TPOWERS

CHAPTER 7

COMPONENT DESIGN

7-1 GENERAL

Automatic weapons are equipped with practically the same components that other weapons need to insure effective and safe (to the operator) performance. Differences lie only in application since the components in the automatic weapon must be geared to automatic performance. These components include feed mechanisms, breech locking systems, sears, firing mechanisms, extractors, ejectors, and cocking mechanisms. Characteristics of other components such as muzzle devices which include silencers are presented in detail in other design handbooks [25] or published reports [26]. Each component generally has features unique to automatic weapons.

7-2 FEED MECHANISM DESIGN

Automatic weapons are fed ammunition from magazines, clips, and belts; the type and capacity depending upon type of weapon. The bolt, moving in counterrecoil, strips the round from the feed mechanism and carries it into the chamber. The withdrawn round is instantly replaced by the next round of the supply.

The first step in designing a feed mechanism is defining the feed path. The feed path is the course of the round from mechanism to chamber. Two requisites take precedence: (1) to have the initial position of the projectile move as close to the chamber as the system permits, and (2) to have the base of the cartridge case as close in line to the center line of the bore as possible at the time of feed. The ideal would have the center lines of round and bore collinear. The ideal is not always possible; therefore, other arrangements must suffice but care must be exercised to avoid impact between bolt face and primer since bolt contacts cartridge during counterrecoil. The primer is the restricting element. The two views of Fig. 7-1 illustrate this characteristic. Unless surface contact is assured at impact, the outer edge of the bolt face must never extend into the primer surface, otherwise the edge may strike the primer with enough resulting penetration to set it off. To preclude premature discharge, a minimum space of 0.010 in. between the edges of the primer and bolt face is necessary. Because of override, impact cannot be eliminated; thereby, obviating this approach as a solution for premature firing. *Override* is the clearance between bolt face and cartridge case base needed to position the round before the bolt moves forward. Interference here cannot be tolerated, otherwise malfunction is inevitable.

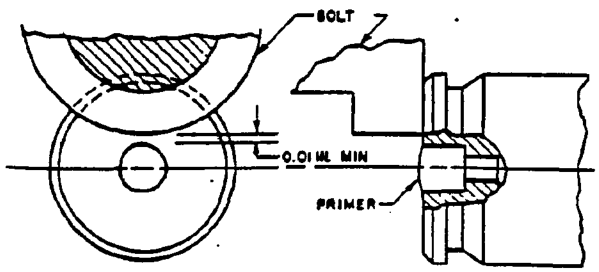

Figure 7-1. Initial Contact of Bolt and Cartridge Case Base

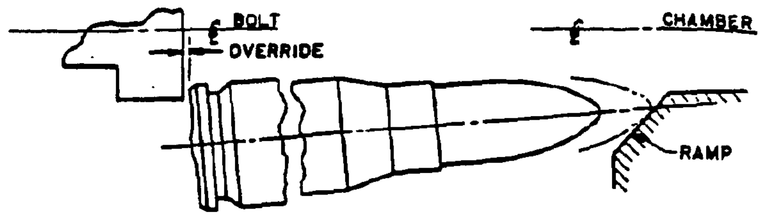

Figure 7-2. Chamber-projectile Contact

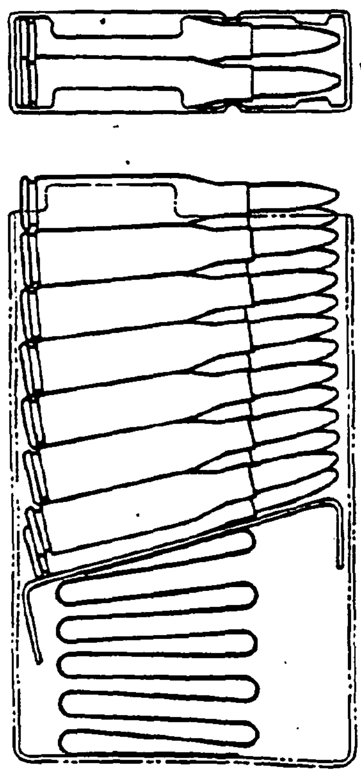

Figure 7-3. Box Magazine

The next design operation is to provide a path for the round between the immediate receptacle and chamber, and guidance along this path. The receptacle—whether magazine, clip, or belt—provides the initial guidance which will be discussed later. The chamber provides the terminal guidance. The entrance to the chamber and the path of the round should be so arranged that any contact between chamber and projectile will take place on the ogive. Fig. 7-2 shows this arrangement. The chamber entrance may be enlarged by a ramp to eliminate the probability of the nose striking the chamber walls first.

7-2.1 MAGAZINES

Magazines, box or drum, are of limited capacity. Box magazines generally hold from 7 to 20 rounds in single or double rows; drums, up to 150 rounds.

7-2.1.1 Box Magazines

A box magazine may be attached to the receiver or it may be an integral part of it. Both types have a spring to keep forcing the rounds toward the bolt as firing continues. The box not only stores the rounds but also restrains their outward motion at the mouth and guides each round as the bolt strips it from the box. The restraining and guiding elements, called lips, are integral with the sides. Fig. 7-3 shows a box magazine with several rounds of ammunition.

Correct lip length is vital to dependable loading. Combined with the direction of the spring force, the lips control the position of the round as it enters the chamber. As indicated in Fig. 7-3, continuous control is exercised by the lips while they restrain the round and so long as the resultant spring force passes within their confines. If the resultant spring force falls forward of the lips, the round will have a tendency to tip excessively

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and increase the probability of jamming. Fig. 7–4 demonstrates how a short lip may fail to guide a round so that it enters the chamber without interference. Fig. 7–4 demonstrates how a longer lip will retain contact with the round long enough for the ogive to hit the ramp just prior to entering the chamber.

The shape of the lip has considerable influence on feeding. The round to be loaded should be restrained by line contact between the cartridge case and lip. Fig. 7–5 shows how this effect can be arranged by making the inner radius of the lip less than the radius of the cartridge case. Absolute assurance of line contact is assured by forming the lip by a right angle bend. The spring load holds the round firmly until the bolt dislodges it. On the other hand, if the lip radius is larger than the cartridge case radius, accurate positioning of the rounds cannot be achieved with any degree of assurance. The cartridge case position, from round to round, may virtually float; thereby, causing an inconsistency in contact area between the bolt face and the rounds. Fig. 7–5 shows how the positions may vary with respect to the fixed bolt position. The larger the radius, the less assurance of sufficient contact area between the bolt face and cartridge case base. In extreme cases the bolt may hit the primer first and initiate it.

The dimensions of the cartridge and the intended capacity determine the size of the magazine. For a single row of cartridges, the width equals the diameter of the base plus 0.005 in.

$$w = D_c + 0.005 \qquad (7\text{–}1)$$

where D_c = diameter of cartridge case base

w = inside width of magazine

Double rows of cartridges are stacked so that the centers form an equilateral triangle as shown in Fig. 7–6 where the inside width of the magazine is

$$w = 1.866\, D_c + 0.005. \qquad (7\text{–}2)$$

The nominal depth of the magazine storage space with double rows is

$$h = \frac{1}{2}\, D_c (N + 1) \qquad (7\text{–}3)$$

where h = depth

N = number of rounds

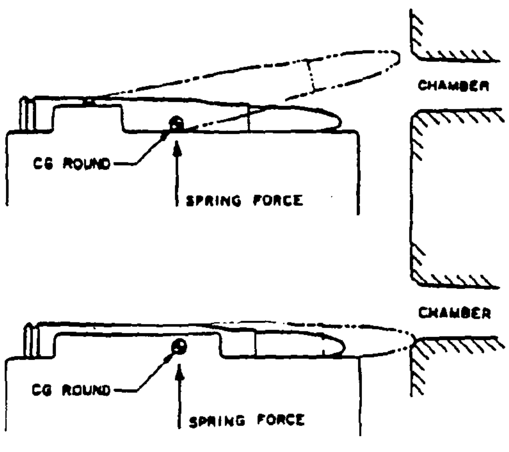
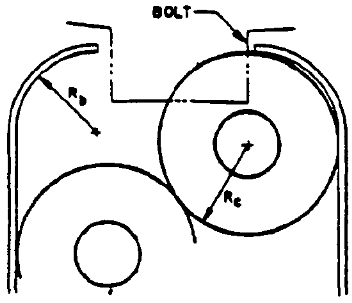


Figure 7–4. Lip Guides



(A) PROPER ARRANGEMENT, $R_b < R_c$

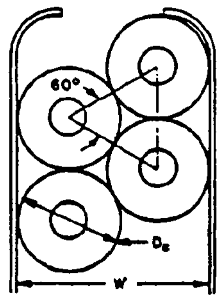
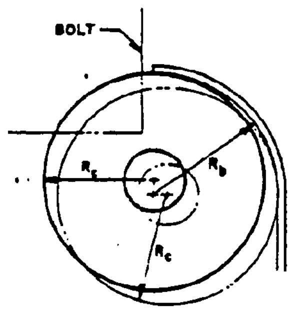


Figure 7–6. Geometry of Double Row Stacking



(B) POOR ARRANGEMENT, $R_b > R_c$

Figure 7–5. Lip-cartridge Case Orientation

7–2.1.2 Box Feed System

The box feed system has three major components, the box which has been discussed, the follower, and the spring. The follower separates the column of cartridges from the spring, transmits the spring force to the cartridges, and provides the sliding surface for the last (single row) or last two (double row) cartridges. The follower also holds the stored rounds in alignment. It should never restrict spring activity. Fig. 7–7 shows three views of a follower. The spring may be a round wire spring shaped into rectangular coils or it may be a flat steel tape folded over at regular intervals to approximate the side view of a helix.

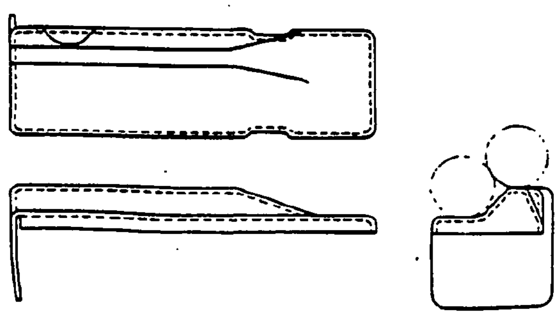


Figure 7–7. Box Magazine Follower

7-2.1.2.1 Flat Tape Spring

The flat steel spring functions in bending rather than in torsion. Each segment behaves as a cantilever beam that has the loaded end restrained from rotating. Fig. 7-8 shows this analogy and the loading diagram. Beginning at the follower, the bending moment M_o at the bend, when the applied load is assumed to be concentrated at the middle of the follower is

$$M_o = -\frac{1}{2}\left(FL\right) \qquad (7\text{–}4)$$

where F = spring force

L = length of each spring segment

The bending moment at the end of the first free segment

$$M = M_o + FL = \frac{1}{2}\left(FL\right). \qquad (7\text{–}5)$$

This moment is identical and, therefore, constant for all segments of the spring. The deflection of one end of each segment with respect to the opposite one is

$$\Delta y = \frac{M_o L^2}{2EI} + \frac{FL^3}{3EI} = \frac{FL^3}{12EI} \qquad (7\text{–}6)$$

where E = modulus of elasticity

I = area moment of inertia of the spring cross section

The total deflection of a spring having N active segments is

$$y = \Sigma \Delta y = N \Delta y = \frac{NFL^3}{12EI} \qquad (7\text{–}7)$$

Solve for the spring constant.

$$K = \frac{F}{y} = \frac{12EI}{NL^3}. \qquad (7\text{–}8)$$

Not only must the spring exert enough force to hold the ammunition in position but it must also provide the acceleration to advance the ammunition and the other moving parts over the distance of one cartridge space in time for the bolt to feed the next round. The equivalent mass of all moving parts in the ammunition box is

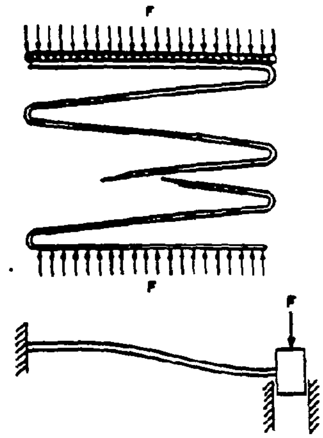


Figure 7-8. Flat Tape Spring and Loading Analogy

$$M_e = \left[(N-1)W_e + W_f + W_{se}\right]/g \qquad (7\text{–}9)$$

where g = acceleration of gravity

N = number of rounds in the box

W_f = weight of follower

W_e = weight of each round

W_s = weight of spring

$W_{se} = \frac{1}{3} W_s$, equivalent weight of spring in motion

The time required for any one particular displacement will be similar to that of Eq. 2-27

$$t = \sqrt{\frac{M_e}{gK}}\ \mathrm{Cos}^{-1}\frac{F_o}{F_m} \qquad (7\text{–}10)$$

where F_m = maximum spring force (preceding one cartridge displacement)

F_o = minimum spring force (following one cartridge displacement)

K = (Eq. 7–5)

M_e = (Eq. 7–6)

ϵ = efficiency of system, generally assumed to be 0.5 for initial design analysis

For initial estimates, provide a spring load of F_i pounds for an empty box and one of F_f for a full box.

The folded flat spring is less desirable than the rectangular coil spring because the latter can be compressed to its solid height whereas total compression of the flat spring is limited by the radius of the folds, thereby, requiring a longer box to house the spring and store the ammunition. Par. 7–2.1.2.2 discusses the rectangular coil spring.

7–2.1.2.2 Rectangular Coil Spring

The rectangular coil spring is a torsion element. Fig. 7–9 illustrates the mechanics of operation. Torsion in each straight segment rotates the adjacent segment. Although bending occurs along the span of each segment, the corners move with respect to each other only by torsional deflection. Bending deflections at the corners are neutralized by equal and opposite bending moments.

Rectangular coil spring characteristics are computed according to procedures similar to helical springs. The applied load is assumed to be concentrated on the axis. The torque T_1 on the long segment is

$$T_1 = \frac{1}{2}\left(aF\right) \qquad (7\text{–}11)$$

and torque T_2 on the short segment is

$$T_2 = \frac{1}{2}\left(bF\right) \qquad (7\text{–}12)$$

where a = length of short segment

b = length of long segment

F = spring force (7–13)

The corresponding angular deflections are

$$\theta_1 = \frac{bT_1}{JG} = \frac{abF}{2JG} \qquad (7\text{–}14)$$

$$\theta_2 = \frac{aT_2}{JG} = \frac{abF}{2JG} \qquad (7\text{–}15)$$

where G = torsional modulus

J = area polar moment of inertia of wire

The axial deflection of each segment of a coil varies directly with the sum of the products of the two segment lengths times the sine of the angular deflection of the adjacent segment (see Fig. 7–9). Stated in algebraic expressions the two deflections are

$$\Delta y_1 = b \sin\theta_1 \qquad (7\text{–}16)$$

$$\Delta y_2 = a \sin\theta_2 \qquad (7\text{–}17)$$

But, according to Eqs. 7–14 and 7–15, $\theta_1 = \theta_2$, and if we let this angle be equal to θ, the deflection of two adjacent segments of a coil is

$$\Delta y = (\Delta y_1 + \Delta y_2) = (a + b)\sin\theta \qquad (7\text{–}18)$$

Since there are 4 segments to each coil, the total deflection of a spring having N active coils is

$$y = 2N\Delta y. \qquad (7\text{–}19)$$

The spring constant, if y is based on a free spring, is

$$K = \frac{F}{y} \qquad (7\text{–}20)$$

The time required for any given displacement can be computed from Eq. 7–10.

7–2.1.3 Example Problems

Compute the spring characteristics for a double row box feed system that holds 20 rounds. Each round weighs 420 grains and has a cartridge case base diameter of 0.48 in. To function properly in the box, the spring should fit in a projected area of 1.75 x 0.75 in. The initial spring load should be approximately 4 pounds.

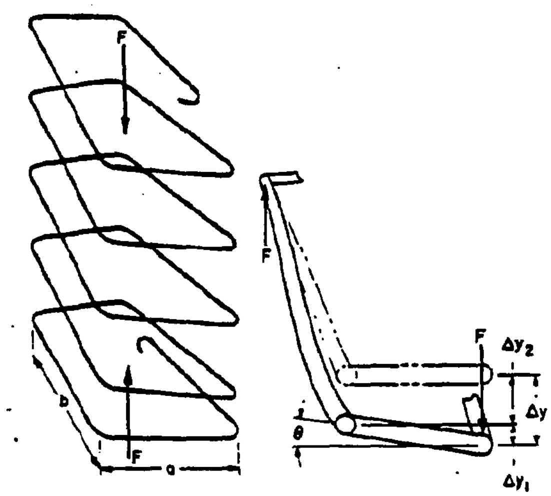


Figure 7–9. Rectangular Coil Spring and Loading Characteristics

7–2.1.3.1 Flat Tape Spring

Set the following initial parameters:

F_i = 4.0 lb, initial spring load

L = 1.75 in., length of each spring segment

N = 14, number of active segments, arbitrary choice but based on previous designs

w = 0.75 in., width of spring

σ_w = 200,000 lb/in.^{2}, working stress of spring

The spring deflection, Eq. 7–3, inside the box caused by the cartridge displacement is

$$y_c = \frac{1}{2} D_i(N+1) = \frac{0.48}{2}(20+1) = 5.04 \text{ in.}$$

where N = 20 rounds.

Assume, as a first estimate, that the deflection on assembly approximates the total cartridge displacement.

$$y_i = 5.0 \text{ in., the initial deflection}$$

According to Eq. 7–8, $K = \frac{F_i}{y_i} = \frac{4.0}{5.0} = 0.8$ lb/in.

Now solving for I in the same equation

$$I = \frac{KNL^3}{12E} = \frac{0.8 \times 14 \times 1.75^3}{12 \times 30 \times 10^6} = \frac{1}{6} \times 10^{-6}$$

Since $I = \frac{1}{12} wt_s^3$, $t_s^3 = \frac{12I}{w} = \frac{8}{3} \times 10^{-6}$

Therefore t_s = 0.014 in. the required spring thickness. The bending moment, Eq. 7–5, is

$$M = \frac{1}{2}(FL) = \frac{1}{2} \times 8 \times 1.75 = 7 \text{ lb-in}$$

where $F = Ky = 0.8 \times 10 = 8$ lb

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The bending stress is

$$\sigma = \frac{Mc}{I} = \frac{7 \times 0.007}{0.1667 \times 10^{-6}} = 294{,}000 \text{ lb/in.}^2$$

where $c = \frac{t_s}{2}$ in.

This stress is too high. To lower it to acceptable levels, the initial and final loads were reduced to 1.0 and 2.0 pounds, respectively. Subsequent computation produced the following data:

K = 0.2 lb/in.

t_s = 0.00874 in.

M = 1.75 lb-in.

σ = 183,000 lb/in.^{2}

The bending stress is still uncomfortably high which almost rules out this type spring for the above application. However, a time analysis will give additional data. The time will be computed for spring action after the first and next to the last round are removed. If the spring weighs 0.063 lb and the follower 0.044 lb, the equivalent moving mass for 19 rounds, according to Eq. 7-9, is

$$M_e = \left(19 \times 0.06 + 0.044 + \frac{0.063}{3}\right) / 386.4$$

$$= 0.00312 \text{ lb-sec}^2\text{/in.}$$

Substitute the appropriate values in Eq. 7-10 to compute the time for the first round

$$t = \sqrt{\frac{M_e}{\epsilon K}} \operatorname{Cos}^{-1} \frac{F}{F_i} = \sqrt{\frac{0.00312}{0.5 \times 0.2}} \operatorname{Cos}^{-1} \frac{1.952}{2.0}$$

$$= \sqrt{0.0312} \operatorname{Cos}^{-1} 0.976 = 0.1765 \times 0.22$$

$$= 0.039 \text{ sec}$$

where ϵ = 0.5, the efficiency of the system.

For the last round

$$M_e = \left(0.06 + 0.044 + \frac{0.063}{3}\right) / 386.4$$

$$= 0.000323 \text{ lb-sec}^2\text{/in.}$$

$$t = \sqrt{\frac{0.000323}{0.5 \times 0.2}} \operatorname{Cos}^{-1} \frac{F}{F_i} = \sqrt{0.00323} \operatorname{Cos}^{-1} \frac{1.0}{1.04}$$

$$= 0.057 \times 0.301 = 0.0172 \text{ sec}$$

The slower of the two is equivalent to 1500 rounds/min which is more than adequate.

7-2.1.3.2 Rectangular Coil Spring

Set the following initial parameters:

a = 0.75 in., length of short segment

b = 1.75 in., length of long segment

F_i = 4.0 lb, initial spring load

N = 7, number of coils, arbitrary choice but based on previous designs

y_c = 5.04 in., cartridge displacement (see par. 7-2.1.3.1)

y_i = 5.0 in., assembled deflection (see par. 7-2.1.3.1)

$$K = \frac{F_i}{y_i} = \frac{4.0}{5.0} = 0.8 \text{ lb/in.}$$

The total deflection for a full box of cartridges is

$$y = y_c + y_i = 10.04 \text{ in.}$$

The deflection for two adjacent segments of a coil from Eq. 7-19 is

$$\Delta y = \frac{y}{2N} = \frac{10.04}{14} = 0.717 \text{ in.}$$

The angular displacement according to Eq. 7-18 is

$$\sin\theta = \frac{\Delta y}{a + b} = \frac{0.717}{2.5} = 0.2868$$

$$\theta = 16°40' = 0.291 \text{ rad.}$$

Solve for J in Eq. 7-15.

$$J = \frac{abF}{2G\theta} = \frac{0.75 \times 1.75 \times 8.032}{2 \times 12 \times 10^6 \times 0.291} = 1.509 \times 10^{-6} \text{ in.}^4$$

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where $F = Ky = 0.8 \times 10.04 = 8.032$ lb, maximum spring load

$G = 12 \times 10^6$ lb/in.^{2}, torsional modulus of steel

Since $J = \left(\frac{\pi}{32}\right) d^4 = 1.509 \times 10^{-6}$

$d^4 = 15.37 \times 10^{-6}$

$d = 0.0626$ in., say, 0.0625 in.

Then $J = 1.5 \times 10^{-6}$ in.^{4} and the maximum spring force F_m is

$$F_m = \frac{2JG\theta}{ab} = \frac{36 \times 0.291}{1.3125} = 8.0 \text{ lb.}$$

The maximum torque, Eq. 7–12, is

$$T_1 = \frac{1}{2} bF_m = \frac{1}{2}(1.75)\ 8.0 = 7.0 \text{ lb-in.}$$

The torsional shear stress is

$$\tau = \frac{T_1 c}{J} = \frac{7.0 \times 0.03125}{1.5 \times 10^{-6}} = 146{,}000 \text{ lb/in.}^2$$

where $c = \frac{d}{2} = 0.03125$ in.

This stress is acceptable.

If the spring weighs 0.036 lb, and the follower 0.044 lb, the moving mass for 20 rounds, according to Eq. 7–9 is

$$M_e = \left(19 \times 0.06 + 0.044 + \frac{0.036}{3}\right) / 386.4$$

$$= 0.0031 \text{ lb-sec}^2\text{/in.}$$

For 19 cartridges,
$y_c = 4.8$ in. and $F_o = (5.0 + 4.8)\ 0.8 = 7.84$ lb.

The time to move this mass through the space left by the departed projectile is computed by Eq. 7–10.

$$t = \sqrt{\frac{M_e}{eK}} \cos^{-1} \frac{F_o}{F_m} = \sqrt{\frac{0.0031}{0.5 \times 0.8}} \cos^{-1} \frac{7.84}{8.0}$$

$$= 0.088 \times 0.201 = 0.018 \text{ sec}$$

where $e = 0.5$, the efficiency of the system.

The time of 18 msec is far less than needed to operate under any existing conditions.

7–2.2 BOLT-OPERATED FEED SYSTEM

The bolt-operated feed system illustrated in Figs. 7–10 and 7–11 represents one of many similar types. The operating features are described by partially isolating each function and then later showing the coordination that exists in the whole system. Fig. 7–10 shows the ammunition belt system including the components directly associated with it. Sketch (A) shows the position of all parts just as the chambered round has been fired. Sketch (B) shows all parts in the same position except that Round 1 and the empty case are partially extracted, and the feed slide has moved to the left with the feed pawl riding on Round 2. Note that if Round 1 had not been extracted from the belt, the pawl arm would ride over this round to lift the feed pawl above Round 2 to preclude engagement between pawl and Round 2. This operation prevents double feeding or jamming. With Round 1 extracted, the feed pawl carried by pawl arm and slide, continues across Round 2 and eventually engages it as shown in Sketch (C). In the meantime the holding pawl prevents the belt from moving backward.

After the slide completes its travel to the left, the extractor pushes Round 1 downward to align it with the chamber and eject the empty case. After this effort, the slide begins its return to the right and since the feed pawl has engaged Round 2, the slide forces the belt to move also. Two positions of the return are shown in Sketches (C) and (D). Round 3 forces the holding pawl downward to permit belt travel. As soon as Round 2 reaches the original position of Round 1 and all other rounds have simultaneously moved up one position, all feed belt activity will stop with all components taking the positions according to Sketch (A).

The feed slide is acuvated by the feed lever which in turn is activated by the bolt. The lever fulcrum is fitted to the cover of the receiver, one end activates the slide while the other end rides in a cam groove in the bolt's top surface. Each end of the cam is straight and parallel to the longitudinal axis of the bolt in order to permit a short dwell period for the slide at the end of each half cycle. Shifting the emphasis between the upper and lower illustrations of Fig. 7–11 provides the opportunity of outlining the whole loading and firing cycle. Assume that the bolt is in battery and firing is imminent. The upper picture shows, in phantom, Round 1 of Fig 7–10 (A) ready to be stripped. The extractor lip is in the extractor groove of the cartridge case. At this same time, the