.0105$ sec

$K_b = 615$ lb/in.

$$F_{ob} = F_{ab} - \frac{1}{2}\left(K_b L_b\right) = 1338 - 423 = 915 \text{ lb}.$$

$$F_{mb} = F_{ob} + K_b L_b = 915 + 846 = 1761 \text{ lb}$$

To realize an acceptable firing rate, the barrel spring force at firing is set as $F_{ot} = 70$ lb, the initial barrel spring force.

The compression time includes the propellant gas period and becomes

$$T_c = t_{rt} + t_a = 0.0105 + 0.006 = 0.0165 \text{ sec.}$$

The surge time $T = \frac{T_c}{3.8} = 0.00434$ sec.

The appropriate spring constant is computed from Eq. 2–67b.

$$1037 K_t T = F_m = F_{ot} + K_t L_t = 70 + 2.525 K_t$$

where $L_t = L_b + x_r = 1.375 + 1.15 = 2.525$ in.

$$K_t = \frac{70}{4.5 - 2.525} = \frac{70}{1.975} = 35.4 \text{ lb/in.}$$

The barrel spring force at end of recoil is

$$F_{mt} = F_{ot} + K_t L_t = 70 + 89.4 = 159.4 \text{ lb.}$$

The buffer spring constant is

$$K_{bs} = K_b - K_t = 615 - 35.4 = 579.6 \text{ lb/in.}$$

The barrel spring force at the end of the propellant gas period is

$$F_{tb} = F_{ot} + K_t x_r = 70 + 35.4 \times 1.15 = 110.7 \text{ lb.}$$

The buffer spring force at the beginning of buffing is

$$F_{obs} = F_{ob} - F_{tb} = 915 - 110.7 = 804.3 \text{ lb.}$$

The maximum buffer spring force is

$$F_{mbs} = F_{mb} - F_{mt} = 1761 - 159.4 = 1601.6 \text{ lb.}$$

The time of barrel recoil from Eq. 2–22 becomes

$$t_{rt} = \sqrt{\frac{\epsilon M_t}{K_b}}\ \mathrm{Cos}^{-1}\ \frac{F_{ob}}{F_{mb}}$$

$$= \sqrt{\frac{0.5 \times 50}{615 \times 386.4}}\ \mathrm{Cos}^{-1}\ \frac{915}{1761}$$

$$= \sqrt{0.000105}\ \mathrm{Cos}^{-1}\ 0.5196 = 0.01025\left(\frac{58.7}{57.3}\right)$$

$$= 0.0105 \text{ sec.}$$

The available energy released by the spring system at the end of buffer travel is

$$E_{crb} = \frac{\epsilon}{2}\left(F_{mb} + F_{ob}\right) L_b = \frac{0.5}{2}\ (1761 + 915)\ 1.375$$

$$= 920 \text{ in.-lb.}$$

The counterrecoil velocity at the end of buffer action becomes

$$v_{crb} = \sqrt{\frac{2E_{crb}}{M_t}} = \sqrt{\frac{1840 \times 386.4}{50}} = \sqrt{14220}$$

$$= 119.2 \text{ in./sec.}$$

The time consumed for counterrecoil by buffer action, **Eq.** 2–27, is

$$t_{crb} = \sqrt{\frac{M_t}{\epsilon K_b}} \operatorname{Cos}^{-1} \frac{F_{ob}}{F_{mb}} = \sqrt{\frac{50}{0.5 \times 615 \times 386.4}} \operatorname{Cos}^{-1} \frac{915}{1761}$$

$$= \sqrt{0.00042}\ Cos\text{-}'\ 0.5196 = 0.0205\left(\frac{58.7}{57.3}\right) = 0.0210 \text{ sec.}$$

The time of counterrecoil for the remaining barrel travel of x_r = 1.15 in. via Eq. 2–26 is

$$t_{crt} = \sqrt{\frac{M_t}{\epsilon K_t}} \left(\operatorname{Sin}^{-1} \frac{-F_{ot}}{Z} - \operatorname{Sin}\text{-}' \frac{-F_{tb}}{Z} \right)$$

$$= 0.0855\ [\operatorname{Sin}\text{-}'\ (-0.1854) - \operatorname{Sin}\text{-}'\ (-0.2934)]$$

$$= 0.0855\ (349.32 - 342.93)/57.3 = 0.0095 \text{ sec}$$

where $\sqrt{\frac{M_t}{\epsilon K_t}} = \sqrt{\frac{50}{0.5 \times 35.4 \times 386.4}} = \sqrt{0.00731}$

$$= 0.0855 \text{ sec}$$

$$Z = \sqrt{F_{tb}^2 + \frac{K_t}{\epsilon} M v_{crb}^2}$$

$$= \sqrt{110.7^2 + \frac{35.4}{0.5}\ 1840}$$

$$= 377.5 \text{ lb}$$

The time elapsed for the complete barrel cycle is

$$t_{ct} = t_a + t_{rt} + t_{crb} + t_{crt}$$

$$= 0.0060 + 0.0105 + 0.0210 + 0.0095$$

$$= 0.0470 \text{ sec.}$$

The time of the barrel cycle is considerably less than the recoil time of the bolt, t_r = 0.0683 sec, and therefore has no influence on the firing cycle if its present operation remains undisturbed. The cyle time of the bolt is

$$t_c = t_a + t_r + t_{cr}$$

$$= 0.0060 + 0.0683 + 0.1459 = 0.2202 \text{ sec.}$$

The firing rate is

$$f_r = \frac{60}{t_c} = \frac{60}{0.2202} = 272 \text{ rounds/min.}$$

This rate is faster than for long recoil (f_r = 195) but slower than the recoil-operated delayed blowback gun (f_r = 420). The rate of the short recoil gun can be improved by resorting to a softer driving spring and the addition of a bolt buffer. The time of bolt travel will then be less in both directions thereby increasing the rate of fire. For example, to initiate the computations, select a driving spring having these preliminary characteristics:

F_o = 12 lb (2 lb greater than the 10 lb bolt weight)

K = 1.0 lb/in., preliminary spring constant

L_b = 0.5 in., buffer travel

The driving spring force when the buffer is reached becomes

$$F_{db} = F_o + K(L - L_b) = 12 + 1 \times 9.5 = 21.5 \text{ lb.}$$

The initial driving spring force at $x = 1.15$ in. is

$$F = F_o + Kx_r = 12 + 1.15 = 13.15 \text{ lb}$$

The energy absorbed during this period will be

$$E_d = \frac{1}{2\epsilon}(F + F_{db})L_d = \frac{1}{2 \times 0.5}\left(34.65\right)8.35$$

$$= 289 \text{ in.-lb}$$

where

$$L_d = L - L_b - x_r = 10.0 - 0.5 - 1.15 = 8.35 \text{ in.}$$

The energy to be absorbed by the combined effort of buffer and driving springs is

$$E_b = E_{rb} - E_d = 736 - 289 = 447 \text{ in.-lb}$$

The velocity of the bolt as it contacts the buffer is also the buffer velocity v_b during recoil.

$$v_b = \sqrt{\frac{2E_b}{M_b}} = \sqrt{\frac{894 \times 386.4}{10}} = \sqrt{34544}$$

$$= 185.9 \text{ in./sec}$$

The time during this decelerating period, based on constant deceleration, is

$$t_d = \frac{2L_d}{v + v_b} = \frac{2 \times 8.35}{238.5 + 185.9} = 0.0393 \text{ sec.}$$

Buffing time, based on constant deceleration, is

$$t_b = \frac{2L_b}{v_b} = \frac{2 \times 0.5}{185.9} = 0.0054 \text{ sec}$$

The total time of driving spring compression will be

$$T_c = t_a + t_d + t_b = 0.0507 \text{ sec.}$$

The spring surge time $T = \dfrac{T_c}{3.8} = 0.0133$ sec.

The required spring constant that supersedes the preliminary $K = 1.0$ is computed from Eq. 2–67b.

$$KT = \frac{F_m}{1037} = \frac{F + KL_d}{1037} = \frac{13.2 + 8.35K}{1037}$$

$$K = \frac{13.2}{13.79 - 8.35} = 2.4 \text{ lb/in.}$$

The spring forces at the limits of L_d are

$$F_d = F_o + Kx_r = 12.0 + 2.4 \times 1.15 = 14.8 \text{ lb}$$

$$F_{db} = F_o + K(L - L_b) = 12.0 + 2.4 \times 9.5 = 34.8 \text{ lb.}$$

The time for this driving spring to compress from the propellant gas period to the buffer is obtained from Eq. 2–22.

$$t_d = \sqrt{\frac{\epsilon M_b}{K}}\left(\mathrm{Sin}^{-1}\frac{F_{db}}{Z} - \mathrm{Sin}^{-1}\frac{F_d}{Z}\right)$$

$$= 0.0735\left(\mathrm{Sin}^{-1}\, 0.7807 - \mathrm{Sin}^{-1}\, 0.3318\right)$$

$$= 0.0735\ (51.18 - 19.28) / 57.3 = 0.0409 \text{ sec}$$

where

$$\sqrt{\frac{\epsilon M_b}{K}} = \sqrt{\frac{0.5 \times 10}{2.4 \times 386.4}} = \sqrt{0.00539} = 0.0735 \text{ sec}$$

$$Z = \sqrt{F_d^2 + \epsilon K M_b v^2}$$

$$= \sqrt{219 + 0.5 \times 2.4 \times 1472} = 44.6 \text{ lb}$$

$$M_b v^2 = 2E_{rb} = 1472 \text{ in.-lb.}$$

t_d is somewhat higher than the initial T_c = 0.0507 sec. Repeating the computation establishes these values. To continue the analysis of the spring system, compute the energy to be absorbed by the buffer system

$$E_b = E_{rb} - \left(\frac{F_d + F_{db}}{2\epsilon}\right) L_d = 736 - 414 = 322 \text{ lb}$$

where

$$L_d = 8.35 \text{ in.}$$

$$\epsilon = 0.5$$

The average buffer spring system is

$$F_{as} = \frac{\epsilon E_b}{L_b} = \frac{0.5 \times 322}{0.5} = 322 \text{ lb.}$$

The velocity at buffer contact is

$$v_b = \sqrt{\frac{2E_b}{M_b}} = \sqrt{\frac{644 \times 386.4}{10}} = \sqrt{24884}$$

$$= 157.7 \text{ in./sec.}$$

For constant deceleration, the time of buffer action in recoil and also the compression time of the spring is

$$T_c = t_b = \frac{2L_b}{v_b} = \frac{2 \times 0.5}{157.7} \; 0.0063 \text{ sec.}$$

The surge time $T = \dfrac{T_c}{3.8} = 0.0016$ sec.

Iterative computation has the spring characteristics converging rapidly. The computed buffer time, according to the procedure which follows, was 0.006 sec. Thus $T_c = t_b = 0.006$ sec and T = 0.00158 sec.

The spring constant is computed from Eq. 2–67b.

$$K_b T = \frac{F_m}{1037} = \frac{F_{as} + \frac{1}{2}\left(L_b K_b\right)}{1037} = \frac{F_{as} + 0.25 K_b}{1037}$$

$$1037 \times 0.00158 K_b = 322 + 0.25 K_b$$

$$K_b = \frac{322}{1.388} = 232 \text{ lb/in.}$$

$$F_{ob} = 322 - 0.25 K_b = 264 \text{ lb}$$

$$F_{mb} = 322 + 0.25 K_b = 380 \text{ lb.}$$

According to Eq. 2–23, the buffing time will be

$$t_b = \sqrt{\frac{\epsilon M_b}{K_b}} \operatorname{Cos}^{-1} \frac{F_{ob}}{F_{mb}}$$

$$= \sqrt{\frac{0.5 \times 10}{232 \times 386.4}} \cos^{-1} \frac{264}{380}$$

$$= 10^{-3} \sqrt{55.78} \operatorname{Cos}^{-1} 0.6947 = 0.00747 \times 0.803$$

$$= 0.0060 \text{ sec}$$

which verifies that t_b = 0.0060 sec and fixes the spring constant at K_b = 232 lb/in. The spring constant of the buffer spring alone becomes

$$K_{bs} = K_b - K = 232 - 2.4 = 229.6 \text{ lb/in.}$$

$$F_{obs} = F_{as} - 0.25 K_b - F_{db} = 322 - 58 - 34.8$$

$$= 229.2 \text{ lb}$$

$$F_{mbs} = F_{obs} + K_{bs} L_b = 229.2 + 114.8 = 344 \text{ lb.}$$

The recoil time of the bolt and, therefore, the compression time of the driving spring is

$$t_{rb} = t_a + t_d + t_b = 0.0060 + 0.0409 + 0.0060$$

$$= 0.0529 \text{ sec.}$$

The time of bolt return from buffing action, Eq. 2–27, is

$$t_{crb} = \sqrt{\frac{M_b}{\epsilon K_b}} \cos^{-1} \frac{F_{ob}}{F_{mb}} = 0.0149 \times 0.803$$

$$= 0.012 \text{ sec.}$$

The energy of the moving bolt at the end of buffer return is

$$E_{crb} = \frac{\epsilon}{2}(F_{ob} + F_{mb})\, L_b = \frac{0.5}{2}(380 + 264)\, 0.5$$

$$= 80.5 \text{ in.-lb.}$$

The time elapsed for completing the bolt return, Eq. 2–26, becomes

$$t_{cr} = \sqrt{\frac{M_b}{\epsilon K}} \left(\mathrm{Sin}^{-1} \frac{-F_o}{Z} - \mathrm{Sin}^{-1} \frac{-F_{db}}{Z} \right)$$

$$= \sqrt{\frac{10}{0.5 \times 2.4 \times 386.4}} \left(\mathrm{Sin}^{-1} \frac{-12}{44.54} - \mathrm{Sin}^{-1} \frac{-34.8}{44.54} \right)$$

$$= 0.1468\,(344.37 - 308.62)/57.3 = 0.0916 \text{ sec}$$

where

$$Z = \sqrt{F_{db}^2 + \left(\frac{K}{\epsilon}\right) Mv_{crb}^2} = \sqrt{34.8^2 + \left(\frac{2.4}{0.5}\right) 161}$$

$$= \sqrt{1984} = 44.54 \text{ lb.}$$

$$Mv_{crb}^2 = 2E_{crb} = 161 \text{ in.-lb.}$$

Time of the complete cycle is

$$t_c = t_a + t_d + t_b + t_{crb} + t_{cr}$$

$$= 0.0060 + 0.0409 + 0.0060 + 0.0120 + 0.0916$$

$$= 0.1565 \text{ sec.}$$

The rate of fire is

$$f_r = \frac{60}{t_c} = \frac{60}{0.1565} = 383 \text{ rounds/min.}$$

This rate is an increase of 28% over the rate of the gun which does not have a buffer for the bolt.

3–6 ACCELERATORS

Recoil-operated machine guns are relatively slow firing because of their slow response to the propellant gas forces. This slow response is due primarily to the large inertial resistance that must be overcome while accelerating the recoiling parts. The entire dynamics structure depends on the velocity of free recoil; the higher the velocity, the higher the rate of fire, but the velocity of free recoil can be influenced only by the mass of the recoiling parts which, for any given gun, is usually limited by structural requirements. High speeds, therefore, must be gained by other means. One of these, as demonstrated in the preceding problem, involves the arrangement of springs whereby somewhat faster action develops by delaying large energy absorption until the buffer is reached. This constitutes the extent of control over firing rates of long recoil guns. However, for short recoil guns, higher rates can be achieved by installing accelerators.

An accelerator, Fig. 3–3, is merely a rotating cam arranged to transfer, over a short distance, some momentum from the rest of the recoiling parts to the bolt, thus augmenting its velocity. At any given instant, the cam and the two masses represent a rotating system. From the law of conservation of angular momentum, the total remains unchanged after an exchange of momentums.

$$r_t M_t v + r_b M_b v = r_t M_t v_t + r_b M_b v_b \qquad (3\text{–}1)$$

where g = acceleration of gravity

$M_b = \frac{W_b}{g}$, mass of bolt

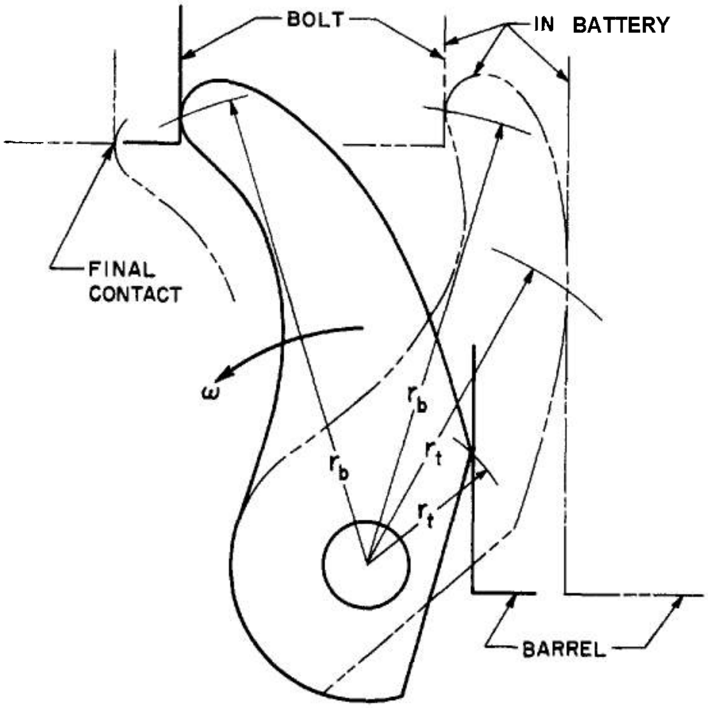


Figure 3–3. Accelerator Geometry

$M_t = \frac{W_t}{g}$, mass of barrel and components

r_b = cam radius to contact point on bolt

r_t = cam radius to contact point on barrel

v = velocity of recoiling parts just prior to accelerator action

v_b = velocity of bolt after accelerator action

v_t = velocity of barrel and components

W_b = weight of bolt

W_t = weight of barrel and components

At the instant of parting from the accelerator, the bolt has acquired a velocity higher than the recoiling barrel. Solving for v_b Of Eq. 3–1

$$v_b = \frac{M_t(v - v_t)}{R_c M_b} + v \qquad (3\text{–}2)$$

where $R_c = \frac{r_b}{r_t}$.

The law of conservation of energy also applies.

$$\frac{1}{2}(M_t + M_b)v^2 = \frac{1}{2}\left(M_t v_t^2\right) + \frac{1}{2}\left(M_b v_b^2\right) \qquad (3\text{–}3)$$

By substituting the expression for v_b, Eq. 3–2, into Eq. 3–3 and collecting terms, we will have a quadratic equation having v_t as the only unknown. The solution for v_t in general terms is too unwieldy and hence not shown. A specific solution is demonstrated in the sample problem.

Other unknown factors are the energy absorbed by the driving and buffer springs and the subsequent change in recoil velocity while the accelerator functions. The procedure for computing these factors is interative. A specific analysis demonstrates this procedure far more readily than a general solution. If follows in the sample problem.

3–7 SAMPLE PROBLEM – ACCELERATOR

3–7.1 SPECIFICATIONS: Identical to long recoil problem (see par. 3–3.1)

3–7.2 DESIGN DATA:

L = 10 in., minimum bolt travel distance

W_b = 10 lb, weight of bolt unit

W_t = 50 lb, weight of barrel unit

e = 0.5, efficiency of spring system

Table 3–3 has the numerical integration for a recoiling weight of 60 lb. The buffer or barrel spring and driving springs resist recoil from the start but are measureably effective only after 1/2 inch of recoil. The buffer spring is not compressed on installation. The accelerator (Fig. 3–3) is so designed that at final contact with the bolt, the bolt has moved 0.56 in., and the barrel, 0.28 in. The radii to the two contact points at this time, are

$$r_b = 0.90 \text{ in. when } \Delta x_b = 0.56 \text{ in.}$$

$$r_t = 0.25 \text{ in. when } \Delta x_t = 0.28 \text{ in.}$$

$$R_c = \frac{r_b}{r_t} = 3.6$$

At the end of the propellant gas period, when $t = 6$ msec, the barrel has recoiled x_r = 1.14 in. and has a velocity of free recoil $v_f = v$ = 234.1 in./sec. A preliminary analysis, conducted by the same procedure that follows showed that the transfer of momentum to the bolt caused the barrel to reverse its direction of motion. Also, appropriate spring constants were selected.

K = 20 lb/in., driving spring constant

K_t = 200 lb/in., barrel spring constant

TABLE 3–3. RECOIL TRAVEL OF 20 mm GUN EQUIPPED WITH ACCELERATOR

t, msec	Δt, msec	A_i, lb-sec/in.^{2}	$FA t$, lb-sec	Δv, in./sec	v, in./sec	v_a, in./sec	Δx, in.
0.25	0.25	3.44	1.77	11.4	11.4	5.7	0.0014
0.50	0.25	10.15	5.24	33.7	45.1	28.2	0.0071
0.75	0.25	11.89	6.13	39.5	84.6	64.8	0.0162
1.00	0.25	11.12	5.74	37.0	121.6	103.1	0.0258
1.25	0.25	9.25	4.76	30.6	152.0	131.9	0.0330
1.50	0.25	7.10	3.76	24.2	176.4	159.3	0.0398
1.75	0.25	5.23	2.69	17.3	193.7	185.0	0.0462
2.00	0.25	3.71	1.91	12.6	206.3	200.0	0.0500
2.25	0.25	2.58	1.34	8.6	214.9	210.6	0.0526
2.50	0.25	1.82	0.94	6.1	221.0	218.0	0.0545
2.75	0.25	1.39	0.72	4.6	225.6	223.3	0.0558
3.00	0.25	1.06	0.55	3.5	229.1	227.4	0.0569
3.263	0.263	1.00	0.52*	3.3	232.4	230.8	0.0607
4.00	0.737	1.34	0.40*	2.6	235.0	233.7	0.1711
5.00	1.00	1.04	0.12*	0.8	235.8	235.4	0.2354
6.00	1.00	0.40	-0.27*	-1.7	234.1	235.0	0.2350

*Reduced by resistance of springs

$$x_r = \Sigma \Delta x = 1.14 \text{ in.}$$

The energy absorbed by the springs during the bolt acceleration period reduced the recoil velocity to 225.4 in./sec. This velocity was obtained by iterative computation. The energy absorbed by the barrel spring is

$$\Delta E_t = \left(\frac{F_t + F_{mt}}{2\epsilon}\right)\Delta x_t = \left(\frac{228 + 284}{2 \times 0.5}\right)0.28$$

$$= 143 \text{ in.-lb}$$

$F_t = K_t x_r = 200 \times 1.14 = 228$ lb, barrel spring force at beginning of accelerator action

$F_{mt} = K_t(x_r + \Delta x) = 200(1.14 + 0.28) = 284$ lb barrel spring force at end of barrel travel

$\epsilon = 0.5$, efficiency of spring system.

The two preceding sets of calculations had the energy absorbed by the driving spring equalling 3948 and 3923 in.-lb, respectively. With the average $E = 3936$ in.-lb, the average driving spring force over the remaining recoil distance becomes

$$F_a = \frac{\epsilon E}{L_{dr}} = \frac{0.5 \times 3936}{8.3} = 237 \text{ b}$$

where

$$L_{dr} = L_d - x_r - \Delta x_b = 10.0 - 1.14 - 0.56 = 8.3 \text{ in.}$$

The driving spring force at the end of acceleration is

$$F_e = F_a - \frac{1}{2}\left(KL_{dr}\right) = 237 - 83 = 154 \text{ lb.}$$

The driving spring force at the end of recoil is

$$F_m = F_a + \frac{1}{2}\left(KL_{dr}\right) = 237 + 83 = 320 \text{ lb.}$$

The driving spring force at assembly is

$$F_o = F_m - KL_d = 320 - 200 = 120 \text{ lb.}$$

The energy absorbed by the driving spring force during acceleration is

$$\Delta E_b = \left(\frac{F_e + F}{2\epsilon}\right)\Delta x_b = \left(\frac{154 + 143}{2 \times 0.5}\right)0.56$$

$$= 166 \text{ in.-lb}$$

where $F = F_o + Kx_r = 120 + 20 \times 1.14 = 143$ lb.

The total recoil energy is

$$E_r = \frac{1}{2}\left(\frac{W_t + W_b}{386.4}\right)v_f^2 = \frac{60 \times 54803}{772.8} = 4255 \text{ in.-lb.}$$

The energy remaining in the moving parts is

$$E = E_r - \Delta E_t - \Delta E_b = 4255 - 143 - 166$$

$$= 3946 \text{ in.-lb.}$$

The corresponding velocity becomes

$$v = \sqrt{\frac{2E}{M_r}} = \sqrt{\frac{7892 \times 386.4}{60}} = \sqrt{50824}$$

$$= 225.4 \text{ in./sec}$$

$$v_b = \frac{M_t(v - v_t)}{R_c M_b} + v = \frac{50(225.4 - v_t)}{3.6 \times 10} + 225.4$$

$$= 538.5 - 1.39 v_t$$

$$v_b^2 = 289982 - 1497.0 v_t + 1.932 v_t^2$$

To solve for v_t, multiply all terms by g and equate the equivalent energies

$$gE = \frac{1}{2}\left(W_t v_t^2\right) + \frac{1}{2}\left(W_b v_b^2\right)$$

$$386.4 \times 3946 = 25 v_t^2 + 5(289982 - 1497 v_t + 1.932 v_t^2)$$

$$v_t^2 - 215.95\, v_t - 2158.8 = 0$$

$$v_t = -9.57 \text{ in./sec.}$$

The low negative velocity indicates a direction change near the end of the accelerating process.

$$v_b = 538.5 - 1.39 v_t = 538.5 + 13.3 = 551.8 \text{ in./sec.}$$

Compute the energy of bolt and barrel

$$E_b = \frac{1}{2}\left(M_b v_b^2\right) = \left(\frac{10}{772.8}\right)304483 = 3940 \text{ in.-lb}$$

$$E_t = \frac{1}{2}\left(M_t v_t^2\right) = \left(\frac{50}{772.8}\right)92 = 6 \text{ in.-lb}$$

$$E = E_b + E_t = 3946 \text{ in.-lb}$$

This energy compares favorably with the earlier computed energy of 3946 in.-lb thereby rendering the last computed data substantially correct.

The time of bolt acceleration period is

$$t_{ab} = \frac{2\Delta x_b}{v + v_b} = \frac{2 \times 0.56}{225.4 + 551.8} = \frac{1.12}{777.2}$$

$$= 0.0014 \text{ sec.}$$

The time of bolt decelerating period during recoil, **Eq.** 2–23, is

$$t_{rb} = \sqrt{\frac{\epsilon M_b}{K}} \operatorname{Cos}^{-1} \frac{F_e}{F_m}$$

$$= \sqrt{\frac{0.5 \times 10}{20 \times 386.4}} \cos^{-1} \frac{154}{320}$$

$$= \sqrt{0.000647} \operatorname{Cos}^{-1} 0.4812 = 0.0254 \left(\frac{}{}\right)$$

$$= 0.0271 \text{ sec.}$$

The time elapsed during recoil, which is the compression time of the barrel spring, is

$$t_{br} = t_{ab} + t_a = 0.0014 + 0.0060 = 0.0074 \text{ sec}$$

where t_a = 0.0060 sec, the propellant gas period.

The time elapsed during bolt recoil, which is the compression time of the driving spring, is

$$t_r = t_{br} + t_{rb} = 0.0074 + 0.0271 = 0.0345 \text{ sec.}$$

The time for the bolt to return as far as the latched barrel, **Eq.** 2–27, is

$$t_{crb} = \sqrt{\frac{M_b}{\epsilon K}} \operatorname{Cos}^{-1} \frac{F}{F_m}$$

$$= \sqrt{\frac{10}{0.5 \times 20 \times 386.4}} \cos^{-1} \frac{148.4}{320}$$

$$= \sqrt{0.002588} \operatorname{Cos}^{-1} 0.4638$$

$$= 0.0508\left(\frac{62.367}{57.296}\right) = 0.0553 \text{ sec}$$

where $F = F_o + K(\Sigma\Delta x + \Delta x_t) = 120 + 20 \times 1.42$ = 148.4 lb, the driving spring force as the barrel latch is released.

The energy of the bolt at this time is

$$E_b = \epsilon\left(\frac{F_m + F}{2}\right)(L - \Sigma\Delta x - \Delta x_t)$$

$$= 0.5\left(\frac{320 + 148.4}{2}\right)8.58 = 1004.7 \text{ in.-lb.}$$

The bolt velocity at barrel pick-up is

$$v_{crb} = \sqrt{\frac{2E_b}{M_b}} = \sqrt{\frac{2009.4 \times 386.4}{10}} = \sqrt{7764}$$

$$278.6 \text{ in./sec.}$$

The velocity of all counterrecoiling parts after the barrel is engaged by the bolt, according to the conservation of momentum, is

$$v_{cr} = \frac{M_b v_{crb}}{M_r} = 10\left(\frac{278.6}{60}\right) = 46.4 \text{ in./sec.}$$

With both springs acting as a unit, the combined spring constant is

$$K_s = K + K_t = 20 + 200 = 220 \text{ lb/in.}$$

The spring force at the time of impact is

$$F_s = F + F_{mt} = 148.4 + 284 = 432.4 \text{ lb.}$$

The spring force at the end of counterrecoil, since the barrel spring force reduces to zero, is

$$F_o = 120 \text{ lb.}$$

According to Eq. 2–26, the time elapsed for completing the recoiling parts return is

$$t_{cr} = \sqrt{\frac{M_r}{\epsilon K}}\left(\text{Sin}^{-1}\frac{-F_o}{Z} - \text{Sin}^{-1}\frac{-F_s}{Z}\right)$$

$$= \sqrt{\frac{60}{0.5 \times 220 \times 386.4}}\left(\text{Sin}^{-1}\frac{-120}{578} - \text{Sin}^{-1}\frac{-432.4}{578}\right)$$

$$= 0.0376\left(\frac{348.02 - 311.55}{57.3}\right) = 0.0239 \text{ sec.}$$

where

$$Z = \sqrt{F_s^2 + \frac{K}{\epsilon}\left(M_r v_{cr}^2\right)}$$

$$= \sqrt{432.4^2 + \frac{220}{0.5}\left(\frac{60}{386.4}\right) 2153}$$

$$= \sqrt{334070} = 578 \text{ lb.}$$

The time consumed for the firing cycle is

$$t_c = t_{dr} + t_{crb} + t_{cr}$$

$$= 0.0345 + 0.0553 + 0.0239 = 0.1137 \text{ sec.}$$

The rate of fire

$$f_r = \frac{b0}{0.1137} = 528 \text{ rounds/min.}$$

Recapitulating, the firing rates of the various types of recoil-operated guns are shown in the table which follows. All guns are identical except for the type of automatic action.

Type	Rate of Fire, rounds/min
Blowback	420
Long Recoil	195
Short Recoil (without bolt buffer)	272
Short Recoil (without bolt buffer)	383
Short Recoil (with accelerator)	528

3-8 RATING OF RECOIL-OPERATED GUNS

The recoil-operated machine guns are idealy suited for large caliber weapons. Their inherent low rate of fire keeps them out of the small caliber field but, for large calibers, the firing rate is relatively high and therefore acceptable. Of the two types involved, the long recoil is superior to some extent although the short recoil has a higher firing rate. Both have the same range but the long recoil is more accurate because the high loading accelerations of the short recoil gun disturb the sighting. Also the higher accelerations require heavier feeders and correspondingly heavier associated parts. The large loads imposed to accelerate these components have a tendency to cause them to wear out faster, thus decreasing the reliability and durability of the weapon.

CHAPTER 4

GAS-OPERATED WEAPONS

4–1 GENERAL REQUIREMENTS

Gas-operated automatic weapons are those weapons that have a gas driven mechanism to operate the bolt and its associated moving components. Except for the externally driven systems, all operating energy for automatic weapons is derived from the propellant gases. Nevertheless, gas-operated weapons are only those that draw a portion of the propellant gas through the barrel wall after the projectile has passed and then use this gas to activate a mechanism to retract the bolt. Timing and pressure are regulated by the location of the port along the barrel and by orifices restricting the gas flow. As soon as the projectile passes the port, propellant gases pour into the gas chamber and put pressure on the piston. The piston does not necessarily move at this time. Motion is delayed by bolt locks which are not released until chamber pressure has dropped to safe levels for cartridge case extraction.

4–2 TYPES OF GAS SYSTEMS

There are four basic types of gas systems: impingement, tappet, expansion, and cutoff expansion.

a. Impingement System: has a negligible gas volume at the cylinder; expansion depending on piston motion. As the piston moves, gas continues pouring through the port until the bullet exits at the muzzle. With the subsequent drop in pressure in the bore, the gas in the cylinder may either reverse its flow and return to the bore or it may exhaust through ports in the cylinder wall as shown in Fig. 4–1. The duration of the applied pressure is short, being dependent solely on the position of the gas port.

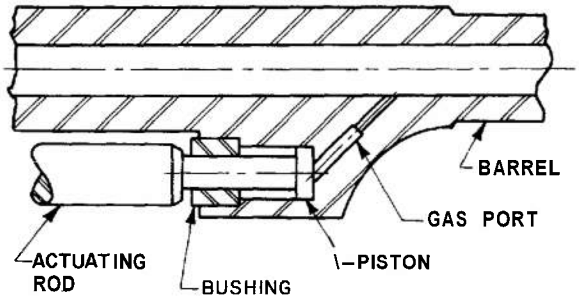


Figure 4–1. Impingement System

b. Tappet System: an impingement system having a short piston travel. (See Fig. 4–7.) The pressure force imparts a relatively high velocity to the piston which moves the operating rod and bolt. The tappet travel is short and its motion ceases as it strikes the end of its cylinder.

c. Expansion System: in contrast, has an appreciable initial volume in its expansion chamber which requires more time to pressurize the chamber, and also more time to exhaust the gas. By judicious selection of port size and location, the required pressurized gas can be drained from the bore.

d. Cutoff Expansion: similar to the direct expansion type, except for a valve which closes the port after the piston moves. As the pressure builds up to a specific value, the piston moves, closing the port and leaving the gas to expand polytropically* to provide the effort needed to operate the moving components of the bolt assembly. Fig. 4–2 shows a cutoff expansion system.

4–3 CUTOFF EXPANSION SYSTEM

4–3.1 MECHANICS OF THE SYSTEM

The final size and location of the gas port are determined by experimental firing. However, for initial design studies, tentative size and location may be computed. The acceleration of the moving parts of the gas-operated system may be expressed generally as

$$\frac{d^2s}{dt^2} = \frac{F}{M_o} = \left(\frac{A_c}{M_o}\right)p_c \qquad (4\text{–}1)$$

*Polytropic is the name given to the change of state in a gas which is represented by the general equation pV^k = constant. $k = c_p/c_v$, where c_p is the specific heat at constant pressure and c_v is the specific heat at constant volume. The specific heats vary with the temperature[27].

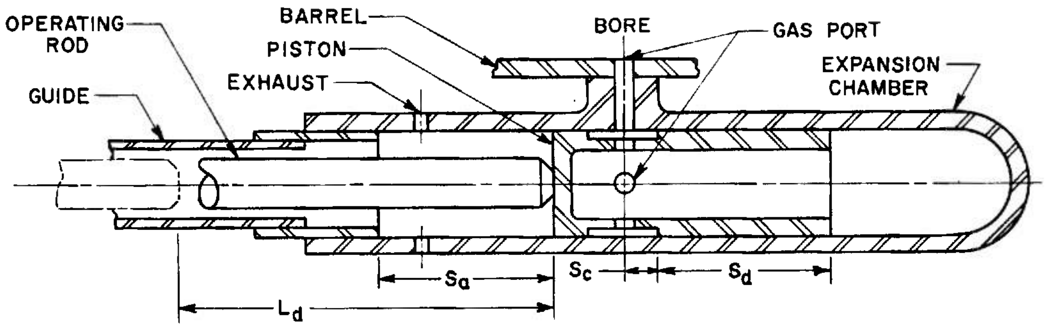


Figure 4–2. Cutoff Expansion System

where

A_c = piston area

M_o = mass of accelerating parts of operating rod

p_c = pressure in the gas cylinder at any time

The gas expands polytropically so that

$$p_c = p_1\left(\frac{V_1}{V_1 + A_c s}\right)^k = p_1\left(\frac{A_c s_o}{A_c s_o + A_c s}\right)^k$$

$$= p_1\left(\frac{s_o}{s_o + s}\right)^k \tag{4–2}$$

where

k = ratio of specific heats

p_1 = initial pressure

s = travel distance of piston

s_o = initial distance equivalent to V_1

V_1 = initial gas volume

Substitute for p in Eq. **4–1**

$$\frac{d^2 s}{dt^2} = \frac{A_c p_1 s_o^k}{M_o}\left[\frac{1}{(s_o + s)^k}\right] \tag{4–3}$$

but

$$\frac{d^2 s}{dt^2} = \frac{dv}{dt} = \left(\frac{ds}{dt}\right)\left(\frac{dv}{ds}\right) = v\left(\frac{dv}{ds}\right)$$

Substitute $v\left(\frac{dv}{ds}\right)$ for $\frac{d^2 s}{dt^2}$ and rearrange the terms in Eq. **4–3.** The expression for vdv appears in Eq. **4–4.**

$$vdv = \frac{A_c p_1 s_o^k}{M_o}\left[\frac{1}{(s_o + s)^k}\right]ds \tag{4–4}$$

Now integrate Eq. **4–4.**

$$v^2 = \frac{2A_c p_1 s_o^k}{(1 - k)M_o}\left[\frac{1}{(s_o + s)^{k-1}}\right] + C_1 \tag{4–5}$$

Solve for C_1

$$C_1 = v_o^2 + \frac{2A_c p_1 s_o}{(k-1)M_o} \tag{4-6}$$

When $s = 0$, $v = v_o$ generally, although $v_o = 0$ before the *piston* begins *to* move.

$$v = \frac{ds}{dt} = \sqrt{\frac{2A_c p_1}{(k-1)M_o}\left[s_o - \frac{s_o^k}{(s_o+s)^{k-1}}\right] + v_o^2} \tag{4-7}$$

$$\frac{ds}{dt} = \sqrt{K_a\left[\frac{1}{s_o^{k-1}} - \frac{1}{(s_o+s)^{k-1}}\right] + v_o^2} \tag{4-8}$$

where

$$K_a = \frac{2A_c p_1 s_o^k}{(k-1)M_o}$$

$$dt = \sqrt{\frac{s_o^{k-1}(s_o+s)^{k-1}}{K_a\left[(s_o+s)^{k-1} - s_o^{k-1}\right] + v_o^2 s_o^{k-1}(s_o+s)^{k-1}}}\, ds$$

$$= s_o^{(k-1)/2}\sqrt{\frac{\left(1+\frac{s}{s_o}\right)^{k-1}}{K_a\left[\left(1+\frac{s}{s_o}\right)^{k-1} - 1\right] + v_o^2 s_o^{k-1}\left(1+\frac{s}{s_o}\right)^{k-1}}}\, ds \tag{4-9}$$

Let $a = 1 + \frac{s}{s_o}$; $ds = s_o da$

$$dt = s_o^{(k+1)/2}\sqrt{\frac{a^{k-1}}{K_a(a^{k-1}-1) + v_o^2 s_o^{k-1} a^{k-1}}}\, da$$

$$dt = \frac{s_o^{(k+1)/2}}{\sqrt{K_a}} \frac{da}{\sqrt{1 - a^{1-k} + v_o^2 s_o^{k-1}/K_a}} \tag{4-10}$$

Let $B = 1 + \left(\dfrac{v_o^2}{K_a}\right) s_o^{k-1}$

$$dt = \frac{s_o^{(k+1)/2}}{\sqrt{K_a}} \left[B\left(1 - \frac{a^{1-k}}{B}\right)\right]^{-1/2} da \qquad (4\text{–}11)$$

Expand according to the series $(1 - x)^{-1/2}$

$$(1 - x)^{-1/2} = 1 + \frac{1}{2}x + \frac{1}{2}\cdot\frac{3}{4}x^2 + \frac{1}{2}\cdot\frac{3}{4}\cdot\frac{5}{6}x^3$$

$$+ \frac{1}{2}\cdot\frac{3}{4}\cdot\frac{5}{6}\cdot\frac{7}{8}x^4 + \ldots. = A + A_1x + A_2x^2 + A_3x^3 + A_4x^4 + \ldots. \qquad (4\text{–}12)$$

where A_1, A_2, A_3, etc., are coefficients of **x** of the expanded series.

Now let $a^{1-k} = x$

$$dt = \frac{s_o^{(k+1)/2}}{\sqrt{K_a}} \left[1 + A_1 \frac{a^{(1-k)}}{B} + A_2 \frac{a^{2(1-k)}}{B^2} + A_3 \frac{a^{3(1-k)}}{B^3} + A_4 \frac{a^{4(1-k)}}{B^4} + \ldots\right] da \qquad (4\text{–}13)$$

Now integrate

$$t = \frac{s_o^{(k+1)/2}}{\sqrt{K_a}} \left[a + A_1 \frac{a^{2-k}}{B(2-k)} + A_2 \frac{a^{3-2k}}{B^2(3-2k)} + A3 \frac{a^{4-3k}}{B^3(4-3k)} + A4 \frac{a^{5-4k}}{B^4(5-4k)} + \ldots\right] + C_2 \qquad (4\text{–}14)$$

when $t = 0$, s = 0 and a = 1.0, thus

$$C_2 = -\frac{s_o^{(k+1)/2}}{\sqrt{K_a}} \left[1 + \frac{A_1}{(2-k)B} + \frac{A_2}{(3-2k)B^2} + \frac{A_3}{(4-3k)B^3} + \frac{A_4}{(5-4k)B^4} + \ldots\right] \qquad (4\text{–}15)$$

Before continuing with the analysis of the mechanics of the gas-operated system, several initial parameters must be established such as the characteristics of driving springs and buffers, time of recoil and counterrecoil, and port sizes and location. Since springs or their equivalent store energy for counterrecoil, they must be given the working capacity for reloading the gun and returning the recoiled parts in the prescribed time. Since the efficiency of the spring system involving recoil activity is relatively low in automatic guns, counterrecoil must necessarily take longer than recoil. Practice sets the preliminary estimate of counterrecoil time as

$$t_{cr} = 1.5t_r = 0.60t_c = \frac{36}{f_r} \text{ sec} \qquad (4\text{–}16)$$

where

f_r = firing rate, rounds/min

t_c = time of firing cycle, sec

t_r = time of recoil, sec

Driving and buffer springs, whether single springs or nests of two or more, are generally installed in series. The driving spring is the softer of the two and, during recoil, seats before the buffer springs begin to move. Its fully compressed load is less than the initial spring load of the buffers. Deflections are consistent with type, the driving spring has a large deflection whereas the much stiffer buffer springs deflect approximately **15%** of the total recoil travel. In counterrecoil, the buffer springs complete their action first, then the driving spring continues the accelerating effort the rest of the way. Another proportion which must be considered during the preliminary design characteristics involves the ratio of counterrecoil energy contributed by the buffer springs to the total counterrecoil energy. A practical value is

$$\frac{E_{bc}}{E_{cr}} = 0.40. \qquad (4\text{–}17)$$

The counterrecoil velocity at the end of buffer spring action is computed from the kinetic energy equation.

$$v_{bc} = \sqrt{\frac{2 \times 0.4E_{cr}}{M_r}} \qquad (4\text{–}18)$$

The velocity at the end of counterrecoil is computed similarly.

$$v_{cr} = \sqrt{\frac{2E_{cr}}{M_r}} \qquad (4\text{–}19)$$

Now

$$\frac{v_{bc}}{v_{cr}} = \sqrt{0.4} \text{ or } v_{bc} = 0.632v_{cr}. \qquad (4\text{–}20)$$

Based on average velocities, the time required to negotiate the buffer spring travel becomes

$$t_{bc} = \frac{2L_b}{v_{bc}} = \frac{3.16L_b}{v_{cr}} \qquad (4\text{–}21)$$

where

L_b = travel of buffer spring.

The time required for the remaining counterrecoil travel is

$$t_{rs} = \frac{2L_d}{v_{bc} + v_{cr}} = \frac{1.225L_d}{v_{cr}} \qquad (4\text{–}22)$$

where

L_d = driving spring travel

The total time of counterrecoil is approximately t_{cr}.

$$t_{cr} = t_{bc} + t_{rs} = \frac{3.16\,L_b + 1.225\,L_d}{v_{cr}} \qquad (4\text{–}23)$$

Since t_{cr} is know, **Eq. 4–16,** the approximate maximum counterrecoil velocity is

$$v_{cr} = \frac{3.16L_b + 1.225L_d}{t_{cr}} \qquad (4\text{–}24)$$

Sufficient data are now known so that, via Eq. **2–26,** practical values of characteristics for the various springs can be estimated. Spring constants may vary greatly among those in the whole system but the loads should be reasonably proportioned. The minimum load on the driving spring should be 3 to **4** times the weight of the recoiling parts. The minimum load of the buffer springs in series should be equal and in turn should be about twice the fully compressed load of the driving spring.

4–3.1.1 Gas Filling Period

The time must also be computed during the period while the initial volume of the operating cylinder is being filled with propellant gas. The operating piston and rod begin to move as soon as the gas force is sufficient to overcome friction. According to the equation of gas flow through an orifice, the rate w of gas flow is [7]

$$w = CA_o p \sqrt{\left(\frac{g}{RT}\right) k \left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}\ , \text{lb/sec} \tag{4–25}$$

where

A_o = orifice area, in.^{2}

C = orifice coefficient

g = **386.4** in./sec^2, acceleration of gravity

k = ratio of specific heats

p = pressure in reservoir, lb/in.^{2}

RT = specific impetus, ft-lb/lb

Eq. **4–25** is valid as long as the discharge pressure does not exceed the critical flow pressure. For gases, the critical flow pressure p_{cr} is

$$p_{cr} = 0.53p. \tag{4–26}$$

All the values of Eq. **4–25** are usually known except for orifice area and pressure so the equation may be simplified by substituting K_w for the product of C times the term under the radical of Eq. **4–25,** i.e.,

$$w = K_w A_o p_a \tag{4–27}$$

where the pressure now becomes p_a, the average pressure over a time interval. The time intervals should be short in duration so that the spread of values is small between average pressure and the maximum and minimum values, thereby minimizing the errors introduced by pressure variations over the interval. The weight ΔW_c of gas flowing into the operating cylinders during the time interval Δt is

$$\Delta W_c = w\Delta t \tag{4–28}$$

Pressures are read from the pressure-time curve, the initial pressure being that corresponding to the position of the orifice in the barrel. A tentative location of the orifice may correspond approximately with bullet location when approximately 50 to 70 percent of its time in the barrel has elapsed. Pressures in the operating cylinder may be adjusted by increasing the orifice area for higher pressures or decreasing it for lower pressures. If higher pressures are needed but a larger orifice is not deemed wise, these higher pressures are available by locating the orifice nearer to the breech. Conversely, lower pressures are available by shifting the orifice toward the muzzle. In both alternatives, these latter effects follow the prescription only if the orifice size remains unchanged.

There are so many variables in this exercise that some values must be assigned arbitrarily in order to approach a practical solution. An initial close approximation of these assigned values saves considerable time, which emphasizes the value of experience. By the time that the gas operating mechanism is being considered, good estimates should have been made on the weights of the bolt and its related moving parts and on the travel distance of these components. With the help of these estimates and with Eqs. **4–16** through **4–24,** the early characteristics of counterrecoil are determined to serve as data for determining the recoil characteristics. As the first step, let the accelerating distance of the gas piston be indicated by s_a.

$$s_a = 0.3(L_d + L_b) \tag{4–29}$$

The velocity of recoil at the end of the gas piston stroke is a function of the counterrecoil velocity at the same bolt position. The two velocities are related by the efficiency of the spring system. The counterrecoil velocity at this position is determined from **Eq. 2–24,** thus

$$v_{cr} = \sqrt{v_{bc}^2 + \frac{F_m (L_d - s_a) - \frac{1}{2} K(L_d - s_a)^2}{M_r}} \tag{4–30}$$

At any given position, the energy remaining to be absorbed by the spring may be expressed generally as ϵE_r

$$\epsilon E_r = \frac{\epsilon}{2}\left(M_r v_r^2\right) = \left(\frac{F_m + F_o}{2}\right) L . \tag{4–31}$$

At the same position during counterrecoil, the energy released by the spring is indicated by **Eq. 4–32.**

$$E_{cr} = \frac{1}{2}\left(M_r v_{cr}^2\right) = \epsilon \left(\frac{F_m + F_o}{2}\right) L \tag{4–32}$$

Solve for $\frac{F_m + F_o}{2} L$ in **Eq. 4–32** and substitute the appropriate terms in **Eq. 4–31** to obtain **Eqs. 4–33** and **4–34.**

$$\frac{\epsilon}{2}\left(M_r v_r^2\right) = \frac{1}{2\epsilon}\left(M_r v_{cr}^2\right) \tag{4–33}$$

$$v_r = \frac{v_{cr}}{\epsilon} \tag{4–34}$$

The work done during a polytropic expansion of a gas is defined in **Eq. 4–35.**

$$W = \frac{p_1 V_1 - p_2 V_2}{k - 1} \tag{4–35}$$

where

k = ratio of specific heats

p_1 = initial pressure

p_2 = final pressure

V_1 = initial volume of gas cylinder

V_2 = final volume of gas cylinder

Since

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^k$$

Eq. 4–35 may be written as

$$W = p_1 \left[\frac{V_1 - \left(\frac{V_1}{V_2}\right)^k V_2}{k - 1}\right] \tag{4–36}$$

By assigning a value to the ratio $\frac{V_2}{V_1}$ so that $V_2 = \left(\frac{V_2}{V_1}\right) V_1$, **Eq. 4–36** may be written

$$W = \frac{p_1 V_1}{k - 1} \left[1 - \left(\frac{V_1}{V_2}\right)^{k-1}\right] . \tag{4–37}$$

Experience has indicated that $k = 1.3$ and that the ratio $\frac{V_1}{V_2} = \frac{3}{5}$ offers a practical beginning in the design study. Substitute these values in **Eq. 4–37** to express the work done by the expanding gas so that

$$W = 0.473 p_1 V_1 . \tag{4–38}$$

This work is equal to the kinetic energy of the recoiling mass. By substitution, the expression for the energy in Eq. 4–38 becomes

$$\frac{1}{2}\left(M_r v_r^2\right) = 0.473 p_1 V_1 \qquad (4\text{–}39)$$

$$p_1 V_1 = 1.06 M_r v_r^2 . \qquad (4\text{–}40)$$

Neither p_1 or V_1 are known but gas volume and corresponding dimensions must be compatible with the dimensions of the gun. The initial pressure must be low enough to assure its attainability and still perform according to time limits. An initial pressure in the neighborhood of 1000 to 1500 psi is appropriate.

On the assumption that the preliminary design is completed to the extent that tentative sizes have been established and the gas port in the barrel located, the pressure in the operating cylinder becomes the primary concern. Note that before the bullet passes the port, the gas operating cylinder is empty. As soon as the bullet passes the port, gases pour into the cylinder and, when the pressure becomes high enough, the piston begins to move. However, a finite time is required, however small, for the gas to fill the cylinder. Also, the pressure in the bore is rapidly diminishing. For this reason, the pressure in the operating cylinder does not have sufficient time to reach bore pressure before cut off when the port is closed. The gas pressure in the operating cylinder is found by establishing a relationship between the gas weight in the cylinder and the total propellant gas weight, and between the cylinder volume and the effective volume of the bore. The effective volume assumes the barrel to be extended beyond the muzzle to correspond with pressure decay. V_b is the effective bore volume after the bullet leaves the barrel.

$$V_b = V_m \left(\frac{p_m}{p_a}\right)^{\frac{1}{k_b}} \qquad (4\text{–}41)$$

where

k_b = ratio of specific heats of propellant gas in bore, usually considered to be 1.2 as compared to 1.3 in operating cylinder since the two locations have different temperatures. See footnote, par. 4–3.1.

p_a = average propellant gas pressure

p_m = propellant gas pressure as bullet leaves muzzle

V_m = chamber volume plus total bore volume

The equivalent gas volume in the cylinder at p_a is shown as

$$V_e = \left(\frac{W_c}{W_g}\right) V_b \qquad (4\text{–}42)$$

where

W_e = EAW_c ,weight of gas in the cylinder

W_g = total weight of propellant gas

The gas pressure in the cylinder becomes

$$p_c = \left(\frac{V_e}{V_c}\right)^k p_a \qquad (4\text{–}43)$$

where

k = ratio of specific heats in operating cylinder

p_a = average bore pressure over a time increment

V_c = cylinder volume

The gas force applied to the piston in the operating cylinder

$$F_c = A_c p_c \qquad (4\text{–}44)$$

The corresponding impulse $F_c \Delta t$ is obtained by multiplying the gas force by the differential time, *At.* Since momentum and impulse are the same dimensionally, the change in velocity of the recoiling unit during any given increment becomes *Av.*

$$Av = \frac{F_c \Delta t}{M_r} \qquad (4\text{–}45)$$

where

M_r = mass of the recoiling unit

The velocity v at the end of any given increment is the summation of the *Av* 's.

$$v = \Sigma \Delta v = v_{n-1} + Av \qquad (4\text{–}46)$$

The distance s traveled by the operating unit at any given time of the s_i

$$s = s_1 + s_2 + s_{n-1} = \frac{1}{2}(\Delta v \Delta t) + v_{n-1} \Delta t + s_{n-1} \qquad (4\text{–}47)$$

and the corresponding gas volume in the cylinder is

$$V_c = V_{co} + A_c s. \qquad (4\text{–}48)$$

The entire computing procedure to arrange the dynamics of the recoiling system so that these data are compatible with counterrecoiling data is iterative and depends on the logical selection of initial values to hold the quantity of exploration to a minimum. Experience is the best guide in this respect but if lacking, definite trends in earlier computations should soon lead to the proper choice.

The initial orifice area is found by first solving for the weight of gas in the operating cylinder at cutoff and then, on the basis of average values, computing the area by **Eq. 4–27.** When the known or available values are substituted for the unkown in **Eqs. 4–41, 4–42, 4–43,** the solution for the weight of the gas becomes

$$W_c = W_g \left(\frac{V_c}{V_m}\right)\left(\frac{p_c^{1/k}}{p_m^{1/k_b}}\right) p_a^{(1/k_b - 1/k)} \qquad (4\text{–}49)$$

Select an initial value of the cylinder pressure p_c. Assume that it is the critical flow pressure, thereby fixing the average bore pressure p_a(p in **Eq. 4–26).** Locate the pressure and the corresponding time on the pressure-time curve. Now measure the area under the pressure-time curve between the limits of the above time and when the bullet passes the gas port. The area divided by p_a gives the time needed to operate at the average pressure.

$$t = \frac{A_{pt}}{p_a} \qquad (4\text{–}50)$$

where A_{pt} = area under pressure-time curve.

The rate of flow is now stated as

$$w = \frac{W_c}{t} \qquad (4\text{–}51)$$

The orifice area A, may now be computed from Eq. 4–27.

$$A_o = \frac{w}{K_w p_a} = \frac{W_c}{K_w A_{pt}} \qquad (4\text{–}52)$$

4–3.1.2 Bolt Locking Cam

The cam that controls the locking and unlocking of the bolt is arranged for the bolt to be released completely only after the propellant gas pressure in the chamber is no longer dangerous and, conversely, completely locks the bolt before the chambered round

is fired. In a gas-operated machine gun such as the 7.62 mm, M60, the operating rod moves a short distance before bolt pickup. During this traverse, about half its axial length, the cam has a shallow constant slope to insure a small angular bolt travel and thereby exposes only the high strength end of the case at its rim. The cam then follows a parabolic curve to complete the unlocking process. A bolt having two locking lugs generally turns about sixty degrees (Fig. 4–3).

To simplify the dynamics of cam operation, all components are assumed to be rigid so that transfer of momentum or energy from translation to rotation is made without considering the elasticity of the system. Therefore, as soon as the cam follower moves under the influence of the operating rod, the bolt immediately assumes the angular kinetics of the moving system. Linear velocity converted to angular velocity becomes

$$\omega = \left(\frac{v_o}{R_c}\right) \tan\beta, \text{ rad/unit time} \qquad (4\text{–}53)$$

where

R_c = cam radius

v_o = operating rod velocity immediately preceding cam action

β = cam rise angle

Apply the law of the conservation of momentum, thus

$$M_o v_o = M_o v + I_b \frac{\omega}{R_c}$$

$$= \left[M_o + M_b\left(\frac{k^2}{R_c^2}\right)\tan\beta\right] v \qquad (4\text{–}54)$$

where

I_b = mass moment of inertia of bolt

k = bolt polar radius of gyration

M_b = mass of bolt

M_o = mass of operating rod assembly

v = axial linear velocity

Fig. 4–4 shows the accelerating force system involved in the cam dynamics. All forces and reactions are derived from the cam normal force and from the geometry of the bolt components.

F_L = transverse force on locking lug; reaction on bolt

N = cam normal force

N_a = axial force on cam and locking lug

N_c = tangential force on cam; reaction on bolt

N_s = reaction normal force

R = bolt radius

R_c = cam radius

R_L = radius of locking lug pressure center

λ = locking lug helix angle

μ_r = coefficient of rolling friction of cam roller

μ_s = coefficient of sliding friction

ω = angular velocity

From the geometry, the axial component of the normal cam force and the cam friction is

$$N_a = N(\sin\beta + \mu_r \cos\beta) \qquad (4\text{–}55)$$

Since the lug cam is analogous to a screw, the other forces and reactions are obtained by resolving the static forces accordingly. Let N_s be the reaction normal to the helix of the lug and $\mu_s N_s$, the frictional resistance. The axial component of these two forces is

$$N_a = N_s(\cos\lambda + \mu_s \sin\beta). \qquad (4\text{–}56)$$

The inertia force of the operating rod and the axial cam force must balance the accelerating force, thus

$$F_c = M_o a + N(\sin\beta + \mu_r \cos\beta) \qquad (4\text{–}57)$$

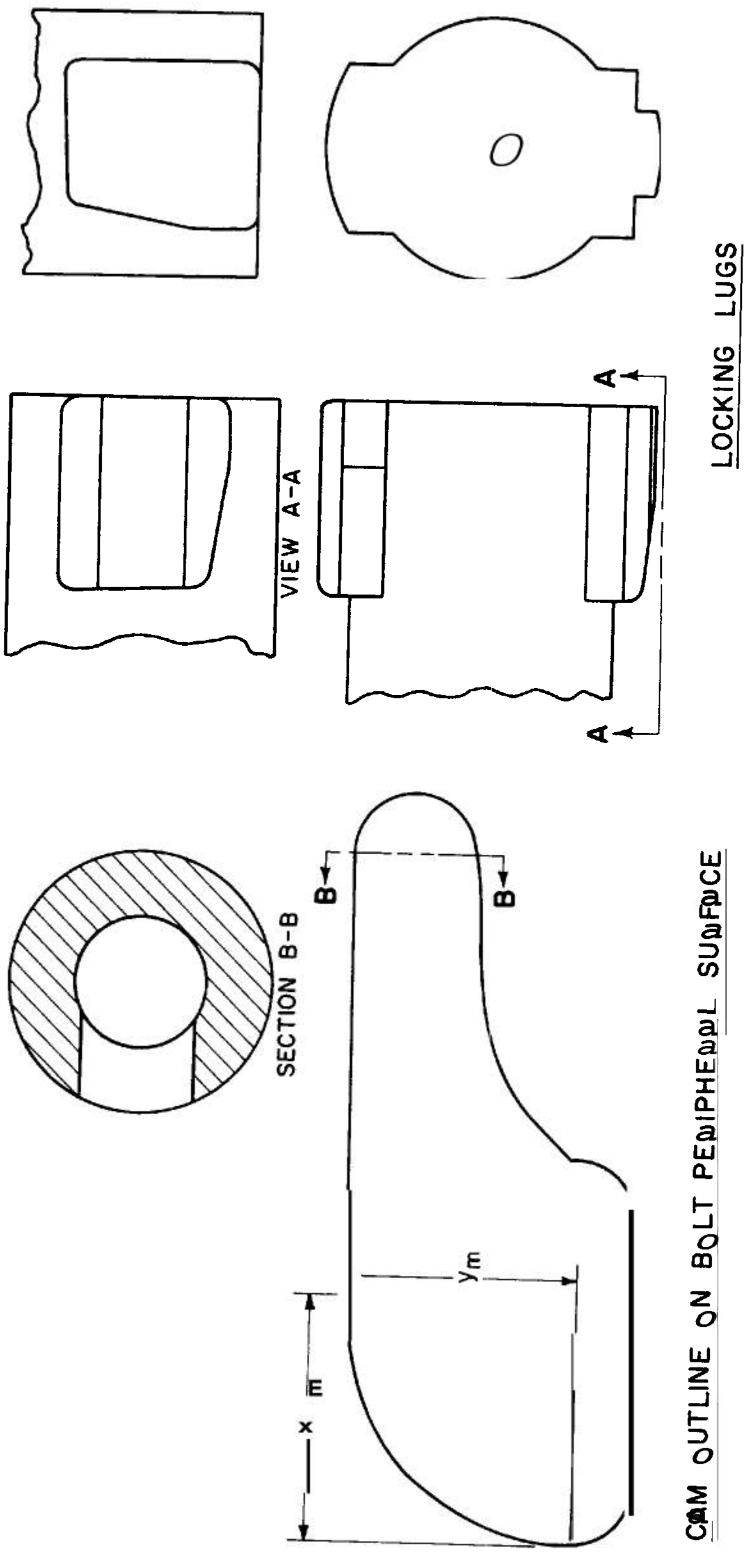


Figure 4–3. Rotating Bolt Lock and Activating Cam

where a = linear acceleration of locking lug.

The applied torque N_cR_c of the transverse cam force must balance the rotational inertia of the bolt, plus the induced torque of the lug, plus the two components of frictional resistance of the bolt.

$$N_cR_c = I_b\alpha - F_LR_L + \mu_sN_cR + \mu_sF_LR \tag{4–58}$$

where

I_b = mass moment of inertia of bolt

N_c = $N(\cos\beta - \mu_r \sin\beta)$

F_L = $N_s(\sin\lambda - \mu_s\cos\lambda)$

$I_b\alpha$ = $M_bk^2\left(\dfrac{a}{R_c}\right)\tan\beta$

a = angular acceleration of bolt

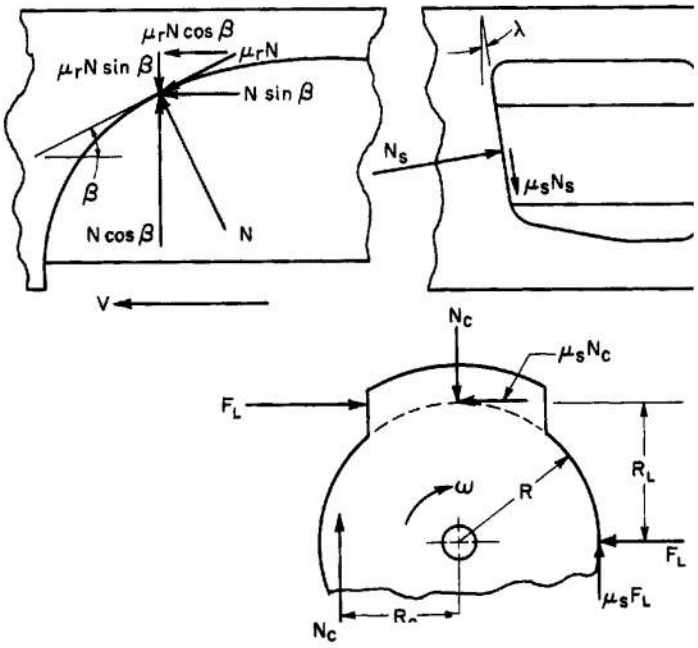


Figure 4–4. Force System of Bolt Cam

After these expressions are substituted into Eq. 4–58 and the terms collected, Eq. 4–59 may be compiled.

$$N(\cos\beta - \mu_r\sin\beta)\,(R_c - \mu_sR) = M_b\,\frac{k^2}{R_c}\,a\tan\beta - N_s\,(\sin\lambda - \mu_s\cos\lambda)\,(R_L - \mu_sR) \tag{4–59}$$

Solve for N_s in terms of N via Eqs. 4–55 and 4–56.

$$N_s = N\left(\frac{\sin\beta + \mu_r\cos\beta}{\cos\lambda + \mu_s\sin\lambda}\right) \tag{4–60}$$

Substitute for N_s in **Eq. 4–59,** collect terms and solve for N.

$$N = M_b a \frac{k^2 \tan\beta}{R_c \left[(R_c - \mu_s R)(\cos\beta - \mu_r \sin\beta) + C_\lambda (R_L - \mu_s R)(\sin\beta + \mu_r \cos\beta)\right]} \tag{4–61}$$

where

$$C_\lambda = \frac{\sin\lambda - \mu_s \cos\lambda}{\cosh + \mu_s \sin\lambda}$$

Now substitute for N in **Eq. 4–57** so that

$$F = a \left\{ M_o + M_b \frac{k^2 \tan\beta}{R_c \left[\dfrac{(R_c - \mu_s R)(\cos\beta - \mu_r \sin\beta)}{\sin\beta + \mu_r \cos\beta} + C_\lambda (R_L - \mu_s R)\right]} \right\} \tag{4–62}$$

For unlocking, F is the gas pressure force on the operating rod. It is the driving spring force during locking. The expression in the braces of Eq. **4–62** is equivalent to an effective mass M_e. Eq. **4–62** may now be written as

$$F_c = M_e a \tag{4–63}$$

$$a = \frac{F}{M_e} \tag{4–64}$$

During the unlocking period, the operating rod derives its accelerating force from the propellant gases which, after cutoff, expand polytropically. The pressure in the operating cylinder at a given position of the rod **is** computed according to Eq. **4–2** and the force according to Eq. **4-44.**

$$p, = p_1 \left(\frac{s_o}{s_o + s}\right)^k \quad \text{(from Eq. 4–2)}$$

$$F_c = A_c p_c \quad \text{(from } Eq. \text{ 4-44)}$$

Both p_c and F_c may be found by assigning small increments to the operating rod travel s.

$$s = s_{n-1} + A s \tag{4–65}$$

The average acceleration over the increment is

$$a_a = \frac{1}{2}(a_{n-1} + a_n) \tag{4–66}$$

The velocity at the end of each increment is obtained by first computing the energy E_o of the operating rod

$$E_o = E_{oi} + \Delta E_o = E_{o(n-1)} + \Delta E_o$$

$$= E_{o(n-1)} + \left[\frac{F_{c(n-1)} + F_{c(n)}}{2}\right] \Delta s \tag{4–67}$$

where E_{oi} = energy of operating rod at gas cutoff

then $$v = \sqrt{\frac{2E_o}{M_o}} \,. \tag{4–68}$$

The incremental time may be computed from the expression

$$\Delta s = v_{n-1}\,\Delta t_e + \frac{1}{2}\left(a_a \Delta t_e^2\right). \tag{4–69}$$

Solving for Δt_e, the incremental time of the gas expansion stroke becomes

$$\Delta t_e = -\frac{v_{n-1}}{a_a} \pm \sqrt{\left(\frac{v_{n-1}}{a_a}\right)^2 + \frac{2\Delta s}{a_a}} \tag{4–70}$$

The time for the total gas expansion stroke is t_e.

$$t_e = \Sigma \Delta t_e \tag{4–71}$$

During the locking period, the applied force is derived from the driving spring. The spring accelerating forces at any position of the operating rod when efficiency is considered is

$$F = \epsilon(F_i + Ks) = F_{n-1} + \epsilon K \Delta s \tag{4–72}$$

where

F_i = initial driving spring force as bolt seats

F_{n-1} = spring accelerating force of previous increment

K = spring constant

s = operating rod travel after bolt seats

Δs = incremental operating rod travel

ϵ = efficiency of the spring system

The breech end of the locking lugs carries the axial reaction on the bolt, thereby relieving the helix end of all loads during cam action. Therefore, in Eq. **4–62**, λ = **0**, and the effective mass becomes

$$M_e = M_o + M_b \left\{ \frac{k^2 \tan\beta}{R_c \left[\frac{(R_c - \mu_s R)(\cos\beta - \mu_r \sin\beta)}{\sin\beta + \mu_r \cos\beta} - \mu_s (R_L - \mu_s R) \right]} \right\} \tag{4–73}$$

Note that if the cam roller is not installed for either locking or unlocking, the Coefficient of rolling friction μ_r changes to μ_s the coefficient of sliding friction in Eqs. 4–62 and 4–73.

4-3.1.3 Cam Curve

The cam curves on the breech end of the bolt are helices, usually having identical slopes. The straight slope merges smoothly with the parabolic curve which may be expressed as

$$y = Ax^2 + Bx + C. \tag{4–74}$$

Locate the coordinate axes so that $y = 0$ when $x = 0$, thus $C = 0$ which reduces the equation to

$y = Ax^2$ [redacted] (4–75)

The slope of the curve is defined as

$$\frac{dy}{dx} = 2Ax + B = \tan\beta \tag{4–76}$$

when $x = 0$; $\beta_o = \frac{dy}{dx} = \tan\beta_o$, and $B = \tan\beta_o$, the slope of the helix and therefore the slope of the parabola where it joins the helix. When x and y reach their respective limits, dimensions that have been assigned and then substituted in **Eq.** 4–75 will yield the value of the coefficient A.

$$A = \frac{y_m - Bx_m}{x_m^2} \tag{4–77}$$

where

x_m = axial length of the parabola (see Fig. 4–3)

y_m = peripheral width of the parabola (see Fig. 4–3)

From Eq. 4–62, the equivalent **mass** for the unlocking system is a constant when the cam action involves the helix. The values assigned to the parameters in the equation are:

k = 0.275 in., radius of gyration

R = 0.39 in., bolt radius

R_c = 0.32 in., cam radius

μ_r = 0.034 coefficient of rolling friction

μ_s = 0.30 coefficient of sliding friction

$\tan\beta_o$ = 0.007465, slope of cam helix

R_L = 0.5 in., radius of locking lug

W_b = 0.75 lb, bolt weight

W_o = 2.5 lb, operating rod weight

$\cos\beta_o$ = 0.999972

$\sin\beta_o$ = 0.0074648

λ = 3°, slope of lug helix

$$M_e = \frac{1}{386.4}\left| W_o + W_b \frac{k^2 \tan\beta_o}{R_c\left[\dfrac{(R_c - \mu_s R)(\cos\beta_o - \mu_r \sin\beta_o)}{\sin\beta_o + \mu_r \cos\beta_o} + C_\lambda (R_L - \mu_s R)\right]}\right|$$

$$= \frac{1}{386.4}\left\{2.5 + 0.75\left[\frac{0.0756 \times 0.007465}{0.32\,(0.203 \times 24.12 - 0.244 \times 0.383)}\right]\right\}$$

where

$$\frac{\cos\beta_o - \mu_r \sin\beta_o}{\sin\beta_o + \mu_r \cos\beta_o} = \frac{0.99997 - 0.00025}{0.00746 + 0.0340} = 24.12$$

$$C_\lambda = \frac{\sinh - \mu_s \cosh}{\cosh + \mu_s \sin\lambda} = \frac{0.05234 - 0.29959}{0.99863 + 0.01570} = -0.244$$

$$M_e = \frac{2.5 + 0.75 \times 1.18 \times 10^{-4}}{386.4} = \frac{2.50001}{386.4} = 0.00647 \text{ lb-sec}^2/\text{in.}$$

The effective mass adds less than one percent to the operating rod mass as the cam follower progresses along the constant cam slope hence it need not be considered in the calculations until the parabolic curvature is reached. Because the entire bolt mass without modification was entered in the analysis of the period before gas cutoff, this analysis may be considered conservative. The effective mass while the cam follower negotiates the parabolic curvature of the cam for the unlocking process becomes (Eqs. 4–63 and 4–62)

$$M_e = 6.47 \times 10^{-3} + \frac{4.58 \times 10^{-4} \tan\beta}{0.203\left(\dfrac{\cos\beta - 0.034 \sin\beta}{\sin\beta + 0.034 \cos\beta}\right) - 0.093}$$

The effective mass for the locking process is obtained from Eq. 4–73.

$$M_e = 6.47 \times 10^{-3} + \frac{4.58 \times 10^{-4} \tan\beta}{0.203\left(\dfrac{\cos\beta - 0.034 \sin\beta}{\sin\beta + 0.034 \cos\beta}\right) - 0.115}$$

The straight slope of the cam is half the axial cam travel distance

$$x_c = \frac{1}{2} s_r = 0.5 \text{ in.}$$

$$x_m = s_r - s = 0.5 \text{ in., axial length of parabola}$$

According to **Eq.** 4–76, $B = \tan \beta_o = 0.007465$. The bolt must turn through an angle of 60° to unlock. The peripheral width of the parabola (see Fig 4–3) is

$$y_m = \left(\frac{60}{57.3} - B\right) R_c = 1.0397 \times 0.32 = 0.3327 \text{ in.}$$

From **Eq.** 4–77, the constant A is

$$A = \frac{y_m - Bx_m}{x_m^2} = \frac{0.3327 - 0.007465 \times 0.5}{0.25}$$

$$= 1.316/\text{in.}$$

The slope of the curve, **Eq.** 4–76, is

$$\tan\beta = \frac{dy}{dx} = 2.632x + 0.007465.$$

4–3.2 SAMPLE PROBLEM FOR CUTOFF EXPANSION SYSTEM

4–3.2.1 Specifications

Gun: 7.62 mm machine gun

Firing Rate: 1000 rounds/min

Interior Ballistics: Pressure vs Time, Fig. 4–5
Velocity vs Time, Fig. 4–5

4–3.2.2 Design Data, Computed

The time for the firing cycle, from **Eq.** 2–29, is

$$t_c = \frac{60}{f_r} = \frac{60}{1000} = 0.060 \text{ sec.}$$

The counterrecoil time t_{cr}, Eq. 4–16, is

$$t_{cr} = 0.60 t_c = 0.036 \text{ sec.}$$

The recoil time

$$t_r = t_c - t_{cr} = 0.060 - 0.036 = 0.024 \text{ sec.}$$

Fig. 4–6 shows a sketch of the various distances involved in the operation of the moving mechanisms during the firing cycle. These initial distances are based on those of an earlier gun.

d_c = 0.874 in., gas cylinder diameter

d_p = 0.135 in., gas port diameter

L_b = 1.00 in., buffer stroke

L_{b1} = 0.75 in., operating distance of primary buffer spring

L_{b2} = 0.25 in., operating distance of secondary buffer spring

L_d = **5.5** in., operating distance of operating rod spring

s_a = 2.0 in., accelerating distance of gas piston

s_b = 4.5 in., distance of bolt retraction during recoil.

s_c = 0.20 in., cutoff distance

s_d = 1.8 in., dwell distance

s_r = 1.00 in., operating rod travel before bolt pickup

V_o = 1.8 in.^{3}, initial gas volume of operating cylinder

W_b = 0.75 lb, weight of bolt

W_o = 2.5 lb, weight of moving operating mechanism

W_r = $W_b + W_o$ = 3.25 lb, weight of recoiling parts

ϵ = 0.50, efficiency of spring system

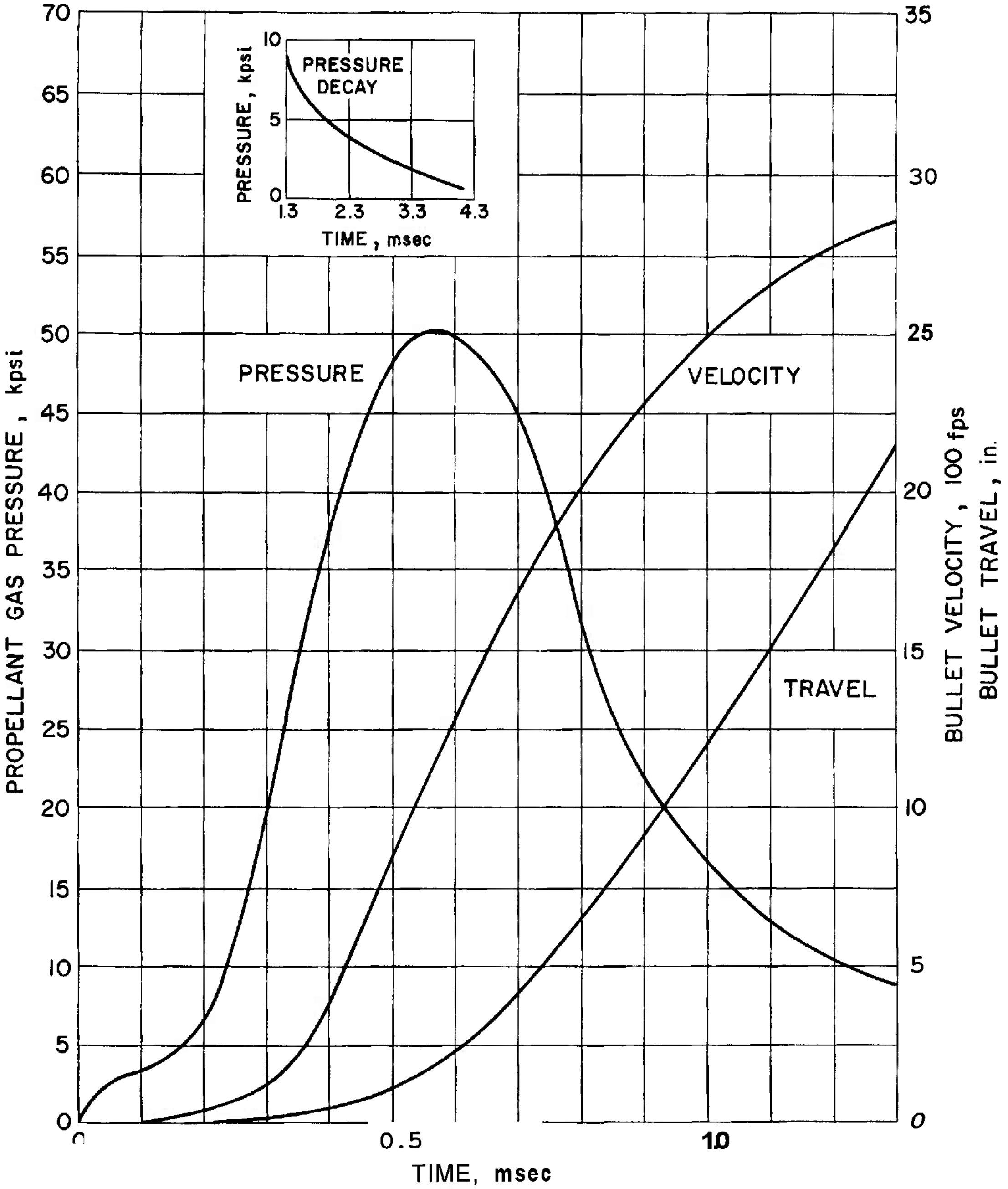


Figure 4–5. Pressure-time Curve of 7.62 mm Round

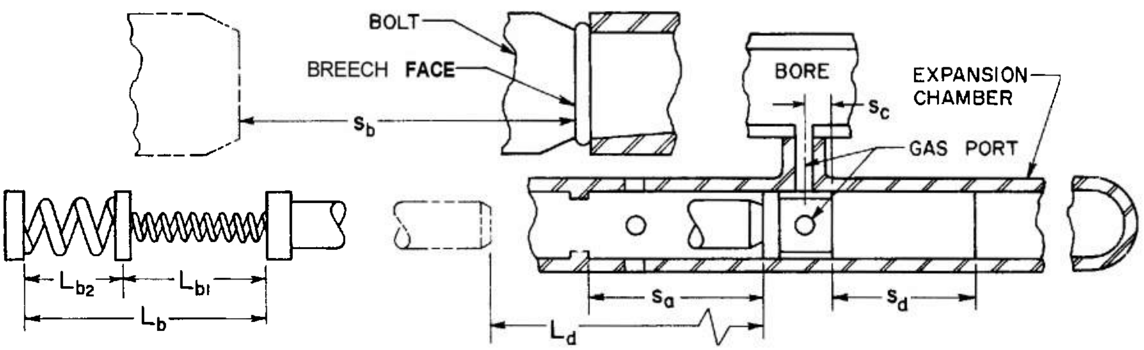


Figure 4–6. Operating Distances of Moving Parts

4–3.2.3 Counterrecoil Computed Data

From Eq. 4–24 the approximate maximum counterrecoil velocity is computed to be

$$v_{cr} = \frac{3.16 L_b + 1.228 L_d}{t_{cr}} = \frac{3.16 + 6.75}{0.036}$$

$$= 275 \text{ in./sec.}$$

From Eq. 4–20 the approximate counterrecoil velocity at the end of buffer spring action becomes

$$v_{bc} = 0.632\, v_{cr} = 174 \text{ in./sec.}$$

According to Eq. 2–24, by substituting F_a for $\left(F_m - \frac{1}{2} Kx\right)$ and L_d for **x**, the expression $L_d F_a$ is

$$L_d F_a = \frac{1}{\epsilon} \left[\frac{1}{2}\left(M_r v_{cr}^2\right) - \frac{1}{2}\left(M_r v_{bc}^2\right)\right]$$

where v_{bc} is equivalent to v_o of Eq. 2–24.

F_a can now be determined.

$$M_r = \frac{W_r}{g}$$

$g \quad = 386.4$ in./sec^2, acceleration of gravity

$L_d \quad = 5.5$ in., operating distance of spring

$$5.5 F_a = \frac{1}{0.5}\left(\frac{1}{2}\right)\left(\frac{3.25}{386.4}\right)(75625 - 30276)$$

$$F_a = 69.3 \text{ lb}$$

For the first trial a spring constant of 4 lb/in. was selected but proved to be too highly stressed. However, the first trial indicated a compression time of 16.6 msec. With an impact velocity above 25 ft/sec, the spring surge time is

$$T = \frac{T_c}{1.8} = \frac{0.0166}{1.8} = 0.0092 \text{ msec,}$$

According to Eq. 2–67b,

$$KT = \frac{F_m}{1037} = \frac{F_a + \frac{1}{2}\left(L_d K\right)}{1037}$$

$$0.0092K = \frac{69.3 + 2.75K}{1037}$$

$$K = 10.2 \text{ lb/in.}$$

$$F_m = F_a + \frac{1}{2}\left(KL_d\right) = 69.3 + 28.1 = 97.4 \text{ lb}$$

$$F_o = F_m - KL_d = 97.4 - 56.1 = 41.3 \textbf{ lb.}$$

For the buffer springs in series, the spring constant K_b is

$$K_b = 60 \text{ lb/in.}$$

$$L_b = 1.0 \text{ in.}$$

$$\frac{1}{2}\left(M_r v_{bc}^2\right) = \frac{1}{2}\left(F_{mb}L_b\right) - \frac{1}{4}\left(K_b L_b^2\right)$$

where $v_o = 0$, at the start of counterrecoil

$$\frac{1}{2} F_{mb} = \frac{1}{2}\left(\frac{3.25}{386.4}\right) 30{,}276 + \frac{1}{4} (60) \quad 1.0$$

$$F_{mb} = 2(127.3 + 15) = 284.6 \text{ lb, say, } 285 \text{ lb}$$

$$F_{ob} = F_m - K_b L_b = 285 - 60 = 225 \text{ lb.}$$

Since the buffer springs in series should have the same terminal loads, spring constants will vary inversely as their deflections. Thus,

$$\frac{K_1}{K_2} = \frac{L_{b2}}{L_{b1}} = \frac{0.25}{0.75} \text{ or } K_2 = 3K_1$$

The spring constants of the primary and secondary springs of the buffer are

$$K_1 = \frac{F_{mb} - F_{ob}}{L_{b1}} = \frac{60}{0.75} = 80 \text{ lb/in.}$$

$$K_2 = 3K_1 = 240 \text{ lb/in.}$$

4-3.2.4 Counterrecoil Time

The counterrecoil time is divided into two periods, i.e., during buffer spring action and during driving spring action. Both are computed according to **Eq.** 2-26. For the buffer springs, $v_o = 0$ therefore

$$t_{bc} = \sqrt{\frac{M_r}{\epsilon K_b}}\left[\text{Sin}^{-1}\left(\frac{K_b L_b - F_{mb}}{F_{mb}}\right) - \frac{3\pi}{2}\right]$$

$$t_{bc} = \sqrt{\frac{2 \times 3.25}{386.4 \times 60}}\left[\text{Sin}^{-1}\left(\frac{1.0 \times 60 - 285}{285}\right) - \frac{3\pi}{2}\right]$$

$$= 0.01675\ [\text{Sin}^{-1}\ (-0.78947) - 4.71241$$

$$= 0.01675\ (5.3733 - 4.7124) = 0.0111 \text{ sec.}$$

From **Eq.** 2-24, the buffer counterrecoil velocity is expressed in terms of spring energy.

$$v_{bc}^2 = \frac{F_{mb}L_b - \frac{1}{2}\left(K_b L_b^2\right)}{M_r}$$

$$= \frac{(285 \times 1.0 - 30 \times 1.0)386.4}{3.25}$$

$$v_{bc}^2 = \frac{255 \times 386.4}{3.25} = 30318 \text{ in.}^2/\text{sec}^2$$

$$v_{bc} = 174.1 \text{ in./sec, buffer counterrecoil velocity}$$

which checks favorably with the first computed velocity of 174 in./sec

The time consumed for counterrecoil during driving spring action also is divided into two periods, when the total recoiling mass is considered, and after the bolt stops when the moving mass consists only of the moving parts of the operating rod unit. According to **Eq.** 2-26, the first time component is

$$t'_{cr} = \sqrt{\frac{M_r}{\epsilon K}}\left[\mathrm{Sin}^{-1}\left(\frac{Ks_b - F_m}{\sqrt{F_m^2 + 2KM_r v_{bc}^2}}\right) - \mathrm{Sin}^{-1}\left(\frac{-F_m}{\sqrt{F_m^2 + 2KM_r v_{bc}^2}}\right)\right]$$

$$= \sqrt{\frac{3.25}{386.4 \times 0.5 \times 10.2}}\left[\mathrm{Sin}^{-1}\ \frac{10.2 \times 4.5 - 97.4}{\sqrt{9487 + 5202}} - \mathrm{Sin}^{-1}\ \frac{-97.4}{\sqrt{9487 + 5202}}\right]$$

$$= 0.04061\left[\mathrm{Sin}^{-1}\ (-0.4249) - \mathrm{Sin}^{-1}\ (-0.8036)\right.$$

$$= 0.04061\ (334.85 - 306.52)/57.3 = 0.04061 \times 0.4944 = 0.0201 \text{ sec}$$

where

F_m	= 97.4 lb	s_b	= 4.5 in.
K	= 10.2 lb/in.	v_{bc}^2	= 30,318 in.^{2}/sec^2
M_r	= $\frac{3.25}{386.4}$ lb-sec^2/in.	ϵ	= 0.50

According to **Eq. 2–24,** the expression for the counterrecoil velocity is

$$v_{cr}^2 = v_{bc}^2 + \frac{2\epsilon\left[(F_m s_b - \frac{1}{2}\left(Ks_b^2\right)\right]}{M_r}$$

$$= 30318 + \frac{97.4 \times 4.5 - 10.2 \times 20.25/2}{3.25}\ 386.4$$

$$= 30318 + 39832 = 70150 \text{ in.}^2/\text{sec}^2$$

$$v_{cr} = 264.9 \text{ in./sec} = 22.08 \text{ ft/sec, maximum bolt velocity on return.}$$

The terminal part of the counterrecoil stroke occurs after the bolt is in battery and the operating rod components are the remaining moving parts. From **Eq. 2–26,** the second time component is

$$t''_{cr} = \sqrt{\frac{M_o}{\epsilon K}}\left[\mathrm{Sin}^{-1}\ \frac{K(L_d - s_b) - F_m}{\sqrt{F_m^2 + 2KM_o v_o^2}} - \mathrm{Sin}^{-1}\ \frac{-F_m}{\sqrt{F_m^2 + 2KM_o v_o^2}}\right]$$

$$t''_{cr} = \sqrt{\frac{2.5}{386.4 \times 0.5 \times 10.2}} \left[\mathrm{Sin}^{-1} \frac{-41.3}{\sqrt{2652 + 9258}} - \mathrm{Sin}^{-1} \frac{-51.5}{\sqrt{2652 + 9258}} \right]$$

$$= 0.03562 \left[\mathrm{Sin}^{-1} (-0.3784) - \mathrm{Sin}^{-1} (-0.4719) \right]$$

$$= 0.03562\,(337.77 - 331.85)/57.3 = 0.0037 \text{ sec}$$

where

$F_m = 51.5$ lb	$M_o = \frac{2.5}{386.4}$ lb-sec^2/in.
$F_o = 41.3$ lb	$s_b = 4.5$ in.
$K = 10.2$ lb/in.	$v_o^2 = 70147$ in.^{2}/sec^2
$L_d = 5.5$ in.	$\epsilon = 0.50$

The total counterrecoil time is

$$t_{cr} = t_{bc} + t'_{cr} + t''_{cr} = 0.0349 \text{ sec.}$$

This computed time compares favorably with the assigned time (0.036 sec) for counterrecoil thus supporting the characteristics of the selected spring.

At the end of counterrecoil the velocity of the operating rod moving parts, according to Eq. 2-24 is

$$v_{cr} = \sqrt{v_o^2 + 2\epsilon \left[F_m (L_d - s_b) - \frac{1}{2} K (L_d - s_b)^2 \right] / M_o}$$

$$= \sqrt{70147 + (51.5 \times 1.0 - 10.2 \times 1.0/2)386.4/2.5}$$

$$= \sqrt{77318} = 278 \text{ in./sec} = 23.08 \text{ ft/sec} = 277 \text{ in./sec}$$

which checks with the first computed velocity of 275 in./sec.

4–3.2.5 Recoil Time

The dynamics of the operating rod and bolt while propellant gases are active in the gas cylinder do not include the resistance of the driving spring since the spring forces are negligible as compared to gas forces. With respect to the dynamics, two periods are involved, i. e., accelerating and decelerating. Initially, the time of each is proportioned according to their respective distance of operation. The decelerating time is computed to be

$$t_d = \left(\frac{L_d + L_b - s_a}{L_d + L_b}\right) t_r = \left(\frac{5.5 + 1.0 - 2.0}{5.5 + 1.0}\right) 0.024$$

$$= 0.0166 \text{ sec}$$

where $t_r = 0.024$ sec (see par. 4–3.2.2) and

where s_a = 2.0 in., the distance that the piston moves while being accelerated by the propellant gas.

The counterrecoil velocity at this position is computed according to Eq. 2–24

$$v_{cr} = \sqrt{v_{bc}^2 + 2\epsilon\left[F_m(L_d - s_a) - \frac{1}{2}K(L_d - s_a)^2\right]/M_r}$$

$$= \sqrt{30318 + (97.4 \times 3.5 - 10.2 \times 12.25/2)386.4/3.25}$$

$$= 251.8 \text{ in./sec.}$$

From Eq. 4–34, the velocity of recoil at this position becomes

$$v_r = \frac{1}{\epsilon} v_{cr} - \frac{251.8}{0.5} = 503.6 \text{ in./sec.}$$

4–3.2.5.1 Recoil Time, Decelerating

As computed above, the recoil velocity at the end of the operating cylinder stroke is 503.6 in./sec which becomes the initial velocity of the spring system as deceleration begins. At this time the driving spring has been compressed to the extent of the two inches that the operating gas piston has traveled. The design data include

F_{sa} = $41.3 + Ks_a = 41.3 + 20.4 = 61.7$ lb

K = 10.2 lb/in., spring constant

M_r = 3.25/g, lb-sec^2 in.

s_a = 2.0 in.

v_o = 503.6 in./sec, initial velocity

ϵ = 0.5, efficiency of spring system

The working distance of the drive spring during the decelerating period of recoil is L_r.

$$L_r = L_d - s_a = 5.5 - 2.0 = 3.5 \text{ in.}$$

Expand Eq. 2–22 to include the limits of $x = 0$ to $x = L_r$, thus the time to compress the drive spring is

$$t_{dr} = \sqrt{\frac{\epsilon M_r}{K}}\left(\sin^{-1}\frac{F_{sa} + KL_r}{\sqrt{F_{sa}^2 + \epsilon K M_r v_o^2}} - \sin^{-1}\frac{F_{sa}}{\sqrt{F^2 + \epsilon K M_r v_o^2}}\right)$$

$$= \sqrt{0.0004123}\left(\sin^{-1}\frac{97.4}{121.2} - \sin^{-1}\frac{61.7}{121.2}\right)$$

$$= 0.0081 \text{ sec.}$$

The velocity of the recoiling parts as the buffer is contacted is

$$v_b^2 = v_o^2 - \frac{(2F_{sa} + KL_r)L_r}{\epsilon M_r}$$

$$= 253613 - \frac{159.1 \times 3.5 \times 386.4}{0.50 \times 3.25}$$

$$v_b = \sqrt{253613 - 132410} = \sqrt{121203} = 348.1 \text{ in./sec.}$$

Since the buffer absorbs the remaining energy,

$$\sqrt{F_{ob}^2 + \epsilon K_b M_r v_b^2} = F_{mb},$$

therefore, Eq. 2–23 becomes the expression for recoil time during buffing,

$$t_b = \sqrt{\frac{\epsilon M_r}{K}}\ \mathrm{Cos}^{-1}\frac{F_{ob}}{F_{mb}} = \sqrt{\frac{0.5 \times 3.25}{60 \times 386.4}}\ \mathrm{Cos}^{-1}\frac{225}{295}$$

$$= 0.00837\ \mathit{Cos}^{-1}\ 0.78947 = 0.00837\left(\frac{37.89}{57.3}\right)$$

$$= 0.0055 \text{ sec.}$$

The total recoil time during deceleration

$$t_r' = t_{dr} + t_b = 0.0081 + 0.0055 = 0.0136 \text{ sec.}$$

4–3.2.5.2 Recoil Time, Accelerating

The time needed to accelerate the recoiling mass consists of two parts, i. e., time before cutoff and time after cutoff. First, the tentative size of the operating cylinder must be determined. According to Eq. 4–40, the work done by the expanding gas is

$$p_1 V_1 = 1.06 M_r v_r^2 = 1.06\left(\frac{3.25}{386.4}\right) 253613$$

$$= 2261 \text{ in.-lb}$$

Select $p_1 = 1250$ lb/in.^{2}

$$V_1 = \frac{p_1 V_1}{p_1} = \frac{2261}{1250} = 1.8 \text{ in.}^3$$

With the ratio $\frac{V_1}{V_2} = \frac{3}{5}$, the volume at the end of the gas expansion stroke is V_2.

$$V_2 = \frac{5}{3} V_1 = \left(\frac{5}{3}\right) 1.8 = 3.0 \text{ in.}^3$$

The selected cylinder pressure is $p_c = p_1 = 1250$ psi. Assume this pressure to be the critical flow pressure. Therefore, the average pressure in the barrel needed to provide it is, according to Eq. 4–26,

$$p_a = \frac{p_c}{0.53} = 2360 \text{ lb/in.}^2$$

Other values are

k = 1.3, ratio of specific heats in operating cylinder

k_b = 1.2, ratio of specific heats in bore

p_m = 9000 lb/in.^{2}, muzzle pressure from Fig. 4–5

V_{co} = V_1 = 1.8 in.^{3}, initial volume of operating cylinder

V_m = 1.74 in.^{3}, chamber volume plus bore volume

W_g = 0.00629 lb, weight of propellant

The weight of the gas in the operating cylinder at its maximum pressure, according to Eq. 4-49, is

$$W_c = W_g\left(\frac{V_{co}}{V_m}\right) p_c^{1/k} p_a^{(1/k_b - 1/k)} / p_m^{1/k_b}$$

$$= 0.00629\left(\frac{1.8}{1.74}\right)\left(\frac{241 \times 1.66}{1970}\right) = 0.00132 \text{ lb.}$$

To insure total access, the diameter of the orifice should not exceed the cutoff distance; thus the two orifices of 0.162 in. diameter.

The measured area under the pressure-time curve between 0.0002 sec and 0.0032 sec is equivalent to $A_{pt} = 16.43$ lb-sec/in.^{2} The first time (0.0002 sec) represents approximately 70% of the bullet travel and the second corresponds with the pressure, $p_a = 2360$ psi. From Eq. 4–52, the first estimate of the orifice area is

$$A_o = \frac{W_c}{K_w A_{pt}} \quad \frac{0.00132}{0.00192 \times 16.43} = 0.042 \text{ in.}^2$$

where, in Eq. 4–25, the expression for the term K_w is

$$K_w = C\sqrt{\frac{g}{RT}\, k\left(\frac{2}{k+1}\right)^{(k+1)/(k-1)}}$$

$= 0.00192$/sec

C = 0.30, orifice coefficient

g = 32.2 ft/sec^2

k = 1.3

RT = 350,000 ft-lb/lb, specific impetus

With A, = 0.042 in.^{2} detailed iterative computations are performed in Table **4–1.** These calculations indicate a cutoff at $s = 0.20$ in. of piston travel. Also, an orifice area of 0.042 in.^{2} seems sufficient. **A** piston area of 0.60 in.^{2} is suitable. Note that in the listed values of the cylinder pressure p_c never exceeds the critical flow pressure of $p_{cr} = 0.53\ p_a$ except for the last increment. Here the computed pressure of 1134 psi via Eq. 4–43 is considered reasonable inasmuch as the pressure of polytropic expansion during the interval drops only to 1105 psi and the continued flow from the barrel should be enough so that the cylinder pressure will approach the 1134 psi.

TABLE 4–1. COMPUTED DYNAMICS OF GAS CUTOFF SYSTEM

$t \times 10^4$, sec	p_a, psi	A_o, in?	w, lb/sec	$\Delta W_c \times 10^4$, lb	$W_c \times 10^4$, lb	V_b, in.^{3}	V_e, in.^{3}	V_c, in.^{3}
9	26000	0.042	2.096	2.096	2.096	0.837	0.0278	1.8000
10	19000	0.042	1.531	1.531	3.627	0.942	0.0543	1.8002
11	14800	0.042	1.193	1.193	4.820	1.147	0.0879	1.8003
12	11400	0.042	0.919	0.919	5.739	1.366	0.1246	1.8006
13	9600	0.042	0.774	0.774	6.513	1.600	0.1657	1.8011
18	6600	0.042	0.532	2.660	9.173	2.255	0.329	1.8080
23	4500	0.042	0.363	1.815	10.988	3.120	0.545	1.8236
28	3200	0.037	0.227	1.135	12.123	4.130	0.796	1.850
33	2400	0.025	0.115	0.575	12.698	5.240	1.058	1.888
36.31	1800	0.007	0.024	0.088	12.786	6.650	1.350	1.920

$t \times 10^4$, sec	p_c, psi	F_c, lb	$F_c\Delta t$, lb-sec	Δv, in./sec	v, in./sec	$s_1 \times 10^4$, in.	$s_2 \times 10^4$, in.	$s \times 10^4$, in.
9	114	69	0.0069	0.82	0.82	0	0	0
10	202	121	0.0121	1.44	2.26	1	1	2
11	294	176	0.0176	2.09	4.35	1	2	5
12	354	212	0.0212	2.52	6.87	1	4	10
13	425	255	0.0255	3.03	9.90	2	7	19
18	717	430	0.2150	25.56	35.46	64	50	133
23	934	560	0.280	33.29	68.75	83	177	393
28	1069	641	0.321	38.17	106.9	95	344	832
33	1128	677	0.339	40.31	147.2	101	535	1468
36.31	1134	680	0.225	26.75	174.0	44	487	1999

At cutoff, the velocity of the moving parts has reached 174 in./sec. This represents a kinetic energy of

$$E_c = \frac{1}{2}\left(Mv^2\right) = \frac{1}{2}\left(\frac{3.25}{386.4}\right) 30276 = 127.3 \text{ in.-lb.}$$

The work done by polytropic expansion, Eq. 4–35, is

$$W = \frac{p_1 V_1 - p_2 V_2}{k - 1} = \frac{1134 \times 1.92 - 635 \times 3.0}{1.3 - 1.0}$$

$$= 907.6 \text{ in.-lb}$$

where

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^k = 1134 \left(\frac{1.92}{3}\right)^{1.3}$$

$$= 1134 \times 0.56 = 635 \text{ lb/in.}$$

$$V_1 = V_{co} + A_c s = 1.8 + 0.6 \times 0.2 = 1.92 \text{ in.}^3$$

The total work done by the operating cylinder is

$$E, = E, + W = 127.3 + 907.6 = 1034.9 \text{ in.-lb}$$

The velocity of the system at the end of the accelerating period becomes

$$v_r = \sqrt{\frac{2E_r}{M_r}} = \sqrt{\frac{2069.8 \quad 386.4}{3.25}} = 496.07 \text{ in./sec.}$$

This velocity compares favorably with the required velocity of 503.6 in./sec (par. 4–3.2.5) and represents an error of only 1.5%. A slight increase in orifice area or location will match the two velocities, but the small error involved does not warrant further computation.

Although the weight of recoiling parts does not become 3.25 lb until the bolt is picked up after 1 inch of travel, the bolt weight is included to compenstate for whatever losses are experienced during the accelerating stroke.

The analysis which follows for the time, t = 2.8 x 10^{-3} sec, illustrates the mechanics for computing the results listed in Table 4–1. The increment of time At = 0.0005 sec and the corresponding average pressure are read from the pressure decay-time curve (see insert) of Fig. 4–5, p_a = 3200 psi. Resorting to Eq. 4–57 to estimate the travel during the interval, observe that Av is unknown. However, in the preceding interval, the difference in Av

$$[Av \text{ for } (t = 2.3 \times 10^{-3})] - [Av \text{ for } (t = 1.8 \times 10^{-3})]$$

< 8 in./sec. Add this increment to Av (t = 2.3 x 10^{-3}). Thus, Av = 33 + 7 = 40 in./sec.

The rate of flow by Eq. 4–27 is computed to be

$$w = K_w A_o p_a = 0.00192 \times 0.037 \times 3200$$

$$= 0.227 \text{ lb/sec.}$$

The weight of the gas flowing through the orifice during the 0.0005 sec increment according to Eq. 4–28 is

$$\Delta W_c = w\Delta t = 0.227 \times 5 \times 10^{-4} = 1.135 \times 10^{-4} \text{ lb.}$$

The total weight of the gas in the cylinder

$$W_c = \Sigma \Delta W_c = W_{c(n-1)} + \Delta W_c$$

$$= (10.988 + 1.135)10^{-4} = 0.0012123 \text{ lb.}$$

By first trial, the distance traveled by the operating unit (Eq. 4–47) is

$$s = \frac{1}{2} \Delta v \Delta t + v_{n-1} \Delta t + s_{n-1}$$

$$= \frac{40}{2} (0.0005) + 68.75 \times 0.0005 + 0.0393$$

$$= 0.0837 \text{ in.}$$

From Eq. 4–48, the cylinder volume

$$V_c = V_{co} + A_c s = 1.8 + 0.6 \times 0.0837 = 1.850 \text{ in.}^3$$

According to Eq. 4–41,

$$V_b = V_m \left(\frac{p_m}{p_a}\right)^{1/k_b} = 1.74 \left(\frac{9000}{3200}\right)^{1/1.2}$$

$$= 1.74 \times 2.37 = 4.13 \text{ in.}^3$$

From Eq. 4–42, the equivalent gas volume in the operating cylinder at the average bore pressure p_a becomes

$$V_e = \left(\frac{W_c}{W_g}\right) V_b = \left(\frac{0.0012123}{0.00629}\right) 4.13 = 0.796 \text{ in.}^3$$

From Eq. 4–43, the gas pressure in the operating cylinder is

$$p_c = \left(\frac{V_e}{V_c}\right)^k p_a = \left(\frac{\;}{\;}\right)^{1.3} 3200$$

$$= 0.334 \times 3200 = 1069 \text{ lb/in.}^2$$

The gas force applied to the operating rod, Eq. 4–44, is

$$F_c = A_c p_c = 0.6 \times 1069 = 641 \text{ lb.}$$

The impulse during the interval is

$$F_c \Delta t = 641 \times 0.0005 = 0.321 \text{ lb-sec.}$$

The velocity at the end of the time interval, Eq. 4–46, is

$$v = v_{n-1} + \frac{F_c At}{M_r} = 68.75 + \frac{0.321 \times 386.4}{3.25}$$

$$= 68.75 + 38.17 = 106.9 \text{ in./sec.}$$

The distance traveled by the operating unit, Eq. 4–47, is

$$s = \frac{1}{2}\left(\Delta v \Delta t\right) + v_{n-1} At + s_{n-1}$$

$$= \left(\frac{38.17}{2} + 68.75\right) 0.0005 + 0.0393$$

$$= 0.0095 + 0.0344 + 0.0393 = 0.0832 \text{ in.}$$

From Eq. 4–48, the chamber volume is

$$V_c = V_{co} + A_c s = 1.8 + 0.6 \times 0.0832 = 1.850 \text{ in.}^3$$

This volume matches the earlier estimate thereby completing this series of calculations. Had the volumes not checked, a new change in volume would be investigated and the series of calculations repeated.

The time elapsed between firing and complete cutoff is the final figure (36.31) in the time column of Table 4–1.

$$t_{co} = 0.0036 \text{ sec.}$$

The time of the remaining accelerating period is determined by Eqs. 4–14 and 4–15. The known data follow:

$A,$ = 0.60 in.^{2}, area of operating piston

k = 1.3, ratio of specific heats

$M_r = \dfrac{3.25}{386.4}\ \dfrac{\text{lb-sec}^2}{\text{in.}}$, mass of recoiling unit

p_1 = 1134 lb/in.^{2}, initial pressure

s = 1.8 in., total travel of piston

s_o = 3.2 in., equivalent travel distance at p_1

v_o = 174 in./sec, initial velocity

Substitute the expression for C_2, Eq. 4–15, in Eq. 4–14, and rewrite, for convenience, the time for the gas expansion stroke

$$t_e = \frac{s_o^{(k+1)/2}}{\sqrt{K_a}} \left[\left(a + \Sigma A_y \frac{a^z}{zB^y} \right) - \left(1 + \Sigma \frac{A_y}{zB^y} \right) \right]$$

$$t_e = \frac{3.81}{1563} (4.658 - 2.763) = 0.0046 \text{ sec}$$

where

$$a = 1 + \frac{s}{s_o} = 1 + \frac{1.8}{3.2} = 1.562$$

$$K_a = \frac{2A_c p_1 s_o^k}{(k-1)M_r} = \frac{2 \times 0.6 \times 1134 \times 4.53 \times 386.4}{(1.3-1)\,3.25}$$

$$= 2{,}443{,}000 \text{ in.}^2/\text{sec}^2$$

$$\sqrt{K_a} = 1{,}563 \text{ in./sec}$$

$$B = 1 + \frac{v_o^2}{K_a} s_o^{k-1} = 1 + \frac{30276 \times 1.418}{2443000} = 1.018$$

$$s_o^{(k+1)/2} = 3.2^{1.15} = 3.81$$

$$y = 1, 2, 3 \ldots\ldots$$

$$z = (y+1) - yk = 1.0 - 0.3y$$

$$\Sigma A_y \frac{a^z}{zB^y} = 3.0959 \text{ (summation of last column in Table 4–2)}$$

$$\Sigma \frac{A_y}{zB^y} = 1.763 \text{ (summation of next to last column in Table 4–2)}$$

The detailed calculations are tabulated for convenience in Table 4–2.

The total recoil time is

$$t_r = t_{co} + t_e + t_r'$$

$$= 0.0036 + 0.0046 + 0.0136 = 0.0218 \text{ sec.}$$

This is only 2.2 msec less that the specified 24 msec (par. 4–3.2.1) and, therefore, is acceptable. The time for the firing cycle is

$$t_c = t_{cr} + t_r = 0.0349 + 0.0218 = 0.0567 \text{ sec.}$$

The rate of fire is

$$f_r = \frac{60}{t_c} = 1058 \text{ rounds/min.}$$

This rate is only 5.8% over the specified rate of 1000 (par. 4–3.2.1) and is acceptable. Firing tests will determine the accuracy of the theoretical rate and any undesirable discrepancies are to be corrected in compliance with the test data.

4–3.3 DIGITAL COMPUTER ROUTINE FOR CUTOFF EXPANSION

This digital computer program is compiled in FORTRAN IV language for the UNIVAC 1107 Computer. The program considers the dynamics of the gas, the bolt operating cam, the bolt, and the operating rod. The specified and computed data are the same as those for the sample problem of par. 4–3.2. The program also follows the same sequence of computations except for the inclusions of the bolt unlocking and locking processes. Each set of computations is discussed in sequential order.

4–3.3.1 Gas Dynamics Before Cutoff

This analysis follows the same procedure as that for the preceding example of Table 4–1 except that the bolt is being turned by the helix portion of the cam. Bolt frictional and rotational inertia forces are considered by substituting the effective mass of Eq. 4–62 for the actual mass of the operating rod. At gas cutoff, the operating rod velocity is set for approximately 350 in./sec and to reach this velocity, the gas port is assigned an initial area of 0.40 in.^{2} If the computed velocity at gas cutoff exceeds ±10 in./sec, the port area is adjusted accordingly and the

TABLE 4–2. GAS EXPANSION TIME CALCULATIONS

Y	A_y	B^y	z	a^z	zB^y	A_y/zB^y	A_ya^z/zB^y
1	0.5000	1.018	0.7	1.366	0.713	0.7013	0.9580
2	0.3750	1.036	0.4	1.195	0.414	0.9058	1.0824
3	0.3125	1.055	0.1	1.045	0.106	2.9481	3.0808
4	0.2734	1.074	-0.2	111.093	-0.215	-1.2716	-1.1634
5	0.2461	1.093	-0.5	1/1.25	-0.547	-0.4490	-0.3599
6	0.2256	1.113	-0.8	111.428	-0.890	-0.2535	-0.1775
7	0.2095	1.134	-1.1	1/1.634	-1.247	-0.1680	-0.1028
8	0.1964	1.154	-1.4	1/1.868	-1.615	-0.1216	-0.0651
9	0.1855	1.175	-1.7	112.14	-1.998	-0.0928	-0.0434
10	0.1762	1.196	-2.0	1/2.44	-2.392	-0.0736	-0.0302
11	0.1682	1.217	-2.3	1/2.79	-2.799	-0.0601	-0.0215
12	0.1612	1.238	-2.6	1/3.19	-3.219	-0.0501	-0.0157
13	0.1550	1.261	-2.9	1/3.65	-3.657	-0.0424	-0.0116
14	0.1495	1.284	-3.2	1/4.17	-4.109	-0.0364	-0.0087
15	0.1445	1.307	-3.5	1/4.76	-4.574	-0.0316	-0.0066
16	0.1400	1.331	-3.8	1/5.45	-5.058	-0.0277	-0.0051
17	0.1359	1.355	-4.1	1/6.22	-5.556	-0.0245	-0.0039
18	0.1321	1.380	-4.4	1/7.11	-6.072	-0.0218	-0.0030
19	0.1286	1.404	-4.7	118.12	-6.599	-0.0195	-0.0024
20	0.1254	1.429	-5.0	119.30	-7.145	-0.0176	-0.0019
21	0.1224	1.455	-5.3	1110.65	-7.712	-0.0159	-0.0015
22	0.1196	1.481	-5.6	1/12.2	-8.294	-0.0144	-0.0011
						Σ = 1.7631	Σ = 3.0959

computation is repeated. Operating rod travel is also computed and should not exceed the axial length of the cam helix. Table 4–5 lists the computed data.

4–3.3.2 Gas Dynamics After Cutoff

4–3.3.2.1 Bolt Unlocking During Helix Traverse

If a portion of the helix remains to be traversed by the cam follower, the time is computed according to Eqs. 4–6 through 4–15. This procedure was also used to compute the values of Table 4–2. The velocity at the end of the helix is obtained from Eq. 4–7. Although the bolt moves axially over the small distance permitted by the locking lug, the effects of this distance and corresponding velocity have a negligible effect on the dynamics of the system and, therefore, are not included in the analysis. Computed data are listed in Table 4–6.

4–3.3.2.2 Bolt Unlocking During Parabola Traverse

The major part of the unlocking process is done by the parabolic portion of the cam (see Figs. 4–3 and 4–4). For the same reason as for the helix analysis, only bolt rotational effects are considered. The axial length of the parabola is divided into short equal increments. Cam curve characteristics are then determined and these are integrated with the rest of the analysis. The equivalent mass of the system is found from Eq. 4–73. The cam constants and variables are determined elsewhere (see par. 4–3.1).

TABLE 4–3. SYMBOL-CODE CORRELATION FOR CUTOFF EXPANSION

Symbol	Code	Symbol	Code	Symbol	Code
A_o	AO	p_a	PA	V_c	VC
A_c	AC	p_c	PC	V_{co}	VCYL
A_y	AY	p_1	P1	V_e	VE
a	A	R	R	v	V
a^z	AZ	R_c	RC	v_{bc}	VBCR
B^y	BY	R_L	RL	v_m	VMAX
C_λ	CLAMDA	s	S	v_o	VO
dx	DX	s_a	SA	v_{sc}	VSCR
E_b	EB	s_c	SCYL	Av	DELV
	EBCR	s_m	SMAX	W_b	WB
E_{cr}	E		SO	W_c	WC
E_r	ER	$s_r - x_m$	HELIX1	ΔW_c	DELW
E_{sc}	ESCR		S(1)	W_o	WO
F	F		Si	w	W
F_{mb}	FBM	s_2	S_2	x	X
F_{ob}	FBO	t	TEP	x_m	HELIX2
AF	DELF	t_{bc}	TBRC	y	Y
g	G	t_{br}	TBR	z	Z
K	DRK	t_{cr}	TDCR	β	B
K_a	SKA	t_{dr}	TDR	ϵ	EPS
K_b	BK		T1	X	DLAMDA
k	RADGYR	At	DELT	μ_r	EMUR
L_b	BL	$\tan\beta_o$	TANBO	μ_s	EMUS
M_e	EM	V	VB		

x_m = 0.5 in., axial length of parabola (Eq. 4–77)

y_m = 0.3327 in., peripheral width of parabola (Eq. 4–77)

A = 1.3161 in. (Eq. 4–77)

B = 0.007465 in. (Eq. 4–77)

$y = Ax^2 + Bx = 1.316x^2 + 0.007465x$ (Eq. 4–75)

$\frac{dy}{dx} = 2Ax + B = 2.632x + 0.007465$ (Eq. 4–76)

$\tan\beta = \frac{dy}{dx}$

At any given position of x,

$$x = \Sigma\Delta x = x_{n-1} + \Delta x$$

the corresponding length of the gas cylinder is

$$s = s_o + x = 3.5 + x \text{ ,in.}$$

The operating cylinder pressure, according to Eq. 4–2, becomes

$$p_c = p_1\left(\frac{s_o}{s}\right)^k = p_1\left(\frac{3.5}{s}\right)^{1.3} \text{ ,psi}$$

According to Eq. 4–7, the velocity at the end of each increment of travel is

$$v = \sqrt{4\left(\frac{p_1}{M_e}\right)\left(3.5 - \frac{5.09675}{3.5 + s^{0.3}}\right) + v_o^2}\ , \text{in./sec.}$$

The corresponding time interval and total time are

$$\Delta t = \frac{\Delta x}{v_a}\ , \text{sec;} \quad t = \Sigma \Delta t, \text{sec}$$

where

v_a = average velocity over the increment.

Computed data are listed in Table 4–7.

4–3.3.2.3 Bolt Unlocked, Bolt Traveling With Operating Rod

At the time that the bolt is completely unlocked, bolt and operating rod begin to travel as a unit. However, one inch of operating rod travel still remains under the influence of cylinder pressure. Except for the initial conditions involving velocity, pressure, distance, and mass; the analytic procedure is the same as mentioned in par. 4–3.3.2.1. Only linear motion prevails since cam action is complete; therefore, the mass of the bolt is no longer influenced by rotational effects. Now the bolt and operating rod are moving as a unit at the same velocity; the bolt acquired its initial velocity just as unlocking became complete. As all of the momentum just prior to this event was concentrated in the operating rod, some of it was transferred to the bolt with a subsequent reduction in velocity. Based on the law of conservation of momentum, this velocity is the velocity of the recoiling parts and has the value

$$v_r = \frac{M_o v_o}{M_r} = \frac{2.5\, v_o}{3.25}\ , \text{in./sec.}$$

where

M_o = W_o/g, mass of operating rod

M_r = W_r/g, mass of recoiling parts

v_o = velocity of operating rod at transition

Table 4–8 has the computed data.

TABLE 4–4. INPUT FOR CUTOFF EXPANSION PROGRAM

Code	Data	Code	Data
AC	0.60	RL	0.5
BK	60.0	SA	2.0
DELV(1)	0.5	SCYL	3.0
DLAMDA	3	SMAX	5.0
DRK	10.2	SO	3.5
DX	0.05	S(1)	0
EMUR	0.034	TANBO	0.007465
EMUS	0.3	T(1)	0.8
EPS	0.5	VCYL	1.8
G	386.4	VMAX	350
HELIX1	0.5	V(1)	0
HELIX2	0.5	WB	0.75
R	0.39	WC1	0
RADGYR	0.275	WO	2.5
RC	0.32		

TABLE 4–5. COMPUTED DYNAMICS BEFORE GAS CUTOFF

I	TIME MSEC	PRESS PSI	PORT AREA SQ-IN	GAS FLOW RATE LB/SEC	GAS IN CYL LB	EQUIV SORE VOL CU-IN	EQUIV CYL VOL CU-IN
2	.900	26000.	.0580	2.895	.00029	.837	.0385
3	1.000	19000.	.0580	2.116	000050	e932	.0743
4	1.100	14800.	.0580	10648	.00067	1.147	.1214
5	1.200	11400.	.0580	1.270	.00079	1.366	.1722
6	1.300	9600.	.0580	1.069	.00090	1.600	.2289
7	1.800	6600.	.0580	.735	.00127	2.255	.4543
8	2.300	4500.	.0580	.501	.00152	3.120	.7529
9	2.800	3200.	.0530	.326	.00168	4.130	1.1035
10	3.300	2400.	.0410	.189	.00178	5.240	1.4788
11	3.665	1800.	.0230	.079	.00180	6.650	1.9074

I	CYL VOL CU-IN	CYL PRESS PSI	PISTON FORCE [redacted]	IMPULSE LB-SEC	DELTA VEL IN/SEC	ROD VEL IN/SEC	ROD TRAVEL IN
2	1.8000	175.6	105.4	.011	1.63	1.6	.0001
3	1.8002	301.1	180.7	.018	2.79	4.4	.0004
4	1.8006	444.5	266.7	.027	4.12	8.5	.0010
5	1.8013	538.8	323.3	.032	5.00	13.5	00021
6	1.8023	656.4	393.9	.039	6.09	19.6	.0038
7	1.8157	1089.8	653.9	.327	50.53	70.2	.0262
8	1.8466	1401.8	841.1	e421	65.00	135.2	.0776
9	1.8981	1581.0	948.6	.474	73.31	208.5	o1635
10	1.9721	1650.7	990.4	.495	76.54	285.0	.2869
11	2.0406	1648.7	989.2	.361	55.81	340.8	.4011

4–3.3.3 Dynamics After Gas Cylinder Operation

After the gas cylinder reaches its total displacement, the recoiling parts, consisting of operating rod and bolt, have only the driving and buffer springs to provide the external forces. These springs stop the recoiling parts and then force them to counterrecoil; the driving spring and momentum of the moving mass finally lock the bolt in the firing position.

Computed spring operating data appear in Table **4–6** and the computed locking data are listed in Table 4–9.

4–3.3.3.1 Recoil Dynamics

The remaining distance for bolt and rod to compress the driving spring fully is

$$L_r = L_d - s_a = 5.5 - 2.0 = 3.5 \text{ in.}$$

The energy to be absorbed by the driving and buffer springs is

$$E_r = 1/2 M_r v_{sa}^2 \text{ in.-lb}$$

TABLE 4–6. COMPUTED DYNAMICS AFTER GAS CUTOFF
BOLT UNLOCKING DURING HELIX TRAVERSE

Y	AY	BY	Z	AZ	ZBY	QUOT1	QUOT2
1.0	.5000	1.0335	.7	1.0203	.7235	.6911	.7051
2.0	.3750	1.0681	.4	1.0115	.4273	.8777	.8878
3.0	.3125	1.1039	.1	1.0029	.1104	2.8308	2.8389
4.0	.2734	1.1409	-.2	.9943	-.2282	-1.1982	-1.1913
5.0	.2461	1.1792	-.5	.9858	-.5096	-.4174	-.4115
6.0	.2256	1.2187	-.8	.9773	-.9749	-.2314	-.2262
7.0	.2095	1.2595	-1.1	.9690	-1.3055	-.1512	-.1465
8.0	.1964	1.3017	-1.4	.9607	-1.8224	-.1078	-.1035
9.0	.1855	1.3433	-	.9524	-. 2871	-.0811	- 73
10.0	.1762	1.3904	-2.0	.9443	-2.7808	-.0634	-.0598
11.0	.1682	1.4370	-2.3	.9362	-3.3051	-.0509	-.0476
12.0	.1612	1.4851	-2.6	.9282	-3.8614	-.0417	-.0387
13.0	.1550	1.5349	-2.9	.9202	-4.4512	-.0348	-.0320
14.0	.1495	1.5863	-3.2	e9123	-5.0763	-.0295	-.0269
15.0	.1445	1.6395	-3.5	.9045	-5.7383	-.0252	-.0228
16.0	.1400	1.6944	-3.8	.8960	-6.4389	-.0217	-.0195
17.0	.1359	1.7512	-4.1	.8891	-7.1800	-.0189	-.0168
18.0	.1321	1.8099	-4.4	.8815	-7.9636	-.0166	-.0146
19.0	.1286	1.8705	-4.7	.8739	-8.7916	-.0146	-.0128
20.0	.1254	1.9332	-5.0	.8665	-9.6661	-.0130	-00112
21.0	.1224	1.9980	-5.3	.8590	-10. 894	-.0116	-.0099
22.0	.1196	200650	-5.6	.8517	-11.5638	-.0103	-.0088
					TOTALS	1.8603	1.9540

EXPANSION TIME DURING HELIX TRAVERSE (TEH) = .00022 SECONDS

V = 381.89 IN/SEC   PC = 1588.4 PSI   S = 3.5000 IN.

TABLE 4–7. COMPUTED DYNAMICS AFTER GAS CUTOFF
BOLT UNLOCKING DURING PARABOLA TRAVERSE

I	PARAB DIST IN	EQUIV CYL LENGTH IN	CAM SLOPE DEG	CYL PRESS PSI	EQUIV RECOIL MASS W/G	ROD VEL IN/SEC	TIME MSEC
13	0050	3.550	7.917	1559.4	.006529	400.05	.1279
14	.100	3.600	15.145	1531.3	.006689	416.71	.2503
15	.150	3.650	21.913	1504.1	.006974	431.83	.3682
16	.200	3.700	28.096	1477.7	.007419	445.33	.4822
17	.250	3.750	33.643	1452.1	.008072	457.19	.5930
18	.300	3.800	38.557	1427.3	.008999	467.39	.7011
19	0350	30850	42.882	1403.3	.010303	475.98	.8071
20	.400	3.900	46.676	1379.9	.012142	483.03	.9114
21	0450	31950	50.003	1357.3	.014778	488.66	1.0143
22	.500	4.000	52.926	1335.3	.018676	492.99	1.1162

TABLE 4–8. COMPUTED DYNAMICS AFTER GAS CUTOFF BOLT AND ROD UNIT RECOILING AFTER CAM ACTION

Y	AY	BY	Z	AZ	ZBY	QUOT1	QUOT2
1.0	.5000	1.0566	.7	1.0859	.7396	.6760	.7341
2.0	.3750	1.1164	.4	1.0482	.4466	.8397	.8802
3.0	.3125	1.1796	.1	1.0118	.1180	2.6491	2.6805
4.0	.2734	1.2464	-.2	.9767	-.2493	-1.0967	-1.0712
5.0	.2461	1.3170	-.5	.9428	-.6585	-.3737	-.3524
6.0	.2256	1.3916	-.8	.9101	-1.1133	-.2026	-.1844
7.0	.2095	1.4704	-1.1	.0785	-1.6174	-.1295	-.1138
8.0	.1964	1.5536	-1.4	.8480	-2.1751	-.0903	-.0766
9.0	.1855	1.6418	1.7	.8185	-2.7907	-.0665	-.0544
10.0	.1762	1.7345	-2.0	.7901	-3.4690	-.0508	-.0401
£1.6	.1682	1.8327	-2.3	e7627	-4e2152	-.0399	-.0304
12.0	.1612	1.9365	-2.6	.7362	-5.0348	-.0320	-.0236
13.0	.1550	2.0461	-2.9	.7107	-5.9337	-.0261	-.0186
14.0	.1495	2.1620	-3.2	.6860	-6.9183	or0216	-.0148
15.0	.1445	2.2844	-	.6622	-7.9953	-.0181	-.0120
16.0	.1400	2.4137	-3.8	,6392	-9.1720	-.0153	-.0098
17.0	.1359	2.5504	-4.1	.6170	-10.4564	-.0130	-.0080
18.0	.1321	2.6947	-4.4	.5956	-11.8569	-00111	-.0066
19.0	e1286	2.8473	-4.7	.5749	-13.3824	-.0096	-.0055
20.0	el254	3.0085	-5.0	.5549	-15.0426	-.0083	-.0046
21.0	.1224	3.1 789	-	.5357	- 6.8479	-.0073	-.0039
22.0	.1196	3.3588	-5.6	.5171	-18.8095	-.0064	-.0033
					TOTALS	1.9459	2.2608

EXPANSION TIME DURING HELIX TRAVERSE (TEH) = .00110 SECONDS

V = 481.66 IN/SEC PC = 1145.7 PSI S = 4.5000 IN.

MINIMUM BUFFER FORC = 176. B
MAXIMUM BUFFER FORCE = 236.8 LB
DRIVING SPRING RECOIL TIME = .000553 SEC
BUFFER RECOIL TIME = .006092 SEC
BUFFER COUNTERRECOIL TIME = .012190 SEC
DH SPRING COUNTERRECOIL TIME = .021407 SEC
BUFFER COUNTERRECOIL VELOCITY = 6.81 IN/SEC
DR SPR COUNTEHRECOIL VELOCITY = 253.81 IN/SEC

where

v_{sa} = velocity of recoiling parts at s_a

The driving spring force at s_a is F_{sa} = 61.7 lb (par. 4–3.2.5.1). Sincc the spring force when fully compressed is F_m = 97.4 lb (par. 4–3.2.4), the energy absorbed by the driving spring is

$$E_d = \frac{1}{2}(F_{sa} + F_m) L_r/\epsilon = 557 \text{ in.-lb}$$

where e = 0.5, the efficiency of the system. The energy consumed by the buffer is

$$E_b = E_r - E_d = E_r - 557 \text{ in.-lb.}$$

The effective spring constant and the buffer stroke are. K_b = 60 lb/in. and L_b = 1.0 in., respectively. The initial and final spring forces are found by equating the spring work to the energy to be absorbed

$$\frac{1}{2}(F_{ob} + F_{mb})L_b/\epsilon = E_b$$

where

$F_{mb} = F_o + K_b L_b$, the initial buffer force is

$$F_{ob} = \frac{\epsilon E_b}{L_b} - 1/2\, K_b L_b = 0.5E_r - 309, \text{lb}$$

The recoil time while the driving spring is functioning is computed by expanding Eq. *2–22* to include the limits of $x = 0$ to $x = L_r$ and by proper substitution for the other variables.

$$t_{dr} = \sqrt{\frac{\epsilon M_r}{K}}\left(\text{Sin}^{-1}\frac{F_m}{Z} - \text{Sin}^{-1}\frac{F_{sa}}{Z}\right)$$

$$= 0.0203\left(\text{Sin}^{-1}\frac{97.4}{Z} - \text{Sin}^{-1}\frac{61.7}{Z}\right), \text{sec}$$

where

$$Z = \sqrt{F_{sa}^2 + \epsilon K M_r v_{sa}^2} = \sqrt{3807 + 10.2E_r}$$

The buffer recoil time is found similarly. However, since the buffer absorbs the remaining energy, the buffer recoil time is computed according to Eq. *2–23*.

$$t_{br} = \sqrt{\frac{\epsilon M_r}{K_b}}\ \text{Cos}^{-1}\frac{F_{ob}}{F_{mb}}$$

$$= 0.00837\ \text{Cos}^{-1}\frac{F_{ob}}{F_{mb}}, \text{sec}$$

4–3.3.3.2 Counterrecoil Dynamics

The time required for counterrecoil during buffer action is found from Eq. *2–27.*

$$t_{bc} = \sqrt{\frac{M_r}{\epsilon K_b}}\ \cos^{-1}\frac{F_{ob}}{F_{mb}} = 0.01675\ \text{Cos}^{-1}\frac{F_{ob}}{F_{mb}}, \text{sec}$$

The velocity at the end of the buffer stroke is found by equating the work done by the springs to the expression for kinetic energy and then solving for the velocity

$$E_{bc} = \left(\frac{F_{mb} + F_{ob}}{2}\right)\epsilon L_b$$

$$= (F_{mb} + F_{ob})/4 = 1/2\, M_r v_{bc}^2$$

$$v_{bc} = \sqrt{2E_b/M_r}\ .$$

TABLE 4–9. COMPUTED DYNAMICS, COUNTERRECOIL BOLT LOCKING DURING PARABOLA TRAVERSE

TRAVEL INCH	FORCE POUND	BETA DEGREE	MASS 1000X	DELTAT MILSEC	VELOCITY IN/SEC	TIME MILSEC
.05	50.99	50.003	.01582	.1963	254.41	.1963
.10	50.48	46.676	.01219	.1958	255.00	.3921
.15	49.97	42.882	.01006	.1954	255.59	.5875
.20	49.46	38.557	.00871	.1950	256.16	.7825
.25	48.95	33.643	.00783	.1945	256.73	.9770
.30	48.44	28.096	.00725	.1941	257.30	1.1712
.35	47.93	21.913	.00687	.1937	257.85	1.3649
.40	47.42	15.145	.00664	.1933	258.40	1.5581
.45	46.91	7.917	.00652	.1929	258.94	1.7510
.50	46.40	.428	.00647	.1925	259.48	1.9435
1.00	41.30	.428	.00647	1.8730	264.45	3.8166

The time required for counterrecoil during driving spring action and while bolt and operating rod are moving as a unit is computed from Eq. 2–26

$$t_{cr} = \sqrt{\frac{M_r}{\epsilon K}} \left(\mathrm{Sin}^{-1} \frac{-F_{sb}}{Z} - \mathrm{Sin}^{-1} \frac{-F_m}{Z} \right)$$

$$= 0.04061 \left(\mathrm{Sin}^{-1} \frac{-51.5}{Z} - \mathrm{Sin}^{-1} \frac{-97.4}{Z} \right)$$

where

$$F_{sb} = F_m - K(L_d - L_b)$$

$$= 97.4 - 10.2 \times 4.5 = 51.5 \text{ lb}$$

$$Z = \sqrt{F_m^2 + KM_r v_{bc}^2/\epsilon} = \sqrt{9487 + 40.8 E_{bc}}$$

The total work done by all springs until locking starts is

$$E_{sc} = E_{bc} + \left(\frac{F_m + F_{sb}}{2} \right) (L_d - L_b)\epsilon$$

$$= E_{bc} + 167.5 \text{, in.-lb.}$$

The velocity at this time is

$$v_{sc} = \sqrt{2E_{sc}/M_r} \text{ , in./sec.}$$

4–3.3.3.3 Bolt Locking Dynamics

When the bolt reaches the breech face, the operating rod continues on its linear path for the remaining one inch of travel. In the meantime, the cam follower on the operating rod locks the bolt, riding over the parabolic cam curve for a half inch of travel and over the helix for the other half inch. Meanwhile, the driving spring continues to force the moving parts into battery.

The cam action during locking is the reverse of that during unlocking but follows a similar pattern. The axial length of the parabola is divided into equal length increments. The spring force is determined at the end of each increment to compute the time and velocity for each increment. Eq. 2–26 yields the differential time with M_e being the effective mass obtained from **Eq.** 4–73. For counterrecoil, the cam follower is a sliding surface, therefore, $\mu_r = \mu_s$.

$$M_e = 0.00647 + \frac{0.000458 \tan\beta}{0.203 \left(\frac{\cos\beta - 0.3 \sin\beta}{\sin\beta + 0.3\cos\beta} - 0.1149 \right)}$$

While the cam follower is traversing the helix during the last stage of locking the bolt, the operating rod continues toward its in-battery position.

The differential time for any increment of travel is

$$t_{cr} = \sqrt{\frac{M_e}{\epsilon K}} \left(\mathrm{Sin}^{-1} \frac{-F_{c2}}{Z} - \mathrm{Sin}^{-1} \frac{-F_{c1}}{Z} \right)$$

$$= 0.4428\sqrt{M_e} \left(\mathrm{Sin}^{-1} \frac{-F_{c2}}{Z} - \mathrm{Sin}^{-1} \frac{-F_{c1}}{Z} \right)$$

where

F_{c1} = spring force at beginning of increment

F_{c2} = $F_{c1} - K\Delta s$, spring force at end of increment

Δs = increment of counterrecoil travel

$Z = \sqrt{F_{c1}^2 + KM_e v^2/\epsilon} = \sqrt{F_{c1}^2 + 40.8 E_{cri}}$

E_{cri} = counterrecoil energy at beginning of increment

The energy at the end of each increment is

$$E_{cr} = E_{cri} + \frac{\epsilon}{2} (F_{c1} + F_{c2})\Delta s$$

$$= E_{cri} + 0.25 (F_{c1} + F_{c2}) \Delta s \text{ , in.-lb.}$$

The counterrecoil velocity at the end of each increment is

$$v_{cr} = \sqrt{2E_{cr}/M_e} \text{ , in./sec.}$$

4–3.3.4 Firing Rate

The time of each firing cycle is the total accumulated by all operations.

Time, sec	Operation	Table
0.003665	Before Gas Cutoff	4–5
0.000220	Bolt Unlocking, Helix	4–6
0.001116	Bolt Unlocking, Parabola	4–7
0.001100	Gas Expansion After Cam Action	4 8
0.000553	Driving Spring Recoil	4–8
0.006092	Buffer Recoil	4–8
0.012190	Buffer Counterrecoil	4–8
0.021407	Driving Spring Counterrecoil	4–8
0.003817	Bolt Locking	4–9
0.050160	Total Firing Cycle	

$$\text{Firing rate } f_r = \frac{60.0}{0.05016} = 1196 \text{ rounds/min.}$$

4–3.4 SPRINGS

4–3.4.1 Driving Spring

The driving spring, in order to comply with the dynamic requirements of the gun, is assigned the following data

K = 10.2 lb/in., spring constant

F_o = 41.3 lb, load at assembled height

F_m = 97.4 lb, load at fully compressed height

L_d = 5.5 in., operating distance of spring

t_b = 0.0055 sec (see par. 4–3.2.5.1)

t_r = 0.0218 sec (see par. 4–3.2.5.2)

v_i = v_r = 503.6 in./sec, impact velocity (see par. 4–3.2.5)

Since $v_i > 25$ ft/sec, select $\frac{T_c}{T} = 1.8$; therefore

$$T = \frac{T_c}{1.8} = \frac{0.0163}{1.8} = 0.0906 \text{ sec, surge time}$$

where $T_c = t_r - t_b$, the compression time.

Set the coil diameter at $D = 0.375$ in. According to Eq. 2–42, the wire diameter is

$$d = 0.27 \sqrt[3]{DKT} = 0.27 \sqrt[3]{0.0333}$$

$$= 0.27 \times 0.322 - 0.087 \text{ in.}$$

The number of coils from **Eq.** 2–41 is

$$N = \frac{Gd^4}{8D^3K} = \frac{11.5 \times 10^6 \times 57.3 \times 10^{-6}}{8 \times 0.0527 \times 10.2} = 152 \text{ coils}$$

where G = shear modulus.

The static torsional stress, Eq. 2–43, is

$$\tau = \frac{8F_mD}{\pi d^3} = \left(\frac{8 \times 97.4 \times 0.375}{3.14 \times 0.659}\right) 10^3$$

$$= 141{,}000 \text{ lb/in.}^2$$

The dynamic torsional stress, **Eq.** 2–44, is

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right] = 141{,}000\left(\frac{2.0}{1.8}\right)$$

$$= 157{,}000 \text{ lb/in.}^2$$

The solid height is

$$H_s = Nd = 152 \times 0.087 = 13.22 \text{ in.}$$

4–3.4.2 Buffer Spring

During recoil, the buffer springs are contacted at an impact velocity of $v_i = v_b = 348.1$ in./sec (see par. 4–3.2.5.1). Since $v_i > 25$ ft/sec, the surge time

$$T = \frac{T_c}{1.8} = \frac{0.0055}{1.8} = 0.00306 \text{ sec}$$

where

$T_c = t_{br} = 0.0055$ sec, compression time of buffer spring.

A nest of two springs is used in both primary and secondary systems. The inner spring has 40% of the load and spring constant of the system. The assigned and computed data are listed in Table 4–10. Design data are also listed for single primary and secondary springs and for a single buffer spring to offer comparative values.

The single buffer spring is, obviously, too highly stressed to be acceptable. Of the two other types, the stresses are satisfactory; this leaves the choice to available space, depending on which is the more critical length or diameter. The nested spring requires less longitudinal space whereas the single units require less diametral space.

4–4 THE TAPPET SYSTEM

The tappet system (Fig. 4–7), by virtue of its extremely short stroke, is usually confined to low muzzle velocity guns and to unlocking mechanisms. Since no initial cylinder volume exists, the delivered gases work at peak pressures immediately, no loss in pressure being suffered because a container must first be pressurized. However, the gas flow calculations will follow the same procedure that is outlined for the cutoff expansion system except that pressure on the tappet is considered to be the initial pressure unless the travel of the tappet creates a gas volume that is not compatible with the critical pressure.

4–4.1 SAMPLE PROBLEM

4–4.1.1 Specifications

Gun: Cal .30 Carbine (7.62mm)

A_b = 0.0732 in.^{2}, bore area

f_r = 600 rounds/min, firing rate

L_{bt} = 16.2 in., length of bullet travel in barrel

V_{ch} = 0.057 in.^{3}, chamber volume

W_g = 13 grains = 0.00186 lb, weight of propellant

Interior Ballistics: Pressure vs Time, Fig. 4–8
Velocity vs Time, Fig. 4–8

4–4.1.2 Preliminary Design Data

d_t = 0.40 in., diameter of tappet

L = 2.5 in., bolt travel

L_t = 0.15 in., tappet travel

W_r = 0.67 lb, weight of recoiling unit

ε = 0.40, efficiency of automatic mechanism

4–4.1.3 Design Data, Computed

The time for the firing cycle, Eq. 2–29, is

$$t_c = \frac{60}{f_r} = \frac{60}{600} = 0.100 \text{ sec.}$$

By employing Eq. 4–21 and assuming constant acceleration, the time for counterrecoil and recoil are, respectively,

$$t_{cr} = \frac{2L}{v_{cr}}, \qquad t_r = \frac{2L}{v_r} = \frac{2\varepsilon L}{v_{cr}}$$

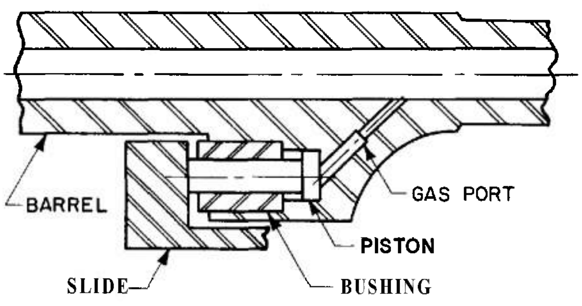


Figure 4–7. Tappet System

TABLE 4—10. BUFFER SPRING DESIGN DATA

Type Data	Primary, Double Inner	 Outer	Secondary, Double Inner	 Outer	Primary Single	Secondary Single	Single Only
K, lb/in.	32	48	96	144	80	240	60
F_m, lb	114	171	114	171	285	285	285
T, msec	3.06	3.06	3.06	3.06	3.06	3.06	3.06
D, in.	0.5	0.875	0.6	1.125	0.5	1.00	0.75
D^3, in.^{3}	0.125	0.67	0.2	1.424	0.125	1.00	0.422
DKT	0.049	0.1285	0.1763	0.4957	0.1224	0.735	0.1377
$\sqrt[3]{DKT}$	0.366	0.505	0.561	0.79	0.497	0.902	0.516
d, in.	0.100	0.136	0.151	0.213	0.134	0.243	0.139
$d^3 \times 10^3$, in.^{3}	1.0	2.52	3.44	9.67	2.4	14.3	2.69
$d^4 \times 10^4$, in.^{4}	1.0	3.42	5.2	20.58	3.22	34.9	3.73
$G \times 10^{-6}$, psi	11.5	11.5	1.5	11.5	1.5	11.5	11.5
N	36	15.3	36	14.4	46.3	20.9	21.2
$\tau \times 10^{-3}$, psi	145	151	51	46	151	51	202
$\tau_d \times 10^{-3}$, psi	162	168	56	51	168	56	225
H_s, in.	3.6	2.08	5.44	3.07	6.2	5.08	2.95

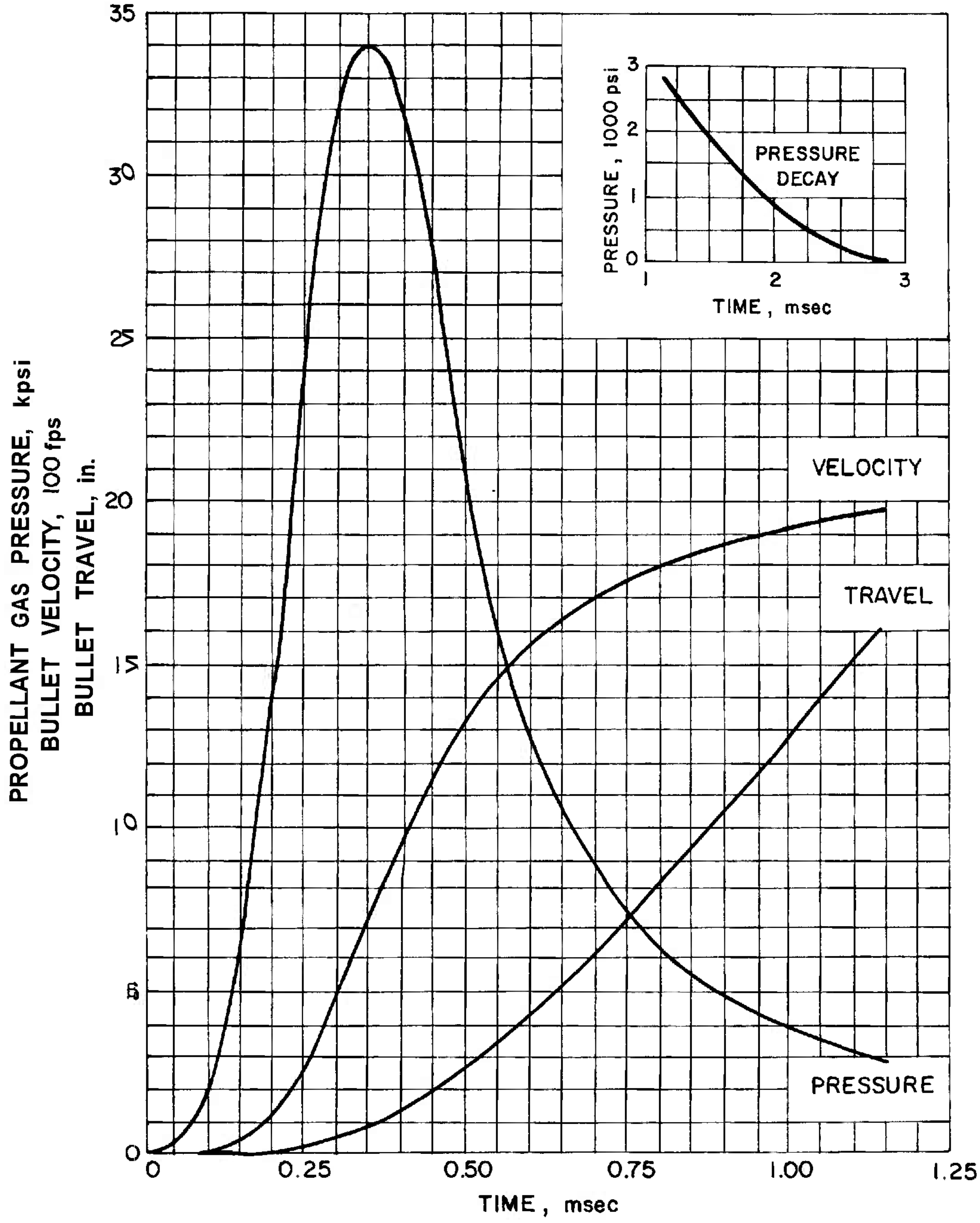


Figure 4–8. Pressure-time Curve of 7.62 mm Carbine Round

Solve both equation for v_r, equate and reduce to the simplest terms

$$t_r = Et_{cr} = 0.40t_{cr}$$

$$t_r + t_{cr} = 1.40t_{cr} = 0.100 \text{ sec}$$

$$t_{cr} = 0.071 \text{ sec}$$

$$t_r = 0.029 \text{ sec}$$

The counterrecoil velocity is

$$v_{cr} = \frac{2L}{t_{cr}} = \frac{2 \times 2.5}{0.071} = 70.4 \text{ in./sec.}$$

The recoil velocity is

$$v_r = \frac{2L}{t_r} = \frac{2 \times 2.5}{0.029} = 172 \text{ in./sec.}$$

The energy of the recoiling part, **Eq.** 2–15, is

$$E_r = \frac{1}{2}\left(\frac{W_r}{g}\right) v_r^2 = \frac{1}{2}\left(\frac{0.67}{386.4}\right) 29600$$

$$= 25.65 \text{ in.-lb.}$$

The average force on the tappet, **Eq.** 2–16, is

$$F_a = \frac{E_r}{L_t} = \frac{25.65}{0.15} = 171 \text{ lb.}$$

The momentum of the recoiling parts Mv_r is

$$Mv_r = \left(\frac{0.67}{386.4}\right) 172 = 0.298 \text{ lb-sec.}$$

Equate momentum and impulse, and solve for time t.

$$t = \frac{Mv_r}{F_a} = \frac{0.298}{171} \text{lb} = 0.00175 \text{ sec}$$

This is the time needed for the tappet to reach the velocity of 172 in./sec. The average pressure in the tappet cylinder is

$$p_t = \frac{F_a}{A_t} = \frac{171}{0.1257} = 1360 \text{ lb/in.}^2$$

where

$$A_t = \frac{\pi}{4} d_t^2 = 0.1257 \text{ in.}^2 \text{, tappet area}$$

$$d_t = 0.40 \text{ in., tappet diameter}$$

Assume that the pressure in the tappet cylinder is the critical pressure, then the corresponding pressure in the bore becomes, **Eq.** 4–26,

$$p_b = \frac{p_c}{0.53} = 2566 \text{ lb/in.}^2$$

The area of the pressure-time corresponding to the impulse of the tappet, **Eq.** 4–50, is

$$A_{pt} = p_b t = 2566 \times 0.00175 = 4.5 \text{ lb-sec/in.}^2$$

According to **Eq. 4–49,** when the bullet is still in the barrel, $p_m = p_b$ and $V_m = V_b$, therefore, if $k_b = k = 1.3$

$$W_c = W_g \left(\frac{V_c}{V_b}\right)\left(\frac{p_c}{p_b}\right)^{1/1.3} = 0.614\, W_g \left(\frac{V_c}{V_b}\right)$$

$$= 0.614 \left(0.00186\right) \frac{0.0189}{1.245} = 1.728 \times 10^{-5} \text{ lb}$$

where

p_c = $0.53p_b$, the pressure in the tappet cylinder, considered to be the critical pressure

V_c = $A_t L_t = 0.0189$ in.^{3}, volume of tappet displacement

V_b = $V_{ch} + A_b L_{bt} = 1.245$ in.^{3}, chamber plus bore volumes

The estimated orifice area, Eq. 4–52, is

$$A_o = \frac{W_c}{K_w A_{pt}} = \frac{1.728 \times 10^{-5}}{1.92 \times 10^{-3} \times 4.5} = 0.002 \text{ in.}^2$$

where

$$K_w = 0.00192/\text{sec (see par. 4–3.2.5.2)}$$

The orifice diameter d_o = 0.0505 in.

If we proceed with the above computed parameters and with the assumed critical pressures, data similar to those in Table 4–11 were computed for the period of time starting at 0.53 msec and extending to the muzzle at 1.15 msec. The area under the pressure-time curve within these time limits equals the 4.5 area computed earlier. Although the required tappet velocity of 172 in./sec was obtained, the tappet travel of 0.072 in. was far short of the required 0.15 in. The required travel could be obtained by merely shifting the gas port toward the muzzle. However, the computed equivalent volumes V_e, Eq. 4–42, were always larger than the computed chamber volume V_c, Eq. 4–48. This created the illusion that the tappet cylinder pressure p_c, Eq. 4-43, was much higher that the available bore pressure, a physical impossibility substantiated by the rate of pressure decline in the bore, so that pressure in the cylinder cannot be maintained higher than that in the bore.

Based on the first computed data, the gas port was moved farther toward the muzzle. Minimum limits on size precluding the use of a port small enough to regulate the pressure to be compatible with velocity and distance had the original port location been kept. Tappet cylinder pressures were assumed to be bore pressure. The assumption is virtually correct since a much higher mass of gas can pass through the port than volume created to accommodate it on the other side by the accelerating tappet.

The pressure in the tappet cylinder now being the same as the bore pressure, the area under the pressure-time curve becomes

$$A_{pt} = p_c t = 1360 \times 0.00175 = 2.38 \text{ lb-sec/in.}^2$$

Now that the gas port has been moved closer to the muzzle, some of the area of the pressure time curve beyond the muzzle must be considered to compensate for that lost by the relocation of the port. The first set of calculations is a good guide for locating the new port position. After the second set of calculations are completed, the exact velocity and tappet travel are determined by manipulating the areas under the pressure-time curve of the first and last increment. That amount subtracted from one must be added to the other to maintain the same area and hence velocity. If the travel distance is too short, the acceleration at the beginning is too high. Lowering the acceleration at the beginning grants the additional time needed at the end to cover the total required distance. A reduction of the area of the pressure-time curve at the start of the activity and its equal added to the end resolves this problem. If travel distance goes beyond that required, an increase in acceleration at the beginning is needed so that the terminal tappet velocity is realized at a shorter distance. A transfer of pressure-time area from the end to the beginning will serve the purpose. When both tappet travel distance and velocity comply with the required values, the gas port becomes located along the length of the barrel corresponding with the time when this activity started.

The data presented in Table 4–11 are the results of a series of computations arriving at a terminal tappet velocity of 172 in./sec on moving a distance of 0.15 in. the required value. The pressure is read from the pressure-time curve in Fig. 4–8 between the time limits 0.867 msec to 2.13 msec, which extends into the decay period after the bullet leaves the muzzle. While the bullet is still in the barrel, the pressures are read at the time interval and are assumed to be constant over the interval. To illustrate the procedure, the sequence of calculations for t = 1.1 msec follows.

At $t = 11.0 \times 10^{-4}$, p_a = 3100 lb/in.^{2} (Fig.4–8)

(The average pressure for 1.65 and 2.13 msec is obtained by dividing the differential area of the pressure-time curve by the corresponding time increment.)

The increment of time, Δt = 0.00005 sec.

The impulse on the tappet is

$$F\Delta t = A_t p_a \Delta t = 0.0195 \text{ lb-sec}$$

where

A_t = 0.1257 in.^{2}, area of tappet

TABLE 4–11. DYNAMICS OF TAPPET

t, msec	$\Delta t \times 10^5$, sec	p_a, psi	s_b, in.	$F\Delta t \times 10^4$, lb-sec	Δv, in/sec	v, in/sec	$\Delta s \times 10^5$, in.
0.876	1.7	5200	9.6	111	6.4	6.4	5
0.90	5	4800	10.4	302	17.4	23.8	44
0.95	5	4300	11.5	270	15.6	39.4	39
1.00	5	3900	12.7	246	14.2	53.6	36
1.05	5	3500	13.9	220	12.7	66.3	32
1.10	5	3100	15.1	195	11.3	77.6	28
1.15	5	2800	16.2	176	10.1	87.7	25
1.65	50	1630	—	1022	58.9	146.6	1472
2.13	48	730	—	440	25.4	172.0	610

t, msec	$\Delta s_t \times 10^5$, in.	$s \times 10^5$, in.	$V_c \times 10^3$, in.^{3}	$\Delta W \times 10^6$, lb	V_b, in.^{3}	$W_c \times 10^6$, lb	$V_e \times 10^3$, in.^{3}
0.876	0	5	0.0063	0.340	0.759	0.34	0.139
0.90	32	81	0.102	0.924	0.819	1.264	0.556
0.95	119	239	0.300	0.826	0.899	2.09	1.010
1.00	197	472	0.590	0.750	0.987	2.84	1.505
1.05	268	772	0.970	0.622	1.075	3.462	2.00
1.10	332	1132	1.42	0.595	1.165	4.06	2.54
1.15	388	1545	1.95	0.538	1.245	4.60	3.08
1.65	4385	7402	9.34	3.130	3.67	7.72	15.26
2.13	7047	15059	18.9	1.344	6.24	9.07	30.4

The velocity at the end of the time interval, Eqs. 4–45 and 4–46, is

$$v = v_{n-1} + \frac{F\Delta t}{M_r} = 66.3 + \frac{0.0195 \times 386.4}{0.67}$$

$$= 66.3 + 11.3 = 77.6 \text{ in./sec.}$$

The distance traveled by the tappet, Eq. 4–47, becomes

$$s = \frac{1}{2}\Delta v \Delta t + v_{n-1}\Delta t + s_{n-1}$$

$$= \left(\frac{11.3}{2} + 66.3\right) 0.00005 + 0.00772$$

$$= 0.00028 + 0.00332 + 0.00772 = 0.01132 \text{ in.}$$

The gas volume in the cylinder according to Eq. 4–48 is

$$V_c = V_{co} + A_t s = 0.1257 \times 0.01132 = 0.00142 \text{ in.}^3$$

where the initial volume $V_{co} = 0$.

The rate of flow, Eq. 4–27, is

$$w = K_w A_o p_a = 0.00129 \times 0.002 \times 3100$$

$$= 0.01190 \text{ lb/sec.}$$

The weight of the gas flowing through the port during the interval, Eq. 4–28, is

$$\Delta W_c = \Delta t w = 0.00005 \times 0.01190 = 5.95 \times 10^{-7} \text{ lb.}$$

The total weight of the gas in the tappet cylinder is W_c.

$$W_c = W_{c(n-1)} + \Delta W_c = (3.462 + 0.595)\, 10^{-6}$$

$$= 4.057 \times 10^{-6} \text{ lb.}$$

The equivalent volume of this gas at 3100 psi pressure, according to Eq. 4–42 becomes

$$V_e = \left(\frac{W_c}{W_g}\right) V_b = \frac{4.057 \times 10^{-6}}{1.86 \times 10^{-3}}\; 1.165$$

$$= 0.00254 \text{ in.}^3$$

where V_b is the gas volume of the barrel.

$$V_b = V_o + A_b s_b = 0.057 + 0.0732 \times 15.1 = 1.165 \text{ in.}^3$$

where

V_o = 0.057 in.^{3}, initial volume (chamber)

A_b = 0.0732 in.^{2}, bore area

s_b = 15.1 in., bullet travel at t = 1.1 msec (Fig. 4–8)

According to Eq. 4–43

$$p_c = \left(\frac{V_e}{V_c}\right)^k p_a = \left(\frac{2.54}{1.42}\right)^{1.3} 3100$$

$$= 2.13 \times 3100 = 6600 \text{ lb/in.}^2$$

This pressure is absurd but it does indicate that more gas is capable of flowing through the port than the cylinder, as the receiver, can admit; therefore, the assumption that cylinder pressure is nearly equal to bore pressure is highly probable particularly since $V_e > V_c$ throughout the operation. Further assurance is available by computing the time needed during each interval to bring the pressure in the cylinder to the critical. In each increment, the gas flow is rapid enough to reach the critical before the moving tappet creates the corresponding volume. This approach is conservative since the differential pressures in the computations were based solely on critical pressures as limits although considerable time is available for additional gas flow into the cylinder, thus tending to approach the bore pressure. For example, continue with the same sequence of calculations for t = 1.1 msec. The critical pressure is

$$p_{cr} = 0.53\, p_a = 0.53 \times 3100 = 1640 \text{ lb/in.}^2$$

The pressure due to expansion of the gas in the cylinder during the interval provided that gas flow ceases is p_e.

$$p_e = p_{cr-1}\left(\frac{V_{c-1}}{V_c}\right)^k = 1860\left(\frac{0.97}{1.42}\right)^{1.3}$$

$$= 1135 \text{ lb/in.}^2$$

where

p_{cr-1} = 1860 lb/in.^{2}, the critical pressure of the previous interval

V_c = 0.00142 in.^{3}, the gas volume in the tappet cylinder

V_{c-1} = 0.00097 in.^{3}, the gas volume of the previous interval

The differential pressure between the expanded gas in the cylinder and the critical pressure provided by gas flow

$$\Delta p_c = p_{cr} - p_e = 1640 - 1135 = 505 \text{ lb/in.}^2$$

The equivalent bore volume of the gas expanded to the critical pressure is

$$V_{ec} = V_b\left(\frac{p_a}{p_c}\right)^k = 1.162\left(\frac{3100}{1640}\right)^{\frac{1}{1.3}} = 1.90 \text{ in.}^3$$

The weight of the gas at the critical pressure in the tappet cylinder is

$$W_c = \left(\frac{V_c}{V_e}\right) W_g = \left(\frac{0.00142}{1.90}\right) 0.00186$$

$$= 1.39 \times 10^{-6} \text{ lb.}$$

The weight of the gas flowing into the cylinder is that needed to increase the pressure from p_e to p_{cr}.

$$\Delta W_{ce} = \left(\frac{\Delta p_c}{p_{cr}}\right) W_c = \frac{505}{1640} (1.39) \quad 10^{-6}$$

$$= 4.28 \times 10^{-7} \text{ lb.}$$

The time needed for the flow is

$$\Delta t_{ce} = \frac{\Delta W_{ce}}{w} = \frac{4.28 \times 10^{-7}}{0.0119} = 3.60 \times 10^{-5} \text{ sec.}$$

The time is about 70% that of the specified interval of 5×10^{-5} sec. The results of the rest of the calculations appear in Table 4–12. On further examination of the tabulated results – since time is available for gas flow beyond the critical – note that the pressure due to expansion p_e would be greater than shown, thereby reducing the time needed to reach the critical pressure, and meanwhile, providing more time for the tappet cylinder pressure to reach the bore pressure.

4–4.1.4 Spring Design Data

Spring characteristics are determined more readily during recoil since more data are immediately available. According to **Eq.** 2–15, after the bolt has traveled its full distance in recoil, the energy to be absorbed by the spring is E_r.

$$E_r = \frac{1}{2}\left(M_b v_r^2\right) = 25.65 \text{ in.-lb}$$

where

$$M_b = \frac{0.67}{386.4} \frac{\text{lb-sec}^2}{\text{in.}}, \text{ mass of bolt unit}$$

v_r = 172 in./sec, recoil velocity

From Eq. 2–67b

$$KT = \frac{F_m}{1037} = \frac{F_a + \frac{1}{2}\left(L_d K\right)}{1037}$$

where

$$K = \frac{4.36}{0.0071 \times 1037 - 1.175} = \frac{4.36}{6.188}$$

$= 0.70$ lb, spring constant

$$T = \frac{T_c}{3.8} = 0.0071 \text{ sec, surge time of spring}$$

$T_c = t_r - t = 0.029 - 0.0021 = 0.0269$ sec, preliminary estimate of compression time of spring

The average spring force, Eq. 2–30, is

$$F_a = \frac{\epsilon E_r}{L_d} = \frac{0.040 \times 25.65}{2.35} = 4.36$$

where

$L_d = L - L_t = 2.35$ in., spring deflection after tappet stops.

TABLE 4—12. CRITICAL PRESSURE TIME REQUIREMENTS

t, msec	p_{cr}, psi	$w \times 10^3$, lb/sec	$V_c \times 10^3$, in.^{3}	p_e, psi	Δp_c, psi	V_{ec}, in.	$W_c \times 10^6$, lb	$\Delta W_{ce} \times 10^6$, lb	$\Delta t_w \times 10^5$, sec
0.867	2760	20.00	0.0063	0	2760	1.24	0.001	0.001	0.05
0.90	2540	18.46	0.102	73	2467	1.34	0.142	0.138	0.77
0.95	2280	16.52	0.300	625	1655	1.47	0.380	0.276	1.67
1.00	1990	15.00	0.590	945	1045	1.61	0.683	0.376	2.50
1.05	1860	12.44	0.970	1040	820	1.75	1.03	0.453	3.64
1.10	1640	11.90	1.42	1135	505	1.90	1.39	0.428	3.60
1.15	1480	10.76	1.95	1088	392	2.03	1.79	0.474	4.40
1.65	870	6.26	9.34	182	688	5.98	2.90	2.290	36.60
2.13	387	2.80	18.90	348	39	10.18	3.46	0.348	12.40

The driving spring force when the bolt is fully retracted is

$$F_m = F_a + \frac{1}{2}\left(KL_d\right) = 4.36 + \frac{1}{2}\left(2.35\right)\ 0.70$$

$$= 5.18 \text{ lb.}$$

The driving spring force when tappet contacts bolt is

$$F_o = F_m - KL_t = 5.18 - 2.35 \times 0.70 = 3.54 \text{ lb.}$$

The time for the bolt to recoil, excluding the time of the initial 0.15 in. of tappet travel, is computed via **Eq.** 2–23.

$$t_r = \sqrt{\frac{\epsilon M_b}{K}}\ \text{Cos}^{-1}\ \frac{F_o}{F_m}$$

$$= \sqrt{\frac{0.4 \times 0.67}{0.70 \times 386.4}}\ \text{Cos}^{-1}\ \frac{3.54}{5.18}$$

$$= 0.0315 \times 0.818 = 0.0258 \text{ sec}$$

Spring data and time are now computed for counterrecoil

$$F_o = F_m - KL = 5.18 - 0.70 \times 2.5 = 3.43 \text{ lb}$$

where

L = 2.5 in., total length of bolt travel including tappet travel.

The time of counterrecoil, **Eq.** 2–27,

$$t_{cr} = \sqrt{\frac{M_b}{\epsilon K}}\ \text{Cos}^{-1}\ \frac{F_o}{F_m}$$

$$= \sqrt{\frac{0.67}{0.4 \times 386.4 \times 0.70}}\ \cos^{-1}\ \frac{3.43}{5.18}$$

$$= 0.079 \times 0.847 = 0.0669 \text{ sec.}$$

The time needed to accelerate the bolt during recoil is obtained from Table 4–11 where $t_a = t = 0.0021$ sec, the last value (rounded to 4 places) in the time column.

The elapsed time for the firing cycle becomes

$$t_c = t_a + t_r + t_{cr} = 0.0021 + 0.0258 + 0.0669$$

$$= 0.0948 \text{ sec.}$$

The new firing rate of

$f_r = \frac{60}{t} = 633$ rounds/min is acceptable since it is only 5.5% higher than the specified rate.

The revised surge time of the spring becomes

$$T = \frac{T_c}{3.8} = \frac{t_a + t_r}{3.8} = \frac{0.0021 + 0.0258}{3.8} = 0.00734 \text{ sec.}$$

Having set the coil diameter at D = 0.25 in., we establish the spring constant at K = 0.70 lb/in., thus the wire diameter according to **Eq.** 2–42 becomes

$$d = 0.27\ \sqrt[3]{DKT} = 0.27\ \sqrt[3]{0.001286}$$

$$= 0.27 \times 0.1088 = 0.0294 \text{ in.}$$

The number of coils, **Eq.** 2–41, is N.

$$N = \frac{Gd^4}{8D^3K} = \frac{11.5 \times 10^6 \times 74.7 \times 10^{-8}}{8 \times 0.0156 \times 0.66} = 104 \text{ coils}$$

The static torsional shear stress becomes

$$\tau = \frac{8F_mD}{\pi d^3} = \frac{8 \times 5.18 \times 0.25}{3.14 \times 25.4 \times 10^{-6}} = 130{,}000 \text{ lb/in.}^2$$

The dynamic stress is

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right] = 130{,}000\left(\frac{4.0}{3.8}\right)$$

$$= 137{,}000 \text{ lb/in.}^2$$

The solid height is

$$H_s = Nd = 104 \times 0.0294 = 3.06 \text{ in.}$$

CHAPTER 5

REVOLVER-TYPE MACHINE GUNS

5–1 SINGLE BARREL TYPE*

Revolver-type machine guns are distinguished from other types by the revolving drum, a feature borrowed from the revolver. The operational characteristics of the two weapons, machine gun and pistol, are basically similar except for refinements in the former that convert it from an ordinary repeater to a machine gun. These refinements involve automatic loading, firing, and ejecting operations. Fig. 5–1 is a schematic of a revolver type machine gun. Its essential components are receiver, drum cradle, drum, barrel, gas operating mechanism, slide, feeder, rammer, driving spring, and adapter.

Fig. 5–1 illustrates a gas-operated gun, however, external power may also be used for this type. When a round is fired, the recoiling parts comprising barrel, drum, and cradle recoil a short distance before being stopped by the adapter. In the meantime, the slide assembly recoils with these parts until a portion of the propellant gas passing from barrel to operating cylinder induces a relative velocity between recoiling parts and slide. As the recoiling parts stop, the slide continues to be accelerated rearward until the piston in the operating cylinder stops. The slide now has sufficient momentum to operate all moving parts until the next round is fired.

Continuing rearward, the slide, through the medium of a cam, imparts motion to the drum and then comes to rest after transferring all its energy less substantial frictional losses to the drum and driving spring. The drum now has the momentum to continue all operations. As it rotates, it actuates the feeder which pulls the ammunition belt far enough to align the next round with an empty chamber and the rammer. Cam action now imparts forward motion to the slide and rammer, the two components being integral. Cam forces – augmented by the driving spring force – drive the slide forward, eject the spent cartridge case, ram a full round into a chamber, and stop the drum as the loaded chamber reaches alignment with the bore just before the round is fired and the whole sequence repeats.

*General information was obtained from Refs. 8, 9, 10, and 11.

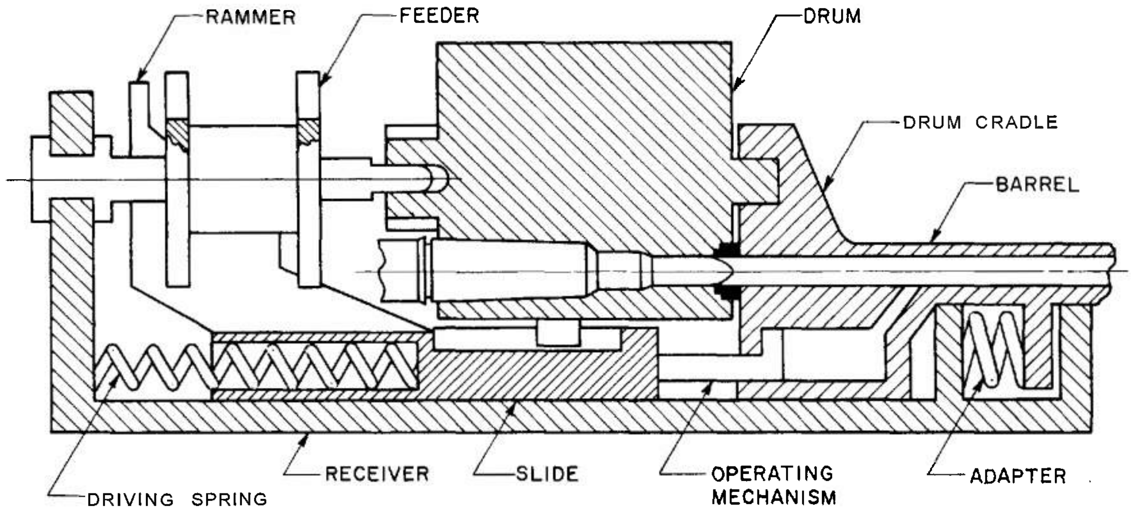


Figure 5–1. Schematic of Single Barrel Revolver-type Machine Gun

Ramming is a two-stage activity. The first stage involves stripping the round from the belt and pushing it about halfway along its path to the chamber. The second stage completes the chambering. Fig. 5–2 shows this two-step action. Actually, the entire process occurs during one cycle but on two adjacent rounds. While first-stage activity is confined to a new round, second-stage activity simultaneously completes the ramming of the round introduced during the preceding firing sequence. This two-stage ramming process represents a major advantage over a single-chambered gun by its ability to reduce the ramming distance to half its usual length thereby decreasing cycle time and increasing the rate of fire. Another contributing factor is the reduction of shocks resulting in higher allowable slide velocities (up to 50 ft/sec) than those usually associated with conventional mechanisms.

5–1.1 PRELIMINARY DYNAMICS OF FIRING CYCLE

The normal approach to the study of the dynamics during the firing cycle is to consider the various operations in their operational sequence. By considering firing as the initial condition, the first response of the gun is recoil. In many applications, since propellant gas forces are appreciable, the effects of recoil mechanism resistance are assumed negligible without introducing serious errors. However, for revolver-type machine guns, the recoil stroke is so short that considerable resistance must be provided immediately to preclude high recoil velocities and to keep recoil travel to the desired minimum. Left unimpeded, the distance of free recoil of a 20 mm barrel (Table 2–2) is almost 5 times that of an existing (M39) gun.

Performing an analysis similar to that defined in Eqs. 2–45 through 2–49, with due attention to the adapter resistance, the following iterative procedure is suggested. Compute the free recoil characteristics similar to those of Table 3–2. After obtaining the velocity and distance of free recoil, efforts must be directed toward reducing the velocity to zero over the prescribed recoil distance. One way of computing a zero velocity is to employ the weighted arithmetic mean of the impulse which yields an average force

$$F_a = \frac{\Sigma F_g \Delta t}{\Sigma \Delta t} \quad (5\text{–}1)$$

Let this average force become the adapter resistance and compute what may be considered to be a resisting

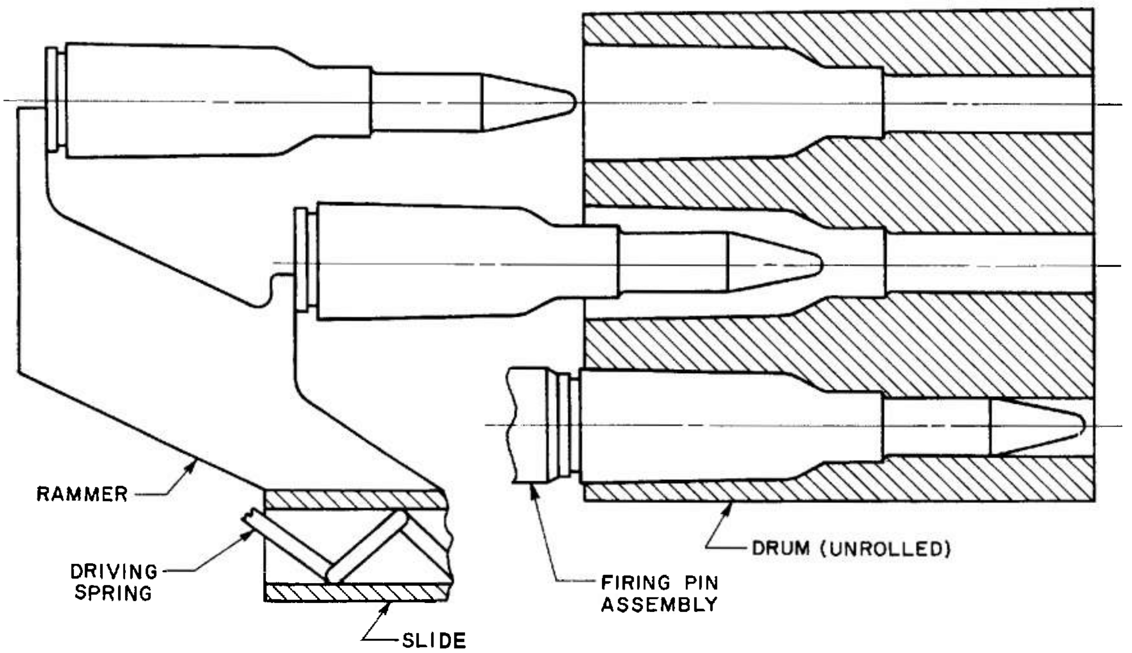


Figure 5–2. Two Stage Ramming

impulse for each increment of time, which, when subtracted from the original impulse, will yield an effective impulse.

$$(F\Delta t)_e = F_g \Delta t - F_a \Delta t \qquad (5\text{–}2)$$

The change in velocity during each time interval will be

$$\Delta v = \frac{(F\Delta t)_e}{M_r} .$$

This procedure will always have $\Sigma \Delta v = 0$, thus meeting one of the design criteria. The recoil distance is obtained from **Eqs. 2–47, 2–48,** and **2–49.**

The data of Table **2–2** can illustrate the above procedure

$$F_a = \frac{\Sigma F_g \Delta t}{\Sigma \Delta t} = \frac{37.42}{0.006} = \underline{6237} \text{ lb}$$

$$(F\Delta t)_e = (F_g \Delta t - F_a \Delta t) = 0$$

$$\Sigma \Delta v = \Sigma \left(\frac{F\Delta t_e}{M_r} \right) = 0$$

$$x = \Sigma \Delta x = \mathbf{0.4313} \text{ in.}$$

If **x** is too large, F_a is increased; if too small, it is decreased. Based on the **0.25** in. recoil distance of the **M39** Machine Gun, F_a must be increased. Although the adapter resistance increases as recoil progresses, the error in assuming F_a constant is minimal since the distance over which it functions is extremely short.

The recoiling parts continue to accelerate until $(F\Delta t)_e$ becomes zero. When this happens, the recoiling parts begin to decelerate but the slide continues to move under its own inertia unless the projectile has already passed the operating cylinder's gas port. In this event, a strong probability, the slide continues to accelerate under the influence of the newly supplied force source. Fig. **5–3** is a force diagram showing the accumulated effect of the various applied and induced forces

where

F_t = adapter force

F_c = operating cylinder force

F_g = propellant gas force

M_s = mass of slide

When slide and recoiling parts act as a unit $M = M_r + M_s$ (otherwise $M = M_r$ the mass of the recoiling parts), the recoil acceleration becomes

$$a_r = \frac{F_g - F_t - F_c}{M_r + M_s} . \qquad (5\text{–}4)$$

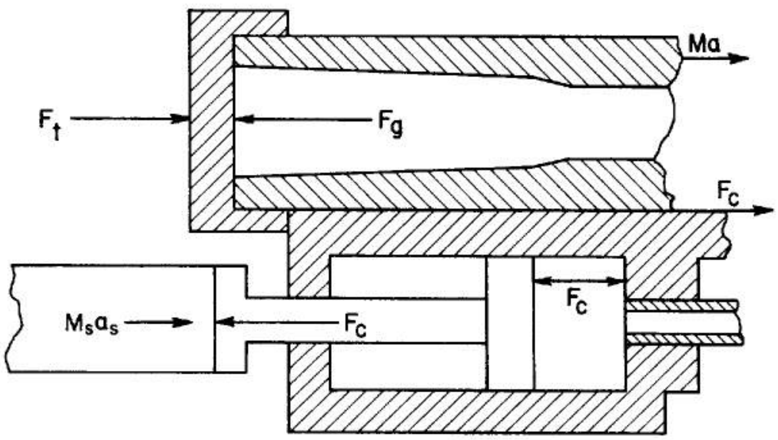


Figure 5–3. Force Diagram of Recoiling Parts and Slide

The slide acceleration becomes

$$a_s = a_r + \frac{F_c}{M_s} \cdot \quad (5\text{–}5)$$

The dynamics of the gas operating cylinder follow a procedure similar to that for the cutoff expansion system (see par. 4–3.1.1).

Before continuing with the gas system analysis, the required operating energy must be estimated which leads to the selection and analysis of the slide and drum dynamics. Transfer of energy of slide to drum and driving spring and then back to slide must be achieved with operative efficiency and tolerable forces. A system such as this is notorious for its high energy losses and large forces. These two characteristics are kept within bounds by an elliptical cam, although other curves may be used if they display similar properties with respect to cam action.

The physical dimensions of the drum are best suited to generate other design parameters. Drum length is dictated by round length. Its outer radius is based on the number of chambers and the strength of the outer wall. A minimum of 4 chambers is sufficient to meet the basic operating requirements of present revolver-type machine gun concepts but may prove awkward in actual practice because of large angular displacement for each firing cycle, thus reducing the firing rate and putting an added burden on the designer to provide more power and acceptable mechanisms such as rammers. A design study at Springfield Armory indicates an optimum number of 5 chambers when based on kinematics alone. When other factors were considered, 5 or 6 chambers showed little difference with 5 having a slightly lower firing rate but definitely lower forces and less weight, thus leaning toward 5 as the recommended number. With the number of chambers established, the linear dimensions are now available from which the mass of the drum can be estimated.

Present practice has the weight of the slide approximately 1/3 the weight of the drum. Another established criterion that provides acceptable design parameters of the cam is the relationship shown in Eq. 5–6a.

$$\frac{M_d}{M_s}\left(\frac{b^2}{a^2}\right) = 1.0 \quad (5\text{–}6a)$$

where

a = major axis of elliptical cam

b = minor axis of elliptical cam

M_d = mass of drum

M_s = mass of slide

When $M_d = 3M_s$,

$$a = 1.732b. \quad (5\text{–}6b)$$

Another design parameter, the index of friction,

$$\mu_i = \pi\left(\frac{a}{b} + \frac{b}{a}\right) \quad \text{(Ref. 14)} \quad (5\text{–}7a)$$

Substitute the value for a in Eq. 5–6b into Eq. 5–7a

$$\mu_i = \pi(1.732 + 0.577) = 7.23. \quad (5\text{–}7a)$$

This index may vary if other ratios of a and b become more attractive.

The slide travel relative to the receiver need be only slightly more than half the round length since ramming takes place in two stages. The addition to the half-length depends on the desired clearances between projectile nose and drum, and between rammer and cartridge case base. Straight portions of the cam provide a dwell period for the drum before and after firing; one over the first part of slide travel during recoil, the other over the last part of counterrecoil. These straight portions may be of different lengths as may be the width of the cam curves for recoil and counterrecoil. Because cam forces are inherently less severe during counterrecoil, a larger sweep of the curve for recoil has the tendency to equalize the forces of the two actions, thereby increasing the efficiency of the system. A separate study of the individual cases is recommended but the relative dimension of an existing system serves as a guide. Fig. 5–4 is a schematic of such an arrangement.

$$L_c = a_{rec} + s_{or} \quad (5\text{–}8)$$

$$L_c = a_{crc} + s_{ocr}$$

$$\frac{a_{crc}}{a_{rec}} = 0.6 \tag{5-9}$$

$$\frac{b_{cr}}{b_r} = 0.75 \tag{5-10}$$

$$\frac{s_{or}}{a_{rec}} = 0.5 \tag{5-11}$$

$$\frac{s_{ocr}}{a_{crc}} = 1.5 \tag{5-12}$$

$$L_c = \frac{1}{2} L_r + C_r \text{ , cam length} \tag{5-13}$$

where

L_r = length of round

C_r = total clearance of the round at both ends

The cam width is

$$w_c = b_r + b_{cr} = \frac{2\pi R_{ch}}{N_c} \tag{5-14}$$

where

N_c = number of chambers in drum

R_{ch} = radius of chamber about drum axis

After the preliminary cam dimensions have been estimated, attention is now directed toward the effort needed to operate all moving parts at speeds commensurate with the firing rate of the gun. Operations that require energy include feeding, ramming, and ejecting. Components that must be activated are slide and rammer, drum, feeder, and loaded ammunition belt. The size of the drum, based on present 20 mm data, has the length L_d and diameter D_d indicated in **Eqs.** 5–15a and 5–15b.

$$L_d = L_r\text{, drum length} \tag{5-15a}$$

$$D_d = 6D_b\text{, drum diameter} \tag{5-15b}$$

where

D_b = bore diameter.

The mass of the drum and rotating feeder components may be estimated as the solid cylinder having the above dimensions. For moving ammunition by the feeder, an additional 10 percent is added to the effort. For ramming, the mass of two rounds is added to that of slide and rammer, whose mass M_g is approximately equal to *1/3* of the drum components M_d, thus

$$M_s = 1/3\, M_d. \tag{5-16}$$

The spent cartridge case should be ejected at a velocity of approximately 70 ft/sec. The velocity of other moving parts depend on the rate of fire f_r. However, since the maximum velocity of the slide should not exceed 50 ft/sec, this limit may be used as the initial estimate of the maximum slide velocity. The energy of the slide at this velocity represents the input energy of the system.

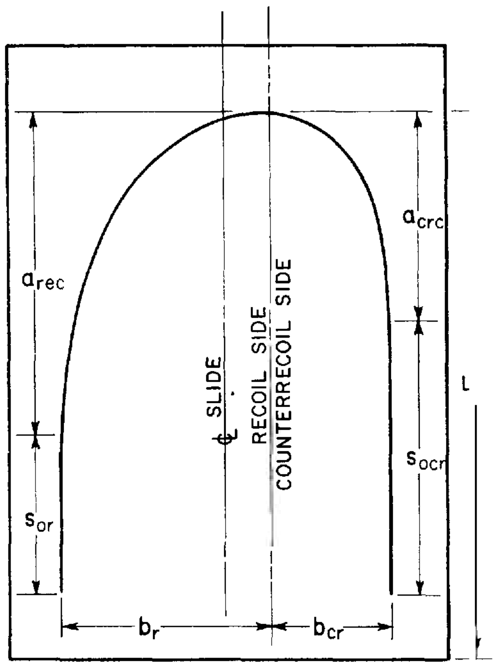


Figure 5–4. Schematic of Cam Geometry

By the time that all moving parts have returned to the firing position, where all motion ceases, considerable energy has been expended to friction, loading, and ejection. According to Ref. 14 when x = L, the loss of energy from slide to drum due to friction is

$$E_{\mu d} = (1.0 - e^{-\mu_i \mu}) E_d \tag{5-17}$$

where E_d is the total energy transferred from slide to drum if the system were frictionless and μ is the coefficient of friction. The energy of the drum alone (belt energy is of no help because it cannot push) just as the slide starts to counterrecoil is computed from **Eq.** 59, Ref. 9, and shown in **Eq.** 5–18

$$E_{dcr} = \left(\frac{M_d}{M_{de}}\right) e^{-\mu_i \mu} E_d \tag{5-18}$$

where

M_d = mass of drum

M_{de} = effective mass of drum and ammunition belt

According to **Eq.** 69, Ref. 9, when x = 0, the frictional energy loss in the slide when fully counterrecoiled is

$$E_{\mu s} = (1.0 - e^{-\mu_i \mu}) E_{dcr} . \tag{5-19}$$

The loss attributed to the driving spring is

$$E_\epsilon = E_s(1 - \epsilon^2) \tag{5-20}$$

where

E_s = energy transferred from slide to driving spring

ϵ = efficiency of spring system

The energy expended to eject the spent cartridge case at velocity v_e is

$$E_e = \frac{1}{2}\left(M_c v_e^2\right) \tag{5-21}$$

where M_c = mass of cartridge case

The total expenditure of energy of the drum and its associated components during a firing cycle is expressed in **Eq.** 5–22.

$$E_\mu = E_{\mu d} + E_{\mu s} + E_\epsilon + E_e \tag{5-22}$$

The energy of the slide derived from normal recoil and the gas operating cylinder is

$$E_{sr} = \frac{1}{2}\left(M_s v_{sm}^2\right) \tag{5-23}$$

where

$v_{sm} \leq$ 50 ft/sec, maximum slide velocity

M_s = mass of slide

A relatively stiff driving spring is recommended to hold the maximum velocity of drum and belt to a minimum (Ref. 9). If ρ is the ratio of spring energy E_s to drum energy E_d,

$$\rho = \frac{E_s}{E_d} . \tag{5-24}$$

Since the slide energy is converted to spring and drum energy, $E_{sr} = E_s + E_d$, the total energy transferred to the drum is shown to be

$$E_d = E_{sr}/(1 + \rho). \tag{5-25}$$

The preliminary firing rate is estimated from the times of recoil and counterrecoil when based on the relative velocity of the cam follower on the drum and the cam in the slide. The recoil time (**Eq.** 98, of Ref. 9) is

$$t_r = \gamma \left(\frac{2s_r}{v_{sm} + v_{dm}}\right) \tag{5-26}$$

where

s_r = cam length for recoil

v_{dm} = maximum peripheral velocity of drum

γ = correction factor

The counterrecoil time will be

$$t_{cr} = \gamma \left(\frac{2s_{cr}}{v_{dm} + v_{scr}}\right) \qquad (5-27)$$

where

s_{cr} = cam length for counterrecoil

v_{scr} = counterrecoil velocity of slide

Based on operating guns, the empirical $\gamma = 0.935$ (Ref. 9).

The firing rate is

$$f_r = \frac{60}{t_r + t_{cr}}, \text{rounds/min.} \qquad (5-28)$$

5-1.1.1 Sample Problem of Preliminary Firing Rate Estimate

Given data W_d = 30 lb, weight of drum

W_s = 10 lb, weight of slide

W_a = 0.6 lb, weight of round

W_{cc} = 0.2 lb, weight of case

R_d = 3 in., radius of cam contact point to drum axis

L_c = 5 in., distance of slide travel (same as cam length)

N_c = 5 chambers

$a_{rec} = \frac{L_c}{1.5} = 3.33$ in. (from Eqs. 5-8 and 5-11)

$a_{crc} = 0.6\, a_{rec} = 2.0$ in. (from Eq. 5-9)

$b_r + b_{cr} = \frac{2\pi R_d}{5} = 3.77$ in., peripheral cam travel

$b_r = \frac{b_r + b_{cr}}{1.75} = 2.15$ (from Eq. 5-10)

$b_{cr} = 3.77 - 2.15 = 1.62$ in.

$s_{or} = L_c - a_{rec} = 1.67$ in.

$s_{ocr} = L_c - a_{crc} = 3.0$ in.

$$s_r = s_{or} + \frac{\pi}{2}\sqrt{\frac{a_{rec}^2 + b_r^2}{2}} = 1.67 + 4.40 = 6.07 \text{ in.,}$$

cam follower travel during recoil

$$s_{cr} = s_{ocr} + \frac{\pi}{2}\sqrt{\frac{a_{crc}^2 + b_{cr}^2}{2}} = 3.0 + 2.86 = 5.86 \text{ in.,}$$

cam follower travel during counterrecoil

v_e = 840 in./sec, maximum recommended ejection velocity of cartridge

v_{sm} = 600 in./sec, maximum allowable slide velocity

$$E_{sr} = \frac{1}{2}\left(M_s v_{sm}^2\right) = \frac{10 \times 360000}{2 \times 386.4}$$

= 4658.4 in.-lb, maximum slide energy of recoil

Select $\rho = 0.25$

$$E_d = \frac{E_{sr}}{1 + \rho} = \frac{4658.4}{1.25}$$

= 3726.7 in.-lb, energy to be transferred to drum

$E_s = E_{sr} - E_d = 931.7$ in.-lb,

energy to be transferred to driving spring

At the end of slide recoil, the energy in the drum, **Eq.** 5–18. becomes

$$E_{dcr} = \left(\frac{M_d}{M_{de}}\right) e^{-\mu_i\mu} E_d = \frac{30}{33}\left(\frac{3726.7}{e^{0.723}}\right)$$

$$= \frac{111801}{68} = 1644.1 \text{ in.-lb}$$

where

$$M_{de} = 1.1M_d = 33/g$$

$$\mu_i\mu = 7.23 \times 0.1 = 0.723$$

$$E_{dcr} = \frac{1}{2}\left(I_d\omega_d^2\right) = \frac{1}{2}\left(M_d k^2 \omega_d^2\right)$$

$$= \frac{1}{2}M_d\left(\frac{R_d^2}{2}\right)\left(\frac{v_{dm}^2}{R_d^2}\right) = \frac{1}{4}M_d v_{dm}^2$$

where I_d = mass moment of inertia of drum

k = radius of gyration

ω_d = angular velocity of drum

$$v_{dm} = \sqrt{\frac{4E_{dcr}}{M_d}} = \sqrt{\frac{4 \times 386.4 \times 1644.1}{30}}$$

$$= \sqrt{84704} = 291 \text{ in./sec, maximum}$$

peripheral velocity of drum

The energy transferred from drum to slide, from Eq. 5–19, is

$$E_{sd} = e^{-\mu_i\mu} E_{dcr} = \frac{1644.1}{2.06} = 798.1 \text{ in.-lb.}$$

The energy transferred from driving spring to slide, assuming 80% efficiency, **Eq.** 5–20, is

$$E_{ss} = \epsilon^2 E_s = 0.64 \times 931.7 = 596.3 \text{ in.-lb.}$$

The energy expended for ejection, **Eq.** 5–21, is

$$E_e = \frac{1}{2}\left(M_{cc}v_e^2\right) = \frac{0.2 \times 705600}{2 \times 386.4} = 182.6 \text{ in.-lb.}$$

The energy in the slide at end of counterrecoil is

$$E_{scr} = E_{sd} + E_{ss} - E_e = 1211.8 \text{ in.-lb.}$$

The velocity of the slide at end of counterrecoil when it bears the additional weight of two rounds is

$$v_{scr} = \sqrt{\frac{2E_{scr}}{M_s + 2M_a}} = \sqrt{\frac{2 \times 1211.8 \times 386.4}{10 + 2 \times 0.6}}$$

$$= \sqrt{83614} = 289 \text{ in./sec.}$$

According to **Eqs.** 5–26 and 5–27, the firing cycle time becomes

$$t_c = 2\gamma\left(\frac{s_r}{v_{sm} + v_{dm}} + \frac{s_{cr}}{v_{dm} + v_{scr}}\right)$$

$$= 1.87\left(\frac{6.07}{891} + \frac{5.86}{580}\right)$$

$$= 1.87\,(0.0068 + 0.0101) = 0.0316 \text{ sec}$$

where γ = 0.935.

The firing rate, **Eq.** 5–28, is estimated to be

$$f = \frac{60}{t_c} = \frac{60}{0.0316} = 1898 \text{ rounds/min.}$$

If the firing rate is too high, the initial velocity of the slide may be reduced proportionately. If too low, other avenues of design improvement must be explored since the upper limit of slide velocity has been incorporated. A stiffer driving spring, variations in moving masses, and efficiency improved by lowering frictional resistance represent three means of achieving a higher firing rate. All involve refinements in design.

5–1.1.2 Analysis of Cam Action

The forces induced by cam action on the slide and drum roller are shown diagrammatically in Fig. 5–5

for both the recoiling and counterrecoiling slide. Because the slide and drum are constrained in the y- and x-directions, respectively, their motions are restricted to axial and peripheral travel, respectively. Other forces are also present; on the slide, the driving spring force and track reactions; on the drum, the thrust and radial bearing reactions. The accelerating forces on either slide or drum are affected only to the extent of the frictional resistances provided by these reactions.

Before resolving the cam forces, the influence of the drum roller must be considered. If the cam follower were a sliding rather than a rolling element, the tangential frictional force on the cam would be merely μN. The roller reduces μN to a lesser value depending on the ratio of pin radius to roller radius. In the drum roller force diagram of Fig. 5–5, the friction resistance is generated between the roller and the pin since no sliding takes place on the rolling surface. Equate the moments about the pin center.

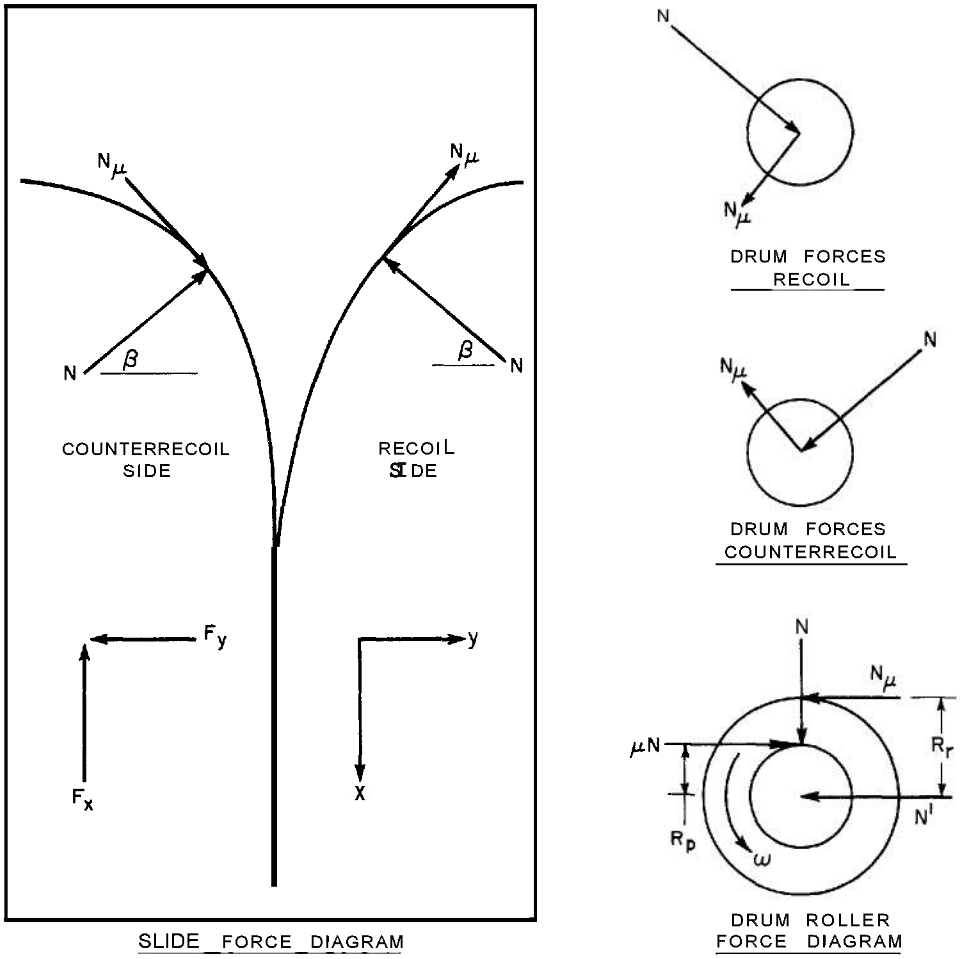


*Figure 5–5. Cam-slide **Force** Diagrams*

$$N_\mu R_r = \mu N R_p \tag{5–29}$$

$$N_\mu = \mu N\left(\frac{R_p}{R_r}\right) \tag{5–30}$$

$$N' = \mu N - N_\mu = \mu N\left(1 - \frac{R_p}{R_r}\right). \tag{5–31}$$

The resultant load on the roller pin becomes

$$F_p = \mu N - N' = \mu N\left(\frac{R_p}{R_r}\right) = N_\mu . \tag{5–32}$$

Resolve the cam forces during slide recoil so that

$$F_x = N\sin\beta + N_\mu\cos\beta$$

$$= N\left[\sin\beta + \mu\left(\frac{R_p}{R_r}\right)\cos\beta\right] = NK_x \tag{5–33}$$

$$F_y = N\cos\beta - N_\mu \sin\beta$$

$$= N\left[\cos\beta - \mu\left(\frac{R_p}{R_r}\right)\sin\beta\right] = NK_y . \tag{5–34}$$

Resolve the cam forces during counterrecoil

$$F_x = N\left[\sin\beta - \mu\left(\frac{R_p}{R_r}\right)\cos\beta\right] = NK_x \tag{5–35}$$

$$F_y = -N\left[\cos\beta + \mu\left(\frac{R_p}{R_r}\right)\sin\beta\right] = NK_y . \tag{5–36}$$

Fig. 5–6 shows the applied and induced forces on the drum. Except for the cam force F_y only the frictional components affect the dynamics. The horizontal reaction *on* the drum shaft is

$$R_y = F_y - \mu F_g \tag{5–37a}$$

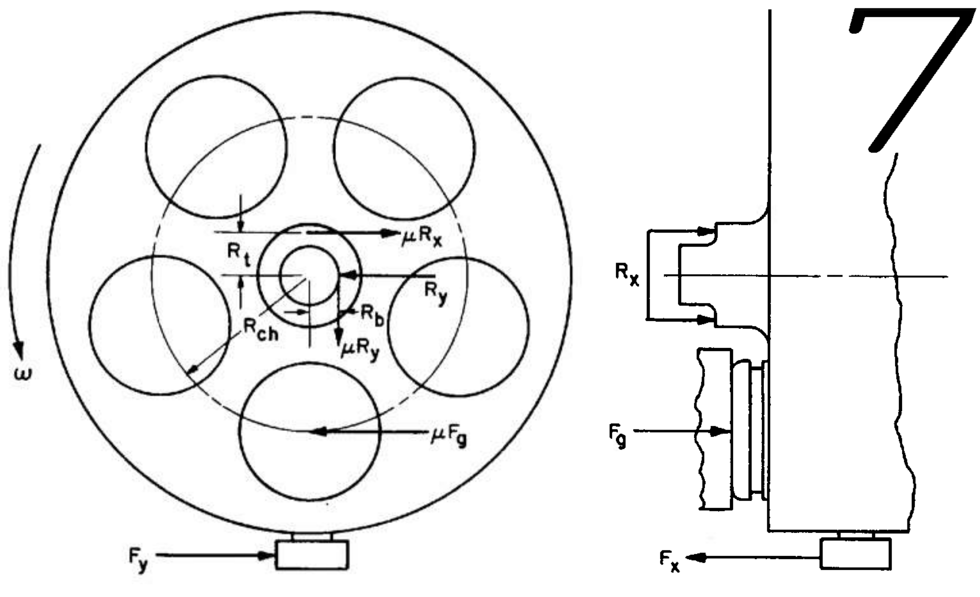


Figure 5–6. Single Barrel Drum Loading Diagram

where

F_g = residual propellant gas force of the round just fired

$$R_x = F_x \tag{5–37b}$$

The frictional force on the thrust bearing μR_x is distributed over the entire annular area and its resultant in any direction is zero. All frictional forces on the drum affect its angular motion. The accelerating torque is expressed as

$$T_\alpha = T_\theta - T_\mu = I_d \alpha_d \tag{5–38}$$

where

I_d = mass polar moment of inertia of drum about its shaft

α_d = angular acceleration of drum

$T_\theta = R_d F_y = N R_d K_y$, applied torque (5–39)

$$T_\mu = R_b \mu R_y + \mu F_x \left[R_t + R_d \left(\frac{R_p}{R_r} \right) \right] + R_{ch} \mu F_g \text{, resisting torque} \tag{5–40}$$

Note that $\mu F_x R_t$ has been substituted for $\mu R_x R_t$. (See Eq. 5–37b)

Observe in Eq. 5–40 that $\mu F_x R_d \left(\frac{R_p}{R_r} \right)$ is the torque resistance contributed by the cam. This expression is derived from the axial component of the cam force, acts in the ydirection, and may be computed by substituting F_x for N in Eq. 5–30.

Substitute for R_x and collect terms, thus

$$T_\mu = \mu \left\{ R_b F_y + \left[R_t + R_d \left(\frac{R_p}{R_r} \right) \right] F_x + (R_{ch} - \mu R_b) F_g \right\} \tag{5–41}$$

Substitute for F_x and F_y and let $\mu(R_{ch} - \mu R_b)F_g = T_g$.

$$T_\mu = \mu N \left\{ \left[R_t + R_d \left(\frac{R_p}{R_r} \right) \right] K_x + R_b K_y \right\} + T_g \tag{5–42}$$

An expression for α can be found from the kinematics of Fig. 5–7. As the cam moves, the relative velocity of the drum roller at any position is v_c. The cam path being curved, the normal acceleration, again at any given position, is

$$a_n = \frac{v_c^2}{R_c} \tag{5–43}$$

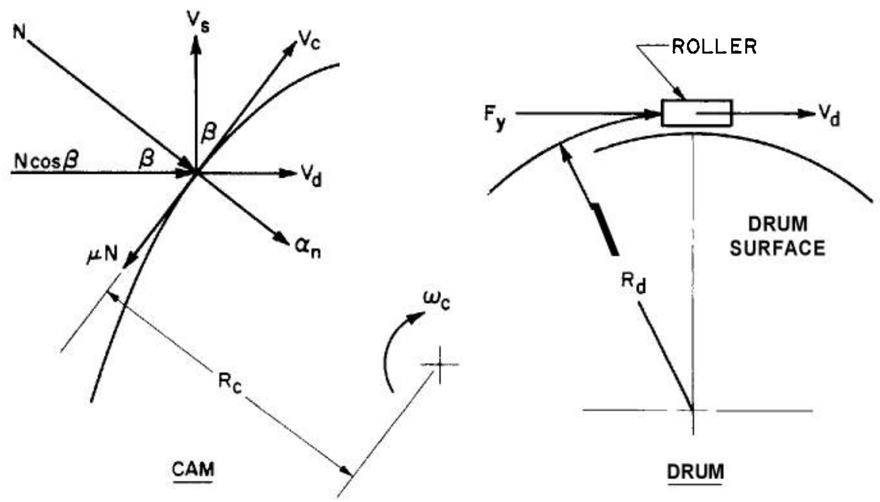


Figure 5–7. Single Barrel Drum Dynamics

However, the roller on the drum can physically travel only in the direction indicated by its tangential velocity v_d, hence the tangential roller acceleration becomes

$$a_d = a_n \cos\beta = \frac{v_c^2}{R_c} \cos\beta \tag{5-44}$$

With $v_s = v_c \cos\beta$, the angular acceleration of the drum may be expressed in terms of the slide velocity.

$$\alpha_d = \frac{a_d}{R_d} = \frac{v_s^2}{R_c R_d \cos\beta} \tag{5-45}$$

Rewrite **Eq.** 5–38 with appropriate substitutions and solve for N. Thus,

$$NR_d K_y - \mu N \left\{ \left[R_t + R_d \left(\frac{R_p}{R_r} \right) \right] K_x + R_b K_y \right\} - T_g = I_d \left(\frac{v_s^2}{R_c R_d \cos\beta} \right) \tag{5-46}$$

$$N = \frac{\dfrac{I_d v_c^2 \cos p}{R_c R_d} + T_g}{(R_d - \mu R_b) K_y - \mu \left[R_t + R_d \left(\dfrac{R_p}{R_r} \right) \right] K_x} \tag{5-47}$$

In the meantime, the slide is subjected to cam forces as well as the driving spring force F and also the frictional resistance μF_y of the slide tracks.

The cam is the medium for transferring energy. The equation of an elliptical cam is

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1 \tag{5-48}$$

or

$$y = \pm \frac{b}{a} \sqrt{a^2 - x^2} \tag{5-49}$$

where

a = half of the major axis in x direction

b = half of the minor axis in y-direction

The slope at any point is $\tan\beta$.

$$\tan\beta = \frac{dy}{dx} = \frac{b}{a}\left(\frac{x}{\sqrt{a^2 - x^2}}\right) \tag{5-50}$$

$$\frac{d^2y}{dx^2} = \frac{ab}{(a^2 - x^2)^{3/2}} \tag{5-51}$$

The radius of the curvature of the cam at any position is R_c

$$R_c = \frac{\left[1.0 + \left(\frac{dy}{dx}\right)^2\right]^{3/2}}{\frac{d^2y}{dx^2}} = \frac{\left[a^2 - \frac{x^2}{a^2}(a^2 - b^2)\right]^{3/2}}{ab} \tag{5-52}$$

The cam dynamics involve an iterative integration procedure for which the law of conservation of energy becomes a convenient basis for computing the values of each increment. For any increment

$$E_i = E_{sr} + E_d + \Delta E + E\mu = \frac{1}{2}\left(Msv_s^2\right) + I_d\left(\frac{\tan^2\beta}{2R_d^2}\right)v_s^2 + \frac{1}{\epsilon}F_a\,\Delta x + E_\mu \tag{5-53}$$

where

ΔE = differential driving spring energy

E_d = drum energy at end of increment

E_i = input energy of each increment

E_{sr} = slide energy at end of increment

F_a = average driving spring force

Ax = incremental travel of slide

= frictional losses during increment

ϵ = spring efficiency.

Note that for the next increment,

$$E_i = \text{preceding } E_i = E_\mu \tag{5-54}$$

The object now is to put E_μ into terms of v_s so that **Eq. 5–53** may be solved. The resultant frictional force in the x-direction is composed of the frictional resistance of slide tracks and that of the cam in the xdirection

$$F_{\mu s} = \mu F_y + \mu F_y\left(\frac{R_p}{R_r}\right) = \mu F_y\left(1.0 + \frac{R_p}{R_r}\right) \quad (5\text{–}55)$$

Write F_y in terms of N.

$$F_{\mu s} = \mu N K_y \left(1.0 + \frac{R_p}{R_r}\right) \quad (5\text{–}56)$$

The energy loss in the slide is

$$E_{\mu s} = \frac{1}{2}(F_{\mu s1} + F_{\mu s2})\Delta x. \quad (5\text{–}57)$$

where subscripts 1 and 2 indicate values for adjacent increments.

The energy loss in the drum becomes

$$E_{\mu d} = \frac{1}{2}(T_{\mu 1} + T_{\mu 2})\Delta\theta. \quad (5\text{–}58)$$

The total frictional losses in drum and slide is the sum of the two components

$$E_\mu = E_{\mu d} + E_{\mu s}. \quad (5\text{–}59)$$

5–1.1.2.1 Sample Calculation of Cam Action

The sample problem is the continuation of the one outlined in par. 5–1.1.1, at a time when the slide has traveled 2.0 inches on the cam. Thus, x_1 = **2.0.** From **Eq. 5–49.**

$$y_1 = \frac{b}{a}\sqrt{a^2 - x^2} = \frac{2.15}{3.33}\sqrt{11.09 - 4}$$

$$= 0.6456 \times 2.6627 = 1.719 \text{ in.}$$

$$\tan\beta_1 = \frac{b}{a}\left(\frac{x}{\sqrt{a^2 - x^2}}\right) = \frac{2.15}{3.33}\left(\frac{2.0}{2.6627}\right)$$

$$= 0.4850 \text{ (from Eq. 5-50)}$$

$$\beta_1 = 25^\circ\ 52.2' = 0.4470 \text{ rad}$$

$$\sin\beta_1 = 0.4364$$

$$\cos\beta_1 = 0.8998$$

$$\frac{d^2y}{dx^2} = \frac{ab}{(a^2 - x^2)^{3/2}} = \frac{3.33 \times 2.15}{18.88}$$

$$= 0.379 \text{ (from Eq. 5–51).}$$

According to Eq. 5–52

$$R_{c1} = \frac{\left[a^2 - \frac{x^2}{a^2}(a^2 - b^2)\right]^{3/2}}{ab}$$

$$= \frac{\left[11.09 - \frac{4.0}{11.09}(11.09 - 4.62)\right]^{3/2}}{7.16}$$

$$= 3.62 \text{ in.}$$

At this time the driving spring has been compressed by

$$L_x = s_{or} + x = 1.67 + 2.0 = 3.67 \text{ in.}$$

The energy absorbed by the spring at this position is E_s.

$$E_s = \frac{3.67}{5.0}\ 931.7 = 684 \text{ in.-lb}$$

The energy confined to the drum-slide system is

$$E_{ds} = E_{s_o} - E_s = 3974.4 \text{ in.-lb.}$$

After losses have been deleted, the energy remaining in the system is

$$E'_{ds} = E_{ds}e^{-\mu_i\mu\frac{x}{a}} \quad \text{(Ref. 14)} \quad (5\text{–}60)$$

$$E'_{ds} = 3974.4\, e^{-0.434} = \frac{3974.4}{1.543} = 2576 \text{ in.-lb}$$

where $-\mu_i \mu \dfrac{x}{a} = -7.23 \times 0.1 \times 2.0/3.33 = -0.434$

$$E'_{ds} = \frac{1}{2}\left(I_{de}\omega_d^2\right) + \frac{1}{2}\left(M_s v_s^2\right)$$

$$= \left(\frac{1}{4} M_{de} \tan^2\beta_1 + \frac{1}{2} M_s\right) v_s^2$$

$$v_s^2 = \frac{2576 \times 386.4}{1.94 + 5.0} = 143{,}425 \text{ in.}^2/\text{sec}^2$$

$$v_s = 378.7 \text{ in./sec}$$

where

$$\frac{1}{4} M_{de} \tan^2\beta_1 = 1.94/g$$

$$\frac{1}{2} M_s = 5.0/g$$

The above given and computed values are assumed to be the values of the parameters at $x = 2.0$ in. To illustrate the integration process, assume an incremental travel of $\Delta x = 0.05$ in.

$$x_2 = x_1 + \Delta x = 2.00 + 0.05 = 2.05 \text{ in.}$$

$$y_2 = \frac{b}{a}\left(\sqrt{a^2 - x^2}\right) = \frac{2.15}{3.33}\sqrt{11.09 - 4.203}$$

$$= 0.6456 \times 2.6243 = 1.694 \text{ in.}$$

$$\tan\beta_2 = \frac{b}{a}\left(\frac{x}{\sqrt{a^2 - x^2}}\right) = \frac{2.15 \times 2.05}{3.33 \times 2.6243}$$

$$= \frac{4.4075}{8.7389} = 0.5044$$

$$\beta_2 = 26^\circ\, 46' = 0.4672 \text{ rad}$$

$$\sin\beta_2 = 0.4504$$

$$\cos\beta_2 = 0.8929$$

$$\Delta y = y_1 - y_2 = 0.025$$

$$\Delta\theta = \frac{\Delta y}{R_d} = \frac{0.025}{3.0} = 0.00833 \text{ rad}$$

$$\frac{d^2y}{dx^2} = \frac{ab}{(a^2 - x^2)^{3/2}} = \frac{3.33 \times 2.15}{18.07} = 0.396$$

$$R_{c2} = \frac{\left[a^2 - \frac{x^2}{a^2}(a^2 - b^2)\right]^{3/2}}{ab}$$

$$= \frac{\left[11.09 - \left(\frac{4.203}{11.09}\right) 6.47\right]^{3/2}}{7.16} = 3.55 \text{ in.}$$

Additional given data are now listed.

F_g = 1000 lb, propellant gas force (residual)

R_b = 1.0 in., radius of radial bearing

R_{ch} = 1.5 in., chamber center to drum axis

R_p = 0.25 in., radius of roller pin

R_r = 0.5 in., roller radius

R_t = 1.25 in., thrust bearing pressure radius

μ = 0.10, coefficient of friction

In Eq. 5–42,

$$T_g = \mu(R_{ch} - \mu R_b)F_g = 0.10(1.5 - 0.10 \times 1.0)1000$$

$$= 140 \text{ in.-lb}$$

$$I_d = M_d k^2 = M_d\left(\frac{R_d^2}{2}\right) = \frac{30}{386.4}\left(\frac{9}{2}\right)$$

$= 0.35$ lb-in.-sec^2, mass moment of inertia of drum

During slide recoil when the ammunition must also be accelerated, the effective mass moment of inertia I_{de} changes from I_d to

$$I_{de} = (1.1)I_d = 0.385 \text{ lb-in.-sec}^2.$$

From Eq. 5–33

$$K_{x1} = \sin\beta_1 + \mu\left(\frac{R_p}{R_r}\right)\cos\beta_1 = 0.4364 + 0.10 \times 0.5 \times 0.8998 = 0.4814$$

$$K_{x2} = \sin\beta_2 + \mu\left(\frac{R_p}{R_r}\right)\cos\beta_2 = 0.4505 + 0.10 \times 0.5 \times 0.8929 = 0.4950.$$

From Eq. 5–34

$$K_{y1} = \cos\beta_1 - \left(\frac{R_p}{R_r}\right)\sin\beta_1 = 0.8998 - 0.10 \times 0.5 \times 0.4364 = 0.8780$$

$$K_{y2} = \mathrm{cosp}, - \left(\frac{R_p}{R_r}\right)\sin\beta_2 = 0.8929 - 0.10 \times 0.5 \times 0.4504 = 0.8704.$$

Since the slide is recoiling, substitute $v_s/\cos\beta$ for v_c in Eq. 5–47, thus

$$N_1 = \frac{\dfrac{I_{de}v_{s1}^2}{R_{c1}R_d\cos\beta_1} + T_g}{(R_d - \mu R_b)K_{y1} - \mu\left[R_t + R_d\left(\dfrac{R_p}{R_r}\right)\right]K_{x1}}$$

$$= \frac{\dfrac{0.385 \times 143425}{3.62 \times 3.0 \times 0.8998} + 140}{(3.0 - 0.10 \times 1.0)\,0.8780 - 0.10\,(1.25 + 3.0 \times 0.5)\,0.4814} - \frac{5651 + 140}{2.546 - 0.132} = \frac{5791}{2.414} = 2399 \text{ lb}$$

$$N_2 = \frac{\dfrac{I_{de}v_{s2}^2}{R_{c2}R_d\cos\beta_2} + T_g}{(R_d - \mu R_b)K_{y2} - \mu\left[R_t + R_d\left(\dfrac{R_p}{R_r}\right)\right]K_{x2}}$$

$$- \frac{\dfrac{0.385\,v_{s2}^2}{3.55 \times 3.0 \times 0.8929} + 140}{2.9 \times 0.8704 - 0.275 \times 0.4950} = \frac{0.0405\,v_{s2}^2 + 140}{2.524 - 0.136} = 0.01696\,v_{s2}^2 + 59.$$

The preliminary characteristics of the driving spring are based on an assumed efficiency of 80% and for $F_m \approx 2F_o$. The average spring force F_a over the full recoil distance is now computed.

$$F_a = \frac{\epsilon E_s}{L_s} = \frac{0.8 \times 931.7}{5} = 149 \text{ lb}$$

$$F_a = \frac{F_o + F_m}{2} = \frac{F_o + 2F_o}{2} = 149$$

$$F_o = \frac{298}{3} = 99.3; \quad F_m = 198.7 \text{ lb}.$$

The spring constant

$$K = \frac{F_m - F_o}{L} = \frac{99.4}{5} = 19.88 \text{ lb/in.}$$

$$F_{x1} = F_o + 3.67K = 99.3 + 73.0 = 172.3 \text{ lb}$$

$$F_{x2} = F_o + 3.72K = 99.3 + 74.0 = 173.3 \text{ lb}.$$

Isolate the components of Eq. 5–53 to compute the combined energy of drum and slide

$$E_i = E'_{ds} = 2576 \text{ in.-lb}$$

$$E_{sr} = \frac{1}{2}\left(M_s v_{s2}^2\right) = \frac{10 v_{s2}^2}{2 \times 386.4} = 0.01294\, v_{s2}^2$$

$$E_d = \frac{1}{2} I_{de}\left(\frac{\tan^2\beta_2}{R_d^2}\right) v_{s2}^2$$

$$= \left(\frac{0.385 \times 0.2544}{2 \times 9}\right) v_{s2}^2 = 0.00544\, v_{s2}^2$$

$$\Delta E = \frac{1}{2} F_{sa} \Delta x = \left(\frac{172.3 + 173.3}{0.8 \times 2}\right) 0.05$$

$$= 10.8 \text{ in.-lb}.$$

Insert the appropriate values in Eqs. 5–42 and 5–56 and compute the torsional and slide frictional resistance.

$$T_{\mu 1} = N_1(0.275K_{x1} + 0.1K_{y1}) + T_g$$

$$= 2399\,(0.1324 + 0.0878) + 140 = 668.3 \text{ lb-in.}$$

$$T_{\mu 2} = N_2(0.275\,K_{x2} + 0.1K_{y2}) + T_g$$

$$= (0.01696 v_{s2}^2 + 59)\,(0.1361 + 0.0870) + 140$$

$$= 0.00378 v_{s2}^2 + 153.2 \text{ lb-in.}$$

$$F_{\mu s1} = 0.15K_{y1}N_1 = 0.15 \times 0.8780 \times 2399$$

$$= 315.9 \text{ lb}$$

$$F_{\mu s2} = 0.15K_{y2}N_2 = 0.15 \times 0.8704(0.01696 v_s^2 + 59)$$

$$= 0.00221 v_{s2}^2 + 7.7 \text{ lb}$$

According to Eqs. 5–57 and 5–58, the energy losses are

$$E_{\mu s} = \frac{1}{2}(315.9 + 7.7 + 0.00221 v_{s2}^2)\,0.05$$

$$= 8.1 + 5.525 \times 10^{-5} v_s^2$$

$$E_{\mu d} = \frac{1}{2}(668.3 + 153.2 + 0.00378\, v_{s2}^2)\,0.00833$$

$$= 3.4 + 1.574 \times 10^{-5} v_{s2}^2$$

Insert computed values in Eq. 5–53 and solve for slide velocity v_s and the energy E_i.

$$2576 = (1294 + 544 + 5.525 + 1.574) v_{s2}^2 \times 10^{-5}$$

$$+ 10.8 + 8.1 + 3.4$$

$$1845.1 \times 10^{-5} v_{s2}^2 = 2553.7$$

$$v_{s2} = \sqrt{138{,}412} = 372 \text{ in./sec}$$

$$E_{\mu} = 11.5 + 7.099 \times 10^{-5}\, v_{s2}^2 = 11.5 + 9.9$$

$$= 21.4 \text{ in.-lb}$$

$$E_i = E_{i(-1)} - \Delta E - E_{\mu} = 2543.8 \text{ in.-lb}$$

5–1.1.2.2 Driving Spring

The average force on each of two driving springs over the decelerating period of the slide is computed from the known slide energy.

$$F_a = \frac{\epsilon E_s}{2a_{rec}} = \frac{0.8 \times 931.7}{2 \times 3.33} = 111.9 \text{ lb.}$$

where

a_{rec} = 3.33 in., slide travel during slide deceleration

E_s = 931.7 in.-lb, slide energy

ϵ = 0.80, system efficiency (assumed)

Investigation shows that K = 20 lb/in. is a practical spring constant. The compression time of the spring

$$T_c = t_r = 1.87 \times 0.0068 = 0.0127 \text{ sec}$$

(see par. 5–1.1.1).

The surge time $T = \frac{T_c}{1.8} = 0.00706$ sec (see par. 2–2.3.5).

Select a spring diameter of D = 1.0 in., then compute the wire diameter according to Eq. 2-42.

$$d = 0.27\sqrt[3]{DKT} = 0.27\sqrt[3]{0.1412} = 0.1406 \text{ in.}$$

Compute the number of coils from Eq. 2-41.

$$N = \frac{Gd^4}{8D^3K} = \frac{11.5 \times 10^6 \times 3.90 \times 10^{-4}}{8 \times 1.0 \times 20} = 28.1 \text{ coils}$$

The spring solid height is

$$H_s = Nd = 28.1 \times 0.1406 = 4 \text{ in.}$$

Since $F_a = \frac{F_o + F_m}{2}$ and $F_o = F_m - Ka_r$

$$F_m = F_a + \frac{1}{2} Ka_{rec} = 111.9 + 33.3 = 145.2 \text{ lb}$$

The static torsional stress, Eq. 2–43, is

$$\tau = \frac{8F_mD}{\pi d^3} = \frac{8 \times 145.2 \times 1.0}{\pi \times 2.78 \times 10^{-3}} = 133{,}200 \text{ lb/in.}^2$$

The dynamic torsional stress, Eq. 2–44, is

$$\tau_d = \tau\left(\frac{T}{T_c}\right)\left[f\left(\frac{T_c}{T}\right)\right]$$

$$= 133{,}200\left(\frac{1}{1.8}\right)2 = 148{,}000 \text{ lb/in.}^2$$

5–1.2 FINAL ESTIMATE OF THE COMPLETE FIRING CYCLE

This sample problem involves the procedures for computing all the data involved for making an accurate estimate of the firing rate for a drum-type machine gun by analyzing the firing cycle in detail. The interior ballistics of Fig. 5–8 are for a 20 mm gun firing a 2000-gram projectile with a 500-grain propellant charge. The pressure has been modified so that the impulse generated by the gas force is congruous with the momentum of projectile and gas from the expression obtained from Eq. 2–14

$$F_g dt = (M_p + \frac{1}{2} M_g)dv$$

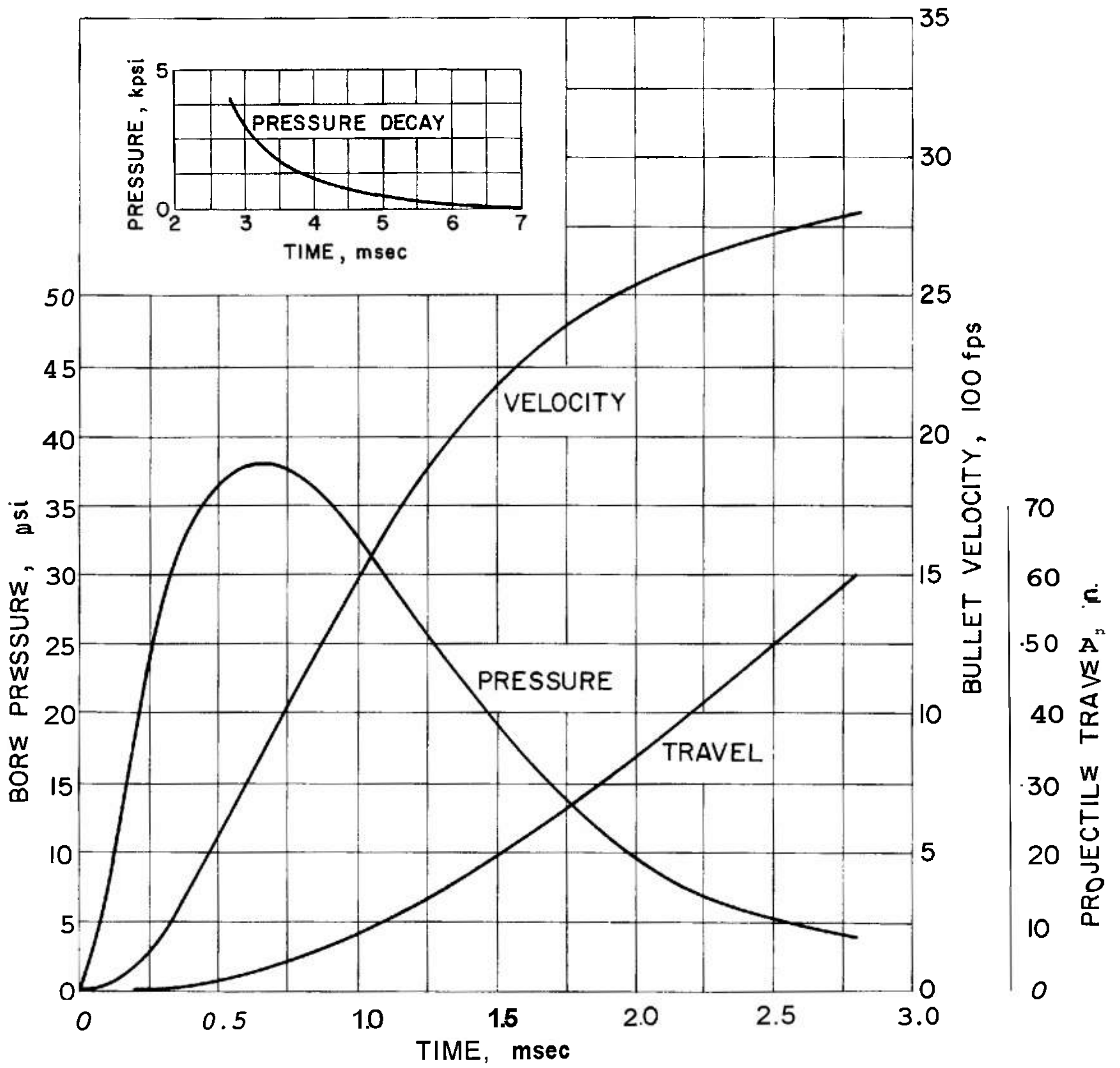


Figure 5-8. Interior Ballistics of 20 mm Revolver-type Gun

Only half the mass of the propellant gas is assumed in motion. This assumption is based on the theory that the velocity of the gas varies linearly as the distance that the projectile has traveled in the bore. It varies from zero in the chamber to the projectile velocity. Thus, if the projectile velocity is v, the momentum of the propellant gas at any time is $M_g v/2$, which indicates that the equivalent mass moving at projectile velocity is $1/2\, M_g$.

5–1.2.1 Control of Recoil Travel During Propellant Gas Period

The given design data include all the given and computed data available from the preliminary firing rate and cam analyses. The additional design data include

L_r = 0.25 in., recoil distance of barrel

L_p = 16 in., location of gas port along barrel

W_r = 96 lb, weight of recoiling parts including 10-lb slide

Table 5–1 shows a free recoil distance of x = 0.572 in. that exceeds the specified travel L_r = 0.25 in. To curb free recoiling tendencies, the adapter resists the propellant gas force at all times. To realize a shorter recoil stroke, the resistance of the adapter and the influence of the gas pressure force in the operating cylinder must be used. This effort is shown in Table 5–2. Since the gas activity in the slide operating cylinder has not been analyzed, its effect on the recoiling parts at this stage is assumed to be included in the adapter performance.

Before computing the data in Table 5–2, some of the data in Table **5–1** are modified to fit more closely the design requirements. The impulse of the propellant gas force forms the basis for the modified values, $F_g\Delta t$ = 29.65 lb-sec. If left uninhibited, a free recoil velocity of **119.4** in./sec is induced, a condition that should not prevail. Lower velocities are achieved by establishing lower effective impulses. The total impulse at t = 1.375 msec when the projectile reaches the gas port $(\Sigma F_g\Delta t)_p$ = 20.31 lb-sec. At this time, the force on the slide operating piston is arbitrarily assumed to exceed the component of the propellant gas force (see Fig. 5–1).

$$F_c > F_g\left(\frac{W_s}{W_r + W_s}\right)$$

Thus separating the slide from the recoiling parts and reducing the latter's weight from 96 to 86 lb tends to increase recoil accelerations, provided that the parts are subjected to the same impulses. The average impulsive force corrected for the change of weight is

$$F_a = \frac{1}{t}\left\{(\Sigma F_g\Delta t)_p + \left[\Sigma F_g\Delta t - (\Sigma F_g\Delta t)_p\right]\frac{86}{96}\right\}$$

$$= \frac{20.31 + 8.37}{0.006} = 4780 \text{ lb}$$

(t = 0.006 sec from last entry in the first column Table 5–2).

The impulse of the propellant gas is present over the entire recoil stroke, therefore, to have the recoil velocity reach zero just at full recoil, a resisting impulse equal to the applied impulse must be made available. This impulse should be distributed so that the full stroke and zero velocity are reached simultaneously which can be achieved by iterative computation with time rather than distance determining the distribution of forces. The initial and final resisting forces are determined from the average with the former low enough not to limit recoil travel too severely, and the latter not to reach loads that cannot be tolerated with respect to handling and structural sizes. During the first 0.25 msec, the average gas force is F_g = 5600 lb. Since motion should not be totally impeded, a resisting force of about half this value, or F_{ot} = 3000 lb may serve as a first trial. The corresponding resistance offered by the adapter is the maximum adapter force.

$$F_{mt} = 2F_a - F_{ot} = 9560 - 3000 = 6560 \text{ lb}$$

The resisting force at any time is based on the initial force of F_{ot}

$$F_t = F_{ot} + \left(\frac{F_{mt} - F_{ot}}{\Delta t}\right)t = 3000 + \left(\frac{3560}{0.006}\right)t$$

$$= 3000 + 593333t$$

TABLE 5–1. FREE RECOIL DATA OF 20 mm REVOLVER-TYPE MACHINE GUN

t, msec	Δt, msec	A_i, lb-sec/in.^{2}	$F_g \Delta t$, lb-sec	Δv_f, in./sec	v_f, in./sec	v_a, in./sec	Δx, in.	x, in.
0.25	0.25	2.72	1.40	5.6	5.6	2.8	0.0007	0.0007
0.50	0.25	8.06	4.15	16.7	22.3	14.0	0.0035	0.0042
0.75	0.25	9.44	4.86	19.6	41.9	32.1	0.0080	0.0122
1.00	0.25	8.84	4.55	18.3	60.2	52.0	0.0130	0.0252
1.25	0.25	7.32	3.77	15.2	75.4	67.8	0.0170	0.0422
1.375	0.125	3.06	1.58	6.4	81.8	78.6	0.0098	0.0520
1.50	0.125	2.72	1.40	5.6	87.4	84.6	0.0106	0.0626
1.75	0.25	4.14	2.13	8.6	96.0	91.7	0.0229	0.0855
2.00	0.25	2.95	1.52	6.1	102.1	99.0	0.0248	0.1103
2.25	0.25	2.06	1.06	4.3	106.4	104.2	0.0260	0.1363
2.50	0.25	1.46	0.75	3.0	109.4	107.9	0.0270	0.1633
2.80	0.30	1.51	0.78	3.1	112.5	111.0	0.0333	0.1966
3.00	0.20	0.73	0.38	1.5	114.0	113.2	0.0226	0.2192
4.00	1.00	1.57	0.81	3.3	117.3	115.6	0.1156	0.3348
5.00	1.00	0.75	0.39	1.6	118.9	118.1	0.1181	0.4529
6.00	1.00	0.24	0.12	0.5	119.4	119.2	0.1192	0.5721

TABLE 5–2. PRELIMINARY RECOIL ADAPTER DATA

t, msec	F_t, lb	$F_t\Delta t$, lb-sec	$F_e\Delta t$, lb-sec	Δv_r, in./sec	v_r, in./sec	v_a, in./sec	Δx, in.	x, in.
0.25	3150	0.79	0.61	2.5	2.5	1.2	0.0003	0.0003
0.50	3300	0.82	3.33	13.4	15.9	9.2	0.0023	0.0026
0.75	3440	0.86	4.00	16.1	32.0	24.0	0.0060	0.0086
1.00	3590	0.90	3.65	14.7	46.7	39.4	0.0098	0.0184
1.25	3740	0.94	2.83	11.4	58.1	52.4	0.0131	0.0315
1.375	3820	0.48	1.10	4.4	62.5	60.3	0.0075	0.0390
1.50	3890	0.49	0.91	4.1	66.6	64.6	0.0081	0.0471
1.75	4040	1.01	1.12	5.0	71.6	69.1	0.0173	0.0644
2.00	4190	1.05	0.47	2.1	73.7	72.6	0.0182	0.0826
2.25	4330	1.08	-0.02	- 0.1	73.6	73.6	0.0184	0.1010
2.50	4480	1.12	-0.37	- 1.7	71.9	72.8	0.0182	0.1192
2.80	4660	1.40	-0.62	- 2.8	69.1	70.5	0.0225	0.1417
3.00	4780	0.96	-0.58	- 2.6	66.5	67.8	0.0136	0.1553
4.00	5370	5.37	-4.56	-20.5	46.0	56.2	0.0562	0.2115
5.00	5970	5.97	-5.58	-25.1	20.9	33.4	0.0334	0.2449
5.74	6410	4.74	-4.65	-20.9	0	10.4	0.0077	0.2526

The resisting impulse at any increment of time is $F_a \Delta t$ and the effective impulse on the recoiling parts is

$$F_e \Delta t = (F_g \Delta t - F_t \Delta t)$$

where $F_g \Delta t$ is read from Table 5–1. The other data of Table 5–2 are computed similarly to those of Table 5–1. When completed, Table 5–2 (for the first trial) shows that the recoiling parts stop before the propellant gas pressure period ends but very close to the 0.25 in. allotted recoil stroke. This close proximity between computed and specified distance is attributed to the choice of F_o = 3000 lb, a shear coincidence. If recoil action over the whole gas period is desired, the adapter force is reduced proportionately and the values of Table 5–2 recomputed.

The recoil data are revised by proportioning the recoiling masses and the corresponding impulses according to time. Since the slide and recoiling parts move as a unit for 1.375 msec and the slide is a separate mass afterwards, the effective mass for the period is

$$M_e = \frac{1.375}{6}(M_s + M_r) + \left(\frac{4.625}{6}\right) M_r$$

$$= \frac{1}{6}\left(\frac{1.375 \times 96 + 4.625 \times 86}{386.4}\right) = \frac{529.75}{2318.4}$$

$$= 0.2285 \text{ lb-sec}^2/\text{in.}$$

A negative velocity of 7.5 in./sec at the end of 6 msec is eliminated by reducing the adapter force of Table 5–2 thereby permitting a larger portion of the gas impulse to act on the recoiling parts. Compute the momentum for the negative velocity and solve for the force of the equivalent impulse.

$$\Delta F = \frac{\Delta Ft}{t} = \frac{M_e \Delta v_r}{t} = \frac{0.2285\,(-\,7.5)}{0.006}$$

$$= -\,286, \text{ say, } -\,300 \text{ lb}$$

By adding −300 lb to each F_t in Table 5–2, a new set of data is computed and arranged in Table 5–3. The final results show practically zero velocity in the allotted time but a larger recoil stroke than the prescribed, which may be corrected by changing the slope of the adapter load while retaining the same area under the force-time curve. However, the data of Tables 5–2 and 5–3 need not be more accurate for preliminary estimates inasmuch as the adapter resistance varies with distance rather than with time. Later when the dynamics of the slide operating cylinder are developed, the recoil analysis will be more precise since the effects of all variables, such as time and distance, will be included.

5–1.2.2 Operating Cylinder Design

Aside from the requirements dictated by the slide, the design data of the operating cylinder are based on four parameters: orifice size, orifice location, cylinder diameter, and stroke. If the cam dwell corresponds with the power stroke, only three parameters need to be resolved. These three parameters may be resolved by searching for compatible relationships among bore pressure, cylinder pressure, and operating cylinder size. An early estimate may be had by calculating the average performance data. The known data at this stage are

s_{or} = 1.67 in., length of power stroke

v_{so} = 62.5 in./sec, slide velocity v_r at t = 1.375 msec (Table 5–2)

v_{sm} = 600 in./sec, maximum slide velocity

W_o = 1.0 lb, weight of moving operating cylinder components

W_{os} = $W_o + W_s$ = 11.0 lb, combined weight of components and slide

W_s = 10 lb, weight of slide

The slide velocity of 62.5 in./sec will be computed more accurately later but is sufficiently accurate for its intended purpose now.

The energy still needed to bring the slide to speed is

$$E_{cs} = \frac{W_{os}}{g}(v_{sm}^2 - v_{so}^2) = \frac{11 \times 356100}{386.4}$$

$$= 10{,}140 \text{ in.-lb.}$$

TABLE 5–3. REVISED PRELIMINARY RECOIL ADAPTER DATA

t, msec	F_t, lb	$F_t\Delta t$, lb-sec	$F_e\Delta t$, lb-sec	Δv_r, in./sec	v_r, in./sec	v_a, in./sec	Δx, in.	x, in.
0.25	2850	0.71	0.69	2.8	2.8	1.4	0.0004	0.0004
0.50	3000	0.75	3.40	3.7	16.5	9.6	0.0024	0.0028
0.75	3140	0.78	4.08	6.4	32.9	24.7	0.0062	0.0090
1.00	3290	0.82	3.73	5.0	47.9	40.4	0.0101	0.0191
1.25	3440	0.86	2.91	1.7	59.6	53.8	0.0134	0.0325
1.375	3520	0.44	1.14	4.6	64.2	61.9	0.0077	0.0402
1.50	3590	0.45	0.95	4.3	68.5	66.4	0.0083	0.0485
1.75	3740	0.94	1.19	5.3	73.8	71.2	0.0178	0.0663
2.00	3890	0.97	0.55	2.5	76.3	75.0	0.0188	0.0851
2.25	4030	1.01	0.05	0.2	76.5	76.4	0.0191	0.1042
2.50	4180	1.04	-0.29	- 1.3	75.2	75.8	0.0190	0.1232
2.80	4360	1.31	-0.53	- 2.4	72.8	74.0	0.0222	0.1454
3.00	4480	0.90	-0.52	- 2.3	70.5	71.6	0.0143	0.1597
4.00	5070	5.07	-4.26	-19.1	51.4	61.0	0.0610	0.2207
5.00	5670	5.67	-5.28	-23.7	27.7	39.6	0.0396	0.2603
6.00	6260	6.26	-6.14	-27.6	0.1	13.9	0.0139	0.2742
		27.98						

The average operating cylinder force is

$$F_{ca} = \frac{E_{cs}}{s_{or}} = \frac{10140}{1.67} = 6072 \text{ lb.}$$

Assume that the gas pressure in the cylinder does not drop below 1000 psi, therefore, flow from bore to operating cylinder ceases at 3.85 msec. The area under the pressure time curve from 1.375 to 3.85 msec is

$$A_t = 17 \text{ lb-sec/in.}^2$$

The average bore pressure for this interval is

$$p_a = \frac{A_t}{\Delta t} = \frac{17}{0.00385 - 0.001375} = \frac{17}{0.002475}$$

$$= 6870 \text{ lb/in.}^2$$

Based on critical flow pressure, the average pressure in the operating cylinder during the same interval is

$$p_c = 0.53 p_a = 3640 \text{ lb/in.}^2$$

The required piston area is

$$A_{cr} = \frac{F_{ca}}{p_c} = \frac{6072}{3640} = 1.67 \text{ in.}^2$$

The corresponding piston diameter d_p = 1.46 in. A nominal diameter of 1.5 in., has an area of

$$A_c = 1.767 \text{ in.}^2$$

The operating cylinder displacement is

$$V_{co} = s_{or} A_c = 1.67 \times 1.767 = 2.95 \text{ in.}^3$$

The initial orifice area is estimated by finding the quantity W_c of gas flowing into the operating clyinder. Eq. 4–49 serves the purpose by substituting p_a for p_m since the muzzle pressure does not apply, hence

$$W_c = W_g \left(\frac{V_{co}}{V_m} \right) \left(\frac{p_c}{p_a} \right)^{1/k} = 0.0039 \text{ lb}$$

where

k = 1.3, ratio of specific heats

V_m = 33.1 in.^{3}, bore volume plus chamber volume

W_g = 500 gr = 0.0714 lb, propellant gas weight

The rate of flow, Eq. 4–51, is

$$w = \frac{W_c}{\Delta t} = \frac{0.0039}{0.002475} = 1.58 \text{ lb/sec.}$$

The first estimate of orifice area, Eq. 4–52, is

$$A_o = \frac{w}{K_w p_a} = \frac{1.58}{0.00192 \times 6870} = 0.120 \text{ in.}^2$$

The orifice diameter = 0.391 in.

Computed data of the operating cylinder are listed in Tables 5–4 to 5–7. The analyses do not consider the influence' of recoil adapter or driving spring since they are an attempt to learn how the gas behaves when entering the operating cylinder. After the nature of gas activity becomes known, all contributing factors to the operating cylinder dynamics will be included in the digital computer program where the effects of their simultaneous activity can be computed in a reasonable time.

Computations in the four tables follow essentially the same procedure. Three values are read directly from Fig 5–8: t, the time; s_b, the bore travel; and p_a, the average bore pressure selected as the pressure falling half way between time intervals. The time of $t \times 10^3$ = 4.00 in Table 5–7 illustrates the procedure. From Eq. 4–27

$$w = K_w A_o p_a = 0.00192 \times 0.06 \times 1700 = 0.196 \text{ lb/sec}$$

where

K_w = 0.00192/sec (see par. 4–3.2.5.2)

The amount of gas capable of passing through the orifice at 1700 psi during the interval of 0.001 sec is

$$\Delta W_c = w \Delta t = 0.196 \times 0.001 = 1.96 \times 10^{-4} \text{ lb.}$$

TABLE 5–4. OPERATING CYLINDER DATA FOR 0.12 in.^{2} ORIFICE (CRITICAL PRESSURE)

$t \times 10^3$, sec	s_b, in.	p_a, psi	w, lb/sec	$\Delta W_c \times 10^3$, lb	$W_c \times 10^3$, lb	V_b, in.^{3}	V_e, in.^{3}	p_c, psi
1.50	19.2	19200	4.424	1.106	1.106	12.1	0.187	4800
1.75	26.2	13700	3.156	0.789	1.895	15.7	0.416	7260
2.00	33.7	9500	2.189	0.547	2.442	19.6	0.669	5040
2.25	41.5	6800	1.567	0.392	2.834	23.6	0.935	3600
2.50	49.7	5200	1.198	0.300	3.134	27.8	1.219	2760
2.80	57.0	4400	1.014	0.304	3.438	31.6	1.519	2330
3.00	60.0	3400	0.783	0.157	3.595	37.9	1.907	1800
4.00		1700	0.392	0.393	3.987	67.5	3.769	900
5.00		740	0.170	0.170	4.157	135.0	7.860	390
6.00		250	0.058	0.058	4.215	334.3	19.73	130

$t \times 10^3$, sec	F_c, lb	$F_c \Delta t$, lb/sec	Δv_s, in./sec	v_s, in./sec	s_1, in.	s_2, in.	s, in.	V_c, in.^{3}
1.50	8480	2.120	74.5	137.0	0.016	0.009	0.025	0.544
1.75	12800	3.200	112.4	249.4	0.034	0.014	0.073	0.629
2.00	8900	2.225	78.2	327.6	0.062	0.010	0.145	0.756
2.25	6400	1.600	56.2	383.8	0.096	0.007	0.248	0.938
2.50	4900	1.225	43.0	426.8	0.082	0.005	0.335	1.092
2.80	4100	1.230	43.2	470.0	0.128	0.005	0.468	1.327
3.00	3200	0.640	22.5	492.5	0.094	0.002	0.564	1.496
4.00	1600	1.600	56.2	548.7	0.492	0.028	1.084	2.415
5.00	690	0.690	24.2	572.9	0.549	0.012	1.645	3.407
6.00	230	0.230	8.1	581.0	0.573	0.004	2.222	4.926

TABLE 5–5. OPERATING CYLINDER DATA FOR 0.12 in.^{2} ORIFICE

$t \times$ 103, sec	s_b, in.	p_a, psi	w, lb/sec	$\Delta W_c \times 10^3$, lb	$W, \times 10^3$, lb	V_b, in.^{3}	V_e, in.^{3}	p_c, psi
1.50	19.2	19200	4.424	1.106	1.106	12.1	0.187	4800
1.75	26.2	13700	3.156	0.789	1.895	15.7	0.416	7690
2.00	33.7	9500	2.189	0.547	2.442	19.6	0.669	7880
2.25	41.5	6800	1.567	0.392	2.834	23.6	0.935	6530
2.50	49.7	5200	1.198	0.300	3.134	27.8	1.219	5200
2.80	57.0	4400	1.014	0.304	3.438	31.6	1.519	
3.00	60.0	3400	0.783	0.157	3.595	37.9	1.907	
4.00		1700	0.392	0.392	3.987	67.5	3.769	
5.00		740	0.170	0.170	4.157	135.0	7.860	
6.00		250	0.058	0.058	4.215	334.3	19.73	

$t \times 10^3$, sec	F_c, lb	$F_c \Delta t$, lb/sec	Δv_s, in./sec	v_s, in./sec	s_1, in.	s_2, in.	s, in.	V_c, in.^{3}
1.50	8480	2.120	74.5	137.0	0.016	0.009	0.025	0.544
1.75	14060	3.515	123.5	260.5	0.034	0.015	0.074	0.631
2.00	13920	3.480	122.2	382.7	0.065	0.015	0.154	0.772
2.25	11540	2.885	101.3	484.0	0.096	0.013	0.263	0.965
2.50	9190	2.298	80.7	564.7	0.121	0.010	0.394	1.196

TABLE 5—6. OPERATING CYLINDER DATA FOR 0.09 in.^{2} ORIFICE

$t \times 103$, sec	s_b, in.	Pa, psi	w, lb/sec	$\Delta W_c \times 10^3$, lb	$W_c \times 10^3$, lb	V_b, in.^{3}	V_e, in.^{3}	p_c, psi
1.50	19.2	19200	3.318	0.830	0.830	12.1	0.141	3360
1.75	26.2	13700	2.367	0.592	1.422	15.7	0.312	5710
2.00	33.7	9500	1.642	0.410	1.832	19.6	0.503	5940
2.25	41.5	6800	1.175	0.294	2.126	23.6	0.702	5140
2.50	49.7	5200	0.898	0.224	2.350	27.8	0.915	4340
2.80	57.0	4400	0.760	0.228	2.578	31.6	1.140	3700
3.00	60.0	3400	0.588	0.118	2.696	37.9	1.430	3220
4.00		1700	0.294	0.294	2.990	67.5	2.826	1700
5.00		740	0.128	0.128	3.118	135.0	5.893	
6.00		250	0.043	0.043	3.161	334.3	14.794	

$t \times 103$, sec	F_c, lb	$F_c \Delta t$, lb-sec	Δv_s, in./sec	v_s, in./sec	s_1, in.	s_2, in.	s, in.	V_c, in.^{3}
1.50	5940	1.485	52.2	114.7	0.016	0.007	0.023	0.541
1.75	10090	2.522	88.6	203.3	0.029	0.011	0.063	0.611
2.00	10500	2.625	92.2	295.5	0.051	0.012	0.126	0.723
2.25	9080	2.270	79.7	375.2	0.074	0.010	0.210	0.871
2.50	7670	1.917	67.3	442.5	0.094	0.008	0.312	1.051
2.80	6450	1.962	68.9	511.4	0.133	0.010	0.455	1.304
3.00	5690	1.138	40.0	551.4	0.102	0.004	0.561	1.491
4.00	3000	3.000	135.0	686.4	0.551	0.068	1.180	2.585

TABLE 5–7. OPERATING CYLINDER DATA FOR 0.06 in.^{2} ORIFICE

t X 103, sec	s_b, in.	p_a, psi	w, lb/sec	ΔW_c X 103, lb	W_c X 103, lb	V_b, in.^{3}	V_e, in.^{3}	p_c, psi
1.50	19.2	19200	2.212	0.553	0.553	12.1	0.094	2000
1.75	26.2	13700	1.578	0.395	0.948	15.7	0.208	3560
2.00	33.7	9500	1.094	0.274	1.222	19.6	0.335	3870
2.25	41.5	6800	0.783	0.196	1.418	23.6	0.468	3550
2.50	49.7	5200	0.599	0.150	1.568	27.8	0.598	3030
2.80	57.0	4400	0.507	0.152	1.720	31.6	0.761	2800
3.00	60.0	3400	0.392	0.078	1.798	37.9	0.954	2490
4.00		1700	0.196	0.196	1.994	67.5	1.884	1580
5.00		740	0.085	0.085	2.079	135.0	3.929	740
6.00		250	0.029	0.029	2.108	334.3	9.866	250

t X 103, sec	F_c, lb	$F_c\Delta t$, lb-sec	Δv_s, in./sec	v_s, in./sec	s_1, in.	s_2, in.	s, in.	V_c, in.^{3}
1.50	3530	0.882	31.0	93.5	0.016	0.004	0.020	0.535
1.75	6290	1.572	55.2	148.7	0.023	0.007	0.050	0.588
2.00	6840	1.710	60.1	208.8	0.037	0.008	0.095	0.668
2.25	6270	1.568	55.1	263.9	0.052	0.007	0.154	0.772
2.50	5350	1.338	47.0	310.9	0.066	0.006	0.226	0.899
2.80	4950	1.485	52.2	363.1	0.093	0.008	0.327	1.078
3.00	4400	0.880	30.9	394.0	0.073	0.003	0.403	1.212
4.00	2790	2.790	98.0	492.0	0.394	0.049	0.846	1.991
5.00	1310	1.310	46.0	538.0	0.492	0.023	1.361	2.905
6.00	440	0.440	15.5	553.5	0.538	0.008	1.907	3.869
		13.975						

The total weight of gas in the operating cylinder is

$$W_c = W_{c(n-1)} + \Delta W_c = (1.798 + 0.196) \times 10^{-3} = 1.994 \times 10^{-3} \text{ lb.}$$

The equivalent volume of the bore, since the projectile has left the muzzle, according to Eq. 4–51 is

$$V_b = V_m \left(\frac{p_m}{p_a}\right)^{1/k_b} = 67.5 \text{ in.}^3$$

where

p_m = 4000 lb/in.^{2}, muzzle pressure

V_m = 33.1 in.^{3}, bore volume plus chamber volume

k_b = 1.2, ratio of specific heats of bore gas

The equivalent gas volume V_e in the cylinder at p_a = 1700 psi is

$$V_e = \left(\frac{W_c}{W_g}\right) V_b = \frac{1.994 \times 10^{-3}}{0.0714} \, 67.5 = 1.884 \text{ in.}^3$$

Since the cylinder volume $V_c = 0.50 + A, s = 0.50 + 1.767 s$ is not known at this time, a trial and error procedure is adopted. First anticipate a change in slide velocity; that for the preceding interval is adequate. Then calculate, in turn, the differential travel, the total travel, the new cylinder gas volume, its pressure, piston force, corresponding impulse and change in slide velocity, and continue until the values converge to prescribed limits. Convergence for these calculations is rapid.

	1st Trial	2nd Trial	3rd Trial
$t = 4 \times 10^{-3}$ sec, $\Delta t = 0.001$ (from Table 5–7)			
n	1	2	3
$\Delta v_s = \Delta v_{s(n-1)}$*	30.9	101.8	98.0
$s_1 = v_{n-1} \Delta t$	0.394	0.394	0.394
$s_2 = \frac{1}{2}\left(\Delta v \Delta t\right)$	0.015	0.051	0.049

*Note that for the first trial $\Delta v_{s(n-1)} = 30.9$ is obtained from Table 5–7 for $t = 0.003$ sec.

	1st Trial	2nd Trial	3rd Trial
$s = s_{n-1} + s_1 + s_2$	0.812	0.848	0.846
$V_c = 0.50 + 1.767s$	1.935	1.998	1.995
$(V_e/V_c)^{1.3}$	0.967	0.927	0.928
$p_c = p_a(V_e/V_c)^{1.3}$	1640	1580	1580
$F_c = A_c p_c = 1.767\, p_c$	2900	2790	–
$F_c \Delta t = 0.001F$	2.90	2.79	
$\Delta v_{s(n)} = \dfrac{F_c At}{M_{cs}} = 35.127\, F_c \Delta t$	101.8	98.0	–

In the third trial, the cylinder pressure equals that of the second trial within three significant figures so that all values after p_c = 1580 psi in the second trial are final and the slide velocity at the end of this interval is

$$v = v_{n-1} + Av = 394.0 + 98.0 = 492.0 \text{ in./sec.}$$

The data in Table 5–4 were computed to ascertain whether the critical flow pressures could be used as the operating cylinder pressure. Under this condition, the travel and corresponding gas volume indicated that the gas flow during the first two increments was sufficient to drive the piston over the rest of its stroke by normal polytropic expansion without the benefit of continued gas flow through the orifice. Since continued gas flow is provided, higher than critical flow pressures are certain. For this reason, the data of Tables 5–5, 5–6, and 5–7 were computed. These tables although similar, show how variations in orifice area lead to specified slide velocity and travel, and help establish acceptable limits in computer programming.

The last values of Table 5–5 indicate that the slide velocity of 50 fps will be reached within less than 30 percent of the stroke. If permitted to function with an orifice of this size, slide velocities would far exceed their limit. A smaller orifice area and hence less pressure would make velocity and travel more compatible. The data of Table 5–6 indicate this trend and those in Table 5–7 almost meet the requirements. A velocity of slightly less than 50 fps is acceptable but to be an accurate estimate, it must be achieved during the complete piston travel. A computed overtravel, a physical impossibility, is not acceptable. Under the conditions enumerated in Table 5–7, the slide velocity lacks approximately 4.5 fps at a travel of 1.67 in. and is definitely acceptable at this stage even though the full 6 msec of effort are not used. The design analysis may now be organized to consider all variables simultaneously.

5–1.2.3 Dynamics of Simultaneous Adapter-Operating Cylinder Action

The resultant force on the recoiling parts (Fig. 5–3) is

$$F_r = F_g - F_c - F_t - F = p_a A_b - p_c A_c - F_t - F \tag{5–61}$$

where

A_b = bore area

$A,$ = piston area of slide operating cylinder

F_t = adapter force

F = driving spring force

F_c = operating cylinder force

F_g = propellant gas force

p_a = average chamber pressure during each increment

p_c = operating cylinder pressure

During each time increment At, the recoiling parts are subjected to the impulse $F_g \Delta t$ that induces a change in velocity defined by Eq 4–55.

$$\Delta v_r = \frac{F_g At}{M_r} \tag{5–62}$$

where M_r represents the mass of all the recoiling parts until the projectile passes the gas port where it loses the burden of the slide and moving operating cylinder components but picks up the operating cylinder force. During counterrecoil it regains the mass of the operating cylinder components. The velocity at the end of each increment is

$$v_r = \Sigma \Delta v_r = v_{r(n-1)} + \Delta v_r . \tag{5–63}$$

The recoil travel is

$$x = x_{n-1} + x_1 + x_2$$

$$- x_{n-1} + \Delta t \Delta v_{r(n-1)} + \frac{1}{2}\left(\Delta t\, \Delta v_r\right) \cdot \tag{5–64}$$

After the projectile passes the gas port and propellant gases begin to act on the slide operating mechanism, the kinematics of the recoiling parts are superimposed on the slide. If the slide unit is isolated, the dynamics of the system follow those expressed in Eqs. 4–51 to 4–58 but modified to fit the prevailing conditions. The resultant force on the piston of the operating cylinder is

$$F_c = A_c p_c - F_d \tag{5–65}$$

The absolute differential velocity of the slide (absolute refers to the nonrecoiling parts as the fixed reference) is

$$Av = \Delta v_r \neq \Delta v_s = \Delta v_r + \frac{F_c \Delta t}{M_{cs}} \tag{5–66}$$

The absolute slide velocity and travel are the same as for the recoiling parts until the projectile passes the gas port. The absolute velocity is

$$v = \Sigma \Delta v = v_{n-1} + \Delta v \ . \tag{5–67}$$

The absolute slide travel is

$$s = s_{n-1} + \Delta t v_{n-1} + \frac{1}{2}\left(\Delta t \Delta v\right) . \tag{5–68}$$

The slide travel with respect to recoil travel is the piston travel, thus

$$x_s = s - x . \tag{5–69}$$

The corresponding gas volume in the operating cylinder becomes

$$V_c = V_{co} + A_c x_s . \tag{5–70}$$

5–1.2.4 Sample Calculation for Complete Firing Cycle

The preliminary calculations summarized in Tables 5–3 and 5–7 provide the initial values for the complete firing cycle analysis. The functioning times of each are identical to the propellant gas period and, although the final results do not conform exactly to specifications, they are close enough to be acceptable; all fall within design specification acceptance limits. To present simultaneous activity, the effects of the adapter and slide forces during the gas period must be synchronized. In Table 5–3, $\Sigma F_t \Delta t = 27.98$ lb-sec; in Table 5–7, $\Sigma F_c \Delta t = 13.98$ lb-sec. Based on time, the average adapter force

$$F_a = \frac{\Sigma F_t \Delta t - \Sigma F_c \Delta t}{t} = \frac{14}{\cancel{0.006}} = 2330 \text{ lb}$$

Maintain the proportions of Table 5–3 where F_a = 27.98/0.006 = 4660 lb, thus the minimum F_{ot} and maximum F_{mt} adapter forces are

$$F_{ot} = \left(\frac{2330}{4660}\right) 2850 = 1425 \text{ lb}$$

where 2850 is first value in F_t column of Table 5–3, and

$$F_{mt} = \left(\frac{2330}{4660}\right) 6260 = 3130 \text{ lb}$$

where 6260 is the last value in F_t column of Table 5–3.

Convert these limits to forces of a ring spring having a conical angle $\alpha = 15^\circ$, a coefficient of friction $\mu = 0.10$, and an efficiency of $\epsilon = 0.45$ (Ref. 15),

$$F_o = F_{ot} = 0.45 \times 1425 = 640 \text{ lb}$$

$$F_m = F_{mt} = 0.45 \times 3130 = 1410 \text{ lb}$$

Distance now, as well as time, becomes a critical parameter in the analysis. For a recoil travel of 0.25 in., the equivalent spring constant of the ring spring is K_t.

$$K_t = \frac{F_m - F_o}{x} = \frac{1410 - 640}{0.25} = 3080 \text{ lb/in.}$$

The average adapter force for any differential recoil travel Δx is

$$F_a = \frac{1}{\epsilon}\left[F_{n-1} + \frac{1}{2}\left(K_t \Delta x\right)\right] = \frac{1}{0.45}\left[F_{n-1} + 1540\Delta x\right]$$

The two driving springs offer a similar but milder effort. Their combined characteristics follow:

K = 40 lb/in., spring constant

F_o = 85 lb, minimum operating load

F_m = 285 lb, maximum operating load

e = 0.80, efficiency

The average driving spring load for any differential travel Δs of the operating slide is

$$F = \frac{1}{\epsilon}\left[F_{(n-1)} + \frac{1}{2}\left(K\Delta s\right)\right] = \frac{1}{0.80}\left[F_{(n-1)} + 20\Delta s\right]$$

where $\Delta s = \Delta x$ until the projectile passes the gas port. Observe that after propellant gases become active in the operating cylinder, F loses its identity by becoming a component of F_c which heretofore had been zero.

5-1.2.4.1 Counterrecoil Time of Recoiling Parts

By restricting the barrel-drum unit to linear travel only during countenecoil, the time required for the activity according to Eq. 2-27 is

$$t_{cr} = \sqrt{\frac{M_r}{\epsilon_t K_t}} \cos^{-1}\frac{F_o}{F_m} = \sqrt{\frac{86}{0.45 \times 3080 \times 386.4}} \cos^{-1}\frac{640}{1410}$$

$$= 0.01303 \times \cos^{-1} 0.4539 = 0.01303 \times 1.1 = 0.01433 \text{ sec.}$$

Earlier, the total time of recoil was estimated to be t_r = 0.0127 sec.* Since 0.006 sec has been consumed for the 1.67 in. of recoil, the remaining time of 0.0067 sec indicates that the counterrecoil of barrel and slide will overlap and a component of the adapter force will be transmitted to the slide-rotating drum combination. Because of the simultaneous activity, the applied force on the slide will be modified according to the involved masses and the cam slope. The components of the adapter force allotted to recoiling parts, drum, and slide are found by a procedure based on the laws of conservation of momentum and energy. Equate the adapter impulse to the linear momentum of recoiling parts and slide so that

$$\bar{F}_t dt = M_r v_{cr} + M_s v_s = M_r v_{cr} + M_s v_c \cos\beta \qquad (5-71)$$

where $\bar{F}_t$ is the average adapter force for the time interval dt and v_c is the velocity of the cam follower along the cam. This form of showing the slide velocity is adopted to avoid the use of tan β which eventually becomes infinite and cannot be used in the digital computer. The energy of the adapter distributed to the various moving element is

$$\frac{1}{\epsilon_t} \bar{F}_t \Delta x = \frac{1}{2}\left(M_r v_{cr}^2\right) + \frac{1}{2}\left(M_s v_c^2 \cos^2\beta\right) + \frac{1}{2}\left(M_{de} v_c^2 \sin 2\beta\right) \qquad (5-72)$$

where

M_{de} = effective mass of the rotating drum.

The two simultaneous equations may be solved by obtaining the expression for v_{cr} in Eq. 5–71 and substituting it into Eq. 5–72. This process merely involves algebraic gymnastics and, since the solution is unwieldly, the expressions for the two velocities are left in their present state. However, with the various constants known, the solutions reduce to a simple quadratic equation of the order

$$Av_c^2 + Bv_c - C = 0 \qquad (5-73)$$

*Par. 5–1.1.2.2

One great advantage inherent in the revolver-type machine gun is the independence of loading and ejecting. Both occur simultaneously with neither interfering with the other. Cam action is illustrated in Figs. 5–9 and 5–10 which show the mechanics of operation. The striker and extractor are fixed to and move with the operating slide whereas the extractor mechanism is fixed to the drum housing and moves with it. As the drum completes its angular travel, the spent cartridge case has moved into contact with the extractor and, in the meantime, lifting the antidouble feed safety switch to break the electric firing circuit so that inadvertent firing of the newly positioned round is precluded. During counterrecoil, the return cam has enough clearance to avoid contact with the extractor cam but, at a prescribed position, the striker hits the extractor cam with the impact needed to rotate the extractor, thereby extracting and ejecting the empty case. After the cam leaves, the antidouble feed safety switch drops into place to reclose the firing circuit. Extraction failure maintains an open circuit until the malfunction is corrected.

During slide recoil, the extractor return cam forces the extractor cam into its normal position. The torsion spring does not activate the extractor, being used primarily as a safety to hold the extractor firmly in position. The cams transmit all effort from slide to extractor. Relative dimensions must comply with required ejection velocity. Once the counterrecoil velocity of the slide is estimated, the ratio of extractor radius to striker radius, r_e/r_s, can be arranged to fit the ejection velocity requirement. The required ratio

$$\frac{r_e}{r_s} = \frac{v_e}{v_{scr}} \qquad (5-74)$$

where

v_{scr} = counterrecoil velocity of the slide

The ejection velocity is assumed immediately at impact of the striker on the ejector cam, resulting in a change of momentum of all involved moving parts. According to the conservation of momentum,

$$M_s v'_{scr} = M_s v_{scr} + M_{ee} v_e \qquad (5-75)$$

where

M_{ee} = effective mass of extractor unit

v'_{scr} = slide velocity before impact.

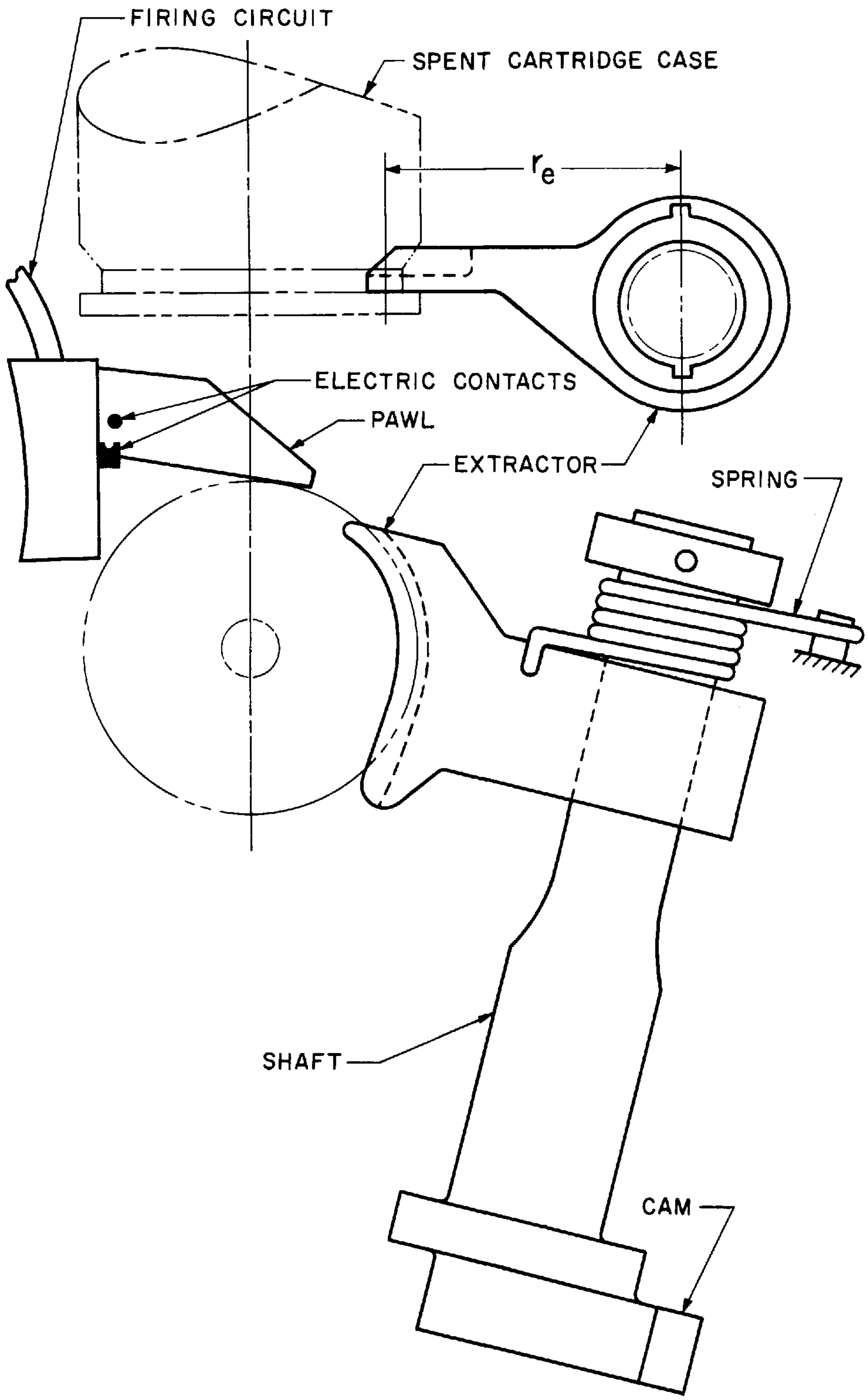


Figure 5–9. Extractor Assembly With Antidouble Feed Mechanism

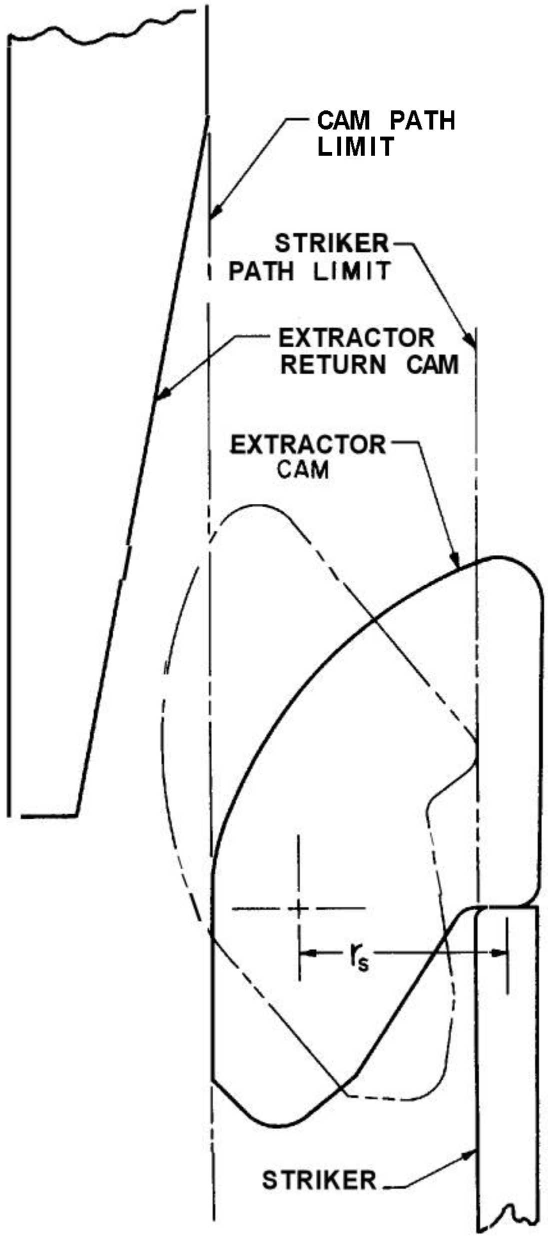


Figure 5–10. Extractor Cam Assembly

5-1.2.4.2 Digital Computer Analyses of Barreldrum Dynamics

Three digital computer programs are compiled in FORTRAN IV language for the UNIVAC 1107 computer. The first computes the data and performance characteristics during the activity of the gas operating cylinder. This program follows the same general procedure that was used for computing the data of Table 5–7 where data cover the effective propellant gas period of *6* msec and the slide travel of 1-2/3 in. The symbol-code relationships are shown in Table **5–8,** the input data as well as the computed results are printed in Table 5–12. In the Appendix, A–1 is the flow chart and **A–8** is the program listing.

The second program begins where the first terminates; just as the follower enters the curved portion of the cam to start the drum rotating, then continuing until the follower has traversed the accelerating portion of the cam and the slide has completed its rearward travel. The third program begins here and computes the data for the decelerating portion of the cam and the first part **of** the slide travel during counterrecoil. Both programs follow the procedure outlined in par. 5–1.2.2. Since the second and third programs are similar, differing only because of direction, the symbol-code relationships (Table 5–9) serve both programs. Inputs are listed in Tables 5–10 and 5–11 for the recoil and the counterrecoil dynamics, respectively.

Computed results are printed in Table 5–13 for the cam and drum dynamics during recoil and in Table 5–14 for the dynamics during counterrecoil. The flow chart and program listing are found in Appendixes A–9 and A–10 for recoil and, in A–11 and A–12, respectively, for counterrecoil.

TABLE 5–8. SYMBOL-CODE CORRELATION **FOR** OPERATING CYLINDER

Symbol	Code	Symbol	Code
A_o	AO	V_b	VB
F	FD	V_c	VC
$F\Delta t$	FDT	V_{co}	VCO
F_c	FC	V_e	VE
$F_c\Delta t$	FCDDT	v	V
F_r	F	v_s	VS
F_t	FA	Δv	DV
M_e	EME	Δv_s	DVS
p_a	PA	W_c	WC
p_c	PC	ΔW_c	DWC
s	S	w	W
s_1	S1	x	X
s_2	s 2	x_1	X1
As	DS	x_2	x 2
t	T	x_s	XS
At	DT	Ax	DX

TABLE 5–9. SYMBOL-CODE CORRELATION FOR CAM DYNAMICS

Symbol	Code	Symbol	Code
C_x	CX	R_d	RD
	CY	s	S
E	E	As	DSI
$E_{bb\ell}$	EBBL	s_x	SX
	DEBBL	Δs_x	DSX
E_d	ED	T_g	TG
E_i	EI	T_μ	TMU
E_μ	EMU	t	TM
$E_{\mu d}$	EMD	At	DELT
$E_{\mu s}$	EMS	Δt_{cr}	DTCR
F	FD	v	V
F_t	FA	v_c	VC
[illegible]	FX	v_d	VD
F_y	FY	v_s	VS
[illegible]	FUS	x	X
I_d	DIE	Ax	DXI
K_y	YK	Δx_o	DXIO
K_x	XK	Δx_1	DXI1
M_r	EMR	Y	Y
M	EMSL	β	BETAD
N	EN	θ	THETA
R_c	RC	$\Delta\theta$	DTHETA

TABLE 5–10. INPUT DATA FOR DRUM DYNAMICS DURING RECOIL

Symbol	Data	Code	Data
CX	0.275	FA (17)	1278.72
CY	2.9	FD (17)	151.8
DF	0.15	S (17)	1.67
DIE	0.385	TM (17)	6.0
EMR	0.22	VC (17)	478.8
EMSL	0.02588	VS (17)	478.8
RD	3.0	X (16)	0.224
TG	140.0	X (17)	0.224

TABLE 5–11. INPUT DATA FOR DRUM DYNAMICS DURING COUNTERRECOIL

Code	Data	Code	Data
CY	2.9	FX (2)	333.0
CX	0.275	FY (2)	14.25
DF	0.15	S (2)	5.0
DIE	0.35	SX (2)	0.0
EMR	0.22	TM (1)	18.6
EMSLR	0.029	TM (2)	19.5968
RD	3.0	TMU (2)	50.014
TG	140.0	V (2)	0.0
BETAD (2)	90.0	VC (2)	199.1
E (2)	848.0 1	VD (2)	199.1
EN (2)	333.0	VS (2)	0.0
FA (2)	1256.0	X (1)	0.21 12
FD (2)	285.0	X (2)	0.21 13
FUS (2)	2.1	Y (2)	0.0

5–1.2.4.3 Firing Rate Computation

Just as the cam completes its cycle, the drum reaches its next firing position and rotation has ceased. According to the print out of the values, neither recoiling parts nor slide has completed its return. The prevailing conditions are

t_e = 0.0329 sec,* elapsed time since firing

s_{or} = 1.67 in., position where slide contacts gas operating unit

F_t = 1210 lb,* recoil adapter force

F = 204 lb,* driving spring force

K_t = 1780 lb/in., recoil adapter spring constant

K = 40 lb/in., driving spring constant of 2 springs

v_{cr} = 8.91 in./sec,* barrel counterrecoil velocity

v_{scr} = 231.6 in./sec,* slide counterrecoil velocity

*Obtained from Table 5–14.

s = 2.974 in.,* remaining counterrecoil travel of slide

x = 0.1854 in.," remaining counterrecoil travel of barrel

W_r = 85 lb, weight of barrel and other recoiling parts

W_{sr} = 11.2 lb, weight of slide with 2 rounds

W_{srp} = 12.2 lb, W_{sr} + weight of gas operating unit

ϵ_t = 0.45, efficiency of recoil adapter

ϵ = 0.80, efficiency of driving spring

According to **Eq.** 2–26, the time of counterrecoil under spring action

$$t'_{cr} = \sqrt{\frac{M_r}{\epsilon_t K_t}} \left(\mathrm{Sin}^{-1} \frac{K_t x - F_t}{f(F)} - \mathrm{Sin}^{-1} \frac{-F_t}{f(F)} \right)$$

where

$$f(F) = \sqrt{F_t^2 + \frac{K_t}{\epsilon_t} \left(M_r v_{cr}^2 \right)}$$

TABLE 5–12. COMPUTED RECOIL AND OPERATING CYLINDER DATA FOR ORIFICE AREA OF 0.042 in.^{2}

TIME MILSEC	BORE VOLUME CU IN	AVERAGE BORE PRESSURE PSI	PROP GAS FORCE LB	ADAPTER FORCE LB	DRIVING SPRING, FORCE LB	RESULT RECOIL FORCE LB	RECOIL IMPULSE LB-SEC	DIFFER RECOIL VEL IN/SEC	RECOIL VEL IN/SEC	DIFFER RECOIL TRAVEL IN	RECOIL TRAVEL IN
.250	2.2	10800.	5562.	880.	85.	3439.	.87	3.5	3.3	.0004	.0004
.500	3.0	32200.	16583.	885.	85.	14510.	3.63	14.6	18.1	.0027	.0031
.750	4.3	37700.	19415.	897.	85.	17315.	4.33	17.4	33.5	.0067	.0099
1.000	6.3	35300.	18179.	917.	86.	16035.	4.01	16.1	51.7	.0109	.0208
1.250	9.0	29300.	15089.	942.	86.	12887.	3.22	13.0	64.6	.0145	.9353
1.375	10.5	24500.	12617.	957.	87.	10382.	1.30	5.2	69.9	.0084	.0437
1.500	12.1	21800.	11227.	973.	87.	7745.	.97	4.4	74.3	.0090	.0527
1.750	15.7	16500.	8497.	1007.	88.	1725.	.43	2.0	76.2	.0188	.0715
2.000	19.6	11800.	6077.	1040.	89.	-2650.	-.66	-3.0	73.2	.0187	.0902
2.250	23.6	8200.	4223.	1072.	91.	-4565.	-1.14	-5.2	68.0	.0177	.1079
2.500	27.8	5800.	2987.	1101.	94.	-4989.	-1.25	-5	62.4	.0163	.1242
2.800	33.1	4500.	2317.	1132.	97.	-5093.	-1.53	-6.9	55.4	.0177	.1418
3.000	37.9	3700.	1905.	1151.	100.	-5042.	-1.01	-4.6	50.8	.0106	.1525
4.000	67.5	1600.	824.	1224.	115.	-4243.	-4.24	-19.3	31.5	.0412	.1937
5.000	135.0	760.	391.	1265.	133.	-3 63.	-3.76	-17.1	14.4	.0230	.2167
6.000	334.3	230.	118.	1278.	152.	-3129.	-3.13	-14.2	.2	.0073	.2240

TIME MILSEC	GAS FLOW RATE LB/SECXM	OPER CYL GAS LBXM	EQUIV CYL VOLUME CU IN	OPER CYL VOLUME CU IN	OPER CYL PRES PSI	OPER PISTON FORCE LB	OPER CYL IMPULSE LB-SEC	DIFFER SLIDE VEL IN/SEC	SLIDE VEL IN/SEC	DIFFER SLIDE TRAVEL IN	SLIDE TRAVEL IN
.250	.0	.000	.000	.000	0.	0.	.000	3.5	3.5	0000	.000
.500	.0	.000	.000	.000	0.	0.	.000	14.6	16.1	.003	.003
.750	.0	.000	.000	.000	0.	0.	.000	17.4	35.5	.007	.010
1.000	.0	.000	.000	.000	0.	0.	.000	16.1	51.7	.011	.021
1.250	.0	.000	.000	.000	0.	0.	.000	13.0	64.6	.015	.035
1.375	.0	.000	.000	.000	0.	0.	.000	5.2	69.9	.008	.044
1.500	1763.0	.220	.037	.501	746.	1318.	.151	5.3	75.2	.009	.053
1.750	1334.4	.554	.122	.510	2566.	4535.	1.106	38.9	114.0	.024	.077
2.000	954.3	.793	.217	.538	3631.	6415.	1.576	55.4	169.4	.035	.112
2.250	663.2	.958	.317	.593	3626.	6406.	1.573	35.3	224.6	.049	.161
2.500	469.1	1.076	.419	.673	3130.	5530.	1.353	47.5	272.2	.061	.222
2.800	363.9	1.183	.349	.797	2770.	4894.	1.432	50.3	322.5	.088	.310
3.000	299.2	1.245	.660	.897	2485.	4390.	.853	30.0	352.4	.067	.377
4.000	129.4	1.374	1.298	1.498	1328.	2347.	2.203	77.4	429.6	.381	.759
5.000	61.5	1.435	2.713	2.239	760.	1343.	1.177	41.3	471.1	.442	1.201
6.000	18.6	1.454	6.805	3.052	230.	406.	.217	7.6	478.8	.468	1.668

TABLE 5–13. CAM AND DRUM DYNAMICS DURING RECOIL

TIME MILSEC	RECOIL ADAPTER FOHCE LB	DRIVING SPRING FORCE LB	NORMAL CAM FORCE LB	AXIAL CAM TRAVEL IN	PERIPH CAM TRAVEL IN	CAM SLOPE DEGREE	DRUM VEL IN/SEC	COUNTER RECOIL VEL IN/SEC	COUNTER RECOIL POSITION IN	SLIDE VEL IN/SEC	SLIDE TRAVEL IN
6.18857	1279.	155.4	2010.	.0900	.0009	1.0	8.3	.4308	.2240	476.2	1.760
6.37862	1278.	159.0	1988.	.1800	.0033	2.0	16.5	.7931	.2238	472.2	1.850
6.57050	1278.	162.6	1967.	.2700	.0072	3.0	24.6	1.1304	.2237	467.9	1.940
6.76435	1278.	166.2	1947.	.3600	.0128	4.0	32.5	1.4427	.2234	463.3	2.031
6.96032	1277.	169.8	1928.	.4500	.0199	5.0	40.4	1.7303	.2231	458.4	2.121
7.15858	1276.	173.5	1911.	.5400	.0286	6.1	48.1	1.9927	.2227	453.2	2.211
7.35930	1276.	177.1	1895.	.6300	.0390	7.1	55.7	2.2296	.2223	447.8	2.302
7.56266	1275.	180.7	1880.	.7200	.0510	8.1	63.2	2.4405	.2218	442.0	.
7.76886	1274.	184.3	1865.	.8100	.0647	9.2	70.6	2.6252	.22 3	436.0	2.483
7.97811	1273.	187.9	1852.	.9000	.0802	10.3	77.9	2.7826	.2208	429.6	2.573
8.19065	1272.	191.6	1840.	.Y9	.0974	11.4	85.0	2.9121	.2202	423.0	2.664
8.40671	1271.	195.2	1829.	1.0800	.1164	12.5	92.1	3.0123	.2195	416.0	2.754
8.62658	1270.	198.8	1819.	1.1700	.1372	13.6	99.0	3 .0823	.2189	408.7	2.845
8.84718	1268.	202.4	1859.	1.2600	.1600	14.8	105.9	3.1123	.2182	401.1	2.934
9.07557	1267.	206.0	1802.	1.3500	.1847	16.0	112.6	3.1089	.2175	393 0 1	3.025
9.30510	1266.	209.6	1848.	1.4400	.2116	17.2	119.2	3.0679	.2168	384.9	3.114
9.53976	1265.	213.1	1845.	1.5300	.2405	18.5	125.6	2.9786	.2161	376.2	3.204
9.78003	1263.	216.7	1842.	1.6200	.2717	19.8	132.0	2.8450	.2154	367.1	3.293
10.02648	1262.	220.3	1841.	1.1100	.3053	21.1	13LI.Z	2.6625	.2147	357.7	3.382
10.28347	1261.	223.9	1777.	1.8000	.3413	22.5	144.3	2.4429	.2140	347.8	3.473
10.54794	1260.	227.5	1775.	1.8900	.380	24.0	150.2	2.1809	.2134	337.5	3.564
10.82061	1259.	231.2	1775.	1.9800	.4215	25.5	156.0	1.8591	.2129	326.7	3.654
11.10237	1258.	234.8	1775.	2.0700	.466	2701	161.6	1.4711	.2124	315.5	3.745
11.39427	1257.	238.4	1777.	2.1600	.5138	28.8	167.1	1.0082	.2120	303.7	3.835
11.69756	1257.	242.0	1781.	2.2500	.5 5	30.6	172.4	.4612	.2118	291.3	3.925
11.93028	1257.	244.7	1785.	2.3166	.6057	32.0	176.2	.0000	.2118	281.7	3.992
12.25745	1257.	248.3	1885.	2.4066	.664	34.0	181.0	.0 00	.2117	268.2	4.082
12.60215	1257.	251.9	1900.	2.4966	.7274	36.2	185.6	.0000	.2117	253.8	4.172
12.96759	1257.	255.5	1918.	2.5866	.7960	38.5	189.9	.000	.2 16	238.5	4.262
A3.35799	1257.	259.1	1940.	2.6766	.8710	41.1	193.9	.0000	.2116	222.3	4.352
13.77909	1257.	262.7	1968.	2.7666	.9535	43.9	197.5	.0 0	.2116	204.9	4.442
14.23911	1256.	266.3	2005.	2.8566	1.0452	47.1	200.6	.0000	.2115	186.2	4.532
14.75046	1256.	269.9	2056.	2.9466	1.1485	50.8	203.2	.0000	.2115	165.7	4.622
15.33356	1256.	273.5	2133.	3.0366	1.2677	55.1	205.0	.0000	.2114	142.9	4.712
16.02666	1256.	277.1	2268.	3.1266	1.4102	60.4	205.6	.0000	.2114	116.7	4.802
16.92005	1256.	280.7	2605.	3.2166	1.5938	67.5	204.1	.0000	.2114	84.7	4.892
19.59680	1256.	285.2	–	3.3300	2.1500	90.0	199.1	.0000	.2113	– 0	5.005

TABLE 5–14. CAM AND DRUM DYNAMICS DURING COUNTERRECOIL

TIME	RECOIL ADAPTER FORCE	DRIVING SPRING FORCE	NORMAL CAM FORCE	AXIAL CAM TRAVEL	PERIPH CAM TRAVEL	CAM SLOPE	DRUM VEL	COUNTER RECOIL VEL	COUNTER RECOIL POSITION	SLIDE VEL	SLIDE TRAVEL
MILSEC	LB	LB	LB	IN	IN	DEGREE	IN/SEC	IN/SEC	IN	IN/SEC	IN
22.17219	1254.	281.0	1405.	.100	.5058	67.9	193.3	.0000	.2103	78.4	4.899
23.26010	1253.	276.9	1346.	.200	.7061	59.1	179.4	.0000	.2093	107.2	4.798
24.11989	1251.	272.9	1238.	.300	.8534	52.6	166.9	.0000	.2083	127.7	4.697
24.86381	1249.	268.8	1167.	.400	.9720	47.2	155.3	.0000	.2073	143.8	4.596
25.53465	1247.	264.8	1114.	.500	1.0715	42.6	144.4	.0000	.2063	157.2	4.495
26.15502	1245.	260.8	1074.	.600	1.1569	38.4	133.8	.0000	.2053	168.5	4.394
26.75740	1244.	256.7	1041.	.700	1.2311	34.7	123.5	.0000	.2043	178.3	4.293
27.28523	1244.	252.7	1029.	.800	1.2960	31.3	113.5	.0000	.2043	186.9	4.193
27.81027	1243.	248.7	1006.	.900	1.3530	28.1	103.7	.2408	.2042	194.3	4.093
28.31741	1243.	244.7	986.	1.000	1.4030	25.1	93.9	.6028	.2040	200.9	3.993
28.81013	1242.	240.7	970.	1.100	1.4467	22.2	84.4	1.0728	.2036	206.7	3.892
29.29120	1241.	236.7	957.	1.200	1.4848	19.5	74.9	1.6378	.2[illegible]	211.8	3.792
29.76285	1240.	232.6	945.	1.300	1.5175	16.8	65.4	2.2897	.2020	216.2	3.691
30.22697	1237.	228.6	936.	1.400	1.5454	14.3	56.0	3.0212	.2008	220.0	3.590
30.68518	1235.	224.5	929.	1.500	1.5686	11.8	46.7	3.8271	.1992	223.3	3.488
31.13890	1231.	220.4	924.	1.600	1.5873	9.4	37.4	4.7035	.1973	226.0	3.386
31.58948	1227.	216.3	920.	1.700	1.6017	7.0	28.0	5.6505	.1950	228.2	3.284
32.03813	1222.	212.2	918.	1.800	1.6119	4.7	18.7	6.6670	.1923	229.8	3.181
32.48601	1217.	208.1	918.	1.900	1.6180	2.3	9.4	7.7509	.1891	231.0	3.078
32.93430	1210.	204.0	918.	2.000	1.6200	.0	.0	8.9058	.1854	231.6	2.974

For counterrecoil of the barrel

$$f(F) = \sqrt{1210^2 + \frac{1780}{0.45}\ (0.22)\ \ 8.91^2} = 1238.2.$$

The time required to complete the counterrecoil of the barrel

$$t'_{cr} = \sqrt{\frac{0.22}{0.45 \times 1780}} \left(\text{Sin}^{-1}\ \frac{1780 \times 0.1854 - 1210}{1238.2} - \text{Sin}^{-1}\ \frac{-1210}{1238.2} \right)$$

$$= 0.01657\ [\text{Sin}^{-1}\ (-0.71070) - \text{Sin}^{-1}\ (-0.97722)]$$

$$t'_{cr} = 0.01657 \left(\frac{314.71 - 282.25}{57.296} \right) = 0.0094 \text{ sec.}$$

The energy of the moving unit at the end of counterrecoil

$$E_{cr} = \frac{1}{2} \left(M_r v_o^2 \right) + \epsilon_t \left(F_t - \frac{1}{2}\ K_t x \right) x$$

$$= \frac{1}{2} \left(\frac{85}{386.4} \right) 8.91^2 + 0.45 \left[1210 - \frac{1780}{2} \left(0.1854 \right) \right] 0.1854$$

$$= 95.9 \text{ in.-lb.}$$

The maximum counterrecoil velocity

$$v_{cr} = \sqrt{\frac{2E_{cr}}{M_r}} = \sqrt{871.9} = 29.53 \text{ in./sec.}$$

Compute the counterrecoil time of the slide in three steps, before the slide contacts the gas operating unit, the effect of cartridge case ejection, and after the slide picks up the operating unit. The distance traveled before contact is

$$s_c = s - s_{or} = 2.974 - 1.67 = 1.304 \text{ in.}$$

The time to traverse this distance, Eq. 2–26,

$$t'_{scr} = \sqrt{\frac{M_{sr}}{\epsilon K}} \left(\text{Sin}^{-1}\ \frac{Ks_c - F}{f(F)} - \text{Sin}^{-1}\ \frac{-F}{f(F)} \right)$$

where

$$\sqrt{\frac{M_{sr}}{\epsilon K}} = \sqrt{\frac{11.2}{0.8 \times 40 \times 386.4}} = \sqrt{0.000906} = 0.0301$$

$$f(F) = \sqrt{F^2 + \frac{K}{\epsilon} M_{sr} v_{scr}^2} = \sqrt{204^2 + \frac{40}{0.8} \times 0.029 \times 231.6^2} = 345.5.$$

Therefore

$$t'_{scr} = 0.0301 \left(\mathrm{Sin}^{-1} \frac{40 \quad 1.304 - 204}{345.5} - \mathrm{Sin}^{-1} \frac{-204}{345.5}\right) = 0.0301 \left[\mathrm{Sin}^{-1}(-0.43946) - \mathrm{Sin}^{-1}(-0.59045)\right]$$

$$= 0.0301 \left(\frac{333.93 - 323.81}{57.3}\right) = 0.0053 \text{ sec.}$$

The slide energy at this position

$$E_s = \frac{1}{2}\left(M_{sr} v_{scr}^2\right) + \epsilon\left(F - \frac{1}{2} K s_c\right) s_c = \left(\frac{0.029}{2}\right) 231.6^2 + 0.8 \left[204 - \quad (1.304)\right] 1.304$$

$$= 963.4 \text{ in.-lb}$$

The corresponding velocity

$$v'_{scr} = \sqrt{\frac{2E_s}{M_{sr}}} = \sqrt{\frac{1926.8}{0.029}} = \sqrt{66441} = 257.8 \text{ in./sec.}$$

To avoid duplication of the above exercise, assume that cartridge case ejection and gas operating piston pick-up occur at the same time. When pick-up occurs, the slide gains one pound. From the conservation of momentum when the ejection velocity, $v_e = 840$ in./sec,

$$\left(\frac{W_{sr}}{g}\right) v'_{scr} = \left(\frac{W_{srp}}{g}\right) v_{scr} + \frac{W_{cc}}{g} v_e$$

$$\left(\frac{11.2}{386.4}\right) 257.8 = \left(\frac{12.2}{386.4}\right) v_{scr} + \left(\frac{0.2}{386.4}\right) 840$$

$$v_{scr} = \frac{2887 - 168}{12.2} = 222.9 \text{ in./sec}$$

where

W_{sr} = weight of slide with 2 rounds

W_{srp} = weight of slide, 2 rounds, and gas operating unit

The time to complete slide counterrecoil is

$$t''_{scr} = \sqrt{\frac{M_{srp}}{\epsilon K}} \left(\mathrm{Sin}^{-1} \frac{Ks_r - F}{f(F)} - \mathrm{Sin}^{-1} \frac{-F}{f(F)} \right)$$

where

$$\sqrt{\frac{M_{srp}}{\epsilon K}} = \sqrt{\frac{12.2}{0.8 \times 40 \times 386.4}} = \sqrt{0.00987} = 0.0314$$

$$F = 204 - Ks_c = 204 - 52.16 = 151.84 \text{ lb.}$$

$$f(F) = \sqrt{F^2 + \frac{K}{\epsilon} \left(M_{srp} v^2_{scr} \right)} = \sqrt{151.84^2 + \frac{40}{0.8} (0.0316) \ 222.9^2} = 318.7.$$

Thus

$$t''_{scr} = 0.0314 \left(\mathrm{Sin}^{-1} \frac{40 \times 1.67 - 151.84}{318.7} - \mathrm{Sin}^{-1} \frac{-151.84}{318.7} \right) = 0.0314 \left[\mathrm{Sin}^{-1} (-0.26683) - \mathrm{Sin}^{-1} (-0.47643) \right]$$

$$= 0.0314 \left(\frac{344.52 - 331.55}{57.3} \right) = 0.0071 \text{ sec.}$$

The total slide counterrecoil time after cam action is

$$t_{scr} = t'_{scr} + t''_{scr} = 0.0053 + 0.0071 = 0.0124 \text{ sec.}$$

The slide energy and velocity at the end of counterrecoil are

$$E_s = \frac{1}{2} \left(M_{srp} v^2_{scr} \right) + \epsilon \left(F - \frac{1}{2} Ks_{or} \right) s_{or}$$

$$= \left(\qquad \right) 222.9^2 + 0.8 \left[151.84 - \frac{40}{2} (1.67) \right] 1.67 = 943.8 \text{ in.-lb.}$$

$$v_{scr} = \sqrt{\frac{2E}{M_{srp}}} = \sqrt{\frac{1887.6}{0.0316}} = \sqrt{59734} = 244.4 \text{ in./sec.}$$

The time of slide counterrecoil exceeds that for the barrel, therefore, the firing rate is based on the former. The total time to complete the firing cycle

$$t_c = t_e + t_{scr} = 0.0329 + 0.0124 = 0.0453 \text{ sec.}$$

The rate of fire

$$f_r = \frac{60}{t_c} = 1324 \text{ rounds/min.}$$

Originally, the maximum recoil velocity of the slide was assumed to be 600 in./sec but, to satisfy some of the other design criteria, this velocity reduces to 478.8 in./sec. Rather than manipulate other variables to reach the 600 in./sec velocity, 478.8 in./sec was accepted to continue the analysis. When this reduced velocity was introduced in the earlier time estimates of par. 5–1.1.1, a revised rate of fire of 1590 rounds/min was computed. These two firing rates – one obtained by means of a digital computer, the other by a short cut estimate – are within 80% agreement of each other.

5–2 DOUBLE BARREL TYPE

The Navy MK 11 Gun is an excellent example of a double barrel revolver-type machine gun. This gun is recoil-operated and fires 4000 rounds/min at a muzzle velocity of 3300 ft/sec (Ref. 16). The high rate of fire is attributed to (1) the simultaneous loading, firing, and ejection of two belts of ammunition for each operation, (2) the use of advance primer ignition technique, and (3) the absence of conventional recoil and counterrecoil shock absorbing elements.

5–2.1 FIRING CYCLE

The firing cycle involves three basic operations: ramming, firing, and case ejection. All perform simultaneously at six chambers of the eight-chambered drum. Fig. 5–11 locates the relative positions of these operations. One belt of ammunition enters the rear drum area from each side of the gun and assumes the respective positions. The rounds in the top and bottom chambers, properly aligned with the barrels, are fired simultaneously. Propellant gases, bled from one barrel only, actuate the rammers and eject the empty cases, all other mechanical functions depend on recoil activity. Since firing must precede ramming and ejecting, an imperceptible lag occurs between firing and the two other functions.

At the start of each burst, only one shot is fired, always from the same *first-fire* barrel. Thus, the momentum of the recoiling parts is equivalent to the impulse generated by the propellant of this single shot. During the latter part of the recoil travel, the energy of translation of the recoiling parts is converted to rotational energy of the drum by cam action. The drum now acts as a flywheel, delivering energy needed to operate the feed system, meanwhile storing the remainder that eventually would be reconverted, by continued action, into the translational energy of counterrecoil. Just before reaching the in-battery position, both barrels are fired. Part of the total impulse of the two shots compensate for the momentum of counterrecoil to stop the moving parts in their forward motion. The remaining impulse induces the recoil that follows. This action continues until the end of the burst when a single shot is fired in the *last-fire* barrel, as opposed to the *first-fire* barrel that starts the burst. The impulse of this shot stops the counterrecoiling parts. Any residual impulse is absorbed by a buffer which also absorbs the full counterrecoil shock in the event of a misfire.

5–2.1.1 Cam Function

The drum has eight elliptical cams cut into its outer surface in the arrangement shown in Fig. 5–12. Forward and rear cam followers, mounted on a pivoting arm, engage alternately forward and rear cams during successive rounds. Fig. 5–12(A) shows the gun in battery with the forward cam follower engaged in a forward cam. As the gun recoils, the follower moves along the straight portion of the cam. The relative motion between cam and follower is augmented by the rocker arm which pivots about its fixed center. As its lower end swings forward during recoil, it draws the cam followers forward thus increasing the relative motion between cam and follower. Fig. 5–12(B) shows the positions at full recoil. By this time the follower has traversed half the curved distance and rotated the drum 22-1/2 deg. All energy is now rotational energy with only the drum and associated parts in motion. As the drum continues to turn, it actuates the follower which induces counterrecoil thereby reversing all translational motion that occurred during recoil. Fig. 5–12(C) shows the positions of the various components after all rotation has reverted to translation. Fig. 5–12(D) shows the respective positions after the return to battery. The front follower has been lowered to disengage it from the cam while the rear follower has been raised to engage the next cam which reaches this position after the drum has completed the 45 deg of travel during the firing cycle.

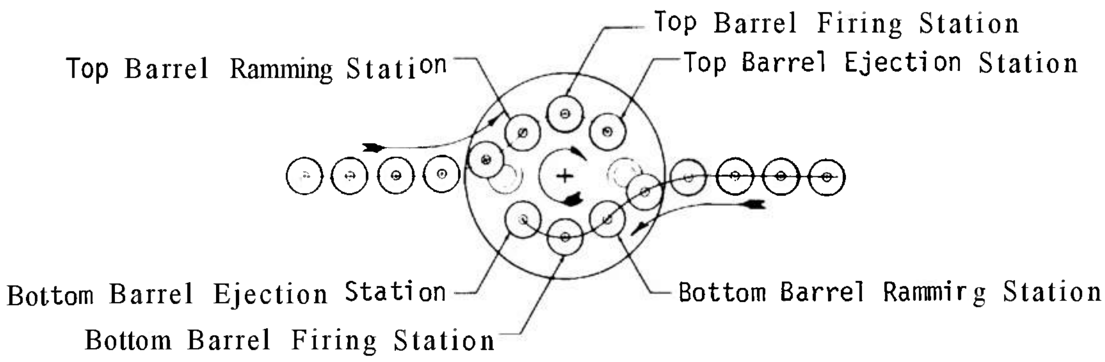


Figure 5–11. Location of Basic Operations

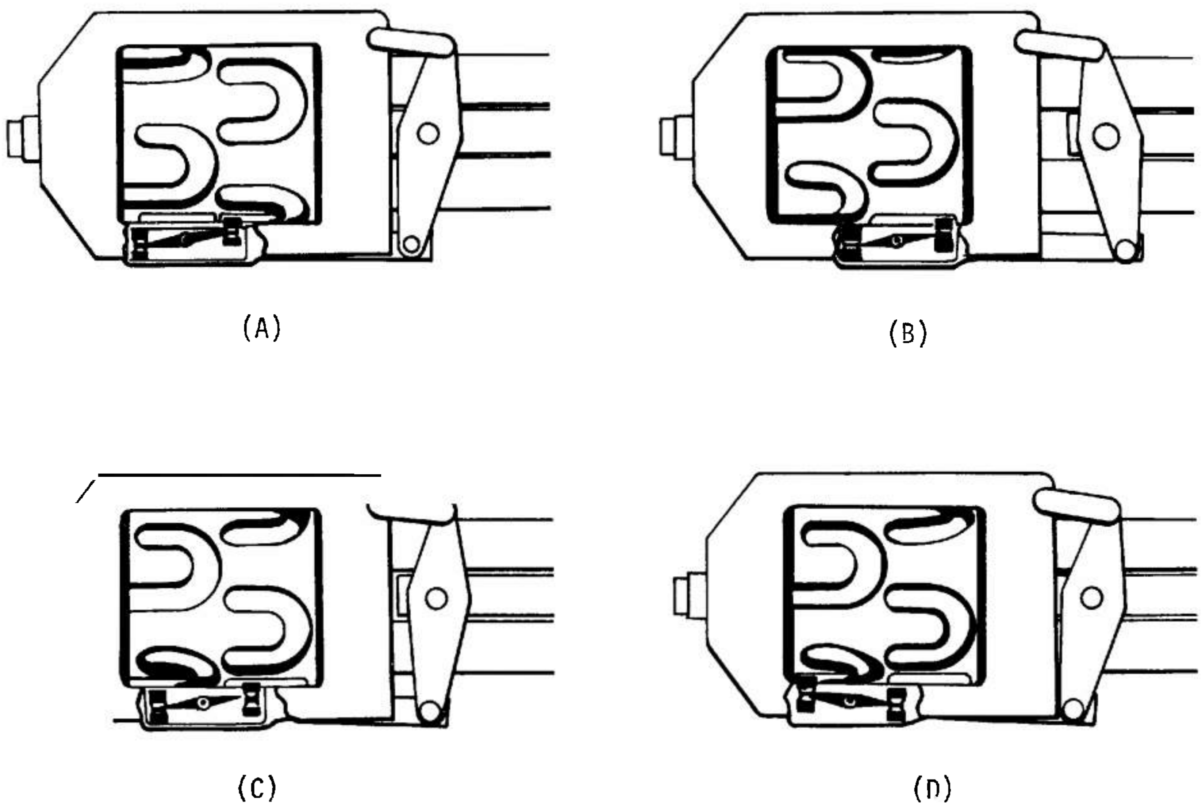


Figure 5–12. Schematic of Double Barrel Drum-cam Arrangements

5-2.1.2 Loading and Ejecting

Ammunition is conveyed to the gun in cylindrical links which are connected to form a belt. Each link has a pin and hook diametrically opposite to each other. The hook of one link engages the pin of the following link to form a joint of limited flexibility that permits the belt to be twisted or folded. Two belts feed the gun, one from each side. The intermittent rotation of the feed sprocket, which is splined to the drum shaft, pulls the ammunition belts into the loader. Two gas-operated rammers, diametrically opposite with reference to the drum, simultaneously strip a round from a link of each belt and ram the ammunition at a speed of 50 ft/sec into the empty chambers adjacent to the barrels. On the next cycle these two rounds are fired. The following cycle, after the drum rotates **45** deg, finds the chambers containing the spent cases and their corresponding links in line with the gas ejector ports and ejection ducts. Propellant gases, tapped from the first-fire barrel, issue from the ports at high velocity and blow the empty cases into the empty links. The case momentum is high enough to seat the cases in the links, slip the belt attachment, and carry the case-link unit into the mouth of the ejection ducts at a velocity of 75 ft/sec which is enough to insure emergence from the ducts at 50 ft/sec. During the next cycle, the emptied chambers remain empty and advance to the 3 and 9 o'clock positions, where the two ammunition belts enter the loader (see Fig. 5-11). By remaining empty, the two idle chambers provide the space requirements for efficient operation.

5-2.1.3 Ammunition Feed System

The ammunition feed system consists of a pneumatic motor, a drive system, and an ammunition magazine. The system functions as a unit – the motor provides the power, the drive transmits the power to the ammunition belts and magazine, while the magazine releases the ammunition and rotates to maintain proper alignment between stored belt and feed throat. Fig. 5-13 is a schematic of the feed system that illustrates the functions in sequence. Although the gun is self-feeding, the pneumatic power boosters at the magazine insure high rate of fire and high functional link belt reliability. The power is transmitted by drive shafts and gear boxes to ***the*** sprockets which (1) engage the ammunition belts, (2) pull the belts from the magazine, and (3) drive the ammunition toward the loader. Another power drive, reduced to low speeds, turns the magazine.

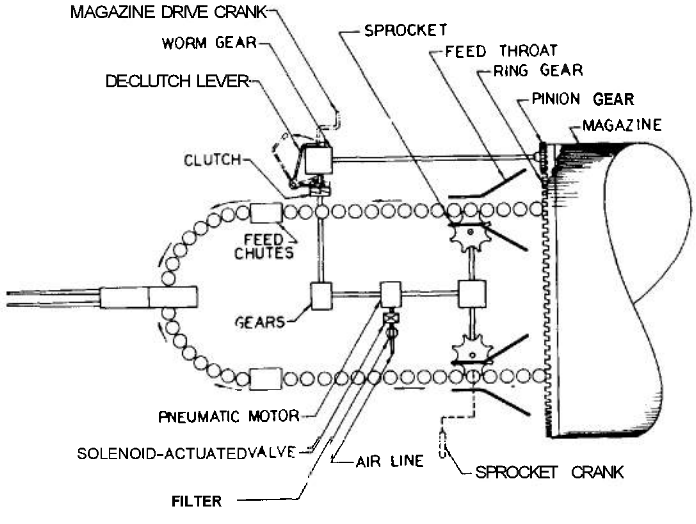


Figure 5-13. Schematic of Ammunition Feed System

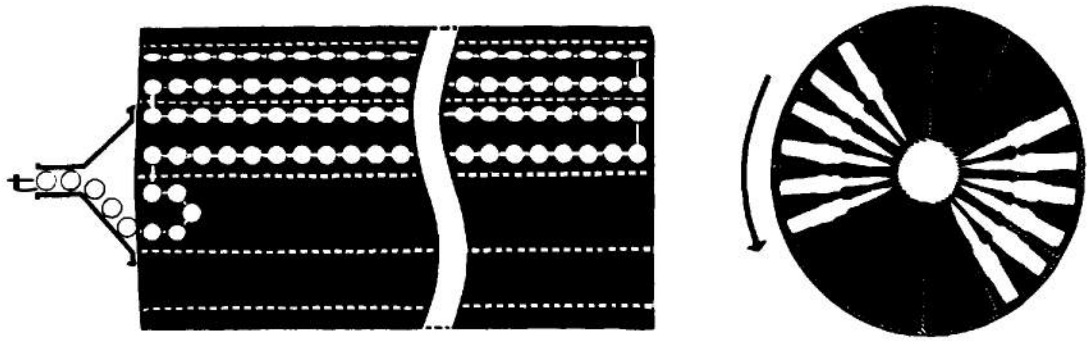


Figure 5–14. Schematic of Ammunition Magazine

The feed system is equipped with two manually operated driving units that are turned by hand cranks. When the magazine is being loaded, a jaw clutch is disengaged, separating magazine drive from ammunition drive so that magazine and feed sprockets can be rotated independently. One unit turns the magazine worm drive which rotates the magazine to the desired loading position; the other manual drive turns the feed sprocket to move the ammunition belt into the magazine.

The magazine is a cylindrical drum that has an even number of radial partitions. Each partition houses 60 rounds of ammunition. The two ammunition belts are loaded symmetrically about the axis so that the belts can be withdrawn simultaneously from two diametrically opposite sectors. Fig. **5–14** shows the arrangement of the stored munition and the method of withdrawal. The end view shows the balanced nature of the stored ammunition while the arrow indicates the direction of rotation. The side view shows how a belt is folded in the partitions. The belts leave the magazine through the feed throat on the left. Ammunition in the sectors is so arranged that only a small segment of each belt is accelerated at any time during a burst. As the ammunition empties from one sector, the next loaded sector – by virtue of the slowly rotating magazine – comes into line with the feed throat. Alignment is assured by synchronizing belt speed with the rotational speed of the magazine.

5–2.2 DYNAMICS OF FIRING CYCLE

To introduce the dynamics of the firing cycle, assume that the machine gun is operating normally, i. e., both barrels are firing simultaneously. Since the gun fires out of battery, the momentum of the counterrecoiling mass must be dissipated before recoil can begin. This momentum is counteracted by the initial impulse of the propellant gas force. The remaining impulse is converted to linear momentum of the recoiling parts and later, by cam action, the linear momentum is converted into angular momentum of the drum and its associated moving parts.

The cam arrangement is such that only linear motion of the recoiling parts occurs shortly before, during, and immediately following firing. The dynamics are readily computed if the firing time is divided into small increments of time. If the acceleration during this short time interval is assumed to be constant, the various parameters can be computed for each time increment. Thus, after the increment of time *At*, the momentum of the counterrecoiling parts at increment *n* is

$$M_n = M_{n-1} + F_g \Delta t \qquad (5\text{–}76)$$

where F_g = propellant gas force during *At*

M_{n-1} = momentum just preceding *At*

During counterrecoil, momentum and velocity are considered to be negative as opposed to positive during recoil. Propellant gas force is always positive. Momentum is defined as **Mu,** therefore the velocity after **At** is

$$v_n = \frac{M_n}{M_r} \qquad (5\text{–}77)$$

where M_r = mass of recoiling parts.

The distance traveled during *At* is

$$\Delta x_r = (v_n + v_{n-1})\,\Delta t/2. \qquad (5\text{–}78)$$

Designate x_r the distance that the recoiling parts are out of battery.

$$x_r = x_{r(n-1)} + \Delta x. \tag{5-79}$$

During recoil, x_r is the recoil travel distance of barrel and drum assembly.

5-2.2.1 Cam Analysis

The forces induced by cam action are shown diagrammatically in Fig. 5-15. Because the cam follower is constrained in the y direction, motion in this direction is restricted to the peripheral travel of the drum. Because of the linkage arrangement shown in Fig. 5-12, the cam-to-cam follower position is determined by the relative displacement of the cam during recoil, or counterrecoil, and the movement of the cam follower that is determined by the link rotation.

The resolution of cam forces and the mutual influence of the various moving parts on one another are illustrated in Figs. 5-15, 5-16, and 5-17; and are defined, except for those referring to the springs and slide, by the expressions of Eqs. 5-29 through 5-47. Because this cam analysis follows the same procedure as that developed for the slide-cam analysis, only the pertinent equations will be given, thus avoiding repetition. According to Eqs. 5-33 through 5-36, the directional coefficients during recoil are

$$K_x = \sin\beta + \mu\left(\frac{R_p}{R_r}\right)\cos\beta \tag{5-80}$$

$$K_y = \cos\beta - \mu\left(\frac{R_p}{R_r}\right)\sin\beta \tag{5-81}$$

and during counterrecoil, they are

$$K_x = \sin\beta - \mu\left(\frac{R_p}{R_r}\right)\cos\beta \tag{5-82}$$

$$K_y = \cos\beta + \mu\left(\frac{R_p}{R_r}\right)\sin\beta \tag{5-83}$$

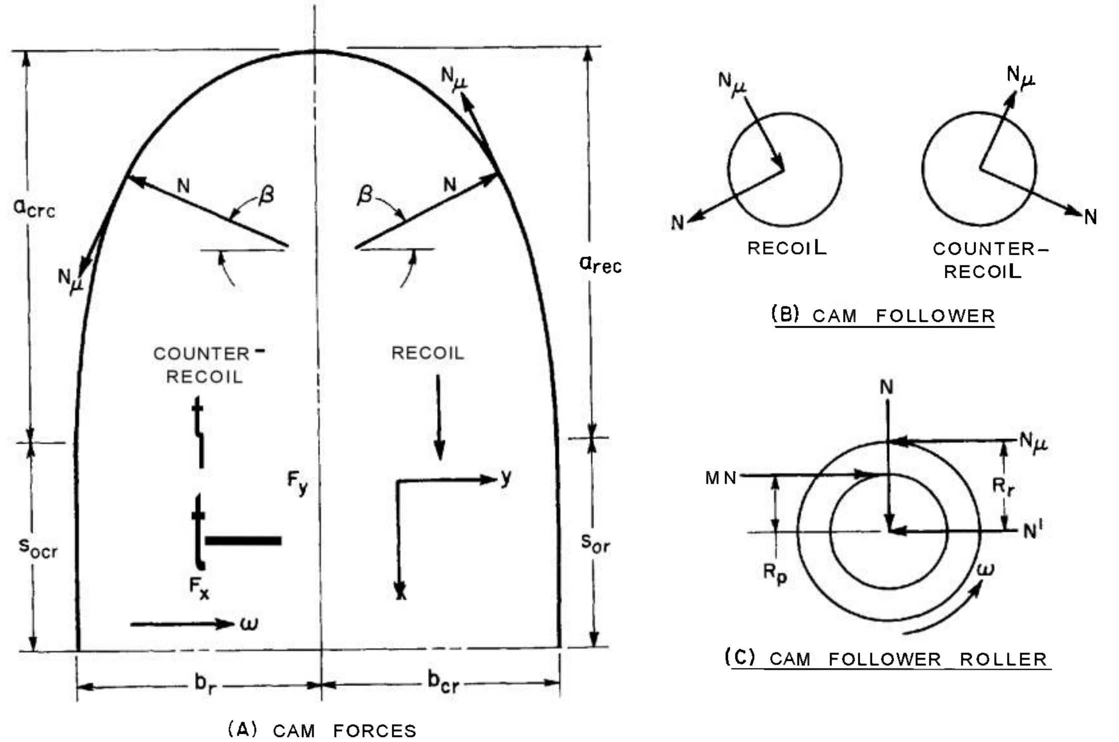


Figure 5-15. Double Barrel Cam Force Diagrams

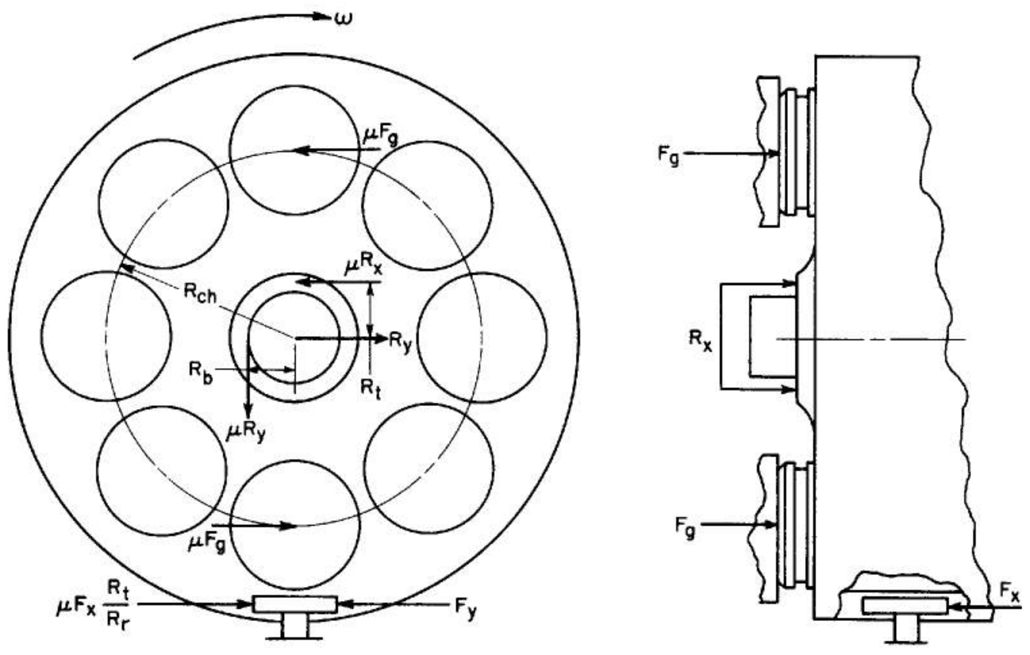


*Figure **5–16.** Double Barrel Drum Loading Diagram*

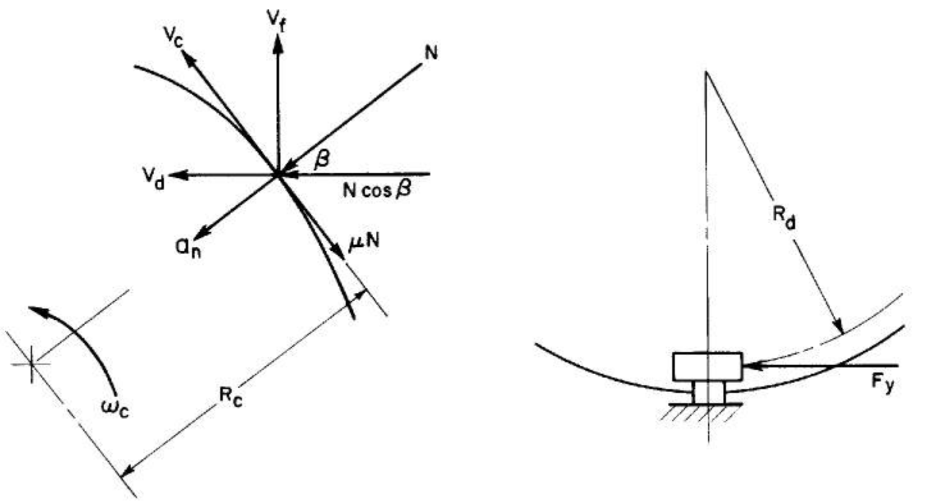


*Figure 5– **17.** Double Barrel Drum Dynamics*

The axial and tangential cam forces are, respectively,

$$F_x = NK_x \qquad (5\text{–}84)$$

$$F_y = NK_y \qquad (5\text{–}85)$$

From Eq. 5–41, the resisting torque induced by the residual propellant gas force is

$$T_g = \mu(R_{ch} - \mu R_b)F_g \qquad (5\text{–}86)$$

The cam normal force, Eq. 5–47, is

$$N = \frac{\dfrac{I_d v_s^2}{R_c R_d \cos\beta} + T_g}{(R_d - \mu R_b) K_y - \mu \left[R_t + R_d\left(\dfrac{R_p}{R_r}\right)\right] K_x} \qquad (5\text{–}87)$$

where v_s is the axial relative velocity between cam follower and cam which is also the slide velocity. The various parameters of the cam geometry are expressed in Eqs. 5–48 through 5–52.

5–2.2.2 Energy Concept

During the early part of the recoil stroke and the latter part of the counterrecoil stroke, the cam follower moves axially in the straight portion of the cam; therefore, no energy is exchanged between rotating and translating parts. When the cam follower is riding in the curved portion of the cam, an exchange of energy takes place. At any given time, by the law of conservation of energy, the total energy remains unchanged. By dividing the cam travel into short increments, the amount of energy E_i in each group of moving parts may be expressed with negligible error in terms of total energy and the changing geometry of cam.

$$E_i = E_r + E_d + E_a + E_\mu \qquad (5\text{–}88)$$

where E_a = energy of ammuntion belt

E_d = energy of rotating parts

E_i = total energy at any given increment

E_r = energy of recoiling parts

E_μ = energy losses due to friction

Note that for the next increment,

$$E_i = E_{i-1} - E_\mu . \qquad (5\text{–}89)$$

The individual energy terms are expressed in terms of the linear velocity of the recoiling parts as shown in the next three equations.

$$E_d = I_d \left(\frac{\tan^2\beta}{R_d^2}\right) v_r^2 \qquad (5\text{–}90)$$

$$E_r = \frac{1}{2}\left(M_r v_r^2\right) \qquad (5\text{–}91)$$

$$E_a = \frac{1}{2}\left(N_a M_a r_a v_r^2\right) \qquad (5\text{–}92)$$

where I_d = mass moment of inertia of drum

M_a = mass of one link of ammunition

M_r = mass of recoiling parts

N_a = number of links of ammunition affected

R_d = distance of cam to center of drum

r_a = effective ratio between v_r and belt velocity

v_r = velocity of recoiling parts

5–2.2.3 Digital Computer Program for Firing Cycle

A digital computer program is arranged to compute the significant data occurring during the firing cycle. It is an iterative procedure that first computes the total impulse. The force-time curve of the interior ballistics is divided into small time increments from which the respective propellant gas force is read. The median force is assumed to be the average. The differential time

between any two adjacent forces is small enough so that the corresponding portion of the curve approximates a straight line; therefore, the assumption is considered accurate. The computed area under the curve is the total impulse. The force-time curve is shown in Fig. 5–18.

The velocity of free recoil is found by equating the impulse to the momentum of the recoiling parts. The energy of free recoil may now be computed. However, this energy is a fictitious value since the gun is fired out of battery. Also, frictional losses in the system account for additional energy loss. To compensate for this loss during the first set of computations, only 70 percent of the energy of free recoil is entered toward computing the counterrecoil velocity plus the initial momentum of counterrecoil just as the first two chambered rounds are fired simultaneously. Each differential impulse is subtracted from the momentum until zero velocity is achieved. Subsequently, the remaining impulse determines the new recoil velocity. The dynamics of the cam system are now computed and if the resulting counterrecoil velocity – after the cam is negotiated by the cam follower – does not match its original value, the initial counterrecoil is adjusted accordingly and the process continued.

The area under the force-time curve for a given time represents the cumulated impulse of the propellant gas force to that term. The area is computed by employing the trapezoid rule.

$$A_n = A_{n-1} + F\Delta t \tag{5–93}$$

$$F\Delta t = \frac{1}{2}\left[F_{g(n-1)} + F_{g(n)}\right]\Delta t. \tag{5–94}$$

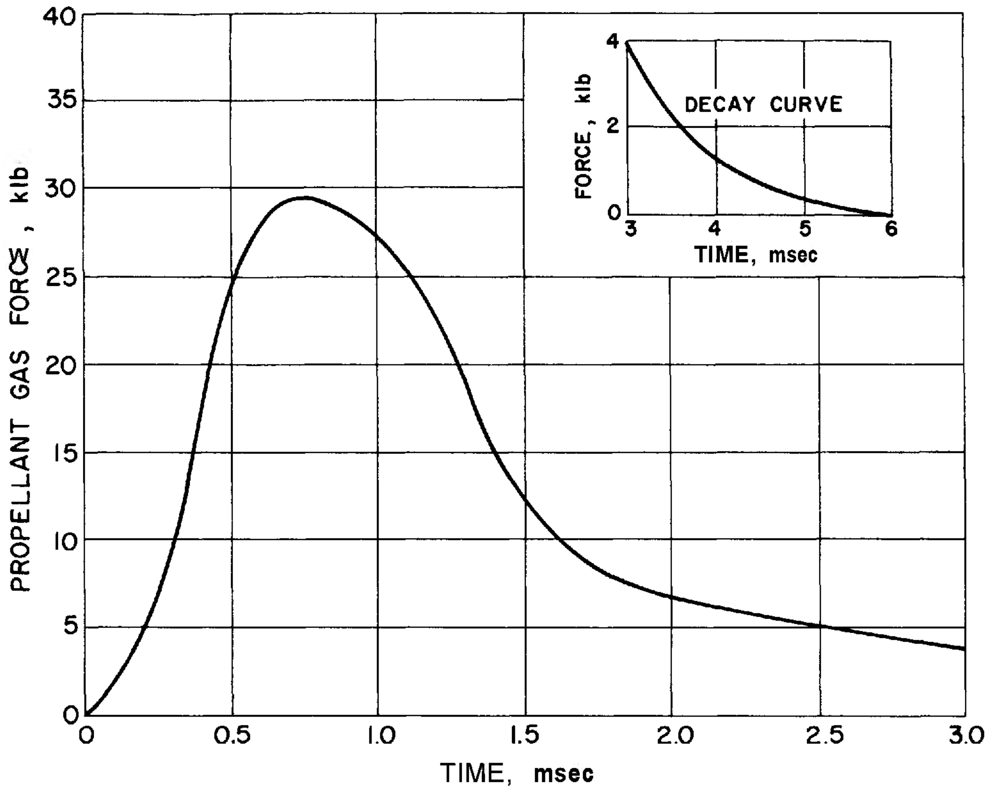


Figure 5–18. Force-time Curve of 20 mm Revolver-type Gun

The momentum during counterrecoil at any given time is

$$M_n = M_{n-1} - F\Delta t. \tag{5-95}$$

The momentum during recoil is

$$M_n = M_{n-1} + F\Delta t. \tag{5-96}$$

The velocity, energy, and distance can now be computed at the end of each time interval. Note that no cam activity must take place while the projectiles are still in the bores. To insure this requirement, the cam follower rides the linear portion of the cam (the dwell period) until the propellant gas pressure has theoretically dropped to zero. Actually some residual gas pressure may still persist and is considered as one of the forces in the cam analysis.

The conservation of energy concept introduced earlier is generally followed with some variations to make the analysis more manageable. These variations are the various components of the energy lost to friction. Two primary components are the linear and angular sliding frictional losses. The linear component consists of the frictional resistance induced by the transverse cam force F_y and its reaction R_y on the drum bearing (see Fig. 5–16). The frictional resistance created by μR_y is relatively small and may be ignored. The work done by the average forces over small increments of travel is

$$E \qquad \left| \left[F_{y(n-1)} + F_{y(n)} \right] \left(\frac{R_p}{R_r} \right) \Delta x + \left[R \qquad {}_{1)} + R_{y(n)} \right] \Delta x_r \right| \tag{5-97}$$

Note that **Ax** is influenced by the friction of the cam follower roller to the extent of the indicated ratio of two radii (see **Eq.** 5–32). According to Fig. 5–19, p = r_c/r_d. Since $R_y = F_y$ and **Ax,** = $\Delta x/\rho$, and according to **Eq.** 5–85, $F_y = NK_y$, therefore

$$E_{\mu s(n)} = \frac{1}{2}\mu \left[\left(NK_y\right)_{(n-1)} + \left(NK_y\right)_{(n)} \right] \left(\frac{1}{\rho} + \frac{R_p}{R_r} \right) \Delta x. \tag{5-98}$$

The respective velocities, v_s and $\tau_,$ of the slide and recoiling parts during cam activity, are

$$v_s = v_c \,\mathrm{cosp}; \quad v_r = v_s/\rho\ . \tag{5-99}$$

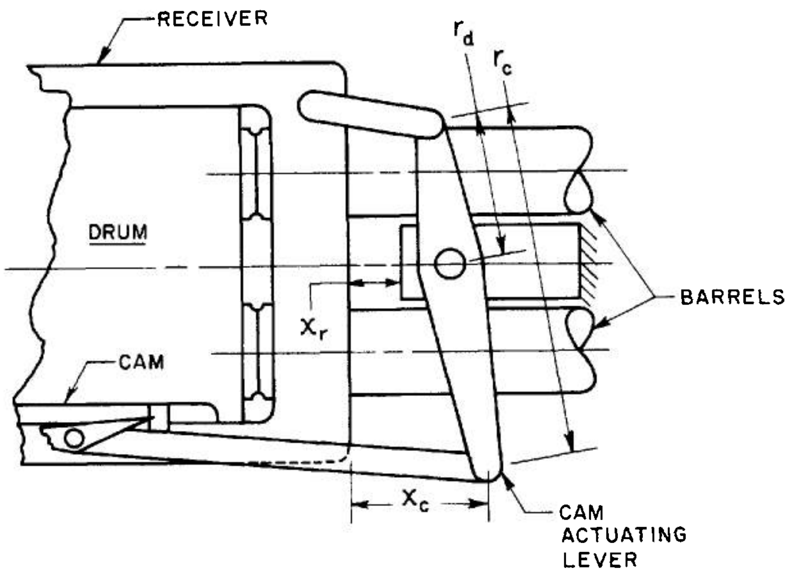


Figure 5–19. Geometry of Cam Actuating Lever

At the same time, the distances traveled are computed from the position of cam follower and cam as indicated in **Eq. 5–49.** Since the distance x on the curve indicates the relative travel along the **x-axis** between cam follower and cam, the total axial distance at any given interval is

$$x_s = s_{or} + x \tag{5–100}$$

where s_{or} is the straight length of the cam. The corresponding travel x, of the recoiling parts (drum and barrel assemblies) is

$$x_r = x_s/\rho. \tag{5–101}$$

During the cam dwell period, the travel of the recoiling parts is

$$x_{rd} = x_{rd(n-1)} + \frac{1}{2}\left[v_{r(n-1)} + v_{r(n)}\right]\Delta t \tag{5–102}$$

where x_{rd} is used instead of x_r to differentiate between the dwell period and the active cam period. The interval Δt_1 spans the time when the follower enters the dwell and until the propellant **gas** forces of the next round become effective. Here, Δt_1 is the first time interval of the firing cycle.

$$\Delta t_1 = \left[\left(s_{or}/\rho\right) - x_{ro}\right]/v_{cr} \tag{5–103}$$

where v_{cr} = counterrecoil velocity

x_{ro} = counterrecoil travel during impulse period

The time interval after the impulse period and until cam action begins during recoil **is**

$$\Delta t_n = \left[\left(s_{or}/\rho\right) - x_{rn}\right]/v_m \tag{5–104}$$

where v_m = max recoil velocity

x_{rn} = recoil travel during impulse period

TABLE 5–15. SYMBOL-CODE CORRELATION FOR DOUBLE BARREL MACH NE GUN

Symbol	Code	Symbol	Code
a	A	s_o	SC
A_n	FTAREA	s_{or}/ρ	SR
b	B	t	T
C_x	CX	At	DT
	CY	t_m	TM
E	E	T_g	TG
F_g	FG	v	V
F_{gr}	FGRES	v_c	VC
$F\Delta t$	FDT	v_d	VD
g	G	v_s	VS
I_d	DI	W_a	WA
K_x	XK	W_r	WR
K_y	YK	x	X
L_c	CL	Ax	DX
L_r	RL	x_s	XC
M_a	EMA	x_{rd}	XR
M_r	EMR	Δx_r	DXR
M_n	EMV	x_r	XREC
N	EN	y	Y
R_b	RB	Δy	DY
R_{ch}	RCH	β	BETA
R_d	RD	θ	THETA
R_p	RP	$\Delta\theta$	DTHETA
R_r	RR	μ	EMU
R_t	RT	ρ	RHO

TABLE 5–16. INPUT DATA FOR DOUBLE BARREL MACHINE GUN

Code	Value	Code	Value
A	1.5	RD	3.25
B	1.276275	RHO	2.5
CL	3.125	RL	1.25
DI	0.81	RP	0.25
EMU	0.1	RR	0.5
FGRES*	200.0	RT	1.25
G	386.4	SR	0.65
RB	1.0	WA	12.256
RCH	2.25	WR	120.0

*The values of FG are listed in the output, Table 2-11.

Throughout the computer program, several combinations of values are repeated. Also, similar terms appearing in more than one equation are lifted from those equations and combined into another unique expression. To avoid recomputation and for programming convenience, each combination is treated as a coefficient which is identified by a new symbol. These coefficients, when isolated, have little physical significance, and therefore, can be defined best by association throughout the development of the computer program.

For any given increment n, both $N_{(n-1)}$ and $K_{y\,(n-1)}$ are known so that

$$E_{\mu s(n-1)} = \frac{1}{2}\left[\mu(NK_y)_{(n-1)}\right]\left(\frac{1}{\rho} + \frac{R_p}{R_r}\right)\Delta x$$

$$= C_{m(n-1)}N(n-1)\Delta x/2. \qquad (5\text{–}105)$$

Note that in general, $C_m = \mu K_y\left(\frac{1}{\rho} + \frac{R_p}{R_r}\right)$

The other part involving NK_y has unkown values and these are expressed in terms of computable parameters. Thus

$$E_{\mu s(n)} = \frac{1}{2}\,\mu\,(NK_y)_n\left[\frac{1}{\rho} + \frac{R_p}{R_r}\right]\Delta x$$

$$= C_{m(n)}N_{(n)}\Delta x/2 \qquad (5\text{–}106)$$

By combining the various expressions of **Eq.** 5–47 into simple terms

$$N = \frac{\left[C_{id}\left(\frac{\cos\beta}{R_c}\right)v_c^2 + T_g\right]}{\left(C_yK_y - C_xK_x\right)}. \qquad (5\text{–}107)$$

$$N = C_{vc}v_c^2 + T_{gn} \qquad (5\text{–}108)$$

Now substitute for N in **Eq.** 5–106

$$E_{\mu s(n)} = E_{\mu sm} + C_{dx}T_{gn} \qquad (5\text{–}109)$$

where

$$E_{\mu sm} = C_{fx}v_c^2 \qquad (5\text{–}110)$$

The resolution of the angular component of sliding friction follows a similar procedure. The total energy lost to friction in the rotating drum is

$$E_{\mu d} = \frac{1}{2}\left[T_{\mu(n-1)} + T_{\mu(n)}\right]\Delta\theta. \qquad (5\text{–}111)$$

where T_μ is the frictional torque. The equation for the frictional torque may be composed by referring to Fig. 5–16.

$$T_\mu = \mu F_yR_b + \mu F_x\left[R_t + R_d\left(\frac{R_p}{R_r}\right)\right] + 2\mu F_{gr}R_{ch} \qquad (5\text{–}112)$$

$$T_\mu = N\ \ \mu R_bK_y + C_xK_x\Big) + T_g = C_{t\mu}N + T_g. \qquad (5\text{–}113)$$

At the beginning of each increment, all data are known.

$$T_{\mu(n-1)} = C_{t\mu(n-1)}N_{(n-1)} + T_g \qquad (5\text{–}114)$$

At the end of each increment, N is not known but may be expressed in terms of computable parameters

$$T_{\mu(n)} = C_{t\mu}N\ \_\ T_g \qquad (5\text{–}115)$$

$$_{t\mu}C_{vc}v_c^2 + C_{t\mu}T_{gn} + T_g. \qquad (5\text{–}116)$$

The various terms for energy may now be expressed by multiplying the frictional torques by $\Delta\theta/2$ and combining the terms when appropriate

Compute the energy of the $N_{(n-1)}$ term of Eq. 5–114.

$$E_{\mu d1} = \frac{1}{2}\left[\Delta\theta\, C_{t\mu(n-1)}N(n-1)\right] \qquad (5\text{–}117)$$

Compute the energy of the v_c term of **Eq.** 5–116

$$E_{\mu dn} = \frac{\Delta\theta}{2}\left(C_{t\mu}C_{vc}v_c^2\right) = C_{tq}v_c^2 \qquad (5\text{–}118)$$

Compute the energy of the T_{gn} terms of **Eqs.** 5–109 and 5–116

$$E_{\mu rtg} = \left[C_{dx} + C_{t\mu}\left(\frac{\Delta\theta}{2}\right)\right]T_{gn} = \left(C_{dx} + C_{dt}\right)T_{gn} \qquad (5\text{–}119)$$

Compute the energy of the T_g terms of **Eqs.** 5–114 and 5–116.

$$E_{\mu dtg} = \frac{\Delta\theta}{2}\left(T_g + T_g\right) = \Delta\theta\, T_g. \qquad (5\text{–}120)$$

The total readily computable energy loss is

$$E_{\mu 1} = E_{\mu s1} + E_{\mu rtg} + E_{\mu dtg} + E_{\mu di}. \qquad (5\text{–}121)$$

The effective masses of the recoiling parts (M_{re}), ammunition (M_{ae}), and drum (I_{de}) with respect to the cam velocity when energy is involved are

$$M_{re} = \frac{M_r}{2\rho^2};\quad M_{ae} = \frac{M_a}{2\rho^2},\quad I_{de} = \frac{I_d}{R_d^2 + \rho^2}. \qquad (5\text{–}122)$$

The energy that remains in the system after the readily computable energy loss is subtracted

$$E_{ie} = E_{i-1} - E_{\mu 1} = E_r + E_a + E_d + E_{\mu sn} + E_{\mu dn}$$

$$= \left| M_{re}\sin^2\beta + \left(M_{ae} + I_{de}\right)\cos^2\beta + C_{fx} + C_{tg}\right| v_c^2$$

$$E_{ie} = \left(C_{mr} + C_{ma} + C_{id} + C_{fx} + C_{tg}\right)v_c^2 = C_e v_c^2 \qquad (5\text{–}123)$$

where C_e is the coefficient of v_c^2

$$v_c = \sqrt{E_{ie}/C_e} \qquad (5\text{–}124)$$

$$E_\mu = E_{\mu 1} + C_{fx} + \left(C_{fx} + C_{tg}\right)v_c^2 \qquad (5\text{–}125)$$

$$E_i = E_{i-1} - E_\mu \qquad (5\text{–}126)$$

The computed results show the time for one cycle to be 26.2 msec (see Table 5–17) which indicates a firing rate of 4580 rounds/min since both barrels are firing simultaneously.

TABLE 5–17. DOUBLE BARREL MACHINE GUN DYNAMICS

I	TIME MSEC	PROPELLANT GAS FORCE LB	IMPULSE LB-SEC	RECOIL VEL IN/SEC	AXIAL CAY VEL IN/SEC	RECOIL TRAVEL IN	AXIAL CAM TRAVEL IN
1	6.071	.0	.00	97.6	243.9	.0577	.1442
2	6.196	2500.0	.16	96.6	241.4	.0456	.1139
3	6.321	7000.0	.75	92.7	231.8	.0337	.0843
	6.446	15200.0	2.14	83.8	209.5	.0227	.0567
5	6.571	23600.0	4.56	68.2	170.4	.0132	.0330
6	6.696	28400.0	7.81	47.2	118.1	.0060	.0150
7	6.821	29500.0	11.43	23.9	59.9	.0015	.0038
8	6.946	28600.0	15.06	.6	1.4	.0000	.0000
9	6.950	28573.6	18.55	.0	.0	-.0000	-.0000
10	7.196	24800.0	21.80	42.8	107.1	.0053	.0132
11	7.321	22000.0	24.72	61.7	154.2	.0118	.0295
12	7.446	16200.0	27.11	77.0	192.6	.0205	.0512
15	7.571	12400.0	28.90	88.6	221.4	.0308	.0771
14	7.696	10600.0	30.30	97.6	243.9	.0425	.1062
15	7.821	8500.0	31.46	105.0	262.6	.0551	.1378
16	7.946	7500.0	32.46	111.5	27R.7	.0h87	.1717
17	8.071	6800.0	33.35	117.2	293.0	.0830	.2074
18	8.196	6400.0	34.17	122.5	306.3	.0979	.2449
19	8.321	6000.0	34.95	127.5	318.8	.1136	.2839
20	8.446	5500.0	35.67	132.1	330.4	.1298	.3245
21	8.571	5200.0	36.34	136.5	341.1	.1466	.3665
22	8.696	4800.0	36.96	140.5	351.2	.1639	.4097
23	8.821	4500.0	37.54	144.2	360.6	.1817	.4542
24	8.946	4200.0	38.09	147.7	369.3	.1999	.4998
25	9.071	3900.0	38.59	151.0	377.5	.2186	.5465
26	9.821	1700.0	40.69	164.5	411.3	.3369	.8423
27	10.571	700.0	41.59	170.3	425.0	.4625	1.1562
28	11.321	200.0	41.93	172.5	431.2	.5910	1.4775
29	12.071	.0	42.01	173.0	432.4	.7206	1.8014
30	12.071	.0	.00	173.0	432.4	.6500	1.6250

TABLE 5–17. DOUBLE BARREL MACHINE GUN DYNAMICS (Con't.)

I	TIME MSEC	NORMAL CAM FORCE LB	CAM SLOPE DEG	PERIPH DRUM VEL IN/SEC	AXIAL CAM VEL IN/SEC	RECOIL VEL IN/SEC	PERIPH DRUM TRAVEL IN	AXIAL CAM TRAVEL IN	RECOIL TRAVEL IN
31	12.2452	8450.0	2.44	18.3	430.6	172.3	.0016	1.7000	.6800
3L	12.4202	8416.9	4.89	36.5	426.6	170.7	.0064	1.7750	.7100
33	12.5970	8396.9	7.36	54.4	421.7	168.7	.0144	1.8500	.7400
34	12.7761	8390.0	9.85	72.2	415.9	166.4	.0258	1.9250	.7700
3s	12.9579	8396.3	12.39	89.9	409.1	163.7	.0405	2.0000	.8000
36	13.1429	8416.1	14.98	107.4	401.4	160.5	.0588	2.0750	.8300
37	13.3319	8449.8	17.64	124.8	392.5	157.0	.0807	2.1500	.8600
38	13.5254	8498.2	20.37	142.1	382.6	153.0	.1065	2.2250	.8900
39	13.7243	8562.2	23.21	159.2	371.4	148.6	.1365	2.3000	.9200
40	13.9297	8642.9	26.16	176.3	358.9	143.6	.1710	2.3750	.9500
41	14.4886	8937.7	34.37	218.9	320.1	128.0	.2815	2.5647	1.0259
42	14.9598	9274.3	41.53	250.0	282.3	112.9	.3920	2.7067	1.0827
43	15.3818	9660.9	48.14	273.6	245.1	98.0	.5026	2.8179	1.1272
44	15.7730	10132.3	54.44	291.2	208.2	83.3	.6131	2.9066	1.1626
45	16.1442	10774.9	60.55	303.9	171.6	68.6	.7236	2.9771	1.1908
4b	16.5027	11819.7	66.54	312.1	135.4	54.2	.8342	3.0321	1.2128
47	16.8545	14079.2	72.45	315.5	99.8	39.9	.9447	3.0735	1.2294
48	17.2056	23522.3	78.32	312.0	64.5	25.8	1.0552	3.1023	1.2409
4Y	17.5623	17832.9	84.17	303.9	31.1	12.4	1.1657	3.1194	1.2477
50	17.9252	96.6	90.00	299.8	.0	.0	1.2763	3.1250	1.2500
51	18.2994	10835.0	84.17	294.9	30.1	12.1	1.3868	3.1194	1.2477
52	18.6825	7595.4	78.32	284.5	58.8	23.5	1.4973	3.1023	1.2409
53	19.0802	6612.2	72.45	272.6	86.2	34.5	1.6079	3.0735	1.2294
54	19.4973	5983.9	66.54	258.4	112.2	44.9	1.7184	3.0321	1.2128
55	19.9399	5470.2	60.55	241.9	136.6	54.6	1.8289	2.9771	1.1908
56	20.4164	5007.2	54.44	222.8	159.2	63.7	1.9394	2.9066	1.1626
57	20.9388	4572.5	48.14	201.1	180.1	72.1	2.0500	2.8179	1.1272
58	21.5256	4157.6	41.53	176.4	199.1	79.7	2.1605	2.7067	1.0027
59	22.2093	3757.6	34.37	147.8	216.1	86.4	2.2710	2.5647	1.0259
60	23.0589	3365.8	26.16	113.3	230.6	92.2	2.3816	2.3750	.9500
61	23.3814	3243.0	23.21	100.6	234.5	93.8	2.4160	2.3000	.9200
62	23.6991	3134.4	20.37	88.2	237.6	95.1	2.4460	2.2250	.8900
63	24.0131	3038.2	17.64	76.3	240.0	96.0	2.4718	2.1500	e8600
64	24.3245	2952.9	14.98	64.7	241.8	96.7	2.4938	2.0750	.8300
65	24.6339	2877.3	12.39	53.4	243.0	97.2	2.5120	2.0000	.8000
66	24.9422	2810.4	9.85	42.3	243.6	97.4	2.5268	1.9250	.7700
67	25.2500	2751.3	7.36	31.5	243.7	97.5	2.5381	1.8500	.7400
68	25.5580	2699.4	4.89	20.8	243.4	97.3	2.5462	1.7750	.7100
69	25.8667	2654.1	2.44	10.3	242.6	97.0	2.5510	1.7000	.6800
70	26.1767	2614.9	.00	.0	241.3	96.5	2.5525	1.6250	.6500

CHAPTER 6

MULTIBARREL MACHINE GUN

6–1 GENERAL

The Gatling Gun type of machine gun provides very high rates of fire by being capable of what may essentially be called simultaneous loading, firing, extracting, and ejecting and still not overexpose any one barrel to the effects of rapid, continuous fire. Each of the above four functions are performed in separate barrels during the same interval thus fixing four as the lower limit for the number of barrels, to the gun. Physical size establishes the upper limit but five or six is the usual number of barrels in each cluster with six being preferred. Several of these six-barreled guns have proved successful and are production items.

6–2 BOLT OPERATING CAM DEVELOPMENT

The closing and opening of the bolts of a multibarreled gun are regulated by a cam attached to or cut into the inner housing wall. Each bolt has a cam follower equipped with a roller that rides in the cam. As the gun rotates, carrying the bolts with it, the cam followers force these bolts into prescribed directions. Fig. 6–1 shows a cam contour. It has two dwell periods, the rear when the bolt is fully retracted and the front when the bolt is closed. The rear dwell provides time to complete the cartridge case ejection and to receive a new round. The front dwell provides firing time and holds the bolt closed until propellant gas pressures reduce to safe limits. The feeding and ejection periods have three intervals: accelerating, constant velocity, and decelerating. Because of the differences in cam force during the two periods, the accelerating distance is generally three times that of the decelerating. The constant velocity period is not absolutely essential but incorporating it has the advantage of distributing power requirements.

6–2.1 CAM ACTION

Parabolic curves are selected for the accelerating and decelerating portions of the cams because of the constant acceleration characteristic. In the same sense, straight lines form the constant velocity portions of the cam. Fig. 6–2 shows the loading diagram on the bolt and cam arrangements during acceleration. Fig. 6–3 isolates the feeding portion of the cam path shown in Fig. 6–1. It consists of two parabolic curves tangent to a straight line. The analysis for feeding or ejecting are identical. Acceleration ends at P_1 and deceleration starts at P_2, the slope β being the same at these two points. The expression for the accelerating curve is

$$y^2 = Kx. \tag{6–1}$$

The slope is

$$\frac{dx}{dy} = \frac{2y}{K} = \frac{2y}{y^2/x} = \frac{2x}{y} = \tan\beta . \tag{6–2}$$

The slope at P_1 is

$$\left(\frac{dx}{dy}\right)_1 = \frac{2x_1}{y_1} . \tag{6–3}$$

The expression for the decelerating curve is

$$(y_2 - y)^2 = K(x_2 - x) . \tag{6–4}$$

Solve for x.

$$x = x_2 - \frac{1}{K}(y_2 - y)^2 \tag{6–5}$$

$$\frac{dx}{dy} = \frac{2}{K}(y_2 - y) \tag{6–6}$$

when $y = 0$, $x = 0$, and $\frac{dx}{dy} > 0$, therefore $K = \frac{y_2^2}{x_2}$

The slope at P_2, where $y = 0$, is

$$\left(\frac{dx}{dy}\right)_2 = \frac{2x_2}{y_2} \tag{6–7}$$

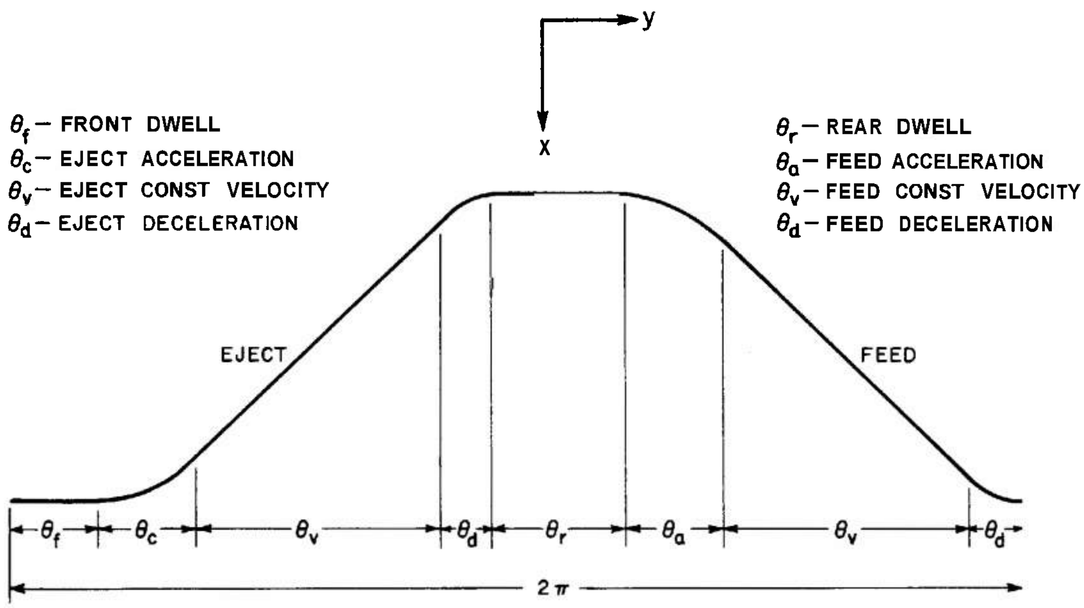


Figure 6–1. Cam Contour of Multibarrel Gun

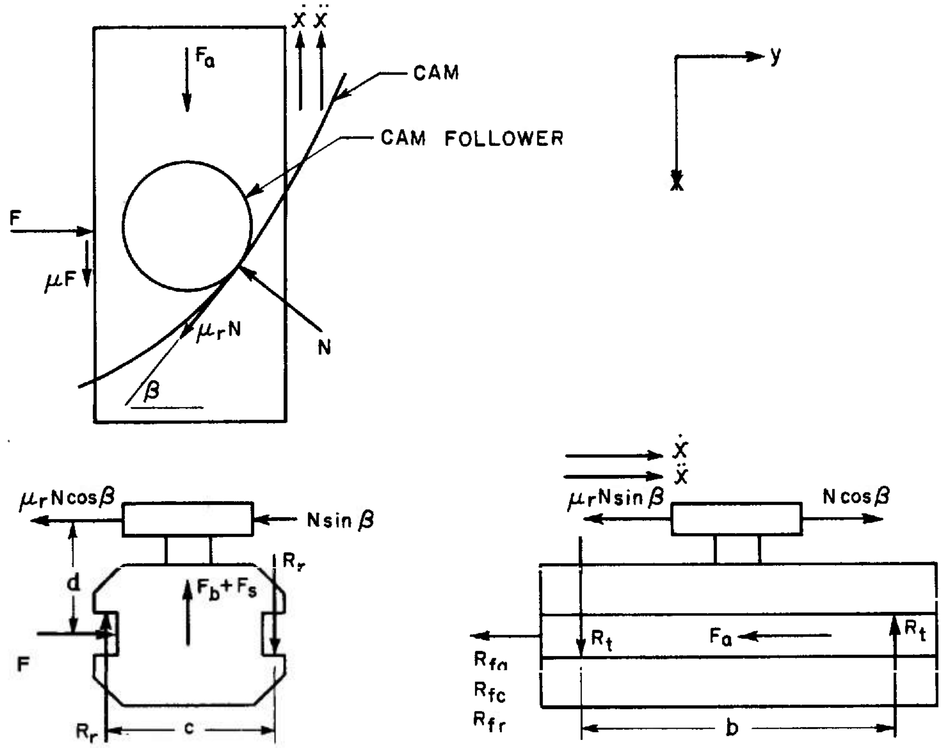
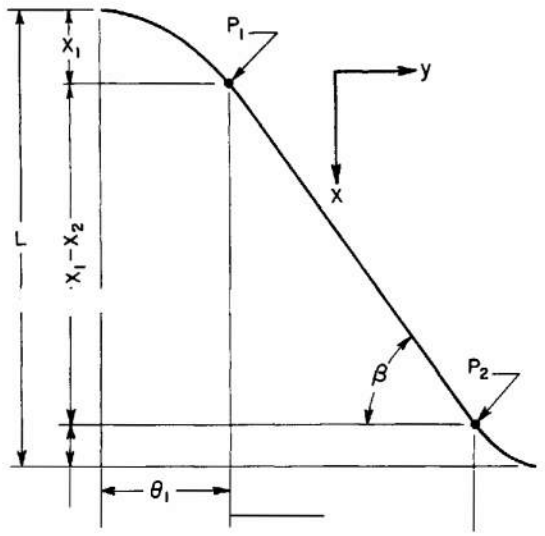


Figure 6–2. Loading Diagram of Bolt and Cam During Acceleration



$$\frac{x_1}{y_1} = \frac{x_2}{y_2} \qquad (6\text{–}8)$$

$$x_2 = \left(\frac{y_2}{y_1}\right) x_1 \,. \qquad (6\text{–}9)$$

The rotor, after its accelerating period turns at a constant velocity and the lengths of y_1 and y_2 are known, is chosen to comply with desired design conditions.

$$y_1 = R_c\theta_1 = R_c\omega t_1 \qquad (6\text{–}10)$$

$$y_2 = R_c\theta_1 = R_c\omega t_1 \qquad (6\text{–}11)$$

where $R,$ = cam radius

ω = angular velocity of the rotor

$L,$ = total peripheral length of the cam.

$$L_c = 2\pi R_c \qquad (6\text{–}12)$$

y_a is the peripheral length of the constant slope of the cam

$$y_a = \theta_a R_c \qquad (6\text{–}13)$$

$$x = \frac{1}{K}\left(y^2\right) = \frac{1}{K}\left(R_c^2\omega^2 t^2\right) \qquad (6\text{–}16)$$

The axial velocity of the bolt becomes

$$\dot{x} = \frac{2}{K}\left(R_c^2\omega^2 t\right). \qquad (6\text{–}17)$$

The corresponding acceleration is

$$\ddot{x} = \frac{2}{K}\left(R_c^2\omega^2\right) \qquad (6\text{–}18)$$

which is constant, conforming to the characteristics of the parabola. For the straight lines connecting the two parabolas where β is constant

$$x = y \tan\beta = R_c\omega t \tan\beta \qquad (6\text{–}19)$$

$$\dot{x} = R_c\omega \tan\beta \text{ (constant)} \qquad (6\text{–}20)$$

$$x = o \qquad (6\text{–}21)$$

The mechanics of the cam during feed deceleration are determined by developing Eq. *6–5*. Substitute $R_c\omega t$ for y.

$$x = x_2 - \frac{1}{K}\left(y_2^2 - 2y_2R_c\omega t + R_c^2\omega^2t^2\right) \qquad (6-22)$$

$$\dot{x} = \frac{2}{K}\left(y_2R_c\omega - R_c^2\omega^2t\right) \qquad (6-23)$$

$$\ddot{x} = -\frac{2}{K}\left(R_c^2\omega^2\right). \qquad (6-24)$$

The ejection part of the cam behaves similarly but in the opposite direction. The curve for ejection acceleration is

$$-y^2 = Kx \qquad (6-25)$$

$$x = -\frac{1}{K}\left(y^2\right) \qquad (6-26)$$

$$\frac{dx}{dy} = -\frac{2}{K}\left(y\right) \qquad (6-27)$$

when $y = y_2$, $x = x_2$, and $\frac{dx}{dy} < 0$,

therefore $K > 0$ and $K = \frac{y_2^2}{x_2}$.

Continue with the mechanics and substitute $R_c\omega t$ for y.

$$x = -\frac{1}{K}\left(R_c^2\omega^2t^2\right) \qquad (6-28)$$

$$x = -\frac{2}{K}\left(R_c^2\omega^2t\right) \qquad (6-29)$$

$$\ddot{x} = -\frac{2}{K}\left(R_c^2\omega^2\right) \qquad (6-30)$$

While ejecting over the straight portion of the cam, the mechanics are

$$x = -R_c\omega t \tan\beta \qquad (6-31)$$

$$x = -R_c\omega \tan\beta \qquad (6-32)$$

$$x = 0. \qquad (6-33)$$

The curve for the ejection deceleration is defined as

$$-\left(y_2 - y\right)^2 = K\left(x_2 - x\right) \qquad (6-34)$$

when $y = 0$, $x = 0$, $x_2 < 0$

$$x = -x_2 + \frac{1}{K}\left(y_2^2 - 2y_2y + y^2\right) \qquad (6-35)$$

$$\frac{dx}{dy} = \frac{1}{K}(-2y_2 + 2y) = \frac{2}{K}(y - y_2) \qquad (6-36)$$

when $y = 0$, $\frac{dx}{dy} < 0$, therefore $K > 0$ and $K = \frac{y_2^2}{x_2}$

Continue the mechanics and substitute $R_c\omega t$ for y

$$x = -x_2 + \frac{1}{K}(y_2^2 - 2y_2R_c\omega t + R_c^2\omega^2t^2) \qquad (6-37)$$

$$\dot{x} = \frac{2}{K}\ (R_c^2\omega^2t - y_2R_c\omega) \qquad (6-38)$$

$$\ddot{x} = \frac{2}{K}\ (R_c^2\,\omega^2) \qquad (6-39)$$

6–2.1.2 Definition of Symbols

b = moment arm for tipping track reactions

c = moment arm for rotating track reactions

d = distance, CG of bolt to center of cam roller surface

F = driving force

F_a = axial inertial force of bolt and round or of bolt and case

F_b = centrifugal force of bolt

F_{ba} = tangential inertia force of bolt

F_s = centrifugal force of round or of case

F_{sa} = tangential inertia force of round or of case

F_α = tangential inertia force of bolt and round or of bolt and case

I_d = mass moment of inertia of all rotating parts

L = length of bolt travel

$M = M_b + M_a$ = mass of bolt unit

M_a = mass of round

M_b = mass of bolt

M_{cc} = mass of case

N = normal force on roller

N_a = axial component of the normal force of the roller

N_t = transverse component of the normal force of the roller

R = radius, gun axis to bolt

$R,$ = cam radius

R_{fa} = frictional resistance due to tangential inertia forces

R_{fc} = frictional resistance due to centrifugal forces

R_{fr} = frictional resistance due to track reactions

R_r = track reactions due to rotational forces

R_t = track reactions due to tipping forces

T = torque about gun axis

x = axial acceleration of bolt

α = angular acceleration of rotor

β = angle of cam path (slope)

θ = angular displacement of rotor

μ_r = coefficient of rolling friction

μ_s = coefficient of friction of case

μ_t = coefficient of friction of track

ω = angular velocity of rotor

6-2.1.3 Cam Forces

The axial inertia force of the bolt and round is

$$F_a = (M_b + M_s)\ddot{x} \; . \tag{6-40}$$

The centrifugal force of the bolt is

$$F_b = M_b R \omega^2 \; . \tag{6-41}$$

The centrifugal force of the round is

$$F_s = M_s R \omega^2 \; . \tag{6-42}$$

The frictional resistance due to centrifugal force is

$$R_{fc} = \pm (\mu_t F_b + \mu_s F_s) \; . \tag{6-43}$$

The tangential inertia force of bolt and round induced by angular acceleration is

$$F_\alpha = F_{ba} + F_{sa} = M_b R \alpha + M_s R \alpha \; . \tag{6-44}$$

The frictional resistance due to the tangential inertia forces is

$$R_{fa} = \pm (\mu_t F_{ba} + \mu_s F_{sa}) \; . \tag{6-45}$$

The axial component of the normal force of the roller is

$$N_a = N \cos\beta - \mu_r N \sin\beta \tag{6-46}$$

where $\mu_r N$ is the resistance induced by rolling friction.

The transverse component of the normal force is

$$N_t = N \sin\beta + \mu_r N \cos\beta \; . \tag{6-47}$$

The driving force of the cam is

$$F = N_t = N(\sin\beta + \mu_r \cos\beta) \; . \tag{6-48}$$

The track reactions due to rotational forces are found by balancing moments in the plane perpendicular to the bolt axis.

$$cR_r = dF$$

$$R_r = \left(\frac{d}{c}\right)F \qquad (6-49)$$

The track reactions due to tipping forces are found by balancing moments in the vertical plane parallel to the bolt axis.

$$bR_t = dN_a$$

$$R_t = \left(\frac{d}{b}\right)N_a \qquad (6-50)$$

The frictional resistance due to track reactions is

$$R_{fr} = \pm 2\mu_t(R_r + R_t).$$

R_{fa}, R_{fc} and R_{fr} have the same algebraic sign as $\dot{x}$.

The normal force on the cam roller is found by balancing the axial forces thus $\Sigma F_x = 0$

$$F_a + \mu_t N_t + R_{fc} - N_a + R_{fr} + R_{fa} = 0 \qquad (6-51)$$

Substitute all values containing N for the terms in Eq. *6–51* and then solve for N. Note that N is always positive.

$$N = \left|(F_a + R_{fc} + R_{fa})/\left[(\mu_r + \mu_t + 2\frac{d}{c}\mu_t - 2\frac{d}{b}\mu_r\mu_t)\sin\beta + (\mu_r\mu_t + 2\frac{d}{c}\mu_r\mu_t + 2\frac{d}{b}\mu_t - 1.0)\cos\beta\right]\right| \qquad (6-52)$$

This force system also applies to the constant velocity portions of the cam, F_a being zero but all other force components acting in the same direction.

Fig. *6–4* shows the loading diagram on the bolt and cam arrangement during deceleration. All the developing equations are the same as for acceleration except

$$N_a = N\cos\beta + \mu_r N\sin\beta \qquad (6-53)$$

$$N_t = N\sin\beta - \mu_r N\cos\beta \qquad (6-54)$$

$$F = N_t = N(\sin\beta - \mu_r \cos\beta). \qquad (6-55)$$

Equate the sum of the forces along the x-axis to zero, $\Sigma F_x = 0$.

$$F_a + \mu_t N_t + R_{fc} + N_a + R_{fr} + R_{fa} = 0 \qquad (6-56)$$

Substitute all values containing N for the terms in Eq. *6–56* and solve for N.

$$N = \left|(F_a + R_{fc} + R_{fa})/\left[(\mu_r\mu_t + 2\frac{d}{c}\mu_r\mu_t - 2\frac{d}{b}\mu_t - 1.0)\cos\beta - (\mu_r + \mu_t + 2\frac{d}{b}\mu_r\mu_t + 2\frac{d}{c}\mu_t)\sin\beta\right]\right| \qquad (6-57)$$

The driving **torque about the gun axis is**

$$T = \Sigma FR_c + I_d\alpha \qquad (6-58)$$

where ΣF is the total driving force on all bolts.

6–2.1.4 Locking Angle

The cam becomes self-locking when the required driving force becomes infinite. Substitution of the expression for N of Eq. *6–52* into Eq. *6–48* indicates that $F = \infty$ when the denominator becomes zero. Equate the denominator to zero; divide by cos β; and then solve for β_L, the locking angle, defined by Eq. *6–59*.

$$\beta_L = \text{Tan}^{-1}\frac{\mu_r\mu_t\left(1 + 2\frac{d}{c}\right) + 2\mu_t\left(\frac{d}{b}\right) - 1.0}{\mu_t\left[2\mu_r\left(\frac{d}{b}\right) - 2\left(\frac{d}{c}\right) - 1\right] - \mu_r} \qquad (6-59)$$

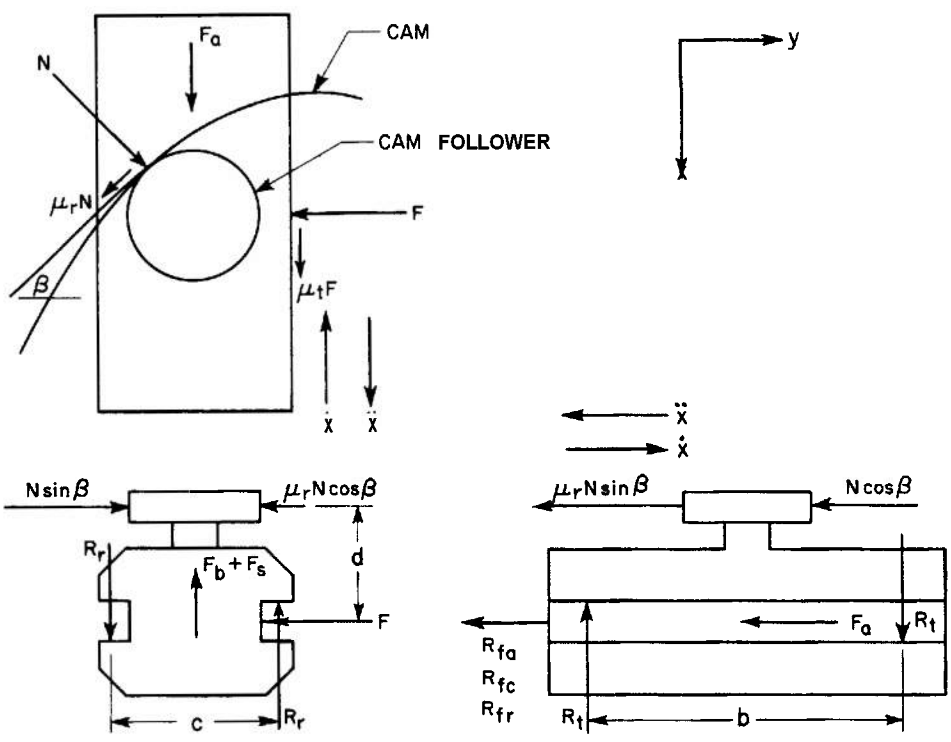


Figure 6–4. Loading Diagram of Bolt and Cam During Deceleration

6–2.2 ROTOR KINEMATICS

The rotor is brought to speed under constant, varying, or a combination of both types angular acceleration during a prescribed period of time t. During the acceleration of the rotor, the acceleration of the round induced by the cam is modified by the components derived from the angular acceleration. If the acceleration is constant at a, the angular velocity is

$$\omega = at \tag{6–60}$$

and the angular travel becomes

$$e = \frac{1}{2}\left(\alpha t^2\right) \cdot \tag{6–61}$$

While the rotor has constant acceleration when traversing the constant slope of the cam, the axial travel of the cam follower according to **Eqs. 6–19** and **6–31** is

$$x = \pm \left(R_c \theta\right) \tan\beta = \pm \frac{1}{2}\left(R_c \alpha t^2\right) \tan\beta. \tag{6–62}$$

Positive indicates feed; negative, ejection. The linear velocity is

$$x = \pm R_c \alpha t \tan\beta. \tag{6–63}$$

The linear acceleration becomes

$$x = \pm R_c \alpha \tan\beta. \tag{6–64}$$

When traversing the feed and ejection acceleration portions of the cam, in this case a parabola, the axial travel, **Eq. 6–16,** while the rotor has constant acceleration is defined by

$$x = \pm \frac{1}{K}\left(R_c^2 \theta^2\right) = \frac{R_c^2}{4K}\left(\alpha^2 t^4\right) \tag{6–65}$$

$$\dot{x} = \pm \frac{R_c^2}{K}\left(\alpha^2 t^3\right) \tag{6–66}$$

$$x = \pm 3\left(\frac{R_c^2}{K}\right)\left(\alpha^2 t^2\right) \tag{6–67}$$

Positive indicates feed acceleration whereas negative indicates ejection acceleration. During feed and ejection deceleration, **Eq.** *6–22,* while the rotor has constant acceleration, **Eq.** *6–61,* the mechanics alternate correspondingly.

$$x = \pm x_2 \mp \frac{1}{K}\left[y_2^2 - y_2 R_c \alpha t^2 + \frac{1}{4}\left(R_c^2 \alpha^2 t^4\right)\right] \tag{6–68}$$

$$\dot{x} = \mp \frac{1}{K}\left(R_c^2 \alpha^2 t^3 - 2 y_2 R_c \alpha t\right) \tag{6–69}$$

$$\ddot{x} = \mp \frac{1}{K}\left(3 R_c^2 \alpha^2 t^2 - 2 y_2 R_c \alpha\right) \tag{6–70}$$

During dwell periods at constant rotor acceleration; the travel, velocity, and acceleration are all zero, i.e., $x = 0, \dot{x} = 0, \ddot{x} = 0$.

When the rotor has a variable acceleration, a constantly decreasing one is generally preferred, thus

$$\frac{d\alpha}{dt} = \frac{d^3\theta}{dt^3} = K_\alpha \tag{6–71}$$

$$\alpha = \frac{d^2\theta}{dt^2} = K_\alpha t + C_1 \tag{6–72}$$

$$\omega = \frac{d\theta}{dt} = \frac{1}{2}\left(K_\alpha t^2\right) + C_1 t + C_2 \tag{6–73}$$

$$\theta = \frac{1}{6}\left(K_\alpha t^3\right) + \frac{1}{2}\left(C_1 t^2\right) + C_2 t + C_3 \tag{6–74}$$

While traversing the constant slope of the cam for feed and ejection, **Eqs.** *6–19* and *6–31*, when $x = \pm R_c \theta$, tan β, the mechanics are alternately

$$x = \pm R_c \left[\frac{1}{6}\left(K_\alpha t^3\right) + \frac{1}{2}\left(C_1 t^2\right) + C_2 t + C_3\right] \tan\beta \tag{6–75}$$

$$\dot{x} = \pm R_c \left[\frac{1}{2}\left(K_\alpha t^2\right) + C_1 t + C_2\right] \tan\beta \tag{6–76}$$

$$\ddot{x} = \pm R_c \left(K_\alpha t + C_1\right) \tan\beta . \tag{6–77}$$

While traversing the increasing slope of the cam, *Eqs. 6–16* and *6–28,* when $x = \pm \frac{1}{K}\left(R_c^2 \theta^2\right)$, the mechanics for feed and ejection appear, respectively,

$$x = \pm \frac{R_c^2}{K}\left[\left(\frac{K_\alpha^2}{36}\right) t^6 + \left(\frac{K_\alpha C_1}{6}\right) t^5 + \left(\frac{4K_\alpha C_2 + 3C_1^2}{12}\right) t^4 + \left(\frac{K_\alpha C_3 + 3C_1 C_2}{3}\right) t^3 + \left(C_1 C_3 + C_2^2\right) t^2 + 2 C_2 C_3 t + C_3^2\right] \tag{6–78}$$

$$\dot{x} = \pm \frac{R_c^2}{K}\left[\frac{K_\alpha^2 t^5}{6} + \left(\frac{5K_\alpha C_1}{6}\right) t^4 + \left(\frac{4K_\alpha C_2 + 3C_1^2}{3}\right) t^3 + \left(K_\alpha C_3 + 3C_1 C_2\right) t^2 + 2\left(C_1 C_3 + C_2^2\right) t + 2 C_2 C_3\right] \tag{6–79}$$

$$\ddot{x} = \pm \frac{R_c^2}{K}\left[\left(\frac{5K_\alpha^2}{6}\right) t^4 + \left(\frac{10}{3} K_\alpha C_1\right) t^3 + \left(4K_\alpha C_2 + 3C_1^2\right) t^2 + 2\left(K_\alpha C_3 + 3C_1 C_2\right) t + 2\left(C_1 C_3 + C_2^2\right)\right] \tag{6–80}$$

If $F\left(K_\alpha t\right)$ represents the function in the brackets of **Eq.** *6–78,* then the last three equations reduce to

$$x = \pm \frac{R_c^2}{K}\left[F(K_\alpha t)\right] \tag{6–81}$$

$$\dot{x} = \pm \frac{R_c^2}{K}\left[F'(K_\alpha t)\right] \tag{6–82}$$

$$\ddot{x} = \pm \frac{R_c^2}{K}\left[F''(K_\alpha t)\right] . \tag{6–83}$$

The mechanics for deceleration during feed, **Eq. 6–22,** and ejection, **Eq. 6–37,** are alternately

$$x = \pm x_2 \mp \frac{1}{K} \left\{ -2y_2R_c \left[\left(\frac{1}{6} K_\alpha \right) t^3 + \left(\frac{1}{2} C_1 \right) t^2 + C_2 t + C_3 \right] + R_c^2 \left[F\,(K_\alpha t) \right] \right\} \qquad (6\text{–}84)$$

$$\dot{x} = \pm \frac{1}{K} \left\{ 2y_2R_c \left[\left(\frac{1}{2} K_\alpha \right) t^2 + C_1 t + C_3 \right] - R_c^2 \left[F'\,(K_\alpha t) \right] \right\} \qquad (6\text{–}85)$$

$$\ddot{x} = \pm \frac{1}{K} \left\{ 2y_2R_c \left(K_\alpha t + C_1 \right) - R_c^2 \left[F''(K_\alpha t) \right] \right\}. \qquad (6\text{–}86)$$

6–2.3 ILLUSTRATIVE PROBLEM

Compute the cam accelerating forces, the torque needed to develop these forces, and all associated data to operate a **20** mm, 6-barreled gun. The assigned data

b = **3.0** in., moment arm of tipping track reactions

c = **1.5** in., moment arm of rotational track reactions

d = **0.732** in., CG of bolt to center of cam roller surface

f_r = **3000** rounds/min, firing rate (equivalent to angular velocity of **52.36** rad/sec)

I_d = **11.2** lb-in.-sec^2, moment of inertia of all rotating parts

L = **6.6** in., length of bolt travel

R = **2.643** in., radius, gun axis to bolt CG

R_c = **3.375** in., cam radius

t_α = **0.35** sec, accelerating time of rotor

W_b = **1.15** lb, bolt weight

W_a = **0.57** lb, weight of total round

W_{cc} = **0.25** lb, weight of empty case

α_m = **200** rad/sec^2, maximum acceleration of rotor

μ_r = **0.063,** coefficient of rolling friction of cam roller

μ_s = **0.22,** coefficient of friction of case

μ_t = **0.125,** coefficient of friction of track

Fig. **6–1** illustrates the developed cam. For an effective firing cycle, the bolt travel in terms of peripheral travel of the rotor

8, = **36°**, dwell while bolt is fully retracted

8, = **42°**, acceleration distance with total round

8, = **90°**, distances at constant velocity in each direction

θ_d = 12°, decelerating distance in each direction

θ_f = **40°**, dwell while firing

θ_c = **38°**, acceleration distance with empty case

The total peripheral length of the cam, **Eq. 6–12**

$$L_c = 2\pi R_c = 6.75\pi = 21.2058 \text{ in.}$$

6–2.3.1 Cam Analysis During Feed, Rotor at Constant Velocity

The peripheral travel of the bolt while carrying the total round via Eq. **6–10**

$$y_1 = R_c\theta_a = 3.375 \left(\frac{42\pi}{180} \right) = 2.4740 \text{ in.}$$

$$y_2 = R_c\theta_d = 3.375 \left(\frac{12\pi}{180} \right) = 0.7069 \text{ in.}$$

The peripheral travel during constant velocity, Eq. 6–13

$$y_a = \frac{\pi}{2} R_c = 5.3014 \text{ in.}$$

From Eq. 6–15

$$x_1 = \frac{Ly_1}{2y_a + y_1 + y_2} = \frac{6.6 \times 2.474}{13.7837} = 1.1846 \text{ in.}$$

From Eq. 6–9

$$x_2 = \left(\frac{y_2}{y_1}\right) x_1 = \left(\frac{0.7069}{2.474}\right) 1.1846$$

$$= 0.3385 \text{ in.}$$

From Eq. 6–2

$$\beta = \text{Tan}^{-1} \frac{2x_2}{y_2} = \text{Tan}^{-1} \frac{0.677}{0.7069}$$

$$= \text{Tan}^{-1}\ 0.9577 = 43°46'$$

$$\sin\beta = 0.69172$$

$$\cos\beta = 0.72216.$$

According to Eq. 6–2, for the accelerating curve

$$K_1 = \frac{2y_1}{\tan\beta} = \frac{4.948}{0.9577} = 5.1665 \text{ in.}$$

For the decelerating curve, K_2 is found when $y = 0$ in Eq. 6–6

$$K_2 = \frac{2y_2}{\tan\beta} = \frac{1.4138}{0.9577} = 1.4762 \text{ in.}$$

The firing rate of 3000 rounds/min is equivalent to 500 rpm since there are six barrels. The angular velocity is

$$\omega = \frac{2 \times 500\pi}{60} = 52.36 \text{ rad/sec.}$$

The bolt acceleration with total round, Eq. 6–18, is

$$\ddot{x} = \frac{2}{K_1}\left(R_c^2\omega^2\right) = \frac{22.78 \times 2742}{5.1665}$$

$$= 12089 \text{ in./sec}^2 = 31.29\ g.$$

The inertia force of bolt and round, Eq. 6–40, is

$$F_a = (M_b + M_a)\,\ddot{x} = 1.72 \times 31.29 = 53.8 \text{ lb.}$$

The centrifugal forces of bolt and round according to Eqs. 6–41 and 6–42 are

$$F_b = \left(\frac{W_b}{g}\right) R\omega^2 = 1.15 \times 18.8 = 21.6 \text{ lb}$$

where

$$\frac{R\omega^2}{g} = \frac{2.643 \times 2742}{386.4} = 18.8\ g$$

$$F_s = M_s R\omega^2 = 0.57 \times 18.8 = 10.7 \text{ lb.}$$

The frictional resistance due to centrifugal force (Eq. 6–43,; $>$ 0)

$$R_{fc} = \mu_t F_b + \mu_s F_s = 0.125 \times 21.6 + 0.22 \times 10.7$$

$$= 5 \text{ lb.}$$

$$R_{fa} = 0 \quad (\text{since } \alpha = 0)$$

If we substitute the numerical equivalents for the general terms, Eq. 6–52 may be written

$$N = \frac{58.8}{0.9234 \cos\beta - 0.3062 \sin\beta}.$$

The locking angle, Eq. 6–59, is

$$\beta_L = \text{Tan}^{-1} \frac{0.9234}{0.3062} = \text{Tan}^{-1}\ 3.0157 = 71°39'.$$

The bolt deceleration with total round, Eq. 6–24, is

$$\ddot{x} = -\left(\frac{2}{K_2}\right) R_c^2\omega^2 = \frac{-22.78 \times 2742}{1.4762}$$

$$= -42{,}313 \text{ in./sec}^2 = -109.5\ g,$$

The inertia force of bolt and round, Eq. 6–40, is

$$F_a = (M_b + M_a)\ddot{x} = 1.72\,(-109.5) = -188.3 \text{ lb.}$$

By proper substitution of numerical equivalents, Eq. 6–57 reads ($\dot{x} > 0$)

$$N = \frac{183.3}{1.0454\cos\beta + 0.3138\sin\beta}$$

While the bolt moves along the constant slope of the cam, $\ddot{x} = 0$, and the normal force on the roller becomes

$$N = \frac{R_{fc}}{0.9234\cos\beta - 0.3062\sin\beta} = \frac{5}{0.6668 - 0.2118} = 11 \text{ lb.}$$

6–2.3.2 Cam Analysis During Ejection, Rotor at Constant Velocity

The peripheral travel of the bolt carrying the empty case during acceleration is computed from Eq. 6–10

$$y_1 = R_c\theta_c = 3.375\left(\frac{38\pi}{180}\right) = 2.2384 \text{ in.}$$

$y_2 = 0.7069$ in., $y_a = 5.3014$ in., same as for total round.

From Eq. 6–15

$$x_1 = \frac{Ly_1}{2y_a + y_1 + y_2} = \frac{6.6 \times 2.2384}{13.5481} = 1.0904 \text{ in.}$$

From Eq. 6–9

$$x_2 = \left(\frac{y_2}{y_1}\right)x_1 = \left(\text{——}\right) 1.0904 = 0.3444 \text{ in.}$$

From Eq. 6–2

$$\beta = \text{Tan}^{-1}\frac{2x_1}{y_1} = \text{Tan}^{-1}\frac{2.1808}{2.2384} = \text{Tan-}' \; 0.97426 = 44°15'$$

$$\sin\beta = 0.69779 \qquad \cos\beta = 0.71630.$$

For the accelerating curve, Eq. 6–2,

$$K_1 = \frac{2y_1}{\tan\beta} = \frac{4.4768}{0.9743} = 4.5944 \text{ in.}$$

For the decelerating curve, K_2 is found when $\dot{y} = 0$ in Eq. 6–6

$$K_2 = \frac{2y_2}{\tan\beta} = \frac{\text{[illegible]}}{0.9743} = 1.4509 \text{ in.}$$

The bolt acceleration with empty case, Eq. 6–30, is

$$\ddot{x} = -\left(\frac{2}{K_1}\right) R_c^2\omega^2 = -\frac{22.78 \times 2742}{4.5944}$$

$$= -13595 \text{ in./sec}^2 = -35.2 \text{ g.}$$

The inertia force of bolt and empty case, Eq. 6–40, is

$$F_a = (M_b + M_{cc})\ddot{x} = 1.40(-35.2) = -49.3 \text{ lb.}$$

The forces during ejection have the same identification as those during feed since the same dynamic equations apply to both periods.

The centrifugal force F_s of the empty case, Eq. 6–42, is

$$F_s = M_{cc}R\omega^2 = 0.25 \times 18.8 = 4.7 \text{ lb.}$$

The frictional resistance due to centrifugal force (Eq. 6–43, $\dot{x} < 0$)

$$R_{fc} = -(\mu_t F_b + \mu_s F_s)$$

$$= -(0.125 \times 21.6 + 0.22 \times 4.7) = -3.7 \text{ lb.}$$

$$R_{fa} = 0$$

After numerical equivalents are substituted, Eq. 6–52 reads

$$N = \frac{53.0}{0.9234 \cos\beta - 0.3062 \sin\beta} \cdot$$

The bolt deceleration with empty case, Eq. 6–39, becomes

$$\ddot{x} = \left(\frac{2}{K_2}\right) R_c^2\omega^2 = \frac{22.78 \times 2742}{1.4509}$$

$$= 43{,}051 \text{ in./sec}^2 = 111.4 \text{ g.}$$

From Eq. 6–40

$$F_a = (M_b + M_{cc})\ddot{x} = 1.40 \times 111.4 = 156 \text{ lb.}$$

The substitution of numerical equivalents makes Eq. 6–57 read ($\dot{x} < 0$)

$$N = \frac{152.3}{1.0454 \cos\beta + 0.3138 \sin\beta}$$

For the constant slope portion of the cam while $\ddot{x} = 0$,

$$N = \frac{R_{fc}}{0.9234 \cos\beta - 0.3062 \sin\beta} = \frac{3.7}{0.4452} = 8 \text{ lb.}$$

6–2.3.3 Cam Analysis During Rotor Acceleration

The rotor is brought to speed under a constant acceleration of 200 rad/sec^2 for 0.1736 sec and then at a constantly reducing acceleration that is defined in Eqs. 6–71 and 6–72. When t = 0 and α = 200, in Eq. 6–72, C_1 = 200; when α = 0, $K_\alpha t$ = -200. Under constant acceleration, the rotor has achieved an angular velocity of 34.72 rad/sec; therefore, for the initial conditions of Eq. 6–73, when t = 0, ω = 34.72 and C_2 = 34.72,

$$\omega = \frac{d\theta}{dt} = \frac{1}{2}\left(K_\alpha t^2\right) + 200t + 34.72 \qquad (6–87)$$

substitute -200 for $K_\alpha t$ and 52.36 rad/sec for ω (see par. 6–2.3.1), the upper limit of the angular velocity

$$52.36 = -100t + 200t + 34.72$$

$$t = \frac{17.64}{100} = 0.1764 \text{ sec}$$

$$K_\alpha = -\frac{200}{0.1764} = -1133.79.$$

Rewrite Eqs. 6–72 and 6–73 and include the time elapsed during constant acceleration

$$\alpha = \frac{d^2\theta}{dt^2} = -1133.79(t - 0.1736) + 200 \qquad (6–88)$$

$$\omega = \frac{d\theta}{dt} = -566.895(t - 0.1736)^2 + 200(t - 0.1736) + 34.72 \qquad (6–89)$$

Substitute the proper values into Eq. 6–74 and integrate

$$\theta = -188.965\, t^3 + 100\, t^2 + 34.72\, t + C_3 \qquad (6–90)$$

when t = 0, θ = 3.014, and C_3 = 3.014. Modify the time by compensating for the constant acceleration period

$$\theta = -188.965(t - 0.1736)^3 + 100(t - 0.1736)^2 + 34.72(t - 0.1736) + 3.014 \qquad (6–91)$$

when t = 0.35 sec

$$\theta = -1.037 + 3.112 + 6.125 + 3.014 = 11.215 \text{ rad.}$$

During constant acceleration of $\alpha = 200$,

$$\omega = \alpha t = 200\,t \text{ when } t \leqslant 0.1736 \tag{6-92}$$

$$\theta = \frac{1}{2}\left(\alpha t^2\right) = 100\,t^2 \text{ when } t \leqslant 0.1736 \tag{6-93}$$

6-2.3.4 Digital Computer Routine for Gun Operating Power

A digital computer program has been compiled in FORTRAN IV language for the UNIVAC 1107 Computer. The program follows, in proper sequence, the computing procedures discussed throughout Chapter 6 i.e., begin with constant acceleration; continue at a reducing acceleration at a constant rate; and then complete the computations while the rotor is turning at constant velocity.

Fig. 6-5 is a visual concept of the analysis. The drum and, therefore, the projected cam periphery are divided into six equal zones, each zone being occupied by one cam follower and corresponding bolt which are numbered in sequence according to zone number. Although all bolts travel the full periphery during each rotor revolution, we assume that each travels, from θ_0 to θ_5 six times in one zone. Thus, in Fig. 6-5, Bolt No. 1 moves from 0° to 60°. On reaching 60° (θ_5) it becomes Bolt No. 2. In the meantime, Bolt No. 2 moves from 60° (θ_0) to 120° (θ_5) where it becomes Bolt No. 3. The sequence continues until Bolt No. 6 becomes Bolt No. 1 and then the cycle repeats. This procedure is a convenient method for defining the dynamics of all bolts at any position on the cam. For the analysis, all bolts are assumed to start from θ_0 and in line, and traverse the distance to θ_5. Fig. 6-1 shows the relative positions on the cam.

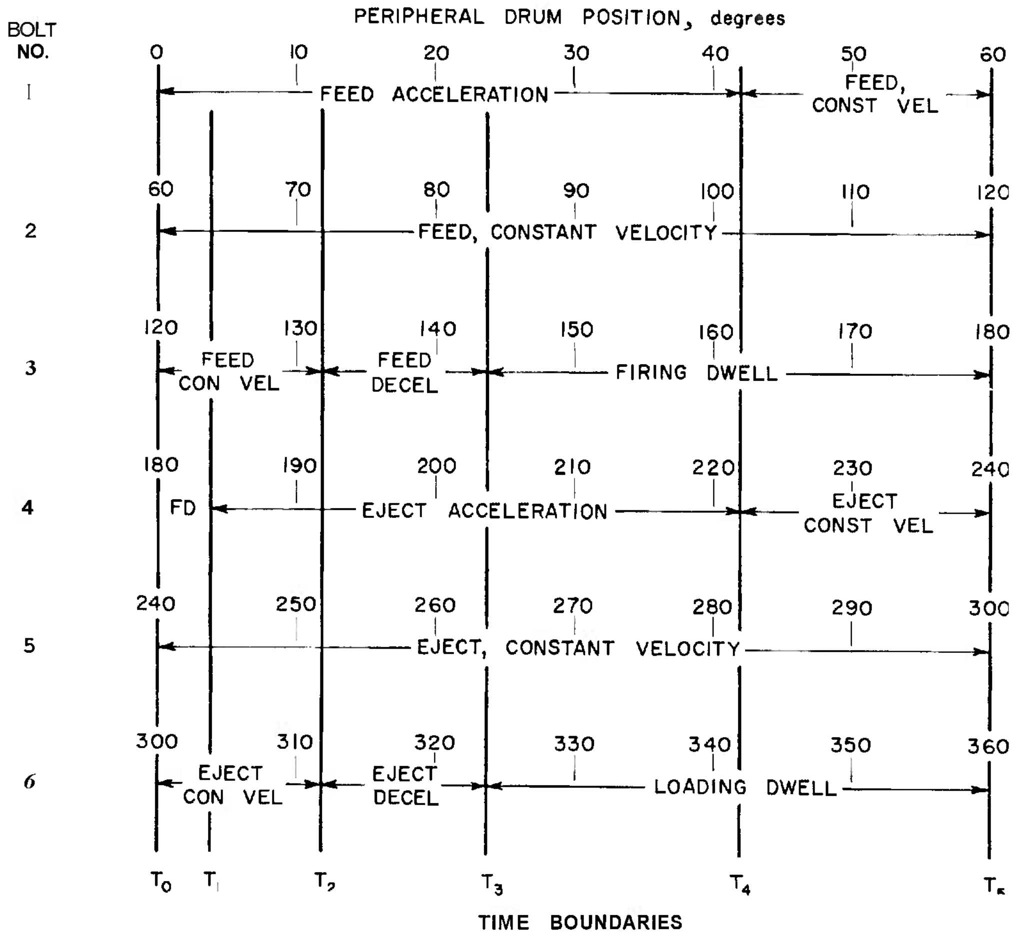


Figure 6-5. Bolt Position Diagram for Computer Analysis

Each accelerating period, constant and varying, is divided into 40 equal differential time increments. A time increment of one millisecond is then assigned to the period of constant velocity. The analysis continues the constant velocity period until one cycle of bolt travel, θ_0 to θ_5 is completed. After the torque, **Eq.** 6–58, has been computed for a given increment, the required horsepower to produce this torque is computed

$$HP = T\omega/6600$$

where 1 horsepower = 6600 in.-lb/sec.

Table 6–1 lists the symbol-code correlation for the computed variables of the computer program. Table 6–2 lists the symbol-code correlation and the numerical values of the constants. Computed cam dynamics for three increments of time are listed in Table 6–3 as a sample output, while Table 6–4 lists the computed torques and horsepower required for each increment. The flow chart, A–13, and the program listing, A–14, are in the Appendix.

6–3 RATING OF GAS-OPERATED AND EXTERNALLY POWERED GUNS

The choice **of** a gas-operated gun, or whether an externally powered weapon should be selected over a gas-operated one, is not the superiority **of** one type over another but rather the tactical purpose of each type. The impingement and tappet types are usually assigned to carbines and subcaliber automatic guns with neither being markedly superior to the other. For small caliber machine guns, cal **.30** and .50, firing at the rate of about 1000 rpm, the cut-off expansion or expansion types are appropriate. Rate of fire within limits is controlled by design detail. Everything being equal, the expansion type should be faster; its gas port is never sealed off from the bore gases.

High rates of fire for large caliber machine guns, cal .60 and above, are generally not feasible for the single chambered type. Long bolt travel for loading and extracting consumes too much time for high cyclic rates. The revolver-and Gatling-types restore the large calibers to the high firing rate category. The revolver-types may be gas-operated or may be driven by external power. Firing rates are in the order of 1000 to 1200 rpm. Gatling-types, because of ducting problems, are consigned to external drives and normally have the highest sustained rate of fire of all machine guns. The multiple barrels of the Gatling-type provide superiority over the single-barreled revolver-type in the areas of fire power, reliability and durability, and generally require less maintenance. On the other hand, the revolver-type has better handling characteristics and is more versatile with respect to efficiency, the Gatling-type being restricted to those periods of highly intensive fire for short periods of time such as in air-to-air combat.

TABLE 6–1. SYMBOL-CODE CORRELATION OF VARIABLES FOR MULTIBARREL GUN

Symbol	Code	Symbol	Code
F	F	t (msec)	TIMEM
F_a	FA	x	X
FR_c	TORKB	x	V
ΣFR_c	BTSUM	x	A
HP	POWER	Y	Y
$I_d\alpha$	TOW	a	ALPHA
N	EN	p	B
R_{fa}	RFA	β°	BDEG
R_{fc}	RFC	8	THETA
T	TORK	8"	THETAD
t	T	ω	OMEGA

TABLE 6-42. SYMBOL-CODE CORRELATION AND INPUT FOR GUN OPERATING POWER

No.	Symbol	Code	Data	No.	Symbol	Code	Data
1	K_{af}	AFK	5.1774	24	α_o	ALPHAO	200
2	K_{df}	DFK	1.4762	25	ω_{max}	OMEGAM	52.36
3	K_{ae}	AEK	4.5944	26	y_{1f}	Y1F	2.4740
4	K_{de}	DEK	1.4509	27	y_{2f}	Y2F	0.7069
5	K_{α}	AK	- 1133.79	28	x_{1f}	X1F	1.1846
6	C_1	C1	200.0	29	x_{2f}	X2F	0.3385
7	C_2	c2	34.72	30	$tan\beta_f$	TANBF	0.9577
8	C_3	c3	3.014	31	$tan\beta_e$	TANBE	0.9743
9	θ_1	ANGLE1	0.0698	32	y_{1e}	Y1E	2.2384
10	θ_2	ANGLE2	0.2094	33	y_{2e}	Y2E	0.7069
11	θ_3	ANGLE3	0.4189	34	x_{1e}	X1E	1.0904
12	θ_4	ANGLE4	0.7330	35	x_{2e}	X2E	0.3444
13	θ_5	ANGLE5	1.0472	36	g	G	386.4
14	L	EL	6.6	37	Δt_1	DELT1	0.00434
15	R	R	2.643	38	Δt_2	DELT2	0.00441
16	R_c	RC	3.375	39	Δt_3	DELT3	0.001
17	W_b	WB	1.15	40	μ_r	EMUR	0.063
18	W_s	WST	0.57	41	μ_s	EMUS	0.22
19	W_{cc}	WSE	0.25	42	μ_t	EMUT	0.125
20	K_{c1}	COEFC1	0.9234	43	I	EYE	11.2
21	K_{c2}	COEFC2	1.0454	44	β_{f1}	BF1	0.76387
22	K_{s1}	COEFS1	0.3062	45	β_{e1}	BE1	0.75230
23	K_{s2}	COEFS2	0.3138				

TABLE 6-3. CAM DYNAMICS

CAM DYNAMICS AT 116.59 MILLISECONDS

BOLT NO	BOLT ACCELERATION IN/SEC/SEC	BOLT VELOCITY IN/SEC	BOLT AXIAL TRAVEL INCH	AXIAL INERTIA FORCE POUND	NORMAL FRICTION FORCE POUND	ANGULAR INERTIA FORCE POUND	CAM CURVE ANGLE DEGREE	CAM NORMAL FORCE POUND	CAM DRIVING FORCE POUND	BOLT DRIVING TORQUE LB-IN
1	54.	5.91	.2	0.	1.	0.	22.143	1.	1.	2.
2	646.	75.37	3.209	3.	1.	0.	43.767	8.	6.	20.
3	536.	38.40	6.512	2.	1.	0.	26.012	5.	2.	8.
4	=34.	-27.30	6.458	-0.	1.	0.	19.133	1.	0.	1.
5	-641.	-74.72	3.530	-2.	1.	0.	43.517	7.	5.	17.
6	-622.	-38.07	.888	-2.	1.	0.	25.817	4.	2.	7.

CAM DYNAMICS AT 118.84 MILLISECONDS

BOLT NO	BOLT ACCELERATION IN/SEC/SEC	BOLT VELOCITY IN/SEC	BOLT AXIAL TRAVEL INCH	AXIAL INERTIA FORCE POUND	NORMAL FRICTION FORCE POUND	ANGULAR INERTIA FORCE POUND	CAM CURVE ANGLE DEGREE	CAM NORMAL FORCE POUND	CAM DRIVING FORCE POUND	BOLT DRIVING TORQUE LB-IN
1	72.	8.06	.293	0.	1.	0.	25.449	2.	1.	3.
2	646.	76.82	3.380	3.	1.	0.	43.767	8.	6.	20.
3	486.	19.75	6.578	2.	1.	0.	13.831	4.	1.	4.
4	-50.	-33.90	6.389	-0.	1.	0.	22.911	1.	1.	2.
5	-6	-76.16	3.361	-	1.	0.	43.517	7.	3.	17.
6	-599.	-19.58	.023	-2.	1.	0.	13.716	4.	1.	4.

CAM DYNAMICS AT 121.08 MILLISECONDS

BOLT NO	BOLT ACCELERATION IN/SEC/SEC	BOLT VELOCITY IN/SEC	BOLT AXIAL TRAVEL INCH	AXIAL INERTIA FORCE POUND	NORMAL FRICTION FORCE POUND	ANGULAR INERTIA FORCE POUND	CAM CURVE ANGLE DEGREE	CAM NORMAL FORCE POUND	CAM DRIVING FORCE POUND	BOLT DRIVING TORQUE LB-IN
1	93.	11.08	.386	0.	1.	0.	28.641	2.	A	3.
2	646.	78.27	3.554	5.	1.	0.	43.767	8.	6.	20.
3	427.	-.02	6.600	2.	1.	0.	-.013	3.	0.	A
4	-69.	-40.85	6.306	-0.	1.	0.	26.555	2	1.	3.
5	-6	- 0	3.188	-	1.	0.	43.31 7	7.	3	17.
6	-567.	.02	.801	-2.	1.	0.	-.013	3.	0.	1.

TABLE 6–4. GUN OPERATING POWER

N-CRI-MENT	TIME MILSEC	ROTOR ANGULAR ACCELERATION RAD/SEC/SEC	ROTOR ANGULAR VELOCITY RAD/SEC	ROTOR ANGULAR TRAVEL DEGREE	BOLT PERIPHERAL TRAVEL INCH	TOTAL CAM TORQUE LB-IN	ROTOR TORQUE LB-IN	REQUIRED TORQUE LB-IN	REQUIRED HORSE-POWER
1	26.420	200.00	5.28	4.0	.236	62.	2240.	2302.	1.8
2	31.255	200.00	6.25	5.6	.330	63.	2240.	2303.	2.2
3	36.090	200.00	7.22	7.5	.440	64.	2240.	2304.	2.3
4	40.925	200.00	8.19	9.6	.565	66.	2240.	2306.	2.9
5	45.760	200.00	Y.13	12.0	.707	67.	2240.	2307.	3.2
6	50.501	200.00	10.10	14.6	.861	51.	2240.	2291.	3.5
7	55.241	200.00	11.05	17.3	1.030	47.			3.8
8	59.982	200.00	12.00	20.6	1.214	48.	2240.	2288.	4.2
9	64.722	200.00	12.99	24.0	1.414	33.	2240.	2293.	4.5
10	69.946	200.00	13.99	28.0	1.651	60.	2240.	2300.	4.9
11	75.169	200.00	13.03	32.4	1.907	72.	2240.	2312.	33
12	80.392	200.00	16.08	37.0	2.181	87.	2240.	2327.	5.7
13	85.615	200.00	17.12	42.0	2.474	107.	2240.	2347.	6.1
14	93.974	200.00	18.79	50.6	2.981	69.	2240.	2309.	6.6
15	102.333	200.00	20.47	60.0	3.534	71.	2240.	2311.	7.2
16	105.688	200.00	21.14	64.0	.236	72.	2240.	2312.	7.4
17	107.291	200.00	21.46	66.0	.351	72.	2240.	2312.	7.5
18	108.893	200.00	21.78	67.9	.468	73.	2240.	2313.	7.6
19	110.496	200.00	22.10	70.0	.586	74.	2240.	2314.	7.7
20	112.098	200.00	22.42	72.0	.707	74.	2240.	2314.	7.9
21	114.344	200.00	22.87	74.9	.878	63.	2240.	2303.	8.0
22	116.590	200.00	23.32	77.9	1.053	55.	2240.	2295.	8.1
23	118.837	200.00	23.77	80.9	1.232	49.	2240.	2289.	8.2
24	121.083	200.00	24.22	84.0	1.414	44.	2240.	2284.	8.4
25	124.168	200.00	24.83	08.3	1.669	46.	2240.	2286.	8.6
26	127.253	200.00	25.45	92.8	1.931	50.	2240.	2290.	8.8
27	130.339	200.00	26.07	97.3	2.199	54.	2240.	2294.	9.1
28	133.424	200.00	26.68	102.0	2.474	60.	2240.	2300.	9.3
29	139.072	200.00	27.81	110.8	2.993	79.	2240.	231Y.	9.8
30	144.720	200.00	28.94	120.0	3.534	81.	2240.	2321.	10.2
a i	147.112	200.00	29.42	124.0	.236	112.	2240.	2322.	10.3
32	148.280	200.00	29.66	126.0	.352	82.	2240.	2322.	10.4
33	149.447	200.00	29.89	128.0	.469	83.	2240.	2323.	10.5
34	150.615	200.00	30.12	130.0	.588	84.	2240.	2324.	10.6
33	151.783	200.00	30.36	132.0	.707	85.	2240.	2325.	10.7
36	153.471	200.00	30.69	135.0	.881	73.	2240.	2313.	10.8
37	155.158	200.00	31.03	137.9	1.056	64.	2240.	2304.	10.8
38	156.846	200.00	31.37	141.0	1.234	57.	2240.	2297.	10.9
39	158.534	200.00	31.71	144.0	1.414	51.	2240.	2291.	11.0
40	160.938	200.00	32.19	148.4	1.673	53.	2240.	2293.	11.2
41	163.341	200.00	32.67	152.9	1.936	56.	2240.	2296.	11.4
42	165.745	200.00	33.15	157.4	2.203	60.	2240.	2300.	11.6
43	168.149	200.00	33.63	162.0	2.474	65.	2240.	2305.	11.7
44	173.600	200.00	34.72	172.7	3.103	89.	2240.	2329.	12.3
45	177.237	195.88	35.44	180.0	3.534	84.	2194.	2277.	12.2

TABLE 6–4. GUN OPERATING POWER (Con't.)

N-CRE-MENT	TIME MILSEC	ROTOR ANGULAR ACCELERATION RAD/SEC/SEC	ROTOR ANGULAR .VELOCITY RAD/SEC	ROTOR ANGULAR TRAVEL DEGREE	BOLT RAL TRAVEL INCH	TOTAL CAM TORQUE F- N	ROTOR TORQUE LB- N	REQUIRED TORQUE B- N	REQUIRED HORSE-POWER
46	179.195	193.66	35.82	184.0	.236	112.	2169.	2281.	12.4
47	180.159	192.56	36.01	186.0	.352	132.	2157.	2289.	12.5
48	181.122	191.47	36.19	188.0	.470	147.	2144.	2291.	12.6
49	182.086	190.38	36.38	190.0	.588	162.	2132.	2294.	12.6
50	383.049	189.29	36.56	192.0	.707	177.	2120.	2297.	12.7
51	184.458	187.69	36.82	195.0	.881	-437.	2102.	1665.	9.3
52	185.867	186.09	37.09	197.9	1.056	-206.	2084.	1878.	10.6
53	187.276	184.49	37.35	200.9	1.233	18.	2066.	2084.	11.8
34	188.685	182.90	37.61	204.0	1.414	247.	2048.	2295.	13.1
55	190.738	180.57	37.98	208.4	1.673	296.	2022.	2319.	13.3
56	192.791	178.24	38.35	212.9	1.930	354.	1996.	2350.	13.7
57	194.843	175.91	38.71	217.4	2.205	419.	1970.	2389.	14.0
38	196.896	173.59	39.07	222.0	2.474	492.	1944.	2437.	14.4
59	200.852	169.10	39.75	230.9	3.000	93.	1894.	1987.	12.0
60	204.808	164.62	40.41	240.0	3.534	93.	1844.	1936.	11.9
61	206.531	162.66	40.69	244.0	.236	120.	1822.	1942.	12.0
62	207.384	161.70	40.83	246.0	.333	141.	1011.	1952.	12.1
63	208.237	160.73	40.97	248.0	.472	155.	1800.	1955.	12.1
64	209.091	159.76	41.10	250.0	.590	170.	1789.	1959.	12.2
65	209.944	158.79	41.24	252.0	.707	186.	1778.	1964.	12.3
66	211.197	157.37	41.44	255.0	.884	-425.	1763.	1337.	8.4
67	212.451	155.95	41.63	258.0	1.060	-196.	1747.	1551.	9.8
68	213.704	154.53	41.83	261.0	1.236	27.	1731.	1757.	11.1
69	214.957	153.11	42.02	264.0	1.414	250.	1715.	1965.	12.5
70	216.801	151.02	42.30	268.5	1.676	300.	1691.	1991.	12.8
71	218.645	148.93	42.58	272.9	1.940	357.	1668.	2025.	13.1
72	220.489	146.84	42.85	277.4	2.206	421.	1645.	2065.	13.4
IS	222.332	144.75	43.12	282.0	2.474	493.	1621.	2114.	13.8
74	225.931	40.67	43.63	290.9	3.000	100.	1575.	1675.	11.1
75	229.530	136.59	44.13	300.0	3.534	100.	1530.	1630.	10.9
76	231.107	134.80	44.35	304.0	.236	126.	1510.	1636.	11.0
77	231.890	133.91	44.45	306.0	.352	147.	1500.	1647.	11.1
78	232.673	133.02	44.56	308.0	.469	161.	1430.	1651.	11.1
79	233.456	132.14	44.66	310.0	.587	176.	1420.	1656.	11.2
80	234.238	131.23	44.76	312.0	.707	192.	1470.	1662.	11.3
81	235.400	129.93	44.91	315.0	.881	-424.	1455.	1031.	7.0
82	236.561	128.62	45.06	318.0	1.057	-194.	14[illegible]0.	1247.	8.5
83	237.722	127.30	45.21	321.0	1.234	29.	1426.	1455.	10.0
84	2318084	125.98	45.36	324.0	1.414	234.	1411.	1665.	11.4
85	240.597	124.04	45.57	328.4	1.675	303.	1389.	1692.	11.7
86	242.311	122.10	43.79	332.9	1.939	33Y.	1367.	1727.	12.9
07	244.025	120.15	45.99	337.4	2.204	423.	1346.	1768.	12.3
88	245.738	118.21	46.20	341.9	2.474	495.	1324.	1819.	12.7
89	249.122	114.37	46.59	350.9	3.001	106.	1281.	1387.	9.8
90	252.505	110.54	46.97	360.0	3.334	106.	1238.	1344.	9.6

TABLE 6–4. GUN OPERATING POWER (Con't.)

INCREMENT	TIME MILSEC	ROTOR ANGULAR ACCELERATION RAD/SEC/SEC	ROTOR ANGULAR VELOCITY RAD/SEC	ROTOR ANGULAR TRAVEL DEGREE	BOLT PERIPHERAL TRAVEL INCH	TOTAL CAM TORQUE LB=IN	ROTOR TORQUE LB=IN	REQUIRED TORQUE LB=IN	REQUIRED HORSE-POWER
91	253.989	108.86	47.13	364.0	.236	132.	1219.	1331.	9.6
92	254.727	108.02	47.21	366.0	.354	153.	1210.	1363.	9.7
93	255.465	107.18	47.29	368.0	.471	167.	1200.	1368.	9.8
94	256.203	106.35	47.37	370.0	.589	182.	1191.	1373.	9.9
95	256.941	105.51	47.45	372.0	.707	198.	1182.	1380.	9.9
96	258.040	104.26	47.57	375.0	.884	-416.	1168.	752.	5.4
97	259.139	103.02	47.68	378.0	1.060	-186.	1154.	968.	7.0
98	260.237	101.77	47.79	381.0	1.237	36.	1140.	1176.	8.5
99	261.336	100.53	47.90	384.0	.414	236.	1126.	1382.	10.0
100	262.965	98.68	48.07	388.5	1.678	307.	1085.	1447.	10.3
101	264.594	96.83	48.22	393.0	1.943	363.			10.6
102	266.222	94.99	48.38	397.5	2.209	426.	1064.	1490.	10.9
103	267.851	93.14	48.53	402.0	2.474	497.	1043.	1540.	11.3
104	271.069	89.49	48.83	411.0	3.004	110.	1002.	1113.	8.2
105	274.288	85.84	49.11	420.0	3.534	110.	46i.	1071.	8.u
106	275.707	84.23	49.23	424.0	.236	136.	943.	1079.	8r1
107	276.415	83.43	49.29	426.0	.355	157.	Y34.	1092.	8.2
108	277.122	82.63	49.35	428.0	.472	172.	925.	1097.	8.2
109	277.830	81.83	49.41	430.0	.590	187.	916.	1103.	b.3
110	278.537	81.02	49.46	432.0	.707	202.	907.	1110.	8.3
111	279.592	79.83	49.55	435.0	.885	-411.	894.	483.	3.6
112	280.648	70.63	49.63	438.0	1.061	-181.	881.	700.	5.3
113	281.	77.43	49.72	441.0	1.238	40.	867.	907.	6.8
114	282.759	76.24	49.00	444.0	1.414	259.	854.	1112.	8.4
115	284.329	74.46	[illegible]	448.5	1.680	30[illegible].	834.	1143.	8.6
116	285.899	72.68	50.03	453.0	1.944	365.	814.	1179.	8.9
117	287.469	70.90	50.14	457.5	2.210	428.	794.	1222.	9.3
118	209.039	69.12	50.25	462.0	2.474	498.	774.	1273.	9.7
119	292.153	65.59	50.46	471.0	3.005	114.	733.	848.	6.5
120	295.266	62.06	50.66	480.0	3.534	113.	695.	808.	6.2
121	296.643	60.49	50.75	484.0	.236	139.	678.	817.	0.3
122	297.330	59.72	50.79	486.0	.355	160.	669.	829.	6.4
123	298.017	58.94	50.83	488.0	.473	1/30	660.	835.	6.4
124	298.704	58.16	50.87	490.0	.591	190.	651.	841.	6.5
[illegible]	[illegible]	57.38	50.91	492.0	.707	205.	643.	848.	6.5
126	300.417	56.22	50.97	495.0	.885	-407.	630.	222.	17
127	301.444	55.05	51.02	498.u	1.062	-178.	617.	439.	3.Y
128	302.470	53.89	51.08	501.0	1.239	43.	604.	647.	5r0
[illegible]	[illegible]	52.72	51.13	504.0	1.414	260.	591.	851.	0.0
130	305.028	50.99	51.21	508.5	1.680	311.	571.	882.	6.8
131	306.559	49.25	51.29	513.0	1.945	367.	552.	918.	7.1
132	308.090	47.52	51.36	517.5	2.211	430.	532.	962.	7.5
133	309.622	45.78	51.44	522.0	2.474	500.	513.	1012.	7.9
134	312.668	42.33	51.57	531.0	3.006	116.	474.	590.	4.6
135	313.715	38.87	51.69	540.0	3.534	115.	435.	31.	4.3

TABLE 6–4. GUN OPERATING POWER (Con't.)

IN-CRE-MENT	TIME MILSEC	ROTOR ANGULAR ACCELERATION RAD/SEC/SEC	ROTOR ANGULAR VELOCITY RAD/SEC	ROTOR ANGULAR TRAVEL DEGREE	BOLT PERIPHERAL TRAVEL INCH	TOTAL CAM TORQUE LB-IN	ROTOR TORQUE LB-IN	REQUIRED TORQUE LB-IN	REQUIRED HORSE-POWER
136	317.064	37.34	51.75	544.0	.236	141.	418.	597.	4.4
137	317.738	36.58	51.77	546.0	.356	162.	410.	572.	4.5
138	310.412	35.81	51.79	548.0	.473	177.	401.	578.	4.5
139	319.086	35.05	51.82	550.0	.591	192.	393.	584.	4.6
140	319.760	34.29	51.84	552.0	.0	207.	384.	591.	4.6
141	320.768	33.14	51.88	555.0	.886	-405.	371.	-34.	-.3
142	321.777	32.00	51.91	558.0	1.062	-176.	358.	182.	1.4
143	322.786	30.85	51.94	561.0	1.239	45.	346.	390.	3.1
144	323.795	29.71	51.97	564.0	1.414	261.	333.	594.	4.7
145	325.304	28.00	52.01	568.5	1.681	312.	314.	625.	4.9
146	326.812	26.29	52.06	573.0	1.946	368.	294.	662.	3.2
147	328.320	24.58	52.09	577.5	2.211	430.	275.	706.	5.6
148	329.029	22.87	52.13	582.0	2.474	500.	256.	756.	6.0
149	332.038	19.46	52.19	591.0	3.006	117.	218.	335.	2.6
150	335.848	16.05	52.25	600.0	3.534	116.	180.	296.	2.3
151	337.183	14.53	52.27	604.0	.236	142.	163.	304.	2.4
152	337.850	13.77	52.28	606.0	.353	163.	154.	317.	2.5
153	338.518	13.02	52.29	608.0	.473	177.	146.	323.	2.6
154	339.185	12.26	52.29	610.0	.591	192.	137.	330.	2.6
155	339.852	11.50	52.30	612.0	.707	208.	129.	336.	2.7
156	340.853	10.37	52.31	615.0	.885	-406.	116.	-289.	-2.3
167	341.854	9.24	52.32	618.0	1.062	-176.	103.	-73.	-.6
158	342.855	8.10	52.33	621.0	1.239	44.	91.	135.	1.1
159	343.855	6.97	52.34	624.0	1.414	261.	78.	339.	2.7
160	345.355	5.27	52.35	628.5	1.680	312.	59.	371.	2.9
161	346.854	3.57	52.35	633.0	1.945	367.	40.	407.	3.2
162	348.354	1.87	52.36	637.5	2.210	430.	21.	451.	3.6
163	350.000	-.00	52.36	642.5	2.474	500.	-0.	500.	4.0
164	333.000	.0	52.36	651.5	3.031	64.	0.	64.	.5
165	356.001	.00	52.36	660.5	3.534	64.	0.	64.	.5
166	331.334	.00	52.36	664.5	.236	96.	0.	96.	.8
167	358.000	.00	52.36	666.5	.381	132.	0.	132.	1.0
168	358.667	.00	32.36	668.5	.498	132.	0.	152.	1.2
169	359.333	.00	52.36	670.5	.616	174.	0.	174.	1.4
170	360.000	.00	52.36	672.5	.707	191.	0.	191.	1.5
171	361.000	.00	52.36	675.5	.911	-626.	0.	-626.	-5.0
172	362.001	.00	52.36	678.5	1.087	-283.	0.	-283.	-2.2
173	363.001	.00	52.36	681.5	1.264	43.	0.	43.	.3
174	364,001	.00	52.36	684.5	1.414	314.	0.	314.	2.5
175	365.501	.00	52.36	689.0	1.706	384.	0.	384.	3.0
176	367.001	.00	52.36	693.5	1.971	457.	0.	457.	3.6
177	360.500	.00	52.36	698.0	2.236	537.	0.	537.	4.3
178	370.000	.0	52.36	702.5	2.474	613.	0.	613.	4.9
179	373.000	.00	52.36	711.5	3.031	64.	0.	64.	.5
180	376.001	.00	52.36	720.5	3.534	64.	0.	64.	.5

CHAPTER 7

COMPONENT DESIGN

7–1 GENERAL

Automatic weapons are equipped with practically the same components that other weapons need to insure effective and safe (to the operator) performance. Differences lie only in application since the components in the automatic weapon must be geared to automatic performance. These components include feed mechanisms, breech locking systems, sears, firing mechanisms, extractors, ejectors, and cocking mechanisms. Characteristics of other components such as muzzle devices which include silencers are presented in detail in other design handbooks [25] or published reports [26]. Each component generally has features unique to automatic weapons.

7–2 FEED MECHANISM DESIGN

Automatic weapons are fed ammunition from magazines, clips, and belts; the type and capacity depending upon type of weapon. The bolt, moving in counterrecoil, strips the round from the feed mechanism and carries it into the chamber. The withdrawn round is instantly replaced by the next round of the supply.

The first step in designing a feed mechanism is defining the feed path. The feed path is the course of the round from mechanism to chamber. Two requisites take precedence: (1) to have the initial position of the projectile move as close to the chamber as the system permits, and (2) to have the base of the cartridge case as close in line to the center line of the bore as possible at the time of feed. The ideal would have the center lines of round and bore collinear. The ideal is not always possible; therefore, other arrangements must suffice but care must be exercised to avoid impact between bolt face and primer since bolt contacts cartridge during counterrecoil. The primer is the restricting element. The two views of Fig. 7–1 illustrate this characteristic. Unless surface contact is assured at impact, the outer edge of the bolt face must never extend into the primer surface, otherwise the edge may strike the primer with enough resulting penetration to set it off. To preclude premature discharge, a minimum space of 0.010 in. between the edges of the primer and bolt face is necessary. Because of override, impact cannot be eliminated; thereby, obviating this approach as a solution for premature firing. ***Override*** is the clearance between bolt face and cartridge case base needed to position the round before the bolt moves forward. Interference here cannot be tolerated, otherwise malfunction is inevitable.

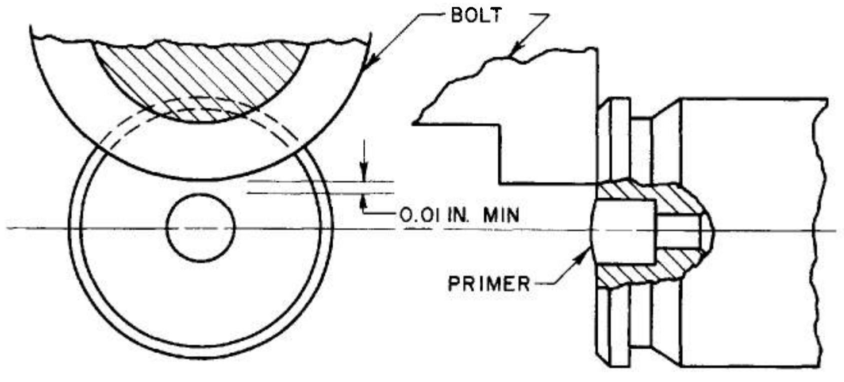


Figure 7–1. Initial Contact of Bolt and Cartridge Case Base

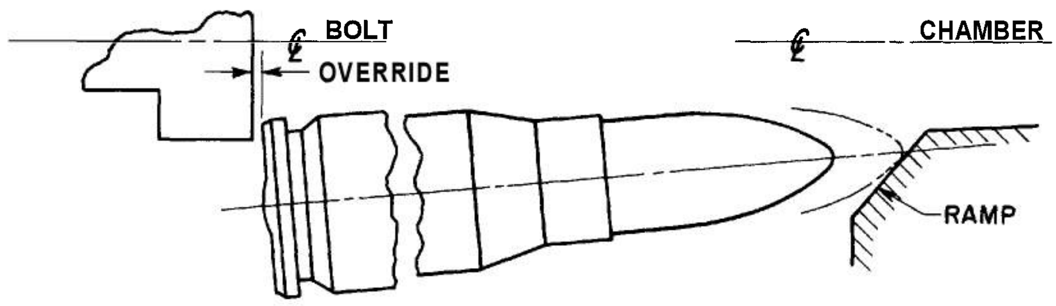


Figure 7–2. Chamber-projectile Contact

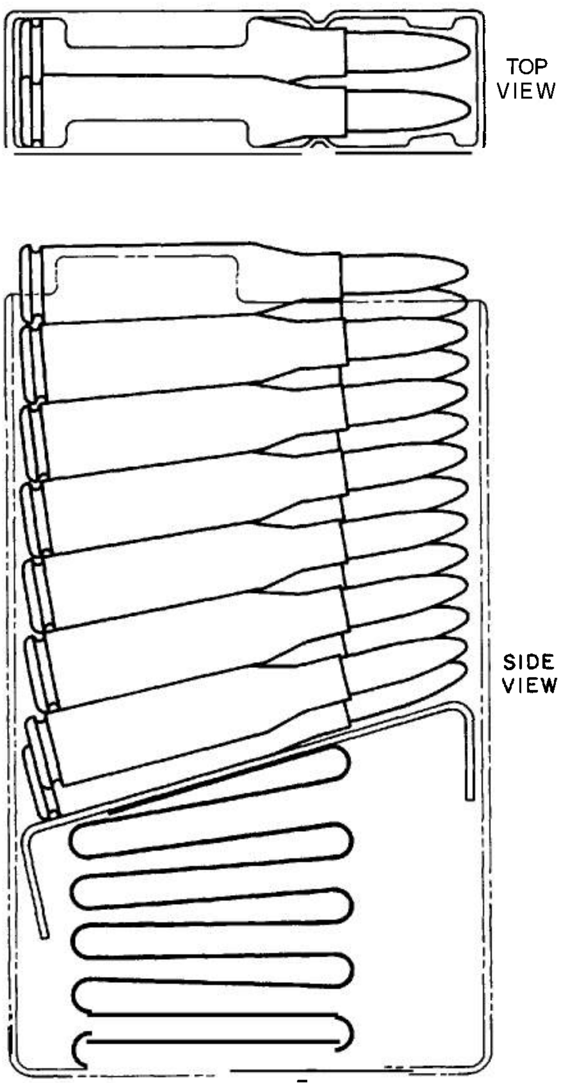


Figure 7–3. Box Magazine

The next design operation is to provide a path for the round between the immediate receptacle and chamber, and guidance along this path. The receptacle–whether magazine, clip, or belt–provides the initial guidance which will be discussed later. The chamber provides the terminal guidance. The entrance to the chamber and the path of the round should be so arranged that any contact between chamber and projectile will take place on the ogive. Fig. 7–2 shows this arrangement. The chamber entrance may be enlarged by a ramp to eliminate the probability of the nose striking the chamber walls first.

7–2.1 MAGAZINES

Magazines, box or drum, are of limited capacity. Box magazines generally hold from 7 to 20 rounds in single or double rows; drums, up to 150 rounds.

7–2.1.1 Box Magazines

A box magazine may be attached to the receiver or it may be an integral part of it. Both types have a spring to keep forcing the rounds toward the bolt as firing continues. The box not only stores the rounds 'but also restrains their outward motion at the mouth and guides each round as the bolt strips it from the box. The restraining and guiding elements, called **lips,** are integral with the sides. Fig. **7–3** shows a box magazine with several rounds of ammunition.

Correct lip length is vital to dependable loading. Combined with the direction of the spring force, the lips control the position of the round as it enters the chamber. **As** indicated in Fig. **7–3,** continuous control is exercised by the lips while they restrain the round and so long as the resultant spring force passes within their confines. **If** the resultant spring force falls forward of the lips, the round **will** have a tendency to tip excessively

and increase the probability of jamming. Fig. 7–4 demonstrates how a short lip may fail to guide a round so that it enters the chamber without interference. Fig. 7–4 demonstrates how a longer lip will retain contact with the round long enough for the ogive to hit the ramp just prior to entering the chamber.

The shape of the lip has considerable influence on feeding. The round to be loaded should be restrained by line contact between the cartridge case and lip. Fig. 7–5 shows how this effect can be arranged by making the inner radius of the lip less than the radius of the cartridge case. Absolute assurance of line contact is assured by forming the lip by a right angle bend. The spring load holds the round firmly until the bolt dislodges it. On the other hand, if the lip radius is larger than the cartridge case radius, accurate positioning of the rounds cannot be achieved with any degree of assurance. The cartridge case position, from round to round, may virtually float; thereby, causing an inconsistency in contact area between the bolt face and the rounds. Fig. 7–5 shows how the positions may vary with respect to the fixed bolt position. The larger the radius, the less assurance of sufficient contact area between the bolt face and cartridge case base. In extreme cases the bolt may hit the primer first and initiate it.

The dimensions of the cartridge and the intended capacity determine the size of the magazine. For a single row of cartridges, the width equals the diameter of the base plus 0.005 in.

$$w = D_c + 0.005 \tag{7–1}$$

where D_c = diameter of cartridge case base

w = inside width of magazine

Double rows of cartridges are stacked so that the centers form an equilateral triangle as shown in Fig. 7–6 where the inside width of the magazine is

$$w = 1.866\,D_c + 0.005. \tag{7–2}$$

The nominal depth of the magazine storage space with double rows is

$$h = \frac{1}{2} D_c(N + 1) \tag{7–3}$$

where h = depth

N = number of rounds

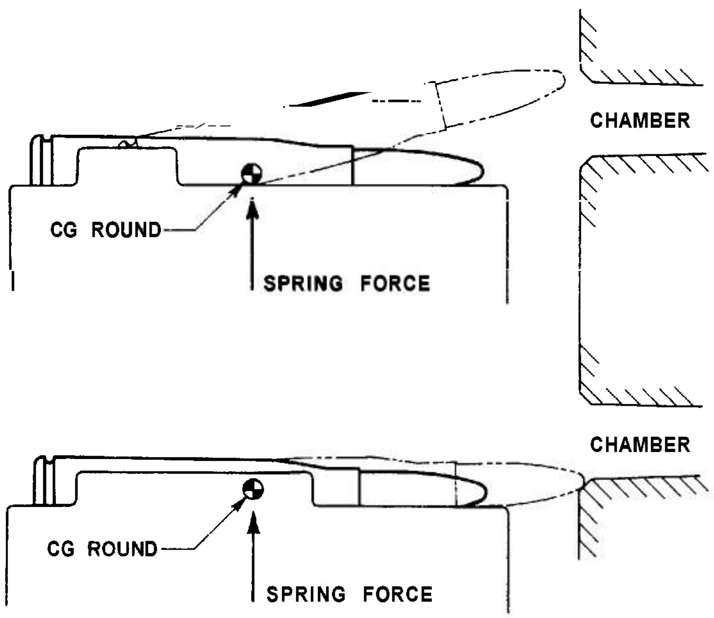
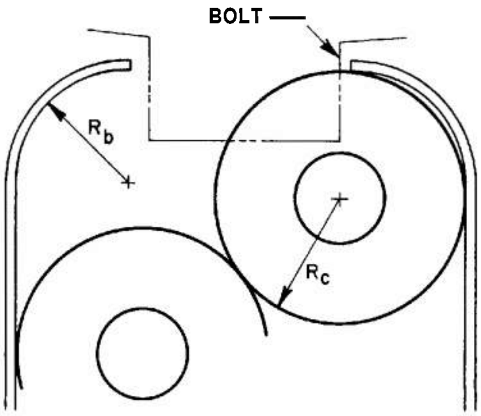
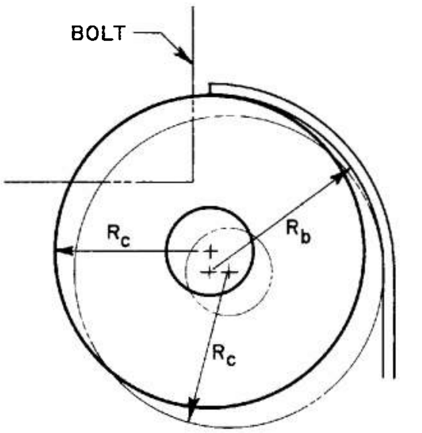


Figure 7–4. Lip Guides



(A) PROPER ARRANGEMENT, $R_b < R_c$



(B) POOR ARRANGEMENT, $R_b > R_c$

Figure 7–5. Lip-cartridge Case Orientation

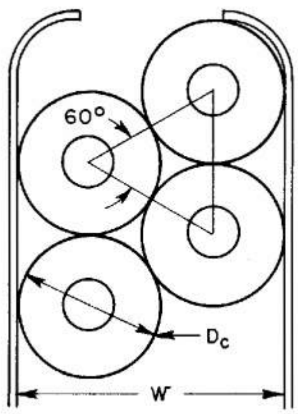


Figure 7–6. Geometry of Double Row Stacking

7–2.1.2 Box Feed System

The box feed system has three major components, the box which has been discussed, the follower, and the spring. The follower separates the column of cartridges from the spring, transmits the spring force to the cartridges, and provides the sliding surface for the last (single row) or last two (double row) cartridges. The follower also holds the stored rounds in alignment. It should never restrict spring activity. Fig. 7–7 shows three views of a follower. The spring may be a round wire spring shaped into rectangular coils or it may be a flat steel tape folded over at regular intervals to approximate the side view of a helix.

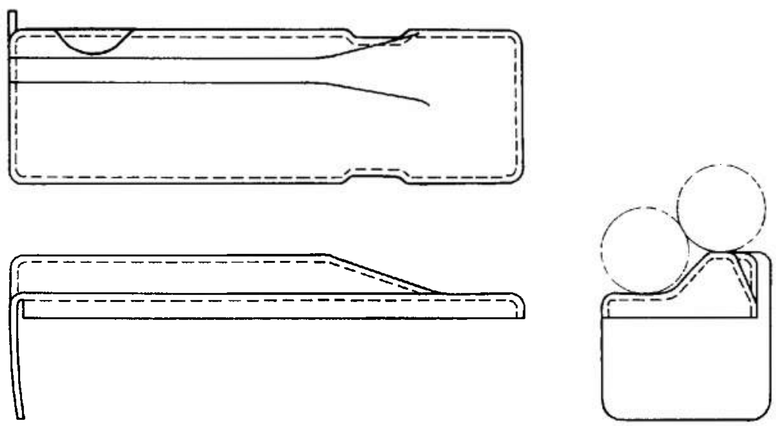


Figure 7–7. Box ***Magazine Follower***

7–2.1.2.1 Flat Tape Spring

The flat steel spring functions in bending rather than in torsion. Each segment behaves as a cantilever beam that has the loaded end restrained from rotating. Fig. 7–8 shows this analogy and the loading diagram. Beginning at the follower, the bending moment M_o at the bend, when the applied load is assumed to be concentrated at the middle of the follower is

$$M_o = -\frac{1}{2}(FL) \tag{7-4}$$

where F = spring force

L = length of each spring segment

The bending moment at the end of the first free segment

$$M = M_o + FL = \frac{1}{2}(FL). \tag{7-5}$$

This moment is identical and, therefore, constant for all segments of the spring. The deflection of one end of each segment with respect to the opposite one is

$$\Delta y = \frac{M_o L^2}{2EI} + \frac{FL^3}{3EI} = \frac{FL^3}{12EI} \tag{7-6}$$

where E = modulus of elasticity

I = area moment of inertia of the spring cross section

The total deflection of a spring having N active segments is

$$y = \Sigma \Delta y = N \Delta y = \frac{NFL^3}{12EI} \tag{7-7}$$

Solve for the spring constant.

$$K = \frac{F}{Y} = \frac{12EI}{NL^3} \tag{7-8}$$

Not only must the spring exert enough force to hold the ammunition in position but it must *also* provide the acceleration to advance the ammunition and the other moving parts over the distance of one cartridge space in time for the bolt to feed the next round. The equivalent mass of all moving parts in the ammunition box is

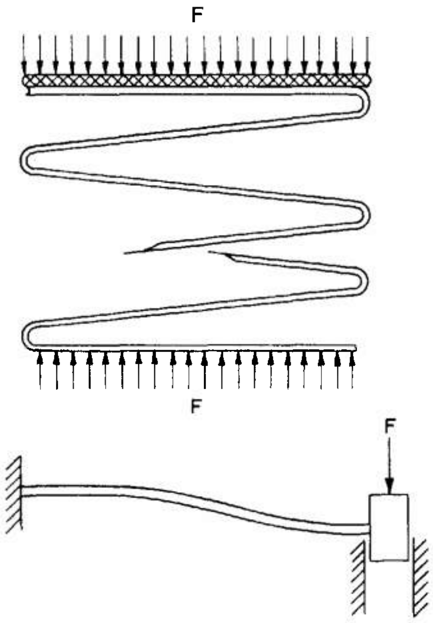


Figure 7–8. Flat Tape Spring and Loading Analogy

$$M_e = \left[(N-1)W_a + W_f + W_{se}\right]/g \tag{7-9}$$

where g = acceleration of gravity

N = number of rounds in the box

W_f = weight of follower

W_a = weight of each round

W_s = weight of spring

$W_{se} = \frac{1}{3} W_s$, equivalent weight of spring in motion

The time required for any one particular displacement will be similar to that of **Eq.** *2–27*

$$t = \sqrt{\frac{M_e}{\epsilon K}} \cos^{-1} \frac{F_o}{F_m} \tag{7-10}$$

where F_m = maximum spring force (preceding one cartridge displacement)

F_o = minimum spring force (following one cartridge displacement)

K = (Eq. 7–5)

M_e = (Eq. 7–6)

E = efficiency of system, generally assumed to be **0.5** for initial design analysis

For initial estimates, provide a spring load of F_i pounds for an empty box and one of F_f for a full box.

The folded flat spring is less desirable than the rectangular coil spring because the latter can be compressed to its solid height whereas total compression of the flat spring is limited by the radius of the folds, thereby, requiring a longer box to house the spring and store the ammunition. Par. 7–2.1.2.2 discusses the rectangular coil spring.

7–2.1.2.2 Rectangular Coil Spring

The rectangular coil spring is a torsion element. Fig. **7–9** illustrates the mechanics of operation. Torsion in each straight segment rotates the adjacent segment. Although bending occurs along the span of each segment, the corners move with respect to each other only by torsional deflection. Bending deflections at the corners are neutralized by equal and opposite bending moments.

Rectangular coil spring characteristics are computed according to procedures similar to helical springs. The applied load is assumed to be concentrated on the axis. The torque T_1 on the long segment is

$$T_1 = \frac{1}{2}\left(aF\right) \qquad \textbf{(7–11)}$$

and torque T_2 on the short segment is

$$T_2 = \frac{1}{2}\left(bF\right) \qquad \textbf{(7–12)}$$

where a = length of short segment

b = length of long segment

F = spring force **(7–13)**

The corresponding angular deflections are

$$\theta_1 = \frac{bT_1}{JG} = \frac{abF}{2JG} \qquad \textbf{(7–14)}$$

$$\theta_2 = \frac{aT_2}{JG} = \frac{abF}{2JG} \qquad \textbf{(7–15)}$$

where G = torsional modulus

J = area polar moment of inertia of wire

The axial deflection of each segment of a coil varies directly with the sum of the products of the two segment lengths times the sine of the angular deflection of the adjacent segment (see **Fig. 7–9).** Stated in algebraic expressions the two deflections are

$$\Delta y_1 = b \sin\theta_1 \qquad \textbf{(7–16)}$$

$$\Delta y_2 = a \sin\theta_2 \qquad \textbf{(7–17)}$$

But, according to **Eqs. 7–14** and **7–15,** $\theta_1 = \theta_2$, and if we let this angle be equal to 8, the deflection of two adjacent segments of a coil is

$$\Delta y = (\Delta y_1 + \Delta y_2) = (a+b) \sin 8 \qquad \textbf{(7–18)}$$

Since there are **4** segments to each coil, the total deflection of a spring having N active coils is

$$y = 2N\Delta y. \qquad \textbf{(7–19)}$$

The spring constant, if y is based on a free spring, is

$$K = \frac{F}{y} \qquad \textbf{(7–20)}$$

The time required for any given displacement can be computed from **Eq. 7–10.**

7–2.1.3 Example Problems

Compute the spring characteristics for a double row box feed system that holds **20** rounds. Each round weighs **420** grains and has a cartridge case base diameter of **0.48** in. **To** function properly in the box, the spring should fit in a projected area of **1.75 x 0.75** in. The initial spring load should be approximately **4** pounds.

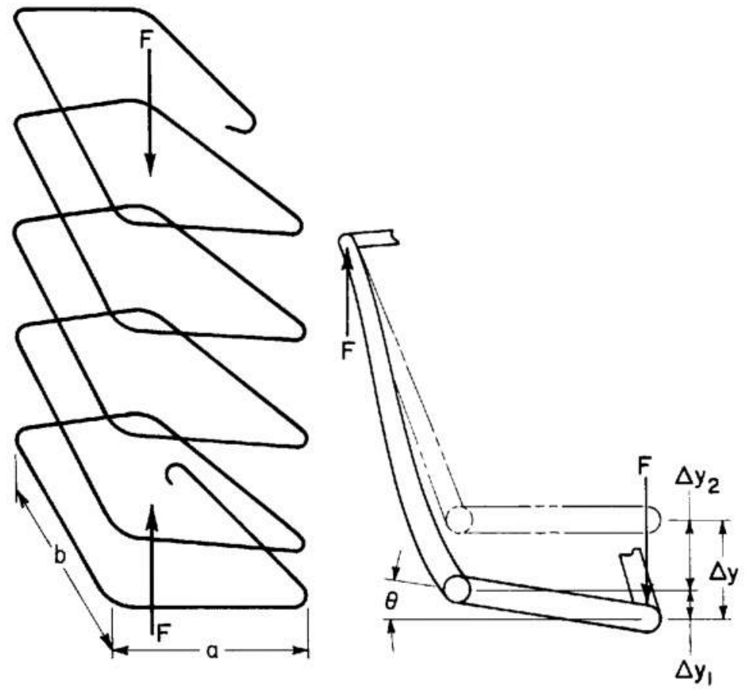


Figure 7–9. Rectangular Coil Spring and Loading Characteristics

7–2.1.3.1 Flat Tape Spring

Set the following initial parameters:

F_i = 4.0 lb, initial spring load

[redacted] 1.75 in., length of each spring segment

N = 14, number of active segments, arbitrary choice but based on previous designs

w = 0.75 in., width of spring

σ_w = 200,000 lb/in.^{2}, working stress of spring

The spring deflection, Eq. 7–3, inside the box caused by the cartridge displacement is

$$y_c = \frac{1}{2} D_c(N + 1) = \frac{0.48}{2} (20 + 1) = 5.04 \text{ in.}$$

where N = 20 rounds.

Assume, as a first estimate, that the deflection on assembly approximates the total cartridge displacement.

y_i = 5.0 in., the initial deflection

According to Eq. 7–8, $K = \frac{F_i}{y_i} = \frac{4.0}{5.0} = 0.8$ lb/in.

Now solving for I in the same equation

$$I = \frac{KNL^3}{12E} = \frac{0.8 \times 14 \times 1.75^3}{12 \times 30 \times 10^6} = \frac{1}{6} \times 10^{-6}$$

Since $I = \frac{1}{12} wt_s^3$, $t_s^3 = \frac{12I}{w} = \frac{8}{3} \times 10^{-6}$.

Therefore t_s = 0.014 in. the required spring thickness. The bending moment, Eq. 7–5, is

$$M = \frac{1}{2}\left(FL\right) = \frac{1}{2} \times 8 \times 1.75 = 7 \text{ lb-in.}$$

where $F = Ky = 0.8 \times 10 = 8$ lb.

The bending stress is

$$\sigma = \frac{Mc}{I} = \frac{7 \times 0.007}{0.1667 \times 10^{-6}} = 294{,}000 \text{ lb/in.}^2$$

wherec $= \frac{t_s}{2}$ in.

This stress is too high. To lower it to acceptable levels, the initial and final loads were reduced to 1.0 and 2.0 pounds, respectively. Subsequent computation produced the following data:

$$K = 0.2 \text{ lb/in.}$$

$$t_s = 0.00874 \text{ in.}$$

$$M = 1.75 \text{ lb-in.}$$

$$\sigma = 183{,}000 \text{lb/in.}^2$$

The bending stress is still uncomfortably high which almost rules out this type spring for the above application. However, a time analysis will give additional data. The time will be computed for spring action after the first and next to the last round are removed. If the spring weighs **0.063** lb and the follower 0.044 lb, the equivalent moving mass for 19 rounds, according to Eq. 7–9, is

$$M_e = \left(19 \times 0.06 + 0.044 + \frac{0.063}{3}\right)/386.4$$

$$= 0.003\,12 \text{lb-sec}^2\text{/in.}$$

Substitute the appropriate values in Eq. 7–10 to compute the time for the'first round

$$t = \sqrt{\frac{M_e}{\epsilon K}} \operatorname{Cos}^{-1} \frac{F}{F_i} = \sqrt{\frac{0.00312}{0.5 \times 0.2}} \cos^{-1} \frac{1.952}{2.0}$$

$$= \sqrt{0.0312} \operatorname{Cos}^{-1} 0.976 = 0.1765 \times 0.22$$

$$= 0.039 \text{ sec}$$

where $\epsilon = 0.5$, the efficiency of the system.

For the last round

$$M_e = \left(0.06 + 0.044 + \frac{0.063}{3}\right)/386.4$$

$$= 0.000323 \text{ lb-sec}^2\text{/in.}$$

$$t = \sqrt{\frac{0.000323}{0.5 \times 0.2}} \operatorname{Cos}^{-1} \frac{F}{F_i} = \sqrt{0.00323} \operatorname{Cos}^{-1} \frac{1.0}{1.048}$$

$$= 0.057 \times 0.301 = 0.0172 \text{ sec}$$

The slower of the two is equivalent to 1500 rounds/min which is more than adequate.

7–2.1.3.2 Rectangular Coil Spring

Set the following initial parameters:

a = 0.75 in., length of short segment

b = 1.75 in., length of long segment

F_i = 4.0 lb, initial spring load

N = 7, number of coils, arbitrary choice but based on previous designs

y_c = 5.04 in., cartridge displacement (see par. 7–2.1.3.1)

y_i = 5.0 in., assembled deflection (see par. 7–2.1.3.1)

$$K = \frac{F_i}{y_i} = \frac{4.0}{5.0} = 0.8 \text{ lb/in.}$$

The total deflection for a full box of cartridges is

$$y = y_c + y_i = 10.04 \text{ in.}$$

The deflection for two adjacent segments of a coil from Eq. 7–19 is

$$\Delta y = \frac{y}{2N} = \frac{10.04}{14} = 0.717 \text{ in.}$$

The angular displacement according to Eq. 7–18 is

$$\sin\theta = \frac{\Delta y}{a+b} = \frac{0.717}{2.5} = 0.2868$$

$$\theta = 16°40' = 0.291 \text{ rad.}$$

Solve for J in **Eq.** 7–15.

$$J = \frac{abF}{2G\theta} = \frac{0.75 \times 1.75 \times 8.032}{2 \times 12 \times 10^6 \times 0.291} = 1.509 \times 10^{-6} \text{ in.}^4$$

where $F = Ky = 0.8 \times 10.04 = 8.032$ lb, maximum spring load

$G = 12 \times 10^6$ lb/in.^{2}, torsional modulus of steel

Since $J = \left(\frac{\pi}{32}\right) d^4 = 1.509 \times 10^{-6}$

$d^4 = 15.37 \times 10^{-6}$

$d = 0.0626$ in., say, 0.0625 in.

Then $J = 1.5 \times 10^{-6}$ in.^{4} and the maximum spring force F_m is

$$F_m = \frac{2JG\theta}{ab} = \frac{36 \times 0.291}{1.3125} = 8.0 \text{ lb.}$$

The maximum torque, **Eq.** 7–12, is

$$T_2 = \frac{1}{2} bF_m = \frac{1}{2} (1.75)\ 8.0 = 7.0 \text{ lb-in.}$$

The torsional shear stress is

$$\tau = \frac{T_2 c}{J} = \frac{7.0 \times 0.03125}{1.5 \times 10^{-6}} = 146{,}000 \text{ lb/in.}^2$$

where $c = \frac{d}{2} = 0.03125$ in.

This stress is acceptable.

If the spring weighs 0.036 lb, and the follower 0.044 lb, the moving mass for 20 rounds, according to **Eq.** 7–9 is

$$M_e = \left(19 \times 0.06 + 0.044 + \frac{0.036}{3}\right) \Big/ 386.4$$

$$= 0.0031 \text{ lb-sec}^2\text{/in.}$$

For 19 cartridges,
$y_c = 4.8$ in. and $F_o = (5.0 + 4.8)\ 0.8 = 7.84$ lb.

The time to move this mass through the space left by the departed projectile is computed by **Eq.** 7–10.

$$t = \sqrt{\frac{M_e}{\epsilon K}} \text{Cos}^{-1} \frac{F_o}{F_m} = \sqrt{\frac{0.0031}{0.5 \times 0.8}} \text{Cos}^{-1} \frac{7.84}{8.0}$$

$$= 0.088 \times 0.201 = 0.018 \text{ sec}$$

where $\epsilon = 0.5$, the efficiency of the system.

The time of 18 msec is far less than needed to operate under any existing conditions.

7–2.2 BOLT-OPERATED FEED SYSTEM

The bolt-operated feed system illustrated in Figs. 7–10 and 7–11 represents one of many similar types. The operating features are described by partially isolating each function and then later showing the coordination that exists in the whole system. Fig. 7–10 shows the ammunition belt system including the components directly associated with it. Sketch (A) shows the position of all parts just as the chambered round has been fired. Sketch (B) shows all parts in the same position except that Round 1 and the empty case are partially extracted, and the feed slide has moved to the left with the feed pawl riding on Round 2. Note that if Round 1 had not been extracted from the belt, the pawl arm would ride over this round to lift the feed pawl above Round 2 to preclude engagement between pawl and Round 2. This operation prevents double feeding or jamming. With Round 1 extracted, the feed pawl carried by pawl arm and slide, continues across Round 2 and eventually engages it as shown in Sketch (C). In the meantime the holding pawl prevents the belt from moving backward.

After the slide completes its travel to the left, the extractor pushes Round 1 downward to align it with the chamber and eject the empty case. After this effort, the slide begins its return to the right and since the feed pawl has engaged Round 2, the slide forces the belt to move also. Two positions of the return are shown in Sketches (C) and (D). Round 3 forces the holding pawl downward to permit belt travel. As soon as Round 2 reaches the original position of Round 1 and all other rounds have simultaneously moved up one position, all feed belt activity will stop with all components taking the positions according to Sketch (A).

The feed slide is activated by the feed lever which in turn is activated by the bolt. The lever fulcrum is fitted to the cover of the receiver, one end activates the slide while the other end rides in a cam groove in the bolt's top surface. Each end of the cam is straight and parallel to the longitudinal axis of the bolt in order to permit a short dwell period for the slide at the end of each half cycle. Shifting the emphasis between the upper and lower illustrations of Fig. 7–11 provides the opportunity of outlining the whole loading and firing cycle. Assume that the bolt is in battery and firing is imminent. The upper picture shows, in phantom, Round 1 of Fig. 7–10 **(A)** ready to be stripped. The extractor lip is in the extractor groove of the cartridge case. At this same time, the

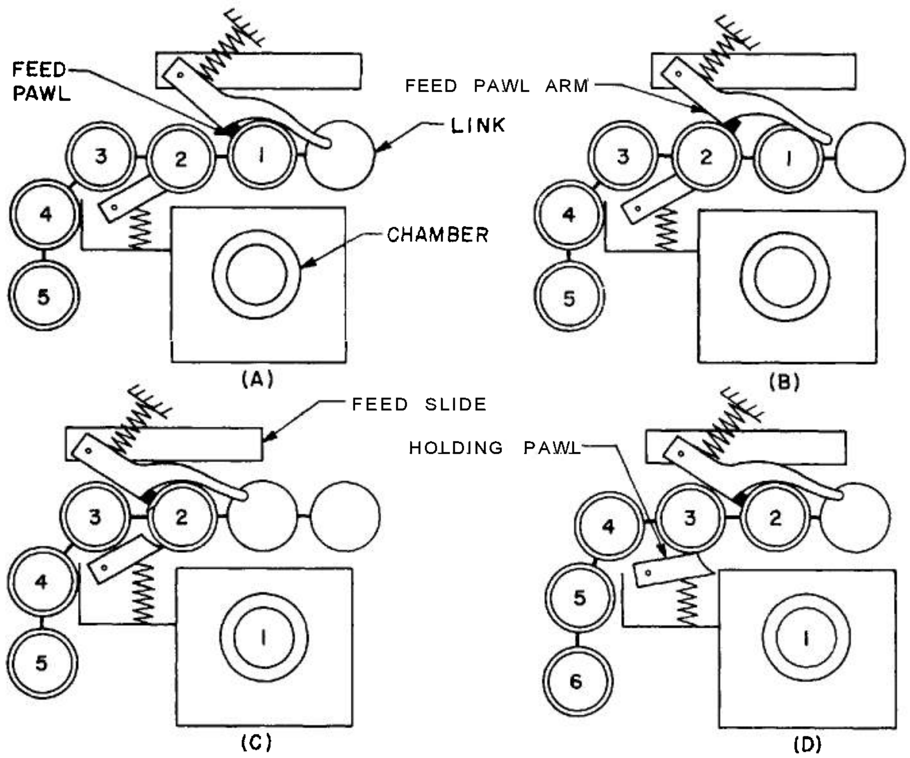


Figure 7–10. Schematic of Feed System, End View

lower picture, in phantom, shows the position of the feed slide and feed lever. None of the components is moving in this stage.

After firing the bolt has recoiled to the position shown in full view in the upper picture. As the bolt travels rearward, the extractor and the T-slot in the face of the bolt strip the round from the link at 1 and extract the empty casy from the chambers at 2. During this time, the bolt feed cam pivots the feed lever counterclockwise to move the feed slide outward, and a cam depresses the extractor to fit the cartridge case base into the T-slot. Also, the unattached link falls free of the belt just as the round is stripped. All of this action has been completed by the time that the rearmost bolt position has been reached. The cutaway of the slide shows the feed pawl in contact with the round it is about to push into the vacated position above the chamber.

The full view of the lower picture shows the bolt shortly after it began to counterrecoil. The cam has continued the downward movement of the extractor to align the live round with the chamber and eject the spent case. The cam on the bolt is now causing the feed lever to pivot clockwise and push the first round into position where the extractor, as the bolt reaches the inbattery position, will be lowered into the extractor groove to complete this cycle.

7–2.3 ROTATING FEED MECHANISM

The rotating feed mechanism operates on the chain-sprocket principle where the chain is represented by the belt of ammunition; the rounds being the rollers. The power that turns the sprockets, or their equivalent, may be derived from recoil or propellant gas operating mechanisms, or from electric motors.

7–2.3.1 Recoil-operated Feed Mechanism

In a recoil-operated mechanism such as shown schematically in Fig. 7–12, the recoil energy is transformed to rotational effort before it reaches the starwheel. Two starwheels are generally used, one engages the cartridge case back of the belt link while

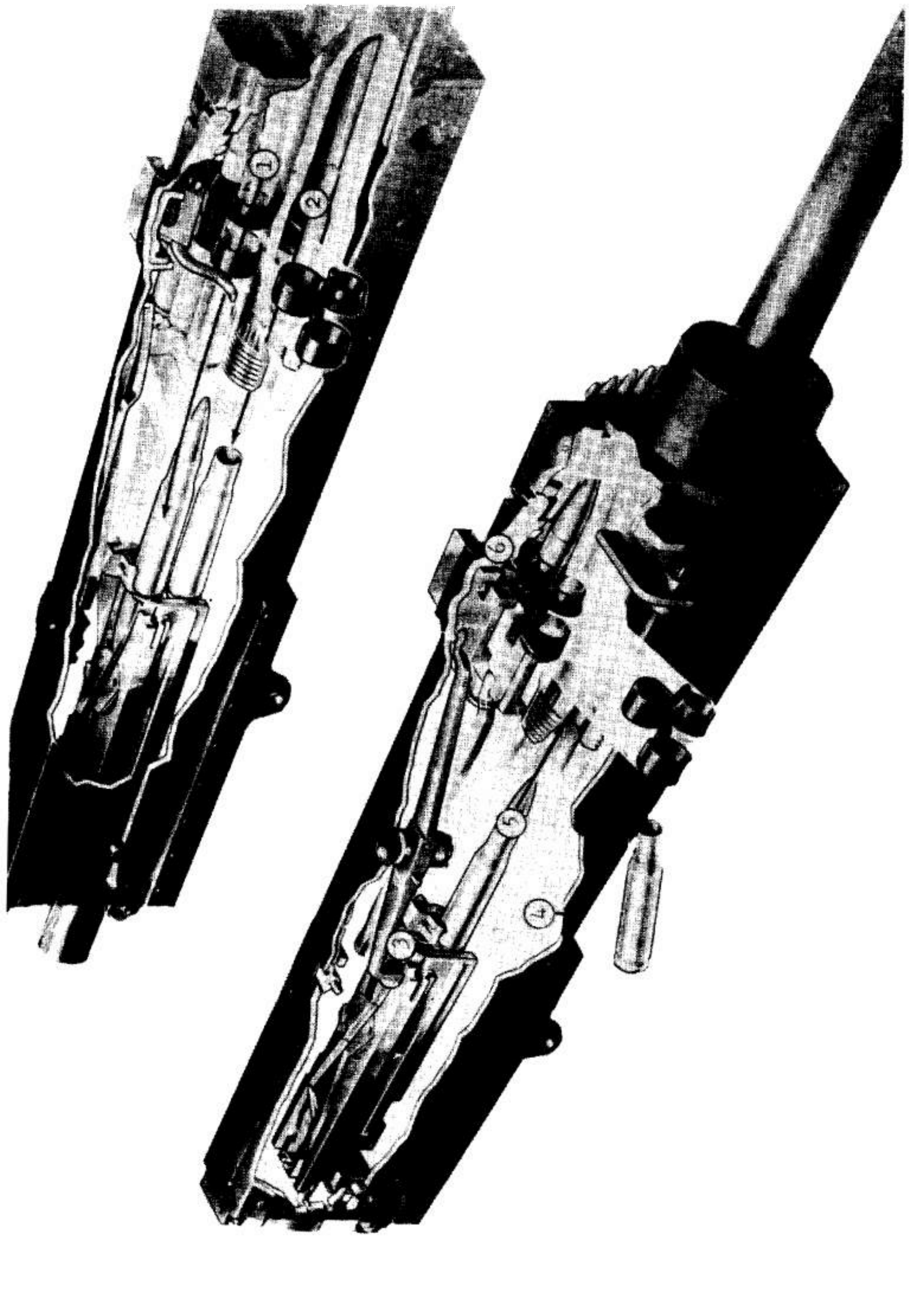


Figure 7–11. Feed System Illustrating Mechanics of Operation

the other engages the projectile just ahead of the rotating band. As the belt and ammunition move with the starwheels, stripper cams wedge between the cartridge case and the clamps of the belt links and pry the links off the case. The freed single end of the link, with its double end still attached to the next round, is guided by the link deflector into the link chute. Freeing the double end releases the link completely from the belt. The detached link falls through the link chute for retrieval or discard. Meanwhile, the link-free round, guided by outer cover and starwheels, continues on its circular path. As it approaches the feed mouth, the round begins to fall away from its cradled position in the starwheels and into the lower contour of the cartridge guides (Fig. 7–12 (B)). The guides complete the path to the feed mouth entrance. Before reaching the mouth, the round contacts the spring-loaded, cartridge holding cams. Forced by the lag tooth of each starwheel, the round pushes the holding cams aside and enters the mouth. As the lag teeth ride over the round, gaps between round and cam surfaces occur to permit the cams to swing back to establish contact between cam surface and round (Fig. 7–12 (B)). The spring loads on the cams force the round downward and simultaneously prevent it from reversing its direction.

The round continues downward until it alights on top of the bolt which is locked in the firing position. It remains in this position until the chambered round is fired and the bolt recoils. The round now moves to the bottom of the mouth where it is retained by a constriction in the mouth. This constriction, or way, is sloped forward at an angle of about three degrees. The round is held in this position by the vertical component of the starwheel force transmitted by the round following. While counterrecoiling, the bolt contacts the lower portion of the cartridge case base and drives the round toward the chamber; the three-degree slope prescribes the desired projectile feed path. As the round clears the ways it is forced downward to become correctly aligned with the bolt. Its former space is now occupied with the next round.

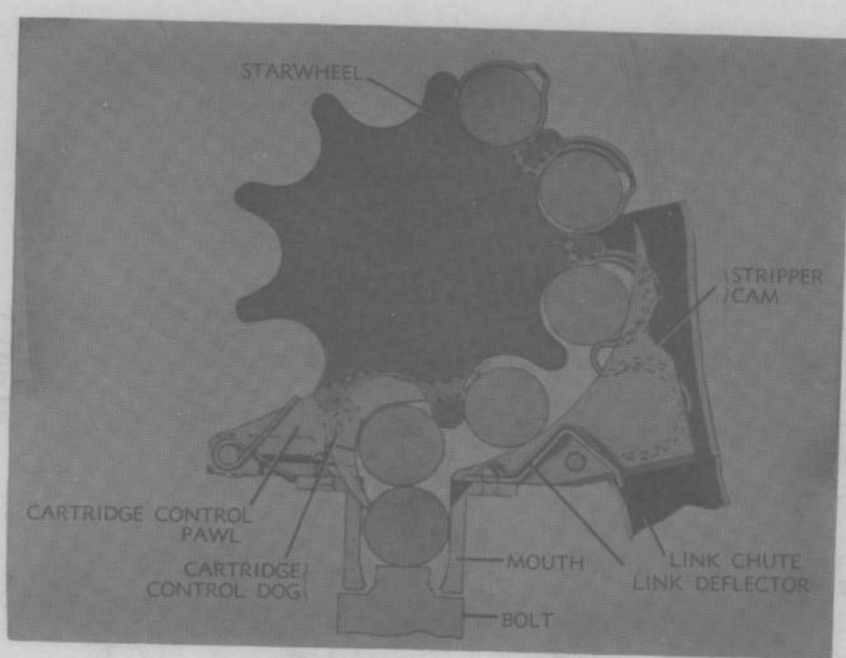
Just prior to entering the feed mouth, the round contacts the lower edge of the spring-loaded cartridge control pawl (Fig. 7–12 (C)). Continued round travel raises the pawl which in turn lifts the holding dog. This action removes the 'obstruction that the normal position of the dog provides and gives free access at the feed mouth entrance to the preceding round. This process is continuous for the entire length of the ammunition belt except for the last round. Because no round follows to lift the pawl, the dog remains undisturbed and holds the last round at the mouth entrance, but at a position low enough to clear the starwheel. If the last round should be able to drop to the top of the bolt as the next-to-last round is fired, all positivi: control over it is lost, thereby increasing the probability of jamming. But, since it is held, the counterrecoiling bolt merely closes on an empty chamber. Action resumes when the first round of a new belt reaches the control pawl.

7–2.3.2 Electrically Driven Feed Mechanism

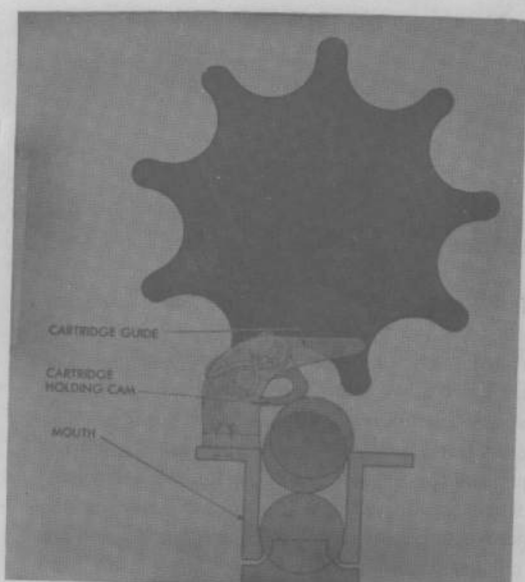
The round and link control of an electrically driven feed mechanism consists of two operating units, the feed wheel unit and the operating lever unit. The feed wheel unit contains two feed wheels, two loading levers, and a bank of three link strippers. The operating lever unit contains two operating levers, two loading guides, and one round retaining finger. The related components between these two units are shown schematically in Fig. 7–13 where the round is used as a common reference. Loading lever and retaining finger are spring loaded; the spring force to the loading lever is transmitted by the operating lever. The retaining finger has its own spring.

The electric motor turns the feed wheel shaft through a gear and clutch system. The two feed wheels draw the belt of ammunition into the feeder at the stripper location (Fig. 7–14 (A)). The link stripper rotates with the feed wheel shaft and its three segments contact the crimp between the leading double and lagging single end of the link. The prying action of the link stripper force on the **link** crimp and the stripper cover reaction snaps the double end of the link off the lead round (Fig. 7–14 (B)). But the single end of the link is still attached to the lag round. However, continued action of the stripper on, the crimp guides the link into the link chute while the freed round continues on its circular path, guided by the feed wheel and link chute support (Fig. 7–14 (C)). As the ammunition belt continues to advance, the prying action of the chute on the double end, combined with the restriction imposed on the lag round by feed wheel and chute support, releases the single end from the round, permitting the now freed link to fall through the link chute.

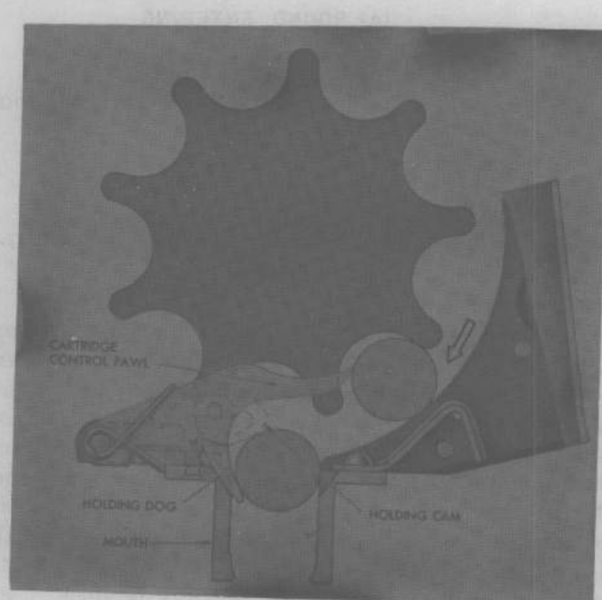
The freed round continues along the curved chute support until it reaches the entrance to the feed mouth where it contacts the loading guides on the opposite side of the mouth. These guides form the path for the round as it moves into the mouth. Here the round contacts the loading levers and retaining finger. As the feed wheel



(A) Stripper Cam Action



(B) Holding Cam Action



(C) Holding Dog and Control Pawl Action

Figure 7–12. Recoil-operated Rotating Feed Mechanism

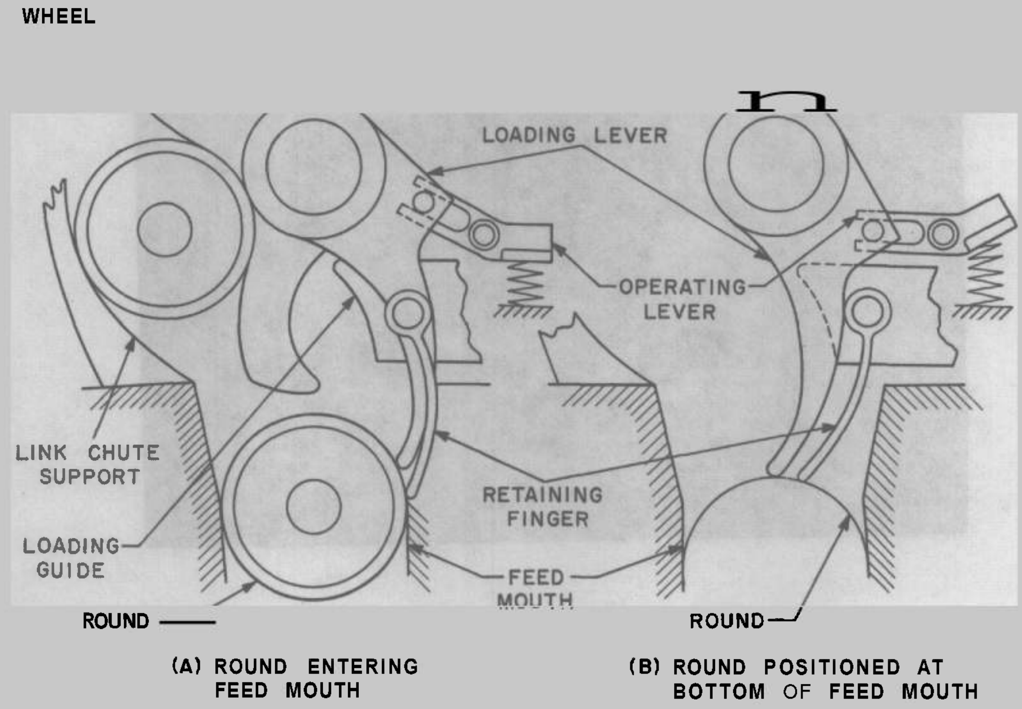


Figure 7–13. Feed Wheel and Operating Lever Units

continues to push the round downward, forces transmitted through the round rotate loading levers and retaining finger outward until the round moves free of the feed wheel. At this stage, being free of the influence of the feed wheel, the loading levers are ready to return to their original position, meanwhile holding the round against the top of the bolt. As soon as the bolt recoils, the loading levers snap the round downward to the ways where it is held in proper alignment by the levers and retaining finger until pushed forward and chambered by the bolt. (This series of events is illustrated in parts (D), (E), (F) of Fig. 7–14.) Unlike the recoil-operated feed mechanism, the last round in the belt may be fired without fear of jamming because of the position control on the round exercised by the loading levers and retaining finger.

While each round is resting on top of the bolt waiting for recoil, the loading mechanism stops. Although these intervals are short, a friction clutch slips a short distance during each interval to prevent the motor from overloading.

7–2.4 LINKLESS FEED SYSTEM

The linkless feed system was developed in order to provide a reliable high speed method of feeding ammunition to a gun without inducing the tremendously high inertia forces that are normally experienced with the conventional link systems. Not only are accelerating forces in the conveyor held to a minimum, but the linkless feed system also provides a large, convenient storage capacity for the ammunition. The major components are the fixed outer drum that stores the ammunition, the rotating inner drum that advances the stored ammunition for loading and feeding purposes, the exit unit that transfers ammunition from drum to conveyor belt, and the conveyor system that carries the loaded ammunition to the gun and carries the spent cases to the entrance unit where these cases are returned to the drum assembly. Two transfer units, at the rear of .ne gun, transfer live and spent ammunition from conveyor belt to gun and later from gun to conveyor belt. There are two general classes of linkless ammunition feed, the double and

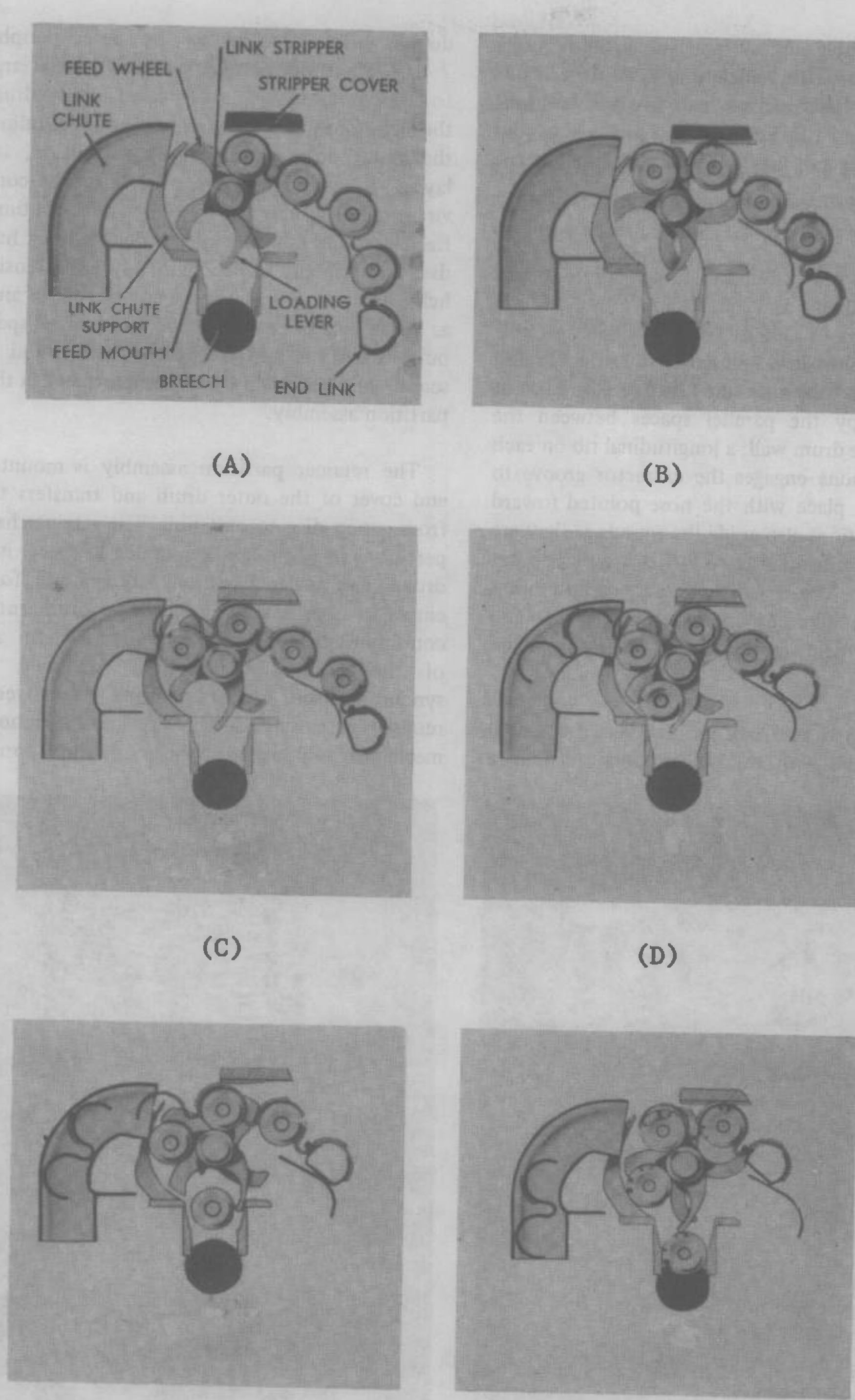


Figure 7–14. Electrically Operated Rotating Feed Mechanism

single end. The double end system has all the features defined above whereas the single end system has only those components that operate with live ammunition; all spent cases and unfired rounds that pass the transfer unit at the gun are dumped from the system by the gun.

The outer drum is a stationary storage compartment that may hold as many as 1200 20 mm rounds. It is a large cylinder that is lined with L-shaped double partitions that extend along the entire drum length and protrude radially toward the axis. Rounds of ammunition occupy the parallel spaces between the partitions. Near the drum wall, a longitudinal rib on each side of the partitions engages the extractor groove to hold the round in place with the nose pointed toward the axis. The partitions also guide the rounds as they are advanced along the drum. Fig. 7–15 is a typical outer drum showing the partition and ammunition arrangement. Two adjacent spaces near the exit and entrance units remain empty at all times to avoid jamming operation.

The inner drum is the rotating member of the two drums. It is a tube with thin sheet metal forming a double helix attached to the outer periphery (Fig. 7–16). The outer periphery of the helix is large enough to engage the ogive of each round. As the drum rotates, the helix advances the ammunition longitudinally along the outer drum. During each revolution, two radial layers of ammunition are carried to the conveyor by virtue of the double helix which assures continuous feed. Each exit and entrance of the double helix has a scoop disc arrangement, which is merely an extension of the helix, to remove or replace a component of ammunition as the scoop passes the respective storage space on the outer drum. A sprocket carries the round along the scoop and deposits it into a compartment in the retainer partition assembly.

The retainer partition assembly is mounted on the end cover of the outer drum and transfers the rounds from scoop disc to exit unit. The retainer has two less partitions ($n-2$) than the number of spaces in the outer drum. The fewer partitions compensate for the two empty storage spaces in the outer drum and permit a continuous flow of ammunition to the gun. All gearing of the rotating components is timed to insure synchronization. The ammunition is removed from the retainer partition assembly by another scoop-sprocket mechanism and loaded in the ammunition conveyor.

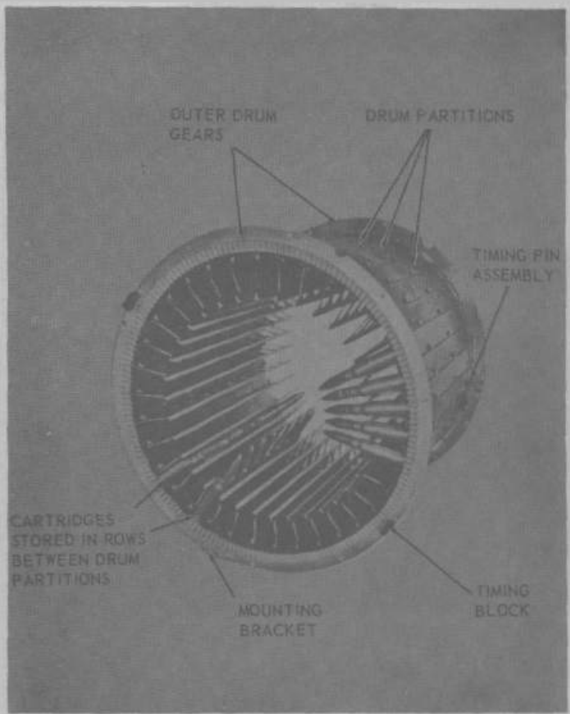


Figure 7–15. Outer Drum

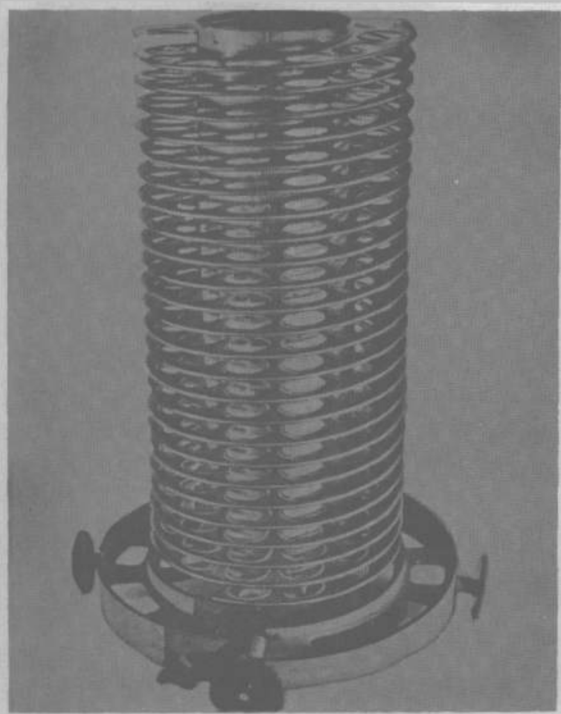


Figure 7–16. Inner Drum Helix

The conveyor is an endless belt made of elements similar to conventional ammunition links. The belt travels in two chutes; the feed chute and the return chute. The former supports and guides the loaded ammunition belt from drum assembly to the gun while the latter supports and guides the empty belt from gun to drum. The chute consist of many links or frames that are hooked together to form a smooth, continuous track which can twist and bend to assume the desired path contours. The chute end frames have snap fasteners for ready attachment to other system components thus providing a good maintenance characteristic.

Each conveyor element is made of two semicircular loops of different size that are held together by a rivet. The larger loop has lugs that engage the extractor groove of the cartridge case. When connected, the smaller loop of one element rests under a tab in the forward part of the larger loop of the adjacent element. Fig. **7–17** shows several elements joined in this manner. One holds a round of ammunition. On the left, a small loop is shown free with its larger counterpart shown on the right. The element does not grip the case tightly and can fully support the round only with aid of the chutes. Once outside the chutes, the rounds or cases are easily lifted from or placed into the elements by the sprockets of the various transfer units. Once the element is relieved of the case, the belt can be easily disconnected or folded over itself. Since the belt is so folded as it passes through the feeder, two elements could part if at least one of them were empty while in the feed chute. Therefore, all elements should be loaded while in the feed chute.

Fig. **7–18** is a schematic of the operating features of a double end linkless feed system. Both conveyor belt and ammunition loop are continuous circuits. The stationary outer drum shows only one row of ammunition. When the rotating elements turn (in the direction indicated by the heavy arrows) the helix on the inner drum advances the ammunition to the right where the scoop picks up the round as it leaves the helix at A. As the scoop continues toward the next stored row of ammunition, a sprocket in the scoop disc assembly carries the round to the retainer partition assembly where the transfer is made at B. To have an empty partition available for the next round, B must travel faster than A. The round leaves the retainer partition at C, the transfer point to the ammunition exit unit. In the meantime, the retainer partition assembly continues to rotate, but the partitions between C and D are empty and will not receiver any rounds until B passes C. The scoop disc at D, the end of the other helix, is diametrically opposite A. It too is collecting a round from each row of stored ammunition and depositing it into a retainer partition. Since the retainer is moving faster than the scoop, all partitions between D and B will be filled by the scoop at D, just as B passes C. However, because the exit unit at C occupies some space, the flow of ammunition from A to B must be interrupted to avoid jamming the rounds against the exit unit. For this reason, two rows of storage space, accurately indexed ahead of C, will interrupt the flow until the required clearance is achieved. Until pick-up is resumed, empty partitions continue to accumulate beyond C. Proper synchronization by gear trains insure continuous ammunition supply to the conveyor.

If there are n storage rows in the outer drum, then there are $n - 2$ rows of stored ammunition. Since two layers of rounds are removed for each revolution of the inner drum, the total number reaches $N = 2(n - 2)$. To retrieve this discharge, the retainer partition assembly must rotate at least twice the speed of the

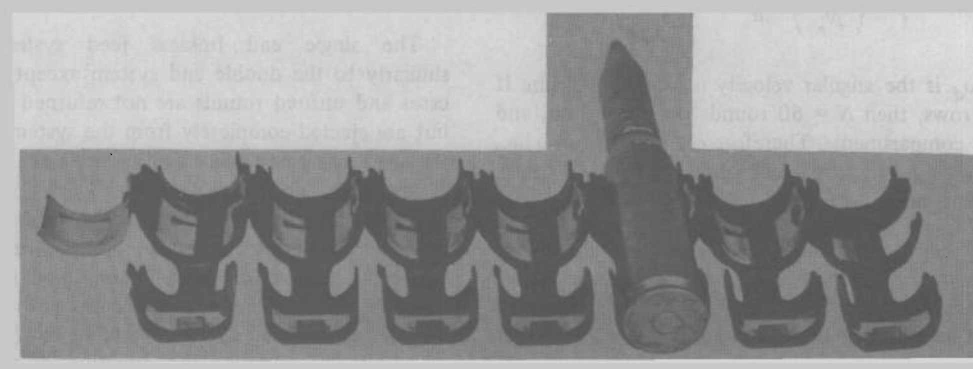


Figure 7–17. Conveyor Elements

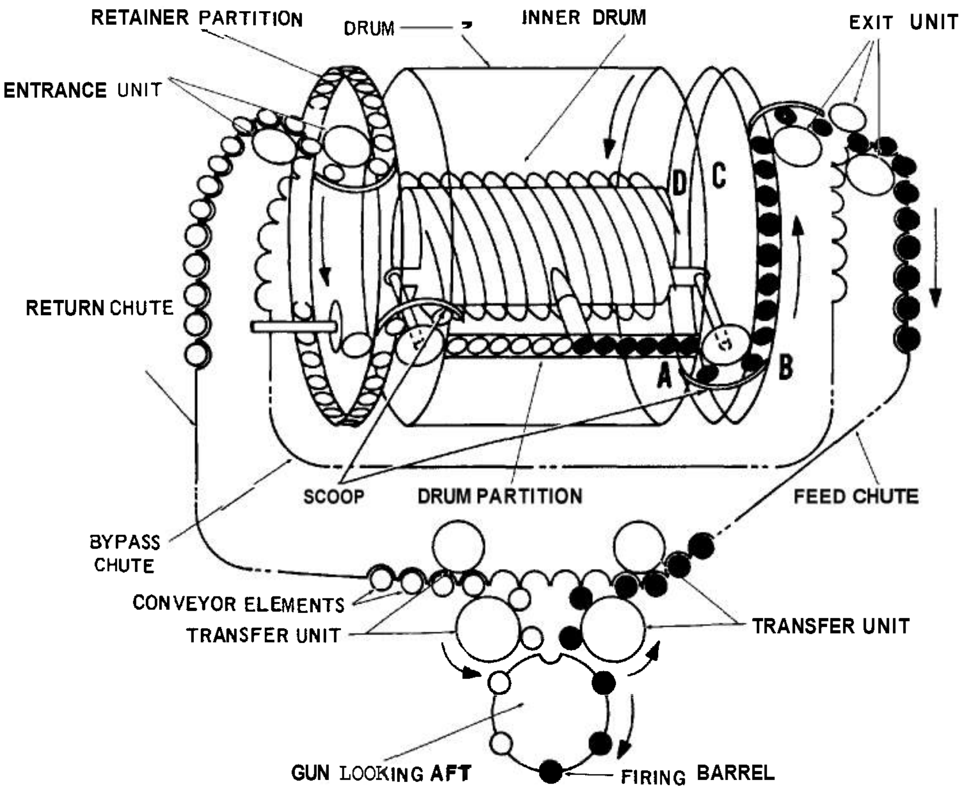


Figure 7–18. Schematic of Linkless Feed System

inner drums. However, while the scoop passes two rows, four empty partitions pass by C. Therefore, for proper indexing, the number of partitions in the retainer $N_r = n - 4$. But the retainer must pick up N rounds, hence its angular velocity ω_r must be

$$\omega_r = \left(\frac{N}{N_r} \right) \omega_d \qquad (7\text{–}21)$$

where ω_d is the angular velocity of the inner drum. If $n = 32$ rows, then $N = 60$ rounds per revolution, and $N_r = 28$ compartments. Therefore $\omega_r = (15/7)\ \omega_d$, i.e., the retainer partition assembly must rotate 2-1/7 times faster than the drum.

After the ammunition leaves the retainer partition at C, it passes through the exit unit where it is loaded into the elements of the conveyor. The conveyor carries the ammunition through the feed chute to the transfer unit where it is loaded into the gun. After firing, the spent case is returned to the transfer unit and reloaded in the conveyor which now moves through the return chute and back to the ammunition entrance unit. The entrance unit removes the cases, and the empty conveyor completes its loop to the exit unit. The empty cases now repeat the same functions as the live rounds but in reverse order, eventually to be stored again in the outer drum.

The single end linkless feed system operates similarly to the double end system except that empty cases and unfired rounds are not returned to the drum but are ejected completely from the system. Therefore, the various ammunition handling units are needed only at the exit end of the drum. Fig. 7–19 shows the operation schematically. Since empty cases and unfired rounds no longer need to be reloaded into the conveyor, the transfer unit is simplified to the point where it actually becomes little more than a feeder. The drum is loaded by disconnecting the system at the exit unit and reversing the direction of moving units and ammunition.

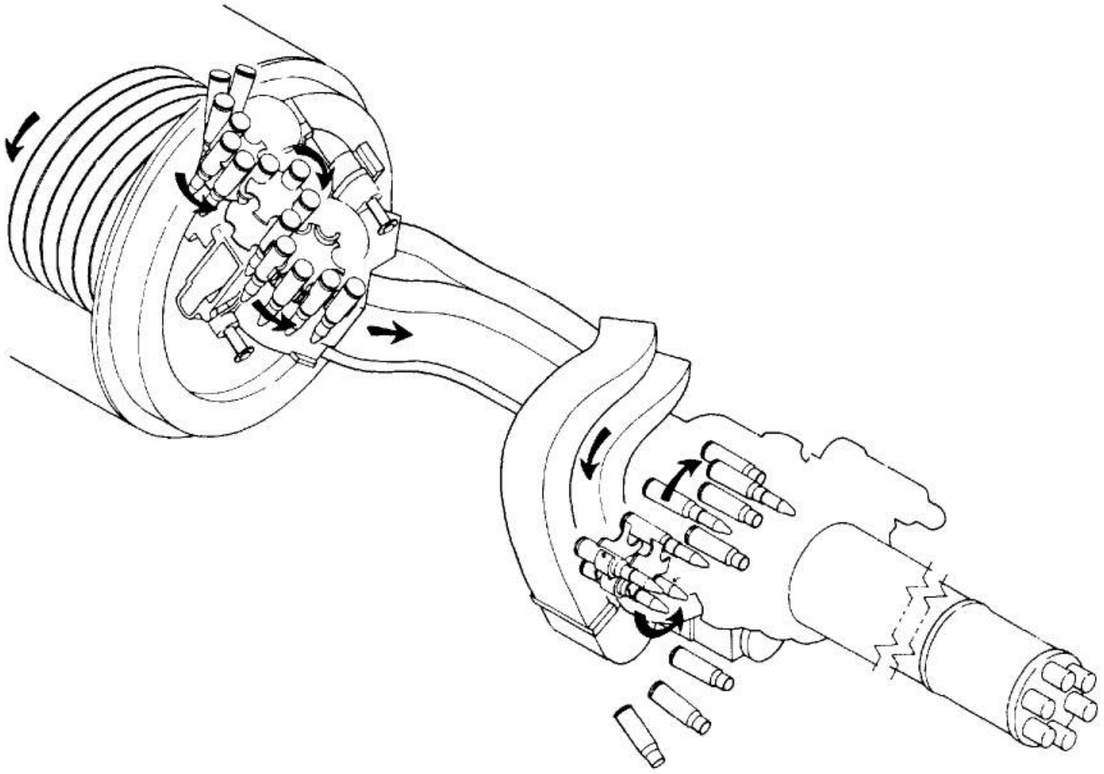


Figure 7–19. Path of Rounds in Single-end System

All linkless feed systems, whether single or double ended, require continuous external power. Also the feeder must be declutched or disengaged from the gun to provide gun clearing after each burst or single shot to prevent "cook-off".

7–2.4.1 Power Required

The power required to operate a linkless feed system includes the power to accelerate the ammunition and all moving components, and that needed to overcome the frictional resistance to all motion. Velocities and accelerations vary from component to component of the feed system; therefore, to maintain a reasonable perspective of the action in each component, the velocity and acceleration of each component is given in terms of its counterpart in the gun. The action throughout the system may be demonstated more clearly by realizing that each time a new round is accepted by the gun: (1) the rounds in each component advance through the respective spaces between rounds, and (2) that the acceleration and velocity for any given component will vary as the linear distance between the rounds. The schematic of Fig. 7–18 illustrates and identifies the components.

The following symbols will be used in the equation for computing the power required to drive a linkless feed system:

a = general term for linear acceleration

F = general term for force

I = general term for the mass moment of inertia

P = general term for power

p = general term for space between rounds (pitch)

N = number of rounds or elements, loaded or empty in each component

R = general term for radius; gun axis to chamber center

T = general term for torque

v = general term for linear velocity

α = general term for angular acceleration

ϕ = double helix drive angle

ω = general term for angular velocity

The following subscripts refer to the specific component of the terms just defined:

c = chute; feed, bipass, or return

d = drum, inner

e = exit or entrance unit

r = retainer partition

t = transfer unit

The peripheral acceleration and velocity at the chamber axis are

$$a = \alpha R \tag{7-22}$$

$$v = \omega R . \tag{7-23}$$

The corresponding accelerations and velocities of the rounds in the other components of the feed system will vary according to the ratio of the pitches. In the transfer unit

$$a_t = a\left(\frac{p_t}{p}\right) \tag{7-24}$$

$$v_t = v\left(\frac{p_t}{p}\right) . \tag{7-25}$$

In any of the chutes

$$= a_t\left(\frac{p_c}{p_t}\right) = a\left(\frac{p_c}{p}\right) \tag{7-26}$$

$$v_c = v_t\left(\frac{p_c}{p_t}\right) = v\left(\frac{p_c}{p}\right) . \tag{7-27}$$

In the ammunition entrance and exit units

$$a_e = a_c\left(\frac{p_e}{p_c}\right) = a\left(\frac{p_e}{p}\right) \tag{7-28}$$

$$v_e = v_c\left(\frac{p_e}{p_c}\right) = v\left(\frac{p_e}{p}\right) . \tag{7-29}$$

In the retainer partition assembly

$$a_r = a_e\left(\frac{p_r}{p_e}\right) = a\left(\frac{p_r}{p}\right) \tag{7-30}$$

$$v_r = v_e\left(\frac{p_r}{p_e}\right) = v\left(\frac{p_r}{p}\right) . \tag{7-31}$$

If the storage drum has N storage spaces of which two are empty, and the retainer has $N - 4$ partitions, the ratio of the kinematics of retainer partition to inner drum is

$$\rho_d = \frac{2(N-2)}{N-4} , \tag{7-32}$$

provided that the inner drum has a double helix drive. If ϕ is the angle of the double helix drive and p_d is the pitch, the slope of the helix is

$$\tan\phi = p_d/2\pi R_d . \tag{7-33}$$

To express the axial acceleration and velocity along the outer drum spaces in terms of their retainer counterparts, the ratio of the radii of inner drum and retainer must also be included. The axial acceleration and velocity of the stored rounds are

$$a_a = \frac{a_r}{\rho_d}\left(\frac{R_d}{R_r}\right)\tan\phi \tag{7-34}$$

$$v_a = \frac{v_r}{\rho_d}\left(\frac{R_d}{R_r}\right)\tan\phi \tag{7-35}$$

Eqs. 7-22 through 7-35 contain the information needed to compute the power required to overcome the

resistance of friction and inertia. The axial force F_a on the helix that will drive the rounds contained in the outer drum is

$$F_a = N_d W_a \left(\frac{a_a}{g} + \mu \right) \qquad (7-36)$$

where g = acceleration of gravity

N_d = total rounds in outer drum

W_a = weight of each round

μ = coefficient of friction

The corresponding power required is

$$P_a = F_a v_a. \qquad (7-37)$$

The torque required to turn the inner drum and overcome the sliding friction on the helix is

$$T_d = I_d \alpha_d + \mu F_a R_d \qquad (7-38)$$

where $\alpha_d = \dfrac{a_a}{R_, \tan\phi}$. The expression for power is

$$P_d = T_d \omega_d = T_d \left(\frac{v_a}{R_d \tan\phi} \right). \qquad (7-39)$$

The torque required to turn the retainer partition will also include that necessary to turn the ammunition in half the partitions since this number of rounds is never exceeded.

$$T_r = I_r \alpha_r + N_r \left(\frac{W_a}{g} \right) a_r R \qquad (7-40)$$

where R_a = radius to the *CG* of the round

$\alpha_r = a_r/R_r$

The corresponding power is

$$P_r = T_r \omega_r = T_r \left(\frac{v_r}{R_r} \right) \qquad (7-41)$$

Similar equations are used, where applicable, for the other components. In the ammunition entrance and exit units the force **is**

$$F_e = N_e \left(\frac{W_u}{g} \right) a_e \qquad (7-42)$$

where $W_u = W_e + W_a$ or $W_e + W_,$, for exit and entrance units, respectively.

$$T_e = I_e \alpha_e = I_e \left(\frac{a_e}{R_e} \right) \qquad (7-43)$$

$$P_e = F_e v_e + T_e \left(\frac{v_e}{R_e} \right) \qquad (7-44)$$

In the chutes

$$F_c = N_c W_c \left(\frac{a_c}{g} + \mu \right) \qquad (7-45)$$

$$P_c = F_c v_c$$

Observe that N_c and $W_,$ represent the number and weight of round and conveyor element for the feed chute, the number and weight of only the empty cartridge case and conveyor element for the return chute, and the number and weight of the element above for the bypass chute. In the transfer unit

$$F_t = N_t \left(\frac{W_t}{g} \right) a_t \qquad (7-46)$$

where $W_,$ may be W_a or W_{cc}, depending on whether the round is entering or the case emerging from the gun.

$$T_t = I_t \left(\frac{a_t}{R_t} \right) \qquad (7-47)$$

$$P_, = F_t v_t + T_t \left(\frac{v_t}{R_t} \right) \qquad (7-48)$$

7-2.4.2 Example Problem for Power Required

Compute the power required for a double end linkless feed system having the following design data:

I_d = 9.32 lb-in.-sec²

I_e = 0.032 lb-in.-sec²

$I_,$ = 2.54 lb-in.-sec²

I_t = 0.095 lb-in.-sec²

N = **32** partitions

N_c = 45 feed; **35** bypass; **25** return

N_d = 1200 rounds

N_e = 3 rounds; 3 cases

N_t = 2 rounds; 2 cases

p = 2.77 in.

p_c = 1.62 in.

p_d = 2.54 in.

p_e = 2.09 in.

p_r = 2.24 in.

p_t = 2.09 in.

R = 2.643 in.

R_a = 6.0 in.

R_d = 7.0 in.

R_e = 2.0 in.

R_r = 10.0 in.

R_t = 2.0 in.

W_a = 0.57 lb, weight of round

W_{cc} = 0.25 lb, weight of case

W_e = 0.12 lb, weight of element

μ = 0.22, coefficient of friction

All data computed from Eqs. 7–22 through 7–48 are put in terms of the gun kinematics.

$$a_t = \left(\frac{2.09}{2.77}\right) a = 0.755\,a \qquad v_t = 0.755\,v$$

$$a_c = \left(\frac{1.62}{2.77}\right) a = 0.585\,a \qquad v_c = 0.585\,v$$

$$a, = \left(\frac{2.09}{2.77}\right) a = 0.755\,a \qquad v_e = 0.755\,v$$

$$a, = \left(\frac{2.24}{2.77}\right) a = 0.809\,a \qquad v_r = 0.809\,v$$

$$\rho_d = \frac{2(32-2)}{32-4} = \frac{60}{28} = 2.143$$

$$\tan\phi = \frac{2.54}{14\pi} - 0.05775$$

$$a_a = \left(\frac{0.809a}{2.143}\right)\left(\frac{7}{10}\right) 0.05775$$

$$= 0.0153\,a,\ \text{in./sec}^2;\quad v_a = 0.0153\,v,\ \text{in./sec}$$

$$F_a = N_d W_a\left(\frac{a_a}{g} + \mu\right)$$

$$= 1200 \times 0.57\,(0.0000396\,a + 0.22)$$

$$= 150.48 + 0.027\,a,\ \text{lb}$$

$$P_a = F_a v_a = (2.302 + 0.00041\,a)\,v,\ \text{in.-lb/sec}$$

$$T_d = I_d\alpha_d + \mu F_a R_d = 9.32\left(\frac{0.0153\,a}{0.404}\right)$$

$$+ 0.22\,(150.48 + 0.027a)7.0 = 0.3530a$$

$$+ 231.7 + 0.0416a = 231.7 + 0.3946a,\ \text{lb-in.}$$

$$P_d = T_d\omega_d = (231.7 + 0.3946a)\left(\frac{0.0153\,v}{0.404}\right)$$

$$= (8.775 + 0.01494a)\,v,\ \text{in.-lb/sec}$$

$$T_r = 2.54\left(\frac{0.809a}{10.0}\right) + 14\left(\frac{0.57}{386.4}\right)\left(0.809a\right) 6.0$$

$$= (0.2055 + 0.1002)a = 0.3057a,\ \text{lb-in.}$$

where

$$N_r = \frac{1}{2}(N - 4) = 14$$

$$R_r = 10.0\ \text{in.}$$

$$P_r = T_r\left(\frac{v_r}{R_r}\right) = \left(0.3057a\right)\left(\frac{0.809\,v}{10.0}\right)$$

$$- 0.02473\,av,\ \text{in.-lb/sec}$$

The combined data in the entrance and exit units according to Eqs. 7–42, 7–43, and 7–44 are

$$F_e = N_e \left(\frac{\Sigma W_u}{g}\right) a_e = \frac{3(0.37 + 0.69)}{386.4} \; 0.755\, a = 0.00621\, a, \text{ lb}$$

where $\Sigma W_u = (W_e + W_{cc}) + (W_e + W_t) = (0.37 + 0.69)$.

$$T_e = 2I_e\left(\frac{a_e}{R_e}\right) = 2 \times 0.032 \times 0.755\, a/2.0 = 0.024160a, \text{ lb-in.}$$

$$P_e = F_e v_e + T_e \left(\frac{v_e}{R_e}\right) = 0.00621\, a \times 0.755\, v + 0.02416a \times 0.755\, v/2.0 = 0.00469av$$

$$+ 0.00912av = 0.01381\, av, \text{in.-lb/sec.}$$

The accumulated data in the chutes according to **Eqs.** 7–45 and 7–46 are

$$F_c = \left[N_f(W_a + W_e) + N_p W_e + N_r(W_{cc} + W_e)\right] \left(\frac{a_c}{g} + \mu\right)$$

$$= (45 \times 0.69 + 35 \times 0.12 + 25 \times 0.37) \left(\frac{0.585\, a}{386.4} + 0.22\right)$$

$$= 9.79 + 0.06737\, a, \text{ lb}$$

where $N_f = N_c = 45$; $N_p = N_c = 35$; $N_r = N_c = 25$.

$$P_c = F_c v_c = 0.585 v F_c = (5.727 + 0.03941\, a)v, \text{ in.-lb/sec}$$

The accumulated data in the transfer unit are computed according to Eqs. 7–44, 7–45, and 7–46.

$$F_t = N_t(W_a + W_{cc})a_t/g = 2(0.57 + 0.25)0.755a/386.4 = 0.0032a, \text{ lb}$$

$$T_t = I_t\left(\frac{a_t}{R_t}\right) = 0.095 \times 0.755a/2.0 = 0.03586a, \text{ lb-in.}$$

$$P_t = F_t v_t + T_t\left(\frac{v_t}{R_t}\right) = 0.0032a \times 0.755v + 0.03586a\left(\frac{0.755\, v}{2.0}\right) = 0.00242av$$

$$+ 0.01353av = 0.01595av, \text{ in.-lb/sec.}$$

The total power required to drive the linkless feed system is

$$P_f = P_a + P_d + P_r + P_e + P_c + P_t = (2.302 + 0.00041a)v + (8.775 + 0.01494a)v$$

$$+ 0.02473av + 0.01381av + (5.727 + 0.03941a)v + 0.01595av$$

$$= 16.804v + 0.1093av = P_v + P_{av}, \text{ in.-lb/sec}$$

The power will be computed for several increments i taken from the tabulated values of Table 6–4, the Gun Operating Power computations. From Eqs. 7–22 and 7–23, the linear acceleration and velocity are

$$a = 2.643\alpha$$

$$v = 2.643\omega$$

The computed data are listed in Table 7–1. The two components of the total power are

$$P_v = 16.804v$$

$$P_{av} = 0.1093av$$

7–3 EXTRACTORS, EJECTORS, AND BOLT LOCKS

7–3.1 EXTRACTORS

Extractors are machined components that pull the cartridge case from the chamber as the bolt recoils. Assembled near the breech face of the bolt, they are generally spring loaded to tilt toward the longitudinal axis of the bolt and thus direct a continuous clamping effort on the cartridge case. This clamping effort is sometimes supplemented by the restraining wall of the receiver or by the induced moment of the axial forces needed for extraction. The source of whatever effort is applied is determined by the type extractor.

Four types of extractor are shown in Fig. 7–20. Of these, (A) and (C) are similar insofar as spring installation is concerned but differ with respect to method of transmitting the tipping action. (A) is the extractor used in the 7.62 mm, M60 Machine Gun. The helical compression spring provides the clamping effort. The plunger transmits the spring force to the outer portion of the extractor while the bolt offers the reaction on the inner portion. Contact between extractor and bolt is effected by a boss on the extractor which rests in a recess in the bolt. The front surface of the boss is conical and is matched by its female counterpart. Tipping action uses this location as the fulcrum.

Springs must be reasonably stiff so that an appreciable effort is demanded to release the case rim. For instance, the nominal spring load is F_s = 15 lb on the extractor, (M60 Machine Gun). The horizontal reaction H_s (Fig. 7–21 (A)) has the same value but since it is on a slope of 20° 40′, it has a vertical component.

$$V_s = H_s \tan 20°40' = 15 \times 0.377 = 5.65 \text{ lb}$$

The vertical reactions on the pads at A and B (Fig. 7–21 (A)) do not contribute to the solution of F_e for the reaction at B gradually disappears as F_e increases to the value that displaces the extractor outward. The value

TABLE 7–1. POWER REQUIRED FOR LINKLESS BELT FEED SYSTEM

i	α, rad/sec²	ω, rad/sec	a, in./sec²	v, in./sec	P_v, in.-lb/sec	P_{av}, in.-lb/sec	P_f, in.-lb/sec	HP
44	200.0	34.72	528.6	91.8	1543	5304	6847	1.04
58	173.6	39.07	458.8	103.3	1737	5180	6917	1.05
60	164.6	40.41	435.0	106.8	1795	5078	6873	1.04
73	144.8	43.12	382.7	114.0	1916	4768	6684	1.01
88	118.2	46.20	312.4	122.1	2052	4169	6221	0.94
100	98.7	48.07	260.9	127.0	2134	3622	5756	0.87
120	62.1	50.66	164.1	133.9	2250	2402	4652	0.70
130	51.0	51.21	134.8	135.3	2274	1993	4267	0.65
140	34.3	51.84	90.6	137.0	2302	1357	3659	0.55
150	16.1	52.25	42.6	138.1	2321	643	2964	0.45
162	1.9	52.36	5.0	138.4	2326	76	2402	0.36

Since the maximum power required to operate the gun is 14.4 HP., at increment i = 58 (see Table 6–4) the total power for gun and feed system totals 15.45 HP.

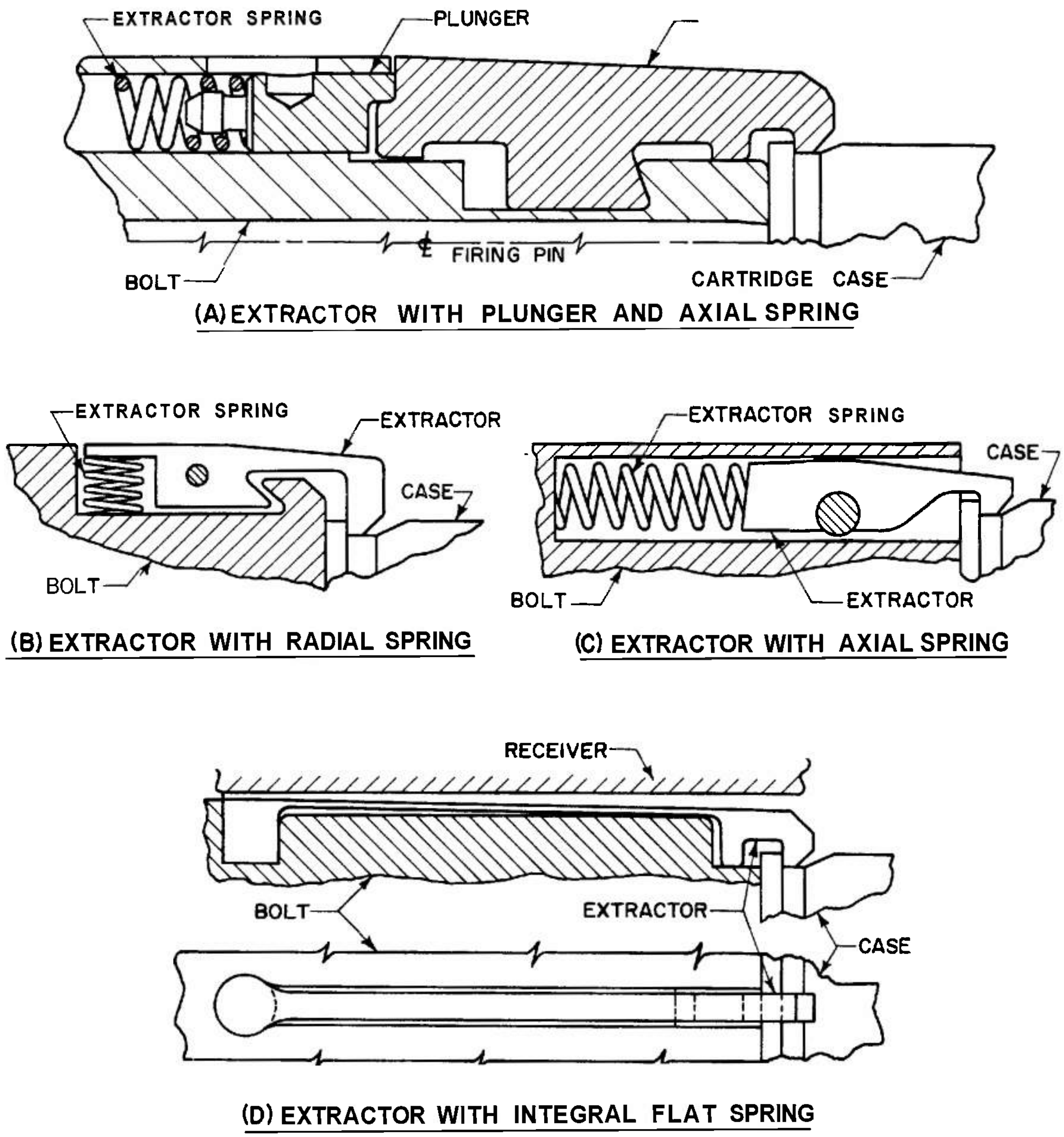
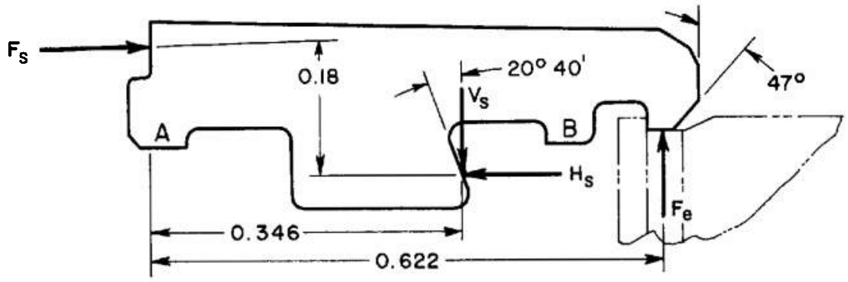
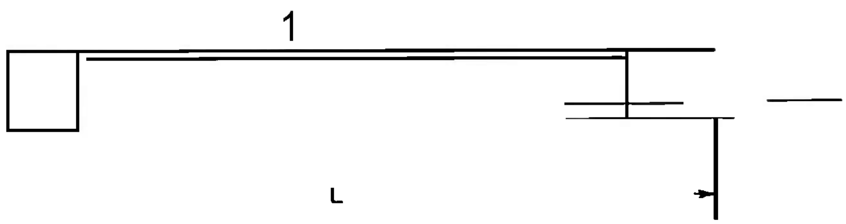


Figure 7–20. Extractors



(A) AXIAL SPRING LOAD



of F_e, thc maximum extractor load to clear the cartridgc case, is found by balancing moments about A.

$$F_e = \frac{0.18F_s + 0.346\,V_s}{0.622} = \frac{2.70 + 1.96}{0.622} = 7.5 \text{ lb.}$$

If thc outer slope of the lip is 8, the horizontal forcc on the extractor that will tilt it is

$$F_c = F_e \tan\theta$$

F_c represents the force that thc new round must exert on the cxtractor for proper engagement during loading. In the prcsent example 8 = 47°, therefore,

$$F_c = 7.5 \times 1.072 = 8.04 \text{ lb.}$$

The other extractors in Fig. 7–20 behave similarly except for (D), the integral flat spring type. Its initial force on the cartridge case base should be such that thc spring is just free of contact with the bolt. Thc maximum outward force will bc the force needed to snap thc extractor far enough outward to clear the casc rim.

The sample problem involves the extractor shown in Fig. 7–21 (B). Assume the maximum load to clear the case F_e = 7.5 lb, the same as in the earlicr example, and the corresponding load when the case is scated, F_{es} = 5 lb. All design data are known exccpt for spring thickness.

L = 1.8 in., spring length

L_e = 0.2 in., extractor length

E = 29 x 10^6 lb/in.^{2}, modulus of elasticity of stccl

Δy = 0.032 in, outward displacement needed to clear rim

b = 0.4 in, spring width

The spring functions as a cantilever. Far components of deflection are involved, the linear and angular deflections, both due to shear and end moment. The total deflection is

$$y = y_e + y_m + L_e\theta_e + L_e\theta_m = \left(\frac{2.66F_e}{EI}\right)$$

where

$$y_e = \frac{F_e L^3}{3EI} = \frac{5.832F_e}{3EI}$$

$$= \frac{1.944F_e}{EI}\text{ , shear deflection}$$

$$y_m = \frac{L_e F_e L^2}{2EI} = \frac{0.2 \times 3.24F_e}{2EI}$$

$$= \frac{0.324F_e}{EI}\text{ , moment deflection}$$

$$\theta_e = \frac{F_e L^2}{2EI} = \frac{3.24F_e}{2EI}$$

$$= \frac{1.62F_e}{EI}\text{ ,angular shear deflection}$$

$$\theta_m = \frac{M_e L}{EI} = \frac{0.2 \times 1.8F_e}{EI}$$

$$= \frac{0.36F_e}{EI}\text{ ,angular moment deflection}$$

The differential deflection from F_{es} to F_e

$$\Delta y = y_1 - y_2 = \frac{2.66}{EI}(F_e - F_{es})$$

$$= \frac{2.66}{29 \times 10^6 I}(7.5 - 5.0) = 0.032 \text{ in}$$

$$I = 7.16 \times 10^{-6} \text{ in.}^4$$

But $I = \frac{1}{12}\left(bt_s^3\right)$,therefore

t_s = 0.06 in., required spring thickness

7–3.2 EJECTORS

Ejectors are simple mechanisms that force the cartridge case from the receiver. They usually are spring-operated but may derive their energy from other sources such as small quantities of the propellant gas. There are perhaps as many kinds of ejectors as there are of their immediate associates, the extractors. Ejectors may be assembled either in the bolt or they may be attached to the receiver. Fig. 7–22 shows four types of ejectors, three are housed in the bolt, one in the chamber; Fig. 7–22 (A) is like that in the 7.62 mm, M60 Machine Gun. The spring force, via the ejector, is always applied to the edge of the cartridge case base. As soon as all radial restraint is removed diametrically opposite, the ejector will flip out the case. The off-center spring force accelerates the case angularly as well as linearly. However, the recoiling velocity at the time will compensate to some degree the forward velocity derived from the spring.

7–3.2.1 Ejector Dynamics

Because of its mass relative to the masses of ejector and cartridge case, the spring must be considered in the dynamics of the ejection mechanism. One-third of the spring mass—when included in the expressions for energy, velocity, and time—will yield approximate but sufficiently accurate results. The equivalent mass of the whole unit is

$$M_e = \frac{1}{3}M_s + M_{ej} + \left(\frac{k^2 + \bar{r}^2}{\bar{r}^2}\right) M_{cc} \qquad (7\text{–}49)$$

where M_{cc} = mass of the case

M_{ej} = mass of the ejector

M_s = mass of the spring

k = radius of gyration of the case about its CG

$\bar{r}$ = distance from tipping point on rim to CG of case

The equivalent mass of the case involves its mass moment of inertia since it is rotating. Fig. 7–23 shows a diagram of the pertinent dimensions. The equivalent mass may now be used in the appropriate formulas to determine the dynamics, Eq. 2–27 for the time, and the conventional equations for energy and velocity.

7–3.2.2 Sample Problem of Ejector Dynamics

The sample problem illustrating the ejector dynamics involves a cal .30 cartridge case. The known data together with the diagram in Fig. 7–23 provide the needed information

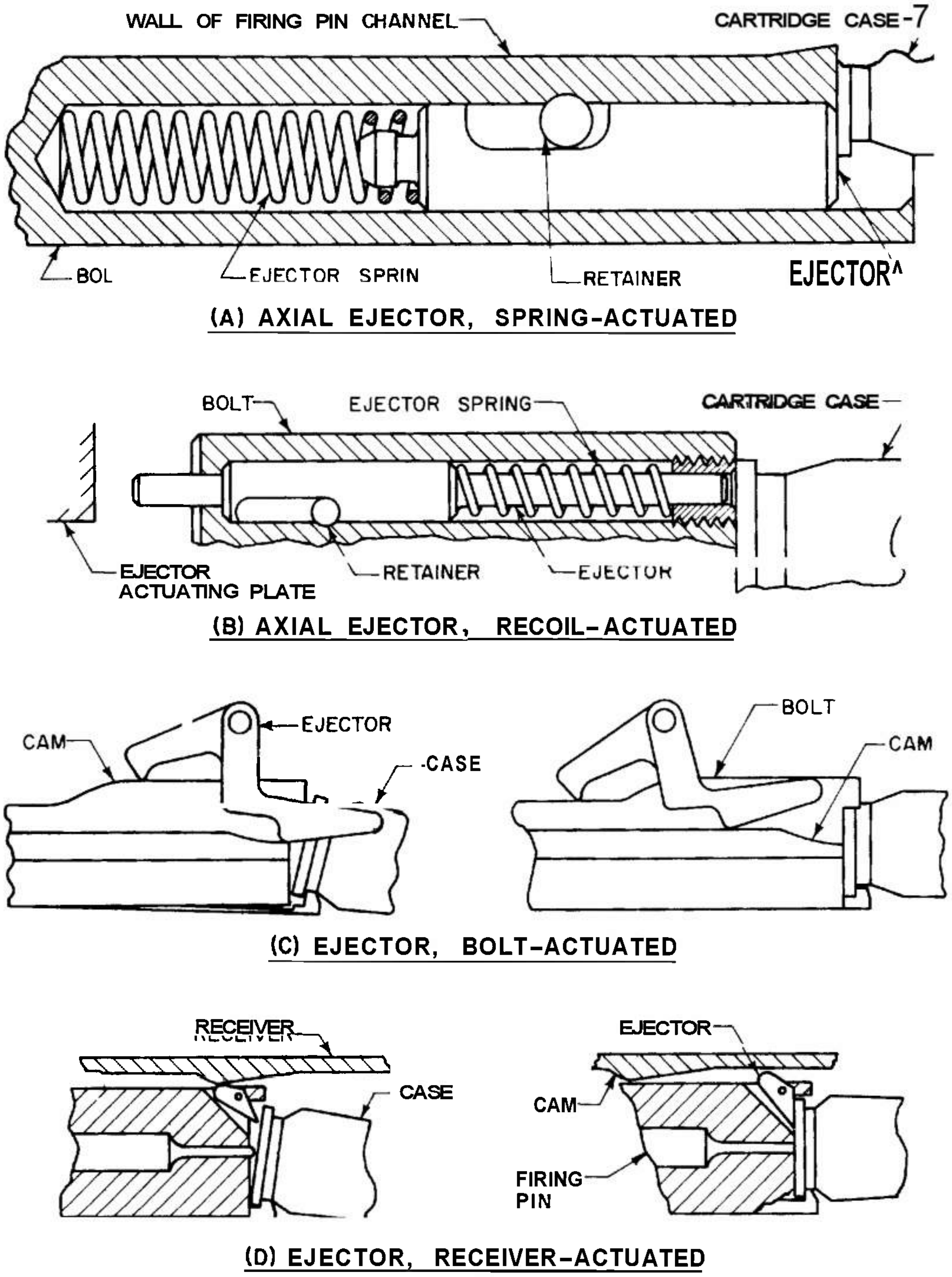


Figure 7–22. Ejectors

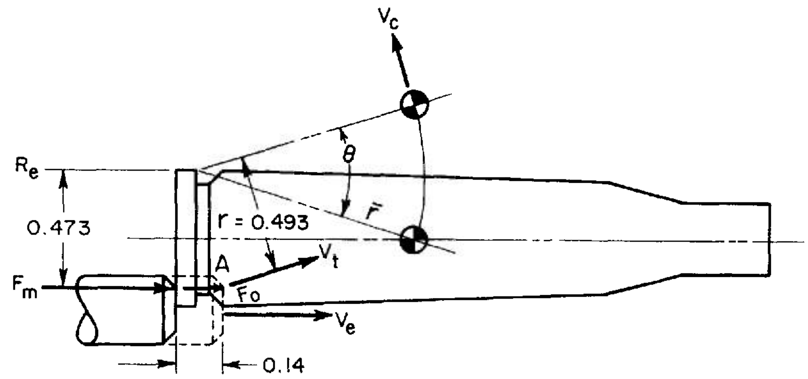


Figure 7–23. Ejector Loading Diagram

F_o = 14.5 lb, minimum spring force

F_m = 24.0, maximum spring force

k = 0.794 in, radius of gyration of case about its CG

K = 68 lb/in., spring constant

$\bar{r}$ = 1.1 in., distance from tipping point on rim to CG of case

W_{cc} = 0.0293 lb, weight of case

W_{ej} = 0.0492 lb, weight of ejector

W_s = 0.0039 lb, weight of spring

Δy = 0.14 in, spring deflection during ejector operation

From **Eq.** 7–49,

$$M_e = \frac{1}{g}\left(\frac{1}{3} W_s + W_{ej} + \frac{k^2 + \bar{r}^2}{\bar{r}^2} W_{cc}\right)$$

$$= \frac{1}{386.4}\left[0.0013 + 0.0492 + \left(\frac{0.63 + 1.21}{1.21}\right) 0.0293\right]$$

$$= 2.46 \times 10^{-4} \text{ lb-sec}^2/\text{in.}$$

To find the time of case ejection, apply **Eq.** 2–27 and then assume that the spring is 90%effective

$$t_e = \sqrt{\frac{M_e}{\epsilon K}} \operatorname{Cos}^{-1} \frac{F_o}{F_m} = \sqrt{\frac{2.46 \times 10^{-4}}{0.9 \times 68}} \operatorname{Cos}^{-1} \frac{14.5}{24}$$

$$= 2.005 \times 10^{-3} \operatorname{Cos}^{-1} 0.604 = 0.002005 \left(\frac{52.8}{57.3}\right)$$

$$= 0.00184 \text{ sec.}$$

The velocity of the ejector at this time is found from the expressions for kinetic energy and the work done by the spring by resorting to the appropriate portions of **Eq.** 2–24 and solving for the velocity.

$$v_e = \sqrt{\frac{\epsilon(F_o + F_m)\,\Delta y}{M_e}} = \sqrt{\frac{0.9 \times 38.5 \times 0.14}{2.46 \times 10^{-4}}}$$

$$= 140 \text{ in./sec}$$

During this time, with the extractor as the center of rotation, the case has traveled through the angle 8 (Fig. 7–23). The last point of contact between base of case and ejector is shown at A. The tangential velocity at this point, since the case is still rotating, is the component of v_e that is perpendicular to the turning radius r.

$$v_t = v_e \cos 8 = 140 \times 0.959 = 134.3 \text{ in./sec}$$

where8 = Tan-' $\frac{0.14}{0.473}$ = Tan-' 0.296 = 16'30'.

The corresponding angular velocity is

$$\omega = \frac{v_t}{r} = \frac{134.3}{0.493} = 273 \text{ rad/sec.}$$

If the case comes free of the extractor at this instant, the tangential velocity v_c becomes the linear ejected velocity.

$$v_c = \omega \bar{r} = 273 \times 1.1 = 300 \text{ in./sec}$$

This velocity is one of two components and is directed in a 16.5" angle forward. Chances are that the case will not become detached from the extractor simultaneously with the ejector, consequently the case path will be even less than 16.5". Regardless of the extractor behavior, the other velocity component, recoil velocity at the time of release may have the influence to direct the ejected case rearward.

The other three ejectors depend on the velocity of recoil for their effectiveness. The ejector in Fig. 7–22 (B) becomes active near the end of the bolt recoil. Recoil velocity here is relatively slow, therefore, this type may not operate quickly enough for fast firing guns. The remaining two, Figs. 7–22 (C) and 7–22 (D), can be activated at any position along the recoil stroke. These two ejectors are cam-operated and the ejection speed is dependent on bolt recoil speed and cam angle. With the cam rise being as abrupt as can be tolerated, the maximum ejection speed becomes available immediately after the cartridge case clears the chamber where useful bolt recoil velocity is highest. But ejection may be delayed because those components assembled near the breech present structural difficulties that prohibit the size opening needed for the ejection port. Many other types of ejectors have been successful but almost all depend on recoil energy directly for ejection effort or indirectly by storing latent energy in springs to be released when appropriate for ejection. Some type machine guns are particularly adaptable to incorporate an ejection effort derived directly from the propellant gas. One such gun is the revolver-type whose case can remain in the chamber and then be blown out by the next round fired. The details of this type ejection appears elsewhere in the text with the discussion on the revolver-type machine gun.

7–3.3 BOLT LOCKS

The bolt is held tightly against the base of the cartridge case during firing. Lugs or some similar type of projection bear against a cammed surface on the receiver, thereby locking it in position to provide the resistance to the rearward thrust of the propellant gas pressure. Some locking devices need not be integral with the bolt. One such is the breech lock shown in Fig. 7–24. This type, used on the M2 Cal .50 Machine Gun, has a breech lock that rises and falls in response to the action of one of two cams. Four positions of bolt and lock are illustrated. In the locked position, just before the round is fired, driving and buffer springs hold the bolt and recoiling parts in battery, with the breech lock holding them together and maintaining this state during the first part of the recoil stroke. Thus, all recoiling parts move as a unit until the lock pin, serving as a cam follower, contacts the depressor. By this time the projectile has emerged from the muzzle and gas pressures have dropped to safe levels for case extraction. When the pin first contacts the cammed surface of the depressor, it has already cleared the locking cam to permit a free downward unlocking movement which is effected by the depressor as the recoiling parts continue rearward. As soon as the lock exits from the bolt recess (Fig. 7–24 (C)), the now unattached bolt is accelerated rearward while the rest of the recoiling parts are stopped by the buffer and held in the fully recoiled position until the returning bolt releases them to reverse the activity. Shortly before reaching battery, the lock pin rides upward on the locking cam and enters the bolt recess to repeat its locking function.

Fig. 4–3 represents a bolt having integral locking lugs near its breech face. Locking and unlocking actions involve bolt rotation which is controlled by a cam cut into the wall of the bolt. The M60 7.62 mm Machine Gun has this type locking device. Bolt activity obtains all needed energy from the gas operating cylinder; the cam actuator, or follower, serving as the rigid link between operating rod and bolt. The actuator, moving rearward, rotates the bolt to unlock it according to the dictates of the cam. When the bolt is unlocked, the actuator forces it open by continued rearward motion. Bolt opening, and therefore case extraction, is delayed until propellant gas pressure drops to a safe level. Delay is controlled by: the location of the gas port along the barrel; the time needed to fill the gas chamber of the operating cylinder; and the time consumed for unlocking the bolt.

Locking action occurs during counterrecoil of the bolt. The driving spring forces the operating rod forward carrying the cam actuator and bolt with it. The locking lugs, riding in guides, prevent rotation while the concave recess at the beginning of the cam surface offers a

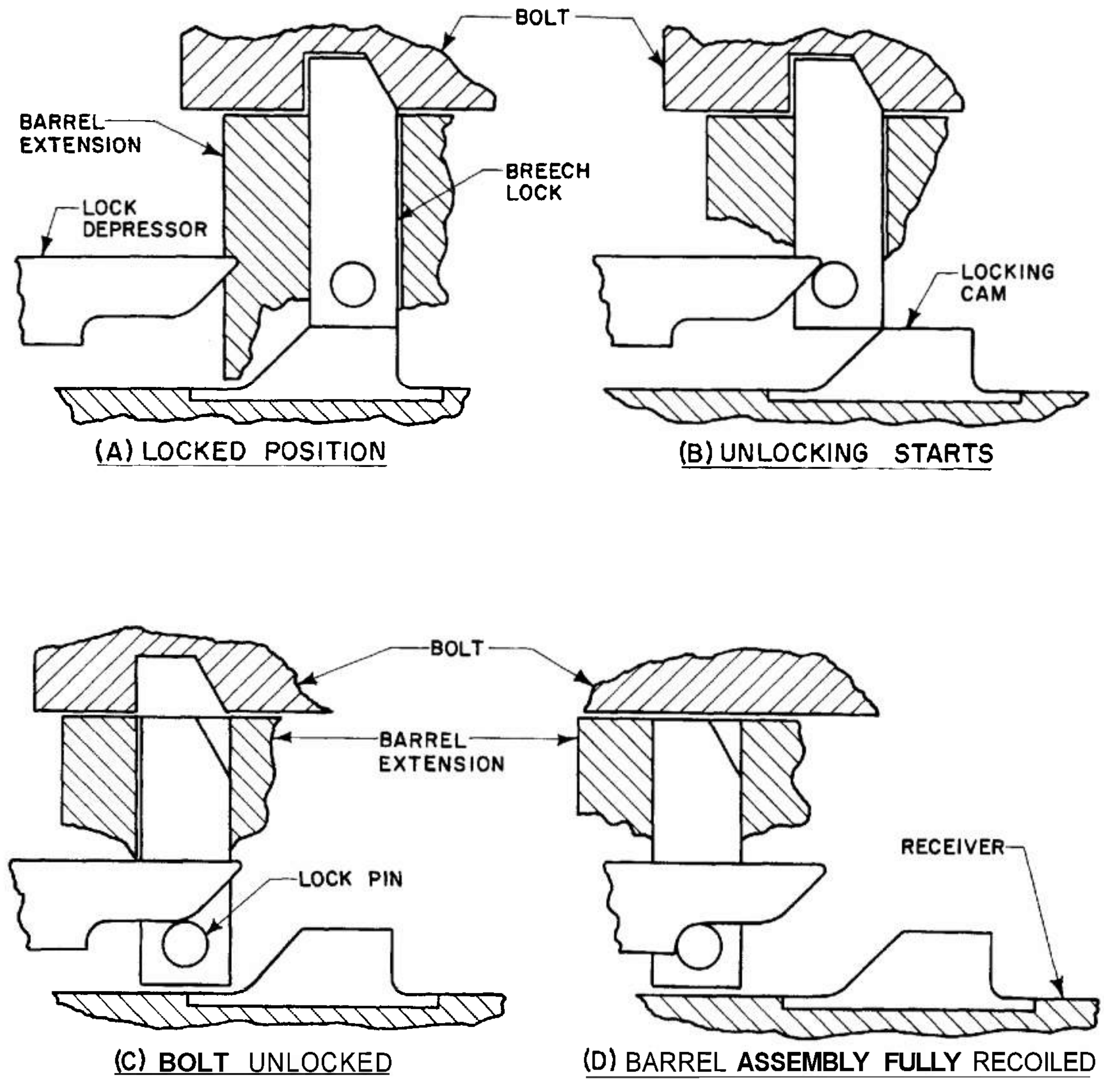


Figure 7–24. Sliding Breech Lock

convenient force transferring area. Somewhere along its return stroke, the bolt picks up a new round. Just as the cartridge case seats: the locking lugs leave the confines of the guides; angular restraint disappears; and the cam actuator, no longer restrained, leaves the recess to continue forward along the cam surface. Since the cam is forced to follow the path of the cam actuator, the bolt rotates into locked position to complete the cycle.

Several variations of the above lug type of bolt lock exist. Two such are the multiple lug lock and the interrupted thread lock. Both are adaptable to either gas- or recoil-operated machine guns. If recoil-operated, the receiver has a sleeve to perform the female function of the lock. As the gun recoils, a cam follower, integral with the sleeve, rotates it to free the lugs or interrupted threads on the bolt which then recoils by itself. The peripheral width of each lug or the length of each thread segment determines the angular distance through which the sleeve must turn to unlock the bolt. On gas-operated machine guns, the bolt is more apt to be the rotating element since the bolt, already actuated by the operating rod of the gas cylinder for linear motion, may just as readily be actuated by the rod for the angular motion of

unlocking. Actually, no set format applies to the unlocking method for any particular type gun. Design expediency usually controls the choice.

Another type of bolt lock that resorts to rotation, but in this case a tipping action, operates in the manner shown in Fig. **7–25.** Rather than rotate in a vertical plane perpendicular to the bore axis, this one tips in a vertical plane along the bore axis. Locking and unlocking are readily accomplished by the action of the operating rod in a gas-operated gun. Locked in position when the round is fired, the bolt remains in this state until propellant gases in the operating cylinder force the rod and carrier rearward. This rearward action causes the unlocking link to rotate forward and pull the locking lug from its notch in the receiver.

7–4 FIRING MECHANISM

7–4.1 COMPONENTS, TYPES, AND ACTION

The firing mechanism is a linkage that releases the firing pin or its equivalent to initiate firing. It has several components including trigger, sear, hammer, firing pin, cocking device, locking device, and safety. Each may be a separate component or may be integral with another. For instance, the trigger may also provide the sear and cocking facility as in some revolvers or pistols. However, in machine guns, the sear is generally a separate link. It engages the sear notch on the hammer, firing pin, or bolt or some appendage attached rigidly to one of those components. Cocking devices are arrangements that arm the firing mechanism by retracting the pertinent components to the position where the sear engages the sear notch to be held until triggered. Loading devices are mostly spring installations that provide the impetus to the firing pin. Safeties are machine elements that lock trigger, sear, or hammer to preclude inadvertent firing. A safety which locks the sear or hammer is more positive than one which locks only the trigger and is to be preferred. Fig. **7–26, 7–27,** and **7–28** shown three types of firing mechanism.

Fig. **7–26** shows a firing mechanism similar to that of the M2, Cal .SO Machine Gun. Three positions are represented: in battery, start of recoil, and fully recoiled positions. Except for a hammer, this example has all the components mentioned earlier. In battery, the spring-loaded sear holds the cocked firing pin by means of the sear notch at the end of the firing pin extension. Downward displacement of the sear releases the firing pin to be snapped toward the primer by the firing pin spring. Being somewhat remote from the sear, the trigger depresses it by lifting the tripper bar on one end thereby rotating the other end downward on the sear. The sear contacting surface of the trigger bar is cammed to minimize impact during counterrecoil when the sear end is held down for continuous firing.

Cocking the firing pin automatically is achieved by the cocking lever which rides in a stationary V-slot actuator. During recoil, the actuator flips the cocking lever forward thus rotating the lower end. The rotating lower end engages the firing pin extension and forces it rearward, meanwhile compressing the firing pin spring. In the fully recoiled position the cocking lever holds the sear beyond the sear notch to provide sufficient

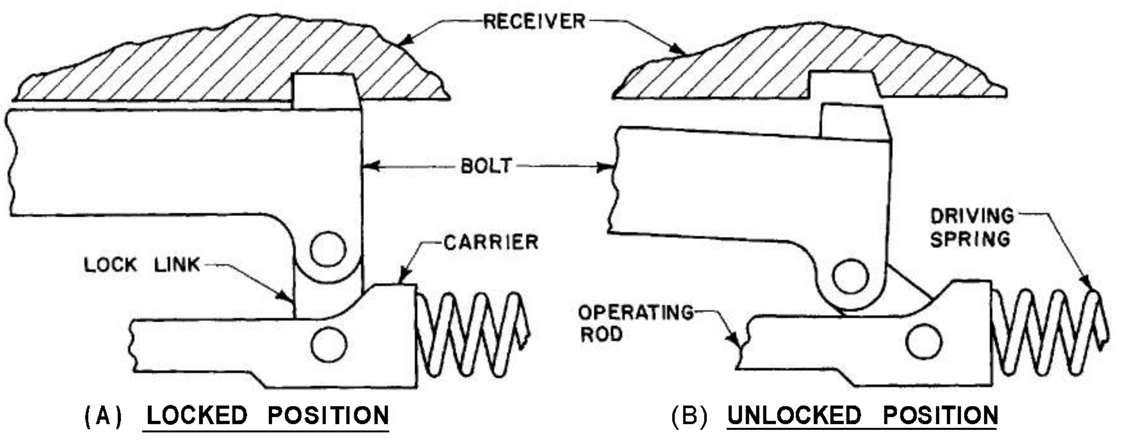


Figure 7–25. Tipping Bolt Lock

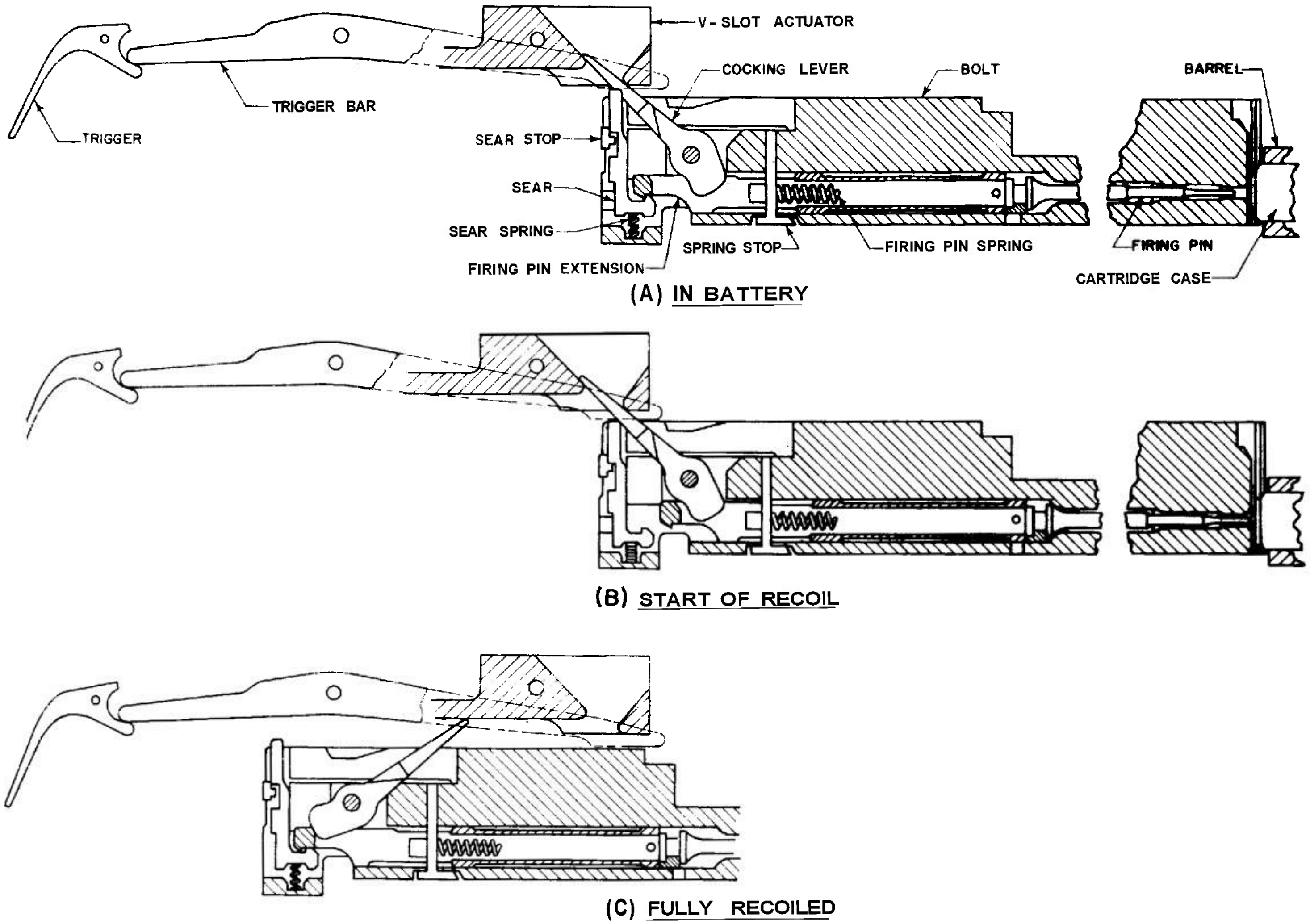


Figure 7–26. Firing Mechanism for Recoil Machine Gun

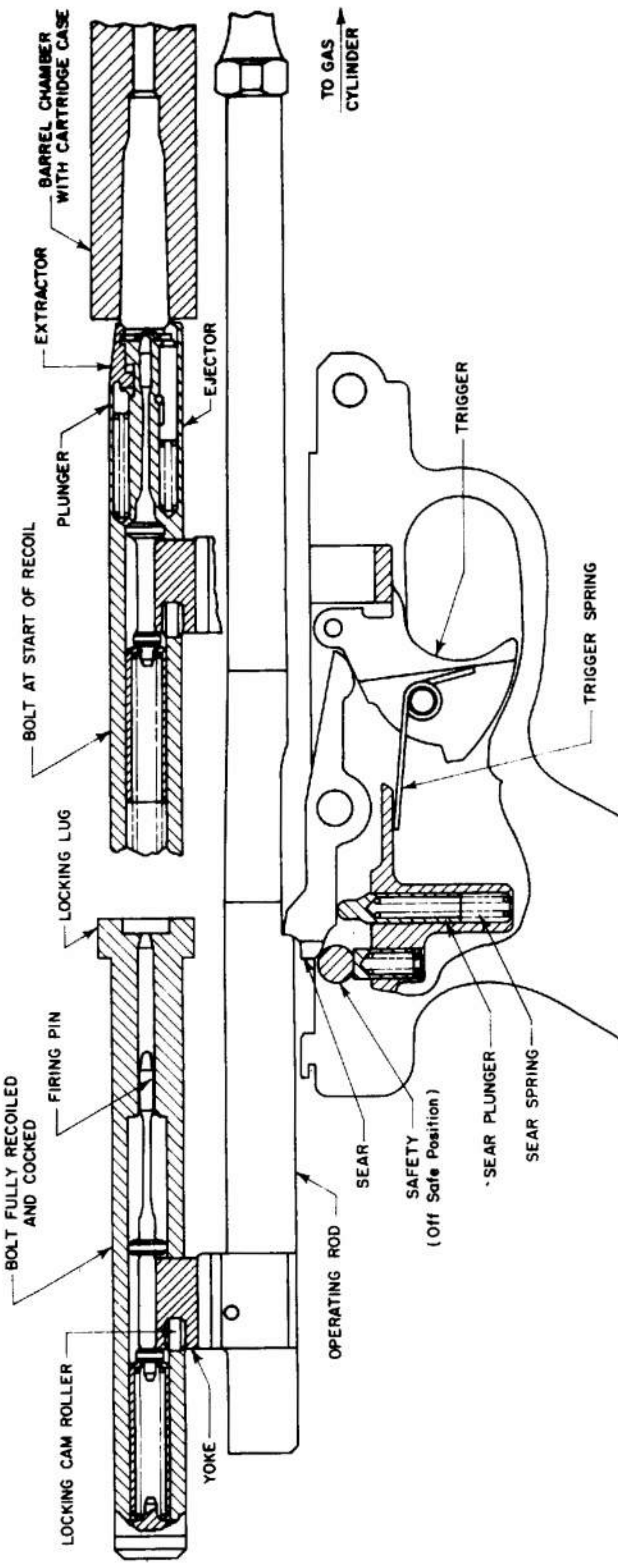


Figure 7–27. Firing Mechanism for Gas-operated Machine Gun

clearance and time for the sear to engage the notch properly thereby reducing the possibility of a prematurely released firing pin. A spring forces the sear upward into the latching position. As the bolt counterrecoils, the cocking lever continues to hold the firing pin in its most rearward position until the lever is rotated by the actuator to free the firing pin and permit it to slide forward a short distance to engage the sear notch. This action is completed when the recoiling parts are almost in battery and just short of the position where the sear passes beneath the trigger.

This firing mechanism is essentially one for automatic operation. Although adaptable to other types, this firing mechanism was designed for a recoil-operated gun, a type whose firing rate is largely determined by the inertial properties of the recoiling parts. Another limit on the firing rate is imposed by the sear spring. Since the spring force must be compatible with trigger pull, the spring may not have the capacity of lifting the sear into latching position before the cocking lever tends to release the firing pin if the bolt is moving too fast. In this event, the unrestrained firing pin will follow the cocking lever and lose the effectiveness of its spring, thus reducing the striking velocity on the primer. Should the firing pin velocity be lowered too much, the primer may not initiate, reducing the gun to inadvertent single shot operation. Planned single shot operation depends on the quick reflexes of the gunner to release the trigger before the sear hits the trigger bar during second round activity. However, positive single shot control is available by installing a bolt latch unit to the receiver. This unit latches to the recoiled bolt and retains it, thus interrupting the firing sequence until released manually. The interruption permits single shot firing.

Fig. **7–27** shows the type firing mechanism used in the M60, **7.62** mm Machine Gun. Two views of the bolt show the travel limits of it and the firing pin. Trigger and sear are shown only in the cocked position but their directions of motion and relative displacements when actuated are readily visualized in the sketch. The sear functions through the notch in the operating rod by holding rod, bolt, and all associated moving parts in the cocked position – the bolt being fully retracted except for the length of buffer travel. Here the sear engages the sear notch to stop all further counterrecoil progress. The sear pivots on a pin and is held in the cocked position by the trigger on one end and by sear plunger and safety on the other. All three are spring loaded. Trigger travel is limited on either end by a fixed limit stop. When the trigger is squeezed, it lifts its end of the sear and depresses the other end against (1) the resistance of sear and safety springs and **(2)** the frictional resistance induced by the driving spring between sear and sear notch. As the sear clears the sear notch, the driving spring forces the operating rod, bolt, and firing pin forward; closing the bolt and firing the round.

A yoke connects the operating rod to the bolt and firing pin. It is fastened rigidly to the operating rod but rides in a cammed slot in the wall of the bolt and cradles the firing pin. As the rod moves, the yoke carries the bolt in the same direction. This action is described in detail in par. **4–3.1.2.** Relative linear motion between the yoke and bolt causes the firing pin to slide inside the bolt. Only linear travel of the firing pin is essential. Any angular motion between it and its adjacent components is inconsequential, contributing nothing to firing efforts. The firing pin rests in the saddle of the yoke, the two integral collars serving as force transmitters, guides, and retainers. As retainers, they prevent relative linear motion between yoke and pin. As a guide, the front collar helps center the firing pins. As force transmitters, the front collar serves during firing activity whereas the rear one serves during retraction as well as the transmitter of the firing pin spring force. The firing pin opening becomes useful after the locking lug engages the lock to provide the necessary external reaction. This arrangement augments the driving spring effort in maintaining a counterrecoil velocity and, subsequently, a firing pin velocity conducive to rapid bolt closing and primer initiation. Designed for automatic operation only, this mechanism continues to fire as long as the trigger is held depressed. When released, all elements return to the cocked position as the sear catches the operating rod during the early part of counterrecoil to stop further firing activity. Afterwards the firing mechanism may be put on safe by rotating the safety until its plunger is seated to establish a rigid link between sear and trigger housing. No matter what position the trigger now assumes, the rest of the firing mechanism is firmly locked to eliminate accidental firing.

Fig. **7–28** illustrates a method whereby firing control is achieved by a three-position lever: the first for automatic, the intermediate for semi-automatic, and the third for putting the gun on safety. The safety is a secondary lever integral with the selector lever. It bears against the sear, holding that component firmly in the slot. When in this position, any pull on the trigger will not disturb the bolt. In either firing position, the safety swings free of the sear and offers no further interference.

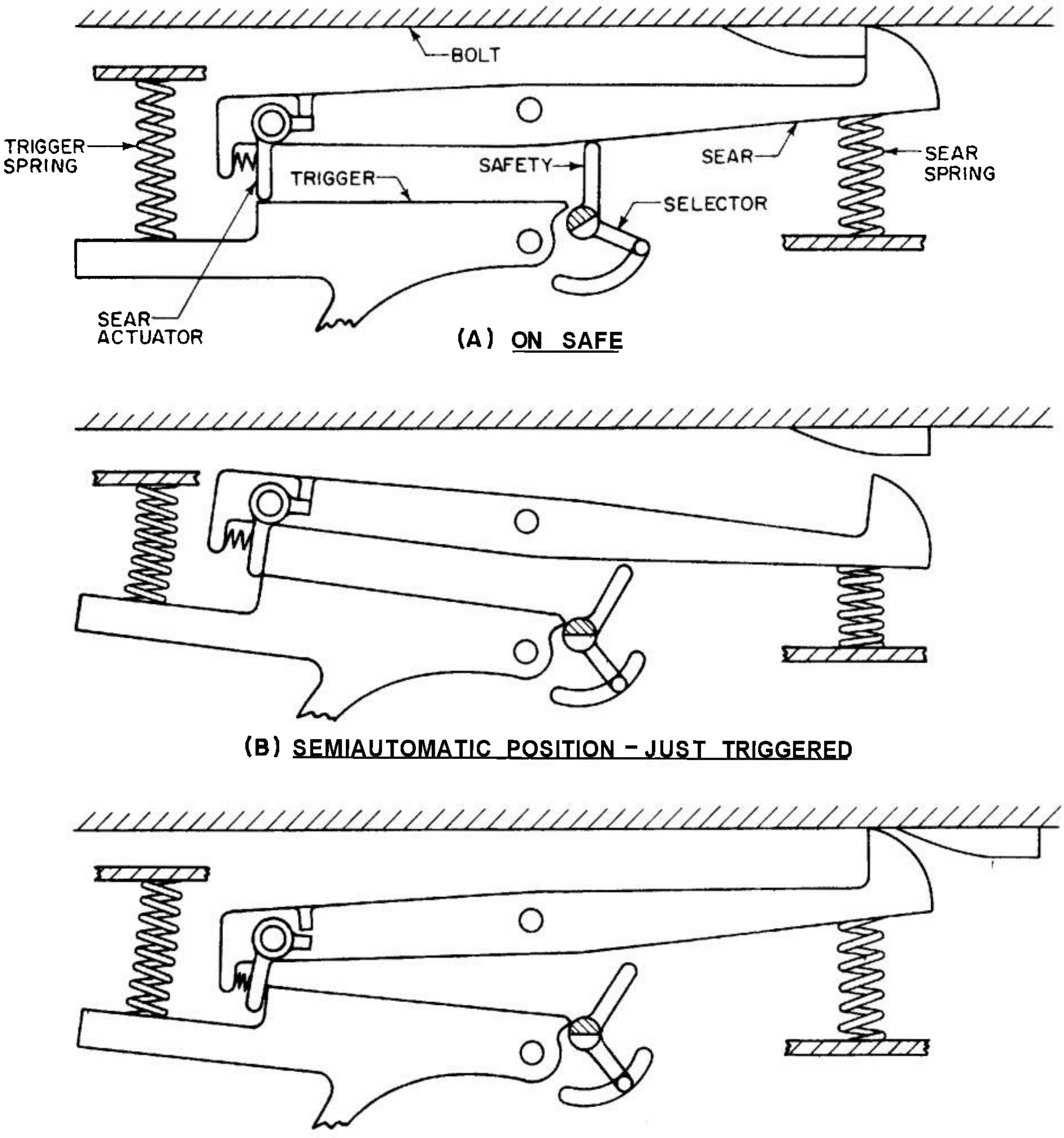
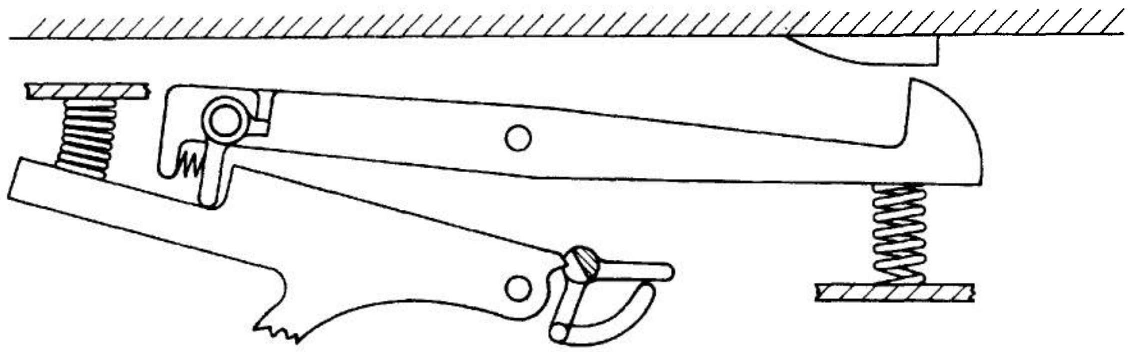


Figure 7–28. Three-position Firing Mechanism (1 of 2)



(D) AUTOMATIC POSITION – SEAR HELD DEPRESSED

Figure 7–28. Three-position Firing Mechanism (2 of 2)

With the selector in the semiautomatic position and as the trigger is being pressed, the selector pushes the actuator upward to rotate the sear and release the bolt. As trigger and sear rotate, the actuator moves rearward on the trigger and eventually slips off the stop to release the sear and permit it to resume its normal position and latch the bolt as it begins to close again–the depressed trigger meanwhile being limited in its movement by the selector. Before the next shot can be fired, the trigger must be released so that the sear actuator too can assume its original cocked position.

Automatic firing is achieved by turning the selector until the trigger can clear it entirely and sweep through the semiautomatic position. The advanced trigger position continues pressure on the sear actuator, thereby, holding the sear in its uncocked position leaving the bolt free to travel at will in either direction, continuing to fire until the trigger is released.

7–4.1.1 Trigger Pull

Computing the trigger pull is primarily an exercise in statics. Fig. **7–29** represents a typical triggering mechanism showing the applied loads on the various links. The trigger pull is found by balancing the moments about S, the pivot of the sear, and therefore, resolving the reaction between sear and trigger. **This** reaction – when applied to the trigger as **a** load – and the effects of the trigger spring, determine the trigger pull by balancing the moments about 0, the pivot of the trigger. Balance moments about S.

$$1.28 R_t = 0.84 F_s + 1.30 F_{sv} + 0.13 F_{sh} + 0.46 F + 1.02 \mu F$$

$$= 13.4 + 26.0 + 2.6 + 12.0 + 2.7 = 56.7 \text{ lb-in}$$

where

F = 26 lb, driving spring force

μF = 2.6 lb, frictional resistance at sear notch

F_s = 16 lb, sear spring force

F_{sv} = 20 lb, safety spring force

F_{sh} = 20 lb, horizontal component of safety spring force

R_t = trigger reaction on sear

μ = 0.10, coefficient of friction

The trigger reaction on the sear is

$$R_t = \frac{56.7}{1.28} = 44.3 \text{ lb}$$

Balance moments about 0,

$$1.06 P_t = 0.23 R_t + T + 0.63 F_t$$

$$= 10.0 + 1.0 + 0.6 = 11.6 \text{ lb-in.}$$

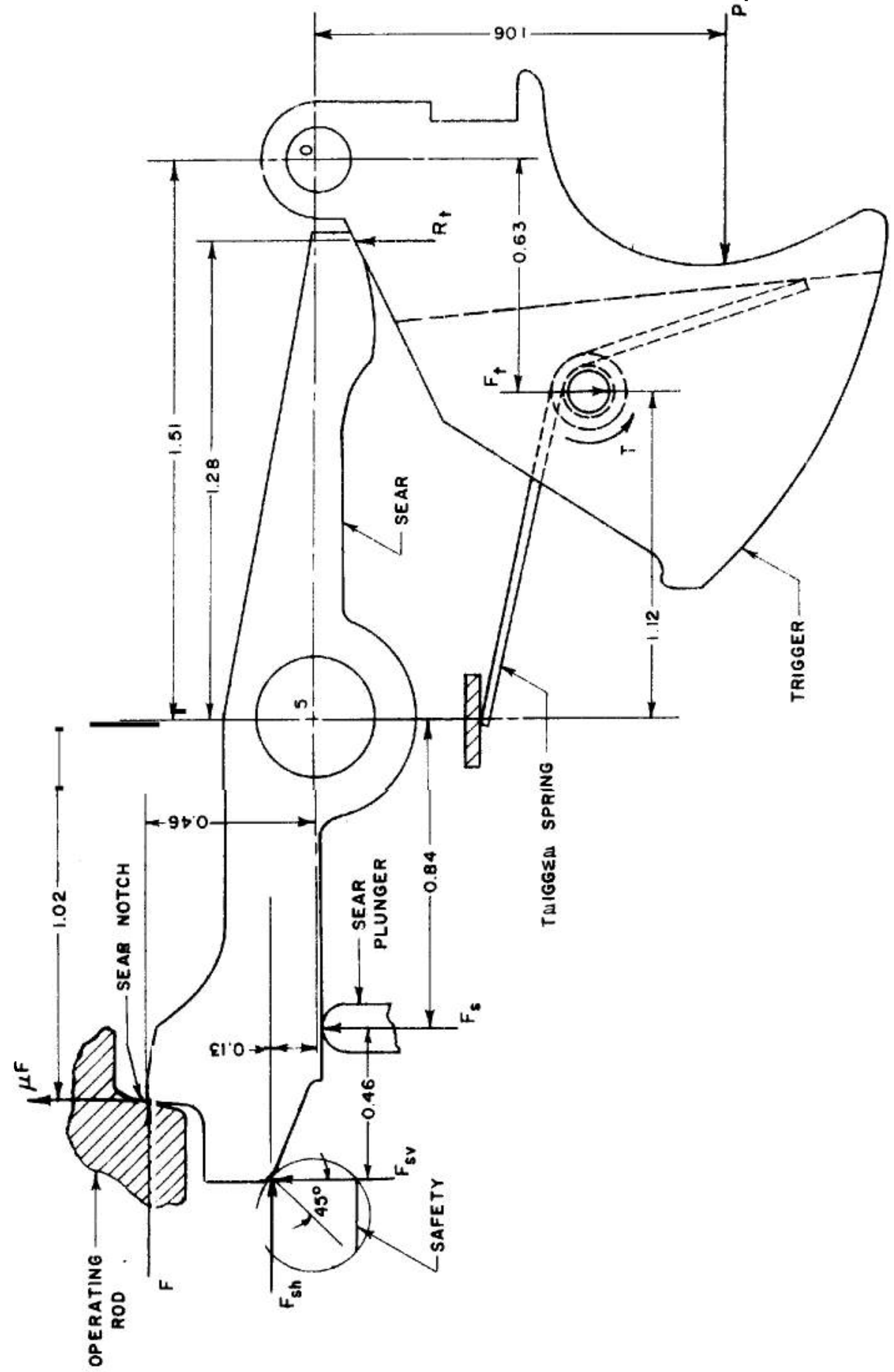


Figure 7–29. Triggering Mechanism Loading

where T = 1.0 lb-in, applied torque of trigger spring, and P_t is the trigger pull.

$$F_t = \frac{T}{1.12}$$

$$= 0.89 \text{ lb, vertical reaction of trigger spring pin}$$

The trigger pull becomes

$$P_t = \frac{11.6}{1.06} = 10.9 \text{ lb}$$

7–4.1.2 Firing Pin Design

Design criteria for firing pins are published elsewhere[22] but two basic requirements are essential for all percussion primers. A minimum amount of energy must be transmitted from firing pin to primer at a minimum striking velocity of 7 ft/sec. The energy is specified in inch-ounces. An upper limit of striking velocity also is specified to avoid puncturing the primer cap. Specifying both energy and velocity removes considerable control over the dynamics of a mechanism; control that normally should be available. For a given firing pin energy, the corresponding striking velocity is

$$v = \sqrt{\frac{2E}{M_e}} \qquad (7-50)$$

where

E = energy available

$M_e = \frac{W_e}{g}$ = equivalent mass of the moving parts

W_e = equivalent weight of the moving parts

A compressed coil spring provides the energy.

$$E = \frac{\epsilon (F_o + F_m)}{2} x \qquad (7-51)$$

where $F_m = F_o + Kx$, maximum spring force (in initial position)

F_o = minimum spring force (in final position)

K = spring constant

x = length of travel

ϵ = spring efficiency

The time elapsed during firing pin action according to Eq. 2–27

$$t = \sqrt{\frac{M_e}{\epsilon K}} \operatorname{Cos}^{-1} \frac{F_o}{F_m} \qquad (7-52)$$

Because M_e is generally small and E relatively large, the striking velocity v will be large. If v exceeds safe limits, the energy should be reduced to its lower limit and the weight of the firing pin increased to proportions that are compatible with good design. Table 7–2 lists various combinations of design parameters and how they affect the velocity and time. The firing pin energy will be held constant at E = 60 in.-oz. The efficiency of the firing pin spring system is also a constant at ϵ = 0.80. By holding the equivalent weight constant and varying the spring characteristics to be compatible with the distance, the time interval increases with respect to distance but the terminal velocity remains constant. But when weight varies and distance is constant, the time increases while terminal velocity decreases. A review of the data in Table 7–2 indicates a wide latitude in spring selection exists for any given firing pin weight. The tabulated data also show that the striking velocity can be lowered only by increasing the firing pin weight. A word of caution should be introduced here. An increase in weight may not be helpful because the vibration of the firing pin mechanism may be out of phase with the mechanical action. Past experience has proved that correcting this type of disorder can be achieved only by reducing the weight of the firing pin; altering the spring characteristics was not effective.

7–5 LINKS

Early machine gun ammunition belts were made of cloth fabric but the susceptibility of cloth to adverse climatic conditions led to its replacement by the modern metallic link belts. The metal belts consist of many links joined in series by some type of mechanical fastener, such as a pin. Many belts use the rounds themselves as pins. In addition to being able to survive most climatic conditions, the metal links have other desirable characteristics, two of which are: (1) the strength needed to transmit the high accelerations imposed by the loading devices of rapid fire guns, and (2) the ability to extend belt lengths quickly by merely joining the last link of one to the first link of another belt.

TABLE 7–2. FIRING PIN DYNAMICS

W_e, oz	x, in.	K, lb/in.	F_o, lb	F_m, lb	t, sec	v, in./sec
0.5	0.50	13.5	6.00	12.75	0.00296	304
1.0	0.50	13.5	6.00	12.75	0.00418	215
1.5	0.50	13.5	6.00	12.75	0.00513	175
2.0	0.50	13.5	6.00	12.75	0.00592	152
0.5	0.75	8.0	3.25	9.25	0.00445	304
1.0	0.75	8.0	3.25	9.25	0.00629	215
1.5	0.75	8.0	3.25	9.25	0.00771	175
2.0	0.75	8.0	3.25	9.25	0.00890	152
0.5	1.00	3.5	2.94	6.44	0.00590	304
1.0	1.00	3.5	2.94	6.44	0.00834	215
1.5	1.00	3.5	2.94	6.44	0.01022	175
2.0	1.00	3.5	2.94	6.44	0.01180	152

7–6.1 TYPES OF LINK

There are three general types of link: the old or extracting type, and the new push through and side stripping types. The extracting type has its round gripped in the cannelure of the cartridge case base and then pulled rearward from the link. When completely withdrawn, the round is lowered into the bolt path and rammed, by the bolt, into the chamber. The push through type depends on a rammer or bolt to push the round directly through the link toward the chamber. The round in the side stripping type link is forced out by applying a force, usually by cam action, perpendicular to the axis of the round. After leaving the link, the round continues its sidewards path until in line with rammer or bolt.

7–6.2 DESIGN REQUIREMENTS

Fig. 7–30 shows a link that may fit any of the above three categories. It components consist of two retaining loops, a connecting loop, and a retaining arm. The retainer loops grip the cartridge case and hold it firmly with respect to any lateral motion between round and link. The retaining arm prevents longitudinal relative motion between round and link. The connecting loop fits loosely over the preceding round to preserve the continuation of the belt. For this link configuration, the rounds are analogous to pins in a chain. Clearances between connecting loop and cartridge case determine the amount of free flexibility in an ammunition belt. The attachment between the connecting loop and the retaining loops, if not rigid, also lend a degree of free flexibility to the belt. ***Free flexibility*** is the flexing of the belt so that it will assume a fan-like position or form a helix, made available by taking up the slack provided by the accumulated clearances in all the links. Its counterpart, ***induced*** or ***forced flexibility***, may be either helical or fanning but the deflection is derived from the elastic deflection within the individual links. Allowable induced flexibility is determined experimentally; helical by measuring the torque necessary to twist the belt through a given angle, and fanning by fixing one end of the belt in a guided circle and hanging a weight on the other end. Either type of induced flexibility must always perform within the elastic range of the link material.

The ammunition belt may assume two positions for fanning flexibility, the nose fanning of Fig. 7–31 where touching is not permissible, and the base fanning of Fig. 7–32 where touching is permissible. Only free flexibility is represented since the ends of the belts are not constrained which is necessary to induce elastic deflection. Fig. 7–33 shows the geometry of two adjacent rounds in a base fanned belt. Free helical flexibility is shown in Fig. 7–34. Another type of belt configuration involves the fold radius. When the connector is a loop over the case (see Fig. 7–30), the linked round can rotate through a complete circle except for the interference of the adjacent round. Thus when a belt of ammunition in 7.62mm M13 Links is housed in

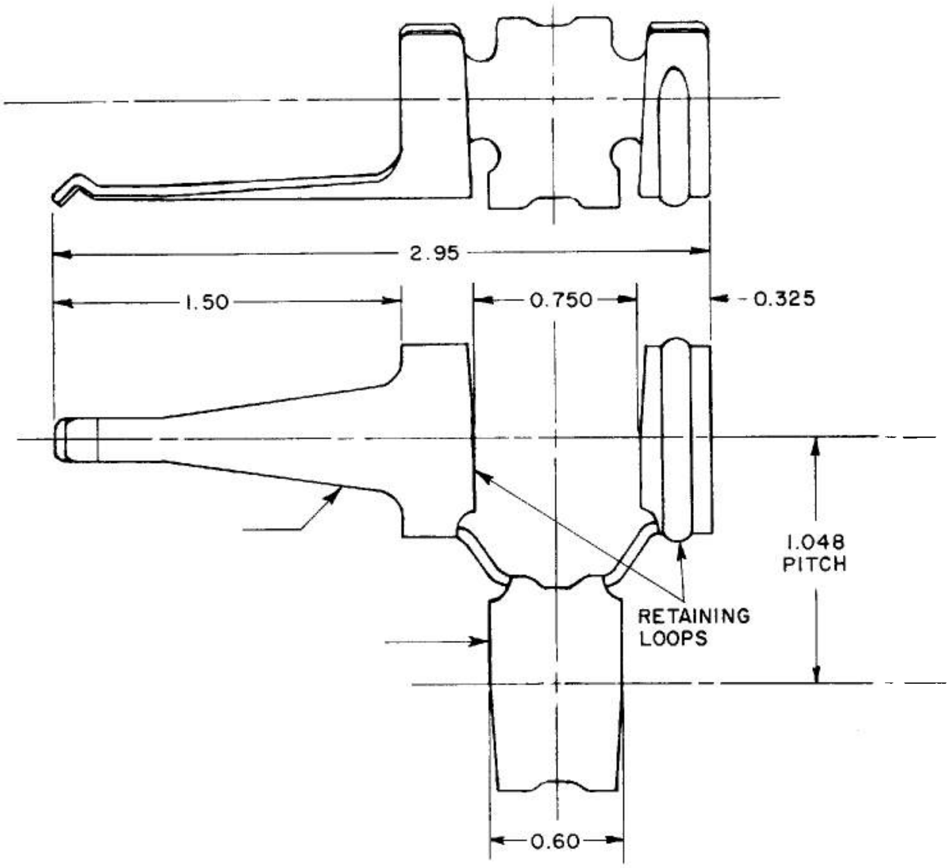


Figure 7–30. Ammunition Link, Cal .50 Round

a magazine or storage container (Fig. 7–35), little space is wasted since the belt can be stacked in horizontal rows. When the connector does not wrap around the case but instead merely joins the retaining loops of adjacent links, the rotation of one round about its neighbor is severely limited since the rotation center is near the case surface instead of being at the axis of the round. Ammunition belts made of links with this type connector will have some waste space when stored (see Fig 7–36 for 7.62 mm links). Another type connector, called a connecting member, operates similarly to a universal joint, i.e., it permits rotation between links about two perpendicular axes. All belts, so equipped, have unlimited free flexibility.

Initial link design is based on past experience. Belt strength is the most important requirement, to be followed closely by retention capability. Forces imposed on the belt are determined by the type of feed system (drum, chute, magazine) and the feed accelerations. Deflections in the ,links and, therefore, in the belt are not necessarily objectionable provided that round retention is maintained. Any variation in pitch due to belt stretch is corrected by the holding pawl which insures constant pitch and, therefore, proper feeding. Later, the feeding pawl controls the round as it is extracted from the link.

Usage determines the configuration and type of link. If the belts are to be discarded after firing, *disintegrating* ones are used where the link drops from the belt immediately after the round is stripped from it, or is forced from the belt by the ejected cartridge case. If the belts are to be retained, then maintaining the empty belt as a unit may be desired. Open links may also be desired. They are good for camming out the round but are relatively poor with respect to belt strength, and round retention can be a problem. For this reason, tolerances are small, to insure a reasonable consistency in retention loads. Should these loads prove to be too high, lightening holes (see Fig. **7–34**) are made to provide more flexibility and less snap-in force when joining two belts. The *snap-in* technique is superior to and preferred over the older *push-through* technique. In contrast to

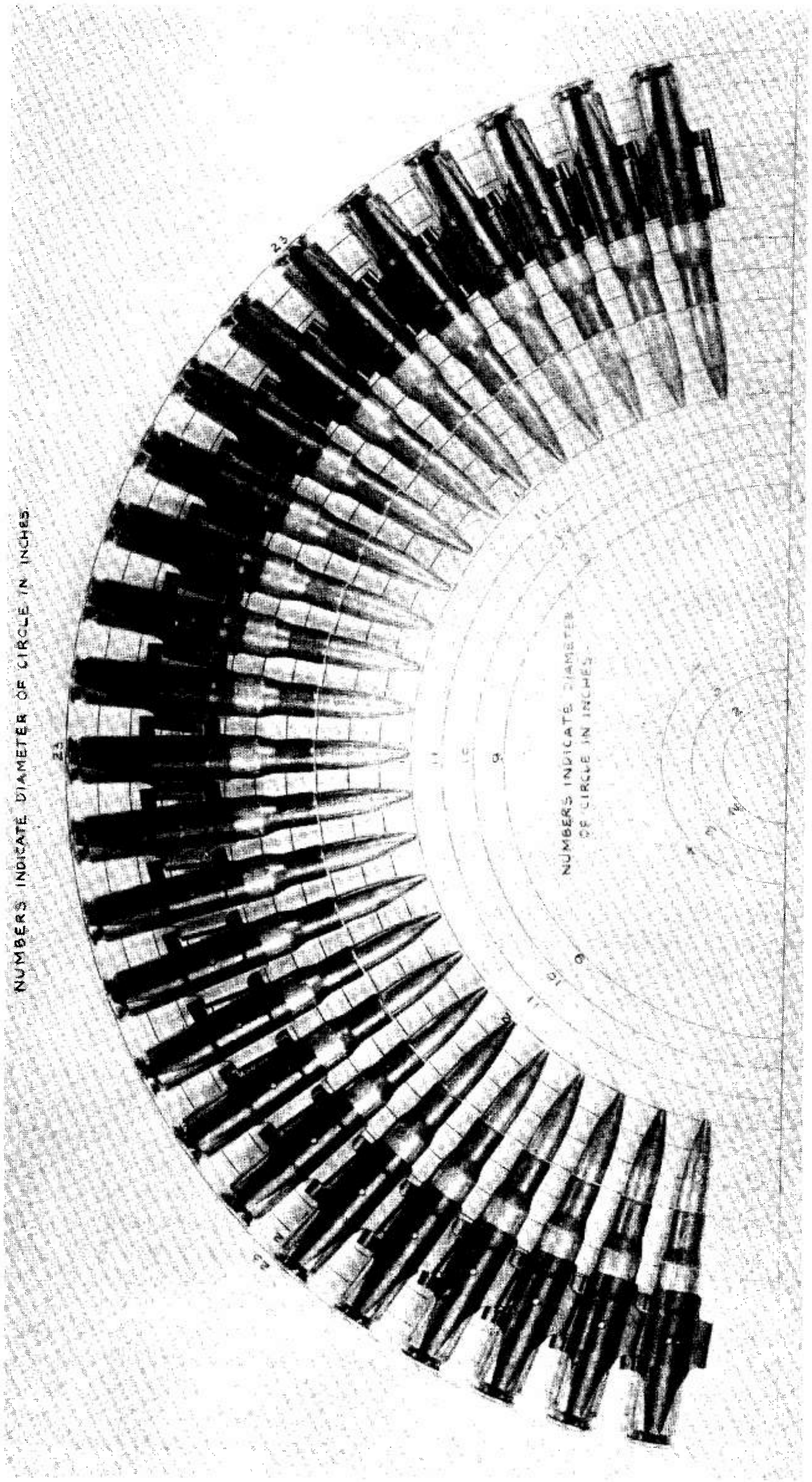


Figure 7–31. Nose Fanning Flexibility, 7.62 mm Link

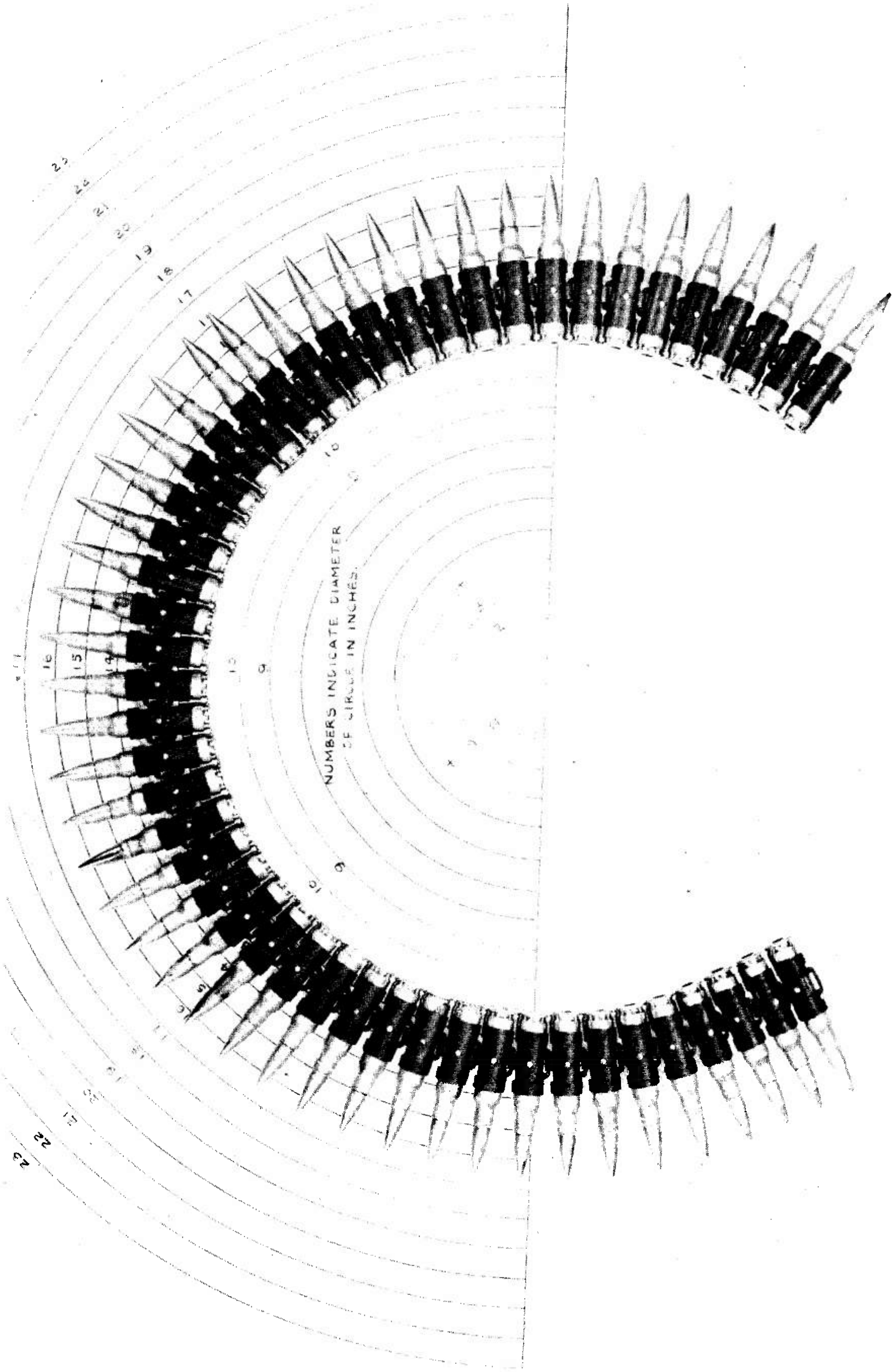


Figure 7–32. Base Fanning Flexibility, 7.62 mm Link

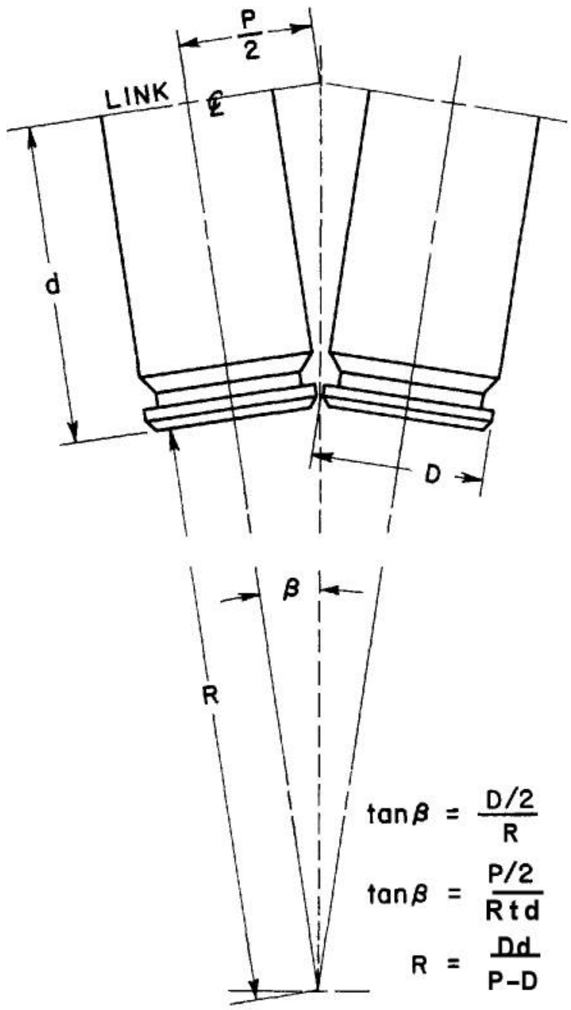


Figure 7–33. Geometry of Base Fanning

open links, closed loop links provide excellent belt strength and retentivity. Extracting the round may be a bit more troublesome than that experienced for the open loop but if a removable cover is used to complete the loop, an effective open loop link can be had by first stripping off the cover thus exposing.the round to the extracting mechanism.

A unique means has been devised to protect electric primers from being initiated inadvertently. This hazard is usually associated with aircraft since radiation emanating from communication (radio), detection (radar), and fire direction (usually on shipboard) facilities are generally associated with air terminals. A compatible antenna, such as a screwdriver, with a different potential in the established radiation field, may induce a spark at the primer to initiate it. This hazard is primarily a ground handling one which is most prevalent during loading, connecting, or breaking ammunition belts. Effective protection is available through the use of a *RADHAZ* (radiation hazard) shown in Fig. 7–37. It is merely an extension of the link bent over the primer to form a cover. The primer is thus shielded from any metal rod that is brought near it.

After the initial design or subsequent modification, pilot lots are made to determine acceptability. The links are stamped out in the annealed state, then heat treated. Extreme care must be exercised to hold the small tolerances. The pilot lots are tested in accordance with operating requirements. One of these is the catenary test to check retention under shock loads. If a free span of belt exists in the installation, a similar length of belt is lifted at midspan to a prescribed height and released to approximate belt whip. If the link is found wanting from this or any of the other tests, the design is modified to strengthen the weak areas, and the manufacturing and testing procedures repeated until an acceptable link evolves. Because of its trial and error nature and because of demanding manufacturing technique, long periods of time, in some cases more than a year, are devoted to designing and producing a successful link.

7–6 MOUNTS

Machine gun mounts are either fixed to vehicles or rest on the ground. Generally simple structures, mounts are adapted to the required limits of elevation and traverse and must be stable within these limits. Stability is readily achieved on vehicles by merely fastening the mount rigidly to the structure of the vehicle. But, if it rests on the ground, a mount such as the tripod type must depend on geometric proportions for stability. For this type, stability is a function of recoil force, command height, total weight, and length of the legs. If traverse is limited to the spread of the rear legs, the position of any given angle of elevation is that at zero traverse.

7–6.1 GEOMETRY AND RESOLUTION OF FORCES

Fig. 7–38 shows the forces involved in the side view projection. Take moments about A and solve for the reaction at B

$$R_b = \left[D_r (W + F_r \sin\theta) - HF_r \cos\theta\right] /L \qquad (7\text{–}53)$$

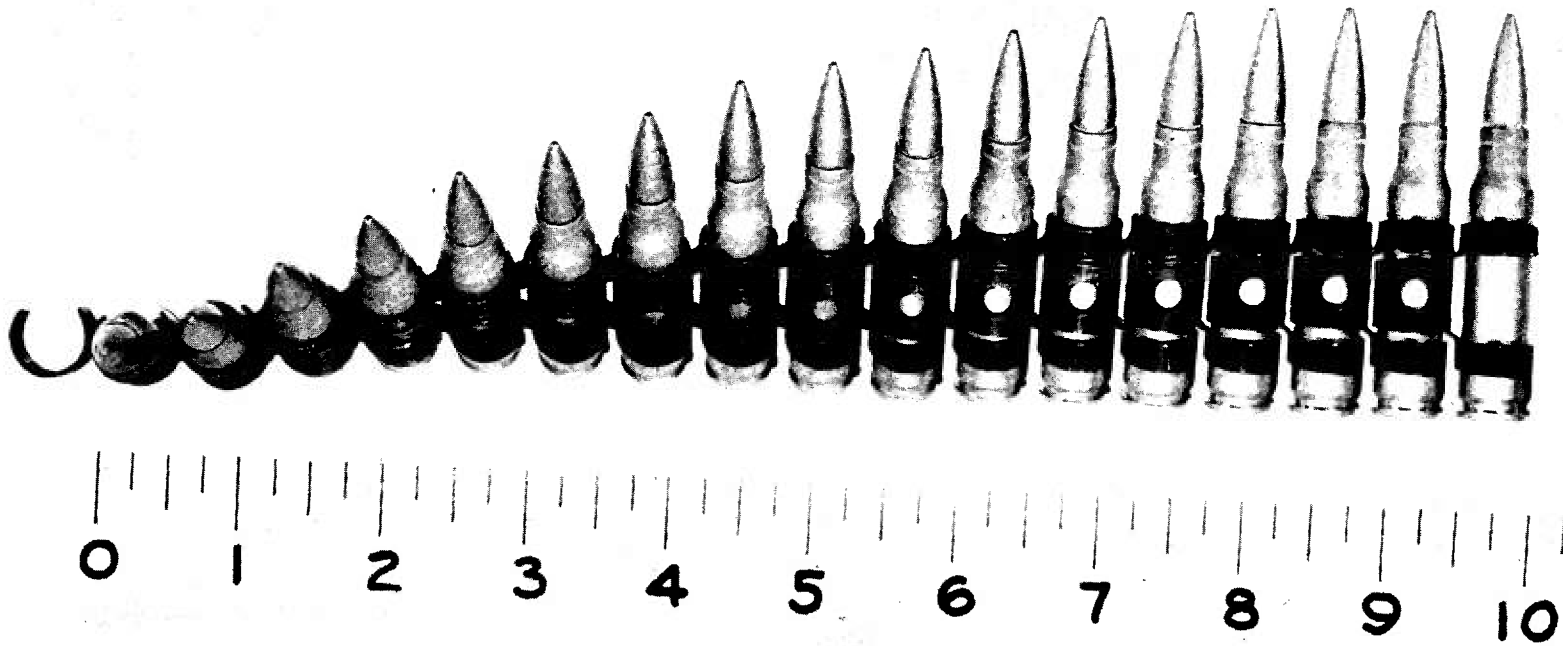


Figure 7–34. Helical Flexibility, 7.62 mm Link

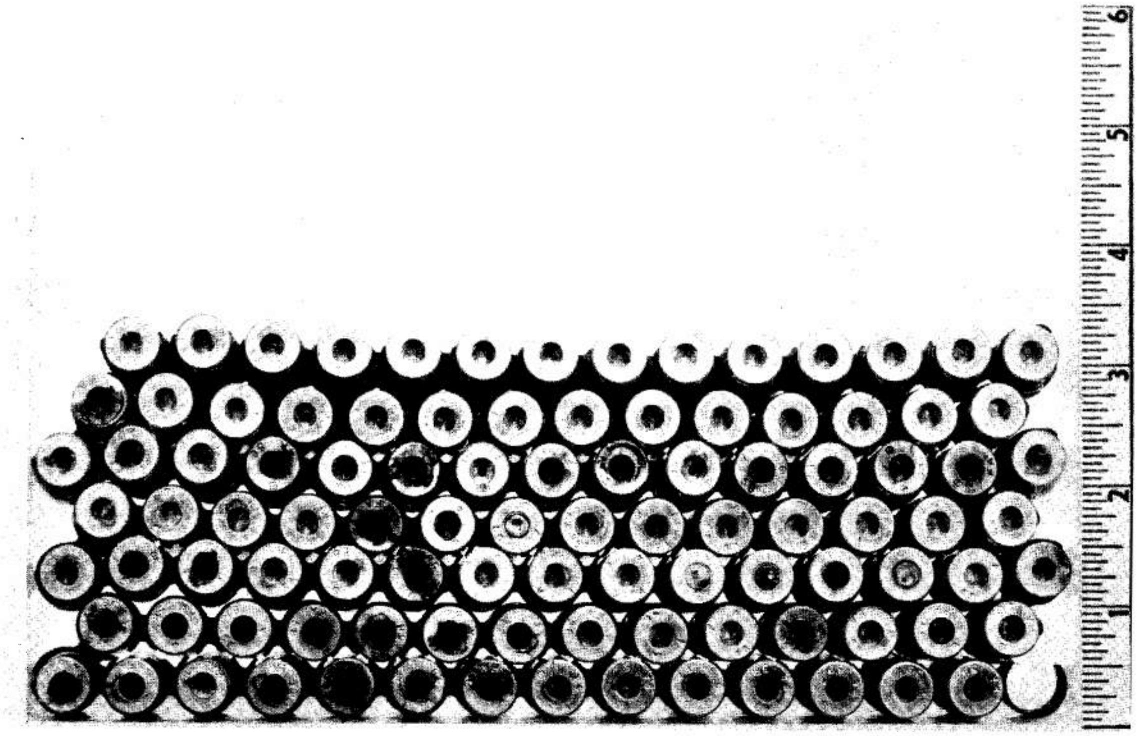


Figure 7–35. Total Folding 7.62mm Ammunition Belt

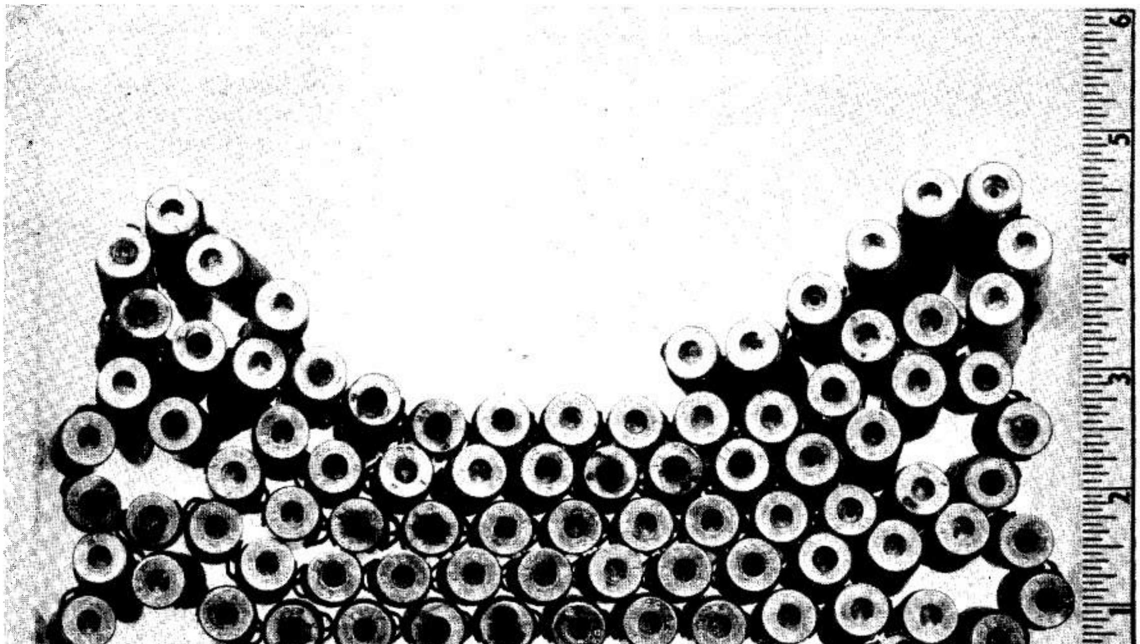


Figure 7–36. Partial Folding 7.62mm Ammunition Belt

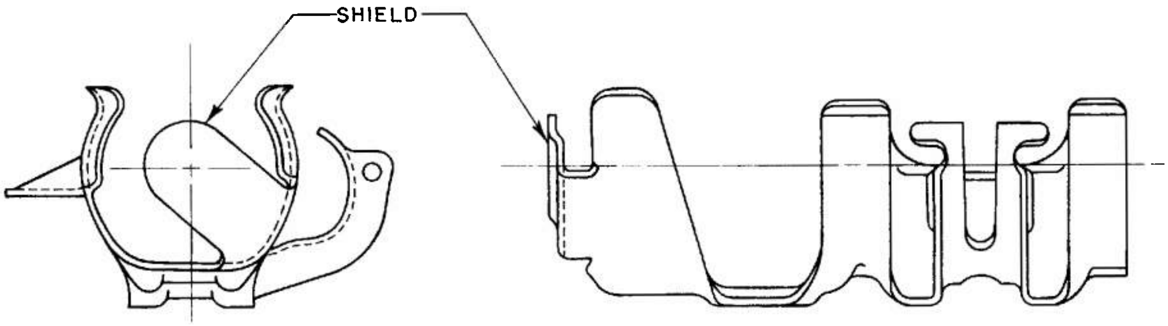


Figure 7–37. Loading Link With RADHAZ Shield

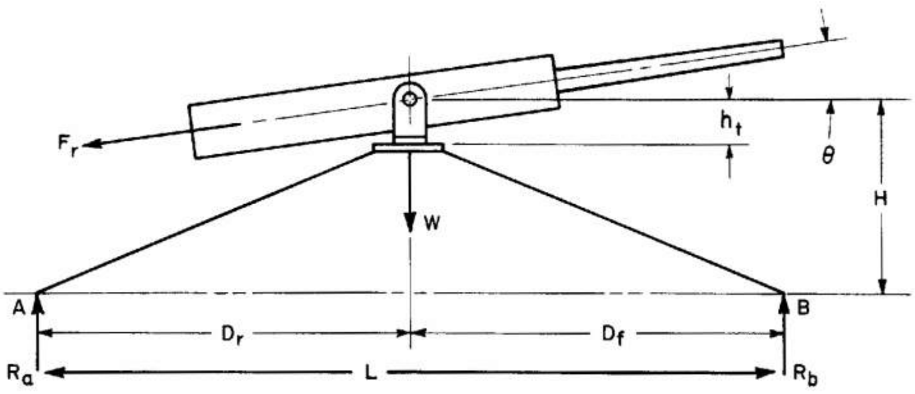


Figure 7–38. Loading Diagram of Mount

where D_r = horizontal distance between trunnion and rear support

F_r = recoil force

H = commandheight

L = distance between front and rear leg supports

R_a = reaction at rear support

R_b = reaction at front support

θ = angle of elevation

If R_a is positive or zero, the weapon is stable.

The recoil force F_r is assumed to be the average force during the recoil cycle. It may be computed by resolving the impulse-momentum characteristics of the recoiling parts. Add the expressions for the time recoil t, and counterrecoil t_{cr} of **Eqs. 2–23** and **2–27** for the total time of one cycle t_c period of oscillation.

$$t_c = \frac{\epsilon + 1}{\sqrt{\epsilon}}\sqrt{\frac{M_r}{K}}\ \mathrm{Cos}^{-1}\frac{F_o}{F_m} \qquad \textbf{(7–54)}$$

where F_m = spring force at end of recoil

F = spring force at beginning of recoil

K = spring constant

M_r = mass of recoiling parts

ϵ = efficiency of spring

The impulse on the recoiling parts induced by the propellant charge may be obtained by measuring the area under the propellant gas force-time curve or by computing the velocity of free recoil and then the momentum of the recoiling parts which is equal numerically to the impulse. The momentum of recoil

$$M_n = \left(\frac{W_r}{g}\right) v_f = (W_p v_m + 4700\, W_g)/g \qquad \text{(Ref. 23)} \tag{7-55}$$

where g = acceleration due to gravity

v_f = velocity of free recoil

v_m = muzzle velocity of projectile

W_g = weight of propellant charge

W_p = weight of projectile

W_r = weight of recoiling parts

4700 = empirical value of the propellant gas velocity in ft/sec, therefore, the other kinematic parameters must be dimensioned in ft and sec.

Since impulse is numerically equal to momentum, Eq. **2–14,** the average recoil force is

$$F_r = \frac{\int F dt}{t_c} = \frac{M_n}{t_c} \ . \tag{7-56}$$

The length of the rear legs extending from the spade to the intersection of leg and pintle is computed by first equating the weight and force moments about **A** at zero elevation.

$$D_r W = H F_r \tag{7-57}$$

The projected horizontal distance between spade and trunnion is

$$D_r = \frac{H F_r}{W} \tag{7-58}$$

The rear leg length becomes

$$L_r = \sqrt{D_r^2 + (H - h_t)^2} / \cos\phi \tag{7-59}$$

where h_t = distance between trunnion and pintle leg intersection

ϕ = half of the angular spread between the rear legs.

Half of the average force during the recoil cycle is assumed to be the applied forward acting force just as the cycle is completed. The distance needed to balance this forward upsetting moment with the weight moment is

$$D_f = \frac{H F_r}{2W} = \frac{1}{2} D_r \ . \tag{7-60}$$

The length of the front leg is

$$L_f = \sqrt{D_f^2 + (H - h_t)^2} \ . \tag{7-61}$$

The structural requirements of the legs, size, strength, and rigidity are satisfied through the usual procedure for computing stresses and deflections of an eccentrically loaded column of uniform cross section[24]. If the leg varies in cross section, the area moment of inertia of the cross section is a function of the distance, and the bending moment is a function of the distance and of the deflection. Unless some simplifying assumptions are made, the alternative rigorous analysis is performed most conveniently with a digital computer.

7–6.2 SAMPLE PROBLEM

Compute the recoil spring characteristics and lengths of the legs of a tripod mount for a gun having the following data:

H = 14 in., command height

h_t = 5 in., height of trunnion above pintle

K = 2000 lb/in., spring constant (ring spring)

L = 0.5 in., length of recoil

v_m = 3000 ft/sec, muzzle velocity

W = 225 lb, estimated weight of weapon

W_g = 0.09 lb, weight of propellant charge

W_p = 0.2 lb, weight of projectile

W_r = 110 lb, weight of recoiling parts

ϵ = 0.50, efficiency of ring spring

θ = 0°, angle of elevation

2ϕ = 50°, spread of rear legs

From **Eq.** 7–55, the velocity of free recoil is

$$v_f = \frac{0.2 \times 3000 + 4700 \times 0.09}{110} = 9.3 \text{ ft/sec.}$$

The energy to absorbed during recoil is

$$E_r = \frac{1}{2}\left(M_r v_f^2\right) = \left(\frac{110}{64.4}\right) 86.49 = 147.73 \text{ ft-lb.}$$

The total average recoil force is

$$F_r = \frac{E_r}{L} = \frac{147.73 \times 12}{0.5} = 3546 \text{ lb.}$$

Since the efficiency of the spring, ϵ = 0.50, assists in stopping the recoiling parts, the actual average spring force is

$$F_{as} = \epsilon F_r = 1773 \text{ lb.}$$

But

$$F_{as} = \frac{1}{2}(F_o + F_m) = \frac{1}{2}(F_o + F_o + KL)$$

$$= F_o + \frac{1}{2} KL$$

$$F_o = F_{as} - \frac{1}{2} KL = 1773 - \frac{1}{2}\left(2000\right) 0.5$$

$$= 1273 \text{ lb.}$$

$$F_m = F_o + KL = 1273 + 1000 = 2273 \text{ lb}$$

According to **Eq.** 7–54

$$t_c = \frac{\epsilon + 1}{\sqrt{\epsilon}} \sqrt{\frac{M_r}{K}} \cos^{-1} \frac{F_o}{F_m}$$

$$= \frac{1.5}{0.707} \sqrt{\frac{110}{2000 \times 386.4}} \cos^{-1} 0.56005$$

$$= 2.122 \times 0.0119 \times 0.976 = 0.0246 \text{ sec.}$$

The average impulsive force during the recoil cycle, **Eq.** 7–56, is

$$F_r = \frac{M_n}{t_c} = \frac{110 \times 9.3}{32.2 \times 0.0246} = \frac{1023}{0.792} = 1292 \text{ lb.}$$

The projected horizontal distance of the rear leg, **Eq.** 7–58, is

$$D_r = \frac{HF_r}{W} = \frac{14 \times 1292}{225} = 80.4 \text{ in.}$$

The length of this rear leg, Eq. 7–59, is

$$L_r = \sqrt{D_r^2 + (H - h_t)^2}/\cos\phi = \sqrt{\frac{6545.16}{0.906}} = 89.3 \text{ in.}$$

The length of the front leg, **Eqs.** 7–60 and 7–61, is

$$L_f = \sqrt{D_f^2 + (H - h_t)^2} = \sqrt{1697.04} = 41.2 \text{ in.}$$

CHAPTER 8

LUBRICATION OF MACHINE GUNS

Conventional, good lubrication design practice is required in machine gun design. Excess, rather than insufficient, lubricant is to be avoided on most sliding parts. If the coat of oil or grease is too thick, dust will readily collect, cause excessive wear, and impede action sometimes to the extent of malfunction. Maintenance instructions stress this fact by emphasizing that all excess lubricant be wiped off all surfaces. Not all self-operating machine guns require reservoirs of lubrication. In some cases, the recoil adapter spring or the driving spring may be lubricated with a graphite grease. In electric or hydraulic-driven machine guns, the driving units are lubricated by applying grease or oil to the moving parts which are usually protected from exposure to dirt by their housings.

A well-designed machine gun is inherently a readily lubricated one, particularly if only a thin coating of lubricant is needed on the sliding part. The lubricant is usually applied after cleaning, which procedure follows after prolonged firing or during periodic inspection and maintenance. Because of this practice, emphasis is generally placed on the lubricant rather than on specific design practices that are controlled by lubrication requirements.

8–1 GENERAL CONCEPT

The machine gun designer must be cognizant of the lubrication requirements for the sliding surfaces of his design. His primary objective is smooth surfaces coupled with his acquaintance with available lubricants, and their general properties and uses. If lubricant properties and required lubricating properties are compatible, the designer's problem is solved. If a proper lubricant is not available, a search for one must be made or the weapon relegated to limited specific conditions, which normally is undesirable. Another alternative involves auxiliary equipment such as heaters to keep the viscosity level of the lubricant in an acceptable range.

The lubrication of machine guns is usually confined to applying a thin film of oil, grease, or other material to sliding surfaces with the expectation that it will last until the next general cleaning time. Military Specifications define in detail the properties of available lubricants. Substitutes are acceptable only after extensive tests prove that the new product has all the necessary properties. The Specifications enumerate all known data from preparation to delivery and storage. A general outline is prepared for illustration.

1. SCOPE. Type of lubricant, general usage, and operating temperature range.

2. APPLICABLE DOCUMENTS

   2.1 A list of Federal and Military Specifications supplementing the given specifications.

   2.2 Standards prepared by accepted private organizations.

3. REQUIREMENTS

   3.1 Qualification. The material must have passed qualifying tests.

   3.2 Material. The ingredients of the material must conform to specification.

   3.3 Physical and chemical requirements. Some of these are listed as flash point, pour point, viscosity at temperature limits, hydroelectric stability, oxidation stability, storage stability, rust prevention, gun performance, workmanship (homogeneous, clear, and with no visibly suspended matter).

4. QUALITY ASSURANCE PROVISIONS

   4.1 Specified inspection procedure.

   4.2 Specified tests.

8–2 EXAMPLES OF LUBRICANTS

Unless a smoother finish is required, an RMS (rough machine surface) of 16 to 32 pin. will provide proper sliding action when covered with a thin layer of lubricant. However, under extremely adverse conditions, the designer may be helpless to cope with the sliding surface preparation. A number of activities associated with machine gun fire at high altitudes demonstrates the difficulties experienced in attempts to eliminate malfunctions. These activities deal with lubricant rather than design.

During World War 11, high-flying airplanes had gun malfunction at temperatures below -20°F. This led to gun heaters, but the added weight and not complete reliability led to attempts to develop new lubricants that would correct the malfunctioning components'[7]. The investigation yielded success in three operations. Low temperature exposure at high altitude followed by condensation at warmer levels and again freezing after returning to high altitudes caused the triggering solenoid to become frozen in place. A free-moving solenoid was assured by spraying the unit with silicone oil to prevent the water condensate from collecting. The material is an open chain methyl silicone having a viscosity of 20 cSt at 77°F and 300 cSt at -65°F, and a pour point of -75°F. Dodecane phosphoric acid (0.1 percent by weight) was added for lubrication. This material was labeled NRL S-75-G interim. In the meantime, special attention to ammunition feeders led to "trouble-free lifetime lubrication" with the application of MIL-G-15793 (BuOrd) grease. Also, synthetic oil MIL-L-17353 (BuOrd) with 2 percent by weight of trecresyl phosphate for wear prevention was discovered to perform adequately for other moving parts of the gun.

In contrast, tests conducted with the Cal .SO M3 Machine Gun, to prove the reliability of removing gun heaters, when lubricated with PD 500 oil gave totally negative results'[8]. Remember that this oil made feasible the removal of heaters from the 20 mm M24A1 Gun without the gun malfunctioning at low temperatures. Apparently some inherent feature in each type of gun rendered acceptance and rejection in the particular weapon. Unfortunately, the tests were not sufficiently broad in scope to determine what design features were responsible.

A semi-fluid grease and an oil blend were developed with satisfactory performance at extreme temperatures for the M61 Multibarreled Gun'[9]. Test results indicate that either lubricant satisfied all requirements, but the semi-fluid grease had longer life and was therefore selected as the lubricant for the M61 Gun.

Dry lubricants are recommended for slowly moving parts with relatively few cycles of operation. Tests of 18 resin systems pigmented with molybdenum disulfide were tested[20]. A pigment-to-resin ratio of 9: 1 was found most effective. Epoxy-phenolic and epoxy-polyamide resin systems were best for both lubrication and storage stability.

8–3 CASE LUBRICANT

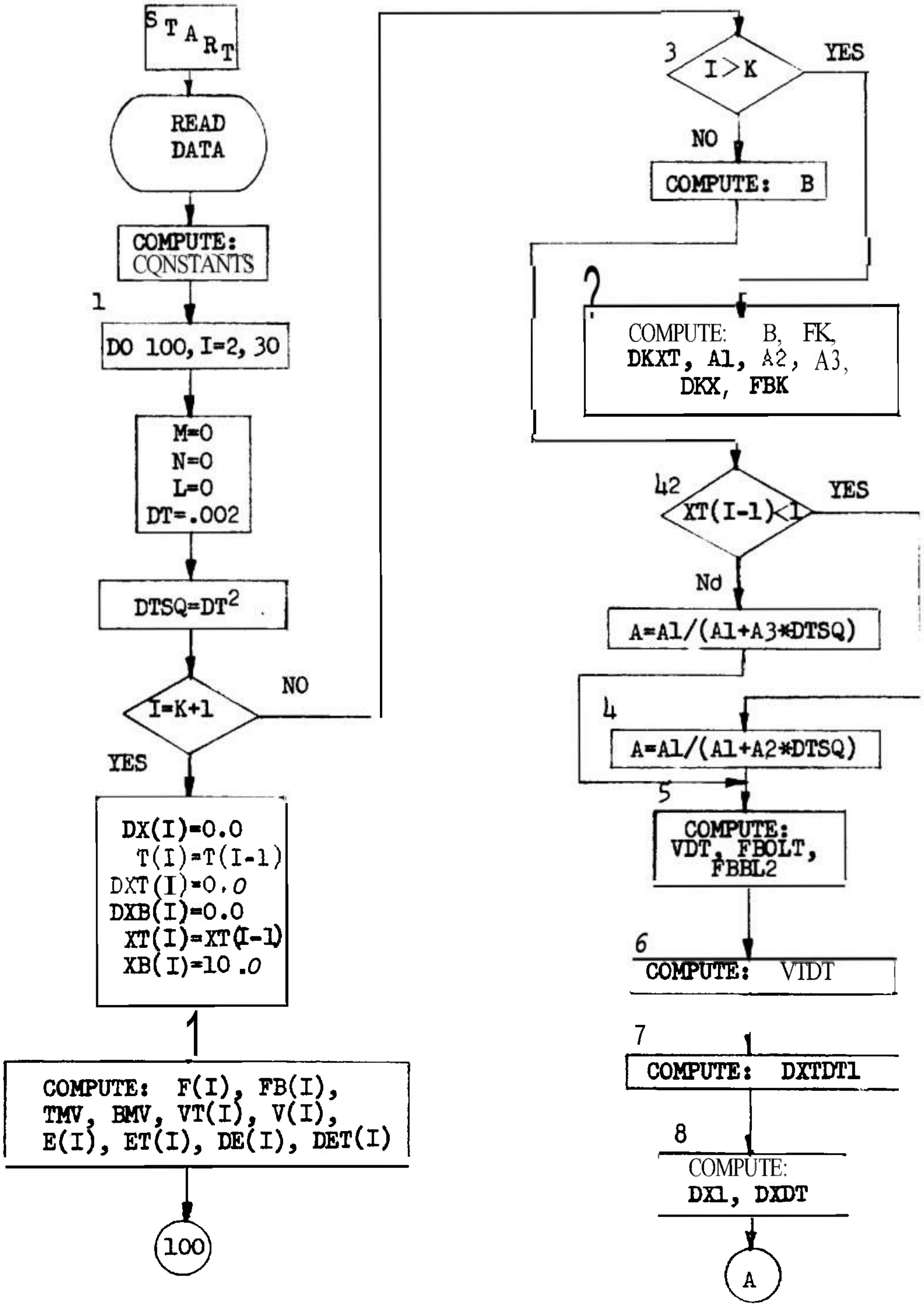
Although the gun designer is not directly involved with ammunition design, he is directly concerned with handling, loading, and extracting during firing. A smooth chamber is essential for extraction and a properly lubricated case is a decided asset. The lubricant should be a dry lubricant and should be applied at the factory. Considerable effort has been made to find suitable lubricants for this purpose. Some success has been achieved but continued search is still being advised, especially since two independent facilities are not in total agreement.

The Naval Research Laboratories conducted tests of brass and steel cartridge cases coated with films of polytetrafluoroethylene (Teflon)[28]. Results were outstanding in meeting required protection and lubrication properties. Laboratory results, later confirmed by firing tests, showed low friction and consequently less wear in gun barrels. Other desirable features include freedom from cartridge malfunction, no chamber deposits, decreased ice adhesion, and less chance of thermal "cook-off'. Teflon can be applied to steel and brass ammunition by mass production methods. Its protective ability permits prebelting and packaging of ammunition since no further handling prior to use is necessary. Its supply is abundant and its cost reasonable. Thus the use of Teflon in this capacity seems ideal.

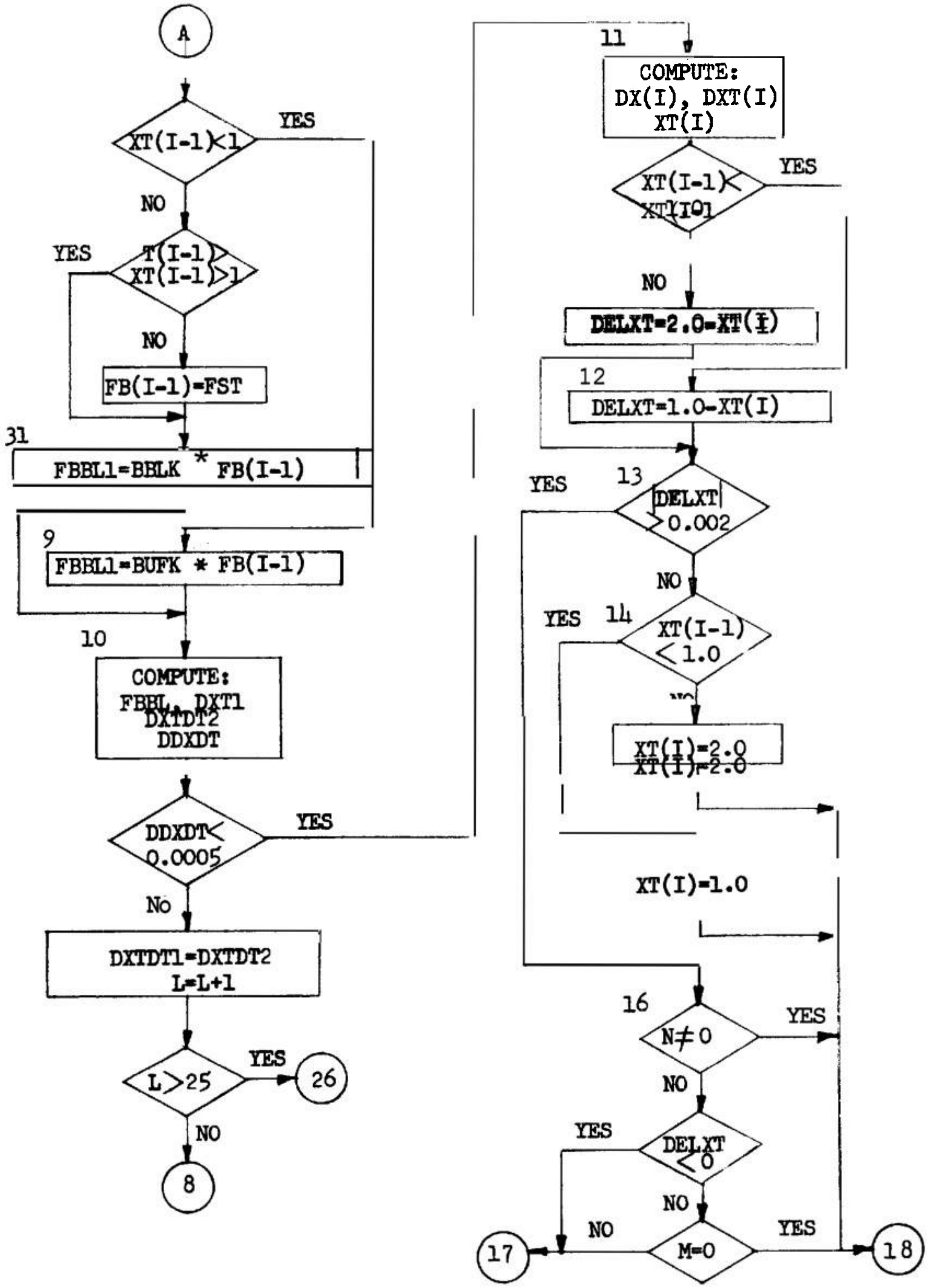
Aberdeen Proving Ground is more reserved in its appraisal of Teflon coating[21]. Whether or not the techniques of applying the coatings were similar, those used at APG were not free of coating defects; a high cull rate existed. When tested with cartridges coated with microcrystalline wax, ceresin wax, and uncoated ammunition; the Teflon-coated wax showed many advantages but was also found wanting in some respects. Teflon and micro-wax had better extraction properties and Teflon left a much cleaner chamber than the others; micro-wax was second best. About SO percent of the Teflon-coated cases had slight bulges after extraction; other types also were similarly damaged but with no apparent significance attached to a definite choice. For dusted ammunition, the Teflon and micro-wax were far superior to the other two types with Teflon having a slight advantage, although when fired in a comparatively rough chamber, Teflon was outperformed by all. Reiterating, the gun designer, aside from providing smooth sliding surfaces, is almost totally dependent on the physical properties of the lubricant to make his gun perform satisfactorily under all assigned conditions.

APPENDIX A

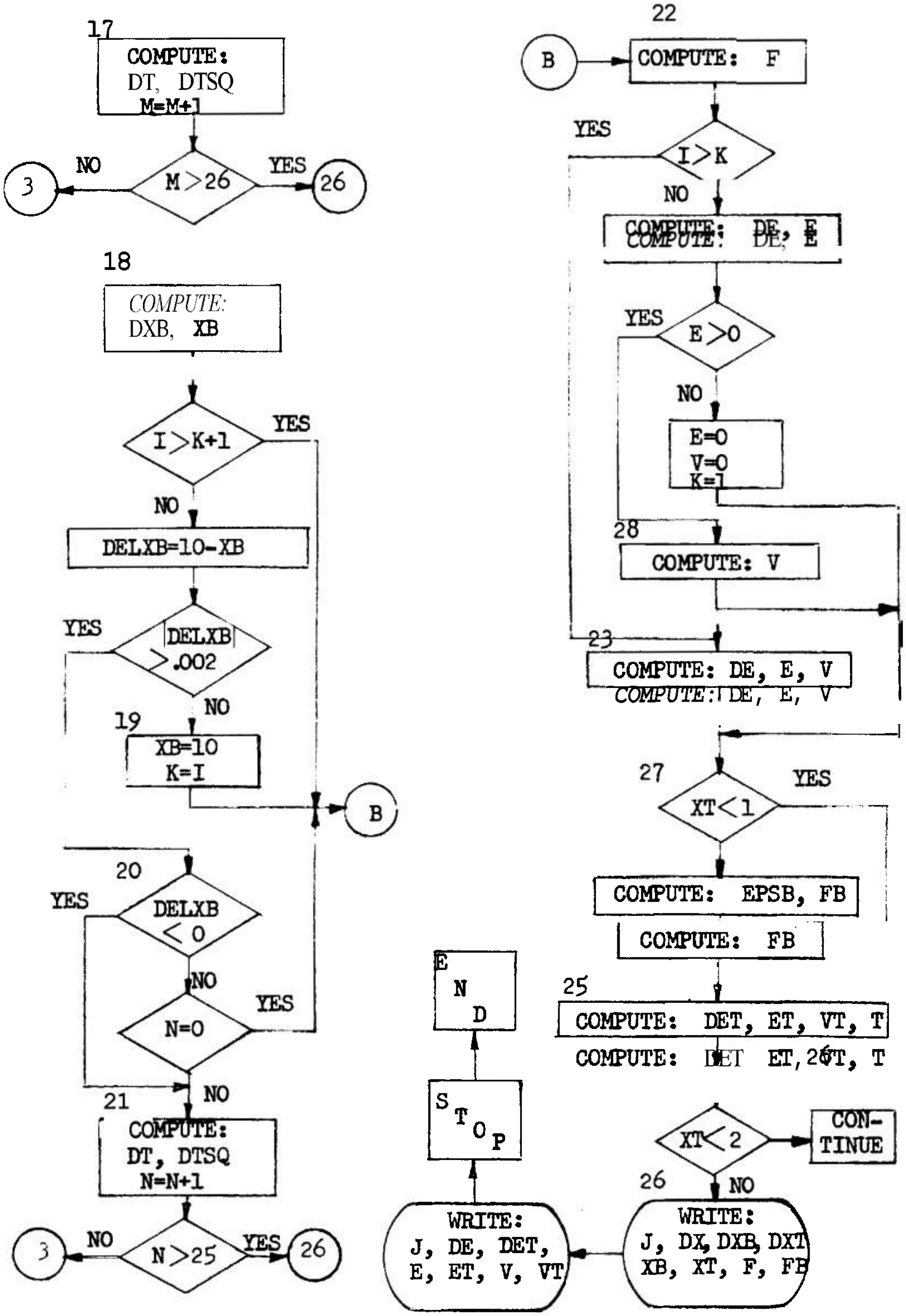
A-1. FLOW CHART FOR DELAYED BLOWBACK



A—1. (Con't.)


A
XT(I-1)<1
YES
NO
T(I-1)>
XT(I-1)>1
YES
NO
FB(I-1)=FST
31
FBBL1=BBLK * FB(I-1)
9
FBBL1=BUFK * FB(I-1)
10
COMPUTE:
FBBL, DXT1
DXTDT2
DDXDT
DDXDT<
0.0005
YES
No
DXTDT1=DXTDT2
L=L+1
L>25
YES
26
NO
8
11
COMPUTE:
DX(I), DXT(I)
XT(I)
XT(I-1)<
XT(I)
YES
NO
DELXT=2.0-XT(I)
12
DELXT=1.0-XT(I)
13
|DELXT|
>0.002
YES
NO
14
XT(I-1)
<1.0
YES
XT(I)=2.0
XT(I)=1.0
16
N≠0
YES
NO
DELXT
<0
YES
NO
M=0
NO
YES
17
18

A—1. (Con't.)


17
COMPUTE: DT, DTSQ M=M+1
M > 26
NO
3
YES
26
18
COMPUTE: DXB, XB
I > K+1
YES
NO
DELXB=10-XB
|DELXB| > .002
YES
NO
19
XB=10 K=I
B
20
DELXB < 0
YES
NO
N=0
YES
NO
21
COMPUTE: DT, DTSQ N=N+1
N > 25
NO
3
YES
26
22
B
COMPUTE: F
I > K
YES
NO
COMPUTE: DE, E
E > 0
YES
NO
E=0 V=0 K=1
28
COMPUTE: V
23
COMPUTE: DE, E, V
27
XT < 1
YES
COMPUTE: EPSB, FB
COMPUTE: FB
25
COMPUTE: DET, ET, VT, T
XT < 2
CON-TINUE
26
NO
WRITE: J, DX, DXB, DXT XB, XT, F, FB
WRITE: J, DE, DET, E, ET, V, VT
STOP
END

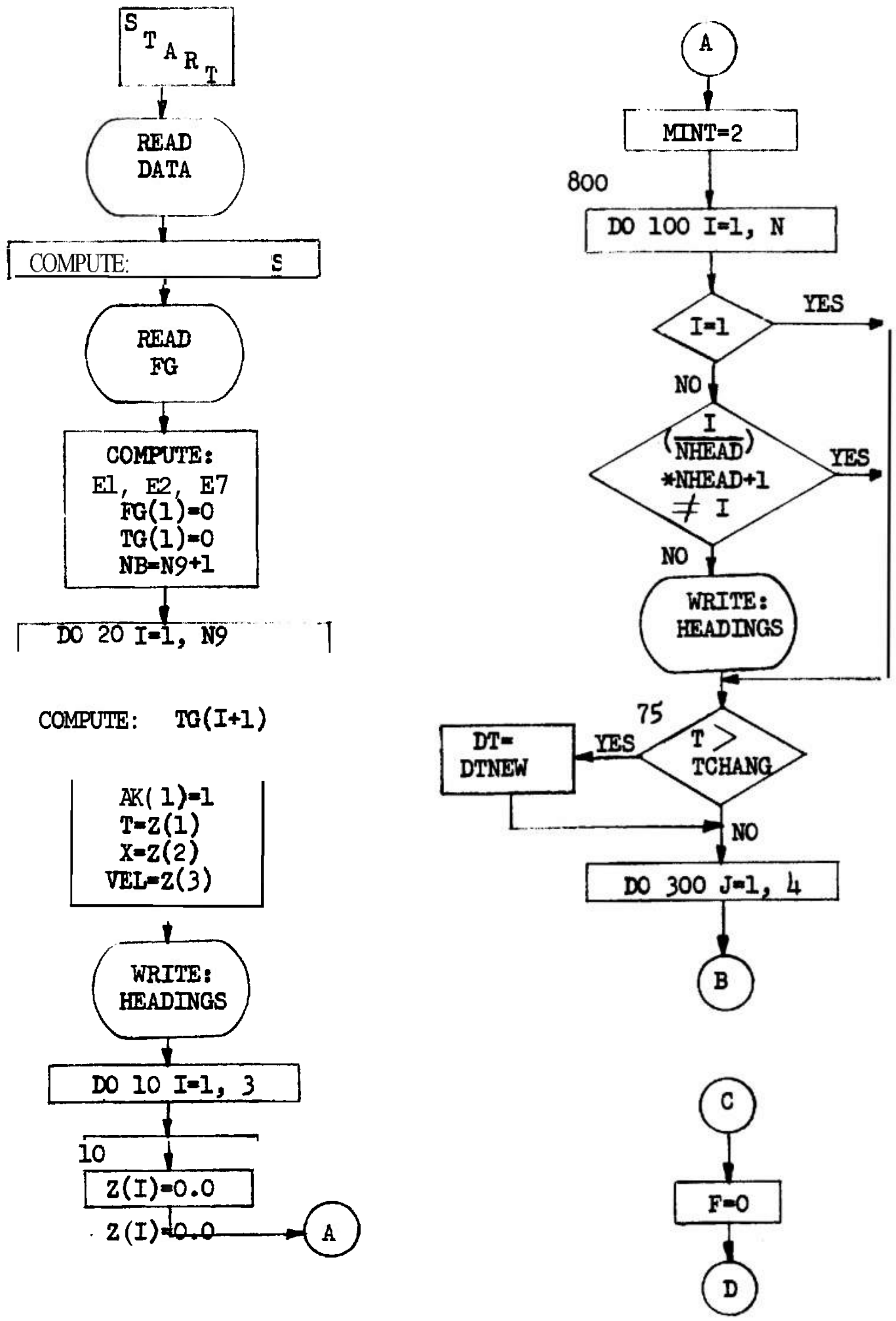
A–2. LISTING FOR DELAYED BLOWBACK PROGRAM

```
00101          DIMENSION  T(30), DX(30), DXB(30), DXT(30), XB(30), XT(30), F(30),     4
00101        1 FB(30), DE(30), DET(30), E(30), ET(30), V(30), VT(30)                 5
00103          READ(5,101) EPS,EPSB,EPST,A1,A2,A3,B1,B2,FK,DKXT,BUFK,                6
00103        1 BBLK, DKX, FBK, SK, SKB, SKT, WB, WBBL, F0, FST, F(1),FB(1),          7
00103        2 V(1),VT(1),XB(1),XT(1),T(1),DX(1),DXB(1),DXT(1),DE(1),              8
00103        3 DET(1),E(1),ET(1),A4,A5,A6,FKCR,DKXTCR,DKXCR,B3,B4                  9
00160      101 FORMAT(7F10.0)                                                         10
00161          G = 386.4                                                              11
00162          K = 100                                                                12
00163          EMB = WB/G                                                             13
00164          EMT=WBBL/G                                                             14
00165        1 DO 100 I=2 30                                                          15
00170          M=0                                                                    16
00171          N=0                                                                    17A
00172          L = 0                                                                  17B
00173          DT=0.002                                                               18
00174        2 DTSQ=DT**2                                                             19
00175          IF(I.NE.K+1)G   3                                                      20
00177          DX(I)=0.0                                                              21
00200          T(I)=T(I-1)                                                            22
00201          DXT(I)=0.0                                                             23
00202          DXB(I)=0.0                                                             24
00203          XT(I)=XT(I-1)                                                          25
00204          XB(I)=10.000                                                           26
00205          F(I)=F0+10.0*SK                                                        27
00206          FB(I)=FB(I-1)                                                          28
00207          TMV=EMT*VT(I-1)                                                        29
00210          BMV=EMB*V(I-1)                                                         30
00211          VT(I)=(TMV+BMV)/(EMT+EMB)                                              31
00212          V(I)=-VT(I)                                                            32
00213          E(I)=0.5*EMB*V(I)**2                                                   33
00214          ET(I)=0.5*EMT*VT(I)**2                                                 34
00215          DE(I)=E(I)                                                             35
00216          DET(I)=ET(I-1)-ET(I)                                                   36A
00217          GO TO 100                                                              36B
00220        3 IF(I.GT.K)GO TO 41                                                     37A
00222          B=B1/(B1+B2*DTSQ)                                                      37B
00223          GO TO 42                                                               37C
00224       41 B=B3/(B3 +B4*DTSQ)                                                     37D
00225          FK= FKCR                                                               37E
00226          DKXT=-DKXTCR                                                           37F
00227          A1=A4                                                                  37G
00230          A2=A5                                                                  37H
00231          A3=A6                                                                  37I
00232          DKX=-DKXCR                                                             37J
00233          FBK=BBLK                                                               37K
00234       42 IF(XT(I-1).LT.1.0)GO TO 4                                              38
00236          A=A1/(A1+A3*DTSQ)                                                      39
00237          GO TO 5                                                                40
00240        4 A=A1/(A1+A2*DTSQ)                                                      41
00241        5 VDT=V(I-1)*DT                                                          42
00242          FBOLT=FK*F(I-1)*DTSQ                                                   43
00243          FBBL2=FBK*F(I-1)                                                       44
00244        6 VTDT=VT(I-1)*DT                                                        05-9
00245        7 DXTDT1=DKXT*VTDT*DTSQ                                                  50
00246        8 DX1=B*(VDT-FBOLT+DXTDT1)                                               51
00247          DXDT=DKX*DX1*DTSQ                                                      52
00250          IF(XT(I-1).LT.1.0)GO TO 9                                              53
00252          IF(XT(I-1).GT.1.0)GO TO 31                                             54
00254          FB(I-1)=FS1                                                            55
00255       31 FBBL1=BBLK*FB(I-1)                                                     56A
00256          GO TO 10                                                               56B
00257        9 FBBL1=BUFK*FB(I-1)                                                     56C
00260       10 FBBL=(FBBL1-FBBL2)*DTSQ                                                57
00261          DXT1=A*(VTDT+FBBL-DXDT)                                                58
00262          DXTDT2=DKXT*DXT1*DTSQ                                                  59
00263          DDXDT=ABS(ABS(DXTDT2)-ABS(DXTDT1))                                     60
00264          IF(DDXDT.LT.0.0005)GO TO 11                                            61
00266          DXTDT1=DXTDT2                                                          62
00267          L = L+1                                                                63A
00270          IF(L.GT.25)GO TO 26                                                    63B
00272          GO TO 8                                                                63C
00273       11 DX(I)=DX1                                                              64
00274          DXT(I)=DXT1                                                            65
00275          XT(I)=XT(I-1)+DXT1                                                     66
00276          IF(XT(I-1).LT.1.0)GO TO 12                                             67
00300          DELXT=2.0-XT(I)                                                        68
00301          GO TO 13                                                               69
```

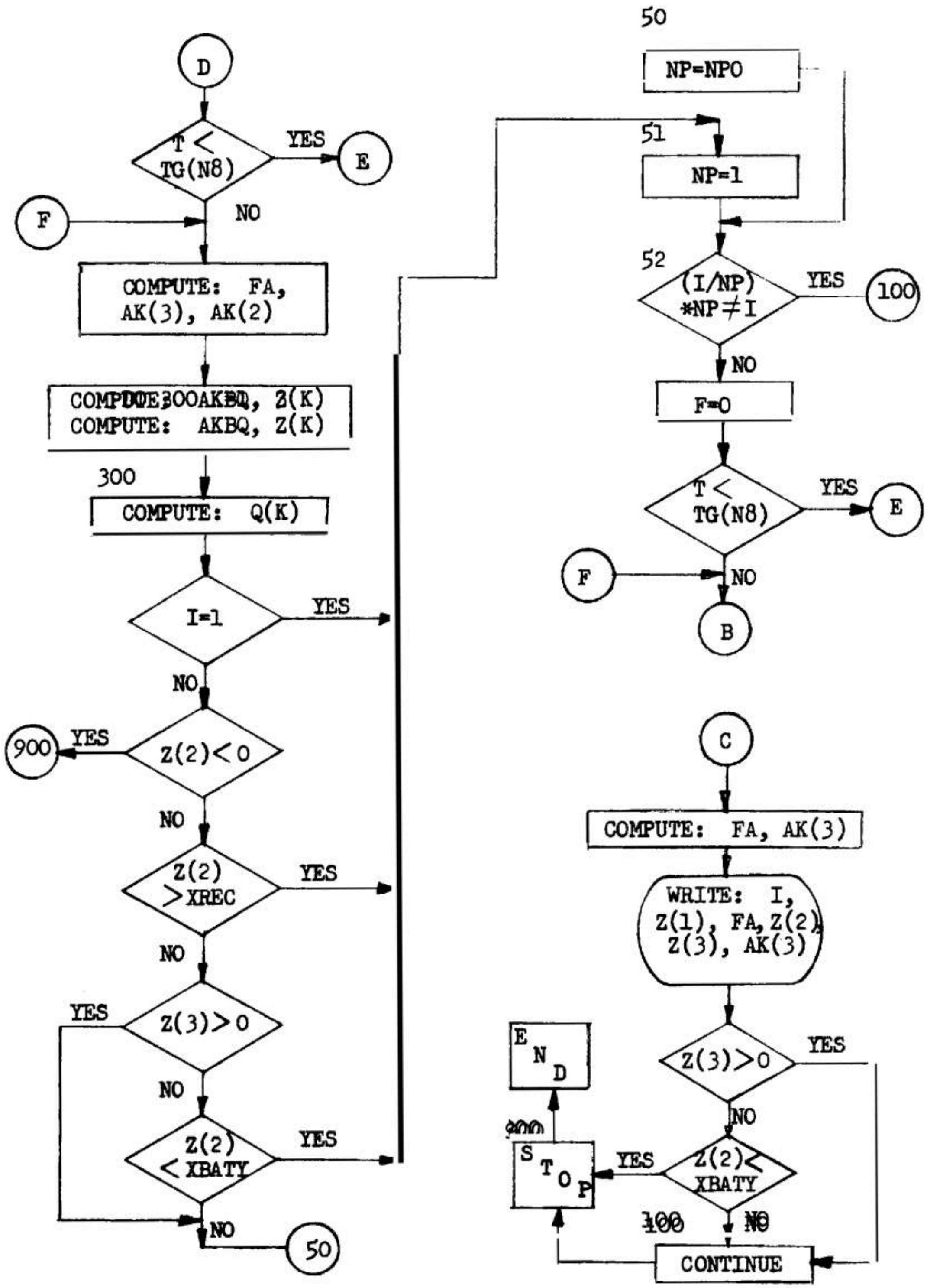
A—2. (Con't.)

```
(30302        12 DELXT=1.0-XT(I)                                                    70
00303         13 IF(ABS(DELXT).GT.0.002)GO TO 16                                    71-2
00305         14 IF(XT(I-1).LT.1.0)GO TO 15                                         73
00307            XT(I)=2.0                                                          74
00310            GO TO 18                                                           75
00311         15 XT(I)=1.0                                                          76
00312            GO TO 18                                                           77
00313         16 IF(N.NE.0)GO TO 18                                                 78
00315            IF(DELXT.LT.0.0)GO TO 17                                           79
00317            IF(M.EQ.0)GO TO 18                                                 80
00321         17 DT=ABS(DT*(DXT(I)+DELXT)/DXT(I))                                   81
00522            DTSQ=DT**2                                                         82
00323            M=M+1                                                              83
00324            IF(M.GT.25)GO TO 26                                                84
00326            GO TO 3                                                            85
00327         18 DXB(I)=DX(I)+DXT(I)                                                86
00330            XB(I)=XB(I-1)+DXB(I)                                               87
00531            IF(I.GT.(K+1))GO TO 22                                             88
00333            DELXB=10.0-XB(I)                                                   89
00334            IF(ABS(DELXB).GT.0.002)GO TO 20                                    90
00336         19 XB(I)=10.0                                                         91
00537            K=I                                                                92
00340            GO TO 22                                                           93
00541         20 IF(DELXB.LT.0.0)GO TO 21                                           94
00343            IF(N.EQ.0)GO TO 22                                                 95
00345         21 DT=ABS(DT*(DXB(I)+DELXB)/DXB(I))                                   96
-00346           DTSQ=DT**2                                                         97
00347            N=N+1                                                              98
00350            IF(N.GT.25)GO TO 26                                                99
00352            GO TO 3                                                            100
00353         22 F(I)=F0+SK*XB(I)                                                   101
00554            IF(I.GT.K)GO TO 23                                                 102
00356            DE(I)=DXB(I)*(F(I)+F(I-1))/(2.0*EPS)                               103
00357            E(I)=E(I-1)-DE(I)                                                  104
00360            IF(E(I).GT.0.0)GO TO 28                                            105A
00362            E(I)=0.0                                                           105B
00363            V(I)=0.0                                                           105C
00364            K = 1                                                              105D
00365            GO TO 27                                                           105E
00366         28 V(I)=SQRT(2.0*E(I)/EMB)                                            105F
00367            GO TO 27                                                           106
00370         23 DE(I)=ABS(DXB(I)*(F(I)+F(I-1))*EPS/2.0)                            107
00371            E(I)=E(I-1)          +DE(I)                                        108
00372            V(I)=-SQRT(2.0*E(I)/EMB)                                           109
00373         27 IF(XT(I-1).LT.1.0)GO TO 24                                         110
00575            EPSB=EPST                                                          111
00376            FB(I)=FST-SKT*(XT(I)-1.0)                                          112
00577            GO TO 25                                                           113
00400         24 FB(I)=FB(I)-SKB*XT(I)                                              114
00401         25 DET(I)=DXT(I)*(FB(I-1)+FB(I))*EPSB/2.0                             115
00402            ET(I)=ET(I-1)+DET(I)                                               116
00403            VT(I)=SQRT(2.0*ET(I)/EMT)                                          117
00404            T(I)=T(I-1)+1000.0*DT                                              118
00405            IF(XT(I).LT.2.0)GO TO 100                                          119A
00407            GO TO 26                                                           119B
00410        100 CONTINUE                                                           120
00412         26 WRITE(6,102)                                                       121
00414            WRITE(6,103)(J,DX(J),DXB(J),DXT(J),XB(J),XT(J), F(J),FB(J),        122
00414           1 J = 1,I)                                                          123
00431            WRITE(6,104)                                                       124
00433        102 FORMAT(1H1/12X,57HTABLE 2-5 COUNTERRECOIL DYNAMICS OF DELAYED BLOW 125A
00433           1 BACK GUN//                                                        125B
00433           2          14X,63H RELATIVE   DELTA   DELTA   TOTAL   TOTAL   D     126
00433           3RIVING    BARREL/77H    INCRE-   DELTA    BOLT    BARREL   BOL     127
00433           4T     BARREL    SPRING   SPRING/6X,70H MENT   TRAVEL   TRAVEL  T   128
00453           5RAVEL   TRAVEL   TRAVEL    FORCE    FORCE/8X,1HI,7X,60H INCH   I   129
00433           6NCH     INCH     INCH     INCH    POUND    POUND//)                130
00434        103 FORMAT(I9,F12.3,2F9.3,F10.3,F9.3,F9.2,F10.1)                       131
00435        104 FORMAT(1H1/9X,62HTABLE 2-5 CONTD-COUNTERRCOIL DYNAMICS OF DELAYED  132A
00435           1 BLOWBACK GUN//                                                    132B
00435           2          25X,14H DELTA    DELTA/6X,6HINCRE-,13X,53H BOLT   BARREL 133
00435           3   BOLT    BARREL    BOLT    BARREL/6X,74H MENT       TIME    ENE  134
00435           4RGY   ENERGY   ENERGY   ENERGY   VELOCITY   VELOCITY/8X,1HI,7X,63H 135
00435           5 MSEC    IN-LB    IN-LB    IN-LB    IN-LB    IN/SEC    IN/SEC//)   136
00436            WRITE(6,105) (J,T(J),DE(J),DET(J),E(J),ET(J),V(J),VT(J),           137
00436           1 J = 1,I)                                                          138
00453        105 FORMAT(I9,F12.2, 2F9.1, F10.1, F9.1,2F10.1)                        139
00454        199 STOP                                                               140
00455            END                                                                141
```

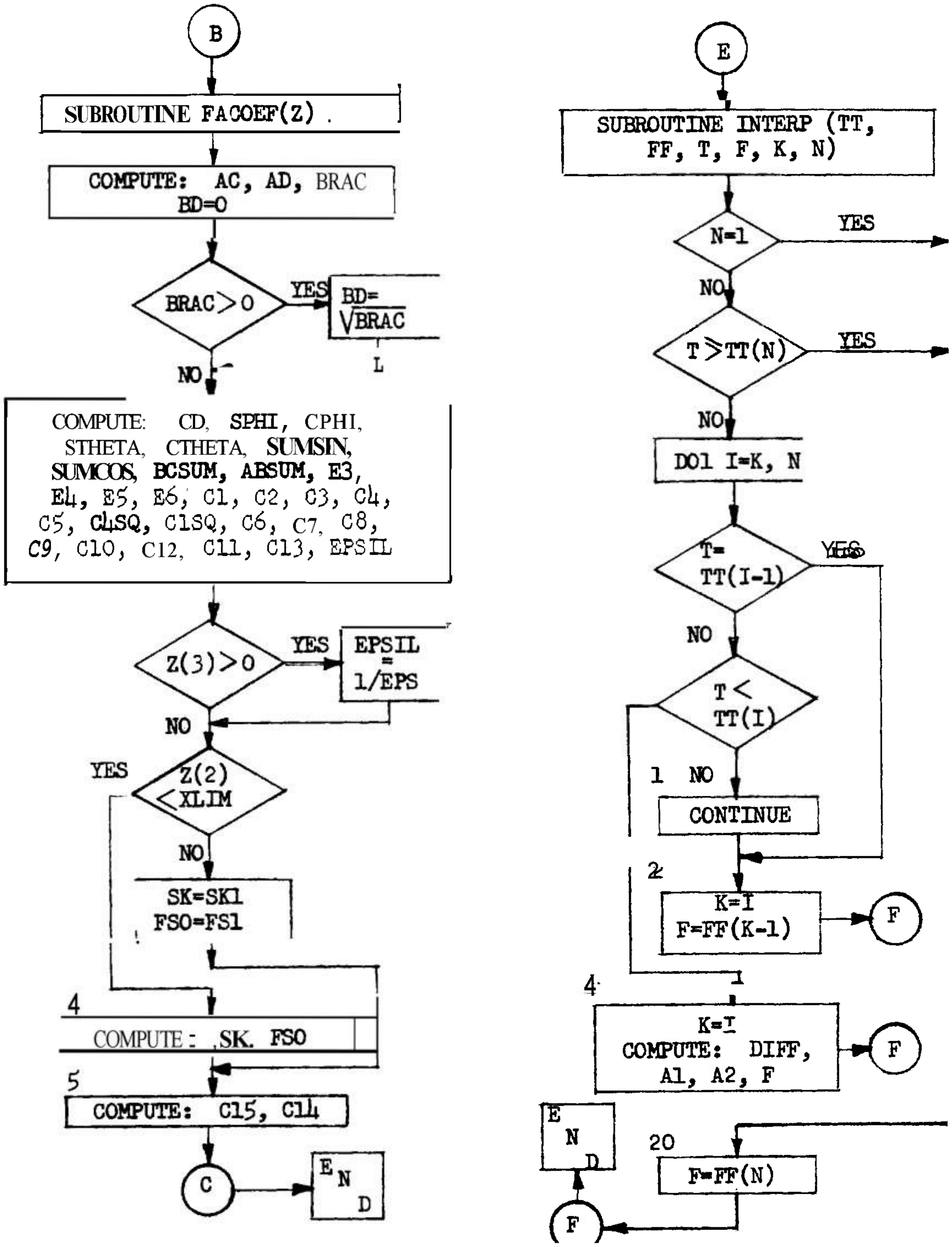
A–3. FLOW CHART FOR RETARDED BLOWBACK



A–3. (Con't.)


D
T < TG(N8)
YES
E
NO
F
COMPUTE: FA, AK(3), AK(2)
COMPUTE: AKBQ, Z(K)
300
COMPUTE: Q(K)
I=1
YES
NO
Z(2)<0
YES
900
NO
Z(2) >XREC
YES
NO
Z(3)>0
YES
NO
Z(2) < XBATY
YES
NO
50
50
NP=NPO
51
NP=1
52
(I/NP) *NP≠I
YES
100
NO
F=0
T < TG(N8)
YES
E
F
NO
B
C
COMPUTE: FA, AK(3)
WRITE: I, Z(1), FA, Z(2), Z(3), AK(3)
Z(3)>0
YES
NO
Z(2)< XBATY
YES
END
900
STOP
100
NO
CONTINUE

A—3. (Con't.)


B
SUBROUTINE FACOEF(Z)
COMPUTE: AC, AD, BRAC
BD=0
BRAC>0
YES
BD=
√BRAC
L
NO
COMPUTE: CD, SPHI, CPHI, STHETA, CTHETA, SUMSIN, SUMCOS, BCSUM, ABSUM, E3, E4, E5, E6, C1, C2, C3, C4, C5, C4SQ, C1SQ, C6, C7, C8, C9, C10, C12, C11, C13, EPSIL
Z(3)>0
YES
EPSIL = 1/EPS
NO
YES
Z(2) <XLIM
NO
SK=SK1
FSO=FS1
4
COMPUTE: SK, FSO
5
COMPUTE: C15, C14
C
END
E
SUBROUTINE INTERP (TT, FF, T, F, K, N)
N=1
YES
NO
T≥TT(N)
YES
NO
DO1 I=K, N
T= TT(I-1)
YES
NO
T< TT(I)
1 NO
CONTINUE
2
K=I
F=FF(K-1)
F
4
K=I
COMPUTE: DIFF, A1, A2, F
F
END
20
F=FF(N)
F

A–4. LISTING FOR RETARDED **BLOWBACK** PROGRAM

```
00101          DIMENSION FG(96),Z(3),Q(3),AK(3),TG(96),A(4),B(4),C(4)                      1
00103          COMMON AZ,ABC,ABSQ,AB,BC,EYEB,E1,E2,EYEC,E7,EMR,EPS,XLIM,SK1,FS1,           8
00103         1SK2,FS2,E3,E4,E5,E6,C1,C2,C3,C4,C5,C6,C7,C8,C9,C10,C11,C12,C13,C14          -3
00103         2,C15,AC,AD,BD,CD,FS0,SUMSIN,SUMCOS,SPHI,CPHI,STHETA,CTHETA                   4
00104          EQUIVALENCE (Z(1),T),(Z(2),X),(Z(3),VEL)                                     2
00105          DATA A(1),C(1),C(4),B(1),B(4),B(2),B(3),A(3),C(3),A(2),C(2),A(4)/           3
00105         D 3*.5,2*2.,2*1.,2*1.70710678,2*.292893219,.166666667/                        4
00122          READ(5,1)N,DT,N9,DTFG,NP0,TCHANG,DTNEW,NHEAD                                 5
00134        1 FORMAT(I6,E12.0,I6,E12.0,I6,2E12.0,I6)                                       6
00135          READ(5,2)XREC,XLIM,AB,BC,WB,WAB,WBC,G,SK1,SK2,FS1,FS2,AZ,EPS,XBATY           7
00156          ABSQ = AB*AB                                                                8A
00157          BCSQ = BC*BC                                                                8B
00160          ABC = ABSQ - BCSQ                                                           8C
00161          EMB = WB/G                                                                  8D
00162          EMAB = WAB/G                                                                8E
00163          EMR = EMB + EMAB                                                            8F
00164          EYEB  = EMAB*ABSQ/12.0                                                      8G
00165          EMBC = WBC/G                                                                8H
00166          EYEC = EMBC*BCSQ/12.0                                                       8I
00167        2 FORMAT(6E12.0)                                                               9
00170          READ(5,15)(FG(I+1),I=1,N9)                                                  10
00176       15 FORMAT(9F8.0)                                                               11
00177          E1 = EMAB*AB/2.0                                                           12A
00200          E2=EMBC*BC/2.0                                                             12B
00201          E7=EYEC/BC-E2/2.0                                                          12C
00202          FG(1)=0.0                                                                  13A
00203          TG(1)=0.                                                                   13B
00204          N8=N9+1                                                                    13C
00205          DO 20 I=1,N9                                                                14
00210       20 TG(I+1)=TG(I)+DTFG                                                          15
00212          AX(1)=1.0                                                                   16
00212    C     T=Z(1)                                                                      17
00212    C     X=Z(2)                                                                      18
00212    C     VEL=Z(3)                                                                   19A
00213          WRITE(6,3)                                                                 19B
00215        3 FORMAT(1H1/25X,37H TABLE 2-8 RETARDED BLOWBACK DYNAMICS/)                  19C
00216          WRITE (6,708)                                                              19D
00220      708 FORMAT(30X,23HAPPLIED          DISTANCE/15X,4HTIME,12X,57HFORCE            19E
00220         1  FROM BREECH        VELOCITY        ACCELERATION/20H      S               19F
00220         2ECOND,11X5HPOUND,11X4HINCH,11X,25H IN/SEC          IN/SEC/SEC)             19G
00221          DO 10 I=1,3                                                                 20
00224          Q(I)=0.0                                                                    21
00225       10 Z(I)=0.0                                                                    22
00227          MINT=2                                                                      23
00230      800 DO 100 I=1,N                                                               24A
00233          IF(I.EQ.1)GO TO 75                                                         24B
00235          IF((I/NHEAD)*NHEAD+1.NE.I)GO TO 75                                         25A
00237          WRITE(6,707)                                                               25B
00241          WRITE (6,708)                                                              25C
00243      707 FORMAT(1H1/22X,44H TABLE 2-8 CONTD. RETARDED BLOWBACK DYNAMICS/)           25D
00244       75 IF(T.GT.TCHANG)DT=DTNEW                                                    25E
00246          DO 300 J=1,4                                                                26
00251          CALL FACOEF(Z)                                                              52
00252          F = 0.0                                                                     64
00253          IF(T.LT.TG(N8))CALLINTERP(TG,FG,T,F,MINT,N8)                                65
00255          FA = F+(C14+C15*Z(2))                                                       55
00256          AK(3)=(FA+C13*Z(3)**2)/C11                                                  56
00257          AK(2)=Z(3)                                                                  67
00260          DO 300 K=1,3                                                                60
00263          AKBQ=A(J)*(AK(K)-B(J)*Q(K))                                                 69
00264          Z(K)=Z(K)+DT*AKBQ                                                           70
00265      300 Q(K)=Q(K)+3.0*AKBQ-C(J)*AK(K)                                               71
```

A—4. (Con't.)

```
00270          IF(I.EQ.1 )GO TO 51                                        71X
00272          IF(Z(2).LT.0.0)GO TO 900                                    72
00279          IF(Z(2).GT.XREC)GO TO 51                                   73A
00276          IF(Z(3).GT.0.0)GO TO 50                                    73B
00300          IF(Z(2).LT.XBATY)GO TO 51                                  73C
00302       50 NP=NP0                                                     73D
00303          GO TO 52                                                   73E
00304       51 NP=1                                                        74
00305       52 IF((I/NP)*NP.NE.I)GO TO 100                                 15
00307          F = 0.0                                                     71
00310          IF(T.LT.TG(N8))CALLINTERP(TG,FG,T,F,MINT,N8)                71
00312          CALL FACOEF(Z)                                              72
00313          FA = F+(C14+C15*Z(2))                                       73
00314          AK(3)=(FA+C13*Z(3)**2)/C11                                  74
00315          WRITE(6,6)I,Z(1),FA,Z(2),Z(3),AK(3)                         76
00325          IF(Z(3).GT.0.)GO TO 100
00327          IF(Z(2).LT.XBATY)STOP                                      77A
00331      100 CONTINUE                                                    77
00333        6 FORMAT(I6,F15.7,F15.1,F17.6,F16.1,F17.1)                    78
00334      900 STOP                                                        79
00335          END                                                         00
```

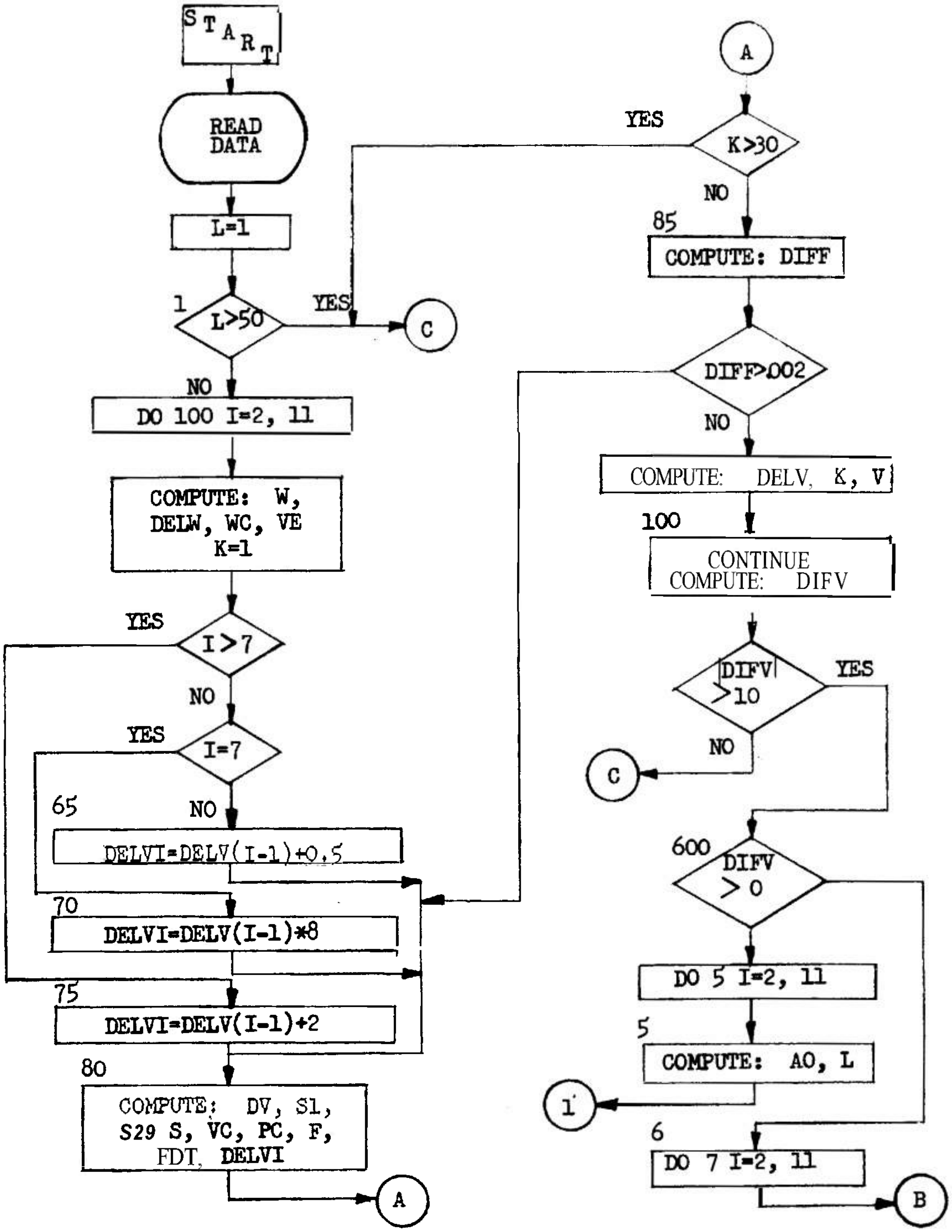
A–4. (Con't.)

INPUT CARD COUNT

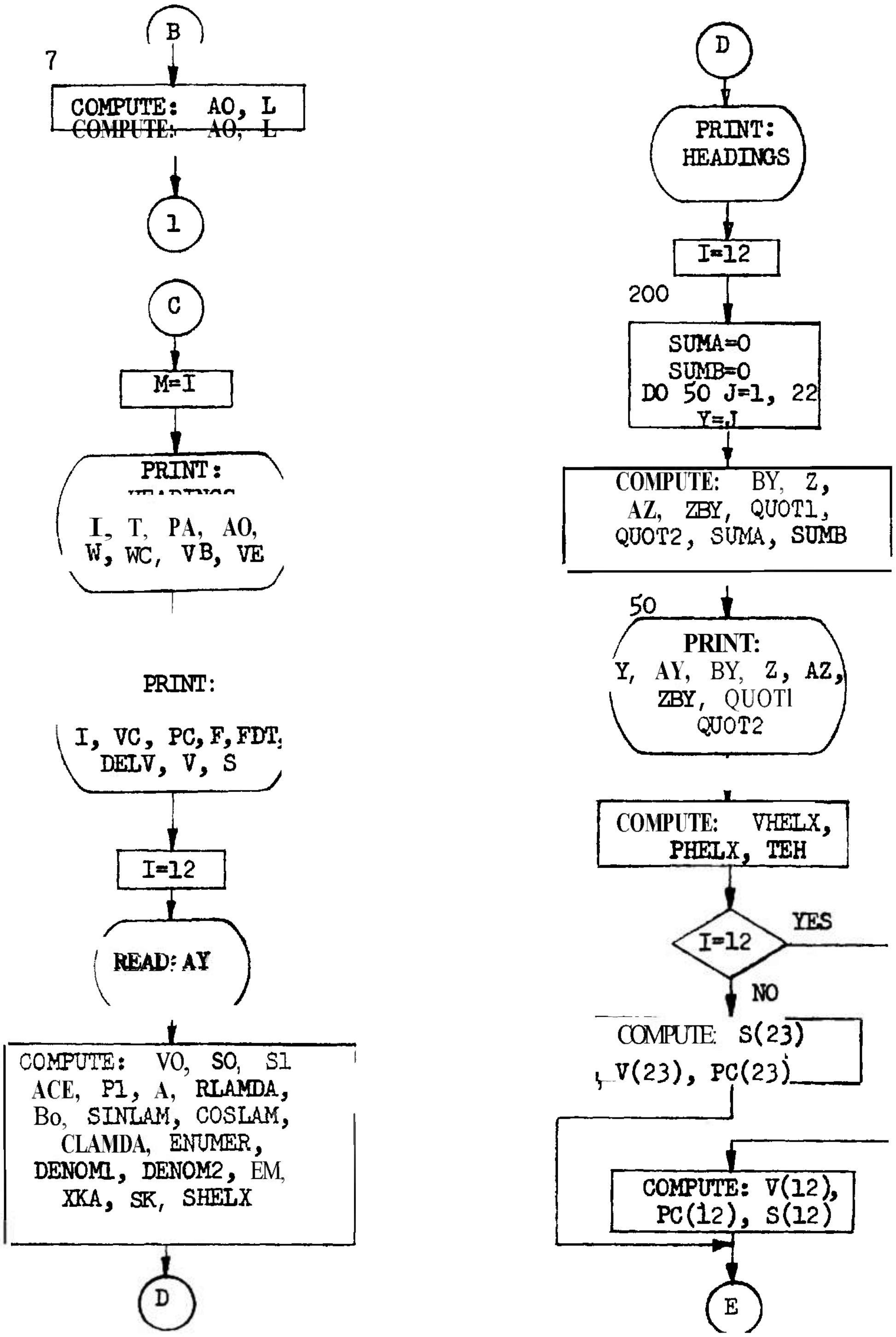
```
      SUBROUTINE INTERP(TT,FF,T,F,K,N)
      DIMENSION TT(1),FF(1)
      IF(N.EQ.1)GO TO 20
      IF(T.GE.TT(N))GO TO 20
      DO 1 I=K,N
      IF(T.EQ.TT(I-1))GO TO 2
      IF(T.LT.TT(I))GO TO 4
    1 CONTINUE
    2 K=I
      F=FF(K-1)
      RETURN
    4 K=I
      DIFF=TT(K)-TT(K-1)
      A1=(T-TT(K-1))/DIFF
      A2=(TT(K)-T)/DIFF
      F=A1*FF(K)+A2*FF(K-1)
      RETURN
   20 F=FF(N)
      RETURN
      END
```

```
00101          SUBROUTINE FACOEF(Z)                                                      S1
00103          COMMON AZ,ABC,ABSQ,AB,BC,EYEB,E1,E2,EYEC,E7,EMR,EPS,XLIM,SK1,FS1,         S2
00103         1SK2,FS2,E3,E4,E5,E6,C1,C2,C3,C4,C5,C6,C7,C8,C9,C10,C11,C12,C13,C14         3
00103         2,C15,AC,AD,BD,CO,FSO,SUMSIN,SUMCOS,SPHI,CPHI,STHETA,CTHETA                 4
00104          DIMENSION Z(3)                                                            54
00105          AC=AZ-Z(2)                                                                27
06106          AD=(ABC+AC**2)/(2.0*AC)                                                   28
00107          BRAC=ABSQ-AD**2                                                           29A
00110          BD=0.                                                                     29B
00111          IF(BRAC.GT.0.)BD=SQRT(BRAC)                                               29C
00113          CD = AC-AD                                                                30
00114          SPHI = BO/AB                                                              31
00115          CPHI = AO/AB                                                              32
00116          STHETA = BD/BC                                                            33
00117          CTHETA = CO/BC                                                            34
00120          SUMSIN = STHETA*CPHI + CTHETA*SPHI                                        35
00121          SUMCOS = CTHETA*CPHI - STHETA*SPHI                                        36
00122          BCSUM = BC*SUMSIN                                                         37
00123          ABSUM = AB*SUMSIN                                                         38
00124          E3 = (EYEB + E1*AB/2.0)/AC                                                39A
00125          E4=E1*SPHI/AC                                                             39B
00126          E5=E2*ABSUM/AC                                                            39C
00127          E6 = (E2*(AB*SUMCOS + BC/2.0) - EYEC)/AC                                  39D
00130          C1 = CPHI/BCSUM                                                           40
00131          C2 = -AB/BCSUM                                                            41A
00132          C3 = -SUMCOS/SUMSIN                                                       41B
00133          C4 = CTHETA/ABSUM                                                         42
00134          C5 = -BC/ABSUM                                                            43
00135          C4SQ = C4**2                                                              44
00136          C1SQ = C1**2                                                              45A
00137          C6 = C2*C4SQ+C3*C1SQ                                                      45B
00140          C7 = C5*C1SQ+C3*C4SQ                                                      46
00141          C8=E3*C4+E6*C1-E4                                                         47
00142          C9=E3*C7+E6*C6-E5*C1SQ                                                    48
00143          C10=(C8*CTHETA+E7*C1)/STHETA                                              49
00144          C12=(C9*CTHETA+E7*C6)/STHETA                                              50
00145          C11=EMR+E2*C1*STHETA-E1*C4*SPHI+C10                                       51
00146          C13=E1*C4SQ*CPHI+E1*C7*SPHI-E2*C1SQ*CTHETA-E2*C6*STHETA-C12               52
00147          EPSIL = EPS                                                               53
00150          IF(Z(3).GT.0.0)EPSIL=1.0/EPS                                              54
00152          IF(Z(2).GT.XLIM)GO TO 4                                                   55
00154          SK = SK1                                                                  57
00155          FSO = FS1                                                                 58
00156          GO TO 5                                                                   59
00157        4 SK = SK1+SK2                                                              60
00160          FSO = FS1+FS2-SK2*XLIM                                                    61
00161        5 C15 = -SK*EPSIL                                                           62
00162          C14 = -FSO*EPSIL                                                          63
00163          RETURN                                                                    54B
00164          END                                                                       S49
```

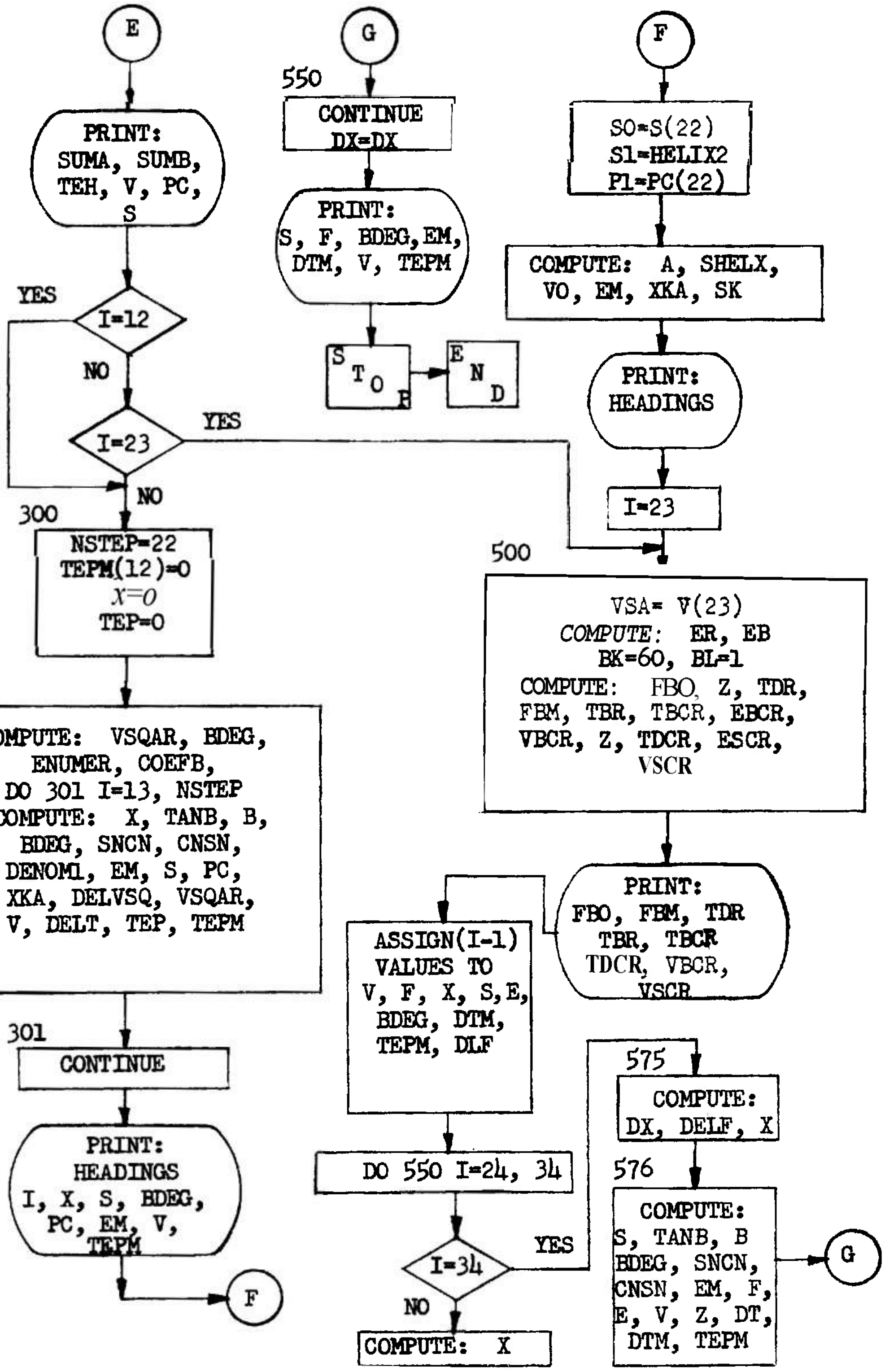
A–5. FLOW CHART FOR CUTOFF EXPANSION



A—5. (Con't.)


B
7
COMPUTE: AO, L
1
C
M=I
PRINT:
I, T, PA, AO,
W, WC, VB, VE
PRINT:
I, VC, PC, F, FDT,
DELV, V, S
I=12
READ: AY
COMPUTE: VO, SO, S1
ACE, P1, A, RLAMDA,
Bo, SINLAM, COSLAM,
CLAMDA, ENUMER,
DENOM1, DENOM2, EM,
XKA, SK, SHELX
D
D
PRINT:
HEADINGS
I=12
200
SUMA=0
SUMB=0
DO 50 J=1, 22
Y=J
COMPUTE: BY, Z,
AZ, ZBY, QUOT1,
QUOT2, SUMA, SUMB
50
PRINT:
Y, AY, BY, Z, AZ,
ZBY, QUOT1
QUOT2
COMPUTE: VHELX,
PHELX, TEH
I=12
YES
NO
COMPUTE: S(23)
V(23), PC(23)
COMPUTE: V(12),
PC(12), S(12)
E

A–5. (Con't.)


E
PRINT: SUMA, SUMB, TEH, V, PC, S
I=12
YES
NO
I=23
YES
NO
300
NSTEP=22
TEPM(12)=0
X=0
TEP=0
COMPUTE: VSQAR, BDEG, ENUMER, COEFB,
DO 301 I=13, NSTEP
COMPUTE: X, TANB, B, BDEG, SNCN, CNSN, DENOM1, EM, S, PC, XKA, DELVSQ, VSQAR, V, DELT, TEP, TEPM
301
CONTINUE
PRINT: HEADINGS I, X, S, BDEG, PC, EM, V, TEPM
F
G
550
CONTINUE
DX=DX
PRINT: S, F, BDEG, EM, DTM, V, TEPM
STOP
END
F
SO=S(22)
S1=HELIX2
P1=PC(22)
COMPUTE: A, SHELX, VO, EM, XKA, SK
PRINT: HEADINGS
I=23
500
VSA= V(23)
COMPUTE: ER, EB
BK=60, BL=1
COMPUTE: FBO, Z, TDR, FBM, TBR, TBCR, EBCR, VBCR, Z, TDCR, ESCR, VSCR
PRINT: FBO, FBM, TDR TBR, TBCR TDCR, VBCR, VSCR
ASSIGN(I-1) VALUES TO V, F, X, S, E, BDEG, DTM, TEPM, DLF
DO 550 I=24, 34
I=34
YES
NO
COMPUTE: X
575
COMPUTE: DX, DELF, X
576
COMPUTE: S, TANB, B BDEG, SNCN, CNSN, EM, F, E, V, Z, DT, DTM, TEPM
G

A–6. LISTING FOR CUTOFF EXPANSION PROGRAM

```
      0DIMENSION T(50),DELT(50),AO(50),PA(50),VB(50),DELV(50),VC(50),
      1F(50),WC(50),AY(50),W(50),S(50),PC(50),BDEG(50),
      3EM(50),V(50),VE(50),FDT(50),X(50),DTM(50),TEPM(50)
      0READ 400,(T(I),I=2,11),(DELT(I),I=2,11),(PA(I),I=2,11),
      1 Q(I),I=2  11),(AO(I) I=2,11)
   40 FORMAT (10F7.3)
      M=100
      READ  30 ,V   X , B  O(1),T(1),AC, O WB,WC(1),VCYL,D AMQ  B , RL,G,
      1SA,EMUR,E  G,TANBO,E B ,RADGYR,          SMAX,DX       B , Q K,V(1),
      3S(1),HEL IX ,HELI X  SC K
   30 FORMAT ( G1 2.0)
      L=1
    1 IF(L.GT.50)GO TO 2
      DO 1 0 I=2,11
      W(1) = .00192*AO(I)*PA(I)
      DELW = (I)*DELT(I)
      WC(I) =C (I -1  D L W
      VE(I) = (I )*  I ) Q   0 29
      K
      IF( I EQ.7)GO TO  
      IF( I GT.7)GO TO  
   65 DELVI=   B O(I-0)+(  5
      GO TO 60
   70 DE VI=   D LV(I-   *  0
      G  TO  6
   75 DELVI=  DELV(I-1)+2.0
    6 DV=DELVI
      S1=0.5*DV*DELT(I)
      S2=V(I-1)*DELT(I)
      S(I)=S1+S2+S(I-1)
      VC(I)=VCYL+AC*S(I)
      PC(I)=(VE(I)/VC(I))**1.3*PA(I)
      F(I)=0.60*FC(I)
      FDT(I)=F(I)* DE.T(I)
      DE.VI = 1 5  56*FDT(I)
      IF(K.GT.30)GO TO 2
   85 DIFF= ABS ( DE.VI- V)/DELVI)
      IF(DIFF.GT..002)GO TO 80
      DELV(I)=DELVI
      K K  1
      V(I )=(I-1)+DE.V(I)
   10 CONTINUE
      DIFV=V(11)-VMAX
      IF(ABS (DIFV).GT.10.0)GO TO 600
      GO TO 2
  600 IF(DIFV.GT.0.0)GO T  6
```

A–6. (Con't.)

```
00247           DO 5 I=2,11
00252         5 AO(I)=AO(1)+0.0010
00254           L = L + 1
00255           GO TO 1
00256         6 DO 7 I=2,11
00261         7 AO(I)=AO(1)-0.0010
00263           L=L+1
00264           GO TO 1
00265         2 M=I
00266           PRINT 23,(I,T(I),PA(I),AO(I),W(I), WC(I),V8(I),VE(1),I=2,11)
00303           PRINT 24,(I,VC(I),PC(I),F(I),FDT(I),DELV(I),V(I),S(I),I=2,11)
00320       230FORMAT (1H1/          9X,46H TABLE 4-5 COMPUTED DYNAMICS BEFORE GAS
00320          1 CUTOFF//33X,31H GAS       GAS     EQUIV     EQUIV/24X,39H PORT      F
00520          2LOW      IN       BORE      CYL/                      7X 56H TIME    PRES
00320          3S       AREA      RATE      CYL        VOL        VOL/64H    I     MSEC     PS
00320          4I        SQ-IN    LB/SEC     LB        CU-IN      CU-IN//(I4,F8.3,F9.0,F9.4
00320          5,F8.3,F9.5,F8.3,F9.4))
00321       240FORMAT(//6X,23H CYL        CYL     PISTON,12X,20HDELTA      ROD       ROD
00521          1/8X,57H VOL       PRESS      FORCE  IMPULSE      VEL       VEL     TRAVEL/6
00321          23H   1      CU-IN       PSI        LB       LB-SEC    IN/SEC   IN/SEC      IN/
00321          4/(I4,F9.4,2F9.1,F8.3,F9.2,F8.1,F9.4))
00322           1=12
00323           READ 350,(AY(J),J=1,22)
00331           V0=V(11)
00332           S0=S(11)+SCYL
00333           S1=HELIX1 -S(11)
00334           ACE=2.0*AC/0.3
00335           P1=PC(11)
00336           A=1.0+S1/S0
00537           RLAMDA=DLAMDA/57.296
00340           B0=ATAN(TANB0)
00341           SINLAM=SIN(RLAMDA)
00342           COSLAM=COS(RLAMDA)
00343           CLAMDA=SINLAM-EMUS*COSLAM/(COSLAM+EMUS*SINLAM)
00344           ENUMER =WB*RADGYR**2*TANB0/RC
00345           DENOM1=(RC-EMUS*R)*(COS(B0)-EMUR*SIN(B0))/(SIN(B0)+EMUR*COS(B0))
00346           DENOM2=CLAMDA*(RL-EMUS*R)
00347           EM(12)=(WC+WB* ENUMER/(DENOM1+DENOM2))/G
00350           XKA=ACE*P1*S0**1.3/EM(12)
00351           SK = S0**1.15/SQRT(XKA)
00352           SHELX=SCYL+HELIX1
Ob353           PRINT 41
00555        41 FORMAT(1H1 /
00555          1  17X,45H TABLE 4-6 COMPUTED DYNAMICS AFTER GAS CUTOFF/21X,37H 80L
00555          2T UNLOCKING DURING HELIX TRAVERSE// 3X,2H Y,6X,3H AY,7X,3H BY,
00555          37X,2H Z, 6X,3H AZ,7X,4H ZBY,6X,6H QUOT1, 5X,6H QUOT2/ )
00356           I=12
00357       200 SUMA=0
00360           SUMB = 0
00361           DO 50 J = 1,22
00364           Y=J
00365           BY = (1.0 + (V0**2/XKA)*S0**0.3)**Y
00366           Z = 1.0 - 0.3*Y
00367           AZ = A**Z
00370           ZBY = Z*BY
00371           QUOT1 = AY(J)/ZBY
00372           QUOT2 = AZ*QUOT1
Ob373           SUMA = SUMA + QUOT1
```

A–6. (Con't.)

```
00374        SUMB = SUMB + QUOT2                                                  102
00375     50 PRINT 21,Y,AY(J),BY, Z, AZ, ZBY, QUOT1, QUOT2                        103
00410     21 FORMAT (F6.1, 2F10.4, F8.1, F10.4, 3F11.4)                           104
00411        VHELX=SQRT(XKA*((1.0/S0**0.3)-1.0/SHELX**0.3)+V0**2)                 105
00412        PHELX=P1*(S0/SHELX)**1.3                                             106
00413        TEH= SK*(A+SUMB - 1.0-SUMA)                                          107
00414        IF(I.EQ.12)GO TO 201                                                 108
00416        S(23)=S(22)+HELIX2                                                   109A
00417        V(23)=VHELX                                                          109B
00420        PC(23)=PHELX                                                         110
00421        GO TO 202                                                            111
00422    201 V(12)=VHELX                                                          112
00423        PC(12)=PHELX                                                         113A
00424        S(12)=SHELX                                                          113B
00425    202 PRINT 51, SUMA,SUMB,TEH,V(I),PC(I),S(I)                              114
00435     51 FORMAT(//49X,6HTOTALS,2F11.4//9X,44HEXPANSION TIME DURING HELIX TR   115
00435       1AVERSE (TEH) = F8.5,8H SECONDS //9X,3HV =F7.2,7H IN/SEC,5X,5H PC =   116
00435       2F7.1,4H PSI,5X,3HS =F7.4,4H IN.)                                     117
00436        IF(I.EQ.12)GO TO 300                                                 118
00440        IF(I.EQ.23)GO TO 500                                                 119
00442    300 NSTEP=22                                                  1—20
00443        TEPM(12)=0.0                                                         121
00444        X(I)=0.0                                                             122
00445        TEP = 0.0                                                            123A
00446        VSQAR=V(12)**2                                                       123B
00447        BDEG(12)=0.4426                                                      124
00450        ENUMER=WB*RADGYR**2/RC                                               125
00451        COEFB=RC-EMUS*R                                                      126
00452        DO 301 I=13,NSTEP                                                    127
00455        X(I)=X(I-1)+DX                                                       128
00456        TANB=2.632*X(I)+TANB0                                                129
00457        B=ATAN(TANB)                                                         130
00460        BDEG(I)=57.296*B
00461        SNCN =SIN(B)+EMUR*COS(B)                                             132
00462        CNSN =COS(B)-EMUR*SIN(B)                                             133
00463        DENOM1=COEFB*CNSN/SNCN                                               134
00464        EM(I)=(W0+ENUMER*TANB/(DENOM1+DENOM2))/G                             135
00465        S(I)=S(I-1)+DX                                                       136
00466        PC(I)=PC(I-1)*(S(I-1)/S(I))**1.3                                     137
00467        XKA=ACE*PC(I)/EM(I)                                                  138
00470        DELVSQ=XKA*(S(I-1)-S(I-1)**1.3/S(I)**0.3)                            139
00471        VSQAR=DELVSQ+VSQAR                                                   140
00472        V(I)=SQRT(VSQAR)                                                     141
00473        DELT=2.0*DX/(V(I)+V(I-1))                                            142
00474        TEP = TEP+DELT                                                       143
00475        TEPM(I)=1000.0*TEP                                                   144
00476    301 CONTINUE                                                             145
00500        PRINT 302,(I,X(I),S(I),BDEG(I),PC(I),EM(I),V(I),TEPM(I),I=13,22)     146
00515   3020FORMAT(1H1/                                                           147A
00515       1 12X,45H TABLE 4-7 COMPUTED DYNAMICS AFTER GAS CUTOFF/14X,40H BOL    147B
00515       2T UNLOCKING DURING PARABOLA TRAVERSE//14X,6H EQUIV,21X,6H EQUIV/55   148
00515       3H      PARAB     CYL      CAM      CYL      RECOIL     ROD/65H       149
00515       4  DIST   LENGTH   SLOPE    PRESS    MASS     VEL      TIME/65H       150
00515       5 1      IN       IN      DEG      PSI      W/G     IN/SEC   MSEC// (  151A
00515                                                                             1518
00516    405 S0=S(22)                                                             152
00517        S1=HELIX2                                                            153
00520        P1=PC(22)                                                            154A
```

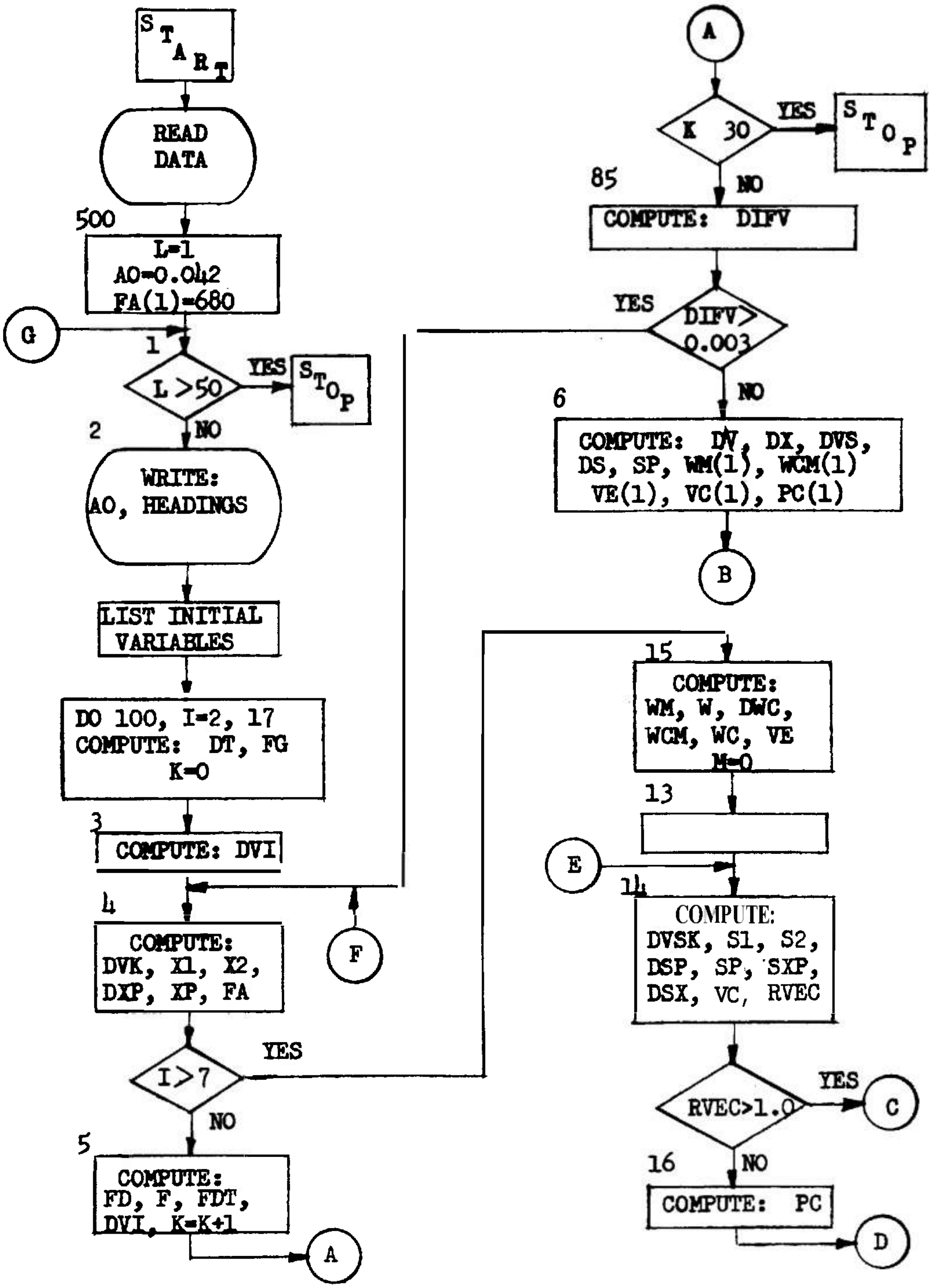
A–6. (Con't.)

```
00521      A=1.0+S1/SC                                                    154B
00522      SHELX=S0+HELIX2                                                154C
00523      V0=V(22)*WC/(W0+WB)                                            155
00524      EM=(W0+WB)/G                                                   156
00525      XKA=ACE*PI*S0**1.3/EM                                          157
00526      SK=S0**1.15/SQRT(XKA)                                          158
00527      PRINT 410                                                      159A
00531      I=23                                                           159B
00532      GO TO 200                                                      159C
00533  410 FORMAT(1H1/      17X,45H TABLE 4-8 COMPUTED DYNAMICS AFTER GAS C  160
00533     1UTOFF/17X,46H BOLT AND ROD UNIT RECOILLING AFTER CAM ACTION//3X,2H  161
00533     2 Y,6X,3H AY,7X,3H BY,7X,1HZ,6X,3H AZ,7X,4H ZBY,6X,6H QUOT1,5X,6H Q  162
00533     3UOT2/ )                                                        163
00534  500 VSA=V(23)                                                      164
00535      ER=EM*VSA**2/2.0                                               165
00536      EB=ER-562.0                                                    166
00537      BK=60.0                                                        167
00540      BL=1.0                                                         168
00541      FB0=EPS*EB-BK*BL/2.0                                           169
00542      Z=SQR((3807.0+20.4*ER)                                         170
00543      TDR=0.00203*(ASIN(97.4/Z)-ASIN(61.7/Z))                        171
00544      FBM=FB0 +BK*BL                                                 172
00545      TBR=0.00837*ACOS(FB0/FBM)                                      173
00546      TBCR=0.01675*ACOS(FB0/FBM)                                     174
00547      EBCR=(FBM+FB0)/4.0                                             175
00550      VBCR=SQRT(2.0*EBCR/EM)                                         176
00551      Z=SQRT(9467.0+40.8*EBCR)                                       177
00552      TDCR=0.04061*(ASIN(97.4/Z)-ASIN(51.5/Z))                       178
00553      ESCR=EBCR+167.5                                                179
00554      VSCR=SQRT(2.0*ESCR/EM)                                         180
00555      PRINT 501,FB0,FBM,TDR,TBR,TBCR,TDCR,VBCR,VSCR                  181
00567  501 FORMAT(///20X,23H MINIMUM BUFFER FORCE =F7.1,3H LB/20X,23H MAXIMUM  182
00567     1 BUFFER FORCE =F7.1,3H LB/20X,29H DRIVING SPRING RECOIL TIME =F9.6  183
00567     2,4H SEC/20X,21H BUFFER RECOIL TIME =F9.6,4H SEC/20X,28H BUFFER COU  184
00567     3NTERRECOIL TIME =F9.6,4H SEC/20X,31H DR SPRING COUNTERRECOIL TIME  185
00567     4= F9.6,4H SEC/20X,32H BUFFER COUNTERRECOIL VELOCITY =F7.2,7H IN/SE  186
00567     5C/20X,32H LR SPR COUNTERRECOIL VELOCITY =F7.2,7H IN/SEC)       187
00570  510 V(23)=VSCR                                                     188
00571      F(23)=51.5                                                     189
00572      X(23)=0.5                                                      190
00573      S(23)=0.0                                                      191A
00574      E=ESCR                                                         191B
00575      BDEG(23)=52.926                                                192
00576      F(23)=51.5                                                     193
00577      DTM(23)=0.0                                                    194
00600      TEPM(23)=0.0                                                   195
00601      DELF=DRK*DX                                                    196
00602      NSTEP=34                                                       197
00603      DO 550 I=24,NSTEP                                              198
00606      IF(I.EQ.NSTEP)GO TO 575                                        199
00610      X(I)=X(I-1)-DX                                                 200
00611      GO TO 576                                                      201
00612  575 DX=HELIX1                                                      202
00613      DELF = DRK*DX                                                  203A
00614      X(I)=0.0                                                       203B
00615  576 S(I)=S(I-1)+DX                                                 204
00616      TANB = 2.632*X(I) + TANB0                                      205
00617      B = ATAN(TANB)                                                 206
```

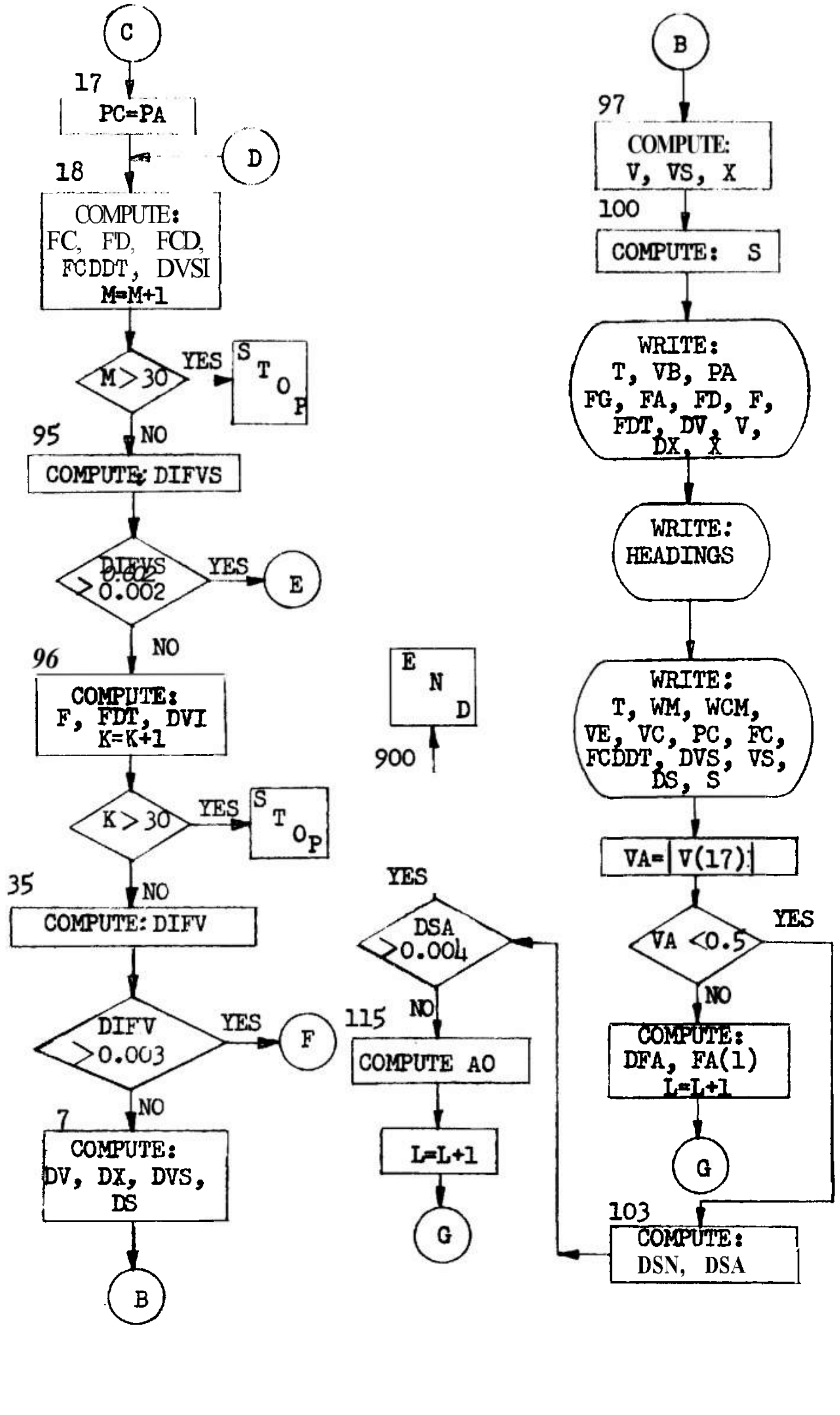
A–6. (Con't.)

```
00620        BDEG(I)=57.296*B                                                  207
00621        SNCN = SIN(B)+EMUR*COS(B)                                         208
00622        CNSN = COS(B)-EMUR*SIN(B)                                         209
00623        EM(I)=.00647 +(0.000345*TANB/(0.203*CNSN/SNCN-0.1149))            210
00624        F(I)=F(I-1)-DELF                                                  211
00625        E=E+0.25*(F(I-1)+F(I))*DX                                         212
00626        V(I)=SQRT(2.0*E/EM)                                               213
00627        Z=SQRT(F(I-1)**2+40.8*E)                                          214
00630        DT=0.4428*SQRT(EM)*(ASIN(F(I-1)/Z)-ASIN(F(I)/Z))                  215
00631        DTM(I)=1000.0*DT                                                  216
00632        TEPM(I)=TEPM(I-1)+DTM(I)                                          217
00633    550 CONTINUE                                                          218A
00635        DX=DX                                                             218B
00636        PRINT 551,(S(I),F(I),BDEG(I),EM(I),DTM(I),V(I),TEPM(I),I=24,NSTEP) 219
00652    551 FORMAT                                                    (IH1/   220
00652       1 15X,42H TABLE 4-9 COMPUTED DYNAMICS,COUNTERRECOIL/17X,37HBOLT LOC 221
00652       2KING DURING PARABOLA TRAVERSE// 2X,6HTRAVEL,4X,5HFORCE,6X,4HBETA, 222
00052       36X,5HMASS ,6X,6HDELTAT,4X,8HVELOCITY,5X,4HTIME,/3X,4HINCH,5X,5HPOU 223
00652       4ND,5X,6HDEGREE,5X,5H1000X,6X,6HMILSEC,5X,6HIN/SEC,5X,6HMILSEC//   224
00652       5     (F7.2,F10.2, F11.3, F11.5, F11.4, F11.2, FI1.4))              225
00653        STOP                                                              226
00654 .      END                                                               227
```

A–7. FLOW CHART FOR OPERATING CYLINDER



A–7. (Con't.)



C
17
PC=PA
D
18
COMPUTE:
FC, FD, FCD,
FCDDT, DVSI
M=M+1
M>30
YES
STOP
95
NO
COMPUTE: DIFVS
DIFVS >0.002
YES
E
96
NO
COMPUTE:
F, FDT, DVI
K=K+1
K>30
YES
STOP
35
NO
COMPUTE: DIFV
DIFV >0.003
YES
F
7
NO
COMPUTE:
DV, DX, DVS,
DS
B
END
900
YES
DSA >0.004
115
NO
COMPUTE AO
L=L+1
G
B
97
COMPUTE:
V, VS, X
100
COMPUTE: S
WRITE:
T, VB, PA
FG, FA, FD, F,
FDT, DV, V,
DX, X
WRITE:
HEADINGS
WRITE:
T, WM, WCM,
VE, VC, PC, FC,
FCDDT, DVS, VS,
DS, S
VA= V(17)
VA <0.5
YES
NO
COMPUTE:
DFA, FA(1)
L=L+1
G
103
COMPUTE:
DSN, DSA

A–8. LISTING FOR OPERATING CYLINDER PROGRAM

```
WIT   FOR REVMGD
COMPILATION BY UNIVAC 1107 FORTRAN I V DATED-     MAY 10,1966  F4010
THIS COMPILATION WAS DONE ON 06 JUN 68 AT 08:46:48

  MAIN PROGRAM            ENTRY POINT 000000

  STORAGE USED (BLOCK, NAME, LENGTH)

       0001   *CODE    000610
       0000   *DATA    001211
       0002   *BLANK   000000

  EXTERNAL REFERENCES (BLOCK, NAME)

       0003   NRDCS
       0004   NIO1$
       0005   NIO2$
       0006   NSTOP$
       0007   NPRTB
       0010   NEXP6$
       0011   NWDU$

  STORAGE ASSIGNMENT FOR VARIABLES (BLOCK, TYPE, RELATIVE LOCATION, NAME)

   0001   000036 1L      0000   000652 10F     0000   001133 101F    0001   000557 103L    0001   000004 1056
   0001   000014 113G    0001   000024 121G    0001   000235 14L     0001   000212 15L     0001   000072 151G
   0001   000303 17L     0001   000305 18L     0001   000436 301G    0001   000501 324G    0000   000650 350F
   0001   000105 4L      0001   000413 97L     0000   001006 98F     0000   001017 YYF     0000 R 000611 AO
   0000 R 000645 DFA     0000 R 000624 DIFV    0000 R 000643 DIFVS   0000 R 000566 DS      0000 R 000647 DSA
   0000 R 000646 DSN     0000 R 000636 DSP     0000 R 000640 DSX     0000 R 000614 DT      0000 R 000210 DV
   0000 R 000016 DVI     0000 R 000617 DVK     0000 R 000231 DVS     0000 R 000632 DVSI    0000 R 000633 DVSK
   0000 R 000627 DWC     0000 R 000336 DX      0000 R 000622 DXP     0000 R 000612 EME     0000 R 000314 F
   0000 R 000063 FA      0000 R 000524 FC      0000 R 000642 FCD     0000 R 000545 FCDDT   0000 R 000252 FD
   0000 R 000535 FDT     0000 R 000273 FG      0000 I 000607 I       0000 I 000613 K       0000 I 000610 L
   0000 I 000631 M       0000 R 000021 PA      0000 R 000503 PC      0000 R 000641 RVEC    0000 R 000146 S
   0000 R 000625 SP      0000 R 000637 SXP     0000 R 000634 S1      0000 R 000633 S2      0000 R 000000 T
   0000 R 000125 V       0000 K 000644 VA      0000 R 000042 VB      0000 R 000462 VC      0000 R 000613 VCO
   0000 R 000441 VE      0000 R 000167 VS      0000 R 000626 W       0000 K 000630 WC
   0000 R 000377 WM      0000 R 000104 X       0000 R 000623 XP      0000 R 000620 X1      0000 R 000620 WCM

 00101     1.        DIMENSION T(17),PA(17),VB(17),FA(17),X(17),V(17),S(17),VS(17),
 00101     2.       1DV(17),DVS(17),FD(17),FG(17),F(17),FDT(17),DX(17),WM(17),WCM(17),
 00101     3.       2VE(17),VC(17),PC(17),FC(17),FCDDT(17),DS(17)
 00103     4.        READ 350, (T(I), I=2,17)
 00111     5.        READ 350, (PA(I), I=2,17)
 00117     6.        READ 350, (VB(I), I=2,17)
 00125     7.    350 FORMAT ( 8F9.0)
 00126     8.    500 L = 1
 00127     9.        AO = 0.042
 00130    10.        FA(1) = 680.0
```

A–8. (Con't.)

```
00131    1-IF(L.GT.50)STOP                                                       13
00133    2 PRINT 10, AO                                                          14
00136      FD(1) = 85.0                                                          15
00137      X(1) = 0.0                                                            16
00140      V(1) = 0.0                                                            17
00141      S(1) = 0.0                                                            18
00142      VS(1) = 0.0                                                           I9
00143      T(I) = 0.0                                                            20
00144      DV(1) - 2.0                                                           a
00145      EME = 0.2285                                                          22
00146      VCO = 0.50                                                            23
00147   10 FORMAT(1H1/16X,74HTABLE 2-12 COMPUTED RECOIL AND OPERATING CYLINDE    24
00147     1R DATA FOR ORIFICE AREA OF F5.3,6H SQ IN//21X,86HAVERAGE      PROP    25
00147     2                   DRIVING    RESULT             DIFFER          DIF  26
00147     3FER/12X,105H LORE        BORE        GAS    ADAPTER    SPRING,   RECO 27
00147-    41L      RECOIL    RECOIL     RECOIL-    RECOIL     RECOIL/I17H  TIME  28
00147     5  VOLUME    PRESSURE    FORCE    FORCE    FORCE    FORCE     IMPUL    29
00147-    6SE      VEL         VEL        TRAVEL     TRAVEL/115H  MILSEC   CU IN 30
00147     7   PSI          LB         LB         L8         LB      LB-SEC  IN/SE 31
00147     8C     IN/SEC          IN           IN/)                               32
00150      DO 100  I=2,17                                                        33
00153      DT =(T(I)-T(I-1))/1000.0                                              34
00154      FG(I) = 0.515*PA(I)                                                   35
00155      K = 0                                                                 36
00156    3 DVI = DV(I-1)                                                         37
00157    4 DVK = DVI                                                             38
00160      X1 =  V(I-1)*DT                                                       39
00I61      X2 = 0.5*DVK*DT                                                       40
00162      DXP= X1 + X2                                                          41
00163      XP = DXP + X(I-1)                                                     42
00164      FA(I) = FA(I-1) + 1780.0*DXP                                          43
00165      IF(I.GT.7)GO TO 15                                                    44
00167    5 FD(I) = FD(I-1) + 40.0*DXP                                            45
00170      F(I) = FG(I)-FA(I)/0.45-FD(I)/0.80                                    46
00171      FDT(I) = F(I)*DT                                                      47
00172      DVI = 4.025*FDT (I)                                                   48
00173      K = K+1                                                               49
00174      IF(K.GT.30)STOP                                                       50
00176   a5 DIFV =  ABS((DVI-DVK)/DVI)                                            51
00177      IF(DIFV.GT.0.003)GO TO 4                                              52
00201    6 DV(I) = DVI                                                           53
00202      DX(I) = DXP                                                           54
00203      DVS(I) = DV(I)                                                        55
00204      DS(I) = DX(I)                                                         56
00205      SP = XP                                                               57
00206      WM(I) = 0.0                                                           58A
00207      WCM(1) = 0.0                                                          58B
00210      VE(1) = 0.0                                                           58C
0021I      VC(1) = 0.0                                                           58D
00212      PC(I) = 0.0                                                           58E
00213      GO TO 97                                                              59
00214   15 WM(I) = 1.92*AO*PA(I)                                                 60A
00215      W = WM(I)/1000.0                                                      60B
00216      DWC = W*DT                                                            61
00217      WCM(I) = WCM(I-1) + 1000.0*DWC                                        62A
00220      WC = WCM(I)/1000.0                                                    62B
00221      VE(I) = I4.0*WC*VB(I)                                                 63
00222      M = 0                                                                 64
```

A–8. (Con't.)

```
00223      13 DVSI = DVS(I-I)                                                  65
00224      14 DVSK = DVSI + DVI                                                66
00225         S1 =  VS(I-1)*DT                                                 67
00226         S2 = 0.5*DVSK*DT                                                 68
00227         DSP= S1 + S2                                                     69
00230         SP = DSP + S(I-1)                                             70+71
00231         SXP = SP - XP                                                    72
00232         DSX = DSP - DXP                                                  73
00233         VC(I) = VC0 + 1.767*SXP                                          74
00234         RVEC = (VE(I)/VC(I))**1.3                                        75
00235         IF(RVEC.GT.1.0)GO TO 17                                          76
00237      16 PC(I) = PA(I)*RVEC                                               77
00240         GO TO 18                                                         78
00241      17 PC(I) = PA(I)                                                    79
00242      18 FC(I) = 1.767*PC(I)                                              80
00243         FD(I) = FD(I-1) + 40.0*DSP                                       81
00244         FCD = FC(I) - FD(I)/0.80                                         82
00245         FCDDT(I) = FCD*DT                                                83
00246         DVSI = 35.127*FCDDT(I)                                           84
00247         M = M+I                                                          85
00250         IF(M.GT.30)STOP                                                  86
00252      95 DIFVS =  ABS((DVSI - DVSK+ DVI)/DVSI  )                          87
00253         IF(DIFVS.GT.0.002)GO TO 14                                       88
00255      96 F(I) = FG(I) - FA(I)/0.45 - FC(I)                                89
00256         FDT(I) = F(I)*DT                                                 90
00257         DVI = 4.546*FDT(I)                                               91
00260         K = K+1                                                          92
00261         IF(K.GT.30)STOP                                                  93
60263      35 DIFV =  ABS((DVI-DVK)/DVI)                                       94
00264         IF(DIFV.GT.0.003)GO TO 4                                         95
03266       7 DV(I) = DVI                                                      96
00267         DX(I) = DXP                                                      97
00270         DVS(I) = DVSI                                                    98
00271         DS(I) = DSP                                                      99
00272      97 V(I) = V(I-1) + DV(I)                                           100
00273         VS(I) = VS(I-1) + DVS(I)                                        101
00274         X(I) = XP                                                       102
00275     100 S(I) = SP                                                       103
00277        0WRITE(6,98)(T(I),VB(I),PA(I),FG(I),FA(I),FD(I),F(I),FDT(I),    104A
00277        1DV(I),V(I),DX(I),X(I),I=2,17)                                  104B
00320         WRITE(6,99)                                                     105
00322        0WRITE(6,101)(T(I),WM(I),WCM(I),VE(I),VC(I),PC(I),FC(I),FCDDT(I),  106A
00322        1DVS(I),VS(I),DS(I),S(I),I=2,17)                                106B
00343      98 FORMAT(F8.3,F10.1,2F10.0,2F9.0,F10.0,F10.2,2F10.1,2F10.4)       107
00344      990FORMAT(//14X,73HGAS     OPER     EQUIV     OPER     OPER     0  108
00344        1PER     OPER    DIFFER,13X,7H DIFFER/12X,105H FLOW      CYL     109
OD344        2 CYL        CYL        CYL      PISTON     CYL     SLIDE    SLIDE  110
00344        3   SLIDE      SLIDE/117H    TIME      RATE      GAS      VOLUME  111
00344        4VOLUME       PRES       FORCE     IMPULSE      VEL      VEL     TRAVEL  112
00344        5   TRAVEL/115H  MILSEC    LB/SECXM    LBXM      CU IN    CU IN  113
00344        6 PSI         LB        LB-SEC    IN/SEC     IN/SEC     IN      IN/ )  114
00345     101 FORMAT (F8.3,F10.1,3F10.3,2F10.0,F9.3,2F10.1,2F10.3)            115
00346         VA =  ABS(V(17))                                                116
00347         IF(VA.LT.0.5)GO TO 103                                          117
00351     105 DFA =0.30*EME*V(17)/(T(17)/1000.0)                              118
00352         FA(1) = FA(1)+DFA                                               119
00353         L = L+I                                                         120
00354         GO TO 1                                                         121
```

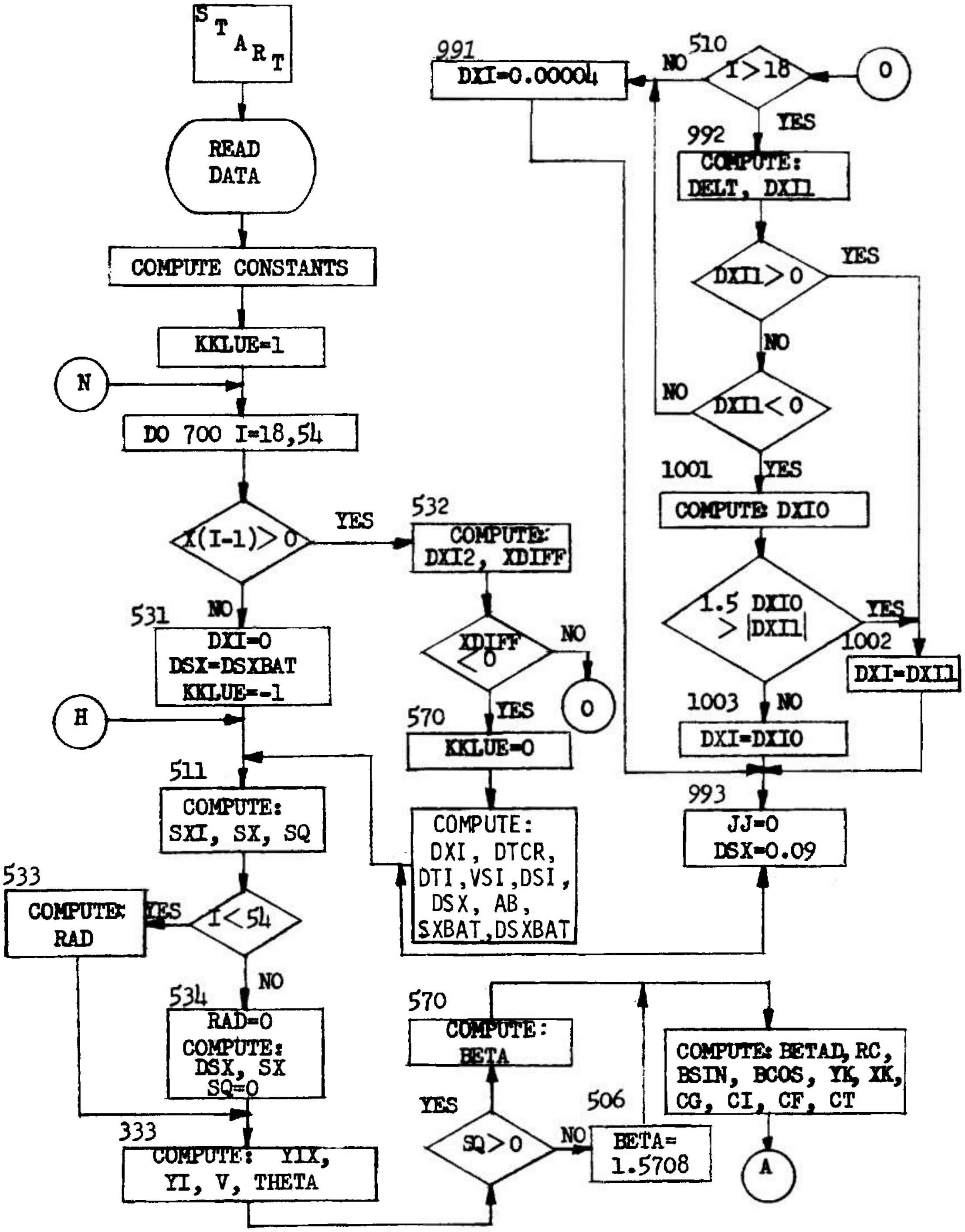
A—8. (Con't.)

```
00355   127.      103 DSN = S(17)-1.67                                  122
00356   128.          DSA =  ABS(DSN)                                   123
00357   129.          IF(DSA.LT.0.004)STOP                              124
00361   130.      115 AO = AO*(1.0 - DSN/1.67)                          125
00362   131.          L = L + 1                                         126
00363   132.          GO TO 1                                           127
00364   133.      900 STOP                                              128
00365   134.          END                                               129

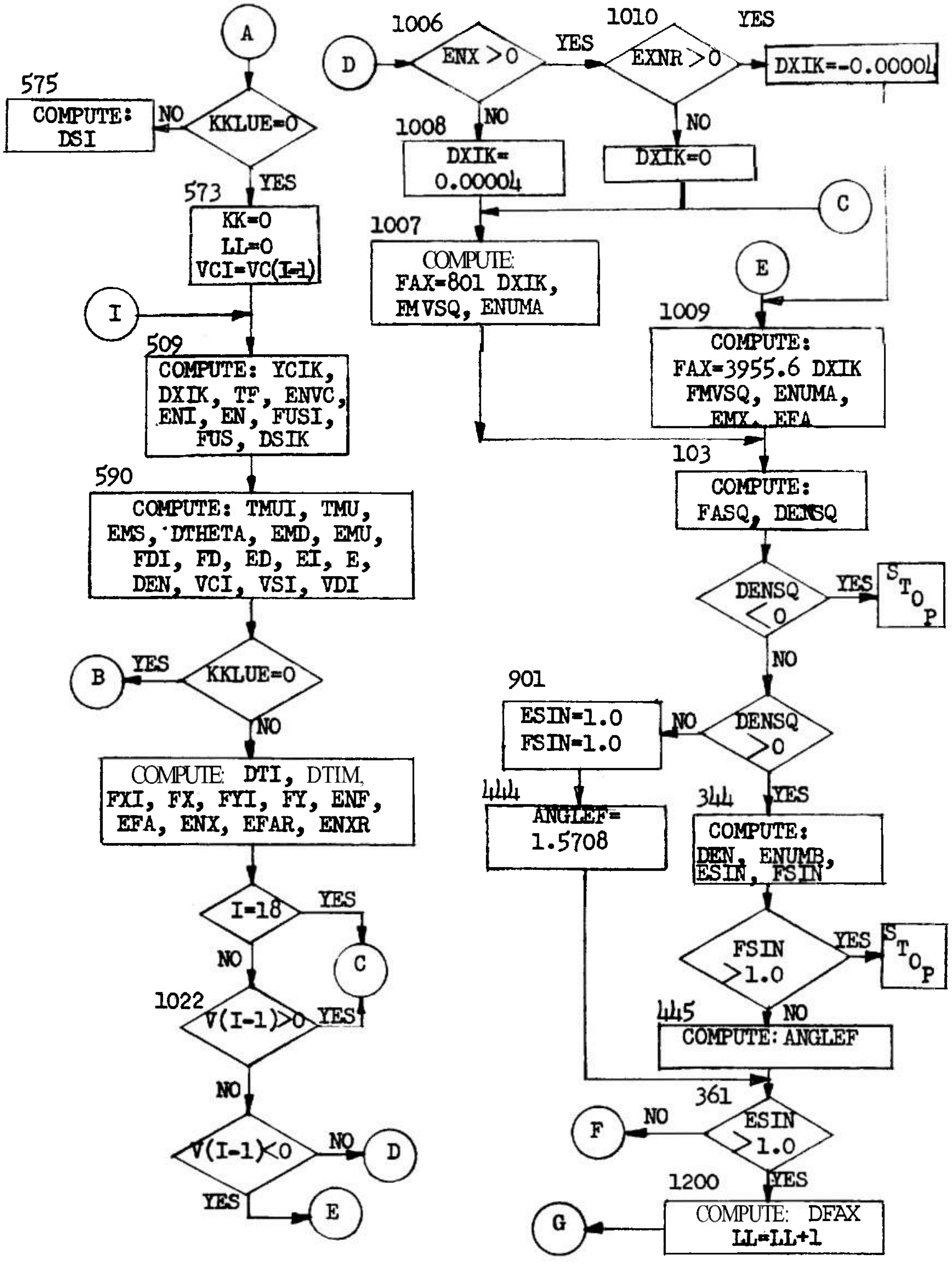
        END OF LISTING.      0 *DIAGNOSTIC* MESSAGE(S).
 PHASE 1 TIME =    2 SEC.
 PHASE 2 TIME =    0 SEC.
 PHASE 3 TIME =    3 SEC.
 PHASE 4 TIME =    0 SEC.
 PHASE 5 TIME =    2 SEC.
 PHASE 6 TIME =    2 SEC.

 TOTAL COMPILATION TIME =   9  SEC
```

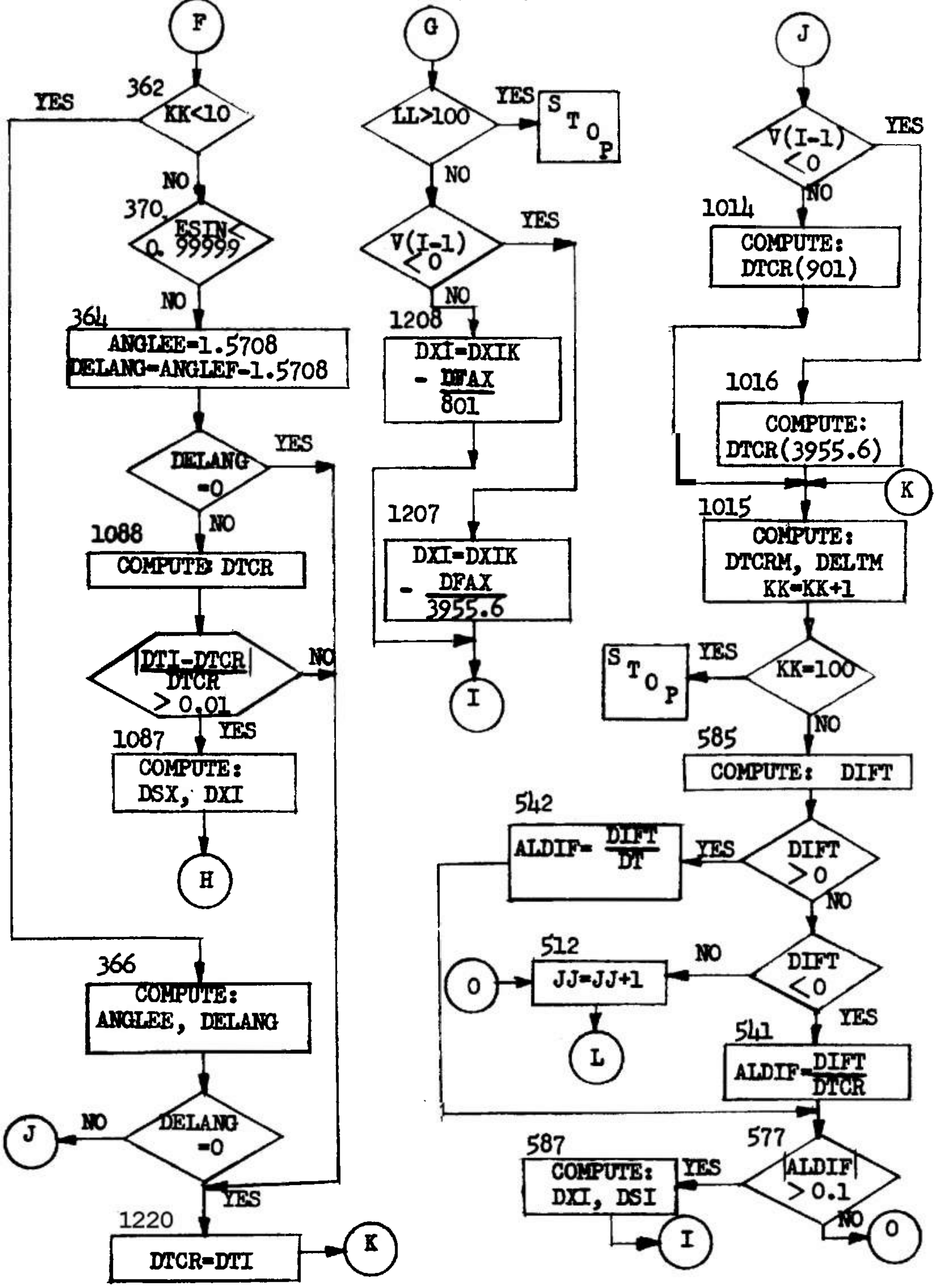
A–9. FLOW CHART FOR CAM AND DRUM DYNAMICS DURING RECOIL



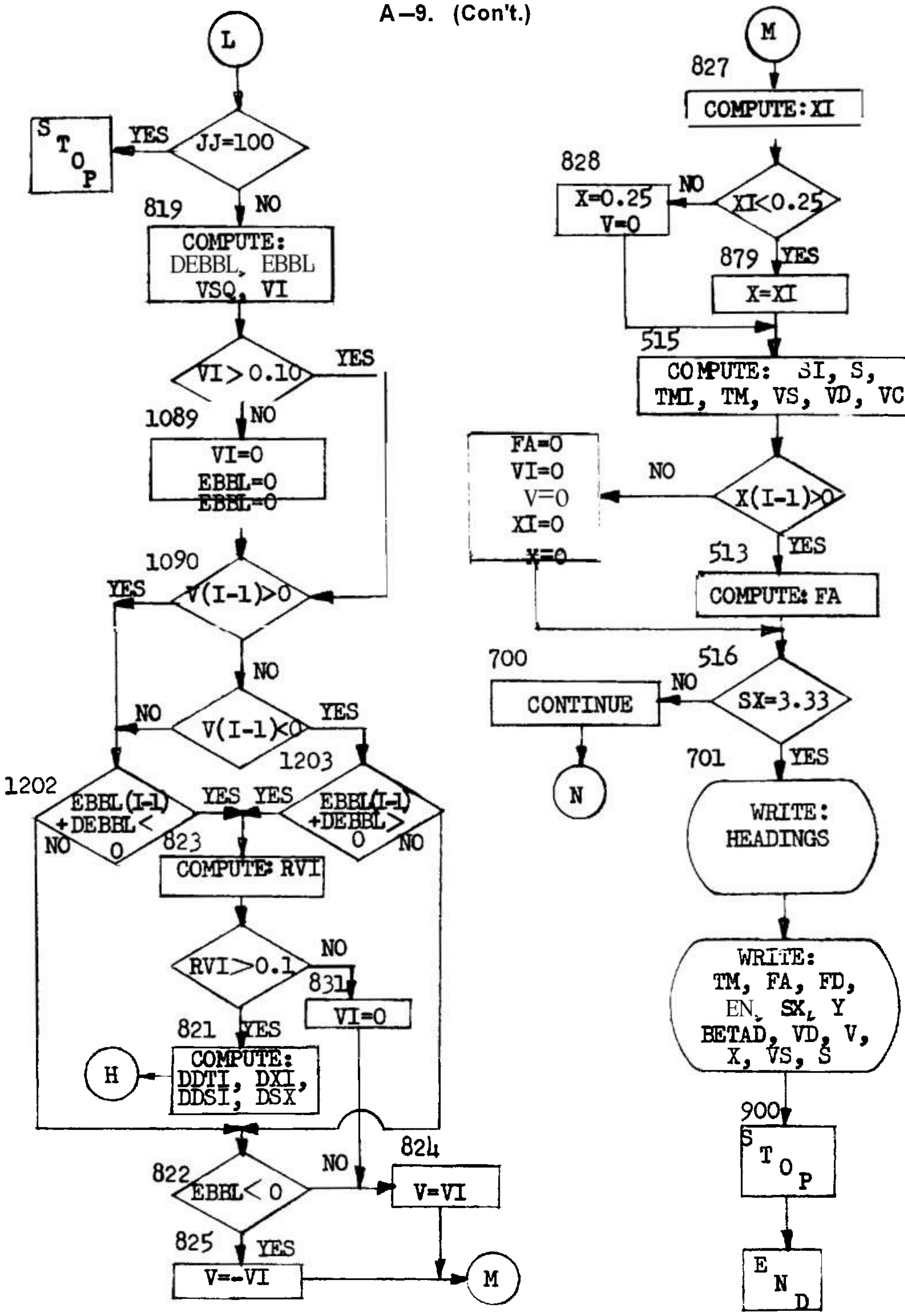
A—9. (Con't.)


A
575
COMPUTE: DSI
NO
KKLUE=0
YES
573
KK=0
LL=0
VCI=VC(I-1)
I
509
COMPUTE: YCIK, DXIK, TF, ENVC, ENI, EN, FUSI, FUS, DSIK
590
COMPUTE: TMUI, TMU, EMS, DTHETA, EMD, EMU, FDI, FD, ED, EI, E, DEN, VCI, VSI, VDI
B
YES
KKLUE=0
NO
COMPUTE: DTI, DTIM, FXI, FX, FYI, FY, ENF, EFA, ENX, EFAR, ENXR
I=18
YES
NO
C
1022
V(I-1)>0
YES
NO
V(I-1)<0
NO
D
YES
E
1006
D
ENX >0
YES
1010
EXNR >0
YES
DXIK=-0.00004
1008
NO
DXIK= 0.00004
NO
DXIK=0
C
1007
COMPUTE: FAX=801 DXIK, FMVSQ, ENUMA
E
1009
COMPUTE: FAX=3955.6 DXIK FMVSQ, ENUMA, EMX, EFA
103
COMPUTE: FASQ, DENSQ
DENSQ <0
YES
STOP
NO
901
ESIN=1.0
FSIN=1.0
NO
DENSQ >0
444
ANGLEF= 1.5708
344
YES
COMPUTE: DEN, ENUMB, ESIN, FSIN
FSIN >1.0
YES
STOP
445
NO
COMPUTE: ANGLEF
361
F
NO
ESIN >1.0
1200
YES
G
COMPUTE: DFAX LL=LL+1

A—9. (Con't.)


F
362
KK<10
YES
NO
370
ESIN< 0.99999
NO
364
ANGLEE=1.5708
DELANG=ANGLEF-1.5708
DELANG =0
YES
NO
1088
COMPUTE: DTCR
|DTI-DTCR| / DTCR > 0.01
NO
YES
1087
COMPUTE: DSX, DXI
H
366
COMPUTE: ANGLEE, DELANG
DELANG =0
NO
J
YES
1220
DTCR=DTI
K
G
LL>100
YES
STOP
NO
V(I-1) <0
YES
NO
1208
DXI=DXIK - DFAX/801
1207
DXI=DXIK - DFAX/3955.6
I
J
V(I-1) <0
YES
NO
1014
COMPUTE: DTCR(901)
1016
COMPUTE: DTCR(3955.6)
K
1015
COMPUTE: DTCRM, DELTM KK=KK+1
KK=100
YES
STOP
NO
585
COMPUTE: DIFT
542
ALDIF= DIFT/DT
YES
DIFT >0
NO
512
JJ=JJ+1
NO
DIFT <0
YES
O
L
541
ALDIF=DIFT/DTCR
577
|ALDIF| >0.1
YES
NO
587
COMPUTE: DXI, DSI
I
O

A–9. (Con't.)


L
JJ=100
YES
STOP
NO
819
COMPUTE: DEBBL, EBBL VSQ, VI
VI > 0.10
YES
1089
NO
VI=0
EBBL=0
EBBL=0
1090
V(I-1)>0
YES
NO
V(I-1)<0
NO
YES
1202
EBBL(I-1) +DEBBL< 0
YES
NO
1203
EBBL(I-1) +DEBBL> 0
YES
NO
823
COMPUTE: RVI
RVI>0.1
NO
831
VI=0
821
YES
COMPUTE: DDTI, DXI, DDSI, DSX
H
822
EBBL< 0
NO
824
V=VI
825
YES
V=-VI
M
M
827
COMPUTE: XI
828
XI<0.25
NO
X=0.25 V=0
879
YES
X=XI
515
COMPUTE: SI, S, TMI, TM, VS, VD, VC
X(I-1)>0
NO
FA=0 VI=0 V=0 XI=0 X=0
513
YES
COMPUTE: FA
700
516
SX=3.33
NO
CONTINUE
N
701
YES
WRITE: HEADINGS
WRITE: TM, FA, FD, EN, SX, Y BETAD, VD, V, X, VS, S
900
STOP
END

A–10. LISTING FOR CAM AND DRUM DYNAMICS DURING RECOIL

```
0000 R 002331 FDI    0000 R 002347 FMVSQ  0000 R 002354 FSIN   0000 R 001210 FUS    0000 R 002322 FUSI
0000 R 002002 FX     0000 R 002337 FX1    0000 R 002070 FY     0000 R 002340 FYI    0000 I 002255 I
0000 I 002266 JJ     0000 I 002312 KK     0000 I 002254 KKLUE  0000 I 002313 LL     0000 R 002277 RAD
0000 R 002303 RC     0000 R 002244 RD     0000 R 002372 RVI    0000 R 000066 S      0000 K 002376 SI
0005 R 000000 SIN    0000 R 002276 SQ     0003 R 000000 SQRT   0000 R 000660 SX     0000 R 002274 SXBAT
0000 R 002275 SXI    0000 R 002317 TF     0000 R 002245 16     0000 R 001452 THETA  0000 R 000572 TM
0000 R 002377 TMI    0000 R 001276 TMU    0000 R 002324 TMUI   0000 R 000154 V      0000 R 001540 VC
0000 R 002314 VCI    0000 R 002315 VCIK   0000 R 001626 VU     0000 R 002335 VDI    0000 R 002371 VI
0000 R 000242 VS     0000 R 002271 VSI    0000 R 002370 VSQ    0000 R 000000 X      0000 R 002262 XDIFF
0000 R 002375 XI     0000 R 002305 XK     0000 R 000746 Y      0000 R 002301 YI     0000 R 002300 YIX
0000 R 002304 YK
```

```
00101          0DIMENSION X(54),S(54),V(54),VS(54),FA(54),FD(54),EN(54),TM(54),        1
00101          1SX(54),Y(54),BETAD(54),BSIN(54),FUS(54),TMU(54),E(54),THETA(54),       2
00101          2VC(54),VD(54),BCOS(54),FX(54),FY(54),EBBL(54)                          3
00103           READ(5,360)RD,TG,EMR,DIE,CY,CX,DF,EMSL,X(16),X(17),S(17),VS(17),       4
00103          1VC(17),FA(17),FD(17),TM(17)                                            5
00125      360 FORMAT (8F10.0)                                                         6
00126           V(17) = 0.0                                                            7
00127           EN(17) = 0.0                                                           8
00130           SX(17) = 0.0                                                           9
00131           E(17) = 0.5*EMSL*VS(17)**2                                             10
00132           FX(17) = 0.0                                                           11
00133           FY(17) = 0.0                                                           12
00134           Y(17) = 0.0                                                            13
00135           FUS(17) = 0.0                                                          14
00136           TMU(17) = TG                                                           15
00137           EBBL(17)=0.0                                                           16
00140           KKLUE = 1                                                              17
00141           DO 700  I = 18,54                                                      18
00144           IF(X(I-1).GT.0.0)GO TO 532                                             19
00146      531 DXI = 0.0                                                               20
00147           DSX = DSXBAT                                                           21
00150           KKLUE = -1                                                             22
00151           GO TO 511                                                              23
00152      532 DXI2 = 2.0*(X(I-2)-X(I-1))                                              24
00153           XDIFF=X(I-1)-DXI2                                                      25
00154           IF(XDIFF.LT.0.0)GO TO 570                                              26
00156      510 IF(I.GT.18)GO TO 992                                                    27
00160      991 DXI = 0.00004                                                           28
00161           GO TO 993                                                              29
00162      992 DELT = (TM(I-1)-TM(I-2))/1000.0                                         30
00163           DXI1 = V(I-1)*DELT                                                     31
00164           IF(DXI1.GT.0.  GO TO 1002                                              32
00166           IF(DXI1.LT.0.0)GO TO 1001                                              33
00170           GO TO 991                                                              34
00171     1001 DXI0 = 0.25-X(I-1)                                                      35
00172           IF(1.5*DXI0.GT.ABS(DXI1))GO TU 1002                                    36
00174     1003 DXI = DXI0                                                              37
00175           GO TO 993                                                              38
00176     1002 DXI = DXI1                                                              39
00177      993 JJ=0                                                                    40
00200           DSX = 0.09                                                             41A
00201           GO TO 511                                                              41B
00202      570 KKLUE = 0                                                               42
```

A–10. (Con't.)

```
00203          DXI   = X(I-1)
00204          DTCR = DXI/V(I-1)
00205          DTI = DTCR
00206          VSI = VS(I-1)
00207          DSI = VSI*DTI
00210          DSX = DSI + DXI
00211          AB = 1
00212          SXBAT = 54.0 - AB
00213          DSXBAT = (5.0 - SX(I-1) - DSX)/SXBAT
00214      511 SX I = SX(I-1) + DSX
00215          SX(I) = SXI
00216          SQ = 11.0889 - SX(I)**2
00217          IF(I.LT.54)GO TO 533
00221      534 RAD = 0.0
00222          DSX=3.33-SX(53)
00223          SX(I)=3.33
00224          SQ = 0.0
00225          GO TO 333
00226      533 RAD=SQRT(SQ)
00227      333 YIX= 0.6456*RAD
00230          YI   = 2.15 - YIX
00231          Y(I) = YI
00232          THETA(I) = 19.0987*Y(I)
00233          IF(SQ.GT.0.0)GO TO 507
00235      506 BETA = 1.5708
00236          GO TO 508
00237      507 BETA =  ATAN(0.6456*SX(I)/RAD)
00240      508 BETAD(I) = 57.296*BETA
00241          RC = (11.0889 - 0.5831*SX(I)**2)**1.5/7.1595
00242          BSIN(I) =  SIN(BETA)
00243          BCOS(I)=  COS(BETA)
00244          YK = BCOS(I)- 0.05*BSIN(I)
00245          XK = BSIN(I) + 0.05*BCOS(I)
00246          CG = CY*YK - CX*XK
00247          CI = DIE*BCOS(I)/(RD*RC*CG)
00250          CF = 0.15*YK
00251          CT = CX*XK + 0.1*YK
00252          IF(KKLUE.EQ.0)GO TO 573
00254      575 DSI = DSX - DXI
00255      573 KK = 0
00256          LL = 0
00257          VCI = VC(I-1)
00260      569 VCIK = VCI
00261          DXIK = DXI
00262          TF = TG/CG
00263          ENVC = CI*VCIK**2
00264          ENI = ENVC + TF
00265          EN(I) = ENI
00266          FUSI    = CF*EN(I)
00267          FUS(I) = FUSI
00270          DSIK = DSI
00271      590 TMUI    = CT*EN(I) + TG
00272          TMU(I)= TMUI
00273          EMS = 0.5*(FUS(I-1) + FUS(I))*DSIK
00274          DTHETA = (Y(I) - Y(I-1))/RD
00275          EMU = 0.5*(TMU(I-1) + TMU(I))*DTHETA
00276          EMU = EMS + EMD
00277          FDI    = FI(I-1) + 40.0*DSIK
```

A–10. (Con't.)

```
00300   102.         FD(I) = FDI                                              101
00301   103.         ED = (FD(I-1) + FD(I))*0.625 *DSIK                       102
00302   104.         EI = E(I-1) - EMU - ED                                   103
00303   105.         E(I) = EI                                                104
00304   106.         DEN = 0.5*(EMSL*BCOS(I)**2+DIE*(BSIN(I)/RD)**2)          105
00305   107.         VCI =  SQRT(E(I)/DEN)                                    106
00306   108.         VSI = VCI*BCOS(I)                                        107
00307   109.         VDI = VCI*BSIN(I)                                        108
00310   110.         IF(KKLUE.EQ.0)GO TO 512                                  109
00312   111.     572 DTI = DSIK/( (VSI + VS(I-1))*0.5)                        110
00313   112.         DTIM = 1000.0*DTI                                        111
00314   113.         FXI = EN(I)*XK                                           112
00315   114.         FX(I) = FXI                                              113
00316   115.         FYI = EN(I)*YK                                           114
00317   116.         FY(I) = FYI                                              115
00320   117.         ENF = 0.5*(FX(I-1)+FX(I))                                116
00321   118.         EFA = 0.45*FA(I-1)                                       117
00322   119.         ENX = ENF-EFA                                            118
00323   120.         EFAR = FA(I-1)/0.45                                      119
00324   121.         ENXR = ENF - EFAR                                        120
00325   122.         IF(I.EQ.16)GO TO 1007                                    121
00327   123.    1022 IF(V(I-1).GT.0.0)GO TO 1007                              122
00331   124.         IF(V(I-1).LT.0.0)GO TO 1009                              123
00333   125.    1006 IF(ENX.GT.0.0)GO TO 1010                                 124
00335   126.    1008 DXIK=0.00004                                             125
00336   127.         GO TO 1007                                               126
00337   128.    1010 IF(ENXR.GT.0.0)GO TO 1012                                127
00341   129.    1011 DXIK = 0.0                                               128
00342   130.         GO TO 1007                                               129
00343   131.    1012 DXIK=-0.00004                                            130
00344   132.    1007 FAX = 801.0*DXIK                                         131
00345   133.         FMVSQ = 801.0*EMR*V(I-1)**2                              132
00346   134.         ENUMA= FAX + ENX                                         133
00347   135.         GO TO 1013                                               134
00350   136.    1009 FAX = 3955.6*DXIK                                        135
00351   137.         FMVSQ = 3955.6*V(I-1)**2*EMR                             136
00352   138.         ENUMA = FAX + ENXR                                       137
00353   139.         ENX = ENXR                                               138
00354   140.         EFA = EFAR                                               139
00355   141.    1013 FASQ = ENX**2                                            140
00356   142.         DENSQ = FASQ + FMVSQ                                     141
00357   143.         IF(DENSQ.LT.0.0)STOP                                     142
00361   144.         IF(DENSQ.GT.0.0)GO TO 344                                143
00363   145.     901 ESIN = 1.0                                               144
00364   146.         FSIN =1.0                                                145
00365   147.         GO TO 444                                                146
00366   148.     344 DENOM =   SQRT(DENSQ)                                    147
00367   149.         ENUMB = ENX                                              148
00370   150.         ESIN  =   ABS(ENUMA/DENOM)                               149
00371   151.         FSIN  =   ABS(ENUMB/DENOM)                               150
00372   152.     356 IF(FSIN.GT.1.0)STOP                                      151
00374   153.     445 ANGLEF=ASIN(FSIN)                                        152
00375   154.         GO TO 361                                                153
00376   155.     444 ANGLEF = 1.5708                                          154
00377   156.     361 IF(ESIN.GT.1.0)GO TO 1200                                155A
00401   157.         GO TO 362                                                155B
00402   158.    1200 DFAX   = (ENUMA - DENOM)                                 156
00403   159.         LL = LL+1                                                157
```

A– 10. (Con't.)

```
00404          IF(LL.EQ.100)STOP                          158
00406   1201.  IF(V(I-1).LT.0.0)GO TO 1207                159
00410   1208   DXI = DXIK - DFAX/801.0                    160
00411          GO TO 509                                  161
00412   1207   GXI = DXIK - DFAX/3955.6                   162
00413          GO TO 509                                  163
00414    362   IF(KK.LT.10)GO TO 366                      164
00416    370   IF(ESIN.LT.0.999990)GO TO 366              165
00420    364   ANGLEE = 1.5708                            166
00421          DELANG=ANGLEF-1.5708                       167
00422          IF(DELANG.GT.0.0)GO TO 1088                168
00424          IF(DELANG.LT.0.0)GO TO 1088                169
00426          GO TO 1220                                 170
00427   1088   DTCR= (SQRT(EMR/801.0))* ABS(DELANG)       171
00430          IF((ABS(DTI-DTCR)/DTCR).GT.0.01)GO TO 1087 172A
00432          GO TO 1220                                 172B
00433   1087   DSX=DTCR*(VS(I-1)+VSI)/2.0                 173
00434          DXI =V(I-1)*DTCR/2.0                       174
00435          GO TO 511                                  175
00436    366   ANGLEE=ASIN(ESIN)                          176
00437    446   DELANG = ANGLEF - ANGLEE                   111
00440          IF(DELANG.GT.0.0)GO TO 1017                178
00442          IF(DELANG.LT.0.0)GO TO 1017                179
00444   1220   DTCR=DTI .                                 180
00445          GO TO 1015                                 181
00446   1017   IF(V(I-1).LT.0.0)GO TO 1016                182
00450   1014   DTCR = (SQRT(EMR/801.0))* ABS(DELANG)      183
00451          GO TO 1015                                 184
00452   1016   DTCR= (SQRT(EMR/3955.67))* ABS(DELANG)     185
00453   1015   DTCRM = 1000.0*DTCR                        186
00454          DELTM = DTIM - DTCRM                       187
00455          KK = KK + 1                                188
00456          IF(KK.EQ.100)STOP                          189
00460    585   DIFT = DTI - DTCR                          190
00461          IF(DIFT.GT.0.0)GO TO 542                   191
00463          IF(DIFT.LT.0.0)GO TO 541                   192
00465          GO TO 512                                  193
00466    541   ALDIF =     (DIFT/DTCR)                    194
00467          GO TO 5rl                                  195
00470    542   ALDIF =     (DIFT/DTI)                     196
00471    577   IF(ABS(ALDIF).GT.0.01)GO TO 587            197A
00473          GO TO 512                                  197B
00474    587   DXI = DXIK*(1.0 + ALDIF/4.0)               198
00475          DSI = DSX + DXI                            199
00476          GO TO 509                                  200
00477    512   JJ=JJ+1                                    201
00500          IF(JJ.EQ.100)STOP                          202
00502    819   DEBBL =  ABS(DXIK)*(-ENUMA)                203
00503          EBBL(I) = EBBL(I-1) + DEBBL                204
00504          VSQ = 2.0*EBBL(I)/EMR                      20s
00505          VI = SQRT(ABS(VSQ))                        206
00506          IF(VI.GT.0.10)GO TO 1090                   207
00510   1089   VI=0.0                                     208
00511          EBBL(I)=0.0                                209
00512   1090   IF(V(I-1).GT.0.0)GO TO 1202                210
00514          IF(V(I-1).LT.0.0)GO TO 1203                211
00516          GO TO 822                                  212
00517   1202   IF(EBBL(I-1)+DEBBL.LT.0.0)GO TO 823        213
```

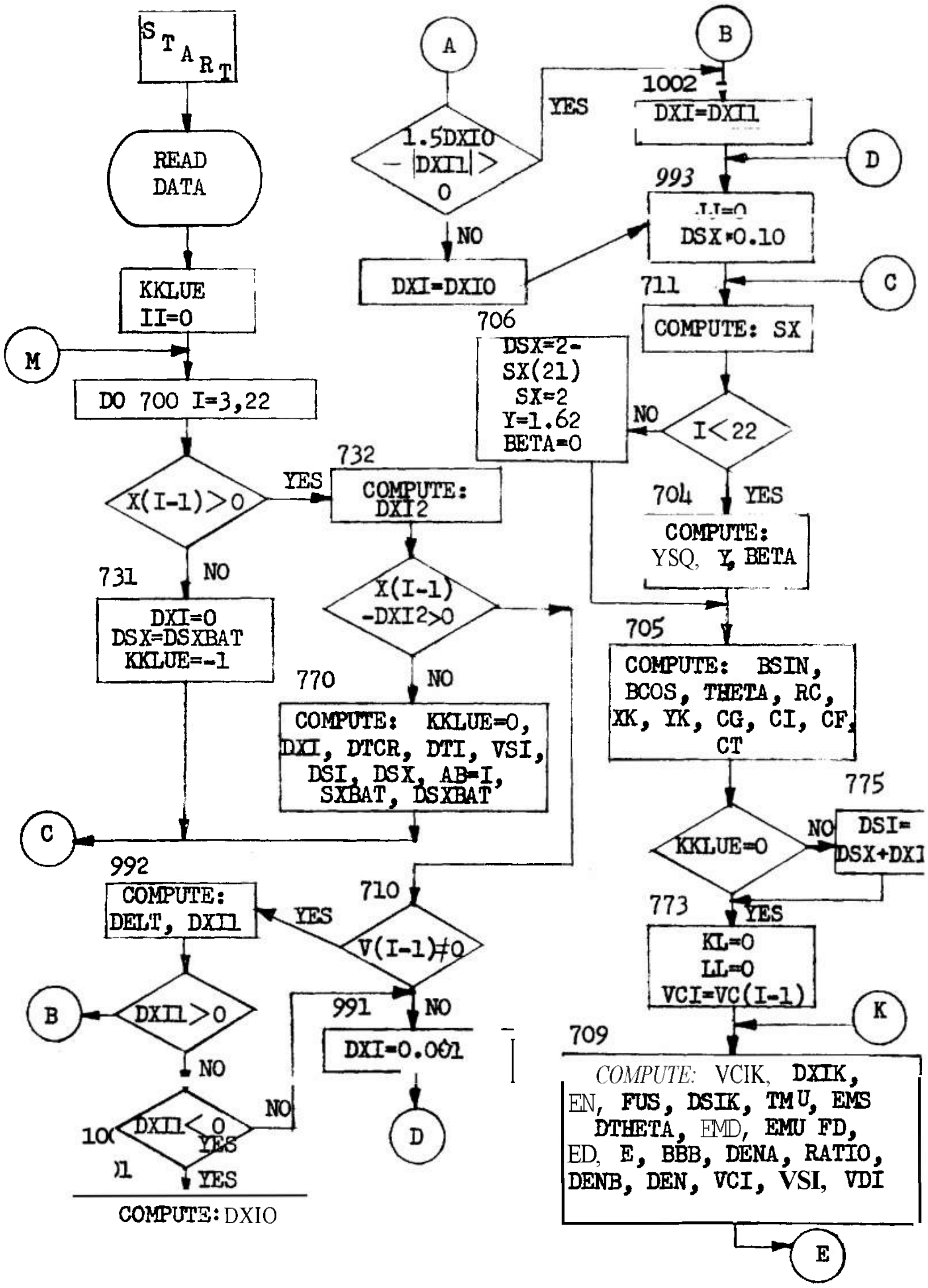
D-WW

A-10. (Con't.)

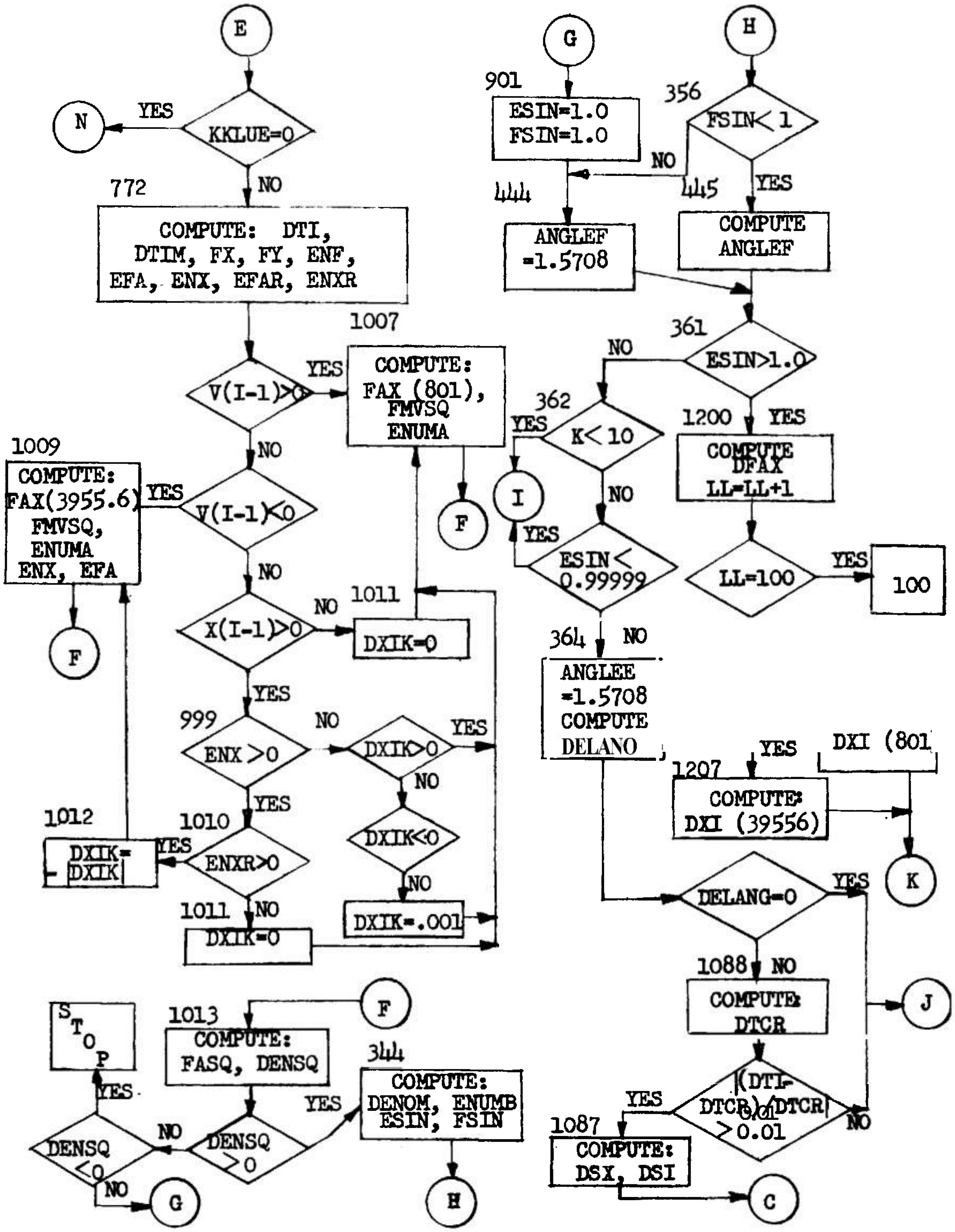
```
00521        GO TO 822
00522   1203 IF((EBBL(I-1)+DEBBL).GT.0.0)GO TO 823
00524        GO TO 822
00525    823 RVI = VI/ABS (V(I-1))
00526        IF(RVI.GT.0.10)GO TO 821
00530    831 VI = 0.0
00531        GO TO 824
00532    821 DDTI = DTI*(1.0 - ( ABS(V(I-1))/(( ABS(V(I-1)) + VI)*4.0)))
00533        DXI = V(I-1)*DDTI
00534        DDSI = VS(I-1)*DDTI
00535        DSX = DDSI + DXI
00536        GO TO 511
00537    822 IF(EBBL(I).LT.0.0)GO TO 825
00541    824 V(I) = VI
00542        GO TO 827
00543    825 V(I) = -VI
00544        [illegible]
00545        IF(XI.LT.0.25)GO TO 829
00547    828 X(I) = 0.25
00550        V(I) = 0.0
00551        GO TO 515
00552    829 X(I) = XI
00553    515 SI = S(I-1) + DSIK
00554        S(I) = SI
00555        TMI = TM(I-1)+ 1000.0*DTI
00556        TM(I) = TMI
00557        VS(I) = VSI
00560        VD(I) = VDI
00561        VC(I) = VCI
00562        IF(X(I-1).GT.0.0)GO TO 513
00564    514 FA(I) = 0.0
00565        VI = 0.0
00566        V(I) = VI
00567        XI = 0.0
00570        X(I) = XI
00571        GO TO 516
00572    513 FAI = 880.0 + 1780.0*X(I)
00573        FA(I) = FAI
00574    516 IF(SX(I).GT.3.33)GO TO 701
00576    700 CONTINUE
00600    701 WRITE(6,110)
00602    110 FORMAT(1H1/38X,46HTABLE 5-13 CAM AND DRUM DYNAMICS DURING RECOIL//
00602       113X,46HRECOIL    DRIVING    NORMAL    AXIAL    PERIPH,22X,17HCOUNT
00602       2ER  COUNTER/12X,105H ADAPTER   SPRING     CAM      CAM      CAM
00602       3    CAM      UKUM     RECOIL    RECOIL    SLIDE    SLIDE/
00602       43X,114H TIME      FORCE     FORCE     FORCE    TRAVEL   TRAVEL
00602       5 SLOPE     VEL       VEL      POSITION    VEL     TRAVEL/
00602       62X,113H MILSEC     LB        LB        LB       IN       IN
00602       7 DEGREE    IN/SEC    IN/SEC     IN      IN/SEC    IN/)
00603        WRITE(6,702)(TM(I),FA(I),FD(I),EN(I),SX(I),Y(I),BETAD(I),VD(I),
00603       1V(I),X(I),VS(I),S(I),I=18,54)
00624    702 FORMAT (F10.5,F9.0,F9.1,F11.0,2F10.4,F8.1,F10.1,2F10.4,F10.1,
00624       1F10.3)
00625    900 STOP
00626        END

           END OF LISTING.      0 *DIAGNOSTIC* MESSAGE(S).
PHASE 1 TIME =    4 SEC.
```

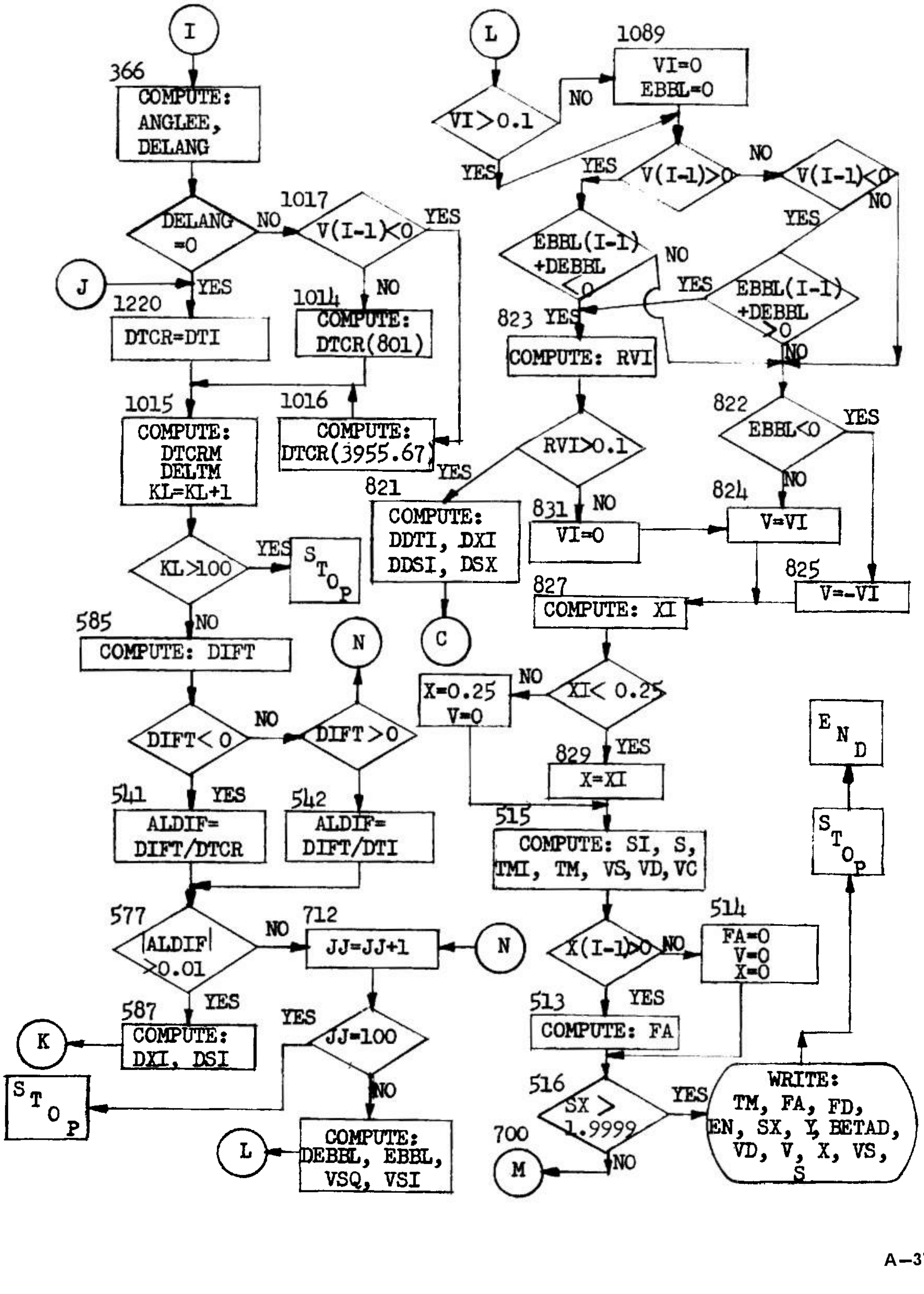
A–11. FLOW CHART FOR CAM AND DRUM DYNAMICS DURING COUNTERRECOIL



A–11. (Con't.)


E
KKLUE=0
YES
N
NO
772
COMPUTE: DTI, DTIM, FX, FY, ENF, EFA, ENX, EFAR, ENXR
V(I-1)>0
YES
1007
COMPUTE: FAX (801), FMVSQ ENUMA
F
NO
1009
COMPUTE: FAX(3955.6) FMVSQ, ENUMA ENX, EFA
V(I-1)<0
F
X(I-1)>0
1011
DXIK=0
999
ENX >0
DXIK>0
DXIK<0
DXIK=.001
1012
DXIK= -|DXIK|
1010
ENXR>0
1011
DXIK=0
G
901
ESIN=1.0 FSIN=1.0
444
ANGLEF =1.5708
H
356
FSIN< 1
445
COMPUTE ANGLEF
361
ESIN>1.0
362
K< 10
I
ESIN< 0.99999
364
ANGLEE =1.5708 COMPUTE DELANO
1200
COMPUTE DFAX LL=LL+1
LL=100
100
DXI (801
1207
COMPUTE: DXI (39556)
K
DELANG=0
J
1088
COMPUTE: DTCR
(DTI-DTCR)/DTCR >0.01
1087
COMPUTE: DSX, DSI
C
1013
COMPUTE: FASQ, DENSQ
DENSQ >0
DENSQ <0
STOP
344
COMPUTE: DENOM, ENUMB ESIN, FSIN
YES
NO

A–11. (Con't.)


I
366
COMPUTE: ANGLEE, DELANG
DELANG =0
NO
1017
V(I-1)<0
YES
J
YES
1220
DTCR=DTI
NO
1014
COMPUTE: DTCR(801)
1015
COMPUTE: DTCRM DELTM KL=KL+1
1016
COMPUTE: DTCR(3955.67)
KL>100
YES
STOP
585
NO
COMPUTE: DIFT
N
DIFT< 0
NO
DIFT >0
541
YES
ALDIF= DIFT/DTCR
542
ALDIF= DIFT/DTI
577
\|ALDIF\| >0.01
NO
712
JJ=JJ+1
N
587
YES
COMPUTE: DXI, DSI
K
JJ=100
YES
STOP
NO
COMPUTE: DEBBL, EBBL, VSQ, VSI
L
L
1089
VI=0 EBBL=0
VI>0.1
NO
YES
V(I-1)>0
YES
NO
V(I-1)<0
NO
YES
EBBL(I-1) +DEBBL <0
NO
YES
EBBL(I-1) +DEBBL >0
NO
823 YES
COMPUTE: RVI
822
EBBL<0
YES
RVI>0.1
YES
821
COMPUTE: DDTI, DXI DDSI, DSX
C
831
NO
VI=0
824
NO
V=VI
825
V=-VI
827
COMPUTE: XI
XI< 0.25
NO
X=0.25 V=0
829
YES
X=XI
515
COMPUTE: SI, S, TMI, TM, VS, VD, VC
X(I-1)>0
NO
514
FA=0 V=0 X=0
513
YES
COMPUTE: FA
516
SX > 1.9999
YES
WRITE: TM, FA, FD, EN, SX, Y, BETAD, VD, V, X, VS, S
700
NO
M
STOP
END

A–12. LISTING FOR CAM AND DRUM DYNAMICS DURING COUNTERRECOIL

0000 R 001040 FASQ	0000 R 001035 FAX	0000 R 000306 FD	0000 R 001036 FMVSQ	0000 R 001045 FSIN
0000 R 000362 FUS	0000 R 000642 FX	0000 R 000670 FY	0000 I 000756 I	0000 I 000755 II
0000 I 000774 JJ	0000 I 000754 KKLUE	0000 I 001006 KL	0000 I 001007 LJ	0000 R 001023 RATIO
0000 R 000777 RC	0000 R 000744 RO	0000 R 001061 RVI	0000 R 000054 S	0000 R 001065 SI
0005 R 000000 SIN	0003 R 000000 SQRT	0000 R 000102 SX	0000 R 000770 SXBAT	0000 R 000745 TG
0000 R 000130 THETA	0000 R 000436 TM	0000 R 001066 TMI	0000 R 000410 TMU	0000 R 000512 V
0000 R 000540 VC	0000 R 001010 VCI	0000 R 001011 VCIK	0000 R 000614 VD	0000 R 001026 VDI
0000 R 001060 VI	0000 R 000566 VS	0000 R 000765 VSI	0000 R 001057 VSQ	0000 R 000000 X
0000 R 001064 XI	0000 R 001000 XK	0000 R 000026 Y	00C0 R 001001 YK	0000 R 000776 YSQ

```
00101          0DIMENSION X(22),Y(22),S(22),SX(22),THETA(22),BETAD(22),BSIN(22),
00101          1BCOS(22),FA(22),FD(22),EN(22),FUS(22),TMU(22),TM(22),E(22),V(22),
00101          2VC(22),VS(22),VD(22),FX(22),FY(22),EBBL(22)
00103           READ(5,370)RD,TG,EMR,DIE,CY,CX,DF, EMSLR,X(2),S(2),SX(2),
00103          1FA(2),FD(2),EN(2),V(2),VC(2),VS(2),FX(2),FY(2),FUS(2),VD(2),E(2),
00103          2TM(2),TMU(2),Y(2),BETAD(2),TM(1),X(1)
00141     370 FORMAT (6F12.0)
00142           KKLUE = 1
00143           II = 0
00144           DO 700 I = 3,22
00147           IF(X(I-1).GT.0.0)GO TO 732
00151     731 DXI = 0.0
00152           DSX = OSXBAT
00153           KKLUE = -1
00154           GO TO 711
00155     732 DXI2 = 2.0*(X(I-2)-X(I-1))
00156           IF((X(I-1)-DXI2).GT.0.0)GO TO 710
00160     770 KKLUE = 0
00161           DXI = X(I-1)
00162           DTCR = DXI/V(I-1)
00163           DTI = DTCR
00164           VSI = VS(I-1)
00165           DSI = VSI*DTI
00166           DSX = DSI - DXI
00167           AB = I
00170           SXBAT = 22.0 - AB
00171           OSXBAT = (2.0 - SX(I-1) - DSX)/SXBAT
00172           GO TO 711
00173     710 IF(V(I-1).GT.0.0)GO TO 992
00175           IF(V(I-1).LT.0.0)GO TO 992
00177     991 DXI = 0.001
00200           GO TO 993
00201     992 DELT = (TM(I-1)-TM(I-2))/1000.0
00202           DXI1 = V(I-1)*DELT
00203           IF(DXI1.GT.0.0)GO TO 1002
00205           IF(DXI1.LT.0.0)GO TO 1001
00207           GO TO 991
00210    1001 DXI0 = 0.25-X(I-1)
00211           IF((1.5*DXI0-ABS(DXI1)).GT.0.0)GO TO 1002
00213    1003 DXI = DXI0
00214           GO TO 993
00215    1002 DXI = DXI1
00216     993 JJ=0
00217           DSX = 0.10
00220     711 SX(I) = SX(I-1) + DSX
```

A–12. (Con't.)

```
00221          IF(I.LT.22)GO TO 704                        46
00223      706 DSX = 2.0 - SX(21)                          47
00224          SX(I) = 20                                  48
00225          Y(I) = 1.62                                 49
00226          BETA = 0.0                                  50
00227          GO TO 705                                   51
00230      704 YSQ =  SQRT(4.0-(2.0 - SX(I))**2)           52
00231          Y(I) = 0.81*YSQ                             53
00232          BETA =  ATAN(0.81*(2.0-SX(I))/YSQ)          54
00233          BETAD(I)=57.296*BETA                        55
00234      705 BSIN(I) =  SIN(BETA)                        56
00235          BCOS(I) =  COS(BETA)                        57
00236          THETA(I) = 19.0987*(Y(I)+2.15)              58
00237          RC = (4.0 - 0.3439*(2.0-SX(I))**2)**1.5/3.24 59
00240          XK = BSIN(I) -0.05*BCOS(I)                  60
00241          YK = BCOS(I) +0.05*BSIN(I)                  61
00242          CG = CY*YK - CX*XK                          62
00243          CI = DIE*BCOS(I)/(RD*RC*CG)                 63
00244          CF = 0.15*YK                                64
00245          CT = CX*XK + 0.1*YK                         65
00246          IF(KKLUE.EQ.0  )GO TO 773                   66
00250      775 DSI = DSX +DXI                              67
00251      773 RL = 0                                      68
00252          LL = 0                                      69
00253          VCI = VC(I-1)                               70
00254      709 VCIK = VCI                                  71
00255          DXIK = DXI                                  72
00256          EN(I)= ABS(CI*VCIK**2 + TG/CG)              73
00257          FUS(I) = CF*EN(I)
00260          GSIK = DSI                                  75
00261          TMU(I) = CT*EN(I) + TG                      76
00262          EMS = 0.5*(FUS(I-1) + FUS(I))*DSIK          77
00263          DTHETA = (Y(I) - Y(I-1))/RD                 78
00264          EMD = 0.5*(TMU(I-1)+ TMU(I))*DTHETA         79
00265          EMU = EMS + EMD                             80
00266          FD(I) = FD(I-1) - 40.0*DSIK                 81
00267          ED = (FD(I-1)+FD(I))*0.40*DSIK              82
00270          E(I) =E(I-1)+ED-EMU                         83A
00271          BBB=BCOS(I)                                 83B
00272          DENA=EMSLR*BBB**2                           84A
00273          RATIO=BSIN(I)/RD                            84B
00274          DENB=DIE*RATIO**2                           84C
00275          DEN=0.5*(DENA+DENB)                         84D
00276          VCI =  SQRT(E(I)/DEN)                       85
00277          VSI = VCI*BCOS(I)                           86
00300          VDI = VCI*BSIN(I)                           87
00301          IF(KKLUE.EQ.0)  GO TO 712                   88
00303      772 DII = DSIK/( (VSI + VS(I-1))*0.5)           89
00304          DTIM = 1000.0*DTI                           90
00305          FX(I) = EN(I)*XK                            91
00306          FY(I) = EN(I)*YK                            92
00307          ENF = 0.5*(FX(I-1)+FX(I))                   93
00310          EFA = 0.45*FA(I-                            94
00311          ENX = ENF-EFA                               95
00312          EFAR = FA(I-1)/0.45                         96
00313          ENXR = ENF - EFAR                           97
00314          IF(V(I-1).GT.0.0)GO TO 1007                 98
00316          IF(V(I-1).LT.0.0)GO TO 1009                 99
```

A—12. (Con't.)

```
00320   1006 IF(X(I-1).GT.0.0)GO TO 999
00322        GO TO 1011
00323    999 IF(ENX.GT.0.0)GO TO 1010
00325   3007 IF(DXIK.GT.0.0)GO TO 1007
00327        IF(DXIK.LT.0.0)GO TO 1012
00331   3008 DXIK=0.001
00332        GO TO 1007
00333   1010 IF(ENXR.GT.0.0)GO TO 1012
00335   1011 DXIK = 0.0
00336        GO TO 1007
00337   1012 DXIK = -  ABS(DXIK)
00340        GO TO 1009
00341   1007 FAX = 801.0*DXIK
00342        FMVSQ = 801.0*EMR*V(I-1)**2
00343        ENUMA= FAX + ENX
00344        GO TO 1013
00345   1009 FAX = 3955.6*DXIK
00346        FMVSQ = 3955.6*V(I-1)**2*EMR
00347        ENUMA = FAX + ENXR
00350        ENX = ENXR
00351        EFA = EFAR
00352   1013 FASQ = ENX**2
00353        DENSQ = FASQ + FMVSQ
00354        IF(DENSQ.GT.0.0)GO TO 344
00356        IF(DENSQ.LT.0.0)STOP
00360        GO TO 901
00361    344 DENOM =  SQRT(DENSQ)
00362        ENUMB = ENX
00363        ESIN  =  ABS(ENUMA/DENOM)
00364        FSIN  =  ABS(ENUMB/DENOM)
00365        GO TO 356
00366    901 ESIN = 1.0
00367        FSIN =1.0
00370        GO TO 444
00371    356 IF(FSIN.LT.1.0)GO TO 445
00373    444 ANGLEF = 1.5708
00374        GO TO 361
00375    445 ANGLEF =ASIN(FSIN)
00376    361 IF(ESIN.GT.1.0)GO TO 1200
00400        GO TO 362
00401   1200 DFAX   = (ENUMA - DENOM)
00402        LL = LL+1
00403        IF(LL.EQ.100)STOP
00405   1201 IF(V(I-1).LT.0.0)GO TO 1207
00407   1208 DXI = DXIK - DFAX/801.0
00410        GO TO 709
00411   1207 DXI = DXIK - DFAX/3955.6
00412        GO TO 709
00413    362 IF(KL.LT.10)GO TO 366
00415   1370 IF(ESIN.LT.0.999990)GO TO 366
00417    364 ANGLEE = 1.5708
00420        DELANG=ANGLEF-1.5708
00421        IF(DELANG.GT.0.0)GO TO 1088
00423        IF(DELANG.LT.0.0)GO TO 1088
00425        GO TO 1220
00426   1088 DTCR= (SQRT(EMR/801.0))* ABS(DELANG)
00427        IF((ABS(DTI-DTCR)/DTCR).GT.0.01)GO TO 1087
00431        GO TO 1220
```

```
00432   1087 DSX=DTCR*(VS(I-1)+VSI)/2.0                                  158
00433        DXI =V(I-1)*DTCR/2.0                                        159
00434        GO TO 711                                                   160
00435    366 ANGLEE=ASIN(ESIN)                                           161
00436    446 DELANG = ANGLEF - ANGLtt                                    162
00437        IF(DELANG.GT.0.0)GO TO 1017                                 163
00441        IF(DELANG.LT.0.0)GO TO 1017                                 164
00443   1220 DTCR=DTI                                                    165
00444        GO TO 1015                                                  166
00445   1017 IF(V(I-1).LT.0.0)GO TO 1016                                 167
00447   1014 DTCR = (SQRT(EMR/801.0))* ABS(DELANG)                       168
00450        GO TO 1015                                                  169
00451   1016 DTCR= (SQRT(EMR/3955.67))* ABS(DELANG)                      170A
06452   1015 DTCRM = 1000.0*DTCR                                         1708
00453        DELTM = DTIM - DTCRM                                        171
00454        KL = KL +1                                                  172
00455        IF(KL.GE.100)STOP                                           173
00457    585 DIFT = DTI - DTCR                                           174
00460        IF(DIFT.LT.0.0)GO TO 541                                    175
00462        IF(DIFT.GT.0.0)GO TO 542                                    164
00464        GO TO 712                                                   177
00465    541 ALDIF =      (DIFT/DTCR)                                    178
00466        GO TO 5f1                                                   179
00467    542 ALDIF =      (DIFT/DTI)                                     180
00470    577 IF( ABS(ALDIF).GT.0.01)GO TO 587                            181
00472        GO TO 712                                                   182
00473    587 DXI = DXIK*(1.0 + ALDIF/4.0)                                183
00474        DSI = DSX + DXI                                             184
00475        GO TO 709                                                   185
00476    712 JJ=JJ+1                                                     186
00477        IF(JJ.EQ. 0 )STOP                                           187
00501    819 DEBBL =  ABS(DXIK)*(-ENUMA)                                 188
00502        EBBL(I) = EBBL(I-1) + DEBBL                                 189
00503        VSQ = 2.0*EBBL(I)/EMR                                       190
00504        VI=SQRT(ABS(VSQ))                                           191
00505        IF(VI.GT.0.10)GO TO 1090                                    192
00507   1089 VI=0.0                                                      193
00510        EBBL(I)=0.0                                                 194
00511   1090 IF(V(I-1).GT.0.0)GO TO 1202                                 195
00513        IF(V(I-1).LT.0.0)GO TO 1203                                 196
00515        GO TO 822                                                   197
00516   1202 IF((EBBL(I-1)+DEBBL).LT.0.0)GO TO 823                       198
00520        GO TO 822                                                   199
00521   1203 IF((EBBL(I-1)+DEBBL).GT.0.0)GO TO 823                       200
00523        GO TO 822                                                   201
00524    823 RVI = VI/ ABS(V(I-1))                                       202
00525        IF(RVI.GT.0.10)GO TO 821                                    203
00527        GO TO 831                                                   204
00530    821 DDTI = DTI*(1.0 -  (ABS(V(I-1))/(( ABS(V(I-1)) + VI)*4.0))) 205
06531        DXI = V(I-1)*DDTI                                           206
00532        DDSI = VS(I-1)*DDTI                                         207
00533        GSX = DDSI + DXI                                            208
00534        GO TO 711                                                   209
00535    831 VI = 0.0                                                    210
00536        GO TO 824                                                   211
00537    822 IF(EBBL(I).LT.0.0)GO TO 825                                 212
00541    824 V(I) = VI                                                   213
00542        GO TO 827                                                   214
```

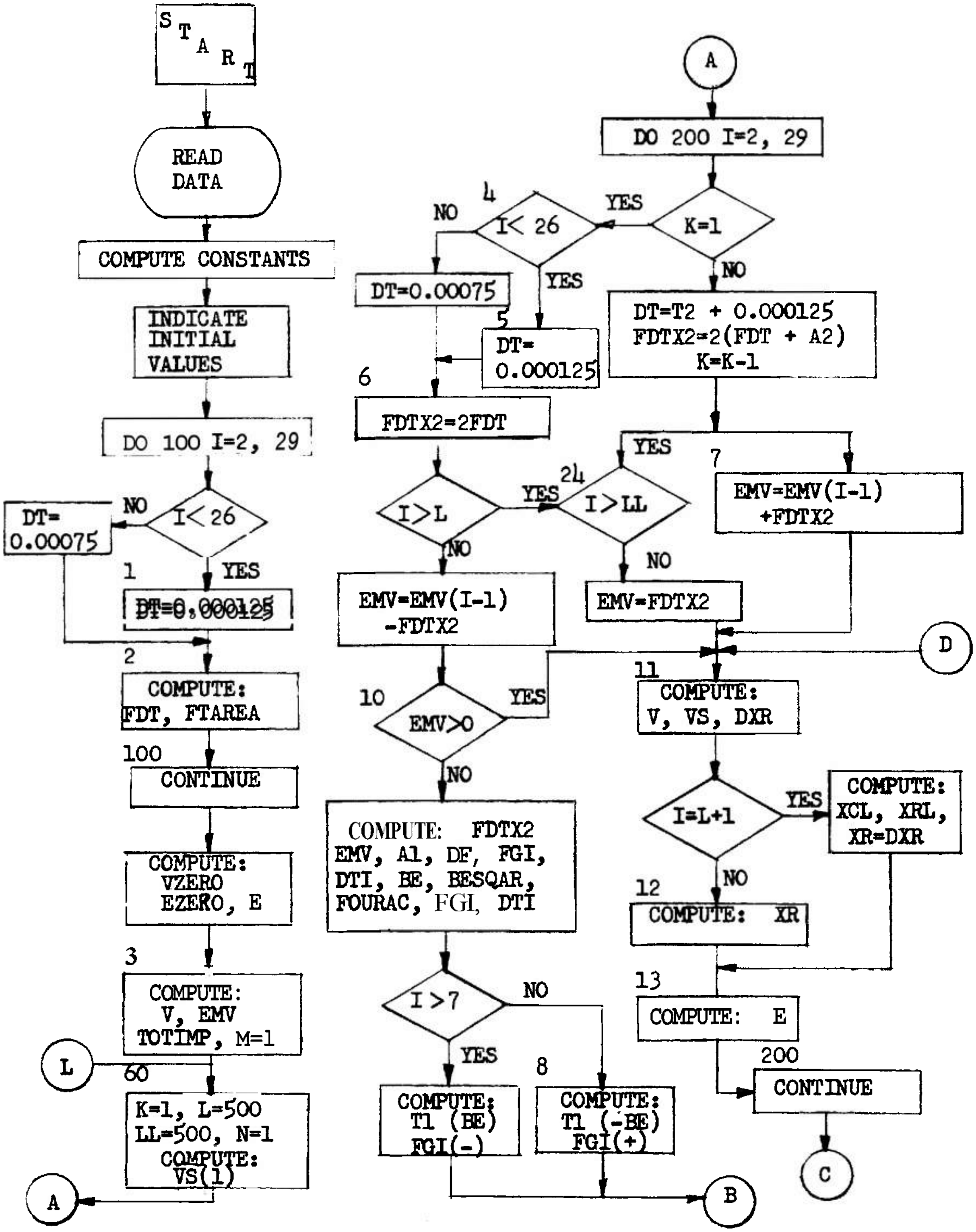
A–12. (Con't.)

```
00543    825 V(I) = -VI
00544    827 XI = X(I-1) - DXI
00545        IF(XI.LT.0.25)GO TO 829
00547    828 X(I) = 0.25
00550        V(I) = 0.0
00551        GO TO 515
00552    829 X(I) = XI
00553    515 SI = S(I-1) - DSIK
00554        S(I) = SI
00555        TMI = TM(I-1)+ 1000.0*DTI
00556        TM(I) = TMI
00557        VS(I) = VSI
00560        VD(I) = VDI
00561        VC(I) = VCL
00562        IF(X(I-1).GT.0.0)GO TO 513
00564    514 FA(I) =    0.0
00565        V(I) = 0.0
00566        X(I) = 0.0
00567        GO TO 516
00570    513 FA(I)=880.0 + 1780.0*X(I)
00571    516 IF(SX(I).GT.1.9999)GO TO 701
00573    700 CONTINUE
00575    701 WRITE(6,170)
00577    170 FORMAT(1H1/34X,53HTABLE 5-14 CAM AND DRUM DYNAMICS DURING COUNTERR
00577       1ECOIL//13X,85HRECOIL     DRIVING     NORMAL        L      PERIPH
00577       2                       COUNTER   COUNTER/
00577       3                 12X,105H ADAPTER   SPRING    CAM     CAM     CAM
00577       4      CAM      DRUM      RECOIL    RECOIL    SLIDE    SLIDE/
00577       53X,114H TIME         FORCE     FORCE     FORCE    TRAVEL   TRAVEL
00577       6 SLOPE        VEL       VEL      POSITIONN   VEL     TRAVEL/
00577       72X,113H MILSEC        LB        LB        LB       IN       IN
00577       8 DEGREE      IN/SEC     IN/SEC      IN      IN/SEC    IN)
00600        WRITE(6,702)(TM(I),FA(I),FD(I),EN(I),SX(I),Y(I),BETAD(I),VD(I),
00600       1V(I),X(I),VS(I),S(I),I=3,22)
00621    702 FORMAT(F10.5,F9.0,F9.1,F10.0,F10.3,F11.4,F8.1,F10.1,F10.4,F10.4,
00621       1F10.1,F10.3)
00622    900 STOP
00623        END

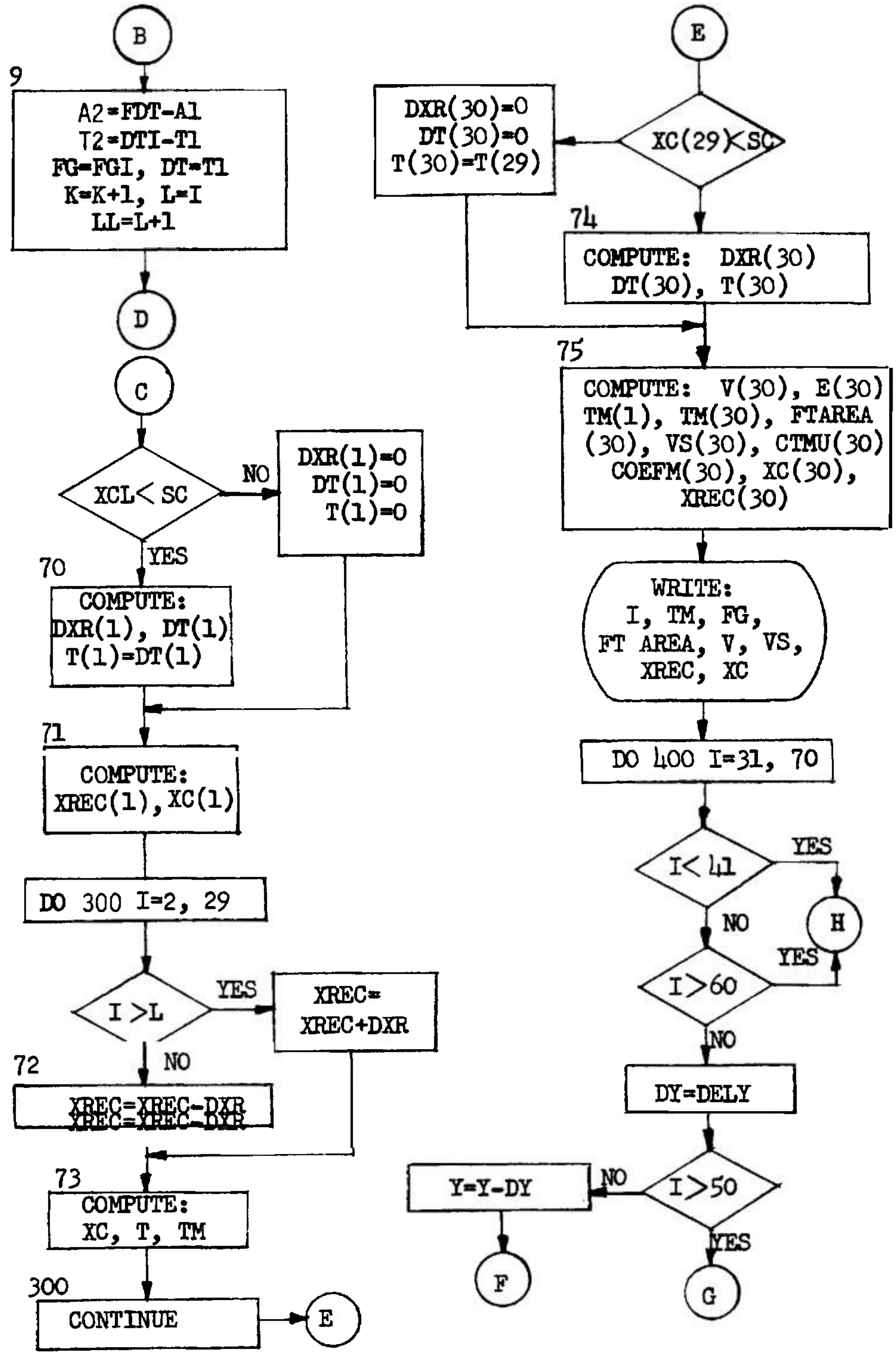
        END OF-LISTING.        0 *DIAGNOSTIC* MESSAGE(S).
PHASE 1 TIME =    3 SEC.
PHASE 2 TIME =    0 SEC.
PHASE 3 TIME =    4 SEC.
PHASE 4 TIME =    1 SEC.
PHASE 5 TIME =    3 SEC.
PHASE 6 TIME =    3 SEC.

TOTAL COMPILATION TIME =   14  SEC
```

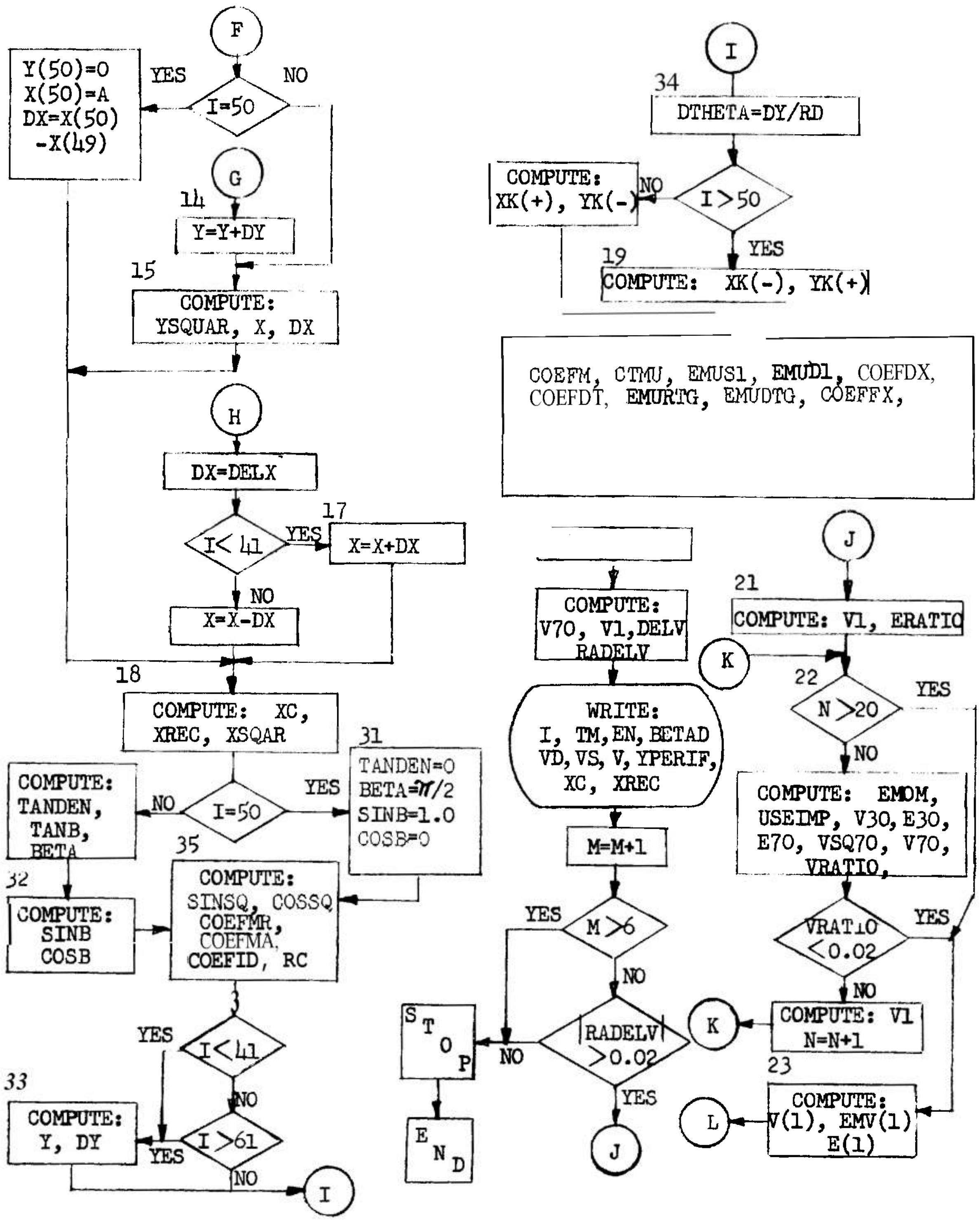
A–13. FLOW CHART FOR DOUBLE BARREL MACHINE GUN



A-13. (Con't.)


B
9
A2=FDT-A1
T2=DTI-T1
FG=FGI, DT=T1
K=K+1, L=I
LL=L+1
D
C
XCL< SC
NO
DXR(1)=0
DT(1)=0
T(1)=0
YES
70
COMPUTE:
DXR(1), DT(1)
T(1)=DT(1)
71
COMPUTE:
XREC(1), XC(1)
DO 300 I=2, 29
I >L
YES
XREC=
XREC+DXR
NO
72
XREC=XREC-DXR
73
COMPUTE:
XC, T, TM
300
CONTINUE
E
E
XC(29)<SC
DXR(30)=0
DT(30)=0
T(30)=T(29)
74
COMPUTE: DXR(30)
DT(30), T(30)
75
COMPUTE: V(30), E(30)
TM(1), TM(30), FTAREA
(30), VS(30), CTMU(30)
COEFM(30), XC(30),
XREC(30)
WRITE:
I, TM, FG,
FT AREA, V, VS,
XREC, XC
DO 400 I=31, 70
I< 41
YES
H
NO
YES
I>60
NO
DY=DELY
I>50
NO
Y=Y-DY
F
YES
G

A–13. **(Con't.)**


F
I=50
YES
NO
Y(50)=0
X(50)=A
DX=X(50)
-X(49)
G
14
Y=Y+DY
15
COMPUTE:
YSQUAR, X, DX
H
DX=DELX
I< 41
YES
17
X=X+DX
NO
X=X-DX
18
COMPUTE: XC,
XREC, XSQAR
I=50
NO
YES
COMPUTE:
TANDEN,
TANB,
BETA
31
TANDEN=0
BETA=π/2
SINB=1.0
COSB=0
32
COMPUTE:
SINB
COSB
35
COMPUTE:
SINSQ, COSSQ
COEFMR,
COEFMA,
COEFID, RC
I<41
YES
NO
33
COMPUTE:
Y, DY
I >61
YES
NO
I
I
34
DTHETA=DY/RD
I >50
NO
COMPUTE:
XK(+), YK(-)
YES
19
COMPUTE: XK(-), YK(+)
COEFM, CTMU, EMUS1, EMUD1, COEFDX,
COEFDT, EMURTG, EMUDTG, COEFFX,
COMPUTE:
V70, V1,DELV
RADELV
WRITE:
I, TM, EN, BETAD
VD, VS, V, YPERIF,
XC, XREC
M=M+1
M >6
YES
NO
STOP
END
|RADELV|
>0.02
NO
YES
J
J
21
COMPUTE: V1, ERATIO
K
22
N >20
YES
NO
COMPUTE: EMOM,
USEIMP, V30, E30,
E70, VSQ70, V70,
VRATIO,
VRATIO
<0.02
YES
NO
COMPUTE: V1
N=N+1
K
23
COMPUTE:
V(1), EMV(1)
E(1)
L

A–14. PROGRAM LISTING FOR DOUBLE BARREL MACHINE GUN

```
0000 I 003353 M        0000 I 003357 N        0000 R 003444 RADELV   0000 R 003306 RB       0000 R 003412 RC
0000 R 003343 RCFRAC   0000 R 003307 RCH      0000 R 003313 RD       0000 R 003330 RDSQ     0000 R 003305 RHO
0000 R 003344 RHOSQ    0000 R 003301 RL       0000 R 003310 RP       0000 R 003322 RPORR    0000 R 003311 RR
0000 R 003312 RT       0000 R 003333 SC       0005 R 000000 SIN      0000 R 003403 SINB     0000 R 003405 SINBB
0003 R 000000 SQRT     0000 R 003317 SR       0000 R 002373 T        0000 R 003402 TANB     0000 R 003401 TANDEN
0000 R 003325 TG       0000 R 003420 TGN      0000 R 003047 TM       0000 R 003352 TOTIMP   0000 R 003372 T1
0000 R 003360 T2       0000 R 003447 USEIMP   0000 R 000454 V        0000 R 003440 VC       0000 R 003433 VCCOEF
0000 R 003434 VCSQ     0000 R 002145 VD       0000 R 003454 VRATIO   0000 R 002032 VS       0000 R 003453 VS070
0000 R 003350 VZERO    0000 R 003442 V1       0000 R 003450 V30      0000 R 003441 V70      0000 R 003277 WA
0000 R 003276 WR       0000 R 001356 X        0000 R 002506 XC       0000 R 003373 XCL      0000 R 003414 XK
0000 R 001015 XR       0000 R 002621 XREC     0000 R 003374 XRL      0000 R 003400 XSQUAR   0000 R 001243 Y
0000 R 003415 YK       0000 R 003162 YPERIF   0000 R 003377 YSQUAR
```

```
00101                                                                              A
00101      1       ,EN(75),Y(75),X(75),BETA(75), COEFM(75),CTMU(75),VS(75),VD(7    3B
00101   25),              BETAD(75),T(75),XC(75),XREC(75),FTAREA(75),
00101   3TM(75), YPERIF(75)                                                        3D
00103      READ 50,(FG(I), I=1,29)                                                 4A
00111      READ 50,WR, WA, DI, RL, CL, A, B, RHO, RB, RCH, RP, RR, RT, RD,         5A
00111   1FGRES ,EMU ,G,SR                                                          58
00135   50 FORMAT(6F12.0)                                                          76
00136      EMR=WR/G
00137      EMA=WA/G                                                                7B
00140      RPORR=RP/RB                                                             7C
00141      CTRIG=EMU*RPORR                                                         8
00142      CID=DI/RD                                                               9
00143      TG=FGRES*(RCH-EMU*RB)*EMU                                               10
00144      CY=RD - EMU*RB                                                          11
00105      CX=EMU*(RT+RD*RPORR)                                                    12
00146      RDSQ=RD**2                                                              13
00147      EMURB=EMU*RB                                                            14
00150      CMUF=EMU*(RPORR+1.0/RHO)                                                15
00151      SC =CL-A                                                                16
00152      DELY=0.0866025*B
00153      DELX=A/20.0                                                             18
00154      ASQUAR=A*A                                                              19
00155      BSQUAR=B*B                                                              20
00156      AOVERB=A/B                                                              21
00157      BOVERA=B/A                                                              22
00160      AB=A*B                                                                  23
00161      RCFRAC=(ASQUAR - BSQUAR)/ASQUAR                                         24
00162      RHOSQ=RHO**2                                                            25
00163      EFMR  =0.5*EMR/RHOSQ                                                    26
00164      EFMA  =0.5*EMA/RHOSQ                                                    21
00165      EFID=0.5*DI/(RDSQ + RHOSQ)                                              28
00166      XR(1)=0.0                                                               29
00167      T(1)=0.0                                                                30
00170      EN(30)=0.0                                                              31
00171      EN(70)=0.0                                                              32
00172      FG(30)=0.0                                                              33
00173      FTAREA(30)=0.0                                                          34
00174      VD(30)=0.0                                                              35
00175      YPERIF(30)=0.0                                                          36
00176      BETAD(30)=0.0                                                           37
00177      COEF M(30)=0.0                                                          38
```

```
00200            CTMU(30)=0.0                                 39
00201            Y(30)=B                                      40
00202            X(30)=0.0                                    41
00203            FTAREA(1)=0.0                                42
00204            DO 100 I=2,29                                43
00207            IF(I.LT.26)GO TO 1                           44
00211            DT(I)=0.00075                                45
00212            GO TO 2                                      46
00213        1   DT(I)=0.000125                               47
00214        2   FDT(I)  =0.5*(FG(I)+FG(I-1))*DT(I)           48
00215            FTAREA(I)=FTAREA(I-1)+FDT(I)                 49
00216      100   CONTINUE                                     50
00220            VZERO=FTAREA(29)/EMR                         51
00221            EZERO = 0.5*EMR*VZERO**2                     52
00222            E(1)=0.7*EZERO                               53
00223        3   V(1)=SQRT(2.0*E(1)/EMR)                      54
00224            EMV(1)=EMR*V(1)                              55A
00225            TOTIMP = 2.0*FTAREA(29)                      55B
00226            M=1                                          56
00227       60   K=1                                          57
00230            L=500                                        58A
00231            LL=500                                       58B
00232            N = 1                                        58C
00233            VS(1)=V(1)*RHO                               59
00234            DO 200 I=2,29                                60
00237            IF(K.EQ.1)GO TO 4                            61
00241            DT(I)=T2+0.000125                            62
00242            FDTX2=2.0*(FDT(I)+A2)                        63
00243            K = K-1                                      64
00244            GO TO 7                                      65
00245        4   IF(I.LT.26)GO TO 5                           66
00247            DT(I)=0.00075                                67
00250            GO TO 6                                      68
00251        5   DT(I)=0.000125                               69
00252        6   FDTX2 = 2.0*FDT(I)                           70
00253            IF(I.GT.L)GO TO 24                           71
00255            EMV(I)=EMV(I-1)-FDTX2                        72
00256            GO TO 10                                     73A
00257       24   IF(I.GT.LL)GO TO 7                           73B
00261            EMV(I)=FDTX2                                 73C
00262            GO TO 11                                     73D
00263        7   EMV(I)=EMV(I-1)+FDTX2                        73E
00264            GO TO 11                                     74
00265       10   IF(EMV(I).GT.0.0)GO TO 11                    75
00267            FDTX2 = EMV(I-1)                             76
00270            EMV(I)=0.0                                   77
00271            A1=FDTX2/2.0                                 78
00272            DF= ABS(FG(I)-FG(I-1))                       79A
00273            FGI = FG(I)                                  79B
00274            DTI=DT(I)                                    79C
00275            BE= FGI*DTI/DF                               80
00276            BESQAR=BE**2                                 81
00277            FOURAC=2.0*DTI*A1/DF                         82
00300            FGI=FG(I-1)                                  83
00301            DTI=DT(I)                                    84
00302            IF(I.LE.7)GO TO 8                            85
00304            T1=BE-SQRT(BESQAR-FOURAC)                    86
00305            FGI=FGI-DF*T1/DTI                            87
```

A–14. (Con't.)

```
00306          GO TO 9                                          88
00307        8 T1=-BE+SQRT(BESQAR+FOURAC)                       89
00310          FGI=FGI+DF*T1/DTI                                90
00311        9 A2=FDT(I)-A1                                     91
00312          T2=DTI-T1                                        92
00313          FG(I)=FGI                                        93
00314          DT(I)=T1                                         94
00315          K=K+1                                            95
00316          L=I                                              96A
00317          LL=L+1                                           96B
00320       11 V(I)=EMV(I)/EMR                                  97A
00321          VS(I)=V(I)*RHO                                   97B
00322          DXR(I)=0.5*(V(I-1)+V(I))*DT(I)                   98
00323          IF(I.NE.(L+1))GO TO 12                           99A
00325          XCL=XR(L)*RHO                                    99B
00326          XRL=XR(L)                                        99C
00327          XR(I)=DXR(I)                                     99D
00330          GO TO 13                                         100
00331       12 XR(I)=XR(I-1)+DXR(I)                             101
00332       13 E(I)=EMR*V(I)**2/2.0                             102
00333      200 CONTINUE                                         103
00335          IF(XCL.LT.SC)GO TO 70                            104
00337          DXR(1)=0.0                                       105A
00340          DT(1)=0.0                                        105B
00341          T(1)=0.0                                         105C
00342          GO TO 71                                         105D
00343       70 DXR(1)=SR-XRL                                    106A
00344          DT(1)=DXR(1)/V(1)                                106B
00345          T(1)=DT(1)                                       106C
00346       71 XREC(1)=XRL                                      107A
00347          XC(1)=XREC(1)*RHO                                107B
00350          DO 300 I=2,29                                    108A
00353          IF(I.LE.L)GO TO 72                               108B
00355          XREC(I)=XREC(I-1)+DXR(I)                         108C
00356          GO TO 73                                         108D
00357       72 XREC(I)=XREC(I-1)-DXR(I)                         108E
00360       73 XC(I)=XREC(I)*RHO                                108F
00361          T(I)=T(I-1) +DT(I)                               108G
00362          TM(I)=1000.0*T(I)                                108H
00363      300 CONTINUE                                         108I
00365          IF(XC(29).LT.SC)GO TO 74                         109A
00367          DXR(30)=0.0                                      109B
00370          DT(30)=0.0                                       109C
00371          T(30)=T(29)                                      109D
00372          GO TO 75                                         109E
00373       74 DXR(30)=SR     -XR(29)                           110A
00374          DT(30)=DXR(30)/V(29)                             110B
00375          T(30)=T(29)+DT(30)                               110C
00376       75 V(30)=V(29)                                      110D
00377          E(30)=E(29)                                      110E
00400          TM(1)=1000.0*T(1)                                110F
00401          TM(30)=1000.0*T(30)                              110G
00402          FTAREA(30)=0.0                                   110H
00403          VS(30)=VS(29)                                    110I
00404          CTMU(30)=CX*CTRIG+EMURB                          110J
00405          COEFM(30)=CMUF                                   110K
00406          XC(30)=SC                                        111A
00407          XREC(30)=XC(30)/RHO                              111B
```

A—14. (Con't.)

```
00410        PRINT 105,( TM(I) ,FG( I),FTAREA(I),V(I),VS(I),XREC(I),XC(I),           111C
00410       1I=1,30)                                                                   111D
00425    105 FORMAT(1H1/12X,47H TABLE 2- 11 DOUBLE BARREL MACHINE GUN DYNAMICS        111E
00425       1          //14X,11H PROPELLANT,19X,6H AXIAL,13X,6H AXIAL/17X,4H GA        111F
00425       2S,14X,33H RECOIL    CAM      RECOIL     CAM/70H  I   TIME     FOR         111G
00425       3CE    IMPULSE    VEL     VEL     TRAVEL     TRAVEL/68H          MSEC      111H
00425       4         LB      LP-SEC   IN/SEC   IN/SEC     IN         IN//             1111
00425       5(I3,F9.3,F11.1,F9.2,2F9.1,2F10.4))                                        111J
00426        GO 400  I=31,70                                                           112
00431        IF(I.LT.41)GO TO 16                                                       113
00433        IF(I.GT.60)GO TO 16                                                       114
00435        DY=DELY                                                                   115
00436        IF(I.GT.50)GO TO 14                                                       116
00440     25 Y(I)=Y(I-1)-DY                                                            117
00441        IF(I.NE.50)GO TO 15                                                       118A
00443        Y(50)=0.0                                                                 1188
00444        X(50)=A                                                                   118C
00445        DX = X(50) - X(49)                                                        118D
00446        GO TO 18                                                                  119
00447     14 Y(I)=Y(I-1)+DY                                                            120
00450     15 YSQUAR=Y(I)**2                                                            121
00451        X(I)=AOVERB*SQRT(BSQUAR-YSQUAR)                                           122A
00452        DX = ABS(X(I-1) - X(I))                                                   1228
00453        GO TO 18                                                                  123
00454     16 DX=DELX                                                                   124
00455        IF(I.LT.41)GO TO 17                                                       125
00457        X(I)=X(I-1)-DX                                                            126
00460        GO TO 18                                                                  127
00461     17 X(I)=X(I-1)+DX                                                            128
00462     18 XC(I)=SC    +X(I)                                                         129A
00463        XREC(I)=XC(I)/RHO                                                         129B
00464        XSQUAR=X(I)**2                                                            129C
00465        IF(I.EQ.50) GO TO 31                                                      129D
00467        TANDEN=SQRT(ASQUAR-XSQUAR)                                                130
00470        TANB=BOVERA*X(I)/TANDEN                                                   131
00471        BETA(I)=ATAN(TANB)                                                        132
00472        GO TO 32                                                                  133
00473     31 TANDEN=0.0                                                                134A
00474        BETA(I)=1.570756                                                          134B
00475        SINB=1.0                                                                  134C
00476        COSB=0.0                                                                  134D
00477        GO TO 35                                                                  135A
00500     32 SINB=SIN(BETA(I))                                                         135B
00501        COSB=COS(BETA(I))                                                         135C
00502     35 SINSQ=SINB**2                                                             135D
00503        COSSQ=COSB**2                                                             135E
00504        COEFMR=EFMR*COSSQ                                                         135F
00505        COEFMA=EFMA*SINSQ                                                         1356
00506        COEFID=EFID*SINSQ                                                         135H
00507        RC=(ASQUAR-XSQUAR*RCFRAC)**1.5/AB                                         136
00510        IF( I.LT.41)GO TO 33                                                      137A
00512        IF(I.GT.60)GO TO 33                                                       1378
00514        GO TO 34                                                                  137C
00515     33 Y(1)=BOVERA*TANDEN                                                        137
00516        DY=ABS(Y(I-1)-Y(I))                                                       138
00517     34 DTHETA=DY/RD                                                              139
00520        IF(I.GT.50)GO TO 19                                                       140
00522        XK=SINB+CTRIG*COSB                                                        141
```

A-14. (Con't.)

```
00523            YK = COSB - CTRIG*SINB                                        142
00524            GO TO 20                                                      143
00525         19 XK=SINB-CTRIG*COSB                                            144
00526            YK=COSB+CTRIG*SINB                                            145
00527         20 DENOMN=ABS(CY*YK-CX*XK)                                       146
00530            COEFVC=CID*COSB/(RC*DENOMN)                                   147
00531            TGN=TG/DENOMN                                                 148
00532            COEFM(I)=CMUF*YK                                              149
00533            CTMU(I)=(CX*XK + EMURB*YK)                                    150
00534-           EMUS1=COEFM(I-1)*EN(I-1)*DX/2.0                               151
00535            EMUD1=CTMU(I-1)*EN(I-1)*DTHETA/2.0                            152
00536            COEFDX=COEFM(I)*DX/2.0                                        153
00537            COEFDT=CTMU(I)*DTHETA/2.0                                     154
00540            EMURTG=(COEFDX+COEFDT)*TGN                                    155
00541            EMUDTG=TG*DTHETA                                              156
00542            COEFFX=COEFDX*COEFVC                                          157
00543            COEFTQ=COEFDT*COEFVC                                          158
00544            HALFFE=EMUS1+EMUD1+EMURTG+EMUDTG                              159
00545            EI=E(I-I)-HALFFE                                              160
00546            VCCOEF=COEFMR+COEFMA+COEFID+COEFFX+COEFTQ                     161
00547            VCSQ=EI/VCCOEF                                                162
00550            EMUR2=COEFFX*VCSQ                                             163
00551            EMUD2=COEFTQ*VCSQ                                             164
00552            ESUBMU=HALFFE + EMUR2 + EMUD2                                 165
00553            E(I)=E(I-1) - ESUBMU                                          166
00554            VC=SQRT(VCSQ)                                                 167
00555            VS(I)=VC*COSB                                                 168
00556            VD(I)=VC*SINB                                                 169
00557            V(I)=VS(I)/RHO                                                176
00560            EN(I)=COEFVC*VCSQ + TGN                                       171
00561            YPERIF(I)=YPERIF(I-1)+ABS(DY)                                 172
00562            BETAD(I)=57.296*BETA(I)                                       173
00563            DT(I)=2.0*DX/(VS(I-1) + VS(I))                                In
00564            T(I)=T(I-1) + DT(I)                                           175
00565            TM(I)=1000.0*T(I)                                             176A
00566        400 CONTINUE                                                      1768
00570            V70=V(70)                                                     176C
00571            V1=V(1)                                                       176D
00572            DELV=V1-V70                                                   176E
00573            RADELV=DELV/V1                                                176F
00574            IF(ABS(RADELV).GT.0.02)GO TO 36                               176G
00576         36 PRINT 110,(I,TM(I),EN(I),BETAD(I),VD(I),VS(I),V(I),YPERIF(I)  177
00576           1,XC(I),XREC(I),I=31,70)                                       178
00615        110 FORMAT(1H1/20X,53H TABLE 2-11 CONT.  DOUBLE BARREL MACHINE GUN DYN  179
00615           1AMICS     //18X,7H NORMAL,11X,15H PERIPH   AXIAL,12X,17H PERIPH  180
00615           2  AXIAL/19X,71H CAM        CAY        DRUM       CAM       RECOIL     DRUM  181
00615           3       CAM       RECOIL/90H  I         TIME       FORCE     SLOPE      VEL  182
00615           4      VEL        VEL       TRAVEL      TRAVEL     TRAVEL/88H           MSEC  183
00615           5      LB         DEG      IN/SEC      IN/SEC     IN/SEC      IN          IN  184
00615           6       IN//(I3,F11.4,F10.1,F9.2,3F9.1,3F10.4))                185-6
00616            M=M+1                                                         187
00617            IF(M.GT.6) GO TO 30                                           188
00621            IF(ABS(RADELV).GT.0.02)GO TO 21                               189
00623            GO TO 99                                                      190
00624         21 V1= (V1+V70)/2.0                                              191
00625            ERATIO = E(70)/E(30)                                          192
00626         22 IF(N.GT.20)GO TO 23                                           193
00630            EMOM=V1*EMR                                                   194
```

)

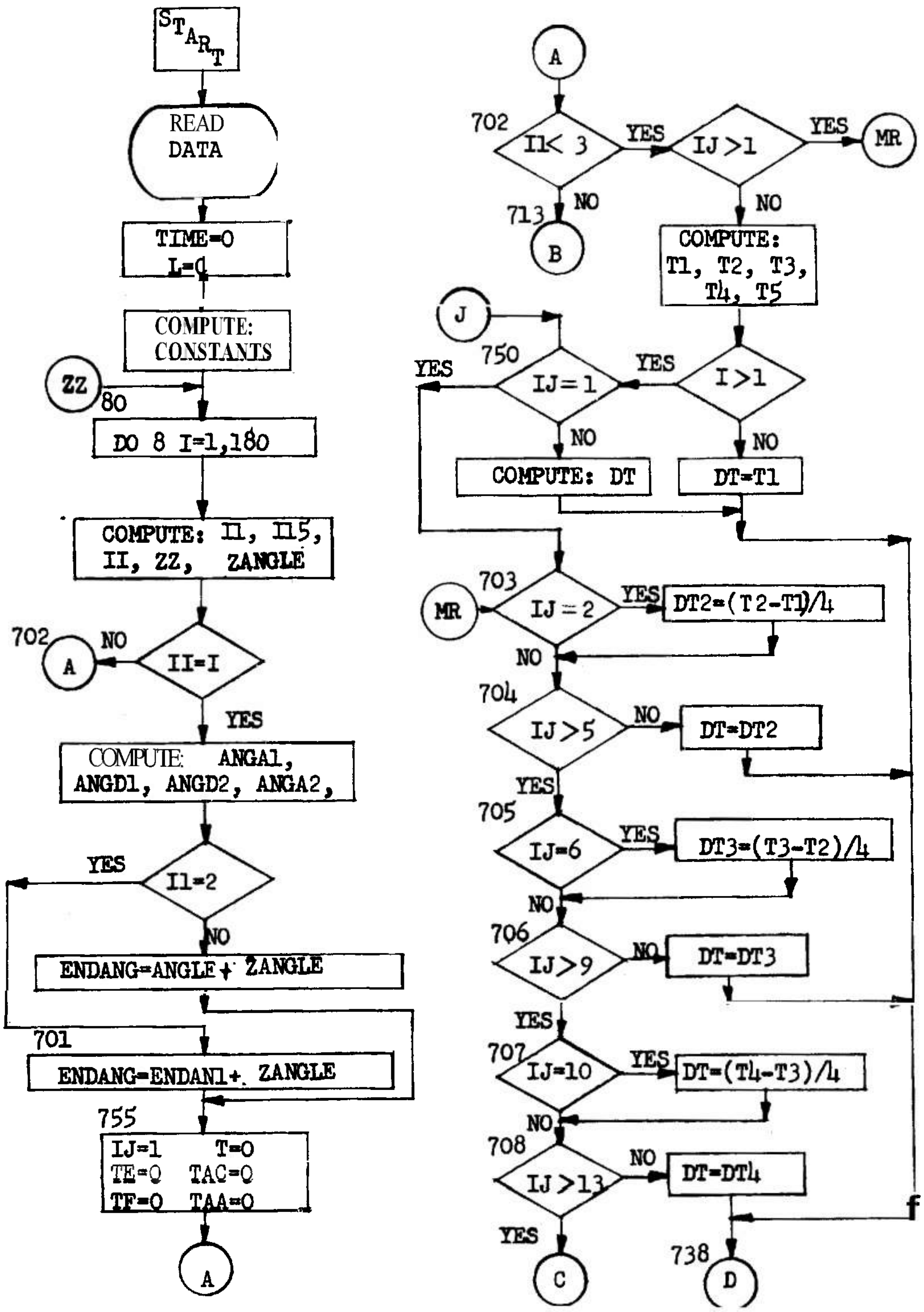
A—14. (Con't.)

```
00631   275.        USEIMP=TOTIMP-EMOM                          195
00632   276.        V30=USEIMP/EMR                              196
00633   277.        E30=EMR*V30**2/2.0                          197A
00634   278.        E70=ERAT10*E30                              197B
00635   279.        VS070=2.0*E70/EMR                           197C
00636   280.        V70=SQRT(VSQ70)                             197D
00637   281.        VRATIO=ABS(1.0-V1/V70)                      197E
00640   282.        IF(VRATIO.LT.0.02)GO TO 23                  197F
00642   283.        V1=(V1+V70)/2.0                             197G
00643   284.        N=N+1                                       198
00644   285.        GO TO 22                                    199
00645   286.     23 V(1)=V1                                     200
00646   287.        EMV(1)=EMR*V(1)                             201
00647   288.        E(1)=EMR*V(1)**2/2.0                        202
00650   289.        GO TO 60                                    203
00651   290.     30 PRINT 80,V(1),T(70),RADELV,M,N              204
00660   291.     80 FORMAT(////3E18.8,2I5)                      205
00661   292.     99 STOP                                        206
00662   293.        END                                         207

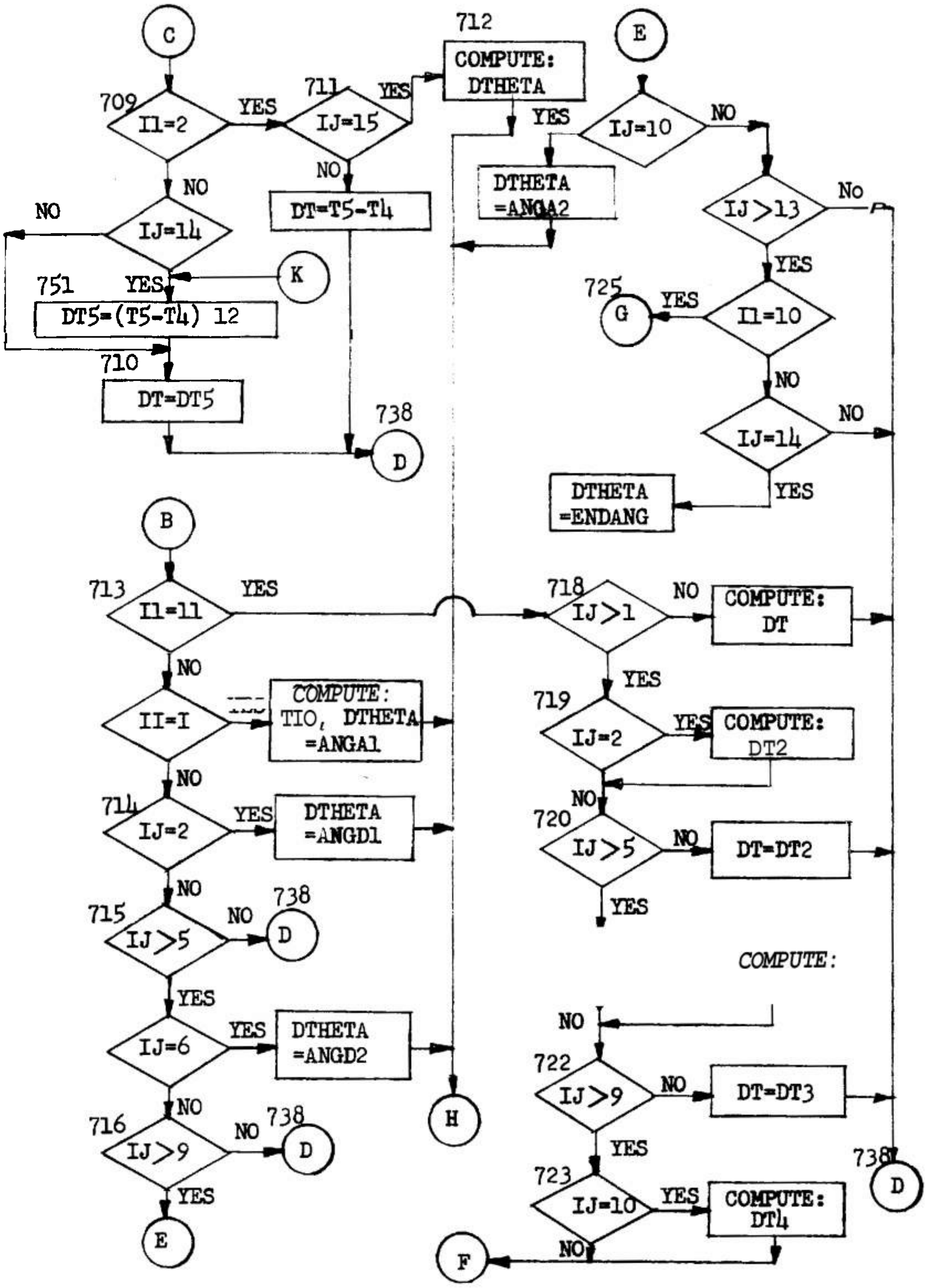
         END OF LISTING.      0 *DIAGNOSTIC* MESSAGE(S).
PHASE 1 TIME =    3 SEC.
PHASE 2 TIME =    1 SEC.
PHASE 3 TIME =    5 SEC.
PHASE 4 TIME =    0 SEC.
PHASE 5 TIME =    4 SEC.
PHASE 6 TIME =    4 SEC.

TOTAL COMPILATION TIME =  17  SEC
```

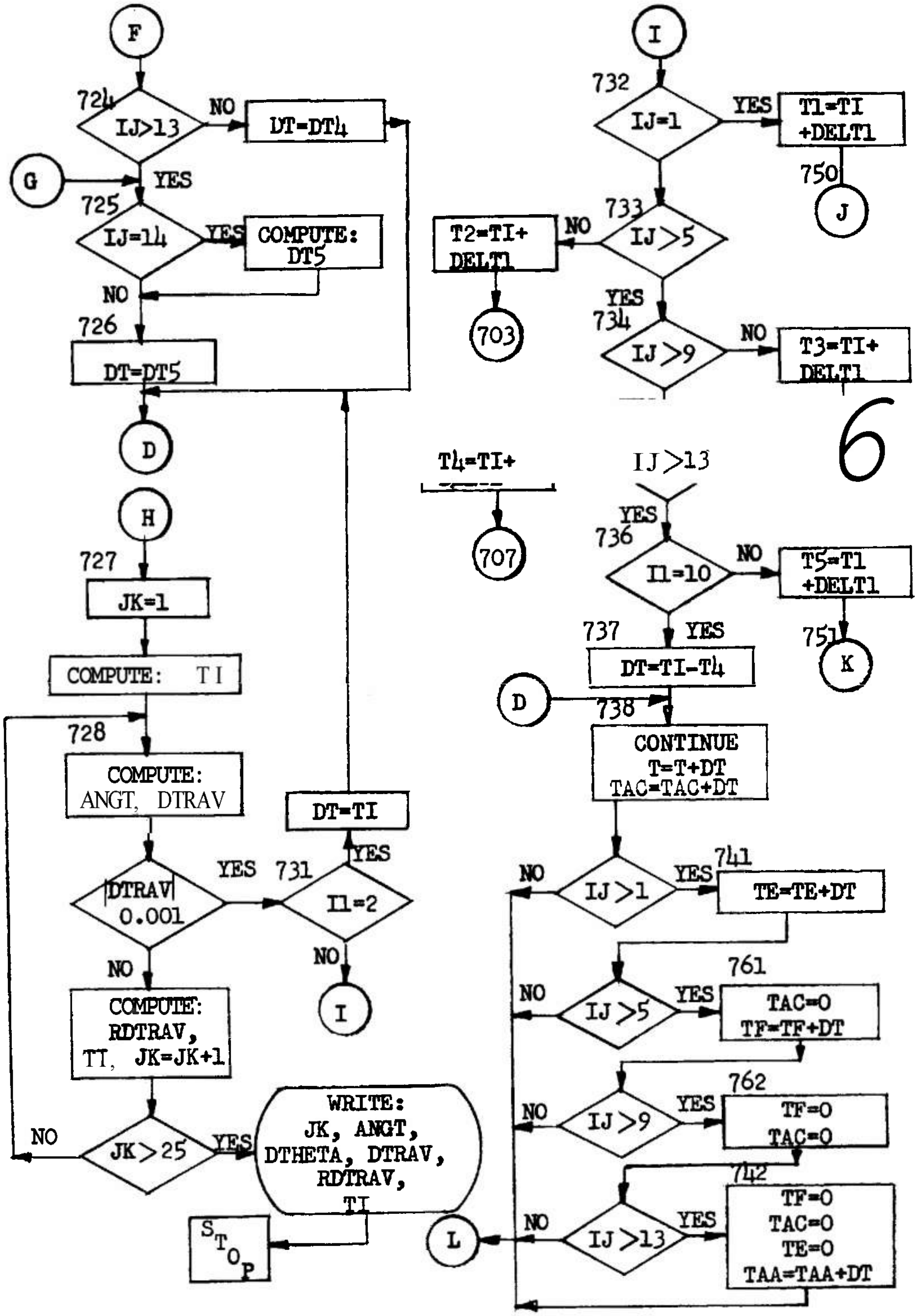
A–15. FLOW CHART FOR MULTIBARREL POWER



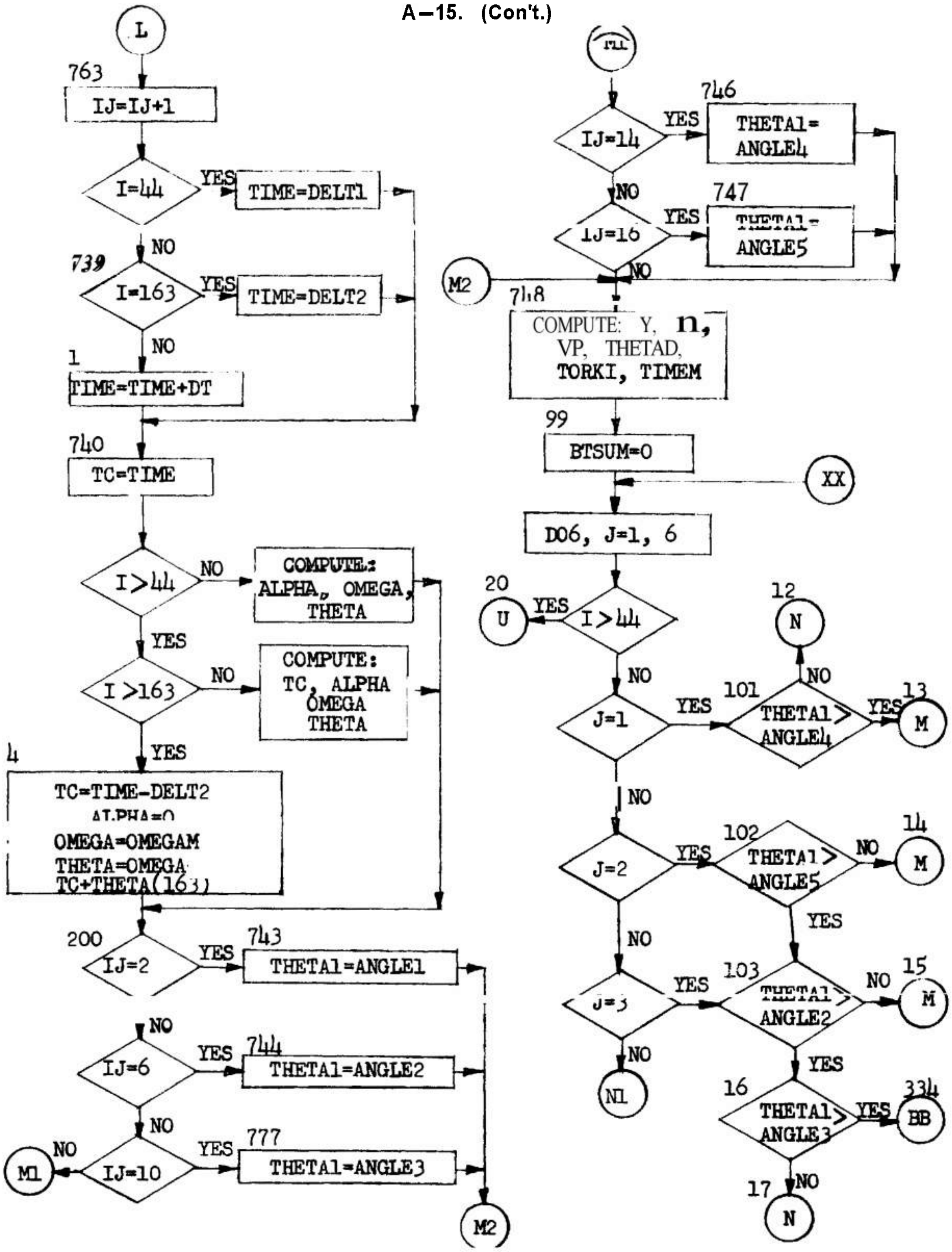
A–15. (Con't.)


C
709
I1=2
YES
711
IJ=15
YES
712
COMPUTE: DTHETA
NO
NO
IJ=14
NO
DT=T5-T4
K
751
YES
DT5=(T5-T4) 12
710
DT=DT5
738
D
E
YES
IJ=10
NO
DTHETA =ANGA2
IJ>13
No
YES
725
G
YES
I1=10
NO
IJ=14
NO
YES
DTHETA =ENDANG
B
713
I1=11
YES
718
IJ>1
NO
COMPUTE: DT
NO
II=I
COMPUTE: TIO, DTHETA =ANGA1
YES
719
IJ=2
YES
COMPUTE: DT2
NO
714
IJ=2
YES
DTHETA =ANGD1
NO
720
IJ>5
NO
DT=DT2
NO
715
IJ>5
NO
738
D
YES
YES
COMPUTE:
IJ=6
YES
DTHETA =ANGD2
NO
NO
722
IJ>9
NO
DT=DT3
H
716
IJ>9
NO
738
D
YES
723
IJ=10
YES
COMPUTE: DT4
738
D
E
NO
F

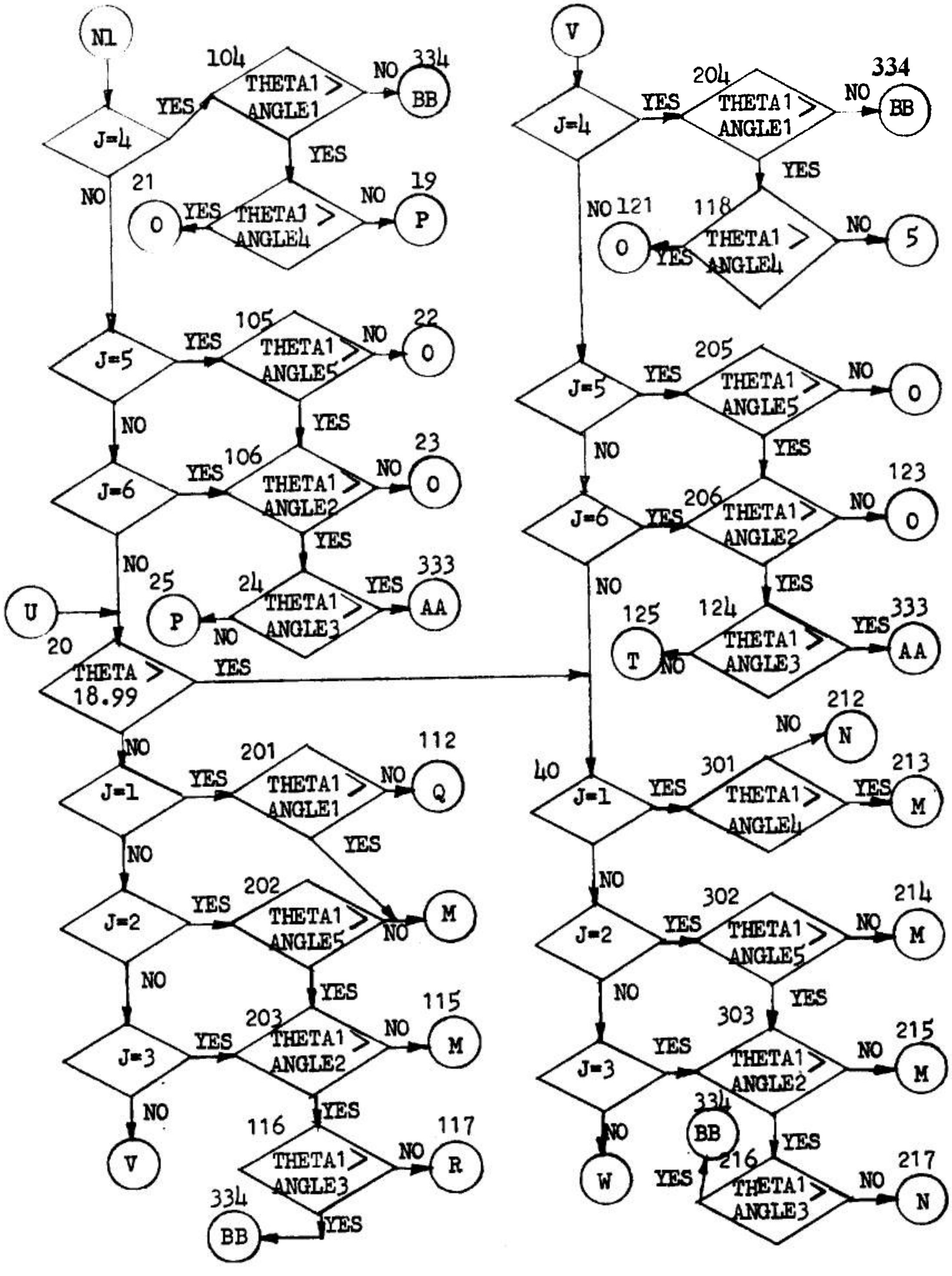
A—15. (Con't.)


F
724
IJ>13
NO
DT=DT4
G
YES
725
IJ=14
YES
COMPUTE: DT5
NO
726
DT=DT5
D
H
727
JK=1
COMPUTE: TI
728
COMPUTE: ANGT, DTRAV
|DTRAV| 0.001
YES
731
I1=2
YES
DT=TI
NO
I
NO
COMPUTE: RDTRAV, TI, JK=JK+1
NO
JK>25
YES
WRITE: JK, ANGT, DTHETA, DTRAV, RDTRAV, TI
STOP
I
732
IJ=1
YES
T1=TI +DELT1
750
J
733
IJ>5
NO
T2=TI+ DELT1
703
YES
734
IJ>9
NO
T3=TI+ DELT1
6
T4=TI+
707
IJ>13
YES
736
I1=10
NO
T5=T1 +DELT1
751
K
737
YES
DT=TI-T4
D
738
CONTINUE T=T+DT TAC=TAC+DT
NO
IJ>1
YES
741
TE=TE+DT
761
NO
IJ>5
YES
TAC=0 TF=TF+DT
762
NO
IJ>9
YES
TF=0 TAC=0
742
L
NO
IJ>13
YES
TF=0 TAC=0 TE=0 TAA=TAA+DT

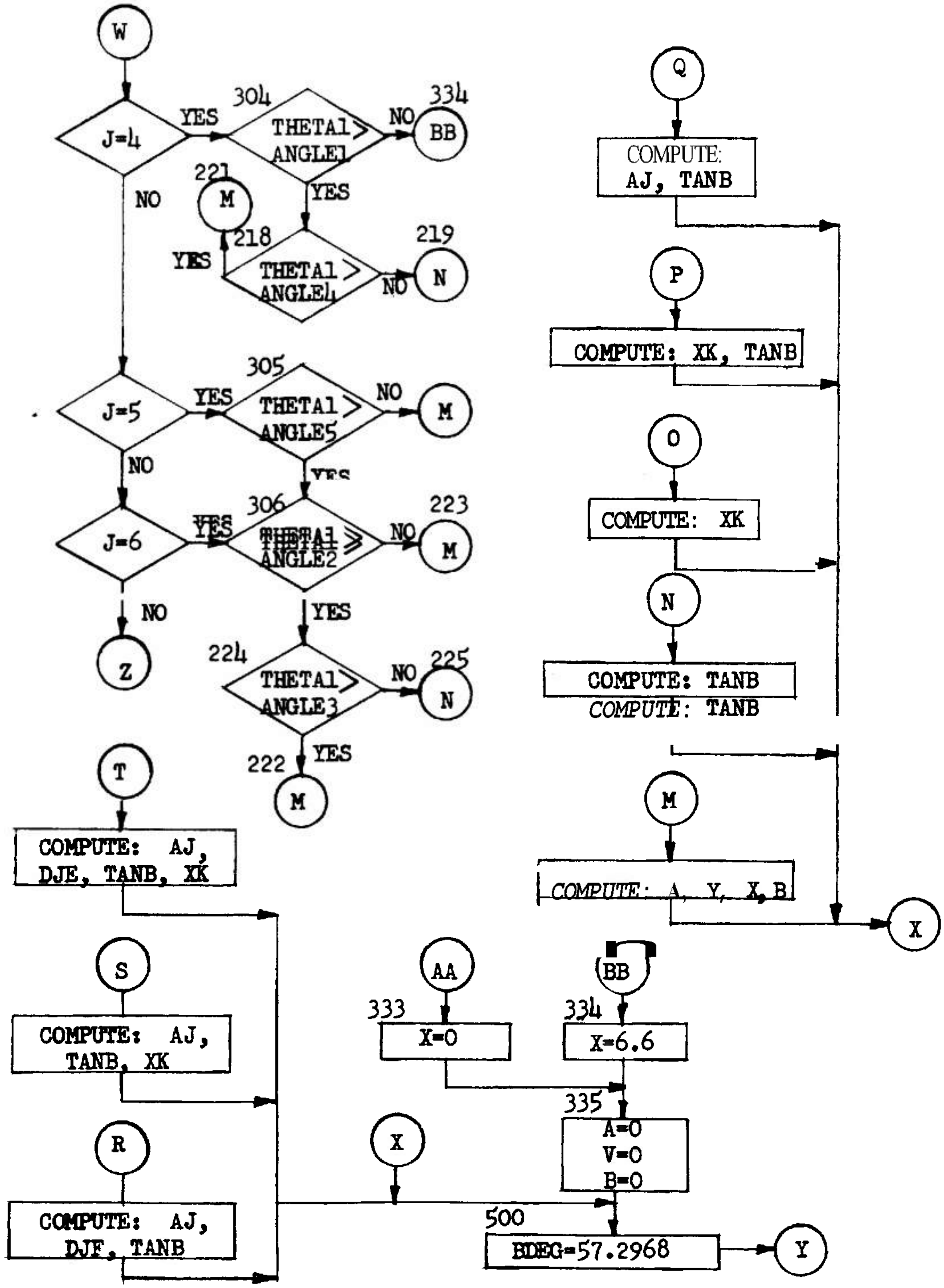
A—15. (Con't.)


L
763
IJ=IJ+1
I=44
YES
TIME=DELT1
NO
739
I=163
TIME=DELT2
1
TIME=TIME+DT
740
TC=TIME
I>44
COMPUTE: ALPHA, OMEGA, THETA
I>163
COMPUTE: TC, ALPHA OMEGA THETA
4
TC=TIME-DELT2
OMEGA=OMEGAM
THETA=OMEGA·TC+THETA(163)
200
IJ=2
743
THETA1=ANGLE1
IJ=6
744
THETA1=ANGLE2
IJ=10
777
THETA1=ANGLE3
M1
M2
746
IJ=14
THETA1=ANGLE4
747
IJ=16
THETA1=ANGLE5
748
COMPUTE: Y, n, VP, THETAD, TORKI, TIMEM
99
BTSUM=0
XX
DO6, J=1, 6
20
U
I>44
J=1
101
THETA1>ANGLE4
12
N
13
M
J=2
102
THETA1>ANGLE5
14
J=3
103
THETA1>ANGLE2
15
N1
16
THETA1>ANGLE3
334
BB
17

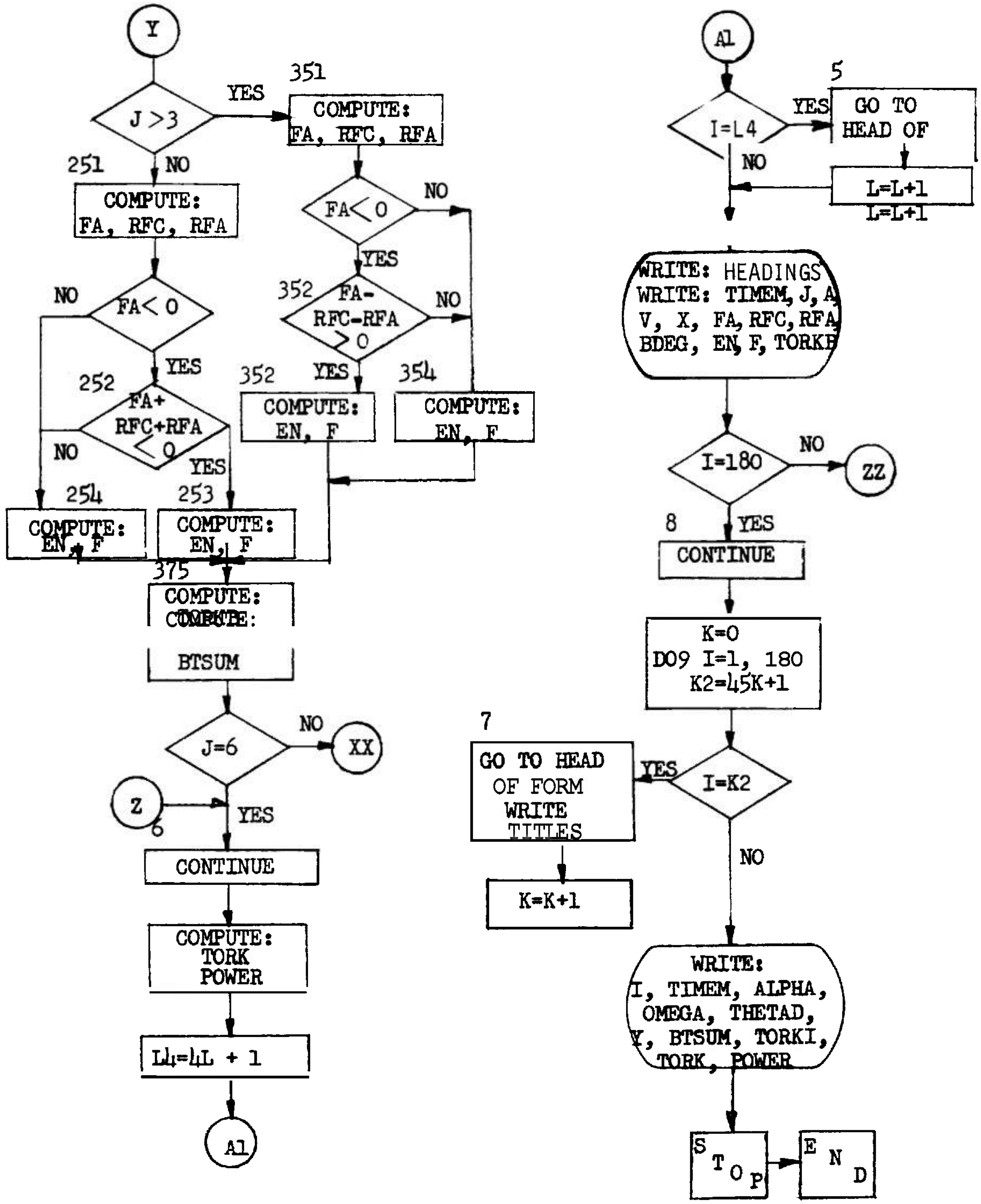
A–15. (Con't.)


N1
J=4
THETA1 > ANGLE1
THETA1 > ANGLE4
J=5
THETA1 > ANGLE5
J=6
THETA1 > ANGLE2
THETA1 > ANGLE3
U
THETA > 18.99
J=1
J=2
J=3
V
W
BB
AA
O
P
Q
M
R
N
T
S
YES
NO
104
105
106
20
21
22
23
24
25
19
334
333
201
202
203
116
117
112
115
40
118
121
123
124
125
204
205
206
212
213
214
215
216
217
301
302
303

A–15. (Con't.)


W
J=4
YES
304
THETA1 > ANGLE1
NO
334
BB
NO
221
M
YES
218
YES
THETA1 > ANGLE4
NO
219
N
305
J=5
YES
THETA1 > ANGLE5
NO
M
NO
YES
306
J=6
YES
THETA1 > ANGLE2
NO
223
M
NO
YES
Z
224
THETA1 > ANGLE3
NO
225
N
YES
222
M
Q
COMPUTE: AJ, TANB
P
COMPUTE: XK, TANB
O
COMPUTE: XK
N
COMPUTE: TANB
COMPUTE: TANB
M
COMPUTE: A, Y, X, B
X
T
COMPUTE: AJ, DJE, TANB, XK
S
COMPUTE: AJ, TANB, XK
R
COMPUTE: AJ, DJF, TANB
AA
333
X=0
BB
334
X=6.6
335
A=0
V=0
B=0
X
500
BDEG=57.2968
Y

A—15. (Con't.)


Y
J > 3
YES
351
COMPUTE: FA, RFC, RFA
251
NO
COMPUTE: FA, RFC, RFA
FA < 0
NO
YES
352
FA-RFC-RFA > 0
NO
FA < 0
NO
YES
252
FA+RFC+RFA < 0
NO
YES
352
COMPUTE: EN, F
354
COMPUTE: EN, F
254
COMPUTE: EN, F
253
COMPUTE: EN, F
375
COMPUTE: BTSUM
J=6
NO
XX
Z
6
YES
CONTINUE
COMPUTE: TORK POWER
L4=4L + 1
A1
A1
I=L4
YES
5
GO TO HEAD OF
NO
L=L+1
WRITE: HEADINGS
WRITE: TIMEM, J, A, V, X, FA, RFC, RFA, BDEG, EN, F, TORKB
I=180
NO
ZZ
8
YES
CONTINUE
K=0
DO9 I=1, 180
K2=45K+1
7
GO TO HEAD OF FORM WRITE TITLES
YES
I=K2
NO
K=K+1
WRITE: I, TIMEM, ALPHA, OMEGA, THETAD, Y, BTSUM, TORKI, TORK, POWER
STOP
END

A–16. PROGRAM LISTING FOR MULTIBARREL POWER

```
      DIMENSION TIMEM(180),ALPHA(180),OMEGA(180),THETA(180),THETAD(180),      3
     1BTSUM(180),TORKI(180),TORK(180),X(6),     V(6),A(6),Y(180),BDEG(6),   -4-
     2FA(6),RFC(6),RFA(6),EN(6),F(6),B(6),TORKB(6),POWER(180)                5
     0READ(5,75)AFK,DFK, AEK, DEK, AK, C1 , C2, C3, ANGLE1,                  6
     1        ANGLE2, ANGLE3, ANGLE4, ANGLE5, EL, R, RC, WB, WST,            7
     2WSE, COEFC1, COEFC2, COEFS1,COEFS2,ALPHA0,OMEGAM,Y1F, Y2F,           8+9
     3        X1F, X2F, TANBF, TANBE, Y1E, Y2E, X1E, X2E, G,                10
     4        DELT1, DELT2,ENDAN1, EMUR, EMUS, EMUT, EYE, BF1, BE1          11
   75 FORMAT (9F8.0)                                                        12
      TIME = 0.0                                                            13
      L = 0                                                                 14
      RCA1 = RC*ANGLE1                                                      15A
      RCTF = RC*TANBF                                                       15B
      RCTE = RC*TANBE                                                       15C
      RSQ = RC**2                                                           15D
      VJF = OMEGAM*RCTF                                                     15F
      VJE = -OMEGAM*RCTE                                                    15G
      AJA3 = 2.0*RC**2*OMEGAM**2                                            15H
      AJAF = AJA3/AFK                                                       15I
      AJDF = -AJA3/DFK                                                      15J
      AJAE = -AJA3/AEK                                                      15K
      AJDE = AJA3/DEK                                                       15L
      AJAP = RCTF*ALPHA0                                                    15M
      AJAN = -RCTE*ALPHA0                                                   15N
      XK1F = RCTF*(ANGLE5-ANGLE4)                                           15O
      XK1E = RCTE*(ANGLE5-ANGLE4)                                           15P
      XK2F = RCTF*(2.0*ANGLE5-ANGLE4)                                       15Q
      XK2E = RCTE*(2.0*ANGLE5-ANGLE4)                                       15R
      XK3F = XK2F + RCTF*ANGLE2                                             15S
      XK3E = XK2E + RCTE*ANGLE2                                             15T
      YF2 = 2.0*Y2F*RC                                                      15U
      YE2 = 2.0*Y2E*RC                                                      15V
      AYF2 = YF2*ALPHA0                                                     15W
      AYE2  =YE2*ALPHA0                                                     15X
      AJ3 = 3.0*RC**2*ALPHA0**2                                             15Y
      AJ3F = AJ3/AFK                                                        15Z
      AJ3E = AJ3/AEK                                                        15AA
      DJ2F = AYF2/DFK                                                       15BB
      DJ2E = AYE2/DEK                                                       15AC
      DJ3F = AJ3/DFK                                                        15AD
      DJ3E = AJ3/DEK                                                        15AE
      AJT4  = 5.0*RSQ*AK**2/6.0                                             15AF
      AJT3 = 10.0*RSQ*AK*C1/3.0                                             15AG
      AJT2 = RSQ*(4.0*AK*C2 +3.0*C1**2)                                     15AH
      AJT1 = 2.0*RSQ*(AK*C3 + 3.0*C1*C2)                                    15AI
      AJT0 = 2.0*RSQ*(C1*C3 + C2**2)                                        15AJ
      BX=C2/C1                                                              15AK
      BSQ=BX**2                                                             15AL
      CX=2.0/C1                                                             15AM
   80 DO 8   I=1,180                                                        16
      I1 = (I-1)/15                                                         A1
      I15 = 15*I1                                                           A2
      II = I15 + 1                                                          A3
      ZZ = II                                                               A4
      ZANGLE = ZZ*ANGLE5                                                    A5
```

A—16. (Con't.)

```
      IF(I1.NE.1)GO TO 702                                     A6
      ANGA1 = ANGLE1 + ZANGLE                                  A7
      ANGD1 = ANGLE2 + ZANGLE                                  A8
      ANGD2 = ANGLE3 + ZANGLE                                  A9
      ANGA2 = ANGLE4 + ZANGLE                                  A10
      IF(I1.EQ.2)GO TO 701                                     A11
      ENDANG = ANGLE5 + ZANGLE                                 A12
      GO TO 755                                                A13
  701 ENDANG = ENDAN1 + ZANGLE                                 A14
  755 IJ = 1                                                   A15A
      T=0.0                                                    A15B
      TE=0.0                                                   A15C
      TAC=0.0                                                  A15D
      TF=0.0                                                   A15E
      TAA=0.0                                                  A15F
  702 IF(I1.GE.3)GO TO 713                                     A16
      IF(IJ.GT.1)GO TO 703                                     A17
      T1 = SQRT(2.0*ANGA1/ALPHA0)                              A18
      T2 = SQRT(2.0*ANGD1/ALPHA0)                              A19
      T3 = SQRT(2.0*ANGD2/ALPHA0)                              A20
      T4 = SQRT(2.0*ANGA2/ALPHA0)                              A21
      T5 = SQRT(2.0*ENDANG/ALPHA0)                             A22
      IF(I.GT.1)GO TO 750                                      A23A
      DT = T1                                                  A23B
      GO TO 738                                                A23C
  750 IF(IJ.NE.1)GO TO 703                                     A23D
      DT = T1 - TIMEM(I-1)/1000.0                              A24
      GO TO 738                                                A25
  703 IF(IJ.NE.2)GO TO 704                                     A26
      DT2 = (T2-T1)/4.0                                        A27
  704 IF(IJ.GT.5)GO TO 705                                     A28
      DT = DT2                                                 A29
      GO TO 738                                                A30
  705 IF(IJ.NE.6)GO TO 706                                     A31
      DT3 = (T3-T2)/4.0                                        A32
  706 IF(IJ.GT.9)GO TO 707                                     A33
      DT = DT3                                                 A34
      GO TO 738                                                A35
  707 IF(IJ.NE.10)GO TO 708                                    A36
      DT4 = (T4-T3)/4.0                                        A37
  708 IF(IJ.GT.13)GO TO 709                                    A38
      DT = DT4                                                 A39
      GO TO 738                                                A40
  709 IF(I1.EQ.2)GO TO 711                                     A41
      IF(IJ.NE.14)GO TO 710                                    A42
  751 DT5 = (T5-T4)/2.0                                        A43
  710 DT = DT5                                                 A44-5
      GO TO 738                                                A46
  711 IF(IJ.EQ.15)GO TO 712                                    A47
      DT = T5-T4                                               A48
      GO TO 738                                                A49
  712 DTHETA = 3.0*ANGLE5                                      A50
      GO TO 727                                                A51-2
  713 IF(I1.EQ.11)GO TO 718                                    A53
      IF(I1.NE.1)GO TO 714                                     A54
      TI0=TIMEM(I-1)/1000.0-DELT1                              A55
      DTHETA = ANGA1                                           A56
      GO TO 727                                                A57-8
```

```
  714 IF(IJ.NE.2)GO TO 715                                      A59
      DTHETA = ANGD1                                            A60
      GO TO 727                                                 A61
  715 IF(IJ.LE.5)GO TO 738                                      A62
      IF(IJ.NE.6)GO TO 716                                      A63
      DTHETA = ANGD2                                            A64
      GO TO 727                                                 A65
  716 IF(IJ.LE.9)GO TO 738                                      A66
      IF(IJ.NE.10)GO TO 717                                     A67
      DTHETA = ANGA2                                            A68
      GO TO 727                                                 A69
  717 IF(IJ.LE.13)GO TO 738                                     A70
      IF(II.EQ.10)GO TO 725                                     A71
      IF(IJ.NE.14)GO TO 745                                     A72
      DTHETA = ENDANG                                           A73
      GO TO 727                                                 A74
  745 GO TO 738                                                 A75-8
  718 IF(IJ.GT.1)GO TO 719                                      A79
      DT = ANGLE1/OMEGAM                                        A80
      GO TO 738                                                 A81
  719 IF(IJ.NE.2)GO TO 720                                      A82
      DT2 = (ANGLE2 - ANGLE1)/(4.0*OMEGAM)                      A83
  720 IF(IJ.GT.5)GO TO 721                                      A84
      DT = DT2                                                  A85
      GO TO 738                                                 A86
  721 IF(IJ.NE.6)GO TO 722                                      A87
      DT3 = (ANGLE3 - ANGLE2)/(4.0*OMEGAM)                      A88
  722 IF(IJ.GT.9)GO TO 723                                      A89
      DT = DT3                                                  A90
      GO TO 738                                                 A91
  723 IF(IJ.NE.10)GO TO 724                                     A92
      DT4 = (ANGLE4 - ANGLE3)/(4.0*OMEGAM)                      A93
  724 IF(IJ.GT.13)GO TO 725                                     A94
      DT = DT4                                                  A95
      GO TO 738                                                 A96
  725 IF(IJ.NE.14)GO TO 726                                     A97
      DT5 = (ANGLE5 - ANGLE4)/ (2.0*OMEGAM)                     A98
  726 DT = DT5                                                  A99
      GO TO 738                                                 A100
  727 JK = 1                                                    A101
      TI=SQRT(BSQ+CX*(DTHETA-C3))-BX                            A102
  728 ANGT = AK*TI**3/6.0 + C1*TI**2/2.0 + C2*TI + C3           A103
      DTRAV = DTHETA - ANGT                                     A104
      IF(ABS(DTRAV).LT.0.001)GO TO 731                          A105
      RDTRAV = DTRAV/(DTHETA-C3)                                A106
      TI =TI*(1.0 + RDTRAV)                                     A107
      JK = JK+1                                                 A108
      IF(JK.GT.25)GO TO 729                                     A109
      GO TO 728                                                 A110
  729 WRITE(6,730)JK,ANGT,DTHETA,DTRAV,RDTRAV,TI                A111
  730 FORMAT(I3,5E15.8)                                         A112
      GO TO 900                                                 A113
  731 IF(II.NE.2)GO TO 732                                      A114
      DT = TI - T5                                              A115
      DT=TI                                                     A115
      GO TO 738                                                 A116
  732 IF(IJ.NE.1)GO TO 733                                      A117
      T1 = T1+DELT1                                             A118
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A–16. (Con't.)

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      GO TO 750                                              A119
  733 IF(IJ.GT.5)GO TO 734                                   A120
      T2 = TI+DELT1                                          A121
      GO TO 703                                              A122
  734 IF(IJ.GT.9)GO TO 735                                   A123
      T3 = TI+DELT1                                          A124
      GO TO 705                                              A125
  735 IF(IJ.GT.13)GO TO 736                                  A126
      T4 = TI+DELT1                                          A127
      GO TO 707                                              A128
  736 IF(II.EQ.10)GO TO 737                                  A129
      T5 = TI+DELT1                                          A130
      GO TO 751                                              A131
  737 DT = TI-T4                                             A132
  738 CONTINUE                                               A133
      T=T+DT                                                 A134
      TAC=TAC+DT                                             A135
      IF(IJ.GT.1)GO TO 741                                  A136A
      GO TO 763                                             A136B
  741 TE=TE+DT                                              A136C
      IF(IJ.GT.5)GO TO 761                                   A137
      GO TO 763                                              A138
  761 TAC=0.0                                                A139
      TF=TF+DT                                               A140
      IF(IJ.GT.9)GO TO 762                                   A141
      GO TO 763                                              A142
  762 TF=0.0                                                 A143
      TAC=0.0                                                A144
      IF(IJ.GT.13)GO TO 742                                 A145
      GO TO 763                                              A146
  742 TF=0.0                                                  17A
      TAC=0.0                                                17B
      TE=0.0                                                  17C
      TAA=TAA+DT                                              17D
  763 IJ=IJ+1                                                 17E
      IF(I.NE.44)GO TO 739                                    17F
      TIME = DELT1                                            17G
      GO TO 740                                               17H
  739 IF(I.NE.163)GO TO 1                                     17I
      TIME = DELT2                                            17J
      GO TO 740                                               17K
    1 TIME = TIME+DT                                         18
  740 TC= TIME                                                19A
      IF(I.GT.44)GO TO 2                                     19B
      ALPHA(I) = ALPHA0                                      20
      OMEGA(I) = ALPHA(I)*TC                                 21
      THETA(I) = OMEGA(I)*TC/2.0                             22
      GO TO 200                                              23
    2 IF(I.GT.163)GO TO 4                                    24-5
      TC= TIME - DELT1                                       26
      ALPHA(I) = AK*TC+ C1                                   27
      OMEGA(I)=AK*TC**2/2.+C1*TC+C2                          28
      THETA(I)=AK*TC**3/6.+C1*TC**2/2.+C2*TC+C3              29
      GO TO 200                                              30
    4 TC= TIME - DELT2                                      31-2
      ALPHA(I) = 0.0                                         33
      OMEGA(I) = OMEGAM                                      34
      THETA(I) = OMEGA(I)*TC+ THETA(163)                     35
```

```
200 IF(IJ.EQ. 2)GO TO  743                                    36A
    IF(IJ.EQ. 6)GO TO  744                                    36B
    IF(IJ.EQ.10)GO TO  777                                    36C
    IF(IJ.EQ.14)GO TO  746                                    36D
    IF(IJ.EQ.16)GO TO  747                                    36E
    THETA1 = THETA(I)-ZANGLE                                  37A
    GO TO 748                                                 37B
743 THETA1 = ANGLE1                                           38A
    GO TO 748                                                 38B
744 THETA1 = ANGLE2                                           38C
    GO TO 748                                                 38D
777 THETA1 = ANGLE3                                           39A
    GO TO 748                                                 39B
746 THETA1 = ANGLE4                                           39C
    GO TO 748                                                 39D
747 THETA1 = ANGLE5                                           39E
748 Y(I) =RC*THETA1                                           40A
    YI = Y(I)-RC*ANGLE2                                       40B
    VP = RC*OMEGA(I)                                          40C
    THETAD(I) = 57.296*THETA(I)                               41
    TORKI(I) = EYE*ALPHA(I)                                   42
    TIMEM(I) = 1000.0*TIME                                    43
 99 BTSUM(I)=0.0                                              44
    DO 6   J=1,6                                              45
    IF(I.GT.44)GO TO 20                                       46
 10 GO TO (101, 102, 103, 104, 105, 106),J                    47
101 IF(THETA1.GT.ANGLE4)GO TO 13                              48
 12 A(J)=AJ3F*T  **2                                          49
    TANB = 2.0*Y(I)/AFK                                       50A
    V(J)=VP**TANB                                             50B
    X(J) = T(I)**2/AFK                                        51
    B(J) = ATAN (TANB)                                        52
    GO TO 500                                                 53
 13 A(J) = AJAP                                               54
    V(J) = TANBF*VP                                           55
    X(J) = X1F + ((Y(I)-Y1F)*TANBF)                           56
    B(J) = BF1                                                57
    GO TO 500                                                 58
102 IF(THETA1.GT.ANGLE5)GO TO 103                             59
 14 A(J) = AJAP                                               60
    V(J) = TANBF*VP                                           61
    X(J) = Y(1)*TANBF + X1F + XK1F                            62
    B(J) = BF1                                                63
    GO TO 500                                                 64
103 IF(THETA1.GT.ANGLE2)GO TO 16                              65
 15 X(J) = Y(I)*TANBF + X1F + XK2F                            66
    A(J) = AJAP                                               67
    V(J) = VP*TANBF                                           68
    B(J) = BF1                                                69
    GO TO 500                                                 70
 16 IF(THETA1.GT.ANGLE3)GO TO 334                             71
 17 A(J)=DJ2F-DJ3F*TE **2                                     72
    TANB = 2.0*(Y2F-YI)/DFK                                   73A
    V(J)=    TANB*VP                                          73B
    X(J)=X2F-(Y2F-YI)**2/DFK+X1F+XK3F                         74
    B(J) = ATAN (TANB)                                        75
    GO TO 500                                                 76
104 IF(THETA1.LE.ANGLE1)GO TO 334                             77
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A—16. (Con't.)

```
   18 IF(THETA1.GT.ANGLE4)GO TO 21
   19 A(J) = -AJ3E*TE **2
      TANB = 2.0*(Y(I)-RCA1)/AEK
      V(J) = -TANB*VP
      XK = (Y(I)-RCA1)**2/AEK
      X(J) = EL-XK
      B(J) = ATAN (TANB)
      GO TO 500
   21 A(J) = AJAN
      V(J) = -TANBE*VP
      XK   = X1E + ((Y(I)-Y1E-RCA1)*TANBE)
      X(J) = EL-XK
      B(J) = BE1
      GO TO 500
  105 IF(THETA1.GT.ANGLE5)GO TO 106
   22 A(J) = AJAN
      V(J) = -TANBE*VP
      XK = X1E + Y(I)*TANBE + XK1E
      X(J) = EL - XK
      B(J) = BE1
      GO TO 500
  106 IF(THETA1.GT.ANGLE2)GO TO 24
   23 A(J) = AJAN
      V(J) = -TANBE*VP
      XK = X1E+XK2E+(Y(I)*TANBE)
      X(J) = EL-XK
      B(J) = BE1
      GO TO 500
   24 IF(THETA1.GT.ANGLE3)GO TO 333
   25 A(J)=-DJ2E+DJ3E*TF **2
      TANB = 2.0*(Y2E-YI)/DEK
      V(J)=-TANB*VP
      XK=X1E+XK2F+X2E-(Y2E-YI)**2/DEK
      X(J) = EL-XK
      B(J) = ATAN (TANB)
      GO TO 500
   20 IF(I.GT.163)GO TO 40
   30 GO TO (201,202,203,204,205,206),J
  201 IF(THETA1.GT.ANGLE4)GO TO 113
      AJ = AJT4*T**4 + AJT3*T**3 + AJT2*T**2 + AJT1*T + AJT0
      A(J)= AJ/AFK
      TANB = 2.0*Y(I)/AFK
      V(J)= TANB*VP
      X(J) = Y(I)**2/AFK
      B(J) = ATAN (TANB)
      GO TO 500
  113 A(J)=RCTF*(AK*TAA+C1)
      V(J) = VP*TANBF
      X(J) = X1F+(Y(I)-Y1F)*TANBF
      B(J) = BF1
      GO TO 500
  202 IF(THETA1.GT.ANGLE5)GO TO 203
      A(J)=RCTF*(AK*T   +C1)
      V(J) = VP*TANBF
      X(J) = X1F + Y(I)*TANBF + XK1F
      B(J) = BF1
      GO TO 500
  203 IF(THETA1.GT.ANGLE2)GO TO 116
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A–16. (Con't.)

```
      A(J)=RCTF*(AK*TAC+C1)                                          133
      V(J) = VP*TANBF                                                134
      X(J) = XIF + Y(I)*TANBF + XK2F                                 135
      B(J) = BF1                                                     136
      GO TO 500                                                      137
  116 IF(THETA1.GT.ANGLE3)GO TO 334                                  138
  117 DJF = YF2*(AK*TF+C1)                                           139
      AJ = AJT4*TF**4 + AJT3*TF**3 + AJT2*TF**2 + AJT1*TF + AJT0     140
      A(J) = (DJF - AJ)/DFK                                          141
      TANB = 2.0*(Y2F-YI)/DFK                                        142
      V(J) = VP*TANB                                                 143
      X(J)=X2F-(Y2F-YI)**2/DFK+X1F+XK3F                              144
      B(J) = ATAN (TANB)                                             146
      GO TO 500                                                      147
  204 IF(THETA1.LE.ANGLE1)GO TO 334                                  148
  118 IF(THETA1.GT.ANGLE4)GO TO 121                                  149
      AJ = AJT4*TE**4 + AJT3*TE**3 + AJT2*TE**2 + AJT1*TE + AJT0     150
      A(J) = -AJ/AEK                                                 151
      TANB = 2.0*(Y(I)-RCA1)/AEK                                     152
      V(J) = -VP*TANB                                                153
      XK=(Y(I)-RCA1)**2/AEK                                          154
      X(J) = EL-XK                                                   155
      B(J) = ATAN (TANB)                                             156
      GO TO 500                                                      157
  121 A(J)=-RCTE*(AK*TAA+C1)                                         158
      V(J) =-VP*TANBE                                                159
      XK = X1E + (Y(I)-Y1E-RCA1)*TANBE                               160
      X(J) = EL-XK                                                   161A
      B(J) = BE1                                                     161B
      GO TO 500                                                      162
  205 IF(THETA1.GT.ANGLE5)GO TO 206                                  163
      A(J)=-RCTE*(AK*T  +C1)                                         164
      V(J) =-VP*TANBE                                                165
      XK = X1E + Y(I)*TANBE + XK1E                                   166
      X(J) = EL-XK                                                   167
      B(J) = BE1                                                     168
      GO TO 500                                                      169
  206 IF(THETA1.GT.ANGLE2)GO TO 124                                  170
      A(J)=-RCTE*(AK*TAC+C1)                                         171
      V(J) =-VP*TANBE                                                172
      XK = X1E+XK2E +(Y(I)*TANBE )                                   173
      X(J) = EL-XK                                                   174
      B(J) = BE1                                                     175
      GO TO 500                                                      176
  124 IF(THETA1.GT.ANGLE3)GO TO 333                                  177
      AJ = AJT4*TF**4 + AJT3*TF**3 + AJT2*TF**2 + AJT1*TF + AJT0     178
  125 DJE = YE2*(AK*TF +C1)                                          179
      A(J) = (DJE + AJ)/DEK                                          180
      TANB = 2.0*(Y2E-YI)/DEK                                        181
      V(J)=-VP*TANB                                                  182
      B(J)=ATAN(TANB)                                                183
      XK=X1E+XK3E+X2E-(Y2E-YI)**2/DEK                                184
      X(J) = EL-XK                                                   185
      GO TO 500                                                      186
   40 GO TO (301,302,303,304,305,306),J                              187
  301 IF(THETA1.GT.ANGLE4)GO TO 213                                  188
  212 A(J) = AJAF                                                    189
      TANB = 2.0*Y(I)/AFK                                            190A
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A–16. (Con't.)

```
      V(J) = VP*TANB                                   190B
      X(J) = Y(I)**2/AFK                               191
      B(J) = ATAN (TANB)                               192
      GO TO 500                                        193
  213 A(J) = 0                                         194
      V(J) = VJF                                       195
      X(J) = X1F + (Y(I)-Y1F)*TANBF                    196
      B(J) = BF1                                       197
      GO TO 500                                        198
  302 IF(THETA1.GT.ANGLE5)GO TO 303                    199
  214 A(J) = 0                                         200
      V(J) = VJF                                       201
      X(J) = Y(I)*TANBF + X1F  + XK1F                  202
      B(J) = BF1                                       203
      GO TO 500                                        204
  303 IF(THETA1.GT.ANGLE2)GO TO 216                    205
  215 A(J) = 0                                         206
      V(J) = VJF                                       207
      X(J) = Y(I)*TANBF + X1F + XK2F                   208
      B(J) = BF1                                       209
      GO TO  500                                       210
  216 IF(THETA1.GT.ANGLE3)GO TO 334                    211
  217 A(J) = AJDF                                      212
      TANB = 2.0*(Y2F-YI)/DFK                          213A
      V(J) = VP*TANB                                   213B
      X(J)=X2F-(Y2F-YI)**2/DFK+X1F+XK3F                214
      B(J) = ATAN (TANB)                               215
      GO TO 500                                        216
  304 IF(THETA1.LE.ANGLE1)GO TO 334                    217
  218 IF(THETA1.GT.ANGLE4)GO TO 221                    218
  219 A(J)= AJAE                                       219
      TANB = 2.0*(Y(I)-RCA1)/AEK                       220A
      V(J)=-VP*TANB                                    220B
      X(J)= EL-(Y(I)-RCA1)**2/AEK                      221
      B(J)= ATAN (TANB)                                222
      GO TO 500                                        223
  221 A(J) = 0                                         224
      V(J) =VJE                                        225
      X(J) = EL-(X1E + (Y(I)-Y1E-RCA1)*TANBE)          226
      B(J) = BE1                                       227
      GO TO 500                                        228
  305 IF(THETA1.GT.ANGLE5)GO TO 306                    229
  222 A(J) = 0                                         230
      V(J) =VJE                                        231
      X(J) = EL - (X1E + Y(I)*TANBE + XK1E)            232
      B(J) = BE1                                       233
      GO TO 500                                        234
  306 IF(THETA1.GT.ANGLE2)GO TO 224                    235
  223 A(J) = 0                                         236
      V(J) =VJE                                        237
      X(J) = EL - (X1E+XK2E+Y(I)*TANBE)                238
      B(J) = BE1                                       239
      GO TO 500                                        240
  224 IF(THETA1.GT.ANGLE3)GO TO 333                    241
  225 A(J) = AJDE                                      242
      TANB = 2.0*(Y2E-YI)/DEK                          243
      V(J) = -VP*TANB                                  244
      X(J)=EL-X1E-XK3E-X2E+(Y2E-YI)**2/DEK             245
```

```
      B(J) = ATAN (TANB)                                              246      462
      GO TO 500                                                       247      463
  333 X(J) = 0.0                                                      248A     464
      GO TO 335                                                       248B     465
  334 X(J) = 6.6                                                      248C     466
  335 V(J) = 0.0                                                      249      467
      A(J) = 0.0                                                      250      468
      B(J) = 0.0                                                      251      469
  500 BDEG(J) = 57.296*B(J)                                           252      470
      IF(J.GT.3)GO TO 351                                             253      471
  251 FA(J) = A(J)*(WB+WST)/G                                         254      472
      RFC(J) = R*OMEGA(I)**2*(EMUT*WB+EMUS*WST)/G                     255      473
      RFA(J) = R*ALPHA(I)*(EMUT*WB+EMUS*WST)/G                        256      474
      IF(FA(J).GE.0.)GO TO 254                                        257      475
  252 IF (FA(J)+RFC(J)+RFA(J))253,253,254                             258      476
  253 EN(J)=(FA(J)+RFC(J)+RFA(J))/(-COEFC2*COS (B(J))-COEFS2*SIN (B(J)))  259  477
      F(J) =EN(J)*(EMUR*COS (B(J))-SIN (B(J)))                        260      478
      GO TO 375                                                       261      479
  254 EN(J)=(FA(J)+RFC(J)+RFA(J))/(COEFC1*COS (B(J))-COEFS1*SIN (B(J)))   262  480
      F(J) =EN(J)*(SIN (B(J))+EMUR*COS (B(J)))                        263      481
      GO TO 375                                                       264      482
  351 FA(J) = A(J)*(WB + WSE)/G                                       265      483
      RFC(J) = R*OMEGA(I)**2*(EMUT*WB+EMUS*WSE)/G                     266      484
      RFA(J) = R*ALPHA(1)*(EMUT*WB+EMUS*WSE)/G                        267      485
      IF(FA(J).GE.0.0)GO TO 354                                       268      486
  352 IF(FA(J)-RFC(J)-RFA(J))354,354,353                              269      487
  353 EN(J)=(FA(J)-RFC(J)-RFA(J))/(COEFC2*COS (B(J))+COEFS2*SIN (B(J)))   270  488
      F(J)=EN(J)*(EMUR*COS (B(J))-SIN (B(J)))                         271A     489
      GO TO 375                                                       271B     490
  354 EN(J)=(FA(J)-RFC(J)-RFA(J))/(COEFS1*SIN (B(J))-COEFC1*COS (B(J)))   272  491
      F(J)=EN(J)*(SIN (B(J))+EMUR*COS (B(J)))                         273      492
  375 TORKB(J) = F(J)*RC                                              274      493
      BTSUM(I) = BTSUM(I)+TORKB(J)                                    275A     494
    6 CONTINUE                                                        275B     495
      TORK(I) = BTSUM(I) + TORKI(I)                                   276      496
      POWER(I) = TORK(I)*OMEGA(I)/6600.0                              277      497
      L3 = L*3+1                                                      278      498
  803 IF(I.NE.L3)GO TO 66                                             279      499
    5 WRITE(6,501)                                                    280      500
      L = L+1                                                         281      501
  501 FORMAT(1H1/37X,43HTABLE 6-2 OUTPUT OF EACH BOLT AT GIVEN TIME//)     282  502
   66 WRITE(6,601)TIMEM(I),(J,A(J),V(J),X(J),FA(J),RFC(J),RFA(J),BDEG(J)   283  503
     1,EN(J),F(J),TORKB(J),J=1,6)                                     284      504
  601 FORMAT(  /41X,15HCAM DYNAMICS AT F7.2,13H MILLISECONDS//33X74HBOLT   285  505
     1    AXIAL     NORMAL    ANGULAR     CAM        CAM        CAM      286      506
     2BOLT/108H BOLT      BOLT        BOLT       AXIAL   INERTIA   FRICTI 287      507
     3ON  INERTIA    CURVE     NORMAL    DRIVING    DRIVING/108H  NO  ACCE 288     508
     4LERATION    VELOCITY   TRAVEL     FORCE      FORCE      FORCE    ANGL 289     509
     5E     FORCE      FORCE      TORQUE/7X,100HIN/SEC/SEC      IN/SEC    290      510
     6INCH      POUND      POUND      POUND     DEGREE      POUND      POUND 291     511
     7    LB-IN//(I4,F12.0,F12.2,F10.3,F9.0,2F10.0,F10.3,3F10.0))     292      512
    F CONTINUE                                                        293A     513
      K = 0                                                           293B     514
      DO 9  I = 1,180                                                 293C     515
      K2 = K*45+1                                                     293D     310
      IF(1.NE.1)GO TO 1051                                            293E     517
      WRITE(6,1052)                                                   293F     518
      GO TO 1054                                                      2936     519
```

A—16. (Con't.)

```
1052 FORMAT(1H1/33X,29HTABLE 6-3 GUN OPERATING POWER/)                  293H
 802 FORMAT(1H1/30X,35HTABLE 6-3 CONTD GUN OPERATING POWER/)            293I
1051 IF(I.NE.K2)GO TO 9                                                 294
   7 WRITE(6,802)                                                       295A
1054 WRITE(6,1053)                                                      295B
     K = K+1                                                            296
   9 WRITE(6,801)I, TIMEM(I),ALPHA(I),OMEGA(I),THETAD(I),Y(I),BTSUM(I)  297
    1,TORKI(1),TORK(I),POWER(I)                                         298
1053 FORMAT (                                  19X,46HROTOR       ROTOR  299
    1  ROTOR       BOLT      TOTAL/5H  IN-,13X,78HANGULAR     ANGULAR   AN  300
    2GULAR  PERIPHERAL    CAM      ROTOR   REQUIRED   REQUIRED/95H CRI-  301
    3 TIME   ACCELERATION  VELOCITY   TRAVEL     TRAVEL     TORQUE   TORQ  302
    4UE   TORQUE      HORSE-/94H MENT  MILSEC   RAD/SEC/SEC   RAD/SEC  303
    5DEGREE      INCH      LB-IN     LB-IN     LB-IN        POWER/ )  304
 801 FORMAT (I4,F9.3,2F11.2,F11.1,F10.3,3F9.0,F11.1)                   305
 900 STOP                                                               306
     END                                                                307
```

APPENDIX B

AUTOMATIC CONTROL OF ROUNDS IN A BURST FOR WEAPON EFFECTIVENESS

Since it is not generally possible to automatically vary the number of rounds in a burst from modern automatic weapons, it is of interest to know whether or not advantage could be taken of such a capability to increase the cost effectiveness of such weapons.

In the most general case, hit probability $(Pr)_{eh}$ is a variational problem because hit probability is represented as

$$(Pr)_{eh} = 1 - \int_0^1 \left[1 - \frac{A}{2\pi\sigma_d^2} x^{\sigma_b^2/\sigma_d^2}\right]^n dx \qquad \text{(B-1)}$$

where

$(Pr)_{eh}$ = engagement hit probability

A = area of target

σ_d^2 = variance of dispersion

σ_b^2 = variance of bias

n = number of rounds in burst (each round assumed independent)

$x = \exp\left[r^2 / 2\sigma_b^2\right]$

r = radial distance from target center

The reference for **Eq.** B-1 and its derivation is **Eq.** 4-413, **AMCP** 706-327, ***Fire Control Systems – General.***

Observe that $(Pr)_{eh}$ is a function of, among other parameters, n. Thus, there is an optimum value of n for a burst. To exceed this optimum value of n increases the use and cost of rounds without appreciably increasing the hit probability. Use of an n smaller than the optimum value decreases hit probability, thereby, decreasing the effectiveness of the weapon.

Extensive studies to date with plotted curves for various values of n have shown that, in terms of $(Pr)_{eh}$, a tight control of n should reduce an excess use of ammunition. Since the value of n is generally under the trigger control of the gunner who cannot concentrate on or control discrete number of rounds in most circumstances, it appears logical that consideration should be given to the evaluation and design of a capability in the trigger or sear area to easily preselect an automatic number of rounds in a burst. Fig. B-1 illustrates the nature of $(Pr)_{eh}$ in relation to the number of rounds n in a burst. In interpreting this figure

$$\alpha = \frac{A}{2\pi\sigma_d^2}$$

$$R = \frac{\sigma_b^2}{\sigma_d^2}$$

For additional information on effectiveness vs the number of rounds in a burst for point fire refer to

1. ***Summary of Test Data and Effectiveness Evaluation for Special Purpose Individual Weapon,*** Ballistic Research Laboratories Technical Note 1542, Aberdeen Proving Ground, Md., August 1964.

2. ***Dispersions for Effective Automatic Small Arms Fire and a Comparison of the M14 Rifle With a Weapon Yielding Effective Automatic Fire,*** Ballistic Research Laboratories Technical Note 1372, Aberdeen Proving Ground, Md., January 1961.

The methods for automatically controlling the number of rounds in a burst are limited only by the ingenuity of the designer. Several methods that have been successfully employed are described briefly:

a. The M61A1 Vulcan Machine Gun employs a burst length control device which is essentially an electrical accessory that is preset by the operator. The accessory controls the length of time that power is supplied to the gun drive and firing circuits. The original design required

bursts of **10, 30,** 60, and 100 rounds. These were later reduced to 10 and 60 because of operational difficulties.

b. **A** second type which performed successfully is a burst circuit located on the side of the gun cradle, which counts the number of rounds and then cams the trip lever down on the last round fired to end the burst. **As** the gun returns to full battery position, a torsion spring is activated which sets the circuit for the next burst. The number of rounds per burst is manually set only once. On the assumption that the circuit is set for a 10-round burst and the trigger is released after 6 rounds have been fired, the lug will cam the lever down and the sear will move over the trip lever. The gun will now settle into full battery position, and the circuit reset and ready to count **10** rounds. The trigger must be pulled and released for each burst.

c. **A** third type, more applicable to self-powered guns, consists of an escapement mechanism which is preset to some desired number of rounds up to maximum capacity. **As** each round **is** fired, the escapement rotates closer to zero or to stopping the gun through holding of the sear or trigger control.

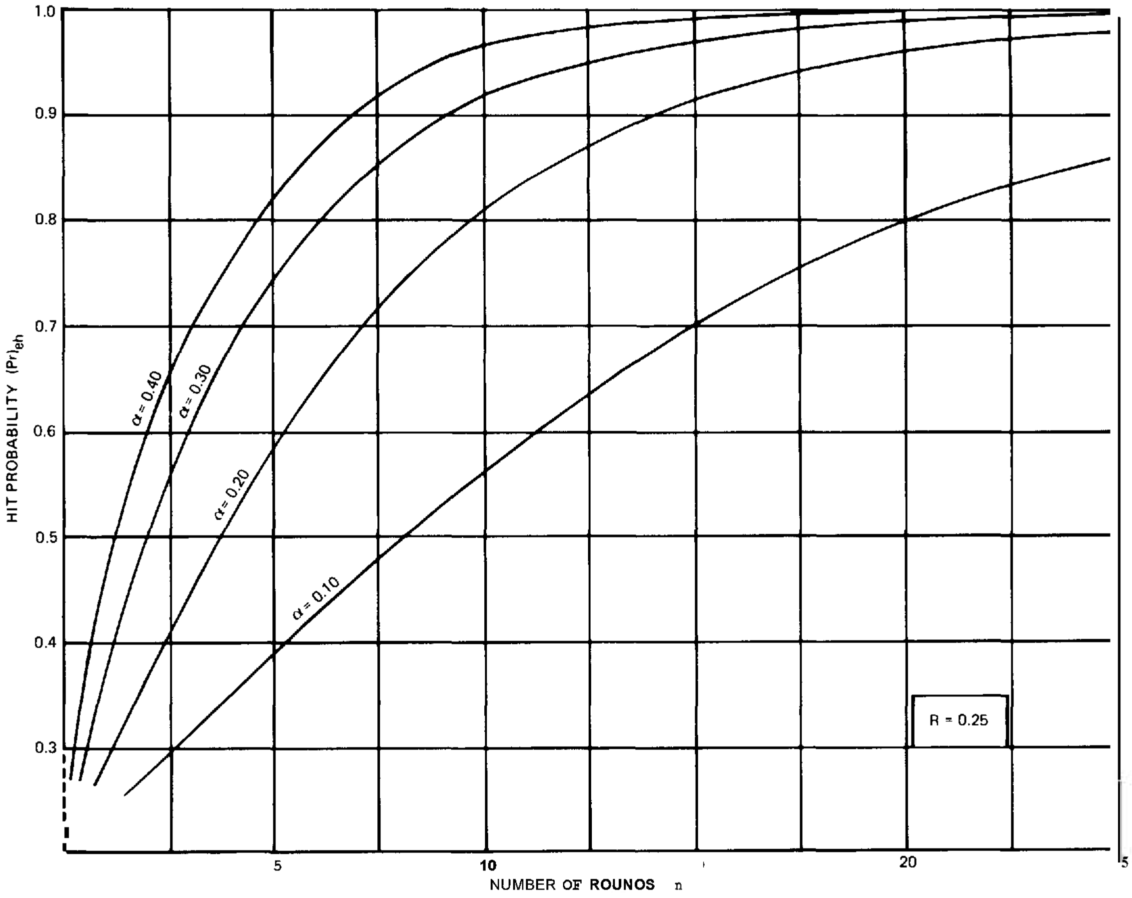

Figure B–1. Hit Probability vs Number of Rounds in a Burst

GLOSSARY

accelerator. A cam arrangement that converts barrel momentum to bolt momentum thereby increasing bolt velocity and decreasing time.

automatic weapon. A rapid, self-firing weapon.

barrel spring. The driving spring equivalent for the barrel.

belt, ammunition. Fabric or metal band with loops for carrying cartridges that are fed from it to an automatic weapon.

belt, disintegrating. An ammunition belt whose empty links are detached as the individual rounds are removed.

blowback. The class of automatic weapon that uses the propellant gas pressure on the cartridge case base to force the bolt open, barrel and receiver remaining relatively fixed.

blowback, advanced primer ignition. A blowback gun that fires before the round is fully chambered.

blowback, delayed. A blowback gun that keeps the bolt locked until the projectile leaves the muzzle.

blowback, retarded. A blowback gun that has a linkage to provide a large, early resistance to recoil.

blowback, simple. A blowback gun that relies on bolt inertia for early recoil resistance.

breech closure. Complete closing of the breech by bolt or breechblock.

buffer spring. A spring that augments either driving or barrel spring during the last stages of either bolt or barrel travel.

compression time. The time during which a spring becomes compressed..

cutoff. The closing of the gas port between bore and operating cylinder.

cutoff expansion system. An expansion system that has a valve to close the gas inlet port after the operating piston moves a prescribed distance.

counterrecoil time. Time required for a counterrecoiling part to return to battery.

critical pressure. The pressure on the discharge end of a nozzle at which flow rate becomes independent regardless of how much the down stream pressure is reduced.

cycle, time of. The time required for a gun to negotiate the firing cycle.

driving spring. The spring that stores some of the bolt recoil energy, stops the recoiling bolt, then drives it into the in-battery position.

ejector. A device in the breech mechanism which automatically throws out an empty cartridge case or unfired cartridge from the breech or receiver.

expansion system. An operating cylinder of a machine gun that has an initial expansion chamber at the gas inlet port.

external power unit. A unit that drives some or all operating components of an automatic weapon by deriving its power from a source other than the propellant gases.

extractor. A device in the breech mechanism that pulls an empty cartridge case or unfired cartridge from the chamber.

firing cycle. The sequential activity that takes place from the time a round is fired until the next round is about to be fired.

firing mechanism. The mechanism that actuates and controls the firing of a gun.

firing pin. The component of a firing mechanism that contacts the primer and relays the detonating energy of the firing mechanism to the primer.

flexibility. The flexing of an ammunition belt so that it will assume a fan-like attitude or form a helix.

flexibility, base fanning. The fan-like flexibility where the cartridge case bases form the inner arc.

flexibility, free. The flexibility that becomes available by taking up the slack provided by the accumulated clearances of the links.

flexibility, helical. The flexibility that forms a helix.

flexibility, induced or forced. The flexibility that is derived from the elastic deflection of the individual links.

flexibility, nose fanning. The fan-like flexibility where the cartridge noses form the inner arc.

gas filling period. The time of gas activity in the operating cylinder.

gas-operated. The class of automatic weapon that uses propellant gases vented through the barrel wall to operate all moving components.

hammer. The striking component of a firing mechanism.

impingement system. A gas-operated gun that has no initial expansion chamber at the gas inlet port.

link. The unit of an ammunition belt that firmly holds and carries one round.

link, extracting type. A link from which the round is removed axially by pulling it rearward.

link, push through type. A link from which the round is removed axially by pushing it forward by bolt or rammer.

line, side stripping type. A link from which the round is removed perpendicular to the axis.

locking cam, bolt. The cam that controls the locking and unlocking of the bolt.

locking period. The time needed to lock the bolt in its closed position.

machine gun. An automatic weapon that can sustain relatively long bursts of firing.

magazine, box. A magazine, usually detachable, of rectangular construction and of small capacity.

magazine, drum. A magazine of drum construction whose capacity is larger than that of the box magazine.

operating cylinder. The gas pressure system that powers a gas-operated machine gun.

override. The clearance between bolt face and cartridge case base when the bolt is in its rearmost position.

propellant gas period. The time that propellant gas pressures are effective.

rate of fire. The number of rounds fired per minute.

recoil, long. A recoil-operated gun that has the barrel recoiling as far as the bolt, both recoiling as a unit but counterrecoiling separately.

recoil-operated. The class of automatic weapon that uses the energy of all recoiling parts to operate the gun.

recoil, short. A recoil-operated gun that has the barrel recoiling a short distance, with barrel and bolt moving as a unit for part of that distance, whereupon the bolt is released to continue its rearward motion.

recoil time. The time required for a recoiling part to negotiate its rearward travel.

recoil time, accelerating. The time required to accelerate the recoiling parts.

recoil time, decelerating. The time required to stop the recoilingparts.

sear. The component of a firing mechanism that releases the hammer.

safety. A locking or cutoff device that prevents a weapon from being fired accidentally.

semiautomatic. A gun that functions automatically except that each round fired must be triggered manually.

specific impetus. The unit energy, ft-lb/lb, of a propellant.

surge time. The period of time required for a compression wave to traverse a spring.

tappet system. An impingement system that has a very short piston travel.

trigger. The component of a firing mechanism that releases the sear to initiate all firing.

trigger pull. The force that is required to actuate the trigger.

unlocking period. The time needed to release the bolt from its closed position.

velocity of free recoil. The maximum velocity that a recoiling part would attain if left unimpeded during recoil.

wall ratio. The ratio of outer to inner diameter of a hollow cylinder.

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