The width and depth of the grooves are usually the same along the entire barrel length, although ideally it would be advantageous to decrease the groove depth towards the muzzle to ensure perfect forward obturation of the bullet. The cross-sectional profile of the bore beyond the forcing cone for jacketed bullets may have no rifling grooves as such but several circular arches whose centre lies outside of the barrel axis. Such a profile is called a polygonal profile and usually provides long service life, good sealing of the bullet and easy maintenance. A polygonal bore with continuous circular arches and four grooves is shown in Fig. 6.7.

The angle of twist of the rifling is usually constant for small calibre weapons. A twist rate is chosen which will stabilise the bullet in both a new barrel and a worn one. It is normally expressed as the length of one complete turn in millimetres. The twist rate at the muzzle and the muzzle velocity determine the spin rate of the bullet.

The force to provide the angular acceleration of the bullet is high and exerts a shearing force on the jacket of the bullet. With constant twist rifling the angular acceleration force reaches a maximum value with maximum pressure in the barrel. The accuracy of the weapon is considerably affected by the muzzle of the barrel which affects the flow of propellant gases around the bullet as it leaves the barrel. It is essential that the end of the muzzle is perpendicular to the barrel axis. The internal edge of the muzzle is usually rounded with a greater radius than the groove depth. For barrels requiring maximum accuracy, such as sniper rifles, the muzzle end is counter bored several calibres in depth, which results in a cylindrical cavity of a diameter which is greater than the diameter of the bullet. This provides constant gas flow around the bullet as it leaves the muzzle.

**BARREL MOUNTING AND GUIDES**

The barrel is attached to the weapon by the breech end to the breech casing and barrel support. In weapons with fixed barrels, the receiver group and the weapon casing are the same and one of the main parts of the weapon. On weapons with a recoiling barrel the barrel is attached to the receiver group which locates the barrel in the weapon casing. The breech block and bolt recoil together with the barrel and the breech casing and unlock from the barrel at a point in the recoil cycle depending on the operating cycle of the weapon. The receiver group has at least one guide, usually to the rear as shown in Fig. 6.8.

The front guide is positioned directly on the barrel or as a sleeve on it. The closer the front guide is towards the muzzle, then the smaller the angular movement of the barrel in the guides. The torque acting on the barrel caused by the bullet spin is resisted by the guide of the receiver group or by a special guide bar which meshes with the breech casing recess as shown in Fig. 6.8. The barrel is either fixed or has a quick change facility. For many small arms the mode of firing does not require the replacement of the barrel under normal operating conditions and therefore the barrel is pressed into the breech casing as shown in Fig. 6.9.
The joint can be of the following design:
- cotter joint
- ribbed joint (bayonet joint with straight ribs)
- sector threaded joint (bayonet joint with threaded ribs).

This joint is often called the barrel nut. A cross wedge bears against a lug on the barrel, or it passes through the cross groove on the barrel as shown in Fig. 6.11. The bevel of the cross wedge presses the barrel into the receiver group. It is locked in position by means of a nut or a spring-loaded pin.

Some machine-guns, such as the SGMT 7.62mm tank machine-gun, are designed so that the barrel nut can adjust the position of the barrel relative to the breech block to adjust the cartridge head space, which if excessive, may cause the cartridge case to break when being extracted. The ribbed joint shown in Fig. 6.12 is composed of ribbed and smooth sectors along the barrel circumference and the barrel beds into the breech casing.

There are usually between two and four ribbed and smooth sectors. One sector is composed of several cross ribs with a rectangular profile. The sector joint is identical in design; only the ribs have a thread shape. This design makes it possible to press the barrel towards the breech block, or to pre-stress the barrel spring slightly. The thread has a rectangular or trapezoidal shape. To join the barrel to the breech casing, the barrel is placed with its threaded sectors into smooth sectors of the breech casing and rotated to lock the two together. The joint is completed when the threaded sectors of the barrel and breech casing engage. To provide for precise positioning of the barrel, the
reciever group is equipped with supports in the form of spigots before and after the joint. It is important that the barrel cannot become loose during firing and a ratchet is usually provided to prevent this. Where the barrel cannot be rotated, due to the use of cross position, the joint is composed of a rotary barrel nut located in the breech casing. The barrel cannot turn if the ritter is used. A flat on the cylindrical bed may be used to prevent this, as shown in Fig. 6.11.

BARREL ATTACHMENTS

The muzzle blast is often used to improve the operation of the weapon. The most common barrel attachments are:

- muzzle brakes
- recoil intensifiers
- flash hiders
- noise suppressors (silencers)
- muzzle deflectors.

Muzzle Brakes

Muzzle brakes decrease the recoil impulse on the barrel by directing propellant gases back towards the breech as they emerge from the barrel. This creates a negative impulse which reduces the overall recoil effect on the barrel. The efficiency of a muzzle brake increases with the quantity of gases discharged and with the angle of deflection. Fig. 6.13 shows an open muzzle brake which uses a plate deflector to direct the gases back towards the weapon.

Fig. 6.13 Open muzzle brake

Fig. 6.14 shows a chamber muzzle brake designed as a vessel to provide an opening for the projectile and with a system of side ports for venting the propellant gases. Its efficiency increases with the ratio of the area of side ports to the area of the muzzle opening. The inside of the brake can be composed of one or more chambers and the outlets comprise of large holes (Fig. 6.14c), slots (Fig. 6.14a) or small holes (Fig. 6.14b). Muzzle brakes can considerably reduce the recoil force acting on the weapon and so allow the weight of the weapon to be reduced. They are simple in design and are lightweight. However, the muzzle blast raised by them can be uncomfortable to bystanders and can stir up dust and obscure the target when firing.

Recoil Intensifiers

Recoil intensifiers have the opposite effect to muzzle brakes. They are used with weapons which are recoil operated and in weapons of small calibre in which it is necessary to increase the velocity of the recoiling barrel. To help accelerate the barrel the use of the muzzle gases, acting in a chamber created by extending the weapon casing up to the muzzle, is made, as shown in Fig. 6.15. Another method is to increase the momentum of discharging gases through a nozzle mounted on the barrel muzzle. Thus a flash hider can also operate as a recoil increaser.
Flash Hiders

The gases flowing out of the barrel are very hot, and because they contain carbon monoxide, hydrogen and methane they combine with oxygen from the air as they emerge from the barrel and a flame may be formed at the muzzle which creates a very visible weapon signature.

Figure 6.14 Chamber muzzle brake

Flash Hiders reduce the temperature of expanding gases at the muzzle to below their ignition temperature. They usually have a conical shape, as shown in Fig. 6.16, which also mechanically screen the flame from view. A greater effect is achieved by sucking cooling air through an opening in the rear part of the flash hider. The mounting of the flash hider must avoid vibration, which would increase the dispersion of the weapon. However, the mounting should also allow its removal for the flash hider to be cleaned.

Noise Suppressors (Silencers)

When a shot is fired by a firearm it creates a loud noise. The sources of this noise are the propellant gases venting from the muzzle, the bullet passing through the air and the weapon mechanism moving backwards and forwards. The greatest component of this noise is caused by the supersonic shock wave of the gases as they leave the barrel. The level of acoustic pressure of small arms noise ranges from 140 to 160 dB and exceeds the threshold of pain by 10 to 30 dB. It decreases to a level of 80-90 dB at a distance of about 1,000m, where the level is not harmful. Ear protection is therefore of the utmost importance. For special small arms the noise level is reduced by means of a noise suppressor, the purpose of which is to reduce the velocity of the gases venting from the muzzle. For maximum effect the gases should be reduced in velocity to below the speed of sound, and to achieve this in the noise suppressor the gases are cooled, their pressure reduced and their exit velocity lowered. The noise suppressor is usually of a cylindrical shape with a large expansion volume and with many baffles and

Figure 6.15 Recoil intensifier

Figure 6.16 Conical flash hider
vents, dividing the internal space into additional expansion spaces in which the temperature and pressure of the gases are reduced. The partitions can be of rubber construction through which the bullet can pass. Examples of noise suppressors are shown in Fig. 6.17.

If the muzzle velocity of the bullet is greater than the speed of sound there will be the high-pitched crack from the shock wave created by the bullet passing through the air. For maximum effect the bullet velocity should therefore be reduced to below the speed of sound, which is approximately 340 m/s. This will affect bullet performance and for automatic and self-loading weapons may give weapon cycling problems if the weapon is designed to operate using full-powered ammunition. Another, often neglected, source of noise is from the operating mechanism of the gun. A sub-machine-gun firing from an open bolt can be heard up to 100 m away when it is dry cycled.

**Muzzle Deflectors**

Muzzle deflectors counteract the muzzle climb associated with automatic weapons firing bursts. They are usually fitted to hand-held small arms and cause a reactive force on the muzzle which acts against muzzle climb, as shown in Fig. 6.18.

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**Figure 6.17 Examples of noise suppressors**

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**Figure 6.18 Examples of muzzle deflectors**

The reactive force is created by the deflection of the muzzle gases in the opposite direction to the required force. Ideally, the reactive force would equal the torque created by firing the weapon, and the weapon would remain stable. The construction of a deflector is quite simple. It is sufficient to machine a muzzle extension with an incline as shown in Fig. 6.18a, or to machine a number of vents beyond the muzzle on the upper surface of a muzzle extension as shown in Fig. 6.18b. The function of the deflector can be substituted by a bayonet which is subjected to the action of gases as shown in Fig. 6.18c.

**STRESSES IN BARRELS**

When firing, the barrel of a weapon is stressed both mechanically and thermally. The barrel must be made to withstand these stresses.

**Mechanical Stresses in Barrels**

The mechanical effects upon the barrel are caused by the propellant gases and the bullet. The propellant gas pressure exerts a radial force that is axially symmetrical in the walls of the barrel. The circumferential stress, $\sigma_c$, and radial stress, $\sigma_r$, are the main stresses and they increase towards the internal surface as shown in Fig. 6.19.

The barrels of most small arms are of monobloc construction and can be analysed as if they were thick-walled tubes. Lameé derived the relationships for
Substituting equations (6.6) and (6.7) into equations (6.4) and (6.5) and rearranging where:

\[ a = r_2/r_1 \text{ and } b = r_2/t \]

the equation for stress in the wall of a monobloc barrel is:

\[ \sigma_r = -p \left( b^2 - 1 \right) / \left( a^2 - 1 \right) \]  

(6.8)

\[ \sigma_\theta = p \left( b^2 + 1 \right) / \left( a^2 - 1 \right) \]  

(6.9)

The variation in stress \( \sigma_r \) and \( \sigma_\theta \) across a monobloc barrel is obtained from equations (6.8) and (6.9) and is dependent on the radius as shown in Fig. 6.19. The actual value of stress may, at certain points in the barrel, differ from the calculated static values because of the dynamic loading. For the practical determination of the barrel strength an empirically derived coefficient of safety is used which may vary along the barrel. The increase in stress towards the muzzle results from an increase in dynamic loading, as shown in Fig. 6.20.

Curves No. 1 to 4 in Fig. 6.20 are the hoop stresses \( \sigma_\theta \) at different points along the barrel during firing. Curve No.1 was taken at the cartridge chamber, curve No.2 at the forcing cone, curve No.3 near the point of maximum pressure and curve No.4 at the muzzle of the weapon. An increase of dynamic stress towards the muzzle is evident from the steepness of the curves. To determine the thickness of the barrel wall it is usual to start at the position of maximum pressure of the propellant gases, \( p_{\text{max}} \), along the barrel, as shown in Fig. 6.21.

Variation in propellant gas pressure is dependent on the bullet displacement, \( p = p(t) \), obtained from the internal ballistic calculations or from actual measurements. The pressure the barrel is designed to withstand, \( p_\sigma \), is given by:

\[ p_\sigma = n \cdot p_{\text{max}} \]  

(6.10)

The safety coefficient, \( n \), includes an allowance for dynamic stresses, the value of which change along the barrel. For most conditions \( n \) is plotted in Fig. 6.21.

To calculate the dimensions of the barrel the elastic limit of the materials used is taken as the maximum allowable stress. There are several ways to determine the elastic limit, such as the Saint-Venant or Mises theories.

The most often used theory is that of greatest elongation, although it underestimates the actual stress of the barrel, and must not exceed the yield stress, \( \sigma_y \). Thus:

\[ \sigma = E \cdot e_y = \sigma_y - \mu \cdot \sigma_y \leq \sigma_y \]  

(6.11)

Substituting \( \sigma \) and \( e_y \) from equations (6.4), (6.5), (6.6) and (6.7) for \( r = r_1 \) and excluding the effect of pressure, the expression for the elastic limit is:

\[ \sigma_\varepsilon = 1.5 \cdot e_y (a^2 - 1)/(2a^2 + 1) \]  

(6.12)

where \( a = r_2/r_1 \).
Using equation (6.12) gives the minimum dimensions of the barrel. However, the dimensions of small arms barrels are not only determined by the need to withstand firing stresses, but also by the requirement for stiffness when being handled and the need for maximum thermal capacity.

**Barrel Heating**

The thermal stress in gun barrels can be significant. Hot propellant gases and the friction of the bullet against the barrel bore are the sources of heat. The transfer of heat from the hot propellant gases into the barrel wall has the greatest effect. This heat transfer, $q$, is mostly by forced convection and is given by:

$$ q = \alpha (T_p - T_w) $$

As the number of rounds fired increases so will the bore temperature, $T_w$. Thus, the flow of heat to the barrel will decrease. The increase in barrel wall temperature is non-steady and is three-dimensional in nature. To simplify the analysis, it is assumed, with little loss of accuracy, that heat flow is two-dimensional in
the radial direction only. The temperature, $T_r$, at a point in the barrel is a function of time, $t$, and radius, $r$, thus $T = f(r, t)$. The temperature distribution in the barrel wall is described by the following differential equation:

$$\frac{dT}{dt} = a \left( \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right)$$

where the thermal diffusivity, $a$, is given by:

$$a = \frac{\lambda}{c \cdot \rho}$$

To solve equation (6.14) it is necessary to give the initial condition $T(r, 0)$ which will depend upon whether firing starts with a cold barrel or a hot barrel. Where the barrel has enough time to equalise temperatures then $T(r, 0)$ will be a constant. Also, it is necessary to give the boundary conditions for both surfaces of the barrel. At the external surface $r = r_e$ and cooling during firing is negligible, thus:

$$\frac{dT}{dr} (r_e, t) = 0$$

At the internal surface $r = r_i$

thus

$$\lambda \frac{dT}{dr} (r_i, t) = [T(r_i, t) - T_e (t)]$$

The solution of equation (6.14) can only be approximate because it is difficult to determine precisely the input quantities, especially the heat transfer coefficient, $\alpha$. The duration of the firing pulse is very short so that heating of the internal layers of the bore is restricted to a very thin layer. The maximum temperature of the internal surface depends on the propellant; it can exceed 900°C but falls very rapidly through the barrel wall as shown in Fig. 6.22. Thus the heating process causes considerable thermal shock at the bore surface which creates large temperature stresses. After only a short time small cracks appear in the surface of the barrel bore. After flowing into the barrel the heat dissipates through the barrel wall until the temperature along the whole wall is equalised.

The heating of the barrel wall takes place in two phases: the transient heating of the bore surface and the bulk heating of the barrel which raises the overall barrel temperature.

A simple and approximate way of calculating the overall rise in barrel temperature, $\Delta T$, from the heat input to the barrel, $q$, is from the following equation:

$$\Delta T = q/(m_b \cdot c)$$

Figure 6.22 Barrel bore temperature profile into the depth of the barrel for different times after firing.

The specific heat, $c$, of barrel steels is between 460 and 520 J/kg°C. Firing trials show that about 10% of the energy released from the burning propellant is transferred to the barrel. For a cold barrel this is approximately 20% and for hot barrels it is about 5%. Thus the temperature rise of the barrel can be calculated using the specific energy of the propellant, $Q_p$, from:

$$\Delta T = (0.1 \cdot m_p \cdot Q_p) / (c \cdot m_b)$$

The muzzle often gets the hottest because of its reduced wall thickness.

If the specific energy of the propellant is not known it is possible to determine the average temperature increase from the kinetic energy of the bullet at the muzzle, $E_0$, and the efficiency of the weapon, $\eta$, thus:

$$\Delta T = (0.1E_0) / (m_b \cdot c \cdot \eta)$$
The efficiency, $\eta$, of small arms is usually between 0.4 and 0.5.

If the increase of the barrel mean temperature per shot is known, it is possible to determine the definite number of rounds, $N$, to heat the barrel to its maximum allowable temperature, $T_{\text{max}}$, thus:

$$N = T_{\text{max}} / \Delta T$$

For automatic weapons $T_{\text{max}}$ may equal 350-450°C, depending on the type of automatic system and the barrel material. The temperature profile for a series of rounds can be determined by superimposing the profiles of individual shots. The rise in barrel temperature for three shots is shown in Fig. 6.23.

When firing continues for a long period with a regular rhythm a steady-state temperature regime is reached in the barrel, when the amount of heat transmitted to the barrel by firing equals the amount of heat lost. The equilibrium temperature will be dependent on the firing regime and the barrel mass. The steady-state temperature regime is of importance because of the onset of thermofrettage, where the thermal stress in the barrel, caused by the expansion of the barrel material, helps the barrel to resist the stress caused by the high-pressure propellant gases. Thermofrettage occurs because of the non-uniform temperature distribution across the barrel wall. The temperature gradient from the inner to the outer barrel diameters for automatic weapons reaches rates in excess of 100°C. The temperature profile across the barrel wall for steady-state conditions and the profile of the reduced thermal stress $\sigma_r$ are shown in Fig. 6.24.

Thermofrettage cannot be guaranteed and is not taken into account when designing barrels. However, the mechanical properties of the materials from which the barrel is made reduce with temperature, as shown in Figs. 6.25 and 6.26. Thermofrettage can therefore compensate for this reduction in material strength.

Fig. 6.27a shows how the internal surface temperature of the barrel and the bulk temperature are affected by breaks in the firing cycle. The breaks in firing have considerable influence on the temperature profile of the inner surface. The increase in temperature for longer bursts is affected by the initial tempera-
ture of the barrel at the end of the previous break. It can be seen from Fig. 6.27a that short breaks in firing reduce the peak bore temperatures of the barrels internal surface, although the bulk temperature increase is little affected.

If the given mode of fire causes the maximum allowable temperature of the barrel to be exceeded, it is necessary either to reduce the requirements for the combat rate of fire, to use cooling, to provide a replacement barrel or to use a
barrel of greater mass. Fig. 6.27 shows the temperature profile at a 5.56mm calibre machine-gun barrel during long-term firing at a relatively high rate of fire (about 50 rounds per minute). The barrel was changed after 250 rounds. In this way, it was possible to fire 2,000 rounds within 40 minutes without exceeding a barrel temperature of 400°C. Further firing shows no substantial increase in temperature because steady-state conditions were reached. The effect on temperature of increasing the barrel mass, for a constant rate of firing for a 5.56mm calibre machine-gun, is shown in Fig. 6.28.

The effect of the combat rate of fire on the increase in barrel temperature for a 7.62mm calibre machine-gun with a barrel of 2.8kg is shown in Fig. 6.29. Cooling helps to stop the increase in barrel temperature, but cannot reduce the thermal shock at the inner surface of the barrel during firing. Barrel cooling can be obtained by natural means or by using artificial cooling. Natural cooling is the widespread method used because of its simplicity, but the cooling effect is small as shown in Fig. 6.30. Whilst heavy barrels heat up slowly they also cool slowly.

Heat loss from the external surface of the barrel is by a combination of radiation and natural convection and is given by:

\[ q = \alpha (T_p - T_\infty) + \varepsilon \sigma (T_p^4 - T_\infty^4) \]

\( T_p \) and \( T_\infty \) are absolute temperatures.

It follows from equation (6.22) that to achieve greater cooling it is necessary to enlarge the external surface of the barrel. Ribs can increase the outer surface area but are not extensively used because of extra costs for barrel production.

Cooling can be increased by additional external or internal cooling, or by both, although, internal cooling cannot be applied when the gun is being fired. Water, water mist or compressed air are frequently-used cooling substances, in which a water cooling jacket fitted to the barrel is the most effective. However, the increase in size, weight and cost of water jackets preclude them from widespread use. A reduction in barrel heating can also be achieved by reducing the heat transmitted to the barrel. This can be achieved by the use of slower burning propellants and additives in the propellant to protect the barrel bore.
When the barrel is heated there are changes in its internal and external dimensions. Bending of the barrel can occur, caused by non-uniform cooling or different thicknesses of the barrel wall as a result of inaccuracy in production. A bent barrel caused by heating increases dispersion. An increase in the dimensions of the barrel bore enlarges the clearance between the barrel wall and bullet, which has the same effect as a worn barrel. An increase in the external dimensions of the barrel results in changes in the clearance between the barrel and guide bushes, which must be taken into account when choosing the guide clearance. The increase in barrel diameter is the sum of thermal expansions and deformations caused by thermal stresses. The resultant relations for the increase of internal (\(\Delta d_1\)) and external (\(\Delta d_2\)) diameter for a barrel starting at an initial temperature of \(T_0\), are:

\[
\Delta d_1 = d_0 \frac{6T}{\ln \frac{a^2}{a_0^2}} \left(1 - \frac{a^2}{a_0^2} \ln \frac{a^2}{a_0^2}\right) + \alpha (T_1 - T_0)
\]

\[
\Delta d_2 = d_0 \frac{6T}{\ln \frac{a^2}{a_0^2}} \left(1 - \frac{a^2}{a_0^2} \ln \frac{a^2}{a_0^2}\right) + \alpha (T_2 - T_0)
\]

\(T_1\) and \(T_2\) are the temperatures of the internal and external surfaces of the barrel and \(\Delta T = T_1 - T_2\).

The component for simple expansion is usually much greater than the deformation caused by thermal stress so that the following simplified equation can be used with little effect on accuracy:

\[
\Delta d_1 = d_0 \alpha (T_1 - T_0)
\]

\[
\Delta d_2 = d_0 \alpha (T_2 - T_0)
\]

**BARREL WEAR**

When a gun is fired a complex set of conditions occur that mechanically and thermally stress the bore, eventually leading to barrel wear. Barrel wear reduces weapon performance and once its performance falls below the required level it is replaced. The service life of a barrel is usually set by the number of rounds it has fired, which is determined by the service life of the barrel bore. The condemnation criteria for barrels is set by:

- reduction in muzzle velocity
- increase in dispersion
- onset of bullet instability
- the weapon becomes a danger to the user.

A common cause of barrel failure is an obstruction in the barrel, most frequently cleaning rods or materials. Mud or snow in the barrel can also form an obstruction. The common effect of firing a gun with an obstruction in the barrel is to cause the barrel to burst.

Another possible cause of catastrophic barrel failure is that of fatigue. While this is a highly unlikely in modern military small arms, it has been known to occur in weapons produced under wartime conditions where the quality control of the barrel materials is not as stringent.

The causes of barrel wear and possible ways of reducing it are shown in Fig. 6.31. An important consideration is that of the rifling lead because the impact of the bullet onto the forcing cone considerably increases barrel wear. Chemical reactions between the burning propellant and the surface of the barrel produce cementation and nitration of the surface in the form of a layer 0.05-0.1 mm thick. Erosion of the bore by the propellant gases appears as a washing away of the bore surface. Erosion is more intensive where the surface is damaged and in narrowing parts of the barrel, such as the transition from the cartridge chamber to the rifling. Erosion effects increase as the bore becomes hotter. The hottest part, the forcing cone, is particularly exposed to erosion effects. Wear causes the forcing cone to move forwards in the bore which in turn results in greater bullet insertion depth and a resultant reduction in the muzzle velocity of the bullet.

The burning propellant has several effects on barrel wear. It causes a high thermal gradient at the surface of the barrel; this leads to high thermal stresses which result in a network of cracks on the bore surface that form the basis of mechanical wear and erosion. Also, heating of the barrel results in a decrease in material strength, as shown in Fig. 6.25, which leads to lower resistance to erosion and abrasion. The cyclic character of surface heating and cooling can lead to mechanical and thermal fatigue, resulting in the development of cracks and a reduction in the resistance to wear.

The first signs of wear appear after only a few shots. The surface of the forcing cone and the first part of the rifling are soon covered with dull spots which then combine in a closed circle. If the surface is viewed through a microscope a network of thin surface cracks can be seen. These cracks become wider, deeper and longer as the number of shots increases. The cracks in the grooves are along the axis of the barrel, while in the lands they are transverse. Lands are exposed to mechanical abrasion which is aggravated by the increased roughness caused by the cracks. Corrosion, engraving of the bullet and erosion by the propellant gases cause the surface of the bore to break away at the beginning of the rifling. The profile of the lands changes by rounding of the edges of the grooves.

Wear is not distributed uniformly along the barrel. The most critical region of wear is the forcing cone and the beginning of rifling up to about the point of maximum pressure, which is the region of maximum wear. Maximum wear
is usually at the beginning of the full depth of the grooves. There follows a region of medium wear which lies near the point of maximum pressure. In the middle part of the bore, the wear is small and uniform.

Ways of increasing barrel life include good barrel and rifling design, the use of wear-reducing materials, correct bullet construction and the use of cooler, less aggressive propellants. Weapon operation, operator training and weapon maintenance all affect barrel life. The service life of automatic weapon barrels also depends on the mass of the barrel: a heavier barrel has greater heat capacity, which results in the barrel temperature rise being less for a given number of rounds fired, as shown in Fig. 6.28. However, for most weapons an increase in barrel mass is undesirable because it reduces maneuverability. Other factors to be considered are the type and shape of rifling. A polygonal bore produces less deformation when engraving the bullet into the forcing cone and better sealing than rectangular rifling which results in a higher initial velocity, a smaller dispersion and greater barrel life. For automatic small arms, extrusion of the barrel to form the rifling using a mandrel can give greater barrel life in the region of the forcing cone. Similar results can be achieved with cold hammer forged barrels.

Better accuracy and longer service life of the muzzle can be achieved through a slight taper of the bore (typical taper for 7.62mm calibre is 0.01 to 0.02mm on the barrel length). This taper can be achieved by cold forging, where the reduction of the barrel bore towards the barrel muzzle is achieved by the taping mandrel.

Hard chromium plating of the bore with a thickness of up to 0.05mm is commonly used on small arms. This can double the barrel life when the weapon is fired under the most severe firing cycles. Stellite liners fitted to the breech end of the barrel were developed during World War II to increase the barrel life of aircraft machine-guns. These liners are still fitted to certain sustained fire medium and heavy machine-guns, and considerably increase the barrel life of these weapons.
Breech Systems

PURPOSE OF THE BREECH SYSTEM
The task of the breech is to close the cartridge chamber safely during firing. It must ensure that it remains closed to support the cartridge case and prevent the escape of propellant gases to assure the safety of the crew as well as avoiding damage to the weapon. Because of its position in the weapon, the breech consists of a group of components which aid the chambering of the cartridge and extraction and ejection of the empty cartridge case. The origin of the breech is closely connected with the development of engineering production technology during the industrial revolution in the second half of the 19th century and with the application of a series of inventions in armament technology, especially the fixed cartridge. Loading the weapon via the breech allowed higher rates of fire, automatic loading, improvement in the sealing of the projectile in the bore and a reduction in dispersion of the projectile. Another advantage was to facilitate the loading of the weapon in the prone position.

CLASSIFICATION OF BREECH SYSTEMS
The breech system can be classified according to the way it locks to the barrel or breech casing. In low-powered weapons it is not necessary to lock the breech directly to the barrel. In high-powered weapons it is usual for the breech to lock into the barrel and for the joint to resist the full load imposed by the propellant gases acting on the base of the cartridge case.

Those breeches which do not lock into the barrel rely on their momentum and the force of the return spring to resist the opening force of the propellant gases. The breech consists of only one piece. For more powerful weapons, it is necessary to apply additional resistance in the movement of the breech block by the use of a delay mechanism and, depending on the delay mechanism, the breech may be made of one or more pieces. If the breech is locked for part of the cycle but is driven to the rear of the weapon by the force of the propellant acting on the base of the cartridge case, it is known as a locked blow-back breech. Fig. 7.1 shows the different types of breech systems.

CONSTRUCTION AND FUNCTION OF THE BREECH

Unlocked Breeches
Unlocked breeches are used with blow-back actions. The return spring acting on the mass of the breech block closes the breech and is shown schematically in Fig. 7.2. Because the breech moves during firing, its weight should be such that extraction of the cartridge case does not occur until the gas pressure has dropped to a safe level.

Blow-back actions with unlocked breeches are used with weapons of relatively low power, such as pistols and sub-machine-guns. The most powerful cartridge used with this type of breech is the 9mm parabellum. Cartridges of greater power than this would require an excessively massive breech block to resist the opening force acting on the base of the cartridge case.

Delayed Breeches
Delayed breeches are similar to unlocked breeches but the resistance to the force on the base of the cartridge case is increased, so that they can be used with cartridges which are more powerful.
Breech with Divided Mass

To slow the breech in its rearward movement a two-part breech block is used, as shown in Fig. 7.3. It reduces the total mass of the breech block.

The breech block is divided into two parts which can move relative to one another. When the breech moves forward, part I in Fig. 7.3 chambers the cartridge which is immediately initiated. Part II continues to move forward and when the bullet starts to move part I starts moving rearwards. The movement of part I is decelerated due to the impact of part II, which slows the opening of the breech.

Advanced Primer Ignition

This method of delaying the opening of the breech also makes use of forward momentum to reduce the mass of the breech block. The front part of the breech is designed so that it chambers the cartridge. The primer is initiated during the forward movement as the striker impacts on the casing lug, as shown in Fig. 7.4. The forward momentum of the breech block resists the initial force applied by the propellant gases to the base of the cartridge case, thus reducing the required weight of the breech block.

Breech Delayed by Friction

Opening of the breech can be delayed by increasing the friction between the moving parts of the breech. This method was used in the early design of the breech of the Thompson sub-machine-gun as shown schematically in Fig. 7.5.

The breech is pushed to the rear in the usual manner by the force acting on the cartridge case base. A lug on the breech block locates in an angled groove in the weapon casing which rotates the breech block, thus imparting both angular and linear momentum to the breech block. Additionally, friction between the lug and the groove acts to resist the motion of the breech block. After the lug enters the horizontal part of the groove the breech acts as a normal unlocked breech. Changes in resistance to motion are achieved by different inclination angles of the oblique groove. By making the angle closer to

92

93
that of a right angle, the resistance to motion increases. However, this angle can be reduced to the point where the breech would not open by the force acting on the base of the cartridge case alone, which is a condition that must be avoided if the weapon is to function. The geometry of the breech arrangement used is shown in Fig. 7.6. Operation of the breech depends on the coefficient of friction between the guide and groove, \( f \), and between the breech block and friction piece, \( f_2 \).

Equilibrium is achieved in the y axis if:

\[
N_1 \sin (\alpha - \beta) - f_2 N_1 \cos (\alpha - \beta) - f_1 N_2 = 0
\]

Normal forces depend on the force acting on the base of the cartridge case, \( F_0 \), and are given by:

\[
N_1 = F_0 \cos \alpha
\]

\[
N_2 = F_0 \cos \beta
\]

Substituting for \( N_1 \) and \( N_2 \) and assuming that \( f_1 = f_2 = f \), then:

\[
\tan (\alpha - \beta) = f \left( 1 + \frac{\cos \beta}{\cos (\alpha - \beta) \cos \alpha} \right)
\]

Because \( \beta \) is small, \( \frac{\cos \beta}{\cos (\alpha - \beta) \cos \beta} \) can be replaced by \( \frac{1}{\cos^2 \alpha} \).

By taking small increments of \( \alpha \) it is possible to approach the condition when the breech will not self-lock:

\[
\tan (\alpha - \beta) \approx 2f
\]

for \( f = 0.1 \) the value of \( (\alpha - \beta) > 11.3^\circ \)

\[
\text{If } \beta = 0 \text{ then } \alpha > 11.3^\circ
\]

**Breech Delayed by Propellant Gases**

Fig. 7.7 shows a blow-back system which uses the propellant gases to slow the breech block as it opens. The barrel is provided with gas holes and the front part of the breech is arranged as a piston. Propellant gases are bled from the holes in the barrel and act on the inner front face of the breech block to slow its movement.

**Breech Using Accelerated Mass**

This type of breech mechanism is the most widely used for delaying the opening of the breech. A two-part breech block is used with an accelerating element between them. A single-part breech can also be used with the accelerating element, in the form of a crank, attached to the barrel as shown in Fig. 7.8. This type of arrangement was used for the Schwarzlose medium calibre machine-gun. The effect of accelerating the breech is to give it a higher velocity so that a lighter breech block can be used for the same breech block momentum.
The transmission ratio, $i$, is the instantaneous ratio of the accelerator element as shown in Fig. 7.9:

$$(7.9) \quad i = \frac{l}{h}$$

The transmission ratio can be changing continuously during the rearward movement of the breech, so that the breech resistance can be controlled as required. Sloiving of the breech is also affected by transmission losses, expressed in terms of efficiency, but the reduced mass has the greatest effect.

The German Heckler Koch G3 7.62mm calibre automatic rifle and the French FAMAS 5.56mm calibre assault rifle are examples of modern automatic weapons using delayed breech mechanisms. A schematic arrangement of the breech of the G3 rifle and how it works is shown in Fig. 7.10.
The transfer of the driving force to the carrier is achieved using two symmetrically positioned rollers which are inserted between the breech block and the carrier. The leverage, or transmission ratio, is \( \geq 2:1 \). The breech starts to move as soon as the bullet moves. The cartridge case acts on the breech block which bears against the rollers and presses them out of the barrel casing beds. The rollers act on the front part of the breech block carrier which is firmly connected to the main part of the breech block carrier. Thus the carrier is accelerated and the rollers move into the space which the carrier previously occupied. Finally, the breech, which is now pure blow-back, is accelerated by the remaining propellant gases fully to the rear. A similar breech is used in the French AAS2 machine-gun shown in Fig. 7.11. The main difference is the use of a lever accelerator. The accelerator bears against the recess in the barrel casing. The shorter arm acts on the breech block while the long arm acts on the breech block carrier.

Locked Blow-Back

Locked blow-back breeches are locked during firing, but unlock while the case pressure is still high. The high-pressure gases acting on the base of the cartridge case drive the breech block to the rear of the weapon. Fig. 7.12 shows a commonly used locked blow-back system.

After taking up the clearance \( \Delta \) (Fig. 7.12) between the breech block and casing, the breech block is stopped when it bears against the weapon casing. The breech block carrier continues to the rear because of its momentum and, after travelling the distance of the underside, it lowers and unlocks the breech block. The breech is further accelerated by the residual pressure of propellant gases acting on the cartridge case base. The time to unlock is governed by the time for the underside travel. The breech appears to be a locked breech. If a static force acts on the breech block, it takes up the clearance \( \Delta \), the breech block bears against the casing but the carrier remains in the front position and the breech remains locked. It requires a dynamic action to operate the mechanism. This simple breech is rarely used with modern weapons because of the difficulty in maintaining the small clearance \( \Delta \).

Locked Breeches

Most modern weapons use a locked breech. The different types of locked breech are:

- axial breech
- wedge-type breech
- fixed breech.
For the kinematic actions of locking, unlocking, opening and closing the breech there are 16 theoretical possible variants. The breech can be unlocked and opened in the x direction, as shown in Fig. 7.13, it can move sideways, up or down, and it can tilt or turn about its axis or about a parallel axis.

However, all theoretical possible variants are not used because a breech that unlocks by moving in the x direction would be an unlocked breech. Breeches are classified as:

- rotary (breech block rotates about the x axis)
- tilting (breech block tilts either about the y or z axis or moves aside in the y or z axis)
- straight (breech block moves along the x axis and needs a special locking piece which can move in the y or z axis).

An advantage of breeches which move in the x axis is that the breech block can be used for feeding the cartridges as well as the extraction and ejection of the empty cases. If a crosspiece at right angles to the barrel axis is used then the breech is known as a wedge-type breech. The group of breeches that unlock and open by rotary motion includes breeches with a tilting drum or tilting block (the tilting breech block of single-shot breech loaders of the Remington system). It also includes those breeches that open by tilting the closing component, or by 'breaking' the barrel. Of this group, only breeches with a tilting drum are suitable for automation. For high rates of fire, revolver breeches are often used where the unlocking and opening operation uses rotary motion.

**Tilting Breeches**

Tilting breeches were fitted to early breech loading military rifles but are now confined to sporting and target weapons. Upward tilting breeches are used the least; an example is the Warrant's breech, as shown in Fig. 7.14a. Downward tilting breeches are typified by the Martini system shown in Fig. 7.14b.

For hunting and sporting shotguns a standing breech is used. Opening of the breech is achieved by tilting or 'breaking' the barrel around a hinge pin. The force acting on the cartridge case base is absorbed by a standing breech behind the receiver. The barrel is secured in the locked position by a wedge acting on lugs attached to the barrel. Fig. 7.15 illustrates three other frequently used ways of locking the barrel to the standing breech.

**Rotating Bolts**

Rotating bolts are the most widely used type of breech mechanism for automatic weapons. The breech block consists of a locking bolt and carrier. The carrier moves only along the axis of the barrel. The rotating bolt is controlled by the carrier and its purpose is to chamber the cartridge, close and lock the breech and extract and eject the empty case. The bolt locks by turning about its longitudinal axis. The reciprocating movement of the carrier produces rotary movement of the bolt, usually by a cam groove on the carrier, as shown in Fig. 7.16. In this case, the control cam groove is machined inside the cylindrical space of the carrier and the guide lugs of the bolt mesh with the cam groove.
The shape of the control cam groove provides a small amount of underslide of the carrier during unlocking, i.e. a short free path of the carrier until the start of unlocking. The amount of underslide is chosen to allow the pressure to drop in the barrel. When closing the breech, the bolt is moved out of the carrier with its guide lugs at the beginning of the cam groove and the cartridge is pushed into the chamber. The bolt does not turn during chambering of the cartridge. At the end of chambering the bolt moves out of the guide and stops against the rear face of the barrel. The carrier continues moving forward due to its inertia and the return spring force, and by the action of the cam groove on the guide lugs of the breech block as it turns the breech block. The locking lugs of the breech block thus engage with a corresponding recess in the breech casing as shown in Fig. 7.16. The angle of rotation and loading capacity of the locking mechanism are determined by the number and arrangement of the locking lugs. A reduction in the angle of rotation can be achieved by using more lugs around the breech block circumference as shown in Fig. 7.17. However, the greater the number of lugs then the smaller they will be.

To increase the loading capacity it is necessary to enlarge the total supporting surface of the locking lugs. If this is not possible it is necessary to use more lugs, especially in a series arrangement, as is the case of heavy machine-guns or automatic cannons, and as shown in Fig. 7.18.

Fig. 7.18 shows a typical position of the control cam on the breech block; the figure also shows part of the control groove for giving the underslide. The rotary bolt is reliable in action because the distance from the locking surfaces (supporting surfaces in the casing) to the breech block face is very small, which
reduces the amount of compression of the breech block or bolt, known as springing. Springing of the breech block increases the clearance between the breech and breech face during firing, as shown in Fig. 7.19.

Fig. 7.19a shows the effect on the deformation of the case when firing a cartridge with no springing of the breech block, and Fig. 7.19b shows the increase in case deformation when springing does occur. In the breech block for which springing does not occur the case stretches by the amount of clearance between the breech block and cartridge case. In the breech for which springing does occur, the case stretches by the amount of case clearance plus the amount of springing of the breech block. If the cartridge stretches more than the ductility of the case, the case will break.

For cartridge cases with a rim or a shoulder, the amount that the cartridge case stretches also depends on the clearance in the chamber cone. In this case it is necessary to add the amount of cartridge case stretch, caused by the cartridge case shoulder bearing on the cone of the chamber due to the propellant gas pressure.

The clearance is affected by the tolerances in the positioning of the breech system, the type of cartridge case and the method of locking. The axial clearance will also increase because of wear of the locking surfaces. Such wear can be reduced by unlocking the breech after the chamber pressure has dropped to a low level: this is achieved by sufficient underslide of the carrier. Also of importance is the heat treatment, or surface protection, of the locking surfaces which can be achieved by plasma-nitriding, hard chrome-plating or case-hardening. If excessive wear of the locking surfaces cannot be avoided, it is necessary to design the locking elements so that they can easily be replaced, or to provide axial adjustment of the barrel.

The clearance for rotary breech blocks can be reduced by arranging the locking lugs into a helix. Thus, when locking, the breech block face moves towards the cartridge chamber. A disadvantage of rotary breeches is that a relatively large breech block diameter is required, which results in the need to lower the cartridge being fed from the loading space to the chamber. For recoil-operated weapons this problem is removed by inserting the cartridge into the breech block recesses when the breech is held to the rear. Another disadvantage of rotary breeches is the high level of friction between the closing and locking surfaces. To reduce this friction the locking groove in the weapon casing is made as a helix. When unlocking the breech block, the cartridge case is then easily released from the cartridge chamber walls. Bolt action rifles also have rotary breeches, the Mauser being the most famous and widely used. The Mauser 66 has a telescopic cylindrical breech and is shown in Fig. 7.20.

The Mauser telescopic breech consists of a prismatic guiding part which moves along the breech block casing by means of longitudinal grooves. A cylindrical locking body with two lugs and a handle, is moved in the guides in the breech body. When unlocking, the internal cylindrical part is turned by means of the handle, but the locking piece does not move. After unlocking, the breech is opened by moving the handle rearward, the cylindrical part engages with the guides and moves rearward also. This telescopic arrangement allows the guides in the breech to be shortened by one half, which results in a shorter weapon length. The overall movement of the breech block face, needed for inserting the cartridge, does not change. Rotary breeches are suitable for some systems with a high rate of fire, especially Gatling guns.
**Tilting Breeches - Automatic Weapons**

In conventional tilting breeches the breech block is directly controlled by the carrier. These breeches are rarely used now, although they were often used between the two World Wars. A diagram of a tilting breech is shown in Fig. 7.21. The breech block is tilted to lock by lifting the rear end to engage with the breech casing. The breech block is guided into this position by a groove in the casing. In the locked position the breech block is supported by the carrier which is stressed during firing by a vertical component of force onto the locking surface. Thus the resistance to motion of the carrier is increased; the carrier is usually driven by propellant gases. During the first phase of unlocking, the carrier makes a short underslide to allow the chamber pressure to fall. Unlocking occurs via a control lug on the carrier acting against the corresponding surface of the breech block.

The breech block can be tilted upwards, downwards or sideward. The biggest disadvantage of this type of breech is the large distance between the breech block face and locking surface which results in the breech block flexing. Thus this type of breech has a tendency to break the cartridge cases. Also, the large tilting body increases the mass of the breech block. These disadvantages can be reduced by using a small breech block which tilts sideways by means of a parallelogram and a groove in the casing, as shown in Fig. 7.22. This principle is used for the NSV 12.7mm calibre heavy machine-gun, in which the breech block carrier is driven by propellant gases.

A similar type of tilting breech can also be used with a transmission piece between the breech block and breech block carrier, as shown in Fig. 7.23. Vertical movement, which is greater than the height of the cartridge, is achieved with a light breech block which can reach high reciprocating velocities, allowing a high rate of fire. To reduce acceleration forces, the transmission ratio is variable.

**Linear Breeches**

The breech block for linear breeches does not move at the instant of locking; locking is achieved by the use of a locking piece. For different locking piece
designs the linear breech also requires different designs. The locking pieces can be part of the breech or positioned in the weapon casings.

Linear breeches with the locking piece as part of the breech block are the most common. The plate breech, designed by Russian designer Dekhtarjev and applied in his machine-gun, the DP 27, is illustrated in Fig. 7.24.

The plates open during locking by means of a shaped firing pin which is connected with, and controlled by, the breech block. After reaching the front position, and after the cartridge has been chambered, the breech block abuts with the rear of the barrel. The breech block carrier and firing pin continue to move forward under the action of the return spring; the firing pin forces the locking plates into recesses in the breech casing. Under slide is achieved by moving the parallel part of the firing pin along the plates before the shaped section of the firing pin is engaged. When the force on the cartridge case base acts on the breech block the plates bear against the weapon casing and the cross component of the breech opening force is resisted by the firing pin. The same type of breech is used in the DShK 12.7mm calibre heavy machine-gun. A disadvantage of most plate breeches is the large distance between the breech block face and the point of locking. Moreover, problems of buckling may occur if the plates are not sufficiently thick. Roller breeches are another type of linear breech and work in the same way, as shown in Fig. 7.25. This type of locking mechanism is used in the German MG-42 and MG-3 machine-guns.

A Czech-designed locking arrangement reduces the problems of flexing and is used in the AR Mod. 58 and Mod. 59 machine-guns. To reduce wear, the slide is hard chrome-plated and the lugs in the weapon casing are induction-hardened. Thus the breech can be unlocked whilst there is still pressure in the chamber. A diagram of this breech in the locked position is shown in Fig. 7.26.

---

The locking lug on the locking piece mesh with the recess in the weapon casing, and the breech block cannot open. The movement of the breech block carrier, which is driven by the propellant gases, unlocks the breech. Because of the slope of the control leg the locking piece moves up out of the recess in the weapon casing. The whole of the breech then moves to the rear. Positioned inside the breech block is the striker.

**Wedge Breech**

The wedge breech is normally used for larger calibre weapons and high chamber pressures. The breech wedge is controlled by either the breech block carrier or by the weapon recoil. Feeding of the cartridge into the chamber requires a separate rammer with an extractor because the breech block moves in the bed of the casing at right angles to the axis of the barrel.
Flexing of the breech is minimal because of the small distance from the rear supporting surfaces and the face of the breech wedge. The breech wedge is of prismatic shape, with the front surface perpendicular to the barrel axis. The rear supporting surface of the wedge is at an angle \( \alpha \), as shown in Fig. 7.27. When the breech opens and closes the wedge moves axially. When the breech is opened there is no friction between the cartridge case and wedge because contact is broken immediately upon movement of the breech. An auxiliary extractor on the wedge is often fitted to provide primary extraction of the case. The wedge should be self-locking. Fig. 7.27 shows a diagrammatic representation of the wedge breech. The conditions for self-locking can be found as follows, the forces acting in the direction of wedge displacement being:

\[ f_1 F_p \cos \alpha + t_c \cdot N - F_p \sin \alpha = 0 \]

where \( N = F_p \cos \alpha \).

After rearrangement and when \( f_1 = f_2 = f \), the condition for self-locking takes the following form:

\[ 2 f \alpha = \tan \alpha \]

For greater safety, the friction between cartridge case and wedge is neglected, so that:

\[ \tan \alpha = f \]

Thus for \( f = 0.1 \), the maximum permissible angle \( \alpha = 5.7^\circ \) and from equation (7.11) \( \alpha = 11.3^\circ \).

The wedge houses the firing pin and part of the trigger and locking mechanism. It is therefore a complex component on which to carry out stress calculations. When calculating the basic dimensions of the wedge, the starting point is the empirically derived dimensions of a simple wedge given in Table 7.1.

<table>
<thead>
<tr>
<th>Wedge dimensions</th>
<th>Multiple of diameter of cartridge chamber base</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>1</td>
</tr>
<tr>
<td>Depth (Height)</td>
<td>3.6 - 3.9</td>
</tr>
<tr>
<td>Width</td>
<td>1.7 - 1.8</td>
</tr>
<tr>
<td>Length under slot</td>
<td>1.5 - 1.7</td>
</tr>
</tbody>
</table>

Table 7.1 Dimensions of the wedge breech shown in Fig. 7.27

The movement of a wedge-type breech is controlled by the breech block carrier as shown in Fig. 7.28. The wedge travels vertically in the breech casing guide which is connected to the barrel. In the figure the breech is open. In the locked position the wedge is supported by the breech block carrier. When firing, the wedge is self-locking and the carrier is not stressed by the force acting on the cartridge chamber base.
To achieve a high rate of fire with acceptable accelerations and velocities of the components, a variable transmission is inserted between the breech carrier and rammer. A diagram of such a wedge-type breech is shown in Fig. 7.29.

The diagram shows the rammer ready to chamber the cartridge. Under the action of the return spring the breech carrier travels forwards, as do the lever accelerator and feeder. The shorter arm of the accelerator locates on the lug in the barrel casing, the accelerator turns and with its long arm accelerates the rammer. After chambering the cartridge, the wedge is closed and at the end of the underside of the carrier the cartridge is fired.

**Revolver Breech**

Revolvers were one of the first and most effective methods of providing a weapon with a multi-shot capability. The breech system they use consists of a fixed standing breech. The same system is used with modern revolver calibers, the fixed standing breech making the weapon very short and rigid. The breech is locked by turning the loaded cartridge chamber between the barrel and the fixed standing breech as shown in Fig. 7.30. Revolver pistols rely on the hammer being cocked by the firer to rotate the chamber cylinder.

When firing, the force from the cartridge case base is transmitted to the breech casing. For automatic revolver systems, rotation of the revolver cylinder around the axis parallel to that of the barrel is achieved by a breech carrier. Transferring the linear movement of the carrier, which is driven either by propellant gases or by barrel recoil, into rotary motion of the drum is achieved by a shaped groove on the outer surface of the revolver cylinder into which a protrusion on the breech carrier is fitted. The control groove, or cam, is in the shape of an ellipse, as shown in Fig. 7.31 and is divided into two branches. One branch accelerates the drum to half the pitch between the chambers and the other branch decelerates the drum to position the cartridge chamber in line with the barrel.
Tilting Drum Breech

The tilting drum breech consists of a fixed standing breech and a cartridge cylinder. However, the cylinder is mounted at right-angles to the barrel as shown in Fig. 7.32. The cylinder contains only one cartridge chamber and the drum does not rotate continuously in one direction but oscillates backwards and forwards. The loading position is reached by turning the cylinder through 90° so that a new cartridge can be chambered. The cylinder is driven by the breech carrier not shown in Fig. 7.32. Two modes of operation can be used. When firing at the standard rate of fire the stroke of the carrier is short and the cylinder reciprocates only once for each stroke of the carrier. When firing at higher rates the carrier stroke is longer, and for one stroke three oscillations occur and a three-round burst is fired. This is shown in Fig. 7.33, which is the time displacement curve for the G11 automatic rifle. This three-round burst mode is made possible in the G11 rifle by the use of caseless ammunition because extraction and ejection of the cartridge case is not required.

![Figure 7.32 Tilting drum breech used in the G11 assault rifle](image)

![Figure 7.33 Time/displacement curve for the breech carrier of the G11 assault rifle](image)

BREECH COMPONENT STRESSES

Rotating Bolt

During firing the force of the propellant gas acting on the cartridge case base is $F_p$. Fig. 7.34 shows a rotating bolt arrangement with two symmetrically positioned lugs.

The propellant force, $F_p$, is divided equally between the two lugs so that one lug is loaded by the force $F_p/2$. The greatest stress applied to the locking lugs is the shear stress. The height of the lug, $h$, is usually chosen as half the width, $b$. It is only necessary to take into account bending when $b/h < 2$. The shear area of the lug is:

$$S_b = bh$$

and the relationship for calculating the shear stress is:

$$\sigma_s = \frac{F_p}{2S_b}$$

where the force on the chamber base with a diameter of $d$ is:

$$F_p = \sigma_p \frac{\pi d^2}{4}$$
The allowable shear stress, \( \sigma_s \), for the material is usually taken to be the shear stress at the yield point in the elastic region. A factor of safety of \( \sqrt{3} \) to 2 is normally used so the allowed value of shear stress for the lugs, \( \sigma_{os} \), is:

\[
\sigma_{os} = \frac{\sigma_s}{\sqrt{3} \text{ to } 2}
\]

Thus the minimum shear area of the lug is:

\[
S_2 = \frac{P_h \cdot \pi \cdot d^2}{8 \sigma_{os}}
\]

To ensure minimum wear of the locking surfaces a sufficiently large surface area is chosen.

The compressive stress acting on the lugs is therefore:

\[
\sigma_{oc} = \frac{\pi \cdot d^2}{bb \cdot h} P_n
\]

This stress should not exceed a value selected according to the type of automatic system, especially the pressure at which the breech unlocks. For automatic weapons this stress ranges from 250 to 600 MPa. Lower values are chosen when unlocking occurs under pressure, which is usually about 400 MPa. Minimum compressed area of one lug is:

\[
S_{oc} = \frac{\pi \cdot d^2}{8 \sigma_{oc}} P_n
\]

The minimum dimensions of the lugs can be found from equations (7.17) and (7.19) and ensuring that \( \frac{h}{b} \geq 2 \).

### Tilting and Linear Breeches

The stresses in both the above breeches are similar and the calculation of the actual values is also similar. It is first necessary to determine the perpendicular forces acting on the individual locking surfaces. The cross-section, \( S_{lb} \), of the breech block is determined by checking the pressure from:

\[
\sigma_{com} = \frac{P_b}{S_{lb}} \leq \sigma_{ot}
\]

For a breech with locking plates it is necessary to check the plate for buckling if its length, \( l \), in relation to its thickness, \( h \), is:

\[
h/l < 1/8
\]

Calculation of the compressive stress, which is used for determining the size of the locking or bearing areas, is usually the determining factor in calculating locking lug size. To determine the forces acting on the individual areas the same approach is used as for a tilting breech or breech with locking plates.

Referring to Fig. 7.35, the force acting on the breech block is:

\[
F_D = \frac{\pi d^2}{4} P_n
\]

This force also acts on the contact area of the breech block and the locking piece. The unknown forces acting at normal to the locking faces, \( F_N \) and \( F_w \), can be found from:

In the direction of the axis \( x \):

\[
F_D - F_N \cdot \cos \alpha - f \cdot F_N \cdot \sin \alpha - f \cdot F_w = 0
\]

In the direction of the axis \( y \):

\[
F_D + f \cdot F_N \cdot \cos \alpha - F_N \cdot \sin \alpha = 0
\]

From equation (7.24) an expression for \( F_N \) can be found:

\[
F_N = F_D (\sin \alpha - f \cdot \cos \alpha)
\]

and substituting (7.25) into (7.23) and rearranging:
\[ F_N = \frac{F_0}{\cos \alpha + 2f \cdot \sin \alpha} \]

The friction between the breech block and the locking piece is small and can be neglected. Also, only a small error will result by neglecting the friction force \( f F_0 \). If the angle \( \alpha \) is small and \( \tan \alpha \leq f \) the bearing of the locking lug will be self-locking and \( F_N = 0 \) and \( F_N = F_0 \).

The individual compression areas are:
- locking piece-breech casing
\[ S_{co} = \frac{F_N}{\sigma_{co}} \]
- locking piece-carrier
\[ S_{co} = \frac{F_0}{\sigma_{co}} \]
- locking piece-breech block
\[ S_{co} = \frac{F_0}{\sigma_{co}} \]

The most important requirement is to determine the locking area between the slide lock and the casing using equation (7.27).

\[ \text{Wedge Breech} \]

The body of the wedge has such a complex shape that it is difficult to calculate the stresses during firing. When designing the weapon the dimensions of the wedge are determined using Table 7.1 and then the stresses checked afterwards. When checking the stress, the wedge is considered to be a beam positioned on two supports and stressed by the pressure of the cartridge case base as shown in Fig. 7.36.

To simplify calculations the continuous pressure of the cartridge case is assumed to act in the centre of gravity of the half-circles of the cartridge chamber base, which is \( 0.21d \) from the barrel axis, and the magnitude of the forces is \( F_0/2 \). The beam is exposed to the maximum bending moment when:

\[ M_{max} = \frac{F_{max} \cdot a}{2} - 0.21d \]

Maximum bending stress is determined from the bending moment and the section modulus, \( l \).

\[ \sigma_{max} = \frac{M_{max}}{l} \leq \sigma_0 \]

Figure 7.35 Forces acting on tilting and linear breeches

Figure 7.36 Forces acting on a wedge breech
Military Small Arms

The design bending stress, \( \sigma_{br} \), for normally used steels is in the region of 200 MPa.

The wedge contains many openings and recesses and so the section modulus is determined as follows:

\[ i = \frac{1}{7} b \cdot h^2 \]

where \( b \) = the length of the wedge.

By substituting equations (7.32) and (7.30) into equation (7.31) and rearranging, the following expression can be derived:

\[ \sigma_{\text{max}} = 2.75 \left( \frac{\sigma_{\text{br}} - 0.21 d}{b \cdot h^2} \right) P_{\text{max}} < \sigma_{L} \]

The safety factor for the wedges is chosen with respect to the concentration of the stress on the edges of openings and is usually taken as 2.5.

The height of the wedge, \( h \), determines the carrying capacity of the wedge and can be approximately determined as follows:

\[ h = (0.8 \text{ to } 1.0) \frac{d}{2} \sqrt{\frac{P_{\text{max}}}{K_{\text{br}}}} \]

All other dimensions are taken from Table 7.1. In addition to bending stresses the rear of the wedge is also subjected to a compressive stress, which is dependent on the contact area of the wedge, \( S_a \). The compressive stress is:

\[ \sigma_c = \frac{F_h}{S_a} \leq \sigma_{\text{max}} \]

An accurate method of calculating the stresses in the breech wedge is to use the finite elements method. For the simplest case, it could be sufficient to use basic space elements (triangular prism or tetrahedron) with linear approximation of movement inside the element. The method is complex and requires the use of a computer.

Automatic Operating Systems

Automatic weapons complete all operations of the functional cycle without any action by the crew and are capable of firing bursts. The duration of a burst is determined by the fire, mode of fire, and magazine capacity. For short target exposure times the rate of fire requirement is continually increasing. This requirement can be fulfilled in two ways:

- increase the velocity or shorten the displacement of the weapon's moving parts
- simplify the functional cycle or operate more cycles simultaneously using several barrels or chambers.

These methods are used with conventional high rate of fire automatic weapon systems. For conventional automatic weapons the whole functional cycle occurs in the interval between each shot being fired. The main component of the system, or that part of the breech block driven by this element, should have a stroke greater than the length of the cartridge so that the cartridge can be fed into the chamber. The velocity of the main element of the system is limited to below 15 m/s when handling a cartridge or cartridge case and is the deciding factor governing the rate of fire. The history of automatic weapon design shows an effort to achieve both the minimum stroke and minimum mass of the reciprocating parts of the main element of the automatic system. Fig. 8.1 shows the different types of conventional automatic systems.

![Figure 8.1 Different types of conventional automatic systems used with small arms](image)
Weapons with very high rates of fire use systems where the rate of fire can be increased without raising the velocity and acceleration of the operating mechanism. The design of the system makes it possible to fire the next shot before all of the functional cycle has been completed. These systems have also been developed to use a simplified functional cycle by changing the ammunition to caseless ammunition or by using different shaped cartridges, and are shown in Fig 8.2.

**BLOW-BACK WEAPONS**

 Blow-back weapons make use of the propellant gases that act on the breech block through the cartridge case and are commonly used with weapons of low or medium ballistic power, such as self-loading pistols and sub-machine-guns.

 During firing the breech is not locked to the barrel. The system relies on the inertia of the breech block and the force applied by the return spring to resist the opening force of the propellant gases. Pure blow-back therefore requires a heavy breech block, which is unsuitable for anything other than low-pressure cartridges. The functional diagram is shown in Fig 8.3. To overcome this problem the blow-back system is modified to increase the resistance of the breech block to the opening force of the propellant gases. The aim of these modifications is to reduce the mass of the breech, and hence the weapon, for cartridges with greater performance.

![Figure 8.2 Different types of high rate of fire automatic systems used with small arms](image1)

---

For a locked blow-back breech, the breech, when unlocked, will have some residue propellant gas pressure in the cartridge case. A pressure pulse acts on the breech and accelerates it to the rear.

The unlocked breech begins to move when the projectile is released from the cartridge case at \( t = t_0 \) (point 0). At point 1 the breech block reaches maximum velocity which then decreases, because of the increasing force applied by the return spring, until it strikes the rear of the weapon casing (point 2). Cartridge feed starts when the breech passes the rear of the cartridge free space between the barrel and the breech. Feeding must be completed within time \( \Delta t_r \), which is before the breech block returns to the base of the cartridge being fed from the magazine. After rebounding from the weapon casing (buffers are not usually fitted) the breech block returns to the front position feeding a cartridge into

![Figure 8.3 Functional diagram for an unlocked blow-back breech](image2)
the chamber as it does so. When the driving force of the return spring and resistance against feeding the cartridge (point 3) are equal, the breech block reaches its maximum forward velocity. At the forward position the breech-block strikes the barrel (point 4) and the cartridge is initiated either immediately with a fixed firing pin or after the striker is released. The next functional cycle starts at point 5 when the projectile of the next cartridge starts to move. Thus the breech block does not move during the interval 4-5. The start of the functional cycle between points 0-1 is expanded in Fig. 8.4 which shows that firing takes place at the very beginning of the functional cycle. One complete functional cycle takes about 300 times longer than the time to fire the cartridge, which takes approximately 1-1.5ms. The rate of fire of these weapons ranges from 500 to 800 rounds per minute. During firing the breech block receives sufficient kinetic energy to power the first part of the functional cycle and for rebound. However, the driving force for accelerating the breech block, $F_{r}$, is many times greater during firing than the resistance that it must overcome during its rearward displacement.

The drive to the breech block ends at point 1 after time $t_{1}$ when the driving force of the last shot is balanced by the resistance to motion of the breech block (point 1). At this point the velocity is at a maximum and the displacement/time curve shows a maximum slope, which is the point of inflection on curve $v_{bh}=x_{bh}(t)$. Maximum acceleration is achieved at $v_{bh \text{ max}}$. Maximum velocity, $v_{bh \text{ max}}$, and maximum kinetic energy of the breech block, $E_{bh \text{ max}}$, is reached at point 1. Maximum kinetic energy of the breech block is given by:

$E_{bh \text{ max}} = \frac{1}{2} m_{bh} \cdot v_{bh \text{ max}}^{2}$

The maximum velocity of the breech block, $v_{bh \text{ max}}$, is determined from equating the breech momentum to the projectile and propellant gas momenta for the muzzle velocity, $v_{r}$:

$\frac{v_{r}}{t_{u}} = m_{bh} \cdot v_{bh} \cdot (v_{bh} + (\beta + 0.5) u_{p} \cdot v_{r} = \int F_{D} \cdot dt = n_{bh} \cdot v_{bh \text{ max}}$

The projectile mass coefficient, $\mu$, usually lies between 1.1 to 1.2.

The coefficient of after-effect for the gases emerging from the barrel, $\beta$, increases the effect of the gases' rearward momentum by acting on the muzzle, which for blow-back weapons has an approximate value of 2.

The next part of the breech block movement depends upon the consumption of energy $E_{max}$. If the energy is fully consumed by friction and the return spring, the breech block will not hit the weapon casing with any force and the impact velocity will be zero. If not all of the kinetic energy of the breech block is consumed then it will hit the weapon casing, resulting in a hard impact. It has been found that if a hard impact does not occur then the ratio of the time for the breech block to move to the rear, $t_{u}$, to the time for the breech block to return to its forward position, $t_{r}$, is approximately 0.78. If a hard impact does occur this ratio is approximately 0.48.

**GAS-OPERATED WEAPONS**

Many small-calibre automatic weapons use the propellant gases taken from ports in the barrel bore to drive the automatic system. The different types of gas system are:

- gas piston
- expansion chamber
- primer operation.

A gas piston is the most frequently used system. It consists of a cylinder connected through a gas port with the barrel bore and a piston positioned in a cylinder. Fig. 8.5 shows a schematic arrangement of this type of system. When the projectile passes the gas port the propellant gases enter the cylinder where they impart a force to the piston. This force is transmitted to the automatic system, usually to the breech block carrier.
The expansion chamber operation is shown in Fig. 8.6. For this system no ports are drilled in the barrel, the gases flowing from the barrel muzzle after the projectile has left the barrel are used to power the system. The expansion chamber consists of a sleeve mounted on the muzzle which can move in the direction of the barrel's longitudinal axis. There is a space before the muzzle where the gases expand and pull the expansion chamber forward. The resultant movement is transmitted through a lever to the breech block carrier. This principle was used for several weapons built between the World Wars. However, the system is heavy and the mechanisms used are intricate, so it is now rarely used. In the primer driven system the function of the gas ports is substituted by blast holes in the cartridge case, and the piston is replaced by the primer and striker. The system is shown in Fig. 8.7.

![Figure 8.5 Operating system using a gas-driven system](image)

![Figure 8.6 Operating system using an expansion chamber at the muzzle of the weapon](image)

The front end of the firing pin, after hitting the primer, bears against the primer and its rear end bears against the breech block carrier. A small portion of propellant gas flows through blast holes in the primer pocket to the movable primer, which pushes the firing pin rearwards until it strikes the breech block carrier. The carrier unlocks the breech block and moves with the cartridge case in the rearward direction. This principle, which has been used in the past, has a disadvantage: it requires special cartridges with a moving primer and the firing pin has a very short service life, although the system is very light in weight. In modern designs, only the gas piston operating system is used except for special purpose weapons. This system is very reliable and is insensitive to barrel wear. The design is divided into two types:

- moving piston with fixed cylinder
- fixed piston with moving cylinder.

Fig. 8.8 shows the system used with the Czech Model 58 automatic rifle. A gas block is mounted on the barrel. A cylinder is machined in the gas block which is connected to the barrel bore through an oblique gas port. A movable piston is housed in the cylinder. Another system used in the Czech Model 59 universal machine gun is shown in Fig. 8.9. In this design, the gas block is placed...
under the barrel with the gas port drilled into the barrel perpendicular to its axis. The port supplies propellant gases into the cylinder through a regulator. The regulator makes it possible to change the cross-sectional area of the port through which the gases flow to compensate for different ammunition natures, fouling of the gas port or adverse weather conditions.

To reduce the length and weight of the piston rod, the gases can be fed through a long tube, as shown in Fig. 8.10. However, the volume of gas required to operate this system is considerably increased.

A somewhat different gas piston system was used for the 7.92mm German G-41W self-loading rifle, as shown in Fig. 8.11. In this design, the barrel had no gas port: the gases were taken from the muzzle in a similar way to that shown in Fig. 8.5. Whilst there was also a chamber before the muzzle it did not move. This chamber acted as the cylinder of the gas system. Inside the cylinder was a piston which was slide-mounted on the muzzle of the barrel. Gases leaving the muzzle after the projectile expanded into the cylinder and pushed the piston backwards. The piston rod, mounted on the barrel, transferred this motion to the breech block carrier.

A fixed piston (with movable cylinder) is used in the 12.7mm NSV machine-gun as shown in Fig. 8.12. The piston is created directly on the body of the gas block and the gas port passes through it. The gas cylinder is connected to a rod which transfers motion to the breech block carrier.
grooves of different depth which connect the gas port with the cylinder. The regulator is adjusted for a small flow area; by turning it in the direction of the arrow the flow area is increased. Better regulation of the propellant gas is achieved using three- or four-way regulators. A three-way regulator of similar design to that in Fig. 8.9 is used in the Russian PK and SG 7.62mm calibre machine-guns. Fig. 8.14a shows a regulator used in the 7.92mm calibre Czech Model ZB 37 medium machine-gun. This machine-gun has ports through its cylindrical core which are used instead of grooves. The regulator of the Czech model 85 14.5mm calibre machine-gun uses a different method to regulate gas flow, as shown in Fig. 8.14b. In this design, the regulator is the gas cylinder which is provided with two vents and can pivot in the gas block. Adjustment is made by pivoting the cylinder around its longitudinal axis.
For gas-operated automatic systems the rate of fire can be raised by using a lever system to increase the rearward velocity of the breech block, as in the 12.7mm calibre Russian A-12.7 aircraft machine-gun shown in Fig. 8.15. The steep angle of the breech block functional diagram shows that the breech block reaches a high mid velocity which reduces the time to complete one functional cycle. If the breech is unlocked early the residual pressure in the chamber further accelerates the breech.

**EXTERNALLY-POWERED WEAPONS**

An example of an externally-powered weapon is the Hughes chain-gun which is fitted as the coaxial machine-gun on the British Warrior Infantry Fighting Vehicle. Energy is supplied by an external electric motor which drives a chain mounted on four sprockets housed in a breech casing. The chain moves with constant speed. Attached to one of its links is a pin which carries a sliding block. The sliding block moves in a transverse groove in the breech carrier. The chain moves the breech carrier backwards and forwards and provides stationary periods for firing and feeding the cartridge. Firing forces are transmitted to a short barrel extension via the rotating breech bolt. The drive system has none of the firing stresses transmitted to it. Fig.8.16 shows the functional diagram for the chain-gun system.

The time taken to complete one cycle is shown on the functional diagram:

\[
T = 2\Delta t + t_1 + t_2 + t_3 + t_4 + t_5 + t_6
\]

because

\[
t_1 = t_3 = t_4 = t_5 \quad \text{and} \quad t_2 = t_6
\]

then

\[
T = 2(\Delta t + t_2 + 2t_1)
\]

The advantage of the chain-gun operating mechanism is that the accelerations and velocities are much lower than in conventional operating systems, with no impacts suffered by the working parts. The drive system does not depend on the weapon being fired and so does not stop if there is a cartridge misfire; the misfired cartridge is ejected from the weapon along with the empty cartridge.
case. The rate of fire can be changed by varying the chain velocity, which is controlled by the electric motor. However, it is necessary to provide an external power source; in the 7.62mm calibre chain-gun the power requirement is 300W. It is also possible to damage the weapon if a hang fire occurs.

RECOIL-OPERATED WEAPONS

In weapons using recoil to cycle them, the recoiling barrel drives the operating mechanism. If the recoil length of the barrel is longer than the cartridge, then it is known as long recoil operation. If the distance that the barrel recoils is shorter than the cartridge length, then it is known as short recoil operation. Weapons driven by recoil were the first successful automatic systems for small arms and are used in automatic weapons to this day. These types of automatic systems have the greatest advantage when used with larger calibres where the recoil energy is sufficient to drive the whole operating system.

Long Recoil-operating Systems

In long recoil-operating systems the barrel moves with the breech block to the rear of the weapon as shown in Fig. 8.17. The breech unlocks and is held to the rear and the barrel returns to the front position, the cartridge case being extracted as it does so. The next cartridge is then fed into the chamber and the breech is closed by the action of the return spring. Fig. 8.17 shows the diagrammatic representation of the system and the time/displacement diagram.

This is a simple and reliable system but one that can only achieve low rates of fire, because each part of the cycle occurs in series and there is a long duration of the recoil and counter-recoil of the barrel. Moreover, maximum velocity of the recoiling parts is limited to values of about 12 m/s because of the mass of the projectile and recoiling parts, and the counter-recoil is much more slower. Therefore, time \( t_1 < t_2 \). The total time of the functional cycle, shown in Fig. 8.17, is:

\[
T = t_1 + t_2 + t_3 + t_4 + \Delta t
\]

Times \( t_1 \) and \( t_2 \) are limited by the velocity of the barrel which is determined by the length of recoil. This length is greater than the length of the cartridge.

Short Recoil-operating System

In order to reduce the time of the functional cycle for recoil-operated weapons, the barrel recoil length needs to be reduced. To reach sufficient breech block velocity and to increase the opening distance of the breech for only a short movement of the barrel an accelerator is used between the barrel and the breech. Fig. 8.18 shows the functional diagram for the short recoil system.
After the breech has reached maximum velocity (at the end of the accelerator action), it continues to move due to inertia until it reaches maximum opening distance. It can be seen that the functional cycle time is lessened by reducing the times $t_1$, $t_2$ and by transferring time $t_b$ to time $t_2$. The functional cycle time is:

\[ T = t_1 + t_2 + t_{mb} + \Delta t \]

The use of an accelerating lever reduces the mass of the moving parts and the overall mass of the weapon. The accelerators are usually of the mechanical type. However, hydraulic or hydro pneumatic accelerators have several advantages, including a modified acceleration/time curve and reduced mass, especially for larger calibre weapons. Mechanical accelerators include:

- lever-operated: impact
- lever-operated: rolling (continuous mesh)
- cam-operated
- spring-operated.

Lever-operated accelerators, which are used most frequently, are shown in Fig. 8.19.

![Diagram of lever accelerators](image)
The rolling lever-operated accelerators have a control profile which, by the position of the barrel, determines the transmission ratio between the barrel and the carrier \( i = \frac{b_1}{b} \), either on the accelerator proper (Fig. 8.19a) or on the weapon casing (Fig. 8.19b). For impact lever-operated accelerators the adjacent element is accelerated by the impact of the main element over the ratio \( i = \frac{2}{3} \) (Fig. 8.19c). Forces \( F_6 \) and \( F_{6\theta} \), by which the accelerators act on the barrel and breech, are internal forces of the system (Fig. 8.19a), and by using an equivalent mass and forces they can be excluded from the equations of motion. Cam-operated accelerators have a shaped profile in contact with the surface of the driving and driven lever. The control cam is usually of a helicoid shape. This type of lever is typical in short recoil automatic systems and is used between the breech block carrier and the breech. A cam-operated accelerator is shown in Fig. 8.20. The breech block carrier is accelerated by the action of the barrel via a pin passing through the breech block on the accelerating profile in the breech block carrier. It is necessary for a slight turning of the accelerator pin because the carrier’s rearward acceleration is implemented through control grooves in the barrel casing. The turning of the accelerator occurs simultaneously with the unlocking motion of the breech block.

Fig. 8.21 shows a spring-operated accelerator, which is not used very frequently. The spring of this accelerator is positioned between the recoiling barrel and breech block. The breech block is also driven by the force of the spring so that its velocity is higher than that of the barrel.
High Rate of Fire Systems of Automatic Weapons

GATLING SYSTEM

In 1861 Richard Jordan Gatling (1818–1903) introduced a weapon with several barrels grouped together that rotated by means of a hand crank with each barrel firing in turn. In its fully developed state, it was capable of rates of fire approaching 1,000 rounds per minute. Within 30 years, this type of weapon was superseded by the self-powered, single-barrelled machine-gun. However, 80 years later this idea became the basis of a group of externally-powered automatic weapons with very high rates of fire known as the Gatling system. The principle of this system consists of joining several barrels in a group rotating about a longitudinal axis in a fixed weapon casing (Fig. 9.1). Each of the barrels must complete one revolution of the barrel group to complete the functional cycle. The cycles of the different barrels are displaced by one barrel pitch, which is through an angle of $2\pi/n$ radians ($n$ being the number of barrels in the group). The interval between shots corresponds to the time taken for the group to turn through such an angle. After pressing the trigger the gun begins rotating until it reaches the operational rotational velocity. The breech blocks move axially along a guiding groove in the weapon casing. During one revolution of the barrel group each breech block moves backwards and forwards in turn with delays at the front locked position for firing and at the rear position for case ejection and feeding the next cartridge. Breech block movement is controlled by a cam formed as a groove in the inner surface of the weapon casing. Each of the breech blocks is provided with a guide with a roller which moves along the groove of the cam as shown in Fig. 9.1. When the barrel group rotates the breech casing remains stationary.

The barrel group drives the feed mechanism and ensures that case ejection and feeding of the next round in front of the corresponding breech block occurs with the breech block in its rear position. Energy for turning the barrel group is usually provided from an external source, such as an electric motor, or from the propellant gases through ports in the barrel.
Barrel Group

The barrel group has between three and seven barrels joined with clamps at the muzzle, at the breech and mid-way between as shown in Fig. 9.2.

The rear clamp is the most massive, with the barrels pressed or screwed into it to take the firing forces. The cartridge chambers may be machined into each barrel or into the rear clamp as shown in Fig. 9.3. The main task of the middle and front clamps is to secure the barrel group and to give it rigidity. The clamps are pressed or slid over the barrels, which are then gripped with bolts. They are often used for other purposes, such as carrying an auxiliary bearing of the barrel group, or for the attachment of the weapon's driving elements.

The bearing taking the firing force is located on the circumference of the rear clamp and is the largest bearing of the whole system. The barrel group rear bearing is located at the end of the breech carrier as shown in Fig. 9.4. The number of barrels used depends on the required rate of fire. For increased barrel life and reliability of the different weapon components the rate of fire per barrel is restricted to similar usual rates of fire for conventional automatic weapons, which is between 600–1,000 rounds per minute.

Gatling guns are very efficient, have a high-firing output, relatively long service lives without excessive maintenance requirements and one of the highest operating reliabilities.

Breech Block Cam

The cam for the breech block has two delays. The rear delay retains the breech in the rear position for the time necessary to eject the cartridge case and feed the next cartridge in front of the breech block. The forward delay ensures that the breech is locked during firing and that the pressure of the gases inside the barrel has time to drop to a safe level before unlocking occurs.

Between these two delays, the cam moves the breech block forwards to chamber the cartridge and backwards to extract the cartridge case. Each cam has three sections: a section for acceleration, a straight section of uniform velocity and a section for deceleration. To ensure a uniform load on the cam and the guide the section for acceleration is three times longer than that for...
deceleration. The section of uniform velocity is not absolutely necessary but it helps in controlling the forces and velocity of the breech.

An essential requirement of the cam design is the angle that the breech group turns through, $\phi_0$, whilst the breech block is in its forward position. The breech must be locked during firing and when the chamber pressure is high. The breech must remain locked for the time $t_p$ for the gas pressure to fall to that of atmospheric. If the barrel is rotating at an angular velocity of $\omega_b$, then:

\[ \phi_0 > \omega_b \cdot t_p \]

There are variations in primer ignition and combustion of the propellant charge, so sufficient reserve must be included for safe operation of the breech.

For the accelerating and decelerating elements of the chambering and extraction segments of the cam, parabolic curves are used. These curves give, at constant barrel velocity, constant acceleration of the breech block.

**Breech Block**

Most Gatling gun systems use rotary breech bolts. The locking and unlocking of the breech is controlled with a helical groove in the breech block and a pin in the carrier, or a fixed dog in the weapon casing. To reduce the passive resistance of the breech, the breech block carrier is often mounted on rollers (usually three) in the breech casing guides, as shown in Fig. 9.5.

**Breech Casing**

The breech casing is attached to the rear end of the barrel group forming the rotor. It guides the breeches longitudinally and transmits the forces necessary for driving the breech blocks. When the drive for the weapon is located at the rear the breech casing also drives the barrel group. The casing must be strong and rigid so that flexing of the breech block cam does not affect the movement of the breech blocks; it must also allow the ejection of the cartridge case and the feeding of the next cartridge. In order to reduce the moment of inertia with respect to the longitudinal axis to a minimum, the breech casing is usually machined away in areas which are not critical to strength or function. Thus its shape can be complex and difficult to machine.

**Case Extraction and Ejection**

Cartridge case ejection is the same as for conventional rotating bolt breeches. The acceleration of the breech block in Gatling weapons during the cartridge case extraction is much lower than in conventional weapons so that the stresses in the extractor and the cartridge case are also lower.

**Gun Drive**

The operation of a Gatling gun relies on the rotation of the barrel group, which is the main element of the system. To begin firing, the barrel group must be rotated around its longitudinal axis and then the angular velocity maintained to give the required rate of fire. The acceleration of the gun to the full rate of fire usually takes from 0.3s to 0.5s, so that the applied torque at the start-up is much higher than during the full rate of fire period.
Gatling weapons use either an external drive or are self-powered. The advantage of an external drive is that weapon design is simplified and the rate of fire can be altered at will. However, the size of the required power source is usually large, which restricts these weapons to being vehicle-mounted.

**OPEN CARTRIDGE CHAMBER (DARDICK SYSTEM)**

The vast majority of modern small arms use conventional bottle-shaped cartridges with metallic cases. However, weapons using this type of cartridge need a long feed and extraction stroke, and the functional cycle is also complex because there are many moving parts. To increase rate of fire, performance and reliability, and to overcome the additional problem of the increase in volume of ammunition needed for feeding these weapons, a new approach to ammunition and weapon design is needed.

One approach is to use open cartridge chambers. This type of weapon was developed in 1949 by David Dardick. He developed several pistols, a heavy machine-gun and a high rate of fire cannon using open chambers. The principle of the open cartridge chamber (Dardick) system is shown in Fig. 9.6.

A cylinder with several open cartridge chambers is used. The cartridges are fed with a magazine or a belt and are rammed from the side into one of these open cartridge chambers. The chamber cylinder rotates and moves the cartridge below a reinforced wall on the breech casing, called a yoke, which overlaps the open side of the cartridge chamber. The base of the chamber is closed with a standing breech, in which the firing mechanism is located.

The axis of the projectile within the loaded chamber is aligned with the barrel and the cartridge is fired. The chamber cylinder rotates to align the next cartridge with the barrel, thus moving the fired chamber to the position where ejection takes place.

Such a weapon can be designed with more than one barrel. The chamber cylinder is a component of the barrel group. The arrangement for such a design is shown in Fig. 9.7. The chamber cylinder rotates continuously together with the barrel group. The open side of the cartridge chamber must be closed during the movement of the chamber by the yoke of the breech casing, which is part of the weapon casting. The breech casing is also more rugged and the surface of the cartridge case is pressed against the yoke during firing. Thus there is a high friction moment opposing the movement of the barrel group which increases the power requirements from the weapon drive. Hence the cartridge case material needs to have a low frictional coefficient.

The method of driving the weapon may be similar to one firing conventional bottle-shaped cartridges. In a two-barrelled Dardick design, it is usual for both barrels to be fired together so as to ensure that the firing stresses are evenly distributed. Bending stresses on the cylinder or the stresses on the bearing only arise if a misfire occurs in one of the chambers (see Fig. 9.7). It can be seen that the smallest number of chambers is six.

In the Dardick system there are no long axial movements of the breech, the cartridge or the cartridge case and there are no high accelerations of the components. The cartridge is held in place by its side surfaces and the loading
movements are short. A large number of small, complicated and highly stressed components, normally associated with automatic weapons, have been eliminated and the design is simple and highly reliable.

Ammunition for the Dardick system has a case made in the form of a tridecal prism with rounded edges. The cartridge consists of a cartridge case, a projectile, a primer, a cap and a propellant charge, as shown in Fig. 9.8.

The cap forms a front cover for the cartridge case, supports the projectile and guides it towards the bore of the barrel during firing. The mechanical or electrical primer is pressed into the cartridge case base. If a fin-stabilised sub-calibre projectile is used, as shown in Fig. 9.8, a sabot is needed to seal the bore and guide the projectile through the barrel. It is discarded after the projectile leaves the muzzle.

The case material must be malleable in order to seal the chamber cylinder and locking yoke. However, it must also be rigid and strong enough to withstand the loading forces. In addition, it must possess good high temperatures fiction characteristics when used in weapons with more than one barrel. There are several plastic materials satisfying these requirements, and hardened cardboard (similar to the material used in shotgun cartridge cases) is a possibility.

Figure 9.8 Typical Dardick cartridge with fin-stabilised projectile

10

Ammunition Feed

PURPOSE OF THE FEED MECHANISM

For a weapon to function, it must be provided with an appropriate supply of ammunition. For automatic weapons the supply must also be continuous so that the weapon can fire bursts until all ammunition in the weapon storage is consumed, without the intervention of the crew. The feeding of cartridges has two phases:

- the supply of cartridges into the weapon feed mechanism
- the feeding of cartridges into the loading space.

The loading space holds the cartridge prior to insertion into the cartridge chamber and occupies an area in front of the breech block when the breech block is in the rear position. In general, the feeding mechanism of an automatic weapon has the following three main parts:

- ammunition box
- method of supply of cartridges into the feed mechanism
- feed mechanism.

The purpose of the ammunition box is to stow the required amount of ammunition. The ammunition stowed in the box must be easily removed by the supply mechanism and the accelerated mass of the ammunition should be as small as possible. However, the ammunition needs to be properly fixed in the box so as to prevent the cartridges bumping against each other or even against the box walls.

The cartridge supply system must deliver ammunition to the weapon under all conditions of traverse and elevation. The feed mechanism is the terminal point of the weapon feed system and links directly with the other mechanisms of the weapon. Its prime function is to provide a flow of cartridges synchronised with the movement of the breech block which moves the cartridges into the cartridge chamber. The two types of feed system normally used with small arms are magazine feed and belt feed.
MAGAZINE FEED

Principles of Magazine Feed

Spring-operated magazines have three main parts: box, spring and follower, as shown in Fig. 10.1.

When cartridges are inserted into the magazine the spring is compressed, thus storing energy to feed the cartridges. The spring then moves cartridges towards the loading space, which is the mouth of the magazine. When the breech moves forward it removes a cartridge from the magazine and feeds it into the chamber. The magazine contains a number of cartridges which the spring must move during each functional cycle; consequently a cartridge will be in position in the loading space in time to be fed into the chamber by the forward moving breech block. The shape of the magazine box depends on the type of magazine and the number of cartridges it contains.

The requirements of magazine construction are:

- It should allow continuous and correct movement of cartridges inside the magazine.
- It should be easy to handle.
- Cartridge capacity should be as great as possible to achieve a high combat rate of fire.
- Cartridges should be held firmly in the magazine mouth (loading space) without excessive force.
- Cartridges in the magazine must not change their position during transport.
- Filling must be quick and easy.

- The magazine should be light and strong.
- It must be simple in construction and cheap to make.

Reliable feed from the magazine is dependent on the mouth of the magazine holding each cartridge in the loading space and directing it towards the cartridge chamber. The side plates of the magazine are bent over at the mouth to form lips to contain the cartridges, the length of the lips being 40–60% of the cartridge length. The gap between the lips is 75–95% of the maximum cartridge case diameter for a single-row opening and 110–130% of maximum cartridge case diameter for double-row opening. The allowable movement of the front part of the cartridge in the magazine mouth should be restricted to ensure that the ogive of the bullet is directed towards the chamber. As the cartridge base leaves the magazine opening, the bullet must already be in the chamber. Cartridges are usually guided in two guide ways formed in the magazine side plates, one in the front and the other in the rear, as shown in Fig. 10.2a and b. The example shown in Fig. 10.2b guides the cartridge at the front by locating on the cartridge case rather than the bullet, which means the magazine can also be used for firing blank rounds during training. The axial clearance between the cartridge and the magazine side plates is about 1mm.

The cartridges in double-stacked magazines are arranged as shown in Fig. 10.3. This is the best use of magazine space and makes the magazine narrower than when the cartridges are stacked side by side. It is important for the magazine to be precisely positioned in the weapon each time and it should be fitted with positive stops in the direction of insertion. Folded leaf springs, helical springs coiled in the shape of the magazine box, helical cylindrical springs in small magazines and spiral springs for disc and drum magazines are used. Magazine springs are not dynamically stressed so that with a full magazine the spring can be compressed to a maximum force with minimum clearance between the coils or leaves. The follower in the magazine must transfer the force of the magazine spring to the cartridges and help to position the cartridges in the mouth of the magazine.

Figure 10.1 Main parts of a spring magazine

Figure 10.2 Different ways of guiding cartridges by using guide ways formed into the magazine side plates
Box Magazine

Box magazines have a case in the shape of a box and can be straight sided or curved, as shown in Fig. 10.4. Straight sided magazine cases are only used for low capacity magazines if bottle-shaped cartridges are used: curved magazine cases are used for larger capacity magazines. Curved magazines cases give better transfer of the force of the spring through the follower to the cartridges in a direction perpendicular to their axes for each cartridge. Curved magazines are fitted to the Czech model 58 automatic rifle and the Soviet AK47 and AK74 automatic rifles and the American M16 automatic rifle.

Drum Magazine

These magazines have a case the shape of a drum. The cartridges are positioned so that their axes are approximately parallel with the axis of the drum which, in turn, is approximately parallel with the axes of the barrel. A drum magazine is shown in Fig. 10.5.

Drum magazines are fitted with a spiral spring. One end of the spring is connected to the case and the other to a rotary shaft. Connected to this shaft, through a knuckle joint, is the follower which moves in a spiral channel in the magazine. Double-drum magazines with one mouth, as shown in Fig. 10.6, have also been developed. Drum magazines can hold large numbers of cartridges. However, they are more complicated and more expensive to make than box magazines. Also, they are heavy, especially when fully loaded, and filling them can be difficult.

Disc Magazine

The disc magazine is also in the shape of a drum. Cartridges are positioned radially, pointing inwards with their axes approximately perpendicular to the magazine case axes and approximately perpendicular to the axes of the barrel. Fig. 10.7 shows this type of magazine. The spring is of the spiral type, when
Disc magazines are fitted to the British Lewis machine-gun (which is gun-driven and therefore does not require a magazine spring) and the Soviet DP light machine-gun. The advantages and disadvantages of disc magazines are similar to those of drum magazines.

**Tubular Magazine**

Tubular magazines are rarely used with military automatic weapons; they are mostly confined to self-loading shotguns. The case of this type of magazine is tube-shaped. Cartridges are positioned one behind the other. A helical cylindrical spring bearing against the follower moves the cartridges in the direction of the tube axes. The tube is positioned below the barrel, or in the stock of the weapon. A tubular magazine is shown in Fig. 10.8.

A major disadvantage of tubular magazines is that the cartridges are positioned with the nose of one being torched into the tail of another by the magazine spring. If the cartridges are centre fire and the bullets have a sharp nose it is possible for the primer to be detonated. Thus the magazine is only used with rim fire cartridges or centre fire cartridges with blunt noses.

The box magazines most frequently used are inserted into the weapon from below, although some weapons are designed with magazines which fit into the weapon from above or from the side. Drum magazines are positioned below or above the weapon, but can also be fitted on the side of the weapon. Disc magazines are only fitted to the upper part of the weapon casing.

**FEED TIMING**

When designing a magazine it is necessary to ensure that the magazine spring provides the correct timing of the feeding of the cartridges. Cartridge feed may only start when the breech block head, with the breech travelling rearwards, passes the base of the cartridge in the mouth of the magazine. Cartridge feed must be completed before the breech block returns to this position, which is

![Figure 10.7 Disc magazine](image)

![Figure 10.8 Tubular magazine](image)

![Figure 10.9 Critical timing of feed for box, drum and disc magazines](image)

the requirement for box, drum and disc magazines as shown in Fig. 10.9. In tubular magazines, with a platform that positions the cartridge on the barrel axes, the time to move the cartridge is substantially longer, so the spring force is not critical.

The cartridge must be fed through the distance \( s \), in the time:

\[
\tau_p \leq t_{bb}
\]

where \( \Delta t_{bb} \) is the time for the breech block head to travel from the base of the cartridge (in the mouth of the magazine) rearwards through the distance \( x \) and return to the base of the cartridges. The time, \( \tau_p \), is required to move the car-
tricle column in the magazine through the distance \( s \), in box magazines. In a drum or disc magazine it is the time to move cartridges along the curve by one pitch. The time \( t_p \) is calculated using Newton's Laws of Motion. For a straight box magazine, when calculating time \( t_p \), the effect of the mass of the accelerated parts (follower and cartridges) is taken into account according to the position of the magazine in the weapon. If the magazine is fitted to the weapon from below, there is a mass acting against the force of the magazine spring, but when inserted from above, the mass helps the spring. When inserted from the side, the mass acts perpendicular to the direction of the force of the magazine spring and thus causes friction between the moving parts in the case. For vertical positioning of the magazine the friction is negligible. For curved box magazines the method of calculation is similar. In drum magazines the cartridges are positioned in a spiral channel, which results in significant frictional forces in the channel which are also enhanced by the centrifugal forces. This friction cannot be neglected, but it is difficult to calculate. The friction differs from cartridge to cartridge because it depends on the curvature of the channel as well as the position of the cartridges in the spiral channel. Fig. 10.10 shows forces acting on the \( i \)-th cartridge in a spiral channel.

Fig. 10.10a shows that forces \( F_{i+1} \) of the front cartridge and \( F_{i-1} \) of the rear cartridge act upon the \( i \)-th cartridge (taken from the magazine mouth). These forces are due to the curvature of the channel inclined against each other so that their resultant \( F_i \) (Fig. 10.10b) pushes the cartridge towards the channel wall by its normal component. The centrifugal force of the cartridge acts in the same direction. Thus the normal component \( F_{ni} \) and centrifugal force \( F_p \) are the cause of friction acting on the cartridge in the channel. To determine time \( t_p \) for drum magazines, the above value should be calculated for all cartridges in the magazine by means of the forces which the follower receives from the spiral spring, or by using equation (10.1) and by solving the force diagram in Fig. 10.10. For all of the cartridges the torque can be calculated for the spiral spring. For disc magazines the problem is much simpler. The spiral magazine spring must develop a torque to overcome the friction of the cartridges and follower, caused by the weight as well as an additional frictional force between the base of the cartridge and the case which is caused by centrifugal force. Tubular magazines are similar to the straight box magazine positioned horizontally. Calculation of the necessary force or torque of the magazine spring is carried out for at least two limiting cases: for a full magazine and for the last cartridge in the magazine.

**BELT FEED**

Individual cartridges are held in special links which form a belt. This passes through the weapon perpendicular to the barrel axes. The linked belt is housed in a special storage box. If the weapon is used for firing on the move, the storage is mounted directly on the weapon. For other applications the storage is mounted on a stand or on the carriage of the weapon. The feed mechanism always moves the belt by one pitch, which corresponds to the distance between two adjacent cartridges in the belt. The feed mechanism is part of the weapon and is driven by some moving part of the weapon.

**Types of Cartridge Belt**

Leather or canvas belts were used by early automatic weapons, but modern weapons use belts with links made from steel sheet. Cartridge belts have either closed links (Fig. 10.11a) or open links (Fig. 10.11b). With closed links the cartridge is held around the whole circumference and cannot be moved forward and directly inserted into the cartridge chamber. The cartridge must be pulled rearwards from the link, lowered and then inserted into the chamber. This type of belt is used with the Soviet 7.62mm calibre SG, model 43 (Gorjunov), SGM, PK and PKT machine-guns.

Open links hold the cartridge by only part of its circumference and are designed so that the cartridge can be pushed forwards directly into the chamber. This type of link is used more frequently than the closed one because
Fixed belts were used with early machine-guns. They were rigid and made from one piece of sheet steel, as shown in Fig. 10.13a. This type of cartridge belt is not normally used with modern machine-guns because they are difficult to control in anything other than short lengths.

Flexible, permanently connected belts are made of links connected together and may be either open or closed. Links are connected either by means of wire in the form of a helical cylindrical spring (Fig. 10.13b), or by a pin (Fig. 10.13c). Disintegrating links deliver the cartridges into the loading space and when ejected from the weapon they disintegrate into separate links, thus helping to avoid jamming of the feed system. Two examples of disintegrating links are shown in Figs. 10.13d and 10.13e. Fig. 10.13d shows a belt made of closed links. Each link has a ring on the right and two rings on the left, with a space between them. The right side ring of the next link engages with this space. As shown in Fig. 10.13d, the individual links are held together by the cartridge. Fig. 10.13e shows an open disintegrating link. Each link is provided with a hook on one side and an eye on the other. The hook engages with the eye of the next link.
To insert the belt into the weapon the first link may be provided with a steel
tongue (Fig. 10.14a), which is pushed through the feed mechanism and then
pulled out the other side of the weapon to a stop. If the belt is not provided
with this tongue then the weapon accessories may contain a hook which is
pulled through the feed mechanism against the belt movement. The hook
catches the first link and inserts it into the weapon. Modern belts with disinte-
grating links are usually placed in position by hand after first lifting the top
feed cover and closing it onto the belt to hold it in place.

**Cartridge Belt Requirements**

The cartridge belt moves by one pitch during feeding, which is the distance
between the axes of two adjacent cartridges. The greater the pitch the
greater the energy required for feeding and the more stress exerted on the feed
mechanism. The pitch of the belt is determined by the cartridge case diameter,
the shape of the links and the way they are connected. Additionally, a reduc-
tion in pitch may also lessen the flexibility of the belt. The cartridge belt of
automatic weapons must be as strong, as it is highly stressed. Cartridges must
be delivered to the loading space in exactly the right position, so need to be
accurately positioned in the belt. However, it is also important to be able to
insert them easily into the belt. Although the force required to remove the
cartridge from the link must not be too great, the cartridge should not be able to
move within the link. The belt must be flexible to allow for movement
between the weapon and belt case. Belt flexibility is characterised by a fan
shape given by radii $R_1$ and $R_2$ (Fig. 10.14a), angle of twist between adjacent
cartridges, $\alpha$ (Fig. 10.14b), and radii of roll-up $R_4$ and $R_5$ (Fig. 10.14c).

Rather than the angle of twist, the minimum number of cartridges, $n$, which can
be moved round the axes parallel to the length of belt by $90^\circ$ is used (first and $n$th
cartridges are perpendicular to each other). Thus the angle of twist is given by:

\[
\alpha = \frac{90^\circ}{n - 1}
\]

The greater the belt flexibility then the smaller are radii $R_1$ and $R_2$, $R_4$ and $R_5$
and the greater is the angle $\alpha$.

**Resistance of the Cartridge Belt**

The feeder which moves the cartridge belt must overcome the resistance to
motion of the belt which is affected by the belt's elasticity. If the belt is elastic,
the force to accelerate it is less than if the belt is rigid. The elasticity of the belt

\[
F = v_1 \sqrt{\frac{m_2}{m_1}} R_4 + (m_2 + m_1) g \sin \alpha + f \cos \alpha
\]

The rigidity of the link is a constant over its elastic deformation and is deter-
mined experimentally using a tensile testing machine.

Using the rigidity of the link, $R_1$, the resistance to movement of the cartridge
belt can be calculated using Rudnev's formula. If the force component of the
belt weight is also included (as shown in Fig. 10.15), the formula for the resis-
tance of the cartridge belt, $F$, for a belt velocity of $v_1$, is given by:

\[
F = v_1 \sqrt{\frac{m_2}{m_1}} R_4 + (m_2 + m_1) g \sin \alpha + f \cos \alpha
\]

Most automatic weapons must function reliably when firing with a treaty
hanging cartridge belt, so that $a = 90^\circ$.

Thus:

\[
F = v_1 \sqrt{\frac{m_2}{m_1}} R_4 + (m_2 + m_1) g \sin \alpha + f \cos \alpha
\]

It can be seen from formula (10.4) that the dynamic component of force $F = v_1
\sqrt{\frac{m_2}{m_1}} R_4$, does not depend upon the length of the belt, but rather on the
velocity of the parts that move the belt, the mass of the cartridge and the link
and its rigidity. To calculate the resistance to movement of the cartridge belt
when designing the feed mechanism it is necessary to know the basic characteristics of the cartridge belt. Examples of these characteristics for some automatic weapons are given in Table 10.1.

**Cartridge Belt Feed Mechanisms**

The cartridge belt is moved by means of the feed mechanism. The different types of construction are table feed mechanisms and ratchet feed mechanisms. Fig. 10.16 shows a typical table feed mechanism.

The main parts of the table feed mechanism are the slide and feed pawl which perform a reciprocating motion. There are also designs in which the slide moves in a circle. The best known example is the Soviet-made Degtyarev model 39 medium machine-gun. The feed pawl catches the link with the cartridge and moves the belt when the slide travels in the direction of the belt's motion. In the reverse movement of the slide the spring-loaded feed pawl engages with the next cartridge while the belt is held by a belt catch. This catch is also spring-loaded and only allows the belt to move in one direction.

![Diagram](image)

**Figure 10.16 Typical table feed mechanism**

<table>
<thead>
<tr>
<th>Weapon</th>
<th>Calibre mm</th>
<th>Cartridge pitch mm</th>
<th>Cartridge weight kg</th>
<th>Link weight kg</th>
<th>Link rigidity kn/m</th>
<th>( F_s ) N</th>
<th>( F_w ) N</th>
</tr>
</thead>
<tbody>
<tr>
<td>LMG model S2</td>
<td>7.62</td>
<td>16</td>
<td>0.0189</td>
<td>0.0022</td>
<td>392</td>
<td>920</td>
<td>940</td>
</tr>
<tr>
<td>MG model 1435 SG, SGMT, PK, PKT</td>
<td>7.62</td>
<td>20</td>
<td>0.023</td>
<td>0.006</td>
<td>657</td>
<td>1,240</td>
<td>1,360</td>
</tr>
<tr>
<td>ShtAKS</td>
<td>7.62</td>
<td>20</td>
<td>0.023</td>
<td>0.011</td>
<td>294</td>
<td>590</td>
<td>880</td>
</tr>
<tr>
<td>MG model 38/46</td>
<td>12.7</td>
<td>33</td>
<td>0.131</td>
<td>0.020</td>
<td>245</td>
<td>2,190</td>
<td>2,190</td>
</tr>
<tr>
<td>DSHKM</td>
<td></td>
<td></td>
<td>0.127</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>UB</td>
<td>12.7</td>
<td>34</td>
<td>0.126</td>
<td>0.037</td>
<td>186</td>
<td>640</td>
<td>1,180</td>
</tr>
<tr>
<td>Browning</td>
<td>12.7</td>
<td>24.3</td>
<td>0.119</td>
<td>0.025</td>
<td>109</td>
<td>390</td>
<td>590</td>
</tr>
</tbody>
</table>

\( F_s \) = belt yield strength \( F_w \) = belt link breaking strength

The table feed mechanism can be also designed without the slide, with the feed pawl mounted directly on the swinging feed lever. This design is used in the 7.62mm calibre Czech model 59 universal machine-gun (Fig. 10.17) and Soviet PKT machine-gun. It was also used in the Czech models 52 and 52/57 light machine-guns.

Other modifications to the table feed mechanism are employed in the 7.92mm calibre German MG 42 machine-gun. In its design the belt is moved half distance by the breech block moving back and half distance by the breech block moving forward. Two slides are used with feed pawls which move in opposite directions and the link and cartridge are transferred from one feed
The ratchet feed mechanism is a widely used for linked ammunition, and is shown in Fig. 10.19.

The belt is moved by a ratchet which rotates round its axes. The ratchet is provided with circular slots for individual cartridges. The ratchet mechanism uses a pawl instead of a belt catch, which allows the rotation of the ratchet in the direction of the belt feed only. The ratchet feed mechanism was used in the 7.62mm calibre Soviet ShKAS machine-gun and the 12.7mm calibre DShK.
model 38 machine-gun. It is also used in almost all high rate of fire automatic weapons of the Gatling type, a typical example being shown in Fig. 10.20.

The drive for the feed mechanism can be divided into four groups. The operation of each mechanism depends upon:

- the breech block movement
- the barrel movement
- the movement of the piston in a gas-operated mechanism
- external power source.

Feed mechanisms driven by the movement of the breech block are often used in automatic weapons with belt feed. Transmission of the breech movement to the belt can be achieved in several ways. Fig. 10.21 shows a mechanism with a two-armed lever meshing with the feed recess in the breech block carrier by one end while the other end meshes with the feed slides. Fig. 10.17 shows a mechanism with a feed lever rotating about the axis parallel to the breech movement, one end of the lever being carried by the feeder while the other is controlled by a recess on the breech block carrier side. Fig. 10.22 shows a mechanism with no feed lever, the slide being kinematically coupled to the carrier directly through the feed recess in the breech block carrier.

The feeding recess transfers the longitudinal movement of the breech block carrier to the transverse motion of the cartridge belt. The recess is often formed directly on the breech block carrier.

Another way of driving the feed mechanism is to maintain the kinematic coupling of the breech block with the slide, by using a shaped lever end, which is in contact with the breech. The shaped lever is a form of feed recess. This system is used for the DSHK 12.7mm calibre machine-gun manufactured in Czechoslovakia during the post-war period and fielded as the model 38/46. Its advantage is that the belt feed is directly connected to the movement of the breech block, which also chambers the cartridge. However, the energy necessary for feeding the belt is taken from the breech block, which results in a slower rate of fire.

The recoiling barrel has considerable kinetic energy. If it is used for driving the feed mechanism, this kinetic energy is reduced. However, there are some design difficulties with this method, one being the synchronization of the belt feed with the breech block movement. The feed mechanism must place the next cartridge into the loading space while the breech block is clearing the loading space during its rearwards movement, until it reaches the cartridge case base during its forwards movement. Driving the feed mechanism by the breech block is easily achieved, but driving it by the barrel is more difficult. Movement of the barrel is much shorter than the movement of the breech block. Also, much of the recoil takes place between unlocking and accelerating the breech block, which is when the breech block has not cleared the loading space and cartridge feed is thus not possible. This problem is solved by either
modifying the breech block or by appropriate design of the feed mechanism. The breech modification usually used is shown in Fig. 10.23, and consists of spring-loading the cartridge rammer.

When the breech moves forward, the rammer is in the upper position so that it catches the cartridge in the loading space and inserts it into the chamber. If the breech is in its front position the rammer, being tilted down, pushes the cartridge into the loading space. Another type of feed mechanism is that shown in Fig. 10.24.

This mechanism is fitted with a feed spring which controls the slide with the feeder. The barrel recoils and compresses the feed spring via the lever with the operating curve. Simultaneously the slide moves against the direction of the belt feed and the feeder catches the next cartridge. In this position a special pawl catches the slide. The slide is released by operating the pawl with the moving breech block. The feed spring moves the slide with the cartridge belt through one pitch. A similar method is used for driving the feed mechanism with a piston using the propellant gases. When firing, propellant gases from a vent in the barrel act on a piston which compresses the feed spring which, in turn, moves the cartridge belt at the moment when the loading space is ready to receive a new cartridge.

An external power source is used to operate some small calibre automatic weapons. Such weapons are based on the Gatling or chain-gun principle. The feed mechanism can be driven directly by the external power source or through the breech mechanism.

**FEED MECHANISM DESIGN**

When designing the feed mechanism the following cartridge belt characteristics should be considered:

- Pitch of cartridges in the belt, \( b \)
- Mass of cartridge with belt link, \( m_{ca} + m_b \)
- Link rigidity, \( k_l \)
- Belt link yield strength, \( F_y \)
- Belt link breaking strength, \( B_r \)

The pitch of the cartridges in the belt is related to the maximum diameter of the cartridge case, \( d_{ca} \), according to the following relationships:

\[
(10.6) \quad b = (1.3 \text{ to } 1.6)d_{ca}
\]

The numbers (1.3 to 1.6) are related to the belt flexibility: the larger the belt the greater is its flexibility. It is possible to determine the travel of the slide and belt moving gear from the pitch of the cartridges. The movement of the belt
during feeding should be greater than the pitch of the cartridge if it is to be caught reliably by the moving gear (before feeding) and by the belt catch (after feeding). To achieve this, an over travel of the slide is necessary, so that the slide travel is:

\[(10.7) \quad b_1 = b + (5 \text{ to } 10) \text{ mm}\]

An increase in slide travel greater than the pitch by about 5mm is usual for automatic weapons up to 8mm calibre, with higher values used for large calibre machine-guns and small calibre automatic cannons. The components of the feed mechanism are stressed by a force which acts against the travel of the slide: the resistance of the cartridge belt, \(F\). If the cartridge belt jams, this resistance can reach the breaking strength of the belt, \(F_{br}\). Thus it is necessary to ensure that the components of the feed mechanism are not damaged and should be designed to withstand the force:

\[(10.8) \quad F_j \geq 2F_{br}\]

This force acts against the travel of the slide with the feed pawl at the point of contact with the cartridge. The movement of the slide and feed pawl can be divided into the three following stages:

- start (acceleration)
- period of constant velocity
- slowing to a stop.

In some designs the constant velocity stage is omitted and the slide acceleration goes directly into slowing to a stop. Constant stopping and starting of the belt causes shocks in the mechanism which can adversely affect component life. The stress in the components of the feed mechanism is greatest during the start period, when the slide and belt moving gear are accelerating. It is therefore necessary that the transmission function of the feed mechanism, which is the relationship between the slide movement and the driving element movement, is such that the force acting between the slides and the accelerating member does not exceed the force \(F_j\). Because of the dynamic character of the forces it is necessary, when designing the feed mechanism, to ensure that for reliable operation that the relationship between the resultant internal forces originates by the accelerating member, \(F_2\), and the force \(F_1\) is:

\[(10.9) \quad F_2 \geq \frac{1}{2} F_1\]

Equations (10.4) and (10.5) show that the force in the cartridge belt depends on its velocity at the start of its movement, which is the same as the velocity of the slide. Thus the cartridge belt has the highest stress at the end of acceler-

When designing the feed mechanism, the values given in equations (10.9) and (10.11) should be used. The correct transmission function must be selected which will determine the ratio of velocities between the slide and the driving element. Thus the transmission function, \(i\), of the feed mechanism is calculated from the velocity of the belt, \(v_b\), and the velocity of the driving component, \(v_{br}\). Thus:

\[(10.13) \quad i = \frac{v_b}{v_{br}}\]

The transmission is therefore a function of the driving component travel, \(x_{br}\), so that:

\[(10.14) \quad i = f (x_{br})\]

This transmission function for the entire feed mechanism includes both the velocity ratio of the feed curve and those between other components of the
mechanism. From equation (10.13) the relationship between the slide travel, $x_s$, and the driving element travel, $x_{de}$, is:

$$i = \frac{dX}{dX_{de}}$$  \hspace{1cm} (10.15)

By integrating the transmission function the relationship between the slide travel and the driving element travel can be obtained:

$$X_i = f(X_{de})$$  \hspace{1cm} (10.16)

Also, by differentiating equation (10.16) the corresponding transmission function can be obtained. Both procedures are used for designing feed mechanisms. Fig. 10.25 shows three examples of transmission functions. Fig. 10.25a shows all periods (start, constant transmission and slowing to a stop). Fig. 10.25b represents only two periods; start and slowing to a stop. Fig. 10.25c shows only the period of constant velocity which is simplified because of impacts at the beginning and at the end of the feed mechanism operation.

Thus when designing a feed mechanism, equation (10.12) is first used to calculate the velocity of the belt. Using this velocity and the velocity of the driving element the maximum velocity ratio, $I_{max}$, is determined. The accuracy to which the velocity of the driving element is known will be dependent on the stage of development of the weapon. In some cases, a first approximation of this velocity can be made. The transmission function is designed not to exceed the value $I_{max}$. It is then checked whether the following condition is met:

$$x_{max} = b_1$$  \hspace{1cm} (10.17)

where $b_1$ is calculated from equation (10.7).

In the second stage of design the requirement in equation (10.9) for maximum resultant internal forces is checked. It is necessary to determine the movements of the breech block and barrel and the movement of the feed mechanism. These calculations should include the velocity and acceleration of the slide and the values of the forces acting on the slide, which must not exceed the values given by equation (10.9). If this cannot be achieved, it is necessary to change the shape of the transmission function, or the relationship given in equation (10.16).

**FEEDING CARTRIDGES INTO THE CHAMBER**

After positioning a cartridge in the loading space it is necessary to feed the cartridge into the chamber. This takes a very short time and is carried out at high speed, but it is important not to damage the cartridge when doing so.
Particular attention must be paid to the shape of the guide which directs the cartridge towards the chamber. Before the cartridge can be chambered it must be removed from the belt link, which can be achieved in three ways:

- pushed forwards
- pulled backwards
- pushed out sideways.

The most simple and frequently used method is to push the cartridge forward from the link, but this can only be applied to rimless cartridges. Pulling the cartridge backwards is used if the cartridge belt is made of closed links. Most Soviet 7.62mm calibre medium machine-guns are designed for a rimmed cartridge (SG, SGM, SGMT, PTK) and have closed links. After being removed from the link the cartridge is lowered and then fed into the chamber. It is a two-level feed system. The cartridge belt is fed in at the upper level and the cartridge is chambered at the lower level. This method is more complicated and requires more parts than the push-through system. These additional parts include a double-sided cartridge extractor, lowering piece and cartridge guide.

Cartridges can only be forced out sideways from open links, as shown in Fig. 10.12. When the belt is fed the cartridge strikes a forcing thumb and follows the curve of the forcing thumb which guides it into the loading space. The forcing thumb can also be used with a ratchet feed mechanism, as shown in Fig. 10.26.

After it has been removed from the belt link the cartridge is chambered by the breech block, which is provided with a projection which passes through the loading space. The breech block strikes the cartridge and pushes it into the chamber. Some weapons use a rammer to chamber the cartridge.

ENERGY REQUIRED BY THE FEED MECHANISM

When feeding the belt, the feed mechanism uses some of the energy of the drive element (breech block, barrel, gas piston or motor drive). When calculating the movement of the automatic operating system, the energy requirement of the feed mechanism must be included. It is therefore necessary to calculate the energy requirement of the feed mechanism from the drive element characteristics using the following expression:

\[
\frac{1}{2} m_{de} \frac{d}{dx} v_{de}^2 = F_{de} \cdot dx
\]

where the equivalent driving element mass, \( m'_{de} \), is given by:

\[
m'_{de} = m_{de} + \sum_{i} n_i \frac{m_i}{n_i}
\]

For the equivalent force, \( F_{de} \), acting on the drive element the following relationship is calculated from the force acting on the i-th element, \( F_i \), and the transmission of the i-th element, \( i_i \), and the efficiency of the i-th element, \( \eta_i \):

\[
F_{de} = \sum F_i + \sum \frac{n_i}{\eta_i} \frac{F_i}{i_i}
\]

Individual elements of the mechanism can be affected by, for example, springs. The main force acting on the last element of the mechanism, which may be the slide with the feed pawl or the ratchet, includes the resistance of the cartridge belt and force, \( F_{de} \), which includes the mean value of the belt catch resistance and friction. Using equation (10.5), this force can be written as:

\[
F_i = v_i \sqrt{m_{ca} + m_i} R_i + (m_{ca} + m_i) g n + F_R
\]

To solve equation (10.21) it is necessary to substitute the velocity of the belt, \( v_i \), for the velocity of the main element using the total transmission of the mechanism given in equation (10.14), thus:

\[
F_i = i_i v_{de} \sqrt{m_{ca} + m_i} R_i + (m_{ca} + m_i) g n + F_R
\]

For solving the equation of motion for the driving element proper, equation (10.18) is used as the starting point. When integrating this equation within the limits \((n-1)\) to \(n\), which are given by the interval \(\Delta t\), corresponding to a change in driving element travel, \(\Delta x_{de}\) the following expression is given:

\[
\frac{1}{2} m'_{de} \frac{dx_{de}}{dt}^2 - \frac{1}{2} m_{de} \frac{dx_{de-1}}{dt}^2 = F_{de} \cdot \Delta x_{de}
\]

Thus the relationship for determining the driving element velocity at the end of the interval is:

\[
\frac{1}{2} m'_{de} v_{de}^2 - \frac{1}{2} m_{de} v_{de-1}^2 = F_{de} \cdot \Delta x_{de}
\]
The third equation for the driving element motion is the differential equation:

\[
\frac{dx_{de}}{dt} = v_{de} \cdot dt
\]

from which the driving element travel can be determined. Thus the numerical solution for the driving element movement during the feed period applies the set of equations (10.22), (10.24) and (10.25), completed with the dependence of the equivalent mass along the driving element travel:

\[
m_{de} = f(x_{de})
\]

It is simpler to substitute equation (26) with a polynomial, for example:

\[
m_{de} = A \cdot x_{de}^2 + B \cdot x_{de} + C
\]

where A, B, C are constants. In a similar way the transmission function \( i = f(x_{de}) \) can be determined.

The above set of equations describing the effect of the feed mechanism on the driving element movement are solved for the start-up of belt movement and constant transmission only. During deceleration the feed mechanism does not use energy.

**11**

**Gun Springs**

**THE IMPORTANCE OF GUN SPRINGS**

Springs are used in the operating mechanisms and in different auxiliary components. Operating springs are mostly subjected to dynamic stresses, while auxiliary springs are commonly subjected to static stresses.

With the exception of externally-powered guns the operation of automatic weapons uses the energy of the propellant gases, which is available for only a very short time and accelerates the breech in the rearwards direction only. Thus only part of the weapon cycle can take place using direct energy from the propellant. To complete the cycle the rearward moving breech block compresses the return spring and the stored energy is used to return the breech block to its forward position.

The force to compress the return spring is relatively high so that the retaining action on the breech is also high, which is detrimental to achieving high rates of fire. If a weak return spring is used only part of the energy of the breech block is absorbed by the spring and the breech block strikes the rear of the weapon casing with the remaining energy. If this remaining energy is low, but only low rates of fire are required, then the impact is acceptable. If the weapon is designed for higher rates of fire, so that the velocity of the breech block is high, the impact becomes too great and large forces are transmitted to the weapon parts. Heavy impact of the breech block on the weapon casing normally occurs in pistols, sub-machine-guns and automatic rifles. The impact is a minor problem in these weapons because the firer's hand or shoulder acts as a spring; otherwise, a buffer must be used to reduce the effect of the impact, the most important component of which is the buffer spring.

In recoil-operated weapons the barrel moves to the rear during firing and must, after recoil, be returned to its forward position using a barrel spring. Operating springs also include firing pin springs and springs used in trigger mechanisms. Auxiliary springs are used with various catches, paws and locks to hold components in place and to enable weapons to be assembled and stripped.

**RETURN SPRINGS**

The return spring must allow a relatively large movement of the breech for only a small increase in force as it is compressed. Simple wire helical springs
with a circular cross-section of wire or stranded helical wire springs are normally used for this. There are exceptions, however, and spiral springs have sometimes been used.

The return spring may be housed in the weapon in a variety of ways. It should be supported to avoid it becoming deformed, and may be guided either on a telescopic guide rod or in a tube. The return spring is often mounted behind the breech block, as shown in Fig. 11.1, but adds to the length of the weapon. To reduce this problem the front part of spring may be inserted into the breech block. In the design of the 7.62mm ShKAS aircraft machine-gun the return spring was completely embedded in the breech block carrier, as shown in Fig. 11.2. To cock the weapon the breech block was pulled by a rod to the rear. The return spring was prestressed by pushing the rod forward and engaging it in the weapon casing.

In some gas-operated machine-guns the return spring is mounted on the piston rod, as shown in Fig. 11.3. This arrangement makes it possible to reduce the total length of the weapon.

Another method of reducing weapon length is shown in Fig. 11.4, where the return spring is mounted on the barrel. This design is usually found in a pistol. However, any spring mounted on the barrel is vulnerable to heating, with the possible loss of the spring temper.

It is possible to mount the return spring in the weapon so that its axis is either oblique to the axis of the breech block travel or parallel to it. The former design, used in the Czech LMG ZB-26, is shown in Fig. 11.5. The return spring is mounted in the stock and its force is transmitted to the breech block by means of a return rod which bears against the breech block carrier.

This return rod is mounted in a tube with the return spring so that it can pivot. This design makes use of the space in the stock and the total length of the weapon can be shortened.

Mounting the return spring axis parallel to the axis of the breech movement is used in some pistols and is shown in Fig. 11.6. This design again reduces the length of the weapon.

Some weapons have two or more return springs. It is common for the breech block to run on twin guide rods, which are also used for mounting and guiding the return springs.

**BUFFERS AND BUFFER SPRINGS**

To reduce the hard impact of the breech block when it reaches its rearward position, and to influence the rate of fire, the weapon casing is often fitted with a buffer. Fig. 11.7 shows two typical buffer arrangements. The buffer spring is much stiffer than the return spring and stores the excess energy of the breech block. The buffer spring does not return all of the original stored kinetic energy to the breech block because of energy losses in the spring. The amount of energy loss depends on the type of spring, as shown in Table 11.1.
Table 11.1
Energy losses for different types of spring

<table>
<thead>
<tr>
<th>Spring type</th>
<th>Percentage energy loss</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simple helical spring</td>
<td>5 - 20</td>
</tr>
<tr>
<td>Simple washer spring</td>
<td>33</td>
</tr>
<tr>
<td>Multilayer washer spring</td>
<td>50</td>
</tr>
<tr>
<td>Ring spring</td>
<td>60 - 70</td>
</tr>
</tbody>
</table>

Springs which have high energy losses are used where impacts must be damped. To eliminate heavy impacts the design of the buffer spring must be able to absorb at least the energy of the breech block before impact. When the breech block strikes the buffer there is an elastic impact between the breech block and buffer. Consequently, the breech block velocity is reduced immediately after impact and the buffer is given a velocity which is higher than the velocity of the breech block after impact. The breech block and buffer move independently of one another. The breech block continues to be slowed by the return spring, but the velocity loss of the light buffer with the stiff buffer spring is greater, so there is another impact between the breech block and buffer. A series of impacts between the two parts produces the time/displacement diagram shown in Fig. 11.8.

The time between these additional impacts shortens and the impacts between both parts become weaker. After several impacts, both parts remain in permanent contact and move together. If the mass of the breech block is large compared to the buffer it is possible, with little loss in accuracy, to treat the series of elastic impacts as one non-elastic impact in which the breech block and buffer move together. The movement of the buffer spring and return spring can then be calculated. However, it is first necessary to consider how the return and buffer springs are mounted. Fig. 11.7 shows the most common methods of mounting the buffer:

- the return spring and buffer spring act in series
- the return spring and buffer spring act in parallel.

Figure 11.4 Return spring mounted on the barrel

Figure 11.5 Return spring contained in the stock

Figure 11.6 Return spring mounted parallel to the breech mechanism

Figure 11.7 Two arrangements for the return and buffer springs mountings
The momentum of the breech block before impact will be the same as the momentum of the breech block and buffer after impact. Thus:

\[ m_{bb} v_{tbb} = \left( m_{bb} + m_{bo} + \frac{1}{3} m_{bus} \right) v_{bo2} \]

**NB** \( \frac{1}{3} m_{bus} \) is used because the buffer spring is stationary with only its compressed end actually moving.

This relationship is valid for both case (a) and (b) shown in Fig. 11.7, because the effect of the return spring mass is negligible. To include the return spring mass, \( m_{rs} \), for case (a) the momenta can be equated before and after the impact:

\[ \left( m_{bb} + \frac{1}{3} m_{rs} \right) v_{tbb} = \left( m_{bb} + m_{rs} + m_{bo} + \frac{1}{3} m_{bus} \right) v_{bo2} \]

For case (a) (Fig. 11.7), the total mass moving on the buffer spring, \( m_{tot} \), is:

\[ m_{tot} = m_{bb} + m_{rs} + m_{bo} + \frac{1}{3} m_{bus} \]

For case (b) (Fig. 11.7), the return spring is compressed simultaneously with the compression of the buffer spring. Thus the total mass, \( m_{tot} \), is:

\[ m_{tot} = m_{bb} + \frac{1}{3} m_{rs} + m_{bo} + \frac{1}{3} m_{bus} \]

This mass is acting on two parallel springs: the common resultant characteristics are obtained by summing the characteristics are the return and buffer springs. When the buffer spring returns the breech towards the barrel it is necessary to include the energy loss in the spring, as given in Table 11.1. Thus:

\[ \eta = \frac{E}{E} = \frac{C}{C} = \frac{F}{F} \]

A line over the symbol corresponds to spring release as shown in Fig. 11.9.

**BARREL SPRINGS**

Weapons operating on the recoil cycle have barrels which move rearwards when the weapon is fired. The barrels of small calibre automatic weapons are returned to the front position by means of a barrel spring. Where there are large barrel movements, the most frequently used spring is the helical type. For small calibre automatic weapons only one barrel spring is used: a typical arrangement is shown in Fig. 11.10.
When the barrel recoils the barrel spring is compressed. This slows the rearward movement of the barrel and brings it to rest. The barrel is then returned to its forward position by the spring. As it is returned to its forward position, the barrel strikes the shoulder in the weapon casing. For low velocities and the light barrels of small calibre weapons, the impact is not a problem. However, for automatic weapons above 10mm calibre the energy of the barrel, when moving forward, is high and creates a heavy barrel impact. To avoid this, the barrel is spring-mounted in recoil and counter-recoil, as shown in Fig. 11.1a and b.

In Fig. 11.1a the barrel is mounted on two springs, the rear spring compressing during recoil and the front spring compressing during counter-recoil. Fig. 11.1b shows how one barrel spring is used for the same purpose.

Fig. 11.12 shows the time/displacement graphs for the two arrangements shown in Fig. 11.11.

When the barrel is spring-mounted in both recoil and counter-recoil, it does not stop in the forward position but rebounds several times. Fig. 11.13 shows the displacement of the barrel with respect to time. These movements must be damped before the next shot is fired. At the moment a shot is fired, it is accelerated by the impulse of the shot to a velocity $v_{A0}$. If another shot is fired at time $t_A$, the barrel will move forward at velocity $v_A$. After the shot has been fired the barrel will stop moving forwards and start moving rearwards.

The resultant rearward velocity of the barrel will be:

$$v_t = v_b - v_A$$  \[11.5\]

Thus the barrel velocity will be less than the original velocity, $v_b$. It is therefore possible that the weapon will not cycle. If the gun is fired at time $t_b$ the barrel will move rearwards at velocity $v_b$, but the barrel is accelerated when another shot is fired. The resultant recoil velocity will be:

$$v_t = v_b + v_b$$  \[11.6\]
This higher recoil velocity may damage parts of the weapon. Thus, designers attempt to ensure that the barrel is stationary before subsequent rounds are fired. There are several ways of doing this without resorting to special hydraulic dampers. One method, using a similar arrangement of barrel springs shown in Fig. 11.11, uses two brake rings, similar to ring springs, mounted on the barrel instead of the front support ring as shown in Fig. 11.14. The inner ring has a single radial cut in it.

When the barrel moves rearwards the inner ring is pressed into the outer one and, because of the radial cut, it closes onto the barrel. During counter-recoil the inner ring is arrested on the shoulder of the weapon casing and the barrel must overcome the friction caused by the compressed inner ring. This friction also acts when the barrel rebounds from the front position to its equilibrium position and considerably dampens barrel movement.

Fig. 11.15 shows a combination of helical and ring spring which is used in the German 15/20mm MG 151. The helical spring allows the necessary barrel recoil. The counter-recoil is spring-loaded by means of a ring spring, which has a relatively high energy loss. These losses dampen the barrel movement.

Most modern automatic weapons use only one ring spring, as shown in Fig. 11.16. Although this design has the advantage of good movement damping qualities, the springs are long and there is a tendency for the split ring to seize on the barrel.

**SPRING CALCULATIONS**

**Types of Springs and their Use**

Automatic weapons mostly use the following types of spring (Fig. 11.17):

(a) – helical wire compression springs of circular section
(b) – helical wire strand springs
(c) – helical compression springs of rectangular section
(d), (e), (f) – leaf springs
(g), (h) – washer springs

---

**(i)** – torsion springs
**(j)** – ring springs
**(k)** – spiral springs.

Helical springs, including wire strand ones, are used as return springs, barrel springs, buffer springs, springs in feed mechanisms, trigger springs and in magazines. Washer springs are used to absorb impacts in buffers, spring-loaded breech sears and for damping the impact of slides in feed mechanism. Ring springs are used mostly as barrel and buffer springs and in recoil units in automatic cannons. Torsion springs are used in firing, trigger and feed mechanisms and also in various paws. Leaf springs are used for extractors, magazines, buffers and in various paws. Spiral springs are used in disc and drum magazines.

**Spring Characteristics**

The spring characteristic is the relationship between the force, $F$, needed to compress the spring and the distance over which it is compressed, $s$:

\[ F = f(s) \]

or for springs with angular deformation, $\phi$, caused by torque, $M$:

\[ M = f(\phi) \]
This relationship is generally linear so that:

\[(11.8) \quad F = c_s \]

\[(11.9) \quad M = c_\phi \]

where the constant of proportionality, \( c_s \), is called the rigidity of the spring.

Fig. 11.18 shows a spring and its force/displacement curve. The different forces correspond to the compression and length of the spring in the following way:

- \( F_T, F_S, F_g \) = spring force for the diagram in Fig. 11.18
- \( S_T, S_g, S_0 \) = compression of spring corresponding to the individual forces
- \( l_0, l_1, l_2, l_3 \) = spring length.

The indices denote the condition of the spring as shown in Fig. 11.18:

- \( 0 \) = no load condition
- \( 1 \) = prestressed condition (minimum load)
- \( 3 \) = fully loaded (maximum working load)
- \( 9 \) = limit condition (spring is compressed to the end of its coils).

The working stroke of a spring is marked by the symbol \( h \). Referring to Fig. 11.18:

\[(11.10) \quad h = S_g - S_1 = l_1 - l_2 \]

The spring rigidity can be expressed as:

\[(11.11) \quad C = \frac{F_0 - F_1}{h} = \frac{F_1 - F_2}{S_1 - S_2} = \frac{F_2 - F_3}{S_2 - S_3} \]

The spring is usually composed of a series of springing elements (coils, washers, rings, leaves) whose sum of the partial deformations gives the total deformation of the spring. For static loading of a spring all spring elements are equally compressed. For dynamic loading (by impact or rapid acceleration), these elements are deformed unequally. Deformation propagates in a wave form so that at each instant the deformation of the spring elements is different (Fig. 11.19). Thus the stress limit of the individual spring elements may be reached.

For return springs there may be a reduction in their service life if the free spring length, \( l_0 \), is shortened: it reduces the force applied by the spring and may cause spring failure. An important consideration is the comparison of the maximum velocity of the moving end of the spring, \( v_{max} \), with the critical velocity of compression, \( v_c \). This critical velocity is the limit of velocity of the impact of the coils due to inertial forces. If \( v_{max} \leq v_c \), impacts between the coils of a compression spring does not occur. This critical velocity of helical compression springs made of a single wire is calculated by means of:

\[ \text{Figure 11.17 Springs used in automatic weapons} \]
(11.12) \[ \nu_\alpha = \frac{T_b \left(1 - \frac{F_b}{F_0}\right)}{\sqrt{2G \cdot \rho}} \times 10^3 \]

and for stranded springs by means of:

(11.13) \[ \nu_\alpha = \frac{T_b \left(1 - \frac{F_b}{F_0}\right)}{\sqrt{1.7G \cdot \rho}} \times 10^3 \]

where:
- \( T_b \) = twisting stress of the spring material in the limit state
- \( F_b \) = force developed by the spring in the fully-loaded state
- \( F_0 \) = force developed by the spring in limit state.

Substituting into equations (11.12) and (11.13) the relative total clearance, \( \delta \), in the fully-loaded state is given by:

\[ \delta = 1 - \frac{F_b}{F_0} \]

Thus the following relationships are obtained:

(11.15) single wire \[ \nu_\alpha = \frac{T_b \delta}{\sqrt{2G \cdot \rho}} \times 10^3 \text{ m/s} \]

(11.16) stranded wire \[ \nu_\alpha = \frac{T_b \delta}{\sqrt{1.7G \cdot \rho}} \times 10^3 \text{ m/s} \]

The limit values of \( \delta \) are:
- for single wire compression springs \( \delta = 0.05 - 0.25 \),
- for stranded compression springs \( \delta = 0.15 - 0.40 \).

It follows from the above that the relationship in (11.17) would be valid for the characteristic of a helical compression spring made of a single wire.
\[ F_8 = (0.75 \text{ to } 0.95)F_9 \]

and for a stranded compression spring it would be:

\[ F_8 = (0.6 \text{ to } 0.85)F_9 \]

It follows from equations (11.15) and (11.16) that the critical velocity is greater for stranded springs than for springs made of a single wire. These are therefore used in weapons where components need to be driven at high velocity.

When calculating spring forces it is necessary to take into account the purpose of the spring. For example, springs which are compressed before firing require an external force. For direct hand cocking this force should not exceed 300N. The spring must also firmly hold the component upon which it is acting against external dynamic forces, which may include sudden movements, blows to the weapon, the weapon falling over, the movement of the vehicle on which the weapon is mounted, aircraft manoeuvring, etc. To resist these forces the prestressed state should be:

\[ F_1 = (5 \text{ to } 8) \text{ mg} \]

Where springs are used for storing excess energy, but are not compressed when cocking the weapon, the spring characteristic is calculated from the amount of kinetic energy, \( E_{KE} \), to be stored by the spring. Using the working stroke, \( h \), the mean force of the spring, \( F_m \), is determined from:

\[ F_m = \frac{E_{KE}}{h} \]

Where:

\[ F_m = \frac{F_1 + F_8}{2} \]

The greater the spring rigidity, the smaller the number of spring elements and the greater the value of force \( F_8 \).

On choosing the characteristic for a torsion or spiral spring the same procedure is followed, the forces, \( F \), being replaced by moments, \( M \), and compression, \( s \), by angular movement, \( \phi \).

**PRINCIPLES OF SPRING CALCULATIONS**

When designing a spring for a weapon, the beginning point is either the chosen characteristic of the spring or the space that it will occupy. The arrangement of the spring elements can be either in series (Fig. 11.20a), or parallel (Fig. 11.20b,c). The Figures show that the arrangement can affect both the compression distance, \( s \), and value of the acting force, \( F \). Fig. 11.20a shows the characteristic of one spring and the characteristic of two springs arranged in series. It is evident that the compression is summed while the force, \( F \), for this double compression is the same as for one spring for compression, \( s \). If these two springs are arranged in parallel (Fig. 11.20b), then for the same compression the force is doubled. Fig. 11.20c shows a parallel arrangement for two different springs, the effect of which is a broken resultant characteristic.
Firing and Trigger Mechanisms

Firing and trigger mechanisms must reliably and intentionally fire the weapon when required, stop firing and allow safe handling of the weapon. The firing mechanism initiates the primer of the cartridge. The trigger mechanism starts and stops the weapon firing.

FIRING MECHANISMS FOR AUTOMATIC WEAPONS

To ignite the propellant charge in a cartridge it is necessary to initiate the primer, which may be either mechanical or electrical. A typical part of a mechanical firing mechanism is the striker, the front of which is pin-shaped, as shown in Fig. 12.1. The striker hits the primer with the firing pin, the primer composition is pressed between the pin and the anvil and is ignited. The flame flashes through the flash holes and ignites the charge. The energy to initiate the primer is usually stored in the firing spring.

Firing mechanisms fall into three groups:

- hammer firing mechanisms
- striker firing mechanisms
- firing mechanisms with a return spring as the firing spring.

The first two types contain a firing spring. In hammer firing mechanisms the spring accelerates the hammer that transmits the energy to initiate the primer. Hammer mechanisms have either linear motion (Fig. 12.2) or angular motion (Fig. 12.3).

Striker firing mechanisms have a firing spring which acts directly on the striker (Fig. 12.4).

Weapons which fire from an open bolt use firing mechanisms with the return spring acting as the firing spring. Part of the energy stored in the breech block return spring is used to initiate the primer after the breech block has reached its forward position. The firing pin may be fixed and mounted in the face of the breech block (Fig. 12.5), or can move in the breech block and is struck after the breech block contacts the breech (Fig. 12.6).

FIRING MECHANISM CALCULATIONS

Firing mechanism calculations are based on the primer properties. Primers are tested by dropping onto them a striker of a given weight from a given height. Table 12.1 shows some examples of the properties of primers. It defines the minimum energy and corresponding velocity required for primer initiation. Primer calculations are based on the energy required to initiate the primer. However, when a primer is tested it is fixed and cannot move, but when the cartridge is placed in a weapon some movement is possible and a greater striker energy is required. Thus 50% higher energy than prescribed by the primer tests is used.
Figure 12.3 Hammer with angular motion

<table>
<thead>
<tr>
<th>Table 12.1 Examples of small arms primers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primer type</td>
</tr>
<tr>
<td>22 long rifle</td>
</tr>
<tr>
<td>9mm pistol</td>
</tr>
<tr>
<td>7.62mm mod 43</td>
</tr>
<tr>
<td>7.62mm mod 59</td>
</tr>
<tr>
<td>Express</td>
</tr>
<tr>
<td>Gevelot</td>
</tr>
</tbody>
</table>

If the energy required for primer initiation is $E_{pr}$ and the striker energy at the moment of initiation is $E_{pr1}$, then:

$E_{pr1} = 1.5 E_{pr}$  \hspace{1cm} (12.1)

Tests have shown that the energy necessary for 100% primer initiation decreases as the velocity of the striker increases, as shown in Fig. 12.7 for a Russian 7.62mm caliber cartridge.

Primer initiation reliability may be influenced by such things as dirt or ice in the firing mechanism, worn parts, etc. The reliability of primer initiation is one of the most important factors in the design of a weapon. The firing mechanism must be designed so that initiation can only occur with the breech locked. The crucial part of the firing mechanism calculation is that of the firing spring. For striker firing mechanisms the firing spring is designed to give the required energy to the striker at initiation. Hammer firing mechanisms are more complex because there is a drop in energy due to the impact of the hammer and striker. Firing mechanisms using the return spring are only checked for reliable functioning because there is usually an excess of energy for primer initiation.

There are two types of hammer firing mechanisms:
- after hitting the striker, the hammer is stopped and the primer is initiated by the striker only
- after hitting the striker, the hammer continues with the striker sharing in primer initiation.

In the first case, the hammer and striker separate after impact. For two elastic components the velocity of component 1 after impacting component 2 is:

$$v'_1 = v_1 + \frac{(v_2 - v_1)(1 + \varepsilon)}{1 + \frac{m_1}{m_2}}$$  \hspace{1cm} (12.2)

If $v_1 = 0$, the striker velocity after impact is:

$$v'_1 = \frac{m_2 - m_1 (1 + \varepsilon)}{m_2 + m_1}$$  \hspace{1cm} (12.3)

Thus the striker kinetic energy after impact is:

$$E'_1 = \frac{1}{2} m_1 \cdot v'_1^2 = E_2 + \frac{m_1 \cdot m_2}{(m_1 + m_2)} (1 + \varepsilon)^2$$  \hspace{1cm} (12.4)
This gives the energy required for the hammer before the impact, which is the energy provided by the firing spring:

\[ E_2 = E'_1 \frac{(m_1 + m_2)^2}{m_1 \cdot m_2} \frac{1}{1 + \varepsilon^2} \]  

for \( \varepsilon = 0.4 \), which is a typical value for two impacting steel components, then:

\[ E_2 = 0.51E'_1 \frac{(m_1 + m_2)^2}{m_1 \cdot m_2} \]  

In the second case, it is rather different. After impact the hammer and striker separate, but the hammer is driven on by the firing spring. It impacts the striker again and shares in the deformation of the base of the primer. This results very closely to an inelastic impact of two bodies. Thus the common striker and hammer velocity after the impact is:

\[ v'_{12} = \frac{m_2}{m_1 + m_2}v_2 \]

Substitution of equation (12.7) into the kinetic energy formula for the striker and hammer after impact gives:

\[ E'_{12} = E_2 \frac{m_2}{m_1 + m_2} \]

which defines the necessary hammer energy before impact:

\[ E_2 = E'_{12} \frac{m_1 + m_2}{m_2} \]

\[ m_p = \frac{J}{r^2} \]

where:

- \( J \) = the moment of inertia of the hammer with respect to the axis
- \( r \) = the radius of the point mass.

Electrical primers not only have a minimum power requirement that must be applied for primer initiation and which will vary from type to type, but they also have a minimum value below which primer initiation will not occur. This latter requirement ensures that stray electrical fields will not accidentally initiate the primer. Electrical primers are not commonly used with small arms.

**FIRING MECHANISM DESIGN**

The stresses acting on firing mechanism components are mostly from impacts, and during one functional cycle there are many of these. This is illustrated by a
hammer firing mechanism with rectilinear hammer motion. After the trigger is pressed the hammer impacts the striker and then stops on the breech block. The striker hits the primer with the firing pin and then again when the breech block stops. During firing, the gas pressure in the cartridge case imparts another short pulse to the striker through the primer. The hammer is caught by the rearward movement of the breech and may strike the weapon casing in the rear position. The final impact occurs when the hammer is arrested. In addition to the firing loads, the firing mechanism components must also withstand dry firing without cartridges: in this case the striker stresses may be higher because there is no cushioning effect created by the primer.

It is important that the impact-stressed parts are correctly proportioned. The following points should be considered during firing mechanism design:

1. Variations in cross-sectional area cause variations in stresses in a component. Thus, if possible, components should have a constant cross-section along their length. It is only acceptable for the cross-sectional area to increase over a short length and there should be no notches to raise stress.
2. The higher the movement of the impacted part, the smaller the force transmitted to it.
3. Micro-notches caused by machining may initiate cracks so should be removed by polishing. The most important parts of the firing mechanism are the striker and the firing pin. The front cross-section of the striker is reduced by the firing pin, which gives shorter life to that part. To increase the life of the firing pin the transition between the striker body and the firing pin should have as large a radius as possible, or the shape of the firing pin should be conical as shown in Fig. 12.8.

To reduce the impact stress to a minimum the striker cross-section should be, if possible, equal along the whole length. It is useful to design the striker as longitudinally grooved as shown in Fig. 12.9 (either in 'A' or 'B' style).

The striker should be guided on a large diameter, but the firing pin should have sufficient clearance in the firing pin hole so as not to bend or break.

\[ \frac{\pi d^2}{4} P_0 < \pi d \delta \sigma_s \]
The firing pin shape may vary as shown in Fig. 12.13. The most often used shape is that in Fig. 12.13a.

For a striker whose axis does not correspond to that of the bore at the instant of initiation, the firing pin is shaped as shown in Fig. 12.13d. Correct firing pin dimensions are shown by the position and shape of the indentation in the primer cup after firing, as shown in Fig. 12.11. Fig. 12.11a shows the firing pins position before firing. The primer is sunk into the cartridge case by the amount ‘a’ and the cup edges are rounded. Fig. 12.11b shows the correct dimension after firing. The primer and cartridge case base are level and the outer edges of the cup and the indentation are rounded. Fig. 12.11c shows the primer pushed out of the cartridge case after firing in which case the space between the breech face and the base of the case is too large. Fig. 12.11d shows sharp edges to the indentation and around the primer, indicating a possible penetration of the primer. Four reasons for penetration are outlined in Fig. 12.14. Fig. 12.14a shows the result of a weak firing spring, so that the shear strength requirement in equation (12.11) is not fulfilled. Fig. 12.14b shows the result of excessive firing pin protrusion and Fig. 12.14c shows the result of excessive clearance of the firing pin in the breech. This clearance may increase during weapon usage. Fig. 12.14d shows what happens if the nose of the firing pin has a sharp edge.

The design of a linear hammer should follow the same principles as striker design. The design of a pivoting hammer should ensure that the hammer pivot axis is at the centre of the impact or close to it. The impact centre is a point where the influence of the revolving pulse and that of the sliding one are equal. Thus it is possible to equate momenta at this point (Fig. 12.15), by equating the mass of the hammer, \( m \), and the velocity of its centre of gravity, \( v_{CG} \), to the mass of the hammer and its angular velocity, \( \omega \). Thus:

\[
(12.12) \quad m \cdot v_{CG} = \frac{\rho^2}{a} \cdot \omega
\]

The inertial radius of the hammer can be found from:

\[
\sqrt{\rho} = \frac{1}{m}
\]

The distance to the centre of impact is:

\[
(12.13) \quad c = \frac{\rho^2}{a}
\]
TRIGGER MECHANISM DESIGN

Introduction

The purpose of the trigger mechanism is to stop and start the weapon firing. There are two main parts, the trigger and the sear, which are shown in Fig. 12.16. The part affected by the control force is the trigger. The part that holds the firing mechanism is the trigger sear.

All trigger mechanisms have these two parts, but the simplest may combine them into one. To hold them in their initial position they are spring-loaded. Trigger mechanisms differ in the position of the breech block at the instant of firing. If the breech block is in the forward position with the cartridge chambered the sear holds the cocked striker or hammer. The task of the trigger mechanism is, in this case, to control the firing mechanism, and it may be designed for firing single shots, bursts or both. If the weapon is fired with the breech block to the rear, it is the breech block that is held by the trigger sear. The task of the trigger mechanism is to release and then catch the breech mechanism, whilst the trigger mechanism is designed to fire either in bursts or in bursts and single shot. The trigger may be released by a force applied by the firer, an electrical solenoid or an air cylinder. Electrical solenoids and air cylinders are used for firing a weapon remotely, usually when mounted on a vehicle.

Figure 12.14 Effect of different firing pin faults on primer indentation

Figure 12.15 Geometry of pivoting hammer

Trigger mechanisms affect firer accuracy. Trigger pull and lift influence the performance of the firer, but minimum values are necessary for safety reasons to ensure that the weapon does not fire when roughly handled or dropped. Shot delay, \( t_{dl} \), is the time interval from the end of aiming until the bullet leaves the muzzle, and is determined from the following:

\[
t_{dl} = t_1 + t_2 + t_3 + t_4 + t_5
\]

where:

- \( t_1 \) = delay caused by the firer

Figure 12.16 Trigger and sear of a firing mechanism
- \( t_1 \) = trigger mechanism operating time
- \( t_2 \) = firing mechanism operating time
- \( t_3 \) = propellant charge ignition and burning time
- \( t_4 \) = bullet motion in the bore.

Time \( t_1 \) depends on the skills of the firer, his response time and training and environmental and physical conditions. The values of \( t_1 \) are, normally, within 0.04 to 0.06 sec. These may increase due to fatigue or worsening environmental conditions. The trigger mechanism operating time is the time from sear release to striker impact and varies for different trigger mechanisms as follows:

- electrical, \( t_5 = 0.005 \) to 0.006s
- mechanical and electromagnetic, \( t_5 = 0.01 \) to 0.02s
- mechanical hand-controlled, \( t_5 = 0.04 \) to 0.06s.

The firing mechanism operating time \( t_5 \) is the time from striker impact to primer impact. Typical values vary from 0.006 to 0.01s. For electrical primer initiation \( t_5 = 0 \). The ignition time for the propellant charge, \( t_6 \), varies between 0.001s and 0.003s and the time for the bullet to travel from breech to muzzle, \( t_7 \), varies from 0.001s to 0.005s, depending on the weapon type, barrel length, \( l_b \), and muzzle velocity, \( v_0 \), and is approximately:

\[
(12.14) \quad t_5 = \frac{2l_b}{v_0}
\]

The total delay time, \( t_{total} \), may be as much as 0.14s, with a minimum value of approximately 0.04s. Shot delay affects accuracy because it influences the angle of jump of the bore axis as the bullet leaves the muzzle. The delay times above are for the weapons firing from a closed breech. In weapons firing from an open breech the time for the breech to close must also be added to the total time, which is approximately 0.05s for a rate of fire of 900 rpm. However, the firing mechanism operating time is shorter.

### Trigger Mechanism Design

Trigger pull requirement varies with the type and use of a weapon and must be sufficiently high to ensure weapon safety. The minimum trigger pull for military pistols is usually 18N and for automatic rifles about 25N. Target rifles have much lighter trigger pulls and may be as low as 2.5N. Trigger pull is not a constant but varies as the trigger is pressed. Military weapons have two-stage triggers: the first stage is light and the second stage heavier, as shown in Fig. 2.17.

Trigger lift for the first stage of trigger pull is much longer than for the second stage. The trigger pull is affected by the springs in the trigger mechanism. Friction acts on pivot pins and the friction force between the sear and striker or hammer. The latter provides the greatest contribution to trigger pull. To ensure that the firing spring does not act to release the sear, the sear surface inclination angle, \( \beta \), (Fig. 12.18) must be \( < 90^\circ \). This angle should normally be greater than 90° so that the sear is self-locking. However, some automatic weapons do not use a self-locking sear, and when firing bursts the sear is released directly by the breech. Firing is halted by supporting the sear. In this case angle \( \beta > 90^\circ + \phi \), where \( \phi \) is the angle of friction.

It is necessary first to decide on the trigger pull so that the whole kinematic chain can be calculated, beginning at the trigger. For the trigger mechanism shown in Fig. 12.18 the trigger pull, \( F_t \), is:

\[
(12.15) \quad F_t = F_1 - \frac{i_1}{\eta_1} F_1 + \frac{i_2}{\eta_2} F_2 - \frac{i_1}{\eta_1} + \frac{i_2}{\eta_2}
\]

It can be seen that the trigger pull is highly dependent on the leverage ratios, \( i \), and mechanical efficiencies, \( \eta \), of the component parts of the trigger. The efficiencies include the friction between the levers and their pivoting pins. The friction force, \( F_f \), can be determined as follows:

\[
(12.16) \quad F_f = f \cdot N = \frac{F_{spring} \cdot f}{\cos \alpha}
\]

The ratings for the different springs are selected to ensure reliable return of the different parts to their initial positions, the most important being the sear.
spring which must ensure that the sear is raised before the breech block returns to the sear after rebounding from the buffer. Trigger mechanisms activated by an electrical solenoid are designed so that the solenoid develops the force, $F_t$.

**Trigger Mechanism Structure**

The basic trigger mechanism is shown in Fig. 12.16. It consists of only a trigger and a sear and is able to fire in bursts, but cannot perform any of the other functions required of an automatic weapon system. Firing single shot requires the hammer to be caught reliably and safely even though the trigger may not have been released. This is achieved by an auxiliary spring-loaded sear called the tripping lever, and is shown in Fig. 12.19, which is the system used for the hammer firing mechanism in the Model 61 7.65mm Czech sub-machine-gun. On pressing the trigger the upper arm of the sear moves forward, releasing the hammer as shown in Fig. 12.19a. The trigger remains pressed, and after firing the hammer is returned by the breech block; it is caught by the tripping lever claw, 2. To fire again the trigger must be released. Thus as the hammer is released from the tripping lever claw its claw is caught by the sear. The trigger and firing mechanisms are therefore returned to their initial positions and are ready for the next shot to be fired. Fig. 12.19b shows an example of a pistol trigger mechanism, with the tripping lever driven by the breech.

Trigger mechanisms designed for firing bursts only differ for open and closed breech firing. Fig. 12.16 shows an example of a trigger mechanism for firing in
bursts from an open breech, used in the Russian DP-27 light machine-gun. The trigger mechanism in Fig. 12.20 for bursts of fire is from a closed breech and contains an 'automatic trigger' that releases the hammer after the breech has locked. The automatic trigger is driven by the breech block carrier at the end of its underside. The Bergmann machine-gun uses this type of trigger mechanism. The main sear is released by pressing the trigger and while it is pressed, sear release is performed by the breech striking the automatic trigger.

Fig. 12.21 shows the trigger mechanism for firing from an open breech used in the Czech ZB-26 light machine-gun. The flat pin of the fire selector/safety catch passes through the tripping lever recess. For the position shown in Fig. 12.21 the mechanism is switched to fire single shot. When the trigger is operated the upper claw of the tripping lever releases the sear. When the returning breech lowers the tripping lever the sear releases claw 1, the lever passes through the opening and the sear rises to hold the breech. To fire the next shot the trigger must be pressed again. To fire bursts, the safety catch is rotated in the direction of the arrow and the sear is deflected by claw 2. The tripping lever is out of reach of the breech, and firing continues until the trigger is released.

If the safety catch is rotated only part way, the lever passes through the opening in the sear without lowering it and the weapon is safe. For weapons that fire from a closed breech, the trigger mechanism is a combination of the mechanisms shown in Figs. 12.19 and 12.20. The selector/safety catch in one position disables the tripping lever and the weapon fires in bursts. In the other position the trigger is either blocked, and so the weapon cannot fire, or the tripping lever engages and the weapon fires single shot. The firing mechanism of the Russian AK-47 is a typical example of this type of mechanism, and is shown in Figs. 12.22 and 12.23.

A two-step trigger can also be used for selecting single shot or burst fire, as shown in Fig. 12.24. When the trigger is pressed to position 1, the spring-loaded claw 'a' moves the sear and releases the breech. The sear is also released from claw 'a' and returns to its original position under the action of the sear spring and catches the breech. Pressing the trigger further results in the sear being pressed down by the fix claw 'b' which remains in the down position until the trigger is released.

The trigger mechanism should be designed for reliability and long service life. The sear is the critically stressed component of the mechanism as the breech or hammer strike it with sufficient energy to cause damage. For small weapons an inelastic impact is acceptable. However, if the energy of the breech is large it is necessary to spring-load the sear, as shown in Fig. 12.25.

The sear is placed in a sliding case that returns with a spring. To avoid excessive rebound, springs with high friction losses are used. Also, the sliding mounting of the sear case absorbs some of the energy of the breech block. The sear can also be damaged by the high velocity of the breech block and the way it is released. The breech may strike the sear when it is not completely lifted, and, in an extreme case, may strike the edge of the sear which may damage it. To avoid this, the trigger mechanism is usually equipped with a device to lift the sear quickly, independently of the trigger movement, known as a quick-break trigger. Two different designs are shown in Fig. 12.26.
Fig. 12.22 Trigger mechanism used on the AK-47 with single shot selected

Fig. 12.26a shows that claw 'a' on the trigger lever pushes the sear down. At the same time, the catch is moved forward by its spring. To stop firing, the trigger is released. The sear, however, is held by the catch until the trigger pushes the catch back through claw 'o'. The sear is released immediately. Fig. 12.26b shows a trigger mechanism with the trigger in its initial position. The sear, however, is held in the released position by the quick-break trigger claw. In this position the rearward-moving breech deflects the quick-break trigger backwards through its protrusion, releasing the sear immediately. The breech is then safely caught.

Fig. 12.23 Trigger mechanism of the AK-47 with automatic fire selected

Figure 12.24 Two-step trigger for firing single shot and automatic

Safety Arrangements

A major safety requirement is that a weapon should not be able to fire when the breech is not fully closed or locked. If the breech is not fully locked, the weapon is safe by means of the relative position of the breech block and the breech block carrier. It is thus guaranteed that the breech block of weapons fired from an open bolt strikes the breech block only after the breech is locked. Weapons fired from a closed breech use the automatic trigger as shown in Figs 12.20 and 12.22, as the safety catch when the breech is unlocked. The automatic trigger is pressed by the breech block carrier only after the breech
Usually the safety catch blocks:
- the trigger
- the sear
- the firing mechanism.

The lever controlling the safety catch may also select the mode of fire which allows the weapon to fire single shot, in bursts or limited burst. An example of a trigger-blocking safety catch can be seen in Fig. 12.27, which shows the trigger mechanism of a Russian AK47 automatic rifle in the safe position. The switch lever presses against the rear lever of the trigger, preventing it from moving. The sear cannot release the hammer.

An example of a mechanism disabling the sear is shown in Fig. 12.21. If the safety catch is turned through an angle, the tripping lever is deflected to pass through the opening in the sear so that it cannot be caught while the trigger is being pressed. A similar example is shown in Fig. 12.26a, where the safety catch (not drawn) moves to lower the trigger lever for claw ‘a’ to pass through the hole in sear ‘b’ disabling it while the trigger is being pressed. Safety catches blocking the firing mechanism are usually auxiliary devices which are additional to the safety features described above.

**Burst Limiters**

As it is difficult to estimate the length of a burst, it is useful to limit the burst with a device which, for small arms, is usually mechanical in operation.
The limiter catches the breech, hammer or striker according to the trigger type. It normally uses the sear engaged in other modes of the weapon operation, and must ensure that it functions after the required number of rounds have been fired. A ratchet limiter, shown in Fig. 12.28, allows three rounds to be fired. When the trigger is pressed, the trigger rod pushes the breech sear to release the breech block. After firing, the rearward-moving breech block lowers the fork lever to turn the ratchet wheel by one tooth. After the last round in the burst the wheel is turned to allow the limiting rod to be released.

A mechanical rack limiter for three rounds, driven by the breech block, is shown in Fig. 12.29. In the cocked position the striker is held on the sear with the breech block in the forward position. Pressing the trigger moves the trigger rod to lower the sear and release the striker. As the breech block, with the striker, moves to the rear a link held by claw A is released and moves to the rear under the action of its spring. At the same time, the link claw moves by one tooth with the rack locked in position by a pawl. As the breech block moves forward, claw A moves the link by another tooth. The last shot in the burst moves the rack to the right, so that claw B hits claw C of the trigger rod and deflects it downward. This releases the sear, which rises under the influence of its spring and catches the striker.
To fire another burst the trigger is released, which frees the pawl so that the rack returns to its initial position under the action of its spring. The trigger rod again contacts the sear by means of its spring. A disadvantage of mechanical limiters is the large number of springs that they use. The simplest and most reliable limiter is the ratchet type, which is also the smallest and most frequently used (as in the French automatic rifle 5.56mm calibre AA 52 FAMAS or Belgian automatic rifle 5.56mm calibre FN CAL).

There is also the requirement for a resetting device to reset the initial value of the round counter if the preceding burst was not completed. The reset device is not always used because of the extra space needed to fit it. Also, burst limiters must not prevent the weapon from working if they fail. Thus they are often separated from the basic firing mechanism.

Adjustable and Set Triggers

To improve trigger control a set trigger can be used, or the trigger made so that it is adjustable. The most important characteristics of a trigger are its pull and movement, which can be designed to be adjustable for target and sniper rifles. By means of screws and springs, the provision for adjustment is to give the trigger the minimum pull and movement whilst ensuring sufficient safety. Trigger pull is measured using weights hung on the trigger.

A set trigger is a method of reducing the trigger pull and movement. It allows the release of the firing mechanism with only a very short movement of the trigger. The velocity of the trigger mechanism components is increased by the set trigger spring. An example of a set trigger is shown in Fig. 12.30, and has two control levers:

- 1 = trigger
- 2 = set trigger lever.

Pressing lever 2 compresses spring 3, and front end 4 of lever 2 is caught by trigger 1. After pressing the trigger (lever 1), front end 4 is released and strikes sear 5. Thus, striker 6 is released.

![Figure 12.30 Set trigger used with hunting and sniper rifles](image)

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13

Auxiliary Mechanisms used in Automatic Weapons

EXTRACTORS

Introduction

After each shot the empty cartridge case must be removed from the chamber by the extractor. The extractor pulls the cartridge case from the chamber by its rim, when the breech block moves to the rear. It also holds the cartridge case on the breech face until the cartridge case is ejected from the weapon.

Extractors are either fixed or spring-loaded. Fixed extractors are machined into the breech block. They may act on one side of the rim, as shown in Fig. 13.1a, or on opposite sides as shown in Fig. 13.1b. Both types retain the cartridge case on the breech face until it is removed by the ejector.

Spring-loaded extractors ride over the rim as the breech closes and are then held in place by a spring. Typical examples are shown in Fig. 13.2.

With a single spring-loaded extractor acting on one side of the rim the cartridge case rim leans against the breech face, as shown in Fig. 13.3b. When a spring-loaded extractor acts on both sides of the rim, the cartridge case is held between the claws by the force of both springs.

A most important feature of an extractor is the claw. The claw is under stress during extraction and during cartridge case ejection: consequently, it must be sufficiently robust to withstand these forces. However, there is limited room in which to fit the extractor. The claw dimensions should allow the cartridge case

![Figure 13.1 Fixed extractors](image)
Cartridge Case Extraction Force

The force to extract a fired cartridge case must be calculated to cover the overall requirement for the weapon cycle and to be able to calculate the necessary dimensions of the claw. An exact solution is difficult because of all the variables involved. It is therefore necessary to make certain simplifying assumptions.

Suppose that the cartridge case is of cylindrical shape with a constant wall thickness along the whole length, as shown in Fig. 13.4. The force necessary to extract the cartridge case from the chamber while there is still propellant gas pressure acting inside the cartridge case is:

\[ F_{EX} = F_T - F_D \]  \hspace{1cm} (13.1)

and

\[ F_T = \pi \cdot d \cdot p_T \cdot \tan \phi \]  \hspace{1cm} (13.2)

\[ F_D = \pi \cdot \frac{d}{4} \cdot p_D \]  \hspace{1cm} (13.3)

After substitution of equations (13.2) and (13.3) into equation (13.1), the following expression is obtained:

\[ F_{EX} = \pi \left( d \cdot 1 \cdot p_T \cdot \tan \phi - \frac{d^2}{4} \right) \]  \hspace{1cm} (13.4)

Equation (13.4) has two unknowns, \( F_{EX} \) and \( p_T \), the pressure between the cartridge case and the chamber walls. Fig. 13.5 shows a half-cartridge case cross-section used to determine \( p_T \). When the cartridge case is in the chamber the forces acting on it are in equilibrium, so that:

\[ p_T \cdot d \cdot 1 = p_D \cdot d_1 \cdot 1 - 2 \cdot \frac{d}{2} \cdot 1 \cdot \sigma_c \]  \hspace{1cm} (13.5)
After rearrangement:

\[
(13.6) \quad p_1 = \frac{d_1^2}{d^2} p_0 - 2 \frac{d}{d} \sigma_c
\]

and substituting into equation (13.4):

\[
(13.7) \quad F_{EX} = \pi \left[ f.1 \left( p_0 - 2 \sigma_c \cdot \delta - p_0 \cdot \frac{d_1^2}{4} \right) \right]
\]

When the cartridge case is being extracted, it is gripped by the chamber walls so that the tangential stress is negative ($\sigma_c < 0$).

This is shown in Fig. 13.6. When the cartridge is fired, the cartridge case deforms until there is no clearance between it and the chamber and then the barrel and chamber expand together. Plastic deformation of the case starts at point 3 and continues to point 4. The barrel is only deformed elastically between points 6 and 7. Thus the barrel wall returns to its original shape after the pressure drops towards point 6 but the cartridge case is permanently deformed, following points 1-2-3-4-5. This causes the cartridge case to be gripped by the barrel. Hook's Law gives:

\[
(13.8) \quad \sigma_c = -E. \Delta
\]

By substituting equation (13.8) into equation (13.7) an expression for the force required to extract a cylindrical cartridge case from the chamber is given:

\[
(13.9) \quad F_{EX} = \pi \left[ f.1 \left( p_0 - 2 \sigma_c \cdot \delta - p_0 \cdot \frac{d_1^2}{4} \right) \right]
\]

If the cartridge case is extracted when there is no gas pressure in the cartridge case ($p = 0$) then:
Because the gases act in all directions there is a force in the bottle cartridge case acting against the extraction force. Thus the force acting on the base of the cartridge case shown in Fig. 13.9 is:

\[ F_D = P_D \cdot \frac{\pi \cdot d_1^2}{4} - P_D \cdot \frac{\pi}{4} \left( d_1^2 - d_2^2 \right) = P_D \cdot \frac{\pi \cdot d_2^2}{4} \]  

Substituting equations (13.12) and (13.13) into equations (13.9) and (13.10) gives:

\[ F_{EX} = \pi \left( P_D \cdot \frac{d_{int}}{2} + 2 \cdot E \cdot \delta_m \left( \Delta - \frac{2 \cdot \alpha \cdot x}{d_m} \right) \right) - \frac{1}{4} \frac{d_2^2}{d_1^2} \cdot P_D \]  

and for \( P_D = 0 \)

\[ F_{EX} = 2 \pi \cdot f.l. \cdot E \cdot \delta_m \left( \Delta - \frac{2 \cdot \alpha \cdot x}{d_m} \right) \]  

Equation (13.15) shows that there is a linear relationship between extraction force and extraction distance, as shown in Fig. 13.10, and that maximum extraction force occurs at the start of cartridge extraction, before the cartridge case has moved. If the cartridge case is extracted when the pressure has dropped to atmospheric the total extraction distance for which there is a force between the case and the chamber is given by:

\[ x_{TX} = \frac{\Delta \cdot d_m}{2 \cdot \alpha} \]  

From Fig. 13.10 the work consumed for cartridge case extraction is:

\[ W_{EX} = \frac{F_{EXmax} \cdot x_{EX}}{2} \]  

If the gases are acting during the cartridge case extraction (i.e. in the period of the additional effect of propellant gases), the acting gas pressure is substituted into equation (13.14). The magnitude of the pressure is obtained from the pressure/time curve.

**EJECTORS**

**Introduction**

The empty cartridge case must be removed from the weapon after extraction from the chamber by the ejector. The position of the extractor and the ejector determines the direction in which the cartridge case is ejected, as shown in Fig. 13.11.
The ejectors may be divided into two groups:

- Ejectors situated in the weapon casing
- Ejectors situated on the breech block.

Ejectors situated in the weapon casing are either fixed, as shown in Fig. 13.12a, or spring-loaded, as shown in Fig. 13.12b. When the breech block moves to the rear the cartridge case hits the ejector and the cartridge case is ejected through an ejection port.

Heavy machine-guns with large cartridge cases impart high-impact energies to the ejectors, which are fitted with springs to reduce the impact, as shown in Fig. 13.12c. Fig. 13.13 is of an ejector in the shape of a two-armed lever. The lever is pivoted in the weapon casing. When the breech is closed the lever swings into a recess in the casing. When the breech block moves to the rear it strikes one arm of the lever and the forward arm pushes the cartridge case from the breech face.

Wedge-type breeches and revolver drums are fitted with lever ejectors, less frequently with rolling ejectors.

Fig. 13.14a shows the ejectors for a revolver drum where the cartridge case is ejected by the opening movement of the carrier. Case extraction and ejection take place in a single movement using the one lever. To reduce the impact and increase the extraction force the movement starts with a rolling motion as shown in Fig. 13.14b for a wedge breech. At the start of the movement the ejector rolls away, the wedge acting through the long arm.

Fig. 13.15 shows examples of ejectors fitted into the breech face. Breech block ejectors have an element that moves in the breech block. When the breech block moves to the rear a protrusion on the weapon casing hits the inclined surface and the ejector hits the cartridge case and ejects it. Fig. 13.15c shows an ejector which is a spring-loaded pin. While the cartridge case is in the chamber the ejector is in the position shown in the figure. When the neck...
of the cartridge case leaves the chamber the ejector is driven forward by a spring and the cartridge case is pushed from the face of the breech block.

Weapons with twin-fx extractors are often not fitted with an ejector: the fired cartridge case is expelled from the weapon by the action of the next cartridge inserted into the extractor claws.

An unusual method of cartridge case ejection was used with the ShKAS 7.62mm calibre aircraft machine-gun. The ejector was fitted with a steel flag which had a mast positioned parallel to the breech block axis. With the breech block to the rear the ejector flag rotated and pushed the empty cartridge case from the ejector claws. When moving forward, the breech block pushed the cartridge case forwards out of the weapon through an ejection tube parallel to the barrel axis. Ejection by a forward-moving breech block is used by several modern automatic systems, e.g. the 12.7mm NSV machine-gun and the McDonnell Douglas chain-gun family of weapons.

**Effect of Case Ejection on Breech Block Movement**

During cartridge case ejection the case is retained by the extractor. The cartridge case base hits the ejector and reduces the velocity of the breech block, as shown in Fig. 13.16a and diagrammatically in Fig. 13.16b.

The mass at point A replaces the ejector, the mass at point B replaces the cartridge case and the mass at point C replaces the breech block and carrier. Fig. 13.17 is a schematic arrangement of the three parts with point masses and the velocity vectors. The velocity vector \( v_y \) is the relative velocity between points B and C.

The equations for calculating the velocity of the breech block carrier after the case impacts with the ejector are derived in reference:

\[
(13.18) \quad v'_A = v_A - \frac{(v_A - v_C - v_y)/(1 + \varepsilon)}{1 + \frac{m_A}{m_A + m_C} \cdot \mu} \left( 1 - \frac{m_A (\mu - 1)}{m'} \right)
\]

\[
(13.19) \quad v'_C = v_C + \frac{(v_A - v_C - v_y)/(1 + \varepsilon)}{1 + \frac{m_A}{m_A + m_C} \cdot \mu} \left( 1 - \frac{m_A (\mu - 1)}{m' \cdot \mu} \right)
\]

\[
(13.20) \quad v'_y = v_y + \frac{(v_A - v_C - v_y)/(1 + \varepsilon)}{1 + \frac{m_A}{m_A + m_C} \cdot \mu} \left( 1 - \frac{m_A (\mu - 1)}{m'} \right)
\]

\[
(13.21) \quad \mu = 1 + \frac{m' (1 - \cos \alpha)}{(m_A - m') i}
\]

\[
(13.22) \quad m' = m_A + m_B \cdot \cos \alpha
\]

\[
(13.23) \quad m_0 = m_A + m_B + m_C
\]
The transmission ratio, $i$, is the transmission ratio at the moment of impact and is given by:

$$i = \frac{V_f}{V_A - V_C}$$

Typical values of the coefficient of restitution, $\varepsilon$, for steel are between 0.35 and 0.5 and for brass are between 0.4 to 0.53. N.B. the velocities marked with an ‘accent’ (') represent velocities after impact; without an ‘accent’ the velocities are before impact.

The cartridge case is held on the face of the breech block until ejection, so that its relative velocity, $v_y'$, equals zero and for $v_y = 0$ equations (13.18), (13.19) and (13.20) become:

$$v_{y}'_{A} = v_{A} - \frac{(V_{A} - V_{C}) (1 + \varepsilon)}{1 + \frac{m_{A}}{m_{B} \cdot t^{2} \cdot \mu}} \cdot \left(1 - \frac{m_{A} (\mu - 1)}{m' \cdot \mu}\right)$$

$$v_{y}'_{C} = v_{C} + \frac{(V_{A} - V_{C}) (1 + \varepsilon)}{1 + \frac{m_{A}}{m_{B} \cdot t^{2} \cdot \mu}} \cdot \left(\frac{m_{A} (\mu - 1)}{m' \cdot \mu}\right)$$

$$v_{y}' = \frac{1}{1 + \frac{m_{B} \cdot t^{2} \cdot \mu}{m_{A}}}$$

The influence of cartridge case ejection on the motion of the weapon casing, breech block and ejected cartridge case can be determined from the above equations. If the weapon casing is fixed during ejection then:
Two basic types of rammer are used:

- axial breech rammer which are part of the breech block
- wedge and revolver breech rammers which are an independent assembly.

Axial breech rammer are lugs or surface projection on the front of the breech block, as shown in Fig. 13.18.

This type of rammer is frequently used for both belt and magazine feed. In magazine feed the breech block rammer passes between the side walls of the mouth of the magazine, catches the cartridge and pushes it into the chamber. The width of the rammer, b, in Fig. 13.18 is the width of the mouth of the magazine with the required clearance. In a breech block with spring-loaded extractor the shape of the rammer and extractor allows the base of the cartridge case to fit into the bed of the breech block head and the extractor claw to fit over the case rim during ramming.

With two-sided fixed extractors, ramming takes place after the transverse movement of the cartridge case into the extractor claws. No special rammer is required and chambering is carried out by the breech block. To ensure that the cartridge does not move out of the extractor claws a spring-loaded retaining catch is fitted into the breech face.

Automatic systems with a breech closing movement not along the axis of the weapon are normally fitted with an independent rammer, as shown in Fig. 13.19. The rammer is usually controlled by the breech block carrier. With wedge breeches, the cartridge is not fully rammed into the chamber, but relies upon the inertia of the cartridge for the final movement.

**RAMMERS**

Rammers are used to move the cartridge from the loading space into the chamber and consist of a variety of slides, retaining catches and lowering pieces. To assist in guiding the cartridge the shape of the rear of the barrel is designed to feed the cartridge into the chamber.
BUFFERS

The breech block and carrier in automatic weapons are driven to the rear of the weapon at a velocity necessary to achieve the required rate of fire and to carry out all of the other functions of the weapon. The return spring is compressed during this action and stores energy to complete the rest of the weapon cycle. A large, heavy spring would increase the length of time to complete the rearward movement of the breech block, so a weak return spring is used with a strong buffer spring to absorb the excess breech block energy. In small calibre weapons with a low breech block kinetic energy a buffer is not used: the breech block hits the weapon casing each time the weapon is fired. Fig. 13.20 shows the displacement/time curves for different methods of buffering.

The buffer shown in Fig. 13.21 influences the rate of fire by its:

- stiffness
- displacement
- spring type.

Spring stiffness is given by the slope of its force/displacement curve. A spring with a greater stiffness for the same displacement and stored energy will give a higher rate of fire, as shown in Fig. 13.22.

![Buffer Diagram](image1)

Figure 13.20 Change in function cycle time for (1) strong return spring, (2) weak return spring and heavy breech impact, (3) weak return spring and buffer

For two springs of the same stiffness but different displacements, the spring that has the shortest displacement is normally used, as shown in Fig. 13.23. For both springs the stored breech kinetic energy, \( E_{\text{kin}} \), is the same and is equal to the area under the force/displacement curve.

![Buffer Diagram](image2)

Figure 13.21 Arrangement of a typical buffer
The buffer spring significantly influences the rate of fire of the weapon. All springs suffer energy losses, so that the energy required to compress them is greater than the energy released when they expand. Helical springs with between 5–20% energy loss, washer springs with between 30–50% energy loss and ring springs with up to 70% energy loss are often used in buffers.

Fig. 13.24 shows how the application of a spring with a low energy loss (A) gives a higher rate of fire than a spring with high energy loss (B) under the same conditions. The breech velocity after striking the buffer is higher for spring A than for spring B.

The main task of the buffer in some automatic systems is to reduce the impact of the breech block against the weapon casing and to absorb most of the impact energy. Where large energy losses are not required, a helical spring is used and the design with the smallest dimensions uses wire of a rectangular cross-section, as shown in Fig. 13.25.

The effect of the buffer spring losses must be included in the buffer design. The effect of the return spring is negligible during buffer movement and is only considered during the breech block impact with the buffer, when a
number of separations occur. Because of the large difference between the breech block and the buffer masses it is usual to ignore these separations and assume that the impact is non-elastic, which is helped by the damping effect of the energy losses in the buffer spring.

**REBOUND BUFFERING**

After the breech has locked, the breech block carrier impacts the breech block, and because the material of both components is elastic, the breech block carrier will rebound. It is possible for this rebound to cause the breech to open as the weapon is being fired, which would damage the weapon and endanger the firer. For low calibre weapons the rebound is small, and the underside in the breech block carrier prevents the breech from opening. Fig. 13.26 shows the time/displacement curve for the breech block carrier for the 7.62mm calibre AR Mod 58 which shows how the breech block carrier underside prevents the breech from unlocking. In larger calibre weapons the rebound energy is sufficiently high to cause problems, and so some form of buffering system must be used.

![Graph showing time/displacement curve for the breech block carrier](image)

*Figure 13.26 Functional diagram for the Czech Mod 58 automatic rifle*

A dynamic method of reducing breech block carrier bounce uses a weight supported by a spring, and is shown in Fig. 13.27. After the breech block has stopped and rebound occurs the weight continues to move in the original direction. Rebound of the carrier will be reduced by the force of the compressed spring and by the impact of the weight with the carrier.

To prevent the breech block carrier from rebounding totally, a pawl or lug may be used. To function correctly, they must lock the carrier before firing and release it before the breech opens. Fig. 13.28 shows a method which uses the barrel recoil. Breech block carrier rebound is prevented by a safety plate which fits into a recess in the breech casing. The breech casing, to which the barrel is attached, moves to the rear. A release lever in the weapon casing acts on the safety plate and releases the breech block carrier at the beginning of barrel recoil.

Rebound catches are also used on gas-operated weapons. Fig. 13.29 shows a system used with larger calibre weapons.

**BOLT HOLDING MECHANISMS**

Automatic weapons are usually designed so that when all of the ammunition in a belt or magazine has been fired the breech block is held to the rear. This signals that the weapon is empty and avoids the necessity to re-cock the weapon after it has been reloaded.

The bolt holding arrangement for weapons firing from a closed breech consists of a helping sear controlled by the floor plate of the magazine, as shown in Fig. 13.30. The helping sear is lifted by the floor plate which holds the breech block in the rear position while the last cartridge is fired. This design is
used with most pistols, sub-machine-guns and automatic rifles. After changing the magazine, a spring sear releases the helping sear or may be released by a bolt release lever.

In weapons which fire from an open breech position the trigger sear is used to hold the bolt open. In small calibre weapons the breech block can be held directly by the floor plate of the magazine, as shown in Fig. 13.31. After the

magazine has been changed the breech block is held on the trigger sear and firing can commence immediately.

Another method used to hold the bolt open after the last cartridge in automatic weapons has been fired, is to employ a transmission lever which can lift the helping sear using the trigger, as shown in Fig. 13.32. The tripping rod contacts the cartridges and after the last has been fired one will move forward to interrupt the contact of the trigger and the sear.

WEAPON COCKING

Before a self-powered weapon can be fired it must be cocked: this action consists of compressing the return spring and the striker spring. In weapons which fire from a closed bolt, the breech block returns to the forward position, inserts
a cartridge and the striker is held on its sear. In recoil-operated weapons, the barrel is also pulled to the rear during cocking. If the force required to compress the return spring is small, the weapon can be cocked manually with one direct pull using a cocking handle or, with pistols, by gripping grooves machined on the rear of the slide.

The cocking handle is either fixed to the breech block carrier or can be separated from the carrier after cocking and pushed forward, where it fixes to the receiver. Fixed cocking handles reciprocate with the bolt and can cause injury or become caught in clothing. To avoid this, in some cases the cocking handle overlaps the weapon's contour lines by only a small amount and is smooth in shape, but such designs are difficult to grasp.

The breech sear in the trigger mechanism can be also used to cock the weapon. The trigger mechanism slides in a slot in the receiver and is held by a pawl. By pushing the trigger mechanism forwards the breech block carrier is caught by the trigger sear. It can then cock the weapon by pulling it to the rear. This method of cocking does not require the hand to be removed from the grip during cocking. It is used with the Czech ZB-37 medium machine-gun, the Mod 52 and 52/57 light machine-guns and the Mod 59 universal machine-gun. Some heavy machine-guns are made so that an empty case can be fitted over the cocking lever to make it easier to grip.

A lanyard with a te handle can also be used for weapon cocking, often where the return spring is directly behind the breech. After cocking, the rope is returned to its original position by means of a spring, as shown in Fig. 13.33. This is the method used in the 30mm AGS-17 grenade launcher.
Dynamics of Automatic Weapons

FORCES ACTING ON THE FIRER

When a shot is fired, in addition to the desired effect of imparting velocity to the projectile, there are also other effects which are undesirable. These include electromagnetic radiation, smoke, toxic fumes as well as thermal and pressure effects. The high-pressure propellant gases impart stresses to the barrel and are the cause of forces and moments transmitted to various parts of the weapon. At the same time a pressure wave is created at the muzzle which acts on the surroundings.

The analysis of the force imparted to the weapon user during firing, and how this force can be reduced, is complex. The solution involves the design of the whole weapon and of every assembly group and component. The loading forces and moments acting along the direction of the axis of the barrel consist of:

- the force acting on the projectile
- resultant force caused by the propellant gases flowing from openings in the barrel
- torque caused by the spin being applied to the projectile by the rifling.

FORCES ACTING ON THE PROJECTILE

When a gun is fired an acceleration is applied to:

- the projectile
- the propellant and its combustion products
- the recoiling parts of the weapon.

In addition to this, there are other weapon parts and the firer's body which are subject to acceleration. To simplify the analysis these additional movements are neglected. The effect of this simplification on the accuracy of the final solution is small.

The projectile and recoiling masses are rigid and of constant mass with fixed centres of gravity. The elements within the cartridge consist of the unburnt propellant and the propellant gases. These moving masses can be defined as a system of points and bodies, the motion of which can be described in terms of the motion of their instantaneous centres of masses and their movements about this point.

The system of projectile, elements within the cartridge case and recoiling parts have a variable mass. The mass varies when the projectile leaves the barrel and when the propellant gases flow from the different barrel openings. The instantaneous mass of the elements in the cartridge case, $m_e$, in the barrel at time $t$ can be expressed as the initial mass of the elements, $m_{e0}$, less the mass flow from the barrel. Thus:

\[ m_e = m_{e0} - \int_0^t \dot{m} \, dt \]

The total instantaneous mass flow of the gases from the barrel, $m$, is given by:

\[ m = \sum m_i \]

In addition to the weight of the projectile and the mass of the elements in the cartridge case there are other forces that affect the system, as shown in Fig. 14.1.

These forces are in dynamic equilibrium and their components in the direction of the longitudinal axis of the barrel are given by:

\[ F_b + F_e - F'_b + \sum F_k \cos \theta_k - R + G \sin \phi = 0 \]

The component of inertia force, $F'_b$, acting on the centre of projectile mass, $m_p$, is given by:

\[ F'_b = F'_b - m_b \dot{x} \]

where the component of inertia force, $F'_b$, caused by the relative movement of the projectile against the barrel is:

\[ F'_b = m_b v \]

The component of inertia force, $F_e$, acting on the instantaneous centre of mass of the elements in the cartridge case at a given instant is:

\[ F_e = F'_e - m_e \dot{x} \]

where the component of inertia force, $F'_e$, caused by the relative movements of the elements in the cartridge case by the outflow of gases from the barrel is given by:

\[ F'_e = m_p \dot{x}_p - 2 m \dot{x} = \dot{m} \dot{x} \]
The reactive force, \( F_{ri} \), caused by the outflow of gases through the i-th opening in the barrel is dependent on the critical cross-sectional area of the i-th opening, \( S_i^* \), and the total (stagnation) pressure for the i-th opening, \( p_{oi} \), and is given by:

\[
F_{ri} = m_i \dot{v}_{si} C_d \frac{S_i^*}{S^*} p_{oi}
\]  
(14.8)

The expressions for the forces \( F_b \) and \( F_c' \) have been simplified because some of the components of the elements within the cartridge case have been neglected where they have a negligible effect on the overall solution.

Values of \( \nu_{si} \) and \( C_d \) are dependent on the actual shape of the i-th orifice from which the gas is flowing. Equation (14.3) can be rearranged in the following form:

\[
m_c \dot{x} = (F_{b} + F_c + F_{rc}) - R_c
\]  
(14.9)

where:

- \( m_c = m_{np} + m_b + m_e \)
- \( m_c = m_{np} + m_e \) when the projectile has left the barrel.

Also, the total reactive force on the barrel is given by:

\[
F_{rc} = \sum_{i=1}^{N} F_{ri} \cos \theta_i
\]  
(14.10)

and the total breaking resistance is given by:

\[
R_c = R - G_n \sin \theta
\]  
(14.11)

To simplify equation (14.3) and (14.11) the concept of a system internal force is used, denoted as the force, \( F_s \), acting on the projectile, as shown in Fig. 14.1. Thus:

\[
F_s = F_b + F_c + F_{rc}
\]  
(14.12)

It is assumed that the force \( F_s \) is non-zero if at least one of the forces \( F_b, F_c \) or \( F_{rc} \) is non-zero.4

Using a system internal force the equation of motion (14.3) is written in the form:

\[
m_{np} \ddot{x} = F_s - R_c
\]  
(14.13)

In practice, such a form of the equation of motion is only of importance if, with little error, the force \( F_s \) can be considered as an external system force. In most cases it has been found that this assumption gives errors of between 2-3% for the value of \( F_s \), which is acceptable when comparing the errors in
determining the internal pressure, velocity and displacement of the projectile. Thus the force acting on the projectile is considered to be an external force. It is also assumed that it depends only on time and is independent of other forces acting on the system, especially the total force \( R_c \) acting against the recoil force.

When solving the equations of motion (14.3), (14.9) and (14.13) the initial conditions of the system at the instant \( t_0 \) when the force acting on the projectile is non-zero, \( x(t_0) = x_0 \) and \( \dot{x}(t_0) = \dot{x}_0 \) must be assigned. The instant of ignition of the primer is taken to be the start of the process, \( t = 0 \).

Analysis of the equation of motion (14.13) shows that the movement of the recoiling parts has two limiting cases:

- If the barrel mounting is absolutely rigid the recoil movement cannot take place, so that \( \ddot{x} = \dot{x} = 0 \). Thus if \( R_{CA} \) is the value of \( R_c \) when the weapon is in a rigid mounting and \( F_{CA} \) is the value of \( F_c \) in the same mount then:

\[
R_{CA} = F_{CA} = F_r + F_{BC}
\]

(14.14)

- It is possible both theoretically and practically for the barrel and recoiling parts to be mounted so that the resisting force is zero, thus:

\[
R_C = 0
\]

(14.15)

Such a case is known as free recoil. The displacement, velocity and acceleration for free recoil is denoted as \( w, \dot{w} \) and \( \ddot{w} \) and for braked or real recoil as \( x, \dot{x} \) and \( \ddot{x} \). It should be noted that the internal ballistic calculations of pressure, velocity, acceleration and displacement are usually carried out for conditions of free recoil.

The equation of motion for free recoil results directly from the equation for braked recoil (14.13). Thus:

\[
m_p \ddot{w} = F_c
\]

(14.16)

If it is assumed that the initial conditions of free recoil are of zero value then \( w(t_0) = 0, \dot{w}(t_0) = 0 \).

Using equation (14.15), equation (14.13) for braked recoil can be written in the following form:

\[
x = w - \ddot{A}
\]

(14.17)

where the deceleration of the recoiling parts, \( \ddot{A} = \frac{k_c}{m_p} \).

Integrating equation (14.17):

\[
\Delta x = x - x_0 = (t - t_0) \ddot{A} = w - \ddot{A}
\]

(14.18)

where

\[
\Delta x = x - x_0 = (t - t_0) \ddot{A} = w - \ddot{A}
\]

(14.19)

where

\[
w = \int_{t_0}^{t} \dot{w} \, dt = \frac{I_p(t)}{m_p}
\]

(14.20)

\[
\ddot{A} = \int_{t_0}^{t} \dddot{A} \, dt = \frac{I_p(t)}{m_p}
\]

(14.21)

Where \( I_p(t) \) is the instantaneous value of the projectile impulse:

\[
I_p(t) = \int_{t_0}^{t} F_c(t) \, dt = m_p \cdot \ddot{w}
\]

(14.22)

\[
I_b(t) = \int_{t_0}^{t} R_c(t) \, dt = m_p \cdot \ddot{A}
\]

(14.23)

The impulse of the projectile is of great importance for the following considerations. Substituting the expression for the external force acting on the projectile \( f_c \), equation (14.12), into equation (14.22) and integrating, then:

\[
I_p(t) = m_p \cdot \dot{v}_s(t) = \Delta H_b(t) + \Delta H_p(t) + I_{BC}(t)
\]

(14.24)

Where the change in projectile momentum is given by:

\[
\Delta H_b(t) = m_p \cdot \dot{v}_s(t)
\]

the instantaneous absolute value of the projectile velocity for free recoil is given by:

\[
\dot{v}_s(t) = v - \ddot{w}
\]

(14.26)

The change in momentum of the elements of the components in the cartridge case for free recoil is \( \Delta H_p(t) \), and the instantaneous value of the total impulse of the reactive forces acting on the recoiling part for free recoil is:

\[
I_{BC}(t) = \int_{t_0}^{t} F_{BC} \, dt
\]

(14.27)

For the different times of action of the reactive forces \( F_{BC} \) of the gases flowing from the barrel the weapons are classified into:

- weapons with no gas flow while the projectile is in the barrel
- recoilless weapons
- gas-operated weapons
- weapons where gas flows past the projectile (worn barrel and mortars)
- weapons with muzzle attachments.

There are also different combinations of the above.

Most weapons consist of the arrangement whereby there is only flow of propellant gas from the muzzle after the projectile has left it. To identify the quantities related to such weapons a superscript zero is used, as in the following examples: \( F_0 \), \( I_0(t) \), \( w_0 \), \( w' \), \( x_0 \). They represent a comparison standard used to compare the same quantities of other barrels.

The internal ballistics cycle is shown in Fig. 14.2 and is divided into three separate periods.

The first period is the ignition of the propellant charge, which begins with the initiation of the cartridge primer and starts at \( t = 0 \) and ends at \( t_p \) which is when the projectile begins to move in the barrel.

The second period is that time for the motion of the projectile in the barrel and starts at time \( t_p \) and ends at time \( t'_{\infty} \) when the base of the projectile has left the barrel. Duration of the projectile motion in the barrel is \( t_p = t'_{\infty} - t_p \).

The third period is when the propellant gases flow from the muzzle of the barrel, and starts at time \( t'_m \) and ends at time \( t'_{\infty} \). This period ends when the reactive forces of the propellant no longer act on any part of the weapon. Duration of this third period is \( t'_{m} = t'_{\infty} - t'_{m} \). Its length is between four and five times greater than the length of time that the projectile is moving down the barrel.

The time \( t'_{m} \) is usually chosen as the moment when the pressure in the cartridge chamber falls to a value of 180 kPa. In previous considerations, which were incorrect, it was assumed that the transition from critical flow of the propellant gases occurred at this pressure.4

The analysis of the force on the projectile, \( F_0(t) \), and its impulse, \( I_0(t) \) with respect to time is now considered. (See Fig. 14.2).

In the first period neither the projectile nor the centre of mass of the elements making up the components within the cartridge case moves with respect to the barrel (\( F_0^b = 0, F_0^d = 0 \)). Propellant gases do not escape from the barrel (\( F_0^e = F_0^g = 0 \)), and therefore the force on the projectile is zero (\( F_p = 0 \)) and the impulse equals zero (\( I_0 = 0 \)).

During the second period the main component of the force acting on the projectile is the inertia force, \( F_p \), whilst the inertia force acting on the centre of mass of the inner cartridge case elements, \( F_p \), is much smaller. Only for the case of sub-calibre projectiles are the values of the two forces similar. At the end of the second period \( (t = t'_{m}) \), the total change of projectile momentum is:

---

Figure 14.2 Internal ballistics cycle divided into three separate periods
Military Small Arms

\[ \Delta H_{bu}^0 - m_b \cdot v_0 \]

and the total change of momentum of the elements in the cartridge case is:

\[ \Delta H_{cu} = m_{cu} \cdot v_{cu} \]

Where \( v_0 \) is the initial projectile velocity and is approximately equal to the absolute velocity of the projectile at free recoil, which is given by:

\[ v_0 = v_c'(t'_{cu}) = v_{cu} \cdot w_u \]

It is assumed that:

\[ v_{cu} = \frac{\alpha_{cu}}{2} \cdot v_0 \]

In practice, \( \alpha_{cu} \) is the coefficient of effect of the distribution of the inner cartridge elements along the barrel axis at time \( t'_{cu} \), can be assumed to be unity with little loss in accuracy.

During the second period, the propellant gases do not escape from the barrel \((F_{i0}^0 = F_{i0}^0 = 0)\), thus:

\[ F_i^0(t) = F_{b0}^0 + F_{e0}^0 \]

and

\[ I_{i0}^0(t) = m_{ip} \cdot \dot{w}_0 = \Delta H_{bu}^0(t) = \Delta H_{cu}^0(t) \]

The impulse of the projectile after the second period, \( t = t'_{cu} \), is given by:

\[ I_{i0}^0 = m_{ip} \cdot \dot{w}_0^0 = \delta_{i0}^0 \cdot \Delta H_{bu}^0 \cdot \left( m_b \cdot \frac{\alpha_{cu}^0}{2} - m_{cu} \right) \cdot v_0 \]

Where:

\[ \delta_{i0}^0 = \left[ 1 + \frac{\alpha_{cu}^0}{2} \left( \frac{m_{cu}}{m_b} \right) \right] / \delta_{v}^0 \]

which is the coefficient expressing an increase in \( I_{i0}^0 \) with respect to the projectile momentum, \( \Delta H_{bu}^0 - m_b \cdot v_0 \), due to the influence of the accelerated inner cartridge elements with respect to the barrel.

\[ \delta_{v}^0 - 1 = \left[ \frac{m_b}{m_{ip}} \right] \left[ 1 + \frac{\alpha_{cu}^0}{2} \left( \frac{m_{cu}}{m_b} \right) \right] \]

\( w_{c0}^0 = \dot{w}_0^0 (t'_{cu}) \) is the free recoil velocity at the instant the projectile leaves the muzzle of the barrel.

\[ \dot{F} = m_{cu}^0 \cdot v_{cu}^0 \]

Its impulse is \( I_{i0}^0(t) \) and \( I_{i0}^0(t) = t'_{cu} \rightarrow 0 \)

where \( m_{cu}^0 \) and \( v_{cu}^0 \) are the values \( m_i \) and \( v_{cu} \) given in equation (14.4).

The mass of gas in the barrel decreases with time. There is also a decrease in the absolute velocity of the gases’ instantaneous mass centres so that their increment of momentum, \( \Delta H_{bu}^0(t) \), decreases until at time \( t'_{cu} \) it is almost zero. This phenomenon is characterised by the general inertia force acting in the opposite direction to that during the second period \( F_{e0}^0 < 0 \) so that:

\[ F_e^0 = F_{e0}^0 - I_{i0}^0 = F_{i0}^0 - |F_{e0}^0| \]

and

\[ I_{i0}^0(t) = m_{ip} \cdot \dot{w}_0 = \Delta H_{bu}^0(t) + \Delta H_{cu}^0(t) + I_{i0}^0_{c0}(t) \]

At the end of the third period the total impulse is generally equal to:

\[ I_{i0}^0 = m_{ip} \cdot \dot{w}_c + \Delta H_{bu}^0 + I_{i0}^0_{c0} = \dot{w}_c \cdot \Delta H_{cu}^0 \]

where:

\[ \dot{w}_c = 1 + \left( \frac{I_{i0}^0_{c0}}{\Delta H_{bu}^0} \right) = 1 + \alpha \left( \frac{I_{i0}^0_{c0}}{\Delta H_{bu}^0} \right) \]

The value of the total impulse of the reactive force \( I_{i0}^0 \) is usually expressed in the form \( I_{i0}^0 = \alpha \cdot I_{i0}^0_{c0} \), where \( \alpha \) is the total characteristic of the barrel gas system, including the leakage of propellant past the projectile. For most weapons \( \alpha \approx 1 \).

To obtain \( I_{i0}^0 \) the relationship introduced in the 19th century by F Krupp is used:

\[ I_{i0}^0 = \dot{w}_c \cdot m_{cu} \cdot v_0 \]

Dynamics of Automatic Weapons
$\beta$ is the coefficient of additional action of the gases on the muzzle of the weapon. Thus for a standard barrel:

\[ F_s = m_p \cdot \dot{w}_s = \delta_s \cdot H_{\beta 0} - (m_b + \beta \cdot m_{\text{gas}}) \cdot v_0 \]

where:

\[ \delta_s = 1 + \beta \left( \frac{m_{\text{gas}}}{m_b} \right) \]

Thus the total impulse during the third period is:

\[ F_{3t} = m_p \left( \dot{w}_s + \dot{w}_a \right) = 1 \frac{s}{2} - 0 \frac{s}{2} + \frac{1}{(s)} \cdot \left( \beta - \frac{\alpha_{0}}{2} \right) m_b \cdot v_0 \]

where the final recoil velocity is:

\[ \omega_0 = \omega_0 \left( t' \right) \]

Actual calculations for a typical weapon give values of $\delta_s$ between 1.3 and 1.7. For sub-calibre projectiles the value can be higher. Thus the effect of the propellant gases flowing from the muzzle cannot be neglected.

The force acting on the projectile, $F_p$, can be expressed as an inner force of the system by means of pressure $p(x, t)$ and friction forces $T(x, t)$ acting on the walls of the barrel. Fig. 14.3a shows a simplified arrangement of a gun barrel with a plain cylindrical bore after the projectile has left the barrel (ts $t'_{s}$).

Atmospheric pressure, $p_a$, (the practical effect of which can usually be neglected), acts on the outside of the barrel. Thus:

\[ F_s = F_p = S \cdot p_D - T - S \cdot \dot{p}_a = \delta_t \cdot S \cdot p_D \]

Where the coefficient expressing the effect of friction forces of the gases on the walls of the barrel and the effect of atmospheric pressure is given by:

\[ \delta_t = 1 - \left( \frac{p_a}{p_D} \right) - \left( \frac{F_t}{S \cdot p_D} \right) \]

Fig. 14.3b shows a section through a more realistic barrel. The coefficient $\delta_t$ may include the cartridge chamber and rifling lead and rifling taper, if any. The effect of the muzzle arrangement is given by the force acting on the muzzle:

\[ F_{me} = \Delta F_{re} = F_{re} - F_{re} = \int_{x_0}^{x_f} p(x, t) \cdot \sin \alpha(x) \cdot \sigma(x) \, dx \]

The circumference of the muzzle arrangement in section $x$, which is acted upon by pressure $p(x, t)$, is given as $\delta(x)$ and the angle between a tangent to the internal surface of the muzzle system and barrel longitudinal axis as $\alpha(x)$.

\[ F_{p} = F_{D} = S_{D} \cdot p_{D} - (S_{D} - S) \cdot p_{a} - F_{T} = \delta_{t} \cdot S \cdot p_{D} \]

where:

\[ \delta_{t} = \left( S \frac{p_{a}}{S} \right) - \left( S \frac{p_{D}}{S} \right) - 1 \left( \frac{p_{a}}{S} \right) - \left( \frac{F_{T}}{S \cdot p_{D}} \right) \]

and:

\[ F_{s} = F_{D} - F_{br} + F_{me} = x(t), F_{D} \]
For most practical purposes the following relationships are used:

\[(14.51)\]  
\[F_p = x(t) . F_D\]

\[(14.52)\]  
\[F_D = \delta_l . S . p_D\]

\[(14.53)\]  
\[x(t) = 1 - \phi_{lb} + \phi_{bc}\]

The value of \(\delta_l\) is usually taken as unity. \(x(t)\) includes the effects of the projectile movement and the barrel gas system.

\(\phi_{lb} = F_{lb}/F_p\) is the coefficient of influence of the frictional resistance against the movement of the projectile in the barrel. For calculation purposes an average value of \(\phi_{lb}\) is used. The actual value is given by: \(\delta_{lb} = 0.03\) to 0.10 for barrels with a smooth rifling lead. \(\delta_{lb} = 0.03\) to 0.10 for barrels with a rifled cylindrical rifling lead. The actual value depends on the projectile construction and design. For jacketed projectiles \(\delta_{lb} \approx 0.1\) As soon as the projectile leaves the barrel \(\phi_{lb} = 0\).

\(\phi_{bc} = \Delta F_{bc} / F_D = (F_{bc} - F_D) / F_D\) is a general coefficient of the effect of the propellant gases flowing from the muzzle and other gas arrangements. For a normal barrel \(\phi_{bc} = 0\).

The variation in the force acting on the projectile, \(F^0_p(t)\), and its impulse, \(I^0_p(t)\), with respect to time is defined by means of \(F_D\) and \(x(t)\) for a normal barrel as shown in Fig. 14.4.

The previously derived expressions can also be applied to a weapon with an unlocked breech (blow-back). The mass of the recoiling parts is given by:

\[(14.54)\]  
\[m_p = m_{pb} + \frac{1}{3} m_n + m_D\]

The force acting on the projectile, \(F_p\), is substituted by the force acting on the breech by the propellant gases:

\[(14.55)\]  
\[F_{pb} = F_D - F_{bx} = x_{bx}(t) . F_D\]

Where the force acting on the cartridge case base, \(F_{bx}\), is given by \(F_D = S_pP_D\). The impulse – force characteristic for an unlocked breech is given by:

\[(14.56)\]  
\[x_{bx}(t) = (S_p/P_D) - (F_{bx}/F_D)\]

The barrel is affected by the propellant gases, projectile and cartridge case, so that:

\[(14.57)\]  
\[F_s = F_{ex} + F_{xb} - F_{bx} - F_{bx} = x_s(t) . F_D\]

The force component along the axis of the barrel, \(F_{bx}\), results from propellant gas pressure, and friction forces acting on the barrel bore, cartridge chamber and forcing cone.

![Figure 14.4 Variation in forces acting on the projectile, and its impulse, with respect to time](image-url)
FLOATING MOUNTS AND ADVANCED PRIMER IGNITION

The power developed by the projectile is given by:

\[(14.58)\]

\[P_\text{p} = F_\text{s} \cdot x\]

Equation (14.18) can be used to determine the recoil velocity as the recoiling parts are slowed. The instantaneous work is given by:

\[(14.59)\]

\[A_\text{w} (t) = \int_{t_0}^{t} P_\text{w} \, dt\]

Substituting the velocity of the recoiling parts as they are slowed, \(v\) by the velocity of free recoil, \(w\), the relationship for power for free recoil, \(P_\text{w}(t)\), and instantaneous work for free recoil, \(A_\text{w}(t)\), can be obtained. This results from the equation of motion for free recoil (14.16) that instantaneous work of the projectile and propellant for free recoil equals the kinetic energy of the recoiling parts for free recoil. Thus:

\[(14.60)\]

\[A_\text{w} (t) \equiv E_\text{KE} (t) = \frac{1}{2} m_{\text{p}} \cdot w^2 (t) = \left( I_2 (t) \right) / (2 m_p)\]

At time \(t'_c\), the end of the action of \(F_\text{w}\), the total work is given by:

\[(14.61)\]

\[A_\text{w} = (1 - \eta_k + \eta_b) A_\text{w}\]

and

\[(14.62)\]

\[A_\text{w} = \frac{1}{2} m_p \cdot w_c^2 = \left( I_2 / (2 m_p) \right)\]

Where \(w_c = \dot{w} (t'_c)\) is the final velocity during recoil and:

\[(14.63)\]

\[\eta_k = (2H_{k}) / (I_{k}) = 2 \cdot (m_{\text{p}} \cdot x_o) / (I_{k}) = 2 \cdot \left( x_o / w_c \right)\]

= coefficient of effects of initial conditions and:

\[(14.64)\]

\[\eta_b = (\Delta A_\text{w}) / (A_\text{w})\]

= coefficient of effect of the slowing resistance, \(R_\text{w}\), or its impulse \(I_\text{w} (t'_c)\).

For most practical systems \(\eta_b\) ranges in value from 0 to 0.25, usually being 0.1. \(\eta_k = 1\) and \(\eta_b = 0\) for absolutely rigid mounting of the recoiling parts.

The significance of the coefficients is shown in Fig. 14.5. If the force acting on the projectile, \(F_\text{e}(t)\), is dependent upon the recoil travel, \(w(t)\), or on the movement during braked recoil, \(x(t)\), the instantaneous values of force, \(F_\text{e}(t)\), at time \(t_b\) are displaced from one another by \(z(t_b) = w(t_b) - x(t_b)\). The graph shown in Fig. 14.5 contains the area \(\Delta A_\text{w}\) which is not undertaken during braked recoil.

If it was assumed that, with little error, the firing force, \(F_\text{f}(t)\), did not depend on the total braking resistance, \(R_\text{f}(t)\), then it can be considered as an external force. Consequently, its impulse, \(I_\text{f}(t)\), could, with little error, be considered as a function of time. However, this assumption cannot be used for \(F_\text{w}(t)\) and \(A_\text{w}(t)\), and it is necessary to use a method of successive approximations in their calculations.

It follows from equations (14.61) and (14.63) that the total work developed when the weapon is fired can be reduced by increasing the mass of the recoiling parts, \(m_\text{w}\). However, the mass of the recoiling parts considerably affects the total mass of the weapon.

By selecting an appropriate initial condition of recoil it is possible for \(A_\text{w} \approx 0\) and for \(1 - \eta_k + \eta_b = 0\). Thus:

\[(14.66)\]

\[x_\text{RE} = - (1 - \eta_b) / (2 m_\text{w})\]

where \(x_\text{RE} \approx 0\) is the ideal (limit) of initial counter-recoil velocity of the recoiling parts of a weapon fired during counter-recoil. The true initial counter-recoil velocity, \(x_\text{RE}\), should be a little smaller so as to safely end the weapon functional cycle and to catch the recoiling parts, or breech block, in the rear, cocked, position.

The Czech Model 53 and 53/59 30mm AA coupled cannons are examples of automatic weapons fired during barrel counter-recoil. The same principle is applied to automatic weapons with an unlocked breech which is fired at the moment that the cartridge is fed into the chamber: these weapons are normally referred to as having advanced primer ignition. A similar effect is achieved when a floating mount is used with burst-fire weapons and the mount is still moving forwards during counter-recoil when the next shot is fired.

Efficiency of the system of firing the weapon during counter-recoil is given by:

\[(14.67)\]

\[\eta_\text{f} = 1 - (E_\text{KE} / A_\text{w})\]

The kinetic energy, \(E_\text{KE}\), is the maximum value of the recoiling parts determined throughout the weapon functional cycle, and \(A_\text{w}\) is the total work done during firing for the same weapon during free recoil. A theoretical efficiency of 75% can be achieved, but in practice an efficiency of 60% is more usual.

The initial kinetic energy of the projectile is given by:

\[(14.68)\]

\[E_{\text{KE}_0} = \frac{1}{2} m_\text{b} \cdot v^2 - H_{\text{b}} / 2 m_\text{b}\]

The work of the firing force during free recoil can be expressed as multiples of \(E_{\text{KE}_0}\):

\[(14.69)\]

\[A_\text{w} = \delta_\text{w} \cdot E_{\text{KE}_0}\]

Where the proportionality coefficient, \(\delta_\text{w}\), is given by:
If the difference in energy, \( \Delta A_p \), is negative, it is necessary to increase the firing impulse, \( I_p \), by means of a recoil intensifier. If this cannot be achieved, it will be necessary to add some work, using an external drive, or to opt for a gas-operated system. Weapons smaller than 25mm calibre often have insufficient energy to drive a recoil-operated system.\(^{13}\)

Recoil intensifiers of the active type (Fig. 6.15) increase the impulse acting on the barrel. They do not affect the total impulse acting on the weapon because the increase in impulse is caused by an additional force applied by the pressure of the propellant gases, which is an internal force of the system.\(^{14}\)

Recoil intensifiers of the reactive type (Fig. 6.16) increase both the impulse acting on the barrel and the weapon as a whole.

**FIRING IMPULSE AND FREE RECOIL OF RECOILING PARTS**

An accurate knowledge of the total firing impulse, \( I_p \), is more important than an accurate knowledge of the firing force, \( F(t) \), and its maximum value, \( F_{max} \). This is because most modern weapon mounts are made to be as light as possible and so have a low resonance frequency, in the range 10–30Hz, which is also the resonance frequency of the human body.

If the resonance frequency of a weapon system, \( f_0 \), has the following value:

\[
(14.70) \quad f_0 = \frac{0.1 \text{ to } 0.3}{t_0}
\]

where \( t_0 \) is the time of movement of the projectile in the barrel. The maximum movement of the weapon system (including the human body), and thus the applied stress, is proportional to the firing impulse, \( I_p \), and is little affected by the firing force, \( F(t) \). For automatic weapons where \( t = 0.5 \) to 3ms then:

\[
(14.71) \quad f_0 = 30 \text{ to } 600Hz
\]

When firing a burst, the situation is more complex. However, it can be shown, even for this case, that the load applied to the system is proportional to an average value of the total braking force, \( \bar{R} \), and the rate of fire of the weapon, \( R(\text{ff}) \):

\[
(14.72) \quad \bar{R} = \frac{I_p}{t_r} = I_p \cdot R(\text{ff}) / 60
\]

and not to the maximum value of the firing force, \( F_{max} \).

The value of the total firing impulse, \( I_p \), can be found either by measurement or calculation. To determine \( I_p \) experimentally it is important to use equipment that provides the closest possible conditions for free recoil. The equipment

\[
(14.69) \quad \delta_{AV} = \frac{m_0}{m_p} \delta_0^2
\]

\( \delta_0 \) is given in equations (14.41) and (14.43).

For the usual values of \( \delta_0, m_0 \), and \( m_p \), the value of work for the total firing force normally reaches 1–5% of the projectile kinetic energy.

Knowing the length of the cartridge of a given weapon and the analysis of the dynamics of the weapon functional cycle for a given rate of fire, it is possible to determine the total energy, \( E_p \), that is required to drive the automatic system for one functional cycle. Comparing the work of the firing force, \( A_p \), to the energy required, \( E_p \), it is possible to see if there is sufficient energy to drive the automatic system. If the difference in work and energy, \( A_p - E_p \), is positive, the automatic system can be driven by barrel recoil. The unused energy, \( \Delta A_p \), can be removed or reduced by using a muzzle brake or firing the weapon during barrel counter-recoil.
includes the ballistic pendulum and ballistic carriage. The impulse can be obtained using stress and force measurements during firing but this leads to considerable errors in determining \( I \).

The ballistic pendulum was first used in 1743 by B Robins in England. He used it for measuring the muzzle velocities of rifle projectiles. The ballistic carriage was first used in the USA by Rodman in 1861 and then by H Sebert of France. As well as measuring the firing impulse, it is necessary to measure the muzzle velocity of the projectile, \( v_0 \).

The procedure for the measurement for a single type of cartridge is as follows:

1. Measure the weapon firing impulse without any barrel gas system and without any gas leakage past the projectile, \( I_0 \). Measure the projectile muzzle velocity, \( v_0 \).
2. Measure the weapon firing impulse with any barrel gas system and with any gas leakage past the projectile, \( I \). Measure the projectile muzzle velocity, \( v_0 \).
3. Using equations (14.40) and (14.43) determine:
   \[ \beta = (m_v/m_\text{mag}) \left[(1/\gamma)/(m_v, v_0) - 1\right] \]
   \[ \beta' = (\alpha, \beta) = (m_v/m_\text{mag}) \left[(1/\gamma, v') - 1\right] \]
   and thus \( \alpha = \beta' / \beta \).

For recoilless weapons it is only necessary to determine \( \beta' \). A series of empirical equations has been developed in the past through statistical evaluation of experimental results. For weapons with a standard barrel and with a thermodynamic efficiency greater than 30% (small arms, less efficient cannons and howitzers), the equation of the Schneider Co. can be used:

\[
\beta = 1300/v_0
\]

However, empirical equations do not allow analysis of the physical properties which affect \( \alpha \) and \( \beta \) and thus the value of \( I_0 \). Thus, theoretically-derived relationships have a greater value.

Hugoniot of France in 1886 theoretically derived and published equations for \( \beta \). Rateau of France in 1919 improved the equations, but the improvements are small and usually neglected. Hugoniot's solution is based on the flow of a gas from a large vessel through a small orifice, in which the gas in the vessel has zero velocity and a total (stagnation) pressure of \( p_v(t) \).

A form of the equation used by Soviet authors is:

\[
b = \phi' \frac{a_\text{mag}}{v_0}
\]

The relative muzzle velocity of the projectile, \( v_{u} \), is found from internal ballistic calculations. Also, \( v_{u} = v + w_{u} \). The function for the isentropic exponent, \( \phi' \gamma \), is found from:

\[
\phi' \gamma = \frac{2}{\gamma} \sqrt{\frac{2}{\gamma + 1}}
\]

Thermal losses and the influence of the co-volume of gases are included in the calculations by using an effective value of the isentropic exponent, the polytropic exponent, denoted as \( n \). For most propellant gases the value of \( n \) is between 1.25 and 1.30. The velocity of sound in the gas, \( a_\text{mag} \), is the value for the gases inside the large vessel (stagnation conditions) and at the moment when gas begins to flow in the barrel (t = \( t' \)).

When using equation (14.74) it is not clear how to calculate the velocity of sound because for the muzzle of a barrel there is little similarity between that and a small orifice in a large vessel for which the original analysis was carried out. To achieve calculated results which agree with those obtained experimentally various methods of calculating the speed of sound have been put forward. One example is that of Prof. V E Sluchcokij:

\[
a_\text{mag} = \sqrt{\frac{\gamma}{T}} \frac{p_v}{\bar{G}_u}
\]

Another example is that of Prof. B V Orlivov:

\[
a_\text{mag} = \sqrt{\gamma} \cdot e_\text{p} (1 - \eta_\text{mag})
\]

The pressure, \( p_v \), is the internal ballistic (mean) pressure at the instant \( t = t' \) and \( \bar{G}_u \) is the average density of the gas in the barrel at the same time, which can be calculated from:

\[
\bar{G}_u = m_{\text{mag}} / V_u
\]

where \( V_u = V_0 + S \Delta_0 \) (total volume of the barrel bore).

The specific energy of the propellant, \( e_\text{p} \), is given by:

\[
e_\text{p} = c_\text{T} \theta_p
\]

The thermodynamic efficiency, \( \eta_\text{mag} \), of the internal ballistic system is calculated from the temperature of the gas \( T_0 \) at \( t' \) and the adiabatic flame temperature for the propellant, \( T_v \), and is given by:

\[
\eta_\text{mag} = 1 - T_0 / T_v
\]

The equation for \( \beta \) can also be written as:

\[
\beta = (2 / \gamma + 1) \left( v_{\text{mag}}^2 / v_{u} \right)
\]
and hence the value of $v^v_{eu}$ at instant $t'_u$:

$$v^v_{eu} = \frac{1}{2} (y + 1) \phi (y) \cdot a_{vo}$$

Equation (14.81)

Additionally, the following relationship for the instantaneous relative pressure, $\Pi$, inside the large vessel results from the Hugoniot theory:

$$\Pi(t) = p_o(t)/p_{vo}(t'_u) = 1/[1 + B (t - t'_u)^{\lambda}]$$

Where:

$$A = 2 \gamma/\gamma - 1$$

Equation (14.83)

$$B = [(\gamma - 1)/2] [m^0(t'_u)/m_{vo}]$$

and

$$m^0(t'_u) = G_{vo} \cdot S \cdot v_u$$

Equation (14.84)

The value of the density of the gas in the large vessel at the instant $t'_u$ is denoted by $G_{vo}$. To estimate the value of $\Pi$, the value of $G_{vo}$ is used, calculated from equation (14.77).

Equation (14.82) is often used for approximation of the true variation in gas pressure acting on the base of the barrel during the period of gas flow from the barrel (third period). It is assumed that:

$$\Pi(t) = p_o(t)/p_{vo}(t'_u) - \Pi_{vo}(t)$$

Suppose that $\delta(t = t'_u) = \text{constant} = \delta_{vo} = \delta(t'_u)$, then using equation (14.44) and (14.51) the following is valid for a standard barrel:

$$I^0_{st} = \int_{t'_u}^{t_c} \int_{t'_u}^{t_c} \int_{t'_u}^{t_c} \Pi(t) dt - F_{vo} \cdot t_o$$

$$= \left[ (\beta - \alpha_{vo}/2) m_{vo} \cdot v_0 \right]$$

$$t_0 = \int_{t'_u}^{t_c} \int_{t'_u}^{t_c} \Pi(t) dt \frac{1}{F_{vo}}$$

Equation (14.87)

The value of the force $F_{vo}$ at $t'_u$ is given by:

$$F_{vo} = \delta_{vo} \cdot S \cdot p_{vo}$$

Equation (14.88)

and $t_o$ is the time constant for the gases flowing from the barrel during the third period.

Equation (14.89)

Equation (14.88) makes it possible to determine the value of the approximation constant $B$ on the basis of the measurement or calculation of $\beta$ by determining the value of $m^0(t'_u)$ and $G_{vo}$ at a value of $\gamma$ selected in advance. 28

$$t_o \int_{t'_u}^{t_c} \Pi(t) dt = 1/B (A + 1)$$

Equation (14.90)

Where $B = 1/(A + 1)$ $t_o$

Equation (14.82) is only an approximate fit to the pressure/time relationship. Also, the relationships for $w$ and $w'$ that were derived from it are not very suitable for numerical calculations. Therefore simpler, but less accurate, relationships for the specific pressure, $\Pi(t)$, are used for the approximation of the pressure, $p_{vo}$, during the third period of the action of the firing forces. An exponential approximation by Admiral Prof. E L Bravin is given by:

$$\Pi(t) = p_o(t)/p_{vo}(t'_u) = \exp[-(t - t'_u)/\tau_{vo}]$$

Equation (14.92)

A parabolic approximation by Prof. AN Kupriyanov is given by:

$$\Pi(t) = p_o(t)/p_{vo}(t'_u) = (1 - (t - t'_u)/\tau_{vo})^n$$

Equation (14.93)

The value of $n$ is usually between 3 and 4. For $n = 1$ then the approximation is linear, as was first used by F Vallier of France in the 19th century.

The approximation constant $t_{vo}$ or duration $t_o$ of the third period for an approximation of $\Pi(t)$ is determined by analogy to the equation (14.90). For the exponential approximation:

$$t_{vo} = t_o/(1 - \Pi_{vo}) - t_o$$

where:

$$t_{vo} = -t_o \cdot \ln \Pi_{vo}$$

and

$$\Pi_{vo} = 0.18 \times 10^6 \cdot p_{vo}/p_{vo}(t'_u)$$

For the parabolic approximation:

$$t_{vo} = (n + 1) \cdot t_o$$

The variation in the following forces with respect to time for any of the above approximations of relative pressure can be expressed as:

$$F_{vo} = \delta_{vo} \cdot S \cdot F_{vo} \cdot \Pi(t)$$

Equation (14.96)
Military Small Arms

\[ F_R^t (t \geq t_u') = F_{R0} \cdot \Pi(t) \]
\[ F_{e}^t (t \geq t_u') = F_{e0} \cdot \Pi(t) \]

where

\[ F_{R0} = F_{R} (t'_u) - m^2 (t'_u) \cdot v_{e0}^{2} \]

where \( m^2 (t'_u) \) is determined from equation (14.85) and \( v_{e0}^{2} \) from equation (14.81)

Finally,

\[ F_{e0} = F_{e} (t'_u) = F_{e0} - F_{Da} < 0 \]

When a barrel gas arrangement is used the forces acting during \( t \geq t_u' \) are found using equation (14.51).

Also,

\[ F_{R} - F_{e} + |P_e^t (t \geq t_u')| \]

Using the approximations given above in equations (14.82), (14.92) or (14.93) and equations (14.96) and (14.101) and by integrating the equation of motion for free recoil (14.10) it is possible to estimate the change in the velocities \( w (t = t_u') \) and displacement \( w (t \geq t_u') \) for free recoil in the third period of the action of the firing force with dependence on the initial conditions \( w_u = w (t_u') \) and \( v_u = v (t_u') \).

An analytical relationship for \( w \) and \( v \) can be obtained for the simplest approximation for specific pressure, \( \Pi (t) \), using Valler's linear approximation:

\[ w (t \geq t_u') = w_u + v_u \left[ \frac{t - t_u'}{2t_R} \right]^2 \]
\[ w (t \geq t_u') = w_u + \left( \frac{t - t_u'}{2t_R} \right) v_u + \frac{1}{2} \left( \frac{t - t_u'}{2t_R} \right)^2 \]
\[ w (t \geq t_u') = w_u + \left( \frac{t - t_u'}{2t_R} \right) v_u + \frac{1}{2} \left( \frac{t - t_u'}{2t_R} \right)^2 \]
\[ w (t \geq t_u') = w_u + \left( \frac{t - t_u'}{2t_R} \right) v_u + \frac{1}{2} \left( \frac{t - t_u'}{2t_R} \right)^2 \]

where \( a_0 = \alpha \cdot \frac{v}{x} \) is the acceleration of free recoil at instant \( t_u' \) (just after the projectile has left the barrel) and \( v_u = F_{R0} / m_{p} \) is the value of \( a_0 \) for a weapon with a standard barrel.

The values of \( w (t) \) and \( w(t) \) at different times throughout the first and second periods of the firing force \( F_e \) acting on the weapon can be obtained by integrating equation (14.16). Details of \( x(t) \) and \( \beta(t) \) can be obtained by measurement or from internal ballistic calculations. However, another method may be used, which will be introduced in the following sections.

For the time when the firing forces no longer act, \( t \geq t_u' \) the equation of motion (14.10) can be integrated for \( F_e = 0 \). Thus:

\[ w(t \geq t_u') = w_u + \frac{1}{2} \cdot m_{p} \cdot \beta(t) \]

and

\[ w(t \geq t_u') = w_u + \frac{1}{2} \cdot m_{p} \cdot \beta(t) \]

where \( w_u = w (t = t_u') \) can be found from equation (14.103). Remember that \( (t'_u - t'_u') = t_R \).

To calculate the variation in velocity, \( w \), and displacement, \( w \), for free recoil during the time that the projectile is traversing down the barrel (the second period), it is necessary to apply a special procedure. The firing force, \( F_e \), is not used but the original equation of motion (14.19) is solved, providing that the kinematic dependence is known (Fig. 14.11):

\[ x(t) = \frac{1}{2} \cdot m_{p} \cdot \beta(t) \]

The kinematic dependence can be expressed in the form:

\[ x(t) = \frac{1}{2} \cdot m_{p} \cdot \beta(t) \]

or by:

\[ x(t) = \frac{1}{2} \cdot m_{p} \cdot \beta(t) \]

thus:

\[ \beta(t) = \frac{1}{2} \cdot m_{p} \cdot \beta(t) \]

Differentiating with respect to time gives the following relationship:

\[ \dot{x}_u = \frac{1}{2} \left( \alpha_u + \dot{v}_u + \alpha_u \cdot \frac{v}{x} \right) \]

\[ \dot{x}_u = \frac{1}{2} \left( \alpha_u + \dot{v}_u + \alpha_u \cdot \frac{v}{x} \right) \]

\[ \dot{x}_u = \frac{1}{2} \left( \alpha_u + \dot{v}_u + \alpha_u \cdot \frac{v}{x} \right) \]

\[ \dot{x}_u = \frac{1}{2} \left( \alpha_u + \dot{v}_u + \alpha_u \cdot \frac{v}{x} \right) \]
where \( x_0 = x_y(0) + l \) and is the distance from the base or the driving band of the projectile to the breech face.

\[ x_0 = x_y(0) + l \]

and is the centre of mass of the cartridge elements from the breech face. \( v_x = \dot{x}_0 \) and is the relative velocity of the centre of mass of the inner cartridge elements with respect to the barrel.

The coefficients \( \alpha \) and \( \alpha_\nu \) are functions of time and/or other parameters. The variation in the coefficients with respect to time is estimated using a simplified model for the motion of the inner cartridge elements in the space behind the projectile. The coefficient \( \alpha \) is found by equating the static moments of the inner cartridge elements from the base of the cartridge chamber:

(14.111a) \[ s_y(t) = m_e(t) \cdot x_y(t) = \int \int G_e(t, x) \, dV = \frac{\alpha}{2} \cdot m_e(t) \cdot x_y \]

\( G_e(t,x) \) is the instantaneous mean density of the 'mixture' at a given point, \( x \), and \( V(t) \) is the instantaneous volume occupied by the inner cartridge elements.

The coefficient \( \alpha \) is determined by equating the momentum of the inner cartridge elements, providing that \( m = 0 \) (i.e. that \( m_e(t) = m_{\text{env}} = \text{constant} \)):

\[ \Delta H_x(t) = m_e(t) \cdot (v_x - v) = \int \int G_e(t, x) \cdot v_e(t, x) \, dV - m_e(t) \cdot \dot{x}_0 \]

(14.111b)

\[ = \left( \frac{\alpha}{2} \cdot v - \dot{x}_0 \right) m_e(t) \]

\( v_e(t,x) \) is the instantaneous relative velocity of the 'mixture' at a given point, \( x \), to the barrel.

The procedure, based on equation (14.107), is more general, because it does not require the condition that \( m_e(t) = \text{constant} \). The principle is evident from the definition of force \( F \), in equation (14.7).

Lagrangian attempted to find the dependence of the relationship given in equation (14.108) in 1790-93. He was not satisfied with the results that he achieved and did not publish them, although they were published by S D Poisson in 1832 after his death. In the 19th century G Robert, St Robert and H Segert followed up the work of Lagrange. In the 20th century additional results were published by J Corner and M E Serebrjakov.

The coefficient \( \alpha \) is explained as follows. Consider a cylindrical barrel bore as shown in Fig 14.6. After the propellant charge has burnt to completion the density of the gases within the whole volume behind the projectile is constant.

In the simple barrel shown in Fig 14.6a the centre of the mass of the cartridge elements, \( C_y \), lies half way between the breech face and the base of the projectile, thus:

(14.112) \[ x_c = x_y/2 \]

Thus \( \alpha = 1 \). The distance of the projectile centre of mass, \( c_y \), from the breech face is \( x_c = x_y + \Delta l_y \).

For a bottle-shaped cartridge chamber (Fig. 14.6b) it is evident that \( x_c < x_y/2 \) so that \( \alpha = 2 x_c / x_y < 1 \). Also, the coefficient \( \alpha \) used in this solution includes the non-uniform distribution of mass of the inner cartridge elements caused by unburnt particles, expansion waves and shock waves.

A simplifying assumption made by most authors is that during the second period (movement of the projectile within the barrel) \( \alpha_m - \alpha_\nu = 1 \). For sub-calibre projectiles this assumption may cause large errors.

The relationship for the velocity and displacement for free recoil are derived for the simplest case of a weapon with a standard barrel. From the equation of motion (14.9) we get (m = 0):

(14.113) \[ m_c \cdot \dot{w} = m_0 \cdot v + m_{\text{env}} \cdot v_x \]

Integrating equation (14.113) gives the law of conservation of system momentum:

(14.114) \[ m_c \cdot w = m_0 \cdot v + m_{\text{env}} \cdot v_x \]

Figure 14.6 The relative position for the centre of mass for the projectile and inner cartridge elements for (a) cylindrical barrel bore and (b) extended cartridge chamber.
and the law of conservation of system centre mass:

\[(14.115)\quad m_0 \cdot \omega_0 = m_0 \cdot l + m_e \cdot (x_e - x_e(0))\]

Substituting equations (14.107) and (14.109) into equations (14.114) and (14.115) gives: \[w_0 = k_1 \cdot v - k_1 \cdot v\]

\[(14.116)\]

\[(14.117)\]

where:

\[(14.118)\]

\[(14.119)\]

\[(14.120)\]

\[(14.121)\]

The values \(w_0^0\) and \(w_0\) are obtained by substituting \(v = v_0\) and \(l = l_0\) into equations (14.116) and (14.117), where \(v_0\) is the projectile velocity and \(l_0\) the displacement of the projectile in the barrel. To analyse the behaviour of a weapon when firing, equations are derived for the rotation of the free weapon around the centre of mass, which is analogous to that of free recoil. Fig 14.7 shows the forces acting on the weapon when a shot is fired.

Rotation of the weapon is caused by the moment of the dynamic couple \(M_0\) = \(F_0\). The equation of motion includes the dynamic couple, \(M_0\), the angular acceleration, \(\omega\), and the moment of inertia of the weapon perpendicular to weapon rotation and passing through the centre of mass of the weapon, \(J_w\). The equation of motion takes the form:

\[(14.122)\]

\(J_w \cdot \omega = M_0\)

or in terms of the weapon mass, \(m_w\), and radius of gyration of the weapon about its centre of mass, \(l_w\):

\[(14.123)\quad J_w = m_w l_w^2\]

After integration:

\[(14.124)\quad J_w (\omega - \omega_0) = \int_{t_0}^{t} F_0 \cdot dt = l \cdot l_0 (t) = l \cdot m_w \cdot \omega (t)\]

\[(14.125)\quad J_w [\phi - \phi_0 - (t - t_0) \omega_0] = \int_{t_0}^{t} t \cdot dt = l \cdot m_w \cdot \omega (t)\]

\(\phi_0 = \phi (t_0)\) and \(\omega_0 = \omega (t_0)\) are the initial conditions of rotary movement of the weapon. This gives the effect of the previous shots fired in a burst on the weapon behaviour or upon the firer.

After rearrangement the final expressions are:

\[(14.125a)\]

\(\omega - \omega_0 = \left(\frac{1}{m_w} \right) \cdot \omega (t)\)

\[(14.126)\]

\(\phi - \phi_0 - (t - t_0) \omega_0 = \left(\frac{1}{L_w} \right) \cdot \omega (t)\)

Substituting the corresponding values for velocity and displacement for free recoil at the instant that the projectile leaves the barrel at \(t'_e\), which is \(w_u\) and \(w_u\), an angle of rotation of the weapon, \(\phi_w\), is obtained from equation (14.125a) and (14.126). This is a component of the angle of jump. The
angular velocity of rotation, \( w_{\text{src}} \), is also obtained, which is a dynamic component of the angle of jump.\(^{27}\)

It follows from equations (14.125a) and (14.126) that the angle of jump can be reduced by minimising the arm of the dynamic couple, \( e_c \), and by increasing the radius of gyration of the weapon, \( I_c \). Reducing the firing effect on the weapon can also be achieved by reducing the velocity and displacement of free recoil by reducing the firing force, \( F_c \), and its impulse, \( I_c \). It is the specific impulse of firing\(^{27}\) that is the basic property affecting the weapon, which is given by:

\[
I_c = \frac{l_c}{G_b}
\]

Where \( I_c = I_0 + H_0 \)

\[
(14.127)
\]

\[
= \frac{l_c}{m_b} \cdot x_c = m_w \cdot w_c + x_c
\]

\[
G_b = m_b \cdot g \quad \text{(where} \quad m_b \text{is the combat weapon weight)}
\]

For hand-held weapons the value used for the final velocity of free recoil is usually given by:

\[
(14.129)
\]

\[
\frac{w_{\text{src}}}{I_c} = \frac{l_c}{m_b}
\]

The design values for \( I_c \) and \( w_{\text{src}} \) will depend on the many factors associated with the weapon, especially the rate of fire and length of burst. A reduction in the firing specific impulse can be achieved by:

- increasing weapon combat weight (this will affect weapon mobility)
- reducing the impulse \( I_c \) by firing during counter-recoil or by reducing the final free recoil velocity.

The final free recoil velocity can be reduced by:

- increasing the mass of the recoiling parts (this usually increases the combat mass of the weapon)
- reducing the projectile mass (this usually reduces weapon effectiveness)
- reducing the muzzle velocity of the weapon (this would also reduce weapon effectiveness)
- reducing the propellant charge weight (to achieve the same muzzle energy a higher specific energy propellant would be required which usually increases barrel wear)
- using a muzzle brake.

A detailed analysis of the establishment of weapon force loading when firing, and of the use of braking recuperators, recoil and counter-recoil brakes and hydropneumatic buffers for controlling weapon force loading can be obtained by consulting other publications.\(^{28}\)

**MUZZLE BRAKES AND MUZZLE DEFLECTORS**

Muzzle brakes and muzzle deflectors use a system of nozzles mounted on the barrel. They use the gases flowing from the muzzle to reduce the firing force, \( F_c \). They can only act during the third period of outflow of the gases from the barrel. Fig 14.8 shows the diagrammatic representation of a muzzle brake and a muzzle deflector.

As soon as the projectile leaves the barrel it is followed by the propellant gases. A barrel without a muzzle arrangement is affected by the firing force, \( F^{0}_c \), that has a component of reactive force from the outflowing gases of \( F^{0}_g = m_w \cdot v_{E0} = m_c \cdot v_c \). The effective velocity of the propellant gases, \( v_{E0} \), is proportional to the muzzle gas pressure. There are few possibilities for changing the velocity by altering the shape and size of the exit from the muzzle, and the effective outflow velocity is little changed by the addition of a muzzle brake or muzzle deflector. A muzzle brake or muzzle deflector divide the flow of gases into several directions, typically three, as shown in Fig. 14.8. It follows from the equation of continuity that:

\[
(14.130) \quad \frac{m_c}{m} = \frac{m_1}{m_2} + \frac{m_3}{m_2}
\]

Multiplying this relationship by the effective outflow velocity gives the basic equation for the composition of the reactive forces:\(^{29}\)

\[
(14.131) \quad \frac{F^{0}_0}{F^{0}_r} = \frac{F^{0}_1}{F^{0}_r} + \frac{F^{0}_2}{F^{0}_r} + \frac{F^{0}_3}{F^{0}_r}
\]

because:

| \( F^{0}_0 \) | = \( m_c \cdot v_{E0} \)
| \( F^{0}_1 \) | = \( m_1 \cdot v_{E1} \)
| \( F^{0}_2 \) | = \( m_2 \cdot v_{E2} \)
| \( F^{0}_3 \) | = \( m_3 \cdot v_{E3} \)

The resultant reactive force acting in the direction of the barrel axis can be obtained from equation (14.131):

\[
(14.132) \quad \frac{F^{0}_r}{F^{0}_r} = \frac{F^{0}_1}{F^{0}_r} + \frac{F^{0}_2}{F^{0}_r} + \frac{F^{0}_3}{F^{0}_r}
\]
Adding the force $\frac{F_o}{e}$ to both sides of equation (14.132) gives:

$$F_i = F_p^o - F_{vb}$$

where:

$$F_{vb} = \left| \frac{F_p}{e} \right| - \left| \frac{F_{R2}}{e} \right| - \left| \frac{F_{R3}}{e} \right| - \left( m_2 + m_3 \right) v_{vu}$$

is the nominal (basic) value of the muzzle brake force and:

$$F_i = F_{R1} - \left| \frac{F_p}{e} \right|$$

is the resultant force of firing. There is also a resultant force acting in a direction perpendicular to the barrel axis given by:

$$\left| \frac{F_p}{p} \right| = \left| \frac{F_{R2}}{e} \right| - \left| \frac{F_{R3}}{e} \right| - (m_2 - m_3) \cdot v_{vu}$$

This perpendicular force acts at distance $L_p$ from the centre of gravity of the weapon and produces the turning moment:

$$\left| M \right| = \left| \frac{F_p}{p} \right| \cdot L_p$$

Muzzle brakes are designed so that the force $\frac{F_p}{e}$ is zero, where as muzzle deflectors are designed to produce a turning moment to counter the moment created by the firing force.

The efficiency of a muzzle brake can be increased by directing the gas through angle $\theta$, as shown in Fig 14.8c. From equation (14.131) and the vector addition of forces $\frac{F_{R2}}{e}$, $\frac{F_{R3}}{e}$, and $\frac{F_{E3}}{e}$, the following relationship for a muzzle brake is found:

$$F_{vb} = \left| \frac{F_{R2}}{e} \right| = \left( \left| \frac{F_{R2}}{e} \right| + \left| \frac{F_{R3}}{e} \right| \right) - \left( \left| \frac{F_{R2}}{e} \right| + \left| \frac{F_{R3}}{e} \right| \right) \cos \theta$$

and thus:

$$F_{vb} = (1 - \cos \theta) F_{vb}$$

where $F_{vb}$ is the nominal (basic) value of the force applied by the muzzle brake. Thus the firing force is given by:

$$F_i = F_{R1}^o - F_{vb}$$

The impulse force characteristic for a muzzle brake or muzzle deflector, which is the special case of the impulse force characteristic for the barrel gas system given by equation (14.50), is given by:

$$\chi(t) = \frac{F_i(t)}{F_p^o(t)} = 1 - \frac{F_{vb}(t)}{F_p^o(t)}$$

where $F_{vb}(t) = F_p^o(t) - F_{vb}(t)$ is the resultant reactive force in the direction of the barrel axis, equation (14.10).
For practical calculations the average values of $\chi$ and $\alpha$ for the third period of the firing force are used. \(^{14}\) 

\[ \chi = \frac{1}{R_0} = \frac{\alpha \beta - 0.5 \alpha \alpha}{\beta - 0.5 \alpha \alpha} \]  

de (14.140) 

\[ \alpha = \frac{1}{R_0} \left[ \frac{\alpha \beta - 0.5 \alpha \alpha}{\beta - 0.5 \alpha \alpha} \right] \]  

de (14.141) 

hence: 

\[ \alpha = \frac{1}{\beta} \left[ 0.5 \alpha \alpha + \chi (\beta - 0.5 \alpha \alpha) \right] \]  

de (14.142) 

where: $\alpha \alpha = 1$.

The efficiency of a muzzle brake or deflector, $\eta_{mbr}$, is defined by the work of the firing force during free recoil, $A_{mbr}$. Thus it can be measured on a ballistic pendulum or ballistic carriage and is independent of the way in which the weapon is braked (equation (14.61)). \(^{14}\) Thus:

\[ \alpha_{mbr} = 1 - \frac{A_{mbr}}{A_{mbr}^{\theta}} = 1 - \frac{m_{mbr}}{m_{mbr}^{\theta} + m_{mbr}} \left( \frac{1}{R_0^{\theta}} \right)^2 \]  

de (14.143) 

The mass of the recoiling parts, $m_{mbr}$, excludes the weight of the muzzle brake, $m_{mbr}^{\theta}$. For equations (14.140) and (14.143) it is valid that ($R_0 - 0$):

\[ \chi = 1 - v_s \left[ \sqrt{\left( 1 - \eta_{mbr} \right) \left( 1 + \frac{m_{mbr}}{m_{mbr}^{\theta}} \right)} - v_s \right] \]  

de (14.144) 

where:

\[ v_s = \frac{P_{mbr}}{R_0} = \frac{\delta}{\delta \theta} \left[ 1 + \frac{\alpha \alpha}{2} \left( \frac{m_{mbr}}{m_{mbr}^{\theta}} \right) \right] \]  

de (14.145) 

The efficiency of a muzzle brake or muzzle deflector increases with the ratio $m_{mbr}/m_{mbr}^{\theta}$ and the muzzle velocity of the weapon, $v_0$, and decreases with the thermal efficiency of the weapon, $\eta_{mbr}$. Thus the efficiency of a muzzle brake with a given design characteristic $\alpha$ is higher for cannons than for howitzers. The efficiency of muzzle brakes and muzzle deflectors is very low for weapons firing pistol ammunition.

Muzzle brakes which vent sideways at 90° usually have an efficiency between 20–30% and the impulse-force characteristic is usually greater than zero. Muzzle brakes which vent sideways at an angle greater than 90° usually have an efficiency of between 50–60% and the impulse-force characteristic is usually negative. The variation in the firing force and its impulse and the firing force work done for free recoil with respect to time are shown in Fig. 14.9 for muzzle brakes of different efficiencies.

The design characteristic $\alpha$ can also be derived for a single chamber muzzle brake as follows: it is known from equation (14.132) that $F_{mbr} = F_{mbr}^{\theta} - F_{mbr}$, thus $F_{mbr} = F_{mbr}^{\theta} - (1 - \cos \theta)$, and $F_{mbr} = F_{mbr}^{\theta} + F_{mbr} \cos \theta$, so that from equation (14.139):

\[ \alpha = \delta + (1 - \delta) \cos \theta \]  

de (14.146) 

where:

\[ \delta = \frac{F_{mbr}}{F_{mbr}^{\theta}} = \frac{m_{mbr} \cdot v_{mbr} - m_{mbr}^{\theta} v_{mbr}^{\theta}}{m_{mbr}^{\theta} \cdot v_{mbr}^{\theta}} - \frac{S_1}{S_1 + C_d (S_2 + S_3)} \]  

de (14.147) 

because $v_{mbr} - v_{mbr}^{\theta} = 1, 2, 3$ and equation (14.130) is valid. Thus for $m_{mbr}$ the following is true:

\[ m_{mbr} = C_d \cdot v_{mbr} \cdot G \cdot S_1 \]  

de (14.148)

For the outflow velocity $v_i$ and for $v_{mbr}$ the following is also true: $v_i = v_{mbr}^{\theta}$, $i = 1, 2, 3$. The cross-sectional area $S_i$ is the flow cross-sectional area of the muzzle brake and $C_d$ is the average discharge coefficient of the side holes (channels) of the muzzle brake measured at right angles to the axis of the channels. $C_d = C_d^{\theta}$ is the average discharge coefficient for the side holes. For chamber type muzzle brakes the value of the discharge coefficient is usually between 0.3 and 0.5 and decreases with the increasing inclination of the side channel, $\theta$. Thus equation (14.145) makes it possible to calculate the design characteristic $\alpha$ of the muzzle brake from its geometric dimensions.

**INCREASE IN THE FIRING IMPULSE OF THE FLASH HIDER**

For small calibre weapons driven by the barrel recoil, it is often necessary to increase the firing impulse to provide sufficient energy to cycle the weapon. When a reactive recoil increaser is used as part of the flash hider the impulse acting on the whole weapon is increased. Fig. 14.10 shows the diagrammatic representation of the flash hider.

An increase in the firing force, and thus the firing impulse, is achieved by increasing the effective outflow velocity of the gases, $v_{mbr}$. Their impulse force characteristic, $\chi$, is greater than 1 and is given by:

\[ \chi = \frac{F_{mbr}}{F_{mbr}^{\theta}} = \frac{F_{mbr} - F_{mbr}^{\theta}}{F_{mbr}^{\theta} - F_{mbr}} = \frac{F_{mbr} / F_{mbr}^{\theta} - 1}{1 - F_{mbr} / F_{mbr}^{\theta}} = \frac{F_{mbr}}{F_{mbr}^{\theta}} = \alpha \]  

de (14.149) 

\[ v_{mbr} = \frac{C_d}{C_d^{\theta}} \]  

de (14.150)
C_d is the discharge coefficient for the nozzle formed by the flash hider and 
C_d_0 is the discharge coefficient for the diffuser opening of the flash hider. The 
discharge coefficient C_d is dependent on the cross-sectional area of the outlet, 
S_v, and the critical area S^*_v = S. The length of the system is chosen so that the 
taper of the supersonic part of the nozzle is less than 10° to 15° to ensure that 
gas separation from the walls does not occur, which would reduce C_d. Table 14.1 
gives values which are approximately valid for θ = 1.2 to 1.3. Using these 
values the design characteristic α can be calculated. 40

<table>
<thead>
<tr>
<th>d/d</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>S/5</td>
<td>1</td>
<td>4</td>
<td>9</td>
<td>16</td>
<td>25</td>
<td>36</td>
</tr>
<tr>
<td>C_d</td>
<td>1.24</td>
<td>1.62</td>
<td>1.72</td>
<td>1.86</td>
<td>1.86</td>
<td>1.89</td>
</tr>
<tr>
<td>α</td>
<td>1.000</td>
<td>1.306</td>
<td>1.387</td>
<td>1.457</td>
<td>1.500</td>
<td>1.547</td>
</tr>
</tbody>
</table>

The variation in the firing force, F(t), its impulse, I(t), and work done 
during free recoil, A(t), for a barrel with a reactive recoil increaser or flash 
hider is shown in Fig. 14.9. Muzzle arrangements usually initiate longitudinal 
barrel vibrations during gas flow through them. Thus the measured recoil 
acceleration is composed of the main component, as previously calculated for 
a rigid barrel, and a superimposed element with a frequency of the resonance 
frequency of the barrel.
FIRING BEHAVIOUR OF SMALL CALIBRE WEAPONS

The firing impulse causes the weapon to move backwards and to rotate, which it does while the projectile is still in the barrel. Thus it will affect the direction in which the bullet leaves the barrel, and so affects weapon accuracy. The angle of jump is calculated for a single round fired and the firing effect is superimposed on the preceding effects to determine the effect of automatic fire.

Effect of Firing a Single Shot

The forces acting when a single shot is fired are shown in Fig. 14.7. The weapon moves to the rear and rotates about its centre of gravity. These movements are resisted by the firer's shoulder and grip on the weapon. Weapon support by the shoulder of the firer is elastic in nature and the force F increases with the duration of the recoil because of the rigidity of the muscles of the shoulder and the body. After recoil, there is counter-recoil, which is controlled by the firer holding the grip of the weapon.

The angle of jump is created by the firing moment F, e. The force F, is dependent upon the weapon ballistics so that to minimise the firing moment it is necessary to minimise the force offset, e. The angle of jump can be affected by the way in which the weapon is held, as well as the shape of the stock and the way in which it is supported. The manner of holding the weapon can affect its movement in two ways:

- Freely holding the weapon so that if the firing force does not pass through the centre gravity the weapon will rotate about the centre of gravity.
- Solid support of the weapon at point B, as shown in Fig. 14.7 so that the weapon rotates about point B.

In practice, the weapon movement will be between these two extremes of movement, but closer to the freely held weapon because when the projectile is moving in the barrel the resistance to the movement of the weapon is small compared to the firing force. The recoil distance during this time is also small (approximately 1mm for an automatic weapon) and the resistance applied by the shoulder for such a short movement is small. It is the angular deflection, θ_u, and the angular velocity, φ_u, at the moment that the projectile leaves the barrel for the two cases above that is important in ascertaining weapon accuracy.

The direction of the bullet as it leaves the muzzle is deflected by the rotation of the weapon. A detailed analysis is given in reference. A simplified expression, using Fig 14.11, is given by:

(14.149) \[ \gamma_u = \delta_u + \phi_u \]
Table 4.2
Angle of jump for two different 7.62mm calibre rifles

<table>
<thead>
<tr>
<th>Angle of jump, rad</th>
<th>Model 58 Rifle</th>
<th>Experimental Rifle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotating about centre of gravity</td>
<td>1.31 0.73</td>
<td>0.61 0</td>
</tr>
<tr>
<td>Rotating about shoulder</td>
<td>1.01 1.15</td>
<td>0.40 0.42</td>
</tr>
</tbody>
</table>

Table 14.2 shows the considerable effect of the change in position of the centre of gravity of the weapon caused by emptying the magazine, which often is more than compensates for the reduction in the total mass of weapon and ammunition.

**Weapon Movement when Firing Bursts**

After the first shot has been fired the weapon recoils rearwards and the muzzle climbs from muzzle jump. During this time the functional cycle is taking place, which results in many accelerating and braking forces acting on the weapon and in changes in the position of the centre of gravity of the recoiling parts. To determine the weapon motion during the functional cycle and the initial conditions for the next functional cycle the force diagram for the weapon is used. The force caused by the propellant gas pressure acts on the barrel axis, but the forces exerted by the springs, case extraction, ejection etc., are more difficult to take into account because they do not always act on the centre of gravity of the different moving components. The dynamic couple that this causes increases the passive resistance to movement of the components and affects the movement of the weapon during firing. Fig. 14.12 shows a typical dynamic couple caused by offset moving masses.

The driving force, $F_d$, in Fig. 14.12 does not act on the centre of gravity of the component and a force couple is created which causes the reactions $N_1$ and $N_2$ with friction forces $f_1N_1$ and $f_2N_2$ that are dependent on the value of the couple $F_d e$. The influence of this couple can be lessened by reducing the offset, $e$, or by increasing the axial distance from the component’s centre of gravity upon which they act.

The point at which the resistance to the recoil acts is taken as the centre of gravity of the weapon and the shoulder reaction force is taken to act one-third or one-half of the way up the butt plate. The point of action of the force exerted on the grip is taken to be on the upper half of the grip. Shoulder resistance is approximately linear and is given by: 47

**Figure 14.12 Schematic arrangement showing the origin of a dynamic couple caused by the centre of gravity of the recoiling parts being offset to the barrel axis**

(14.154) $R_{1s} = R_{2s} + c_s x$

Shoulder muscle rigidity, $c_s$, is approximately 100N/mm and varies between 75–120N/mm according to the position and the clothes worn by the firer. Variation in the shock absorption with time is highly dependent on the initial force, $F_d$, with which the rifle is pushed into the shoulder. From the rigidity of the shoulder it is possible to draw some important conclusions concerning the resonant frequency in the longitudinal direction of the weapon, $f_r$. For a weapon of between 2-5kg and a rigidity range of 75–120N/mm, the resonant frequency will be between 19–40Hz. Thus it follows that rates of fire in excess of 19 shots per second (1,100 rounds per minute) should be avoided. However, the bending rigidity of a firer who is standing is substantially lower and results in a resonant frequency of approximately 7Hz, which is equivalent to a rate of fire of 400 rounds per minute. The resonant frequency is affected by the position of the firer. This will have different effects on the way that the rate of fire reacts with the body of the firer, and will also be affected by the way in which the weapon is held and by the weapon mass. Weapon movement when firing bursts may be determined from the sum of the movements from successive shots.49

The lower the rate of fire, then the lower is the number of rounds fired before the barrel returns to its original position and again it is important to reduce the offsets $e$ and $f_1$ to minimise weapon rotation. The position of the grip is also critical in overcoming the deflecting forces. The effect of the rate of fire on weapon movement is complex.49
For accurate determination of the effect of rate of the fire on weapon movement it is necessary to take into account the dynamics of the weapon’s automatic system, barrel vibrations, and the way in which the weapon is supported.\textsuperscript{50}

**DAMPING EFFECT OF THE WEAPON MOUNT**

For automatic weapons held in any type of mount it is necessary to know the forces acting on the different parts of the mount during firing. Neglecting the inertial and centrifugal forces exerted by the firer on the mount, the following forces act on the barrel during firing:

- Firing force acting on the barrel, $F_v$.
- Weight, $G$.
- Starting torque of the barrel group for Gatling-type weapons.

When the barrel is fixed relative to the rest of the weapon then all components attaching the weapon to the mount are loaded by the above forces and transmit them to the mount. Weapon weight and starting torque for Gatling guns have a small effect compared to the firing force. The firing force is represented by a vector acting on the centre of gravity of the recoiling parts. Additionally, the dynamic couple, $F_v e$, will be applied to the mount. The variation of the applied force with respect to time depends on the firing force, its damping and the type of automatic system used.

**Gas-operated Weapon with Fixed Barrel**

The forces applied to the mount by this type of weapon are periodic in nature. They consist of the force applied to the gas piston, $F_p$, the buffer force, $F_b$, the return spring force, $F_s$, and the shock impulse of the moving parts hitting the front of the weapon casing, $I_{sh}$. These forces are shown in Fig. 14.13.

The sum of these forces transmitted from the weapon to the mount is called the weapon force-impulse action on the mount and is represented by the force-impulse diagram, as shown in Fig 14.14.

The force-impulse diagram for a weapon of known construction can be determined from the firing force acting on the barrel, $F_v$, and by using the functional diagram of the weapon or force and acceleration diagram. It can also be found experimentally. The detailed construction of a force-impulse diagram is described in references.\textsuperscript{51}

**Gas-operated Weapon with Short Barrel Recoil**

For this example, the firing force does not act directly on the mount but the short time that the force is acting is replaced by a recoil resistance effect which
is longer in duration. A recoil mechanism consists of an elastic element and a damping element. The recoil mechanism is positioned between the recoiling parts and a cradle. Fig 14.15 shows a typical force-impulse diagram.

**Measurement of Recoil Forces applied to the Mount**

It is possible to measure the firing forces that act on the mount for weapons and mounts that have already been built. A combination of displacement transducers and strain gauges is used. An example of how these measurements are made can be illustrated with the 30mm automatic cannon shown in 14.16.

The output from the strain gauges is shown in Fig. 14.17. The output signals were filtered and converted into digital form using an analogue to digital

---

Figure 14.14 Force-impulse diagram for a gas-operated weapon with a fixed barrel

Figure 14.15 Force-impulse diagram for a gas-operated weapon with a recoiling barrel
converter for processing by computer. The variation in firing force over a six-shot burst is shown in Fig. 14.17.

It can be seen that the force transmitted to the mount has a maximum value for the first shot fired. For the second shot there is a reduction in the firing force which is still further reduced for the third shot.

Fig. 14.18 shows the output from the displacement transducer attached to the barrel during the same six-round burst, and Fig. 14.19 is the functional diagram for the weapon. It can be seen that the maximum force applied to the mount occurs at the instant that the barrel is arrested and the breech carrier begins to act on the buffer.

The spectral density variation of the firing force acting on the mount is shown in Fig. 14.20. This shows that the basic frequency of 8.15Hz is given by the rate of fire. Frequencies of 16.5Hz, 23.77Hz and 30.66Hz are higher harmonics. The impulse of the force acting on the mount is the same as the impulse of the firing force, which allows quick and simple calculations of the force applied to the mount to be made.$^2$

**SHOCK ABSORBERS FOR AUTOMATIC WEAPONS**

To reduce the load on the mount, it is common to mount a medium calibre automatic weapon by means of a recoil system, or a small calibre weapon by dampers or shock absorbers. All of these devices reduce the load applied to the mount by inserting a flexible damping element between the weapon and the mount. Shock absorbers are usually situated below the weapon and
symmetrically alongside the weapon. However, if fitted below the weapon a shock absorber can cause high frictional forces and high wear rates in the guide- ways. Symmetrical shock absorbers eliminate this problem, but are a more complex solution.

The requirements of shock absorbers are as follows:

- To ensure that the direction of recoil is along the axis of the barrel by the correct design of the guideways.
- To ensure minimum guideway clearance and maximum guideway length.
- To remove any shocks during recoil and counter-recoil of the weapon.
- To ensure that the minimum force is transmitted to the mount and that there is minimum weapon recoil, and to ensure that the weapon is stable during firing.

The maximum allowable recoil length is dependent upon the cartridge feed system, and must not be greater than that allowed by the flexibility of the cartridge belt (for belt-fed weapons). A short recoil distance simplifies the construction and fitting of the weapon’s aiming system.\(^{33}\)

The basic action of a shock absorber is shown in Fig. 14.21, and is dependent upon the mass of the weapon, \(m_w\), the rigidity of the spring, \(k_s\), and the friction force acting on the guideways, \(F_{fr}\). The force transmitted to the mount is determined from the methods discussed earlier in this chapter.

Figure 14.19 Functional diagram for a 30mm calibre AA cannon

Figure 14.20 Spectral density of forces applied to a 30mm calibre AA cannon mount

The characteristics of a shock absorber can influence the functioning of the automatic system. If the shock absorber used with a gas-operated weapon has a functional time that is shorter than the time for one functional cycle then:

- The rearward breech block velocity increases because of the effect of the initial velocity of the whole weapon up to the beginning of the gas take-off.
- The absolute displacement of the breech block carrier increases because of the whole displacement of the weapon.

Figure 14.21 Diagrammatic representation of a spring shock absorber
- The ratio of the breech block carrier velocity before and after impact varies.
- The functional cycle time may be shorter and therefore the rate of fire may be higher if the breech block impacts the buffer when the weapon is moving foreword.

Shock absorbers used with recoil-operated weapons reduce the rate of fire because of the reduction in the counter-recoil velocity of the barrel and the breech block.\textsuperscript{54}

**Spring Shock Absorbers**

A simplified arrangement of a spring shock absorber is shown in Fig 14.22. The cylinder of the shock absorber is attached to the mount and the central rod to the weapon, which compresses the spring during weapon recoil. However, this simple arrangement does not give shock damping during counter-recoil.

To provide damping during counter-recoil two springs are used as shown in Fig 14.23. One spring is compressed during recoil and the other is compressed during counter-recoil. However, the prestressed value for the twin springs is zero. Thus the weapon may not reach a stable position before subsequent shots are fired.

Twin spring shock absorbers are used on the PKT 7.62mm calibre machine-gun. Fig. 14.24 shows the displacement/time trace for this weapon when firing a six-round burst. At the end of firing the weapon is displaced 2mm to the rear.

Fig. 14.25 shows the damping of counter-recoil movement by using fibre washers. Rubber washers may be used instead, as in the KPVT 14.5mm calibre heavy machine-gun.

A shock absorber, using a fibre cone and a split sleeve to damp out vibrations, is shown in Fig. 14.26. The spring presses onto the cone and the split sleeve, and creates friction between the sleeve and the inner body of a tube.

\begin{equation}
F_i = 2.1F_p \cos \alpha /2
\end{equation}

**Gas Shock Absorbers**

Recoil damping can be applied between the barrel and the weapon casing (usually called a recuperator), or between the weapon and its mount. For both types the propellant gas can be used, which results in a small, lightweight system. A typical arrangement is shown in Fig. 14.27.
Propellant gas is fed from a vent in the barrel into a gas cylinder which is used to brake the recoil of the barrel, or the whole weapon, and to return the recoil system to its forward position. A spring is also fitted to the system, but its main purpose is to hold the recoil system in its forward position when the weapon is not firing, or between shots if the shock absorber is designed to return the recoil system to its forward position. The design may be such that the gas vent is open during the whole cycle of recoil and counter-recoil, or the vent may be closed after only a small initial recoil movement so that the propellant gases in the cylinder are compressed for the remainder of the recoil stroke.

During the first part of the recoil movement, the restoring or braking force is that of friction and the spring of the shock absorber. After the base of the projectile has passed the gas vent in the barrel, the propellant gases will flow into the gas cylinder and exert a force on the gas piston to brake the recoil movement. Gas flow into the cylinder will continue while the pressure in the barrel is greater than the pressure in the cylinder. When the barrel pressure falls below that of the gas cylinder there will be gas flow from the cylinder back into the barrel. A certain amount of gas leakage will also occur from the cylinder because of the clearance between cylinder and piston. The system is returned to its forward position by the force of the gases acting on the piston and by the shock absorber spring. During counter-recoil, the gas pressure is low compared to during recoil so that the counter-recoil velocity is low. Counter-recoil velocity can be increased by closing the gas vent after only a short recoil movement to retain a high gas pressure in the gas cylinder. Fig. 14.28 shows the displacement/time curve throughout the recoil cycle for an open and closed vent.

For low rates of fire the weapon can be returned to its forward position between shots. With high rates of fire there is insufficient time to do this. A floating mount is therefore used in which subsequent shots are fired while the recoil system is still moving forward during counter-recoil. Fig. 14.29 shows the displacement/time curve for an eight-round burst for a floating mount.
If the recoil system returns to the forward position after each shot, it is necessary to damp the vibrations and the impact caused by counter-recoil. This can also be achieved with a gas recoil system by braking the counter-recoil movement immediately before it reaches its fully forward position. To do this, a space in front of the piston is created and part of the propellant gases supplied to it.Compression of this gas creates a high pressure in the front of the piston, which opposes its forward motion. A vent is also provided to release the pressure at the end of counter-recoil so that the gas pressure does not push the piston rearwards. Fig. 14.30 shows a typical arrangement used.

The propellant gas leaking from the shock absorber through the clearance between the piston and the cylinder is used, i.e. through the area $S_{pl} - S_{pr}$. The damper cylinder is connected to the atmosphere by means of a vent of cross-sectional area $S_c$. During the initial phase of recoil the recoil system begins to move and the piston is pushed to the rear. The volume of the cylinder increases and air is drawn in through the vent in the cylinder of cross-sectional area $S_c$. During the second phase the projectile passes the vent in the barrel and propellant gases flow into the shock absorber cylinder. Three different periods can occur during this second phase. 55

- 1st $P_{cy2} < P_{cy1}$, $P_{cy2} < P_{atm}$
- 2nd $P_{cy2} < P_{cy1}$, $P_{cy2} > P_{atm}$
- 3rd $P_{cy2} > P_{cy1}$, $P_{cy2} > P_{atm}$

Gas shock absorbers are analogous to the gas-operating systems described in Chapter 16. The main difference is that in gas shock absorbers the gas piston is used to slow the system, whereas in a weapon operating cycle the gas piston is used to accelerate the system. If a counter-recoil gas damper is used the analysis of the system becomes considerably more complex.

The forces acting on the recoil system are shown in Fig. 14.17. The force exerted by the propellant gas via the shock absorber is calculated from the gas pressure in the cylinder, $P_{cy}$, acting on the piston area, $S_{cy}$, if it is a moving piston, or on the cylinder area, $S_{cy}$, if it is a moving cylinder. Forces acting on the breech, $F_B$, are calculated from the cross-sectional area of the bore, $S$, and the propellant pressure acting on the breech, $p_{D}$:

\[
F_B = S_p D
\]

where $S = (0.805$ to $0.820)d = \text{cross-sectional area of the rifled part of the bore}$; $d = \text{calibre of the weapon}$. From this force must be subtracted the resistance to motion of the bullet, $F_B$. This is calculated from the bullet mass, $m_B$, and bullet acceleration, $a_B$:
An allowance for the weight of the recoil system is only made if the weapon is elevated and will be $G \cdot \sin \alpha$. Friction of the slide ways is calculated from:

\[
F_G = G \cdot f_{gw} \cdot \cos \alpha
\]

If the counter-recoil damper is used the force applied by the gas piston during the counter-recoil must also be included. Chapter 16 gives the gas laws used with a gas cylinder for calculating the forces acting on the gas piston throughout the recoil cycle.

The equation of motion for the recoil system is given by:

\[
m \cdot \frac{d^2x}{dt^2} = F_D - F_R - F_{gw}. \ (\sin \alpha) - F_{ps} - F_{sa} - G \cdot \sin \alpha
\]

Gas shock absorbers are very similar in action to the gas systems used for cycling gas-operated automatic weapons. The analysis for the gas system in gas-operated weapons is covered in Chapter 16, Section 2, and the equations developed there are similar in nature to those for gas shock absorbers. Care should be taken to ensure that the mathematical signs used are correct and correspond with the different direction of gas flow through the gas ports.
15

Accelerator Mechanisms

TRANSMISSION FUNCTIONS AND MECHANICAL EFFICIENCY

Many automatic weapons rely on lever and cam mechanisms to control the relative movement of their components. The shape of the control curve is of crucial importance for the correct operation of the weapon, especially between the main functional element (the driving element) and the secondary element (the driven element). The high velocities and accelerations of the driving element make it necessary to avoid any sudden changes of motion of the driven element. There is an instantaneous ratio of velocity between the driving element, \( v_d \), and the driven element, \( v_k \), which is called the transmission ratio, \( i \), and is expressed as:

\[
(15.1) \quad i = \frac{v_d}{v_k}
\]

The expression of dependence of the change in transmission ratio with respect to the displacement of the driving element is known as the transmission function \( i = i(x) \). The shape of this function describes the quality of the automatic weapon mechanism. For continuous motion of the driven element and to achieve a high rate of fire it is necessary to ensure that the transmission function is continuous and without any sudden changes. The shape of the acceleration and deceleration curves is particularly important. To avoid impacts, high stresses and energy losses in the mechanism, it is necessary to pay particular attention to transitions from an acceleration phase to a deceleration phase or into a phase of uniform motion.

Graphical and analytical methods are used to solve the transmission function. The transmission ratio \( i = \frac{v_k}{v_d} \) is a derivative of the function \( y = y(x) \) with respect to displacement which performs as the controlling profile as shown in Fig. 15.1, so that:

\[
(15.2) \quad i = \frac{v_k}{v_d} = \frac{dy}{dx} = \tan \alpha
\]

The mechanisms used in automatic weapons are usually some form of kinematic couple. Fig. 15.2 shows a typical example used in weapons design. Their analysis is similar to that for the basic type shown in Fig. 15.1.

The element 'A' in Fig. 15.1 acts as the driving element, which may be the barrel or breech carrier: it is operated by a driving force which is dependent upon time, \( F_d(t) \). This driving force is normally the firing force (the force applied by the gas pressure on the cartridge chamber) or it may be the piston force in a gas-operated system. Additional forces, in the form of resistances to movement, may also act on element 'A'. The driven element 'B' (which may be the breech block, feed mechanism slide, etc.) is also acted on by forces which are dependent upon the displacement \( F_k(y) \). The controlling profile which gives the change in transmission ratio with respect to element 'A' is acted upon by either element 'A' or element 'B'. The instantaneous value of the transmission ratio is given by equation (15.2).

Fig. 15.3 shows an example of a control curve which is commonly used. It consists of two circular arcs and one straight section connecting them. The highest value of the transmission ratio can be found from \( \tan \alpha = i \). For the straight section the transmission ratio is constant. A graph of the transmission function can be constructed as shown in Fig. 15.3. It can be seen that the continuous path of the control curve is not always ideal because the sudden changes in the transmission ratio occur during the transition between sections, which causes abrupt changes in the acceleration of the components, leading to non-uniform and excessive wear.
Figure 15.2 Typical mechanisms with variable transmission ratios used in automatic weapons.

To achieve correct evaluation of the control curve and the quality of the transmission function it is not only the continuity and acceptability of the control curve in terms of its first order derivative which is important, but also the second order derivative, with respect to time and displacement, because:

\[ y = f(x(t)) \]

This is illustrated in Fig. 15.4. Curve ‘A’ is the controlling profile of a continuously varying transmission and curve ‘B’ is the transmission function shown in Fig. 15.3. If the control curve is a composite function of the type given in equation (15.3), then its first and second order derivatives with respect to time are:

\[ \frac{dy}{dt} = i \times \frac{dx}{dt} \]

\[ \frac{d^2y}{dt^2} = i \times \frac{d^2x}{dt^2} - \frac{di}{dx} \times \frac{x^2}{x} \]

Thus it is necessary to estimate not only derivatives of the function \( x = x(t) \) with respect to time, but also the derivative with respect to displacement, i.e.
dy/dx = i and its derivative \( \frac{d^2y}{dx^2} = \frac{di}{dx} \) shown in Fig.15.4. The basis of correct design for an automatic weapon is to minimise the acceleration of any given element.

For transition into the deceleration part of the control curve the highest velocity of the driven element is achieved and then deceleration begins. The change in sign of \( di/dx \) (or \( y \)) causes the driven element to interrupt contact with the driving element immediately after transition into the deceleration part of the control curve; then any clearance between the two parts is eliminated and an impact occurs between the two elements. The linkage must therefore be double-sided and the control curve must be in the form of a groove. To reduce the wear at the transition the clearances are reduced to a minimum and a continuous change in the transmission ratio is used.

\[
\frac{i}{x_1 + x_2} \left( 1 - \cos \left( \frac{\pi}{x_1} x \right) \right)
\]

(15.6)

Fig. 15.5 shows examples of the continuously changing transmission function \( i(x) \) (curve no. 2) and the corresponding control curve \( y(x) \) (curve no. 1) which are used for weapons with high rates of fire. This is illustrated by the curve of its second order derivative (curve no. 3).

The transmission function is only created for the accelerating section \( x_1 \) and by the immediately following decelerating section \( x_2 \) and can be expressed as:

- for acceleration \( x < x_1 \)

\[
\frac{y_1}{x_1} \left( 1 - \cos \left( \frac{\pi}{x_1} \right) \right)
\]

\[
\frac{y_2}{x_2} \left( 1 - \cos \left( \frac{\pi}{x_2} \right) \right)
\]

\[
\frac{y_3}{x_3} \left( 1 - \cos \left( \frac{\pi}{x_3} \right) \right)
\]

- for deceleration \( x > x_2 \)

\[
\frac{y_1}{x_1} \left( 1 - \cos \left( \frac{\pi}{x_1} \right) \right)
\]

\[
\frac{y_2}{x_2} \left( 1 - \cos \left( \frac{\pi}{x_2} \right) \right)
\]

\[
\frac{y_3}{x_3} \left( 1 - \cos \left( \frac{\pi}{x_3} \right) \right)
\]

- for acceleration \( x_1 < x < x_2 \)

\[
\frac{y_1}{x_1} \left( 1 - \cos \left( \frac{\pi}{x_1} \right) \right)
\]

\[
\frac{y_2}{x_2} \left( 1 - \cos \left( \frac{\pi}{x_2} \right) \right)
\]
for deceleration $x_2 \geq x \geq x_1$,

$$i = \frac{y_1}{x_1 + x_2} \left[ 1 + \cos \left( x - x_1 - \frac{\pi}{x_2} \right) \right]$$

where $y_1$ is the complete stroke of the driven element.

After integration or differentiation with respect to displacement an analytical expression for the control curve $y(x)$ or the curve for the derivative of the transmission function $i' = f(x)$ is found:

$$x < x_1 : y = \frac{y_1}{x_1 + x_2} \left[ x + \frac{x_1}{\pi} \sin \left( \frac{\pi}{x_1} - x \right) \right]$$

$$x > x_1 : y = \frac{y_1}{x_1 + x_2} \left[ x + \frac{x_2}{\pi} \sin \left( \frac{\pi}{x_2} (x - x_1) \right) \right]$$

$$x < x_1 : i' = \frac{y_1}{x_1 + x_2} \left[ \frac{\pi}{x_1} \sin \left( \frac{\pi}{x_1} - x \right) \right]$$

$$x > x_1 : i' = \frac{y_1}{x_1 + x_2} \left[ -\frac{\pi}{x_2} \sin \left( \frac{\pi}{x_2} (x - x_1) \right) \right]$$

Transmission functions with sudden changes in transmission ratio should be avoided in the design of automatic weapons. A straight control curve, as shown in Fig. 15.6, produces impacts at the transition between phases. Mechanisms with constant transmission ratios are usually only used where they are driven by hand, such as cocking levers.

In every mechanism that moves relative to another there are energy losses caused by friction. The ratio of energy transmitted by a mechanism to the energy supplied by the mechanism is known as its mechanical efficiency, $\eta$. A means of expressing mechanical efficiency is to define the energy input and output in terms of forces needed to be applied to and transmitted by the mechanism. This is best done by means of a resultant of the lost force in the direction of movement of the driving element, based on the dynamic couple shown in Fig. 15.1. This resultant of the lost force is considered to be an internal force, $F_{RB}$, which resists the movement of the driving element, as shown in Fig. 15.7. The force $F_{RB}$ is considered to be an internal force acting on the element $B$. For the ideal case the resultants of the lost forces $F_{RA}$ and $F_{RB}$ would be equal so that:

$$F_{RA} = F_{RB} \text{ so that } F_{RA} = i, F_{RB}$$

However, there are energy losses so that:

$$F_{RA} = F_{RB}(i/\eta)$$

where the efficiency is defined as:

$$\eta = i (F_{RB}/F_{RA})$$

To find the resultant of the lost force in the direction of motion of both elements it is necessary to separate the elements and to develop equations from Fig. 15.7. The equations of equilibrium for the element 'A' are given as:

$$\Sigma x : R.f.\cos \alpha + R.\sin \alpha + F_{NA}.f - F_{RA} = 0$$

$$\Sigma y : R.f.\cos \alpha - R.\sin \alpha + F_{NA} = 0$$

It is then possible to express $F_{RA}$ in terms of these equations:

$$F_{RA} = R(2.f.\cos \alpha + \sin \alpha - f \cdot \sin \alpha)$$

The value of $f \cdot \sin \alpha$ is small and is usually neglected so that:

$$F_{RA} = R(2.f.\cos \alpha + \sin \alpha)$$

The equations of equilibrium of the element $B$ are given as:

$$\Sigma x : F_{NB} - R.f.\cos \alpha - R.\sin \alpha = 0$$

$$\Sigma y : R.\cos \alpha - R.f.\sin \alpha - F_{NB} + F_{RB} = 0$$

By analogy:

$$F_{RB} = R(\cos \alpha - 2.f.\sin \alpha)$$

Substituting the values of $F_{RA}$ and $F_{RB}$ into equation (15.14) gives the relationship for efficiency:
The turning moment \( F_{\text{RA}} \cdot r_B \) is that of the lost forces resultant which achieves equilibrium for element B.

For other arrangements, or for multi-element mechanisms, it is possible to use the above approach. The force \( F_{\text{RA}} \) is used acting in the direction of movement of the main element and for the nth element a force \( F_{RB} \) is used in the direction of the nth element of such a magnitude that it balances the mechanism. Thereafter it is necessary to determine all of the internal forces and express the conditions of equilibrium for all of the elements. This procedure allows the forces \( F_{\text{RA}}, F_{\text{RB}} \) to be determined. Fig. 15.8 shows a multi-element mechanism which has a central accelerating element which pivots through an angle \( \phi \).

The other two elements have linear movements. It should be noted that it is necessary to take into account the inertia of the pivoting element. When denoting the transmission ratio and efficiency from the driving element to the pivoting element, \( (l_1, \eta_1) \), and from the pivoting element to the driven element, \( (l_2, \eta_3) \), and when expressing the equilibrium conditions for all of the elements of the mechanism (the resultant of the lost forces for the pivoting element is found from the equilibrium of its moment), then the following can be written:

\[
\eta_1 = \frac{l_2 - f \cdot l_1}{(\sin \alpha + 2 \cdot f \cdot \cos \alpha) l_1} \cdot \eta_1
\]

Figure 15.8 Diagram of an accelerator lever
(15.25) \[ \eta_2 = \frac{(1 - f \cdot l_1)}{(l_2 - f \cdot l_1)} \left( \frac{\sin \phi - 2 f \cdot \cos \phi}{\sin \alpha + 2 f \cdot \cos \alpha} \right) \]

Values for the transmission ratio for \( \eta = 1 \) when \( f = 0 \):

(15.26) \[ i_1 = \frac{1}{l_4} \sin \alpha ; i_2 = \frac{1}{l} \sin \frac{\sin \alpha}{\sin \phi} \]

In short recoil weapons with a pivoting accelerator the accelerator mass can be ignored because it is low compared to the other elements. Thus equation (15.22) can be used: the instantaneous transmission ratio is the lever ratio for the pivoting element. It is necessary to estimate the conditions of equilibrium and the influence of the two contact areas and associated friction as described in section 15.2.

Fig. 15.9 shows how, for different values of friction coefficient, the efficiency rises, reaches a maximum and then falls as the transmission ratio increases. At higher values of transmission ratio the efficiency falls to zero and, according to equation (15.22), can become negative, which is not physically possible. When the efficiency falls to zero the mechanism becomes locked.

By substituting \( f = \tan \phi \) into equation (15.22) (where \( \phi \) is the friction angle) and \( i = \tan \alpha \) (where \( \alpha \) is the mean angle between the tangent and the controlling curve), then the point at which locking occurs can be found:

(15.27) \[ \frac{1}{\tan \alpha + 2 \tan \phi}, \tan \alpha = 0 \]

There are two solutions:

1. \( \tan \alpha = -1 = 0 \) and therefore \( \alpha = 0 \)
2. \( 1 - 2 \tan \phi \tan \alpha = 0 \)

Locking occurs when \( \alpha \) does not equal 0, so that:

(15.28) \[ \tan \alpha = \frac{1}{2 \tan \phi} \]

Thus the movement of the mechanism can occur when:

(15.29) \[ \tan \alpha < \frac{1}{2 f} \]

If the small elements of the mechanism are neglected then:

(15.30) \[ \tan \alpha < \frac{1}{2 f} \]

When the coefficient of friction \( f = 0.1 \) then the minimum angle at which locking can occur is \( \alpha = 78.69^\circ \) for equation (15.29) and \( \alpha = 78.58^\circ \) for equation (15.30). The maximum possible transmission ratio is \( i = 5 \). The maximum value of efficiency occurs if the transmission ratio satisfies the condition:

(15.31) \[ \frac{d\eta}{di} = 0 \]

Thus from equation (15.22):

(15.32) \[ \frac{d\eta}{di} = -\frac{2 f \cdot i^2 - 8 f^2 \cdot i + 2 f}{(i + 2 f)^2} = 0 \]

The roots of the quadratic equation are:

(15.33) \[ i_{1,2} = \frac{-8 f^2 \pm \sqrt{D}}{4 f} ; D = 64 f^4 + 16 f^2 \]

for which only the positive root is valid.

The curves in Fig.15.9 show the advantages of reducing the coefficient of friction to a minimum. Mechanisms with high transmission ratios benefit from having rollers to reduce friction losses. Good maintenance is essential to ensure that frictional losses do not increase. The mechanism shown in Fig. 15.1, for example, becomes locked if the coefficient of friction exceeds 0.25.
VARIABLE COEFFICIENT OF FRICTION

So far, the coefficient of friction has been considered to be constant, which is usually the case if the surface velocities do not exceed 20m/s. For weapons with a high rate of fire it is possible for velocities to exceed this and the coefficient of friction will vary with the velocity. There is no theory relating surface velocity to coefficient of friction so that experimental results must be used to determine the relationship.

Consider a linkage with a groove in which the element is guided by a roller or a guide lug. For the driven member the roller will lift off the groove during deceleration, as shown in Fig. 15.10, and will then be guided by the top surface of the groove.

During this phase the driven element will not be taking energy from the driving element but will actually be returning energy. Energy losses during this time phase will be taken into account by means of an inverted value for efficiency relating to the driven element 'B'. After changing from being guided on the bottom of the groove to the top of the groove, the efficiency will be found by considering element 'B' as the driving element. It is necessary to determine the instant in time when both members change from acceleration to deceleration by finding when the inertial braking force overcomes the resistance acting on element 'B'.

To find the efficiency relating to element B the following equation is used:

\[ \eta_B = \frac{F_{RA}}{F_{RB}} \cdot \frac{1}{i} \]  

where \( i = \frac{1}{1 - v_i/v_f} \).

By expressing the conditions of equilibrium (in a similar way to which equation (15.22) was derived), it will be found that only changes in sign has occurred, except for the friction forces (including the resultant of the lost forces) and \( \eta_B \) possesses the following resultant form:

\[ \eta_B = \frac{1 - 2 f}{1 + 2 f} \cdot \frac{1}{i} \]

Equation (15.35) cannot be used in this form, and so the efficiency must be rewritten as:

\[ \eta = \frac{1}{\eta_B} \]

Thus:

\[ \eta = \frac{1 + 2 f \cdot i}{1 - 2 f \cdot i} \]

When \( i > 2f \) then \( \eta > 1 \) and is only referred to as the efficiency for use in the equations of motion, the values \( h_1, h_2 \) being substituted. (\( \eta > 1 \)) indicates that energy is given to the secondary element.

By analysing equation (15.35), it can be shown that \( i = 2f \) is the limiting value for the element B to be the driving element, because when \( \eta_B = 0 \), \( \eta_B = \infty \) and the function \( \eta(i) \) has a discontinuity of the second order at this point, as shown in Fig.15.11

For \( i < 2f \) then \( \eta < 0, \eta_B < 0 \), which is physically not possible. This can be explained by a change in the sign of the force \( F_{RA} = R(i - 2f) \cos \theta \), which prevents element B from being a driving element for small transmission ratios.

The minimum value of \( \eta \) and thus the maximum efficiency for the element B, \( \eta_{max} \), occurs when \( \frac{ds}{di} = 0 \) and by analogy the above root satisfies:

\[ i = \frac{8f^2 + \sqrt{D}}{4f} \]

For the curve no. 1 in Fig.15.11 where \( f = 0.1 \), \( \eta_{min} = 1.49 \) and occurs at \( i = 2.22 \). This corresponds to an efficiency of the element 'A' to 'B' of 67%.

If the efficiency is expressed in terms of the displacement of the first element then a graph can be constructed which complements the transmission function. For the example where contact of the roller on the driven element...
changes from the bottom to the top surface of the controlling curve coincides with \( \eta_{\text{max}} \), for the transmission ratio \( x_1/x_2 = 4/9 \) shown in Fig. 15.5, the curves are shown in Fig. 15.12.

Curve no. 1 is for the example where the driving element remains the driving element throughout the stroke of the mechanism. Curve no. 2 is the efficiency for the element 'B' for the displacement \( x > x_f \), which is when it becomes the driven element. Curve no. 3 is for \( \eta = \frac{1}{\eta_{\text{max}}} \). When the mechanism reaches the displacement for which \( i = \tan \alpha = 2f \) then \( \eta_{\text{th}} = 0 \) and \( \eta \rightarrow \infty \). For \( f = 0.1 \) then \( \tan \alpha = 11^\circ \) and the mechanism will lock.

The analysis becomes more complex if the driving force does not act on the centre of gravity of the driving element, because a dynamic couple will be set up and there will be an increase in the passive resistance of the mechanism.

**MECHANISMS WITH VARYING TRANSMISSION RATIOS**

Varying transmission ratios are used between the driving and driven elements in automatic weapons to reduce the accelerating and decelerating forces. It is not possible to employ the commonly used equations of motion of the form 'force = mass x acceleration', and it is necessary to perform a reduction of the forces and masses.

Displacement, velocity and acceleration of a point on the element of a mechanism can be determined by the equations of motion if the forces acting on the different elements are known. The reverse of this can be carried out whereby the forces acting on the different points of the element can be determined from given values of velocity and acceleration.

Kinematic and dynamic analysis of the operating mechanism of a weapon is not only used in the design of the weapon, but also as a method of understanding the function of the mechanism and to find the critical points in it.
The basis of design consists, especially when computer-aided design is used, of the development and solution of the equations of motion of the main elements of the mechanism.

Using the basic couple shown in Fig. 15.1 and Lagrange's equation for one degree of freedom (the main driving element is only able to move in the x-axis) the equation of motion is constructed. Rigidity and damping effects are small, so the initial Lagrange equation is:

\begin{equation}
\frac{d}{dt} \left( \frac{\partial S}{\partial \dot{\theta}} \right) - \frac{\partial S}{\partial \theta} = Q
\end{equation}

The kinetic energy, \( E \), is for all of the system elements with a general velocity, \( v \). The force \( Q \) is the general force doing work because of the displacement, \( x \). Lagrange's equations can also be used in polar co-ordinates for elements rotating about an axis, as for the examples shown in Fig. 15.2.

To derive the equations of motion for mechanisms with varying transmission ratios equation (15.37) and Fig. 15.1 are used. The total kinetic energy of the system is:

\begin{equation}
E = \frac{1}{2} m_A \cdot v_A^2 + \frac{1}{2} m_B \cdot v_B^2
\end{equation}

Rearranging and substituting for \( i = v_A/v_A \):

\begin{equation}
E = \frac{1}{2} v_A^2 (m_A + m_B \cdot i^2)
\end{equation}

thus:

\begin{equation}
\frac{\partial E}{\partial v_A} = v_A (m_A + m_B \cdot i^2)
\end{equation}

and:

\begin{equation}
\frac{d}{dt} \left( \frac{\partial E}{\partial \dot{v}_A} \right) = (m_A + m_B \cdot i^2) \frac{dv_A}{dt} + 2 v_A \cdot i \cdot m_B \cdot \frac{di}{dt}
\end{equation}

\begin{equation}
= (m_A + m_B \cdot i^2) \frac{dv_A}{dt} + 2v_A \cdot i \cdot m_B \cdot \frac{di}{dx}
\end{equation}

After substitution of the above differentiations into equation (15.37) and rearranging:

\begin{equation}
(m_A + i^2 \cdot m_B) \dot{v}_A + v_A \cdot m_B \cdot i \frac{di}{dx} = Q_A
\end{equation}

The expression in the parenthesis can be replaced by a single value known as the reduced mass, \( m_r \), so that:

\begin{equation}
m_r = (m_A + m_B \cdot i^2)
\end{equation}

Equation (15.45) is valid for an ideal linkage in which there are no losses. If losses are taken into account then:

\begin{equation}
m_a = m_A + \frac{i^2}{\eta} \cdot m_B
\end{equation}

The generalised force, \( Q_A \), results from all of the external forces acting onto the couple and reduced to the element \( A \) and is determined from equating the outputs of the work done by all of the forces and the output of the work done by the reduced force:

\begin{equation}
Q_A = F_A (t) \cdot v_A - F_A (x) \cdot v_A - F_B (y) \cdot v_B
\end{equation}

After rearrangement and using \( i = v_B/v_A \) and adding the mechanical efficiency between the elements:

\begin{equation}
Q_A = F_A (t) - F_A (x) - \frac{i}{\eta} - F_B (y)
\end{equation}

Equations (15.46) and (15.48) can also be used for n-member mechanisms with n-1 variable transmission ratios by using expressions for reduced mass and reduced force. For n successive elements (n=1,2,3,...):

\begin{equation}
m_n = m_1 + m_2 + \ldots + m_i + \frac{i^2}{\eta_1} + \frac{i^2}{\eta_2} + \ldots + \frac{i^2}{\eta_{n-1}}
\end{equation}

\begin{equation}
Q_A = F_1 \pm F_2 + \frac{i_1}{\eta_1} \pm F_3 + \frac{i_2}{\eta_2} \pm \ldots \pm F_n + \frac{i_{n-1}}{\eta_{n-1}}
\end{equation}

where:

\begin{equation}
i_1 = \frac{v_1}{v_A}, i_2 = \frac{v_2}{v_1}, i_3, \ldots, i_{n-1} = \frac{v_{n-1}}{v_{n-2}} , i_{n-2}
\end{equation}

The equations (15.46) and (15.48) are valid for all types of mechanism shown in Fig. 15.1 and Fig. 15.2. For mechanisms with an inserted element the elasticity and the mass are neglected. If it is wished to take the mass into account then the mechanism is considered to be multi-element with the second element being pivoted and its moment of inertia used in the solution of the equation of motion.
Dynamic Analysis of Classic Automatic Systems

BLOW-BACK SYSTEMS

The basic analysis of the automatic operating cycle is based on the pure blow-back system which can then be extended to the delayed blow-back system, and the other self-powered operating systems. Fig. 16.1 shows the diagrammatic representation of a pure blow-back system and the forces acting on the basic components. The forces shown in Fig. 16.1 do not all act at the same time but at different displacements of the breech block.

**Force acting on the Cartridge Case Base,** $F_D$

This is the most important force acting on the system with the greatest magnitude and is comprised of the force exerted by the propellant gas, $P_{ig}$, acting on the internal base area, $S_D$, of the cartridge case, thus:

$$F_D = P_{ig}S_D$$  \hspace{1cm} \text{(16.1)}

---

It is the force $F_D$ that drives the breech block to the rear of the weapon and provides the energy to cycle the weapon.

**Force of the Return Spring,** $F_{rs}$

This force acts on the breech block throughout the whole functional cycle. It is usually linear and is given by:

$$F_{rs} = F_{rs0} + c.s$$  \hspace{1cm} \text{(16.2)}

The pre-tension force, $F_{rs0}$, is usually in the range (5 to 8) $Q_{bb}$, where $Q_{bb}$ is the weight of the breech block. The return spring force has a major effect on the functional cycle of the weapon and depends on the spring constant, $c$, the amount by which it is compressed, $s$, and its efficiency, $\eta$. Helical wire springs are normally used with an efficiency of between 80-95%. Thus the spring constant, $c'$, during the return movement of the breech block is given by:

$$c' = \eta_s F_{rs}$$  \hspace{1cm} \text{(16.3)}

**Friction Force,** $F_f$

Friction acts to oppose motion throughout the whole functional cycle. Its value varies considerably from weapon to weapon. A dynamic couple acting on the breech block (when the function forces are offset from the centre of gravity of the breech block) increases passive resistance to movement, as was discussed in Chapter 14.

Fig. 14.12 shows a breech block where the centre of gravity is offset from the centre line of the acting friction forces.

The individual friction forces, $N_1$ and $N_2$, can be calculated from:

$$N_1 = N_2 = \frac{F_D \cdot e}{x_1 + x_2 + f(y_1 - y_2)}$$  \hspace{1cm} \text{(16.4)}

$x_0y_0$ = distance from the centre of mass of the moving parts to the point at which the forces $N_1$, $N_2$, and $F_D$ act.

It can be seen that it is possible to reduce the influence of a dynamic couple by reducing the offset of the breech block mass, $e$, or by increasing the distances $x_1$ or $x_2$, or the height $y_1$ and $y_2$. If the offset, $e$, is zero the dynamic couple will be zero.

**Braking Force,** $F_B$

The simplest case is when $F_B$ = constant, such as an additional friction force acting on the breech block. For a more complex solution, such as delayed
blow-back, it is necessary to develop the equations of motion for the two-piece breech block. An example of delayed blow-back is the roller locking breech block used on the German G3 automatic rifle, which is shown diagrammatically in Fig. 16.2.

Angles $\beta$ and $\gamma$ are chosen so that the breech rollers remain engaged until the bullet leaves the barrel. If the surfaces are flat the transmission ratio remains constant between the two breech block masses. The forces shown in Fig. 16.2, with the exception of $F_D$, are internal forces and do not affect the overall equations of motion.

Forces $F_p$ and $F_n$ are the forces transmitted to the weapon casing and breech block carrier respectively by the rollers and can be obtained from:

$$F_p = F_D \frac{\sin \psi}{\sin (\phi + \psi)}$$  \hspace{1cm} (16.5)

and

$$F_n = F_D \frac{\sin \phi}{\sin (\phi + \psi)}$$  \hspace{1cm} (16.6)

If it is assumed that the forces in the $x$ direction are in equilibrium then:

$$F_D = F_{px} + F_{nx} + m_r \cdot \dot{x} - F_p \cdot \cos \phi + F_n \cdot \cos \psi$$  \hspace{1cm} (16.7)

if the inertia force $m_r \cdot \dot{x}$ of the rollers is small compared to that of the breech block.

The transmission ratio, $i$, between the two part breech block is $i > 1$ and for $\beta > \gamma$ is $i > 2$. Thus the return spring force is $\frac{i^2}{\eta} F_n$ and the equivalent mass of the breech:

$$m_i = m_{bc} + m_r + \frac{i^2}{\eta} \left( m_{bc} + \frac{1}{3} m_n \right)$$  \hspace{1cm} (16.8)

Only one-third of the mass of the return spring is used to approximate the actual moving mass of the spring. The roller mass, $m_r$, is the mass of both rollers.

**Cartridge Case Extraction Force, $F_{EX}$**

The force required to remove the empty cartridge case from the chamber acts until the chamber is completely clear of the cartridge case. In automatic weapons using the blow-back method of operation the cartridge case is being extracted from the moment the bullet starts to move. The extraction force is determined by the friction force between the cartridge case and the chamber wall, which is determined by the propellant gas pressure acting on the inner surface of the cartridge case.

For blow-back systems the force acting on the base of the cartridge case is much higher than the required extraction force, i.e. $F_D >> F_{EX}$. Thus an extractor is only required to remove misfired or unfired cases and to hold the extracted case for ejection.

Where cartridge case extraction forces are high, as when steel cases are used, a fluted chamber may be used, as shown in Fig. 16.3. This allows high-pressure gas between the cartridge case and chamber wall, which helps to balance the propellant force pushing the case onto the chamber wall thus reducing the friction force.

The resistance of the cartridge case extraction can be taken into account in a simplified manner by multiplying the breech mass by the coefficient $\phi_r$. An approximate solution for pistol systems using parallel cartridge cases is for...
\( \phi_2 = 1.35 \). The breech mass can then be expressed as 
\( m'_{\text{bb}} = \phi_2 (m_{\text{bb}} + \frac{1}{3} m_{\text{n}}) \).
This mass increase only occurs over the displacement of the breech block equal to the length of the cartridge case.

**Breech Block Buffer Force, \( F_{\text{bu}} \)**

The breech block buffer only affects the movement of the breech block towards the end of its stroke. Most blow-back weapons are not fitted with a buffer and buffer spring, but rely on the breech block impacting the weapon casing. To calculate this effect the breech casing is treated as a buffer with a spring of high rigidity and small movement with high energy on impact. Thus a proper buffer system is dealt with in the same manner, but with the appropriate buffer characteristics.

If energy is absorbed by the weapon casing on impact by being displaced by \( x_n' \), then the equivalent spring stiffness during compression is:

\[
C_1 = \frac{m_{\text{bb}} \cdot v_{\text{bb}}^2}{x_n'}
\]

If there are losses of energy at impact the spring stiffness during expansion is:

\[
C_2 = \eta \cdot C_1
\]

The efficiency, \( \eta \), is obtained from the ratio of impact energy after impact to that of the energy before impact.

When the breech block reaches its rear position the velocity will be zero. The buffer force will then change sign as it then acts in the same direction as the return spring. For a true buffer the same procedure is applied so that:

\[
F_{\text{bu}} = F_{\text{bo}} + C_{1/2} x
\]

The mass of the buffer, \( m_{\text{bu}}' \). and part of the mass of the buffer spring, \( m_{\text{bu}}' \), are added to the mass of the breech block. The effect of rebound of the buffer is small and can usually be ignored. Thus during the initial return of the breech block the equivalent mass of the breech block is:

\[
m'_{\text{bb}} = m_{\text{bb}} + \frac{1}{3} m_{\text{n}} + m_{\text{bu}} + \frac{1}{3} m_{\text{bo}}
\]

An approximate solution for a hard breech block impact using Hertz's impact theory can be applied, for which the breech block velocity after impact, \( v'_{A'} \), is given by:

\[
v'_{A'} = v_A - \frac{(v_{\text{bb}} - v_n) (1 + \varepsilon)}{1 + \frac{m_{\text{bb}}}{m_w}}
\]

The weapon velocity before impact, \( v_w \), can be taken to be zero, so that:

\[
v_{\text{bb}}' = v_{\text{bb}} - \frac{v_{\text{bb}} (1 + \varepsilon)}{1 + \frac{m_{\text{bb}}}{m_w}}
\]

**Cartridge Case Ejection**

Cartridge case ejection for blow-back systems consists of a rigid fixed projection on the body of the weapon which strikes the cartridge case as the breech block is driven to the rear of the weapon and pushes the empty case through the ejection port. This results in the interaction between the weapon casing, the cartridge case and the breech block. The mass of the cartridge case is small compared to the mass of the breech block for most of these types of weapon, so the effect on the velocity of the breech block is usually small and can be ignored. If it is thought to be significant for a particular example, then the method given in the section 'Effect of Case Ejection on Breech Block movement' in Chapter 13 can be used.

**Effect of the Firing Mechanism**

For blow-back systems either a solid striker is used, which initiates the primer by direct impact or, more often, a hammer firing mechanism is used. Hammer release when firing a burst is performed when the breech block is fully forward and stationary. Energy released by the hammer therefore has a negligible effect. Thus it is only cocking of the hammer or striker by the breech block during its rearward movement that is considered. Fig. 16.4 shows a typical hammer arrangement.

The resistance of the hammer being cocked by the rearward-moving breech will have the effect of adding the mass of the hammer, and firing spring mass, \( m_{\text{h}} \), to the other rearward-moving masses when it is a hammer with linear movement that is being considered.

Thus:

\[
m'_{\text{bb}} = m_{\text{bb}} + \frac{1}{3} m_{\text{n}} + m_{\text{h}} + \frac{1}{3} m_{\text{ho}}
\]

The forces resisting the breech movement are:

\[
F = F_{\text{n}} + F_{\text{h}} + F_{\text{ho}}
\]

For a rotating hammer the expressions for the total mass of the breech and the resisting forces are:
Thus:

(16.20) \[ F_m = F_{T1} + F_{T2} \]

Forces \( F_{T1} \) and \( F_{T2} \) are the friction forces between the cartridge and magazine guide and between cartridges and their values during movement equal to:

(16.21) \[ F_{T1} = f_1 \cdot F_m = f_1 \cdot \frac{F_{ms}}{\cos \alpha} \]

For the contact between two cartridges:

(16.22) \[ F_{T2} = F_2 \cdot F_n = F_2 \cdot \frac{F_{ms}}{\cos \alpha} \]

If \( f_1 = f_2 = f \) then:

(16.23) \[ F_m = 2f \cdot \frac{F_{ms}}{\cos \alpha} \]

During firing the force applied by the magazine spring, \( F_{ms} \), will reduce in steps as a cartridge is removed.

During feeding of a fresh cartridge there is an increase in breech block mass by the mass of the cartridge, \( m_{cr} \), from the instant that the breech block contacts the base of the cartridge. Thus the new breech block mass is:

(16.24) \[ m'_{bo} = m_{bo} + \frac{1}{3} m_{ns} + m_{ca} \]

From Fig. 16.5 it can be seen that the friction force increases as the contact angle \( \alpha \) between cartridges increases until the cartridges sit side by side and three cartridges are in contact with one another. There are additional frictional
forces involved by the cartridge being guided into the chamber and the springing of the spring-loaded extractor onto the cartridge case rim. For small calibre cartridges these are usually small and can normally be neglected.

Equation of Motion for an Automatic Weapon

The equation of motion for the breech block of an automatic weapon is:

(16.25) \[ m'_{bb} \cdot \frac{d^2x}{dt^2} = F_D - F_n - F_{EX} - F_{bb} - F_A - F_f \]

(2) Forward movement:

(16.26) \[ m'_{bb} \cdot \frac{d^2x}{dt^2} = F_n + F_{bb} - F_m - F_f \]

The basic expression for \( m'_{bb} \) is:

(16.27) \[ m'_{bb} = m_{bb} + \frac{1}{3} m_n \]

This is valid for the whole movement of the breech block from the front to the rear and back again to the front.

At certain positions masses are added, such as those for the cartridge case and the buffer, as discussed above. To solve equations (16.25) and (16.26) it is necessary to use a numerical method of integration. For first order differential equations the fourth order Runge Kutta usually gives the required results.

The equation of motion is a second order differential equation, so that it is necessary to convert it into a system of two first order differential equations by the substitution of \( \frac{dx}{dt} = v \).

Initial Conditions for the Solution

Initial conditions are when the bullet begins to move in the barrel. At this instant \( t = 0 \), velocity, \( x = 0 \) and displacement, \( x = 0 \). However, acceleration, \( \ddot{x} \) does not = 0 because pressure builds up in the cartridge case prior to the bullet being released and the initial driving force, \( F_D = P_D \cdot S_D > 0 \). The force of the return spring is equal to its pre-stressed value, i.e. \( F_n = F_{no} \). Thus the initial value of acceleration as expressed by equation (16.25) is:

(16.28) \[ \ddot{x}(0) = \frac{p_D \cdot S_D - F_f - F_{po} - F_{EX}}{m'_{bb}} \]

where

\[ m'_{bb} = m_{bb} + m_{ca} + \frac{1}{3} m_n \]

If an equivalent breech mass is used to compensate for the extraction force, then \( m'_{bb} = \frac{1}{3} (m_{bb} + m_n) \) and thus the extraction force \( F_{EX} = 0 \). The final conditions exist when the breech block has returned to the forward position so that the displacement \( x = 0 \). The solution to the equation of motion consists of a solution for displacement, velocity and acceleration with respect to time, i.e. \( x, \dot{x}, \ddot{x} = f(t) \). The movement of the breech block is characterised by a varying breech block mass and varying resistance to motion with respect to the displacement of the breech block. The length of the integrating steps should, therefore, be chosen to suit each point of the cycle.

GAS-OPERATED WEAPONS

It is possible to apply a similar method of analysis to gas-operated weapons as was previously undertaken for blow-back weapons. The driving force on the breech block, however, is now provided by a force acting on a piston, the force being derived from the propellant gas pressure at a port in the barrel. Thus the driving force acting on the piston, \( F_{pi} \), is determined by the area of the piston and the gas pressure acting on the piston, \( p_{pi} \), and is given by:

\[ F_{pi} = S_{pi} \cdot p_{pi} \]

(16.29)

The main problem in obtaining a solution to the equations of motion for a gas-operated system is the determination of the pressure \( p_{pi} \) in the gas cylinder with respect to time. For the purposes of analysis the gas cylinder arrangement shown in Fig. 16.6 is used.

![Figure 16.6 Arrangement of gas cylinder used for the analysis of gas flow through the gas ports](image)
The whole process can be analysed as gas flowing through a channel connecting two vessels of varying volume. In the first vessel, the barrel, the pressure rises steeply to a typical value between 300–400MPa, which then falls due to the movement of the bullet. In the second vessel, the gas cylinder, the piston moves due to the gas pressure acting in it, thus increasing the cylinder volume. Additionally, there is a loss in pressure caused by gas leaking between the piston and the cylinder walls.

The process begins when the bullet has passed the gas port, the pressure in the barrel being $p_{b1}$. Gas will flow into the cylinder from the barrel, the initial flow usually being critical (the velocity of the gas being greater than the speed of sound in the gas). The pressure in the cylinder, $p_{cy}$, will increase and a point is reached when the flow becomes subcritical. When $p_{p1} = p_{cy}$, the gas flow will cease and when $p_{b1} < p_{cy}$ the gas flow will reverse in direction. The process is shown in Fig. 16.7. For the purpose of analysis the process is divided into two parts: when $p_{b1} > p_{cy}$, and when $p_{b1} < p_{cy}$.

To obtain a solution for the movement of the piston and the force applied to the return spring, $F_{rs}$, the following equations are used with $F_{i}$ denoting all the other forces resisting the motion of the breech block:

**Equation of motion for the piston:**

$$m \frac{d^2 x}{dt^2} = S_{pi} \left( p_{cy} - p_{g} \right) - F_{n} - \sum F_{i}$$

**Equation for calculating the instantaneous cylinder volume:**

$$V = V_{o} + S_{cy} \cdot x_{pi}$$

**Equation for the instantaneous return spring force:**

$$F_{n} = F_{rs} + c_{rs} \cdot x_{pi}$$

**Equation of state for the gas in the cylinder:**

$$p \cdot V = R \cdot T$$

**Equation for the piston velocity:**

$$v_{pi} = \frac{dx}{dt}$$

The following equations are used for the two different periods mentioned above:

**Period $p_{b1} > p_{cy}$:**

**Equation of gas energy change in the gas cylinder:**

$$\frac{d(p_{cy} \cdot V)}{dt} = \gamma \cdot R \left( G_{gp} \cdot T_{b1} - G_{cyw} \cdot T_{cy} \right) - \left( \gamma - 1 \right) p_{cy} \cdot S_{pi} \cdot v_{pi}$$

**Equation of mass change in the gas cylinder:**

$$\frac{d \left( \frac{V}{T} \right)}{dt} = G_{gp} - G_{cyw}$$

**Equation of state for the gas in the barrel:**

$$p_{b1} \cdot V_{b1} = R \cdot T_{b1}$$

**Period $p_{p1} < p_{cy}$:**

**Equation of gas energy change in the cylinder:**

$$\frac{d \left( p_{cy} \cdot V \right)}{dt} = - \gamma \cdot R \cdot T_{cy} \left( G_{gp} + G_{cyw} \right) - \left( \gamma - 1 \right) p_{cy} \cdot S_{pi} \cdot v_{pi}$$

**Equation of mass change in the gas cylinder:**

$$\frac{d \left( \frac{V}{T} \right)}{dt} = G_{gp} - G_{cyw}$$

---

**Figure 16.7 The pressure/time curve for the pressure in the barrel and the pressure in the gas cylinder**

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The following relationships are used for calculating the mass flow at state (1) to state (2) through an orifice. For subcritical flow:

\[ G = C_d \cdot S \cdot \sqrt{\frac{2}{\gamma - 1}} \cdot \left( \frac{p_1}{p_2} \right)^{\frac{2}{\gamma}} \cdot \left( 1 - \left( \frac{p_2}{p_1} \right)^{\frac{\gamma + 1}{\gamma}} \right) \]

For critical flow:

\[ G = C_d \cdot S \cdot \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma + 1}{2(\gamma - 1)}} \cdot \sqrt{\frac{\gamma - 1}{\gamma}} \cdot \frac{p_1}{v_1} \]

To determine the mass flow through the gas port, \( G_{wp} \), and the mass flow from the cylinder between the piston and cylinder walls, \( G_{cy} \), the values given in Table 16.1 are used in the above equations.

| Table 16.1 Values used for determining the gas flow |
|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| \( p_1 \) | \( p_2 \) | \( p_{wp} \) | \( p_{cy} \) | \( p_{cy} \) | \( p_{cy} \) |
| \( v_1 \) | \( v_{wp} \) | \( v_{cy} \) | \( v_{cy} \) | \( v_{cy} \) | \( v_{cy} \) |
| \( c_d \) | \( c_d \) | \( c_d \) | \( c_d \) | \( c_d \) | \( c_d \) |

Fig. 16.8 shows a typical flow diagram for the flow of gas through a typical gas port, where the flow begins as critical and changes to subcritical part of the way through. The curves \( p_{wp} = f(t) \) and \( p_{cy} = f(t) \) are measured values. The curves \( G_{wp} = f(t) \) and \( M_{cy} = f(t) \) are calculated values.

Three commonly used gas characteristics for calculating the pressure history in the gas system are the ratio of specific heats, \( \gamma \), the gas constant, \( R \), and the gas temperature in the barrel, \( T_{b1} \). Accurate estimation of these characteristics is difficult. Figs. 16.9, 16.10 and 16.11 show the effect of a typical pressure/time history curve for a typical gas cylinder for different values of these characteristics.

Figure 16.8 Gas flow diagram for the gas flow through a typical gas port where the flow changes from critical to subcritical
It can be seen that the effect of quite large variations in these characteristics results in only a small difference in the pressure/time curve. The recommended values for two of the characteristics are:

- $\gamma = 1.26$
- $R = 350 \text{J/kgK}$.

The influence of the gas pressure in the barrel at the entrance to the gas port is shown in Fig. 16.12. Case 1 had an initial pressure in the barrel $p_{bi} = 261 \text{MPa}$ and for case 2 the same pressure was $109 \text{MPa}$.

When a gas flows through a channel there are losses which affect the mass flow rate. The losses are characterised by the discharge coefficient, $C_d$. During the period when $p_{g1} > p_{s}$, the high velocity of the gases in the barrel considerably affects the gas flow through the gas port. The flow features are shown in Fig. 16.13, which depicts two different types of gas port.

The way in which the gas port wears shows that gases do not flow parallel to the axis of the gas port. High velocity gases impact on the opposite side of the port wall as they enter the port owing to their original flow direction. They are then deflected onto the opposite wall as a series of impacts. This causes greater losses in flow rate than the usual entry losses associated with a conventional discharge coefficient. Comparing one flow period with another it can be seen that the discharge coefficient for flow from the barrel to the cylinder, $C_{d_{1}}$, is less than the discharge coefficient for flow from the cylinder to the barrel, $C_{d_{2}}$.

For gases which flow from a vessel in which they are stationary, through a circular channel, the values of discharge coefficient are usually in the range of 0.60-0.65. The deflection of high velocity gases from wall to wall as described above would indicate that the value of $C_{d_{1}}$ will fall at a greater rate with gas velocity than $C_{d_{2}}$. This has been proved experimentally.\(^1\)
Figure 16.13 Two different geometries used for typical gas ports

When the gas port is perpendicular to the barrel axis the variation in the discharge coefficient with respect to the gas velocity in the barrel is shown in Fig. 16.14. The straight line relationship is given by:

$$\text{Cd}_1 = 0.650 - 0.00016v_{bl} \tag{16.42}$$

The velocity of the propellant gases in the barrel, \(v_{bl}\), is the same as that of the bullet.

The discharge coefficient is also affected by the angle of the gas port and the length of the channel. Any change in the direction of the gases, especially through a regulator, also affects the discharge coefficient. For simple gas port arrangements equation (16.42) is used for calculating \(\text{Cd}_1\). The value of \(\text{Cd}_1\) is usually taken as 0.65.
Gas flow through the clearance between the piston and the cylinder also results in losses which are characterised by a discharge coefficient. Fig. 16.15 shows how the clearance varies according to the concentricity, or otherwise, of the piston and cylinder. If the piston and cylinder are concentric, the radial clearance is:

\[ \delta = R_{cy} - R_{pi} \]

(16.43)

If there is maximum eccentricity between piston and cylinder, the largest gap between the two is 2 \( \delta \), as shown in Fig. 16.15. There are cases between the two extremes. The influence of the clearance between the piston and the cylinder on the measured curves for \( p = f(t) \) for different piston diameters is shown in Fig. 16.16.

The initial pressure in the barrel, \( p_{i0} = 261 \text{MPa} \), the initial volume of the cylinder, \( V_0 = 2.03 \text{cm}^3 \), the diameter of the gas port was 1.6mm and the diameter of the cylinder was 15mm. Gas flow through the clearance between the piston and the cylinder is through an area considerably different to that of the circular port. Thus \( C_d_{cy} \) is considerably different to \( C_d_1 \) and \( C_d_2 \). Data from a variety of sources were used to obtain the following expression for the discharge coefficient through an annulus of thickness \( \delta \):

\[ C_d_{cy} = 1.123 (\delta + 0.04)^{0.44} - 0.29 \]

(16.44)

Fig. 16.17 shows a graphical plot of equation (16.44).

If there is any eccentricity between cylinder and piston it is recommended that the discharge coefficient \( C_d_{cy} \) is increased by 25\%. If labyrinth packing or piston sealing rings are used, then this will considerably affect the mass flow.

Using the preceding recommendations for the above discharge coefficients and the appropriate values for the other variables in the gas system the equations (16.30) and (16.41) can be solved. From these solutions the dependence of the pressure in the gas cylinder, velocity and displacement of the gas piston with respect to time can be obtained, a typical set of curves being shown in Fig. 16.18. These results are then used with the equations of motion for the automatic system of the weapon.

**LONG RECOIL-OPERATED WEAPONS**

In long recoil-operated weapons the barrel recoil is the same as that of the breech block and is the main element in the automatic system. The equation of motion for the barrel differs for individual phases of the operating cycle. The basic equations include the mass of the barrel, \( m_b \), the mass of the recuperator, \( m_{rp} \), and are:
Figure 16.17 Effect of annulus clearance between the piston and cylinder on discharge coefficient

(1) Barrel and breech recoil to the rear:

\[ (16.45) \quad \left( m_b + m_{bb} + \frac{1}{3} m_w + \frac{1}{3} m_c \right) \ddot{x} = F_D - F_w - F_n - F_{rl} - F_{bu} \]

(2) Counter-recoil, which includes the feeding of the round and the closing of the breech:

\[ (16.46) \quad \left( m_b + \frac{1}{3} m_w + \frac{i}{n_{le}} m_w \right) \ddot{x} = F_t + F_{re} - F_{rl} - F_{le} - F_{ob} - \frac{i}{n_{le}} F_{le} \]

The signs in the equations for the friction forces and the intensity of the buffer force change with the direction of the barrel movement, the buffer spring usually having large energy losses so that the return energy is considerably reduced.

The final phase of the operating cycle is the ramming of the cartridge and the locking of the breech, which requires the force \( F_{bc} \), under the action of the return spring. The equation of motion is:

\[ (16.47) \quad \left( m_{bb} + \frac{1}{3} m_w + m_c \right) \ddot{x} = F_n - F_{rl} - F_{bl} \]

**SHORT RECOIL-OPERATED WEAPONS**

As with long recoil-operated weapons, it is the barrel which is the main element of the operating cycle in short recoil-operated weapons. However, there is also the need to calculate the period of inertia of the breech block after it separates from the barrel. The remainder of the components of the cycle are similar to those previously discussed.

Short recoil systems use a variable transmission ratio between the barrel and the breech block. It is usually in the form of a lever accelerator whose mass is small compared to the barrel, and so can be ignored in the analysis. The phase for the rearward-moving barrel and breech mechanism is divided into four separate periods for the development of the equations of motion.
First Period of Motion

The barrel and all recoiling parts are locked together and move towards the rear of the weapon. The first period starts when the projectile begins to move and ends when the accelerator is set in motion. The actual equation of motion is the same as that of the long recoil cycle given by equation (16.45). The force for the buffer is omitted because the first period ends before the buffer is reached.

Second Period of Motion

This is the period during which there is breech block underslide, so that there is relative movement between the barrel and the breech block carrier which is achieved by the accelerator. The breech block, however, is still locked to the barrel. Equations (15.46) and (16.15) are used to develop the equations of motion for the barrel and the acceleration of the breech block carrier. Fig. 16.19 shows the action of the accelerator lever between the barrel and breech block carrier.

Equivalent mass of the barrel:

\[ m'_{b} = m_{b} + \frac{i^2 \eta}{\eta} m_{b} \]  

(16.48)

\[ \frac{i}{\eta} = \frac{l_2}{l_1} \cdot \frac{\sin \alpha + 2f \cdot \cos \alpha}{\sin \alpha - 2f \cdot \cos \alpha} \]  

(16.51)

\[ \frac{i}{\eta} = \frac{l_2}{l_1} \cdot \frac{\tan \alpha + 2f}{\tan \alpha - 2f} \]  

(16.52)

Equivalent force:

\[ Q_{A} = F_{A}(t) - F_{A}(x) - \frac{i}{\eta} F_{B}(x) \]  

(16.53)

where: \( F_{A}(t) = F_{11}(t) - 0.98 \ p(t) \ \cdot \ S \)

and: \( F_{A}(x) = F_{re} + F_{11} \)

\( F_{B}(x) = F_{re} + F_{12} \)

The equation of motion is obtained by substituting the relationship for \( m'_{b} \) and \( Q_{A} \) and equation (16.51) into equation (16.47) for a variable transmission. Thus,

\[ m'_{b} \cdot \frac{d^2x}{dt^2} + \frac{1}{2} \ v_{A}^2 \cdot \frac{dm'_{b}}{dx} = Q_{A} \]  

(16.54)

where:

\[ \frac{1}{2} \ v_{A}^2 \cdot \frac{dm'_{b}}{dx} = v_{A}^2 \ m_{b} \frac{i}{\eta} \frac{di}{dx} \]  

(16.55)

and \( \frac{di}{dx} \) is a derivative of the transmission function with respect to \( x \). If \( i(x) \) is a constant, then this element can be neglected because \( \frac{di}{dx} = 0 \).

Third Period of Motion

This is the period during which the breech unlocks. The accelerator acts on the breech block carrier which controls the movement of the breech block through a cam system, as shown in Fig. 16.19. If all other mechanisms controlled by
the barrel or breech block carrier are ignored the system is simplified to a three-element mechanism with two transmission ratios and associated mechanical efficiencies. The mass of the additional accelerator element for the breech block is small and can usually be ignored. During the third period there is also the force exerted by the residue propellant pressure on the internal base of the cartridge case, \( F_p(t) \), because for weapons with high rates of fire the gas pressure when the breech unlocks can be as high as 10–20 MPa.

To determine the equation of motion, equation (15.46) is used, but extended for more elements, and modifying equations (15.49) and (15.50) for the reduced mass and force. Thus:

\[
\begin{align*}
    m_1 &= m_1 + \frac{I_1}{\eta_1}, \\
    m_2 &= \frac{I_1}{\eta_2}, \\
\end{align*}
\]

and:

\[
\begin{align*}
    Q_A &= F_3(t) - F_{RI} - F_{Re} - \frac{I_1}{\eta_1} (F_{R1} + F_{Re}) - \frac{I_2}{\eta_2} (F_{Rb} + F_{El})
\end{align*}
\]

where:

\[
\begin{align*}
    m_1 &= m_b + \frac{1}{3} m_{re} \\
    m_2 &= \frac{1}{3} m_{re} \\
    m_3 &= m_{bb} \text{ etc}
\end{align*}
\]

To determine the efficiency of the ratio \( \frac{1}{\eta} \), the simplified arrangement shown in Fig. 16.20 is used. As in the case for the second period the accelerator is simplified and its inertia is neglected. Fig. 16.20 shows all internal forces and the resultant of these losses, \( F_{RI} \). The efficiencies can be determined as follows:

\[
\begin{align*}
    \eta_1 &= \frac{F_{R2}}{F_{R1}} \frac{l_1}{I_1} \\
    \eta_2 &= \frac{F_{R3}}{F_{R1}} \frac{l_2}{I_1}
\end{align*}
\]

Where:

\[
\begin{align*}
    F_{R1} &= R_1 (\sin \alpha + 2 f \cos \alpha) \\
    F_{R2} &= R_2 (\sin \alpha - 2 f \cos \alpha) - R_3 (\sin \beta + 2 f \cos \beta) \\
    F_{R3} &= R_3 (\cos \beta - 2 f \sin \beta)
\end{align*}
\]

The reaction \( R_2 - R_3 \) is excluded from the analysis because the individual parts of the mechanism are in equilibrium.

The transmission ratio is given by:

\[
\begin{align*}
    i_1 &= \frac{l_2}{l_1} \\
    i_2 &= \frac{v_2}{v_1} = \frac{l_2}{l_1} \tan \beta
\end{align*}
\]
Fourth Period of Motion

The fourth period lasts from the end of the unlocking of the breech until the end of the accelerator action. The breech block and carrier are driven to the rear of the weapon and the cartridge case is extracted from the chamber. If unlocking occurs whilst propellant gas pressure is above atmospheric pressure there will be a force acting on the base of the cartridge case, \( F_{c}(t) \), which will add to the acceleration of the breech block.

The equation of motion, including efficiency and transmission ratio, is similar to that of the second period. The masses in terms of equivalent mass change are:

\[
(16.68) \quad m_{k} = m_{b} + \frac{1}{3} m_{w}
\]

\[
(16.69) \quad m_{b} = m_{1b} + m_{w} + \frac{1}{3} m_{n}
\]

In the expression for equivalent force, the force \( F_{g}(t) \) will not be included. However, the cartridge case extraction force, \( F_{EX} \), will be added during the period of case extraction and there may be the gas pressure force acting on the base of the cartridge case, \( F_{D} \).

The reduced force acting on the main element is:

\[
(16.70) \quad Q_{A} = F_{c}(t) - F_{w} - F_{D} - \frac{1}{\eta} (F_{n} + F_{12} + F_{EX} - F_{D})
\]

At the end of the fourth period the remainder of the operating cycle is similar to that of a blow-back system.

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Chapter 15


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Chapter 16

1 Popelínský, op. cit. in Ch. 8, note 1.


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Symbols

a = velocity of sound
a = surface area
a = thermal diffusivity
a = constant
A = work done
b = width of locking lug
b = cartridge belt pitch
bc = breech closing
Br = breaking strength
c = specific heat
c = integration constant
c = spring constant
cm = maintenance cost of a single weapon
cw = cost of a single weapon
Cd = discharge coefficient
d = diameter
e = strain
e = specific energy
e0 = specific energy at the muzzle
E = aiming error
E = energy
E = Young's Modulus
E0 = absolute aiming error
EO = muzzle energy
E = energy of projectile at impact
f = coefficient of friction
f = frequency
fp = fire-power
F = force
g = acceleration due to gravity
G = gas density
G = modular of elasticity in shear
G = mass flow
G = weight
h = height of locking lug
h = working length of a spring
i = radius of gyration
l = transmission (leverage) ratio

I = section modulus
J = moment of inertia
k = coefficient
k = factor of safety
Kr = readiness coefficient
l = length
m = mass
m = average number of rounds to incapacitate the target
mw = weapon mass
mz = muzzle
m = total instantaneous mass flow from the barrel
M = torque
M = bending moment
n = barrel design safety coefficient
n = number of rounds fired in a burst
n = polytropic exponent of gas expansion
n = number of weapons in a production batch
nb = number of rounds fired before barrel replacement
nd = daily ammunition expenditure
nnm = number of rounds in a magazine
N = total ammunition expenditure
Nd(n) = number of bursts of n rounds
nf = number of rounds between failures
Nf = number of failures
Nw = number of weapons
p = pressure
pi = piston
P = power
Ph = hit probability
P(t) = probability of failure
P(t) = probability of incapacitation
Ph1 = hit probability for a single round
Ph(1) = hit probability for the first round
q = heat transfer
Q = specific energy
r = radius
R = braking resistance
R = gas constant
R = rigidity
R = ratio of muzzle energy to weapon mass
RF = combat rate of fire
R(t) = rate of fire
RSP = relative stopping power
s = locking lug cross-sectional area
Military Small Arms

s = spring displacement
s = bullet cross-sectional area
t = time
ta = time to aim the weapon
tb = time to change the barrel
tf = time of firing
tm = time to fire one magazine of ammunition
tp = time for repair
tr = time between shots
trm = time to replace the magazine
T = temperature
v = specific volume
v = velocity
v₀ = muzzle velocity
V = volume
x = displacement

cyw = cylinder walls
CG = centre of gravity
d = design value
de = design element
dl = shot delay
D = cartridge case
e = engagement of target
e = equivalent
e = elements within the cartridge case
E = elastic limit
EX = extraction
f = friction
fe = feed
gp = gas port into cylinder
gw = guideways
h = hammer
hs = hammer spring
KE = kinetic energy
l = link
m = mean
m = magazine
ms = magazine spring
mζ = muzzle
p = cartridge position
p = propellant
pi = piston
pr = primer
pri = primer initiation
r = radius
r = rifling
r = resultant
r = rollers
re = recuperator
rp = recoiling parts
rs = return spring
rso = return spring pre-tension
R = reactive effect of flow of propellant gases
R = resistance
s = internal system
s = shear
sp = spring
t = trigger
tot = total
T = temperature

Subscripts

a = axial
at = atmosphere
b = bullet
b = barrel
bb = breech block
bc = breech block carrier
bl = breech locking
bp = barrel port
br = braking point
br = barrel friction
bt = bullet travel
bu = buffer
bus = buffer spring
B = braking
c = chamber
c = circumference
c = cycle
cα = cartridge
cβ = cartridge belt
cο = compressive
cοm = combustion space
cr = commencement of rifling
cr = critical
cr = counter-recoil
cy = cylinder
the specific energy of the propellant.

(8) The pressure measured by crusher gauges or piezo electric pressure transducers does not give the actual pressure acting on the base of the chamber: the pressure being measured is the mean pressure behind the bullet at the moment when maximum pressure is achieved.

It can be seen that precise internal ballistics calculations are not necessary, especially at the beginning of a problem. It is more convenient to apply simplified analytical methods developed for large calibre guns with corrections to account for the differences in the internal ballistics.

5

Functional Cycle and Automatic Weapon Main Parts

If a weapon is to operate at all, there must be a set sequence of events within it. First, the cartridge must be inserted into the chamber and the breech block closed behind it to support the cartridge case. The primer is then initiated. After the shot has been fired the breech must be opened and the empty cartridge case removed from the chamber and ejected from the weapon. The next cartridge must be loaded into the weapon by removing a round from the magazine and feeding it into the chamber. Again the bolt must be closed and locked to facilitate the next shot. The sequence of operations that occurs between two subsequent shots is known as the functional cycle of the weapon.

In automatic weapons the continuous transfer from one function to the next is controlled by the operating system of the weapon. Automatic functioning can be achieved for part of the functional cycle, to give semi-automatic operation, or for all of the cycle to give fully automatic operation.

In semi-automatic weapons some functions are carried out by its semi-automatic system, while the rest are carried out by the crew. In self-loading weapons, after the weapon has been fired one complete functional cycle takes place and the weapon is ready for the next shot. Before the next shot is fired it is necessary to release and then press the trigger. Self-loading weapons are used for single aimed shots. Automatic weapons continue to fire with one functional cycle following the other until the trigger is released, so the weapon is capable firing bursts.

The time interval between two subsequent shots is dependent on the rate of fire. For a conventional automatic weapon the time interval between two subsequent shots is the same as the time interval for one functional cycle.

For weapons designed with a high rate of fire and with multiple barrels or chambers the time interval between two subsequent shots is less than that of one functional cycle. This is shown in the diagrams in Fig 5.1.
Fig. 5.1 Functional cycle for different types of automatic weapons

These diagrams show those parts of the functional cycle which are related to the cartridge, or empty cartridge case. They are:

1. Unlocking and opening the breech block
2. Extracting the cartridge case
3. Ejecting the cartridge case
4. Inserting the cartridge into the chamber
5. Closing the breech
6. Locking the breech.

Fig. 5.1 shows that the high rate of fire of multi-barrelled or multi-chambered weapons is achieved by overlapping the functional cycle of one cartridge chamber or barrel with that of another.

The rate of fire of a weapon is dependent on the individual elements of the functional cycle, the speed of the bolt, the cartridge dimensions and the passive resistance to motion of the system components. Fig. 5.1 indicates how the rate of fire of conventional automatic weapons can be increased. Table 5.1 shows two typical functional cycles for conventional automatic weapons in which the positioning of the cartridge before firing differs. In the first example the cartridge is already chambered with the bolt closed when the trigger is pulled: this is known as closed bolt operation and is normally used with pistols and rifles. In the other example the cartridge is held in the magazine or ammunition belt ready to be inserted into the cartridge chamber, and is known as open bolt operation and is normally used with machine-guns and sub-machine-guns. Table 5.1 lists the individual elements of the functional cycle and the mechanisms associated with those elements. For automatic cycling of a weapon the following are required:

- barrel group (barrel, receiver, driving and/or holding elements of the barrel, muzzle system)
- bolt
- recoil system (for weapons with a recoiling barrel)
- cartridge feeding system
- extraction mechanism
- ejection mechanism
- feed mechanism
- trigger mechanism
- firing mechanism
- return and striker springs.

Additionally, there must be a source of energy for driving the system. The body of the weapon plays an important role in the weapon operating system and houses the barrel group, bolt, recoil system, feed system, trigger, firing mechanism and other weapon mechanisms and elements. If the barrel is firmly attached to the receiver then the weapon body fulfills the role of the receiver.

To the components of the basic weapon must be added the handling and auxiliary parts, which include the stock, grip, hand guard, bipod, sights, sling swivels and hand guards. These basic gun parts are illustrated in Fig. 5.2.
Two methods are used to represent graphically the relative movements and timings of the different components and mechanisms in a weapon. Fig. 5.3 shows a typical cyclograph depicting the sections of the individual parts and mechanisms of the weapon which are dependent on the displacement of the main part of the automatic system, usually the breech block carrier. This cyclograph can be produced for an existing weapon by measuring the displacements of the main part of the automatic system and those parts connected kinematically to it. When designing a new weapon, the establishment of the cyclograph is the first step, as

this makes it possible to determine the basic length of the weapon to provide the required automatic function.

Another method is the functional diagram. Whereas the cyclograph represents only length dimensions, the functional diagram depicts the displacement, velocity and acceleration as a function of time for the different mechanisms. Fig. 5.4 shows two examples of a functional diagram depicting the displacement of components with respect to time. Fig 5.4a shows the displacement of the bolt for a weapon firing from a closed bolt. Fig. 5.4b shows the displacement of the breech block carrier and recoiling barrel for a weapon firing from an open bolt.

The functional diagram of an existing weapon can be found experimentally by measuring the displacements of individual components with respect to time using a displacement transducer. Differentiating the functional diagram gives the velocities and acceleration with respect to time. When designing a new weapon the functional diagram is calculated and used for assessment of the functional diagram and to calculate the rate of fire. Fig. 5.5 shows a section through a model 58 7.62mm sub-machine-gun. Fig. 5.6 shows the functional diagram for the same weapon and Fig. 5.7 shows its cyclograph.
Figure 5.4 Functional diagram for a weapon firing from (a) a closed bolt and (b) an open bolt.

1 - initiation of primer
2 - unlocking
3 - end of breech acceleration
4 - impact on buffer
5 - end of buffer action
6 - breech caught by sear

$x_1$ = barrel recoil
$x_2$ = breech acceleration
$x_3$ = breech displacement from the sear to front position
$x_4$ = compression of buffer spring

Figure 5.5 Section through the Model S8 sub-machine-gun.
Figure 5.7 Cyclograph for the model 58 sub-machine-gun
Barrels

PURPOSE AND CONSTRUCTION OF BARRELS

The purpose of the barrel of a weapon is to impart velocity and direction to the bullet. It is a thick walled pressure vessel in which the propellant is burnt and must be able to withstand high pressures and temperatures which are developed each time the weapon is fired. The rear of the bore of the barrel is used to locate the cartridge, while the front part is used for the expansion of propellant gases which impart velocity to the bullet. This is shown schematically in Fig. 6.1.

In rifled weapons the barrel also imparts spin to the bullet by using helical grooves along the bore. The main dimension of a barrel is its calibre which is the dimension of the bore before the rifling grooves are machined. Barrel length, L, is an important parameter because it affects the bullet muzzle velocity, \( v_0 \), which can be calculated from the following empirical formula:

\[
v_0 = k(L)^{0.5}
\]

The coefficient of weapon ballistic power, \( k \), is usually between 90 and 130 m/s. The barrel of a gun resembles the piston of a combustion engine operating at high pressures: it achieves a high specific output (8 to 120 MW dm\(^{-3}\)) and has a very short service life (several tens of minutes in actual operation). Because of the very high specific power output the barrel operates under a high mechanical and thermal stress. It is important that the barrel exploits the full potential of the ammunition under all firing conditions,\(^1\) and should contribute to reducing the dimensions and weight of the weapon.

Gun Barrel Construction

The barrel comprises an independent block referred to as the barrel group which consists of the following parts:

- barrel
- receiver
- barrel supports
- barrel attachments.

The barrel is fitted into the receiver, which closes the barrel bore from the end of the cartridge chamber and serves to position the bolt, to which the firing loads are transferred. For automatic weapons loaded from the rear, the barrel breech is formed by the closing bolt, or breech block, which is connected to the receiver when firing. Thus, the receiver supports the bolt and transfers the force of the propellant gases to it.

Barrel supports serve to mount and locate the barrel in the weapon. The mounting may be either of a fixed or a sliding design. For small calibre weapons the mounting is fixed and the supports hold and steady the weapon. For larger calibre weapons the supports make it possible to spring-load the housing with the barrel sliding in guides in the weapon casing.

Barrel attachments consist of a variety of components, the most important being:

- muzzle brakes
- recoil increasers
- flash hiders
- noise suppressors (silencers)
- muzzle deflectors
- gas blocks (gas-operated weapons only).

Barrel Types

Small arms are fitted with either monobloc or compound barrels. In monobloc barrels the wall consists of a homogeneous layer which offers sufficient
strength for normal performance and is simple to make. Composite barrels comprise a monobloc barrel with an insert fitted to protect the breech end, the most vulnerable part of the barrel.

**General Barrel Requirements**

The barrel has a major effect on the construction of the whole weapon. The basic requirements are:

- required strength at maximum operating load
- high rigidity to minimise vibrations
- maximum straightness
- concentricity between inner and outer diameters
- adequate service life
- optimum mass to achieve the required strength and stiffness
- low manufacturing costs.

**BARREL BORE**

The bore is the internal part of the barrel. It has a long tapered opening at the breech which houses the cartridge and guides the bullet during firing. The bore has the following parts as illustrated in Fig. 6.2:

- breech part
- cartridge chamber
- forcing cone
- rifled section.

The bore of the barrel is symmetrical about the longitudinal axis. Its diameter diminishes towards the forcing cone while the rifled section is usually parallel. The basic parameters of the barrel bore are:

```plaintext
- calibre or diameter across the grooves, d
- length of the initial combustion space, l_{comb}
- length of the cartridge chamber, l_c
- length of the rifled section, l_r
- length of bullet travel inside the barrel, l_{bt}
- barrel length, l_b.
```

**Cartridge Chamber**

The cartridge chamber houses the cartridge in the barrel. The distance between the base of the bullet and the base of the cartridge case determines the initial combustion space. The cartridge case seals the rear end of the barrel and prevents the propellant gases leaking from the breech end. The layout and shape of the cartridge chamber depend upon the design of the cartridge. A circular cartridge chamber is usual. Other shapes can only be used for systems with open chambers or with caseless ammunition. Each part of the cartridge chamber follows the shape of the cartridge case. However, radial and axial dimensions of the cartridge case differ from those of the chamber by the amount of clearance between them as shown in Fig. 6.3.

The longitudinal shape of the cartridge case should provide for both easy loading and extraction after firing. To meet these requirements, the cartridge case and the chamber are of conical design and a clearance is provided between the chamber and the cartridge. The value of this clearance is affected by production tolerances of the cartridge and chamber, by the barrel and cartridge temperatures and by fouling of the chamber by combustion products and lubricant. The limiting value of clearance is given by the cartridge case.
strength. In the axial direction it is more affected by a displacement of the}
{breach block due to the action of the high pressure propellant gases. Radial}
clearance for automatic weapons is usually between 0.05 and 0.25mm. The}
taper of the basic cone governs the conditions of cartridge case extraction after}
firing and is greater with longer cartridge cases operating at high pressure and}
with high rate of fire weapons. The taper is usually between 1:30 to 1:60. Taper}
of the chamber cone depends on the bottle shape of the cartridge chamber, the}
bottle shape ratio \( X \), being:

\[
X = \frac{l_o}{l_{com}}
\]

where \( l_o \) is calculated from the volume of the original combustion space, \( v_{com} \),
and the cross-sectional area of the rifling, \( s \), so that:

\[
l_o = \frac{v_{com}}{s}
\]

The bottle shape ratio, \( X \), of the chamber is usually between 1.05 and 2.5.
Automatic systems firing high performance cartridges have the highest bottle
shape ratios. This results in a shorter cartridge and shorter strokes in automatic
systems which give a higher rate of fire. For pistols and sub-machine-guns the
ballistic performance of the cartridge is low and the bottle shape ratio of the
chamber should be a minimum without increasing the length of the chamber.
The taper of the cone is usually between 1:1.1 to 1:2.5. The taper of the neck
part is very small, being between 1:200 and parallel. The construction of the
cartridge case and chamber should reliably position the loaded cartridge in the
chamber. Location of the cartridge can be achieved in the following ways, as
shown in Fig. 6.4:

- by the cartridge case rim for a rimmed case
- by the chamber cone for a rimless case
- by the belt for a belted case.

For a rimmed round reliable and simple positioning is achieved by squeezing
the cartridge case rim between the barrel and the breach as shown in Fig. 6.4a.
A disadvantage with rimmed rounds is that the protruding rim causes weapon
feed problems. Also, when the breach block closes there is a hard impact
against the barrel. For rimless cartridges, seating the cartridge into the chamber
cone results in a longer distance between the breach block and the seating
point as shown in Fig. 6.4c. Rimless cartridges cause the fewest feed problems;
consequently this is the most widely used design for cases. An additional
advantage of this method of positioning the cartridge is that the impact by the
breach block is reduced when it closes. The use of a belted case, as shown in
Fig. 6.4b is a compromise between rimmed and rimless cases. The belt does not
protrude so much as the rim on a rimmed round but can still cause feed prob-
lems on automatic weapons. It is also expensive to manufacture.

Figure 6.4 Rimmed, belted and rimless cartridge cases

The distance between the breach block and the seating point of the cartridge
case in the chamber is known as the cartridge head space. To ensure that the
weapon functions correctly it is important that this dimension is correctly
maintained. Special gauges are used to measure and maintain the correct
dimension when the weapon is in service. If the cartridge head space is too
short the breech block will not fully close. If it is too long the cartridge will not be fully supported during firing and the case will be distorted, leading to extraction problems.

Extraction of the cartridge case may require excessive force depending upon the operating cycle of the weapon, case material, cartridge case taper and the surface finish of the chamber. To ease this problem some systems coat the cartridge case with oil or wax. However, this presents additional problems because dirt and debris are attracted to the chamber which cause a build up of material that foul the chamber.

The reliable extraction of fired cartridge cases is very important to the operation of automatic weapons. Another method of easing the problem is to provide the chamber with longitudinal grooves, known as fluting, which allow propellant gases to enter between the chamber wall and cartridge case and thus facilitate extraction of the cartridge case.

**The Forcing Cone**

The forcing cone is designed to provide smooth and continuous engraving of the bullet jacket by the rifling grooves so that deformation of the jacket takes place smoothly with no shearing of the driving material. When the jacket of the bullet is fully engraved the bore of the barrel is fully sealed. The forcing cone is the place of greatest wear in the barrel and has a major effect on the service life of the weapon. When the forcing cone is excessively worn the accuracy of the barrel decreases because of variations and reductions in muzzle velocity.

As wear increases, the forcing cone tends to move towards the muzzle and its steepness diminishes. The forcing cone dimensions are such as to allow loading of cartridges of maximum length (positive production tolerances) into the shortest cartridge chamber (negative production tolerances). There is usually a clearance between the jacket and the forcing cone which affects the forcing cone service life because propellant gases leak past the bullet and heat the forcing cone. Further forward displacement of the forcing cone caused by wear results in greater clearance and the rate of wear increases with the number of rounds fired. This effect can be reduced by choosing a narrow tolerance limit and greater steepness for the forcing cone. The taper of the forcing cone for automatic weapons is between 1:5 and 1:10 and for pistol ammunition it is usually 1:30.

**Rifling Profile**

With the exception of shotguns all small arms barrels are rifled, the purpose of the rifling being to impart spin to give the projectile gyroscopic stability. In cross-section the rifling appears as a number of lands and grooves as shown in

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**Figure 6.5 Rectangular rifling profile**

**Figure 6.6 Trapezoidal rifling profile**