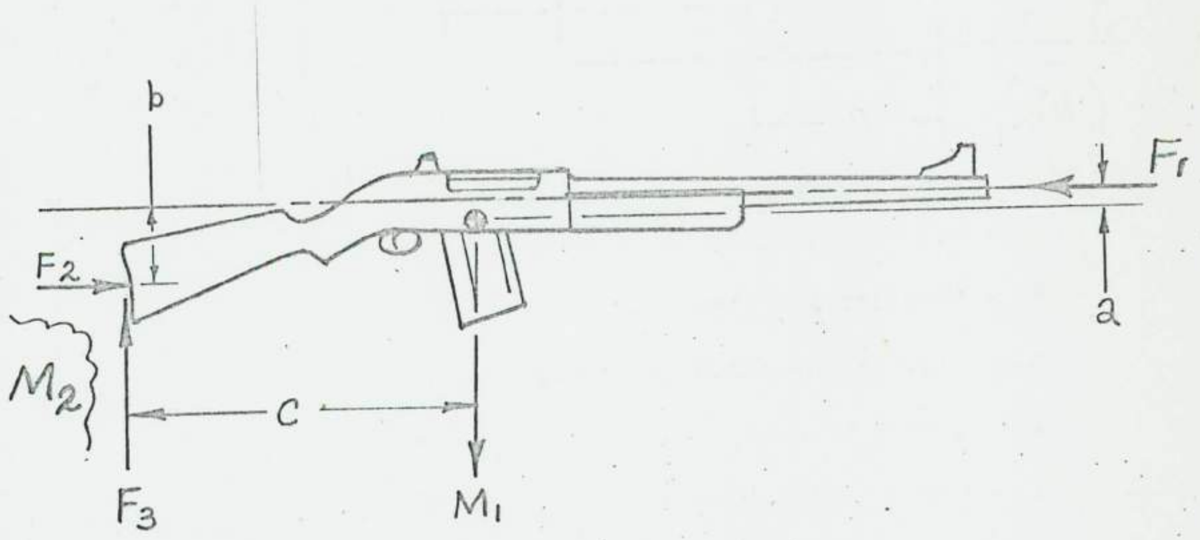
DYNAMICS of AUTOMATIC RIFLES (Typical)



◉ = Center of Gravity

RESULTANT WORKING FORMULA:

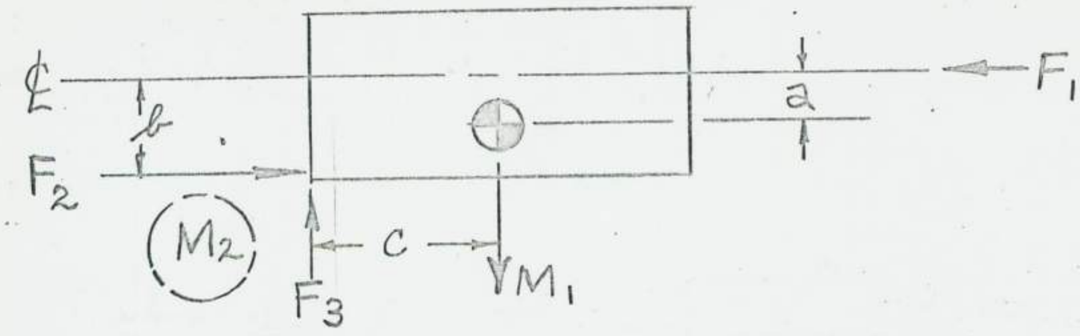
$$\omega = M \left[\frac{M_1 a + M_2 b}{(I + M_1 c^2)(M_1 + M_2) + M_1 M_2 (b-a)^2} \right]$$

ω = angular velocity (radians per second)

I = moment of inertia (ft.-lb.-sec.^{2})

M = Momentum (lb.-sec.)

Consider the following force diagram:



F_1 - Propelling Force

F_2 - Shoulder Reaction (horizontal)

F_3 - Shoulder Reaction (vertical)

a - Bore ℄ to c.g.

b - Bore ℄ to shoulder

c - c.g. to shoulder

$\bar{a}$ - angular acceleration of weapon

w - angular velocity of weapon

I - moment of inertia about c.g.

M_1 - Weapon mass

M_2 - Mass of shooter (equivalent)

M - Momentum of bullet and charge

Dynamics of Automatic Rifles

The moments and reactions about the c.g. are as follows:

$$(1) \quad I\,\bar{a} = F_1\, a + F_2\,(b-a) - F_3\, c$$

The value of M_2 varies greatly and cannot be calculated, but it has been observed that a value of 1/4 the shooter's weight is reasonable. This is an empirical value for the shooter's supporting weight plus the force applied by the shooter to resist recoil. In any event, since it is many times the rifle weight, variations have little practical effect.

The horizontal acceleration of the gun butt is: ($F = m\ a$)

$$(2)\quad \frac{F_2}{M_2} = \frac{F_2 - F_1}{M_1} + \quad (b-a)\ \bar{a}$$

so that

$$(3)\quad F_2 = \frac{M_2\ F_1 - M_1\ M_2\ (b-a)\ \bar{a}}{M_1 + M_2}$$

The vertical acceleration moment is:

$$(4)\quad \frac{F_3}{M_1} - C\ \bar{a} = 0$$

so that

$$(5)\quad F_3 = M_1\ C\ \bar{a}$$

Substituting equations (3) and (5) into (1)

$$I\bar{a} = F_1\ a + \left[\frac{M_2\ F_1 - M_1\ M_2\ (b-a)\ \bar{a}}{M_1 + M_2}\right](b-a) - M_1\ c^2 a$$

$$\bar{a} = F_1 \left[\frac{M_1\ a - M_2\ b}{(I + M_1\ C^2)(M_1 + M_2) + M_1\ M_2\ (b-a)^2}\right] = F_1 \left[\emptyset\right]$$

Since angular velocity is a function of acceleration as M is a function of F_1,

$$(\int \bar{a}\ dt = w \text{ and } \int F_1\ dt = M)$$

therefore $w = M \quad [\emptyset]$

(w in radians/sec)

By inspecting this formula, you see that if a and b are near zero, (an in-line condition), w vanishes. Also note that due to the M2/M1 ratio the "w" is more sensitive to the "drop" of the stock than to the positive of the c.g.

However, rifles with drop stock configurations are designed for facility in sighting, maintaining a low profile. A compromise between profile and automatic burst control is therefore necessary.

Most weapons are symmetrical with respect to the vertical plane, otherwise they would rotate respectively e.g. T148 semi-automatic grenade launcher. This weapon had a horizontal magazine, which caused the center of impact of rounds to shift laterally as the launcher fired successive rounds.

Applying the above formula to a typical rifle, in which "a" = 1 inch, "b" = 4 inches, "c" = 20.5 in., "M_1" = 9.5 lb., "I" = 1000 ft.lb.-sec.^{2}, "M_b" = 150 gr., "M_p" = 50 gr., "V" = 2700 fps. Firing at a full automatic rate of 750 spm, what angular deviation can be expected between rounds 1 and 2 by gunners ranging in weight from 140 to 240 lb.?

For the 140 lb. man, w = .534 rad/sec., or 534 mils/sec.

For the 240 lb. man, w = .572 rad/sec., or 572 mils/sec.

Applying the cyclic rate of 750 rpm; results in the following deviations:

For the 140 lb. man: 42.7 mils.

For the 240 lb. man: 45.7 mils.

Thus it is obvious that in full automatic fire, accuracy cannot be expected; which is normal. Also, note the small variation due to body weight; also, that the heavier body causes greater dispersion. To understand this, consider the extremes in weight; a body that approached zero weight would offer no resistance to recoil, and the only deviation would be the force about the moment arm to the c.g. At the other extreme, a rigid body, or block would cause a greater turning moment, due to the fact that the buttstock would not be free to drift rearward. (as it does against a shoulder)

Therefore, in summary, the point of contact of the buttstock with the shoulder should be in-line with, or slightly above, the bore axis. The cyclic rate should be high. Short bursts are recommended.

A rubber butt-pad is useful in reducing recoil load effects for two reasons:

(1) The rubber pad distributes the bearing load over a wider area of the shoulder,

(2) The shooter intuitively presses the buttplate harder against his shoulder, which is a favorable attitude for this purpose.

OUTLINE STUDY OF SMALL-ARMS AUTOMATIC WEAPONS

I Introduction

Scope and Purpose of Outline

This outline is a summary of all of the factors that must be considered in designing a new weapon. There may be additional factors depending upon the specific requirements of each new weapon system, but this outline represents a minimum list of factors. It may be utilized as a checklist of weapon design parameters and is provided as a guide to insure that no important weapon element is overlooked when a new concept feasibility study is conducted.

II Basic Weapon Operational Cycles (Blowback, Recoil, and Gas)

A. Simple Blowback

1. Definition of simple blowback cycle

a. Cycle of operation

(1) Kinematic and dynamic analysis of cycle

(2) Construction of calculated T-D curve.

b. Ammunition characteristics required for simple blowback cycle

(1) Case and chamber design

(a) Headspace

(b) Breeching space

(2) Case Material

(3) Lubrication

2. Strength requirements of breech mechanism

3. Advantages and disadvantages of simple blowback systems

B. Modification of Simple Blowback Cycle - Delayed Blowback

1. Definition of delayed blowback

a. Cycle of operation

(1) Kinematic and dynamic analysis of cycle

(2) Construction of calculated T-D curve

II 1. b. Ammunition characteristics required for delayed blowback cycle

(1) P-T curve at chamber

(2) Case and chamber design

(a) Headspace

(b) Breeching space

(3) Case Material

(4) Lubrication

2. Strength requirements of breech mechanism

3. Advantages and disadvantages

C. Modification of Simple Blowback Cycle - Retarded Blowback

1. Definition of retarded blowback

a. Cycle of operation

(1) Kinematic and dynamic analysis of cycle

(2) Construction of calculated T-D curve

b. Ammunition characteristics required for retarded blowback cycle

(1) P-T curve at chamber

(2) Case and chamber design

(a) Headspace

(b) Breeching space

(3) Case Material

(4) Lubrication

D. Modification of Simple Blowback Cycle - Advanced Ignition

1. Definition of blowback with Advanced Ignition

a. Cycle of operation

(1) Construction of calculated T-D curve

(2) Kinematic and dynamic analysis of cycle

II . b. Ammunition characteristics required for advanced ignition blowback cycle

(1) P-T curve at chamber

(2) Case and chamber design

(a) Headspace

(b) Breeching space

(3) Case Material

(4) Lubrication

E. Gas Operation

1. Definition of Gas Operation Cycle

a. Cycle of operation

(1) Kinematic and dynamic analysis of cycle

(2) Construction of calculated T-D curve

(3) Gas systems

(a) Open - impingement

(b) Closed - cut off and expansion

(c) Pressure - time curves of chamber and bore at gas port.

b. Ammunition characteristics required for gas operation cycle

(1) Case and chamber design

(a) Headspace

(b) Breeching space

(c) Pressure data

(2) Case material

(3) Lubrication

2. Strength requirements of breech mechanism

3. Advantages and disadvantages

II F. Recoil Operation

1. Definition of recoil operation

    a. Cycle of operation

        (1) Kinematic and dynamic analysis of cycle

        (2) Construction of calculated T-D curves

        (3) Recoil systems

            (a) Short recoil system

                (1) "Browning" cycle

                (2) German cycle

                (3) Linked action cycle

            (b) Long recoil systems

        (4) Accelerators

            (a) Definition

            (b) Variable lever

            (c) Integral cam

    b. Ammunition characteristics required for recoil operation

        Short recoil   Long recoil

        (1) P-T curves of interior ballistics system

        (2) Case and chamber design

            (a) Headspace

            (b) Breeching space

        (3) Case material

2. Strength requirements of breech mechanism

3. Advantages and disadvantages

G. Externally Powered Systems

II G. 1. Revolver and multichamber systems

a. Cycle of operation

b. Kinematic and dynamic analysis of cycles

c. Ammunition characteristics required for revolver and multichamber systems.

(1) Case and chamber design

(2) Dynamic seals

2. Multi-barrel systems

a. Cycle of operation

b. Kinematic and dynamic analysis of cycles

c. Operational cam design

3. Drive systems

a. Hydraulic

b. Electric

c. Mechanical

III Weapon Components

A. Breech Locking Systems

1. Integral locks

a. Rotary locks

b. Tilting locks

c. Sliding locks

2. Separate locks

a. Rotary

b. Tilting

c. Sliding

3. Strength requirements of locking systems

III B. Feed Systems

1. Magazines

   a. Box type

      (1) Single column

      (2) Double column

   b. Rotary types

      (1) Fixed

      (2) Removable

   c. Magazine springs

   d. Kinematic analysis of feeding operation

2. Strip and clip feed systems

   Kinematic analysis of feeding from strips and clips

3. Belt Feed System

   a. Pull out or "U" type

   b. Push through

   c. Fundamentals of link design

   d. Kinematic analysis of belt feeding systems

C. Extraction and Ejection Systems

1. Kinematic and dynamic analysis of extractor and ejection

2. Functional requirements of extractors and ejectors

D. Fire Control Mechanisms

1. Sear systems

   a. Automatic fire systems

   b. Semi-automatic fire systems

   c. Trigger pull requirements

III D. 2. Ignition systems

a. Percussion firing systems

(1) Single striker

(2) Multiple strikers

(3) Firing pin design

(a) Tip configuration

(b) Tip support at time of firing

(c) Types

Fixed - Movable - Inertia

b. Electric firing systems

(1) Electrical energy

(2) Typical circuitry

Insulation problems

(3) Firing pin design

Tip configuration

E. Barrel Design for Small Arms Automatic Weapons

1. Chamber design

Headspacing

(1) Rimmed and belted cartridges

(2) Rimless cartridge

(a) Shouldered

(b) Straight

2. Rifling

Types and twists

3. Thermal problems of barrels at high firing rates

F. Muzzle Attachments

III F. 1. Muzzle brakes

Design

2. Muzzle boosters

a. Design

b. Special considerations

3. Flash hiders and flash suppressors

G. Chargers

1. Manually operated

2. Externally powered

a. Air

b. Hydraulic

c. Electric

d. Cartridge actuated

H. Mounts - Mounting conditions with respect to weapon functioning

1. Vibrational characteristics of automatic weapons

2. Mounting conditions

a. Shoulder fired or hand held

b. Bipod and tripod mounts

c. Vehicular mounts

(1) Armor vehicles

(2) Aircraft

(a) Fixed wing

(b) Helicopter

3. Recoil adapter design

Buffer design

IV Areas of Special Consideration for Small Arms Automatic Weapons

A. Open and Closed Bolt Cycles

B. Weapon Safety Requirements

C. Economics of Fabrication

1. Sheet metal construction

2. Die castings; sintered metals

3. Special machining techniques

a. Milling

b. Broaching

c. Turning

D. Service Life Requirements

V Tactical and Logistical Requirements Affecting Small Arms Automatic Weapon Design

A. Human Engineering Factors

1. Handling characteristics

2. Operating characteristics

B. Environmental Conditions

1. Temperature extremes

2. Mud and dust problems

3. Lubrication problems

IV TYPES OF WEAPON SYSTEMS OF OPERATION

(AUTOMATIC WEAPONS)

Classes of weapon systems may be outlined, in general, as follows:

I Blowback

A. Pure blowback (Cal. .45 M3) (Cal. .22 LB)

B. Delayed blowback (Cal. .45 Reising SMG)

C. Retarded blowback (Cal. .45 Thompson SMG) (earlier versions)

D. Advanced primer ignition (Oerlikon)

II Recoil

A. Long recoil (misc. shotguns)

B. Short Recoil

1. Toggle (maxim)

2. Propped lock (Browning)

3. Tilting barrel (Cal. .45 M1911)

4. Cammed cross-bolt (M73)

5. Hinged bolt (M85)

6. Linkage (proportioned) (R. Robinson)

C. Misc. revolver types

III Gas Operation

A. Piston (rearward)

1. Impingement (M1)

2. Cut-off & expansion (M14)

B. Bleed-off (M16) (French tube)

C. Muzzle blast (Bang)

D. Lever (Colt Browning)

III E. Primer actuated (AA1)

F. Piston (forward) (misc. European)

IV Misc. self-powered

A. Floating chamber (win. shotgun)

B. Set-back (impulse) (garand exp.)

C. Alternating barrel (Hughes)

V External power

A. Electric drive (Vulcan, M75)

B. Hydraulic or pneumatic (Vigilante)

C. Gear drive off vehicle (ZB-80)

Factors that determine what type of operating system a new weapon will assume are outlined following this paragraph. Note that the military requirements are the first item, and the most important.

Ammunition characteristics are physically responsible for limiting certain types of operation, due to the nature of pressure development and available power.

SELECTION OF SYSTEM OF OPERATION

1. Military requirements

2. Weapon environment

   a. Mount flexibility or rigidity

   b. Noxious gases and/or muzzle blast

   c. Permissible trunnion load

   d. Space requirements for

      (1) Installation,

      (2) Maintenance, and

      (3) Ammunition feed, spent case and link disposal

3. Ammunition

   a. Cartridge case design

   b. Internal ballistics

   c. Ignition system (percussion/electric)

4. Effectiveness

   a. Rate of fire

   b. Belt pull capacity

   c. Weight, size, and shape

   d. Reliability

   e. Maintainability

5. Cost

<u>Blowback Operation</u>

The most important factor in the blow-back operated weapon is the behavior of the cartridge case, since it is in motion at the instant of firing. Blowback is usually reserved for low power cartridges, and for military weapons, is most usually found in sub-machine guns firing the cal. .45 and 9mm pistol cartridges.

Cartridges for blowback weapons are essentially cylindrical; with no neck or shoulder, and very little body taper, if any. The base is sufficiently thick to support the chamber pressure for the time and travel that the cartridge case moves during blowback. It is this initial movement that limits the blowback principle to low-powered weapons with a cartridge design that can move rearward in the chamber without danger of case stretch.

Calculations for this system are quite straight-forward. The force acting rearward on the bolt equals the chamber pressure multiplied by the cross-sectional area of the mouth of the case. The bolt impulse is equal to the projectile and gas impulses.

The cartridge case is designed to resist seizure by the chamber wall at the onset of blowback motion. At this point lubrication will make a difference in function, reflecting higher rates of fire. However, the U.S. systems use unlubricated cases, while some European systems do lubricate their ammunition. The force expended in overcoming friction between the cartridge case and the chamber may reduce the energy transferred to the bolt sufficiently to cause a short recoil, or failure to feed.

As an exercise in determing recoil velocities, the following formula is usually given:

$M_r V_r = M_p V_p + M_c$ 4700, where MV is impulse

This impulse is given in lb. sec.

The peak velocity to which the bolt will be accelerated at the instant of bullet exit is easily found. However, a correction factor of approximately .75 should be applied to compensate for inerti effects. This is, the bolt energy going into battery must be absorbed by the blowback energy before bolt recoil begins.

The impulse may also be determined by measuring the area under the pressure time curve, and multiplying by the bore area.

After determining bolt impulse, either the bolt weight or velocity are calculated next, depending on what system requirements are known, such as: rate of fire, geometric limitations of the bolt, (hence weight), and allowable recoil travel.

The calculation of impulse is useful in determining free recoil velocities of, say, a rifle or other complete weapon at time of firing since the weapon mass will be in a status of recoil, just as the bolt mass of the blowback system.

For example, since I=mV and F = ma, then F = I/t = 2.3/.002 = 1150 lb. for an average force of a 7.62mm NATO firing weapon, rigidly mounted. Of course, weapon recoiling travel and time reduces this to loads that are easily managed. For example, an average reaction of 18 msec. reduces this load to 1/9, or a little over 100#. Thus acceleration is very important in the amount of "punishment" or effective load felt by the shooter.

Advanced Primer Ignition

In this system, the bolt is moving forward at a significant velocity at the start of ignition. Therefore, considerable rearward impulse is absorbed in decelerating the bolt to a stop prior to recoil. This principle is employed in the Oerlikon family of weapons, the XM140 firing cycle, and in heavier gun tubes. The maximum efficiency of this system occurs when the closing velocity equals the opening velocity, so that the recoil velocity is then one half of a comparable plain blowback system. Since energy is a function of V^2, then reducing the velocity by 1/2 reduces the energy to 1/4; thus reducing trunnion loads by 75%.

The weapon will be lighter, firing rate can be increased, although special features must be incorporated into the cartridge case design for this type of weapon cycle of operation.

Delayed Blowback System

These systems, of which there are many variations, employ a lock to the breechblock for only a portion of the peak pressure time. After unlocking, the remaining breech pressure acts on the bolt in "blowback" fashion. Again, special ammunition design features are required, such as a heavy case head, headspace control during opening to prevent case splitting, fluting chambers, etc.

Bolt velocity is a result of several vectors. That is, the initial energy absorbed by the unlocking linkage and bolt carrier, if any. In addition, effect of residual bore pressure acts to implement the bolt recoil. This system results in higher rates of fire, particularly if the bolt weight is light.

Retarded Blowback Systems

In this system, the bolt is not positively locked, but must act against a mechanical disadvantage in opening. The mechanism employed utilizes essentially lightweight components, with a high inertia to be overcome. This is analogous to a wheel and crank slightly off a dead-center position.

A toggle joint mechanism embodies this principle. It is important that the retarding mechanism should act against a mechanism constantly decreasing mechanical disadvantage throughout the bolt stroke. In fact, the retarding element should act so that its delaying force reflects the shape of the pressure time curve; high at the start and low through most of its action. The proper location of toggle pivot joints effectively act to produce this motion. The resultant mechanism is relatively smooth in action, but must be limited to relatively low to medium powered classes of ammunition.

Since the mass effect of rotating linkages vary, the bolt motion cannot be determined in simple terms of impulse and momentum.

Set-Back (Impulse)

This is a combination blowback and recoil operation, in which the bolt is free to blow back against a locking abutment for a short stroke, in the order of .060-.090 inches. A heavier bolt carrier continues to recoil rearward through a dwell travel before it unlocks the bolt and reciprocates it. This system is limited by cartridge case design considerations.

Recoil Operation

There are many variations of this principle of operation, in which the bolt, barrel, and lock recoil together until the breech can be safely unlocked. The energy generated during this motion is then distributed throughout the cycle of operations, and the manner in which this distribution is made is a measure of the efficiency of the weapon.

The Long Recoil system will not be discussed at length because it is of limited military interest for small arms, due to practical rate of fire restrictions.

The Short Recoil mode is popular because of its universal effectiveness. In this system, the reciprocating bolt receives energy from three vectors: First, the momentum from its basic recoil motion; Second, an acceleration from the barrel, due to an accelerator mechanism that in turn helps buff the barrel stroke, and Third, an added impulse from the blowback action of residual pressure at the time of bolt unlocking. In addition, some weapons use a muzzle booster, in which the muzzle gases are trapped within an exterior fixed housing and impinge upon the barrel as though the barrel muzzle end were a piston, accelerating it further.

Some weapons, notably the Browning Machine Gun Series, fire as the bolt, barrel, and lock are ending their counter-recoil (forward) motion. The initial recoil thrust is checked, and the need for counter-recoil buffers to absorb and dissipate the forward kinetic energy of the counter-recoiling parts is eliminated. This principle is quite similar to that employed in the advanced primer ignition type of Blowback operation discussed earlier.

In the analysis of a short recoil system, a review of system forces is in order.

Consider the following schematic:

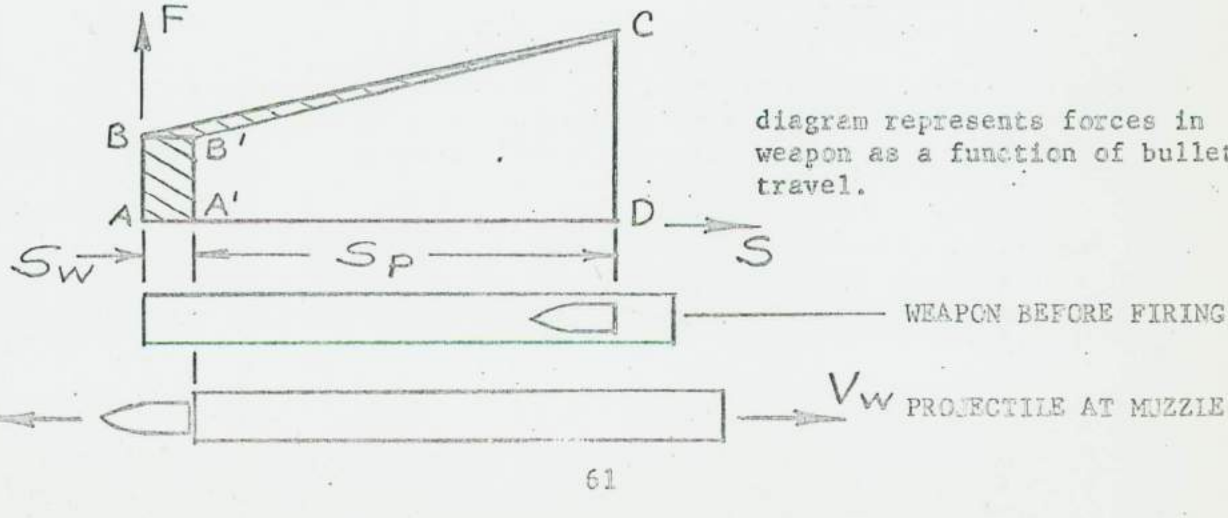


diagram represents forces in weapon as a function of bullet travel.

For the initial few milliseconds, while the masses are accelerated to their operating velocities, the following relationships are true:

Since $F = ma$

acceleration of the projectile is

$a_p = F/W_p\,g$ (W_p = projectile wgt. + 1/2 charge wgt)

while the acceleration of the weapon in the opposite direction is:

$a_w = F/W_w g$

The accelerations of the projectile and the weapon are inversely proportional to their weights:

$a_p/a_w = W_w/W_p$

The same is true of the velocities and travels of projectile and weapon:

$V_p/V_w = S_p/S_w = W_w/W_p$

Now the impulse of the projectile is at all times equal to the impulse of the weapon.

$I = mV$

$I_p = W_p/g\ V_p$

$I_w = W_w/g\ V_w$

therefore: $\dfrac{I_p}{I_w} = \dfrac{W_p \times V_p}{W_w \times V_w}$ Since $\dfrac{V_p}{V_w} = \dfrac{W_w}{W_p}$

then $\dfrac{I_p}{I_w} = \dfrac{W_p}{W_w} \times \dfrac{W_w}{W_p} = 1.$

The kinetic energy of the projectile at the muzzle corresponds to the area A'B'CD of the schematic diagram since the ordinate represents force and the abscissa is travel, and the product of force and travel represents energy.

The energy of recoil of the weapon Ew is indicated by the small area ABCB'A'. Note that the same force acts on the weapon as on the projectile but the travel (thus velocity) of the weapon is much smaller.

The energies of the projectile and weapon are in the same ratio as their travels, which is also the inverse ratio of their weights.

Therefore:

$$\sqrt{\frac{Ep}{Ew}} = \frac{Sp}{Sw} = \frac{Ww}{Wp}$$

In summation, on firing, the muzzle momentum of the projectile is equal to the recoil momentum (or impulse) of the weapon, but the kinetic energies of weapon and projectile are not equal.

The sum of energy of weapon and projectile, is equal to the whole energy available. The projectile energy is expended during flight along the trajectory and at the target, while the weapon energy goes to work performing the automatic cycle of operations of the weapon.

For a short-recoil type of weapon, a general study of energy requirements is as follows:

MBBL = Mass of barrel and associated components in motion

MBO = Mass of bolt assembly

I = impulse of projectile at muzzle

Since $I = MV$ and $E = 1/2MV^2$ then $V^2 = I^2/M^2$ and $E = I^2/2M$

$E\ (BBL + BO) = I^2/2\ (MBBL + MBO)$

Using a mass ratio of $r = MBO/MBBL$

$E\ (BBL + BO) = I^2/2\ MBBL\ (1 + r)$

The energy of the Bolt above is:

$EBO = I^2/2MBBL \cdot r/(1 + r)^2$

The energy of the Barrel above is:

$E\ BBL = I^2/2MBBL \cdot 1/(1 + r)^2$

After the pressure in the barrel has decreased to a safe value (dependent upon cartridge case design) then separation of Bolt and Barrel can start.

Since the barrel assembly mass is quite heavier than the bolt, the bolt requires additional energy to perform its assigned tasks.

Here an accelerator is used which transfers much of the remaining barrel energy to the bolt. The percentage of barrel energy transferred to the bolt, in this case, "n" is assumed to be about 70%. This acceleration also serves to buff the barrel. (absorb excess barrel energy)

The maximum energy which the bolt can have is then

$$\text{Max } E\ BO = I^2/2M\ BAR \cdot \frac{r + n}{(1 + r)^2}$$

This energy must enable the bolt:

1. To perform work such as feeding, cocking etc. = Ew (recoil)

2. Overcome friction losses = E f (recoil)

3. Store energy in the drive spring for counter-recoil = Es

Also, the bolt will have a certain amount of energy which is partially absorbed by the buffer = E B

Thus

$$\text{Max } E_{BO} = E_w + E_f + E_s + E_B$$

For the counter-recoil stroke a similar equation is developed, so that the bolt energy at the end of C' recoil is:

$$E_{BO}\,(C'R) = Es + \Delta E_B - Ef\,(C'R) - Ew\,(C'R).$$

Ew (C'R) is the work done in C' recoil such as stripping the cartridge from the link, chambering, etc.

ΔE_B is the energy given back by the buffer, (coefficient of restitution) and may vary widely dependent upon the intention of the designer, and in this instance is assumed to be 50 to 60%.

However, E_B may be neglected so that the minimum energy requirement can be derived. That is, the bolt should still function without assistance of energy from the buffer.

$$\text{Max } E_{BO}\,(\text{OFF Accelerator}) - E_{BO}\,(C'R) = E$$

Where E is all the energy used to perform the cycle of operations. Keeping this simple formula in mind, a relationship between starting bolt energy and remaining bolt energy at the end of the cycle will be developed.

Recoil time is determined by the initial velocity, which is a function of the square root of the initial energy, and the time for counter-recoil a function of the remaining counter-recoil velocity, likewise a function of the square root of the end energy on counter-recoil.

The time ratio for recoil and counter-recoil $TR/TC'R = \beta$ where β is about 75 to 80%.

Therefore, since

$$TR/TC'R = \frac{\sqrt{E\ C'R}}{\sqrt{\max E\ BO}}$$

Then $E\ C'R = \beta^2\ (\max E_{BO})$

Now since $\text{Max } E_{BO} = E + E\ C'R$,

(That is, all energy used on recoil and c'recoil plus energy remaining at end of c'recoil.)

Putting the above two equations together,

$$\text{Max } E_{BO}\ (1 - B^2) = E$$

$$\text{Max } E_{BO} = E/\ (1 - B^2)$$

Using $B = .80$, then

$$\text{Max } E_{BO} = E/(1 - .64) = 2.8E$$

In summary, this means that the initial energy of the bolt, which is powered mainly by the barrel weight, must be about three times greater than the energy which is used for the weapon function.

The details of a mathematical analysis of a short recoil type of weapon design depends upon the mechanism design selected or invented for performing the sequence of operational functions. Therefore it is not feasible to set up an analytical method which will apply universally to all short recoil weapons.

Of course, the cartridge must be defined as to velocity, pressure-time data, and case strength.

Vallier's formula for approximating the duration of the residual pressure is

$$T_{RES} = \frac{Mc}{AP}(9400 - Vp)$$

P = Muzzle pressure
Vp = Proj. vel.
Mc = Charge Mass.
A = Bore area

The residual pressure time span is required in order to determine forces that act on the bolt at the moment of unlocking.

The action on the accelerator should not be too abrupt, as this will create shock loads, abrasion and wear of parts. At least 4 msec. should be given as an action time for accelerator function. Too long a time will not adequately accelerate the bolt and the firing rate will be reduced. The formula for Kinetic Energy is used to determine velocity increase in the bolt.

The change in velocity will enable forces on the accelerator to be computed by using the formula $F = m\,a$

It must be remembered that the bolt velocity must be modified to compensate for inertia effects, drag, etc. This may vary from 70 to 85% of the indicated velocity.

In counter-recoil motion, two separate approaches to the control of the barrel motion may be used, depending upon the locking system selected. In one, the barrel remains locked back until the bolt approaches it while chambering a new cartridge. Then, as the bolt is locked, it moves forward with the barrel. The Browning machine gun family is of this type.

The other approach does not latch the barrel, but allows it to return forward immediately. Here, the barrel must be damped out in its motions before the bolt is ready to lock and fire.

The K.E. of components coming into battery is usually so great as to require that the weapon fire, while the components are still biased with forward momentum. Otherwise a substantial buffer mechanism will be required. Precision in the timing of the firing mechanism strike will be required.

Gas System of Operation

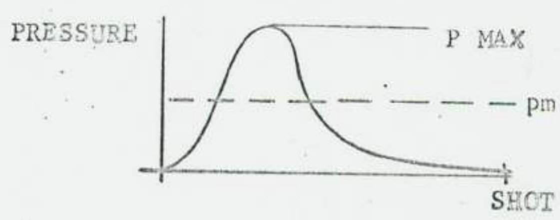
This is usually descriptive of weapons in which a transverse hole in the barrel, a gas port, is used to draw off gas, in order to operate the unlocking and other functional components.

In considering the utilization of gases vs. the effect upon muzzle velocity, there is only an insignificant loss of projectile velocity, averaging about .5%. The gas operated gun usually has an ample supply of available energy for actuating the weapon mechanism.

This is useful, particularly in the early development phase of a new weapon, if there is a lack of operating power, the gas port is merely opened up several thousandths of an inch. However, the most abused method of determining gas port size required, in the initial test weapon is to begin with an unusually small hole, approximately .030" in dia. and progressively open the hole, while measuring weapon function by a time-displacement curve of the operating rod or carrier. The many variables, from one weapon to another, of propellant characteristics (burning rate), mass ratios, moments of inertia, friction, geometry of control surfaces in sliding, render initial calculations vulnerable to errors in assumed values, so that, in the long run, the trail and error process is resorted to, within limitations.

In a gas operated weapon, the gas port should be located at least further than the "point of all burnt" of most of the propellant. If the gas port is too close to the chamber particules of unburned propellant will enter the gas cylinder and either burn, causing erratic power development, or gather in such a manner as to cause a secondary explosion during some subsequent firing.

Determination of gas pressure along the bore:



pm = mean gas pressure, where the area under pm = the area under the curve. Relating the pressure curve to energy of projectile:

$$\text{ENERGY} = pm \times A \times Se \simeq \frac{Wp + .5\ Wc}{2g} \times V^2$$

The ratio of pm to p max is called the pressure ratio "N"

$$N = Pm/P\ max$$

This is a typical empirical value of a variety of weapons and propellant types. A value of .4 is typical for an efficiently used small arms propellant. The smaller "N" is, the more efficiency realized.

The length of barrel may be expressed as a function of the pressure ratio with the formula given as follows:

$$Se = \frac{Wp + .5\ Wc}{2g\ N\ Pmax\ A} \times V^2$$

Demonstrating the 7.62mm example:

$$Se = \frac{150/7000 + 23/7000 \times 2650^2}{64.4 \times .4 \times 48{,}000 \times 144^2 \times \pi/4\ .308^2/144^2}$$

$$Se = 1.88 \text{ ft.} = 22.6''$$

Characteristics of Pressure, Velocity and Time curves during Projectile Travel.

A series of empirical curves have been developed from observations of the pressure curve form as adapted to the pressure ratio and projectile travel. These are all listed in the Oerlikon Pocket Book, available in most ordnance engineering libraries.

For a sample pressure ratio of .4, the following values at the moment of maximum gas pressure are:

Short travel: $S_1 = Se \times .074$

Time: $t_1 = 2\ Se/V \times .358$

Velocity: $V_1 = V \times .383$

At the muzzle:

Gas pressure: $pe = pm \times .400$

$= p\ max \times .16$

Time: $te = 2\ Se/V \times .946$

For plotting values of shot travel S, corresponding to the ratio $\lambda = s/s_1$ we can obtain the gas pressure, velocity, and time of which typical values are:
V_1; t_1; S_1 = values at point of p max

$P = p\ max \times a$

$v = V_1 \times b$

$t = t_1 \times c$

ratio λ	a	b	c
1.0	1.0	1.0	1.0
2.0	.769	1.46	1.218
5.0	.397	2.046	1.672
10.0	.214	2.453	2.267
15.0	.144	2.665	2.794
20.0	.108	2.812	3.286

With these values, as functions of λ(travel) we can caluclate the variation of gas pressure, velocity, and time with shot travel in the bore. There are variations due to changes in projectile and/or propellant weight, temperature, loading density, ignition variables, shot start, rifling dimensions, and other factors. Gas pressure is measured either by piezo-electric gauges or by copper crusher gauges. The copper crusher method is quite old, and is based merely on the permanent deformation of a copper cylinder due to a load proportional to the gas pressure.

Major caliber guns use "loose" pressure gages that are merely put into the combustion chamber, while small arms systems have "fixed" pressure gages that are mounted on a test barrel. The crusher gage is suitable for checking manufacturing of ammunition lots for comparative measurements. It is limited in that only relative results are obtained, because the evolution of gas pressure is a dynamic process, while the gauge can be calibrated only statically.

The piezo-electric method for measuring pressures is based on the property of certain crystals of developing electric charges on certain surfaces if they are mechanically stressed. Quartz is the one type most suitable for ballistic pressure measurements, because of its great compressive strength, independence of temperature, and high natural frequency. The full curve of gas pressure in the combustion chamber can be reflected by piezo-electric equipment.

Measurements may be determined at the selected gas port location (in a test barrel) in order to determine variations, if any, between lots of ammunition. However, the pressure data after the gases throttle turbulently through the gas port and expand in the gas cylinder are only of academic interest. They should be of utmost important to the designer, but usually only the pressure at the gas port is measured.

In selecting a gas port location, power requirements are of primary concern. Gas temperature should not be too high, as this will cause gas port erosion and resultant erratic power development, and it should not be too low, with a port location far down the barrel, or the gas cylinder will foul too quickly, particularly in a cut-off and expansion gas system. Location of the gas port also effects the time delay between ignition and start of operating rod travel, in turn, affecting dwell travel before the bolt begins to unlock.

The following variables affect operating characteristics of a gas operated weapon:

1. Location of gas port

2. Piston diameter

3. Initial volume of the gas cylinder

4. Weight of primary components (operating rod assembly)

5. Weight of secondary components (bolt assembly)

6. Dwell travel between primary and secondary components.

7. Ammunition characteristics, including cartridge case design as well as interior ballistic data.

8. Finally, the gas port dia., which can function as a control element, but only within limitations.

Of the piston operated types of gas systems there are two basic types, the "impingement" system and the "cut off and expansion", or White, system. In the former, gas acts directly on the piston end of the operating rod after passing through the gas port, while in the latter, the gas cylinder incorporates a large volume that the gas must fill before the hollow piston begins its motion. After the piston has moved a short distance, the gas port in the barrel is blocked by the piston wall, so that the previously trapped volume of gas in the piston cavity does the work of driving the piston by its expansion. The first type was used on the rifle, M1, while the second is used in the rifle, M14 and machine gun M60. The latter type has the advantage or providing a limited measure of compensating for differences of pressure between different lots of ammunition. That is, a higher bore pressure would cause the piston to move back quicker, causing the gas port to close earlier, limiting the amount of gas that entered the gas cylinder.

Some military rifles have been developed that incorporate a method of manually adjusting either the gas port or some element of the gas system. This is intended to adjust the gas power for differences in environment, wear, and/or ammunition. Devices of this type should be avoided if possible, as field experience indicates that it will only be accidental if the weapon is adjusted properly. That is, after a G.I. has increased power to adjust for sand, dust, fouling, etc., he is not likely to decrease power to the proper level after cleaning his rifle. The result will be excessive velocity of moving parts, with attendant battering, wear, and possible breakages.

The components accelerated directly by the gas are referred to as the primary mass, and includes the piston, operating rod, and any other directly connected mass, depending upon the weapon design. As this primary mass moves to the rear, it unlocks the bolt and carries the bolt assembly, which is called the secondary mass. For smooth functioning, the primary mass should be considerably heavier than the secondary mass. That is, the recoiling operating rod or bolt carrier develops a given kinetic energy from the gas pressure. When it unlocks the bolt and starts carrying it rearward, kinetic energy is transferred from the operating rod to the bolt. This is reflected in a sudden drop in operating rod velocity. The nature of this velocity shift determines the degree of inertia loss of the primary mass and if this is high, function will be erratic. The primary mass divided by the secondary mass is called the MASS RATIO, and this should be at least 3. As a note, the Soviet AK-47 weapon has a mass ratio of 5.4, a highly impressive figure.

Residual pressure in the chamber should act to help the secondary mass, in the work of accelerating the bolt to its peak recoil velocity. For this reason, mere gas port adjustment does not efficiently constitute proper gun development, as the timing of the moment of bolt unlocking can improve the efficiency of the weapon cycle.

In the continuation of rearward motion, the operations of extraction, ejection, cocking, and feeding are performed. At buffer contact, 40 to 60% of the remaining energy is returned by the buffer to assist the drive spring in maintaining the required rate of fire. The closing, or counter-recoil slide velocity provides energy to strip the cartridge from the link, chamber the cartridge and lock and fire the weapon. At slide closing, the velocity should not exceed 8 fps or a condition of slide rebound may prove to be a problem.

In evaluating a gas system, consider the fact that the operating piston load of necessity is eccentric to the bore. Therefore frictional force between the operating rod, bolt, and receiver, or frame, may be high unless adequately long bearing surfaces are provided. The frictional drag is aggravated by marginal lubrication conditions due to gas fouling on the piston and cylinder after extensive firing.

Attempts have been made to incorporate annular pistons concentric to the barrel, but barrel heating limits the effectiveness of this approach in extended firing.

The forces in the recoiling slide bearing on the receiver can be solved thus a series of moment equations.

The standard three equations of equilibrium $\sum F_V = 0; \sum F_H = 0$

$\sum$ Moments $= 0$ should balance.

As a matter of detail, in the gas piston design there must be a minimum clearance of several thousandths between piston and gas cylinder to permit function with thermal expansion. This will allow a small amount of gas leakage, which can be minimized by incorporating grooves on the piston. These grooves help block gas escape by the fact that turbulant flow in the grooves acts to keep the piston concentric in the cylinder, and the turbulence blocks the flow of gas to a marked degree.

Note that with the formulae $K.E. = 1/2m\ V^2$ and $I = mV$ then $K.E. = I^2/2M$

Therefore, it is seen that K.E. is inversely proportional to the mass. As a result, the lighter the piston and operating rod the higher the energy. This is due to the effect of V^2, but the operating rod weight must be balanced against bolt weight, the required stroke with the rate of fire, and momentum with work. That is; too light an operating rod will create frictional forces due to the high velocity, that will dissipate operating rod energy quicker than a slower-moving, heavier operating rod.

The required velocities of moving parts to attain the desired rate of fire, of course, is a function of all the variables of gas port location and size, piston diameter and stroke, and mass and stroke of operating slide and bolt. The common peak velocities used are approximately 32 feet per second, but may range from 24 fps up to 50 fps, depending upon design requirements. The location of slide guide ways with respect to center of gravity, piston and bore is quite critical with respect to loads on the receiver, location of return spring and buffer. Moment arms should be minimized wherever possible.

As for the operating rod energy, after estimates have been made of the required peak velocity, this can be related directly to the piston size. Using $I = m\ V$, the required impulse is determined. Integrating under the pressure time curve between the time the bullet passes the gas port and the time that pressure is cut-off (or exhausted at the muzzle) determines available power.

Gas piston area is $\left(\text{Impulse required}/\text{Impulse available}\right)$ X 1. in. sq.

P = Pressure
A = Area
M = Mass
dt = time diff.

Also, change in velocity of the piston is a function of $\frac{PA}{M}\,dt$

Then, after dwell travel, the operating rod momentum will unlock and carry the bolt back. At this instant their combined momentum will be slightly lower than the single operating rod momentum because of the work done.

Typical Analysis of Gas Systems

A brief resume of factors affecting gas system design was given in the previous chapter of this series; however, a number of particular studies of gas systems analysis will be reviewed here, in order to provide a comparative basis, or guideline, for other gas system studies.

The studies reviewed are as follows:

(A) Energy extracted from the bore to operate a 15.mm Spotting Rifle

(B) Effect on weapon function of Ammunition Pressure Variance (7.62mm)

(C) Maximum Pressure in M14 gas cylinder

(D) Maximum Pressure in M60 gas cylinder

NOTE: These studies are based upon actual instumented values, and therefore are realistic.

15mm Spotting Rifle, XM90 Gas System Analysis

The spotting round was developed so that through proper selection of ballistic coefficient (projectile weight, caliber, form factor) and muzzle velocity the trajectories of spotter and major weapon would match at long ranges. Much of the preliminary ammunition development was conducted using single shot Mann barrel test fixtures, so the object of this study is to determine how much variation in velocity is caused by tapping some of the gas from the bore and into the gas cylinder.

Since the projectile is affected by gas taken from the bore before projectile exit, the pressure-time history of the gas cylinder up to the time of projectile exit will indicate the magnitude of the energy extracted from the interior ballistic system of the weapon.

Calculated Energy extracted:

Impulse = Ft = APt

$I = 1.59$ lb. - sec.

$E = I^2/2M = \frac{I^2 g}{2W}$

$E = \underline{5.83 \text{ ft.-lb.}}$

A = Gas piston area

$= \pi/4\ (.765)^2 = .459$ in.^{2}

P = Gas cylinder Pressure

= 2800 PSI

t = 1.2 msec.

W = 7.0 lb. (Inertia)

Piston velocity at time of bullet exit = 7.2 fps. This continues to accelerate to a maximum of 11.6 ft./sec. after bullet exit)

$$E = \frac{m V^2}{2} = \frac{W V^2}{2g} = \frac{7. \times 7.2^2}{64.4} = \underline{5.64} \text{ ft.-lb.}$$

The gas system functions at an efficiency of approximately 30%, therefore/comparing energy delivered to the inertia mechanism vs. energy extracted from the bore, we have

$$E_c = \frac{5.83}{.3} = 19.5 \text{ ft.-lb.}$$

Muzzle Energy of Projectile:

$$E_m = \frac{Wp\ Vm^2}{64.4}$$

Wp = 1240 grains

Vm = 1750 fps

$$= \frac{1240 \times 1750^2}{7000 \times 64.4}$$

$Em = \underline{8423 \text{ ft.}} - \text{lb.}$

Percentage Energy loss

$$Ls = Ec/Em \times 100 = \frac{19.5}{8423} \times 100 = .232\%$$

This is verified by data taken from six different lots of ammunition, in which five lots showed no significant difference in velocity loss, measuring with and without gas system. In fact, the velocity variation for the test barrel without the gas system was greater than the variation going from "no-gas-system" to "gas system".

Therefore, other factors, such as powder measure, rifling tolerance, and projectile tolerances offer greater deviations in muzzle velocity than the energy given to the gas system.

Effect on Weapon Function (M14 Rifle) of Ammunition Gas Pressure Variance at the Gas Port

This is a brief study of the effect on weapon function caused by significant variations in gas pressure at the gas port area.

This study should result in a determination as to what pressure variations at the gas port positions are acceptable for weapon function.

It will be shown that a relationship exists between the pressure in the barrel and the pressure in the gas cylinder together with the velocity of the operating rod.

In the function of the gas cut-off and expansion system, as utilized in the M14 rifle system, there are two different periods. The first is the movement of the piston and the operating rod, until the cut-off, and the second is the ploytropic expansion of the gas in the gas cylinder after cutoff. The study deals only with the first period, which will give an adequate relationship between the gas pressure in the cylinder and the velocity of the operating rod with respect to the pressure in the barrel at the time of cut-off.

The velocity of the gas flow "W" through the gas port is constant because the pressure in the cylinder is under the critical pressure.

Therefore, equation #1 is:

$$(1)\quad W = \sqrt{2g\ Cp\ T_1\ \Delta}$$

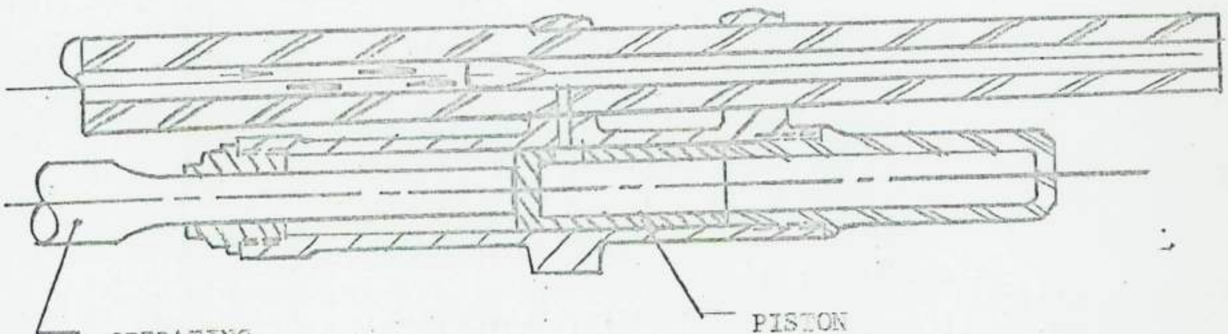
$$(1a)\ \Delta = (P_2/P_1)^{2/K} - (P_2/P_1)^{(K+1)/K}$$

$$(1b)\quad P_2/P_1 = (\not{f}) \left[2/K+1\right]^{K/K-1}$$

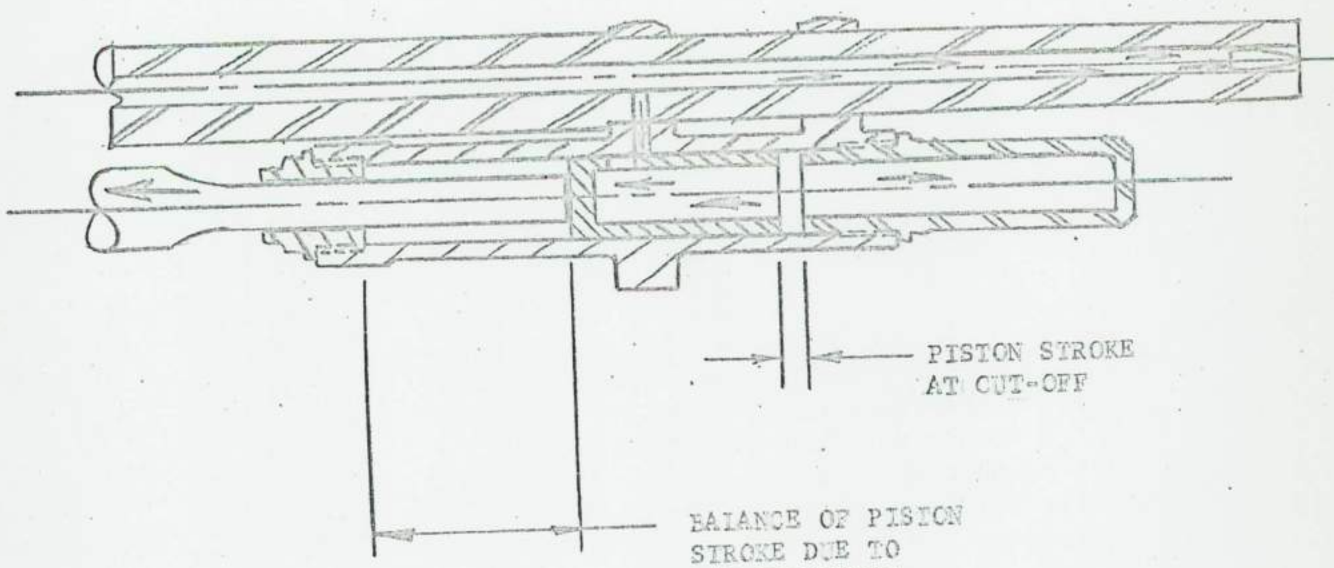
W = Velocity of the gas
Cp = Specific heat of the gas
P1 = Pressure in the barrel
P_2 = Pressure in the gas cylinder
K = exponent of polytropic expansion
T_1 = gas temperature in the barrel

CUT-OFF & EXPANSION

GAS SYSTEM OF OPERATION



BULLET NEAR GAS PORT



It should be noted that P1 is not the instantaneous pressure when the bullet passes over the gas port, but is the average pressure between bullet passage and time to cut-off. This will not induce an error, since it is assumed that P1 is proportional to the bullet passage pressure. This study will deal with qualitative proportions, rather than absolute values.

To determine realvalues, the equations require coefficients that are found only by experimental tests. These values were not available at the time of this study, and will be noted as the formulae are developed.

The mass flow of gas per unit time is given as:

(2) $\mathring{M} = \frac{Ao}{V2} W$ $\qquad V2 = CC - \text{Volume}$

and can be written thermodynamically as:

(3) $\mathring{M} = \frac{Ao\ P1}{\sqrt{RT1}} \times \emptyset\ (K)$

(3a) where

$$\emptyset = \sqrt{\frac{2g\ K}{K-1}\ \Delta}$$

Ao = area of gas port which is fully open until cut-off time.

After a period of time "t" the gas weight "G" in the cylinder is:

(4) $G = \frac{Ao\ P1}{\sqrt{RT}} \cdot \emptyset\ (K)\ t$ $\qquad$ note that $G = \mathring{M}\ t$

The pressure P2 in the gas cylinder is:

(5) $P_2 = \frac{Ao\ P1\ R}{\sqrt{RxT_1Vo}} \cdot \emptyset\ (K)\ T_2 \cdot t$

Vo = Volume of the piston cavity
(Note relationship of Vo with P2)
T2 = Temperature of gas flowing through the orifice

Another thermal relationship is

(5a) $T_2/T_1 = 2/K+1$

which produces the final expression:

(5b) $P_2 = \frac{Ao}{Vo} \cdot P_1 \sqrt{RT_1} \cdot \psi (K) \cdot t$

(5c) where $\psi (K) = 2/K+1 \cdot \emptyset (K)$

The maximum pressure in the gas cylinder, therefore, is at the time of the cut-off.

This time can be determined by the equations of motion for the piston and operating rod.

(6) $M\ddot{x} + F_o + K\bar{x} = P_2 tA$

M = mass of piston and operating rod
Fo = Spring force
K = Spring rate
A = Gas cylinder area

$K \bar{x}$ can be neglected since the displacement $\bar{x}$ is quite small.

The solution of (6) is given by:

(7) $X = \frac{\dot{P}_2}{6M} \cdot t^3 - \frac{Fo\ t^2}{2M}$

Velocity is given by:
$V = dx/dt$

(8) $V = \frac{P_2\ t^2}{2M} - \frac{Fot}{M}$

The equation is limited, of course, by the diameter of the gas port in the piston Dp. Therefore, Xmax = Dp

(7) $\frac{P_2\ t^3}{6M} - \frac{Fo\ t^2}{2M}$ DP

For all practical purposes Fo (Spring force) can be neglected.

The equation then reduces to:

(9) $tc = \sqrt[3]{\frac{6\ Dp\ M}{P_2}}$

As a result of inspecting equations (5), (7), (8), and (9) the relationship with bore pressure P1 is as follows:

$$(10)\quad \text{max. } P2 = \left(P_1\right)^{2/3} \times \text{CONSTANT COEF.}$$

$$(11)\quad V = \left(P_1\right)^{1/3} \times \text{CONSTANT COEF.}$$

The constant coefficients are a function of temperature, effective area of orifice, orifice length, propellant, and mass.

V is the operating rod velocity at cut-off.

Therefore, from equations (10) and (11) it is seen that for a pressure variation of P1 of 30%, the variation of gas cylinder pressure Po will be 20% and velocity variation V will be 10%.

Maximum Gas Cylinder Pressure (M14 Rifle)

Applying numerical values as follows:

$T = 2800\ ^{o}K$
$R = 119$ ft./degree Kelvin
$K = 1.246$

Gas port dia. = .073 in.
Dia. cylinder = .50 in.
Mass of piston and rod = .018 lb. - sec^2/ft.
Initial cylinder volume .291 in^3
The barrel pressure at the port, P1, is in the order of 15,000 psi, but values will be shown for pressures of 10,000 and 20,000, as well as 15,000 psi for comparative values.

P (bore) = 10,000 psi	P (cyl) = 1540 psi	V = 13.5 fps
P (bore) = 15,000 psi	P (cyl) = 2170 psi	V = 15.0 fps
P (bore) = 20,000 psi	P (cyl) = 2750 psi	V = 17.2 fps

Maximum Pressure in M60 Gas Cylinder

For comparative purposes, the same method may be applied to the M60 machine gun, since that weapon also employs the expansion and cut-off system.

	M60	M14
Diameter of gasport	.135 in.	.073 in.
Cylinder dia.	.875 in.	.50 in
Initial cylinder volume	1.879 in.^{3}	.291 in.^{3}
Weight inertia	1.53 lb.	.59 lb.
Piston motion to cut-off	.19 in.	.075 in.

The pressure in the barrel for each weapon is approximately 15,000 psi.

from equation (5a):

$$P2 = \frac{Ao}{Vo} \cdot \text{CONSTANT COEF.}$$

Ao = area of gas port
Vo = initial cylinder volume
P_2 = pressure in gas cylinder

Relating M60 values to M14 values ;

$$(60)\ P_2 = \frac{(60)\ Ao}{(14)\ Ao} \cdot \frac{(14)\ Vo}{(60)\ Vo} \cdot (14)\ P_2$$

(14) = M14
(60) = M60

therefore,

$$(60)\ P_2 = 3.42 \cdot 1/6.5 \cdot (14)\ P_2$$

Since (14) P_2 = 2170 psi, then (60) P_2 = 1150 psi

M60 gas pressure is then adjusted for the loss at the forward vent hole:

$$P = P_2\ (1 - .055) = 1090\ \text{psi}$$

Max force on piston:

$$F = \pi/4\ D^2 \cdot P = 660\ \text{lb.}$$

Definition of terms:

W = Velocity of gas flow
g = gravity, = 32.2
Cp = specific heat of gas
T_1 = Temperature gas in bore
Δ = Pressure ratio coefficient
K = Exponent of polytropic expansion
P_1 = Bore pressure (average at port)
P_2 = Gas cylinder pressure
$\dot{M}$ = Mass of gas
Ao = Area of gas port
V_2 = Covolume of gas
R = Gas constant PV/T
ϕ = function of g, K, and Δ
G = Weight of gas
t = time
Vo = Volume of piston cavity
T_2 = Temperature of gas through orifice
γ = Coefficient of K
M = Mass of inertia parts
A = Area of piston (gas cylinder)
tg = Time to gas cut-off
Dp = Diameter of gas port

V Weapon Design

In this section a number of ordnance design practices are reviewed which should serve to instruct the uninitiated. In this respect a number of elements of "art" are practiced, but ordnance engineering is predominantly a "science" as are most branches of mechanical engineering.

Experience, therefore, will lead the ordnance engineer toward utilizing the principles disclosed in these chapters. Creativity, or inventiveness, cannot be taught, but comes with practice in solving problems and "thinking out" new approaches to problems that are either old or new. This is therefore an individual quality which each engineer must cultivate.

Headspace

"Headspace" is simply the critical dimension that relates the chamber size to the cartridge size. In order that prolonged bursts of automatic fire may be executed without stretching or splitting the cartridge case the headspace dimension in the weapon must be maintained and not be affected by temperature, strain, or fatigue.

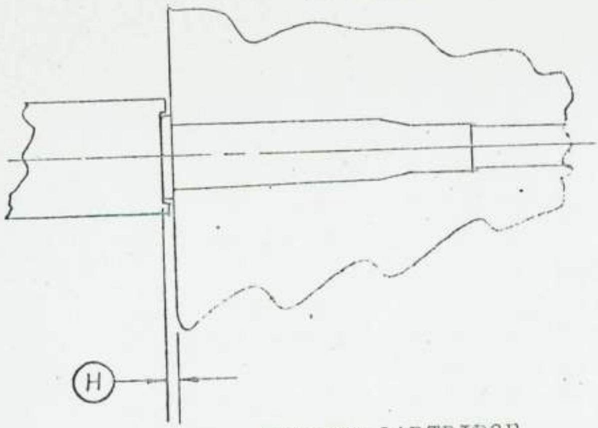
The term "headspace" was derived as a specified distance, or space, between the bolt face and barrel face, in the days of rimmed cartridges. Early cartridges, including those of high power class, were rimmed; therefore headspace indentification and control was simplified.

The use of rimless and semi-rimmed cartrdige cases necessitated other techniques for identifying "headspace" in a weapon. It, therefore is designated as the distance from the bolt face to the surface that stops forward movement of the cartridge case in chambering.

The standard necked case will have a reference diameter near the middle of the tapered shoulder area that corresponds with the same diameter in the barrel. The variations between projectile headspace and weapon headspace is the clearance, and this is usually the effect of tolerance accumulation. A number of machine guns have been developed that utilize an adjustment nut to draw the barrel closer to or further from, the breechblock face, thus compensating for changes in headspace that may be due to wear, replacement of components, case stretch indications, etc. Modern machine methods permit production of weapons with a "fixed" headspace, such as the M60 machine gun.

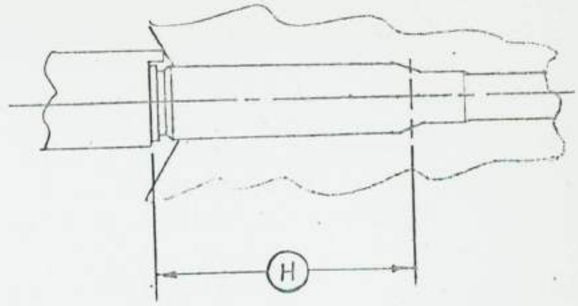
As mentioned above, the headspace in modern chambers is the distance from the locked breechface to a reference plane along the cartridge shoulder. To measure, or check, this a special set of

BASIC FORMS OF HEADSPACING



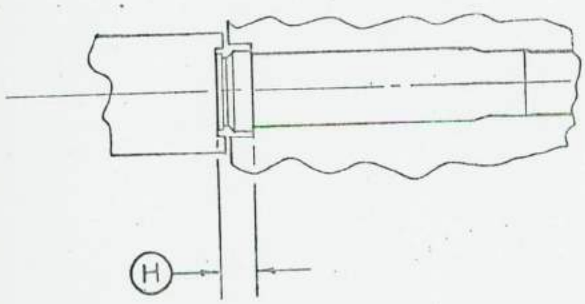
RIMMED CARTRIDGE

(30-30 Win.)



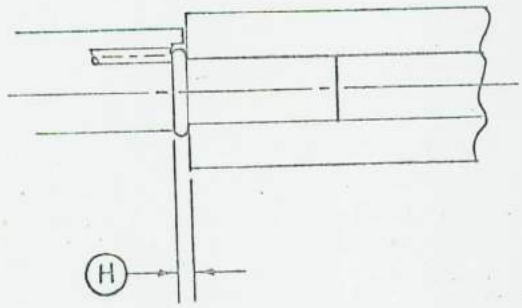
RIMLESS CARTRIDGE

(7.62 mm NATO)



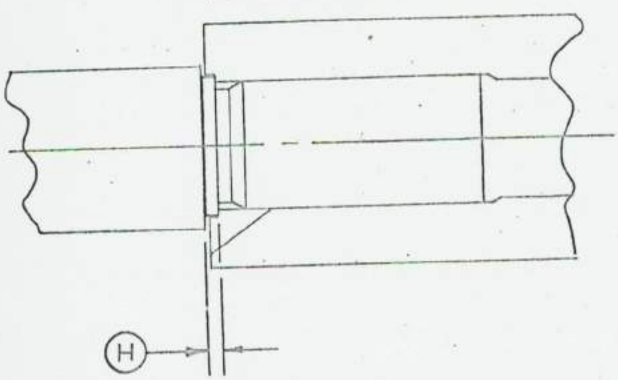
BELTED CARTRIDGE

(.375 H&H Magnum)



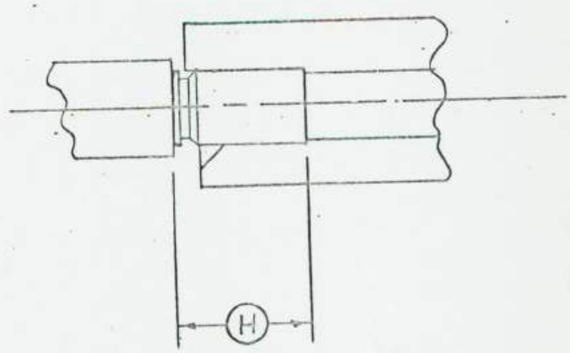
RIMFIRE CARTRIDGE

(cal..22)



SEMI-RIMMED CARTRIDGE

(cal..38 ACP)



RIMLESS CARTRIDGE

(no shoulder) (cal..45 ACP)

(H) indicates HEADSPACE CONTROL DIMENSION headspace gages is required. A "master ring" is used to check headspace gages. .

For example, in the cal. .30 M1, the headspace specified is 1.940 MIN, 1.944 MAX., a variation of only .004. This is measured from the breech face to a reference diameter of .4425 on the shoulder taper of 34°26' incl. An additional .002 headspace is permitted when inspecting overhauled rifles, with yet another .004 additional permitted as a field headspace limit for serviceable rifles, of 1.950. This is a total of .010 headspace variation permitted. This is close to the limit for a brass case, in which .016 stretch would adversely stretch the case body at the area behind the shoulder.

Another headspace control technique utilizes a "belt" or shoulder that functions as a secondary rim in front of the base rim. This short stopping shoulder stops on a ledge in the chamber shortening the headspace length, thus minimizing deflections in firing.

Control of headspace is necessary, because excessive headspace will cause irregular ignition as well as overstretched cases. Excessive headspace may cause case rupture, resulting in a stoppage, if not serious breakage. Excessive plastic deformation of the cartridge case may be observed as a shiny circle approximately 1/4"-3/8" behind the cartridge shoulder inside the case.

The following stress-strain diagrams demonstrate the conditions of either clearance or case interference, after firing, in the radial direction:

The following assumptions are made:

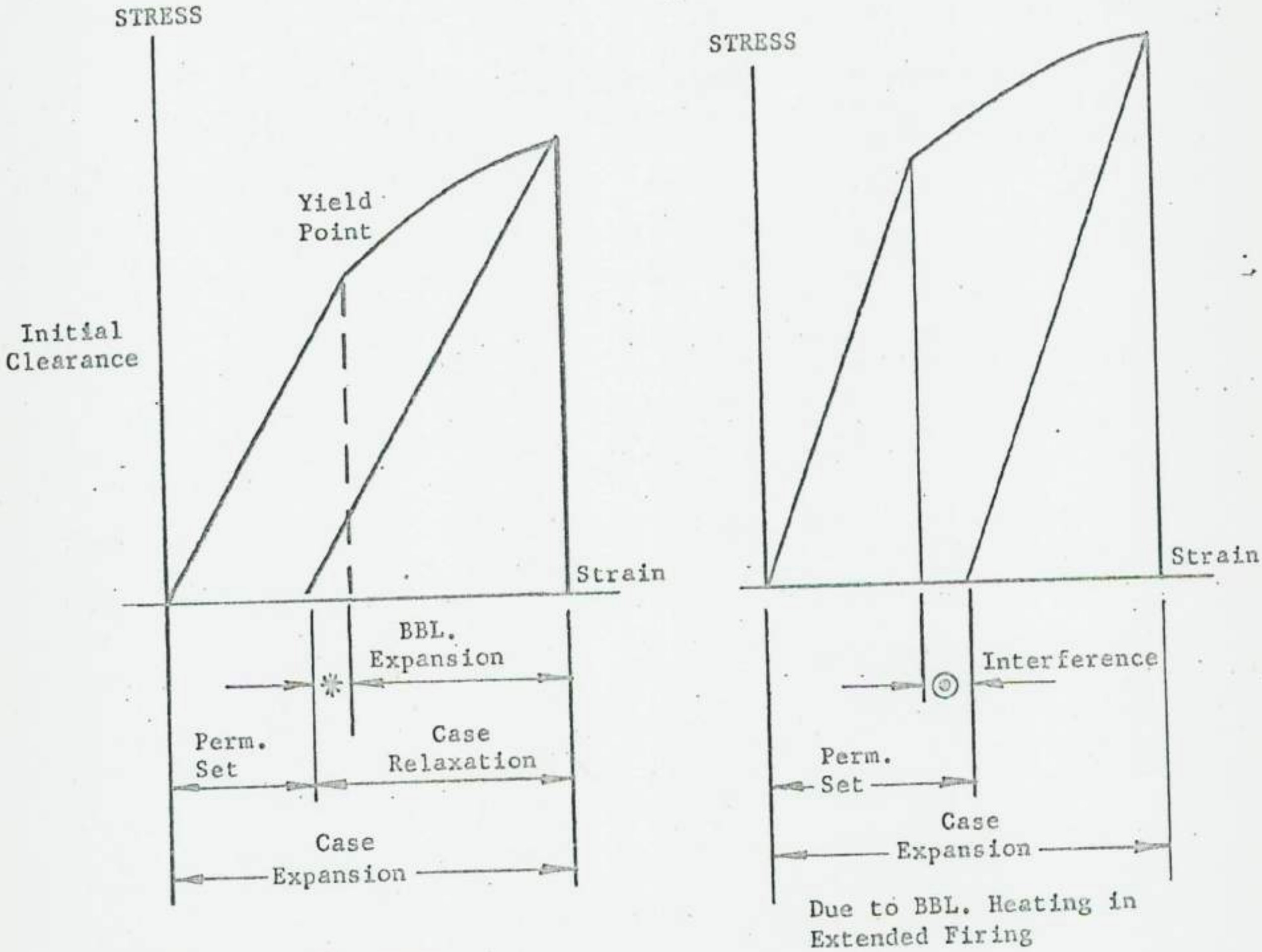
(1) The chamber recovers completely from the deformation due to propellant gas pressure, otherwise the chamber would be most improperly designed.

(2) The yield strength of quarter-hard 70-30 cartridge case brass is 40,000 psi.

RADIAL EXPANSION

CARTRIDGE CASE VS. CHAMBER

Stress-Strain diagrams of cartridge case radial expansion in chamber during firing for conditions of normal and excessive barrel expansion.



※ = Resulting Clearance (Case will be free for extraction)

The obvious method of preventing interference is to increase the chamber wall thickness or reduce case/chamber clearance.

Longitudinal clearance, of course, can cause serious stoppage, if excessive.

In a number of systems, crunch-up is intentionally designed into the case and chamber. Sufficient residual energy must be available in the moving breechblock to perform this work. (M75) weapon, as an example exhibits this method of cartridge control in the chamber. One approach to chamber design is to assign a metal-to-metal contact between a minimum cartridge and a maximum chamber. If there is not available energy to chamber a round under conditions of interference; then the chamber must be deepened, at the cost of excessive headspace when tolerances result in an excessive clearance.

Chamber sidewall taper is calculated so as to assist in ready extraction. A slight primary extraction effort will free the case from the chamber wall.

A forcing cone at the start of the rifling prepares the rotating band of the bullet for engraving by the rifling, with a minimum of free run, or "bullet jump" to minimize impact.

The Browning Machine Gun, for example, required headspace adjustment each time the weapon was reassembled or each time another barrel was inserted. The M60 machine gun has "fixed headspace" and does not require field adjustments. This is due to coordination between gun design and production techniques.

For example, in the BMG too tight headspace would cause:

(1) Failure of lock to enter recess in bolt.

(2) Failure to fire, the bolt not reaching the firing position

(3) Failure to reach and extract cartridge from the belt.

(4) Sluggish action due to excessive locking friction. This is the most frequent consequence of tight headspace.

Loose headspace would tend to cause:

(1) Excessive battering of the lock, locking recess, and lock cam.

(2) Ruptured or separated cases.

(3) Poor shot patterns due to pressure leakage at the breech.

American cartridge brass is softer than the European brass, to allow greater headspace variation. U.S. ordnance practice is for 100% interchangeability of components, whereas European practice is for greater use of selective assembly and individual fitting at assembly. For this reason, the European hard brass is not usually given to excessive case stretch.

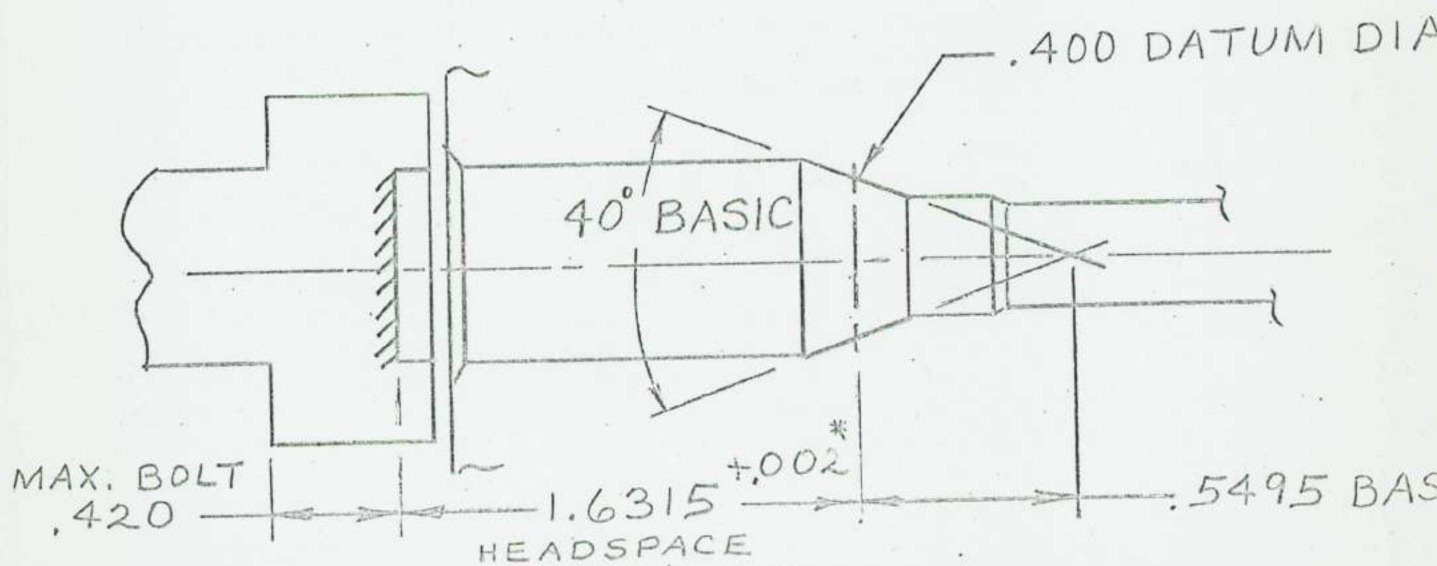
Front locking lugs are desirable for designs that tend to reduce excessive case stretch in firing. Rear locking lugs, or locking surfaces located a long distance from the case seat cause stretching of the receiver or frame and compression of the bolt body, both according to the formula $\delta = P\ell/AE$. Headspace change also occurs through temperature change over this length. For this reason, most modern weapons are front-locking rotary bolts with fixed headspace, resulting in less distortion of the cartridge case in extended firing.

Symptoms of excessive headspace are obvious in visual inspection. A case stretch is indicated by a bright zone extending all around the case about a half inch from the base. As headspace increases a fine crack will begin to show up on the cartridge body.

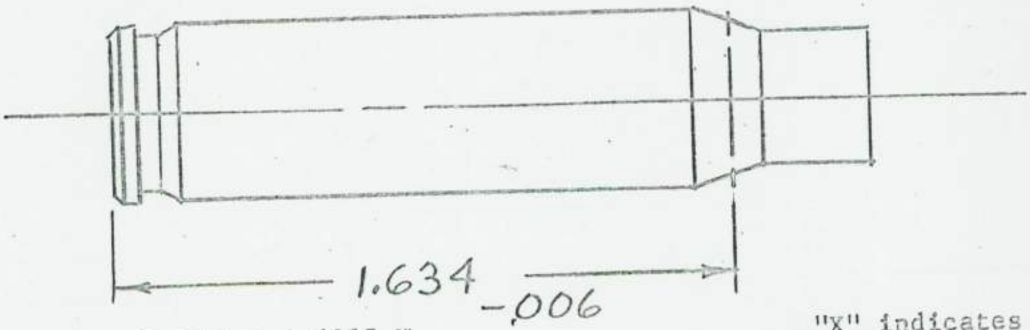
Another danger of excess headspace is in the free run it permits the cartridge case and bolt in setting back against the locking abutment. This impact intensifies the shock loading factor, and is dangerous for receivers tempered to a high hardness. The rimless cartridge can characteristically be driven forward an excess amount by the bolt slamming into battery, particularly if there is excessive forward clearance between the bolt and the barrel face.

Here a practical exercise is given in studying dimensions and accumulating tolerances in a typical weapon.

As an example, study M60 headspace diagram as follows:



Compare this with cartridge case dimensions of:



chamber 1.6345 X 1.6315 N
case 1.628 N 1.634 X

"X" indicates maximum di
"N" indicates minimum di

.0065 X .0025 INTERFERENCE, OR CRUSH.
CLEARANCE

* 1.6315 + .003 AFTER PROOF FIRING

Looking further into the development of this headspace data, study the ability of the case to stretch: The section just to the rear of the case shoulder tapers from a thickness of .010 inches to .030 in a length of 1.310 inches. The formula for determining allowable stretch is:

$d = \frac{sl}{E}$ where s = yield stress of 40,000 psi

E = 18,000,000 psi

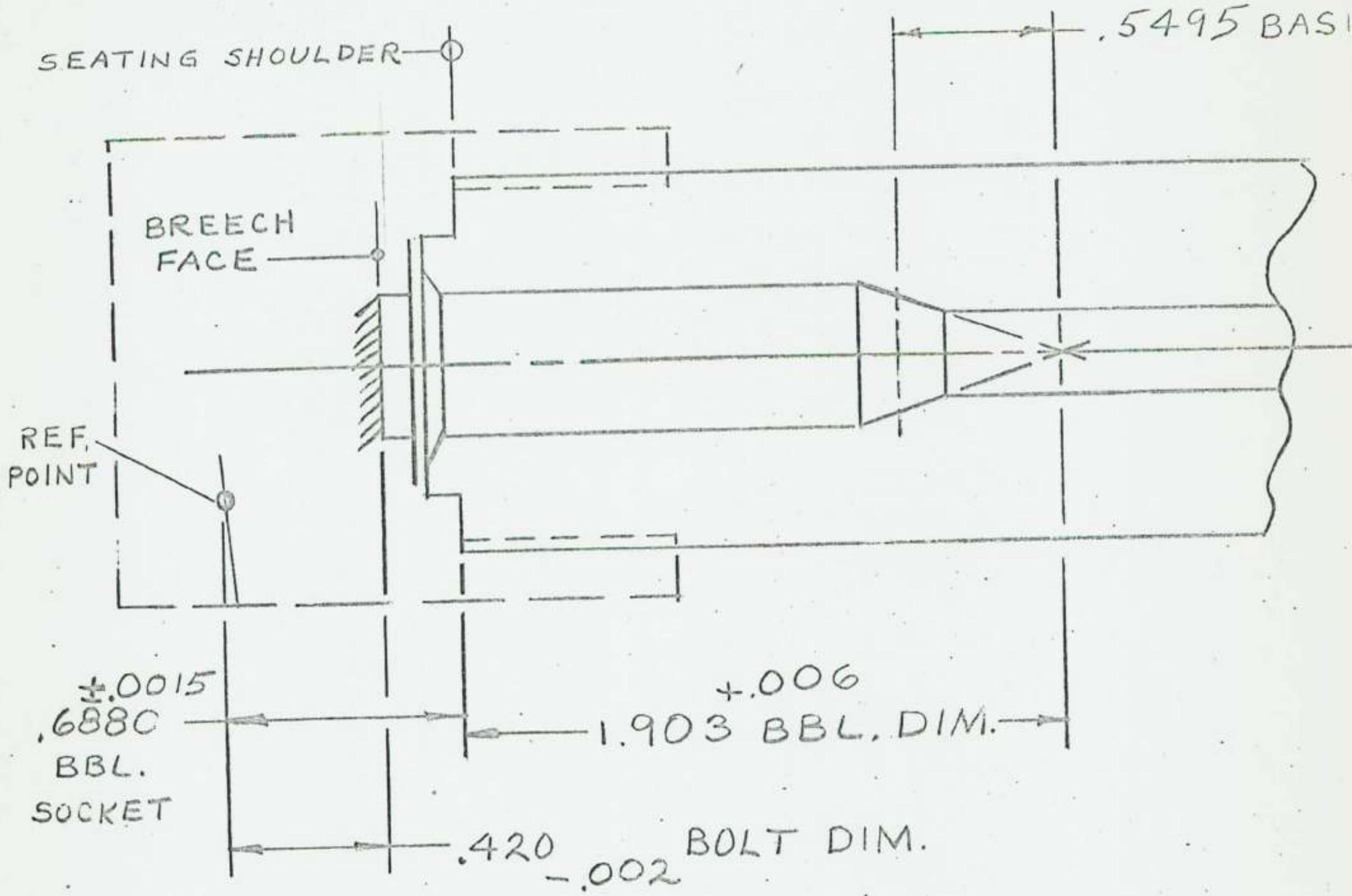
l = 1.31 in.

$$d = \frac{.404 \times 1.310}{18.}$$

$d = .003$ "

Therefore, any stretching over .003" permanently deforms the case, which illustrates why handloaders resize their cartridge cases.

A look at the individual dimensions in the M60 Machine Gun to demonstrate how tolerances can accumulate:



ACCUMULATION OF TOLERANCES:

	1.909X	1.903 N
	.6895X	.6865N
	2.5985X	2.5895N
-	.9675N	.9695X
	1.6310X	1.6200N

.420 X	.418 N
.5495	.5495
.9695X	.9675N

Therefore, after the closest tolerances, reasonably maintained in production, the final headpsace dimension, reamed at assembly, compares as follows:

1.6335X	1.6315N	
1.6200N	1.6310	
.0135X	.0005N	METAL REMOVED

For machine guns, the barrel chamber area may increase in temperature by as much as 850° F.

The temperature expansion, then is:

$\delta_T = E(\Delta T)\ell$ $\quad\quad$ $d = (.0000065)(850)(1.63)$

$d = .009''$

It is obvious that for prolonged firing excessive stretching will ensue, which may be compensated for in the cartridge case physical properties, which are generally as follows:

Properties of cartridge brass:

70% Cu

30% ZN

	HARD	SOFT
Tensile Strength kpsi	76	47
% elongation in 2 in.	8	62
Yield strength (5% extension under load) kpsi	63	15
Rockwell Hardness	B82	F64
Melting Point	1750°F.	
Density #/ft^3	532	
Coefficient of expansion per	$^{\circ}F \times 10^{-6}$	11.1
Thermal conductivity (BTU/hr/ft^2/ft/F°)		70

U.S. cartridge brass is approximately quarter hard in order to take advantage of the high percentage of permissible elongation (ductility)

In closing this topic it may be recalled that during days of adjustable headspace machine guns, a gunnery instructor could adjust the weapon so that, after firing briefly, cases would begin to split, and the students would have to quickly correct headspace, in the dark, in order to complete the course.

Typical Factors of Safety in Small Arms Design

The determination of allowable stresses for small arms components is quite important in that small arms design demands a minimum of weight and volume commensurate with the work being done by the weapon mechanism. The factor of safety is generally an educated guess and is based upon the endurance life of the subject weapon. It should take into account the non-uniformity of the material and its heat treatment, inside corners of various degrees as stress raisers, tool marks, grain direction, and a myriad of other factors.

If the allowable stresses selected in design are too high, the service life of the component is compromised, whereas if it is too low, a penalty is paid in excess weight, particularly for an infantry weapon.

One comparative design area that may be used for study purposes is the bolt locking lug design. This area is usually highly stressed, since the bolt lug size is preferably kept to a minimum. This is because, for rotary and tilting bolts, if the rotation to lock/unlock is minimized, the cam profile will be reduced, and bolt carrier bulk minimized. The locking lug height is minimized, as this determines the size of the receiver or locking ring. As a result, allowable stresses are as high as any other component in a weapon; with the possible exception of the barrel, which has additional stresses due to temperature, vibration, external loads, etc. Therefore, locking lug design practice is more representative of a small arms design structure than barrel design.

Accordingly, maximum stresses and resultant factors of safety for three bolt lug designs are compared.

These include:

(a) 7.62mm M60 Machine Gun
(b) 7.62mm M14 Rifle
(c) Cal. .30 M1 Carbine

The maximum firing force is assumed to be the maximum service pressure, multiplied by the maximum inside area of the cartridge case, since the wall of the case does not transmit bearing forces. (Except in the instance in which the case ruptures from excessive headspace).

Stresses calculated are typically shear, pressure, and bending. The shear and bending stresses are then combined into a comparable multi-axial stress, according to the Hencke-Von Mises theory.