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RESEARCH AND DEVELOPMENT OF MATERIEL

ENGINEERING DESIGN HANDBOOK

CARRIAGES AND MOUNTS SERIES RECOIL SYSTEMS

U.S. ARMY MATERIEL COMMAND

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(AMCRD)

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PREFACE

This handbook on Recoil Systems has been prepared as one of a series on Carriages and Mounts. It presents information on the fundamental operating principles of recoil systems and the design of recoil systems and their components.

Text and line illustrations were prepared by The Franklin Institute, under contract with Duke University, with the technical assistance of the Ordnance Weapons Command.

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LIST OF SYMBOLS*

a = average recoil acceleration during Δt
a_b = acceleration during buffing
a_c = counterrecoil orifice area
a_o = recoil orifice area
a_{o_b} = buffer orifice area
a_r = acceleration during counterrecoil
A = effective area of recoil piston
A_1 = contact area of packing on cylinder wall
A_b = effective area of buffer piston
A_c = cross section area of control rod
A_{cr} = area of counterrecoil piston (independent system)
A_g = bore area of gun tube
A_o = peripheral discharge area of recoil throttling valve
A_r = root area of thread
A_R = recuperator area; same as area of floating piston
A_s = root area of valve stem
A_v = effective pressure area of throttling valve
b = width of packing
c = open periphery of valve head
C_c = counterrecoil orifice coefficient
C_o = orifice coefficient in general or for recoil orifice
D_1 = inside diameter of recoil cylinder
E_{cr} = kinetic energy of counterrecoil
E_r = kinetic energy of recoil
f_c = total frictional resistance of packing in recoil brake and counterrecoil cylinder
f_{cr} = hydraulic resistance of recoil orifice during counterrecoil
f_o = hydraulic resistance at each restriction in flow path other than controlled orifice
f_p = total frictional resistance of packings
f_p' = frictional resistance of a packing assembly
f_r = frictional resistance of packing in recuperator
F = force tending to accelerate recoiling parts
F_1 = static force of recuperator in battery
F_1' = static resistance to counterrecoil
F_2 = recuperator force, end of recoil
F_a = net accelerating force or inertia force of recoiling parts

* Symbols peculiar to double recoil systems are listed in paragraph 149 and those for recoil systems for small arms in paragraph 187.

F_{ac} = available counterrecoil force before throttling
F_b = total buffer force
F_b' = buffer net decelerating force
F_{bc} = hydraulic resistance of counterrecoil orifice
F_B = initial Belleville spring load
F_c = coil spring force of recoil throttling valve
F_c' = initial coil spring load
F_{cr} = net counterrecoil accelerating force
F_{fc} = frictional resistance of slides during counterrecoil
F_g = propellant gas force
F_o = hydraulic resistance of recoil brake
F_p = force on counterrecoil piston
F_r = force on counterrecoil piston
F_R = recuperator force; same as force on floating piston
F_s = spring load in general; the combined load of coil and Belleville springs
F_s' = combined initial spring load of coil and Belleville springs
F_{s_2} = spring force when buffers are contacted
F_θ = radial force of packing on cylinder
h = velocity head
h_c = lift of recoil throttling valve, coil spring active
h_s = lift of recoil throttling valve, both springs active
k = stress concentration factor
K = total resistance to recoil
K_a = resistance offered by elastic medium of recuperator
K_b = Belleville spring rate, recoil throttling valve
K_c = coil spring rate, recoil throttling valve
K_f = frictional resistance of cradle slides during recoil
K_s = spring rate in general; spring rate of combined coil and Belleville springs
K_p = pressure factor
K_R = recoil rod force
L = length of recoil
L_{cr} = length of counterrecoil stroke to contact buffers
M = mass equivalent of projectile and propellant gas
M_b = bulk modulus of fluid
M_r = mass of recoiling parts

n = polytropic exponent
n = subscript denoting value at stated interval
$n-1$ = subscript denoting value preceding the stated interval
P = pressure in recoil brake cylinder
ΔP = pressure drop across counterrecoil orifice
P_{a_c} = oil pressure before throttling through counterrecoil orifice
P_{a_o} = oil pressure before throttling through recoil orifice during counterrecoil
P_0 = minimum recuperator pressure, in-battery
P_1 = gas pressure at end of recoil
P_2 = gas pressure when buffers are contacted
P_a = axial pressure in packing
P_b = buffer pressure
P_g = propellant gas pressure
P_h = pressure rise caused by orifice
P_m = maximum fluid pressure
ΔP_o = pressure drop across recoil orifice during counterrecoil
P_p = proof pressure
P_r = recuperator pressure or equivalent pressure of spring
P_R = radial pressure in packing
P_s = axial pressure in packing produced by spring
P_x = gas pressure at any position of recoil
P_Θ = fluid pressure on packing
Q = rate of flow
Q_c = rate of flow through counterrecoil orifice
Q_o = rate of flow through recoil orifice during counterrecoil
R = secondary recoil force
S_b = length of buffer stroke
S_f = factor of safety
t = time
At = change in time
v = recoil velocity
Av = change in velocity
v_a = average velocity during At
v_b = velocity of counterrecoil during buffing
v_f = maximum velocity of free recoil
Δv_f = change in free recoil velocity
v_{f_1} = velocity of free recoil at time, t_1
v_{f_2} = velocity of free recoil at time, t_2
v_m = muzzle velocity of projectile
v_o = velocity of flow through orifice
$v,$ = counterrecoil velocity
$Av,$ = change in counterrecoil velocity
v_R = velocity of floating piston
V_0 = gas volume, in-battery
V_1 = gas volume at end of recoil
V_2 = gas volume when buffers are contacted
AV = gas displacement
w = density of fluid
W_{cr} = available energy in recuperator for counter-recoil
W_g = weight of propellant charge
W_p = weight of projectile
W_r = weight of recoiling parts
W_s = energy required to overcome static resistance
x = distance of recoil at time t
Ax = distance recoiled during At
x_b = distance of buffer travel at any time t
x_c = displacement of control rod
x_r = distance of counterrecoil at any time t
$Ax,$ = distance counterrecoiled during At
e = angle of elevation
λ = in-battery sustaining factor
μ = coefficient of friction
ν = leakage factor
ρ = mass density of fluid
σ_r = radial stress
σ_t = tensile stress; hoop stress
σ_y = yield strength
τ_m = maximum shear stress

CARRIAGES AND MOUNTS SERIES
RECOIL SYSTEMS*

I. INTRODUCTION

A. GENERAL

1. This is one of a series of handbooks on Carriages and Mounts. This handbook deals with the design of recoil mechanisms.

B. FUNCTION OF A RECOIL MECHANISM

2. A recoil mechanism moderates the firing loads on the supporting structure by prolonging the time of resistance to the propellant gas forces. As the gas pressure propels the projectile toward the muzzle, it exerts an equal and opposite force on the breech, which tends to drive the gun backward. The main purpose of a recoil mechanism is to cushion this force and limit the rearward movement.

3. The dynamics of recoil presents a study in the conservation of momentum. From mechanics we have the expression for a force,

$$F = \frac{d(mv)}{dt}, \tag{1}$$

where: m = mass,

v = velocity.

The forces tending to separate two bodies are equal and opposite in direction, thus, equating the forces in the above equation we have

$$F_1 = \frac{d(m_1v_1)}{dt} = F_2 = \frac{d(m_2v_2)}{dt}, \tag{1a}$$

and

$$d(m_1v_1) = d(m_2v_2). \tag{1b}$$

Integrating we have

$$m_1v_1 = m_2v_2. \tag{1c}$$

This principle is directly applicable to the recoil activity of guns where one side of Equation (1c) represents the momentum of the recoiling parts and the other side represents the total momentum of the projectile and propellant gases moving in the opposite direction. The only unknown value, the velocity of free recoil, can be determined by appropriate substitutions in Equation (1c). Once this velocity is found, the kinetic energy of the recoiling mass can be calculated (see Equation 3a Par. 78). At this stage, the method by which the kinetic energy is dissipated becomes the sole basis of design of the recoil mechanism.

4. It would be possible to attach the gun tube rigidly to its carriage, thereby exposing the structure to the full propellant pressure force which may exceed two million pounds in large guns. However, to be strong enough to sustain this tremendous load, the structure would become overwhelmingly large and unwieldy. The expanse of the base to provide stability, that is, to prevent tipping over, would be enormous. Pistols and shoulder arms of the closed breech type are designed to this concept, but they rely upon the human body to provide the recoil resistance.

5. In practical design of larger weapons, the gun is permitted to recoil, or move back, a prescribed distance and against some predetermined resistance. Figure 1 shows a weapon with a recoil mechanism. The function of a recoil mechanism is to absorb the energy of recoil effectively and then return the gun to the "in-battery" position. The large rearward thrust acts for a very short time, only so long as the propellant gas pressure acts. In order to confine the supporting structure to reasonable size and weight, and to achieve stability with a relatively small firing base, it is necessary to prolong the duration of resistance to the impulsive force of the propellant gas.

6. The propellant gas pressure force, instead of being applied directly to the carriage structure, merely accelerates the gun and other recoiling parts in their recoil motion. This motion is retarded by a predetermined and controlled force. The retarding force is the one which must be taken by the

* Prepared by Martin Regina, Laboratories for Research and Development of The Franklin Institute.

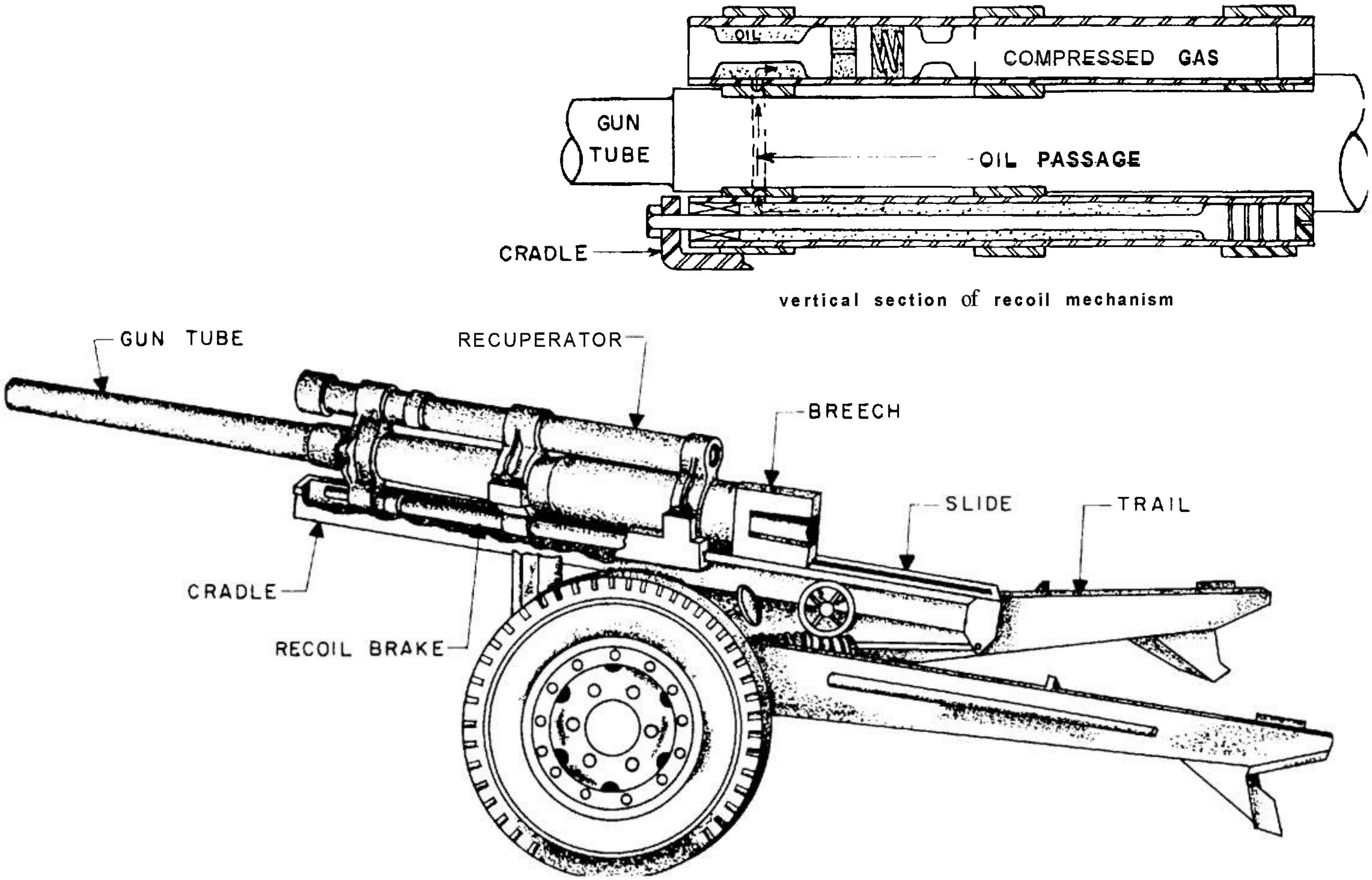


Figure 1. Weapon Showing Recoil Mechanism (Recoil Brake and Recuperator)

structure. It is much smaller than the original propellant gas force, because it acts over a much greater interval of time and an appreciable distance; the longer the distance, the smaller the force. The resistance to motion is provided mostly by the recoil mechanism and partly by gun slide friction. A detailed analysis of this activity is discussed in Parts D.1 and D.2 of Chapter IX.

7. The product of recoil distance and retarding force is recoil energy, which is a primary criterion in the design of a recoil mechanism. When the weapon is fired, recoil begins immediately. The energy of recoil is developed in the short time the propellant gas forces are acting. This energy is expended in several ways, namely: (1) a small amount is stored in deflecting the structure and ordinarily may be safely ignored; (2) some is absorbed by gun-slide friction; (3) the greatest portion is dissipated by the recoil mechanism; and (4) a sufficient amount is stored in the recuperator to return the gun to the in-battery position.

8. While returning to the in-battery position, the moving parts acquire a counterrecoil energy. Some means must be provided to absorb this energy and ease the unit into the in-battery position. This is accomplished by the counterrecoil buffer.

11. THE RECOIL SYSTEM

A. DEFINITIONS

9. A recoil system is defined as an assembly of components whereby the forces acting on a gun and its related mount during a firing cycle can be controlled and limited to certain parameters by one or more recoil mechanisms.

10. The recoil mechanism is that component of a recoil system which absorbs and stores the recoil energy and then returns the gun to battery position.

11. There are two types of recoil systems: a single and a double system. A single recoil system is one wherein the recoil mechanism (or mechanisms) has its recoiling parts moving, as a single coordinated unit, in one direction (see Chapter III, Part A).

12. A double recoil system is one in which the recoil mechanisms control two separate units of recoiling parts, with both coordinated units moving in the same general rearward direction.,but in paths not necessarily parallel (see Chapter 111, Part B, and Chapter XI).

B. DESCRIPTION OF THE BASIC COMPONENTS

13. A recoil mechanism is comprised of three basic components : a recoil brake, a counterrecoil mechanism, and a buffer as shown diagrammatically in Figs. 2, 3, and 4. The recoil brake consists of a hydraulic cylinder and piston assembly. As the piston moves within the cylinder, a force is generated by restricting the flow of hydraulic fluid from the pressure chamber of the cylinder. The magnitude of this restricting force is a function of the flow of fluid through one or more orifices, whose size is regulated to provide the desired recoil velocity and pressure curves. The recoil energy absorbed by this restricting force is dissipated as heat.

14. The counterrecoil mechanism is composed of a recuperator and counterrecoil cylinder assembly.

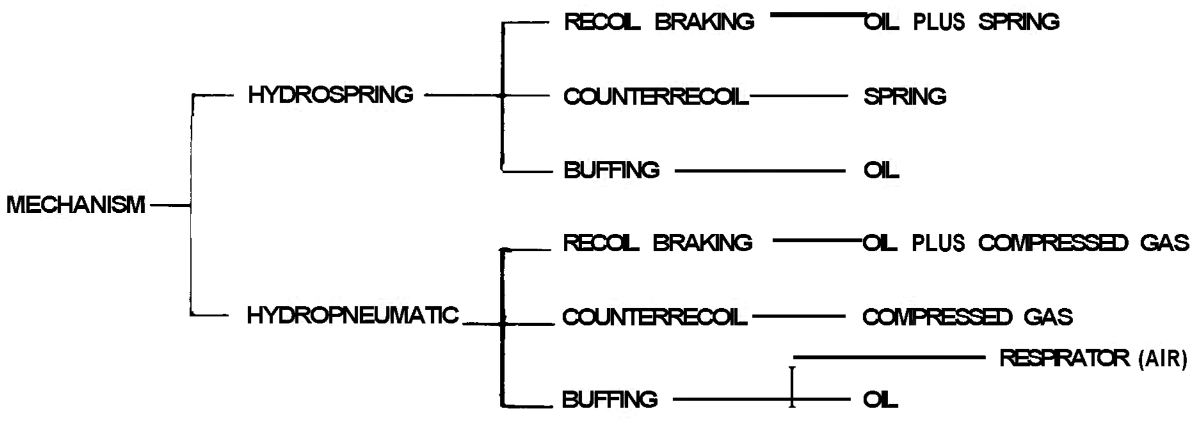


Figure 2. Diagram of Recoil Mechanism Types (General)

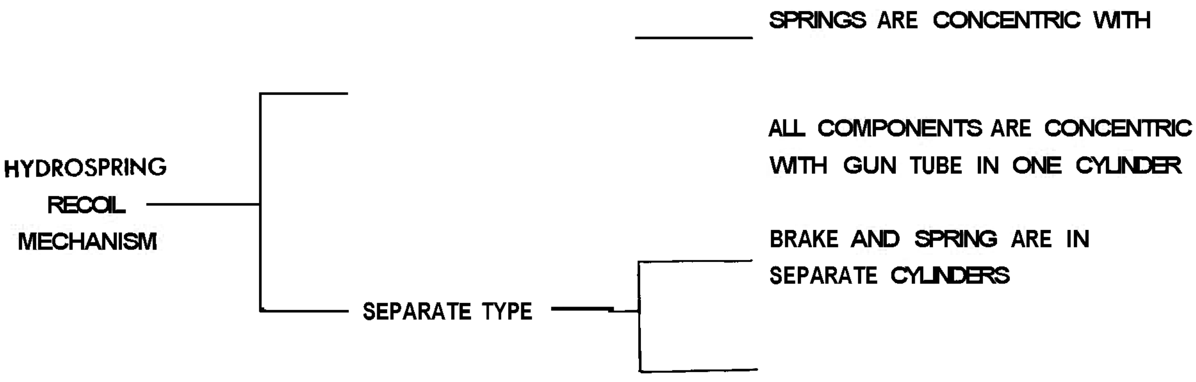


Figure 3. ***Diagram* of *Hydrospring Types***

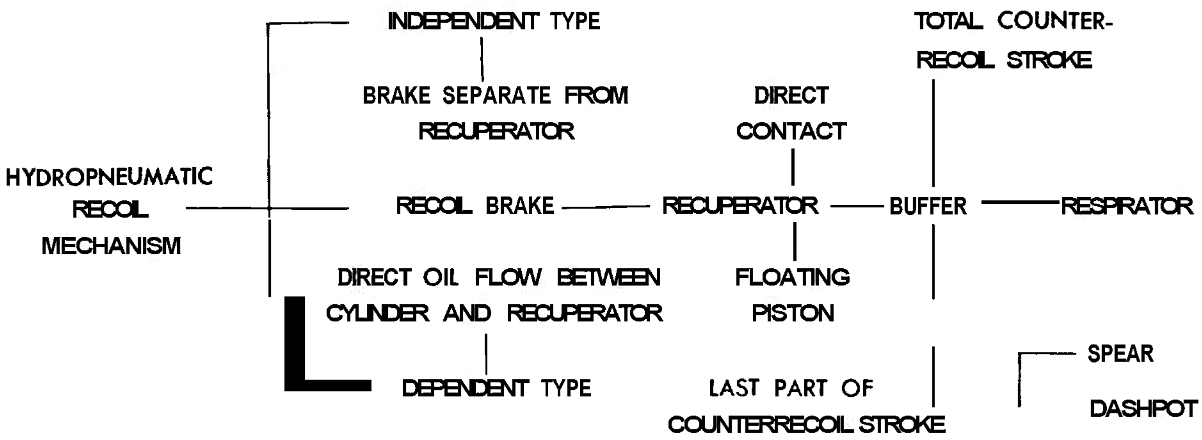


Figure 4. ***Diagram* of *Hydropneumatic Types***

The latter may be a separate unit or it may be the recoil brake components operating in reverse. The terms counterrecoil mechanism and recuperator are sometimes used as synonyms. However, to avoid confusion, the recuperator is defined here as the equipment which stores some of the recoil energy for counterrecoil, whereas the counterrecoil mechanism is defined as the unit which returns the recoiling parts to battery. It derives its energy from the recuperator. The recuperator can be of either the hydrospring type or the hydropneumatic type. The hydrospring type stores the energy required to return the gun to the battery position in a mechanical spring, or springs. The hydropneumatic type stores this energy in compressed gas. There is always some recuperator force present to hold the recoiling parts in battery at all angles of elevation. During recoil, the spring or gas is compressed further, storing the additional energy needed for counterrecoil. While in transit, gun locks, either with or without the aid of the in-battery force, hold the recoiling parts in position.

15. The buffer functions similarly to the recoil brake; it absorbs the energy of counterrecoil. There is sufficient recuperator energy to drive the recoiling parts into battery at an appreciable velocity. If this were not controlled, an impact would occur, which might cause the weapon to nose over, create structural damage, or both. The buffer is usually a dashpot type of device.

16. The components are described above as separate units, which sometimes is the case. Frequently, though, they are integrated into a single mechanism. However, whether separate or inte-

gral, all components are interdependent and function as one unit.

III. DESCRIPTION OF THE RECOIL CYCLE

A. SINGLE RECOIL SYSTEM, SEQUENCE OF OPERATION

17. As soon as the gun is fired and the projectile starts forward, propellant gas pressure accelerates the recoiling parts backward. This motion is resisted by the inertia of the recoiling parts, friction, and the recoil mechanism. The force exerted by the recoil mechanism comes from both the recoil brake and recuperator. Acceleration of the recoiling parts takes place during the time of travel of the projectile in the bore plus the time of pressure decay after the projectile leaves the muzzle.

18. The retarding force occurs over the entire recoil stroke. At the very instant of firing, only recuperator and friction forces are available. After motion begins, these forces are augmented by the hydraulic throttling force. The recoiling parts reach maximum velocity when the retarding force is equal to the propellant gas force, and then decelerate until motion ceases. Meanwhile, due to the further compression of its spring or gas, the recuperator force increases gradually, storing the energy needed for counterrecoil.

19. At the completion of the recoil stroke, the recuperator begins to return the recoiling parts into battery. Its force can never be less than that required to hold the gun in battery. Therefore, that part of the area of the force-distance curve which represents stored counterrecoil energy is somewhat predetermined.

20. A quick return to batiery is an advantage in rapid-fire guns but is undesirable in single-fire guns. Here, high forces cause problems in strength and stability. As counterrecoil velocities are sometimes limited to 2 or 3 feet per second, even more restriction of hydraulic flow than in recoil may be necessary.

21. Before the counterrecoil stroke is completed, the moving parts contact the buffer, which is designed to absorb the remaining counterrecoil energy. The moving parts should stop just as they reach the in-battery position. The recuperator force is still acting, but at its minimum value. The weapon is now in battery and ready to be loaded and fired again.

B. DOUBLE RECOIL SYSTEM, SEQUENCE OF OPERATION

22. In a double recoil system, two masses are joined to each other and to the fixed structure by recoil mechanisms. The primary recoiling parts consist of gun tube, breech housing and operating mechanism, breechblock assembly, and those parts of the recoil mechanisms not fixed to the cradle. In some instances, the recoil piston rod, the counterrecoil piston rod, and the buffer are attached to and move with the recoiling parts; in others, these items are fixed to the cradle while their associated cylinders constitute a portion of the recoiling parts. The secondary recoiling parts are composed of the cradle, top carriage, that portion of the primary recoil mechanism that is affixed to the cradle, and those parts of the secondary recoil mechanism which move with the top carriage. The secondary system action is somewhat modified by the fact that the secondary system does not begin to move until the primary is definitely under way. Its recuperator resistance and inertia are sufficient to delay the start of motion. The primary system is well on the way in counterrecoil before the secondary has fully recoiled. Ordinarily, the primary is in battery while the secondary is still in counterrecoil.

IV. PRINCIPAL TYPES OF RECOIL MECHANISMS

A. HYDROSPRING TYPE

23. The hydrospring mechanism utilizes a mechanical spring for the recuperator and a hydraulic system for recoil and buffing. Sometimes, the spring is mounted concentric to the gun tube; in other arrangements, it is concentric to the recoil mechanism; or the spring may be a separate unit. The manner of mounting depends upon the size of spring needed, the available space and its location, and on the effects of eccentric forces.

B. HYDROPNEUMATIC TYPE

24. The hydropneumatic mechanism uses compressed gas for its recuperator, usually dry nitrogen

because of its relative inertness. It may be either "independent" or "dependent," or variations of these two.

25. In the independent type, the recuperator is an entirely separate unit from the recoil brake (refer to Figures 9 and 10). The piston rods of both brake and recuperator are joined directly to the recoiling parts. As the gun recoils, hydraulic fluid or oil is forced into the compressed gas chamber. As the fluid compresses the gas still further, the pressure increases. During counterrecoil, this action is reversed. The gas and fluid may be separated by a floating piston, or they may be in direct contact. If in direct contact, sufficient oil must be provided so that gas cannot escape through the port to the counterrecoil cylinder.

26. In the dependent case, only the recoil piston rod is joined to the recoiling parts (refer to Figures 8 and 11). Fluid is forced from the recoil brake cylinder into the recuperator, where it is throttled. The recuperator normally is connected directly to the recoil brake cylinder, but a floating piston separates gas and fluid.

C. TYPES OF COUNTERRECOIL BUFFER

27. Buffers operate by means of a controlled restriction of hydraulic flow and are of two general types. One acts over a short distance during the end of the counterrecoil stroke. The other, where lower forces and finer control over velocity are needed, acts during the entire length of stroke.

V. OPERATING CHARACTERISTICS

A. GENERAL

28. All recoil mechanisms work on some combination of the same basic principles; that of providing a controlled resistance over a set distance to check the motion of the recoiling parts, then returning them to the firing position and providing a sufficient restraint to hold them in that position at maximum elevation. This resistance to recoil should be nearly constant, since, for a prescribed recoil distance, this will produce the smallest possible force on the structure (see Figure 5). The area under the force-distance curve represents energy and, clearly, a rectangular curve will yield the lowest peak force. However, a rectangular curve is not applicable at the beginning because the recoil resistance should not exceed the propellant gas force if prompt recoil motion is desired.

29. The total resistance is a combination of a hydraulic force, a spring force, which may be mechanical or from compressed gas, and friction. Whichever combination is used, it works as a unit, the parts affecting each other at; they act simultaneously. Therefore, the entire system must be analyzed as a unit.

B. THE RECOIL BRAKE

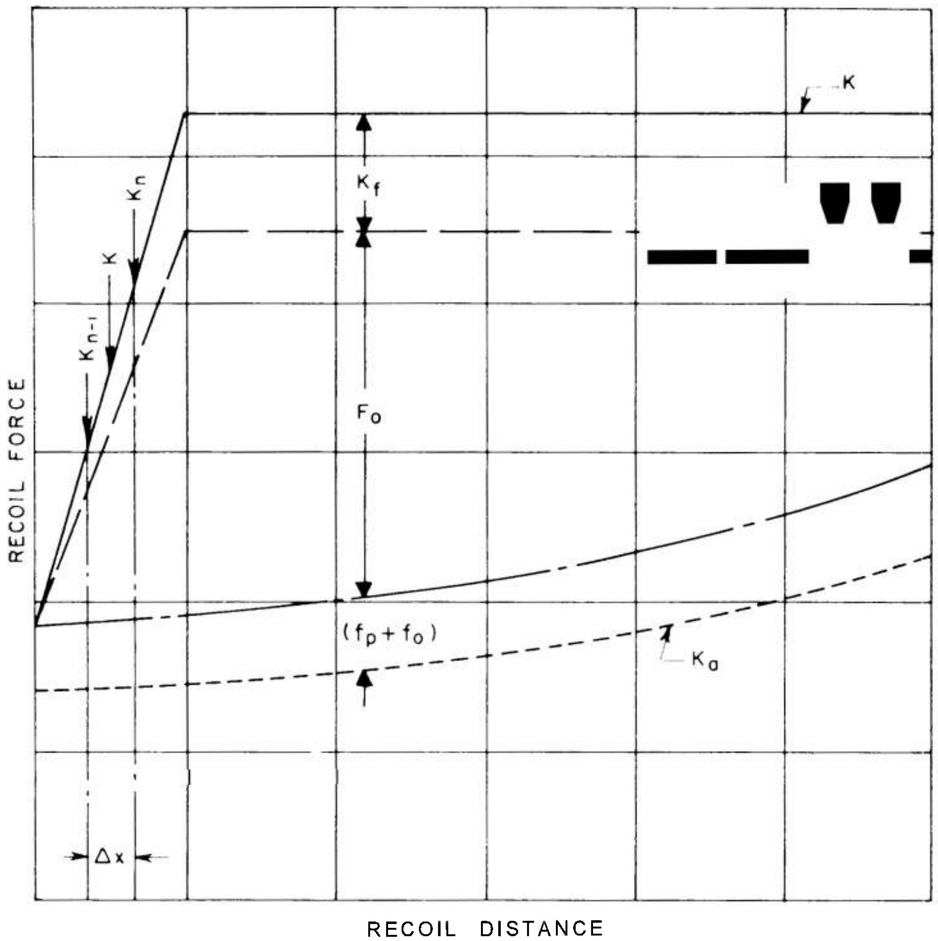
30. Since the recuperator force-distance curve is somewhat determined by its battery position force, it is necessary to adjust the hydraulic brake curve so that the total curve will be as desired (see Figure 5). After the friction and recuperator curves are known, their ordinates may be subtracted from those of the total resistance curve. The differences of the ordinates form the design brake curve.

31. The recoil brake force, at any point along the stroke, depends upon the recoil velocity and orifice area at that point. It is therefore necessary to vary the orifice area from point to point to suit the changing velocity and force. This may be done in any of several ways, or in combinations thereof. Figure 6 illustrates some of these methods.

32. A throttling bar (Fig. 6a), whose cross section varies along its length, is fastened along the cylinder in such manner that it cannot move longitudinally. This bar passes through a fixed-area orifice in the piston. As the piston moves, the net orifice area changes with corresponding change in restriction to fluid flow. The same effect may be had with a solid piston and a varying groove cut into the cylinder wall (Fig. 6b). Either method offers excellent control over the pressure curve. Two bars or grooves diametrically opposed are recommended for a balanced pressure load on the piston.

33. Another method varies orifices through the piston (Fig. 6c). A rotatable disk with matching holes is assembled to the piston. A projection of the disk is guided by a spiral groove in the cylinder wall. As the disk rotates, the orifices change in size. Again, excellent control is possible.

34. Controlled throttling may be attained by use of a perforated sleeve inside the cylinder (Fig. 6d). Holes are properly spaced so that those back of the piston provide the restriction during the first part



K = TOTAL RESISTANCE TO RECOIL
K_{n-1} = TOTAL RESISTANCE **AT** BEGINNING OF Δx
K_n = TOTAL RESISTANCE AT END OF **Ox**
K_R = NET FORCE ON RECOIL ROD
K_a = RESISTANCE OFFERED BY ELASTIC MEDIUM OF RECUPERATOR
K_f = FRICTIONAL RESISTANCE OF SLIDING SURFACES
Ax = INCREMENT OF RECOIL DISTANCE
F_a = RESISTANCE OFFERED BY THROTTLING HYDRAULIC FLUID
f_o = RESISTANCE OFFERED BY FLUID CONNECTING PORTS
f_p = FRICTIONAL RESISTANCE OF PACKINGS

Figure 5. Recoil Mechanism Force Chart

of the stroke and those in front during the last part.

35. The flow may be regulated by a spring-loaded valve, in which case the pressure is constant or nearly so (Fig. 6e).

C. HYDROSPRING MECHANISM

36. A schematic drawing of a hydrospring system is given in Figure 7, showing each element separately. In practice, it is preferable to combine two or all three elements into one housing to Secure compactness. Sometimes the spring is wrapped around the gun tube, permitting the use of a larger spring, resulting in a more compact gun assembly.

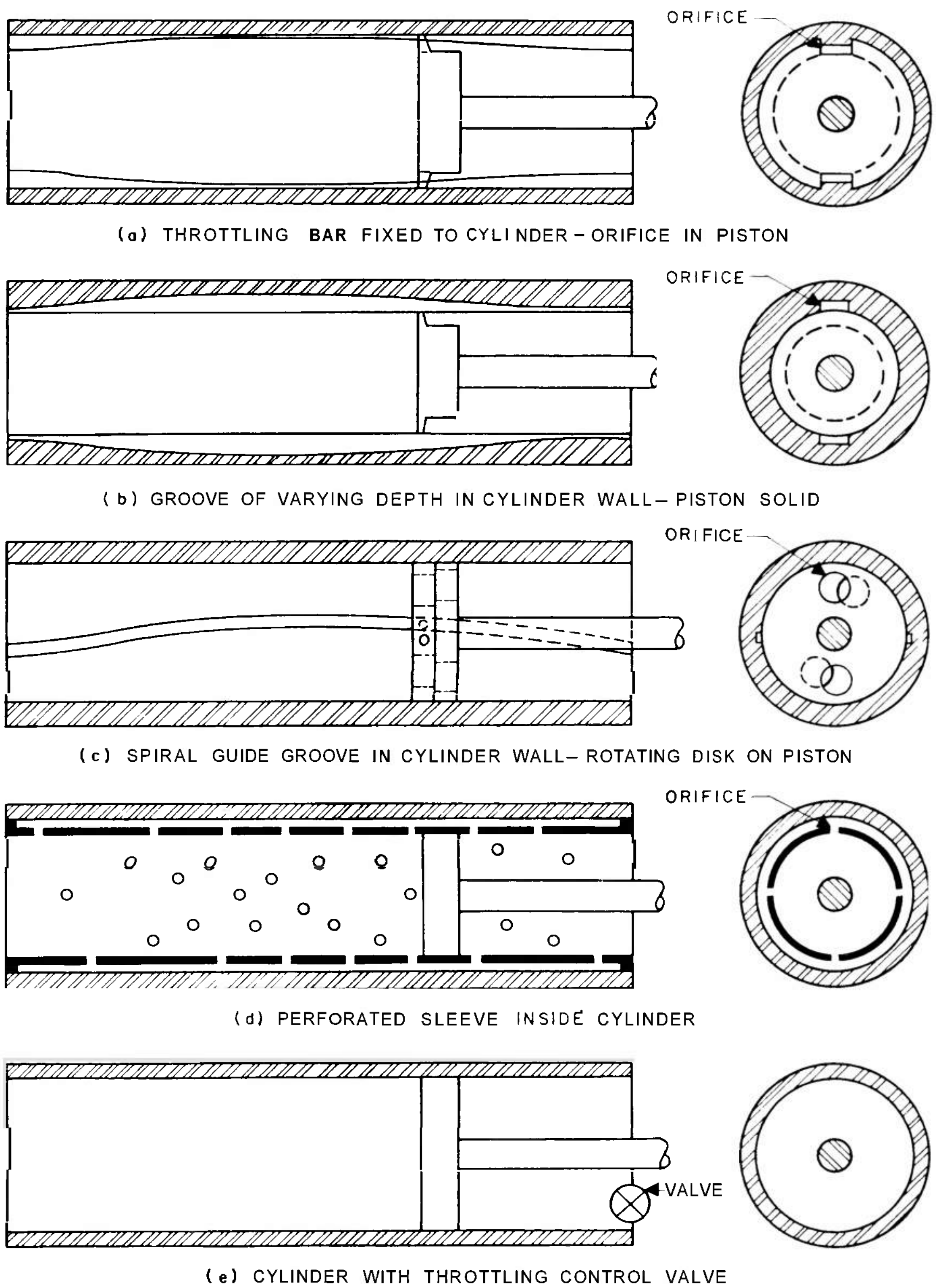


Figure 6. Methods of Orifice Area Control (Right Sections Are at Pistons)

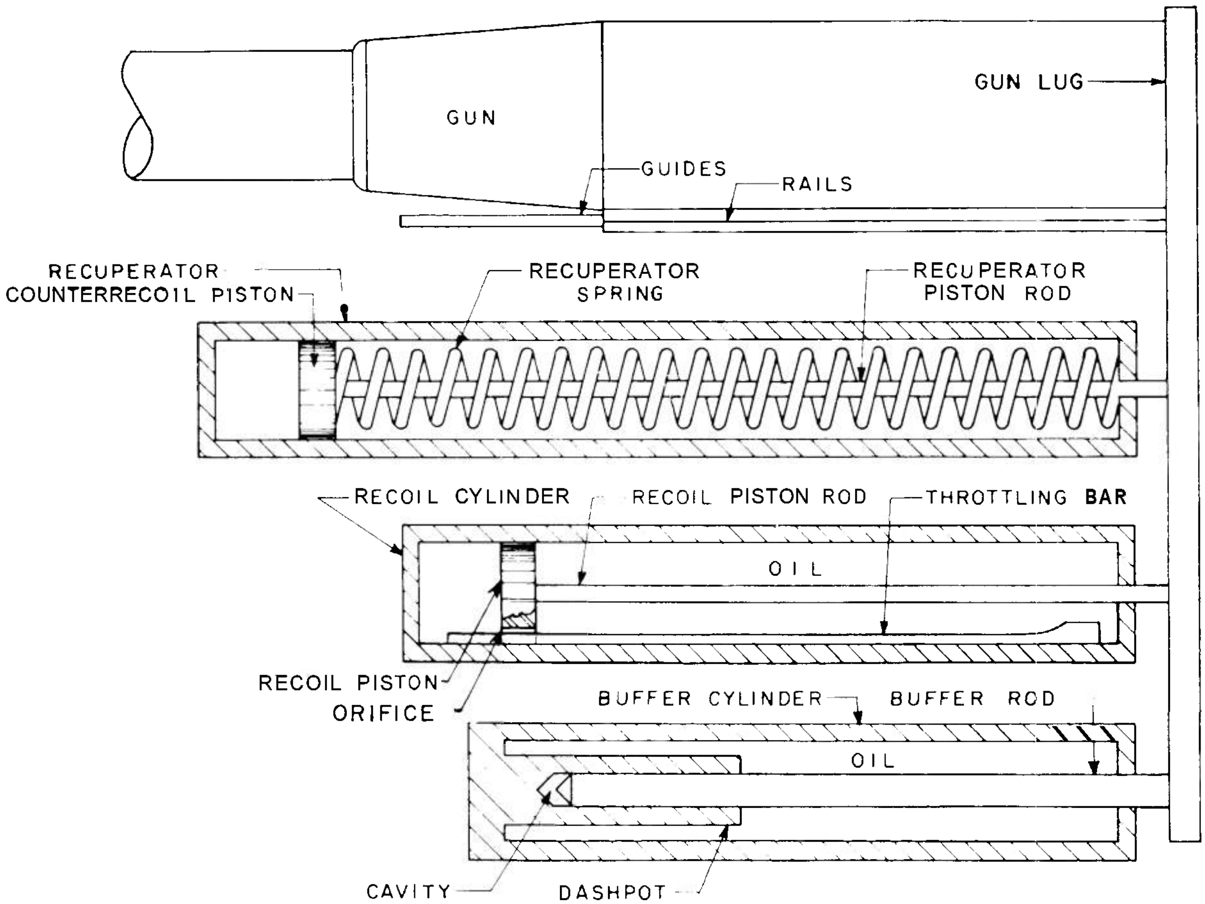


Figure 7. Hydrospring Recoil Mechanism (Schematic)

37. The advantages of a hydrospring system are:
   a. Simplicity of design.
   b. Ease of manufacture.
   c. Low initial cost.
   d. Rapidity of repair in field.
   e. Fewer seal or packing problems.

The disadvantages are:
   a. Unpredictable spring life.
   b. High replacement rate.
   c. Bulkiness.

D. HYDROPNEUMATIC MECHANISM

38. The points in favor of the hydropneumatic system are:
   a. Reliability.
   b. Durability because of little mechanical articulation.
   c. Smooth action, because gas pressure can be finely adjusted to varied conditions.
   d. The capacity to absorb small modifications of the weapon without requiring recoil system redesign.
   e. Relatively long recoi! is possible.
   f. Flexibility of design approach.
   g. Adequate warning of imminent failure.
   h. Low field maintenance.

Disadvantages are:
   a. Specialization required in manufacture, leading to high cost and some difficulty in procurement; although it lends itself to mass production, fitted or select assembly is usually necessary.
   b. Maintenance in storage requires great care to avoid deterioration and damage by internal corrosion, particularly with leather packing.
   c. Variation of gas pressure with ambient temperature affects recoil velocity and

distance. This may require some form of compensation.

d Greater number of internal cylinder walls requiring accuracy of form and high surface finish. Dents or scratches in the inside walls cause rapid failure of the packing passing over them.

e. Difficulty of maintaining high rate of fire because of effect of heat on packings and antifriction metal.

39. There is a great variety of possible designs of hydropneumatic recoil systems for the same general performance, as in the case of the hydrospring. In the following sections, from **E** to H, several existing designs are described. These are presented as some examples of past experience, but are not intended to put any limit on new ideas or resourcefulness.

E. THE PUTEAUX MECHANISM

40. The Puteaux mechanism in Figure 8 illustrates a hydropneumatic, dependent type of recoil mechanism. It consists of a hydraulic brake, directly connected by a port to the recuperator, which also houses the controls. The recoil brake is not self-sufficient, being a simple hydraulic unit which merely provides a force to retard recoil. The magnitude of the force is regulated by throttling in the recuperator. The recoil brake comprises a cylinder, piston, and piston rod. The recuperator contains a regulator, a throttling or control rod, a floating piston, and other associated parts.

41. The regulator has three cylindrical sections, the ends, or heads, being much larger in diameter than the middle section. It is fixed in position, being held in place by the closure at the breech end which is threaded to the recuperator. The front head is hollow and fits the cylinder bore. Its rear wall contains one-way valves which permit fluid passage only during recoil. The front wall is a flat plate having a central orifice. The regulator is bored axially through the rear head and middle section into the chamber of the front head to form a cylindrical housing for the control rod and a return passage for the fluid during counterrecoil. The bore may be grooved longitudinally for flow control.

42. The control rod is tapered and passes through the orifice. At its forward end, it is attached to, and centered by, a diaphragm. The breech end of the control rod is centered in its housing by a piston

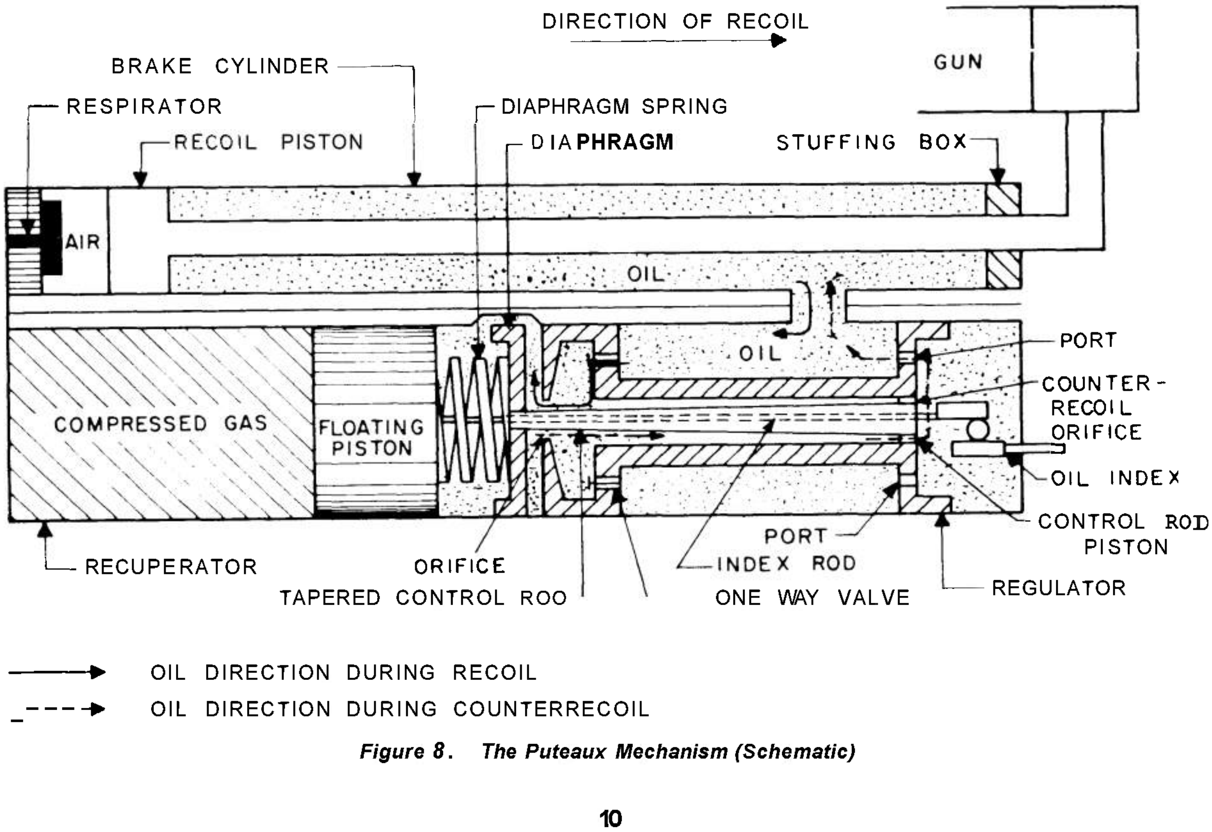


Figure 8. The Puteaux Mechanism (Schematic)

which provides restriction to fluid flow during counterrecoil when that feature is desired. The control rod is drilled through its entire length to accommodate the fluid gage (index) actuating rod.

43. The floating piston separates the gas from the liquid and also indicates the volume of fluid in the system. It lies directly in front of the diaphragm, separated from it by a compression spring. The spring insures proper positioning of the control rod just before recoil starts, as it forces the diaphragm against the orifice plate. The volume of fluid between the piston and diaphragm is the fluid reserve. A slender rod, attached to the piston, extends through the hollow throttling rod to actuate the fluid gage. Thus, the position of the piston indicates the amount of liquid in reserve.

44. The above description does not include any reference to a counterrecoil buffer because the buffer arrangement has no bearing on the identification of the Puteaux mechanism. For light artillery, where the energy to be absorbed is small, a buffer may be built into the front end of the recoil cylinder. For heavy artillery, separate buffers may be necessary to insure adequate performance.

45. During the recoil stroke, the retarding force is created by pressure built up on the rod end of the recoil piston. The piston forces fluid to flow into the regulator, where it opens the one-way valves and continues on its way through the orifice. The fluid forces the diaphragm and floating piston forward against the recuperator gas pressure. As the diaphragm moves forward, it draws the throttling rod through the orifice and, becaus'e of the proper taper of the rod, adjusts the net orifice to the desired area at each increment of stroke. The energy of recoil is principally absorbed by throttling through the orifice. Some is stored in compressing the gas and a small amount is consumed in overcoming the combined friction of all moving parts.

46. At the very start of recoil, the diaphragm is pressed against the orifice plate and no flow can occur. This n.eans that, for a brief instant, the resistance is provided only by the recuperator gas pressure and almost no control exists over the hydraulic pressure curve. As soon as an appreciable recoil velocity is attained, the orifice is regulated to produce the desired resistance.

47. As recoil ends and counterrecoil begins, the flow of fluid reverses. The gas pressure pushes the floating piston toward its original position, thus forcing the fluid back through the orifice. However, in this direction, the fluid takes a different path. The one-way valves in the regulator head are closed and the fluid is diverted to the center bore of the regulator where it flows along the control rod. To preclude excessive counterrecoil velocities, the flow is usually restricted at' the breech end of the control rod, either by slots in the control rod piston or grooves in the wall of the regulator bore. This restriction to flow is sometimes construed as buffing over the entire counterrecoil stroke, but would better be considered as a way to restrict the maximum counterrecoil velocity.

48. The Puteaux recoil mechanism has these particular advantages:

a. Compactness.
b. Light weight.
c. Provision for a fluid index.
d. One rod connection to the breech lug or to the front end of the cradle.

49. It also has these characteristic disadvantages:

a. Inadequate fluid reserve may allow the gun to fall out of battery at high elevation.
b. Control rod is not positively tied to the gun, therefore its correct position is not inherently assured.
c. Repairs require special facilities and expert mechanics.

F. THE SCHNEIDER MECHANISM

50. The Schneider mechanism (Fig. 9) illustrates a hydropneumatic, independent type recoil mechanism. It comprises a recoil cylinder, a counterrecoil cylinder, a recuperator, and a built-in buffer. There is no communicating passage between recoil cylinder and either counterrecoil cylinder or recuperator. All controls are contained in the recoil cylinder; the counterrecoil cylinder and recuperator simply store energy. The recoil and counterrecoil piston rods are separately attached to the cradle and are stationary. All three cylinders are mounted on, and move with, the recoiling parts.

51. The recoi! brake consists of three concentric components: the outside cylinder, the recoil piston and hollou piston rod, and the centra: control roo.

52. The control rod is rigidly attached to the cylinder and, therefore, also moves with ths recoiling parts. It extend- througr the orifice and into the hollou piston rod Its contour is such that it properly regulates the orifice as i passes through it

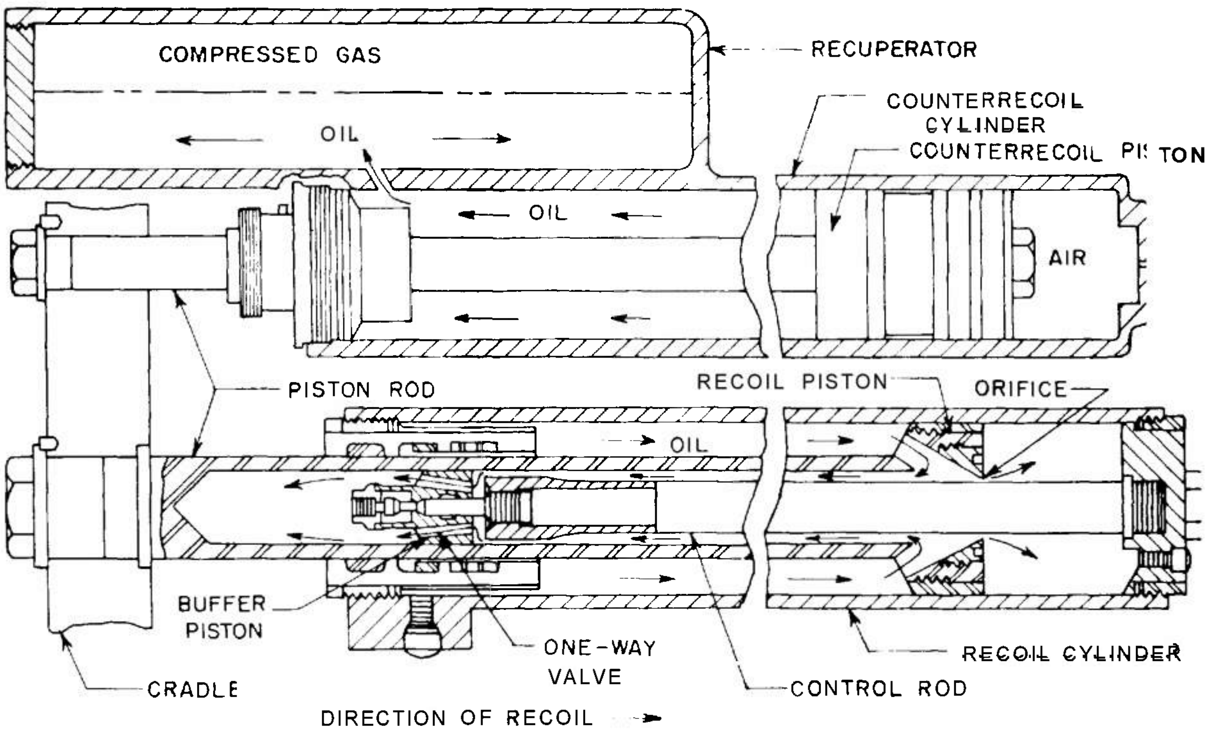


Figure 9. The Schneider Recoil Mechanism

and also permits clearance inside the hollow rod for free flow of the fluid.

53. The buffer consists of a piston at the breech end of the control rod. It slides a short distance on a spindle and thereby acts as a one-way valve. During recoil, pressure forces it away from the end of the control rod and uncovers the ports, allowing free flow to the void created by the withdrawal of the control rod. During counterrecoil the valve is forced shut and the flow must be bypassed around the buffer piston. The bore of the hollow piston rod is slightly conical for the last part of the counterrecoil stroke, which further restricts the flow and provides the necessary buffing force.

54. Fluid movement is not impeded except by gas pressure between the counterrecoil cylinder and the recuperator, as no control is attempted in these units. The recuperator is of the direct contact type with no floating piston between gas and liquid.

55. While in battery position, all compartments of the recoil brake cylinder are filled with fluid. During recoil, the control rod is withdrawn from the piston rod while the piston rod moves out of the cylinder, each motion enlarging the volume of its respective compartment. The fluid which is displaced on the pressure side of the piston is much greater in volume than the void created by the withdrawal of the control rod. Consequently, enough fluid is available to control the pressure as it is forced through the orifice.

56. The space from which the control rod has been displaced is readily filled with fluid through the one-way valve, which is open during recoil. However, when recoil ceases and counterrecoil begins, the valve closes and the fluid is forced between the buffer piston and the wall of the hollow piston rod. Buffing occurs, then, over the entire counterrecoil stroke, and the moving parts are finally brought to rest by the narrowing of the restriction described in Paragraph 53.

57. The Schneider recoil mechanism has these merits:

a. It provides adequate counterrecoil buffing.
b. No floating piston is used.
c. The control rod is secured to the gun, insuring correct position.
d. Maintenance in the field is relatively simple because assembly and disassembly are readily accomplished.

58. It has the following drawbacks:

a. The recoil and counterrecoil cylinders require separate filling.
b. No fluid index is included.

G. THE FILLOUX MECHANISM

59. The Filloux recoil mechanism (Fig. 10) is an example of the hydropneumatic, independent type, incorporating variable recoil. It comprises a recoil brake and an entirely separate counterrecoil cylinder with attached recuperator.

60. The recoil brake cylinder contains the recoil piston, a hollow piston rod, a control rod, and a buffer. It is similar in some respects to the Schneider mechanism. The piston has two ports, 180° apart, leading from the pressure side to the inside of the hollow piston rod and to the tapered throttling grooves in the control rod. In this case, the control rod does not taper but instead there are two pairs of longitudinal throttling grooves. One pair is short and regulates the fluid flow for high angles of elevation. At high elevation, stability of the weapon is not a serious problem, but ground clearance for the recoiling parts very often is. Therefore, a short recoil stroke with relatively high force may be advantageous. At low angles of elevation, the situation is reversed and a long stroke with smaller force is desirable. This latter is accomplished by bringing into play, additionally, the other pair of throttling grooves which are long. The control rod can be rotated so that only the short grooves, or both long and short, or a continuous graduation in between, are exposed to the discharge from the ports in the piston. This rotation is accomplished directly and positively from the elevating motion by a cam and gear arrangement.

61. No attempt at throttling during counterrecoil is made in the recoil cylinder, except for buffing during the final part of the stroke. Instead, a regulator valve, located in the recuperator, restricts fluid flow in counterrecoil. The recuperator is of the floating piston type, where gas and liquid are separated.

62. The operation of this recoil mechanism is characteristic of hydropneumatic systems and need not be repeated here. Finally, counterrecoil buffing is accomplished by a spear buffer located in the recoil cylinder.

63. The peculiar advantages of the Filloux mechanism are:

a. Variable recoil, to suit all angles of elevation, is provided.
b. Adequate counterrecoil buffing is provided.
c. A fluid index is provided.

64. The special disadvantages are:

a. Inadequate fluid reserve may permit the gun to fall out of battery at high elevation.
b. Repairs require special facilities and expert mechanics.
c. The recoil and counterrecoil cylinders require separate filling.

H. THE ST. CHAMOND MECHANISM

65. The St. Chamond mechanism, Figure 11, is a hydropneumatic dependent recoil mechanism, featuring variable recoil. It comprises a recoil cylinder, a recuperator with floating piston, and an independent buffer assembly. The recoil cylinder and recuperator are interconnected.

66. During recoil, the flow of fluid from the recoil cylinder to the recuperator is regulated by a spring loaded throttling valve located between them. Variable recoil is obtained by altering the limit of valve opening. The pressure which produces the retarding force is determined by the amount of valve opening and the recoil velocity.

67. During counterrecoil the one-way counterrecoil valve opens and fluid flows back to the recoil cylinder by this path. In the last part of the stroke the parts are brought to rest by an external dashpot buffer.

68. The desirable features of the St. Chamond mechanism are:

a. Variable recoil is provided at all elevations.
b. It is compact.
c. It is light in weight.

69. The undesirable features are:

a. An inadequate fluid supply may permit the gun to fall out of battery at high elevation.
b. No fluid index is provided.
c. Repairs require special facilities and expert mechanics.

I. DOUBLE RECOIL SYSTEM

70. All mechanisms heretofore discussed are

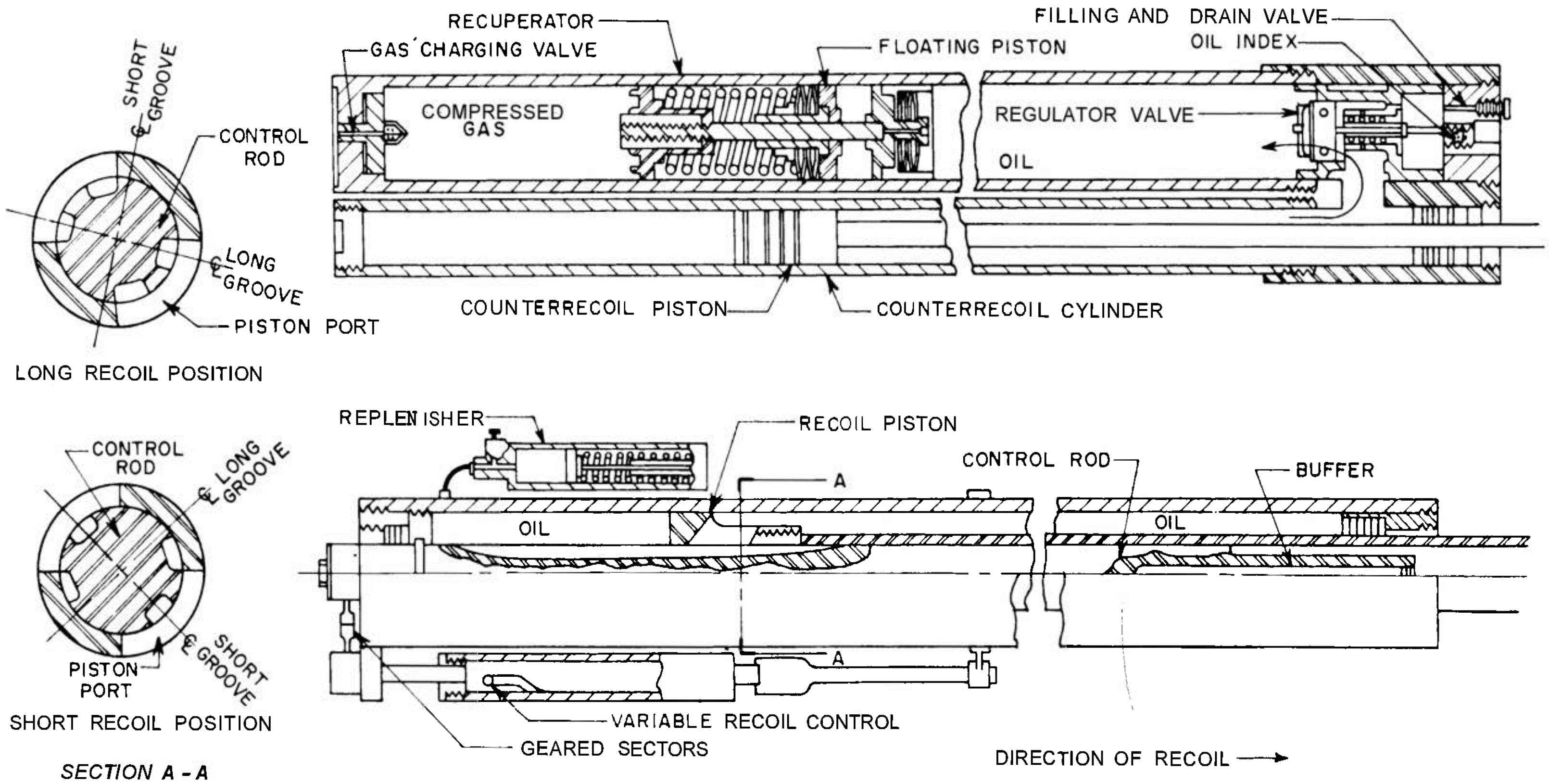


Figure 10. The Filloux Recoil Mechanism

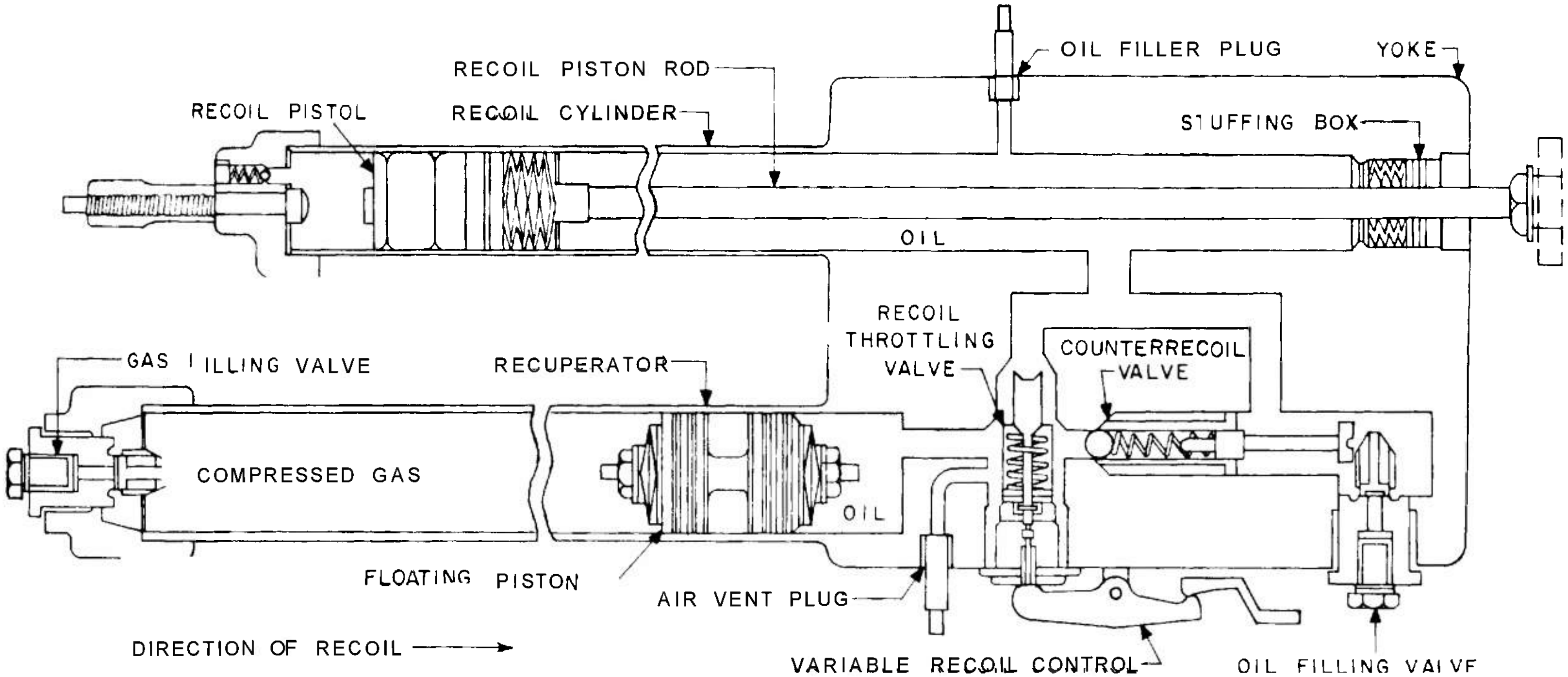


Figure 71. The St. Chamond Recoil Mechanism

single recoil systems. Sometimes, particularly with heavy weapons, it is advantageous to introduce a secondary recoil system between top and bottom carriages. Double recoil systems are discussed in detail beginning with Paragraph 147.

VI. SELECTION OF A RECOIL SYSTEM

A. GENERAL

71. Selection of the type of recoil system is governed by the characteristics of the weapon, such as size, purpose, rate of fire, and range of elevation angles. Hydrospring systems are usually limited to light artillery and short recoil distances. Hydropneumatic systems can be adapted to either light or heavy artillery. Heavy mobile weapons may require double recoil systems.

72. The options as to whether the mechanism shall be independent or dependent, variable or constant recoil stroke, floating piston or direct contact, internal or external buffer, all are strongly influenced by basic factors such as recoil force and distance, space available, stability, and ground clearance. The foregoing discussion of several designs, and their merits and shortcomings, is intended as a guide for future determinations.

B. REQUISITES OF THE RECOIL SYSTEM

73. A long recoil stroke is usually desirable to minimize recoil forces. However, the length of stroke may be limited by ground clearance, especially at high angles of elevation. At low elevations, where stability is critical, clearance is available for a longer stroke. This suggests the use of variable recoil or double recoil.

74. The recoil distance is also influenced by a high rate of fire. The recoil cycle must be completed quickly to be ready for the next round. It may be necessary to shorten the stroke and design the structure to withstand the higher forces which result. A rapid counterrecoil stroke requires a large energy storage in the recuperator. Even more critical is the large buffer force required.

75. The most important single factor having the greatest influence on the selection of the recoil system is the space available. This may dictate the use of a hydrospring mechanism instead of hydropneumatic, or the choice between dependent and independent systems, or upon the type of buffer selected.

76. Another requisite of extreme importance is ease of maintenance. Ability to be repaired in the field is a prime asset. Ruggedness and durability should be intrinsic in the design, *so* that ordinary wear and tear may be withstood for long periods of time without overhaul. When maintenance work does become necessary, it will be greatly eased by simplicity in the mechanism. A minimum number of parts facilitates disassembly and replacement. Special techniques should be eliminated so that mechanics, with only ordinary skills, can make repairs merely by following instructions. Damaged parts of one unit should be replaceable by serviceable ones from disabled weapons. The advantages of using standard and commercially available parts cannot be overemphasized. They cost less, are readily procurable, and can be made in less time than special parts.

VII. PRELIMINARY DESIGN DATA

A. VELOCITY OF FREE RECOIL

77. The original design data required for the recoil mechanism are the length of recoil and the recoil force. These items are interdependent and their values are based on the momentum of the recoiling parts and the combined momentum of projectile and propellant charge. Preliminary figures for recoil force and length of stroke are based on the velocity of free recoil, which is determined from the momentum :*

$$v_f = \frac{W_p v_m + 4700 W_g}{W_r}, \tag{2}$$

where: v_f = maximum velocity of free recoil, ft/sec,
v_m = muzzle velocity of projectile, ft/sec,
W_g = weight of propellant charge, lb,
W_p = weight of projectile, lb,
W_r = weight of recoiling parts, lb.

Free recoil defines the condition where no resistance is offered to the recoiling parts. The value of 4700 feet per second is the assumed velocity at which the propellant gases leave the muzzle. It is

* **Page 242 of reference 1. References are found at the end of this handbook.**

an empirical value based on firing tests. This formula is approximate but is sufficiently accurate for its intended application. If desired, more accurate methods are available in texts on ballistics.

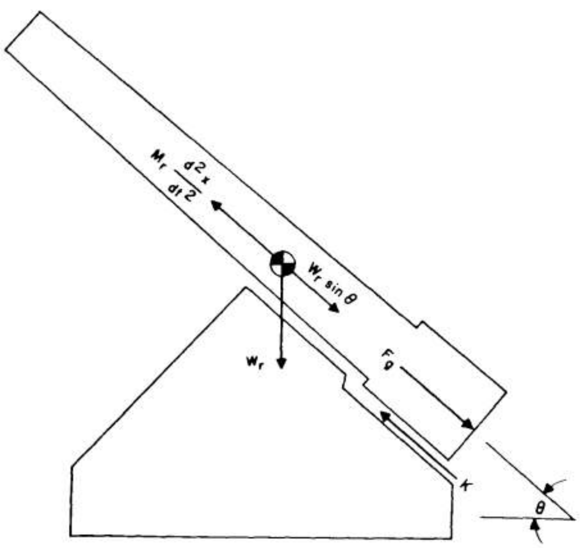
B. RECOIL FORCE

78. The general equation for the forces acting on the recoiling parts is:

$$F_g + W_r \sin\theta - K = M_r \frac{d^2x}{dt^2}, \qquad (3)$$

where: F_g = propellant gas force,
K = total resistance to recoil,
M_r = mass of recoiling parts,
W_r = weight of recoiling parts,
t = time of recoil,
x = length of recoil at time t,
θ = angle of elevation.

Figure **12** illustrates this force system. The expression $M_r \frac{d^2x}{dt^2}$ according to D'Alembert's principle, represents the inertia force. The propellant gas force soon becomes zero and, since K always opposes recoil and is greater than the weight component W_r sine, $M_r \frac{d^2x}{dt^2}$ changes directions. The exact value of K in Equation (3) is eventually determined by trial through a step-by-step integration (see Chapter IX, Part C) but, first, a reasonably close value must be found to put it in the working range. The energy of free recoil and the length of recoil are used for this purpose.



F_g = PROPELLANT GAS FORCE
K = TOTAL RESISTANCE TO RECOIL
M_r = MASS OF RECOILING PARTS
W_r = WEIGHT OF RECOILING PARTS
8 = ANGLE OF ELEVATION

Figure 72. Recoil Force System

$$E_r = \tfrac{1}{2} M_r v_f^2, \qquad (3a)$$

where: E_r = kinetic energy of free recoil,
M_r = mass of the recoiling parts,
v_f = maximum velocity of free recoil (Eq. 2).

This energy, divided by the length of recoil, gives the average resistance necessary to stop the moving mass. To this resistance must be added the static force component of the weight of recoiling parts (W_r sine). The first approximation of the total resistance to recoil is:

$$K = \frac{E_r}{L} + W_r \text{ sine}, \qquad (3b)$$

where: L = length of recoil,
e = angle of elevation.

Although the recoil rod force, K_R, is reduced somewhat from K by the frictional forces of the cradle, K, as defined in Equation (3), will be used without modification as a preliminary design load for the recoil mechanism. The error involved will be small and conservative. For final design, these frictional forces may be considered.

C. IN-BATTERY FORCE

79. The minimum force required of the recuperator is that which is sufficient to hold the recoiling parts in battery plus the force necessary to overcome all frictional resistance. In equation form:

$$F_1 = \lambda(W_r \text{ sine} + \mu W_r \cos\theta + f_p), \qquad (4)$$

where: F_1 = static force of the recuperator in battery,
f_p = total frictional resistance of packing,
W_r = weight of recoiling parts,
e = angle of elevation,
λ = in-battery sustaining factor,
μ = coefficient of friction.

Present design practice assigns the following values:

$f_p = 0.30 F_1$,

$\lambda \;\; = 1.30,$*
$\mu \;\; = 0.30.$*

The value $0.30\,F_1$ is used as a preliminary estimate. After the sizes of the recoil cylinder and the recuperator have been established, a more correct figure for the packing friction can be obtained by the method outlined in Paragraph 90. The in-battery sustaining factor, λ, can vary. Other factors have been used such as $\lambda = 1.15$ for the 175mm gun.†

D. VELOCITY OF COUNTERRECOIL

80. The velocity of counterrecoil is usually critical because of its influence on the stability of the weapon. High velocities may mean large buffing forces sufficient to nose over the gun. Hence, low counterrecoil velocities must be maintained, except for rapid fire weapons. For large guns the total time of the firing cycle is relatively long: Velocities as low as two feet per second are common.

81. The energy available to return the gun to battery is stored in the recuperator. In a hydro-pneumatic recuperator with polytropic expansion of the gas, the available energy is:

$$W_{cr} = \frac{P_1V_1 - P_2V_2}{n-1}, \tag{5}$$

where: P_1 = gas pressure at end of recoil,
P_2 = gas pressure when buffers are contacted,
V_1 = gas volume at end of recoil,
V_2 = gas volume when buffers are contacted,
n = polytropic exponent defined in the relationship PV^n = constant.‡

If the recuperator is a mechanical spring, the available energy is:

$$W_{cr} = \tfrac{1}{2}(F_2 + F_{s_2})L_{cr} \tag{6}$$

where: F_2 = recuperator force at end of recoil,
F_{s_2} = spring force when buffers are contacted,
L_{cr} = length of counterrecoil stroke to point where buffers are contacted.

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* **Part 10 of Reference 2.**
† **Page 32 of Reference 3.**
‡ **Past practice has used the value of n = 1.3. Page 385 of Reference 4.**

Procedure for designing mechanical spring and compressed gas recuperators to obtain the desired characteristics are discussed in Chapter VIII.

82. Some of the recuperator energy must be used to overcome the static resistance of the system. The work expended by this resistance is:

$$W_s = F_1'L_{cr}, \tag{7}$$

where F_1' is the static resistance to counterrecoil, expressed as:

$$F_1' = W_r\sin\theta + \mu W_r\cos\theta + f_p \tag{7a}$$

If no resistance is offered to fluid flow, the kinetic energy of the counterrecoiling mass when it first contacts the buffers is:

$$E_{cr} = W_{cr} - W_s \tag{8}$$

and the approximate maximum counterrecoil velocity is:

$$v_r = \sqrt{\frac{2E_{cr}}{M_r}}, \tag{9}$$

where: M_r = mass of recoiling parts.

Although this method is not final, it indicates what must be done to meet the specified counter-recoil velocity. If it is too low, more of the recoil energy must be stored. If it is too high, the fluid flow must be restricted. Usually the required orifice area for return is smaller than that for recoil. The procedure for obtaining the correct velocity follows a step-by-step integration involving a trial and error approach. This procedure is discussed in detail in Chapter IX, Part E.

E. BUFFER FORCE

83. The buffer force is based on the kinetic energy of the counterrecoiling parts, the static resistance, and the recuperator pressure. That part of the force which stems from recuperator pressure does not affect weapon stability. If an approximately constant retarding force is desired, the energy component of the resistance is:

$$F_b' = \frac{E_{cr}}{S_b}, \tag{9a}$$

where: S_b = length of buffer stroke.

The required buffer force is:

$$F_b = F_b' - F_1' + F_r \tag{10}$$

where: F_1' = the static resistance, (**Eq.** 7a)
F_r = force on counterrecoil piston

and its pressure is:

$$P_b = \frac{F_b}{A_b}, \tag{11}$$

where: A_b = effective area of buffer piston.

84. The recoil, counterrecoil, and buffer forces having been approximated as set forth in Paragraphs 77 to 83, it is now possible to turn to the design of recoil mechanism components.

VIII. DESIGN OF RECOIL MECHANISM COMPONENTS

A. SUGGESTED MATERIALS

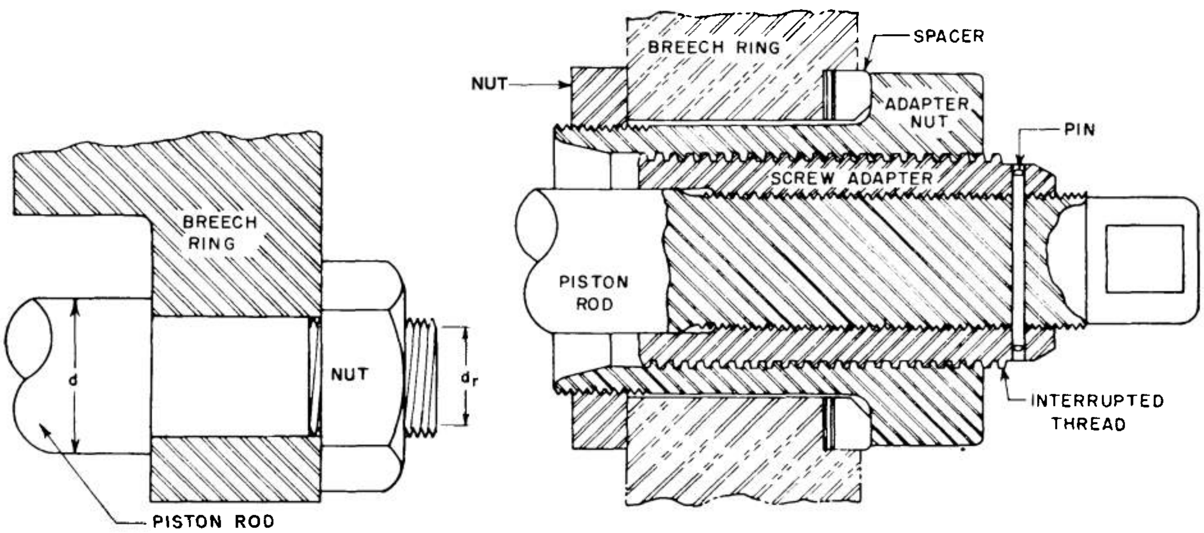
85. Except for bearings, bushings, and packings, all components of the recoil mechanism are made of steel. Bearings are of antifriction metal, comprised chiefly of tin, antimony, and copper, similar to Specification QQ-T-390, Grade 2. Bushings are of bronze. Packing materials are discussed later in Part D. High strength steels should be avoided unless there is a positive advantage in their use. Moderate yield strengths of about 70,000 psi are recommended. The factor of safety, based on the yield strength, should not be less than 1.5. Usage will often influence the choice of material. Where high strength-to-weight ratios are needed, high tensile steel is indicated. But, where rigidity is essential, a lower strength steel is more economical and will serve as well.

B. RECOIL PISTON ROD

86. The recoil piston rod is a tension member. One end is attached to the recoil piston, the other, usually, to the breech ring. It may be threaded to the piston (see Fig. 14), or it may be integral with it. Figure 13 shows two methods of attaching the rod to the breech ring. Figure 13(a) illustrates the conventional method while Figure 13(b) illustrates a quick detachable method which has a screw adapter with an interrupted thread. Large diameter, hence rigid, rods are an asset because distortion prone, slender rods may soon damage packings to the extent where leakage is inevitable. The strength of the rod is readily found because it is a simple tension member. However, there are some abrupt changes in diameter, especially at the threads, which introduce stress concentration. These can be taken into account with factors that are found in available references.* For example, the maximum stress in the rod of Figure 13(b) is the tensile stress of the root area of the thread, increased by the concentration factor, ***k*:**

$$\sigma_t = k\frac{K_R}{A_r}, \tag{12}$$

* **Table XVII of Reference 5.**



(a) CONVENTIONAL ATTACHMENT *(b) QUICK DETACHABLE ASSEMBLY*

Figure 13. **Rod-Breech Ring Attachments**

where: σ_t = rod tensile stress,
K_R = recoil rod force,
A_r = thread root area,
k = stress concentration factor depending on the ratios $\frac{r}{d_r}$ and $\frac{d}{d_r}$ of Type 11 of Reference **5**.

C. RECOIL PISTON

87. The thickness of the piston is controlled by the space needed for the packing and so is greater than would normally be required for strength. The net piston area, and hence the diameter, is governed by the maximum fluid pressure. This pressure is limited by the ability of the packing to seal. Past practice set the limit at 4500 psi* but because of better packing materials, higher pressures are now permissible.

* Part 17 of Reference 2.

$$A = \frac{K_R}{P_m}, \tag{13}$$

where: A = effective piston area,
P_m = maximum fluid pressure.

The piston diameter **is** determined from :

$$A = \frac{\pi}{4}(D_1^2 - d^2), \tag{13a}$$

where: D_1 = piston diameter, also **ID** of cylinder,
d = piston rod diameter.

All diameters are selected to the nearest nominal size in order to conform to commercially available stock and materials.

D. PACKINGS

88. Figure 14 depicts a typical packing assembly. The packing illustrated is proportioned after those already in use, so that previous experience **is** an

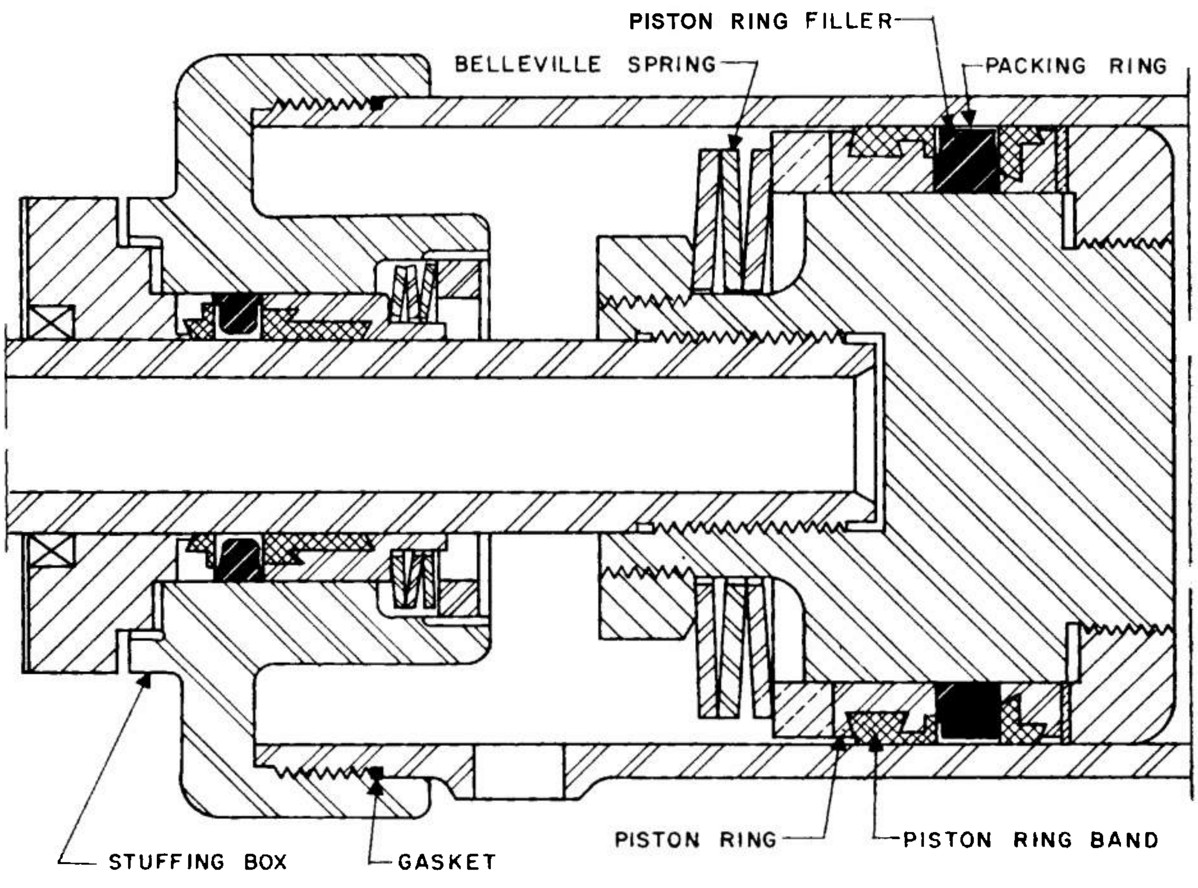


Figure 14. Typical Packing Assembly

important factor in its design. Packings prevent leakage past moving parts, such as pistons and piston rods. The packings are forced firmly against the moving surfaces both by the pressure of the fluid itself and by springs. Because of the nearly hydrostatic behavior of the packing material, axial pressure is nearly equal to the radial pressure which is necessary for sealing. The ratio of the radial pressure to the applied axial pressure is a property of the packing material and is called the "pressure factor." It is somewhat analogous to Poisson's ratio. To insure positive sealing, the radial pressure must be greater than the maximum fluid pressure. This is possible because of the force applied by the springs. This ratio is known as the "leakage factor" and is usually at least 1.0. Sometimes a small amount of leakage is desirable for lubrication; at such times the leakage factor is less than 1.0.*

The radial pressure exerted by the packing expressed in terms of fluid pressure is:

$$P_R = K_p(P_s + P_m) = \nu P_m, \tag{14}$$

where:
K_p = pressure factor,?
ν = leakage factor,
P_m = maximum fluid pressure,
P_s = axial pressure in packing produced by spring.

Solving for P_s:

$$P_s = \frac{\nu - K_p}{K_p} P_m. \tag{14a}$$

89. The packing filler has been made of rubber. **The** liner or packing ring in contact with the cylinder has been made of leather. Silver rings, having a right angled cross section, were commonly used to confine the corner of the leather packing to prevent it from extruding between piston ring and cylinder. Recent developments have shown alternative materials, such as polytetrafluoroethylene (Teflon) for leather and an aluminum alloy for silver, to be satisfactory.

90. The spring pressure being known, the packing friction can now be found. The total axial pressure on the packing equals the spring pressure plus the fluid pressure:

$$P_a = P_s + P_\theta, \text{ axial pressure on packing;} \tag{15}$$

$$P_R = K_p P_a, \text{ radial pressure on packing,} \tag{15a}$$

$A_1 = \pi D_1 b$, contact area on cylinder wall,

where:
D_1 = ID of cylinder,
b = width of packing,
P_θ = fluid pressures on packing at any recoil position,

$$F_\theta = A_1 P_R, \text{ radial force of packing on cylinder} \tag{15b}$$

$$f_p' = \mu F_\theta, \text{ frictional force of a packing assembly,} \tag{15c}$$

where, for leather, μ = 0.05 and for silver, μ = 0.09.* The force f_p' is the general expression for the packing frictional forces: f_r in the recuperator and f_c in the recoil brake cylinder. If the recoil mechanism is an independent type, the total frictional force due to packings is:

$$f_p = f_c + f_r. \tag{15d}$$

But, if it is a dependent type, then:

$$f_p = f_c + \frac{A}{A_R} f_r$$

where:
A = the effective area of the recoil piston,
A_R = the area of the floating piston.

E. BELLEVILLE SPRINGS

91. Springs are used to augment the packing pressure. Belleville springs are selected because they require very little space and provide large loads at small deflections. The design of Belleville springs is outlined in most spring manufacturers' handbooks,† These springs are extremely sensitive to small changes in dimensions, and manufacturing variations can produce large load differences. Therefore, each spring assembly should be tested for load before installation. The required spring load is:

$$F_s = \pi(r_2^2 - r_1^2)P_s, \tag{16}$$

where: P_s = packing pressure required of spring

* This value is taken from the design specifications. For example, specifications for the recoil mechanism of the 175mm Gun Carriage, T76, call for a leakage factor of 1.3 for the floating piston and 0.88 for the stuffing box. Lubrication was desired for the latter.

† For rubber filler, K_p = 0.73. Page 12 of Reference 6.

* Page 12 of Reference 6.

† Reference 7.

load (determined from Equation 14a),
r_1 = inside radius of piston ring,
r_2 = outside radius of piston ring.

F. RECOIL CYLINDER

92. The inside diameter of the cylinder is determined by the piston size (Paragraph **87**). The outside diameter depends upon the pressure due to fluid or packing, and the yield strength of the material. In this and other pressure vessels of the recoil mechanism, rigidity is of more concern than high strength to weight ratio. Thick walls minimize the possibility of local damage and prevent excessive dilation which makes the seals less effective. Therefore, steels of moderate yield strength are recommended. To detect defects, all pressure vessels should be subjected to hydrostatic proof tests, usually at one and one-half times the working pressure. That is:

$$P_p = 1.5 P_f,$$

where: P_f = maximum fluid pressure,
P_p = proof pressure.

The proof pressure, being higher, becomes the basis for design. The maximum shear stress, according to the maximum shear stress theory of Tresca and Saint Venant,* is given by:

$$\tau_m = \frac{\sigma_t - \sigma_r}{2} = \frac{\sigma_y}{2}, \qquad (17)$$

or

$$\sigma_t - \sigma_r = \sigma_y, \qquad (17a)$$

where: σ_r = radial stress,
σ_t = hoop stress of cylinder,
σ_y = yield strength,
τ_m = maximum shear stress.

A vessel should not be stressed beyond the yield strength at proof pressure; thus, to be slightly conservative, a factor of safety of 1.5 is introduced. Equation (17a) now becomes:

$$\sigma_t - \sigma_r = \frac{\sigma_y}{S_f}, \qquad (17b)$$

where S_f = 1.5 = the factor of safety.
If we arbitrarily select σ_y to be twice σ_t, then:

$$\sigma_t = \tfrac{1}{2}\sigma_y. \qquad (17c)$$

* **Page 39 of Reference 8.**

It is also known that:

$$\sigma_r = -P_p. \qquad (17d)$$

Substituting in Equation (17b) the values of 1.5 for S_f, σ_t from Equation (17c), and σ_r from Equation (17d), we have:

$$\tfrac{1}{2}\sigma_y - (-P_p) = \tfrac{2}{3}\sigma_y, \qquad (17e)$$

or, collecting terms,

$$P_p = \frac{\sigma_y}{6}. \qquad (18)$$

Thus, for the selection of σ_t in Equation (17c), we find that the proof pressure, P_p, should never exceed one-sixth of the yield strength **of** the material, or, conversely, the yield strength should be six times the proof pressure. This establishes a minimum yield strength. **A** higher yield strength may be specified if a larger factor of safety is desired. On occasion, it may be desired to **use** a material having a higher yield strength than $\sigma_y = 6P_p$. **To** maintain the factor of safety of 1.5, the hoop stress should be:

$$\sigma_t = \frac{\sigma_y}{1.5} - P_p, \qquad (18a)$$

where P_p is the same proof pressure used in **Equation (18).**

93. When a maximum allowable working pressure of **4500** psi is used,

$$P_p = 1.5P_m = 6750 \text{ psi},$$

and, for the conditions leading to Equation (18), the minimum yield strength **is:**

$$\sigma_y = 6P_p = 40{,}500 \text{ psi}.$$

The dimensions of the pressure vessel may be found using the equation of Lamé:

$$\sigma_t = P_p\left(\frac{D_2^2 + D_1^2}{D_2^2 - D_1^2}\right) \qquad (18b)$$

where: D_1 = inside diameter of cylinder,
D_2 = outside diameter of cylinder.

Solving for D_2 gives:

$$D_2 = D_1\sqrt{\frac{\sigma_t + P_p}{\sigma_t - P_p}}. \qquad (18c)$$

Using the known values of σ_t and P_p and considering that D_1 is determined by the diameter of the piston, the minimum outside diameter may be found. For example, the conditions leading to Equation (18), $\sigma_t = \tfrac{1}{2}\sigma_y$ and $P_p = \tfrac{1}{6}\sigma_y$, when substituted in Equation (18c) give:

$$D_2 = \sqrt{2D_1^2} = 1.41D_1. \qquad (18d)$$

After the outside diameter becomes known, the various stresses may be computed by conventional methods for high pressure vessels.

G. RECUPERATOR, HYDROPNEUMATIC

94. The recuperator is the energy reservoir of the recoil system. Its gas pressure holds the gun in battery. During recoil, the gas is compressed further to store the energy required for counterrecoil. There are, then, both maximum and minimum operating pressures to be considered.

95. The area of the counterrecoil piston and eventually the size of the recuperator of an independent system (Paragraph 25) is determined by the in-battery force, Equation (4), and the minimum gas pressure P_0:

$$A_{cr} = \frac{F_1}{P_0}, \tag{19}$$

For a dependent system, the piston area was determined by Equation (13). The minimum pressure is also the charging pressure; consequently, it is dependent upon the source of supply, usually high-pressure bottled gas. As the source is, in effect, exhausted when its pressure becomes equal to that of the recuperator, the initial difference in pressures should be large. For efficient use of bottled gas at 2000 or 2500 psi, a recuperator minimum pressure of about 800 psi is recommended. The maximum pressure at the end of recoil is selected at about twice the in-battery pressure but not to exceed a pressure which would induce leakage past the packings. It must, however, be adequate to assure prompt counterrecoil.

96. The maximum and minimum pressures having been established the recuperator size can be found as follows:

$P_1 \approx 2P_0$, pressure at end of recoil,
$\Delta V = LA_{cr}$, change of gas volume during recoil,
V_1 = gas volume at end of recoil,
$V_0 = V_1 + \Delta V$, gas volume, in battery.

From the equation of polytropic expansion:

$$\frac{P_1}{P_0} = \left(\frac{V_0}{V_1}\right)^n, \tag{20}$$

where: $n = 1.3$.*

The quantities P_0, P_1 and ΔV being known, the values of V_1 and V_0 are readily determined.

* Page 385 of Reference 4.

H. RECUPERATOR, SPRING TYPE

97. The upper and lower limits of the spring forces are set up similarly to those of the hydropneumatic recuperator. The in-battery force is obtained by Equation (4). Then:

$F_2 \approx 2F_1$,
where: F_1 = in-battery force,
F_2 = force at end of recoil,

$$K_s = \frac{F_2 - F_1}{L}, \text{ spring rate.} \tag{21}$$

The applied loads and spring rate being known and the available space ascertained, the remaining parameters necessary for design of the spring (or springs) are the torsional modulus of rigidity and allowable torsional stress (see Paragraph 144). These may be obtained from standard references.* A spring having a slenderness ratio (free length divided by mean coil diameter) greater than four may tend to buckle, as does a column. Curves which indicate when buckling may be expected are available.* The ends must be restrained from lateral movement or buckling may occur at lengths less than shown on the curve. The equations for stress and deflection may be found in textbooks or handbooks of spring manufacturers.† Because of the interdependence of the variables, several trials may be necessary before a satisfactory spring is evolved. Also, if a single spring cannot be worked out, it is possible that multiple springs will satisfy the requirements.

I. COUNTERRECOIL BUFFER

98. Counterrecoil buffers may be hydraulic or pneumatic. The hydraulic type is a form of dashpot and may be an external, separate unit or an integral part of the interior of the recoil mechanism. In either case the stroke is so selected that the buffer force will not unduly disturb the stability of the weapon. An external buffer is illustrated in Figure 15. As the counterrecoiling parts contact the piston rod head, hydraulic fluid is forced through a confined space around the piston to generate the buffing force. At the same time the spring is compressed. During the next recoil stroke, the spring forces the piston to return to its

* Reference 9.
† Reference 10.

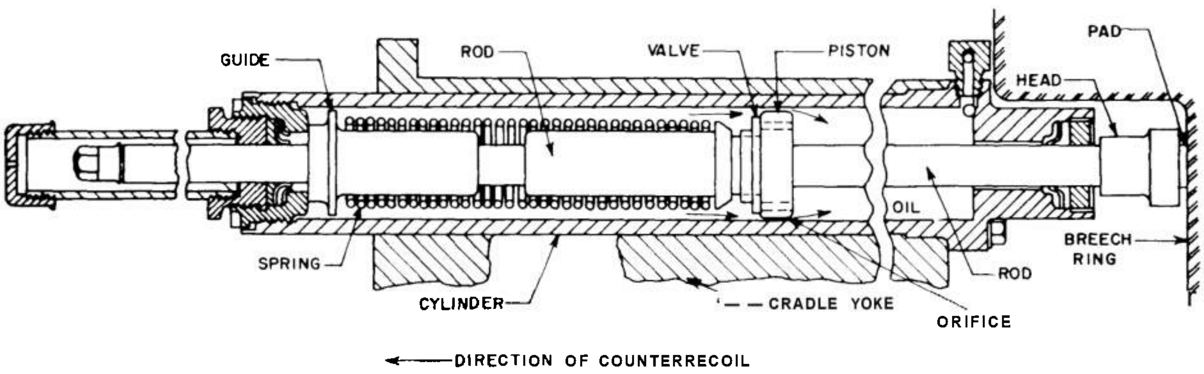


Figure 75. External Buffer

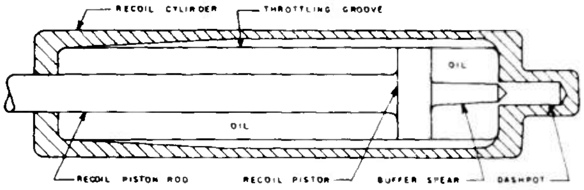


Figure 76. Internal Buffer

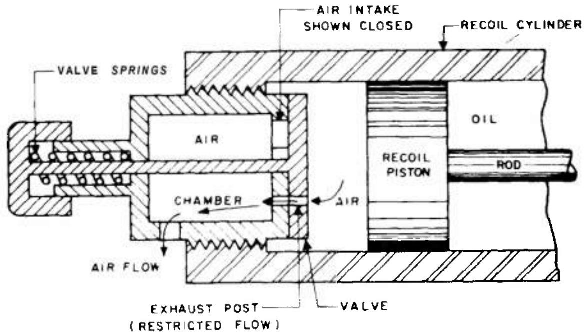


Figure 77. Respirator

buffing position, the one-way valve being open to facilitate this movement. Figure 16 shows an internal buffer, consisting of a dashpot and buffer spear. The spear is fixed to the recoil piston. During the first part of the counterrecoil stroke the dashpot is filled with fluid. As the spear enters the dashpot, this fluid is forced out through the clearance, and the restriction of flow creates the force needed for buffing.

99. Another hydraulic buffer, first mentioned in Chapter IV, Part *C,* controls the velocity along the full counterrecoil stroke. Not a true buffer in the sense that it does not absorb the shock of a moving mass, it merely is an orifice (Figures 8 and 20) which provides a controlled restriction in the path of the returning fluid that precludes the counterrecoiling parts from exceeding a desired maximum velocity. The mechanics are discussed in Chapter IX, Part E.

100. A pneumatic buffer is known as a respirator (*see* Fig. 17). It consists of an air chamber at the end of either recoil or counterrecoil cylinder, depending upon the type of recoil mechanism. As the operating piston is withdrawn during recoil, the check valve is open and atmospheric air flows freely into the chamber to fill the space vacated. When counterrecoil begins, the one way check valve closes, trapping the air in the chamber. A small, hand-adjustable orifice remains open, permitting the air to escape at a controlled rate, thus regulating the pressure which stops the counterrecoiling parts. A tendency is present for the inner cylinder walls to rust from exposure to the atmosphere. Proper lubrication will reduce the tendency.

J. FLOATING PISTON

101. The floating piston separates the liquid from the gas within the recuperator. It has no piston rod and moves freely as the gas changes in volume. During recoil, hydraulic fluid forces the piston to compress the. gas; then during counterrecoil, the gas pressure forces the piston and fluid to return to their original positions. The piston has two heads joined integrally by a **shank (Fig. 18).** Each head contains a packing as described in Paragraph 88. The void around the shank is packed with grease for lubrication. In some applications an index rod is attached to the fluid side of the piston to gage the amount of fluid reserve. The floating piston must move smoothly and, con-

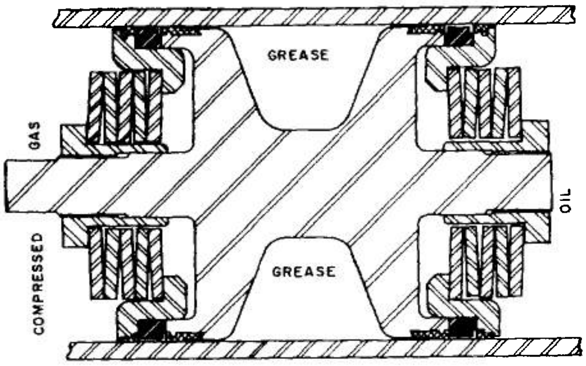


Figure 78. Floating Piston

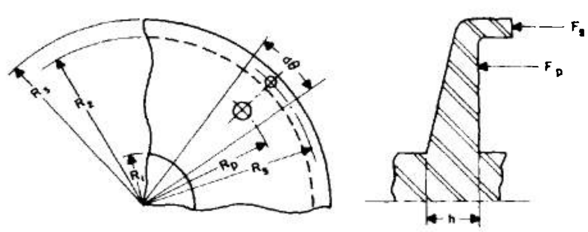


Figure 79. Piston Flange loading Diagram

sequently, must be long enough to prevent binding; in practice, usually one and one-third diameters.

102. The strength of the flange is determined, conservatively, by treating a sector cut out by the angle, $d\theta$, as a cantilever beam acting independently of adjacent sectors (see Figure 19). We obtain an expression for the center of gravity of the pressure area of a very small circular element described by R_1, R_3, and $d\theta$. The pressure load may be considered as concentrated here. For the remainder of these calculations, the angle, $d\theta$, need not be assigned a value because, as will be seen, it divides out in the summation of equations. The total spring load, F_s, is concentrated at R_s, midway between R_2 and R_3. The bending moment from the pressure load, F_p, is:

$$M_p = F_p (R_p - R_1)$$
$$= p\frac{d\theta}{2}(R_3^2 - R_1^2)(R_p - R_1) = k_p\, d\theta, \quad (22)$$

where: R_p is the radius to the center of pressure,

$$R_p = \frac{2 \sin \frac{d\theta}{2} (R_3^3 - R_1^3)}{3 \frac{d\theta}{2} (R_3^2 - R_1^2)} \quad (22a)$$

The moment due to the spring load, F_s, is:

$$M_s = \frac{F_s}{2\pi} d\theta (R_s - R_1) = k_s\, d\theta, \quad (22b)$$

where: R_s is the distance to the center of the spring load.

The total bending moment is:

$$M = (k_p + k_s)\, d\theta. \quad (22c)$$

The section modulus at the shank is:

$$Z = \tfrac{1}{6}bh^2 = \tfrac{1}{6}R_1 d\theta h^2 = k_z\, d\theta. \quad (22d)$$

The bending stress is:

$$\sigma = \frac{M}{Z} = \frac{k_p + k_s}{k_z}. \quad (22e)$$

K. REGULATOR

103. The regulator, Figure 20, when used, is housed in the recuperator cylinder as, for example, in the Puteaux mechanism, Figure 8. Its components are discussed in Chapter V, Part E. Since it provides the means of regulating pressures during recoil and counterrecoil, it must control the flow of hydraulic fluid in either direction. The design is essentially one of configuration; reasonable proportions generally insuring adequate strength. The diameter should be large enough to provide the flow channels. Pressures are controlled by restricting the flow with **orifices.** The orifices should exercise most of the control; the channels, including open valves, being relatively free of restriction. To realize this control, the channels must be much greater in cross section than the orifice area; a ratio of 5:1 seems reasonable. According to Equation (36), when the combined area of all parallel channels leading toward an orifice is at least five times the largest orifice area, the rise in pressure accountable to the channels will not exceed four percent of that due to the orifice.

L. RECOIL THROTTLING VALVE

104. A spring loaded throttling valve is used in some dependent types of mechanism for control of the hydraulic resistance to flow from the recoil cylinder into the recuperator. It is adaptable to variable recoil and is a feature of the St. Chamond mechanism (Paragraphs 65 to 69). It usually has two springs, as shown in Figure **21. A** coil spring is

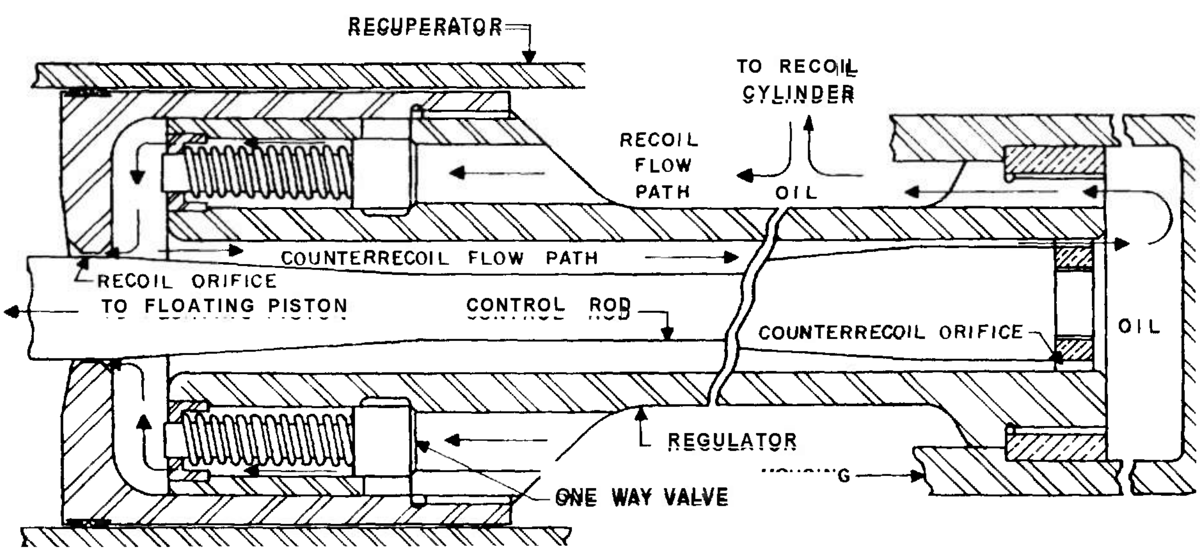


Figure 20. Regulator, Showing Oil Now Paths

used for light loads and, because of space limitations, a stiffer Belleville spring for heavy loads. The springs may be preloaded if necessary.

105. At low angles of elevation, with long recoil stroke and relatively small force, the throttling

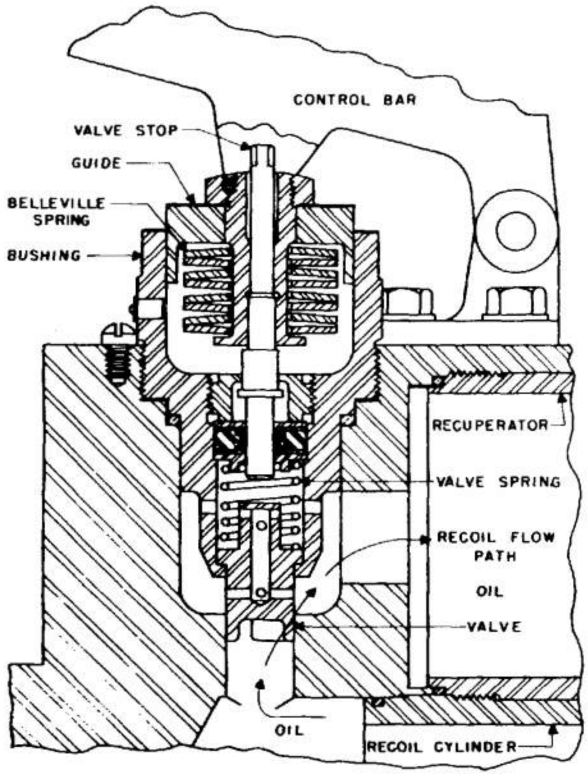


Figure 21. Recoil Throttling Valve

valve's resistance to opening comes only from the coil spring and recuperator pressure. This permits the valve to open wide, thus providing the large orifice needed. When the gun is elevated to intermediate angles, the control arm, actuated by the elevating motion, moves the upper spindle closer to the lower. When the valve is partially open, the two spindles come into contact, bringing into action the higher capacity Belleville spring to provide the increased resistance to further valve travel. At maximum elevation, the two spindles are in contact from the very start of valve travel, and the valve resistance is that of both springs.

106. The design of the recoil throttling valve is based upon recoil velocity and required orifice pressure. As a starting point, the velocity of free recoil (Eq. (2)) and the approximate recoil force (Eq. (3b)) are used. Friction is neglected at this time, but the recuperator pressure is an important factor and must be considered.

107. The orifice area to obtain the necessary increase in recoil cylinder pressure may be found from Equation (36), rewritten in terms of pressure rise. The peripheral discharge area of the recoil throttling valve is:

$$A_o = \frac{Av}{C_o}\sqrt{\frac{\rho}{2P_h}}. \tag{23}$$

Symbols are defined in Paragraph 121. The largest orifice is used for low elevation angles and the corresponding valve travel is:

$$h_c = \frac{A_o}{c}, \quad (23a)$$

where: c = the open periphery of the valve head.

For good control, most of the flow restriction must be in the orifice. Therefore, to minimize the effect of the valve port, its area must be at least five times that of the through flow. Thus:

$$\frac{\pi}{4} D_p^2 \geq 5A_o, \quad (23b)$$

where: D_p = diameter of valve port.

This constitutes a preliminary design procedure. Later, in Chapter IX, Part E, a method is presented for the analysis of flow through a throttling valve along the entire recoil stroke. This analysis determines the advanced design of the valve components.

M. REGULATOR VALVE

108. A regulator valve (Fig. 22) is sometimes used, as in the Filloux mechanism, as the counter-

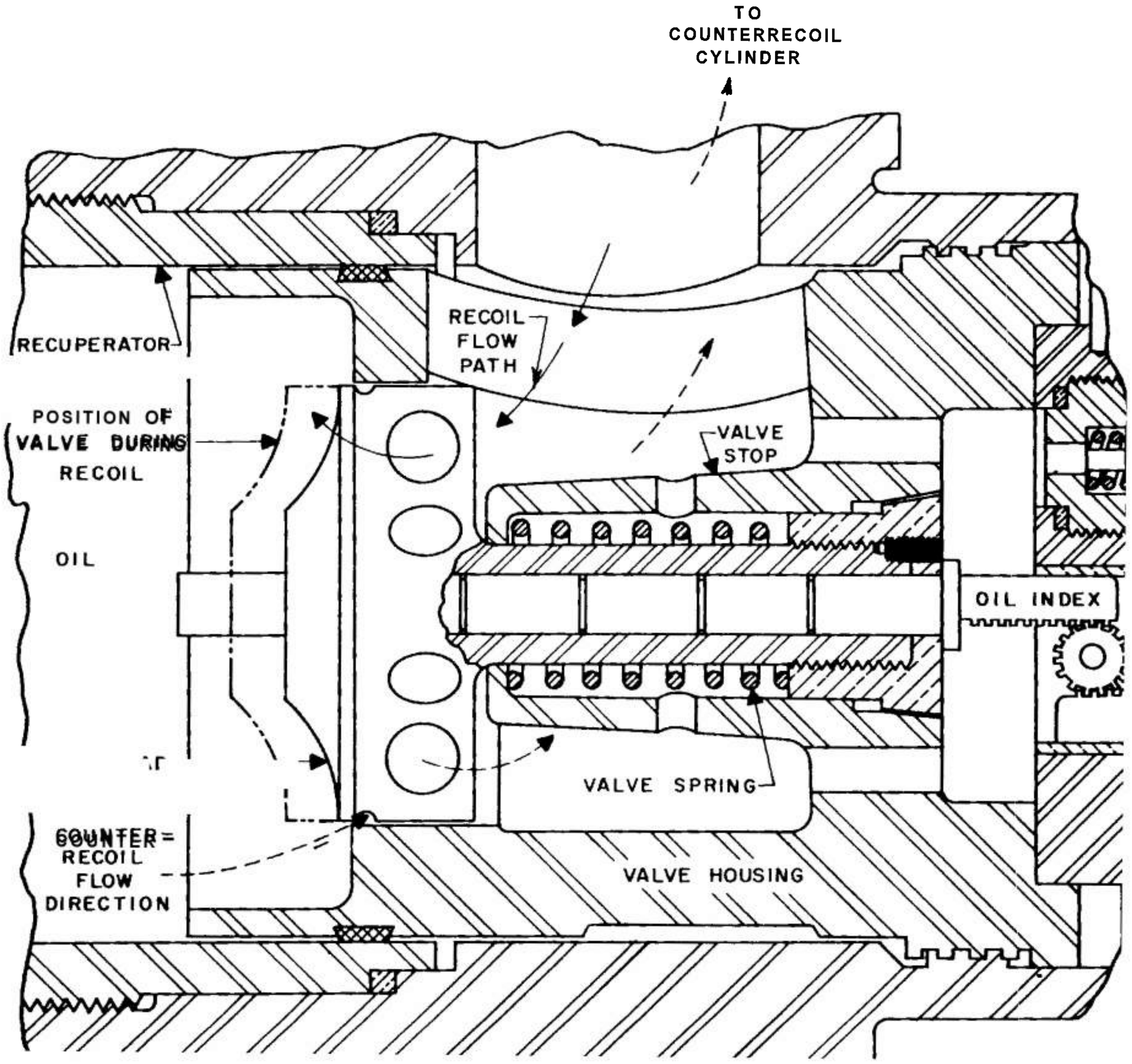


Figure 22. Regulator Valve (Valve Closed for Counterrecoil)

recoil brake. It is housed in the liquid end of the recuperator and regulates the flow of hydraulic fluid from recuperator to counterrecoil cylinder throughout the counterrecoil stroke. During recoil, fluid under pressure opens the valve, permitting relatively free passage through the ports. As counterrecoil begins, the reversed flow of hydraulic fluid, plus the valve spring force, seats the valve. The valve spring must be stiff enough to do this promptly. There are now only small orifices in the valve available for fluid flow. These are designed to provide the proper restriction and to maintain specified counterrecoil velocity as discussed in Chapter IX, Part E.

N. MANUFACTURING PROCEDURES

109. No special techniques are involved in the fabrication of the recoil mechanisms. They can be constructed by standard machine shop operations. It should be emphasized however that surface finishes and clearances for moving parts are critical. According to Ordnance Finishes Specification URAX6, the limit on surface finishes of cylinder bores is 4 microinches RMS, that on antifriction metal is 16 RMS, and that on parts sliding in packings is 8 RMS. To obtain these finishes, the cylinder bore is radially honed and draw polished. The bearing on the piston is turned on a lathe. The one in the stuffing box is bored by lathe. The piston rod is ground, honed (hone fixed, rod rotated), and draw polished. Suggested clearance between sliding members of the recoil brake is 0.0030 ± 0.0005 in. and of the recuperator is 0.005 ± 0.001 in. The antifriction metal of the bearing, whether on piston or in stuffing box, is fitted on assembly to assure the proper sliding fit.

O. MAINTENANCE FEATURES

110. A sound recoil mechanism, always essential for a well-performing weapon, demands proper maintenance. Ease of maintenance thus becomes a principal design criterion. Today's recoil mechanisms, with their high pressures or large spring forces, are potentially too dangerous for any attempt at disassembly in the field. Maintenance activity, except for inspection, is limited to the confines of an arsenal or a specially equipped base shop where experienced crews are available. Apart from the danger involved, field disassembly is discouraged because the interior parts become exposed to dirt that may cause leaks by scratching highly polished surfaces or that may become lodged in valves and packings. The ideal recoil mechanism from this viewpoint, which presents a challenge to the future designer, is one having ready and complete field maintenance features.

111. Maintenance activities may be performed either in the field or in the shop. The latter pertains to all work associated with repair and replacement if disassembly of the whole or part of the recoil mechanism is necessary. Inspections and minor adjustments may be performed in the field. These include checking for damage, wear, failure, or impending failure of all structural components. A dented cylinder, scored or worn sliding surfaces, a bent piston rod, or crushed threads are positive indicators of need for repair. Check points, filling, and drain plugs should be readily accessible. The oil index or replenisher should show a sufficient oil reserve at all times. Both gas and oil should be measured for proper working pressures. Leaks should be detected and, if minor, stopped by tightening the fitting involved. If leaks persist, requiring continued refills, defects at the packings or seals are indicated and call for shop repairs.

112. Recoil mechanisms have relatively large, highly finished interior surfaces which are subject to deterioration when in prolonged disuse such as during storage periods. This progressive damage may be retarded considerably by the use of oil containing corrosion inhibitors but such help is confined to those parts containing oil. Formerly the principal cause of deterioration has been the corrosive effect of unlubricated packings on cylinder walls and rods attributed to the residual acid in the leather. Exercising the mechanism by moving rods and pistons to re-establish the oil film between packings and sliding surfaces practically eliminates all corrosion tendencies. Exercising may be accomplished by pulling the recoiling parts with a winch or similar apparatus or, better still, by firing the weapon if this is practical. The operation should be performed at least once in each six-month period in normal climates and, unless inspections indicate otherwise, oftener in extreme climates. Tests show that newly adopted Teflon packings have eliminated all traces of corrosion. However, until the improvements shown by this de-

velopment have been firmly established over long periods, the practice of exercising the mechanism should continue.

IX. SINGLE RECOIL SYSTEM CALCULATIONS

A. PROPELLANT GAS FORCE *VS.* TIME CURVE

113. Recoil calculations are based on the principle of linear impulse and momentum which states in essence that the linear impulse equals the change in linear momentum. Expressed mathematically,

$$\int_{t_1}^{t_2} F_g dt = M(v_2 - v_1), \qquad (24)$$

where:

M = mass equivalent of projectile and propellant gas,
v_1 = projectile velocity at time, t_1,
v_2 = projectile velocity at time, t_2.

The expression $\int_{t_1}^{\cdot} F_g dt$ represents the impulse of the applied force and is obtained by measuring the area under the force-time curve between the time limits of t_1 and $t2$. This curve is obtained from the interior ballistician (see Fig. **23).** The subject of interior ballistics is covered in a separate Ordnance Corps Pamphlet, ORDP 20-247, Design for Projection **(C).**

$$F_g = P_g A_g, \text{ propellant gas force,} \qquad (24a)$$

where: P_g = propellant gas pressure
$A,$ = bore area.

Since the expression $M(v_2 - v_1)$ represents a change in the momentum of projectile and propellant charge, then, according to the law of the conservation of momentum,

$$M(v_2 - v_1) = M_r(v_{f_2} - v_{f_1}) \qquad (24b)$$

where: M_r = mass of recoiling parts,
v_{f_1} = velocity of free recoil at t_1,
v_{f_2} = velocity of free recoil at t_2.

B. RECOIL FORCE CHART

114. The length of recoil stroke and the approximate retarding force were determined earlier in Paragraph **78.** It is now desired to separate the retarding force into its several parts. A chart is drawn representing increments of stroke as abscissae and the corresponding forces as ordinates (see Figure *5).* First, the total retarding force is plotted. Next, frictional resistance, K_f, of the cradle guides and recoil brake packings is subtracted and the net result, K_R, plotted. Now the recuperator force, K_a, plus its packing friction and other losses, $(f_p + f_o)$, are shown. The difference between the net retarding force, less friction, and the total recuperator force, including friction, must be provided by the hydraulic resistance of the recoil brake, F_o. Finally, the net recuperator force, less friction, is that which is available to return the gun to the in-battery position.

C. RECOIL CALCULATIONS

115. **A** long recoil stroke is desirable as it reduces forces, but the length will necessarily be limited by the size and arrangement of the weapon. Calculations of recoil forces and length of stroke are based on the mass of recoiling parts, the gas force-time curve (Figures 23 and **24)** and the recoil force-distance curve (Figure *5).* The gas force acts from the time the projectile first begins to move in the tube until after it has left the muzzle and the gas pressure has decayed to zero.

116. On the force-time curve, select an increment of time, *At,* small enough that the included curve may be reasonably represented by a straight line, and take for the average gas force, , a value halfway along *At.* The average force tending to accelerate the recoiling parts is this gas force plus the component of weight parallel to the bore:

$$F = F_g + W_r \sin\theta. \qquad (25)$$

The force is opposed by the total retarding force, K (refer to Figure 5), so that the net accelerating force is:

$$F_a = F - K = M_r \frac{d^2x}{dt^2} = M_r a, \qquad (25a)$$

where: a = average recoil acceleration during Δt

According to D'Alembert's principle, F_a also represents the inertia force of the recoiling parts. Despite all endeavors to the contrary, the recoil (retarding) forces seldom work out to be constant during the first part of the stroke and so must be

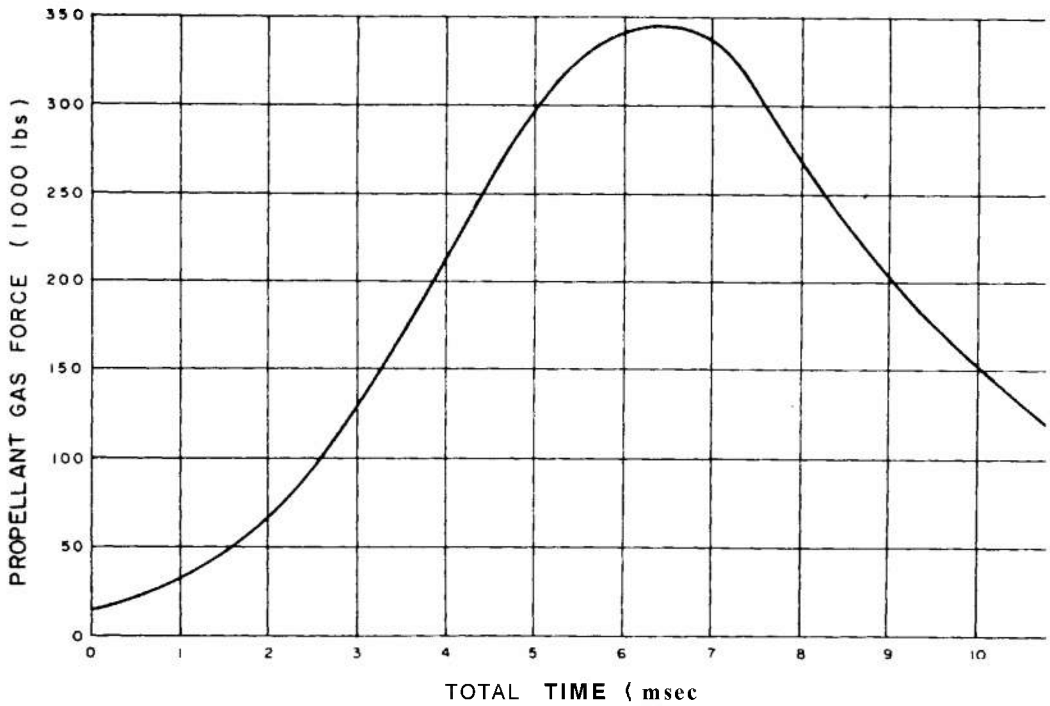


Figure 23. *Propellant Gas Force–Time Curve During Projectile Travel in Bore*

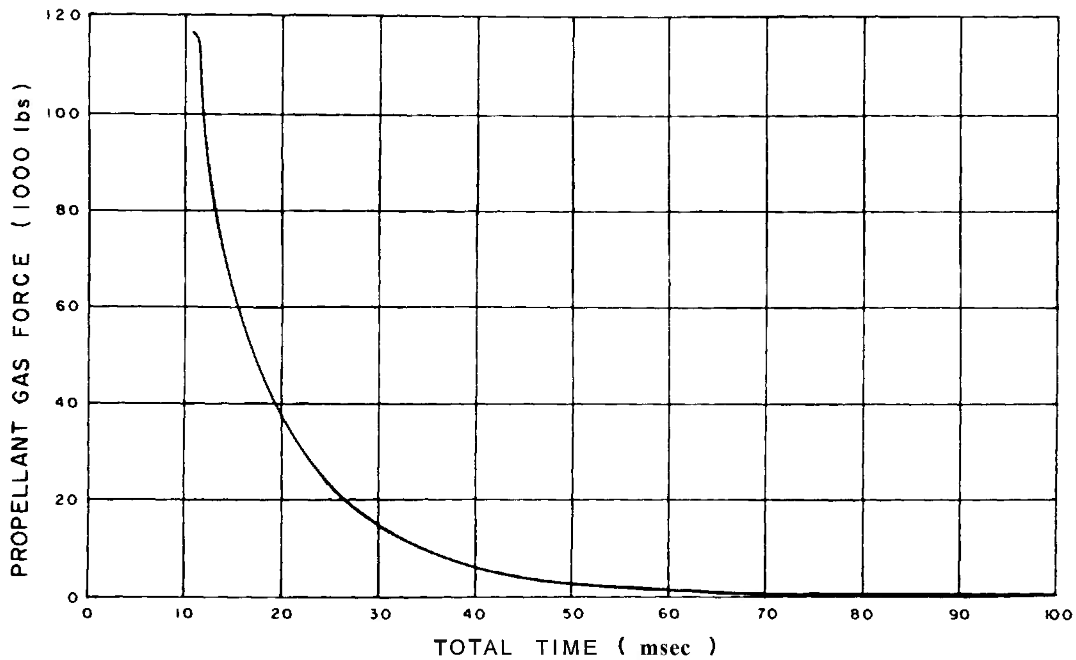


Figure 24. *Propellant Gas Force–Decay Curve After Projectile leaves Muzzle*

considered as an average along *Ax,* the recoil travel during Δt. Then:

$$K = \tfrac{1}{2}(K_{n-1} + K_n). \tag{26}$$

K_n is selected as the point on the curve where it is anticipated the recoiling parts will have reached at the end of *At* (Paragraph 78 and Fig. 5). After Δx has been calculated, it is checked for compliance with K_n. If $K_{\Delta x}$ is significantly different from K_n, another trial must be made and the process repeated. After K has been established and the net accelerating force calculated, the acceleration of the recoiling mass is found from Equation (25a):

$$a = \frac{F_a}{M_r}, \tag{27}$$

where: M_r = mass of the recoiling parts.

The change in velocity is:

$$\Delta v = a\Delta t, \tag{28}$$

and the total velocity at the end of each interval:

$$V_r = \Sigma\Delta v. \tag{29}$$

The average velocity during *At* becomes:

$$v_a = \tfrac{1}{2}(v_{n-1} + v_n), \tag{30}$$

where v_{n-1} is velocity of recoil at beginning of *At*, and the distance traveled during *At* is:

$$\Delta x = v_a At. \tag{31}$$

The total travel becomes:

$$x = \Sigma\Delta x. \tag{32}$$

117. At the beginning of recoil, where the recoil velocity is zero, the total resistance must be provided by the recuperator. If the orifice area is zero at this point, no fluid flow can occur, so the initial recoil travel depends solely on fluid compressibility. However, there is no assurance in some recoil mechanisms that the orifice will be opened after the fluid is compressed, with the result that high peak pressures will occur. This condition may be corrected by providing the minimum orifice at the beginning of recoil stroke to assure oil passage during the entire stroke. Then, the recoil force may be varied from the minimum, or recuperator force at the beginning, to the maximum at some arbitrary point after motion begins. The rate at which the recoil force increases should be as near that of the propellant gas force as possible until the maximum retarding force is reached so as to assure the maximum efficiency of the recoil mechanism and still not develop peak pressures.

118. When recoil motion ceases, the distance traveled should be equal to the specified recoil distance. If there is a lack of agreement of more than 1.5% the recoil force, K, must be adjusted to compensate for the discrepancy. The entire performance analysis must then be repeated until close agreement between specified and computed recoil distances is achieved. *(See* sample problem, Paragraphs 210 through 213.) It is possible that the above recoil calculations, as well as those for counterrecoil (see Paragraph 120), can be adapted to high-speed computers.

D. ANALYSIS OF FLUID BEHAVIOR DURING RECOIL

1. Recoil Force

119. The total retarding force is the sum of the hydraulic (brake) force, the recuperator force, and the sliding frictional force of the recoiling parts. Sliding friction is governed by the weight of recoiling parts plus the effects of eccentricities. Assuming a dependent type of mechanism, with only one rod attached to the breech ring, the reaction on the front cradle bearing is calculated by taking moments about the intersection of R_2 and K_f and solving for R_1 (see Fig. 25):

$$R_1 = \frac{d_r}{a} K_R + \frac{b}{a} W_r \cos\theta + \frac{d_f}{a}(F_a - W_r \sin\theta) - \frac{e}{a} F_g. \tag{33}$$

The lengths a and b are defined in Fig. 25. The directions of R_1 and R_2 may be the same or opposite, depending on the value of R_1 compared with $W_r \cos e$. If $R_1 < W_r \cos e$, then the reaction on the rear bearing is:

$$R_2 = W_r \cos\theta - R_1, \tag{34}$$

acting in the direction of R_1. If $R_1 > W_r \cos e$, then:

$$R_2 = R_1 - W_r \cos\theta, \tag{34a}$$

acting in the opposite direction. When computing friction of the slides, the reactions are added arithmetically, thus,

$$K_f = \mu(R_1 + R_2), \tag{35}$$

is the frictional force. The total resistance to recoil consists of two components:

$$K = K_R + K_f, \tag{35a}$$

where K_R = recoil brake rod force or rod pull. According to Equations (25) and (25a),

$$F_a = F_g + W_r \sin\theta - K. \tag{35b}$$

Now, by making the proper substitutions in Equa-

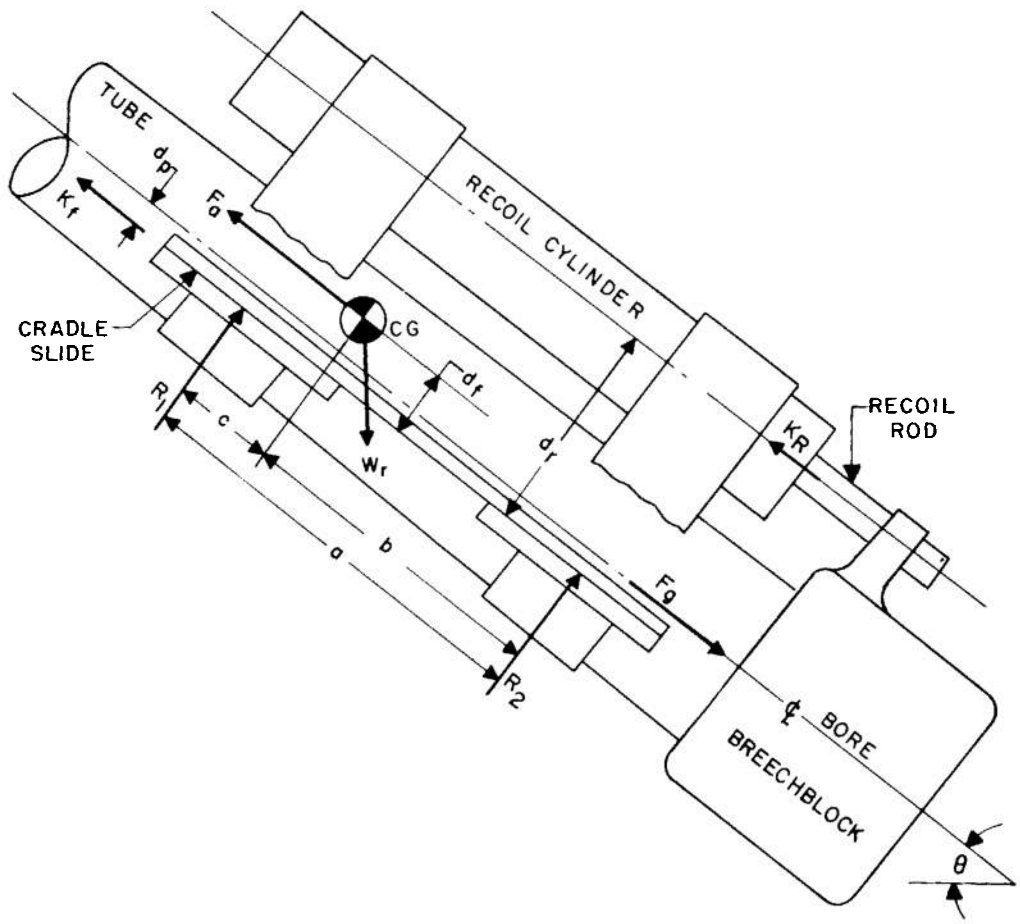


Figure 25. forces and Reactions on Recoiling Parts

tion (33) with the values obtained from Equations (34) through (35b), the front reaction is:

$$R_1 = \frac{d_r - d_f}{a} K + \frac{b - \mu d_r}{a} W_r \cos\theta + \frac{d_f - d_p}{a} F_g, \tag{35c}$$

when $R_1 < W_r \cos\theta$, or

$$R_1 = \frac{d_r - d_f}{a + 2\mu d_r} K + \frac{b - \mu d_r}{a + 2\mu d_r} W_r \cos\theta + \frac{d_f - d_p}{a + 2\mu d_r} F_g, \tag{35d}$$

when $R_1 > W_r \cos\theta$. This is a tedious process if made for each step of the detailed analysis. Since the propellant gas force is present for only a short recoil distance, the error introduced is negligible if F_g is taken as zero and R_1 and R_2 are assumed constant over the total recoil distance.

120. The recoil rod force, K_r, comprises the force, F_o, generated by the restriction of the controlled orifice, the force due to the elastic medium of the recuperator, K_a, the packing frictional force, f_p, and the resistance, f_o, of oil flow through the various ports. The force required to overcome the inertia of hydraulic fluid and the moving internal parts of the system is so small in comparison with other forces that it can be excluded with negligible error. Also, f_o may be so small as to be negligible. A spot check only is required to decide when it should be considered.

2. Orifice Size

121. From hydraulics, the rate of flow:

$$Q = Av = C_o a_o v_o,$$

where: A = effective area of recoil piston,
a_o = area of orifice,
v = recoil velocity,
v_o = velocity of flow through orifice,
C_o = orifice coefficient or coefficient of discharge.

In terms of the velocity head, h:

$$v_o = \sqrt{2gh}.$$

Now if, P = hydraulic pressure in recoil brake cylinder,
P_r = recuperator pressure or equivalent pressure of spring,
$P_h = P - P_r$ = pressure rise due to orifice,

then $P_h = wh$ = hydraulic pressure head,

where: w = density of fluid,

and $F_o = P_h A$ = hydraulic force generated by orifice,
$\rho = \frac{w}{g}$ = mass density of fluid.

Substituting these values in the flow equation for Q,

$$Av = C_o a_o \sqrt{\frac{2F_o}{A\rho}}.$$

Solving for the orifice area,

$$a_o = \frac{v}{C_o}\sqrt{\frac{\rho A^3}{2F_o}}. \tag{36}$$

122. The orifice coefficient, or coefficient of discharge, depends to a great extent on Reynolds number and on whether the orifice is sharp or round-edged and to a lesser extent on its shape, e.g., circular or rectangular. A list of coefficients previously used and their manner of application follows*:

C_o = 0.95, throttling bar (round),
C_o = 0.95, rectangular groove (counterrecoil),
C_o = 0.77 to 0.91, throttling valve (recoil),
C_o = 0.71 to 0.83, rectangular groove (buffer),
C_o = 0.60, for sharp edge orifice.

Flow through the orifice in a recoil system is confined to the turbulent range. Also, the ratio of orifice area to piston area is generally less than 0.02. Consequently, according to Rouse,† a discharge coefficient of 0.60 is recommended. Ballistics Research Laboratories* offers another approach but consigns it to concentric recoil mechanisms. It incorporates the use of drag coefficients in relation to the ratios of piston areas to orifice areas.

* These values were provided by Watertown Arsenal.
† Page 261 of Reference 11.

123. Equation (36) assumes that all the fluid displaced by the recoil piston passes through the orifice. If some of the displaced fluid does not flow through the orifice but remains to fill the void left by the moving control rod, as in the Puteaux mechanism, the equation must be revised. The derivation of the revised equation appears in Paragraph 135. The orifice area becomes:

$$a_o = \frac{v}{C_o}\sqrt{\frac{\rho A^3}{2F_o}\left(\frac{A_R - A_c}{A_R}\right)^3}, \tag{37}$$

where: A_R = recuperator cross section area,
A_c = control rod cross section area.

The area of the control rod varies but, for use in Equation (37), it may be considered as constant. A_c is small compared with A, and the error introduced will be negligible.

3. *Losses in the Hydraulic System*

124. Considering the losses in the hydraulic system, the orifices generate the resistance:

$$F_o = K_R - K_a - (f_p + \Sigma f_o). \tag{38}$$

Paragraph 90 discusses the packing frictional forces, f_p. In some recoil mechanisms, the fluid does not flow directly from the recoil piston to the orifice but passes through other restrictions such as ports between cylinder and recuperator and through valves of regulators. These restrictions are not necessarily deleterious to the proper functioning of the recoil mechanism as they merely contribute to the throttling pressure. For good control, however, their total influence must be small compared to that of the regulated orifice. According to Equation (36), the force on the recoil piston necessary to develop the pressure at each restriction in series is:

$$f_o = \frac{\rho A^3 v^2}{2C_o^2 A_i^2} \tag{39}$$

C_o may be taken as unity. The terms in Equation (39) are defined in Paragraphs 120 and 121, except for A_i, the port area. As stated in Paragraph 120, a spot check quickly determines whether this force can be treated as negligible.

* Reference 12.

4. *Compressibility of Hydraulic Fluid*

125. If the fluid were incompressible, the distance travelled by the moving parts would be directly proportional to the volume of displaced fluid. But the fluid does compress, and the recoil piston will move farther than the distance indicated by the displaced volume. When this additional distance becomes appreciable, allowance must be made for it in design. Experimental data are insufficient to conclude precisely when this factor should be considered, hence it remains a matter of engineering judgment. As a conservative measure, compressibility should be considered if it permits an increase in recoil stroke of more than 29 percent of the overtravel. Overtravel is defined as an additional distance provided in case the recoil energy, for some uncontrollable reason, is not completely absorbed within the limits of the specified recoil travel distance. Consider the mechanism in Figure 26. The travel of the piston (and, therefore, the recoiling parts) and of the control rod are both influenced by the compressibility of the fluid. The volume of the hydraulic fluid between brake piston and orifice plate is represented by:

$$V = V_0 - xA, \tag{40}$$

where: V_0 = the initial fluid volume,
A = effective area of the piston,
x = travel distance of recoiling parts at any time t.

The change in volume due to compressibility is:

$$\Delta V = V\frac{\Delta P}{M_b}, \tag{41}$$

where: M_b = bulk modulus of the fluid,
$\Delta P = P - P_0$,
P = fluid pressure in brake cylinder at x,
P_0 = fluid pressure, in-battery position (minimum recuperator pressure).

The recoil travel distance then becomes:

$$x = x' + \Delta x, \tag{42}$$

where: x' = travel distance if the fluid is incompressible,

$$\Delta x = \frac{\Delta V}{A},$$

The calculations showing the control rod displace-

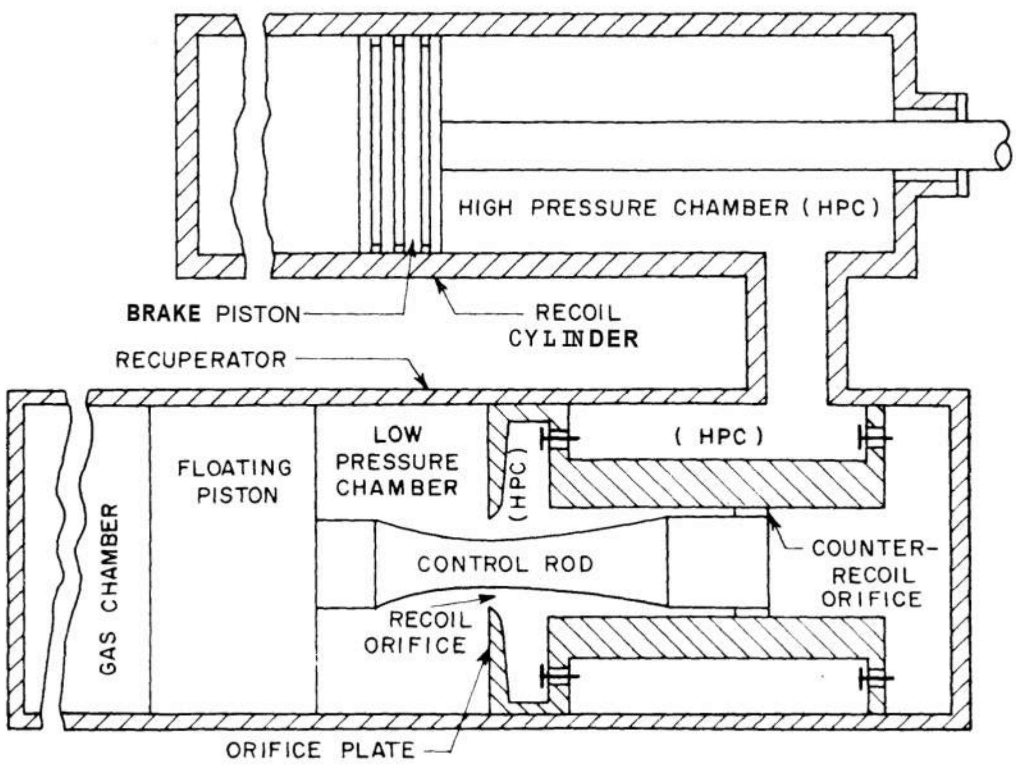


Figure 26. Oil Chambers of a Recoil Mechanism

ment are somewhat more involved. All the liquid forced from the brake cylinder does not flow through the orifice. A small amount, equal in volume to that part of the control rod which passes through the orifice, fills that void. The volume of fluid flowing through the orifice is:

$$V_{f_1} = (xA - \Delta V) - V_c, \tag{43}$$

where: V_c = displaced volume of control rod.

Although the diameter of the control rod varies, the mean area of the rod is assumed to extend along its total length to simplify the calculations. This is permissible as the area of the rod at any point is small in comparison with the area of the recuperator; hence, only a negligible error is involved. The total displacement of floating piston plus control rod is:

$$A_R x_R = (xA - \Delta V), \tag{44}$$

where x_R = travel of control rod,
and the displacement of the control rod is:

$$V_c = A_c x_R. \tag{45}$$

Subtracting the two volumes:

$$(A_R - A_c)x_R = (xA - \Delta V) - V_c,$$

or: $$\frac{A_R - A_c}{A_R} A_R x_R = (xA - \Delta V) - V_c.$$

But $A_R x_R = (xA - \Delta V)$, hence,

$$V_{f_1} = (xA - \Delta V)\frac{A_R - A_c}{A_R}, \tag{46}$$

where: A_R = area of the recuperator cylinder,
A_c = average area of the displaced portion of the control rod.

During recoil, the gas is compressed and the pressure increases. thus:

P_0 = fluid pressure: in-battery position,
P_x = gas pressure at any position of recoil.

The liquid in the low pressure chamber between the floating piston and the orifice plate has a slightly larger volume than if it were subjected to the pressure in the brake cylinder. The total volume in the low pressure chamber becomes:

$$V_{f_2} = V_{f_1} + V_{f_0}, \tag{47}$$

where: V_{f_0} = the initial fluid volume.

The compressibility changes this volume to this value:

$$V_{f_2} = V_{f_1}(1 - M_b \Delta P_x) + V_{f_0}(1 - M_b \Delta P_0), \tag{47a}$$

where: $\Delta P_x = P_x - P$,
$\Delta P_0 = P_x - P_0$.

Thus the component V_{f_1} will increase in volume and V_{f_0} will decrease. The distance traveled by the control rod now becomes:

$$x_R = \frac{V_{f_2} - V_{f_0}}{A_R - A_c}. \tag{48}$$

The control rod diameters will be the same as those for the incompressible fluid but at different locations along its length.

5. *Analysis for Recoil Throttling Valve*

126. The calculations for a recoil mechanism that is controlled by a throttling valve (Figure 21) such as for the St. Chamond are basically the same but somewhat more involved than for the mechanism whose orifice size is controlled by the position of the recoiling parts. In the latter, the orifice area is determined from established recoil forces and velocities as discussed in Chapter IX, Parts C, D-1 and D-2. The behavior of the throttling valve is more complex because neither recoil force nor velocity can be determined independently of the fluid flow. For example, suppose that a constant pressure rise through the spring-loaded throttling valve is desired during the total recoil stroke. A constant pressure rise and, hence, a constant force applied to the valve means a constant spring deflection, and, consequently, a constant orifice area. But, according to Equation (23), to maintain all these functions constant, recoil velocity must remain constant. Since the recoil velocity does vary (from some positive value to zero) the above example is unsound. Generally then, the pressure rise, and, hence, the recoil force cannot be evaluated on the basis of recoil distance and velocity alone. However, an acceptable recoil force-distance curve can be obtained by the selection of valve springs with appropriate spring rates.

127. The characteristics of the springs cannot be determined in one operation but must depend on a trial and error analysis. Assuming that the various parameters, including the recoil brake diameter, the recuperator size and pressure, the length of recoil,

and the valve dimensions, have been given fixed values, preliminary valve spring rates can be determined.

128. Two conditions determine the spring rates. The rate of the coil spring for long recoil depends on horizontal firing whereas the combined rate of Belleville and coil springs for short recoil depends on firing at the maximum angle of elevation. In either case velocity and energy of free recoil are found from Equations (2) and (3a), respectively, and the first approximations of K, the total resistance to recoil, is found from Equation (3b). Equation (3b) shows that K is appreciably smaller for horizontal firing, where recoil distance is long, than for high angle firing, where recoil distance is short. The maximum value of F_o, the hydraulic resistance, is obtained from force-distance curves similar to Figure 5. The curves are based on the value of K, above, and on the recuperator pressure found according to Part G, Chapter VIII. It is assumed that maximum F_o is acting at the maximum velocity of free recoil. Since $F_o = AP_h$ (Paragraph 121), Equation (23) may be solved for the maximum discharge area of the valve. Thus, writing Eq. (23), but substituting the maximum velocity of free recoil v_f for v,

$$A_o = \sqrt{\frac{\rho A^2}{2 C_o^2 P_h}}\, v_f, \qquad (49)$$

where: A_o = peripheral area of valve.

But from Eq. (23a),

$$A_o = ch_c, \qquad (49a)$$

where: c = the open periphery of the valve head,
h_c = coil spring deflection or valve lift.

According to Paragraph 121,

$$P = P_h + P_r. \qquad (49b)$$

Thus, for the force on the coil spring,

$$F_c = A_v P, \qquad (49c)$$

where: A_v = effective pressure area of the valve.

Solving for the coil spring rate:

$$K_c = \frac{F_c - F_c'}{h_c}, \qquad (49d)$$

where: F_c' = the initial coil spring load.

For the first trial assume, $F_c' = \frac{F_c}{2}$.

Values are determined similarly for the maximum angle of elevation. For this condition P_h is larger and A_o is smaller.

$$A_o = ch_s, \qquad (49e)$$

where: h_s = deflection of the combined coil and Belleville springs.

The force on the springs is:

$$F_s = A_v P, \qquad (49f)$$

and the combined spring rate is:

$$K_s = \frac{F_s - F_s'}{h_s}, \qquad (49g)$$

where: F_s' = the combined initial spring load. The initial Belleville load is:

$$F_B = F_s' - F_c'. \qquad (49h)$$

For the initial analysis assume that $F_s' = \frac{F_s}{2}$.

The required Belleville spring rate is:

$$K_b = K_s - K_c. \qquad (49i)$$

129. The values obtained from the equations of the 49 series are preliminary and are used for the first step-by-step analysis of the recoil system. The procedure followed in the analysis is developed in Chapter IX of Reference 4. First consider the horizontal firing position for long recoil:

$$v_n = v_{(n-1)} + (v_{f_n} - v_{f_{(n-1)}}) - \frac{AP_{(n-1)} + K_f}{M_r}\Delta t, \qquad (50)$$

the recoil velocity at interval n,

where: A = effective recoil piston area,
K_f = frictional resistance to recoil,
M_r = mass of recoiling Darts,
$P_{(n-1)}$ = recoil brake pressure during the past interval.

The change in free recoil velocity:

$$v_{f_n} - v_{f_{(n-1)}} = \Delta v_f, \qquad (50a)$$

is determined by the method discussed in Paragraph 116.

Select a short time interval (say Δt = 0.001 sec) on the propellant gas force curve (Fig. 23) and read the force F_g at $\frac{1}{2}\Delta t$, assuming it acts throughout Δt. The accelerating force on the recoiling parts becomes, according to Equation (25),

$$F = F_g + W_r \sin e.$$

At e = 0, zero angle of elevation, the accelerating force simply becomes the gas force. Since the acceleration,

$$a = \frac{F}{M_r}, \qquad (50b)$$

the change in free recoil velocity is:

$$\Delta v_f = a\Delta t = \frac{F}{M_r}\Delta t. \qquad (50c)$$

During this time, resistance to recoil is offered by K_f, the frictional forces, and ***AP***, the recoil brake force. This resistance can be expressed in terms of "change in velocity", similar to Equation (50c),

$$\Delta v = \frac{AP_{(n-1)} + K_f}{M_r}\Delta t. \qquad (50d)$$

If **At is** small, the use of $P_{(n-1)}$ rather than *P*, which is unknown, is permissible as the error involved is small.

130. The pressure in the recoil brake cylinder is solved by the equation

$$P = m\left[\frac{N^3}{2} + \left(\frac{N^6}{4} - \frac{1}{730}\right)^{\frac{1}{2}}\right]^{\frac{1}{3}} + m\left[\frac{N^3}{2} - \left(\frac{N^6}{4} - \frac{1}{730}\right)^{\frac{1}{2}}\right]^{\frac{1}{3}} - \frac{2}{3}m + P_r \qquad (51)$$

where: P_r = the recuperator pressure, oil end.

$$N^3 = \frac{2}{27} + \frac{B}{m^3}, \qquad (51a)$$

$$B = \frac{\rho A^2 K_c^2}{2C_o^2 c^2 A_v^2} v_n^2, \qquad (51b)$$

$$m = \frac{Jv_n^2 - F_c'}{A_v}, \qquad (51c)$$

$$J = \frac{\rho A^2}{2A_v}, \qquad (51d)$$

p = mass density of fluid.

Except for short distances at the beginning and end of recoil, 1/730 becomes negligible in comparison with $N^6/4$ and may be omitted without serious error. Equation (51) therefore reduces to

$$P = m(N - \tfrac{2}{3}) + P_r. \qquad (51e)$$

It is well to remember that for the first step the recoil velocity is:

$$v_1 = v_{f_1} - \frac{A\left(P_r - \frac{F_c'}{A_v}\right) + K_f}{M_r}\Delta t. \qquad (51f)$$

After the analysis **is** completed, the recoil force-distance curve is shown to compare it with the original assumed curve. If the calculated distance,

$$x = \Sigma v_n \Delta t, \qquad (52)$$

lacks agreement by more than 1.5 percent with the specified distance and if the resistance to recoil,

$$K = AP + K_f, \qquad (52a)$$

peaks at any time, the springs should be redesigned and the analysis repeated until the results are acceptable.

131. For the shortest recoil, i.e., for maximum angle of elevation, the recoil calculations follow the same general procedure **as** for long recoil. K_s, the combined spring rate, is acting throughout the recoil stroke and the expressions for ***B*** and *m* change accordingly:

$$B = \frac{\rho A^2 K_s^2}{2C_o^2 c^2 A_v^2} v_n^2, \qquad (52b)$$

$$m = \frac{Jv_n^2 + A_s P_r - F_s'}{A_v}, \qquad (52c)$$

where: ***A,*** = area of the valve stem.

For the first step:

$$v_1 = v_{f_1} - \frac{A\left(P_r - \frac{A_v - A_s}{A_v} + \quad\right) \quad K_f}{M_r} \qquad (52d)$$

132. After the recoil particulars have been established for zero and maximum angles of elevation, the intermediate firing positions **are** evaluated. The problem, here, is to deterniine when to bring the Belleville springs into action *to* augment the coil spring. At low angles of elevation, this action is delayed until near the end of recoil. ***At*** higher angles, it is brought into play near the beginning. The space between the valve stem and the Helleville spring spindle sets the distance through which the coil spring must deflect before the Belleville springs begin *to* function. This space is regulated **by** a cam attached *to* the tipping parts so that the higher the elevation, the smaller

the space. During the early part of recoil, the calculations are the same as for zero elevation. During the later part of recoil, the calculations follow the procedure for the maximum angle of elevation with the exception that:

$$m = \frac{J v_n^2 + A_r P_r - F_s''}{A_v} \tag{52e}$$

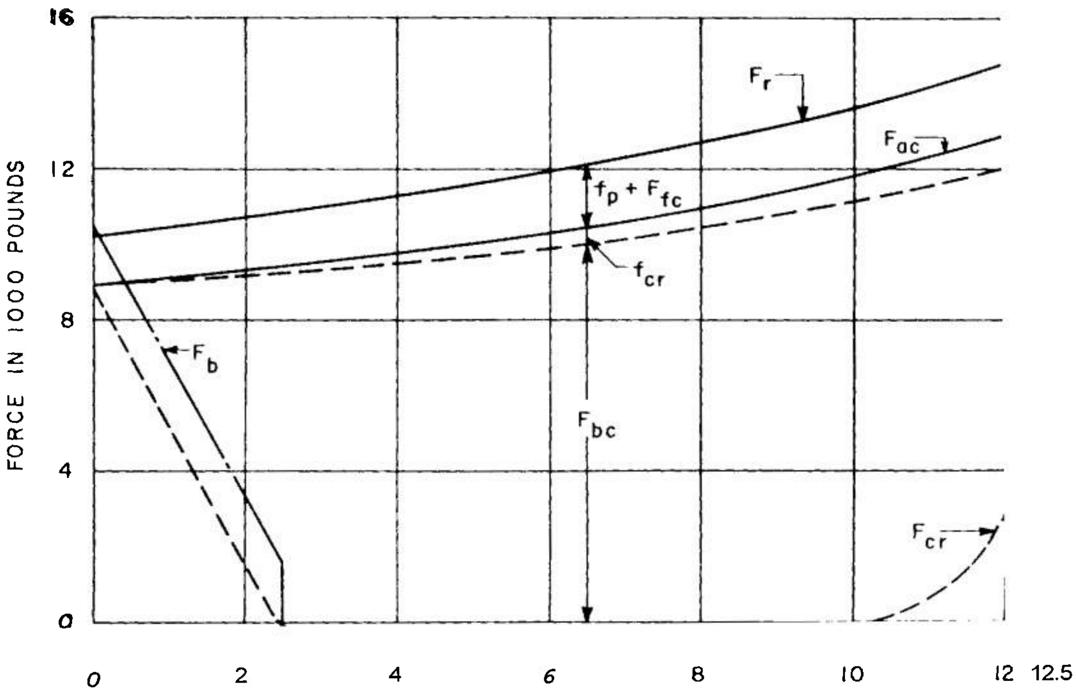
where:

$$F_s'' = F_c + F_B. \tag{52f}$$

The F_c, here, is the coil-spring load just as the valve stem contacts the valve stop to bring the Belleville springs into play.

E. COUNTERRECOIL CALCULATIONS

133. If free counterrecoil is permitted, i.e., no additional restrictions to fluid flow exist other than those during recoil, the velocity of counterrecoil



DISTANCE OUT OF BATTERY

THE ARROW INDICATES THE DIRECTION OF COUNTERRECOIL

F_r = FORCE ON COUNTERRECOIL PISTON

F_{ac} = AVAILABLE COUNTERRECOIL FORCE BEFORE THROTTLING

F_{cr} = NET COUNTERRECOIL ACCELERATING FORCE

F_b = TOTAL BUFFER FORCE

F_{bc} = HYDRAULIC RESISTANCE OF COUNTERRECOIL ORIFICE

F_{fc} = FRICTIONAL RESISTANCE OF SLIDES DURING COUNTERRECOIL

f_{cr} = HYDRAULIC RESISTANCE OF RECOIL ORIFICE DURING COUNTERRECOIL

f_p = TOTAL FRICTIONAL RESISTANCE OF PACKINGS

x_r = COUNTERRECOIL STROKE AT ANY TIME t.

Figure 27. Counterrecoil Force Chart

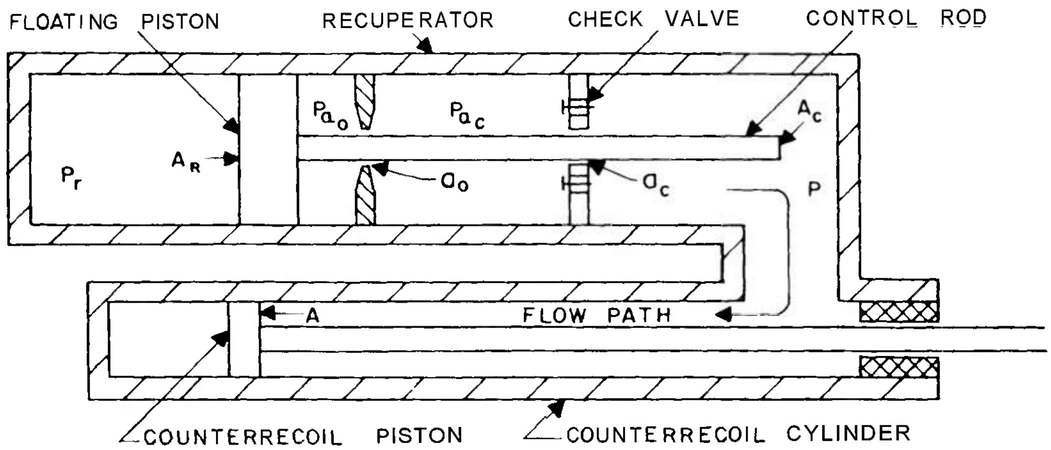


Figure 28. Functioning Components During Counterrecoil

usually will be excessive. For rapid-fire guns, more freedom of flow is required and other flow channels must be provided. However, when low counterrecoil velocities are specified, additional restrictions are necessary. The calculation procedures for this last type are discussed in detail, although the computations are essentially the same for all types.

134. The counterrecoil force is provided by the recuperator and is obtained from a chart similar to Figure 27. Before it accelerates the counterrecoiling mass, it loses some of its potential in frictional resistance of slides and packings. The packing losses are found, according to the method in Paragraph 90, for the in-battery position and the position just before counterrecoil begins. The mean of the two results is assumed, without serious error, to act over the total stroke, thus:

$$f_r = \tfrac{1}{2}(f_{pi} + f_{po}), \tag{53}$$

$$f_c = \tfrac{1}{2}(f'_{pi} + f'_{po}). \tag{53a}$$

Then:

$$F_{fc} = \mu W_r \cos\theta, \tag{54}$$

where:
F_{fc} = frictional resistance of slides during counterrecoil,
f_c = the frictional force of packings of the recoil brake and the counterrecoil cylinder,
f_{pi} = the in-battery packing frictional force of recuperator,
f_{po} = the out-of-battery packing frictional force of recuperator,
f'_{pi} = the total in-battery packing frictional force of the recoil brake and the counterrecoil cylinder,
f'_{po} = the total out-of-battery packing frictional force of the recoil brake and the counterrecoil cylinder,
f_r = frictional force of packings in recuperator.

135. The counterrecoil force is further reduced by the restriction of flow through the orifice determined for recoil regulation. If a further reduction is necessary to maintain low counterrecoil velocities, the hydraulic fluid is directed through another orifice before it reaches the counterrecoil cylinder. Figure 28 is a schematic drawing of the functioning components during counterrecoil,

where:
A = area of counterrecoil piston,
A_c = area of control rod,
A_R = area of floating piston,
a_c = area of orifice for counterrecoil,
a_o = area of orifice for recoil,
P = oil pressure in recoil brake cylinder after throttling through a_c,
P_{a_c} = oil pressure before throttling through a_c,
P_{a_o} = oil pressure before throttling through a_o,
P_r = recuperator gas pressure,
ρ = mass density of hydraulic fluid,
C_c = orifice coefficient of a_c,
C_o = orifice coefficient of a_o.

During counterrecoil there are two pressure drops, one at a_c,

$$\Delta P = P_{a_c} - P, \tag{55}$$

and the other at a_o,

$$\Delta P_o = P_{a_o} - P_{a_c}. \tag{55a}$$

From hydraulics, the flow rate through a_c is:

$$Q_c = C_c a_c \sqrt{\frac{2\Delta P}{\rho}}, \qquad (56)$$

likewise, at a_o,

$$Q_o = C_o a_o \sqrt{\frac{2\Delta P_o}{\rho}}. \qquad (56a)$$

Solving for ΔP and P_o:

$$\Delta P = \frac{Q_c^2 \rho}{2C_c^2 a_c^2}, \qquad (57)$$

$$\Delta P_o = \frac{Q_o^2 \rho}{2C_o^2 a_o^2}; \qquad (57a)$$

but $Q_o = Q_c$, so

$$\Delta P_o = \frac{Q_c^2 \rho}{2C_o^2 a_o^2}. \qquad (57b)$$

From Figure 28, it is observed that:

$$P_{a_o} = P + \Delta P + \Delta P_o; \qquad (58)$$

substituting for ΔP and P_o, we have:

$$P_{a_o} = P + \frac{Q_c^2 \rho}{2C_c^2 a_c^2} + \frac{Q_c^2 \rho}{2C_o^2 a_o^2}. \qquad (58a)$$

Since the forces on the floating piston must balance,

$$F_R = P_{a_o}(A_R - A_c) + PA_c + f_r, \qquad (59)$$

where: F_R = the recuperator force.

Substitute for P_{a_o} of Equation (58a) into Equation (59) and collect terms so that:

$$F_R = PA_R + (A_R - A_c)\left(\frac{Q_c^2 \rho}{2C_c^2 a_c^2} + \frac{Q_c^2 \rho}{2C_o^2 a_o^2}\right) + f_r. \qquad (59a)$$

The force on the counterrecoil piston is:

$$F_p = PA. \qquad (60)$$

F_p can be determined from Equation (59a) by substituting the expression for P obtained from Equation (60) and solving for F_p:

$$F_p = F_R \frac{A}{A_R} - A\frac{A_R - A_c}{A_R}\left(\frac{Q_c^2 \rho}{2C_c^2 a_c^2} + \frac{Q_c^2 \rho}{2C_o^2 a_o^2}\right) - \frac{A}{A_R} f_r. \qquad (60a)$$

136. The flow rate through the counterrecoil orifice, Q_c, may also be written in terms of velocity and area:

$$Q_c = (A_R - A_c)v_R, \qquad (60b)$$

where: v_R = velocity of floating piston and the rate of flow into the counterrecoil cylinder is:

$$Q = Av_r = A_R v_R, \qquad (60c)$$

where: v_r = the counterrecoil velocity;

solving for v_R,

$$v_R = \frac{Av_r}{A_R}. \qquad (60d)$$

Therefore,

$$Q_c = Av_r\left(\frac{A_R - A_c}{A_R}\right). \qquad (60e)$$

Now substitute the expression for Q_c from Equation (60e) into Equation (60a):

$$F_p = F_R \frac{A}{A_R} - \frac{\rho}{2C_c^2}\left(\frac{A_R - A_c}{A_R}\right)^3 \frac{A^3}{a_c^2} v_r^2 - \frac{\rho}{2C_o^2}\left(\frac{A_R - A_c}{A_R}\right)^3 \frac{A^3}{a_o^2} v_r^2 - \frac{A}{A_R} f_r. \qquad (60f)$$

By subtracting the frictional resistance of Equations (53a) and (54), the net accelerating force is:

$$F_{cr} = F_p - f_c - F_{fc}. \qquad (61)$$

For convenience, simpler terms are provided for the expressions in Equation (60f). Thus, the resistance provided by the counterrecoil orifice is:

$$F_{bc} = \frac{\rho}{2C_c^2}\left(\frac{A_R - A_c}{A_R}\right)^3 \frac{A^3}{a_c^2} v_r^2 \qquad (61a)$$

and the resistance supplied by the recoil orifice is:

$$f_{cr} = \frac{\rho}{2C_o^2}\left(\frac{A_R - A_c}{A_R}\right)^3 \frac{A^3}{a_o^2} v_r^2. \qquad (61b)$$

Equation (37) has the solution for a_o. Substituting it in Equation (61b) yields:

$$f_{cr} = \left(\frac{v_r}{v}\right)^2 F_r. \qquad (61c)$$

The values for v and F_o are available from the recoil calculations. The available static force on the counterrecoil piston transposed from the force due to the recuperator gas pressure on the floating piston is:

$$F_r = \frac{AF_R}{A_R}, \qquad (61d)$$

and similarly, by transposing the frictional resistance of the recuperator floating piston packing to the counterrecoil piston and adding it to the total of the recoil brake and counterrecoil cylinder, the total frictional resistance of the packings is:

$$f_p = \frac{Af_r}{A_R} + f_c. \qquad (61e)$$

The simplified equation for the net accelerating force is obtained by subtracting frictional and hydraulic resistances from the available static force on the counterrecoil piston:

$$F_{cr} = F_r - (f_p + f_{cr} + F_{bc} + F_{fc}). \qquad (61f)$$

If $F_{ac} = F_r - f_p - F_{fc}$ is the available counterrecoil force before throttling, then,

$$F_{cr} = F_{ac} - f_{cr} - F_{bc}. \qquad (62)$$

Note that when the fluid flow is restrained to limit the counterrecoil velocity, F_{cr} should theoretically become zero when this limit is reached. Actually, for a constant orifice, the limit may be exceeded during the early part of the stroke. Then, in order that the velocity not exceed the limit when buffing starts, F_{cr} becomes negative over a portion of the stroke. This is demonstrated in the example problem (see Tables 5 and 6).

137. The counterrecoil calculations follow a step-by-step integration procedure and each step involves trial and error computations. The critical but unknown factor is the size of the constant orifice required (similar to the one shown in Fig. 20) to control the counterrecoil velocity. As a first approximation of an orifice of reasonable area, assume that it is 10 percent of the maximum orifice area used during recoil. If, in the process of making the counterrecoil calculations, it appears that the velocity will be too high, the area must be reduced; if too low, it must be increased. After the orifice area has been selected, the counterrecoil calculations are made. The values of f_p and F_{fc} are determined from Equations (61c) and **(54)** and are assumed constant. The values of F_{bc} and f_{cr} are calculated from Equations (61a) and (61c). The value of F_r is read from the curve in Figure 27. Select what appear to be reasonable values of Δv_r and Ax_r, the changes in counterrecoil velocity and distance, respectively. Obtain F_r from the curve, half way along x_r at $(x_{r_{n-1}} + \frac{1}{2}\Delta x_r)$. Because v_r is not constant, the following procedure is not exact but, if Δx_r is short enough, the approximations will be sufficiently close to be acceptable.

Then: $$x_r = x_{r_{n-1}} + \Delta x_r, \qquad (63)$$

and $$v_r = v_{r_{n-1}} + \Delta v_r. \qquad (64)$$

Find the values **of** v and F_o **corresponding** to the distance x_r in the recoil calculations and obtain f_{cr} as the average value over the increment:

$$f_{cr} = \left(\frac{v^2_{r_{n-1}} + v^2_{r_n}}{v^2_{n-1} + v^2_{n}}\right)F_o. \qquad (64a)$$

Similarly, **from** Equation (61a), F_{bc} is calculated as the average value over the increment:

$$F_{bc} = \frac{\rho}{2C_c^2}\left(\frac{A_R - A_c}{A_R}\right)^3 \frac{A^3}{a_c^2}\frac{\left(v^2_{r_{n-1}} + v^2_{r_n}\right)}{2}; \qquad (64b)$$

and **find** F_{cr} from Equation (61f) or (62). The counterrecoil acceleration is then found:

$$a_r = \frac{F_{cr}}{M_r}; \qquad (65)$$

and the velocity checked from the equation:

$$v_r^2 = v^2_{r_{n-1}} + 2a_r\Delta x_r. \qquad (66)$$

If the calculated velocity in the last equation does not equal the selected value at the beginning, then the selected value must be revised and the calculations repeated. After the two values agree, the time is calculated:

$$\Delta t = \frac{v_r - v_{r_{n-1}}}{a_r} \qquad (67)$$

and: $$t = t_{n-1} + \Delta t. \qquad (68)$$

138. The buffer, at the end of the stroke, has two force components. One is the decelerating force required to stop the counterrecoiling parts. The other is the net recuperator force determined by Equation (62).

If a buffer is selected with a constant orifice, the calculations continue as for counterrecoil with the addition of another factor, F_b, the total buffing force. The buffer net decelerating force is:

$$F_b' = F_{cr} - F_b; \qquad (69)$$

and: $$F_b = \frac{\rho A_b^3}{2C_o^2 a^2_{o_b}} v_b^2, \qquad (69a)$$

where:
- A_b = area of buffer cylinder,
- a_{o_b} = buffer orifice area,
- C_o = orifice coefficient,
- ρ = mass density of buffer fluid,
- v_b = velocity of counterrecoil during buffing.

The acceleration during buffing **is:**

$$a_b = \frac{F_b'}{M_r}; \qquad (69b)$$

and the velocity of counterrecoil during buffing becomes:

$$v_b^2 = v_{b_{n-1}}^2 + 2a_b \Delta x_b. \tag{69c}$$

If the orifice is variable, constant deceleration provides a constant F_b'. Thus solving Equation (69) for F_b, the total buffer force becomes:

$$F_b = F_{cr} - F_b'. \tag{70}$$

Considering constant deceleration, the buffer velocity can be found from:

$$v_b^2 = v_{rb}^2 + 2\, a_b\, x_b, \tag{70a}$$

where: a_b = buffer acceleration,
v_{rb} = counterrecoil velocity at buffer contact,
x_b = buffer travel at any time, t.

When $x_b = S_b$, the total buffer stroke, $v_b = 0$, and

$$a_b = (-)\frac{v_{rb}^2}{2S_b}, \tag{70b}$$

the net decelerating force becomes:

$$F_b' = M_r\, a_b. \tag{70c}$$

All the factors are now known and the orifice area can be calculated for the total buffer stroke:

$$a_{ob} = v_b \sqrt{\frac{\rho A_b^3}{2C_o^2 F_b}}. \tag{71}$$

X. DESIGN OF CONCENTRIC RECOIL MECHANISMS*

A. INTRODUCTION

139. The concentric recoil mechanism is similar in principle to other types, having a hydraulic brake and a mechanical spring recuperator. However it is frequently mounted on tanks whose requirements differ sharply from those of field artillery. Recoil and counterrecoil forces can be set high because of the rigidity of the carriage. Actually they have to be high because of the limited travel imposed by the turret. This is one example which illustrates that under most conditions the design requirements of the tank take precedence over the recoil mechanism although it is always attempted to design the latter to the usual high standards of precision.

* **The material on Concentric Recoil Mechanisms was provided by Ordnance Tank-Automotive Command, Detroit, Mich.**

B. TYPES OF CONCENTRIC RECOIL MECHANISM

140. There are three types of concentric recoil mechanism. All have the same basic hydraulic system, concentric with the gun tube but differ in spring arrangements. The first has one spring, concentric with the tube (Fig. **29**). When the diameter **of** the gun is large, it is impractical to use a concentric spring. More space is required to house it and the spring itself is difficult to manufacture to the prescribed specifications. Then, the second type, a multicylinder recoil mechanism, is used (Fig. **30**). Usually, four smaller spring assemblies, each consisting of two springs concentric with each other, are located 90" apart around the periphery of the system. This permits flattening the cylinder housing between springs, thus conserving space. The third or separate type has the counterrecoil mechanism completely divorced from the recoil brake (Fig. 31). It may be used if space is available for the spring assembly because it has two distinct advantages. The spring design is not dependent upon the dimensions of the recoil brake mechanism and there is freedom in handling the complete arrangement without disturbing the recoil brake.

C. RECOIL CALCULATIONS, CONCENTRIC TYPES

141. Recoil calculations are made either graphically or analytically. The graphical method is discussed for the concentric system. First an interior ballistics curve of pressure **vs.** time is obtained from the proper source. From this information, the propellant gas force is computed, added to the weight component of the recoiling parts and plotted against time on large scale paper (Fig. **32**). Simply by adjusting the scale by a factor, $1/M$ (from $F = Ma$, where M **is** the mass of the recoiling parts), the same curve becomes the acceleration-time curve. The retardation of the recoil mechanism is plotted as a deceleration **vs.** time curve and the resultant of the two becomes the net effective acceleration curve. From the beginning of recoil, the retarding force is made to vary linearly with respect to time from zero to a maximum at the time of maximum propellant gas pressure, where it becomes constant. This is an arbitrary practice and is not inviolable, but any deviation from it should

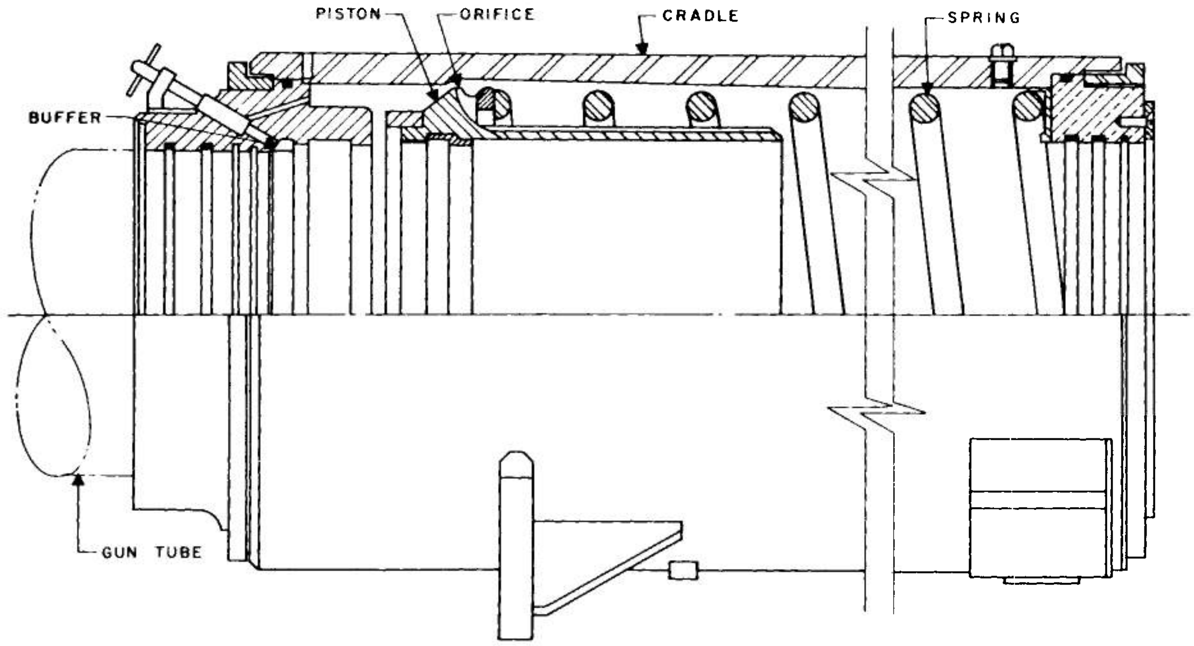
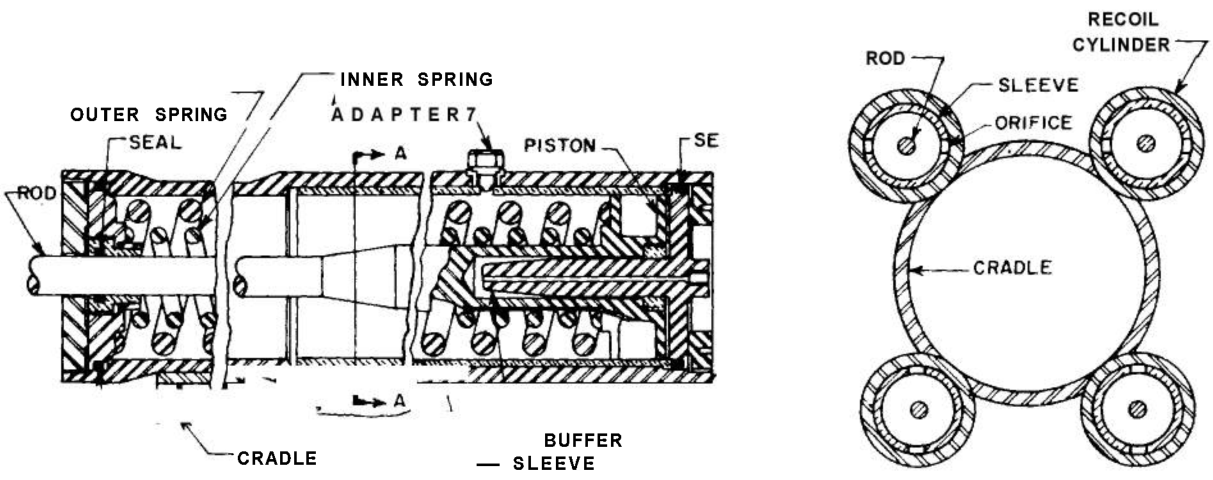


Figure 29. Concentric Recoil Mechanism (Concentric Spring Type)



longitudinal section of one recoil cylinder

Section at A – A (through complete unit)

Figure 30. Concentric Recoil Mechanism (Multiple Cylinder Type)

not disturb proper recoil motion (see Paragraph 117).

142. For the first trial, the retarding force may be obtained from Equation (3b). After the net acceleration curve is plotted, it is integrated by planimeter, by counting blocks, or other suitable means to obtain the velocity-time curve (Fig. **33).** Integrating the velocity-time curve yields velocity *vs.* distance which is plotted (Fig. 34). If the calculated distance differs from the specified recoil distance, the retarding force is adjusted and the process repeated. After some proficiency has been acquired, more than two trials are seldom necessary.

D. ORIFICE DESIGN, CONCENTRIC TYPES

143. The orifice area is determined by the

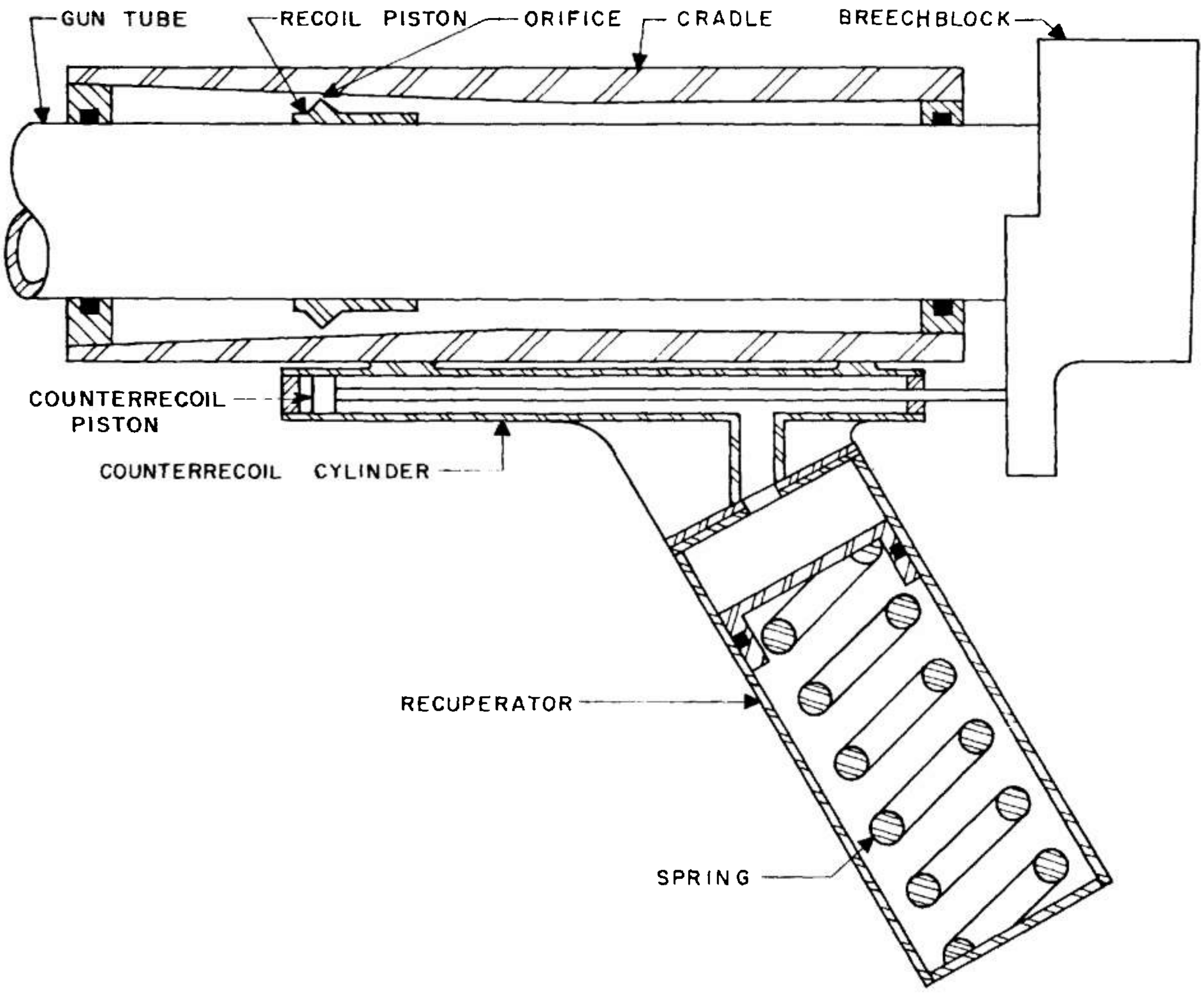


Figure 37. Concentric Recoil Mechanism with Separate Counterrecoil Assembly

method described in Paragraph 121 and based on a sharp-edged orifice with a discharge coefficient of 0.60. To assure the sharpedge effect, the lip of the piston adjacent to the orifice should not exceed 1/16-inch in width. The orifice area may be regulated by varying the inside diameter of the cylinder or by cutting grooves of varying width or depth in the inner wall. Care must be taken with the former method as accumulated tolerances on piston and cylinder diameters may result in annular areas which are much too large. Hydraulic systems now in use operate at pressures from 1000 to 5000 psi, depending on individual requirements. O-rings or T-rings are used for seals. In most cases, spring resistance is small and is not considered in computing recoil distance. However, this is a matter of judgment and it may sometimes be advisable to include the spring force, in which instance the hydraulic resistance will be reduced accordingly.

E. SPRING DESIGN, CONCENTRIC TYPES

144. Springs are designed by conventional formulas and are stressed to the elastic limit. Dynamic stresses are not considered because a life of 100,000 cycles is satisfactory and, in practice, springs so designed usually exceed this number. Buckling of springs must be prevented, either inherently in the design of the spring itself or by providing guides to insure lateral stability. Sometimes as many as 15 or 20 springs, with variations in wire diameter, coil size, and number of coils, may be investigated before one is selected. Each end should be provided with an inactive three-

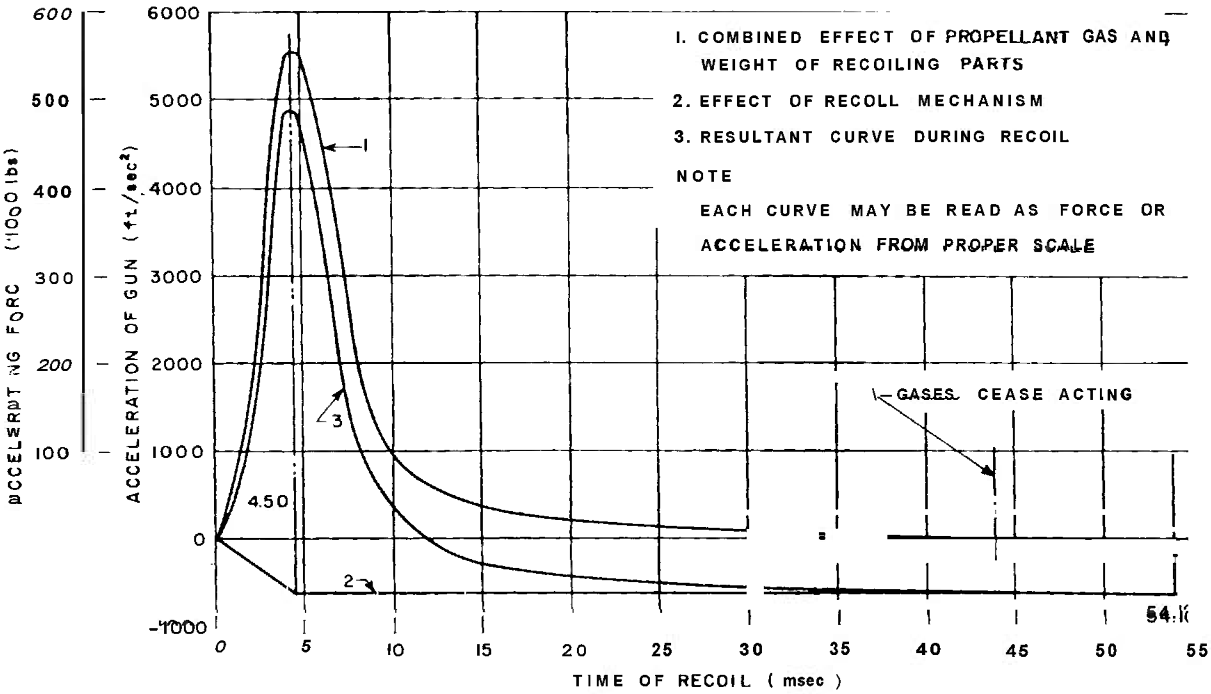


Figure 32. Force-Time and Acceleration-Time Curves

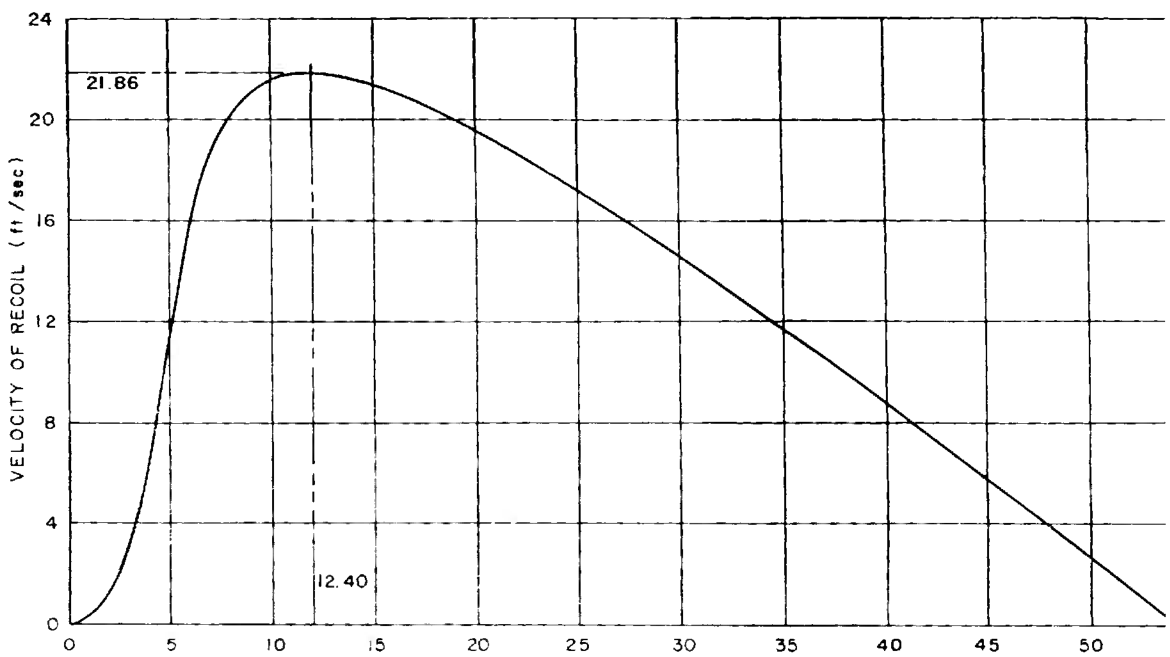


Figure 33. Velocity-Time Curve of Recoil

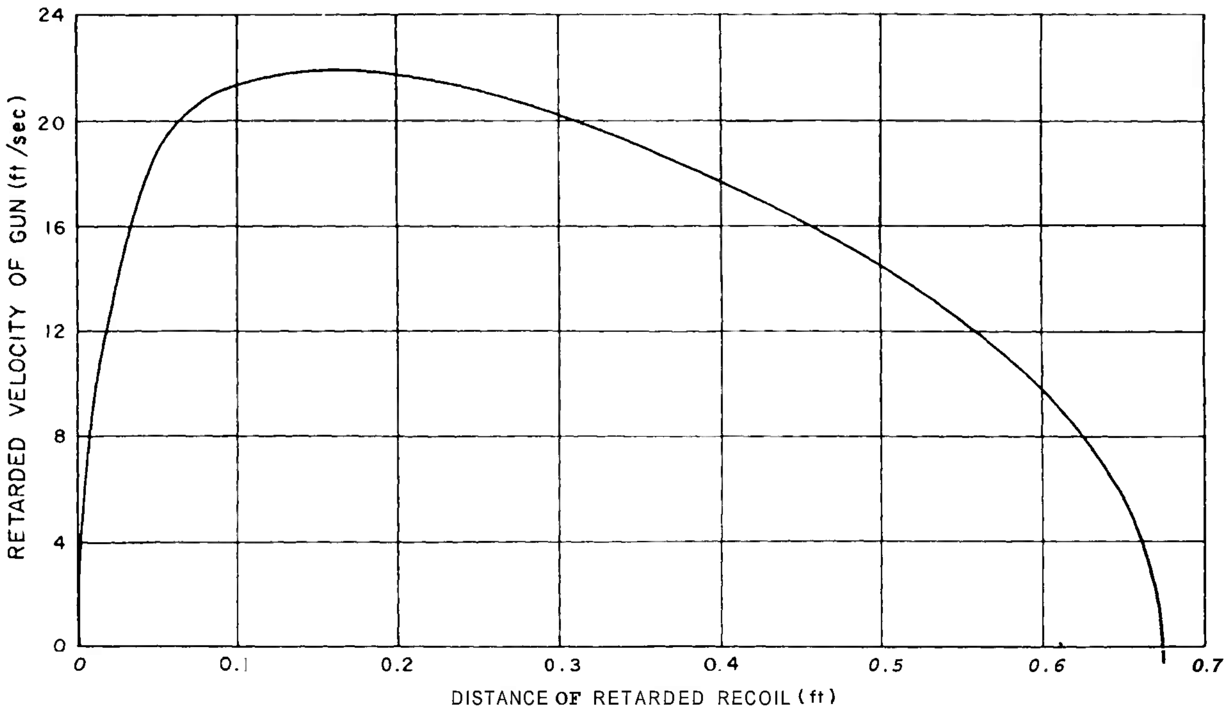


Figure 34. Velocity-Distance Curve of Recoil

quarter coil. The spring is never permitted to reach solid height at maximum recoil, but should have approximately ⅛-inch between coils so that hydraulic fluid may flow freely between them. Special, noncommercial manufacturing and inspection procedures are used. Tolerances are severe. Ground stock is always used, generally heat-treated and shot peened.

145. If guns are to be fired at high angles of elevation, up to or approaching 90°, the installed, or in-battery, spring-load should be 130 percent of the weight of recoiling parts. Where high elevations are not required, as in tank guns, this figure should be 100 percent, if possible. Sometimes, particularly in tanks, space limitations may dictate still lower loads, down o the equivalent of 45° elevation (see Paragraph 39).

XI. DESIGN OF DOUH .E RECOIL SYSTEMS*

A. INTRODUCTION

146. A double recoil gun has two separate recoil systems, one designated the primary and the other the secondary recoil system (see Fig. 35). There are certain advantages to this type of gun over one with a single recoil system, particularly in large calibers where a long recoil is required. For a limited space, a larger recoil distance results thus reducing the horizontal ground reaction as well as providing greater stability of carriage in or out of battery. It also permits a lower silhouette and shorter guides or rails.

147. The primary recoiling parts consist of gun tube, breech, either guides or sleigh, and certain parts of the primary recoil mechanism that are not fixed to the secondary system. It recoils on the guides in the cradle of the top carriage and is included among the elevating parts. The secondary recoiling parts consist of top carriage, cradle, part of the primary recoil mechanism that does not recoil with the primary, and that part of the secondary recoil mechanism which is not fixed to the bottom carriage; in fact, everything that recoils upon the bottom carriage, exclusive of the primary recoiling system. The secondary guides on the bottom carriage are fixed in a horizontal position. A turntable beneath, and connected to the bottom carriage, allows for traverse of the equipment.

* The theoretical approach is based on original work done by Dr. Kupen Eksergian of The Franklin Institute.

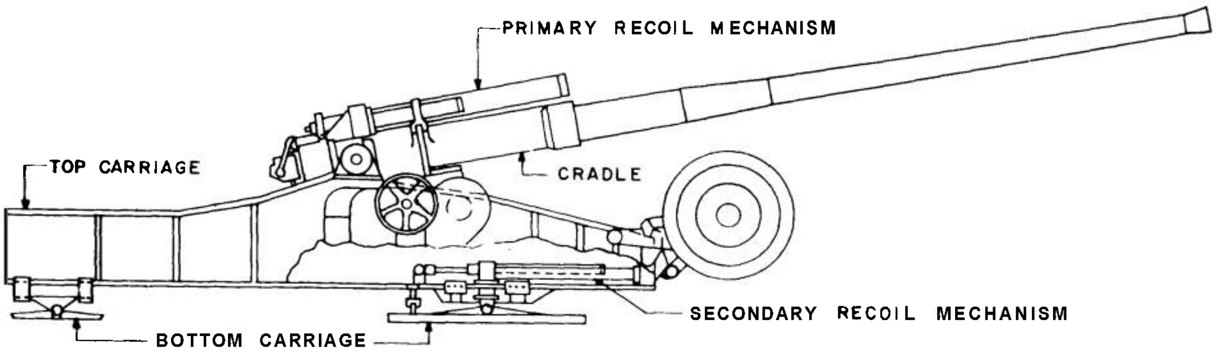


Figure 35. Gun With Double Recoil Mechanism

B. RECOIL FORCES

148. In a gun with a double recoil system, there are two recoil forces, one between primary and secondary recoiling parts, known as the primary recoil force, and one between secondary recoiling parts and bottom carriage, known as the secondary recoil force. The primary recoil force, denoted by K, consists of three components: the hydraulic resistance to recoil in the recoil cylinder, which is the principal resistance to recoil; the force due to pressure in the recuperator cylinder; and the force due to friction on the primary guides. The secondary recoil force, denoted by R, also consists of a hydraulic force, a recuperator force, and a friction force.

C. PROCEDURE FOR DYNAMIC ANALYSIS

149. The final calculations for the recoil action of a double recoil system are long and laborious. In order to reduce the work to a minimum, several preliminary solutions are initially made. These usually enable the final step-by-step integration of the equations of motions of primary and secondary recoiling masses to be performed but once or twice, effecting a considerable saving in time and labor. The complete procedure is summarized below followed by a detailed description of each step.

Step 1. Assume a primary and secondary recoil distance.

Step 2. Find the preliminary primary and secondary recoil forces using the assumed primary and secondary recoil distance.

Step 3. Determine accurately by trial and error the primary and secondary recoil forces while assuming that the forces are constant throughout the length of recoil.

Step 4. Determine the components of the primary and secondary recoil forces in order to design the various parts of the recoiling systems.

Step 5. Determine the variation of the relative velocity of the primary recoiling mass with the relative primary recoil distance and of the velocity of the secondary recoiling mass with the secondary recoil distance.

Step 6. Determine the primary and secondary forces in the recoil cylinders by a step-by-step integration of the equations of motion of the primary and secondary recoiling masses.

Step 7. Investigate counterrecoil.

1. *Nomenclature*

The following nomenclature applies to the double recoil calculations:

F_g = propellant gas pressure force
g = acceleration of gravity
K = primary recoil force
L_1 = primary recoiling distance relative to the secondary mass
L_2 = secondary recoil distance
L_m = total horizontal movement of center of gravity of m during recoil
m_1 = primary recoiling mass
m_2 = secondary recoiling mass
m = $m_1 + m_2$ = total recoiling mass
N_1 = normal reaction of primary recoiling parts
N_2 = normal reaction of secondary recoiling parts
R = secondary recoil force
s = displacement of center of gravity of m during recoil

At = increment of time
v_1 = velocity of primary recoiling mass, absolute
v_2 = velocity of secondary recoiling mass
v_{2i} = initial velocity of secondary recoiling mass
v_f = initial velocity of primary mass (velocity of free recoil)
v_m = horizontal velocity of center of gravity of m
v_{mi} = initial horizontal velocity of center of gravity of m
o, = velocity of primary recoiling mass relative to the secondary mass
v_{ri} = initial velocity of primary mass relative to secondary mass
W_1 = weight of primary mass
W_2 = weight of secondary mass
x_r = primary stroke relative to secondary stroke
θ = angle of elevation of gun measured from the horizontal.

2. *Detailed Discussion*

150. ***(Step 1) Recoil Distances.**** **A** primary and secondary recoil distance must be selected first. This selection is based upon size of equipment, space available for the recoil mechanisms, high or **low** silhouette, and other ordnance considerations. Once the recoil distances have been established, the problem resolves itself into making a design to meet these requirements. Full advantage should be taken of these recoil distances, for greater distances mean lower recoil forces and lighter equipment.

151. ***(Step** 2) **Preliminary Recoil Forces.*** With the primary and secondary recoil distance determined, preliminary primary and secondary recoil forces may be found from the expressions derived below. The formulas for the primary recoil force K and the secondary recoil force, R, are:

$$K = \frac{m_1 v_f^2}{2L_1}\left[1 - \frac{L_2 \cos^2\theta}{L_1 \cos\theta + L_2\left(\frac{m_1 + m_2}{m_1}\right)}\right] + W_1 \sin\theta; \tag{72}$$

and:

$$R = \frac{\frac{1}{2} m_1 (v_f \cos\theta)^2}{L_1 \cos\theta + L_2\left(\frac{m_1 + m_2}{m_1}\right)}. \tag{73}$$

The maximum velocity of free recoil is imparted instantaneously to the primary recoiling parts, i.e., the propellant gas forces, F_g, have dissipated. Figure **36** is a diagram of the preliminary forces of a double recoil system. To simplify the derivation for the preliminary values of the recoil forces, several assumptions are made: (1) the **primary** and secondary recoiling masses reach **zero** velocity simultaneously, (2) when the initial recoil velocity is imparted to the primary mass, the secondary velocity is still zero, and **(3)** in the derivation, K and R are constant.

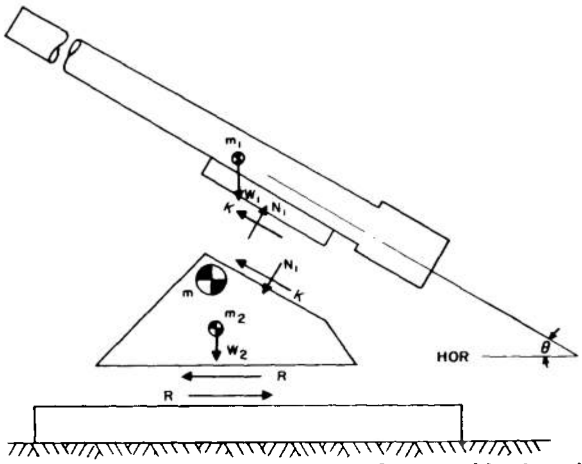


Figure 36. Preliminary Forces of a Double Recoil System

***Derivation*:**

$$m = m_1 + m_2. \tag{74}$$

From conservation of momentum:

$$m v_m = m_1 (v_r \cos\theta + v_2) + m_2 v_2; \tag{75}$$

the initial momentum:

$$m v_{mi} = m_1 (v_{ri} \cos\theta + v_{2i}) + m_2 v_{2i}; \tag{76}$$

but by assumption, v_{2i} is zero. Then,

$$m v_{mi} = m_1 v_{ri} \cos\theta. \tag{77}$$

Also, $$m L_m = m_1 (L_1 \cos\theta + L_2) + m_2 L_2, \tag{78}$$

and: $$m \frac{dv_m}{dt} = m_1\left(\cos\theta \frac{dv_r}{dt} + \frac{dv_2}{dt}\right) + m_2 \frac{dv_2}{dt}. \tag{79}$$

For the primary recoiling mass, from the law, $F = ma$:

$$m_1\left(\cos\theta \frac{dv_r}{dt} + \frac{dv_2}{dt}\right) = -K\cos\theta + N_1 \sin\theta; \tag{80}$$

* **Step numbers** of **sub-headings in Paragraphs 150 to 182 refer to steps** summarized **in Paragraph 149.**

and for the secondary recoiling mass:

$$m_2 \frac{dv_2}{dt} = + K\cos\theta - R - N_1 \sin\theta, \quad (81)$$

where N_1 **is** the normal reaction on the primary recoiling parts and also is the normal load applied **to** the **secondary** recoiling parts.

Now, adding Equations (80) and (81):

$$m_1\left(\cos\theta \frac{dv_r}{dt} + \frac{dv_2}{dt}\right) + m_2 \frac{dv_2}{dt} = -R; \quad (82)$$

and combining Equations (79) and (82):

$$m \frac{dv_m}{dt} = -R.$$

But:

$$m \frac{dv_m}{dt} = m \frac{dv_m}{ds} \cdot \frac{ds}{dt} = mv_m \frac{dv_m}{ds};$$

therefore,

$$mv_m \frac{dv_m}{ds} = -R;$$

and: $$\int_{v_{mi}}^{0} mv_m dv_m = -Rds;$$

and: $$\frac{-mv^2_{mi}}{2} = -RL_m;$$

from which:

$$R = \frac{mv^2_{mi}}{2L_m}.$$

Now, substituting in this equation the following values of m, v_{mi}, L_m,

$$m = m_1 + m_2, \quad (74)$$

$$v_{mi} = \frac{m_1 v_{ri} \cos\theta}{m_1 + m_2}, \quad (77)$$

$$L_m = \frac{m_1(L_1\cos\theta + L_2) + m_2L_2}{m_1 + m_2}, \quad (78)$$

there results:

$$R = \frac{(m_1 + m_2)\left(\frac{m_1 v_{ri} \cos\theta}{m_1 + m_2}\right)^2}{2\left[\frac{m_1(L_1\cos\theta + L_2) + m_2L_2}{m_1 + m_2}\right]} = \frac{\frac{1}{2} m_1^2 (v_{ri}\cos\theta)^2}{m_1(L_1\cos\theta + L_2) + m_2L_2}.$$

Substitute v_f for v_{ri}. Although $v_f = v_{ri} + v_{2i}$, the expression is written $v_f = v_{ri}$ because v_{2i} is assumed to be zero:

$$R = \frac{\frac{1}{2} m_1 (v_f \cos\theta)^2}{L_1\cos\theta + L_2\left(\frac{m_1 + m_2}{m_1}\right)}. \quad (73)$$

Solving for the primary recoil force, K, from the law of the conservation **of** energy, and assuming now that K as well as R is constant:

$$\tfrac{1}{2} m_1 v_f^2 = KL_1 + RL_2 - W_1L_1\sin\theta;$$

from which: $$K = \frac{\frac{1}{2} m_1 v_f^2 - RL_2 + W_1L_1\sin\theta}{L_1}$$

Substituting for R:

$$K = \frac{\frac{1}{2} m_1 v_f^2 - \left[\frac{\frac{1}{2} m_1 (v_f\cos\theta)^2 L_2}{L_1\cos\theta + L_2\left(\frac{m_1 + m_2}{m_1}\right)}\right]}{L_1} + W_1\sin\theta;$$

and:

$$K = \frac{m_1 v_f^2}{2L_1}\left[1 - \frac{L_2\cos^2\theta}{L_1\cos\theta + L_2\left(\frac{m_1 + m_2}{m_1}\right)}\right] + W_1\sin\theta. \quad (72)$$

152. (Step 3) Recoil Forces, Constant. While the primary and secondary recoil forces obtained in Paragraph 151 are preliminary, they yield a good starting point. These values will not give the proper distances because they are derived from assumptions which are not rigorously applicable. It is necessary that the forces be more nearly correct, which is accomplished by a trial and error method. The approach is described below, first for the primary recoiling mass and then for the secondary recoiling mass. Both recoil mechanisms are designed for their respective critical conditions, maximum elevation for the primary and zero elevation for the secondary.

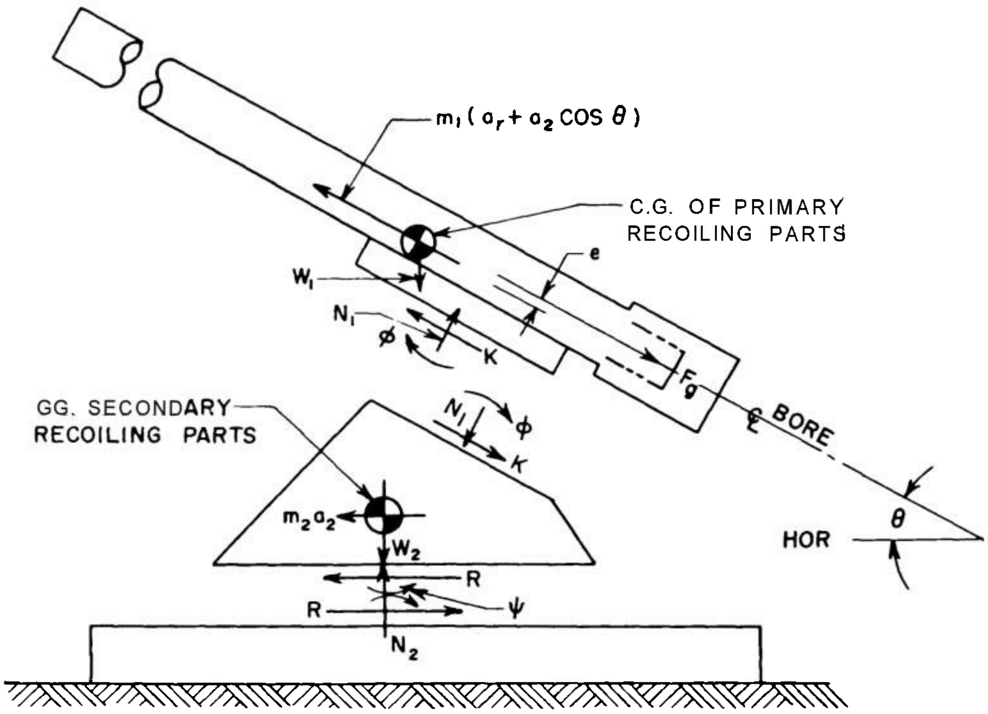
153. Calculations for the primary recoil distance are divided into two parts: (a) the distance traveled while propellant gas pressure is acting and (b) the distance traveled after the gases cease to act. In both cases, the preliminary values for K and R found in Paragraph 151 are used. As just mentioned, they will not immediately yield the correct recoil distances but, by adjusting their magnitudes several times and solving for the distances in each case, the trend toward the correct values can be found and proper values of K and R will soon be determined. It is also possible to adapt double recoil calculations to high **speed** computers. The dynamics of the primary and secondary recoiling masses of a gun with a double recoil action are presented in the following derivation.

154. Figure 37 shows the forces on a double recoil system. Also, refer to the schematic sketch in Figure 36 and the vector diagram in Figure 38. Substituting in the equation $F = ma$, there results for the force along the primary guides the equation,

$$m_1\left(\frac{dv_r}{dt} + \frac{dv_2}{dt}\cos\theta\right) = F_g - K + W_1 \sin\theta; \quad (83)$$

and for the force perpendicular to the primary guides, the equation,

$$m_1 \frac{dv_2}{dt}\sin\theta = N_1 - W_1\cos\theta. \quad \textit{(84)}$$



F_g = PROPELLANT GAS FORCE
K = PRIMARY RECOIL FORCE
R = SECONDARY RECOIL FORCE
N_1 = NORMAL REACTION OF PRIMARY PARTS
N_2 = NORMAL REACTION OF SECONDARY PARTS
W_1 = WEIGHT OF PRIMARY PARTS
W_2 = WEIGHT OF SECONDARY PARTS
$m_1(a_r + a_2 \cos\theta)$ = AXIAL INERTIA FORCE PRIMARY PARTS
$m_2 a_2$ = INERTIA FORCE SECONDARY PARTS
ϕ = TURNING MOMENT OF PRIMARY PARTS
ψ = TURNING MOMENT OF SECONDARY PARTS
θ = ANGLE OF ELEVATION

Figure 37. Forces on a Double Recoil System

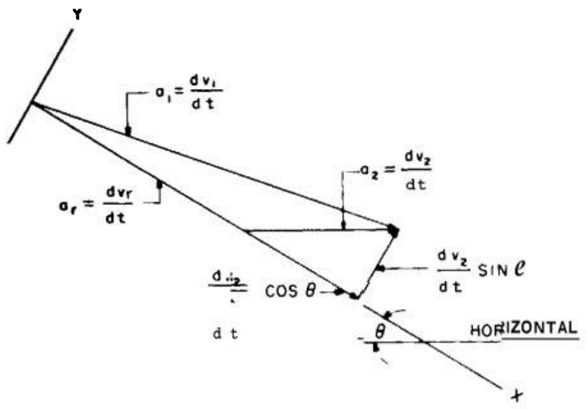


figure 38. ***Acceleration Diagram of a Double Recoil System***

For **the** secondary recoiling mass, the equation of force is:

$$m_2 \frac{dv_2}{dt} = K \cos\theta - N_1 \sin\theta - R. \qquad (85)$$

Now, from Equation (84):

$$N_1 = m_1 \frac{dv_2}{dt} \sin\theta + W_1 \cos\theta.$$

Substituting this value of N_1 in Equation (85) there results:

$$m_2 \frac{dv_2}{dt} = K \cos\theta - m_1 \frac{dv_2}{dt} \sin^2\theta - W_1 \cos\theta \sin\theta - R;$$

transposing:

$$\left(m_2 + m_1 \sin^2\theta\right) \frac{dv_2}{dt} = K \cos\theta - W_1 \cos\theta \sin\theta - R. \qquad (86)$$

Now, to find the relative acceleration of the primary recoiling mass, from Equation (83):

$$m_1 \frac{dv_r}{dt} = F_g - K + W_1 \sin\theta - m_1 \frac{dv_2}{dt} \cos\theta; \quad (87)$$

from Equation (86):

$$\frac{dv_2}{dt} = \frac{K \cos\theta - W_1 \cos\theta \sin\theta - R}{m_2 + m_1 \sin^2\theta}. \qquad (88)$$

Then, substituting the value of dv_2/dt obtained in Equation (88) into Equation (87), there results:

$$m_1 \frac{dv_r}{dt} = F_g - K + W_1 \sin\theta - \left(\frac{K \cos^2\theta - W_1 \cos^2\theta \sin\theta - \cos\theta}{m_2 + m_1 \sin^2\theta}\right) m_1.$$

Dividing by m_1:

$$\frac{dv_r}{dt} = \frac{F_g - K + W_1 \sin\theta}{m_1} - \frac{K \cos^2\theta - W_1 \cos^2\theta \sin\theta - R \cos\theta}{m_2 + m_1 \sin^2\theta}. \qquad (89)$$

155. ***(Part a).*** If an increment of time and the corresponding increment of velocity are known, the product of the two will be the increment of distance traveled during that time. Therefore a convenient increment of time, Δt, is assumed and the corresponding increment of relative velocity, Δv_r, is calculated from Equation (89):

$$\Delta v_r = \left(\frac{F_g + W_1 \sin\theta - K}{m_1}\right) \Delta t - \left(\frac{K \cos^2\theta - W_1 \sin\theta \cos^2\theta - R \cos\theta}{m_2 + m_1 \sin^2\theta}\right) \Delta t. \quad (90)$$

The increment of distance, Δx, is found by multiplying the time by the corresponding velocity.

$t \;= \Sigma \Delta t$,
$v_r \;= \Sigma \Delta v_r$,
$\Delta x_r \;= v_r \Delta t$, the increment of distance during acceleration,
$x_r \;= \Sigma \Delta x_r$.

These calculations may be conveniently tabulated as shown in Table 1.

TABLE 1

Δt	t $(\Sigma \Delta t)$	F_g	Δv_r	v_r $(\Sigma \Delta v_r)$	Δx_r	x_r $(\Sigma \Delta x_r)$

156. ***(Part b).*** To calculate the distance traveled after the propellant pressure ceases, it is necessary to find acceleration and time for the remainder of the primary recoil stroke. With the propellant gas force, F_g, equalling zero, Equation (89) becomes:

$$\frac{dv_r}{dt} = a_r = \frac{W_1 \sin\theta - K}{m_1} - \frac{K \cos^2\theta - W_1 \sin\theta \cos^2\theta - R \cos\theta}{m_2 + m_1 \sin^2\theta}, \quad (89a)$$

and since all terms are constants, a_r is also constant, and negative, indicating a constant deceleration. The total time of deceleration is:

$$t_b = \frac{v_{r_1}}{a_r}. \tag{91}$$

where v_{r_1} is the relative primary velocity when propellant pressure ceases. The primary recoil stroke during deceleration is:

$$x_{r_2} = \tfrac{1}{2} a_r t_b^2. \tag{91a}$$

Therefore, the total relative primary recoil distance traveled during recoil will be the sum of that traveled during the propellant pressure period and that traveled after the propellant pressure period.

157. In order to determine the secondary recoil distance, the calculations are divided into three parts, (a) the distance traveled until the time when the relative primary velocity,* v_r, becomes zero, (b) the distance traveled from the time the relative primary velocity becomes zero until it reaches a predetermined maximum counterrecoil velocity,? and (c) the distance traveled from the time that the relative primary velocity attains the predetermined counterrecoil velocity to the time that the secondary velocity becomes zero. As before, the preliminary values for K and R are used.

158. (Part a). Calculate the distance traveled x_{2_a} during the time, t, of primary recoil. From Equation (88):

$$a_{2_a} = \frac{K\cos\theta - W_1 \sin\theta\cos\theta - R}{m_2 + m_1 \sin^2\theta},$$

$$x_{2_a} = \tfrac{1}{2} a_{2_a} t^2, \tag{91b}$$

$$v_{2_a} = a_{2_a} t. \tag{91c}$$

159. (Part b). Calculate the distance traveled from the end of primary recoil until the primary mass attains its specified maximum counterrecoil velocity v_{r_n}. From Equation (89a), the relative primary counterrecoil acceleration is:

$$a_r = \frac{W_1 \sin\theta - K_1}{m_1} - \frac{K_1\cos^2\theta - W_1 \sin\theta\cos^2\theta - R\cos\theta}{m_2 + m_1 \sin^2\theta}$$

In this case, K_1 is the primary counterrecoil force of the recuperator less friction forces. The time required for the primary mass to reach the specified velocity is:

$$t_n = \frac{v_{r_n}}{a_r}. \tag{91d}$$

And from Equation (88), the acceleration of the secondary recoiling mass is:

$$a_{2_b} = \frac{K_1\cos\theta - W_1 \sin\theta\cos\theta - R}{m_2 + m_1 \sin^2\theta}$$

The distance traveled by the secondary mass is:

$$x_{2_b} = v_{2_a} t_n + \tfrac{1}{2} a_{2_b} t_n^2. \tag{91e}$$

160. (Part c). Calculate the distance traveled from the time of maximum primary counterrecoil velocity, v_{r_n}, to the time that the secondary velocity becomes zero. The secondary velocity at the end of t, becomes:

$$v_{2_b} = v_{2_a} + a_{2_b} t_n. \tag{91f}$$

From Equation (88) the secondary acceleration becomes:

$$a_{2_c} = \frac{K\cos\theta - W_1 \sin\theta\cos\theta - R}{m_2 + m_1 \sin^2\theta},$$

the time is:

$$t_{2_c} = \frac{v_{2_b}}{a_{2_c}}; \tag{91g}$$

and the distance is:

$$x_{2_c} = v_{2_b} t_{2_c} + \tfrac{1}{2} a_{2_c} t_{2_c}^2. \tag{91h}$$

The total secondary recoil distance is the sum of the above three distances. Thus:

$$x_2 = x_{2_a} + x_{2_b} + x_{2_c}. \tag{92}$$

161. (Step 4) Components of the Recoil Forces. The foregoing calculations provide for the determination of the total primary and secondary recoil forces for the given recoil distance. However, in order to design the recoil and counterrecoil mechanisms and the recuperators, it is necessary to know the values of the components of the total recoil forces; i.e., how much resistance to recoil the recoil cylinder offers, how much the recuperator offers, and how much friction offers. Stated in equation form, the primary recoil force is:

$$K = K_{h_s} + K_a + K_f, \tag{93}$$

and the secondary recoil force is

*** The velocity of the primary recoiling mass relative to the secondary recoiling mass is referred to as the relative primary velocity, for brevity. Similarly, the velocity of the secondary recoiling mass is referred to as the secondary velocity.**

† This assumes that there will be a buffing device provided to curtail the counterrecoil motion.

$$R = R_{h_s} + R_a + R_f, \qquad (94)$$

where:

K_{h_s} = hydraulic resistance in primary system,
K_a = recuperator resistance in primary system,
K_f = friction resistance in primary system,
R_{h_s} = hydraulic resistance in secondary system,
R_a = recuperator resistance in secondary system,
R_f = frictional resistance in secondary system.

Now K and R have been calculated and the K_a and R_a forces throughout the length of recoil may be calculated from the formula for polytropic compression :

$$P_0 V_0^n = P_x V_x^n;$$

but:

$$V_x = V_0 - Ax,$$

$$P_x = P_0\left(\frac{V_0}{V_0 - Ax}\right)^n, \qquad (95)$$

and the recuperator resistance at any recoil distance is:

$$K_{ax} = P_x A,$$

where: A = effective recoil piston area,
n = polytropic exponent,
P_0 = recuperator gas pressure in battery,
P_x = recuperator gas pressure at x,
V_0 = gas volume in recuperator in battery,
V_x = gas volume in recuperator at x,
x = any recoil distance.

162. The recuperator gas pressure for the primary system is found in the same way as for single recoil systems. Rewrite Equation **(4)** to associate it with a double recoil system.

$$K_{a1} = \lambda_1 (W_1 \sin\theta + \mu W_1 \cos\theta + f_p). \qquad (96)$$

Refer to Chapter VIII, Part G for the discussion on the method for determining the primary recuperator characteristics. The characteristics of the secondary recuperator can be determined in the same way except for the initial force in battery which is:

$$R_{a2} = \lambda_2 (W_1 + W_2), \qquad (97)$$

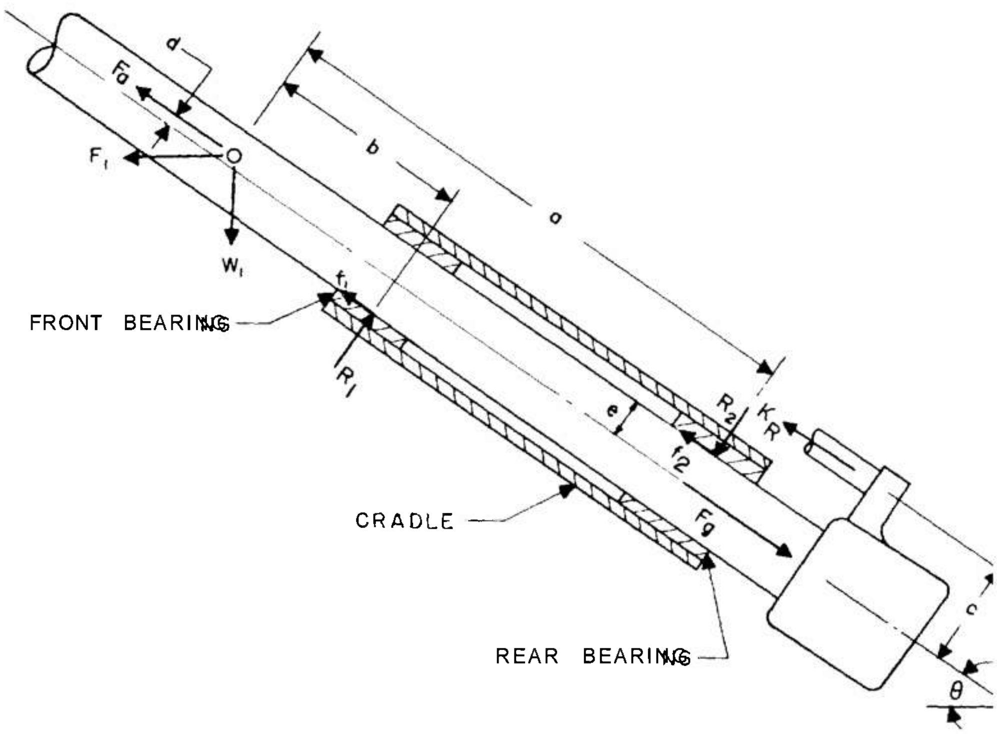
where, based on previous experience, $\lambda_2 = 0.25$.

163. The frictional forces resisting both primary and secondary recoil are determined from the forces normal *to* the guides. In the case of the secondary guides, the total frictional force is readily determined because these guides are always horizontal, and the sum of normal forces is the static weight of the mass resting on them plus the vertical component of the K force. But for the primary guides, the normal forces are influenced by several contributing factors, all of which are not so simply evaluated. One is the normal component of the static weight of the primary rec'oiling mass; another is the dynamic forces due to the accelerations of the primary and secondary recoiling masses; and a third factor which develops because the accelerating forces are located so that they produce a moment about the cradle guides or bearings and thus may increase the normal reactions. Figure 39 shows these forces and reactions. Pamphlet No. ORDP 20-341 on Cradles discusses in detail how the reactions are obtained. If R_1 and R_2 act in the same direction, their sum equals the sum of the forces parallel to them. However, if R_1 and R_2 are opposite in direction, their arithmetical sum is greater than the sum of the normal components of the forces, thus increasing the frictional forces.

164. There will always be a normal component of the static weight of the primary recoiling parts which will produce friction on the guides. The dynamic force on the primary guides resulting from the acceleration of secondary recoil varies with the angle of elevation simply because, at high angles of elevation, the normal component of the dynamic force due to secondary acceleration is a larger part of the total. Experience has shown that, at high angles of elevation, the secondary accelerations may be as large as those at low angles, but occur for a much shorter time interval.

165. Theoretically this dynamic force should be considered when designing the primary recoil cylinder but, from a practical viewpoint, it is better to omit it. If it were included, the design hydraulic brake force would become smaller because a greater proportion of the total retarding force would be composed of friction. Then, if the recoil cylinder were designed to this reduced hydraulic force and the frictional force failed to be realized, there would be insufficient hydraulic recoil resistance over most of the stroke. Recoil velocity would exceed specified limits; there would be a greatly increased hydraulic force toward the end of the stroke, and the recoil distance might easily become inadequate, perhaps to the extent of causing damage.

166. On the other hand, if the friction force of the dynamic load is excluded from the calculations but does occur in reality, no great harm is done. It



F_a = INERTIA FORCE DUE TO PRIMARY ACCELERATION

F_g = PROPELLANT GAS FORCE

F_1 = INERTIA FORCE DUE TO SECONDARY ACCELERATION

f_1 = FRICTIONAL FORCE ON FRONT CRADLE BEARING

f_2 = FRICTIONAL FORCE ON REAR CRADLE BEARING

K_R = RECOIL ROD FORCE

R_1 = NORMAL REACTION ON FRONT CRADLE BEARING

R_2 = NORMAL REACTION ON REAR CRADLE BEARING

W_1 = WEIGHT OF PRIMARY RECOILING PARTS

θ = ANGLE OF ELEVATION

Figure 39. Applied Loads and Reactions on Cradle

simply means a shorter recoil distance, because the recoil resistance is greater than calculated leading to a conservative design.

167. Another source of friction which may be figured to be very large is the moment on the primary guides mentioned in Paragraph 163 (see ϕ in Fig. 37). In the initial stages of design, however, this moment cannot be accurately calculated because of the vagueness of the weights and centers of gravity of the various parts. Even in final design it is difficult to determine closely, due to the uncertainty of the friction coefficient, which may be 0.1, 0.2, or even 0.3. Not too much dependence, then, should be placed upon friction as a means of resisting recoil. So, for these reasons, and because neglecting it is conservative, friction caused by the moment on the primary guides is omitted from the computations. For primary recoil calculations, the only guide friction considered will be that caused by the normal components of static weight

of the primary recoiling parts. The coefficient of friction will be assumed as 0.15 and the resulting friction is the K_f force.

168. The friction of the secondary guides has been separated into two parts for ease of computation. That which is due to the static weight resting on the guides is designated as R_{f_1}, and the friction arising from the vertical component of K, the primary recoil force, is called R_{f_2}. Again a coefficient of friction of 0.15 is used.

169. Now, at any point during recoil, the forces K, R, K_a, K_f, R_{f_1} and R_{f_2} are known and therefore it is possible to calculate K_h at any point. From these data, curves are drawn indicating the variation of hydraulic, gas pressure, and friction forces with recoil distance for the primary and secondary recoil systems.

170. *(Step* **5***) Variation of Recoil Velocities with Recoil Distances.* The previous paragraphs explain the derivation of hydraulic forces in the recoil cylinders for any recoil distance. In order to design the recoil cylinders, it is also necessary to have the variation of primary relative velocity with primary relative recoil distance, of secondary velocity with primary relative distance, and of secondary recoil velocity with secondary recoil distance. These are calculated as follows: The variation of velocity of the primary recoiling mass with distance has been calculated through the end of the propellant gas period (see Paragraphs 154 and 155). After the propellant gases cease to act and until the relative primary recoil velocity is zero, the following formula is used to determine the primary velocity at any distance of recoil:

$$v_s = \sqrt{v_{r_1}^2 + 2a_r x_p}, \qquad (98)$$

where:

v_{r_1} = the relative primary velocity at the end of the propellant period,

a_r = the acceleration; if a deceleration, the sign before it must be minus,

x_p = any distance from the position at the end of the propellant period to the end of recoil.

171. For the variation of velocity of the secondary recoiling mass, the calculations are divided into three steps in accordance with the three divisions previously established for the determination **of** the secondary recoil distance (see Paragraphs 158, 159, and 160). The three steps are listed below with the formula to be used for each.

a. During the period of primary recoil,

$$v_{2r_a} = \sqrt{2\, a_{2_a}\, x_{2_a}}; \qquad (99)$$

b. From the instant that the primary relative recoil velocity equals zero until the primary recoiling mass is reaccelerated to a constant maximum velocity (in counterrecoil):

$$v_{2r_b} = \sqrt{v_{2_a}^2 + 2\, a_{2_b}\, x_{2_b}}; \qquad (100)$$

c. From the instant that the primary counterrecoil velocity attains the constant value until the secondary velocity equals zero:

$$v_{2r_c} = \sqrt{v_{2_b}^2 + 2\, a_{2_c}\, x_{2_c}}, \qquad (101)$$

where in each of the expressions the letters v_{2r}, a_2, and x_2 pertain to velocity, acceleration, and distance of the secondary recoil. Due regard for positive and negative signs must be exercised.

172. From these data, curves are drawn, one showing the variation of the relative primary recoil velocity with the relative primary recoil distance and one showing the variation of the secondary recoil velocity with the secondary recoil distance. For convenience in the design of the recoil rods, these curves may be drawn on the same graphs which show the variation of recoil forces with recoil distances (explained in Step **4**, Paragraph 161). With these sets of curves, then, the primary recoil orifice may be determined by substituting the appropriate values in Equation (36):

$$a_o = \frac{v_r}{C_o}\sqrt{\frac{\rho A^3}{2 K_h}}, \qquad (102)$$

where:

a_o = orifice area,

C_o = orifice constant,

A = effective piston head area in the recoil cylinder,

ρ = mass density of hydraulic fluid,

K_h = hydraulic force,

v_r = velocity of recoil.

The **orifice** area for the secondary recoil mechanism **is** calculated similarly.

173. ***(Step*** 6) *Recoil Forces, Variable.* In the design of the recoil control rods, it was first assumed that the recoil forces wkre constant, which **is** not true; in **fact,** it is impossible. **Peaks** will occur at points along the travels of both primary and secondary recoil masses, particularly near the end of the recoil stroke. It is now necessary to ascertain the magnitude of these peaks and check

the recoil rods with due consideration of the peaking. The primary constant recoil force was obtained for its critical condition, maximum gun elevation. Then, using primary constant force, the secondary constant recoil force is computed for its critical condition, zero degrees gun elevation. The control rods are tentatively designed to satisfy these conditions. To check both recoil rod designs for excessive peaking, the two equations expressing the acceleration of primary and secondary recoiling masses are used. From Equation (89),

$$dv_r = \left(\frac{F_g - K + W_1 \sin\theta}{m}\right) dt - \left(\frac{K\cos^2\theta - W_1\cos^2\theta\sin\theta - R\cos\theta}{m_2 + m_1\sin^2\theta}\right) dt; \tag{103}$$

and from Equation (88),

$$dv_2 = \left(\frac{K\cos\theta - W_1\cos\theta\sin\theta - R}{m_2 + m_1\sin^2\theta}\right) dt. \tag{104}$$

These equations are organized in tabular form in Table 2, Paragraph 174, and are integrated step-by-step. K and R forces are divided into their respective components, as shown in the table. These two computations must be made at zero degrees and maximum gun elevation. If, upon completion, the calculations disclose that recoil force peaks too high near the end of recoil, indicating the need for more recoil distance, the recoil rod or rods must be redesigned. If this be the case, the entire procedure must be repeated from the beginning. The precise method for carrying out the step-by-step integration is given below. The table headings are shown in Paragraph 174 and each heading is explained in detail in Paragraph 176.

174. The form for stepby-step integration of equations of motion of double recoil action for the determination of recoil force is shown in Table 2.

TABLE 2

1	2	3	4	5	6	7
Δt	Δv_r	v_r	Δx_r	x_r	v_s	$\left(\frac{v_r}{v_s}\right)^2$
sec	in/sec	in/sec	in.	in.	in/sec	
8	9	10	11	12	13	14
K_{h_s}	K_h	K_a	K_f	K	F_g	Δv_2
lb	$K_{h_s}\left(\frac{v_r}{v_s}\right)^2$	lb	lb	$K_h + K_a + K_f$	lb	in/sec
15	16	17	18	19	20	21
v_2	Δx_2	x_2	v_{2r}	$\left(\frac{v_2}{v_{2r}}\right)^2$	R_h	R_h
in/sec	in.	in.	in/sec		lb	$R_{h_s}\left(\frac{v_2}{v_{2r}}\right)^2$
22	23	24	25	26	27	28
R_a	R_{f_1}	R_{f_2}	R	$k_1 F_g$	$k_2 K$	$k_3 R$
		$\mu K \sin\theta$	$R_h + R_a + R_{f_1} + R_{f_2}$			
lb	lb	lb	lb			

TABLE 2 (Cont.)

29	30	31	32	33	34
k_4	Δv_r	$k_3 K$	$k_5 R$	k_6	Δv_2
	$k_1 F_g + k_2 K + k_3 R + k_4$				$k_3 K + k_5 R + k_6$

175. Definitions of Symbols in Table 2:

v_r = relative velocity of primary recoiling mass with respect to the secondary recoiling mass,
v_2 = velocity of secondary recoiling mass,
x_r = relative recoil distance of primary recoiling mass with respect to the secondary recoiling mass,
x_2 = recoil distance of secondary mass,
v_s = recoil velocity with constant recoil force,
K_{h_s} = hydraulic force in primary recoil cylinder with the recoil forces K and R assumed constant,
K_h = hydraulic force in primary recoil cylinder with the recoil forces K and R variable,
K_a = primary recuperator force,
K_f = friction force on primary guides from static weight only,
R_a = secondary recuperator force,
R_{f_1} = friction force on secondary guides from static weight only,
R_{f_2} = friction force on secondary guides produced by the primary recoil force, K,
R_h = hydraulic force in the secondary recoil cylinder with the recoil forces K and R variable,
R_{h_s} = hydraulic force in secondary recoil cylinder with the recoil forces K and R constant,
μ = coefficient of friction,
e = angle of gun elevation measured from the horizontal,

$$k_1 = \frac{1}{m_1} \Delta t,$$

$$k_2 = -\frac{m_1 + m_2}{m_1 (m_2 + m_1 \sin^2 \theta)} \Delta t,$$

$$k_3 = \frac{\cos \theta}{m_2 + m_1 \sin^2 \theta} \Delta t,$$

$$k_4 = \frac{(m_1 + m_2) g \sin \theta}{m_2 + m_1 \sin^2 \theta} \Delta t,$$

$$k_5 = -\frac{1}{m_2 + m_1 \sin^2 \theta} \Delta t,$$

$$k_6 = -\frac{W_1 \sin \theta \cos \theta}{m_2 + m_1 \sin^2 \theta} \Delta t.$$

176. Following are explanations of the column headings of Table 2 according to the numbered sequence of columns:

1. Time is divided into intervals that are arbitrarily selected to fulfill a compromise. The smaller the interval, Δt, the more accurate the calculation but the greater the labor. A good approach is to assume small increments of about 0.001 to 0.004 second during the propellant pressure period, where forces are large and changing rapidly. After the propellant period, larger intervals, 0.020 to 0.040 second, may be used. However, near the end of recoil, it is good practice to again take small increments in order not to miss the peaks which may build up very suddenly.
2. Δv_r is the estimated change in primary relative recoil velocity during Δt and is later to be checked against Column 30.
3. v_r is the primary relative recoil velocity and is the summation of Δv_r.
4. Δx_r is the increment of displacement during Δt, of the primary recoiling mass relative to the secondary recoiling mass and equals $v_r \Delta t$.
5. x_r is the displacement of the primary recoiling mass relative to the secondary recoiling mass and is the summation of Δx_r.
6. v_s is the velocity of the primary recoiling mass, which is obtained from the curve drawn by assuming the primary and secondary recoil forces, K and R, to be constant (see Paragraphs 154 to 172).
7. $(v_r/v_s)^2$ is the ratio used to convert the primary recoil force, K, from a constant to a variable to conform with the hydraulic force, K_h. This ratio is applied at each increment of time throughout the length of recoil. The piston rod force of a recoil cylinder is:

$$K_h = \left(\frac{\rho}{2C_o^2}\right)\left(\frac{A^3}{a_o^2}\right)\left(v_r^2\right).$$

(102 rearranged)

See Paragraph 172 for symbols. From this equation it is evident that the force is directly proportional to the square of the relative primary recoil velocity which is the same as the relative primary piston rod velocity. Since no dimensions or constants have been altered, the only term in the expression that changes is the velocity, v_r; hence the ratio $(v_r/v_s)^2$.

8. K_{h_s} is the hydraulic force acting on the primary recoil piston head, obtained from the curve drawn with K and R assumed constant (see Paragraph 169).
9. K_h is the hydraulic force acting on the same piston head with variables K and R, and is obtained by multiplying K_{h_s} by the ratio $(v_r/v_s)^2$.
10. K_a is the primary recuperator force and is taken from the graph drawn according to instructions in Paragraph 169.
11. K_f is the friction force on the primary guides assumed as the product of the weight component normal to the guides and the coefficient, μ. Assume μ equal to 0.15.
12. K is the total primary recoiling force consisting of $K_h + K_a + K_f$ where each force is as explained above.
13. F_g is the force acting on the projectile due to the propellant pressure and equals the pressure multiplied by the bore area. The pressure is obtained from the ballistic curve of propellant pressure plotted against time.
14. Δv_2 is the increment of velocity of the secondary recoiling mass and is obtained from Column 34.
15. v_2 is the velocity of the secondary recoiling mass and is the summation of Δv_2.
16. Δx_2 is the increment of displacement of the secondary recoiling mass and equals $v_2 \, At$.
17. x_2 is the displacement of the secondary recoiling mass and is the summation or Δx_2.
18. v_{2_r} is the velocity of the secondary recoiling mass obtained from the curve drawn by assuming the recoil forces, K and R, constant (see Paragraphs 171 and 172).
19. $(v_2/v_{2_r})^2$. See description of Column 7.
20. R_{h_s} is the hydraulic force acting on the secondary recoil piston head obtained from the curve drawn with K and R assumed constant.
21. R_h is the secondary recoil piston rod force with varying forces, K and R, and is the product of R_{h_s} and $(v_2/v_{2_r})^2$.
22. R_a is the secondary recuperator force, from the curve drawn according to Paragraph 169.
23. R_{f_1} is the friction force on the secondary guides due to the weight only and equals the coefficient of friction, μ, multiplied by total static weight normal to the secondary guides (this includes primary and secondary masses). μ is assumed as 0.15.
24. R_{f_2} is the force due to friction produced by the primary recoil force, K, and equals μK sine, where μ is assumed as 0.15.
25. R is the total secondary recoil force consisting of the secondary R_h, the secondary R_a, and the secondary R_{f_1} and R_{f_2}.

26, 27, 28, and 29. These are the different components of Av, simplified so as to aid in the computation. The various constants have been defined in Paragraph 175.

30. $\Delta v_r = k_1 F_g + k_2 K + k_3 R + k_4$. This is the increment of the relative primary recoil velocity. It represents in condensed form Equation (103):

$$\Delta v_r = \left(\frac{F_g - K + W_1 \sin\theta}{m_1}\right)\Delta t - \left(\frac{K\cos^2 \theta - W_1 \cos 2\theta \sin\theta - \quad \cos\theta}{m_2 + m_1 \sin^2\theta}\right) At. \qquad (103a)$$

31, 32 and 33. The items in these columns are the different components of Δv_2 simplified so as to aid in computation. The various constants are defined in Paragraph 175.

34. $\Delta v_2 = k_3K + k_5R + k_6$. This is the increment of the secondary recoil velocity. It represents in condensed form Equation (104):

$$\Delta v_2 = \left(\frac{\text{Kcos } 8 - W_1 \cos 8 \sin\theta - R}{m_2 + m_1 \sin 28}\right) At. \quad (104a)$$

177. Following is the procedure for step-by-step integration, using Table 2. Assume an increment of time, At. Estimate the change, Δv_r, in relative primary recoil velocity which occurs during At. The summation of this plus all preceding increments in velocity is the total relative primary velocity at time, t, and is designated v_r. Ax,, the increment of primary travel during At, is computed from Ax, $=$ $v_r\ \Delta t$, and the summation of Ax, is x,, the total travel at time, t. From the graph, based on constant recoil forces, of primary relative velocity versus primary relative distance, the velocity, v_s, at this distance is found. Calculate $(v_r/v_s)^2$. Find K_{h_s} from the graph, based on constant recoil forces, or primary force versus distance. K_h is calculated from $K_h = K_{h_s}(v_r/v_s)^2$. K_a is determined from the graph of primary recuperator force versus primary relative distance, based on constant recoil forces. K_f is the primary static weight component, normal to the primary guides, multiplied by the coefficient of friction. K is the sum of K_h, K_a, and K_f. The propellant pressure force, F_g, is found from the curve of propellant pressure versus time. Δv_2 is found from Column **34** and v_2 is the summation of Δv_2. x_2 is the summation of $v_2\,At$. Knowing x_2, the secondary velocity, v_{2_r}, is obtained from the graph, based on constant recoil forces, of secondary recoil velocity versus secondary distance. Calculate $(v_2/v_{2_r})^2$. R_{h_s} is found from the graph, based on constant recoil forces, of the secondary hydraulic force versus the secondary recoil distance. R_h is equal to $R_{h_s}(v_2/v_{2_r})^2$. R_a is taken from the graph of secondary recuperator force versus secondary distance. R_{f_1} is the secondary guide friction due to static weight only. The secondary friction due to primary recoil is $R_{f_2} = \mu K \sin 8$. The total secondary recoil force is $R = R, + R_a + R_{f_1} + R_{f_2}$. **All** the constants, k_1, $k2$, k_3, k_4, k_5, and k_6 are calculated for the increment of time selected, and k_1F_g, k_2K, k_3R, Δv_r, k_3K, k_5R, and Δv_2 are all readily determined. If the final value of Δv_r is not in conformity with the value estimated at the beginning of the computation, the calculations must be repeated until agreement is reached.

178. The above procedure is necessary for the first increment of time and may be applied to each additional interval but a somewhat less laborious method is recommended. For each new increment of time, the average primary recoil velocity is taken as the velocity at the end of the preceding interval, thereby eliminating the tedious and incessant trials and errors. This simplified method introduces some inaccuracy, which can be minimized by assuming very small increments of time throughout the entire procedure. Even though a greater number of small intervals are used, there will still be a substantial saving of time and effort. The greater accuracy of the trial and error method seldom justifies its tedium. Whichever procedure is employed, it is carried through until the recoil distances have reached their maximum values. If, within these distances, the recoil motions have become zero and the forces have not peaked excessively, nor gone to infinity, the design is satisfactory. Otherwise, the need for redesign is indicated.

179. *(Step 7) Counterrecoil.* The procedure used for recoil calculations is also applied to counterrecoil. However, design requirements are less stringent than for recoil. All that is necessary is to return the gun to the in-battery position with forces sufficient to overcome friction, but not to create such high velocities that the weapon can overturn during buffing. In guns of large caliber, where double recoil systems are used to best advantage, there is ample time between rounds for counterrecoil and very low velocities are acceptable. Generally, forces large enough to overcome friction under all design conditions are adequate to insure a timely return.

180. The forces producing counterrecoil motions are imparted by the recuperators and the velocities are regulated by buffer mechanisms, the type of which depends upon the recoil system employed. Careful control of buffing is necessary as the in-battery position is approached so that the inertia forces in retardation remain commensurate with stability. The counterrecoil velocities sometimes are permitted to increase at will, until the buffers are contacted, and then decelerated to rest. Or, they may be controlled throughout their strokes, so that they increase from zero to some predetermined values, are held there for the greater part of their strokes, and then are buffed to rest.

181. To calculate counterrecoil action, the same

basic equations are used as in the primary and secondary recoil computations. It is a continuation of the same calculations, but with the addition of primary and secondary buffer forces, K_b and R_b. Friction is reversed, and recoil forces K and R become negative as the recuperator forces now exceed the other components. Thus,

$$K = K_h - K_a + K_f + K_b, \tag{105}$$

and

$$R = R_h - R_a + R_{f_1} + R_{f_2} + R_b, \tag{106}$$

where, as before, forces assigned the symbol, K, refer to the primary action and those designated R refer to the secondary action.

182. Forces constituting K and R are all predetermined except K_b and R_b. Since both Δv_r and Δv_2 are functions of K and R, plus a constant weight component, it is possible, by assigning proper magnitudes to the buffer forces, to control v_r and v_2 in counterrecoil. It is in this manner, by manipulating the buffing, or throttling forces, K_b and R_b, that counterrecoil motion is regulated.

XII. RECOIL SYSTEMS FOR SMALL ARMS*

A. INTRODUCTION

183. Designers of modern, lightweight weapons, particularly for aircraft, are confronted with conflicting demands for increased fire-power and decreased weapon weight. These requisites, involving high muzzle energy, reduced recoiling mass, and short recoil and counterrecoil time and distance, combine to increase transient recoil forces. Large, cyclic pulsations in these forces result in severe vibration problems in airframes and associated equipment. It is necessary that these recoil force oscillations be mechanically damped to the smallest possible amplitudes without impairing rate of fire, operating dependability, or accuracy. This is done through the application of what are known as "soft mounting" techniques.

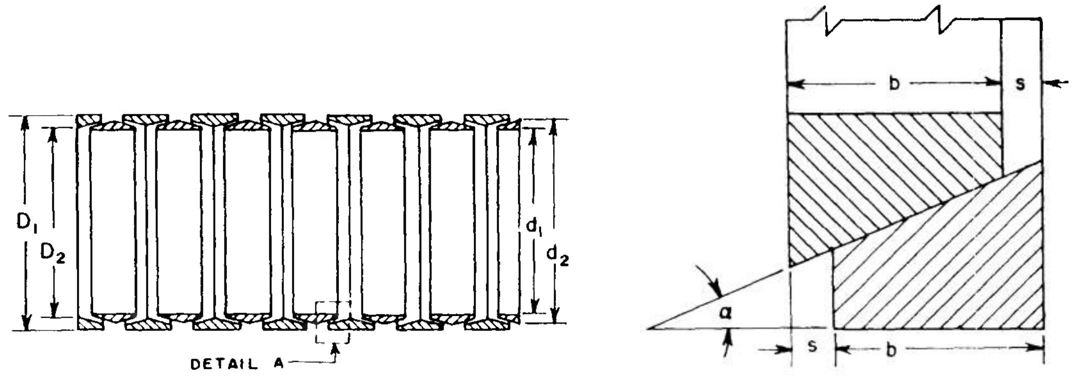
184. Recoil mechanisms for small arms are termed "recoil adapters." Of the several types that have been or are being developed, three will be discussed. (1) The ring spring represents the initial attempt to produce a soft recoil mounting. Simplicity of design and operation, plus durability, are its advantages. (2) The sleeve brake derives its retarding force from friction between a braided wire sleeve and a brake-liner covered rod oscillating within the sleeve. This force is always opposite in direction to the motion of the recoiling mass to which the brake rod is attached. (3) The hydrospring adapter generates its braking force by restricting the flow of a hydraulic fluid by an annular orifice. The fluid flow is deliberately made turbulent to increase the frictional loss for energy dissipation and rapid stabilization of the oscillations. The sleeve brake and the hydrospring adapters are more effective than the ring-spring type as they reduce peak-to-peak variations in trunnion force to about 50 percent of those permitted by the ring spring.

* Reference 13.

B. DESIGN AND OPERATING CHARACTERISTICS OF RING SPRINGS

185. The ring-spring assembly consists of a number of inner and outer closed rings contacting each other along conical surfaces. See Figure 40. When axial force is applied, the rings telescope into each other. A nearly uniform distribution of circumferential stress is obtained in both inner and outer rings; the outer rings being stressed in tension and the inner ones in compression. The rings are highly elastic and, as the spring assembly is compressed, the outer rings dilate and the inner contract as each conical surface telescopes into the adjacent one. Each pair of telescoping conical surfaces is considered a spring element. The total deflection of a spring assembly is the sum of the travels of the individual elements. Figure 40 also shows a cross section of a single spring element. The dimensions indicated, together with the number of elements, determine the design of a ring spring.

186. In designing ring springs for recoil adapters, the outside diameter and solid height are dictated by space considerations. The highest resistance of the spring is equal to the maximum recoil force. The maximum deflection is based upon the maximum travel of the recoiling mass and upon the spring preload, if any. A specific slope of the conical surfaces must be assigned. Generally, a minimum angle of approximately 14 degrees ensures release of the spring. Dimensions and certain characteristics of the ring elements and spring assembly are determined by the mathematical relationship given below.



LONGITUDINAL SECTION OF A WHOLE SPRING

DETAIL A – SHOWING DIMENSIONS OF SINGLE SPRING ELEMENT

Figure 40. Ring Spring With Single Spring Element

187. Nomenclature for ring spring design calculations. See also Figure 40.

A_1 = sectional area of outer ring element ($\frac{1}{2}$ area of outer ring)
A_2 = sectional area of inner ring element ($\frac{1}{2}$ area of inner ring)
b = half the width of each outer and inner ring
C = constant used in Equation (107) and pertaining to the recoil motion
C_1 = constant used in Equation (108) and pertaining to the counterrecoil motion
D_1 = outer diameter of outer ring
D_2 = inner diameter of outer ring
D_o = mean diameter of outer ring
d_1 = outer diameter of inner ring
d_2 = inner diameter of inner ring
d_i = mean diameter of inner ring
E = modulus of elasticity
F = axial force required to compress spring to its solid height
h = solid height of spring
n = total number of spring elements (twice the number of outer rings)
R = load deflection rate during compression
R_1 = load deflection rate during return
r = mean radius of the rings
S = total deflection or travel of spring
s = maximum travel of each spring element
V = total volume of ring spring assembly
V_1 = volume of outer ring element
V_2 = volume of inner ring element
W = work done during compression of spring
W_o = elastic work done during compression of spring
α = angle of conical surface
ϵ = mechanical efficiency of spring
μ = coefficient of friction
σ = numerical average of all circumferential stresses at load, F, disregarding signs
σ_c = average circumferential compressive stress in inner ring at load, F
σ_t = average circumferential tensile stress in outer ring at load, F.

188. Mathematical Relationships

$$C = \frac{\tan\alpha + \mu}{\tan\alpha\,(1 - \mu \tan\alpha)} \quad (107)$$

$$C_1 = \frac{\tan\alpha - \mu}{\tan\alpha\,(1 + \mu \tan\alpha)} \quad (108)$$

$$W = \frac{FS}{2} \text{ (no preload)} \quad (109)$$

$$W_o = \frac{W}{C} \quad (110)$$

$$V = \frac{2EW_o}{\sigma^2} \quad (111)$$

$$s = \frac{D_1 + d_2}{2} \frac{\sigma}{E\tan\alpha} \quad (112)$$

$$n = \frac{S}{s} \quad (113)$$

$$h = nb \quad (114)$$

$$A_1 \sigma_t = A_2 \sigma_c \quad (115)$$

$$D_o = \frac{D_1 + D_2 + b\tan\alpha}{2} \quad (116)$$

$$d_i = \frac{d_1 + d_2 - b\tan\alpha}{2} \quad (117)$$

$$s = \frac{D_o\sigma_t + d_i\sigma_c}{2E\tan\alpha} \tag{118}$$

$$R = \frac{2\pi E A_1 C \tan^2\alpha}{nr\left(1 + \frac{A_1}{A_2}\right)} \tag{119}$$

$$R_1 = \frac{2\pi E A_1 C_1 \tan^2\alpha}{nr\left(1 + \frac{A_1}{A_2}\right)} \tag{120}$$

$$\sigma_t = \frac{ES\tan\alpha}{nr\left(1 + \frac{A_1}{A_2}\right)} \tag{121}$$

$$\epsilon = \frac{R_1}{R} = \frac{C_1}{C}. \tag{122}$$

189. F, **S**, h, and D_1 are set up by the design requirements of the problem. a is assigned and is subject to some manipulation. E and σ are properties of the material to be used. μ is not known accurately and must be assumed, possibly supplemented by physical testing. With all these quantities at hand, Equations (109), (110), and (111) are used to find the required spring volume, V. It is permissible to work the inner rings to their elastic limit because the stresses are compressive and the danger of breakage is remote. However, the outer rings are stressed in tension that should be sufficiently below the elastic limit to insure safe operation of the spring. The inside diameter, d_2, of the inner ring, may be determined as the quantities D_1, h, and V are now known. Equation (112) yields the magnitude of **s**, the maximum travel of each spring element, and Equation (113) gives the value of n, the number of spring elements. If n is not an even number, the next higher even number is selected and s is recalculated. The dimension, b, is found from Equation (114). Equation (115) is available for finding the areas, A_1 and A_2, which **are** necessary to determine d_2. After finding D_o and d_i from Equations (116) and (117), respectively, the maximum travel, **s**, of the spring element can be checked with Equation (118). If close agreement with the original value of s is not reached, the spring must be revised.

190. It is convenient to have a chart, as in Figure **41**, to read directly the values of C, C_1, and ϵ for

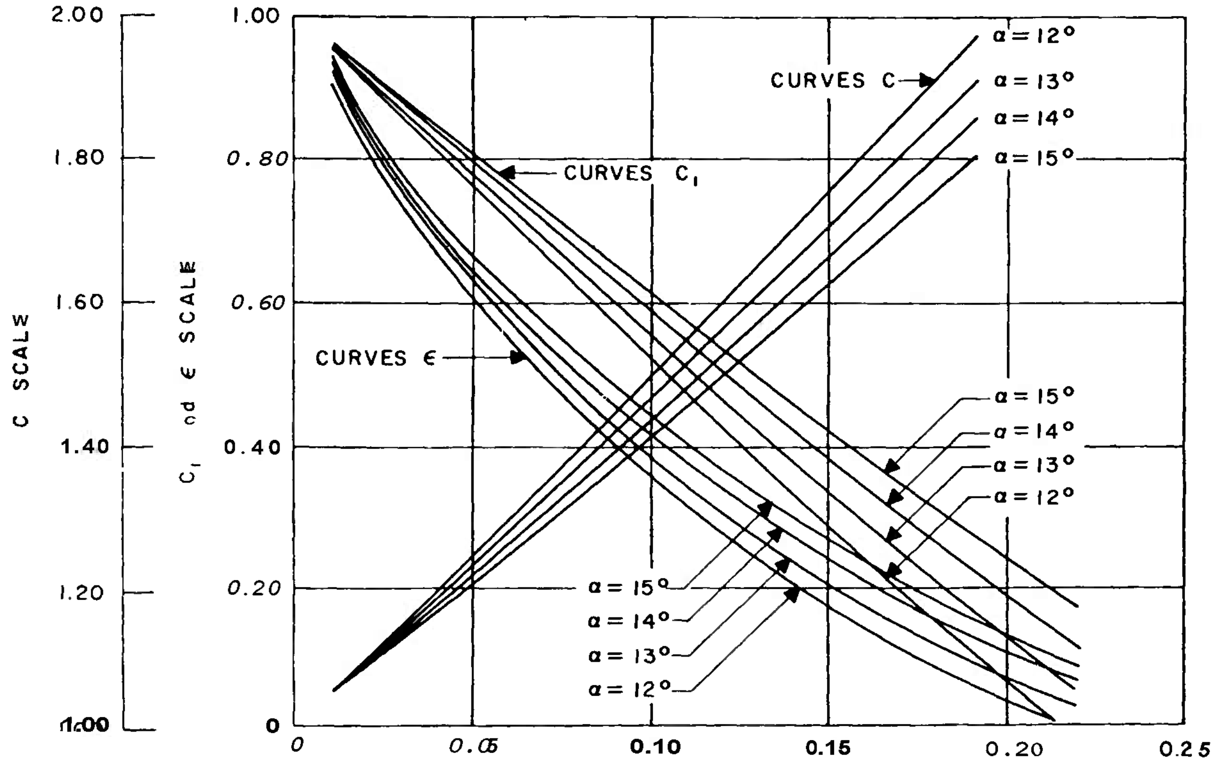


Figure 47. Ring Spring Constants and Efficiency

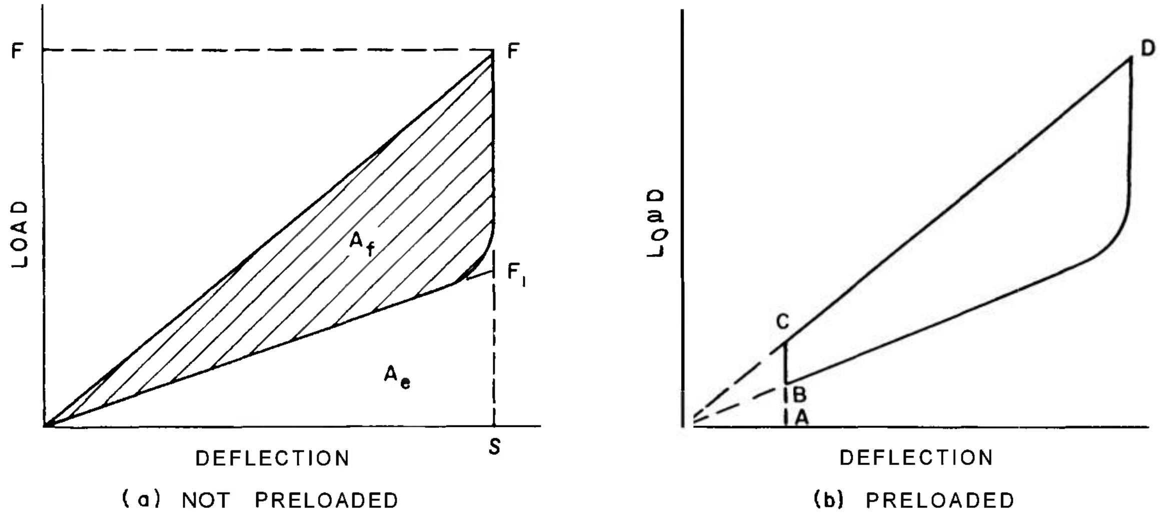


Figure 42. Load-Deflection Diagrams of Ring Springs

various combinations of α and μ. There is a wide variation in the coefficient of friction (from 0.01 to 0.22) depending upon type of lubricant and finish of the conical surfaces.

191. The theoretical load deflection rates (R and R_1) for the compression and return motions of a ring spring are calculated from Equations (119) and (120) and plotted as in Figure 42(a). The shaded area, A_f, represents the energy dissipated by the spring in the form of friction. Area, A_e, represents energy stored in the spring during the compression stroke and is available for returning the mass to its original position. The mechanical efficiency is:

$$\epsilon = \frac{SF_1}{SF} = \frac{A_e}{A_f + A_e} \tag{122a}$$

192. Ring springs are generally preloaded when used as recoil adapters. Figure 42(b) is a load-deflection diagram of a preloaded ring spring. The amount of preload is given by ordinate AC. The line CD represents compression of the spring while the curve DB is the load deflection relationship during its return to the initial preloaded condition. The dotted lines complete the curve to illustrate the diagram applied to the same spring if it were not preloaded.

193. Figure 43 shows the force deflection curves for two similar ring springs, using the same lubricant. The striking difference in their efficiencies illustrates the importance of correct contact between inner and outer conical surfaces. Graph (a) shows the effect of imperfect surfaces while graph (b) shows the characteristic of practically perfect surface-to-surface contact.

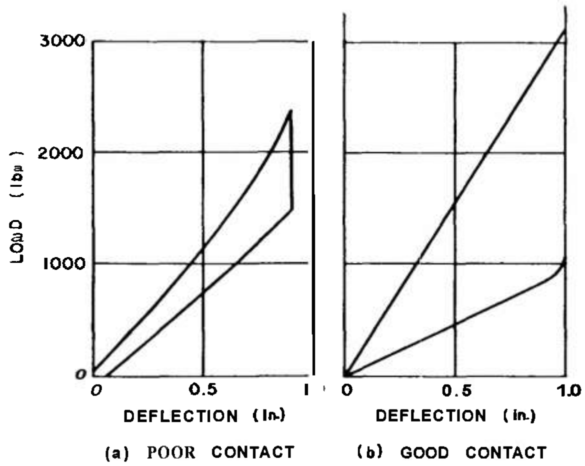


Figure 43. Load-Deflection Diagrams of Similar Ring Springs

C. DESIGN AND OPERATING CHARACTERISTICS OF THE SLEEVE BRAKE

194. The sleeve brake adapter consists of recoil spring, brake spring, and brake. Figure 44 shows the components of the mechanism. The recoil spring stores the energy for counterrecoil as it is compressed during recoil. The brake is essentially a braided wire sleeve which, when stretched by the brake spring, decreases in diameter and causes a

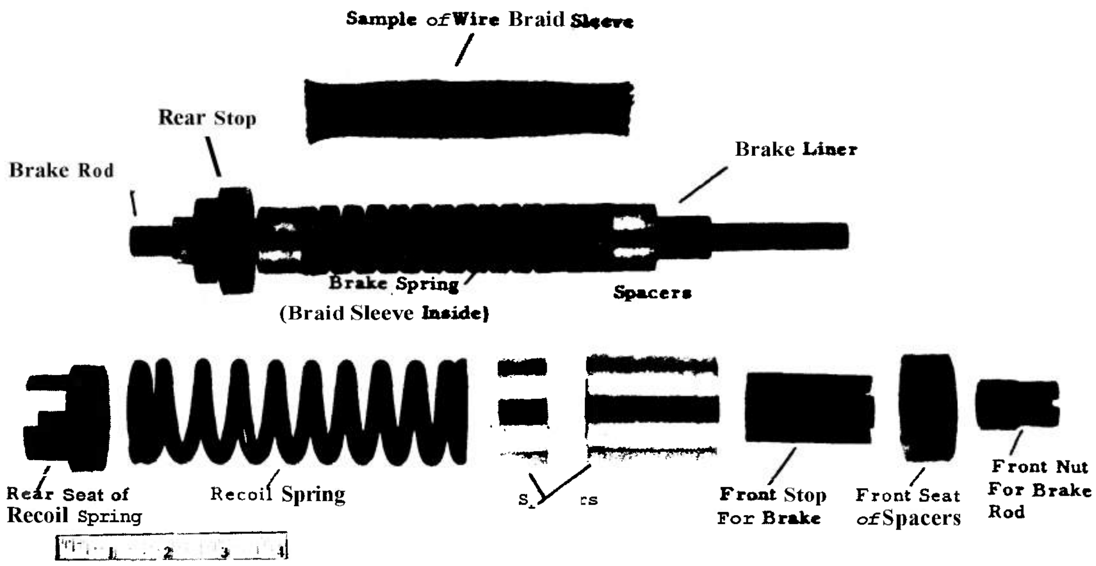


Figure 44. ***Sleeve Brake Recoil Adapter Disassembled***

frictional drag on the moving brake-liner covered rod. The relation of the various parameters affecting the sleeve friction force is given by Equation (123):*

$$\frac{F_f}{F_b} = 1 - e^{\left(-\mu \frac{L}{r} \tan^2 \beta\right)}, \qquad (123)$$

where:

$\tan\beta$ = ratio of circumference of wire strands to pitch,

F_f = rod force required to overcome friction,

F_b = brake spring force,

L = length of contact between sleeve and rod,

r = radius of rod,

β = braid angle or angle between wire strands and the axis of the sleeve,

μ = coefficient of friction between rod ana sleeve.

Figure 45 is a plot of Equation (123) for various values of L/r and for $\beta = \pi/4$ radians. From this, it is evident that if the ratio L/r is large, the braking force, F_f, will be nearly constant, provided the friction coefficient is reasonably large. The variation in F_f/F_b is less than 10 percent for a friction coefficient varying from 0.25 to 1.00 for an L/r ratio of 10 and with a β of $\pi/4$. Equation (123) will give the maximum friction force for a given friction brake without using any lengthy numerical or experimental procedures. A tubular wire braid for the sleeve brake is completely specified by the following data:

1. tube inner diameter,
2. type of wire (material),
3. diameter of wire,
4. number of wires per strand,
5. number of strands,
6. strand pitch.

195. The following represents a typical braid design:

1. 0.75 in. inner diameter of tube,
2. stainless steel (TS = 250,000 psi) or brass coated steel wire,
3. 0.01 in. wire diameter,
4. 9 wires per strand,
5. 24 strands,
6. 2.555 in. pitch.

Using 0.76 in. as an approximate mean diameter:

$$\tan\beta = \frac{0.76\,\pi}{2.555} = 0.938,$$

$$\beta = 43^\circ.$$

Fatigue failure of the braided sleeve is dependent upon the tensile force applied to it by the brake

* For derivation of Equation (123), *see* Reference 14.

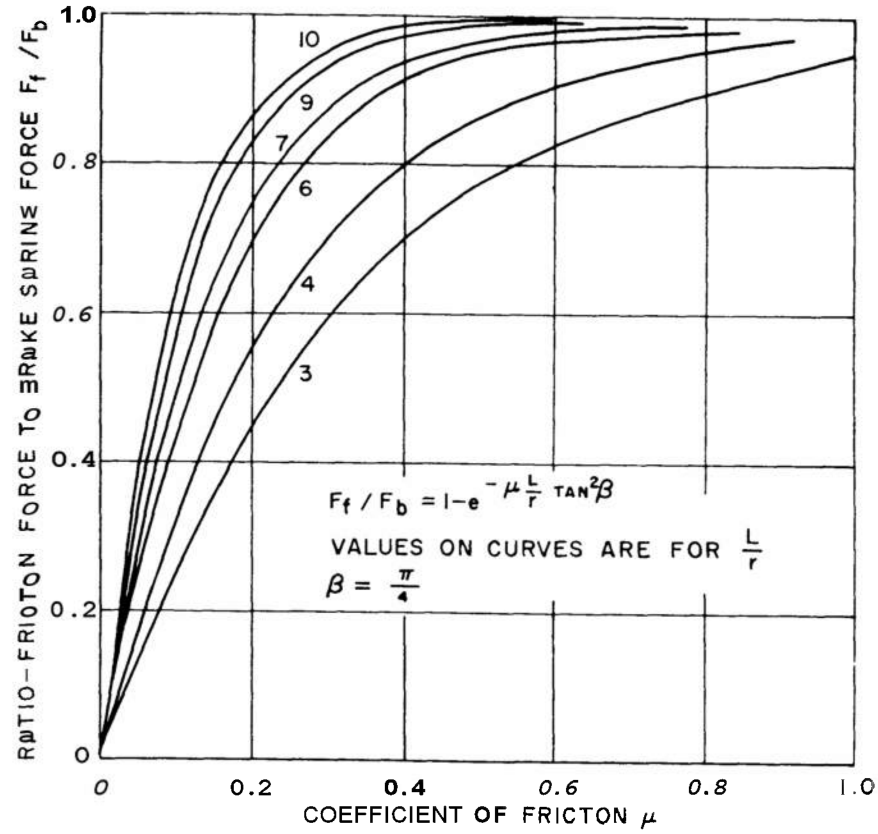


Figure 45. Force-Friction Relations of a Sleeve Broke Mechanism

spring. Sleeves similar to the one exemplified showed **signs** of fatigue failure at the end of 3000 cycles with a tensile force of 4000 pounds. When this tensile force was reduced to 225 pounds, no failure occurred during 5000 cycles. In static tests, degreased sleeve brakes produced higher brake forces than brakes tested in the greased, or oiled conditions. As the length of the sleeve is in-

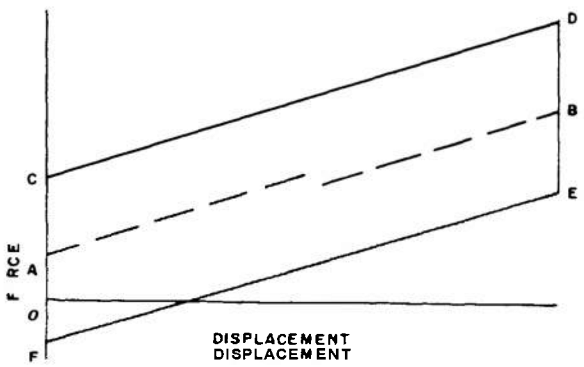


Figure 46. Force-Displacement Diagram of a Sleeve Brake Adapter

creased, the braking force increases so as to approach the brake spring force and the ratio F_f/F_b of Equation (123) approaches 1.00 as a limit. Static tests results also indicate that the coefficient of friction decreases with increased braid tension.

196. Figure 46 shows an idealized force — displacement diagram of a sleeve brake adapter. This adapter contains a single recoil spring which has a preload equal to the force *OA* on the diagram. The constant frictional force opposing recoil is represented by the line *AC* and $AC/A0 > 1.00$. The frictional force *AF* during counterrecoil equals *AC*. In actual practice, these frictional forces are larger than the spring preload and, on the basis of force alone, would not permit the gun to return to battery. However, the inertia of the moving counterrecoiling mass is sufficient to overcome the excess friction near the in-battery position.

197. Point 0 represents the initial static condition of the adapter. The curve 0*CD* represents the recoil conditions of the brake rod. When the force of the propellant gases is transmitted to the brake rod, no motion occurs until this force ex-

ceeds OC. Line CD represents the force — displacement curve during recoil. Its slope is established by the rate of the recoil spring. Counterrecoil activity is represented by the curve DEF. The dotted line is the force—displacement curve of the preloaded recoil spring. Since the area under the line CD represents the total recoil energy and that under line EF is the recoverable recoil energy, the frictional loss is given by the area within the parallelogram $CDEF$.

D. DESIGN AND OPERATING CHARACTERISTICS OF THE HYDROSPRING ADAPTER

198. The hydrospring adapter for small arms is similar to the hydrospring recoil mechanisms of larger weapons. The basic principle is the dissipation of energy resulting from the restricted flow of a hydraulic fluid through an orifice. The orifice can be either of constant or of varying area. Equation (36) reads

$$a_o = \frac{v}{C_o}\sqrt{\frac{\rho A^3}{2F_o}}.$$

If the orifice area, a_o, is held constant and the velocity of recoil, v, decreases with travel, the hydraulic resisting force, F_o, must also decrease so that the ratio, $v/\sqrt{F_o}$, is a constant. This will result in either a long recoil stroke or high initial forces. Neither choice is desirable; hence, in almost all cases, a constant orifice is not recommended.

199. The most efficient recoil mechanism is one providing a constant resistance to the recoiling mass. If the only resistance to recoil is by hydraulic means, then, to have a constant force, the orifice area must vary parabolically. However, an appreciable part of the recoil resistance comes from the recoil spring. Since this force is not uniform, the hydraulic resistance must also vary so that the total resistance is constant. Thus, the orifice area must vary somewhat differently from a parabolic curve. At the instant that the propellant gases cease to act on the gun, the energy of the recoiling mass is:

$$E_r = \frac{Mv_f^2}{2}, \qquad (124)$$

where:

v_f = velocity of free recoil;

assuming a constant recoil force which may be determined from the equation:

$$FL = \frac{Mv_f^2}{2}, \qquad (125)$$

where:

L is the recoil distance over which the force, F, acts.

The total recoil resistance has two components:

$$F = F_s + F_o, \qquad (126)$$

where:

F_s = spring force,
F_o = hydraulic force.

The value of F_s may be found for any distance, x, along L, and F_o is then determined:

$$F_s = F_{s_o} + Kx, \qquad (127)$$

where:

K = spring rate,
F_{s_o} = initial spring force.

After F_o is known, the orifice area, a_o, may be calculated from Equation (36). This can be done readily because the deceleration is uniform and all other parameters are known:

$$v^2 = v_f^2 - 2ax. \qquad (128)$$

The above method neglects the effects of the retarding force during the propellant gas period because its sole purpose is to illustrate the mechanics of combined spring and hydraulic forces. A more refined analysis must follow according to the procedure outlined in Chapter X. Another method for recoil analysis is found in Reference 15.

XIII. SUPPLEMENTAL DESIGN FEATURES

A. MUZZLE BRAKES

200. The propellant gas forces are totally responsible for recoil motion both during the time of projectile travel in the bore and after it leaves the muzzle until the gas pressure becomes ineffective. The curve of gas pressure versus time is obtained from the interior ballistician. Interior ballistics are treated in another series of Ordnance *Corps* Pamphlets beginning with Pamphlet No. ORDP 20-150.

201. After the projectile has left the muzzle, the propellant pressure persists for a short time and the gases are expelled at high velocity. The muzzle brake utilizes some of the energy still present in these gases to impede the rearward motion of the

recoiling parts. The brake has a definitely beneficial effect on recoil forces. When a muzzle brake is used, the designer of the recoil mechanism depends upon the gun designer to provide the performance data for the muzzle brake. The method for obtaining performance characteristics of muzzle brakes is discussed in the Ordnance Corps Pamphlets on Guns, beginning with **ORDP 20-250.**

B. LIQUID RESERVE INDICATOR, OR OIL INDEX

202. The liquid reserve indicator, or oil index, is a convenient means of showing the quantity of hydraulic fluid in reserve. Not all recoil mechanisms are equipped with fluid reserve indicators. The Puteaux and Filloux mechanisms are two that are *so* **equipped** (Refer to Chapter **V,** Parts **E** and *G).* **A** liquid indicator is an advantage in that it provides a quick, positive way to check for an adequate fluid reserve.

203. Figure **47** illustrates one type of oil index. It consists of two parallel racks on either side of, and engaged **by,** a pinion. The three components are assembled back of the regulator (Fig. 8) and are always subjected to the oil pressure which tends to force the lower rack outward. This tendency is restrained by a rod, attached to the floating piston and butting against the upper rack. When the liquid reserve between piston and regulator diaphragm diminishes, the rod moves outward, forcing the upper rack to move in the same direction. The lower rack, through action of the pinion, pulls the indicator into the housing; the length left extending indicates the amount of reserve fluid. During recoil, the floating piston withdraws the rod from the upper rack leaving the index free to move outward. When all components of the recoil mechanism resume their in-battery position, the oil index too assumes its normal position and again correctly indicates the oil reserve.

C. REPLENISHER

204. A replenisher is the oil reservoir for the hydraulic brake on some independent recoil mechanisms. It functions mainly as an expansion chamber

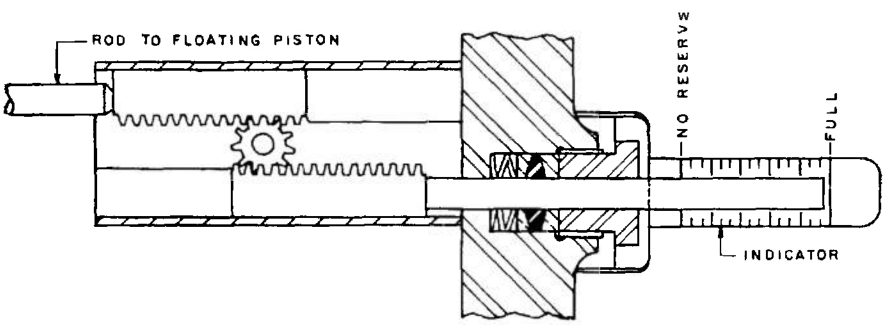


Figure 47. Oil Index

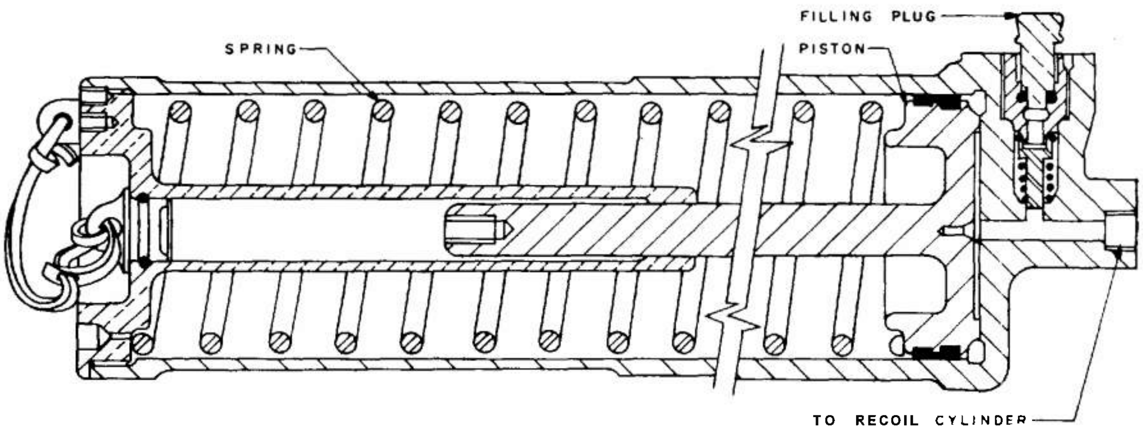


Figure 48. Replenisher

to offset the efforts of thermal expansion or contraction of the fluid brought about by changes in air temperature or by activity of the mechanisms. It consists of a cylinder and a spring loaded piston (Fig. 48). A line connects the replenisher to the low pressure end of the brake cylinder. The spring, by continually applying pressure to the fluid, maintains a completely filled brake cylinder at all times. The replenisher also serves as an oil index providing a means for checking the liquid volume. An access hole is provided on the spring end of the replenisher cylinder from which the position of its piston can be measured. The distance is calibrated to indicate whether the fluid volume is in the working range. If the piston is too near the spring end, liquid is in excess and must be removed. If too far from that end, fluid must be added to the replenisher until the correct volume is attained.

XIV. SAMPLE CALCULATIONS, SINGLE RECOIL

205. To exemplify the procedure followed in making recoil calculations, a hypothetical weapon is selected with given characteristics upon which the recoil mechanism is designed. This example problem applies to a single recoil system. The calculations are set up for the weapon firing at its maximum angle of elevation. The recoil mechanism is of the Puteaux type (Fig. 8).

206. Weapon Data.

L = 12.5 in., total recoil stroke,
v_m = 3400 ft/sec, muzzle velocity,
W_r = 6400 lb, weight of recoiling parts,
W_g = 10 lb, weight of propellant charge,
W_p = 15 lb, weight of projectile,
θ = 65°, maximum angle of elevation.

207. Effective Piston Area.

From Equation (2):

$$v_f = \frac{W_p v_m + 4700\, W_g}{W_r},$$

$$= \frac{15 \times 3400 + 4700 \times 10}{6400},$$

$$= 15.31 \text{ ft/sec, velocity of free recoil.}$$

From Equations (3a) and (3b):

$$K = \frac{M_r v_f^2}{2L} + W_r \sin\theta$$

$$= \frac{6400 \times 234}{32.2 \times 2 \times \frac{12.5}{12}} + 6400 \times 0.906 = 28{,}100 \text{ lb.}$$

This is the total recoil force used in the first series of calculations. It also determines the maximum pressure in the brake cylinder. Assume a recoil piston diameter of D_1 = **4.0** in. and a piston rod diameter of d = 1.5 in.:

$$A = \frac{\pi}{4}(D_1^2 - d^2) = 10.80 \text{ in}^2 \text{ effective piston area.}$$

From Equation (13):

$$P_m = \frac{K}{A} = \frac{28{,}100}{10.8} = 2600 \text{ psi pressure in recoil cylinder.}$$

This pressure is well within the former limit of 4500 psi (Refer to Paragraph 87) and so is acceptable.

208. Recuperator Gas Capacity.

From Equation (4):

$$F_1 = \lambda(W_r \sin\theta + \mu W_r \cos\theta + f_p), \text{ in-battery force,}$$

$$= 1.15(6400 \times 0.906 + 0.30 \times 6400 \times 0.423) + 1.15 \times 0.30\, F_1 = 10{,}100 \text{ lb.}$$

The force, F_1, is required to hold the gun in battery and to overcome all frictional resistance.

From Equation (19):

$$P_0 = \frac{F_1}{A} = \frac{10{,}100}{10.8} = 935 \text{ psi, initial recuperator pressure,}$$

$$\Delta V = LA = 12.5 \times 10.8 = 135 \text{ in}^3\text{, gas displacement,}$$

$$P_1 = 1.5\, P_0 = 1400 \text{ psi, recuperator pressure at end of recoil.}$$

From Equation (20):

$$\frac{P_1}{P_0} = \left(\frac{V_0}{V_1}\right)^n,$$

but $V_1 = V_0 - \Delta V;$

and $n = 1.3.$

Then: $$\frac{V_0}{V_0 - 135} = 1.5^{1/1.3} = 1.366,$$

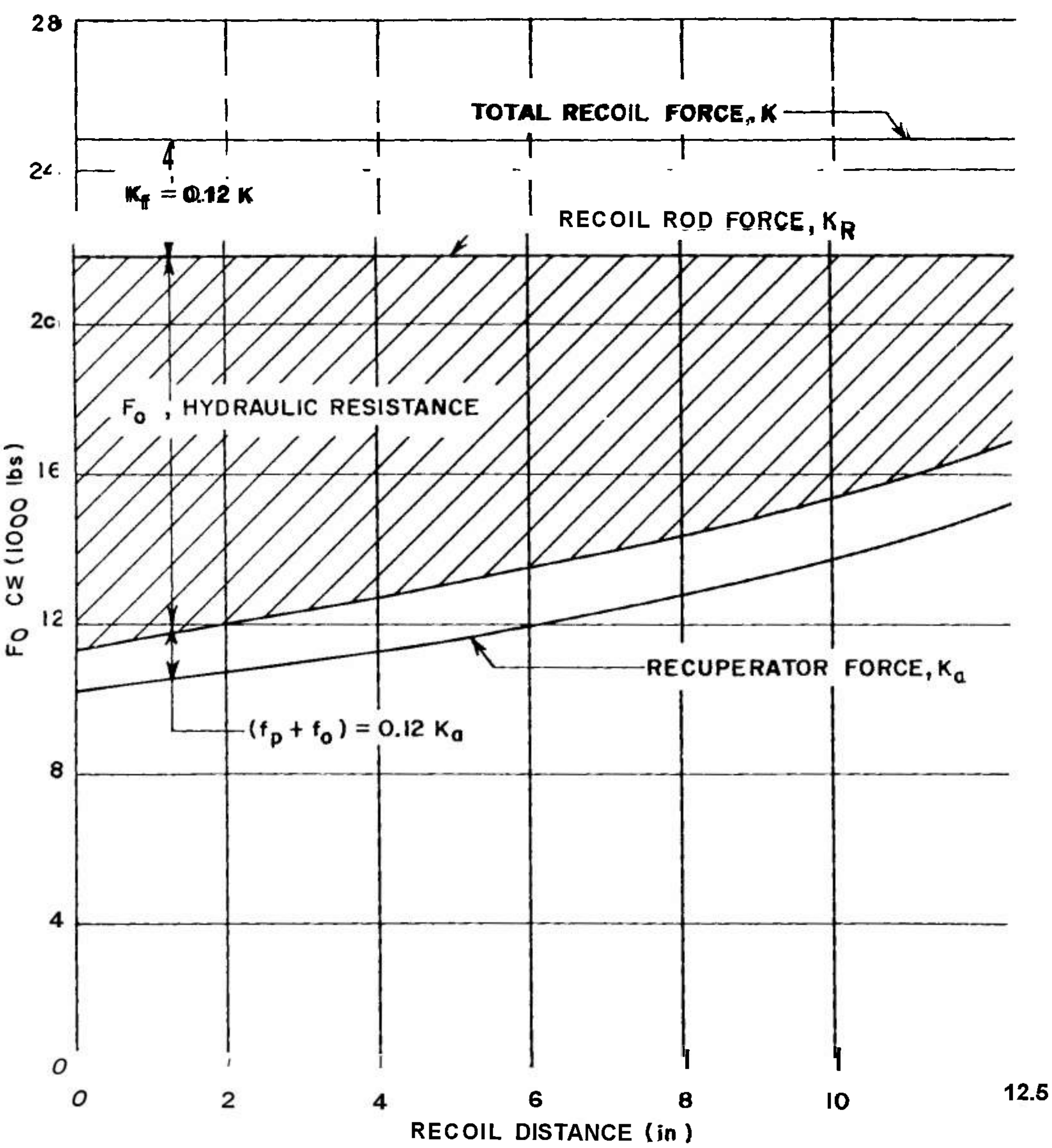


Figure 49. Recoil Force Chart for Sample Problem

V_0 = 503.8 in3 initial gas volume in recuperator.

Returning to Equation (20), the gas pressure at any distance of recoil, *x*, is:

$$P_x = P_0 \left(\frac{V_0}{\quad} \right)^n \quad \overline{503.8 - 10.8x}$$

and the recuperator force is:

$$K_a = AP_x = 10.80\ P_x.$$

The chart showing the various recoil forces may now be drawn (see Fig. **49**). It may be assumed for purpose of analysis that the friction of the recoil brake and sliding surface constitutes **12** percent of the total recoil force and also constitutes **12** percent of the recuperator force. For a finished design, these figures can be later substantiated by a rigorous analysis and the proper corrections made. The force-time curves have been prepared for the time that the projectile is in the bore and for the time required for the pressure to become zero after the projectile leaves the muzzle. The values are obtained from Figures **23** and **24**. **To** simplify the sample problem, the total recoil force is assumed constant for the entire recoil stroke.

209. The recoil calculations follow a step-by-step integration procedure. Results are more readily observed if in tabulated form as presented in Table **3**, Paragraph **215.** To illustrate the procedure, one step will be isolated here. Consider the time from 0.006 to 0.0065 sec:

$$At = 0.0065 - 0.006 = 0.0005 \text{ sec.}$$

From the curve in Figure 23, half way between $t = 0.006$ and $t = 0.0065$,

$F_g = 343{,}000$ lb, propellant gas force.

From Equation (25):

$F = F_g + W_r \sin\theta = 343{,}000 + 6400 \sin 65° =$ 348,800 lb, the accelerating force on the recoiling parts;

K = 24,800, the total resistance to recoil. The value for K is assumed constant for the entire recoil stroke;

$F_a = F - K = 324{,}000$ lb, the net accelerating force on the recoiling parts;

$a = \dfrac{F_a}{M_r} = 19{,}570$ in/sec², acceleration of the recoiling parts;

$M_r = \dfrac{W_r}{g} = \dfrac{6400}{386.4}$' mass of recoiling parts when $g = 386.4$ in/sec²;

$Av = a\Delta t = 9.79$ in/sec, change in velocity between 0.006 and 0.0065 sec;

$v = v_{n-1} + Av = 48.50 + 9.79 = 58.29$ **in/sec,** velocity of recoiling parts at end of 0.0065 sec and v_{n-1} is the velocity at the end of 0.006 sec;

$v_a = \frac{1}{2}(v_{n-1} + v) = 53.40$ in/sec, the average velocity between 0.006 and 0.0065 sec;

$Ax = v_a At = 0.0267$ in., the distance that the recoiling parts travel between 0.006 and 0.0065 sec;

$x = x_{n-1} + Ax = 0.0782 + 0.0267 = 0.1049$ **in.,** the total distance that the recoiling parts have travelled in 0.0065 sec.

210. The recoil calculations involve a trial and error procedure with the total recoil force, K, being the determining factor. All results are based on this value. In Paragraph 207, the first value of $K =$ 28,100 lb was computed. Subsequent calculations (not shown) proved that, using this resistance, the length of stroke necessary to stop the recoiling parts is 10.237 in., or 2.263 in. less than the stipulated stroke of 12.5 in. Since this result is not acceptable, another series of calculations must be made based on a corrected K.

211. Increasing the calculated 10.237 in- stroke to 12.5 in. lowers the net recoil resistance to $K =$ 22,300 lb after propellant gas forces cease. Let us assume that this is the force that would be applied over the last 2.263 in. if the full stroke is to be realized. The recoil energy for this distance becomes:

$$AE_r = 2.263 \times 22{,}308 = 50{,}500 \text{ in-lb.}$$

Applied over the total stroke of 12.5 in., the resulting change in recoil resistance is:

$$\Delta K = \frac{\Delta E_r}{12.5} = 4000 \text{ lb.}$$

The calculated stroke being less than the desired one indicates that the original K was too large, hence the new quantity becomes:

$$K = 28{,}100 - 4{,}000 = 24{,}100 \text{ lb.}$$

212. A second series of calculations is then made. This time the total recoil stroke is 12.985 in. or 0.485 in. over the desired distance: indicating that K is slightly too low. The new recoil. resistance over the last part of the stroke is $K =$ 18,300 lb. Correcting again, the additional energy required to stop the recoiling mass in the allotted distance is:

$$\Delta E_r = 0.485 \times 18{,}300 = 8846 \text{ in-lb;}$$

then:
$$\Delta K = \frac{\Delta E_r}{12.5} = 700 \text{ lb.}$$

The corrected total recoil resistance becomes:

$$K = 24{,}100 + 700 = 24{,}800 \text{ lb.}$$

213. The third series of calculations is made based on this new magnitude of K and is shown in Table 3 of Paragraph 215. The total recoil stroke is 12.365 in. This distance is acceptable as it is only one percent less than the required stroke of 12.5 in.

214. A convenient check of the calculations is provided by computing the work done for each increment of travel. Thus

$$AE = F_a \Delta x$$

where F_a is the net recoil accelerating force and is positive during the first part of the stroke and negative during the second part when the recoiling mass is decelerating. If the summations of negative and positive results are numerically equal, the calculations may be considered correct. Thus:

$$ZAE = (-)Z(-)AE$$

For the example just completed,

$$ZAE = 155{,}350 \text{ in-lb,}$$

$$\Sigma(-)\Delta E = -155{,}150 \text{ in-lb.}$$

215. TABLE 3. *Recoil Calculations*

t (sec)	Δt (sec)	F_g (1000 lb)	F (1000 lb)	K (1000 lb)	F_a (1000 lb)	a (in/sec^2)
0				24.8		
	0.001	23	28.8		4	242
0.001				24.8		
	0.001	47	52.8		28	1691
0.002				24.8		
	0.001	92	97.8		73	4407
0.003				24.8		
	0.001	172	177.8		153	9237
0.004				24.8		
	0.001	258	263.8		239	14429
0.005				24.8		
	0.0005	316	321.8		297	17931
0.0055				24.8		
	0.0005	334	339.8		315	19018
0.006				24.8		
	0.0005	343	348.8		324	19562
0.0065				24.8		
	0.0005	343	348.8		324	19562
0.007				24.8		
	0.001	308	313.8		289	17448
0.008				24.8		
	0.001	233	238.8		214	12920
0.009				24.8		
	0.001	178	183.8		159	9600
0.010				24.8		
	0.0005	140	145.8		121	7305
0.0105				24.8		
	0.0005	129	134.8		110	6641
0.011				24.8		
	0.001	90	95.8		71	4287
0.012				24.8		
	0.002	69	74.8		50	3020
0.014				24.8		
	0.002	51	56.8		32	1933
0.016				24.8		
	0.002	43	48.8		24	1450
0.018				24.8		
	0.002	37	42.8		18	1087
0.020				24.8		
	0.005	28	33.8		9	544
0.025				24.8		
	0.005	19	24.8		0	0
0.030				24.8		
	0.005	12	17.8		−7	−423
0.035				24.8		
	0.005	8	13.8		−11	−664
0.040				24.8		
	0.005	5	10.8		−14	−846
0.045				24.8		
	0.005	3	8.8		−16	−966
0.050				24.8		
	0.010	2	7.8		−17	−1027
0.060				24.8		
	0.010	1	6.8		−18	−1087
0.070				24.8		
	0.010	0.3	6.1		−18.7	−1129
0.080				24.8		
	0.010	0	5.8		−19	−1148
0.090				24.8		
	0.0695	0	5.8		−19	−1148
0.1595				24.8		

TABLE 3. *Recoil Calculations* (Continued)

t (sec)	Δv (in/sec)	v (in/sec)	v_a (in/sec)	Δx (in.)	x (in.)	ΔE (in-lb)
0		0			0	
	0.24		0.12	0.00012		
0.001		0.24			0.0001	
	1.691		1.09	0.00109		30
0.002		1.93			0.0012	
	4.407		4.14	0.00414		302
0.003		6.34			0.0054	
	9.237		10.96	0.01096		1677
0.004		15.58			0.0163	
	14.429		22.80	0.02280		5450
0.005		30.01			0.0391	
	8.966		34.50	0.01725		5120
0.0055		38.98			0.0564	
	9.509		43.74	0.02187		6890
0.006		48.49			0.0782	
	9.781		53.38	0.02669		8650
0.0065		58.27			0.1049	
	9.781		63.16	0.03158		10230
0.007		68.05			0.1365	
	17.448		76.78	0.07678		22190
0.008		85.50			0.2133	
	12.920		91.96	0.09196		19680
0.009		98.42			0.3052	
	9.600		103.22	0.10322		16416
0.010		108.02			0.4084	
	3.653		109.85	0.05493		6647
0.0105		111.67			0 4634	
	3.320		113.33	0.05667		6234
0.011		114.99			0.5200	
	4.29		117.14	0.11714		8320
0.012		119.28			0.6372	
	6.04		122.30	0.2446		12230
0.014		125.32			0.8818	
	3.866		127.26	0.2543		8138
0.016		129.19			1.1360	
	2.90		130.64	0.2612		6269
0.018		132.09			1.3973	
	2.174		133.18	0.2663		4793
0.020		134.26			1.6636	
	2.720		135.62	0.6781		6100
0.025		136.98			2.342	
	0		136.98	0.6849		0
0.030		136.98			3.027	
	−2.12		135.92	0.6796		−4760
0.035		134.86			3.706	
	−2.32		133.20	0.6660		−7330
0.040		131.54			4.372	
	−4.23		129.38	0.6469		−9060
0.045		127.22			5.019	
	−4.83		124.81	0.6241		−9990
0.050		122.39			5.643	
	−10.27		117.26	1.1726		−19930
0.060		112.12			6.816	
	−10.87		106.69	1.0669		−19200
0.070		101.25			7.883	
	−11.29		95.61	0.9561		−17880
0.080		89.%			8.839	
	−11.48		84.22	0.8422		−16000
0.090		78.48			9.681	
	−78.48		39.24	2.684		−51000
0.1584		0			12.365	

216. The orifice area and eventually the control rod diameter are determined from Equation (37):

$$a_o = \frac{v}{C_o}\sqrt{\frac{\rho A^3}{2F_o}\left(\frac{A_R - A_c}{A_R}\right)^3};$$

then:
- A = 10.8 in², effective area of recoil piston,
- A_R = 19.64 in², recuperator area,
- A_c = 0.99 in², control rod area (see Paragraph 123),
- C_o = 0.70, orifice coefficient,
- F_o = hydraulic resistance,
- g = 386.4 in/sec²,
- v = recoil velocity,
- w = 0.0315 lb/in³, density of fluid,
- ρ = w/g, mass density of fluid.

The value of F_o is obtained from Figure **49.** It is assumed that the losses in the hydraulic system, the $f_p + f_o$ of Equation (38), constitute 12 percent of the net recuperator force. Substituting the constants in Equation (37), the orifice area becomes:

$$a_o = 0.2996\frac{v}{\sqrt{F_o}}.$$

The hole in the **orifice** plate has a diameter of $1\frac{1}{8}$ in. Now solve for d_c, the control rod diameter at the orifice. Calculations are shown in Table **4,** Paragraph 217.

$$a_o = \frac{\pi}{4}(1.125^2 - d_c^2);$$

$$d_c = \sqrt{1.2656 - \frac{4}{\pi}a_o}.$$

Since the areas of the recoil piston and the recuperator are not equal, the distance that the control rod moves varies as the ratio of their areas

$$x_c = x\frac{A}{A_r};$$

$$x_c = 0.55x.$$

217. TABLE **4.** *Calculations for Control Rod Dimensions*

x (in.)	x_c (in.)	F_o (lb)	$\sqrt{F_o}$	v (in/sec)	a_o (in²)	$\frac{4}{\pi}a_o$ (in²)	d_c^2 (in²)	d_c (in.)
0	0	10,500	102.47	0	0	0	1.2656	1.125
0.0001	0.00006	10,500	102.47	0.24	0	0	1.2656	1.125
0.0012	0.00066	10,500	102.47	1.93	0.0056	0.0071	1.2585	1.122
0.0054	0.0030	10,500	102.47	6.34	0.0185	0,0236	1.2420	1.115
0.0163	0.0090	10,500	102.47	15.58	0.0456	0.0581	1.2075	1.099
0.0391	0.022	10,500	102.47	30.02	0.0878	0.1118	1.1538	1.074
0.0564	0.031	10,500	102.47	38.99	0.1140	0.1451	1.1205	1.058
0.0782	0.043	10,500	102.47	48.50	0.1418	0.1805	1.0851	1.042
0.1049	0.058	10,500	102.47	58.29	0.1704	0.2170	1.0486	1.024
0.1365	0.075	10,400	101.98	68.08	0.2000	0.2546	1.0110	1.005
0.2133	0.117	10,400	101.98	85.50	0.2512	0.3198	0.9458	0.973
0.3052	0.168	10,400	101.98	98.42	0.2891	0.3681	0,8975	0.947
0.4084	0.225	10,400	101.98	108.02	0.3173	0.4040	0.8616	0.928
0.4634	0.255	10,300	101.49	111.67	0.3296	0.4196	0.8460	0.920
0.5200	0.286	10,300	101.49	114.99	0.3394	0.4321	0.8335	0.913
0.6372	0.350	10,300	101.49	119.28	0.3521	0.4483	0.8173	0.904
0.8818	0.485	10,200	101.00	125.32	0.3717	0.4732	0.7924	0.890
1.136	0.625	10,200	101.00	129.19	0.3832	0.4879	0.7777	0.882
1.399	0.768	10,100	100.50	132.09	0.3938	0.5014	0.7642	0.874
1.664	0.915	9900	99.50	134.26	0.4042	0.5146	0.7510	0.867
2.342	1.288	9700	98.49	136.98	0.4167	0.5305	0.7351	0.857
3.027	1.665	9500	97.47	136.98	0.4210	0.5360	0.7296	0.854
3.706	2.038	9200	95.92	134.86	0.4212	0.5362	0.7294	0.854
4.372	2.405	9000	94.87	131.54	0.4154	0.5289	0.7367	0.858
5.019	2.760	8700	93.27	127.22	0,4086	0.5202	0.7454	0.863
5.643	3.104	8500	92.20	122.39	0.3977	0.5064	0.7592	0.871
6.816	3.749	8000	89.44	112.12	0.3755	0.4781	0.7875	0.887
7.883	4.336	7500	86.60	101.25	0.3503	0.4460	0.8196	0.905
8.839	4.861	7000	83.67	89.96	0.3221	0.4101	0.8555	0.925
9.681	5.325	6600	81.24	78.48	0.2894	0.3685	0.8971	0.947
11.066	6.086	5800	76.16	54.57	0.2146	0.2732	0.9924	0.996
11.957	6.576	5200	72.11	30.55	0.1269	0.1615	1.11041	1.051
12.365	6.800	4900	70.00	0	0	0	1.2656	1.125

218. The sample calculations for counterrecoil are based on the same hypothetical weapon that was used for the recoil calculation. The maximum counterrecoil velocity is specified as 2 ft/sec or 24 in/sec over a distance of 9.685 in. before the buffer is contacted. According to Equation (62),

$$F_{cr} = F_{ac} - f_{cr} - F_{bc}.$$

The packing frictional force, f_p, and the slide frictional force, F_{fc}, are assumed to constitute 12 percent of F_r in Equation (61d). Thus the available counterrecoil force, F_{ac}, is plotted and can be read directly from the curve in Figure 27. By substituting the average of F_o over each increment, Equation (64a) may be written

$$f_{cr} = \left(\frac{v_{r_{n-1}}^2 + v_{r_n}^2}{v_{n-1}^2 + v_n^2}\right)\left(\frac{F_{o_{n-1}} + F_{o_n}}{2}\right). \quad (129)$$

The force, F_o, generated by the orifice during recoil and the recoil velocity, v, are obtained from the recoil calculations. The value of v_r, the counterrecoil velocity, is determined by trial and error. From Equation (64b):

$$F_{bc} = \frac{\rho}{2C_c^2}\left(\frac{A_R - A_c}{A_R}\right)^3 \frac{A^3}{a_c^2}\left(\frac{v_{r_{n-1}}^2 + v_{r_n}^2}{2}\right).$$

Since the constants are the same as those for the recoil calculations (Paragraph 216), the equation reduces to

$$F_{bc} = \frac{0.0898}{a_c^2}\,\frac{v_{r_{n-1}}^2 + v_{r_n}^2}{2} \quad (130)$$

219. The maximum orifice area during recoil is 0.4212 in². In accordance with Paragraph 137, a constant orifice area of 0.04212 in² was tried but it held the counterrecoil velocity to less than 17 in/sec. The orifice was increased to 0.07 in² and subsequent calculations showed a counterrecoil velocity of 22.55 in/sec just as contact with the buffer begins (at x = 2.5, Table 5, Paragraph 222). This velocity is reasonably close to the desired 24 in/sec and, until more exact frictional data become available, it may be considered adequate. One step of the calculations is detailed below:

x = distance from in-battery position, from Table 4, Paragraph 217.

x_r = travel in counterrecoil, 12.365 − x.

$\Delta x_r = x_{r_n} - x_{r_{n-1}}$.

at x = 9.681 in., Δx_r = 1.385 in. and x_r = 2.684 in.

From the chart of Figure 27, F_{ac} = 12,300 lb.
From Table 4, Paragraph 217, the recoil calculations show:

$$F_{o_n} = 6600 \text{ lb}, \; F_{o_{n-1}} = 5800 \text{ lb};$$

$$v_n = 78.48 \text{ in/sec}, \; v_{n-1} = 54.57 \text{ in/sec}.$$

After two trials, v_{r_n} = 25.02 in/sec is finally selected and from the fourth column of Table 5, when $x_{r_{n-1}}$ = 1.299 in., $v_{r_{n-1}}$ = 24.90 in/sec.

Then:

$$f_{cr} = \left(\frac{24.90^2 + 25.02^2}{54.57^2 + 78.48^2}\right)\left(\frac{5800 + 6600}{2}\right) = 845 \text{ lb}.$$

This represents the throttling through the recoil orifice. Substituting known quantities in Equation (130a):

$$F_{bc} = \frac{0.0898}{0.0049}\left(\frac{620 + 626}{2}\right) = 11{,}420 \text{ lb},$$ the throttling through the counterrecoil orifice;

F_{cr} = 12,300 − 845 − 11,420 = 35 lb, the counterrecoil accelerating force;

$$a_r = \frac{F_{cr}}{M_r} = \frac{35}{16.56} = 2.11 \text{ in/sec}^2,$$ the counterrecoil acceleration;

$$M_r = \frac{W_r}{g} = \frac{6400}{386.4} = 16.56 \text{ lb-sec}^2\text{/in}.$$

From Equation (66):

$$v_r^2 = v_{r_{n-1}}^2 + 2a_r\,\Delta x_r,$$
$$= 620.0 + 2 \times 2.415 \times 1.385,$$
$$= 620.0 + 5.8 = 625.8 \text{ in}^2\text{/sec}^2,$$
$$v_r = 25.02 \text{ in/sec}.$$

This agrees with the selected velocity at the beginning of the calculations.

220. The sample buffer calculations illustrate the method of constant deceleration and variable orifice. From Equation (70b):

$$a_b = \frac{v_{rb}^2}{2S_b} = -101.64 \text{ in/sec}^2,$$

when: v_{rb}^2 = 508.2 in²/sec² (see counterrecoil calculations, Table 5, Paragraph 222),

S_b = 2.5 in., buffer stroke.

From Equation (70a):

$$v_b^2 = v_{b_{n-1}}^2 + 2a_b x_b,$$

$$v_b^2 = v_{b_{n-1}}^2 - 203.28\, x_b.$$

From Equation (70c)

$$F_b' = M_r a_b = 16.56 \times (-)\ 101.6 = -1683\ \text{lb};$$

and from Equation (70)

$$F_b = F_{cr} - F_b' = F_{cr} + 1683\ \text{lb}.$$

221. The calculations for one increment of buffer travel are shown at the buffer travel of x_b = 1.0 in. This corresponds to the counterrecoiling parts being at x = 1.5 in. from the in-battery position:

$\Delta v_b^2 = -203.28 \times 0.25 = -50.82\ \text{in}^2/\text{sec}^2$ (Table 6, Paragraph 223),

$$v_b^2 = 355.74 - 50.82 = 304.92\ \text{in}^2/\text{sec}^2,$$

$$v_b = 17.46\ \text{in/sec}.$$

From recoil calculations (Table 4, Paragraph 217) interpolate between x = 1.664 in. and x = 1.399 in. to find v for x = 1.5 in.

$$v = 132.92\ \text{in/sec},$$

$$F_o = 10{,}000\ \text{lb (Fig. 49)}.$$

From the chart in Figure 27:

$$F_{ac} = 9250.$$

From Equation (61c) when x = 1.5,

$$f_{cr} = \left(\frac{v_{b_{n-1}}^2 + v_{b_n}^2}{v_{n-1}^2 + v_n^2}\right) F_o,$$

$$= \left(\frac{355.74 + 304.92}{18{,}117 + 17{,}668}\right) 10{,}000 = 184\ \text{lb}.$$

According to Paragraphs 219 and 220:

$$F_{bc} = \frac{0.0898}{a_o^2}\left(\frac{v_{b_{n-1}}^2 + v_{b_n}^2}{2}\right),$$

$$= \frac{0.0898}{0.0049}\left(\frac{355.74 + 304.92}{2}\right) = 5589\ \text{lb}.$$

From Equation (62):

$$F_{cr} = F_{ac} - f_{cr} - F_{bc},$$

$$= 9250 - 184 - 5589 = 3477,$$

and

$$F_b = F_{cr} + 1683 = 3477 + 1683 = 5160\ \text{lb}.$$

From Equation (71):

$$a_{ob} = v_b\sqrt{\frac{\rho A_b^3}{2C_o^2 F_b}} = 0.033\,\frac{v_b}{\sqrt{F_b}};$$

A_b = 2.356 in², buffer piston area,

C_o = 0.70 orifice coefficient,

ρ = w/g, mass density,

w = 0.0315 lb/in³, oil density,

$$a_{ob} = 0.033\,\frac{1746}{71.8} = 0.0080\ \text{in}^2.$$

222. *TABLE 5. Counterrecoil Calculations*

x_r (in.)	Δx_r (in.)	F_{ac} (lb)	v_r (in/sec)	f_{cr} (lb)	F_{bc} (lb)	F_{cr} (lb)	a_r	$2a_r\Delta x_r$ (in²/sec²)	v_r^2 (in²/sec²)
0			0						0
	0.408	13,300		2063	3496	7740	467.4	381.4	
0.408			19.53						381.4
	0.891	12,800		1408	9178	2214	133.7	238.2	
1.299			24.90						619.7
	1.385	12,300		845	11,420	35	2.11	5.8	
2.684			25.02						625.2
	0.842	11,800		591	11,347	−138	−8.33	−14.0	
3.526			24.74						612.0
	0.956	11,400		477	11,067	−145	−8.70	−16.6	
4.482			24.40						595.5
	1.067	11,000		398	10,745	−143	−8.64	−18.4	
5.549			24.02						577.2
	1.173	10,600		340	10,397	−137	−8.09	−19.4	
6.722			23.61						557.6
	0.624	10,300		305	10,127	−132	−7.97	−9.9	
7.346			23.40						547.5
	0.647	10,100		287	9944	−131	−7.91	−10.2	
7.993			23.18						537.5
	0.666	9900		273	9758	−131	−7.91	−10.5	
8.659			22.96						527.0
	0.679	9700		264	9566	−130	−7.85	−10.7	
9.338			22.73						516.3
	0.527	9500		235	9392	−127	−7.67	−8.08	
9.865			22.55						508.2

223. TABLE 6. *Buffer Calculations*

x (in.)	x_b (in.)	Δv_b^2 (in²/sec²)	v_b^2 (in²/sec²)	v_b (in/sec)	V (in/sec)	v^2 (in²/sec²)	F (lb)
2.50	0	0	508.20	22.55	136.98	18,764	9500
2.25	0.25	50.82	457.38	21.39	136.61	18,662	9400
2.00	0.50	50.82	406.56	20.16	135.61	18,390	9380
1.75	0.75	50.82	355.74	18.86	134.60	18,217	9300
1.50	1.00	50.82	304.92	17.46	132.92	17,668	9250
1.125	1.375	76.23	228.69	15.12	129.02	16,646	9150
0.75	1.75	76.23	152.46	12.35	122.08	14,960	9050
0.375	2.125	76.23	76.23	8.73	105.15	11,070	8950
0	2.50	76.23	0	0	0	0	8900

x (in.)	F_o (lb)	F_{bc} (lb)	f_{cr} (lb)	F_{cr} (lb)	F_b (lb)	$\sqrt{F_b}$ –	a_{ob} (in²)
2.50	9,600	9315	260	−75	1,608	40.1	0.0185
2.25	9,700	8384	250	766	2,449	49.5	0.0143
2.00	9,800	7452	228	1700	3,383	58.2	0.0114
1.75	9,900	6521	206	2573	4,256	65.3	0.0095
1.50	10,000	5589	184	3477	5,160	71.8	0.0080
1.125	10,200	4192	159	4799	6,482	80.5	0.0058
0.75	10,250	2795	124	6131	7,814	88.4	0.0046
0.375	10,400	1397	92	7461	9,144	95.6	0.0030
0	10,500	0	72	8828	10,583	102.5	0

GLOSSARY

brake pull. The force applied to the recoil brake rod during recoil.

breechblock. The structure used for closing the rear end of a gun tube.

breech housing. The structure attached to the rear of the gun tube which houses the breechblock and its components.

buffer. A device used to absorb the energy of counterrecoil and bring the recoiling parts to a stop without shock. See: **recoiling parts.**

buffer, independent. A buffer whose action is not controlled by the recoil brake or recuperator.

control rod. A rod articulated by the recoiling parts, and whose motion regulates the size of the orifice of a recoil mechanism.

counterrecoil. The motion of the recoiling parts as they return to the in-battery position after recoiling. See: **recoiling parts.**

counterrecoil cylinder. The cylinder that houses the counterrecoil mechanism.

cradle. That element of a gun carriage or mount that supports a cannon and allows movement of the recoiling parts.

floating piston. An unattached piston that is used to separate the gas from the hydraulic fluid in the recuperator of a hydropneumatic unit.

gun carriage. The structure which transmits to the ground the forces resulting from firing of a weapon. In mobile weapons it also serves as part of the structure during transport.

gun mount. The supporting structure of a gun.

gun tube. The gun barrel, that part of a gun which controls the initial direction of the projectile.

in-battery. The position of the recoiling parts in the extreme forward position in the cradle.

leakage factor. The ratio of the radial pressure of a packing to the applied fuid pressure.

liquid reserve indicator. See: **oil reserve indicator.**

muzzle brake. A unit attached to a gun muzzle, to divert the propellant gas rearward, utilizing the gas momentum to decrease the total recoil momentum.

muzzle velocity. The velocity of the projectile as it leaves the muzzle.

oil index. Same as *oil reserve indicator.*

oil reserve. A quantity of oil available in a recoil mechanism to replenish the supply of working oil as the latter becomes depleted from leakage.

oil reserve indicator. A gage indicating the quantity of reserve oil in a hydropneumatic recoil cylinder. Also called *liquid reserve indicator.*

orifice. An opening of controlled size through which fluid passes for the purpose of absorbing energy.

port. A passage of sufficient size to transmit fluid without appreciable loss of energy.

pressure factor. The ratio of the radial pressure of a packing to the applied axial pressure.

propellant charge. The quantity of propellant used in firing a gun.

propellant gas force. The force due to the propellant gases that drives the gun rearward in recoil.

recoil. The movement of the gun tube and attached parts in direction opposite to the projectile travel.

recoil adapter. A recoil mechanism for small arms.

recoil brake. The part of a recoil mechanism that develops the resistance to recoil.

recoil, constant. A recoil mechanism that provides the same recoil-stroke distance regardless of the angle of elevation.

recoil cylinder. The cylinder that houses the recoil brake.

recoil energy. The energy of the recoiling parts during recoil.

recoil force. The total resistance to movement of the recoiling parts.

recoil mechanism. The unit that absorbs the energy of recoil, storing some for returning the recoiling parts to battery.

recoil mechanism, dependent. The type of hydropneumatic recoil mechanism that has direct oil flow between the recoil cylinder and the recuperator. The recoil brake rod is the only attachment to the recoiling parts.

recoil mechanism, Filloux. A hydropneumatic, independent, variable recoil mechanism with a floating piston. Hydraulic resistance is developed by a control rod with axial grooves over which the recoil piston rod slides.

recoil mechanism, hydropneumatic. A type of recoil mechanism that forces hydraulic fluid through an orifice to develop the recoil brake force and uses gas under pressure to store some of the recoil energy for counterrecoil.

recoil mechanism, hydrospring. A type of recoil mechanism which operates similarly to the hydropneumatic except that a spring is used to store the energy for counterrecoil.

recoil mechanism, independent. The type of hydropneumatic recoil mechanism that has the recoil brake independent of the recuperator and counterrecoil cylinder. Each has its own rod attached to the recoiling parts.

recoil mechanism, Puteaux. A hydropneumatic, dependent type recoil mechanism of constant recoil with a floating piston in the recuperator; hydraulic resistance is regulated by a control rod passing through an orifice and attached to and positioned by the floating piston.

recoil mechanism, Schneider. A hydropneumatic, independent type recoil mechanism of constant recoil. It has a direct contact recuperator. The hydraulic resistance is regulated by a control rod passing through an orifice and attached to the recoiling parts.

recoil mechanism, St. Chamond. A hydropneumatic, dependent type recoil mechanism of variable recoil with floating piston recuperator. Its hydraulic resistance is regulated by a throttling valve.

recoil rod. The rod that transmits the resistance of the recoil brake to the recoiling parts.

recoil system. The complete unit that involves the recoil and counterrecoil processes.

recoil system, double. The system that has two complete recoiling systems; the primary system and the secondary system.

recoil system, primary. The system of the double recoil type which includes the recoil mechanism for the gun tube and its components.

recoil system, secondary. The system of the double recoil type which permits the top carriage to recoil, thus effectively using the mass of the structure to reduce recoil forces.

recoil system, single. The system that has only the gun tube and its components as recoiling parts.

recoil, variable. A recoil mechanism having a stroke that varies in accordance with the angle of elevation; at high angles of elevation, the recoil stroke is shortened.

recoiling parts. The components of a gun and its supporting structure that move during recoil.

recoperator. The portion of the recoil mechanism that stores some of the energy of recoil for counterrecoil.

recoperator, direct contact. A recuperator in which the gas acts directly against the hydraulic fluid, without separating piston.

recuperator, hydropneumatic. A recuperator that contains a compressed gas for its energy storing medium.

recuperator, spring. A recuperator that has one or more coil springs for its energy storing medium.

regulator. A structure used in some hydropneumatic recoil systems, located in the oil end of the recuperator, which contains the means provided to control the hydraulic pressure during recoil and counterrecoil.

regulator valve. A valve housed in the liquid end of the recuperator which regulates the flow of liquid during the counterrecoil stroke, to produce retardation.

replenisher. A reservoir for the hydraulic brake fluid that maintains nearly uniform pressure on the fluid and keeps the brake cylinder filled with fluid.

respirator. A pneumatic type buffer which admits air during recoil and releases the air during counterrecoil through a small orifice. See: **buffer.**

ring spring. The spring of a recoil adapter made of a series of inner and outer overlapping rings, the mating surfaces being conical.

sleeve brake. The part of a recoil adapter consisting of a sleeve made of braided wire covering a brake rod to form the braking mechanism.

throttling bar. A bar of varying cross-sectional area, usually circular or rectangular, which changes the orifice area of a recoil mechanism.

throttling valve. A spring loaded valve that controls the hydraulic pressure during recoil, as a means of obtaining variable recoil.

velocity of counterrecoil. The velocity of the recoiling parts as they move in counterrecoil.

velocity of free recoil. The velocity that the recoiling parts would attain if no resistance were provided during recoil.

velocity of recoil. The velocity of the recoiling parts during recoil.

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