ENGINEERING DESIGN HANDBOOK

CARRIAGES AND MOUNTS SERIES

EQUILIBRATORS

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ENGINEERING DESIGN HANDBOOK

EQUILIBRATORS

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EQUILIBRATORS

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EQUILIBRATORS

LIST OF FIGURES

LIST OF SYMBOLS

A = Effective piston area

c = Distance from trunnion to fixed equilibrator pivot on carriage

D = Subscript denoting maximum angle of depression

d = Distance from movable equilibrator pivot to r

E = Modulus of elasticity

E_t = Total spring energy

F = Equilibrator force or spring force

F_A = Frictional force produced at carriage bearing

F_B = Frictional force produced at cradle bearing

F_c = Chain tension

F_f = Total frictional force in rod

F_g = Gas force on equilibrator piston

F_m = Maximum spring force

F_p = Radial force exerted by packing

F_R = Net equilibrator rod force

F_w = Weight components causing unbalanced moment in traverse

f_p = Packing frictional force

$f(T)$ = A function of T

I = Moment of inertia of cross section

K_p = Pressure factor

K_s = Spring rate

K_t = Spring rate, torsion bar

L = Length of equilibrator, general

L_m = Maximum equilibrator length

L_s = Actual length of clock-type spring

L_θ = Length of equilibrator at any angle of elevation

ΔL = Equilibrator travel or stroke, at any position

M = Bending moment

M_e = Equilibrator moment about trunnions

M_s = Torsion bar torque applied to tipping parts

M_w = Weight moment

m = Reciprocal of Poisson's ratio

n = Defined in the relation PV^n = a constant

P = Gas pressure, general

P_a = Gas pressure, absolute

P_m = Maximum fluid pressure

P_s = Packing pressure produced by spring

P_θ = Gas pressure at any angle of elevation

Q = Location of c.g. of tipping parts

R = Turning radius of equilibrator about trunnions

R_s = Radius of chain drum

R_t = Distance, trunnion axis to c.g. of tipping parts

r = Equilibrator moment arm, perpendicular from trunnion to L_θ

S = Spring deflection at maximum weight moment

S_m = Maximum spring deflection

T = Absolute temperature; also required torque

T_A = Frictional torque at carriage bearing

T_B = Frictional torque at cradle bearing

T_e = Torque required by elevating mechanism

T_s = Torque provided by torsion bar

T_t = Torque required of torsion bar

u = Subscript denoting $u°$ of elevation

V = Gas volume, general

V_θ = Gas volume at any degree of elevation

ΔV = Change in gas volume

W_t = Weight of tipping parts

Z = Section modulus

0 = Subscript denoting zero position

α = Angle between R and R_t

θ = Angle of elevation

θ_t = Slope of terrain

μ = Coefficient of friction

ν = Leakage factor

σ = Stress, general

σ_c = Compressive stress

σ_t = Tensile stress

ϕ = Angle locating c.g. of tipping parts from horizontal line through trunnion

ϕ_a = Angle locating c.g. of traversing parts from horizontal line through turret center

Ψ = Angle between R and c, variable

EQUILIBRATORS

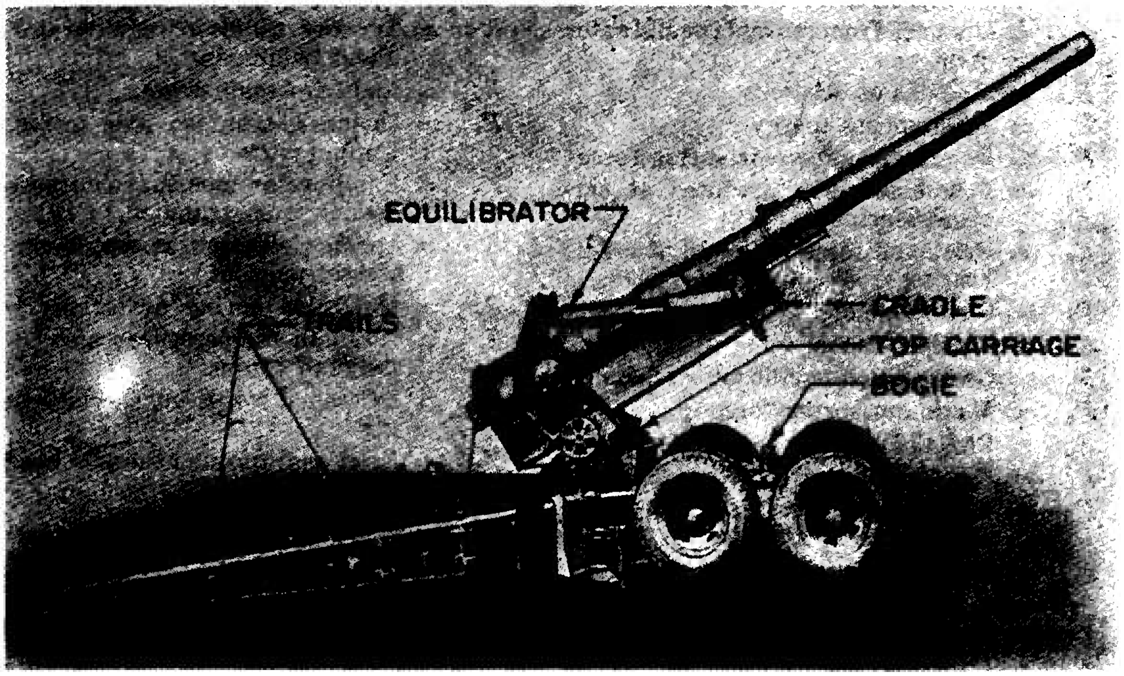

Figure 1. Weapon Showing Typical Equilibrator Installation

PREFACE

The Engineering Design Handbook Series of the U.S. Army Materiel Command is a coordinated series of handbooks containing basic information and fundamental data useful in the design and development of Army materiel and systems. The handbooks are authoritative reference books of practical information and quantitative facts helpful in the design and development of Army materiel so that it will meet the tactical and technical needs of the Armed Forces.

This handbook has been prepared as one of a series on Carriages and Mounts. It presents information on the fundamental operating principles of equilibrators, on that part of the artillery assemblage which overcomes the unbalance of the tipping parts, or in the case of an azimuth equilibrator, compensates for the effect of tilt of the mount. Comparisons of various types of equilibrators are presented with guides for the selection of the desirable type.

This handbook was prepared by the Franklin Institute Research Laboratories, under subcontract to the Engineering Handbook Office of Duke University, prime contractor to the U.S. Army Materiel Command for the Engineering Design Handbook Series. Technical assistance was given by the U.S. Army Weapons Command.

The handbooks are readily available to all elements of AMC, including personnel and contractors having a need and/or requirement. The U.S. Army Materiel Command policy is to release these Engineering Design Handbooks to other DOD activities and their contractors and to other Government agencies, in accordance with current AR 70-31, 9 September 1966. Procedures for acquiring these handbooks follow:

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Comments and suggestions on this handbook are welcome and should be addressed to Army Research Office-Durham, Box CM, Duke Station, Durham, North Carolina 27706.

Equilibrators *

INTRODUCTION

1. This handbook, one of a series on Carriages and Mounts, describes equilibrators, their characteristics, functions, requirements and design features.

2. Mobile artillery should have a low silhouette and low center of gravity and yet be able to fire at high, as well as low, elevation. To provide clearance for recoil at high elevation, it is necessary that the recoiling parts be placed well forward with respect to the trunnions. This places the center of gravity of the tipping parts ahead of the trunnions, and creates a muzzle preponderance, or weight moment. This weight moment, without further provision is balanced by a couple applied at the trunnions and elevating gear, but the large force on the gear requires more effort for elevating. Hence, it is desirable to eliminate, or at least reduce, the weight moment by balancing either with counterweights or some mechanical device. A mechanical device is preferred to counterweights because of saving in weight, space, and moment of inertia of tipping parts.

3. The most effective and desirable method of balancing the tipping parts is by the use of an equilibrator. An equilibrator is a force-producing mechanism whose function is to provide a balancing moment. One such moment to be balanced is the muzzle preponderance of the tipping parts. Figure 1 shows a typical equilibrator installation.

4. The muzzle preponderance of artillery is little affected by changes in ammunition weight. However, this must be considered in dealing with other types of weapons, such as missile launchers. Here, the weight of the missile is large compared with that of the tipping parts. After a missile is launched, the weight moment has changed sufficiently to affect equilibrium. Provisions must be made to balance the new weight moment. Equilibrators now being designed will respond to the changing moment.

5. Operators of combat vehicles can neither select level terrain at will nor stop to level an emplacement from which to fire. Invariably, they will be traveling on a grade. As long as their power units are functioning, these weapons can traverse their gun turrets without difficulty. But, if power fails and firing is still demanded, personnel may find it next to impossible to traverse by hand. For conditions such as these, some weapons have azimuth equilibrators to relieve the hand wheel loads. Their use also reduces traversing power requirements. Although the functions of the two equilibrators are similar, the elevation type seldom moves through an arc greater than 90 degrees; whereas, the azimuth type moves through 360 degrees in either direction. The latter can do this readily when equipped with the proper linkage.

*This publication was prepared by Martin Regina and the late Dudley H. Wimer, Jr.

TYPES OF EQUILIBRATORS

6. The equilibrator provides its force either by gas pressure or a spring. The direction in which its force must be applied determines whether it is to be a "pull type" or a "push type". (See figures 2 and 3.) With a pull type, the piston rod is in tension; with a push type the rod is in compression. The location and type of equilibrator depend upon other characteristics of the mount, such as, available space, clearances and silhouette.

7. All equilibrators must be able to apply a force over a variable distance. This is done by having an effective elastic medium in the system. There are three basic types: The most simple is the strut type in which the spring element is self-contained. The second is a gas strut type with a conveniently located accumulator. Both of these can be either pull or push type. In the third type, the force is carried to the tipping parts by means of a chain. This chain may act directly or be wrapped around a cam to control the moment arm more effectively. (See figure 4.)

8. Equilibrators are further identified according to their methods of producing a force. Types, in this sense, are:

(1) Spring
 Coil spring
 Torsion bar
 Clock spring
(2) Pneumatic
(3) Hydropneumatic
(4) Spring-Hydraulic

9. There are three types of spring equilibrators; (1) the coil spring, (2) the torsion bar, (3) and the clock spring. The coil spring type is shown in figure 5. Note that, as illustrated, this is also a pull type. The free end of the rod is pin-connected to the tipping parts, while the far end of the housing is pinned to the top carriage. Maximum compression of the spring is at minimum elevation. As the tipping parts are elevated, the weight moment decreases. Concurrently the springs expand, thereby reducing the equilibrating force. The torsion bar type (fig. 6),

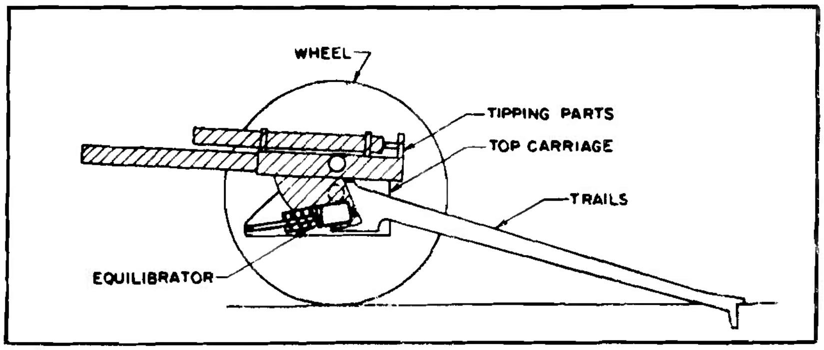


Figure 2. Pull-Type Equilibrator

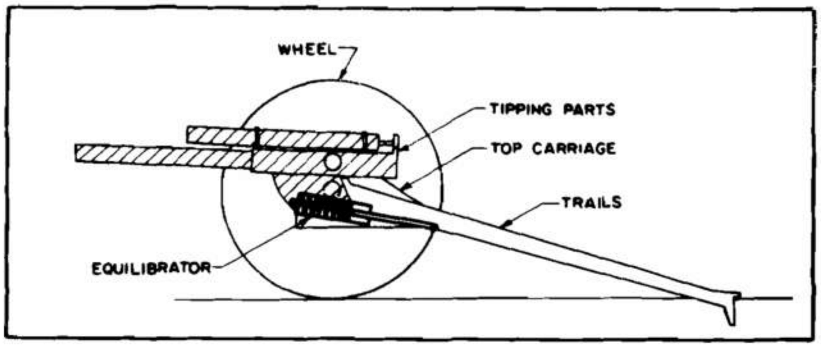


Figure 3. Push-Type Equilibrator

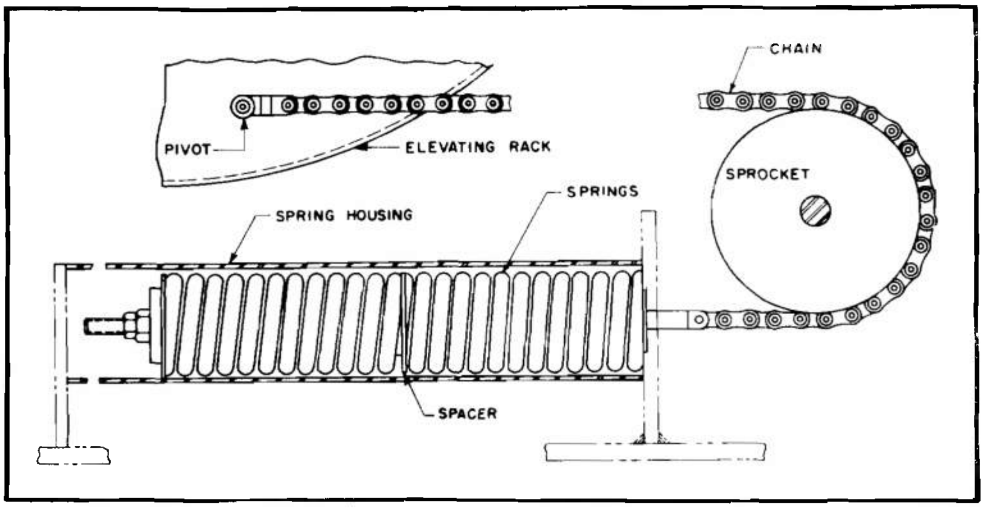


Figure 4. Chain-Type Equilibrator

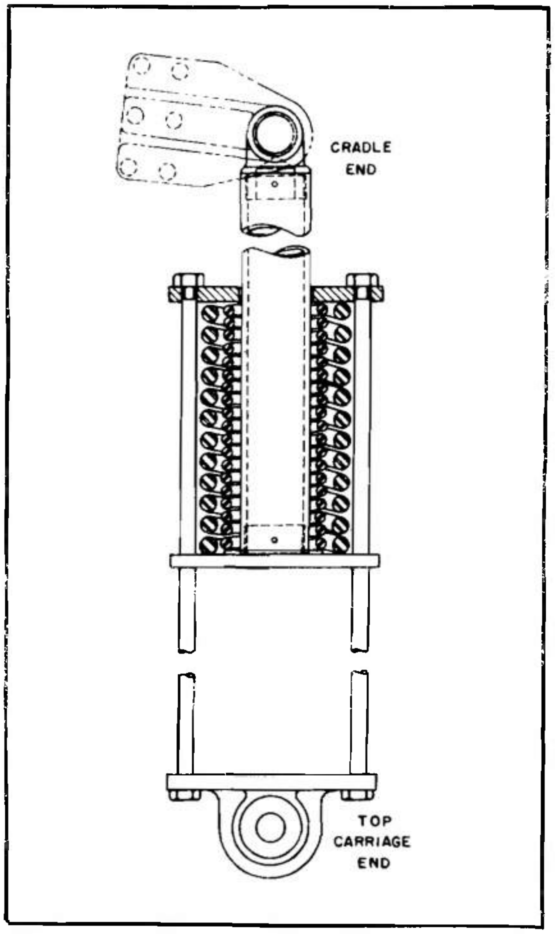


Figure 5. Spring Equilibrator

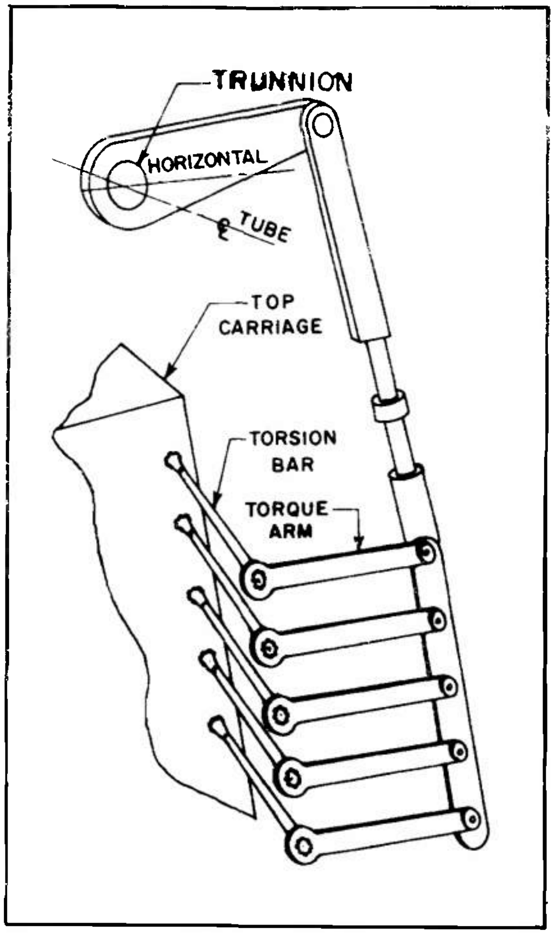


Figure 6. Torsion Bar Equilibrator

instead of coil springs, has one or more torsion bars. The far end of each bar is rigidly connected to the top carriage; the free end being fixed to the torque arm. As the latter rotates during elevation, the bars unwind to reduce the equilibrator moment. And, conversely, when the gun is lowered, the bars wind tighter to approximate the increasing weight moment. The clock-spring type equilibrator (fig. 7) consists of a series of flat, spiral springs. It functions similarly to the torsion bar type by providing a resisting torque to the equilibration system.

10. The pneumatic equilibrator consists of a cylinder and piston. (See figure 8.) It is similar in application to the spring type, but its elastic medium is compressed gas rather than coil springs. Maximum pressure occurs at minimum elevation. As the volume of the gas expands during elevation, the pressure diminishes and the equilibrating force is reduced.

11. In the hydropneumatic equilibrator (fig. 9), the elastic medium is compressed gas. The system consists of a hydraulic cylinder, piston and rod, reservoir, and flexible connecting line. The pressure chamber of the hydraulic strut, all of the line and part of the reservoir are filled with hydraulic fluid. The remainder of the reservoir is filled with gas under pressure. Again, maximum pressure occurs at minimum elevation. As the size of the pressure chamber increases with elevation, fluid under pressure flows from the reservoir through the line, adapter, piston rod, and ports into the chamber. This changes the volumes of fluid and gas in the reservoir, resulting in a reduction of pressure and, hence, a reduction of the equilibrating force. The reservoir may be single or multiple, as required. The hydropneumatic equilibrator lends itself to weapons where weight moments are large, and space may be saved by storing the compressed gas remote from the equilibrator strut.

12. The spring-hydraulic equilibrator combines coil springs with hydraulic pressure. It has been specifically designed to compensate for sudden surges in weight moment, as in missile launchers. Because of its classified status and because it is still in the test stage, it is not further discussed here.

13. Figure 10 is a chart systematizing the types of equilibrators described in paragraphs 6 through 12 and showing their relationship.

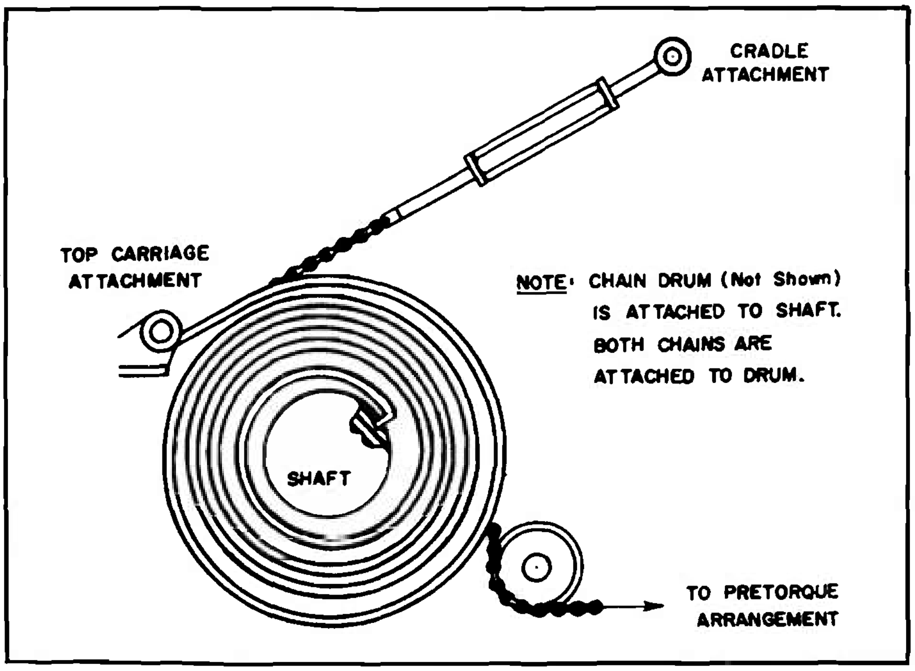


Figure 7. Clock Spring Equilibrator

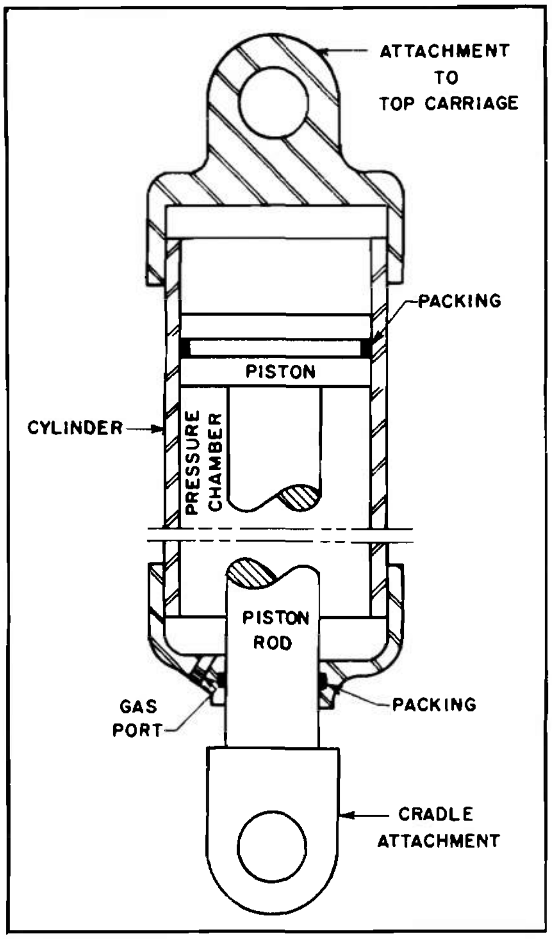


Figure 8. Pneumatic Equilibrator

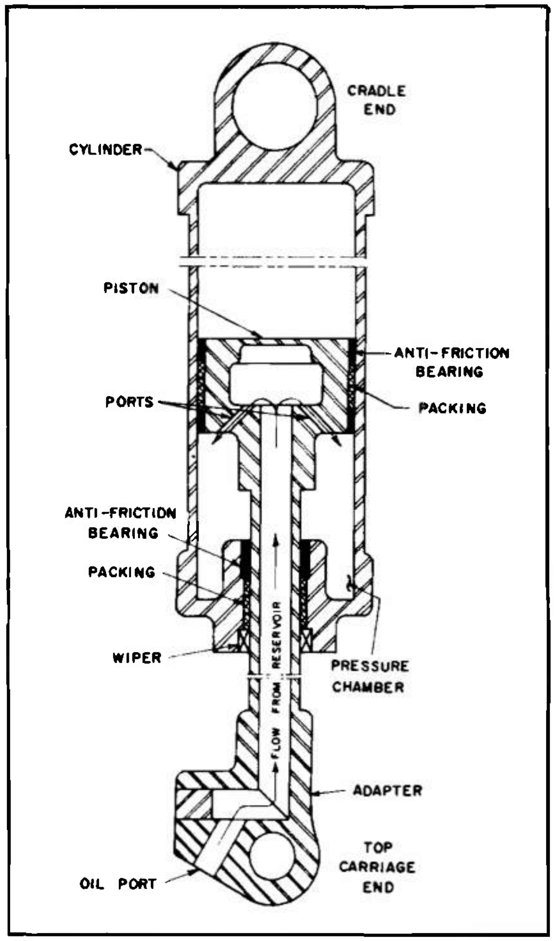


Figure 9. Hydropneumatic Equilibrator

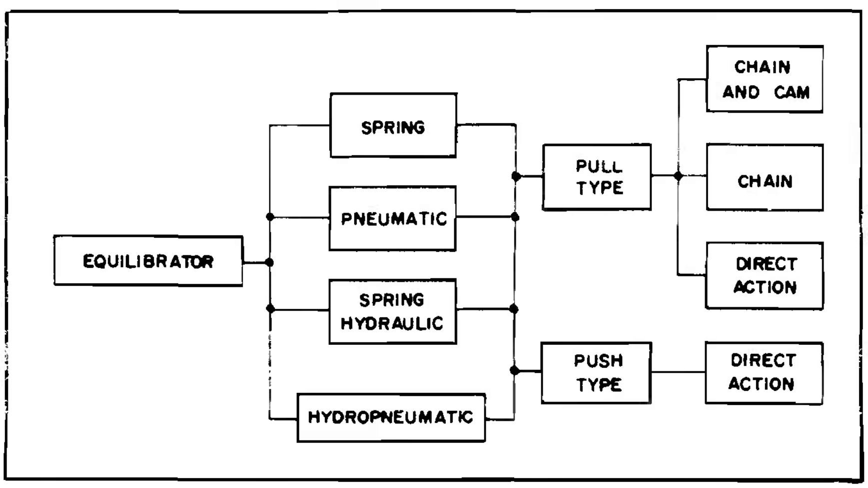


Figure 10. Types of Equilibrators

SELECTING AN EQUILIBRATOR

14. The accompanying table evaluates the four general types of equilibrators according to their desirable characteristics. The lower the number, the higher is the rating.

15. Where mechanical springs are used, small variations in dimensions, such as manufacturing tolerances, affect the spring rate. On the other hand, the rate is not affected by the speed of operation or the atmospheric temperatures, which do influence gas springs.

16. The selection is not simply a matter of choice but a matter dictated by the problems encountered. A spring type is most desirable because of its simplicity. However, the designer may find it necessary to use one of the other types because of space, length of stroke, force required or other factors.

	Spring	Pneumatic	Hydropneumatic	Spring Hydraulic
Simplicity	1	2	3	4
Space*	3	1	2	4
Length of Stroke	4	3	1	?
Minimum Weight	3	1	2	4
Dependability	1	3	2	?
Ease of Maintenance	1	2	2	?
Flexibility of Operation	2	1	1	?
Versatility	2	1	1	1
Minimum Cost	1	2	3	4

*This means space in a local area where it is at a premium. The hydropneumatic type may be larger over-all than some others, but the gas storage can be located outside this critical area.

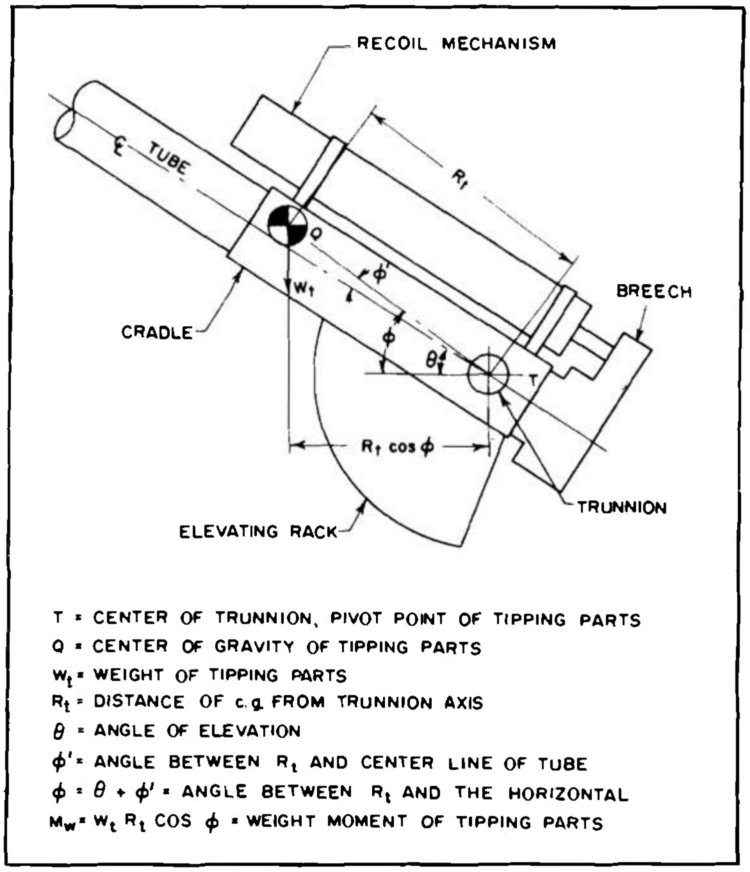


Figure 11. Geometry of Weight Moment of Tipping Parts

EQUILIBRATOR DESIGN

FACTORS INFLUENCING DESIGN

Weight Moment of Tipping Parts

17. The muzzle preponderance or weight moment is the product of the weight of the tipping parts and the horizontal distance of their center of gravity from the trunnion axis. The tipping parts consist of all parts supported by the trunnions. (See figures 11 and 12.)

18. The weight moment is a cosine function, $M_w = W_t R_t \cos \phi$. By manipulating the equilibrating force or its moment arm, or both, in such a way that their product varies as $\cos \phi$, it is theoretically possible to achieve perfect balance for any angle of elevation. This is particularly true of equilibrators having constant spring rates. In others, it is usual to accept a close approximation. Even if the ideal were attempted, other factors such as friction would impair its precision.

Friction

19. Friction in the equilibrator system increases the required input forces and detracts from the performance; thus its magnitude must be computed. Frictional forces cannot be compensated for by the equilibrator, since they change direction with elevation and depression. Friction occurs in the journal bearings at connections to the carriage and cradle and in packings, where used. Cradle trunnion friction may be neglected where low friction bearings are used.

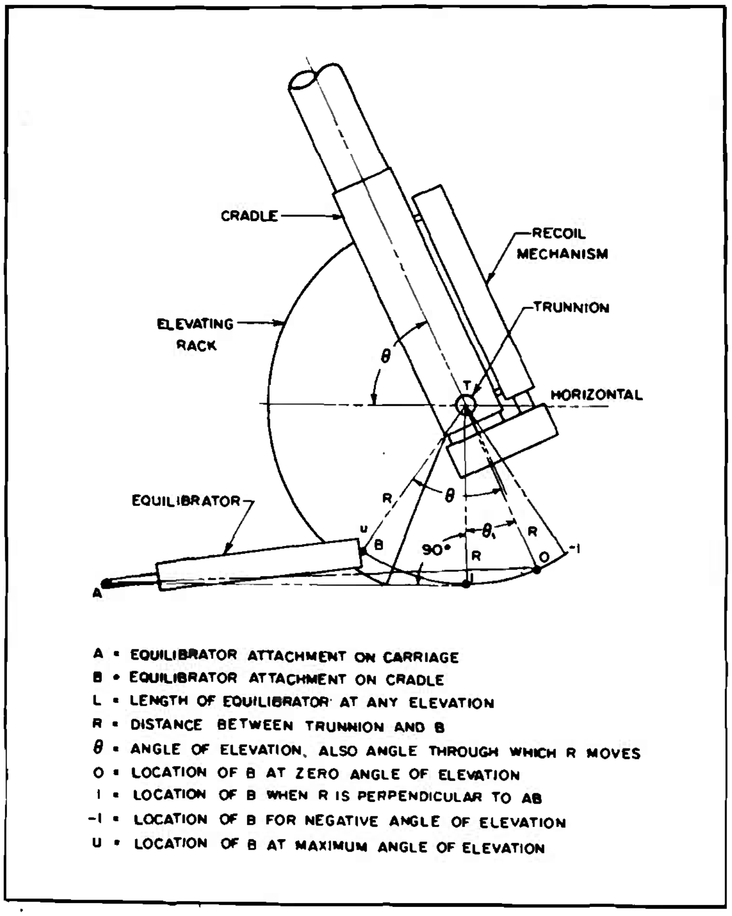


Figure 12. Equilibrator Geometry Showing Components

Packing Friction

20. In pneumatic and hydropneumatic equilibrators, friction of the packings in liquid or gas cylinders must be considered. 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. (See figure 13.) Because of the nearly hydrostatic condition 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.

*For leather, $\mu = 0.05$. For silver, $\mu = 0.09$. (See bibliography, reference 1, page 12.)

21. 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. The ratio of radial pressure to the maximum fluid pressure 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.

22. The packing frictional force is the product of the total radial force exerted by the packing and the coefficient of friction:

$$f_p = \mu F_p \qquad (1)$$

where: f_p = packing friction

F_p = radial force exerted by packing (See detailed discussion in paragraph 71.)

μ = coefficient of friction*

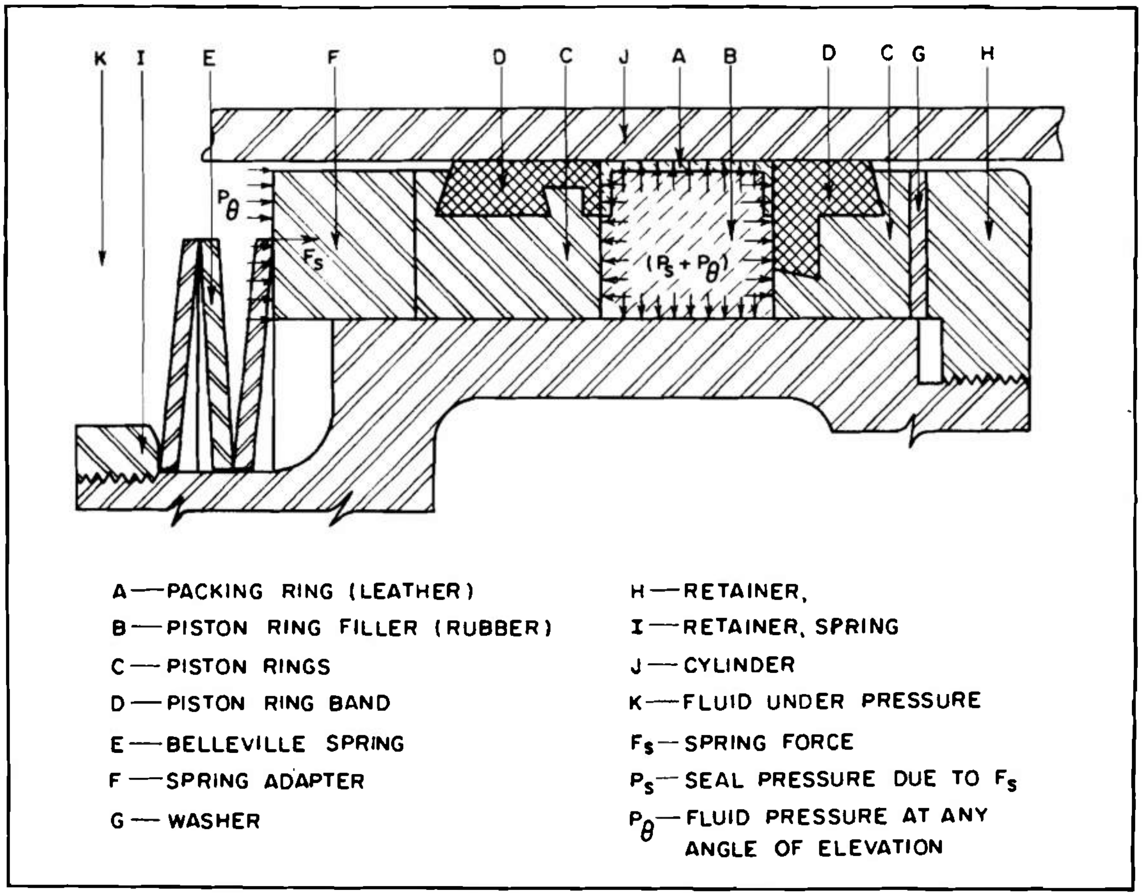


Figure 13. Piston Seal Assembly Showing Pressure Distribution and Applied Loads

BASIC EQUILIBRATOR CALCULATIONS

23. The equilibrator force is applied to the tipping parts, eccentric to the trunnions, to counterbalance the weight moment. As the tipping parts are elevated, the weight moment decreases. Sometimes it is impractical to strive for perfect balance at all angles of elevation, but it is feasible to keep the unbalance very small. The criterion for permissible unbalance is that it must be easily overcome by the elevating handwheel. Even though the weapon may have power elevation, it is still necessary that it be readily hand-elevated in case of power failure.

Equilibrator Placement

24. The position of the equilibrator on the mount is a critical design feature. A layout must be made of the gun carriage in the initial design stage to show the space available at all angles of elevation. From it is determined the most effective geometry, including the proper turning radius, so that the equilibrator can provide the required torque about the trunnions. The geometry is determined first for a perfectly balanced system. If it is not compatible with the available space, the geometry is modified but only to the extent where it still closely approximates the ideal system. This requires a trial and error approach, hence, previous experience or familiarity with the technique becomes a decided asset. Sample problems are presented in Appendixes I and II as a guide for future designs. Once the geometry has been established and if it is maintained, the equilibrator may be located at any convenient place on the mount without changing its effectiveness.

Mechanics of Perfect Balance Equilibrators

25. A method has been established to determine the geometric properties necessary for a spring equilibrator to achieve perfect balance. (See bibliography, reference 2.) It can be used for layout purpose of other types not having constant spring rates but, in these, some unbalance will exist. The force required of an equilibrator depends on the moment arm and stroke available. (See figures 14 and 15.) The turning radius R should be as large as the proportions of the cradle permit. The ratio of c/R determines the efficiency of the spring; the optimum being 1.0, although a ratio as high as 3.0 is not unduly inefficient. (See paragraph 28.) The length of the equilibrator, L_θ, at any angle of elevation θ, may be found by the law of cosines; thus,

$$L_\theta^2 = c^2 + R^2 - 2cR \cos \Psi \quad (2)$$

By the law of sines, $$\frac{R}{\sin \epsilon} = \frac{L_\theta}{\sin \Psi}, \quad (2a)$$

but $$\frac{r}{c} = \sin \epsilon; \quad (2b)$$

therefore, the equilibrator moment arm

$$r = \frac{cR}{L_\theta} \sin \Psi, \quad (3)$$

and stroke $$\Delta L = L_0 - L_\theta, \quad (4)$$

where L_0 is the length of the equilibrator when ϕ equals zero. It should be noted that if the geometry is maintained, the pin on the tipping parts may be located at any point on the circle described by B about T.

26. Perfect balance may be achieved by analyzing the mechanics of the equilibrator system. Either figure 14 or 15 may be used. For perfect balance, the equilibrator moment equals the weight moment at all angles of elevation.

$$M_w = W_t R_t \cos \phi, \text{ weight moment.} \quad (5)$$

When $\phi = 0$, $$M_{w_0} = M_w = W_t R_t, \quad (5a)$$

$$M_e = rF, \text{ the equilibrator moment,} \quad (6)$$

$$M_e = M_w = M_{w_0} \cos \phi. \quad (6a)$$

Combining Equations (5) and (6),

$$Fr = W_t R_t \cos \phi. \quad (6b)$$

But, $$r = \frac{cR}{L_\theta} \sin \Psi. \quad \text{(See eq. 3.)}$$

According to figure 14,

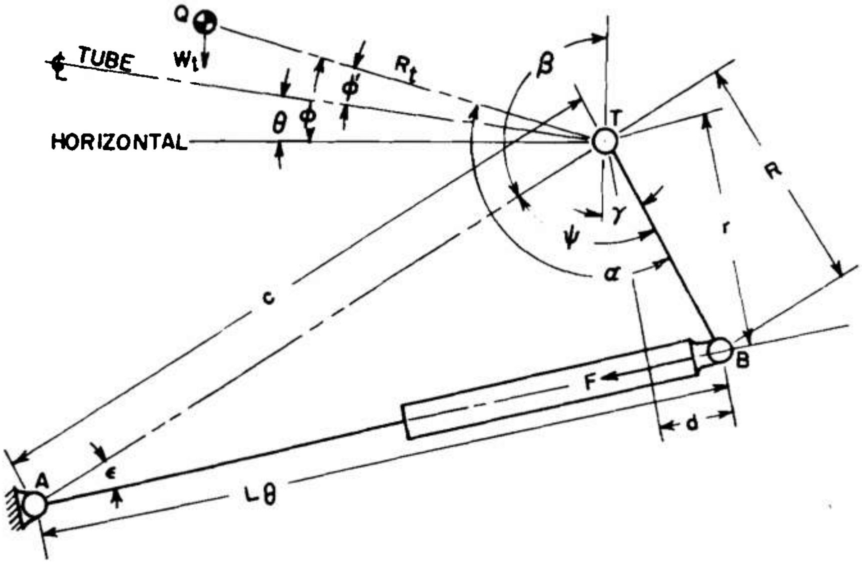
$$\Psi = \alpha - \phi - (\beta - 90) = 90 - (\phi - \alpha + \beta); \quad (7)$$

or, according to figure 15,

$$\Psi = 90 - \phi - \alpha + \beta = 90 - (\phi + \alpha - \beta). \quad (7a)$$

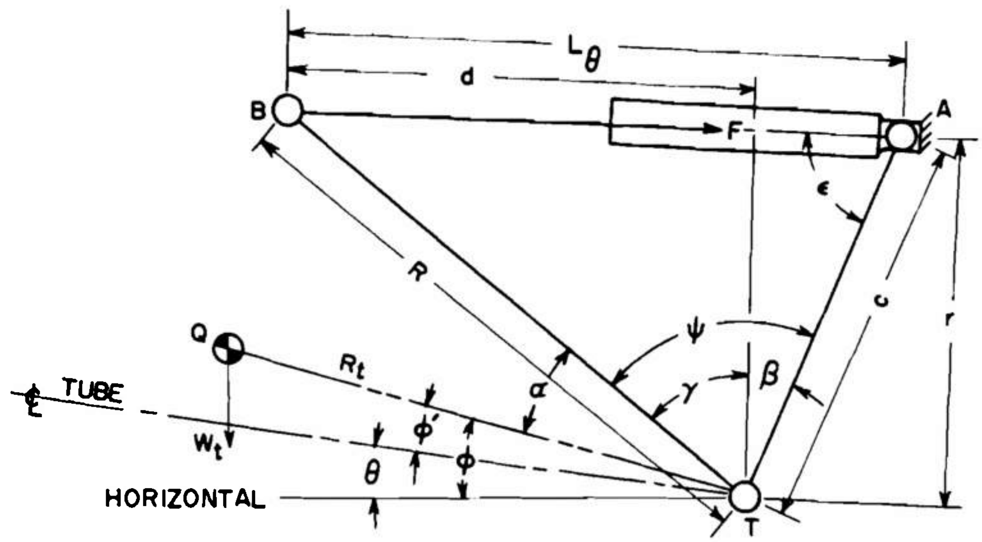
When $\alpha = \beta$, $$\Psi = 90° - \phi; \quad (7b)$$

Therefore, $$\sin \Psi = \cos \phi. \quad (7c)$$



T = CENTER OF TRUNNIONS, PIVOT POINT OF TIPPING PARTS
A = PIVOT POINT OF EQUILIBRATER ON CARRIAGE, FIXED
B = PIVOT POINT OF EQUILIBRATER ON CRADLE, ROTATES ABOUT T
c = DISTANCE BETWEEN A AND T, FIXED
d = DISTANCE FROM B TO PERPENDICULAR FROM T TO L_θ
L_θ = LENGTH OF EQUILIBRATOR, VARIABLE
Q = LOCATION OF C.G. OF TIPPING PARTS
R_t = DISTANCE BETWEEN TRUNNIONS AND Q
R = TURNING RADIUS, CONSTANT
r = EQUILIBRATOR MOMENT ARM, VARIABLE
W_t = WEIGHT OF TIPPING PARTS
F = EQUILIBRATOR FORCE
α = ANGLE FORMED BY R_t AND R, CONSTANT
β = ANGLE FORMED BY VERTICAL LINE AND c, FIXED
γ = ANGLE FORMED BY VERTICAL LINE AND R, VARIABLE
ψ = ANGLE FORMED BY R AND c, VARIABLE
θ = ANGLE OF ELEVATION, VARIABLE
ϕ = ANGLE FORMED BY HORIZONTAL LINE AND R_t, VARIABLE
ϕ' = ANGLE FORMED BY R_t AND ℄ TUBE, CONSTANT

Figure 14. Geometry of Equilibrator Shown Below Trunnions



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W_t = WEIGHT OF TIPPING PARTS
F = EQUILIBRATOR FORCE
α = ANGLE FORMED BY R_t AND R, CONSTANT
β = ANGLE FORMED BY VERTICAL LINE AND c, FIXED
γ = ANGLE FORMED BY VERTICAL LINE AND R, VARIABLE
ψ = ANGLE FORMED BY R AND c, VARIABLE
θ = ANGLE OF ELEVATION, VARIABLE
ϕ = ANGLE FORMED BY HORIZONTAL LINE AND R_t, VARIABLE
ϕ' = ANGLE FORMED BY R_t AND ℄ TUBE, CONSTANT

Figure 15. Geometry of Equilibrator Shown Above Trunnions

When substituting Equations (3) and (7c) in Equation (6b),

$$F\ \frac{cR}{L_\theta}\ \cos\phi = W_t\ R_t \cos\phi, \qquad (8)$$

and
$$F = \frac{W_t\ R_t}{cR}\ L_\theta \qquad (8a)$$

where F is the equilibrator spring force and is directly proportional to the linear distance between A and B, or the length of the equilibrator. Thus the spring force at any position is

$$F = K_s L_\theta, \qquad (8b)$$

where
$$K_s = \frac{W_t\ R_t}{cR}.$$

Now consider the position of the equilibrator when $\phi=0$. The general equation,

$$\Psi = \Psi_0 - \phi, \qquad (9)$$

becomes
$$\Psi = \Psi_0; \qquad (9a)$$

and r becomes
$$r_0 = \frac{cR}{L_0}\ \sin\ \Psi_0. \qquad (10)$$

Likewise, $M_{w_0} = M_{e_0} = r_0 F_0 = W_t R_t, \qquad (11)$

and
$$F_0 = \frac{M_{e_0}}{r_0}. \qquad (11a)$$

The spring force at any angle ϕ is

$$F = F_0 + K_s \Delta L, \qquad (12)$$

where K_s is the spring rate.

Substituting Equations (11a) and (4) in Equation (12) yields

$$F = \frac{M_{e_0}}{r_0} + K_s\ (L_\theta - L_0)\ ; \qquad (12a)$$

or, substituting for M_{e_0} and r_0,

$$F = \frac{M_{w_0}\ L_0}{cR\ \sin\ \Psi_0} + K_s\ (L_\theta - L_0). \qquad (12b)$$

Multiply both sides of Equation (12b) by r and substitute the value for Fr from Equation (6b), the value for r from Equation (3), and the value for Ψ from Equation (9):

$$M_{w_0}\ \cos\ \phi = \frac{cR}{L_\theta}\ \sin(\Psi_0 - \phi) \left[\frac{M_{w_0}\ L_0}{cR\ \sin\ \Psi_0} + K_s\ (L_\theta - L_0)\right] \qquad (13)$$

Rearranging Equation (13),

$$M_{w_0}\left[\frac{L_\theta\ \cos\ \phi}{\sin\ (\Psi_0 - \phi)} - \frac{L_0}{\sin\ \Psi_0}\right] = cRK_s\ (L_\theta - L_0). \qquad (13a)$$

This is the general balance equation. By inspection, the only value of Ψ_0 that will make it independent of the variable angle ϕ is 90 degrees. Inserting this value, Equation (13a) reads

$$M_{w_0}\left[\frac{L_\theta\ \cos\ \phi}{\sin\ (90 - \phi)} - \frac{L_0}{\sin\ 90}\right] = cRK_s\ (L_\theta - L_0). \qquad (13b)$$

But $\sin\ (90-\phi) = \cos\ \phi$ which reduces Equation (13b) to

$$M_{w_0} = cRK_s. \qquad (13c)$$

Two conditions necessary to obtain perfect balance throughout the operating range of spring equilibrators are controlled by the geometry. They are

$$\Psi_0 = 90^\circ \qquad (14)$$

and $K_s = \dfrac{M_{w_0}}{cR}$, the spring rate. (14a)

27. The spring rate equation (14a) has two independent variables, the dimensions c and R (fig. 14 or 15). The turning radius, R, is selected to conform in size to the rest of the structure (par. 25). A discussion on spring efficiency tells how to find a practical length for c. Figure 16 shows the Force-Deflection curve of a spring. Substituting for M_{e_0} and r_0, Equation (11a) may be written:

$$F_0 = \frac{M_{w_0}\ L_0}{cR\ \sin\ \Psi_0}. \qquad (15)$$

But $\Psi_0 = 90^\circ$ (Eq. 14); therefore,

$$F_0 = \frac{M_{w_0}\ L_0}{cR} = K_s\ L_0, \qquad (15a)$$

L_0 is equivalent to the spring deflection when ϕ equals zero. The maximum spring deflection, S_m, occurs at the maximum angle of depression where L_θ becomes L_m and ϕ becomes ϕ_D. Written in terms of the maximum force, F_m, and the spring rate, K_s,

$$S_m = \frac{F_m}{K_s}. \qquad (16)$$

The area under the curve (fig. 16) represents the total spring energy.

$$E_t = \frac{S_m\, F_m}{2} = \frac{F_m^2}{2K_s}. \qquad (17)$$

From Equations (12) and (12b),

$$F_m = F_0 + K_s\,(L_m - L_0). \qquad (18)$$

By substituting the value for F_0 (Eq. 15a) and collecting terms, Equation (18) becomes

$$F_m = K_s\, L_m. \qquad (18a)$$

It should be noted that the values of L_m and S_m are identical. Substituting for F_m in Equation (17) and then for K_s from Equation (14a), the total energy is

$$E_t = \frac{K_s^2\, L_m^2}{2K_s} = \frac{M_{w_0}\, L_m^2}{2cR}. \qquad (19)$$

From Equation (2)

$$L_m^2 = c^2 + R^2 - 2\,cR \cos\Psi, \qquad (19a)$$

but $\Psi = \Psi_0 - \phi_D$, (See eq. 9)

$\Psi_0 = 90°$, (See eq. 14)

and $\cos(\Psi_0 - \phi_D) = \sin\phi_D$.

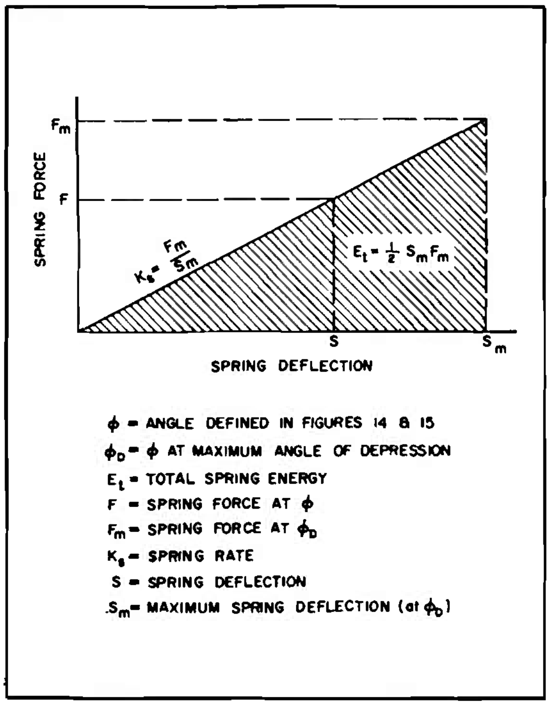


Figure 16. Force-Deflection Curve of Spring

Substituting for L_m^2 in Equation (19),

$$E_t = M_{w_0}\frac{c^2 + R^2 - 2cR\sin\phi_D}{2cR}. \qquad (19b)$$

Rearranging the values,

$$E_t = \frac{M_{w_0}}{2}\left(\frac{c}{R} + \frac{R}{c} - 2\sin\phi_D\right). \qquad (19c)$$

(Note: ϕ_D may be negative.)

This is the general equation for the total spring energy required. It shows that the total energy requirement of the equilibrator spring is dependent on M_{w_0}, the weight moment, when $\phi = 0$; on the value of ϕ_D, when angle of depression is maximum; and on the ratio of c to R, which is the only variable.

Let $$K = \frac{c}{R}. \qquad (19d)$$

By substitution in Equation (19c),

$$E_t = \frac{M_{w_0}}{2}\left(K + \frac{1}{K} - 2\sin\phi_D\right). \qquad (19e)$$

The first derivative of E_t with respect to K is

$$\frac{dE_t}{dK} = \frac{M_{w_0}}{2}\left(1 - \frac{1}{K^2}\right). \qquad (20)$$

Equate the first derivative to zero and solve for K:

$$\frac{M_{w_0}}{2}\left(1 - \frac{1}{K^2}\right) = 0 \qquad (20a)$$

$$K^2 = 1 \qquad (20b)$$

$$K = \pm 1. \qquad (20c)$$

Negative values of K have no physical significance, since c and R are always positive. The second derivative of E_t with respect to K is

$$\frac{d^2E_t}{dK^2} = M_{w_0}\left(\frac{1}{K^3}\right) \qquad (21)$$

When $K = 1$, the second derivative has a value greater than zero, which means that $K = 1$ is a minimum and $c = R$. The total energy requirement (E_t) of an equilibrator spring is, therefore, a minimum when the distance c and R are equal.

28. The effect of a ratio of c to R other than one is shown in figure 17, where K and the corresponding values of E_t are plotted. To simplify matters, the calculations are based on unity. A value is arbitrarily assigned to M_{w_0} that has $E_t = 1.0$ when $K = 1.0$. This is sufficient to show the trend in the curve as K changes. In Equation (19e), assume that

$$M_{w_0} = 0.852,$$

and $$\phi_D = -10°,$$

so that

$$E_t = 0.426 \left(K + \frac{1}{K} + 0.348\right). \quad (21a)$$

Figure 17 shows that the minimum value of E_t occurs when $K = 1$. It is not always feasible to design an equilibrator with the minimum ratio but this does not prove objectionable. The ratio K may be increased to 3.0 with little more than 50 percent additional spring energy required. The equilibrators of some modern weapons have this feature.*

Mechanics of Approximately Balanced Systems

29. When a perfect balance equilibrator is not feasible due to obstructions that limit its geometry, the engineer must resort to other methods that closely approximate the perfect system. For the first attempt to establish a suitable geometry, it is recommended that the geometry of a perfect balance system be attempted first, and then altered to conform to the available space. The weight moments to be balanced at various angles of elevation divided by the corresponding moment arms of the equilibrator about the trunnions determine the theoretical required forces. These forces, when plotted against equilibrator travel, will not fall on a straight line. (See figure 18.)

30. For determining the equilibrator's spring constant and initial force, the following three methods are available:

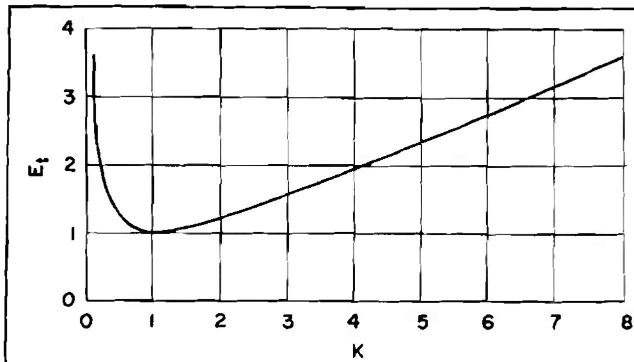
Method 1 — Calculate forces so as to balance the weight and equilibrator moments at two angles of elevation; one very low, the other very high. Usually, the resulting unbalance throughout the rest of the elevating range is slight enough to be acceptable. Ordinarily, the low angle is selected at zero. The high angle is from 5 to 10 degrees less than maximum elevation. For example, assume that the two angles selected are $\theta = 0$ and $\theta = u$. The weight moments at these elevations are M_{w_0} and M_{w_u}, and the corresponding equilibrator moment arms are: r_0 and r_u.

$$F_0 = \frac{M_{w_0}}{r_0}, \text{ equilibrator force at } 0°, \quad (22)$$

$$F_u = \frac{M_{w_u}}{r_u}, \text{ equilibrator force at } u°, \quad (22a)$$

$$K_s = \frac{F_0 - F_u}{\Delta L}, \text{ spring rate}, \quad (22b)$$

*One of these is a 105mm Howitzer, M2A2.



K	E_t
1/8	3.61
1/4	1.96
1/2	1.21
1	1.00
2	1.21
4	1.96
8	3.61

Figure 17. Trend Curve of Equivalent Total Energy

where ΔL is the equilibrator stroke between 0° and u° elevation. After the spring rate is known, the equilibrator force at any angle of elevation may be found.

Method 2—Draw a straight line that most closely fits the theoretical force *vs.* elevation curve (fig. 18). The initial force, F_0, is read directly from the curve and the slope of the straight line determines the spring constant.

Method 3 — The most accurate method is to apply the theory of least squares to the forces on the equilibrator that are required to balance the weight moments at all angles of the elevation.

If F = equilibrator force, lb,

F_0 = initial equilibrator force, lb,

K_s = spring constant (or rate) of equilibrator, lb/in

and ΔL = equilibrator travel, in,

then $F = F_0 - K_s \Delta L$. (23)

Summing up the forces at all increments of travel:

$$\Sigma F = NF_0 - \Sigma K_s \Delta L. \quad (24)$$

Summing up the moments of the forces at all increments about point 0 (fig. 18):

$$\Sigma F \Delta L = F_0 \Sigma \Delta L - K_s \Sigma \Delta L^2. \quad (25)$$

Solving Equations (24 and 25) simultaneously, values for K_s and F_0 are obtained. For an example, determine the equilibrator's spring rate and initial force for the theoretical required forces given in figure 18. If ΔL is taken in one-inch increments over a total distance of 12 inches, the terms can be tabulated as follows:

N	ΔL	ΔL^2	F	FΔL
1	0	0	1300	0
2	1	1	1130	1130
3	2	4	1070	2140
4	3	9	1020	3060
5	4	16	960	3840
6	5	25	930	4650
7	6	36	880	5280
8	7	49	860	6020
9	8	64	820	6560
10	9	81	780	7020
11	10	100	730	7300
12	11	121	670	7370
13	12	144	620	7440
	Σ=78	650	11770	61810

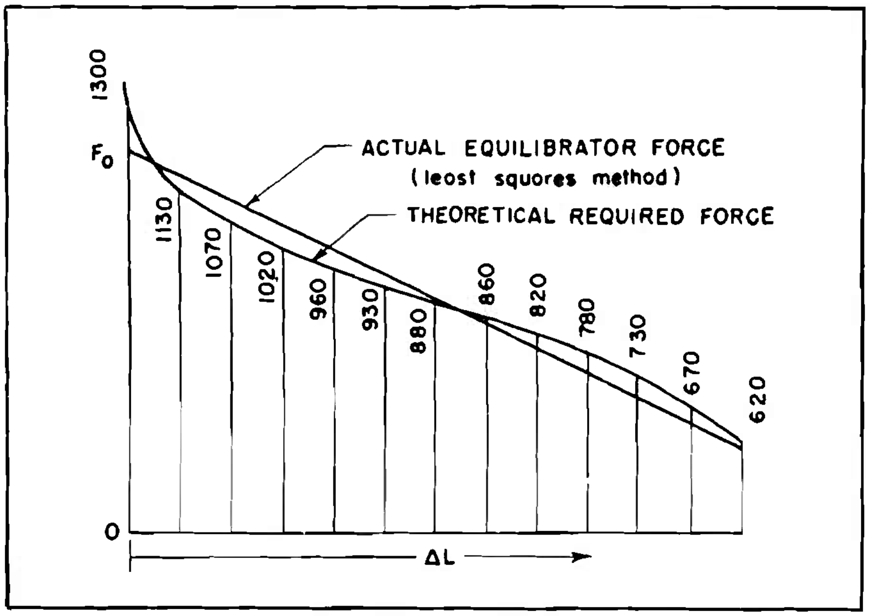


Figure 18. Equilibrator Force Curve

Writing the equations for ΣF and $\Sigma F \Delta L$, and solving for K_s,

$$11770 = 13F_0 - 78K_s,$$

$$61810 = 78F_0 - 650K_s,$$

therefore, $K_s = 48.406$ = spring rate of equilibrator, lb/in.

and $F_0 = 1196$ initial force, lb.

Of the three methods discussed above, the first is simplest and is usually satisfactory. The others are suggested for applications where greater accuracy is required for spring equilibrators not having perfect balance.

Mechanics of a Torsion Bar Equilibrator

31. The torsion bar equilibrator (fig. 6) may be considered to be another spring equilibrator having a different linkage, since its torque is developed by twisting a round bar. Although it does not lend itself to a perfect balanced system, the torsion bar equilibrator can provide a torque that closely approximates the weight moment. If the proportions of the links in figure 19 are maintained, the torque and weight moment will not vary more than 3 percent over a span of 60 degrees. Other linkages or variations in this type may improve the stated performance

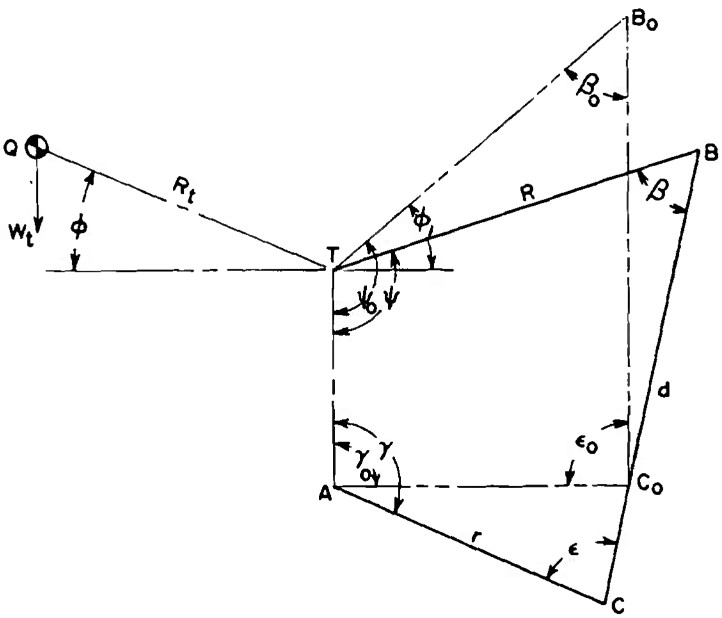


Figure 19. Torsion Bar Linkage

limits. The four-bar linkage of figure 19 amply illustrates the fundamental features of this type equilibrator. As the gun elevates, link R rotates clockwise about the trunnion T, and link d forces link r to turn in the same direction, untwisting the torsion bar at A.

32. The rigidity of the torsion bar is determined by the weight moment of the tipping parts. This is achieved by equating the required torsional resistance at two positions of the gun, usually at zero degrees elevation and at an elevation 10 degrees less than the maximum. These elevation limits may be varied if better balance can be realized. Returning to figure 19,

$M_w = W_t R_t \cos\phi$, weight moment, (See eq. 5)

$$F_d = \frac{M_w}{R \sin\beta}\text{, tensile load in link } d, \quad (26)$$

and $T_t = F_d r \sin\epsilon$, required torque, torsion bar. (26a)

Assume that the weight moments are balanced at elevations where $\phi = 0$ and $\phi = \phi_1$. For each of these elevations determine the angles and the values of M_w, F_d and T_t. Arrange the geometry so that γ and ϵ are right angles when $\phi = 0$, and that Ψ is a right angle at ϕ_1; hence $\Psi_0 = 90° + \phi_1$. This size is not absolutely essential, but it offers a good starting point. Sufficient information is now available to determine the spring constant of the torsion bar:

$$\Delta\gamma_1 = \gamma_1 - \gamma_0 \text{ (rad.)}, \quad (27)$$

$$K_t \Delta\gamma_1 = (T_0 - T_1), \quad (27a)$$

$$K_t = \frac{T_0 - T_1}{\Delta\gamma_1}, \quad (27b)$$

where K_t = torsional spring constant.

With the spring constant known, the counterbalancing moment at the trunnion can be calculated for all elevations:

$$T_s = K_t \Delta\gamma, \quad (28)$$

$$M_s = T_s \frac{R \sin\beta}{r \sin\epsilon}, \quad (29)$$

where T_s = torque produced by the torsion bar,

M_s = torque of torsion bar applied at the trunnion.

33. All that is known of the torsion bar unit is its spring constant K_t which may be that of one bar or the total of several bars. Should a unit that uses only one bar prove cumbersome, it is well to consider a multiple torsion bar assembly, principally to conserve weight and space. The torque on each bar becomes

$$T_s' = \frac{T_s}{n}, \quad (29a)$$

where n = number of torsion bars.

In some units, the spring consists of a torsion bar assembled inside a tube, both serving as components of the spring. Two parameters govern the design of the torsion bar; the deflection and the allowable stress. Most torsion bars have integral serrated hubs at each end which alter the torsional rigidity of the active length; thus, the ordinary deflection formulas for round bars do not apply. Modified versions of these formulas are available for this type of construction. (See bibliography, reference 3.)

34. Another arrangement of a torsion bar assembly is shown in figure 20. This one uses a chain to join the torsion bar arm to the tipping parts. Its size and proportions are restricted to the available space on the mount. Throughout the elevation range, the response is computed similarly to that of other equilibrators:

$$M_w = W_t R_t \cos\phi. \quad \text{(See eq. 5)}$$

The chain tension is $$F_c = \frac{M_w}{r}, \quad (30)$$

where $$r = \frac{cR}{L}\sin\Psi, \quad \text{(See eq. 3)}$$

and $$T_s = r_1 F_c, \quad (30a)$$

where $$r_1 = \frac{c_1 R_1}{L_1}\sin\Psi_1. \quad \text{(See eq. 3)}$$

The computed values of T_s near the angles of elevation of $\phi = 0$ and $\phi = \phi_{max}$ determine the spring rate. For the first trial, the angles, Ψ and Ψ_1, should be approximately 90 degrees. If the difference between weight moment and equilibrator moment is large, the geometry should be revised to make them more nearly equal.

Mechanics of a Clock-Spring Type Equilibrator

35. The geometry of the linkage for the clock-spring type equilibrator (fig. 21) is similar to the compression spring type to the point where perfect balance can be attempted. Perfect balance cannot be achieved theoretically since point A is not fixed. However, point A may be assumed fixed for the initial conditions. Thus, at $\phi = 0$ and $\Psi = 90°$,

$$K_s = \frac{M_{w_0}}{cR}, \quad \text{(See eq. 14a)}$$

and

$$F_c = K_s L, \quad \text{(See eq. 8b)}$$

where F_c = chain tension.

The distance, L, also is equivalent to the distance that the chain must move from A to B to wind the spring to the desired torque. The torque on the spring becomes

$$T_s = F_c R_s, \quad (31)$$

where R_s = radius of the chain drums; and the bending stress is

$$\sigma = \frac{6T_s}{bh^2}, \quad (31a)$$

where b = width of spring,
h = thickness of the spring material.

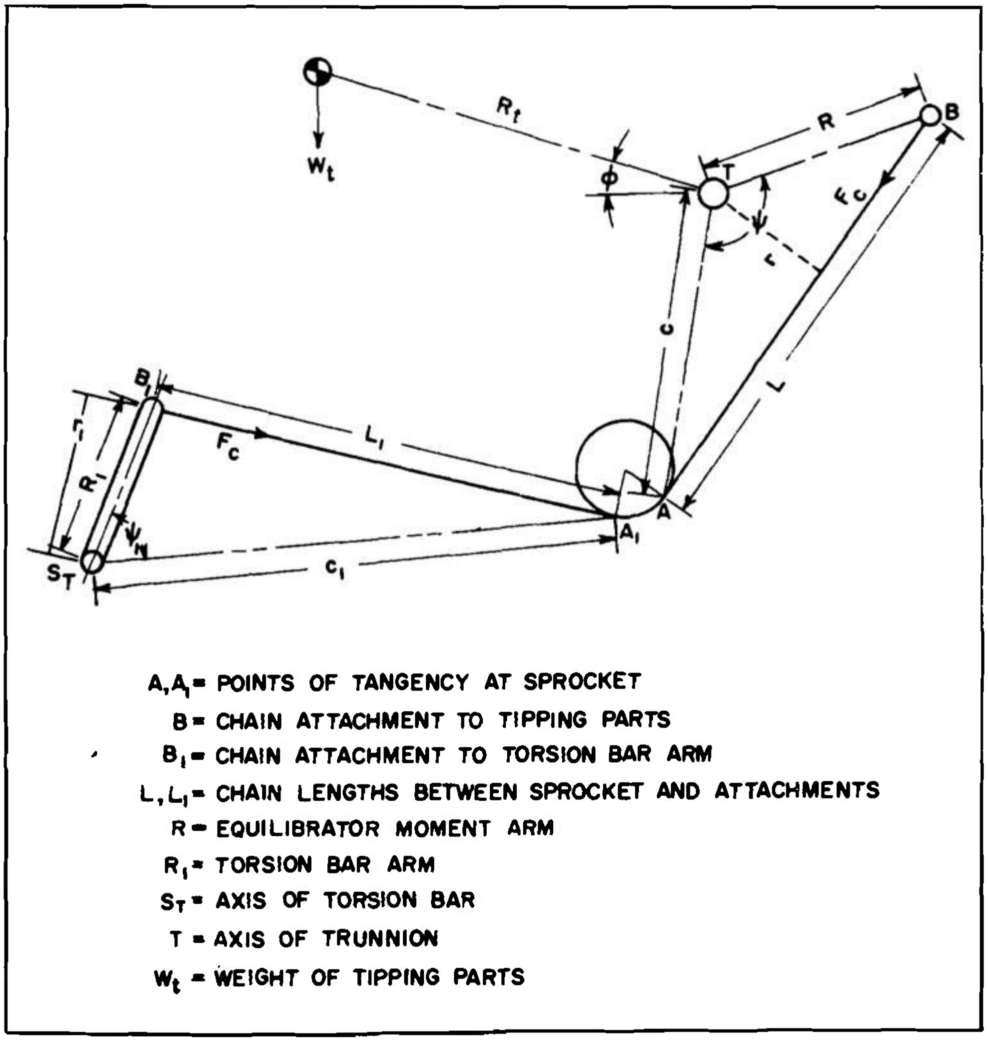


Figure 20. Torsion Bar Equilibrator — Chain Type

If the width is large and unwieldly, several springs may be used, with b representing the total width. The active length of the spring is

$$L_s = \frac{LEI}{F_c R_s^2},^* \qquad (31b)$$

where E = modulus of elasticity,

I = moment of inertia of the total cross section.

The investigation to obtain the appropriate size of the spring and chain drum is largely a matter of a trial and error procedure. After a spring has been selected, a detailed analysis of the equilibrator moment may be computed.

Referring to figure 21, angle ψ' is known for any angle ϕ. Then

$$d^2 = R^2 + e^2 - 2eR \cos \psi'. \qquad (31c)$$

Because L is always perpendicular to R_s,

$$L^2 = d^2 - R_s^2.$$

Sufficient information is now available to solve for all the remaining dimensions of triangles ABA', $BA'T$, and BAT necessary to complete the equilibrator analysis.

*This formula has been derived from spring formulas found in bibliography reference 7.

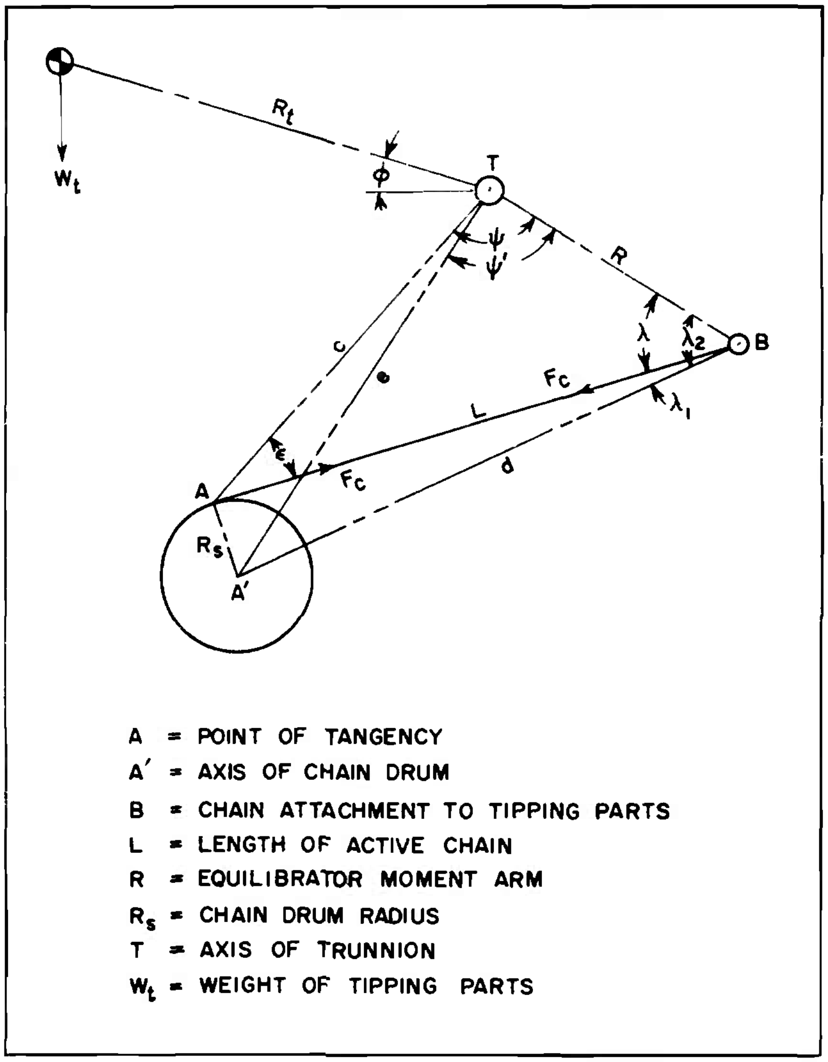


Figure 21. Geometry of Clock Spring Equilibrator

Mechanics of a Phase Adjustment Device

36. Equilibrators are designed to balance the tipping parts while the weapon is resting on a horizontal plane. If the plane becomes oblique, the weight moment changes, and the built-in equilibrator moment no longer matches it; i.e., the equilibrator will be out of phase in terms of some angle. To bring the equilibrator into phase, an adjustable linkage, assembled to the unit, changes the equilibrator force and moment arm to re-establish the balanced condition. A sketch of this linkage is shown in figure 22.

37. The mechanics of the equilibrator are identical to those of paragraphs 25 through 30, if the geometry of the line diagram in figure 23 is adopted; i.e., the general shape of $TT_1B_1B_1'$ is a parallelogram, becoming rectangular when $\phi = 0$. The system offers perfect balance when the weapon rests on a horizontal plane. Now suppose that the weapon has a pitch of angle ϵ which changes the weight moment to

$$M_w = M_{w_0} \cos (\phi + \epsilon). \qquad (32)$$

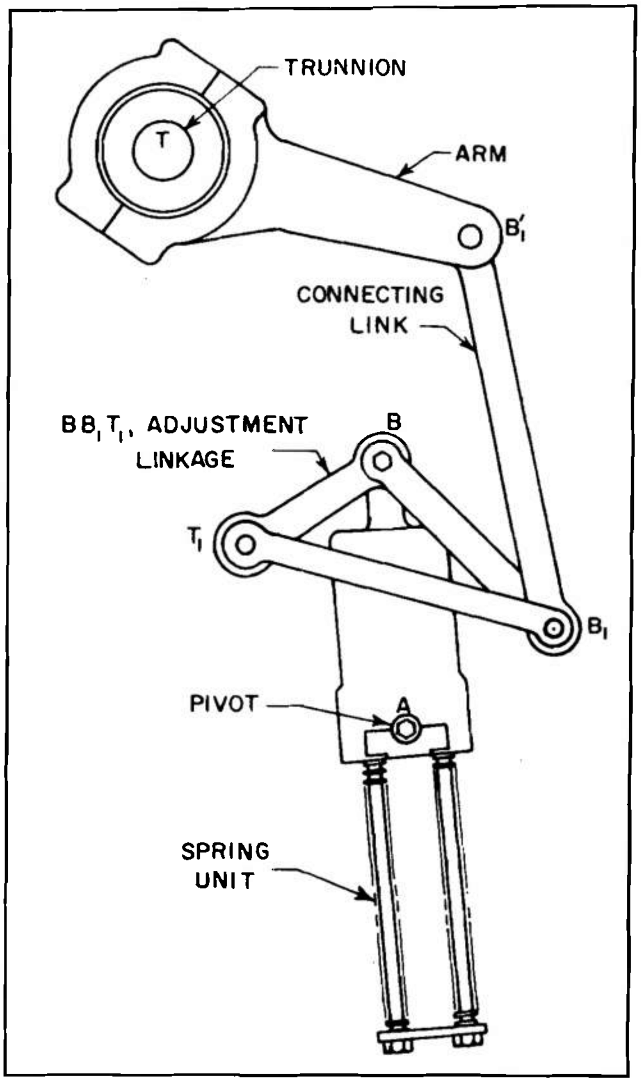


Figure 22. Spring Equilibrator with Phase Adjustment

We know that the equilibrator moment should balance the weight moment, and by manipulating the values of Equations (5a), (6a), (6b), (8a), and (14a), we arrive at the equation

$$\frac{M_{w_0}}{cR} L_\theta r = M_w. \tag{32a}$$

But $r = \frac{cR}{L_\theta} \sin \Psi$ (Eq. 3), and substituting for M_w in Equation (25) we have

$$\frac{M_{w_0}}{cR} L_\theta \cdot \frac{cR}{L_\theta} \sin \Psi = M_{w_0} \cos (\phi + \epsilon). \tag{32b}$$

Reducing to the simplest terms,

$$\sin \Psi = \cos (\phi + \epsilon)$$

$$= \sin [90 - (\phi + \epsilon)], \tag{32c}$$

and

$$\Psi = 90 - (\phi + \epsilon). \tag{32d}$$

Thus, if the weapon pitches through an angle, ϵ, angle Ψ must be changed by that angle. This can be readily accomplished by making link BB_1 an adjustable strut. Then with proper automatic sensing devices and controls, suitable balance can be achieved whatever the pitch angle may be.

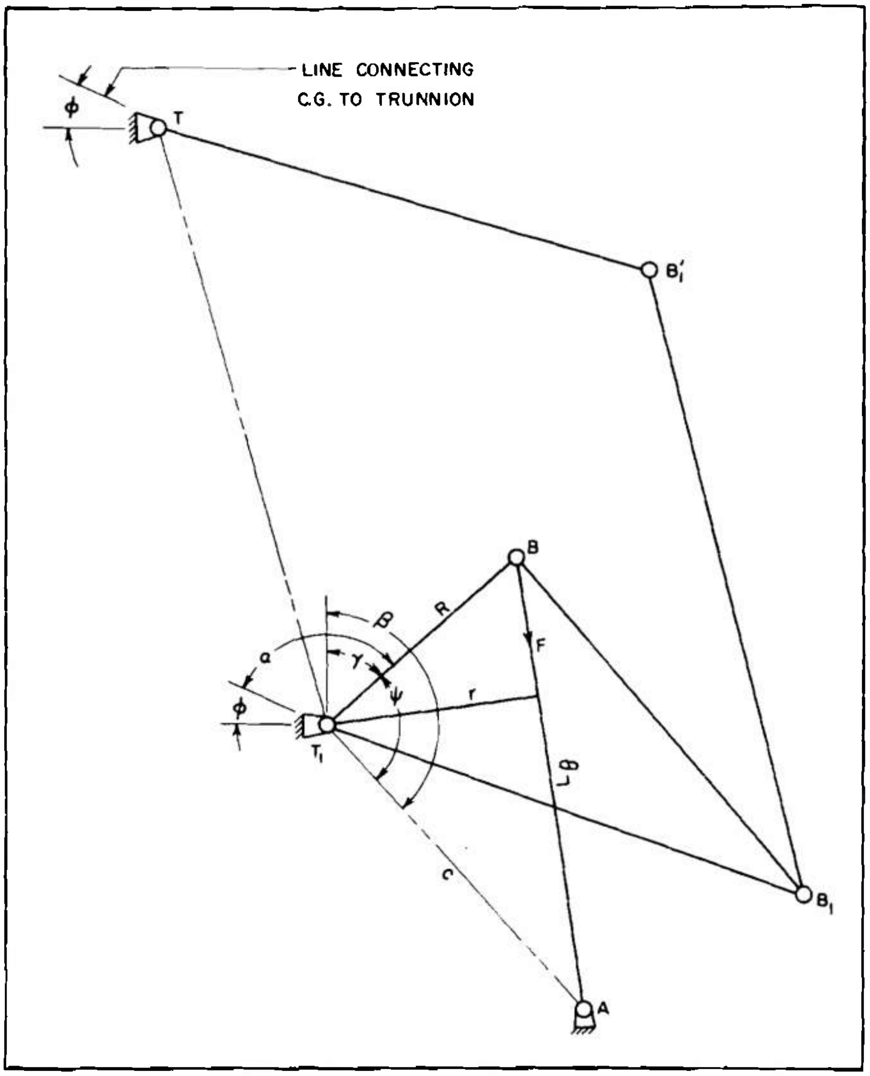


Figure 23. Line Diagram of Phase Adjustment Linkage

Mechanics of an Azimuth Equilibrator

38. Figure 24 shows a spring-type azimuth equilibrator. This type lends itself readily to a perfect balance system; the geometry being similar to its elevation counterpart. The line diagram in figure 25 shows the various positions and geometry of the equilibrator as the turret traverses 360 degrees. Following the same procedure as outlined in paragraphs 25 through 30, with reference to figure 25, the equilibrator moment can be shown to equal the weight moment for all positions of the turret after the spring rate has been established. The component of the weight causing the unbalance is

$$F_w = W_t \sin \theta_t, \qquad (33)$$

where W_t = weight of traversing unit,

θ_t = slope of the terrain.

From Equation (5), substituting ϕ_a for ϕ, the weight moment becomes

$$M_w = F_w R_t \cos \phi_a, \qquad (33a)$$

where ϕ_a locates the center of gravity of the traversing parts with respect to the horizontal line passing through the center of the turret.

When $\phi_a = 0$, set $\Psi = 90°$ and according to Equation (14a), the spring rate is

$$K_s = \frac{F_w R_t}{cR}; \qquad (33b)$$

and from Equation (8b) the spring force at any position is

$$F = K_s L. \qquad (33c)$$

39. The preceding discussion involves an azimuth equilibrator which balances the weight moment for a weapon resting on a constant slope. As neither the slope of the terrain nor the position of the vehicle on it remain constant, the equilibrator must have some means of compensating for the unbalanced moment changes. As the slope increases or decreases, the length of link R changes

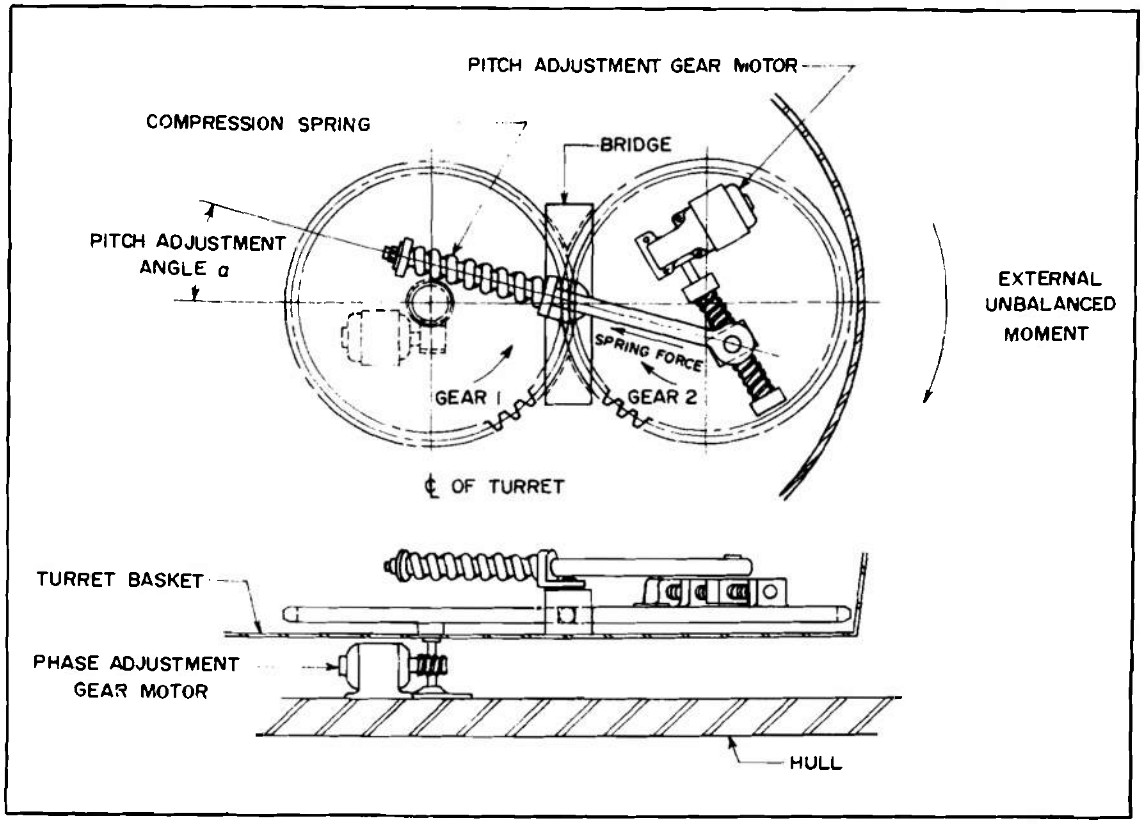


Figure 24. Spring-Type Azimuth Equilibrator

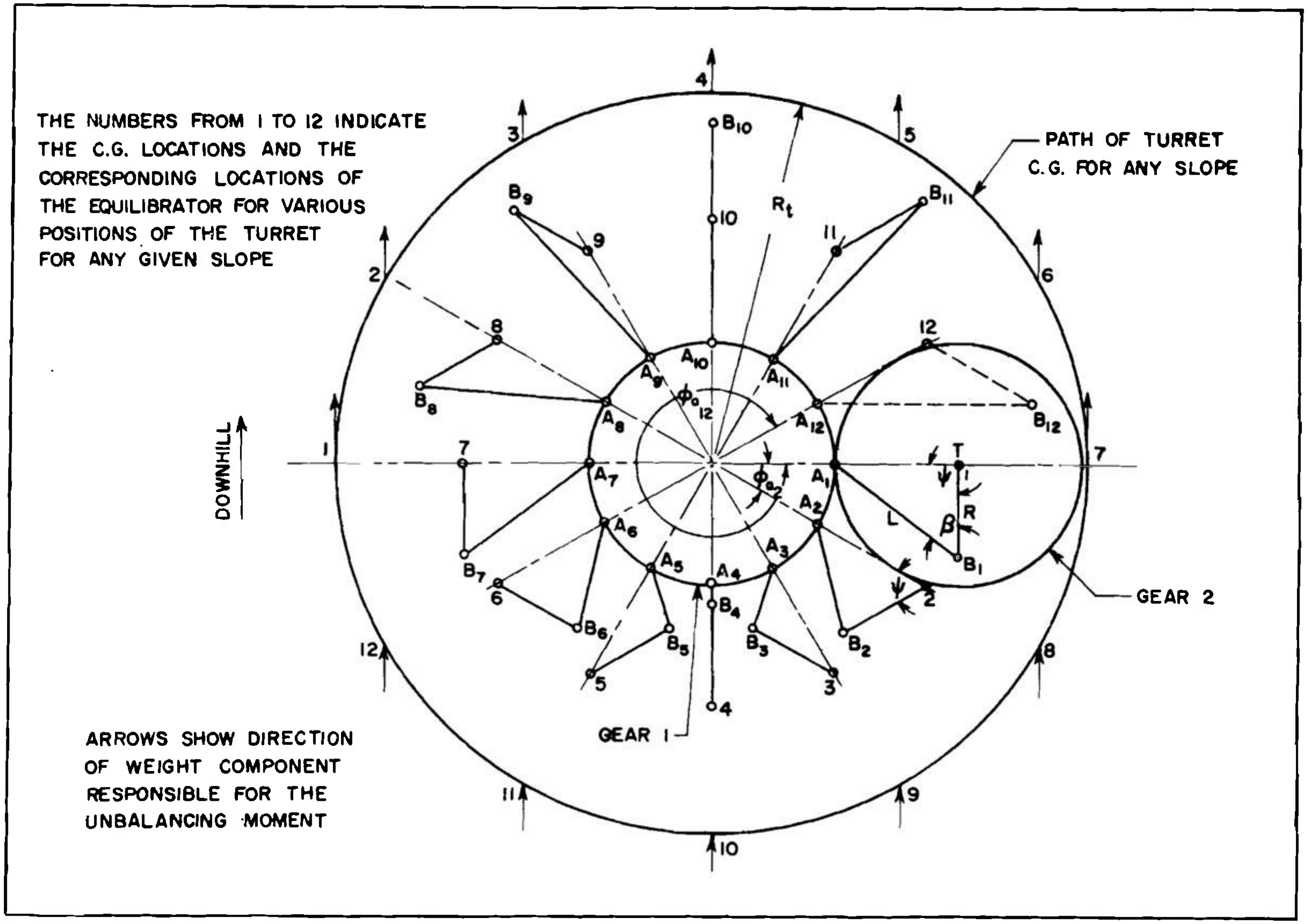


Figure 25. Azimuth Equilibrator Positions

correspondingly. (See figures 24 and 25.) If the vehicle shifts its position on a given slope so that the equilibrator and weight moments are no longer compatible, the phase angle β is adjusted by turning the gears.

Performance Calculations

40. Equilibrator performance calculations are most conveniently presented in tabular form. The calculations listed in the following paragraphs should be made for each value of θ at regular increments over the entire elevating range. Sample calculations are given in Appendixes I and II.

Performance of a Spring-Type Equilibrator

41. For the spring-type equilibrator, the performance is identical for either manual or power operation. In addition to the geometry and weight moment calculations, three sets of calculations are needed. (See sample calculations in Appendix II.)

42. First, the equilibrator moment must be calculated without considering friction. This determines the required spring characteristics including spring rate, size, and strength. Referring to figures 11, 12, 14 and 15, the weight moment, lb-in, from Equation (5) is

$$M_w = W_t R_t \cos \phi;$$

the equilibrator moment, lb-in, from Equation (6) is

$$M_e = Fr.$$

In the above,

ϕ = angle of elevation of center of gravity of tipping parts

θ = angle of elevation, degrees

$F = K_s [S - (L_0 - L)]$ = spring force, lb, equivalent to F of hydropneumatic equilibrators (34)

$[S - (L_0 - L_\theta)]$ = net spring deflection, in.

S = spring deflection, in., when $\phi = 0$

S_m = maximum spring deflection

L_0 = equilibrator length, in., when $\phi = 0$

L_θ = equilibrator length, in., at any angle, θ

r = moment arm, in.

In conformance with *Method 1* in paragraph 30, the equilibrator force should be made equal to M_w/r at some angle near the maximum angle of elevation and at $\theta = 0°$, although the minimum angle may be less than 0°.

43. Next, the performance during elevation is computed, including the effect of friction. Friction is considered only in the end connection bearings, and is computed by isolating the equilibrator (line AB) as shown in figure 26. The calculations for the equilibrator moment are similar to those in paragraph 42 except that friction is considered. The motion and positions of the cradle and equilibrator may be observed in figures 12, 14 and 15.

$F = K_s [S - (L_0 - L_\theta)]$ = spring force, lb (Eq. 34)

$F_R = F$ = net rod force used for computing friction

$T_A = \mu F_R r_A$ = frictional torque, lb-in, at bearing A, on carriage (fig. 26) (34a)

$F_A = f(T_A)$ = load in rod, lb, due to friction in bearing A (fig. 26) (34b)

$T_B = \mu F_R r_B$ = frictional torque, lb-in, at bearing B, on cradle (fig. 26) (34c)

$F_B = f(T_B)$ = load in rod, lb, due to friction in bearing B, (fig. 26) (34d)

$F_f = F_A + F_B$ = total frictional load in rod, lb (34e)

$F = F_R - F_f$ = net force on equilibrator rod; i.e., spring load, lb (34f)

$M_e = F r$ = equilibrator moment, lb-in (34g)

$T_e = M_w - M_e$ = torque required to elevate, lb-in (at trunnions). (34h)

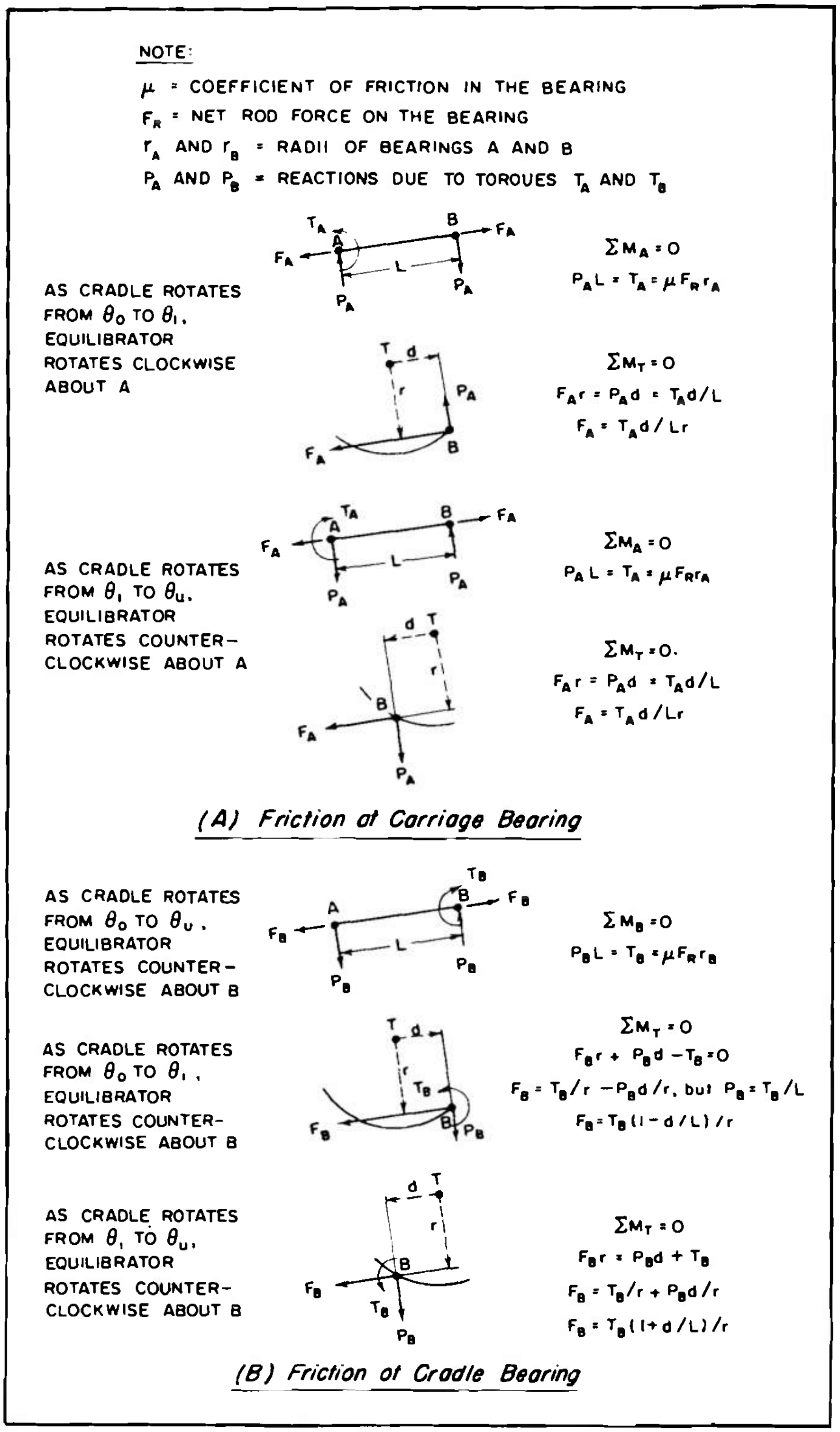


Figure 26. Friction Moments During Elevation

44. Finally, the performance during depression is computed. The procedure is the same as that for elevation except that the frictional forces (fig. 27) change direction; i.e., F_f will be negative:

$F = F_R - F_f$ = net force on rod, lb (See eq. 34e)

$M_e = Fr$ = equilibrator moment, lb-in (See eq. 34g)

$T_e = M_e - M_w$ = torque required to depress, lb-in, (at trunnions). (35)

Performance of Pneumatic and Hydropneumatic Equilibrators

45. For pneumatic and hydropneumatic types, power operation performance differs from manual. Manual operation is slow, and expansion or contraction of the gas may be considered isothermal. Power operation is rapid, and expansion or compression is polytropic. There are, consequently, more tables of calculations necessary than with the spring type. Usually, there are six:

(1) The weight moment, as given by Equation (5).

(2) The equilibrator moment, based on the gas force obtained from isothermal expansion and neglecting friction.

(3) The equilibrator moment during manual elevation, considering friction.

(4) The equilibrator moment during manual depression, considering friction.

(5) The equilibrator moment during power elevation, considering friction.

(6) The equilibrator moment during power depression, considering friction.

46. When based on isothermal expansion with friction neglected, the gas spring behaves like a mechanical spring. The equilibrator moment equals the weight moment at two places, as in paragraphs 30 and 42. This fixes the initial gas pressure and volume from which all other values are derived. Atmospheric pressure of 15 psi is used instead of the more correct value of 14.7 psi, with negligible error.

P_θ = gas pressure at any angle of elevation,

$P_\theta = P_a - 15$ = gas pressure, psig (36)

$P_a = \frac{(P_0 + 15)\ V_0}{V_\theta}$ = pressure, psia (36a)

$F_g = AP_\theta$ = gas force, lb, where A is the effective piston area, in^2 (36b)

P_0 = gas pressure at zero elevation, psig

V_0 = gas volume at zero elevation, in^3

$\Delta L = (L_0 - L_\theta)$ = equilibrator stroke, in. (See eq. 4)

$V_\theta = V_0 + A\Delta L$ = gas volume, in^3 (36c)

$F = F_g$ = equilibrator force, lb, since friction is not considered.

$M_e = Fr$ = equilibrator moment, lb-in (See eq. 6)

$M_w = W_t R_t \cos \phi$ = weight moment, lb-in. (See eq. 5)

The equilibrator force is made equal to $\frac{M_w}{r}$ at $\theta = 0°$, and at some other angle near maximum elevation.

47. During manual elevation, the gas still behaves isothermally. Friction must be considered and now includes packing friction as well as bearing friction.

$P = \frac{(P_0 + 15)\ V_0}{V_\theta} - 15$ = gas pressure, psig (37)

$F_g = AP_\theta$ = gas force, lb (See eq. 36b)

f_p = friction of packing, lb (par. 22)

$F_R - F_g - f_p$ = net rod force used for computing friction in bearing, lb (37a)

$T_A = \mu F_R r_A$ = frictional torque, lb-in, at bearing A, on carriage (fig. 27) (See eq. 34a)

$F_A = f\ (T_A)$ = load in rod, lb, due to friction in bearing A (See eq. 34b)

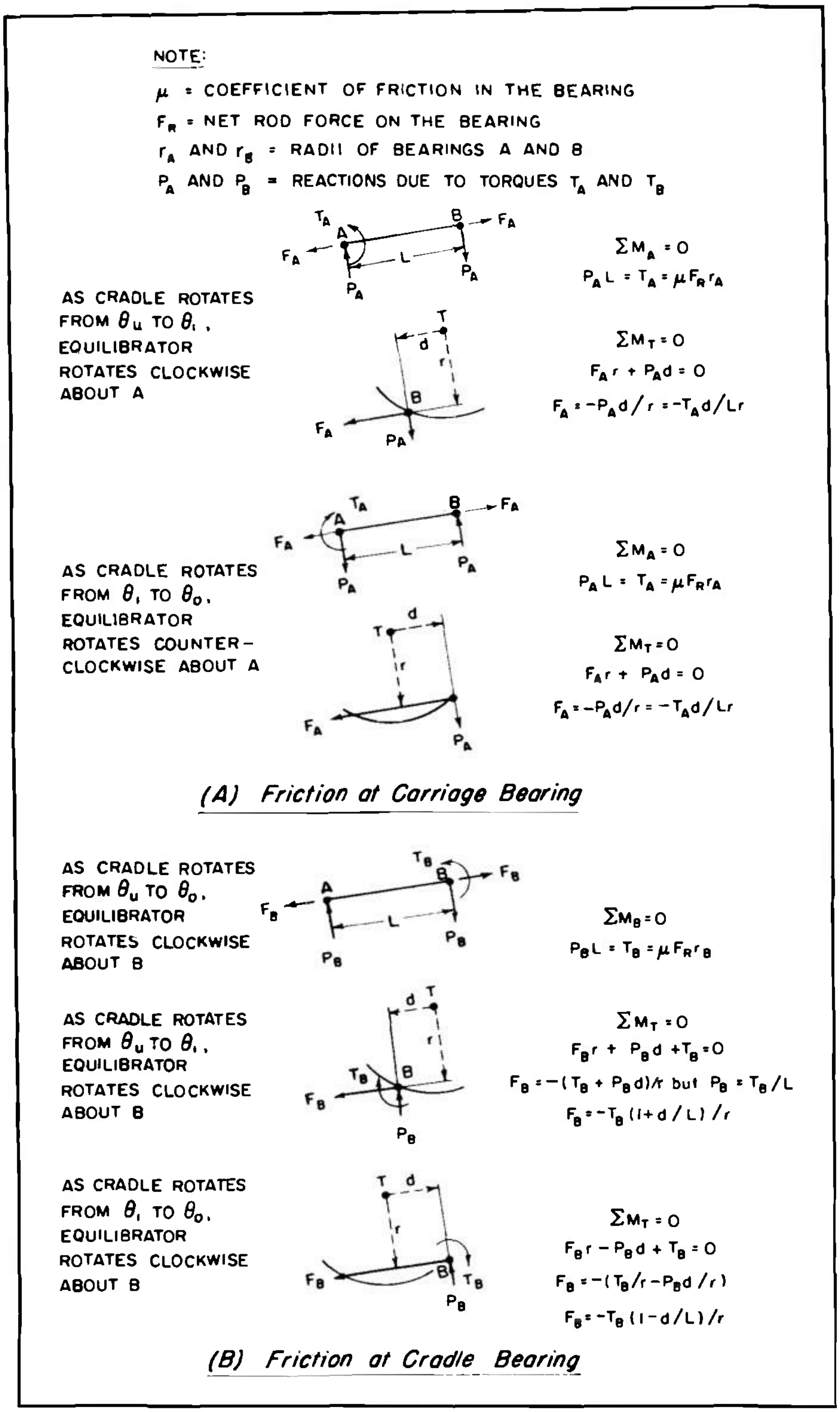


Figure 27. Friction Moments During Depression

$T_B = \mu F_R r_B =$ frictional torque, lb-in, at bearing B, on cradle (fig. 27) (See eq. 34c)

$F_B = f(T_B) =$ load in rod, lb, due to friction in bearing B (See eq. 34d)

$F_f = F_A + F_B =$ total load due to friction of bearings (See eq. 34e)

$F = F_R - F_f =$ net force, lb, in equilibrator rod (See eq. 34f)

$M_e = Fr =$ equilibrator moment, lb-in (See eq. 34g)

$T_e = M_w - M_e =$ torque, lb-in, required to elevate. (See eq. 34h)

48. In manual depression, the calculations are similar to manual elevation, except that the frictional forces are reversed. F_f and f_p will have negative values:

$F = F_R - F_f =$ net rod force, lb (See eq. 34e)

$T_e = M_e - M_w =$ torque, lb-in, required to depress. (See eq. 35)

49. During power elevation, the motion is rapid and the gas expands polytropically. Elevation begins at the loading angle, θ_L. Frictional forces are considered, and, aside from the method of obtaining pressures, the calculations are identical with those for manual elevation.

From the equation, $P_1 V_1^n = P_2 V_2^n$, for polytropic expansion,

$$P_\theta = \frac{(P_0 + 15)\ V_0^n}{V_\theta^n} - 15 \text{ psig,}$$

(See eq. 37)

The value of n varies, depending on the speed of elevation. The constant, $n = 1.29$, was determined from tests on the 280mm Carriage, T72E1, when the time to elevate or depress took 35 to 45 seconds. (See bibliography, reference 4.) In power elevation, a heat sink* (fig. 28) consisting of a mass of thin copper tubes in the gas system has been used to keep the gas cycle nearer to an isothermal than to an adiabatic function. Corrections from $PV^{1.35} = C$ to $PV^{1.05} = C$ have been accomplished by this method. However, when determining the power required for elevating, the designer may be justified to assume the limit $n = 1.4$ in order to provide some margin for error.

50. In power depression, the gas compresses polytropically. The initial conditions are those for the angle to which the gun has just been elevated. The remaining calculations are the same as for manual depression.

51. In paragraph 50, it is assumed that the gun is power-elevated to some angle, θ, and immediately depressed to the loading angle. The gas pressure then rises along the same

*The heat sink concept was introduced by Watertown Arsenal.

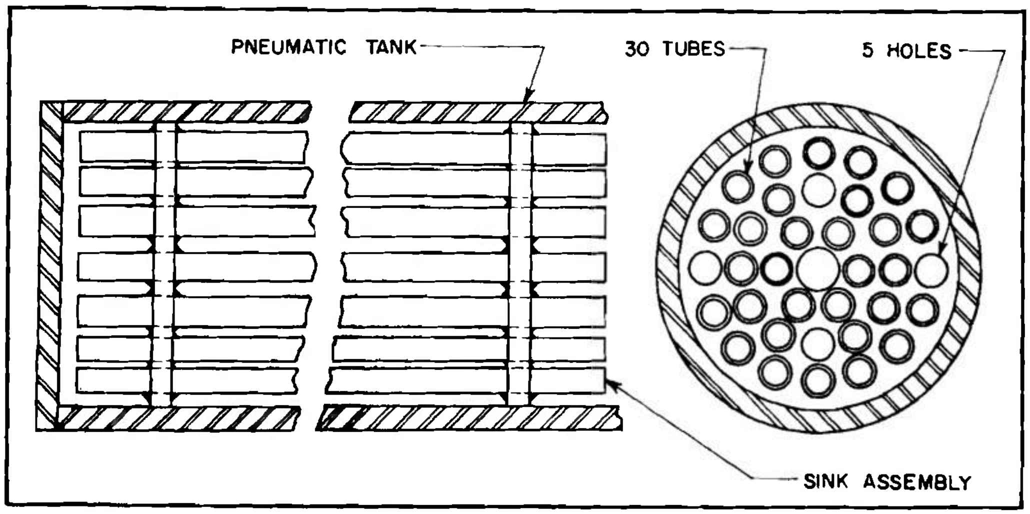


Figure 28. Equilibrator Heat Sink

curve it came down. It is possible, however, for it to be held in the elevated position until the gas recovers ambient temperature. Then, upon power depression, it will follow a different curve. This is true if the gun is manually elevated and then power depressed. This gives rise to another set of calculations which may sometimes be required.

52. After all of the above calculations have been made, the moment *vs.* angle of elevation should be plotted. This will show graphically the differences between the equilibrator moments and the weight moment.

53. The adjacent table lists the order in which performance calculations are made for pneumatic and hydropneumatic equilibrators. The order reads from top to bottom.

Item	Isothermal	Elevation	Depression
θ	*	*	*
r	*	*	*
r_1	*	*	*
ΔV	$A \bullet \Delta L$	$A \bullet \Delta L$	$A \bullet \Delta L$
V_θ	$V_0 + \Delta V$	$V_0 + \Delta V$	$V_0 + \Delta V$
P_a**	$P_{a_0}(V_0/V_\theta)$	$P_{a_0}(V_0/V_\theta)^n$	$P_{a_0}(V_0/V_\theta)^n$
P	$P_a - 15$	$P_a - 15$	$P_a - 15$
F_g	AP_θ	AP_θ	AP_θ
f_p	0	μF_p	$-\mu F_p$
F_R	0	$F_g - f_p$	$F_g - f_p$
T_A	0	$\mu F_R r_A$	$\mu F_R r_A$
F_A	0	$f(T_A)$	$f(T_A)$
T_B	0	$\mu F_R r_B$	$\mu F_R r_B$
F_B	0	$f(T_B)$	$f(T_B)$
F_f	0	$F_A + F_B$	$F_A + F_B$
F	F_g	$F_R - F_f$	$F_R - F_f$
M_e	Fr	Fr	Fr
M_w	$W_t R_t \cos\phi$	$W_t R_t \cos\phi$	$W_t R_t \cos\phi$
T_e	$M_w - M_e$	$M_w - M_e$	$M_e - M_w$

* Determined from geometry.

** $P_{a_0} = P_0 + 15$, absolute pressure at initial conditions.

SPRING EQUILIBRATOR DESIGN

54. Coil springs provide an equilibrator with a simple, efficient, force-producing device. Although bulky, they are applicable to any mount that has sufficient space. Sometimes, to conserve space, two concentric springs are used. Figure 5 illustrates such an installation. The sum of the individual spring rates equals the total required. The springs are wound counter to each other to preclude binding. The inner spring generally provides about 40 percent of the outer spring force.

55. Compression springs are used for several reasons: They cannot be stressed higher than their solid heights permit. They need no additional strength for end loops, as do extension springs. In case of breakage, a compression spring will still work, although at reduced load.

56. The design of a spring to provide the required characteristics is well outlined in textbooks, pamphlets, and handbooks of spring manufacturers. (See bibliography, references 5, 6, and 7.) The equations are not complicated, but because of the interdependence of the variables, several trials may be necessary before a suitable spring is evolved.

57. 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 those shown on the curve. (See bibliography, reference 5.) If a spring is so long that it is unstable, several shorter lengths may be used with spacers in between. However, the total number of active coils of the several shorter springs must be equal to that of the full-length spring, and the spacers must be guided. In some equilibrator designs, the spring housing offers sufficient restraint to prevent buckling.

PNEUMATIC EQUILIBRATOR DESIGN

Gas Volume

58. In a pneumatic unit, the equilibrator force is provided by gas pressure. The design should be based on minimum handwheel loads, and since manual operation is a slow process, the assumption that the gas behaves isothermally is warranted.

59. The gas volume is determined by equating the equilibrator moments with the weight moments at $\theta = 0°$ and at $\theta = u°$ (near maximum elevation), by solving for the corresponding forces and pressures, and subsequently by solving for the initial gas volume.

$M_e = M_w$ at 0° and $u°$ elevation (See eq. 6)

$$F_0 = \frac{M_{w0}}{r_0} = \text{equilibrator force at zero elevation, lb} \quad (38)$$

$$P_0 = \frac{F_0}{A} = \text{gas pressure at zero elevation, psig} \quad (38a)$$

$$F_u = \frac{M_{w_u}}{r_u} = \text{equilibrator force at } u° \text{ elevation, lb} \quad (38b)$$

$$P_u = \frac{F_u}{A} = \text{gas pressure at } u° \text{ elevation, psig} \quad (38c)$$

$$\Delta L_u = \text{equilibrator stroke at } u° \text{ elevation, in.} \quad (38d)$$

$$\Delta V_u = A \Delta L_u = \text{displacement at } u° \text{ elevation, in}^3 \quad (38e)$$

$$V_u = V_0 + \Delta V_u = \text{gas volume at } u° \text{ elevation, in}^3 \quad (38f)$$

$V_0 =$ gas volume at 0° elevation, in^3

$P_a = P_\theta + P_A =$ psia

where $P_A = 15$ psi, atmospheric pressure.

According to Boyle's Law of isothermal expansion:

$$P_{a_0} V_0 = P_{a_u} V_u \quad (39)$$

$$V_0 = \left(\frac{P_{a_u}}{P_{a_0}}\right) V_u = \left(\frac{P_{a_u}}{P_{a_0}}\right)(V_0 + \Delta V_u) \quad (39a)$$

P_{a_u}, P_{a_0} and ΔV_u are known; therefore, V_0 can be solved

$\Delta V = A \Delta L_\theta =$ displacement at any angle of elevation, in^3

$$V_\theta = V_0 + \Delta V = \text{gas volume, in}^3\text{, at any angle of elevation} \quad (39b)$$

$$P_\theta = P_{a_0}\frac{V_0}{V_\theta} - P_A = \text{gas pressure, psig, at any angle} \quad (39c)$$

$$F_\theta = P_\theta A = \text{equilibrator force, lb, at any angle} \quad (39d)$$

$$M_e = r_\theta F_\theta = \text{equilibrator moment, lb-in} \quad (39e)$$

$T_e = M_w - M_e =$ moment required to elevate, lb-in. (See eq. 34h)

Maximum Pressure

60. The maximum gas pressure, P_{max}, for pneumatic equilibrators is limited to the capacity of the packings in the cylinders. The initial pressure at zero elevation, P_0, must be selected such that P_{max} will not exceed this limit. This requires a trial-and-error solution because the displacements, volumes, and piston are interdependent. The greatest increase in pressure under normal operation would take place during polytropic compression from maximum to minimum volume as the gun depresses from maximum to minimum elevation:

Assume adiabatic compression from θ_{max} to θ_{min}

$$P_{max} = P_{a(min)}\left(\frac{V_{max}}{V_{min}}\right)^{1.4} - 15 = P_d \text{ psig,} \quad (40)$$

where P_d is the limit design pressure of the packing.

Find value for $P_{a(min)}$, minimum absolute pressure, and solve for P_{0_m},

$$P_{0_m} = P_{a(min)}\left(\frac{V_{max}}{V_0}\right) - 15 \text{ psig} \quad (40a)$$

equilibrator pressure at 0° elevation or the maximum charging pressure.

$$A = \frac{F_0}{P_0} = \text{effective area of piston, in}^2. \quad (41)$$

A theoretical piston area, A, must first be found, then appropriate sizes selected for piston and piston rod to approximate it closely. The actual area must not be less than the theoretical area, so that the pressures never exceed their design values.

HYDROPNEUMATIC EQUILIBRATOR DESIGN

61. The gas volume of a hydropneumatic system is determined by the same method as for the pneumatic. The principles are the same, except that the hydropneumatic transmits the gas pressure to the operating piston by means of hydraulic fluid. One fluid available for this purpose is MIL-H-6083B, Hydraulic fluid, petroleum base, preservative. Figure 29 shows a general arrangement of this type of equilibrator. The gas in the accumulator tends to force the liquid from the reservoir through a flexible tube and hollow piston rod into the equilibrator cylinder. (See figure 9 and paragraph 11.) Sufficient liquid must be available to complete the stroke and still have some in reserve. This small additional volume is necessary to permit adjustments and to prevent gas from entering the line. This establishes the minimum liquid level. Usually, the gas has direct contact with the fluid, the reservoir being mounted upright with the liquid port located in its bottom. If a separate tank, or accumulator, is used to store the gas, a maximum liquid level must be established so that the liquid cannot spill into the tank, even when the weapon is tilted.

DESIGN OF EQUILIBRATOR COMPONENTS

62. The forces of an equilibrator are usually treated as being confined to one unit. When only one equilibrator is necessary, it is located in a vertical plane passing through the centerline of the gun tube to preclude eccentric loads. However, a weapon may have two identical equilibrators if space in the center of the mount is not available, or if one unit would be too bulky. When this condition occurs, the forces are divided into two equal parts, each half representing the design loads for the equilibrators which are arranged symmetrically about the center of the mount.

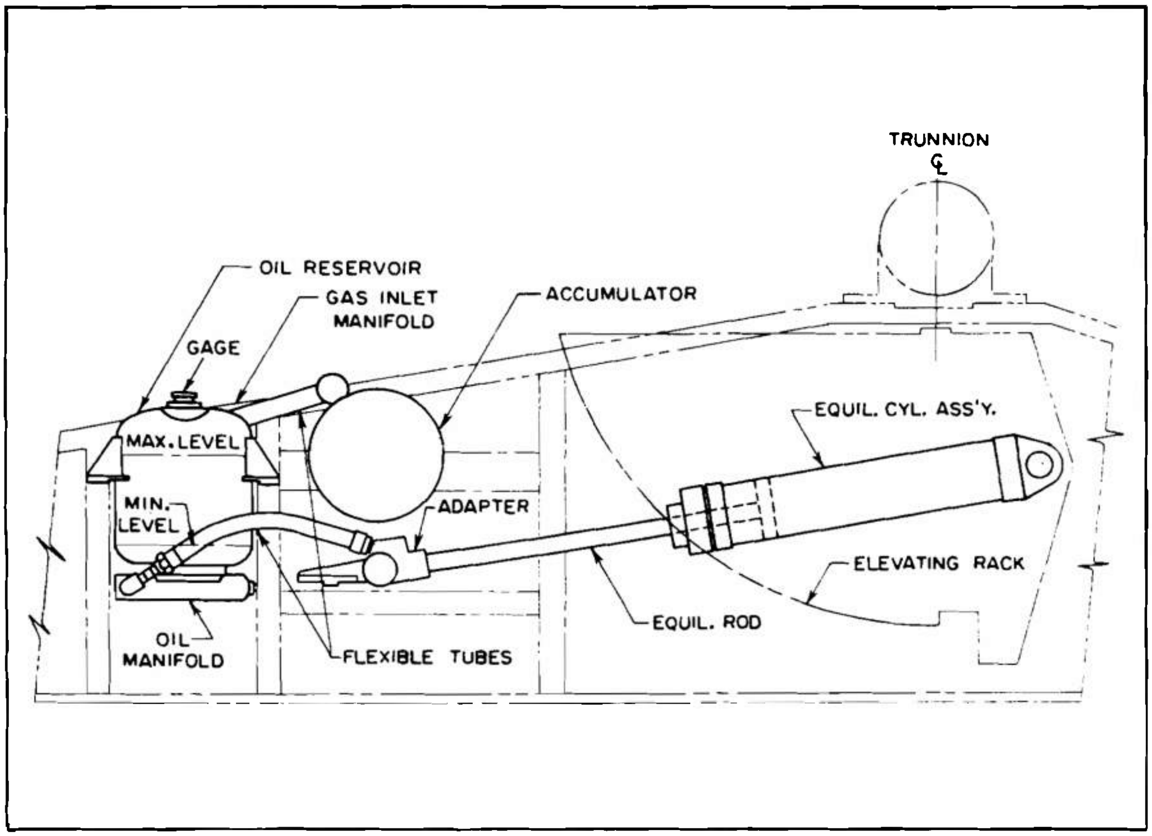


Figure 29. Equilibrator System Arrangement

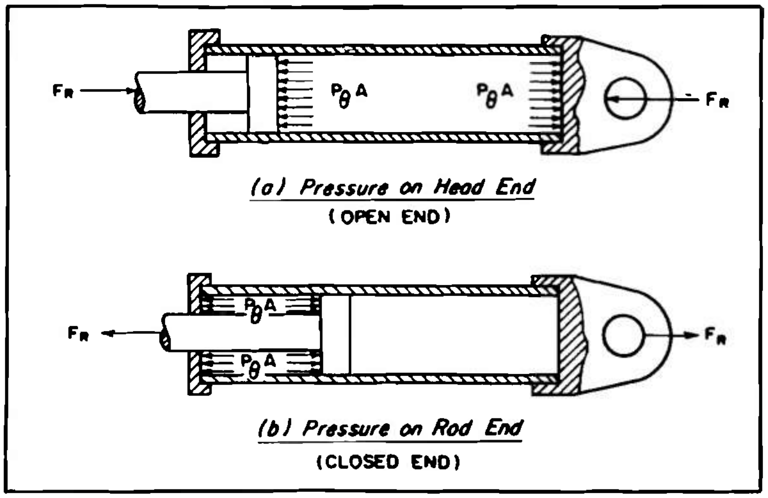


Figure 30. Two Methods of Applying Pressure in Cylinder

Cylinder Design

63. The inside dimensions of the cylinders are determined by the piston size and the length of stroke. The wall and head thicknesses are set up by the pressures to be withstood. The cylinder may be either closed or open ended, depending upon which side of the piston is pressurized. (See figure 30.)

64. The circumferential stress, σ_t, of the cylinder wall (due to fluid or packing) may be found by the Lamé formula for thick-walled cylinders:

$$\sigma_t = P_{max}\left(\frac{d_2^2 + d_1^2}{d_2^2 - d_1^2}\right). \qquad (42)$$

When the pressure is applied to the rod end of the piston, there is also a longitudinal stress. Present also are bending stresses from the frictional torque, T_B, but these are so small as to be negligible. The cylinder is an application where medium strength steel is suggested. Rigidity is very desirable to minimize the possibility of local damage and to prevent excessive dilation, which makes the seals less effective.

65. The cylinder head (rod end) is treated as a flat plate with a concentric hole, its outer edge fixed and supported, and a uniform load over the entire actual surface. (See figure 31.) The maximum stress occurs at the inner edge. (See bibliography, reference 8.)

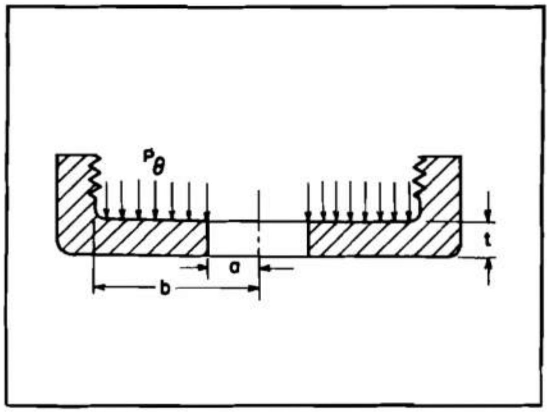


Figure 31. Cylinder Head, Rod End

$$\sigma_t = \frac{3P_\theta\,(m^2 - 1)}{4mt^2}\left[\frac{a^4 - b^4 - 4a^2b^2 \log a/b}{a^2(m-1) + b^2(m+1)}\right] \qquad (43)$$

where: a = outer radius

b = inner radius

m = reciprocal of Poisson's ratio

P_θ = pressure

t = plate thickness.

The cylinder head at the terminal end is more than adequately reinforced by the lug and, therefore, requires no analysis.

Piston Rod

66. If the equilibrator is a pull type, the piston rod will be subjected to a tension load, and the stresses can be calculated accordingly. Note that if threaded, the critical section is at the root of the thread. If the equilibrator is a push type, the rod is subjected to compression, and column action must be considered.

67. In addition to axial loads, the rod is subjected to a bending moment because of the frictional torque, T_A (fig. 32). Usually, this is small enough to be ignored, but it must not be forgotten. With a long slender rod, there is a possibility that this bending might become appreciable. For this condition, the bending moment (bibliography, reference 9) is

$$M = T_A \sec \tfrac{1}{2} U, \qquad (44)$$

and the maximum compressive stress is

$$\sigma_c = \frac{M}{Z} + \frac{F_R}{A_r} \qquad (45)$$

where

$T_A = \mu F_R r_A$ (See eq. 34a)

$U = \frac{L_r}{j}$

$j = \sqrt{\frac{EI}{F_R}}$

E = modulus of elasticity, psi

L_r = length of rod, in.

A_r = cross section area at mid-span, in^2

Z = section modulus at mid-span, in^3.

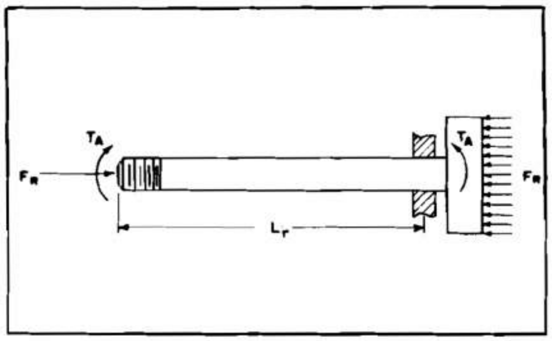


Figure 32. Rod Showing Applied Loads and Moments

Terminals

68. The equilibrator is pin-connected to the tipping parts at one end and to the top carriage at the other. The terminals should be bushed and lubricated to reduce friction. Intermittent lubrication is considered satisfactory. If the bearing is a free-fit with frequent relative motion but unprotected from dirt, the allowable bearing stress is 8000 psi (bibliography, reference 10). If protected, the allowable bearing stress becomes 12,000 psi (bibliography, reference 10).

69. The maximum stress in the lug (fig. 33) occurs at m and is (bibliography reference 11):

$$\sigma = a \frac{8 F_R}{\pi r_0^2 t}, \qquad (46)$$

where $a = 4.35$ when $2 \leqslant \frac{r_0}{r_i} \leqslant 4$

F_R = applied equilibrator load

r_i = radius of hole

r_0 = radius of lug

t = thickness of lug.

The lug must satisfy both bearing and tensile strength requirements. The hole size used above is that of the lug, and not of the bushing.

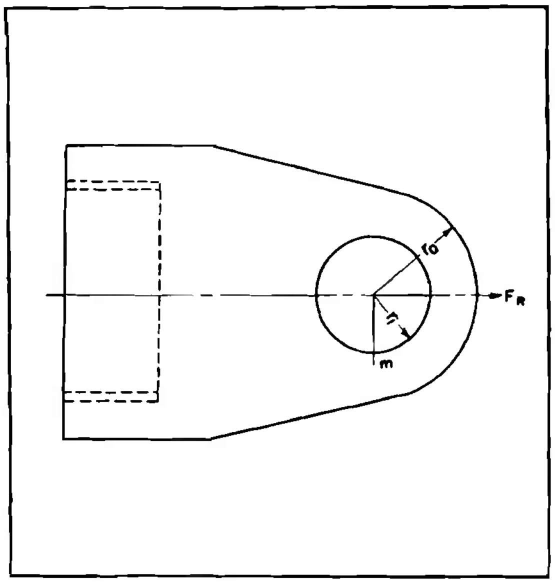


Figure 33. Cylinder Terminal

70. Pins are made of high tensile steel and are subjected to shear and bending stresses. The load distribution is shown in figure 34. Both shear and bending should be checked. Bearing pressure also may be critical because of its low permissible value (8000 psi). Whichever of these three conditions is most critical will determine the size of the pin.

Packings

71. Figure 35 shows a typical packing assembly. The packing illustrated is proportioned after those already in use, so that previous experience is an important factor in its design. The Belleville spring and fluid pressure force the piston ring against the rubber filler. The pressure developed in the rubber then presses the packing ring against the cylinder wall to provide a seal. The method used to determine the amount of sealing pressure to be produced by the spring force follows:

*The pressure factor is the ratio of the radial pressure to the applied axial pressure. It is somewhat analogous to Poisson's ratio. The pressure factor would be 1.0 if the packing behaved hydrostatically. For rubber filler, $K_p = 0.73$. (See bibliography, reference 1, page 12.)

**The value of ν is taken from the design specifications. For example, specifications for the recoil mechanism of the 175mm Gun Carriage, T76, call for $\nu=1.3$ for the floating piston and 0.88 for the stuffing box. Lubrication was desired for the latter.

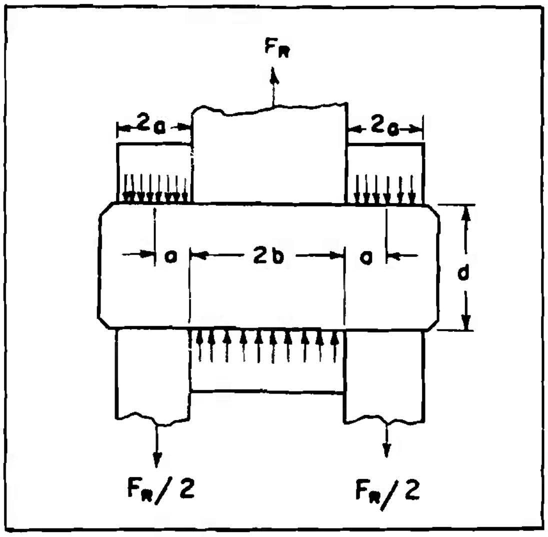


Figure 34. Loading Diagram of Pin and Bearing

The radial force:

$$F_p = \pi D w K_p (P_\theta + P_s), \text{ lb.} \qquad (47)$$

Expressing radial pressure in terms of fluid pressure,

$$K_p (P_s + P_m) = \nu P_m, \qquad (47a)$$

from which the required spring pressure is

$$P_s = \frac{\nu - K_p}{K_p} P_m \text{ psi}, \qquad (47b)$$

where D = outside diameter of packing, in.

K_p = pressure factor*

ν = leakage factor**

P_m = maximum fluid pressure, psi

P_s = axial pressure in packing produced by spring, psi

P_θ = fluid pressure at elevation angle θ, psig

w = width of packing, in.

72. Silver rings, whose cross sections are right angles, may be used to confine the corner of the packing to prevent it from extruding between the piston ring and the cylinder. Lately, considerable success has been attained in substituting polytetrafluoroethylene (Teflon) for leather and an aluminum bearing alloy for silver.

73. Another type of packing, usually limited to stuffing boxes, is the chevron packing. It is commercially available and can be selected, according to requirements, from manufacturers' catalogues.

Belleville Springs

74. Springs are used to augment the packing pressure. Belleville springs are selected because they require little space and provide large loads at small deflections. The design of Belleville springs is outlined in most spring manufacturers' handbooks (bibliography, reference 7). 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

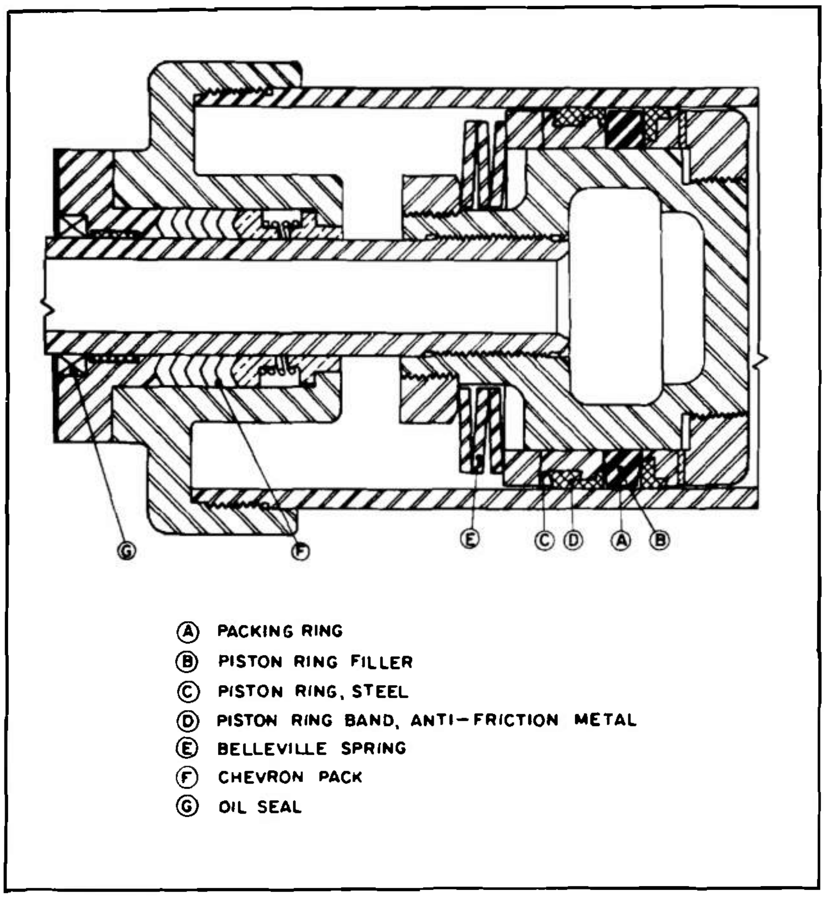


Figure 35. Typical Packing Assembly

$$F_s = \pi\,(r_2^2 - r_1^2)\,P_s, \qquad (48)$$

where P_s = packing pressure due to spring loads (par. 71)

r_1 = inside radius of piston ring assembly

r_2 = outside radius of piston ring assembly.

Tanks, Liquid and Gas

75. Capacities of the tanks, reservoirs, or accumulators are determined by the amounts of hydraulic fluid and gas required for proper functioning of the equilibrator. Figure 36 is a sectional view of a fluid reservoir with liquid level gage, gas inlet manifold pipe and liquid outlet port. The manifold extends into the tank near the top, and, except for several small gas ports, is completely sealed

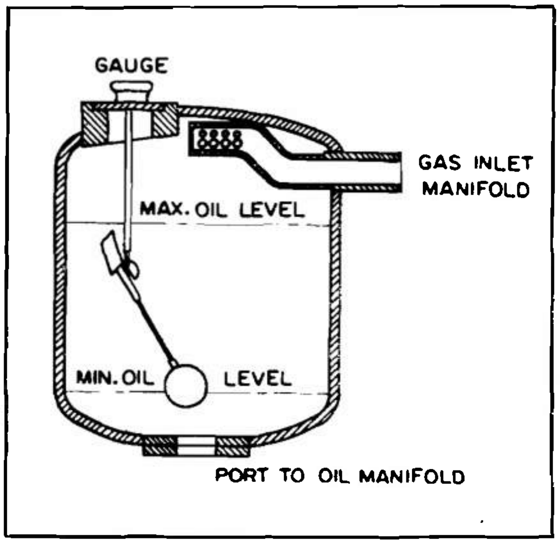


Figure 36. Section of Oil Reservoir

to prevent oil from splashing into the gas line. An internal float-type gage indicates the liquid level. It is preferred to an external gage because it is protected by the tank walls.

76. Dished heads are used to save space, and the structure should conform to the ASME Unfired Pressure Vessel Code. The gas tank, or accumulator, may be similarly constructed but without the gage and manifold.

Equilibrator Adjustment

77. Adjustment devices provide a convenient means of compensating for changes in gas pressure brought on by temperature changes. They eliminate frequent adding and releasing gas to adjust the pressure. With the volume of the gas being held constant, the pressure varies directly as the absolute temperature:

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}.$$

At 70°F., $T_1 = 530$ degrees absolute. Thus, a variation of plus or minus 25 degrees represents a deviation in pressure of about 5 percent each side of the mean. This deviation is reflected in the equilibrator force and equilibrator moment about the trunnion. The adjustment is designed to compensate for this amount of deviation. If further adjustment is needed, gas is either added to or released from the tank. The adjustment may be achieved in either of two ways: adjust the moment arm, or adjust the gas pressure.

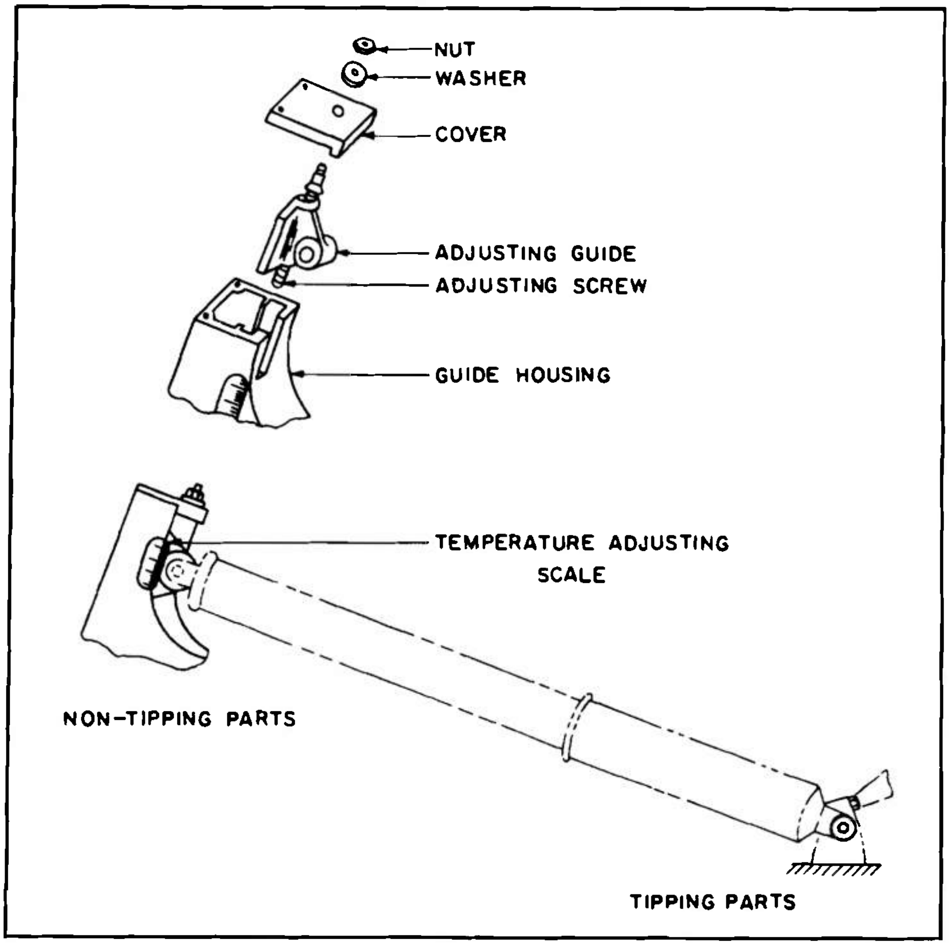


Figure 37. Variable-Moment-Arm Adjustment

78. The variable-moment-arm-method of adjusting for temperature variations is illustrated in figure 37. The top carriage end of the equilibrator is pin-connected to a slider, or adjusting guide. This sliding guide is housed in a machined guide housing attached to the top carriage. The guide includes an adjusting screw, which is used to raise or lower the guide along the housing. This shifts the line of action of the equilibrator and increases or decreases the length of the moment arm, as may be required. An index and a scale, calibrated in degrees, indicate what the temperature setting should be.

79. The procedure for determining the direction and length of travel of the adjusting guide will now be described. Figure 38 is a schematic sketch of the system. The adjusting movement of the guide is perpendicular to the normal position of the equilibrator. Since only 5 percent deviation in movement is required, the limits of the adjusted moment arm, r' are $0.95r$ and $1.05r$. At the limits of r',

$$\Psi = \sin^{-1}\frac{r}{R}$$

$$r' = 0.95r \text{ and } 1.05r$$

$$\Psi' = \sin^{-1}\frac{r'}{R}$$

$$e = L\tan(\Psi - \Psi')$$

$$L' = \frac{L}{\cos(\Psi - \Psi')}.$$

A change in length, L, means a change in gas volume and pressure. Thus, the equilibrator moment changes in two respects: a change in moment arm, and a change in force; although the latter is negligible and its effect may be ignored.

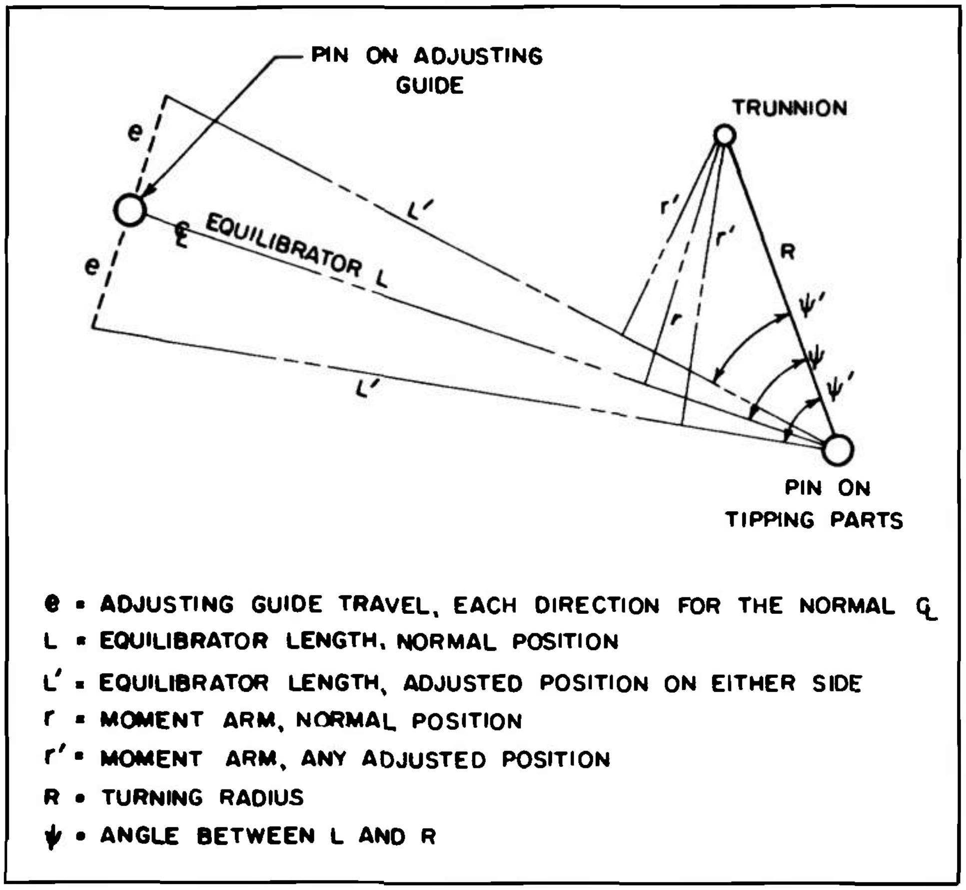


Figure 38. Equilibrator Geometry for Adjustment

80. The adjusting guide travel having been determined, the adjusting scale can now be calibrated in terms of temperature change. At points along the travel line on both sides of the normal position, small increments, e, are selected for which L' and r' are calculated. From the change in equilibrator length, $\Delta L = L - L'$, the new gas volumes are determined (par. 59). The equilibrator moment is found for normal temperature and position. Next, the moment is calculated for the changes in pressure and moment arm caused by the change in position. These results determine how much the pressure must be increased or decreased to provide a moment equal to the normal one. The change in pressure is indicative of the change in temperature necessary to produce it. This change in temperature is marked on the scale at the appropriate point. This process is repeated until the required temperature range is spanned.

81. The pressure-control method of adjustment entails adding fluid to the reservoir to raise the gas pressure or removing fluid to lower the pressure. The correct pressure is usually indicated by a specified handwheel torque at a given elevation. A hand pump may be used to add fluid, if this is required. The adjustment range is limited by the amount of reserve fluid in the accumulator.

Semiautomatic Adjustment

82. Figure 39 shows a semiautomatic device for maintaining proper equilibrator gas pressure. This unit controls the pressure by adding or draining hydraulic fluid from the accumulator by means of a pressure relief valve and a hydraulic pump. The fluid reserve is contained in the pump tank, or reservoir (as distinguished from the equilibrator reservoir, or accumulator). Power should be available to operate a hydraulic pump capable of delivering fluid at a rate that will compensate for a 25-degree temperature change in about 10 minutes. The motor is started and stopped by a manually operated switch, and adjustment takes place only when the motor is running. At all other times, the equilibrator is isolated by the equilibrator check valve (A, fig. 39).

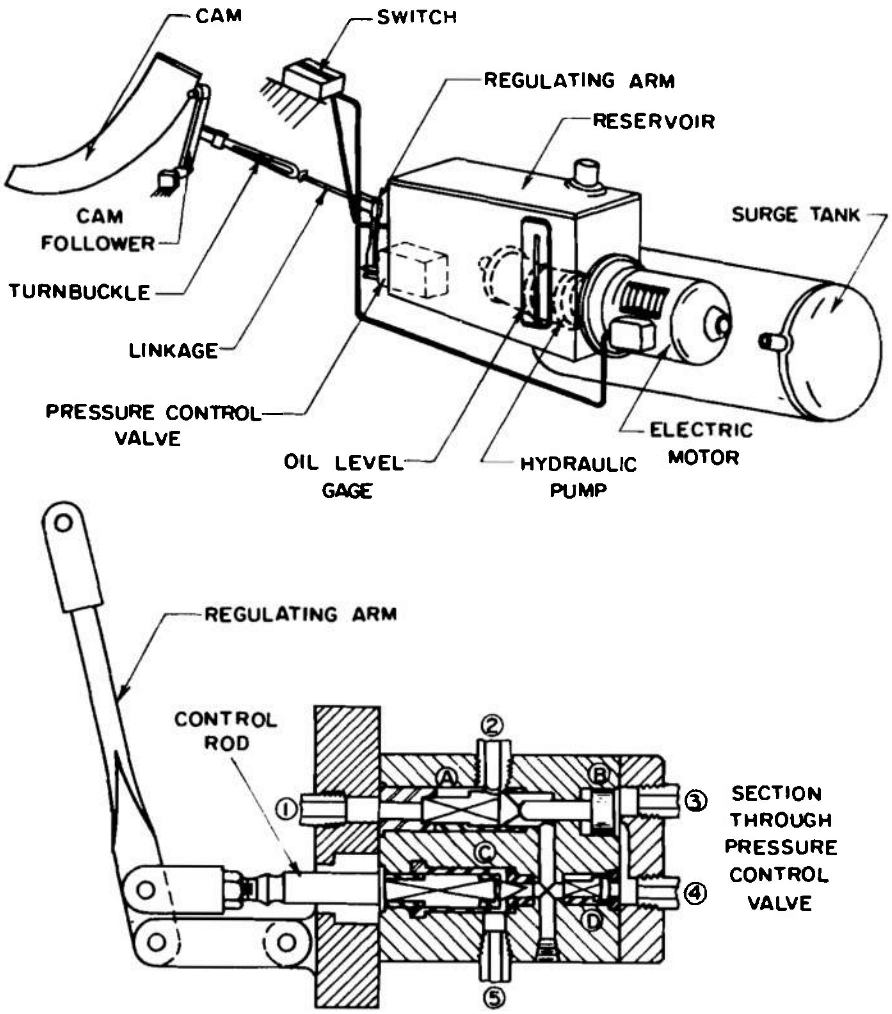
83. The correct pressure for any angle of elevation is known, and this knowledge is built into a cam which moves with the tipping parts. (See figure 39 for this and all other references in this description.) The cam follower is connected by a linkage to the control-valve spring (pressure relief valve, C). Thus, the spring always exerts the proper force corresponding to the correct pressure for any angle of elevation.

84. With the pump running and the equilibrator pressure low, equilibrator check valve (A) is opened by the operating piston (B), whose area is larger than that of the valve. Pump check valve (D) opens by hydraulic pressure from the pump. Fluid flows into the accumulator through valves (D) and (A) and outlet (1) until the proper pressure is reached. At that time, the pressure relief valve (C) opens and any additional fluid from the pump flows through inlet (5) to the pump reservoir.

85. With the pump operating and the equilibrator pressure high, valves (A) and (D) open, as before. Pressure relief valve (C) opens because of excessive pressure in the system. Fluid from both the surge tank and the pump flows into the pump reservoir. When the correct pressure is reached, valve (C) closes and only pump fluid goes to the reservoir. When the pump motor is shut off, after either high or low pressure compensation, valve (A) closes and the equilibrator is again isolated.

86. An oil level gage is provided on the pump reservoir. When the fluid level is at either extreme and an unbalanced condition persists, gas should be added or released so as to bring the level to its proper position.

87. Check valves (A) and (D) are spring seated, but, since they do not seat against any pressure, the springs need only enough stiffness to insure prompt seating when the pump stops. Check valve (D) prevents a reverse flow back to the pump. The pressure relief, or control valve (C) is a different matter. Its spring must be capable of adjusting its seating load to correspond to the equilibrator pressure at any elevation. The spring rate is based on maximum and minimum equilibrator pressures and on a convenient small displacement. The initial spring load is adjusted to suit different pressures exerted by the control rod, which compresses the spring to a greater or lesser degree in response to the cam follower. The linkage



① OUTLET TO SURGE TANK
② OUTLET (PLUGGED) FOR GAGE, GAS PRESSURE
③ OUTLET (PLUGGED) FOR GAGE, PUMP
④ INLET FROM HYDRAULIC PUMP
⑤ INLET TO RESERVOIR
Ⓐ EQUILIBRATOR CHECK VALVE
Ⓑ OPERATING PISTON
Ⓒ PRESSURE RELIEF VALVE
Ⓓ PUMP CHECK VALVE

Figure 39. Semiautomatic Adjustment

between the cam follower and the control rod has a mechanical advantage of approximately 4 to 1, which provides desired sensitivity of the pressure relief valve (C), and at the same time, results in a cam of reasonable proportions. The cam radii are based on the control rod displacements: (See figure 40.)

A_v = valve pressure area

P_θ = equilibrator pressure, any position

$F_v = A_v P_\theta$ = valve load, any position

F_m = maximum valve spring load

F_0 = minimum valve spring load

Δ_s = spring deflection between F_0 and F_m

Δ_0 = spring deflection at minimum pressure

R_0 = minimum cam radius

λ = mechanical advantage of linkage

θ = angle of elevation

$$K_s = \frac{F_m - F_0}{\Delta_s} = \text{spring rate}$$

$$\Delta = \frac{F_v}{R_s} = \text{spring deflection at } \theta$$

$$d_s = \Delta - \Delta_0 = \text{control rod travel}$$

$$R_c = R_0 + \Delta R_c = \text{cam radius at } \theta,$$

where $\Delta R_c \approx \lambda d_s$. The value of ΔR_c can be obtained from the geometry of the system shown in figure 40.

Two fine adjustments are provided for the control-valve operating mechanism: A slotted hole in one end of the cam permits it to be slightly rotated about the other end for exact positioning. A turnbuckle in the link between the cam follower and the regulating arm provides adjustment of initial spring loads of the pressure relief valve. The cam follower is spring-loaded to insure proper contact at all times.

Suggested Materials for Equilibrator Components

88. Hard bearing bronze, properly grooved for lubrication, is recommended for sliding or rotating parts not freely lubricated. The choice of packing materials may depend on present and future developments. However,

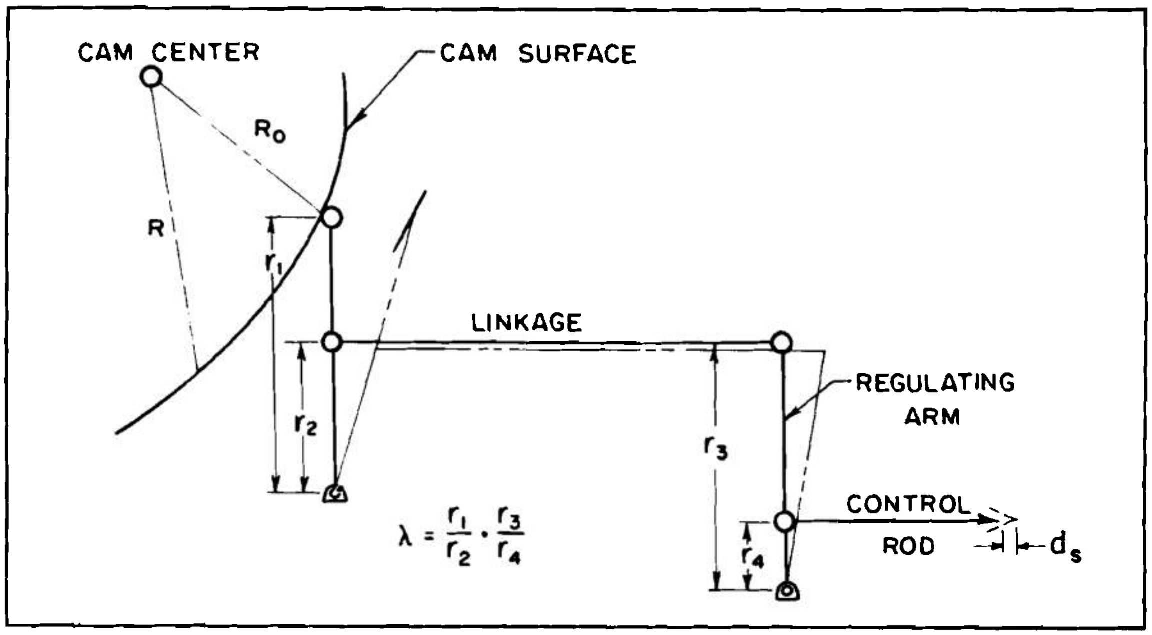


Figure 40. Schematic of Linkage for Adjustment

some units now in use, such as shown in figure 35, have a leather packing ring, WXS-157, with a piston ring filler made of molded synthetic rubber compound SC 610 ABFF, MIL-R-3065. Oil seals may be of leather or neoprene, CLHX2. As stated in paragraph 73, chevron packings are commercially available. Where tubes are used to transmit fluid (fig. 29), seamless, flexible metal tubing is recommended similar to American Brass Co., Type S-2-C.

89. High strength steels should be avoided unless there is a positive advantage in using them. Moderate tensile strengths of about 70,000 psi are recommended. The factor of safety, based on yield strength, should be 1.5. The choice of material will be influenced by the purpose. Where high strength-to-weight ratios are needed, as in mobile weapons, a high strength steel is indicated. But where rigidity is the prime requirement, a medium strength steel will serve just as well.

Manufacturing Procedure

90. No special techniques are involved in the fabrication of equilibrators since they can be constructed by ordinary machine shop methods. It should be pointed out, however, that cylinders are honed and pistons are fitted on assembly.

MAINTENANCE OF EQUILIBRATORS

91. Adequate care of the equilibrators usually insures that it will be in working order at all times. Periodic inspection, lubrication of sliding surfaces, and adjustments to compensate for temperature changes (par. 77), constitute preventive maintenance. The adjustment of spring tension, gas pressure, or oil volume, the replacement of broken parts, or the repair of slightly damaged parts, constitute corrective maintenance. The gun crew attends to preventive maintenance but only skilled ordnance personnel, unless directives state otherwise, perform corrective maintenance.

92. Inspection comprises several activities including observations of whether a failure or impending failure of structural components is present; whether sliding surfaces are smooth, clean, and lubricated; and whether the tipping parts can be elevated readily. The last is the key to a normally functioning equilibrator, provided that the elevating system is in good working order. The elevating mechanism and equilibrator are treated as one unit for this phase of the inspection. Maximum specified hand wheel loads should not exceed 20 pounds, but 12 pounds is a more attractive limit. An ordinary spring scale attached to the handle may be used to measure these loads. During inspection, when the hand wheel loads exceed the maximum or are uneven, either the equilibrator or the elevating mechanism is faulty. If the elevating system is unbroken and is neither clogged with dirt, in need of lubrication, nor its gears meshed too tightly, then the malfunction stems from the equilibrator.

93. A dented cylinder or spring case, and broken or worn parts cause uneven as well as excessive hand wheel loads. Set springs of spring equilibrators should be adjusted, or replaced, if the set is too pronounced. Broken springs must be replaced. High hand wheel loads are also caused by low gas pressure or insufficient oil supply which result from leakage. On the other hand, excessive gas pressures also contribute to high hand wheel loads. Pressure increases are caused by frequent and rapid operation or by hot weather. If corrections are needed beyond the limited range of the adjustment devices (par. 77), gas or oil must either be added to or removed from the system. It is not advisable to check gas pressure unless it is evident that the equilibrators are not functioning properly. During each check, about 15 psi of pressure are lost. Also, the operation exposes the valve seat to dirt, causing leakage and eventual ineffectiveness. Filter elements should be checked, and when necessary, cleaned or replaced. When dirty, equilibrators are to be disassembled, washed in a nonflammable dry cleaning fluid, dried thoroughly, and lubricated before reassembly.

94. Lubrication provides the most effective means of preventive maintenance. Not only does it reduce friction, and subsequent wear or galling of sliding surfaces, but it also serves as a preservative, guarding against corrosion. Although several oils and greases make good equilibrator lubricants, the one now used generally is MIL-G-10924A grease. This grease is suitable for lubrication where ambient temperatures range from 65°F to 125°F.

APPENDIX I
SAMPLE CALCULATIONS FOR PNEUMATIC EQUILIBRATOR PERFORMANCE

A. EQUILIBRATOR MOMENT ARM

95. As an example two pneumatic equilibrators performing as one unit in a hypothetical weapon have been chosen. The equilibrator geometry is shown in figure 41. For calculating the moment arm, the known terms (in inches and degrees) are:

when $\theta = 0, \Psi = 86° 48'$

$c = 56.989$

$\theta_{max} = 65°$

$R = 19$

$\beta = 123° 12'$

$\theta_{min} = -5°$

From Equation (2)

$$L_\theta^2 = c^2 + R^2 - 2cR \cos\Psi$$
$$= 3608.75 - 2165.58 \cos\Psi, \text{ in}^2.$$

From Equation (3)

$$r = cR \sin\Psi/L_\theta$$
$$= 1082.79 \sin\Psi/L_\theta, \text{ in.}$$

$$d = \sqrt{R^2 - r^2}$$
$$= \sqrt{361 - r^2}, \text{ in.}$$

B. WEIGHT MOMENT

96. The tipping parts of the hypothetical weapon weigh 10,000 lb and conform to figure 42. The weight moment is calculated from the known data:

$W_t = 10{,}000$ lb

$R_t = 29.39$ in.

$\phi' = 3° 28'$

and from Equation (5),

$$M_w = W_t R_t \cos\phi, \text{ lb-in.}$$

$$\Delta L = L_0 - L_\theta.$$

For convenience L_0 is selected to be the equilibrator length when θ equals zero rather than when ϕ equals zero. (Reference paragraph 25.)

C. GAS VOLUME

97. The gas volume is determined by equating the equilibrator and weight moments at 0° and 55° elevation, and then solving for the initial gas volume. Assuming that for

MOMENT ARM CALCULATIONS

θ	Ψ	sin Ψ	cos Ψ	L_θ^2	L_θ	r	d
—5°	91° 48′	0.9954	—0.0314	3676.8	60.64	17.85	6.51
0°	86° 48′	.9984	.0558	3487.9	59.06	18.30	5.11
10°	76° 48′	.9736	.2284	3114.2	55.80	18.89	2.04
20°	66° 48′	.9191	.3939	2755.6	52.49	18.96	1.23
30°	56° 48′	.8368	.5476	2423.0	49.22	18.41	4.70
40°	46° 48′	.7290	.6846	2126.3	46.11	17.12	8.24
45°	41° 48′	.6665	.7455	1994.4	44.66	16.16	9.98
50°	36° 48′	.5990	.8007	1874.7	43.30	14.98	11.69
55°	31° 48′	.5270	.8499	1768.2	42.05	13.57	13.30
60°	26° 48′	.4509	.8926	1675.8	40.94	11.92	14.80
65°	21° 48′	.3714	.9285	1598.0	39.98	10.06	16.12

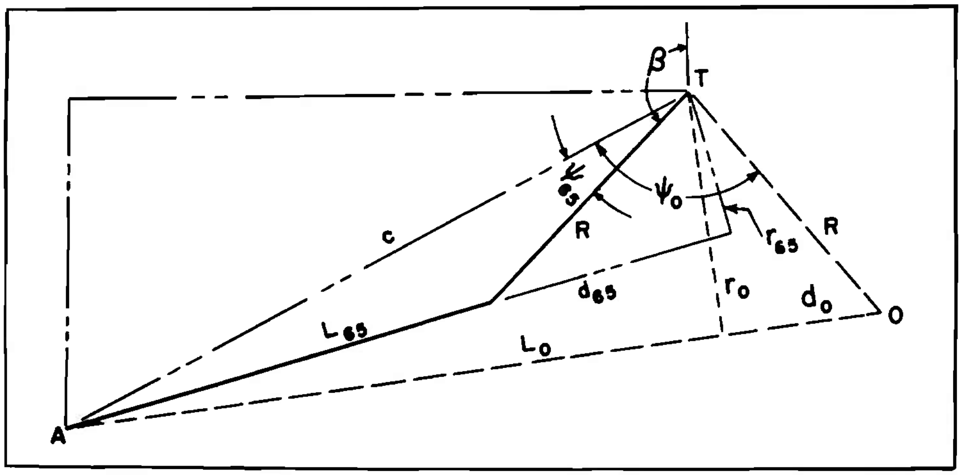


Figure 41. Geometry of a Typical Pneumatic Equilibrator

WEIGHT MOMENT CALCULATIONS

θ°	ΔL	φ	cos φ	R_t cos φ	M_w
—5°	—1.58	—1° 32′	0.9996	29.38	293800
0°	0.00	3° 28′	.9982	29.34	293400
10°	3.26	13° 28′	.9725	28.58	285800
20°	6.57	23° 28′	.9173	26.96	269600
30°	9.84	33° 28′	.8342	24.52	245200
40°	12.95	43° 28′	.7258	21.33	213300
45°	14.40	48° 28′	.6631	19.49	194900
50°	15.76	53° 28′	.5953	17.50	175000
55°	17.01	58° 28′	.5230	15.37	153700
60°	18.12	63° 28′	.4467	13.13	131300
65°	19.08	68° 28′	.3670	10.79	107900

this hypothetical equilibrator $D = 3.75$ in. (piston diam.) and $d = 1.25$ in. (rod diam.), then the effective pressure area of two pistons is:

$$A = 2\frac{\pi}{4}(D^2 - d^2) = 19.635 \text{ in}^2.$$

Equating the weight moment, M_w, to the equilibrator moment, Fr, at two elevations:

Rod force at 0° elevation,

$$F_0 = \frac{M_{w_0}}{r_0} = \frac{293400}{18.30} = 16030 \text{ lb},$$

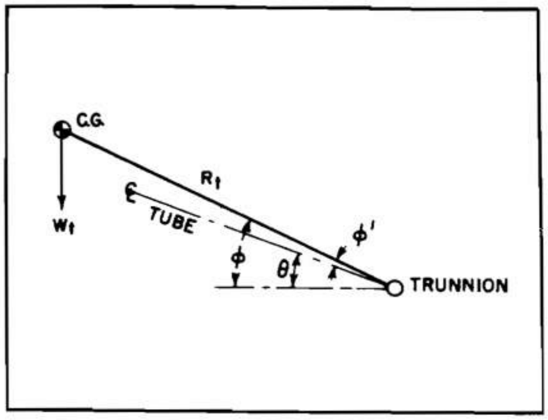


Figure 42. Geometry for the Weight Moment

Rod force at 55° elevation,

$$F_{55} = \frac{M_{w_{55}}}{r_{55}} = \frac{153700}{13.57} = 11330 \text{ lb}.$$

The respective gas pressures must be:

$$P_0 = \frac{F_0}{A} = 816 \text{ psig},$$

$$P_{55} = \frac{F_{55}}{A} = 577 \text{ psig};$$

or, in terms of absolute pressure,

$$P_{a_0} = 816 + 15 = 831 \text{ psia},$$

$$P_{a_{55}} = 577 + 15 = 592 \text{ psia}.$$

98. To determine initial gas volume, first compute gas volume at 55° elevation:

$$V_{55} = V_0 + \Delta V_{55},$$

where $\Delta V_{55} = A\Delta L_{55}$ = displacement at 55° elevation; and

$\Delta L_{55} = 17.10$ in. = equilibrator stroke at 55° elevation.

Thus, from Boyle's Law:

$$P_{a_0} V_0 = P_{a_{55}} V_{55}$$

or: $831 V_0 = 592 (V_0 + 334)$

$$V_0 = 827 \text{ in}^3 = \text{initial gas volume.}$$

99. If friction is neglected, the following equations are sufficient for calculating the torque required to elevate $(T_e = M_w - M_e)$:

Change in gas volume,

$$\Delta V = A\Delta L = 19.635\, \Delta L \text{ in}^3$$

Gas volume, $V_\theta = V_0 + \Delta V = 827 + \Delta V$

Absolute pressure, $P_{a_\theta} = \dfrac{P_{a_0} V_0}{V_\theta} = \dfrac{687237}{V_\theta}$ psia

Gas pressure, $P_\theta = P_{a_\theta} - 15$ psig

Rod force, $F_g = P_\theta A = 19.635\, P_\theta$ lb

Equilibrator moment, $M_e = F_g r$ lb-in.

The calculations are tabulated in the accompanying table, and figure 43 shows the relation of the weight (M_w) and gas force (M_e) moments about the trunnion during frictionless elevation.

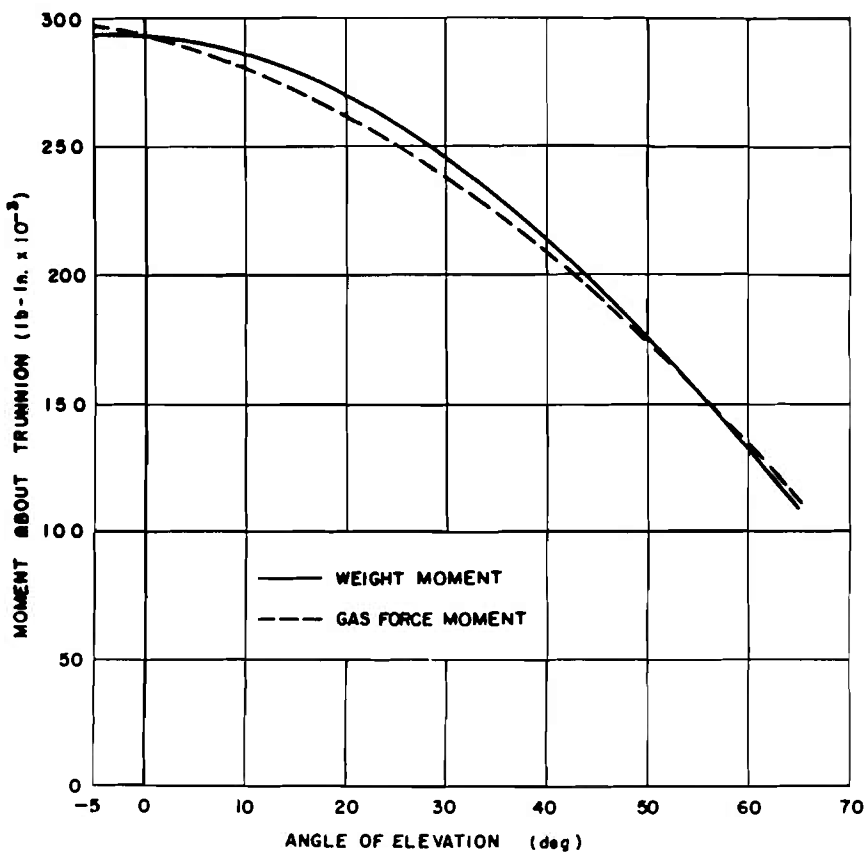


Figure 43. Equilibrator Performance Curves for a Frictionless System

TORQUE REQUIRED TO ELEVATE

(Friction Not Considered)

θ	ΔL	ΔV	V_θ	P_{a_θ}	P_θ	F_g	r	M_e	M_w	T_e	θ
—5	—1.58	—31	796	863	848	16650	17.85	297200	293800	—3400	—5
0	0	0	827	831	816	16030	18.30	293400	293400	0	0
10	3.26	64	891	771	756	14844	18.89	280400	285800	5400	10
20	6.57	129	956	719	704	13823	18.96	262100	269600	7500	20
30	9.84	193	1020	674	659	12939	18.41	238200	245200	7000	30
40	12.95	254	1081	636	621	12193	17.12	208700	213300	4600	40
45	14.40	283	1110	619	604	11860	16.16	191600	194900	3300	45
50	15.76	309	1136	605	590	11585	14.98	173500	175000	1500	50
55	17.01	334	1161	592	577	11330	13.57	153700	153700	0	55
60	18.12	356	1183	581	566	11113	11.92	132500	131300	—1200	60
65	19.08	375	1202	572	557	10937	10.06	110000	107900	—2100	65

D. MANUAL ELEVATION

100. For manual elevation, the handwheel torque must overcome friction as well as the unbalance of the equilibrator and weight moments. The equivalent spring pressure of a typical pneumatic equilibrator (Eq. 47b) is:

$$P_s = \left(\frac{\nu - K_p}{K_p}\right) P_m = \frac{0.88 - 0.73}{0.73} 848 = 174 \text{ psi}$$

where $\nu = 0.88 =$ leakage factor

$K_p = 0.73$ pressure factor for rubber filler

$P_m = 848$ psi = maximum applied pressure (max. P_θ).

Then the frictional force on the piston seals is determined (Eqs. 1 and 47). At the piston seal:

$$f_{p_1} = \pi D w \mu K \; (P_\theta + P_s) \; = 0.269 \; (P_\theta + 174),$$

where $w = 0.625 =$ width of seal, in

$\mu = 0.05 =$ coefficient of packing friction

$P_\theta =$ fluid pressure, psig at any elevation.

At the rod seal: $f_{p_2} = \pi d w \mu K \; (P_\theta + P_s) = 0.090 \; (P_\theta + 174)$.

The total frictional force of the packing then is:

$$f_p = f_{p_1} + f_{p_2} = 0.359 \; (P_\theta + 174), \text{ lb.}$$

Therefore, the total rod force,

$$F_R = F_g - f_p, \text{ lb.}$$

101. To compute bearing torque at the connections, assume that the bearing diameter at both ends is 2 in. ($r_A = r_B = 1$ in.), and the coefficient of friction $\mu = .08$; then (fig. 26):

$$T_A = \mu F_R r_A = 0.08 \, F_R, \text{ lb-in,}$$

and $$T_B = \mu F_R r_B = 0.08 \, F_R, \text{ lb-in.}$$

From $\theta = -5°$ to $\theta = 16° \, 17'$, where $16° \, 17'$ represents position 1 in figure 12,

$$F_A = T_A d/Lr = 0.08 \, F_R d/Lr, \text{ lb,}$$

$$F_B = T_B \, (1 - d/L)/r = 0.08 \, F_R \, (1 - d/L)/r, \text{ lb.}$$

From $\theta = 16° \, 17'$ to $\theta = 65°$,

$$F_A = T_A d/Lr = 0.08 \, F_R d/Lr, \text{ lb,}$$

$$F_B = T_B \, (1 + d/L)/r = 0.08 \, F_R \, (1 + d/L)/r, \text{ lb.}$$

The frictional load on the rod then is:

$$F_f = F_A + F_B, \text{ lb.}$$

MANUAL ELEVATION CALCULATIONS

θ	L	r	d	P_θ	F_g	f_p	F_R	F_A	F_B	F	M_w	M_e	T_e	θ
—5	60.64	17.85	6.51	848	16650	367	16283	8	65	16210	293800	289300	4500	—5
0	59.06	18.30	5.11	816	16030	355	15675	6	63	15606	293400	285600	7800	0
10	55.80	18.89	2.04	756	14844	334	14510	2	59	14449	285800	272900	12900	10
20	52.49	18.96	1.23	704	13823	315	13508	1	58	13449	269600	255000	14600	20
30	49.22	18.41	4.70	659	12939	299	12640	5	60	12575	245200	231500	13700	30
40	46.11	17.12	8.24	621	12193	285	11908	10	66	11832	213300	202600	10700	40
45	44.66	16.16	9.98	604	11860	279	11581	13	70	11498	194900	185800	9100	45
50	43.30	14.98	11.69	590	11585	274	11311	16	77	11218	175000	168000	7000	50
55	42.05	13.57	13.30	577	11330	270	11060	21	86	10953	153700	148600	5100	55
60	40.94	11.92	14.80	566	11113	266	10847	26	99	10722	131300	127800	3500	60
65	39.98	10.06	16.12	557	10973	262	10711	34	120	10557	107900	106200	1700	65

102. Thus, the equilibrator rod force is:

$$F = F_R - F_f, \text{ lb},$$

and the equilibrator moment

$$M_e = Fr, \text{ lb-in}.$$

The torque required to elevate is:

$$T_e = M_w - M_e, \text{ lb-in}.$$

The calculations for manual elevation are tabulated above and shown graphically in figure 44.

E. MANUAL DEPRESSION

103. During manual depression, the packing frictional forces are the same as during manual elevation except that their direction will be reversed:

$$f_p = -0.359\ (P_\theta + 174), \text{ lb}.$$

The total rod force,

$$F_R = F_g - f_p, \text{ lb}.$$

The bearing torques, T_A and T_B, equal μF_R or .08 F_R as before. Then according to figure 27, from $\theta = 65°$ to $\theta = 16°\ 17'$,

$$F_A = -T_A\ d/Lr = -0.08\ F_R\ d/Lr, \text{ lb}.$$

$$F_B = -T_B\ (1+d/L)/r$$
$$= -\ 0.08\ F_R\ (1+d/L)/r, \text{ lb}.$$

From $\theta = 16°\ 17'$ to $\theta = -5°$,

$$F_A = T_A d/Lr = -\ 0.08\ F_R d/Lr, \text{ lb}.$$

$$F_B = -\ T_B\ (1-d/L)/r$$
$$= 0.08\ F_B\ (1-d/L)/r, \text{ lb}.$$

Then, the rod frictional load will be:

$$F_f = F_A + F_B, \text{ lb};$$

and the equilibrator rod force,

$$F = F_R - F_f, \text{ lb}.$$

104. As before, the equilibrator moment is:

$$M_e = Fr, \text{ lb-in}$$

and the torque required to depress,

$$T_e = M_e - M_w, \text{ lb-in}.$$

The accompanying table lists the results obtained from the manual depression calculations. (See figure 44.)

F. POWER ELEVATION

105. In power elevation, polytropic expansion must be considered in calculating packing friction. Power elevation is assumed to begin at the loading angle, in this case 10 degrees elevation. For polytropic expansion,

$$P_{a_\theta} V_\theta^n = P_{10} V_{10}^n,$$

which yields the absolute gas pressure:

$$P_{a_\theta} = P_{10}\left(\frac{V_{10}}{V_\theta}\right)^n = 771\left(\frac{891}{V_\theta}\right)^{1.4},$$

MANUAL DEPRESSION CALCULATIONS

θ	L	r	d	P_θ	F_E	f_p	F_R	F_A	F_B	F	M_w	M_e	T_e	θ
65	39.98	10.06	16.12	557	10973	—262	11235	—36	—125	11396	107900	114600	6700	65
60	40.94	11.92	14.80	566	11113	—266	11379	—28	—104	11511	131300	137200	5900	60
55	42.05	13.57	13.30	577	11330	—270	11600	—22	—90	11712	153700	158900	5200	55
50	43.30	14.98	11.69	590	11585	—274	11859	—17	—80	11956	175000	179100	4100	50
45	44.66	16.16	9.98	604	11860	—279	12139	—13	—73	12225	194900	197600	2600	45
40	46.11	17.12	8.24	621	12193	—285	12478	—10	—69	12557	213300	215000	1700	40
30	49.22	18.41	4.70	659	12939	—299	13238	—6	—63	13307	245200	245000	—200	30
20	52.49	18.96	1.18	705	13823	—315	14138	—1	—61	14200	269600	269200	—400	20
10	55.80	18.89	2.04	756	14844	—334	15178	—2	—62	15242	285800	287900	2100	10
0	59.06	18.30	6.01	816	16030	—355	16385	—7	—64	16456	293400	301100	7700	0
—5	60.64	17.85	6.51	848	16650	—367	17017	—8	—68	17093	293800	305100	11300	—5

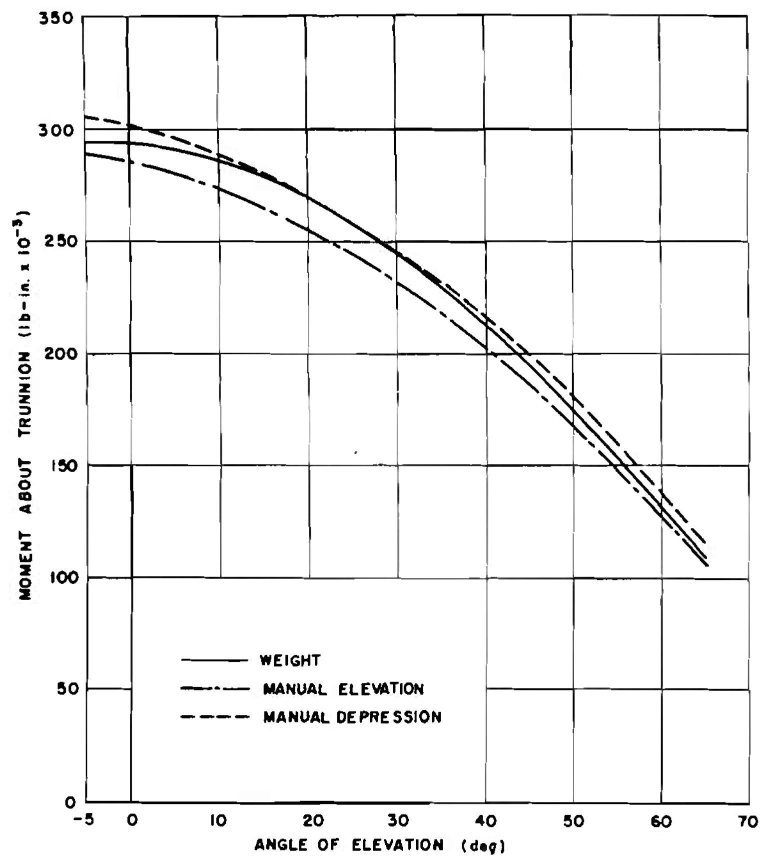


Figure 44. Equilibrator Performance Curves for Manual Operation

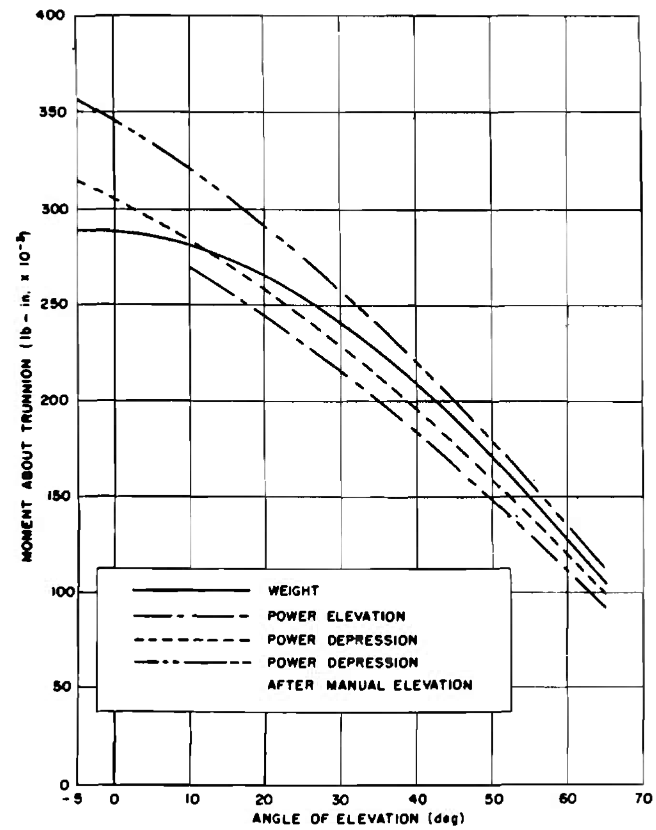


Figure 45. Equilibrator Performance Curves for Power Operation

POWER ELEVATION CALCULATIONS

θ	d/L	r(in.)	V_θ	$\left(\frac{V_{10}}{V_\theta}\right)^{1.4}$	P_θ	F_E	f_p	F_R	F_A	F_B	F	M_w	M_e	T_e	θ
10	0.036	18.89	891	1.000	756	14844	334	14510	2	59	14449	285800	272900	12900	10
20	.022	18.96	956	0.906	683	13411	308	13103	1	56	13046	269600	247400	22200	20
30	.095	18.41	1020	.829	624	12252	286	11966	5	57	11904	245200	219200	26000	30
40	.179	17.12	1081	.762	572	11230	268	10962	9	61	10892	213300	186500	26800	40
45	.224	16.16	1110	.736	552	10838	261	10677	12	64	10501	194900	169700	25200	45
50	.270	14.98	1136	.712	534	10485	254	10231	15	69	10147	175000	152000	23000	50
55	.316	13.57	1161	.690	517	10151	248	9903	18	77	9808	153700	133100	20600	55
60	.362	11.92	1183	.673	504	9896	243	9653	23	88	9542	131300	113700	17600	60
65	.403	10.06	1202	.657	492	9660	239	9421	30	105	9286	107900	93400	14500	65

where $P_{10} = 771$ psia = gas pressure at 10° elevation (par. 99)

$V_{10} = 891$ in³ = gas volume at 10° elevation

$n = 1.4$, assumed.

Gage gas pressure then is:

$$P_\theta = P_{a_\theta} - 15, \text{ psig}$$

The force of the gas on the equilibrator rod:

$$F_g = AP_\theta = 19.635\, P_\theta, \text{ lb.}$$

Summing the packing frictional forces:

$$f_p = .359\ (P + 174)\ \text{(par. 100), lb.}$$

Net rod force then is:

$$F_R = F_g - f_p, \text{ lb.}$$

106. Frictional torque at the bearings and subsequent calculations are exactly as in paragraphs 101 and 102 (manual elevation). The calculations are tabulated in the accompanying table, and the moments about the trunnion are plotted in figure 45.

G. POWER DEPRESSION

107. It is unlikely that the maximum angle of elevation will be maintained for any appreciable length of time. Therefore, it is reasonable to assume that the expanded gases will not regain any heat before power depression begins and the pressure-volume curve for compression will coincide with that for expansion. From polytropic compression, then, with the exponent $n = 1.4$, the absolute gas pressure can be computed.

$$P_{a_\theta} V_\theta^n = P_{65} V_{65}^n$$

$$P_{a_\theta} = P_{65} \left(\frac{V_{65}}{V_\theta}\right)^{1.4}, \text{ psia,}$$

where $P_{65} = 492$ psig = gas pressure at 65° elevation (par. 106)

$V_{65} = 1202$ in³ = gas volume at 65° elevation

Gage gas pressure then is:

$$P_\theta = P_{a_\theta} - 15, \text{ psig.}$$

Gas force on the equilibrator rod:

$$F_g = AP_\theta = 19.635\, P_\theta, \text{ lb.}$$

Total packing friction:

$$f_p = .359\ (P_\theta + 174)\ \text{(par. 100), lb.}$$

Net rod force:

$$F_R = F_g - f_p, \text{ lb.}$$

108. Bearing torque and subsequent calculations are identical with those shown in paragraphs 103 and 104 (manual depression). Power depression calculations are shown in the accompanying table. (See figure 45.)

POWER DEPRESSION CALCULATIONS

θ	d/L	r	V_θ	$\left(\frac{V_{65}}{V_\theta}\right)^{1.4}$	P_θ	F_g	f_p	F_R	F_A	F_B	F	M_w	M_e	T_e	θ
65	.403	10.06	1202	1.000	492	9660	—239	9899	—32	—110	10041	107900	101000	—6900	65
60	.362	11.92	1183	1.023	504	9896	—243	10139	—25	—93	10257	131300	122300	—9000	60
55	.316	13.57	1161	1.050	517	10151	—284	10435	—19	—81	10499	153700	142500	—11200	55
50	.270	14.98	1136	1.082	533	10485	—254	10739	—15	—73	10917	175000	163500	—11500	50
45	.224	16.16	1110	1.120	552	10838	—261	11099	—12	—67	11178	194900	180600	—14300	45
40	.179	17.12	1081	1.160	573	11251	—268	11519	—10	—63	11592	213300	198500	—14800	40
30	.095	18.41	1020	1.258	623	12233	—286	12519	—5	—60	12584	245200	231700	—13500	30
20	.022	18.96	956	1.377	683	13411	—308	13719	—1	—59	13779	269600	261200	—8400	20
10	.036	18.89	891	1.520	756	14844	—334	15178	—2	—62	15242	285800	287900	—2100	10
0	.086	18.30	827	1.688	841	16513	—364	16877	—6	—67	16951	293400	310200	16800	0
—5	.107	17.85	796	1.781	888	17436	—381	17817	—9	—71	17897	293800	319500	25700	—5

H. POWER DEPRESSION AFTER MANUAL ELEVATION

109. The final condition of equilibrator operation concerns power depression after manual elevation. This condition imposes the maximum pressure under which the system may operate. The equations are the same as those for normal power depression (pars. 107 and 108) except for P_{65}, which is 572 psig in the isothermal condition prior to depression. The following table lists the calculations. (See figure 45.)

CALCULATIONS FOR POWER DEPRESSION AFTER MANUAL ELEVATION

θ	d/L	r	V_θ	$\left(\frac{V_{65}}{V_\theta}\right)^{1.4}$	P_θ	F_g	f_D	F_E	F_A	F_B	F	M_w	M_e	T_e	θ
65	.403	10.06	1202	1.000	557	10937	—262	11199	—36	—125	11360	107900	114300	6400	65
60	.362	11.92	1183	1.023	570	11192	—267	11459	—28	—105	11592	131300	138200	6900	60
55	.316	13.57	1161	1.050	586	11506	—273	11779	—22	—91	11892	153700	161400	7700	55
50	.270	14.98	1136	1.082	604	11860	—279	12139	—18	—82	12239	175000	183300	8300	50
45	.224	16.16	1110	1.118	624	12252	—287	12539	—14	—76	12629	194900	204100	9200	45
40	.179	17.12	1081	1.160	649	12743	—295	13038	—11	—72	13121	213300	224600	11300	40
30	.095	18.11	1020	1.258	705	13843	—316	14159	—6	—67	14232	245200	262000	16800	30
20	.022	18.96	956	1.377	773	15178	—340	15518	—1	—67	15586	269600	295500	25900	20
10	.036	18.89	891	1.520	854	16768	—367	17137	—3	—70	17210	285800	325100	39300	10
0	.086	18.30	827	1.688	951	18673	—404	19077	—8	—75	19160	293400	350600	57200	0
—5	.107	17.85	796	1.781	1004	19714	—423	20137	—9	—81	20227	293800	361000	67200	—5

APPENDIX II

SAMPLE CALCULATIONS FOR SPRING EQUILIBRATOR PERFORMANCE

A. PERFECT BALANCE EQUILIBRATOR

110. To illustrate the procedure for calculating the parameters of a perfect balance equilibrator system (par. 26) we have chosen the configuration shown in figure 46 with the following fixed conditions:

$$\alpha = \beta$$

$$\Psi = 90°, \text{ when } \phi = 0°$$

$$\phi' = 5°$$

$$c = 12.15 \text{ in.}$$

$$R = 36.33 \text{ in.}$$

$$R_t = 18 \text{ in.}$$

$$W_t = 2000 \text{ lb.}$$

From Equation (5a),

$$M_{w_0} = R_t W_t = 36000 \text{ lb-in.}$$

From Equation (14a),

$$K_s = \frac{M_{w_0}}{cR} = \frac{36000}{441}$$

$$= 81.5 \text{ lb/in, spring rate.}$$

The maximum spring force will occur at the minimum angle of elevation or

$$\theta = -5°,$$

and $\phi = \theta + \phi' = 0°.$

From Equation (2),

$$L^2 = c^2 + R^2 - 2cR \cos \Psi$$

$$L^2 = 1467 - 0 = 1467$$

$L = 38.3$ in., the maximum spring deflection.

From Equation (8b),

$$F_m = K_s L$$

$= 3120$ lb, the maximum spring force.

Sufficient information is now available to design the spring. A sample problem appears in paragraph 113. If desired, perfect balance can be shown throughout the elevation cycle by substituting values in Equations (5) and (6).

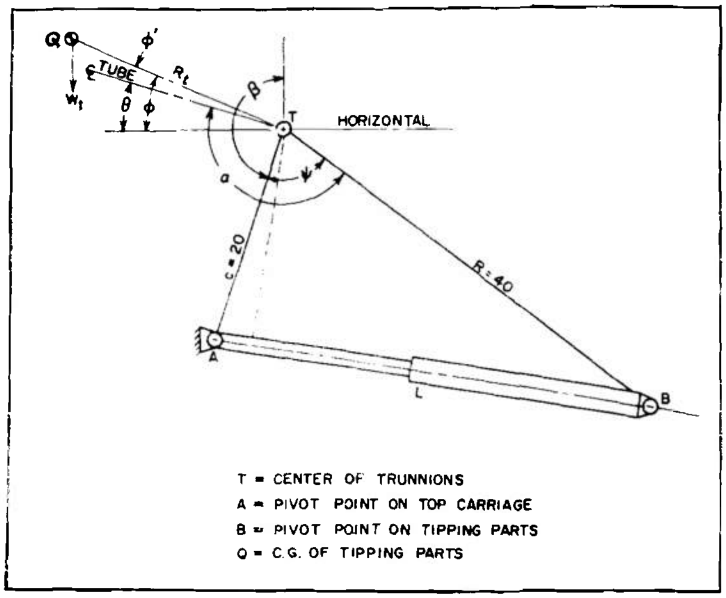


Figure 46. Geometry of Spring Equilibrator

WEIGHT MOMENT CALCULATIONS

θ	ϕ	$\cos\phi$	$R_t \cos\phi$	M_w
—5°	—1° 32′	0.9996	18.00	36000
0°	3° 28′	.9982	17.96	35900
10°	13° 28′	.9725	17.50	35000
20°	23° 28′	.9173	16.50	33000
30°	33° 28′	.8342	15.02	30000
40°	43° 28′	.7258	13.06	26100
45°	48° 28′	.6631	11.94	23900
50°	53° 28′	.5953	10.72	21400
55°	58° 28′	.5230	9.42	18800
60°	63° 28′	.4467	8.04	16100
65°	68° 28′	.3670	6.60	13200

MOMENT ARM CALCULATIONS

θ	Ψ	$\sin\Psi$	$\cos\Psi$	L^2	L	r
—5	120	0.8660	—0.5000	1908.9	43.69	8.75
0	115	.9063	—.4226	1840.6	42.90	9.32
10	105	.9659	—.2588	1696.0	41.17	10.36
20	95	.9962	—.0872	1544.4	39.30	11.19
30	85	.9962	.0872	1390.5	37.29	11.79
40	75	.9659	.2588	1239.0	35.20	12.11
45	70	.9397	.3420	1165.6	34.14	12.15
50	65	.9063	.4226	1094.4	33.08	12.09
55	60	.8660	.5000	1026.1	32.03	11.93
60	55	.8192	.5736	961.1	31.00	11.66
65	50	.7660	.6428	900.0	30.00	11.27

B. APPROXIMATE BALANCE EQUILIBRATOR

111. When space limitations preclude the installation of a perfect balance equilibrator, one must be designed that will closely approximate the ideal. To illustrate the procedures employed in this case, figure 46 is again used, but this time

$$\alpha = \beta,$$

$$\Psi = 115°, \text{ when } \phi = 0,$$

and $\phi' = 3° 28'$.

Geometry

From Equation (2),

$$L_\theta^2 = c^2 + R^2 - 2cR\cos\Psi$$

$$= 1467.5 - 882.8\cos\Psi.$$

From Equation (3),

$$r = cR\sin\Psi/L_\theta$$

$$= 441.3\sin\Psi/L_\theta$$

$$\Psi = \Psi_0 - \theta = 115° - \theta.$$

Weight Moment

112. The tipping parts of this hypothetical weapon weigh 2,000 lb and the dimensions conform to figure 46. The known dimensions are:

$$W_t = 2{,}000 \text{ lb}$$

$$R_t = 18.0 \text{ in.}$$

$$\phi' = 3° 28'.$$

The weight moment can be calculated from Equation (5)

$$M_w = W_t R_t \cos\phi, \text{ lb-in.}$$

Spring Analysis

113. The characteristics of the springs are determined by equating the equilibrator and weight moments at 0° and 55° elevation, and then solving for the spring rate. For this analysis, two concentric springs are considered, where the inner one supports 40 percent of the outer spring load. The total spring loads at 0° and 55° elevation, are, respectively:

$$F_{s_1} = \frac{M_{w_0}}{r_0} = \frac{35900}{9.32} = 3852 \text{ lb},$$

and

$$F_{s_2} = \frac{M_{w_{55}}}{r_{55}} = \frac{18800}{11.93} = 1576 \text{ lb}.$$

Values for M_w and r were obtained from the table entitled *Moment Arm Calculations* and from paragraph 118, respectively.

114. The portion of these loads borne by the outer spring, therefore, are

$$F_{o_1} = \frac{F_{s_1}}{1.4} = 2752 \text{ lb},$$

$$F_{o_2} = \frac{F_{s_2}}{1.4} = 1126 \text{ lb.}$$

The corresponding spring deflection is

$$\Delta L = L_0 - L_{55} = 10.87 \text{ in,}$$

which yields the spring rate:

$$K_{s_0} = \frac{F_{o_1} - F_{o_2}}{\Delta L} = 149.5 \text{ lb/in.}$$

For convenience L_0 is selected to be the equilibrator length when θ equals zero rather than when ϕ equals zero. (Reference paragraph 25.)

115. The procedure and relationships, outlined in bibliography reference 5, are used to determine other parameters of the spring. Assume that the mean diameter of the spring should be

$$D = 5.25,$$

and also assuming that the allowable stress in the spring wire at 0° elevation is

$$\tau_a = 120000 \text{ psi.}$$

Then the wire diameter is given by:

$$d^3 = \frac{2.55\ F_{o_1}D}{\tau_a} = \frac{2.55 \times 2752 \times 5.25}{120000} = 0.303,$$ (bibliography reference 7)

$$d = 0.671 \text{ in.};$$

and the number of coils by:

$$N = \frac{S\,Gd^4}{8\,F_{o_1}D^3} = \frac{18.4 \times 11.5 \times 10^6 \times 0.2027}{8 \times 2752 \times 144.7} = 13.5,$$ (bibliography reference 7)

where S = deflection of the assembled spring

$= F_{o_1}/K_s = 2752/149.5 = 18.4$ in.

G = torsion modulus of rigidity

$= 11.5 \times 10^6$ psi.

(bibliography reference 7)

116. To compute the maximum spring stress, the maximum deflection and maximum load must first be determined.

Maximum deflection:

$$S_m = S + (L_{-5} - L_0) = 18.40 + 0.79 = 19.19 \text{ in.}$$

Maximum load (at −5° elevation):

$$F_m = K_{s_0} S_m = 149.5 \times 19.19 = 2869 \text{ lb.}$$

Therefore, the maximum stress:

$$\tau = \frac{2.55\ F_m D}{d^3} = 127000 \text{ psi.}$$

(bibliography reference 7)

The curve (page 23 of bibliography reference 25) for alloy steel wire heat treated after coiling, shows that the maximum design stress for a wire of 0.67 in. diameter is 132,000 psi, rendering the calculated stress of 127,000 psi acceptable.

117. The characteristics of the inner spring are found in the same manner as shown above. Since the inner spring is known to support 40 percent of the outer spring load, the loads borne by the inner spring at 0° and 55° elevation are, respectively:

$$F_{i_1} = 0.4F_{o_1} = 1100 \text{ lb}$$

$$F_{i_2} = 0.4F_{o_2} = 450 \text{ lb,}$$

and the spring rate,

$$K_{s_i} = \frac{F_{i_1} - F_{i_2}}{\Delta L} = \frac{650}{10.87} = 59.8 \text{ lb/in.}$$

Assume that the mean diameter (D) of the inner spring = 3.5 and that the maximum allowable stress is 120,000 psi. Then the wire diameter will be

$$d^3 = \frac{2.55\,F_{i_1}D}{\tau_a} = \frac{2.55 \times 1100 \times 3.25}{120000} = 0.076,$$

$$d = 0.424 \text{ in.}$$

The number of coils required:

$$N = \frac{SGd^4}{8F_{i_1}D^3} = \frac{18.4 \times 11.5 \times 10^6 \times 0.0323}{8 \times 1100 \times 42.875} = 18.1,$$

where

$$S = \frac{F_{i_1}}{K_{s_i}} = \frac{1100}{59.8} = 18.4 \text{ in.}$$

= deflection of the assembled spring.

Maximum deflection:

$$S_m = S + (L_{-5} - L_0)$$
$$= 18.4 + .79 = 19.19 \text{ in.}$$

Maximum load (at —5° elevation):

$$F_m = K_s S_m = 59.8 \times 19.19 = 1147 \text{ lb.}$$

Maximum stress

$$\tau = \frac{2.55\ F_m D}{d^3} = 135000 \text{ psi.}$$

Bibliography reference 5, page 23, shows that the maximum design stress for alloy steel wire heat treated after coiling is 154,000 psi for a diameter of 0.42 in.

118. It remains to compute the performance of the double spring. The total spring rate:

$$K_s = 149.5 + 59.8 = 209.3 \text{ lb-in.}$$

Spring force on the rod:

$$F_s = K_s\ (S - L) \text{ lb,}$$

where $S = 18.4$ = spring deflection at 0° elevation

$\Delta L = L_0 - L_\theta$ = equilibrator stroke.

The equilibrator moment, when friction is neglected, is

$$M_e = F_s r \text{ lb-in.}$$

and the unbalanced moment:

$$T_e = M_e - M_w \text{ lb-in.}$$

The spring performance calculations are as follows:

θ	Δ	S	F_s	r	M_e	M_w	T_e
—5	—0.79	19.19	4016	8.75	35100	36000	—900
0	0	18.40	3853	9.32	35900	35900	0
10	1.73	16.67	3489	10.36	36100	35000	1100
20	3.60	14.80	3098	11.19	34700	33000	1700
30	5.61	12.79	2677	11.79	31600	30000	1600
40	7.70	10.70	2240	12.11	27100	26100	1000
45	8.76	9.64	2018	12.15	24500	23900	600
50	9.82	8.58	1796	12.09	21700	21400	300
55	10.87	7.53	1576	11.93	18800	18800	0
60	11.90	6.50	1360	11.66	15800	16100	—300
65	12.90	5.50	1151	11.27	13000	13200	—200

119. The problem is not complete because friction has not yet been considered. The method of calculating friction in the bearings follows the procedures as for the pneumatic type in paragraphs 101, 102, and 103, and need not be repeated here.

GLOSSARY

Angle of Elevation — the angle between the center line of the bore and the horizontal when the weapon is resting on a horizontal plane.

Artillery, Fixed — artillery mounted on a permanent foundation, thus not readily transported.

Artillery, Mobile — artillery mounted on conveyances and readily transported.

Belleville Springs — cupped, washer-type springs.

Carriage, Gun — a gun mount equipped with wheels for traveling; wheels are not removed from the mount when it is in the firing position.

Carriage, Top — primary structural unit of a weapon; it supports the tipping parts and moves with the cradle in traverse.

Cradle — the nonrecoiling structure of a weapon which houses the recoiling parts and rotates about the trunnions to elevate the gun.

Cradle Trunnions — cylindrical structure of the cradle which rotates in the elevating bearings as the gun is elevated.

Depress — the process of lowering the gun toward its horizontal position.

Depression — the angle that the gun can be lowered below its horizontal position.

Elevating Gear — the gear which moves the tipping parts in elevation.

Elevating Mechanism — the system which controls and moves the tipping parts.

Elevating Range — the angular distance through which the tipping parts move from the minimum to the maximum angle of elevation.

Elevating System Brakes — a brake unit which prevents the tipping parts from moving after the elevating mechanism ceases to operate.

Elevation — the process of raising the gun through the elevating range.

Emplacement — the process of positioning the weapon for firing.

Equilibrator — the force-producing mechanism whose function is to provide a moment about the cradle trunnions equal and opposite to that caused by the muzzle preponderance of the tipping parts.

Equilibrator, Chain-Type — an equilibrator which transmits its force to the tipping parts by means of a chain; it usually is fixed rigidly to the top carriage.

Equilibrator, Hydropneumatic-Type — an equilibrator operated by a hydraulic fluid which is subject to pneumatic pressure.

Equilibrator, Pivot-Type — an equilibrator which transmits its force to the tipping parts with a piston rod; it is pivoted at both ends and maintains its alignment by rotating about the pivots.

Equilibrator, Pneumatic-Type — an equilibrator which derives its force from compressed gas applied to a piston.

Equilibrator, Pull-Type — an equilibrator whose force is applied in the direction which tends to draw its attachment points toward each other.

Equilibrator, Push-Type — an equilibrator whose force is applied in the direction which tends to force its attachment points away from each other.

Equilibrator, Spring-Hydraulic-Type — an equilibrator operated by springs and controlled by hydraulic pressure.

Equilibrator, Spring-Type — an equilibrator which derives its force from a coil spring.

Equilibrator, Telescoping-Type — an equilibrator whose moving parts consist of two concentric cylinders having an action similar to the sliding tubes of a telescope.

Firing Cycle — the sequence of activity when a gun is fired; in-battery, load, fire, recoil, counterrecoil, buffing, ejection of cartridge case, in-battery.

Ground Clearance — the space between the breech and ground at the end of recoil at highest angle of elevation, also referred to as the under-carriage clearance when traveling.

Gun — weapon consisting of a tube through which projectiles are discharged by the gases generated by a rapidly burning powder.

Gun Assembly — usually considered as the tube, breech housing, and breechblock.

Housing, Spring — the structure which contains a spring.

In-Battery Position — the position of the recoiling parts of a weapon prior to firing.

Launcher, Guided Missile — the structure which supports and aims a guided missile during launching.

Launcher, Rocket — the structure which supports and aims a rocket during launching.

Leakage Factor — the ratio of the radial pressure of a packing to the maximum fluid pressure.

Loading Angle — the angle of elevation specified for loading weapon with ammunition.

Moment, Equilibrator — the moment produced by the equilibrator force about the cradle trunnions.

Moment, Weight — the moment produced by the weight of the tipping parts about the cradle trunnions.

Muzzle Preponderance — the unbalance of the tipping parts; same as the weight moment.

Packing — a seal which converts axial pressure to radial pressure to preclude passage of fluid past two moving surfaces.

Pressure, Absolute — gage pressure plus atmospheric pressure.

Pressure Factor — the ratio of the resulting radial pressure of a packing to the applied lateral pressure.

Pressure, Gage — pressure of a vessel beyond atmospheric pressure.

Recoil Cycle — the sequence of activity after the gun is fired — recoil, counterrecoil, buff.

Recoiling Mass — the mass of the recoiling parts of a weapon.

Recoiling Parts — tube, breech housing, breechblock assembly and parts of recoil mechanism that move during recoil.

Stability — a condition generally associated with a slender structural member whose limit in compression is less than the yield point of the material. A compressive stress above this limit will cause failure and collapse of the structure.

Stuffing Box — the exit chamber of a piston rod containing packings and bearings.

Temperature Adjuster — a device on pneumatic and hydropneumatic equilibrators which compensates for changes in equilibrator force caused by changes in temperature.

Tipping Parts — the assembled structure of a weapon which rotates about the cradle trunnions as it is being elevated.

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Smith and Duncan: *Sampling Statistics and Applications,* McGraw-Hill Book Co.

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(AMCRD-TV)

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ENGINEERING DESIGN HANDBOOKS

Listed below are the Handbooks which have been published or are currently under preparation. Handbooks with publication dates prior to 1 August 1962 were published as 20-series Ordnance Corps pamphlets. AMC Circular 310-38, 19 July 1963, redesignated those publications as 706-series AMC pamphlets (e.g., ORDP 20-138 was redesignated AMCP 706-138). All new, reprinted, or revised Handbooks are being published as 706-series AMC pamphlets.

No.	Title
100	• Design Guidance for Producibility
104	• Value Engineering
106	Elements of Armament Engineering, Part One, Sources of Energy
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186	Military Pyrotechnics, Part Two, Safety, Procedures and Glossary
187	Military Pyrotechnics, Part Three, Properties of Materials Used in Pyrotechnic Compositions
188	• Military Pyrotechnics, Part Four, Design of Ammunition for Pyrotechnic Effects
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* *UNDER PREPARATION – not available at present – will be announced when printed.*

† *OBSOLETE – out of stock or replaced by another handbook.*

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MIL-HDBK-212	Gasket Materials (Nonmetallic)
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MIL-HDBK-692	Guide to Selection of Rubber O-Rings
MIL-HDBK-693 (MR)	Magnesium and Magnesium Alloys
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MIL-HDBK-697 (MR)	Titanium and Titanium Alloys
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