ENGINEERING DESIGN
HANDBOOK
CARRIAGES AND MOUNTS
EQUIVILBRATORS

HEADQUARTERS, U.S. ARMY MATERIEL COMMAND  
DECEMBER 1970
# ENGINEERING DESIGN HANDBOOK

## EQUILIBRATORS

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*This pamphlet supersedes AMCP 706-345, April 1960 (formerly ORDP 20-345).*
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### 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"**: Radial force exerted by packing  
- **F_R**: Net equilibrator rod force  
- **F_K**: Weight components causing unbalanced moment in traverse  
- **f**: 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_a**: Actual length of clock-type spring  
- **L_o**: 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_g**: Gas pressure at any angle of elevation  
- **Q**: Location of c.g. of tipping parts  
- **R**: Turning radius of equilibrator about trunnions  
- **R_e**: Radius of chain drum  
- **R_t**: Distance, trunnion axis to c.g. of tipping parts  
- **r**: Equilibrator moment arm, perpendicular from trunnion to \( L_0 \)  
- **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_e**: Gas volume at any degree of elevation  
- **ΔV**: Change in gas volume  
- **W_t**: Weight of tipping parts  
- **Z**: Section modulus  
- **δ**: Subscript denoting zero position  
- **a**: Angle between \( R \) and \( R, \)  
- **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_0**: Angle locating c.g. of traversing parts from horizontal line through turret center  
- **ψ**: Angle between \( R \) and \( c, \) variable
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.
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 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 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.

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<td>Flexibility of Operation</td>
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<td>1</td>
<td>1</td>
<td>?</td>
<td></td>
</tr>
<tr>
<td>Versatility</td>
<td>2</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>Minimum Cost</td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td></td>
</tr>
</tbody>
</table>

*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.
EQUILIBRATORS

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, \( M_w = W \cdot R \cdot \cos \phi \). 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.
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:

\[
f_p = \mu F_p
\]

(1)

where:

- \( f_p \) = packing friction
- \( F_p \) = radial force exerted by packing

(See detailed discussion in paragraph 71.)

\( \mu = \) 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, $L_e$, at any angle of elevation $\theta$, may be found by the law of cosines; thus,

$$L_e^2 = c^2 + R^2 - 2cR \cos \psi$$  \hspace{1cm} (2)

By the law of sines,

$$\frac{R}{\sin \epsilon} = \frac{L_e}{\sin \psi}, \hspace{1cm} (2a)$$

but

$$\frac{r}{c} = \sin \epsilon; \hspace{1cm} (2b)$$

therefore, the equilibrator moment arm

$$r = \frac{cR}{L_e} \sin \psi, \hspace{1cm} (3)$$

and stroke

$$\Delta L = L_o - L_e, \hspace{1cm} (4)$$

where $L_o$ is the length of the equilibrator when $\phi$ 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,

$$M_w = W_1 R \cos \phi, \text{ weight moment.} \hspace{1cm} (5)$$

When $\phi = 0$, $M_w = M_w = W_1 R_1$, \hspace{1cm} (5a)

$$M_e = r F, \text{ the equilibrator moment,} \hspace{1cm} (6)$$

Combining Equations (5) and (6),

$$F \tau = W_1 R_1 \cos \phi \hspace{1cm} (6b)$$

But,

$$\tau = \frac{cR}{L_e} \sin \psi. \hspace{1cm} (6b)$$

According to figure 14,

$$\psi = 90 - (\beta - 90) = 90 - (\phi + \alpha + \beta); \hspace{1cm} (7)$$

or, according to figure 15,

$$\psi = 90 - (\phi + \alpha + \beta) = 90 - (\phi + \alpha + \beta). \hspace{1cm} (7a)$$

When $\alpha = \beta$, \hspace{1cm} $\psi = 90 - \phi$; \hspace{1cm} (7b)

Therefore, \hspace{1cm} $\sin \psi = \cos \phi$. \hspace{1cm} (7c)
T = CENTER OF TRUNNIONS, PIVOT POINT OF TIPPING PARTS
A = PIVOT POINT OF EQUILIBRATOR ON CARRIAGE, FIXED
B = PIVOT POINT OF EQUILIBRATOR ON CRADLE, ROTATES ABOUT T
C = DISTANCE BETWEEN A AND T, FIXED
D = DISTANCE FROM B TO PERPENDICULAR FROM T TO LG
LG = LENGTH OF EQUILIBRATOR, VARIABLE
Q = LOCATION OF C.G. OF TIPPING PARTS
RL = DISTANCE BETWEEN TRUNNIONS AND Q
R = TURNING RADIUS, CONSTANT
r = EQUILIBRATOR MOMENT ARM, VARIABLE
Wt = WEIGHT OF TIPPING PARTS
F = EQUILIBRATOR FORCE
α = ANGLE FORMED BY RL 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 RL, VARIABLE
ϕ' = ANGLE FORMED BY RL AND TUBE, CONSTANT

Figure 14. Geometry of Equilibrator Shown Below Trunnions
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 $L_\theta$  
$L_\theta$ = 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  
$\alpha$ = ANGLE FORMED BY $R_t$ AND R, CONSTANT  
$\beta$ = ANGLE FORMED BY VERTICAL LINE AND c, FIXED  
$\gamma$ = ANGLE FORMED BY VERTICAL LINE AND R, VARIABLE  
$\psi$ = ANGLE FORMED BY R AND c, VARIABLE  
$\theta$ = ANGLE OF ELEVATION, VARIABLE  
$\phi$ = ANGLE FORMED BY HORIZONTAL LINE AND $R_t$, VARIABLE  
$\phi'$ = 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),

\[ F = \frac{cR}{L_0} \cos \phi = W_t R_t \cos \phi, \quad (8) \]

and

\[ F = \frac{W_t R_t}{cR} L_0 \quad (8a) \]

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

\[ F = K_s L_0, \quad (8b) \]

where

\[ K_s = \frac{W_t R_t}{cR}. \]

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

\[ \Psi = \Psi_0 - \phi, \quad (9) \]

becomes

\[ \Psi = \Psi_0; \quad (9a) \]

and \( r \) becomes

\[ r_0 = \frac{cR}{L_0} \sin \Psi_0. \quad (10) \]

Likewise, \( M_{w_0} = M_0 = r_0 F_0 = W_t R_t \),

\[ (11) \]

and

\[ F_0 = \frac{M_{w_0}}{r_0}. \quad (11a) \]

The spring force at any angle \( \phi \) is

\[ F = F_0 + K_s \Delta L, \quad (12) \]

where \( K_s \) is the spring rate.

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

\[ F = \frac{M_{w_0} L_0}{cR \sin \Psi_0} + K_s (L_0 - L_0); \quad (12a) \]

or, substituting for \( M_{w_0} \) and \( r_0 \),

\[ F = \frac{M_{w_0} L_0}{cR \sin \Psi_0} + K_s (L_0 - L_0). \quad (12b) \]

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

\[ M_{w_0} \cos \phi = \frac{cR}{L_0} \sin (\Psi_0 - \phi) \]

\[ \left[ \frac{M_{w_0} L_0}{cR \sin \Psi_0} + K_s (L_0 - L_0) \right] \quad (13) \]

Rearranging Equation (13),

\[ M_{w_0} \left[ \frac{L_0 \cos \phi}{\sin (\Psi_0 - \phi)} - \frac{L_0}{\sin \Psi_0} \right] = cR K_s (L_0 - L_0). \quad (13a) \]

This is the general balance equation. By inspection, the only value of \( \Psi_0 \) that will make it independent of the variable angle \( \phi \) is 90 degrees. Inserting this value, Equation (13a) reads

\[ M_{w_0} \left[ \frac{L_0 \cos \phi}{\sin (90 - \phi)} - \frac{L_0}{\sin 90} \right] = cR K_s (L_0 - L_0). \quad (13b) \]

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

\[ M_{w_0} = cR K_s. \quad (13c) \]

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

\[ \Psi_0 = 90^\circ \quad (14) \]

and

\[ K_s = \frac{M_{w_0}}{cR}, \quad \text{the spring rate.} \quad (14a) \]

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 \( M_{w_0} \) and \( r_0 \), Equation (11a) may be written:

\[ F_0 = \frac{M_{w_0} L_0}{cR \sin \Psi_0}. \quad (15) \]

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

\[ F_0 = \frac{M_{w_0} L_0}{cR} = K_s, \quad L_0. \quad (15a) \]

\( L_0 \) is equivalent to the spring deflection when \( \phi \) equals zero. The maximum spring deflection, \( S_m \), occurs at the maximum angle of depression where \( L_0 \) becomes \( L_m \) and \( \phi \) becomes \( \phi_m \). Written in terms of the maximum force, \( F_m \), and the spring rate, \( K_s \),

\[ S_m = \frac{F_m}{K_s}. \quad (16) \]
The area under the curve (fig. 16) represents the total spring energy.

\[ E_t = \frac{S_m F_m}{2} = \frac{F_m^2}{2K_s}. \]  

(17)

From Equations (12) and (12b),

\[ F_m = F_o + K_s (L_m - L_0). \]  

(18)

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

\[ F_m = K_s L_m. \]  

(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

\[ E_t = \frac{K_s^2 L_m^2}{2K_s} = \frac{M_{w_0} L_m^2}{2cR}. \]  

(19)

From Equation (2)

\[ L_m^2 = c^2 + R^2 - 2cR \cos \psi, \]  

(19a)

but

\[ \psi = \psi_0 - \phi_D, \]  

(See eq. 9)

\[ \psi_0 = 90^\circ, \]  

(See eq. 14)

and

\[ \cos (\psi_0 - \phi_D) = \sin \phi_D. \]

Substituting for \( L_m^2 \) in Equation (19),

\[ E_t = M_{w_0} \frac{c^2 + R^2 - 2cR \sin \phi_D}{2cR}. \]  

(19b)

Rearranging the values,

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

(19c)

(Note: \( \phi_D \) 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 \( \phi_D \), when angle of depression is maximum; and on the ratio of \( c \) to \( R \), which is the only variable.

Let

\[ K = \frac{c}{R}. \]  

(19d)

By substitution in Equation (19c),

\[ E_t = \frac{M_{w_0}}{2} \left( K + \frac{1}{K} - 2 \sin \phi_D \right). \]  

(19e)

The first derivative of \( E_t \) with respect to \( K \) is

\[ \frac{dE_t}{dK} = \frac{M_{w_0}}{2} \left( 1 - \frac{1}{K^2} \right). \]  

(20)

Equate the first derivative to zero and solve for \( K \):

\[ \frac{M_{w_0}}{2} \left( 1 - \frac{1}{K^2} \right) = 0 \]  

(20a)

\[ K^2 = 1 \]  

(20b)

\[ K = \pm 1. \]  

(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

\[ \frac{d^2E_t}{dK^2} = M_{w_0} \left( \frac{1}{K^3} \right). \]  

(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

\[
\phi_0 = -10^\circ,
\]

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 \( K = 1.0 \). 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 \( \theta \) to 10 degrees less than maximum elevation. For example, assume that the two angles selected are \( \theta = 0^\circ \) and \( \theta = u^\circ \). The weight moments at these elevations are \( M_{w_0} \) and \( M_{w_u} \), and the corresponding equilibrator moment arms are: \( r_0 \) and \( r_u \).

\[
F_0 = \frac{M_{w_0}}{r_0}, \text{ equilibrator force at } 0^\circ, \quad (22)
\]

\[
F_u = \frac{M_{w_u}}{r_u}, \text{ equilibrator force at } u^\circ, \quad (22a)
\]

\[
K_s = \frac{F_0 - F_u}{\Delta L}, \text{ spring rate, } \quad (22b)
\]
where $\Delta L$ is the equilibrator stroke between $0^\circ$ and $u^\circ$ 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, $F_o$, 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,  
$F_o =$ initial equilibrator force, lb,  
$K_s =$ spring constant (or rate) of equilibrator, lb/in  
and $\Delta L =$ equilibrator travel, in,  
then $F = F_o - K_s \Delta L$. (23)

Summing up the forces at all increments of travel:

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

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

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

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

<table>
<thead>
<tr>
<th>$N$</th>
<th>$\Delta L$</th>
<th>$\Delta L^2$</th>
<th>$F$</th>
<th>$F\Delta L$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>0</td>
<td>1300</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>1</td>
<td>1130</td>
<td>1130</td>
</tr>
<tr>
<td>3</td>
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<td>4</td>
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<td>960</td>
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<td>930</td>
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<td>6020</td>
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<td>820</td>
<td>6560</td>
</tr>
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<td>9</td>
<td>81</td>
<td>780</td>
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<td>11</td>
<td>10</td>
<td>100</td>
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<td>7300</td>
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<td>12</td>
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<td>121</td>
<td>670</td>
<td>7370</td>
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<tr>
<td>13</td>
<td>12</td>
<td>144</td>
<td>620</td>
<td>7440</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$\Sigma = 78$</td>
<td>650</td>
</tr>
</tbody>
</table>

Figure 18. Equilibrator Force Curve
Writing the equations for $\Sigma F$ and $\Sigma F\Delta L$, and solving for $K_s$,

\[ 11770 = 13F_0 - 78K_s \]
\[ 61810 = 78F_0 - 650K_s \]

therefore, $K_s = 48.406 = \text{spring rate of equilibrator, lb/in.}$

and $F_0 = 1196 \text{ initial force, lb.}$

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**

A = CENTER OF TORSION BAR
B = TIPPING PARTS ATTACHMENT
T = TRUNNIONS
Q = C.G., TIPPING PARTS
$W_t$ = WEIGHT, TIPPING PARTS
AC = TORSION BAR ARM
TB = EQUILIBRATOR MOMENT ARM
BC = CONNECTING LINK

SUBSCRIPT SHOWS POSITION WHEN $\phi = 0$
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)

$$F_d = \frac{M_w}{R \sin \beta},$$

tensile load in link $d$, (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 $\gamma$ and $\epsilon$ are right angles when $\phi = 0$, and that $\psi$ is a right angle at $\phi_1$; hence $\psi_0 = 90^\circ + \phi_1$. 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:

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

(27)

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

(27a)

$$K_t = \frac{T_0 - T_1}{\Delta \gamma_1},$$

(27b)

where $K_t$ is torsional spring constant.

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

$$T_s = K_t \Delta \gamma_1,$$

(28)

$$M_s = T_s \frac{R \sin \phi}{r \sin \epsilon},$$

(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

$$T_s' = \frac{T_s}{n},$$

(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:

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

The chain tension is

$$F_c = \frac{M_w}{r},$$

(30)

where

$$r = \frac{cR}{L} \sin \psi,$$ (See eq. 3)

and

$$T_s = r_1 F_c,$$ (30a)

where

$$r_1 = \frac{c_1 R_1}{L_1} \sin \psi_1.$$ (See eq. 3)

The computed values of $T_s$ near the angles of elevation of $\phi = 0$ and $\phi = \phi_{\text{max}}$ determine the spring rate. For the first trial, the angles $\psi$ and $\psi_1$, 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 \( \psi = 90^\circ \),

\[
K_s = \frac{M \omega_0}{cR}, \quad \text{(See eq. 14a)}
\]

and

\[
F_c = K_s L_c, \quad \text{(See eq. 8b)}
\]

where \( F_c \) = chain tension.

The distance, \( L_c \), 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

\[
T_s = F_c R_s \quad \text{(31)}
\]

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

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

where \( b \) = width of spring,

\( h \) = thickness of the spring material.

---

![Diagram of a torsion bar equilibrator](image)

**Figure 20. Torsion Bar Equilibrator — Chain Type**

- \( A, A_1 \) = Points of tangency at sprocket
- \( B \) = Chain attachment to tipping parts
- \( B_1 \) = Chain attachment to torsion bar arm
- \( L_c, L_1 \) = Chain lengths between sprocket and attachments
- \( R \) = Equilibrator moment arm
- \( R_s \) = Torsion bar arm
- \( S_T \) = Axis of torsion bar
- \( S_T \) = Axis of trunnion
- \( W_T \) = Weight of tipping parts
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If the width is large and unwieldy, several springs may be used, with \( b \) representing the total width. The active length of the spring is

\[
L_a = \frac{LEI}{F_c R_s^2},
\]

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

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

Referring to figure 21, angle \( \psi \) is known for any angle \( \phi \). Then

\[
d^2 = R^2 + e^2 - 2eR \cos \psi.
\]

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 \( ABA', BAT \), and \( BAT \) necessary to complete the equilibrator analysis.

**Figure 21. Geometry of Clock Spring Equilibrator**
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 $TT,B_1B_1'$ 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 $\epsilon$ which changes the weight moment to

$$M_w = M_w \cos (\phi + \epsilon).$$

(32)
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

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

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

$$\frac{M_{w_0}}{cR} L_\theta \frac{cR}{L_\theta} \sin \psi = M_w \cos (\phi + \epsilon). \quad (32b)$$

Reducing to the simplest terms,

$$\sin \psi = \cos (\phi + \epsilon) = \sin [90 - (\phi + \epsilon)], \quad (32c)$$

and

$$\psi = 90 - (\phi + \epsilon). \quad (32d)$$

Thus, if the weapon pitches through an angle, \( \epsilon \), angle \( \psi \) 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

$$F_w = W_t \sin \theta_t,$$  \hspace{1cm} (33)

where

- $W_t =$ weight of traversing unit,
- $\theta_t =$ slope of the terrain.

From Equation (5), substituting $\phi_a$ for $\phi$, the weight moment becomes

$$M_w = F_w \cos \phi_a \cos \theta_t \cos \alpha$$  \hspace{1cm} (33a)

where $\phi_a$ 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 $\psi = 90^\circ$ and according to Equation (14a), the spring rate is

$$K_s = \frac{F_w R}{c R}$$  \hspace{1cm} (33b)

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

$$F = K_s L.$$  \hspace{1cm} (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.
The numbers from 1 to 12 indicate the C.G. locations and the corresponding locations of the equilibrator for various positions of the turret for any given slope.

Arrows show direction of weight component responsible for the unbalancing moment.

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 \( \beta \) 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 \( \theta \) 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

\[
M_w = W_r R \cos \phi;
\]

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

\[
M_e = F r.
\]

In the above,

\( \phi \) = angle of elevation
of center of gravity of tipping parts
\( \theta \) = angle of elevation, degrees
\( F = K_s [S - (L_0 - L)] \)
= spring force, lb,
equivalent to \( F \) of hydropneumatic equilibrators
\( [S - (L_0 - L)] = \) net spring deflection, in.
\( S = \) spring deflection,
in., when \( \phi = 0 \)
\( S_m = \) maximum spring deflection

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.

\[
F = K_s [S - (L_0 - L)] = \text{spring force, lb (Eq. 34)}
\]

\[
F_R = F = \text{net rod force used for computing friction}
\]

\[
T_A = \mu F_R r_A = \text{frictional torque, lb-in, at bearing A, on carriage (fig. 26)} \quad (34a)
\]

\[
F_A = f (T_A) = \text{load in rod, lb, due to friction in bearing A (fig. 26)} \quad (34b)
\]

\[
T_B = \mu F_R r_B = \text{frictional torque, lb-in, at bearing B, on cradle (fig. 26)} \quad (34c)
\]

\[
F_B = f (T_B) = \text{load in rod, lb, due to friction in bearing B, (fig. 26)} \quad (34d)
\]

\[
F_f = F_A + F_B = \text{total frictional load in rod, lb} \quad (34e)
\]

\[
F = F_R - F_f = \text{net force on equilibrator rod; i.e., spring load, lb} \quad (34f)
\]

\[
M_s = F r = \text{equilibrator moment, lb-in} \quad (34g)
\]

\[
T_e = M_w - M_r = \text{torque required to elevate, lb-in (at trunnions)} \quad (34h)
\]

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 \( \theta = 0^\circ \), although the minimum angle may be less than \( 0^\circ \).
NOTE:

\[ \mu = \text{COEFFICIENT OF FRICTION IN THE BEARING} \]
\[ F_n = \text{NET ROD FORCE ON THE BEARING} \]
\[ r_A \text{ AND } r_B = \text{RADII OF BEARINGS A AND B} \]
\[ P_A \text{ AND } P_B = \text{REACTIONS DUE TO TORQUES } T_A \text{ AND } T_B \]

AS CRADLE ROTATES FROM \( \theta_0 \) TO \( \theta_1 \), EQUILIBRATOR ROTATES CLOCKWISE ABOUT \( A \)

\[ F_A \]
\[ T_A \]
\[ P_A \]
\[ d \]
\[ \Sigma M_A = 0 \]
\[ P_A L + T_A \mu F_n A \]

AS CRADLE ROTATES FROM \( \theta_1 \) TO \( \theta_2 \), EQUILIBRATOR ROTATES COUNTER-CLOCKWISE ABOUT \( A \)

\[ F_A \]
\[ T_A \]
\[ P_A \]
\[ d \]
\[ \Sigma M_A = 0 \]
\[ P_A L + T_A \mu F_n A \]

\( (A) \) Friction at Carriage Bearing

AS CRADLE ROTATES FROM \( \theta_2 \) TO \( \theta_3 \), EQUILIBRATOR ROTATES COUNTER-CLOCKWISE ABOUT \( B \)

\[ F_B \]
\[ T_B \]
\[ P_B \]
\[ d \]
\[ \Sigma M_B = 0 \]
\[ P_B L + T_B \mu F_n B \]

AS CRADLE ROTATES FROM \( \theta_3 \) TO \( \theta_4 \), EQUILIBRATOR ROTATES COUNTER-CLOCKWISE ABOUT \( B \)

\[ F_B \]
\[ T_B \]
\[ P_B \]
\[ d \]
\[ \Sigma M_B = 0 \]
\[ F_B + P_B d - T_B \]
\[ F_B + T_B \]
\[ F_B + T_B (1 + d/L)/r \]

\( (B) \) Friction at Cradle Bearing

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., \( F_r \) will be negative:

\[
F = F_r - F_f = \text{net force on rod, lb}
\]

(See eq. 34e)

\[
M_e = Fr = \text{equilibrator moment, lb-in}
\]

(See eq. 34g)

\[
T_e = M_e - M_w = \text{torque required to depress, lb-in, (at trunnions)}.
\]

(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.

\[
P_o = \text{gas pressure at any angle of elevation},
\]

\[
P_o = P_o - 15 = \text{gas pressure, psig}
\]

(36)

\[
P_o = \frac{(P_o + 15)}{V_o} = \text{pressure, psia}
\]

(36a)

\[
F_o = A P_o = \text{gas force, lb, where } A \text{ is the effective piston area, in}^2
\]

(36b)

\[
V_o = \text{gas volume at zero elevation, in}^3
\]

(36c)

\[
\Delta L = (L_o - L_a) = \text{equilibrator stroke, in. (See eq. 4)}
\]

\[
V_o = V_o + A \Delta L = \text{gas volume, in}^3
\]

(36d)

\[
F = F_o = \text{equilibrator force, lb, since friction is not considered.}
\]

\[
M_e = Fr = \text{equilibrator moment, lb-in}
\]

(See eq. 6)

\[
M_c = W_i R_i \cos \phi = \text{weight moment, lb-in. (See eq. 5)}
\]

The equilibrator force is made equal to \( \frac{M_w}{r} \) at \( \theta = 0^\circ \), 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.

\[
P = \frac{(P_o + 15)}{V_o} - 15 = \text{gas pressure, psig}
\]

(37)

\[
F_o = A P_o = \text{gas force, lb (See eq. 36b)}
\]

\[
f_p = \text{friction of packing, lb (par. 22)}
\]

\[
F_r = f_p = \text{net rod force used for computing friction in bearing, lb}
\]

(37a)

\[
T_A = \mu F_R \tau_A = \text{frictional torque, lb-in, at bearing } A, \text{ on carriage (fig. 27)} (\text{See eq. 34a})
\]

\[
F_A = f (T_A) = \text{load in rod, lb, due to friction in bearing } A
\]

(See eq. 34b)
EQUILIBRATORS  
AMCP 706-345  

NOTE:
\( \mu \) = COEFFICIENT OF FRICTION IN THE BEARING
\( F_a = \) NET ROD FORCE ON THE BEARING
\( r_A \) AND \( r_B \) = RADIUS OF BEARINGS A AND B
\( P_A \) AND \( P_B \) = REACTIONS DUE TO TORQUES \( T_A \) AND \( T_B \)

AS CRADLE ROTATES FROM \( \theta_u \) TO \( \theta_i \),
EQUILIBRATOR ROTATES CLOCKWISE ABOUT A

\[
\begin{align*}
\Sigma M_A &= 0 \\
F_a L + T_A - \mu F_a r_A &= 0
\end{align*}
\]

AS CRADLE ROTATES FROM \( \theta_i \) TO \( \theta_o \),
EQUILIBRATOR ROTATES COUNTER-CLOCKWISE ABOUT A

\[
\begin{align*}
\Sigma M_A &= 0 \\
F_a L + T_A - \mu F_a r_A &= 0
\end{align*}
\]

\[
\begin{align*}
\Sigma M_f &= 0 \\
F_a r + P_a d &= 0
\end{align*}
\]

\[
\begin{align*}
F_a &= -P_a d/r = -T_a d/Lr
\end{align*}
\]

(A) Friction at Carriage Bearing

AS CRADLE ROTATES FROM \( \theta_o \) TO \( \theta_d \),
EQUILIBRATOR ROTATES CLOCKWISE ABOUT B

\[
\begin{align*}
\Sigma M_B &= 0 \\
P_B L + T_B - \mu F_B r_B &= 0
\end{align*}
\]

AS CRADLE ROTATES FROM \( \theta_d \) TO \( \theta_i \),
EQUILIBRATOR ROTATES CLOCKWISE ABOUT B

\[
\begin{align*}
\Sigma M_T &= 0 \\
F_B r + P_B d + T_B &= 0
\end{align*}
\]

\[
\begin{align*}
F_B &= -T_B d/L
\end{align*}
\]

AS CRADLE ROTATES FROM \( \theta_i \) TO \( \theta_u \),
EQUILIBRATOR ROTATES CLOCKWISE ABOUT B

\[
\begin{align*}
\Sigma M_T &= 0 \\
F_B r - P_B d - T_B &= 0
\end{align*}
\]

\[
\begin{align*}
F_B &= -(T_B + P_B d)/r \text{ but } P_B = T_B/L
\end{align*}
\]

\[
\begin{align*}
F_B &= -T_B (l + d/L)/r
\end{align*}
\]

(B) Friction at Cradle Bearing

Figure 27. Friction Moments During Depression

27
EQUILIBRATORS

\[ T_B = \mu F_B \tau_B = \text{frictional torque, lb-in, at bearing } B, \text{ on cradle (fig. 27)} \]  
\[ F_B = f(T_B) = \text{load in rod, lb, due to friction in bearing } B \]  
\[ F_B = F_A + F_B = \text{total load due to friction of bearings} \]  
\[ F = F_B - F_A = \text{net force, lb, in equilibrator rod} \]  
\[ M_e = M_r - T_e = \text{equilibrator moment, lb-in} \]  
\[ T_e = M_r - M_e = \text{torque, lb-in, required to elevate} \]  
\[ F = F_H - F_B = \text{net rod force, lb} \]  
\[ T_e = M_e - M_w = \text{torque, lb-in, required to depress} \]

48. In manual depression, the calculations are similar to manual elevation, except that the frictional forces are reversed. \( F_B \) and \( f \) will have negative values:

\[ F = F_B - F_A = \text{net rod force, lb} \]

\[ T_e = M_e - M_w = \text{torque, lb-in, required to depress} \]

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

From the equation, \( P_1 V_1^* = P_2 V_2^* \), for polytropic expansion,

\[ P_t = \frac{(P_a + 15) V_2^*}{V_1^*} - 15 \text{ psig} \]

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 \( PV^{1.65} = C \) 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, \( \theta \), 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.

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<th>Depression</th>
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<td>*</td>
<td>*</td>
</tr>
<tr>
<td>(\tau)</td>
<td>*</td>
<td>*</td>
<td>*</td>
</tr>
<tr>
<td>(\tau_e)</td>
<td>*</td>
<td>*</td>
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<td>(P_\theta(V_\theta/V_\theta))</td>
<td>(P_\theta(V_\theta/V_\theta))</td>
</tr>
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<td>(AP_\sigma)</td>
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</table>

* Determined from geometry.
** \(P_\theta = P_a + 15\), absolute pressure at initial conditions.

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 \( \theta = 0^\circ \) and at \( \theta = u^\circ \) (near maximum elevation), by solving for the corresponding forces and pressures, and subsequently by solving for the initial gas volume.

\[
M_e = M_w \text{ at } 0^\circ \text{ and } u^\circ \text{ elevation (See eq. 6)}
\]

\[
F_0 = \frac{M_w}{r_0} = \text{equilibrator force at zero elevation, lb (38)}
\]

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

\[
F_u = \frac{M_w}{r_u} = \text{equilibrator force at } u^\circ \text{ elevation, lb (38b)}
\]

\[
P_u = \frac{F_u}{A} = \text{gas pressure at } u^\circ \text{ elevation, psig (38c)}
\]

\[
\Delta L_u = \text{equilibrator stroke at } u^\circ \text{ elevation, in. (38d)}
\]

\[
\Delta V_u = A \Delta L_u = \text{displacement at } u^\circ \text{ elevation, in}^3 (38e)
\]

\[
V_u = V_0 + \Delta V_u = \text{gas volume at } u^\circ \text{ elevation, in}^3 (38f)
\]

\[
P_u = P_e + P_A = \text{psia (39)}
\]

where \( P_A = 15 \text{ psi, atmospheric pressure.} \)

According to Boyle’s Law of isothermal expansion:

\[
P_0 \frac{V_0}{P_u} = P_u \frac{V_u}{P_0} = \text{gas pressure, psig, at any angle (39c)}
\]

\[
F_e = P_e A = \text{equilibrator force, lb, at any angle (39d)}
\]

\[
M_e = \tau_0 F_e = \text{equilibrator moment, lb-in (39e)}
\]

\[
T_e = M_e = \text{moment required to elevate, lb-in. (See eq. 34h)}
\]

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 \( \theta_{max} \) to \( \theta_{min} \)

\[
P_{max} = P_0(\min) \left( \frac{V_{max}}{V_{min}} \right)^{1.4} = 15 = P_d \text{ psig, (40)}
\]

where \( P_d \) is the limit design pressure of the packing.

Find value for \( P_0(\text{min}) \), minimum absolute pressure, and solve for \( P_0(\text{min}) \),

\[
P_0(\text{min}) = P_0(\text{min}) \left( \frac{V_{max}}{V_0} \right) - 15 \text{ psig (40a)}
\]

equilibrator pressure at \( 0^\circ \) elevation or the maximum charging pressure.

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

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.
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, $\sigma_t$, of the cylinder wall (due to fluid or packing) may be found by the Lamé formula for thick-walled cylinders:

$$\sigma_t = P_{max} \left( \frac{d_2^2 + d_1^2}{d_2^2 - d_1^2} \right)$$  \hspace{1cm} (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.)

$$\sigma_t = \frac{3P_t (m^2 - 1)}{4mt^2} \left[ \frac{a^4 - b^4 - 4a^2b^2 \log a/b}{a^2(m-1) + b^2(m+1)} \right]$$  \hspace{1cm} (43)

where:

- $a$ = outer radius
- $b$ = inner radius
- $m$ = reciprocal of Poisson’s ratio
- $P_t$ = pressure
- $t$ = plate thickness.

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

\[
M = T_A \sec \frac{1}{2} U, \tag{44}
\]

and the maximum compressive stress is

\[
\sigma_c = \frac{M}{Z} + \frac{F_R}{A_r} \tag{45}
\]

where

\[
T_A = \mu F_R r_A
\]

\[
U = \frac{L_r}{j}
\]

\[
j = \sqrt{\frac{EI}{F_R}}
\]

\[
E = \text{modulus of elasticity, psi}
\]

\[
L_r = \text{length of rod, in.}
\]

\[
A_r = \text{cross section area at mid-span, in}^2
\]

\[
Z = \text{section modulus at mid-span, in}^3.
\]

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):

\[
\sigma = a \frac{8 F_R}{\pi r_0 z t}, \tag{46}
\]

where

\[
a = 4.35 \text{ when } 2 \leq \frac{r_0}{r_i} \leq 4
\]

\[
F_R = \text{applied equilibrator load}
\]

\[
r_i = \text{radius of hole}
\]

\[
r_0 = \text{radius of lug}
\]

\[
t = \text{thickness of lug}
\]

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.
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:

\[ F_p = \pi D w K_p (P_a + P_f), \text{lb.} \]  
(47)

Expressing radial pressure in terms of fluid pressure,

\[ K_p (P_a + P_m) = v P_m, \]  
(47a)

from which the required spring pressure is

\[ P_s = \frac{v - K_p}{K_p} P_m, \text{psi} \]  
(47b)

where \( D \) = outside diameter of packing, in.

\[ K_p \] = pressure factor*

\[ v \] = leakage factor**

\[ P_m \] = maximum fluid pressure, psi

\[ P_s \] = axial pressure in packing produced by spring, psi

\[ P_0 \] = fluid pressure at elevation angle \( \theta \), 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

\[ F_a = \pi (r_2^2 - r_1^2) P_s \]  

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

\[ \frac{P_1}{P_2} = \frac{T_1}{T_2}. \]

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 $r'$,

$$
\psi = \sin^{-1} \frac{r}{R} \\
\psi' = \sin^{-1} \frac{r'}{R} \\
e = L \tan (\psi - \psi') \\
L' = \frac{L}{\cos (\psi - \psi')}
$$

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](image-url)
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
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.)

\[
\begin{align*}
A_v &= \text{valve pressure area} \\
P_e &= \text{equilibrator pressure, any position} \\
F_v &= A_v P_e = \text{valve load, any position} \\
F_m &= \text{maximum valve spring load} \\
F_0 &= \text{minimum valve spring load} \\
\Delta &= \text{spring deflection between } F_0 \text{ and } F_m \\
\Delta_0 &= \text{spring deflection at minimum pressure} \\
R_0 &= \text{minimum cam radius} \\
\lambda &= \text{mechanical advantage of linkage} \\
\theta &= \text{angle of elevation} \\
K_s &= \frac{F_m - F_0}{\Delta_0} = \text{spring rate}
\end{align*}
\]

\[
\Delta = \frac{F_v}{R_s} = \text{spring deflection at } \theta \\
d_s = \Delta - \Delta_0 = \text{control rod travel} \\
R_c = R_0 + \Delta R_c = \text{cam radius at } \theta,
\]

where \( \Delta R_c = \lambda d_s \). The value of \( \Delta R_c \) 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,
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 \( \theta = 0, \phi = 86° 48' \)

- \( c = 56.989 \) in.
- \( \theta_{\text{max}} = 65° \)
- \( R = 19 \) in.
- \( \beta = 123° 12' \)
- \( \theta_{\text{min}} = -5° \)

From Equation (2)

\[
L_e^2 = c^2 + R^2 - 2cR \cos \phi
\]

= 3608.75 - 2165.58 \cos \phi, \text{ in}^2.

From Equation (3)

\[
r = cR \sin \phi / L_e
\]

= 1082.79 \sin \phi / L_e, \text{ in.}

\[
d = \sqrt{R^2 - r^2}, \text{ in.}
\]

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:

- \( W_t = 10,000 \text{ lb} \)
- \( R_t = 29.39 \text{ in.} \)
- \( \phi' = 3° 28' \)

and from Equation (5),

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

\[
\Delta L = L_0 - L_e.
\]

For convenience \( L_0 \) is selected to be the equilibrator length when \( \theta \) equals zero rather than when \( \phi \) 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

<table>
<thead>
<tr>
<th>( \theta )</th>
<th>( \phi )</th>
<th>( \sin \phi )</th>
<th>( \cos \phi )</th>
<th>( L_e^2 )</th>
<th>( L_0 )</th>
<th>( r )</th>
<th>( d )</th>
</tr>
</thead>
<tbody>
<tr>
<td>-5°</td>
<td>91° 48'</td>
<td>0.9964</td>
<td>-0.0314</td>
<td>3676.8</td>
<td>60.64</td>
<td>17.85</td>
<td>6.51</td>
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<td>86° 48'</td>
<td>0.9984</td>
<td>0.0558</td>
<td>3487.9</td>
<td>59.06</td>
<td>18.30</td>
<td>5.11</td>
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<td>0.2284</td>
<td>3114.2</td>
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<td>18.89</td>
<td>2.04</td>
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<tr>
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<td>0.3939</td>
<td>2755.6</td>
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<td>18.96</td>
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<td>56° 48'</td>
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<td>0.5476</td>
<td>2423.0</td>
<td>49.22</td>
<td>18.41</td>
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<td>46° 48'</td>
<td>0.7290</td>
<td>0.6846</td>
<td>2126.3</td>
<td>46.11</td>
<td>17.12</td>
<td>8.24</td>
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<td>44.66</td>
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</tr>
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<td>36° 48'</td>
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WEIGHT MOMENT CALCULATIONS

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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,

$$F_0 = \frac{M_w}{r_0} = \frac{293400}{18.30} = 16080 \text{ lb},$$

Rod force at $55°$ elevation,

$$F_{55} = \frac{M_w}{r_{55}} = \frac{153700}{13.57} = 11380 \text{ lb}.$$

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,

$$P_{55} = 816 + 15 = 831 \text{ psia},$$

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

\[ \Delta L_{55} = 17.10 \text{ in.} = \text{equilibrator stroke at 55° elevation.} \]

Thus, from Boyle's Law:

\[ P_a V_0 = P_{a55} V_{55} \]

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 \( (T_e = M_w - M_r) \):

\[ \Delta V = A \Delta L = 19.635 \Delta L \text{ in}^3 \]

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

Absolute pressure, \( P_\theta = \frac{P_a V_0}{V_\theta} = \frac{687237}{V_\theta} \text{ psia} \)

Gas pressure, \( P_\theta = P_a - 15 \text{ psig} \)

Rod force, \( F_r = P_a A = 19.635 P_a \text{ lb} \)

Equilibrator moment, \( M_r = F_r r \text{ lb-in.} \)

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

---

**Figure 43. Equilibrator Performance Curves for a Frictionless System**
## EQUILIBRATORS

### TORQUE REQUIRED TO ELEVATE

**(Friction Not Considered)**

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### 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:

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

where

- $r = 0.88$ = leakage factor
- $K_p = 0.73$ pressure factor for rubber filler
- $P_m = 848$ psi = maximum applied pressure (max. $P_s$).

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

$$f_{s1} = \pi dw \mu K (P_s + P_x) = 0.269 (P_s + 174),$$

where

- $w = 0.625$ = width of seal, in
- $\mu = 0.05$ = coefficient of packing friction
- $P_s$ = fluid pressure, psig at any elevation.

At the rod seal: $f_{s2} = \pi dw \mu K (P_x + P_s)$

$$= 0.090 (P_s + 174).$$

The total frictional force of the packing then is:

$$f_s = f_{s1} + f_{s2} = 0.359 (P_s + 174), \text{ lb.}$$

Therefore, the total rod force,

$$F_R = F_s - f_s, \text{ lb.}$$

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

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

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

From $\theta = 16^\circ 17'$ to $\theta = 65^\circ$,

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

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

The frictional load on the rod then is:

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

**AMCP 706-345**

**MANUAL ELEVATION CALCULATIONS**

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<th>L</th>
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<th>F₀</th>
<th>t₀</th>
<th>F₀</th>
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102. Thus, the equilibrator rod force is:

\[ F = F_R - F_r, \text{ lb}, \]

and the equilibrator moment

\[ M_e = F_r r, \text{ lb-in}. \]

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:

\[ f_r = -0.359 (P_\theta + 174), \text{ lb}. \]

The total rod force,

\[ F_R = F_v - f_r, \text{ lb}. \]

The bearing torques, \( T_A \) and \( T_B \), equal \( \mu F_R \) or \( 0.08 F_R \) as before. Then according to figure 27, from \( \theta = 65^\circ \) to \( \theta = 16^\circ 17' \),

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

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

From \( \theta = 16^\circ 17' \) to \( \theta = -5^\circ \),

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

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

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:

\[ M_e = F_r r, \text{ lb-in}. \]

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,

\[ P_{ae} V_{f}^* = P_{10} V_{10}^*, \]

which yields the absolute gas pressure:

\[ P_{ae} = P_{10} \left( \frac{V_{10}}{V_f} \right)^* \approx 771 \left( \frac{891}{V_f} \right)^{1.4}, \]
### MANUAL DEPRESSION CALCULATIONS

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*Figure 44. Equilibrator Performance Curves for Manual Operation*
**Figure 45. Equilibrator Performance Curves for Power Operation**

**POWER ELEVATION CALCULATIONS**

| \( \theta \) | \( d/L \) | \( r \) (in.) | \( V_{\theta} \) | \( \left( \frac{V_{\theta}}{V_{\theta}} \right)^{1.4} \) | \( P_{e} \) | \( F_{e} \) | \( F_{p} \) | \( F_{m} \) | \( F \) | \( M_{w} \) | \( M_{s} \) | \( t_{s} \) | \( \theta \) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 10 | 0.036 | 18.89 | 891 | 1.000 | 14844 | 334 | 14510 | 2 | 59 | 14649 | 285800 | 272900 | 12900 | 10 |
| 20 | 0.022 | 18.96 | 956 | 0.906 | 683 | 13411 | 308 | 13103 | 1 | 56 | 13046 | 269600 | 247400 | 22200 | 20 |
| 30 | 0.095 | 18.41 | 1020 | 0.829 | 624 | 12252 | 286 | 11966 | 5 | 57 | 11904 | 245200 | 219200 | 26000 | 30 |
| 40 | 0.179 | 17.12 | 1081 | 0.762 | 572 | 11230 | 268 | 10962 | 9 | 61 | 10892 | 213800 | 186500 | 26800 | 40 |
| 45 | 0.224 | 16.16 | 1110 | 0.736 | 552 | 10838 | 261 | 10677 | 12 | 64 | 10501 | 194900 | 169700 | 25200 | 45 |
| 50 | 0.270 | 14.98 | 1136 | 0.712 | 534 | 10485 | 254 | 10231 | 15 | 69 | 10147 | 175000 | 152000 | 23000 | 50 |
| 55 | 0.316 | 13.57 | 1161 | 0.690 | 517 | 10151 | 248 | 9903 | 18 | 77 | 9808 | 153700 | 133100 | 20600 | 55 |
| 60 | 0.362 | 11.92 | 1183 | 0.673 | 504 | 9896 | 243 | 9653 | 23 | 88 | 9542 | 131300 | 113700 | 17600 | 60 |
| 65 | 0.403 | 10.06 | 1202 | 0.657 | 492 | 9660 | 239 | 9421 | 30 | 105 | 9286 | 107900 | 93400 | 14500 | 65 |
where $P_{10} = 771$ psia = gas pressure at 10° elevation (par. 99)

$V_{10} = 891$ in$^3$ = gas volume at 10° elevation

$n = 1.4$, assumed.

Gage gas pressure then is:

$$P_g = P_{ag} - 15, \text{ psig}$$

The force of the gas on the equilibrator rod:

$$F_g = AP_e = 19.635 \times P_e, \text{ lb.}$$

Summing the packing frictional forces:

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

Net rod force then is:

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

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.

$$P_{ag} = \frac{P_{65} V_{65}^{1.4}}{V_{65}^{1.4}}, \text{ psia},$$

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

$V_{65} = 1202$ in$^3$ = gas volume at 65° elevation

Gage gas pressure then is:

$$P_g = P_{ag} - 15, \text{ psig}.$$

Gas force on the equilibrator rod:

$$F_g = AP_e = 19.635 \times P_e, \text{ lb.}$$

Total packing friction:

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

Net rod force:

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

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

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49
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 $P_{es}$, which is 572 psig in the isothermal condition prior to depression. The following table lists the calculations. (See figure 45.)

| $\theta$ | d/L | $r$ | $V_\theta$ | $\left(\frac{V_c}{V_\theta}\right)^{1.4}$ | $P_\theta$ | $F_r$ | $f_r$ | $F_s$ | $f_s$ | $F_a$ | $M_r$ | $M_s$ | $M_t$ | $T_c$ | $\theta$ |
|----------|-----|-----|------------|---------------------------------|----------|------|------|------|------|------|------|------|------|------|------|------|
| 65       | .403| 10.06| 1292      | 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| 8900 | 50   |
| 45       | .224| 16.16| 1110      | 1.118                          | 624      | 12252| —287 | 12559| —14  | —76  | 12629| 194900| 204100| 9200 | 45   |
| 40       | .179| 17.12| 1081      | 1.160                          | 649      | 12748| —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| 235500| 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   |
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:

- $a = \beta$
- $\psi = 90^\circ$, when $\phi = 0^\circ$
- $\phi' = 5^\circ$
- $c = 12.15$ in.
- $R = 36.33$ in.
- $R_t = 18$ in.
- $W_t = 2000$ lb.

From Equation (5a),

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

From Equation (14a),

$$K_s = \frac{M_{w_0}}{cR} = \frac{36000}{441} \approx 81.5 \text{ lb/in, spring rate.}$$

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

$$\theta = -5^\circ,$$

and

$$\phi = \theta + \phi' = 0^\circ.$$

From Equation (2),

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

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

$$L = 38.3 \text{ in., the maximum spring deflection.}$$

From Equation (8b),

$$F_m = K_s L$$

$$= 3120 \text{ lb, the maximum spring force.}$$

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
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

\[ \alpha = \beta, \]

\[ \psi = 115^\circ, \text{ when } \phi = 0, \]

and

\[ \phi' = 3^\circ 28'. \]

Geometry

From Equation (2),

\[ L_\phi^2 = c^2 + R^2 - 2cR \cos \psi \]

\[ = 1467.5 - 882.8 \cos \psi. \]

From Equation (3),

\[ r = cR \sin \psi/L_\phi \]

\[ = 441.3 \sin \psi/L_\phi \]

\[ \psi = \psi_0 - \theta = 115^\circ - \theta. \]

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:

\[ W_t = 2,000 \text{ lb} \]

\[ R_t = 18.0 \text{ in.} \]

\[ \phi' = 3^\circ 28'. \]

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:

\[ F_{t1} = \frac{M_{w0}}{r_0} = \frac{35900}{9.32} = 3852 \text{ lb}, \]

and

\[ F_{t2} = \frac{M_{w55}}{r_{55}} = \frac{18800}{11.93} = 1576 \text{ lb}. \]

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

\[ F_{e1} = \frac{F_{t1}}{1.4} = 2752 \text{ lb}, \]
The corresponding spring deflection is

\[ \Delta L = L_0 - L_{25} = 10.87 \text{ in.} \]

which yields the spring rate:

\[ K_{t_0} = \frac{F_{s_1} - F_{s_2}}{\Delta L} = 149.5 \text{ lb/in.} \]

For convenience \( L_0 \) is selected to be the equilibrator length when \( \phi \) equals zero rather than when \( \phi \) 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 \( 0^\circ \) elevation is

\[ \sigma_a = 120,000 \text{ psi.} \]

Then the wire diameter is given by:

\[
d^3 = \frac{2.55 F_a D}{\sigma_a} = \frac{2.55 \times 2752 \times 5.25}{120000} = 0.303, \]  

(bibliography reference 7)

\[ d = 0.671 \text{ in.;} \]

and the number of coils by:

\[
N = \frac{S G d^4}{8 F_{s_1} D^3} = \frac{18.4 \times 11.5 \times 10^6 \times 0.2027}{8 \times 2752 \times 144.7} = 13.5, \]  

(bibliography reference 7)

where \( S = \text{deflection of the assembled spring} \)

\[ = F_{s_1}/K_s = 2752/149.5 = 18.4 \text{ in.} \]

\( G = \text{torsion modulus of rigidity} \)

\[ = 11.5 \times 10^6 \text{ psi.} \]

(bibliography reference 7)

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

Maximum deflection:

\[ S_m = S + (L - L_{25}) = 18.40 + 0.79 = 19.19 \text{ in.} \]

Maximum load (at \( -5^\circ \) elevation):

\[ F_m = K_{t_0} S_m = 149.5 \times 19.19 = 2869 \text{ lb.} \]

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 \( 0^\circ \) and \( 55^\circ \) elevation are, respectively:

\[ F_{i_1} = 0.4F_{s_1} = 1100 \text{ lb}, \]

\[ F_{i_2} = 0.4F_{s_2} = 450 \text{ lb,} \]

and the spring rate,

\[ K_{i_0} = \frac{F_{i_1} - F_{i_2}}{\Delta L} = \frac{650}{10.87} = 59.8 \text{ lb/in.} \]

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

\[
d^3 = \frac{2.55 F_i D}{\sigma_a} = \frac{2.55 \times 1100 \times 3.25}{120000} = 0.076, \]

\[ d = 0.424 \text{ in.} \]

The number of coils required:

\[
N = \frac{S G d^4}{8 F_{i_1} D^3} = \frac{18.4 \times 11.5 \times 10^6 \times 0.0323}{8 \times 1100 \times 42.875} = 18.1, \]

53
where

\[ S = \frac{F_t}{K_t} = \frac{1100}{59.8} = 18.4 \text{ in.} \]

= deflection of the assembled spring.

**Maximum deflection:**

\[ S_m = S + (L_0 - L_0) = 18.4 + .79 = 19.19 \text{ in.} \]

**Maximum load (at -5° elevation):**

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

**Maximum stress**

\[ \tau = \frac{2.55 F_n D}{d^3} = 135000 \text{ psi.} \]

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:

\[ K_t = 149.5 + 59.8 = 209.3 \text{ lb-in.} \]

**Spring force on the rod:**

\[ F_s = K_t (S - L) \text{ lb,} \]

where \( S = 18.4 \) = spring deflection at 0° elevation

\[ \Delta L = L_0 - L_0 = \text{equilibrator stroke}. \]

The equilibrator moment, when friction is neglected, is

\[ M_e = F_r r \text{ lb-in.} \]

and the unbalanced moment:

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

The spring performance calculations are as follows:

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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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      MIL-HDBK-700 (MR) Corrosion and Corrosion Protection of Metals Glass
      MIL-HDBK-722 Steel and Wrought Iron Products