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.

According to this theory, when there is a two axial stress consisting of a normal stress and a tangential (shear) stress the equivalent stress is:

$$Se^2 = S^2 + 3\tau^2$$

S = BEARING
τ = SHEAR

The equivalent stress must be smaller than the allowable stress in tension. Therefore, the maximum shear stress is equal to .57 of the allowable stress in tension. (as a maximum)

It is recommended that the actual shear stress is 1.5 times the calculated average shear stress, because in bending, the shear is not distributed evenly over the cross section. Summarizing, then,

$$1.5\,\tau \leqq .57\ S\ (\text{ALLOWABLE})$$

or

$$\tau \leqq .38\ S\ (\text{ALLOWABLE})$$

As for the bending stress on the locking lugs, it is assumed that the breech thrust force is uniformly distributed, so that the lever arm of the bending moment is taken as half of the height of the locking lugs.

Under these assumptions, the following values were determined:

	M60/M14	CARBINE
Service Pressure	50,000 psi	34,200 psi
Cartridge I. D.	.41 in	.31 in
Force	6,600 lb.	2,600 lb.

Resultant stresses are:

	M60	M14	CARBINE Left Lug	CARBINE Right Lu
Shear stress	12,700	31,000	17,500	17,000
Bearing stress	70,000	64,000	90,000	22,000
Bending stress	7,800	50,000	10,000	40,000
Combined stress	23,400	73,400	21,000	50,000

The safety factor is a ratio of the dynamic yield stress to the above stresses. The dynamic yield stress is assumed to be equal to the static tensile strength of the material.

The static tensile strength is:

'S' TENSILE = 168,000 psi for M60 & M14
'S' TENSILE = 172,00 psi for Carbine

Safety factors resulting:

	M60	M14	CARBINE
Shear	5.3	2.2	3.9/4.0
Bearing	2.4	2.6	1.9/7.8
Bending	21.4	3.4	17.2/4.3
Combined	7.3	2.3	8.2/3.4

Therefore, as a guideline, the preliminary average factors of safety for design is recommended as follows:

Shear	3.0*	Minimum
Bearing	2.5	Minimum
Bending	4.0	Minimum
Combined	3.5	Minimum

*Shear stress safety factor $\leqq$ .38 static tensile strength

Stresses in Barrels

Much of the small arms barrel strength is designed to resist external forces, such as bayonet thrust loads, pull from sling, grenade launchers and other muzzle attachments, parachute drop, and other adverse or abusive forces.

An excellent arrangement of standard formulae on this subject is given in Pamphlet AMCP-706-252 "Gun Tubes"

However, additional confirmatory data is obtained experimentally, and strain gages are applied to the outside diameter of the barrel in order to measure actual loads (strains) imposed during firing. The internal pressure in terms of the strains measured at the outside diameter is given by the following formula:

$$Pi = 1/2\ E\,\varepsilon_c\,(w^2-1)$$

E = Young's Modulus (30×10^6 for steel)

ε_c = Strain in/in measured at O.D.

w = ratio of O. D. to I.D.

For a simple comparative value, the following formula for the maximum tangential stress, which occurs at the inner wall, is frequently useful:

$$\sigma_{tp} = P\,\frac{w^2+1}{w^2-1}$$

w = the wall ratio, Do/Di

As a general rule of thumb, for rifles, the barrel thickness over the chamber should be at least 2/3 of the dia., Di, measured at the midpoint of the cartridge body.

$$
\begin{aligned}
\text{For 7.62mm NATO: dia} &= .465, \\
\text{then th.} &= .310 \\
\text{O. D.} &= 1.085 \\
w = 1.085/.465 &= 2.34 \\
w^2 &= 5.45
\end{aligned}
$$

Using a proof pressure of 65,000 psi, the resulting maximum tangential stress at that point in the barrel is approximately:

$$\sigma_{tp} = 65{,}000 \times 6.45/4.45 = \underline{94{,}000 \text{ psi}}$$

For machine guns, where temperature stresses are a factor, the wall thickness should be increased so that the maximum tangential stress is in the order of 80,000 psi. Another useful formula is that :

for thin walled cylinders: $(t < 1/10\ D)$

hoop stress $= pr/t$

axial stress $= pr/2t$

Stress on Bolt Locking Lugs

In a rifle design, the bolt lugs are usually highly stressed, as noted in the previous chapter on "Factors of Safety".

The Cal. .30 M1 rifle is used as an example, since, as a rotary bolt with front locking lugs, it typifies most of the modern military rifles.

Several methods may be used to determine the load developed aginst the bolt face during firing.

The first shown is in knowing the shape of the pressure-time and pressure-travel interior ballistic curves and equating impulse to momentum, so that

$$Ft = mV$$

In firing a 150 grain bullet with 50 grains of propellant, and a muzzle velocity of 2700 ft./second.

The peak pressure is 50,000 psi, and acts for .2 msec. before a projectile travel of 1.5 inches with a velocity of 1200 fps causes a decrease of pressure in typical hyperbolic fashion. For this period of projectile acceleration, $F = m\,a = \dfrac{m\,v}{t}$

$$F = \frac{150 \times 1200}{7000 \times 32.2 \times .0002} = \underline{4000 \text{ lb.}}$$

This is the thrust load acting on the projectile at the time of peak acceleration.

Compare this with the projected peak pressure, acting on the bore diameter and the cartridge case.

$F = SA = 50,000 \times \pi/4\ .308^2 = 3750$ lb., which compares with the force of 4000 lb.

Inside the cartridge case, the inner dia is approx. .410", therefore

$F = 50,000 \times \pi/4\ \ .41^2 = 6,600$ lb.

This may be taken as the peak thrust against the bolt face. The bolt thrust forces are developed by the peak pressure acting across the inner case dia, not the chamber dia. In the event of a case rupture, then the pressure does act across the chamber dia, increasing the bolt thrust considerably. The difference between projectile and bolt thrust is the force acting against the annular case neck, obturating the gases. In this case it is approximately 2850 lb. and acts to stretch the cartridge case in the chamber, requiring headspace control. This will be discussed in detail in the chapter on Headspace. It should be noted that any looseness in headspace will increase the impact effect of the case and bolt against the locking lugs.

In calculating the stress on locking lugs, four stresses are considered:

(1) Bearing

(2) Shear

(3) 45° shear

(4) Bending

(5) Combination of above 4 stresses

Using the .30 cal. M1 example, the bearing surfaces are approximately .081 in.^{2} which results in a direct bearing stress of 5,500 lb./.081, of 68,000 psi. Apply a factor of 1.35 as an impact load factor, due to the nature of dynamic loading, resulting in a stress of 92,000 psi.

Now the direct shear area should be approximately at least twice as long as the direct bearing height. This will provide, in this instance, a shear stress of 46,000 psi, and provide a lug that will be essentially free of high bending stresses in the axial direction. In the actual M1, the lug is longer than necessary; therefore the shear stress is lower. (31,000 psi) Bending stresses in the radial direction are more complex, and formulae may be found in Roark's "Formulae for Stress and Strain".

The 45° shear stress is known as Lueder's slip lines, and is a plane 45° to the bearing surface. This is the plane of maximum shear stress. In this example, the shear stress from this source is $92000/\sqrt{2}$ or 65,000 psi.

The radial (inward) bending stress depends upon the construction of the bolt body. Hollow shell type bolts, may suffer high stresses in this respect, but a fairly solid bolt head, as in the M1, is rugged in this respect.

It cannot be overemphasized that sharp inside corners, particularly on the stressed corner where the locking lug is developed, should be avoided since a sharp inside corner is the beginning of a crack. On some bolts considerable inletting of grooves are specificially machined in order to avoid the presence of a sharp corner. This is done on the M60 bolt, for example.

Note that in the M1 bolt structure the locking lugs are positioned high with respect to the lateral centerline. This results in a definite longitudinal torque to the entire bolt body, thus positively defining the stress path for each shot. That is, on each round, the upper surface of the bolt body is in tension, the lower surface in compression. This geometry is necessary, so that the feed lips may be raised to a favorable position for feeding.

One common formula for combined stress is:

$$\text{Max. } S_p = \sqrt{(1/2\ S)^2 + S_s^2} = 79{,}000 \text{ psi}$$

S_p = Max shear stress

S = Bearing Stress = 68,000 X 1.35 = 92,000 psi

S_s = 65,000 psi = Shear Stress at 45°

The principal stress, then is

$$S^1 = 1/2\ S \pm \sqrt{(1/2\ S)^2 + S_s^2} = 125{,}000 \text{ psi}$$

This is why points of stress concentration should be avoided, as a plague. Note also that the peak loading time of .2 milliseconds develops a safety factor, since the bolt lugs undoubtedly react to a lower average load.

However, the 125,000 psi combined stress value is a good reference point since the yield strength of this material is in the order of 160,000 psi.

There are a number of methods by which a bolt lock may function. This includes rotary bolts, tilting bolts or locks, propped locks, wedge locks, ad infinitum. However, each good locking system should have a number of distinct characteristics that enable it to function efficiently with the high pressure characteristics of the cartridge case. The most prominent mechanical design feature is that commonly referred to as "primary extraction" or "initial extraction".

It is that characteristic in design that enables the cartridge to slip rearward only several thousandths of an inch in the chamber prior to unlocking and full extraction. In this way, the cartridge case is pried away from its sealed-in position in the chamber, and some semblance of an acceleration to the cartridge is applied. At this time of initial extraction the chamber pressure is still in the order of 700 - 1000 psi, therefore, the cartridge case is still in a condition of stress, particularly at a point just to the rear of the shoulder, where the wall thickness is near a minimum. This is where case stretching occurs, and will be discussed in further detail in the chapter on "headspace".

However, the bolt locking lugs, the subject of this topic, should provide initial extraction because, for rotary bolts, the lug bearing surface is machined at a helix angle in the order of 4°. Angles larger than 4° are not recommended, as the bolt lugs will not be self-locking, but will tend to transmit a rotary pulse upon firing, therefore battering the locking cam and/or stud. This helix angle not only cams the cartridge into the chamber but also helps back out the cartridge case, unseating it from the chamber walls preparatory to the formal extraction stroke and this distance is equal to the <u>arc length rotated by the base of the locking lug</u> times the <u>tangent of 4°</u>. Therefore, rotary bolts with large numbers of small lugs are limited in this extraction distance because of the small angle or rotation.

As a simple experiment in observing the self-locking stability of a bolt, merely assemble the bolt into the locked position without the operating rod or bolt carrier, and with the gun held vertically, muzzle upward. Carefully drop a long, brass rod down the bore, so that it will impact sharply on the bolt face. While the bolt will prance up and down, if the locking surfaces are self locking, the bolt will not rotate and fall out. Repeat this several times, particularly if the bolt creeps in rotation. Bolts that tilt, as the cal. .50 M8 Spotting Rifle and the 7.62mm FN rifle, or have tilting locks, as the B A R, also exhibit primary extraction. As the bolt locking surface rides up the locking abutment during unlocking, it will be observed that the bolt face moves rearwardly a proportionately small distance. The same is true of the Browning Machine Guns, in which the locking face is not vertical, but inclined rearwardly approximately 3° - 4°.

The M73 machine gun has a cross-sliding bolt that moves, not perpendicular to the bore, but at a receding angle of 4°. Again this cams the cartridge into the chamber, during feeding, and pries the spent case out during initial extraction.

The cam driven, externally powered weapons, such as the M61, M75, M129, and XM140 have this primary extraction feature built in to a marked degree because of the low slope of the initial opening phase of the cam.

In addition, it may be noted that most of the modern military and commercial weapons employ front locking lugs on the bolt. This shortens the amount of breech deflection at the instant of peak pressure, thereby improving headspace control.

A practical note may be added here in that the bolt should have an adequate impact surface against the barrel or breech face when the operating rod is released without ammunition being fed, that is, an empty weapon. This impacting will occur frequently, therefore adequate bearing surface should be provided in order to prevent peening or jamming.

Firing Mechanism Design

The starting point for designing the elements of a firing mechanism is a consideration of the primer sensitivity, or the energy required to fire 100% of the primers 100% of the time. A large number of misfires will result if the striker blow is less than the maximum of the drop test specification for that primer. For example, in testing primers, a sampling of .05% is taken (one in 2000) so that, statistically, if one primer of 1000 samples misfired at the high drop level, the chances are about one in five of obtaining misfires at this energy level of a group of 200 primers taken from this lot.

Inspection methods are usually required to insure that a given small arms weapon has adequate striker energy delivered to the firing pin. This may be done by the firing pin indenting a standard copper crusher gage when struck. A test fixture is designed in which the primer is replaced by a copper cylinder.

Coordination is required between ammunition and weapon designers as to the accepted level of primer indent energy required for each system. Then, the striker, or firing mechanism energy is designed for twice (or 1.5 as a minimum) the specified "all must fire" primer strike energy.

The following standard primer drop test specifications are summarized:

CALIBER	WGT. OF BALL	HEIGHT AT WHICH ALL MUST FIRE	ENERGY
.30 Rifle	4 oz.	15. in.	3.75 in. lb.
.30 Carbine	2 oz.	18. in.	2.25 in. lb.
.45 Cal.	2 oz.	16. in.	2.0 in. lb.
.50 Cal.	8 oz.	17. in.	8.5 in. lb.

An indent test of various weapons (using copper pressure cylinders) was performed, and could be made part of an automatic burst, since the instrumented cartridge case could be loaded in a clip or belt.

A. Correlation with drop test (4 oz.)

DROP HEIGHT	INDENT
9"	.013 in.
12"	.015 in.
15" (All fire spec.)	.017 in.

B. Rifles

M1903	.020 in.
M1917	.017 in.
M 1	.016 in.
BAR - first shot	.014 in.
slow automatic	.016 in.
full automatic	.013 in.

C. Machine Guns

M1917 - first shot	.015 in.
full auto	.023 in.
M2 AC - first shot	.017
full auto	.020 in.

M1918 A4 first shot	.018 in.
full auto	.021

Therefore, a number of weapons appear to be marginally close to the 15" drop specification.

To better understand primer test procedure the following is repeated from a specification for the M42 (Fuze) primer, in which the drop-weight is a steel ball weighing 1.94 ± .02 ounces.

Procedure of Test

Drop test 50 primers at 8 inches. Then drop test 50 more primers at 9 inches, at 10 inches, etc., until a height is reached at which all the sample of 50 fire. If the test is run to 13 inches and misfires still occur, reject the lot. Then for all other lots, test 50 primers at 7 inches, 6 inches, etc. until a height has been reached at which none of the samples fire. If the test is run to 2", and fires still occur, reject the lot. The test procedure is terminated as soon as a height at which none fire has been reached"

When the striker, firing pin, or hammer hits the primer, the primer pellet is compressed between the anvil and the firing pin tip. This blow must be sharp, so that the firing pin energy will not be dissipated, causing misfires. The firing pin tip should not be too large in diameter or the resulting pressure force will shear the primer cup into the firing pin recess, either plugging the hole or severely eroding the breech face by high pressure gas leakage.

The primer strike is one end of the firing mechanism, while the other end, which starts the train of events is the trigger pull. Here is where a firing mechanism is so unique with respect to all other mechanisms and devices. The trigger pull must be a smooth, carefully gauged motion, or "feel". Trigger pull is in the order of 8 lb. ± 2 lb. and the trigger pull is usually taken in two phases, a slack, and a squeeze. The mechanical motion must be smooth, and not gritty, so that the marksman does not know when the hammer is released. If he did know, he would flinch, quite involuntary, and thus miss his shot. If, at any time during the trigger pull, the shooter changes his mind and decides not to shoot, the linkages should return to their starting position, and not be "hung up" partway along the sear surfaces.

For purposes of demonstration, the M14 firing mechanism, derived from the M1 Rifle, shall be discussed in detail. Mechanically, if contains many ideal features, as well as all of the required elements and characteristics of any typical firing mechanism.

Outwardly, it is compact, modular, rugged, foolproof in assembly, contains few components, and is accessible for cleaning and inspection. It contians one coil spring and one formed wire spring; and two cross pins, both of which are arrested from cross motion during firing, by the receiver.

The safety lever is ideal, in that its position is instantly recognized by sight or feel. Further, once the safety is applied, it is not easily unsafed by accident. The trigger finger "safes" or "unsafes" the weapon, minimizing motion of the shooter's hand or arm.

The components, or elements, required in a typical firing mechanism, and as contained in the M14 firing mechanism, are as follows:

1. Housing - This is the form of a three-walled box-type structure, and is highly rigid, supporting the lower receiver bridge and magazine housing. In conjunction with the trigger guard, it clamps the stock tightly to the receiver. This is important, in that it eliminates looseness in the housing, receiver, and stock groups.

2. Trigger Guard - Made of sheet metal, it cam-locks the housing to the receiver, clamping the stock up tightly. It is formed so as to be spring-tight in the latched position, with a bullet tip hole provided for use with tight samples. As the trigger guard is swung open, the hammer is automatically cocked.

3. Trigger - Simple, rugged, and comfortable to the finger. The spring moment-arm is extremely short, in order to reduce loads to an acceptable level. After hammer release, finger follow-through motion is short and comfortable.

4. Primary Sear - In the M14 mechanism, this is integral with the trigger. In many other mechanisms, it is the component that the trigger moves to release the hammer. The hammer "claw", "hook" or "notch" must work at a slight pulling angle when the primary sear is in motion. In this way, the hammer/sear will not slip or creep due to vibration, but kinematically is self-holding.

5. Secondary Sear - This is the component that engages the hammer during the cycle of operations after firing while the trigger is still pulled. The secondary sear should hold the hammer in a lower position than the primary sear. When pressure on the trigger is relaxed, control of the hammer then passes from the secondary sear to the primary sear. During trigger pull, the primary sear moves partway from the hammer claw in the "slack" phase. Then the back of the hammer claw bears against the secondary sear for increased pressure in the "squeeze" phase. An extension on the side of the secondary sear provides a bearing surface for the sear release, in full automatic fire.

6. Sear Release - Mounted on receiver, in the M14 rifle. In automatic fire, this component functions to release the hammer upon the timely traverse of the operating rod. That is, the hammer is held until the bolt has been closed and positively locked by the operating rod.

7. Connector - Mounted on receiver. In automatic fire, this component is actuated by the operating rod to rotate the sear release, firing the weapon. The timing of release is a function of operating rod position. (Bolt is locked, bounce under control)

8. Safety - The safety locks the hammer and blocks the trigger. It cannot be engaged unless the hammer is cocked. This is important, as a firing mechanism can be broken otherwise. That is, if a safety mechanism is first engaged, then the hammer cocked, then the cocking motion would cause the hammer to interfere with the safety. As the safety engages the hammer, it cams the hammer lower than the primary or secondary sear. This is necessary for tolerancing purposes so that, when the safety is released, the primary sear will always be in position to re-engage the hammer.

9. Hammer - In this mechanism, the hammer can be repeatedly released in "dry firing" without breakage. The striking face is accompanied by an extension that engages the base of the bolt, camming it closed so that the weapon cannot fire in an unlocked position. This is important. The left side of the hammer also contains an integral stop lug, so that the hammer is positively stopped during inertia travel, preventing breakage of parts. (cocking) Note that the position of the hammer claws is far removed from the hammer pivot point, thereby minimizing loads on the hardened gripping surfaces, thus reducing chipping, galling, and gritty trigger pull.

Note that the point of application of the hammer spring is such that the maximum spring leverage is applied when the hammer is striking the firing pin. That is, as the hammer rotates to a cocked position, the lever arm rotates to a lesser value, thus reducing cocking forces.

The hammer energy may be taken directly as the spring energy, or FXS; that is, mean spring force times spring stroke.

Time for hammer fall or firing pin strike is given by the formula:

$$t = \sqrt{\frac{2WS}{gf}}$$

s = travel (c.g.)
g = 32.16
F = avg. spring force
W = Wgt. striker + 1/2 spring

If at all possible, a rotating hammer should strike the firing pin at the center of percussion of the hammer, This is the thereoetical point through which the line of action passes of the resultant of all forces acting on the body. It is given by the formula:

$$\ell = Jm/xoM$$

where

ℓ = distance (ft) from axis of rotation to center of percussion

Jm = mass moment of inertia (lb./g - ft^2)

m = mass (lb./g)

Xo = distance (ft) between axis of rotation to center of gravity

The center of percussion is correctly developed by re-distributing hammer mass.

10. Hammer Plunger - a headed pin that bears against the hammer notch.

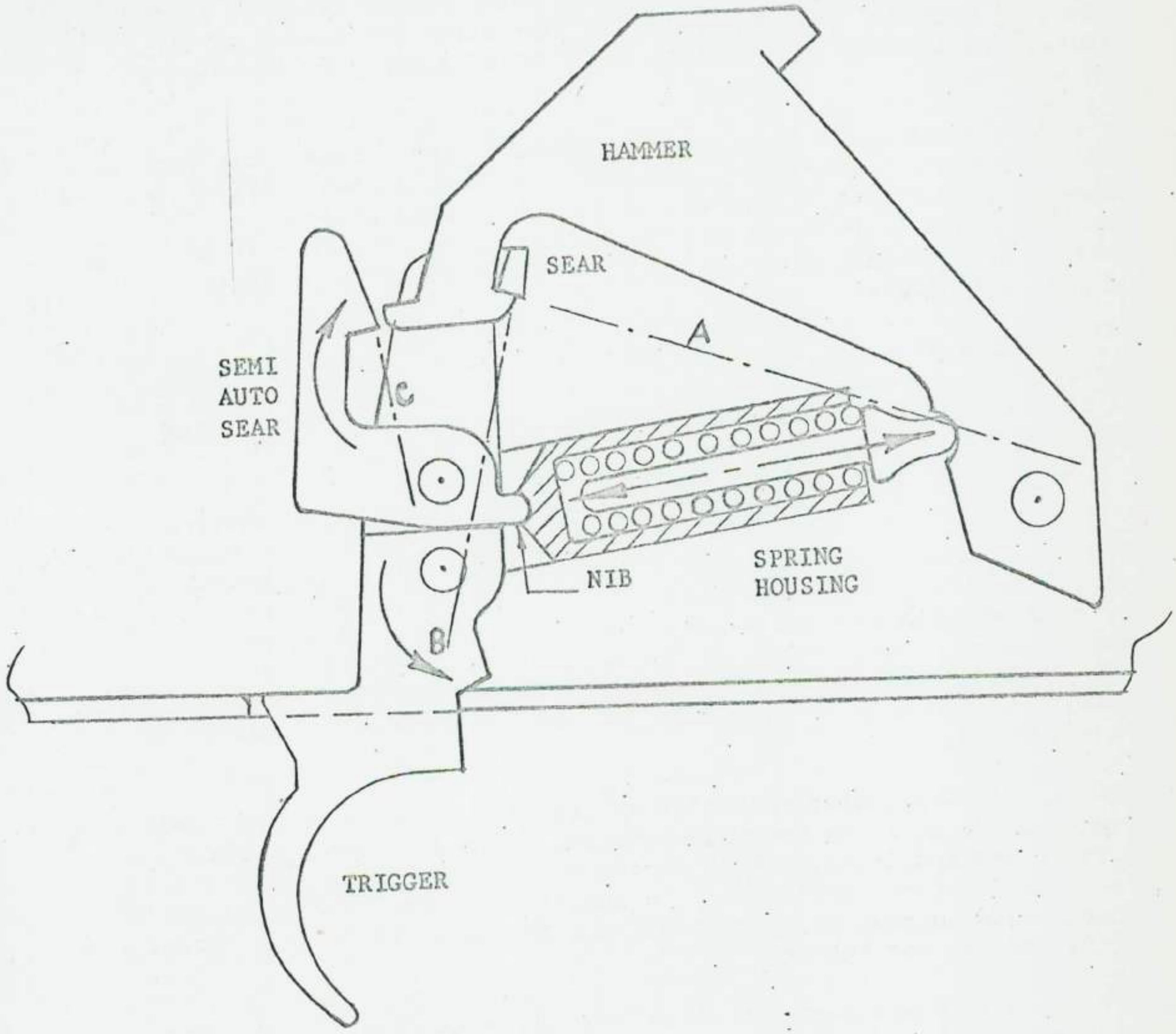
11. Hammer Spring - a plain coil spring. Spring design will be discussed in detail in the chapter on "Spring Design".

12. Safety Spring - a formed wire spring that firmly latches the safety in either the safe or fire position. Note that the bottom leg of this wire form, in the M1 Rifle, also functioned as the clip ejector spring.

13. Housing, Spring - This component is simple in appearance but performs a delicate function, that of directing the spring force on the tips of the secondary sear bits. Looking back at the trigger assembly, a force on these bits produces a short counter-clockwise moment on the trigger, and a clockwise moment on the secondary sear.

To ensure that the bits engage properly for each assembly, the housing is slotted on one side, so as to fit around the safety, thus preventing reversed assembly. Note the two small holes that provide seats for the sear bits.

In summarizing the M14 firing mechanism, it is dis-assembled only with the bullet tip, and re-assembled without any tools. It permits dirt, sand, etc., to work through the mechanism without any inside pockets for same to collect in it.



M 14/M1 TRIGGER GEOMETRY

Note that the position and engagement angles between the hammer claw and sear surfaces is a delicate balance between a self locking grip and trigger pull. The alignment plane of the sear and hammer hook, (A) extended longitudinally is just above the hammer pivot point. If it were below, there would be a tendency of the hammer to slip off.

A perpendicular plane to this surface, (B) , comes just forward of the trigger pivot point, giving the trigger a counter-clockwise moment, as required. When the secondary sear is engaged, a perpendicular to this surface, (C) , is just to the rear of the sear's pivot point, giving it a clockwise moment, as required for a self-holding grip.

Self-styled marksmen at times alter their sear surfaces in order to lighten trigger pull. T H I S I S A D A N G E R O U S P R A C T I C E, as the slope could be transformed to a creeping self-release angle, which would be extremely dangerous.

The M16 rifle firing mechanism also has all of the required functions. If the bolt is held unlocked, the bolt carrier would hold the hammer head back from falling prematurely. The hammer has three distinct notches, in different locations, for the trigger sear, the secondary sear, and the full automatic sear. The secondary sear, here called the disconnect, is unlatched by the trigger being returned to rest, while the trigger sear enters its own notch. The automatic sear is released by the bolt carrier.

At times a firing mechanism is required to perform additional functions, such as in the SPIW program, where an automatic burst counter was specified. In this, the weapon would fire three rounds and automatically stop. A counter mechanism was incorporated which would automatically trip off the sear when the third segment of its travel was executed.

Then it would re-cycle itself for the next three-round burst. If the selector was then set for semi-automatic fire, and only one or two rounds fired, then the mechanism re-selected to controlled burst fire, and the counter wheel has to discriminate this fact and re-set itself.

Machine gun firing mechanisms are quite simple, usually requiring only a primary sear and a safety. Trigger squeeze is not a significant factor, and the minimum of parts insures ease of maintenance. The most critical problem may be in sear bounce, causing run-away firing. This may be resolved by shaping the notch to a self-locking angle of approximately 5^{o}, and permitting adequate dwell travel in the notch length so that the sear has sufficient time to assert itself.

TABLE OF PRIMER ENERGY

TYPE	FT - LBS. *
Small Pistol 500	5.5
Small Piston Magnum 550	8.8
Small Rifle 400	6.0
Small Rifle Magnum 450	7.2
Large Pistol 300	7.1
Large Pistol Magnum 350	9.15
Large Rifle 200	9.2
Large Rifle Magnum 250	9.45

* Energy imparted to a piston in a test cylinder.

A uniform striker energy of .45 ft. - lb. was used for all primers.

The pressure in the cartridge case is raised several thousand psi by this primer energy.

Feeding

Feeding is the action of placing each cartridge, in turn, into the receiver at a position in back of the chamber. The forward, or counter--recoil motion of the bolt then pushes the cartridge into the barrel for firing. This very simple process is the most critical in the design and development of a weapon, as "ff" (Failure to Feed) is the most frequent type of stoppage encountered in the development testing of an automatic weapon. This is because there usually is a portion of the feed process in which the cartridge is not controlled This may occur at the following points of the feed cycle:

A. For a magazine feed automatic rifle:

(1) Bolt over-ride may occur because the spring-biased stack of rounds was surging and the base of the top round bounced below its pick-up position.

(2) In pushing the top cartridge forward, the cartridge rim rides along under the magazine lip. At the end of this lip, the cartridge base is suddenly released upward. If the cartridge is not controlled in the barrel mouth, receiver, and/or bolt face, a jam may occur.

B. for a belt-fed machine gun:

(1) The leading linked round can bounce about in the feed tray, unless it is controlled, usually by depressor pawls and a holding pawl.

(2) At the point where the round is freed of the (push-through) type of link, the round base is subjected to an uncontrolled transverse force.

(3) When the cartridge is being pushed between the feed tray and the chamber, it may be free to bounce laterally, so that the bullet tip can jam against an inside contour or corner, jamming the action in a half-open position.

C. Sprocket-fed machine guns:

(1) The linked round entering the feeder should be controlled so that it does not bind upon entering the sprockets. This is because of some tendency of the belt to "fishtail". Control is facilitated by pulling on the center-of-gravity of the linked round.

(2) The cartridge, in being "passed off" from sprocket to bolt face is usually held in a momentary "finger-tip" position, and may be subject to "bounce" caused by vibration.

These "danger zones" are aggravated by external vibrations of the vehicle, such as truck, tank, helicopter, fixed wing aircraft, tripod mount, or human shoulder. The importance of maintaining mechanical control over every phase of the cartridge traverse cannot be over-emphasized, if a low stoppage rate is desired.

Usually, the cartridge feed stroke into the chamber is complicated by the fact that the mouth of the magazine or feedway is not in line with the bore, so that the cartridge must be fed in compound directions. In this case a ramp is constructed between the feed tray or magazine and the chamber.

For best results, the cartridge should be fairly close to the chamber, and the ramp angle, particularly for magazines, should not exceed 40°. Machine guns are usually found to have longer ramp distances, so the ramp slope is shallower. A series of layouts should depict, in small increments, the cartridge motion. In doing this, avoid any inside corners in the receiver or barrel extension in which the bullet tip may stub if the bullet deviates from the predicted path.

In feeding a machine gun, the cartridge is usually placed in the feed tray in the vertical axis of the bore, but in a rifle, because of the double stack magazine, the cartridges are alternatively to the left and right center. Layouts of bullet ramps from magazine to chamber should reflect this compound angle.

The starting point for a weapon design is in placing the cartridge in a favorable position for feed. This affects the bolt design, bolt lugs, barrel extension, and receiver. Too often the locking system is designed first, then the feed system designed afterward. They should be integrated. The radial distance from bore to cartridge should be minimized. This is why multiple chamber revolver guns are more reliable: because the cartridge is already in line with the bore for the critical ramming step.

The Browning and Maxim families of machine guns draw the cartridge rearwardly out of the belt in recoil and transfer the cartridge to the bore axis in c'recoil. This places a greater demand upon the feed system, as it requires that the round is already in belt lead position during the recoil stroke. As a result, the belt feed stroke must occur during the counter-recoil motion. Belt feed is the advancing of the entire belt (or portion thereof) one pitch distance. The feeding of the belt is the portion of the weapon cycle that requires the greatest amount of energy to perform efficiently. Paradoxically, the weapon has more energy in its recoil motion than in counter-recoil, therefore, available energy for the Browning or Maxim system is limited. In the M60 machine gun, the belt feed occurs during recoil motion of the bolt.

Revolver-type machine guns such as the M61 and XM134 have continuous motion feeders, as does the XM140, which is ideal in this respect: <u>The Feed System Should Be Designed To Feed During A Maximum Portion Of The Weapon Cycle</u>.

Since the belt feed requires the greatest amount of work in the weapon cycle, this work should be distributed over the widest time frame, in order to reduce peak loads.

In Chinn's "Machine Gun" Vol. IV, a wide variety of feed mechanisms is shown in an excellent array of sketches. However, study only those mechanisms that were in an actual weapon. (See Appendix A.) Those designs designated as someone's "Patent" should be disregarded, since feasibility is not proven. Before designing a feed system, it is well to review these mechanisms, conveniently displayed as to mechanism group, such as lever type, cam operated, sprocket type, etc. Note that, particularly with the lever type, the belt feeding is accomplished, in many designs during both recoil and counter-recoil motions. A common form is a dual lever (scissors-type) that provides 35% to 50% of the feed during recoil and 50% to 65% during counter-recoil, depending upon design philosophy.

The lever form or cam shape that controls feed velocity should be designed to produce a minimum belt velocity at all times. The ideal form of constant minimum velocity feeder is incorporated in the XM140, XM134 and M61; the feeder operating during 100% of the weapon cycle time. The resultant velocity diagram then is constant velocity. This philosophy is unique to the design of feed cams, and will result in minimizing belt surge, belt loads, and belt separation or breakage. Minimum constant velocity produces a more "even" pull on all of the rounds in the belt. Other cam forms would produce higher loads on subsequent rounds in a bèlt. That is, during a feed stroke, not all rounds are pulled at once. This varies from shot to shot and depends upon the condition of the belt in, and adjacent to, the feed tray.

Initial acceleration is not as high as supposed, because of the flexibility of the belt, feed levers, cam path backlash, etc.

The link should not be too stiff, then, in order for the linked belt assembly to function somewhat as a shock absorbing spring during any peak loading. The formula for belt load (max effect of acceleration) is given as:

(1)

$$P = W + \left[\left(W/2\right)^2 + \frac{W_T\ K}{g}\ V^2 \right]^{1/2}$$

W = Total weight of belt

W_T = Weight of linked rounds being accelerated

K = Spring rate of linked belt (lb./in.)

V = Maximum velocity of belt

g = 32.16

Note how the force, which is the feed mechanism loading factor, increases with Velocity squared, and with K, belt stiffness.

The formula is developed as follows:

Consider the energy imparted to the ammunition belt:

$$(2)\quad E = \frac{W_T V^2}{2g} + W\,(X_d - X_s)$$

X_d = dynamic deflection of belt

X_s = static deflection of belt

Also:

$$(3)\quad E = K X_d^{\,2}/2 - K X_s^{\,2}/2$$

therefore:

$$(4)\quad \frac{W_T V^2}{2g} + W X_d - W X_s = \frac{K}{2}\,(X_d^{\,2} - X_s^{\,2})$$

so:

$$(5)\quad X_d^{\,2} - X_s^{\,2} = \frac{W_T V^2}{gK} + \frac{2WX_d}{K} - \frac{2WX_s}{K}$$

transposing;

$$(6)\quad X_d^{\,2} - \frac{2W X_d}{K} = \frac{W_T V^2}{gK} + X_s^{\,2} - \frac{2WX_s}{K}$$

completing the square:

$$(7)\quad X_d^{\,2} - \frac{2WX_d}{K} + \left(\frac{W}{K}\right)^2 = \frac{W_T V^2}{gK} + X_s^{\,2} - \frac{2WX_s}{K} + \left(\frac{W}{K}\right)^2$$

factoring:

$$(8)\quad \left(X_d - \frac{W}{K}\right)^2 = \frac{W_T V^2}{gK} + \left(X_s - \frac{W}{K}\right)^2$$

Taking the square root:

$$(9)\quad X_d - \frac{W}{K} = \sqrt{\frac{W_T V^2}{gK} + \left(X_s - \frac{W}{K}\right)^2}$$

Now, by definition, static deflection equals $W/2K$.

therefore; $X_s = W/2K$

$$(10) \quad X_d = \frac{W}{K} + \sqrt{\frac{W_T V^2}{gK} + \left(-\frac{W}{2K}\right)^2}$$

The load developed is simply the dynamic deflection times the spring rate of the linked round.

$$(11) \quad P = X_d K$$

Therefore, combining the above two formulas:

$$(12) \quad P = W + K\sqrt{\frac{W_T V^2}{gK} + \left((-)\frac{W}{2K}\right)^2}$$

$$(13) \rightarrow (1) \quad P = W + \sqrt{\frac{W_T K V^2}{g} + \left(\frac{W}{2}\right)^2} \quad \left\{ \begin{array}{l} \text{WORKING} \\ \text{FORMULA} \end{array} \right.$$

This load should be considered in calculating the strength of every component (the feed mechanism) such as shear strength of feed pawl pivot pin, compressive stress between roller and cam, etc.

The pressure required to dent the cartridge case should also be considered, and may determine the required bearing area between feed pawl and case. (esp. caseless ammo.)

$$P_{cr.} = \frac{E h^3}{4(1-\mu^2) R^3}$$

E = modulus of elasticity of cartridge brass

h = wall thickness of case

u = Poisson's ratio, = .28

R = outside case diameter at contact

Link Design

The link design is integrated with the feed system, and links are generally classed as side-stripping, push-through, or pull-out, depending upon the type of motion used to separate the round from the link. There is an excellent display of link designs shown in Vol. IV of "Machine Gun" by G. Chinn, page 285.

Links should have the following characteristics:

(1) Pull the cartridge at the linked round center of gravity

(2) Lock onto the cartridge, so as to prevent "walk-off" or a progressive vibrating off its position on the cartridge (axially)

(3) Be tight enough to grip the cartridge so that during maximum belt pull forces, the round does not loosen (radially)

(4) Not be tight enough to prevent easy round stripping in feeding

(5) Belt be flexible in a butt fan radius

(6) Be flexible in a nose fan radius

(7) Be flexible in twist

(8) Be able to fold in parallel layers in an ammo can. (front & back)

(9) Must permit the layers to stack evenly.

(10) Linked belt must not disintegrate when armorer handles belt outside of weapon and chuting (Infantry load)

(11) Links must disintegrate upon ejection from weapon

(12) Linked round must not have protrusions that cause belt to snag in ammo can, chuting, feeder, weapon, or ejection chute

(13) Links must be re-usable (for development test purposes) without significant change in characteristics upon re-use

(a) stripping force

(b) gripping force

(14) Capable of long term storage without change in gripping force and without corroding.

(15) Cost should be low

(16) Adaptable for either left or right hand feed

The link, in pulling on the center of gravity, should have cartridge gripping sections that straddle the grip area, without being loosened as belt pull increases. The gripping force between link and round is calculated by treating the gripper as a cantilever beam extending from a fixed end. Apply the formulae for stress and deflection, the deflection being the interference between cartridge and free link.

$$S = \frac{Mc}{I} \quad \text{and} \quad \delta = \frac{W\ell^3}{3EI}$$

I = moment of inertia of tab section

This is a good approximation, with more detailed analysis given in Roark's "Stress and Strain" on curved beams.

Some links have been designed with separate "Cap" sections so that the belt pull forces do not tend to spread open the link. The cap is stripped off the link during entry into the feeder.

The M75 weapon utilizes an extremely lightweight link in relation to the cartridge size because the link is a closed loop encircling the cartridge. In this way, belt loads tighten the grip between link and round, rather than loosen it.

The linked belt pitch distance should be as small as possible for maximum efficiency in feeding, yet be long enough to permit parallel stacking of rows. (particularly at the row ends) A short pitch distance means a short stroke of the feed system (levers, cams, etc.) and reduces the weapon profile. For links that pivot on a succeeding cartridge, the pitch distance is approximately 1.3 times the cartridge body dia., while for links that pivot in the hooks between rounds, the pitch distance is approximately 1.4 to 1.45 times the cartridge body diameter.

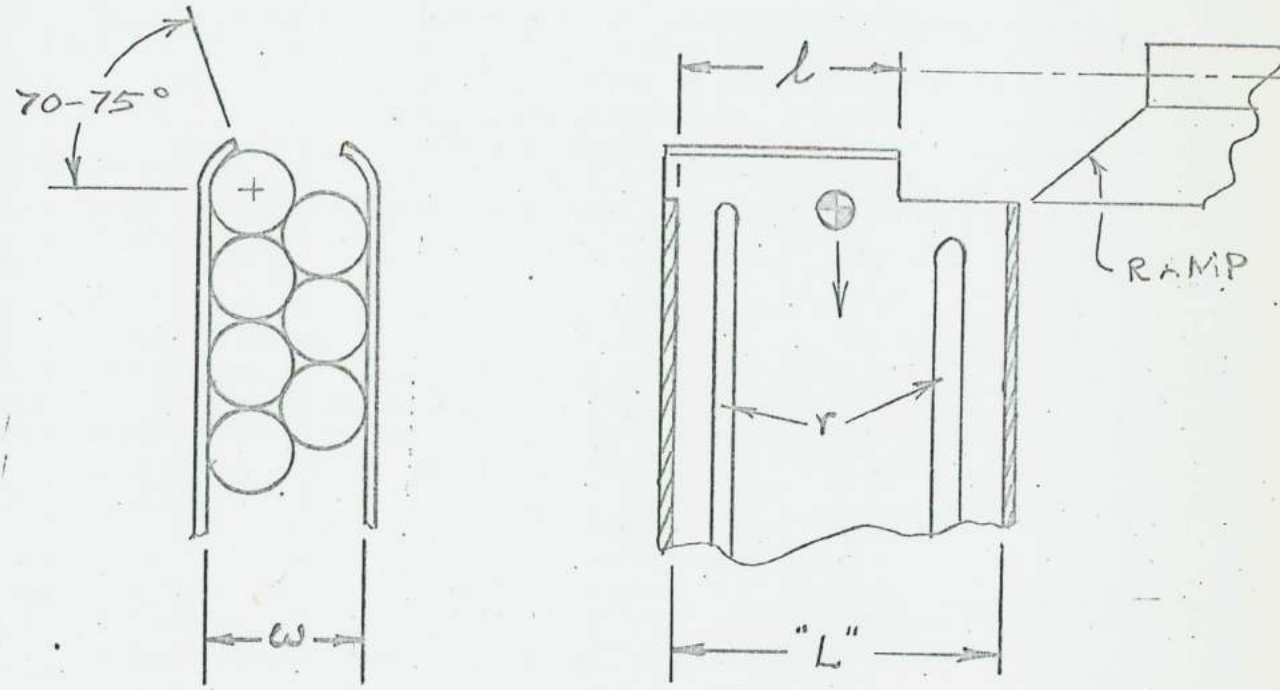
Linkless helical drum type feeders are also employed in some weapons such as the Lewis machine gun (radially stacked drum) and Thompson sub-machine gun (axially stacked drum). Large capacity linkless feed drums are also used (M61 and XM134) which control several thousands of rounds with a conveyor system. This item is a specialized subject of its own.

A recent problem has evolved in which electric primed ammunition becomes over-sensitive in the presence of electronic equipment over a period of time. To prevent the primer from forming an induced electro-magnetic field, a tab of the link is incorporated as an integral shield.

Magazine Design

The magazine unfortunately was conceived as a cheap, throw-away, one-time-use item, and, accordingly is constructed of rather light gauge sheet metal, which is subject to damage in rough handling. Unfortunate, because the magazine lip is one of the most critical surfaces in the entire rifle. Attempts have been made to incorporate a fixed machined lip as an integral part of the receiver in order to eliminate this weak point in the system.

The magazine should be as high as possible into the receiver, commensurate with bolt configuration, in order to optimize cartridge ramping. The magazine feed lips should wrap around the top cartridge so as to prevent cartridge pop-out in handling. This is usually an angle of 70° to 75° as depicted in the following sketch.



The magazine lip length "ℓ" should be long enough to contain the cartridge center of gravity, yet short enough so that the following cartridge, or follower, can give the base an upward moment upon cartridge release, preventing jamming between bolt face and ramp area.

The inside magazine width, "W" for a double stack, should be approximately equal to the cartridge diameter plus cosine 30° times cartridge diameter. If the width is greater than this circle contact, then greater forces are transmitted to the magazine side wall, causing hard feeding and subsequent binding. If the width is shorter, the magazine depth becomes unnecessarily longer.

The magazine side walls should have longitudinal ribs "r" as a cartridge bearing surface to permit easier feeding. Sand, grit, and other matter will then collect harmlessly in the void between the ribs. Secondly, the ribs reinforce the sheet metal wall, greatly increasing its section modulus against deflection.

Inside length "L" should be minimized in order to reduce impact of the bullet tip during automatic fire.

In summary, when designing a magazine, first study other successful magazines in detail, in order that all required characteristics are incorporated. Magazine spring design will be discussed in the chapter on "Spring Design".

Magazine depth must be carefully controlled. This should not be so from an engineering viewpoint, but for practical reasons is necessary. For example, in a 20 round magazine, if the user can squeeze in a twenty-first round, he will do so. Then, when this is inserted in the rifle it may cause hard stripping, or, in bearing tightly against the bolt, may cause a short recoil, or may spread the magazine sidewalls, deforming it enough to cause subsequent jams. The accumulated tolerances of max/min cartridges plus magazine tube tolerances are involved.

The magazine follower and spring is usually designed so that the load is concentrated approx. 35% of "L" from the cartridge base. This is to prevent cartridge base drop during surging of the stack. Also, it helps in lifting the full stack properly, since the bottom cartridge will be inclined approx 15° due to body taper accumulation. For a thirty round magazine, the accumulated incline is so high that the magazine body usually is curved to compensate for this inclination.

Ammunition Storage and Boosters

Weapon systems on aircraft usually have such a large ammunition supply, or such a long, torturous path from ammunition can to weapon, that the weapon feed mechanism cannot pull the load without reducing firing rate or causing excessive wear of cam surfaces and components. For this reason, boosters are used; that is, auxiliary feed devices usually in the form of a motor driven sprocket.

The main problem is in correlating booster delivery with weapon rate of fire. This is not simple, since weapon rate of fire is not constant, hence a sensing device must be incorporated that will control the booster speed and delivery.

A. One method is to used a mechanical pitch sensor. When ammunition is bunched up between booster and gun, the belt pitch will shorten and a mechanical device will stop the booster. As ammunition is used, the belt will stretch out and increase pitch, so that the sensor switches the booster on. The response has to be quick, because as the pitch is increasing, the weapon is firing; the slow inertia of the booster motor and drive may not supply the next round that the gun demands. Consequently, belt pull will be high.

B. The same approach as above, but based upon belt catenary. Catenary is the droop of a length of belting causing a switch to close, controlling the booster.

C. A dual speed booster, where the sensor switch does not shut off the booster, but switches in a resistor that slows the booster slightly. Here response time is quick, because the booster does not have to build up speed from a zero start.

D. Time delay may be a critical factor, as in intermittent fire, when firing stops, the booster has sufficient inertia to feed a fraction of a round more. A few spasmodic bursts will soon "overfill" the feed tray causing a gun jam. A short-time delay control is incorporated in the booster circuit on situations of this type in order to clear the pile-up.

E. A round counter mounted on the feeder that pulses a feed station on the booster sprocket.

Spring Design

In designing a spring, usually a helical compression spring, for some part of a weapon mechanism, it usually happens that only a minimum of space is left for the spring; that is, if the spring is not carefully designed, it will be vulnerable being over-stressed in service.

Firstly, the mechanical requirements of each component, then the strength and durability levels must be satisfied. Secondly, the load requirements of the spring are specified, and, finally, the amount of spring stroke is determined. For the reason that the stress levels of weapon mechanism springs are high, the material to be used is music wire, which has elastic limit stress values of 150,000 to 180,000, depending upon the wire diameter. The smaller the wire size, the greater the permissible yield stress. Springs subjected to high temperature levels are usually stainless steel, such as "Elgiloy". Common spring steels are reliable when stressed up to 80,000 psi at temperatures of 350°F - 400°F. or less.

The use of square, rectangular, or other than round, wire shapes should be avoided for several reasons. Firstly, the stock is not readily available to most vendors, therefore the cost will be higher. Secondly, these special shapes are not produced in tonnage, as compared with round wire, hence have not had the refining development which has been given to round wire. The resultant yield strength is not equal to that of good grades of round wire.

In specifying spring wire, a number of standard gauge series are commonly used, therefore, in order to avoid error, the wire size should be specified in decimals.

The starting point for designing a coil spring is to determine the outside wire diameter permissible, commensurate with available space, and the minimum operating height of the coil. Then, calculate the load and stress levels of a sample coil that will have a D/d ratio of 6.5 to one, and a number of coils that permit only 3 to 5 thousandths inch between coils. "D" is the mean coil diameter, while "d" is the wire size. The compression of a coil spring is technically, not "compression", but "torsion". That is, the coil of wire is being subjected to a torsional, or twisting, force. As the ratio of D/d decreases, an additional shear stress acts, and this becomes dis-proportionally higher, as the coil becomes tighter. A D/d ratio of 6.5 is quite satisfactory, but this may be adjusted between 5.5 and 7.5, depending upon load desired and resulting stress. In no case should the D/d ratio be lower than 4. The component of additional shear stress due to curvature is known as the "Wahl" factor, and is well covered in any standard work on spring design.

Typical values of "K", the Wahl factor, are:

D/d	"K"	"R"
9.	1.15	1.07
7.	1.21	1.1
6..	1.25	1.12
5.	1.32	1.16
4.	1.4	1.2
3.5	1.47	1.23

The calculated spring stress is multiplied by "K". However, in gun design, a spring does not cycle for hundreds of thousands of cycles, therefore, the level of fatigue does not approach that of other fields, such as valve, automotive, etc. Therefore the factor "R" is more realistic, being midway between 1.0 and "K".

The basic formula for stress in torsion for round wire is:

$$S = \frac{Mc}{I}$$

in which $M =$ torsional moment $= PR = \frac{PD}{2}$

$C = d/2$
$I =$ polar moment of inertia $= \frac{\pi d^4}{32}$

so $S = \frac{8\ PD}{\pi d^3}$

However, this static stress is not a positive indicator of the dynamic performance of the spring. Impact loading can cause a much higher set in springs than solid height stress. A testing machine with the same deformation velocity is required.

The other principal spring design formula is that for rate of deflection, or spring rate

$$R = \frac{G\ d^4}{8ND^3} \text{ lb./in.}$$

G = torsional modulus of rigidity, of steel wire, or 11.5×10^6

Note that the Load/Stress formula does not contain a factor for number of coils. This is because the coils are in series and the load required to deflect one coil to solid height is the same required to deflect the entire spring to solid height.

In designing a spring, the formulae for stress and rate are worked together. For example, if a lower rate spring is desired, either "d" is decreased or "D" is increased. This, in turn, reduces the peak load for a given stress, each to a varying degree. Use or a commercial "slide rule" type of spring calculator permits rapid trials of various combinations.

The spring rate should be as low as reasonably attainable. One reason is that stress and stress range govern the life of springs. The wider the stress range, the quicker the spring will fatigue. Comparatively high stresses can be used where the working range is short, and reducing the spring rate accomplishes this. Spring rate is reduced by maximizing the number of coils. This, in addition, reduces the deflection range of coils during spring surge. That is the free distance that individual coils of wire can deflect. For this reason, the spacing between coils, at minimum operating height, is .004 inch, and may even be reduced to .002 inch.

The factor "N" is the active number of coils. That is, springs with squared, or squared and ground ends have one inactive coil on each end.

Springs are generally wound right hand, but when one spring is inside another, they are wound opposite hand, in order to prevent clashing, or pinching, of coils.

As a spring is compressed, the outside diameter increases, due to the closing of the coils. The formula for calulating diameter increase is:

$$D_1 = 1/2 \sqrt{P^2 + 4\ D^2 - d^2}$$

D = mean dia.
D_1 = mean dia., solid height
p = pitch at free length
d = wire dia.

The spring natural frequency should also be checked, and should be at least 11 to 13 times higher than the weapon rate. If not, the spring will surge and the stresses greatly augmented. Some surging can be absorbed by having a few closed coils built-in.

One general formula for determining spring frequency, n, is: $n = 826{,}500\ d/ND^2$ cycles/min.

The surge wave of a spring, when loaded, requires a specific time to travel from one end to another. Velocity is constant for a particular spring and does not depend on the load or velocity of the load. However the magnitude of the wave is affected by the velocity at which it is struck.

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The equation for the surge-wave velocity "C" in a compression spring is:

$$C = \frac{d}{D}\sqrt{\frac{G\ 2}{2\gamma}} \qquad \text{(in./sec.)}$$

d = wire dia.
D = mean Coil dia.
$G = 11.5 \times 10^6$
$\gamma = .283$ lb./in.^{3}

thus $C = 88{,}600\ (d/D)$ in./sec.

the surge time T:

$$T = \frac{3.17\ D\ N}{C} \text{ sec.}$$

or $$T = \frac{N\ D^2}{27{,}900\ d} \text{ sec.}$$

When the ratio of the load vs. the spring weight is reduced to 4/1, then the spring weight begins to affect the calculation for energy stored in the spring. The load is then modified by adding 1/3 the weight of the spring.

The limiting factor in the velocity that a spring can drive a component is not the energy stored, but the rate at which this can propagate down the turns of the spring. This velocity is related to speed of sound in the spring material and is limited by the allowable stress in the material.

This ultimate velocity is:

$$V_{max} = \frac{S}{131} \text{ in./sec. (steel)}$$

This is about 115 ft/sec.

At times, the behavior of some springs cannot be predicted, but can be observed by using a stroboscope.

Of course, proper heat treatment is necessary, and modern equipment in most facilities assures this, but if springs are over-heated grain structure will be coarse and the fatigue life poor. One simple method of determining proper spring design and heat treatment is to close a spring to solid height three times and measure the free length. Any "set", or permanent deformation, should then be apparent.

Compression springs must be guided, either by means of a rod inside the coil, or the coil inside a hole, if the free spring length is 4 or more times greater than the mean coil diameter. Otherwise, buckling would be likely to occur.

When dual (inner and outer) springs are used, one spring is wound right-handed while the other is wound left-handed, as mentioned previously, but the outer spring should be designed to carry approx. 2/3 the load, the inner 1/3.

When designing odd shape compression springs, such as a magazine tube spring, two methods may be used. The first, is to take each segment of a single coil and treat that as a cantilever beam with a flexible support, and summing the load and deflections for the four segments (of a rectangular coil) of the single helix. The load will be the same for the total spring, with the rate varying inversely as the number of coils. This method is tedious, but a simpler method is to add the total circumference of one coil of wire and convert that to a round coil of the same circumference. Use the same wire dia., number of coils, etc, and solve for loads and stresses. This result will be within 5 - 10% of actual loads and stresses.

Extension Springs

In designing extension springs, the factors are the same as in the design of compression springs except that extension springs can be wound tightly with an initial tension between the coils so that a load must be applied to separate them.

The types of ends vary widely, depending upon loop, hook, or end desired.

Extension springs are not usually found on weapon mechanisms, because the stress concentration where the end coil is turned to form the loop, or hook, is vulnerable to breakage. Also, for a very practical reason, when an extension spring is being assembled and/or disassembled, there is no positive stop as there is in compression springs (solid height) and thus an extension spring can be very easily over stressed in handling.

A special form of a compression spring is a so-called "garter spring" in which a close-coiled spring is assembled into the form of a ring. This was done in a rifle grenade launcher where the spring acted to retain the grenade. But after it was set into position, a twist buckled all the spring coils so that they locked the grenade tube to the muzzle device. This caused a serious accident, which pointed out how dangerous that type of spring could be in practice.

Torsion Springs

A torsion spring can also be quite useful in weapon mechanisms, being a coil of wire subjected to torque, that is a wind-up of the coil, as in a common "rat-trap" spring. The ends may be configured in a variety of styles, and this is usually useful in reaching out to remote distances to perform load, or retaining, functions.

It is most useful in rotating components, or to cushion shock on rotating parts.

A torsion spring should always be actuated in a direction that would tend to "wind up" the coil, or reducing the diameter of the coil. Otherwise, the end coils would tend to bend outward, carrying all the load. Also bear in mind that when a torsion spring is loaded, its coil length increases, as one full turn makes the coil one wire diameter longer. Allow clearance for this increase in length or the coil will bind or break. The spring leg will also shorten, and if not contained, will snap free.

There is almost no limit to the possible configurations of a torsion spring, but bear in mind that designing shapes beyond those commonly tooled increases cost astronomically.

The specifications for a torsion spring should include the load as a torque in inch-pounds, as well as the mechanical dimensions, including right or left hand coiling. If possible, the spring should be reversible in assembly.

Two formulae required to develop a torsion spring are as follows:
(1) Equation for deflection (round wire):

$$M = \frac{E\,d^4\,T}{10.8\,D\,N}$$

M = Torque (in.-lb.)
N = Number of coils of wire
T = Number of turns spring is torqued
E = 30×10^6
D = Mean diameter of coil

(2) Equation for stress: (round wire):

$$S = \frac{32\,M}{\pi d^3}$$

M = Torque (inch - lb.)
d = wire size

As in coil springs the ratio of D/d must also be considered, since the stress for a straight beam must be modified for the curvature of the coil. For torsion springs, the stress correction factor (K) increases sharply as the D/d ratio is reduced below 2.5 to 3.0.

In general, the lower the maximum stress, and the shorter the range of stress between initial and final working positions, the longer the service life.

Flat Springs

At times, limitations in space prohibit the use of coil springs, and a flat "strip" type of spring may be utilized. Flat springs have the added advantage of being able to perform a combination of functions.

Special blanking and forming tools are used to produce this item, and a number of specialty vendors are able to produce these quite economically.

Straight carbon steels are used in these springs, with .70 - .80% carbon lending itself better to sharp bends, while the .90 - 1.05% carbon exhibiting higher elastic limits. In some cases alloy steels are also used. Sharp inside corners should be avoided at all times, with holes punched at the ends of slits.

The controlling factor on the loads of small flat springs is the thickness of the material, as the deflection formula shows the load to be a function of the thickness of the material cubed.

Calculations of flat springs are similar to those of beams, and the comparable form should be selected, such as cantilever with fixed end, free end, etc., or beam with distributed load, or concentrated load, etc.

For example, consider a cantilever beam clamped at one end with a concentrated load at the other.

The formulae for stress and deflection are worked together, such as:

$$S = \frac{Mc}{I} = \frac{6\ PL}{b\ h^2} \qquad h = \text{thickness} \quad b = \text{width}$$

$$F = \text{deflection} = \frac{PL^3}{3EI} = \frac{4PL^3}{Ebh^3}$$

Of course, stress raisers, such as holes and sharp inside corners, must be minimized, particularly where the base of the spring (max. stress) is bending.

Bends in the spring also affect stress, generally according to the following table:

Inside radius of bend			Stress factors "k"
.5 X metal thichness			2.0
.75	"	"	1.66
1.0	"	"	1.5
1.25	"	"	1.4
1.5	"	"	1.32
2.0	"	"	1.25
3.0	"	"	1.16
4.0	"	"	1.12

Theoretically, the shape of a flat spring can be modified to improve its efficiency, such as the uniformly stressed spring and the built up (leaf) spring, but this advantage is offset by production costs.

Belleville Springs

This form of washer-type spring is quite common in gun mechanisms, particularly buffers, and have the advantage of being quite compact, the range of motion being quite short.

A series of empirical curves are used to calculate this type of spring, and may be found in most up-to-date spring design manuals. (Associated Spring Handbook). Otherwise, one would have a formula with 4 variables, resulting in extensive work.

A typical solution of a Belleville Spring configuration is as follows:

$$P = \frac{E\,f}{(1-\sigma^2)\,Ma^2}\left[\left(h-\frac{f}{2}\right)(h-f)\,t+t^3\right]$$

P = load
E = 30×10^6
f = deflection = .004
σ = Poisson's ratio = .3
h = free height minus thickness = .005
t = thickness = .084
O.D. = .730
I.D. = .380
a = 1/2 O.D. = .365
M = Constant taken from chart relating O.D./I.D. ratio to stress constants C_1 and C_2
C_1 = 1.21
C_2 = 1.35
M = .675
$\therefore$ P = 867 lb.

Likewise, the stress equation is:

$$S = \frac{Ef}{(1-\sigma^2)\,Ma^2}\left[C_1\,(h - f/2) + C_2\,t\right]$$

$\therefore$ S = 172,000 psi

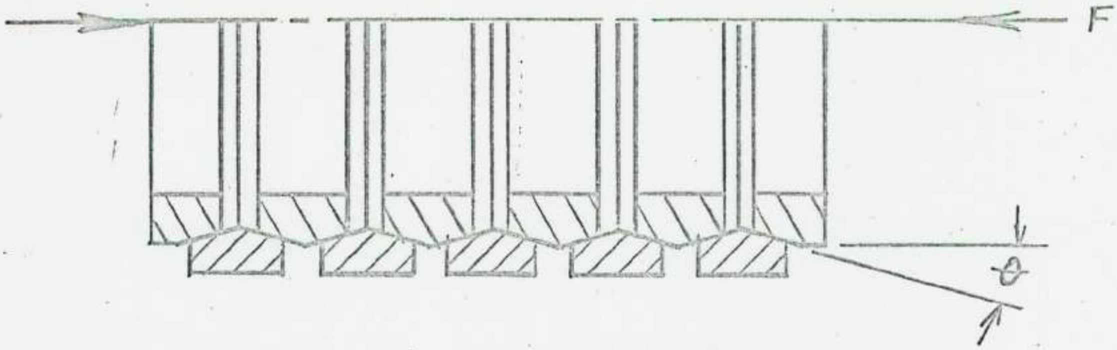
The spring rate of Belleville springs is controlled by specifying the method of stacking and the number of elements.

Belleville spring packages are usually used as buffers, to absorb shock, or to take up slack, somewhat as a spring washer. In a commercial application, Belleville type springs are used as pressure disks for power brakes, so the range of application is wide.

As may be expected the thickness of the plate greatly affects the load characteristics, also the ratio of h/t; that is, the free height vs. the thickness. This ratio affects the load deflection curve, and for certain curves, the springs have a near zero rate for a portion of their travel. Thus, changes in assembled heights (tolerances, wear, etc.,) would not change the load.

Ring Springs

Ring springs have also been used extensively in ordnance, particularly as barrel buffer springs for short-recoil operated weapons. Ring springs consist of a series of inner and outer elements that utilize friction as the means of absorbing load. In compression, the outer elements expand while the inner elements contract, with high frictional loads bearing on the mating surfaces.



Ring springs inherently have a high load capacity for their size and weight and can absorb shock with low recoil.

The spring elements act as inclined planes through the angle θ. The force F is uniformly distributed throughout the circumference of the ring. The resultant force acting on the wedging surface of each sector can be divided into two components, one normal to the surface Fn and the other tangential (friction) equal to Fn times u (coef. of friction)

Charts of ring spring compression constants and recoil constants in terms of taper angles and various coefficients of friction (usually from (.10 to .18) have been prepared and include taper angles of from 10° to 30°.

The axial load is less during unloading than during loading, of course, but the radial loads are the same.

Some recommended proportions for ring spring packages are:

(1) The compressed height should be at least 4 times the deflection.
(2) The ring height should be 15 to 20% of the outside diameter.
(3) The outside diameter should be as large as space permits.
(4) Normal ring taper is approximately 14° (θ)
(5) Allowable stresses are usually 160,000 psi for steel not machined after heat treatment and 200,000 psi for those machined after heat treatment. The ring spring can continue to function even though several of the rings are broken.

The lowest capacity commercial ring spring is in the order of 2 tons.

Another variation of the ring spring is the split ring spring in which each ring is slit on one side. The spring rate is greatly decreased, while the frictional action is retained. The contour of the spring is modified, so that the ring thickness is maximum 180° from the slit and tapers toward each side. The spring package is then arranged with alternating slit and thick sides.

Stranded Wire Springs

When an axial load is applied at the ends of a stranded helical spring, the material forming the helix is subject to a twisting moment. In this respect the stranded spring is not essentially different than a conventional spring (helical) made from a single homogeneous wire element. The outstanding characteristic of the stranded spring is an inherent tendency for the damping of high-velocity displacement of its coils, a characteristic not shared with the conventional spring. The damping in the stranded spring is due to a binding action existing between the twisted wires in consequence of the twisting moment acting on the strand.

In the stranded spring it is essential that the helix of the strand be opposite in direction to that of the coils of the spring. An applied load causes a twisting moment which tends to cause a wind-up of each helically formed wire. The wires of the strand are in contact with each other even in the unloaded state; and since there can be no appreciable wind-up, binding between the wires results. However, if both the strand and the coils of the spring have the same turn of helix, the twisting moment tends to unwind the strand, in which case the binding action is lost and the spring deflects as though it had a subnormal value of shear modulus.

For the purpose of load-deflection computations the stranded wire spring may be resolved into as many partial springs acting in parallel as there are wires in the strand. The rate of deflection R may then be determined by the following formula:

$$R = Kn \frac{Gd^4}{8D^3N}$$

K = a factor = 1.05 for 3 wire strands
n = number of wires in the strand
G = shear modulus = 11,500,000
d = diameter of wire in the strand
D = pitch diameter of spring coils
N = number of active coils

Stranded springs usually have one coil closed at ends but the ends cannot be closed as effectively as they can in conventional springs. For such unground end construction the number of active coils "N" can be estimated as the total number of coils less 1.2.

An accurate computation of the numerical value of the shear stress in the stranded spring is a complicated affair. However, if the spring is resolved into partial springs acting in parallel, as is assumed for load-deflection computations, an average value of the shear stress "S" may be obtained for each partial spring from the following formula:

$$S = \frac{Gdf}{D^2N}$$

G = shear modulus
d = diameter of wire in the strand
f = deflection of the spring
D = pitch diameter of spring coils
N = number of active coils

For a given application in which the spring coils have a high velocity of displacement, springs which have relatively low values of statically computed stress at solid compression fail earlier than springs in which the stress at solid compression is relatively high. The performance of the gun driving springs apparently cannot be predicted by stress computations based on static assumptions. Driving springs should be designed to have the lowest possible mass.

Neglecting the effect of a difference in end coil construction and in the number of inactive coils, a 3-wire stranded spring, having wire diameters which are 68% of the diameter of the wire in a given conventional spring, will have substantially the same rate, pitch diameter, solid height and average computed stress as the conventional spring. Maximum fatigue life will result, when 3-wire strands are used, the strand being so proportioned that the ratio of the length along the strand axis in which a single wire makes one turn, to the strand diameter is between 5.0 and 5.5.

Two wire strands have a longer fatigue life than conventional round wire helical springs; but the results were very much inferior to those of 3-strand springs. Four or more wires are not stable unless a center wire is used.

The diameter which circumscribes the three round wires woven into a strand (and in contact with each other), will be 2.155 times the wire diameter; assuming the wires to lay on the corners of an isosceles triangle. Three wire strands having a ratio of twist in the strand to the strand diameter of 5. and 5.5, it is found that the strand diameter is very close to 2.18 times the wire diameter.

For either a conventional single-wire spring or a stranded spring in which the pitch angle of the coils is not substantially in excess of 10. degrees, a fair approximation of the time in milliseconds for the wave to travel from one end of the spring to the other can be obtained from the following formula:

$$T = .0354 \frac{D^2N}{d}$$

D = pitch diameter of spring coils
N = number of active coils
d = diameter of wire

As the wave of displacement passes thru the coils of the spring immediately after firing, each element of the spring acquires the velocity of the free end upon the arrival of the wave at the element. The elements then tend to move at the acquired velocity until the motion is affected by the reflection of the wave from the fixed end of the spring. The initial reflection of the wave at the fixed end of the spring and the subsequent reflection from the free end are the causes of the extremely high dynamic stresses imparted to the spring.

According to transient wave theory the change of stress of an element of a spring varies directly as the change of velocity of the element. The damping effects of stranded springs become more effective as the velocity is increased. Stranded springs have little damping action at low velocities of displacement, and hence, compare most favorably with conventional springs when the displacement velocity is high.

It is believed that the greater damping action of stranded springs is effective in absorbing energy so that the reflected wave in a stranded spring has much lower energy content than the wave in a conventional spring. The substantial decrease in the energy content of the reflected waves decreases the dynamic displacement of the coils of the spring, proportionately reducing the stress in the spring.

The stress range is a more critical factor in the fatigue life of springs than the stress level, or the mean stress in the operating cycle.

Music wire is currently the most satisfactory stranded spring material. The fatigue life of pretempered chrome-silicon wire was distinctly inferior to that of music wire. The relatively low yield point of music wire enables it to yield and adjust itself to the strains incident to the stranding and coiling operations in making a stranded spring.

So long as the ratio of the pitch diameter of the coils to the diameter of the wires in the strand is not substantially smaller than approx. (13.) thirteen, stranded springs can be coiled automatically on standard spring coiling equipment.

As in the case of contentional music wire springs, it is essential that the stranded springs be stress relieved after coiling-heated to approx. 450°F for a minimum of 30 minutes. It is also essential to coil the stranded springs somewhat longer than finished length, and to remove the excess length of pressing the springs from free to solid height a sufficient number of times to assure that subsequent compression to solid will produce no further reduction in spring length. This pressing operation produces a beneficial residual stress pattern across the wire section in the unloaded spring; and as the operation promotes a minimum of set during the subsequent operation of the springs, it is just as vital where stranded springs are concerned as it is for conventional single wire springs.

The wave motion of the coils of a driving spring generates stresses which may exceed the stresses produced by static compression to solid height. If so, the spring will take additional set during the gun firing, especially during the first few rounds.

Shot peening increases the life of conventional single wire springs by up to 60%. Shot peening is not effective in increasing the fatigue life of stranded drive springs.

The Neg'ator Spring

This is a flat strip of coiled metal that has a nearly constant force level, at times even decreasing in force with deflection. A constant force had previously been achieved with dead weights or intricate cam or lever systems. Neg'ator is a trade name patented by the Hunter Spring Co. of Lansdale, Pa. that fabricates this item.

Principal features of this spring are:

1. Flat force - deflection curves
2. Extremely long deflections, or extensions, up to 50 times the length of the original spring
3. Ability to act without losses around corners

The important consideration in attaching a bayonet to the muzzle is the effect upon accuracy, as well as strength. Accuracy firings with and without a bayonet attached show a definite change in the center of impact, due to the eccentric mass. Close fits are of added importance. The bayonet latch should be concealed, so that it cannot be inadvertantly released when sparring. Barrel strength should be adequate to sustain severe thrust and chop loads via the bayonet.

While on the subject of muzzle and barrel, a discussion of barrel length will be followed by muzzle devices, such as muzzle boosters, flash hiders, silencers, and of most interest, muzzle brakes.

Barrel Length

The longer the barrel is in proportion to the bullet diameter, the less powder will burn at the muzzle. The .256mm Jap rifle with 31.4 inch barrel offers practically no flash and little smoke because powder burning is essentially completed before the bullet reaches the muzzle. Tests conducted of a 36 inch cal..30 barrel (M2 ball ammunition) show absolutely no flash and little smoke. Of course, a hot gun increases flash, and not even a 40 inch barrel was effective in eliminating flash, when firing long bursts.

Barrel lengths do not vary widely among commercial as well as military models. They usually conform to an original barrel length used in standardizing the cartridge. Of course, barrel length in handguns vary widely, and this is due to the intended use of the gun. Target models require a long sight radius, police models require lightness, and concealed weapons shortness. Velocities are not significantly affected, a 6 inch cal..38 barrel being 3 to 5% higher in velocity than a 4 inch barrel, and 10% higher than a 2" barrel.

For rifles, the loss in velocity (for several inches) is quite small, and can be overshadowed by other variables, such as tolerance in land and groove, powder temperature, powder measure, etc. The only reliable method of measuring effect of barrel length on muzzle velocity is to take one barrel, and using carefully measured loads of the same lot of ammunition, measure the velocity of a sample (at least 32 rounds) then cut the barrel off an inch (for example) at a time. This data should also confirm the interior ballisitcs theories formulated for that cartridge.

A typical table of muzzle velocities versus length of barrel is given as follows:

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Principal features of this spring are:

1. Flat force - deflection curves

2. Extremely long deflections, or extensions, up to 50 times the length of the original spring

3. Ability to act without losses around corners

4. High initial force

5. Ability to store and deliver twice as much energy as a common coil spring of the same initial volume.

The flat strip of metal is prestressed so that it possesses a strong natural curvature, and force must be exerted to straighten it. The spring gradient may be varied by the amount of prestress applied to each section.

The variables that affect spring force are:

1. Modulus of elasticity

2. Thickness of strip material

3. Width of strip material

4. Natural radius of curvature

This type of spring was tried as a magazine follower spring, for rifles, and as a drive spring for the T148 and T148E1 semi-automatic 3-shot 40mm grenade launchers.

Muzzle Devices

In the firing of a typical small arms cartridge, up to 42% of the potential energy of the propellant is exhausted at the muzzle. The topic of this subject will be on methods and devices utilized to either take advantage of this energy or to control it.

Items of mechanical hardware, such as front sights, bipods, bayonets, grenade launchers, etc. are routine in nature and need not be covered in detail. Grenade launchers are usually powered by blank cartridges, and the grenade weight causes the gas pressure level in the bore to be maintained at a higher level for a much longer period of time, therefore, for gas operated automatic weapons, the power delivered to the operating rod is substantially higher. In the M14 rifle this is controlled by a closure valve that blocks the gas orifice when grenades are to be launched. In the M1 rifle, the grenade launcher body includes a solid pin that opens a valve on the gas cylinder plug, thereby automatically venting some of the gases. Recoil impulse to the weapon structure, particularly the stock, barrel, and receiver is also high.

Conversely, for firing blank cartridges (without launching grenades or other missiles) the bore pressure is exhausted all too quickly to power the automatic weapon, so a "blank firing attachment" is added to the muzzle. In its simplest form, this is a muzzle cap with an orifice that restricts the outflow of gases. Blank firing attachments are usually colored bright red with a flag section visible along the sight line for safety purposes. It must be removed when firing ball ammunition, or the weapon will be severely damaged.

The important consideration in attaching a bayonet to the muzzle is the effect upon accuracy, as well as strength. Accuracy firings with and without a bayonet attached show a definite change in the center of impact, due to the eccentric mass. Close fits are of added importance. The bayonet latch should be concealed, so that it cannot be inadvertently released when sparring. Barrel strength should be adequate to sustain severe thrust and chop loads via the bayonet.

While on the subject of muzzle and barrel, a discussion of barrel length will be followed by muzzle devices, such as muzzle boosters, flash hiders, silencers, and of most interest, muzzle brakes.

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A typical table of muzzle velocities versus length of barrel is given as follows:

Barrel Length	.270 Win. 150 gr.	.30-06 180 gr.	.300 Savage 150 gr.
24	2800	2700	2670
23	2770	2690	2655
22	2740	2675	2640
21	2705	2660	2620
20	2670	2640	2600
19	2635	2620	2575
18	2595	2590	2550
17	2550	2560	2520
16	2505	2525	2490

Shotguns, 12, 16, and 20 gauge were found to lose velocity linearly by 7 feet per second per inch, from a length of 27 in. to 20 in.

While we are on the subject of effect of barrel length on muzzle velocity, it would be well to discuss other factors that affect muzzle velocity. One particular facet is the constant demand for higher performance, in this case, a higher velocity.

Consider the question, "What is the limiting velocity obtainable in the small arms class?" Of course, this means increasing the charge/mass ratio to unusually high levels. Theoretically the absolute upper limit is approx. 13,000 feet per second. However, a number of penalties would have to be paid to achieve this. (For nitro-cellulose powder, it is approx. 9100 fps.)

An experiment was conducted to determine the effect of changing the charge/mass ratio upon muzzle velocity. To assure near-instantaneous and complete powder burning the grains were ground to a fine dust. (This is normally a dangerous practice). Pressures were in the order of 100,000 psi, and for high c/m ratios, extremely small bullets were used.

Accordingly, the following table was compiled:

(Note how the ratio of powder weight increase compares with the rate of muzzle velocity increase.)

c/m Ratio	Muzzle velocity (upper limit)
.2	2600 fps
.3	3000 fps
.8	4200 fps
3.2	6400 fps
5.8	7400 fps
11.0	8000 fps
22.0	9000 fps
44.0	9200 fps

Then, taking a typical commercial cartridge to correlate powder burning rate with c/m ratio vs. m.v.,

Powder	c/m RATIO	m.v.
Fast burning	.295	2800
Med. burning	.307	2800
Slow burning	.347	2800

As a practical example of extreme velocity effects, a classic example is the so-called Paris Gun of World War I. The Germans were bogged down on the Hindenburg line 75 miles from Paris, and developed three samples for the purpose of spreading panic among the civilians. These were reconstructed 15 inch 45 caliber artillery pieces tubed down to 8.26 inches. It is reported that the projectiles were waist high and the powder bags per shot twice as high as a man. The gun was set at 50^{o}, fired at a muzzle velocity over 5200 fps and went about 12 miles up, so that approx. 3/4 of the trajectory was in near-vacuum.

One gun blew up, and the other two soon wore out, to be re-barrelled. A total of 200 rounds were fired.

Summarising the three methods of achieving high velocity:

(1) Use of light projectile: This is inefficient due to the extremely poor ballistic coefficient.

(2) Use of long gun with large powder charge: The barrel tube is too long to be practical, with serious barrel erosion, high pressures and recoil.

The ratio of bullet mass to gun volume must be small, and reviewing the interior ballistics theory that the muzzle energy is equal to the work done on the projectile,

$$(P\ avg) \times AL = 1/2\ m\ V^2$$

A = bore area
L = barrel length
P avg. = mean bore pressure

$$\therefore \frac{2\ P}{m/V_o} = V^2$$

The smaller the m/V_o ratio, the higher the velocity.
V_o = Bore volume

(3) This brings us to the third method of achieving a high velocity, that of a small caliber projectile in a large caliber gun, made possible by using a sabot of lightweight material.

One method of removing the sabot at the muzzle is by using a muzzle attachment called a "stripper". The bore is smooth, but the stripper is a short length of rifling, that imparts a spin to the segmented sabot, centrifugal force causing it to break apart evenly. Alloys of exotic metals are required for strippers of any endurance. The stripper length is approx. 4 to 6 calibers long.

Muzzle Booster

Muzzle boosters are used in recoil operated weapons to augment the energy delivered to the recoiling mass. For a conventional recoil cycle, the recoil mass is accelerated rearward while the projectile is accelerated forward. If the recoiling mass is too heavy, it will not have enough energy to complete the cycle satisfactorily for all firing conditions, particularly in adverse conditions and in extended firing schedules.

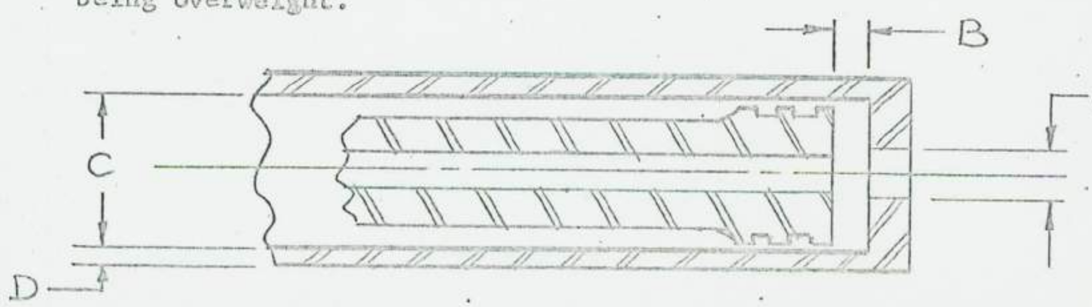
Of course, the weight is necessitated for strength and "heat sump" purposes, so a muzzle booster is incorporated into the barrel jacket which is fixed to the weapon receiver or frame, and does not recoil with the barrel. It traps muzzle gases just as the projectile exits, so that the gases impinge between the capped end of the barrel jacket (muzzle booster) and the barrel muzzle face, thus adding to the recoil impulse. In this way, some of the excess energy that normally escapes at the muzzle is trapped and put to work. Critical dimensions include the following:

A. Exit bore diameter, which must not interfere with maximum bullet yaw angle, nor be too large.

B. Inboard length, which must compensate for increase in barrel length due to thermal expansion in prolonged firing.

C. Barrel bearing diameter, which should not be too small, thus binding when barrel muzzle expands thermally, nor should it be large enough to permit excessive barrel vs. jacket mis-alignment during muzzle vibration, causing interference of bullet with dia. "A".

D. Thickness of jacket, to provide tensile strength sufficient to resiststrain caused by gas pressure acting on booster cap, without being overweight.



Flash Suppression

Muzzle flash is caused by excessive temperature and pressure of muzzle gases. A number of chemical additives have been added to the propellant to reduce the tendency to flash. Most have been salts of alkali metals. The most effective was desium iodide, but the most commonly used are salts of potassium, or potassium sulfate.

Mechanical flash suppressors are most effective. A cone type flash hider was first used, but is limited in effective flash suppressive action. The average cone type flash hider has a 12 to 18^{o} included angle, and a length of 3 to 6 inches. (7.62mm)

The bar type flash suppressor is most effective in the manner that the gases expand through the slots, breaking up the continuity of the flow, thus preventing shock formation. An odd number of bars is usually necessary, and the end of the slot nearest the muzzle is at right angles, because gas increases in velocity when it turns at right angles, externally, mixing quicker with the air, for a cooling effect.

The width of the bar is slightly wider (approx. .02 inch) than the slot opposite it, on symmetrical designs. For rifles, the lower bar is usually wider, in order that it may function as compensator in reducing muzzle climb.

The slot area facing the bore should be at least 20 times the bore area. The suppressor is usually open-ended, except on ground weapons, in which a closed end is used to prevent brush-spearing discomfort.

Silencers

The use of silencers is usually limited to low powered weapons. A silenced weapon has advantages on certain applications where firing is necessary without revealing one's position.

The principle of a silencer is similar to that of an automobile muffler, in which the energy of the gases is reduced by an expansion chamber and baffle system. Silencers are usually cylindrical in shape and project in front of the barrel as well as around it. The weight and bulk increases as a function of the degree of noise suppression. That is, there may be a compromise between silencing and volume.

Only projectiles traveling at sub-sonic velocities can be completely silenced, since a projectile at supersonic velocity sets up a shock wave in the atmosphere, creating a sharp cracking sound along the trajectory until the velocity drops to sub-sonic.

The weapon discharge, however can be silenced, and, of course, the silencer is heavy, since it must withstand the muzzle pressure.

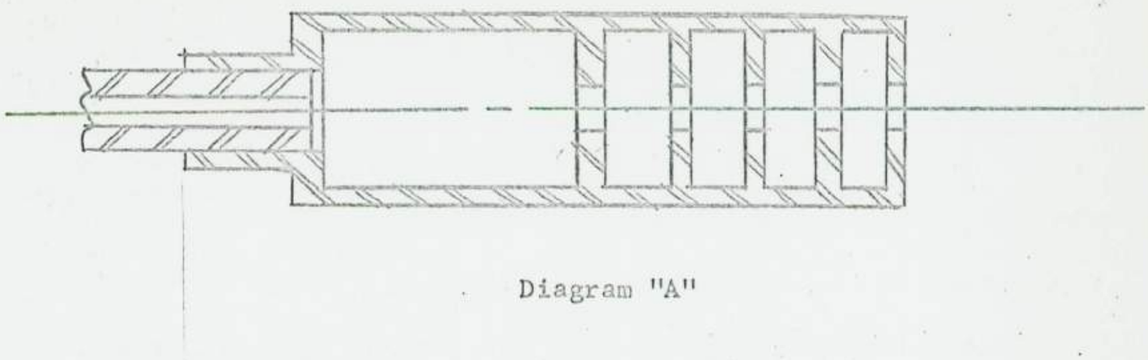


Diagram "A"

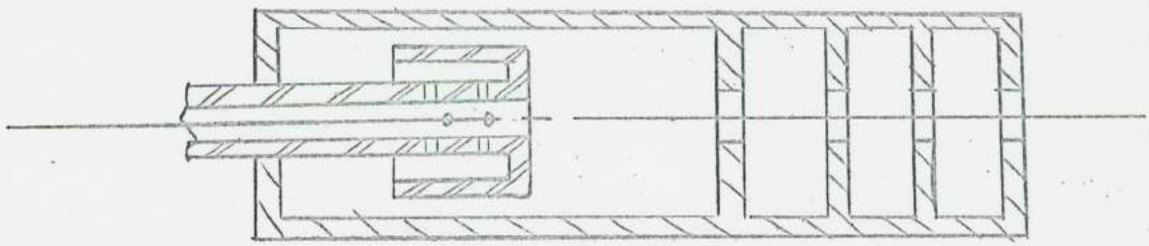


Diagram "B"

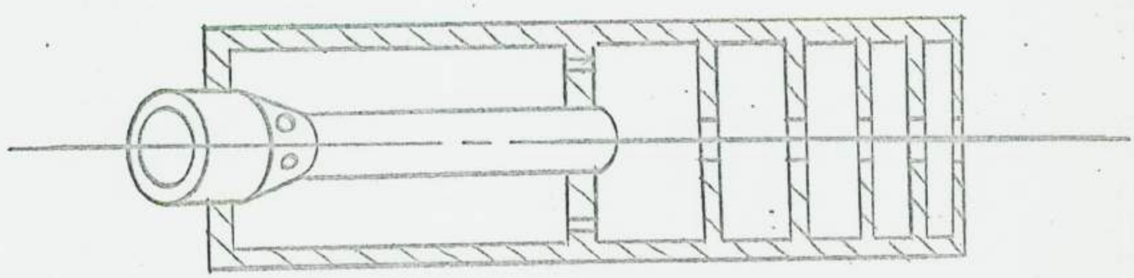


Diagram "C"

TYPICAL SILENCER DESIGNS

Silencers are usually fitted to weapons with fixed barrels. This is because the weight of a recoiling primary mass with/without the silencer would vary too greatly. In this case, the silencer could be mounted on a non-recoiling barrel jacket.

The common revolver cannot be completely silenced due to the gas leakage at the barrel/cylinder interface.

One typical silencer is shown in diagram A.

An initial chamber allows muzzle blast to be reduced and the following smaller ones dissipate gas energy, so that noise is reduced. The bullet hole should be as small as possible commensurate with bullet yaw. In some cases a self-sealing rubber pad is inserted, but this eventually burns, and is limited to a few rounds before replacement, if desired.

The muffler, or silencer, will act as a muzzle brake, become overheated in extended firing, and requires frequent cleaning.

In diagram B the sharp muzzle blast peak is reduced, being vented about an additional baffle. Muzzle velocity is not appreciably changed.

In order to reduce velocity of standard supersonic rounds (to avoid special ammo) some of the gas may be exhausted into an expansion chamber. For example, in diagram C the 9mm Parabellum cartridge (m.v.1330 fps) could employ this system for optimum silencing.

By law, silencers are not permitted on commercial weapons, but certain types of "sound moderators" may be available.

The noise level attained can be measured by a microphone connected to a cathode ray oscillograph, with photographs taken of the sound waves.

Muzzle Brake

A muzzle brake functions to reduce recoil effects by trapping gas at the muzzle and causing a forward impulse to act on the weapon.

Usually a series of baffles are formed in a muzzle brake so that the gases pass through and are diverted to the rear.

The gases should not be completely reversed or a blast wave may reach the shooter's face, eyes, or ears. The ports should be symmetrical, so that the gun is not turned, and should not be directed downward where they would raise a dust cloud. A muzzle brake can also be integrated with a compensator, to keep upward muzzde jump to a minimum.

Essentially, the action of the muzzle gases on a muzzle brake is the same as that of (for comparative and demonstrative purposes) a jet of high pressure steam on the turbine blades of a turbine. The kinematic analysis of each is similar. The gas pressure reduces quickly at the muzzle and the gases escape with a kinetic energy equal to the pressure difference only in the case in which a Laval expansion nozzle is fixed at the muzzle. The gases must pass with a high velocity from the muzzle to the blade (brake). The design of the nozzle must accordingly not impede this flow.

Outflow velocities in the case of simple parallel openings reveal that the critical velocity is far surpassed.

The cross-sectional boundary where the gases are not yet mixed with the air is shown in the accompanying sketch, in graphic scale. This represents the "jet border" within which the muzzle brake must act, also shown in part b of the sketch.

The time that the projectile travels from the muzzle to the brake vane, or blade, is the effective period of the muzzle brake. Gas mass is also significant, so the higher charge-to-mass ratio (c/m) loads are more efficient in muzzle brake action. That is, weapons using ammunition with low charge weights cannot effectively benefit from use of a muzzle brake.

To determine the efficiency of the muzzle brake, fire the weapon at 0^{o} elevation without the muzzle brake, then with the muzzle brake.

As the difference in the lengths of the recoils is only a few millimeters, one can assume in practice that in both cases, with the maximum recoil velocity, the recoil lengths are the same, and that the maximum recoil velocity occurs at the end of the after effect. The brake force K_x may have a constant value.

K_x = The constant brake force without muzzle brake,

K'_x = The constant brake force with muzzle brake,

Gr = Weight of recoiling parts.

I. Without muzzle brake: $K_x \cdot S_3 = \frac{Gr}{2 \cdot g} \cdot V^2 \max = E$,

(1) $$K_x = \frac{2 \cdot g \cdot S3}{Gr} \cdot V^2 \max$$