THE MACHINE GUN

VOLUME IV, PARTS X AND XI

UNITED STATES NAVY

BUREAU OF ORDNANCE

THE MACHINE GUN

VOLUME IV

PARTS X AND XI

THE MACHINE GUN

Design Analysis of
Automatic Firing Mechanisms
and Related Components

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VOLUME IV, PARTS X AND XI

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PREFACE

Held to the strictest interpretation of the definition[1] and to the means by which the function is accomplished, there is only one primary force that actuates any automatic weapon: namely, the energy generated by the explosion of the powder charge contained in the chamber of the barrel. There have been, to date, only two known means that can be derived from this source of power that have resulted in successful operation: (1) the rearward thrust of the recoiling mass; and (2) pressure generated in the bore by the expanding gas of the progressive burning charge. The former is known as recoil actuation, while the latter is labeled gas operation. All known means used in making an automatic weapon complete a full cycle fall into this broad classification, whether the mechanics employed be reciprocating or rotary.

The recoil-operated type of weapon can be further broken down into two distinct classifications: short and long recoil. Gas operation, however, seems to have no limit in its application. For instance, the residual pressure remaining in the bore a few milliseconds after the projectile has cleared has been, for lack of a better term, called blowback, while in reality it is but another form of gas operation.

However, the most common method of employing the energy created by the gas of the exploding propellant is to tap the barrel and let the expanding gas be brought to bear on an actuating device such as a piston, lever, etc. The system is universally referred to as "gas operation", erroneously implying that this is the only way gas pressure is utilized as a source of power.

If one were satisfied only with generalities, it would be quite in order to state that there are only five known practical applications for accomplishing sustained fire as outlined in the definition of an automatic gun: (1) short recoil; (2) long recoil; (3) gas pressure in the bore bled off externally through an orifice (gas operation); (4) residual pressure remaining in the bore a few milliseconds after the projectile has cleared (blowback); and (5) blast energy generated by the expanding gases after being released from the confines of the barrel at the muzzle end (muzzle blast actuation).[2] These are considered the basic principles and from these simple variants of power, more than 3,000 patents have been issued since 16 June 1884 on operational features of machine guns.

So thoroughly have the gun designers of the past covered the subject that since World War I the individual was indeed skilled in his profession if he could even make an improvement on a feature that had already been in existence a long time, much less originate something that could rise to the dignity where it could truthfully be called an invention.

There is a tendency to use general terms too loosely in describing certain types of actions. For instance, the word "blowback" is invariably employed when describing any weapon that uses this form of actuation either wholly or in part. This unusual power supply has been exploited to such a degree that it takes at least four distinctly different classifications to cover the application of this method of utilizing residual pressure for completing a cycle of operation: (1) pure blowback (Bergmann); (2) retarded blowback (Schwarzlose); (3) delayed blowback (Scotti); and (4) advanced primer ignition (Becker). Each system is strictly adaptable to certain types of actions and is utterly impractical other than for a specific purpose. For example, the caliber .22 Colt Woodsman pistol uses *pure blowback* and is a well-balanced, highly efficient hand arm, and for this type of weapon such a method practically defies improvement. However, if this system were applied to a conventional 20 mm cannon, the bolt alone would need to weigh in the neighborhood of 380 pounds, with an approximate rate of fire of 200 rounds a minute, both of which would be totally unacceptable. If *advanced primer ignition* were used, the weight of the 20 mm weapon could be held to 90 pounds and

[1] AUTOMATIC MACHINE GUN—A weapon capable of sustained fire with its operating energy being derived wholly from the force generated by the explosion of the propellant charge.

[2] Although relatively unimportant, two other systems should be mentioned to complete the picture. "Blowforward" is a method in which the barrel is held to the rear by heavy spring pressure against a solid non-recoiling breech that supports the cartridge, gas pressure driving the projectile forward through the bore to move the barrel off the empty cartridge case. "The Gast system" is a double-barrel arrangement whereby the firing of one barrel furnishes the power to load, lock, and feed the other barrel, with unlocking being tied in with the first part of recoil movement.

the rate of fire would be about 600 rounds a minute, while if *delayed blowback* were employed, the overall weight of the gun would be slightly more (about 100 pounds), but the weight of the bolt or recoiling parts could be held to a bare minimum (6 pounds), and the rate of fire could be brought up to as much as 1,000 rounds a minute. On the other hand, the maximum elasticity of the conventional type cartridge case, coupled with the inevitable high chamber pressures of today, makes the timing factor too critical to permit any consideration of *retarded blowback* in the design of a large caliber automatic weapon. Such comparisons are limitless when based on the fundamental principles governing the design of automatic weapons.

It is the purpose of Volume IV of "The Machine Gun" to analyze these principles in such a way that the designer has before him at all times the minimum and maximum potentialities of any system of his choosing. Each and every one has its strong and weak points. No two are alike, but all have one thing in common. For every obvious virtue there is a hidden feature that is extremely critical and the designer who meets with any degree of success will still have to depend not so much on some clever way to utilize an unlimited power source but on his thorough knowledge of all systems so that he can work his way out of traps of his own creation.

The conventional 20 mm cartridge has been used arbitrarily in both the text and illustrations of this work as it represents the first step above rifle caliber and the starting bore diameter in automatic cannon design. It seems to be the most acceptable reference point for the designer regardless of the eventual caliber of his product. Likewise, emphasis has been placed on the airborne automatic weapon because its use under widely varied conditions requires a mechanism that must approach the ultimate of perfection. Having mastered its difficulties, the designer should find most of his other problems have been made relatively easy.

Part X of "The Machine Gun" provides detailed engineering and mathematical analyses of the basic sources of energy that set automatic weapons into activity. This information is the result of observations and practical experience accumulated over the years and translated by the mathematician into a theoretical yardstick for practical design. In Part XI the illustrator replaces the writer. In the chapters of this portion of the book, outstanding design features such as feed mechanisms, locking systems, revolver actions, accelerators, sears, extractors, ejectors, and other components are depicted for the guidance and stimulation of the designer. The sources for the drawings range from early patents and prototypes of actual weapons to theoretical designs of the future. These sources are intentionally omitted from the pages where the drawings are reproduced so that they may be considered on their merits without any conscious or unconscious influence by the fame or obscurity of the originators. Appendix A provides identification of these sources

Additional appendices contain an annotated bibliography listing source materials on machine guns and other automatic arms by topic and a similar subject listing of important patents of the 19th and 20th centuries, each with an abstract of its most outstanding features.

If this work can help in any small way to revive in this country the almost forgotten art of automatic weapon design, a field so thoroughly dominated by Americans in the past, the effort and toil that have been spent in preparing the book will have been well repaid.

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Having no precedent as a guide, Mr. James Fleming has performed an outstanding task in setting down the mathematical and engineering analysis of the basic systems of automatic fire. Pioneering in one of the most debatable fields of ordnance, he carried out this analysis from the practical, as well as the theoretical, side of ordnance, making use of field reports, research studies, personal observations, and actual physical participation in firing tests any time the apparent solution to the problem contained an element of doubt.

Mr. Edward Hoffschmidt, internationally known for his drawings and articles on firearms, has demonstrated great originality and resourcefulness in preparing the illustrations for this work. Despite their great volume, he has carried the same high quality and attention to detail into each picture. His unusual skill in bringing out features of automatic mechanisms of varying calibers and projecting them in a standardized form demonstrates his thorough knowledge of weapon design.

One cannot overestimate the importance of the part played by Master Sergeant George McGarry, USMC, in the completion of this publication. Sergeant McGarry's task in disassembling practically every known automatic firing mechanism and studying it in order to highlight its outstanding features was performed in a skillful manner that reflects the quality of his extensive background in automatic weapons.

The text has been edited by Mr. Franklin W. Clark, who displayed not only professional competence in the technical details of organizing a complex manuscript into readable form, but a unique ability in coordinating controversial subject matter varying from the elementary to the technical in a manner that minimizes the inevitable questions that arise.

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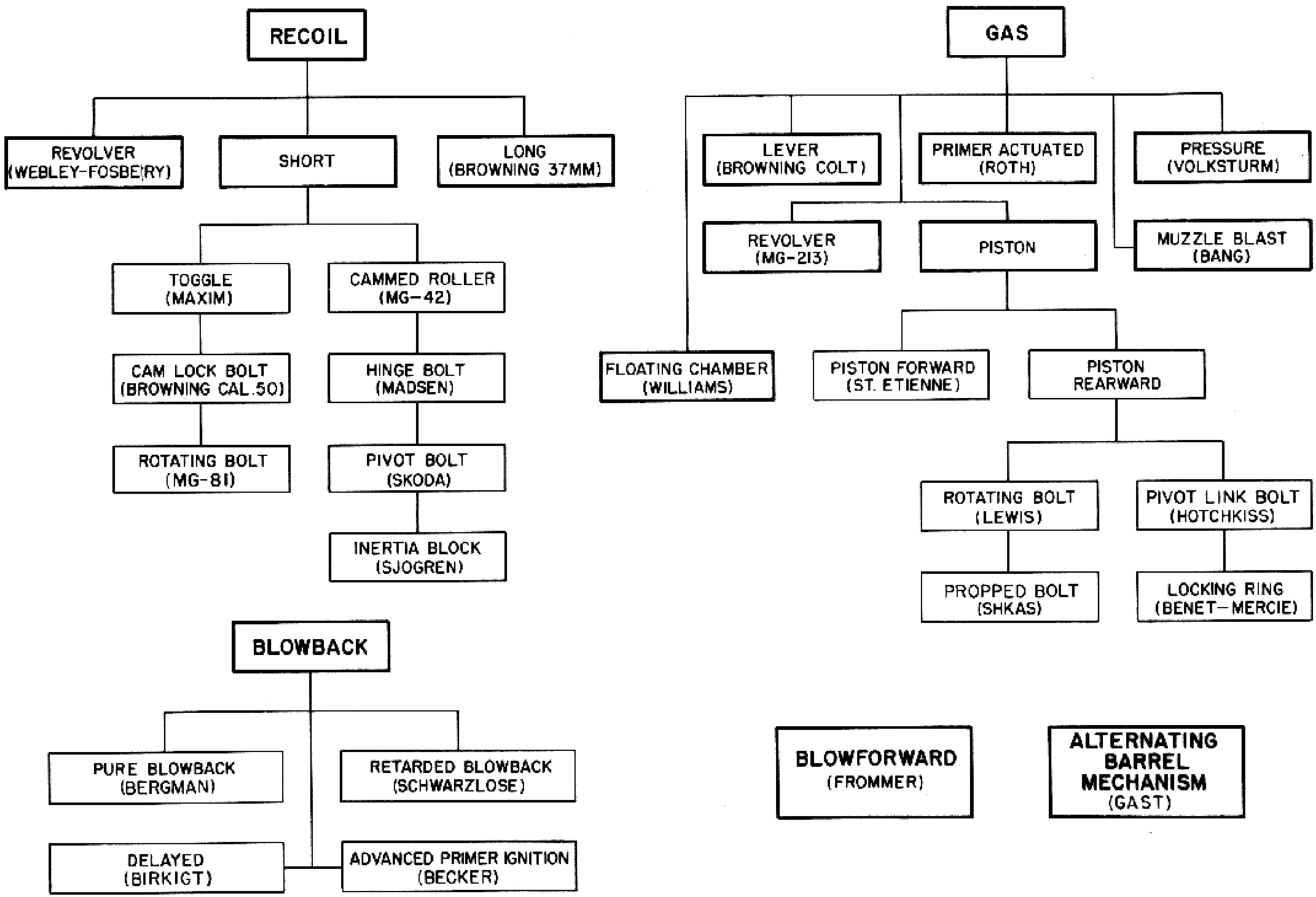
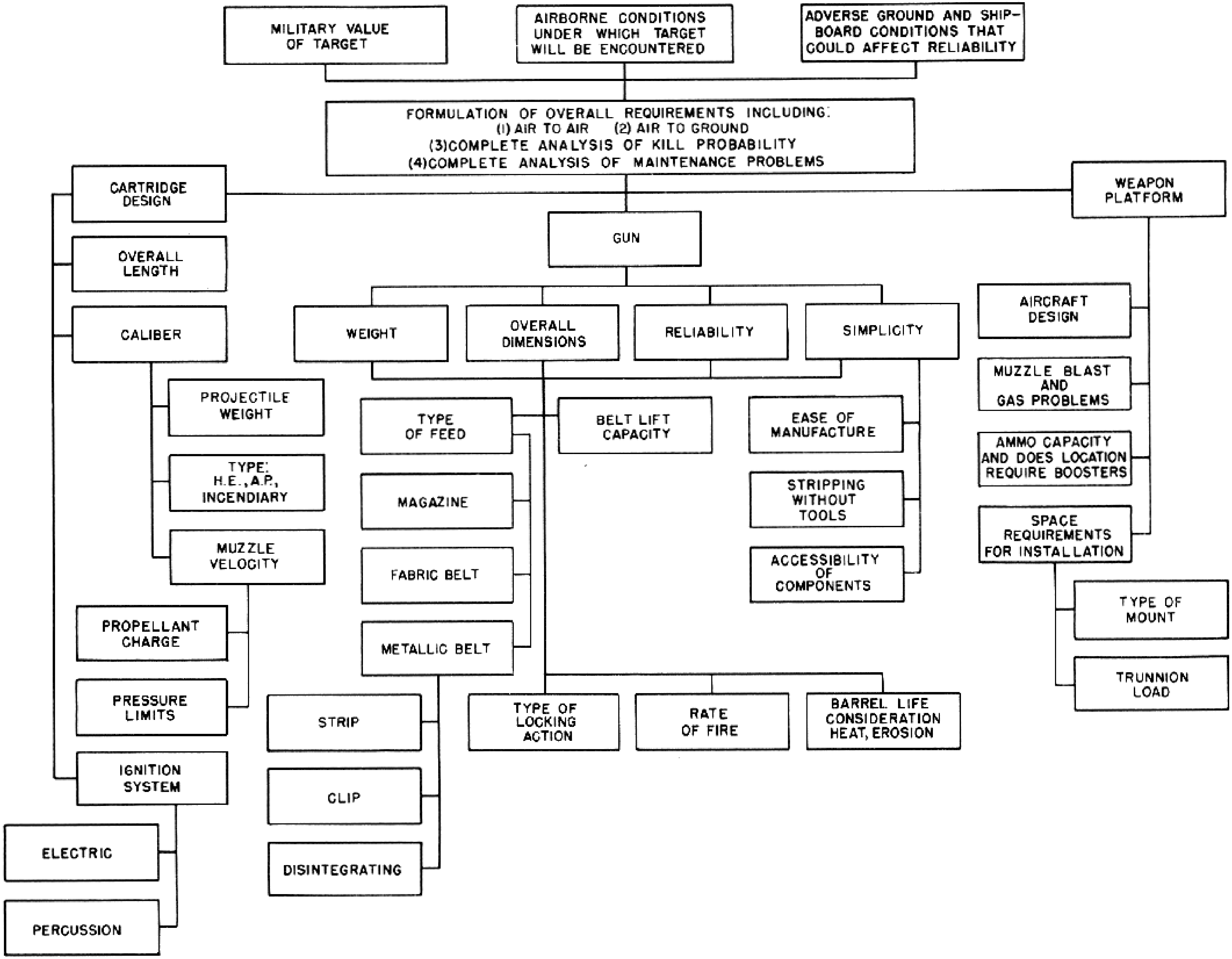


FACTORS THAT DICTATE TYPE OF SYSTEM USED IN DESIGN OF AN AIRBORNE AUTOMATIC WEAPON



PART X

ANALYSIS OF SYSTEMS

Chapter 1

BLOWBACK

BLOWBACK OPERATION

In the design of any gun, great care must be exercised to be sure that the cartridge case is adequately supported to withstand the tremendous rearward thrust exerted by the powder gases. In many guns, this is accomplished by providing locking devices of great strength and positive action which hold the bolt rigidly closed until the pressure of the powder gases has dropped to zero or at least has decreased to an operating limit at which the breech may be opened safely. However, absolutely rigid support is not mandatory and some movement of the bolt and cartridge case is permissible, providing that the movement is controlled in accordance with definite principles. This controlled movement can then be utilized as a source of energy for automatic operation of the gun mechanism.

The system of operation in which the power required for operating the gun mechanism is derived from the motion of the cartridge case as the case is thrust to the rear by the pressure of the powder gases is called for lack of a better name the "blowback" system. In some guns (which invariably must employ low-powered ammunition) all of the energy used for performing the entire cycle of operation is derived from plain blowback; in other guns, the energy derived from blowback may perform only certain functions in the cycle or may merely supplement the operational energy derived from another system of automatic operation.

In the larger sense, blowback might well be considered to be a special form of gas operation. This is reasonable because the cartridge case may be conceived of as a sort of piston driven by the powder gases.

Actually, blowback involves so many special problems that it is best considered to be in a class by itself. The question of whether or not it should be included within the more general classes of gas operation or recoil operation is purely academic. The important point is that it partakes of some of the properties of both classes and, depending on the particular problem at hand, may be considered to be in either one.

The distinguishing characteristic of a blowback weapon is that the cartridge case must move under the direct action of the powder gas pressure. Thus, any gun in which the bolt is permitted to move while there is pressure in the chamber will be subject to some blowback action. The extent to which blowback is utilized depends on the manner in which the bolt movement is controlled and on what proportion of the operating energy is derived from other systems of operation. The major problem encountered in blowback operation is the problem of controlling the bolt movement so that the motion of the cartridge case is kept to a bare minimum (based on the ultimate strength of the case) during the action of the tremendous gas pressures existing until the projectile leaves the muzzle of the gun. This is necessary in order to maintain effective sealing and also because excessive motion under peak pressures can result in separation or rupture of the cartridge case.

Behavior of the Cartridge Case

The most important single factor to be considered in analyzing blowback operation is the behavior of the cartridge case as it is affected by the pressure of the powder gases. Since this primary factor is so critical and controls all of the basic design features which distinguish blowback operated guns, it will be described fully in the following paragraphs before any further discussion of the details of blowback systems. (Most of the numerical values mentioned in this description apply to a 20-mm gun and typical 20-mm ammunition. This caliber has been selected for purposes of illustration because it is representative of high-powered heavy machine guns.)

When a gun is fired, the explosion of the propellant charge results in the generation of exceedingly high pressures which vary with extreme rapidity. (See fig. 1–1.) Although the curves for

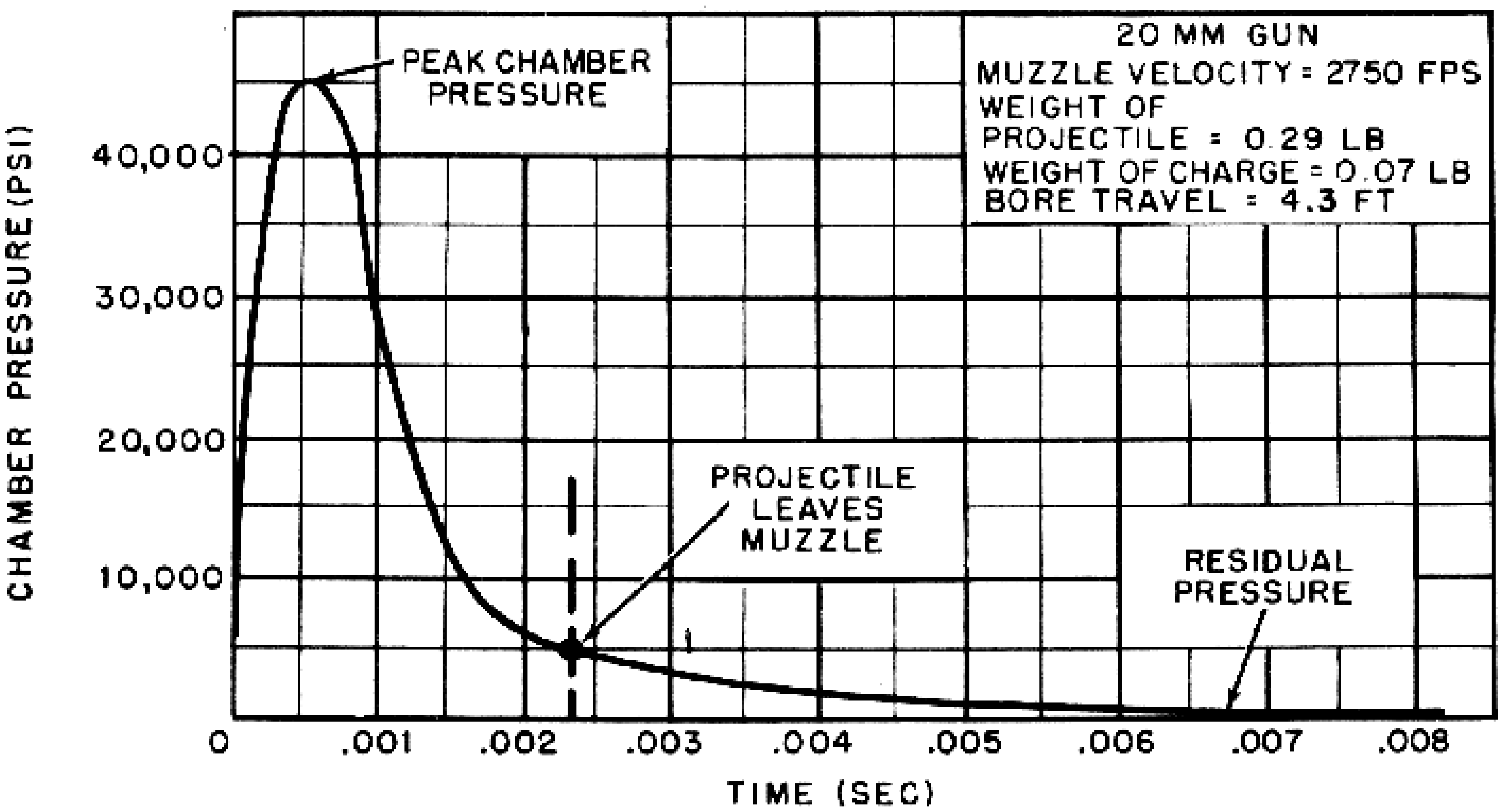


Figure 1–1. Variation of Chamber Pressure With Time.

varying types of 20-mm ammunition, when used in different types of guns, are not all exactly the same, the values shown by the curve in the figure are more or less typical of modern ammunition of this caliber. As shown in fig. 1–1, the gas pressure builds up to a peak of 45,000 psi within 0.0005 second after ignition of the primer, then (for the particular barrel length cited in fig. 1–1) decreases until it is 5000 psi when the projectile leaves the muzzle at 0.0023 second, and finally falls off quickly to zero as the gases leave the muzzle. The pressure which exists during the small fraction of a second after the projectile leaves the muzzle is called the "residual" pressure. This pressure may be considered to decrease exponentially with time and to be practically zero within 0.008 or 0.009 second after ignition of the primer.

The gas pressure acts uniformly in all directions against the inside of the cartridge case as shown in fig. 1–2. The radial components of the pressure act on the walls of the case to expand the case against the walls of the chamber, thus creating a seal which prevents the escape of the powder gases to the rear. (It should be realized that the radial forces acting on the case walls are of tremendous magnitude. In fact, the walls of the case are crushed so tightly against the chamber walls and the temperature in the chamber is so high that there is almost a tendency for the case to be spot-welded to the chamber.)

The net sum of the axial pressure components acting against the cartridge case creates a force which tends to drive the case to the rear against the resistance offered by the bolt. When the case

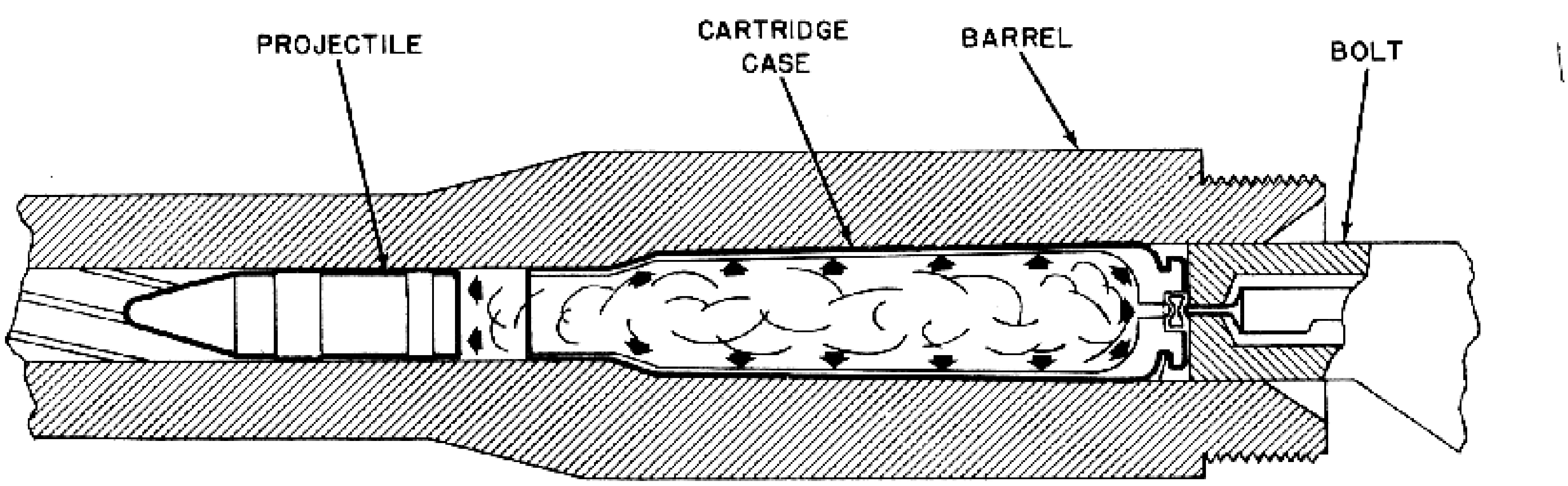


Figure 1–2. Pressure in Chamber.

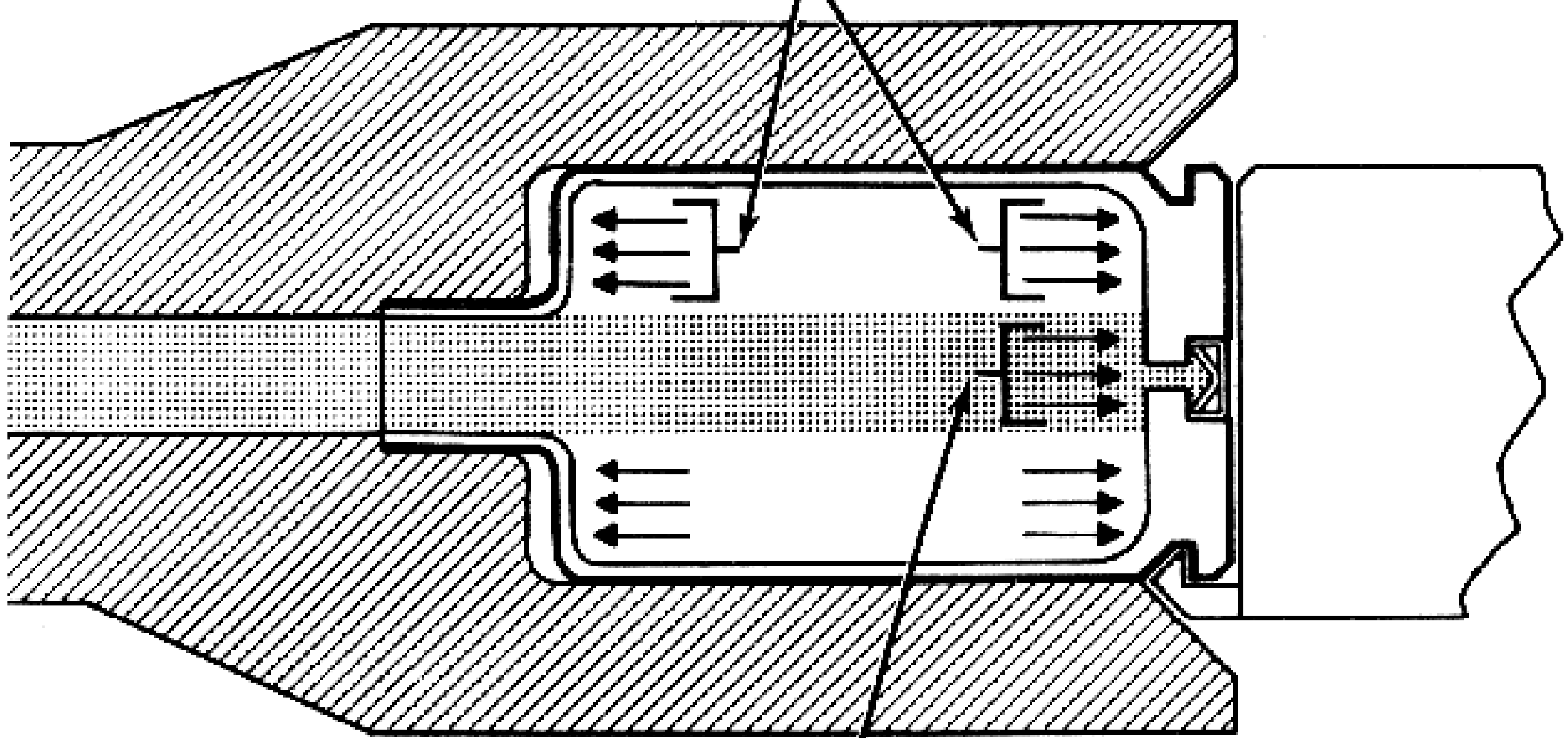


Figure 1–3. Effects of Axial Pressure.

is free to slide in the chamber, this net force is equal to the chamber pressure multiplied by the total cross-section area of the mouth of the case (not the projected area of the inside of the base). (See fig. 1–3.) If the projected area of the base of the cartridge case is greater than the area of the mouth, the force on the base is of course greater than the net force. However, the difference between the forces is cancelled by the axial component on the tapered or necked portion of the case. (This component is directed forward.) Note that if the entire case were free to move, the difference between the net force and the force on the base would cause a tension in the walls of the case.

If the bolt were rigidly locked and held firmly against the base of the cartridge case (that is, if there were zero excess head space), the gas pressure would merely compress the material of the cartridge case against the chamber walls and bolt face. However, with blowback operation, the case and the bolt must be free to move at some predetermined time during the action of the gas pressure so that the required energy can be transmitted to the bolt. For purposes of examining the effects of this movement, it will be assumed that the bolt is not locked at any time and that it resists movement only by its inertia. (As will be pointed out later, these conditions do not apply for all blowback weapons.) The movement of the cartridge case can be considered to occur in the following three phases:

Phase 1. The pressures which exist during the first 0.0001 second of the propellant explosion are relatively low but are sufficient to expand the thin brass near the mouth of the cartridge case against the chamber wall, thus forming the seal which prevents the powder gas from escaping to the rear. Since the radial pressure is not extremely high during this phase, the friction between the cartridge case and the chamber is not excessive. Therefore, the axial component of the pressure causes the entire case to stretch or slide back slightly in the chamber so that it first takes up any excess head space which may exist and then starts to impart motion to the bolt. Because of the high inertia created by the mass of the bolt, the velocity of this motion is relatively low.

NOTE: If the excess head space is very large, it may not be taken up entirely before the chamber pressure builds up to a high value. In this event, the conditions described in Phase 2 are applicable.

Phase 2. The second phase of the cartridge case movement occurs during the period of extremely high pressure on either side of the peak shown in fig. 1–1. The behavior of the cartridge case during this phase depends entirely upon whether or not the case is suitably lubricated.

If the cartridge case is entirely unlubricated or is insufficiently lubricated, it is expanded heavily against the chamber walls, producing a high-pressure metal-to-metal contact which results in a very high friction between the case and the chamber. Because of the friction between the walls of the case and the chamber, the pressure acting against the base of the case will result in a tensile stress in the walls of the case. In fact, the forward portion of the case may be gripped so tightly during the period of high chamber pressure that the frictional resistance exceeds the tensile yield strength of the case walls. In this event, the forward portion of the case sticks to the chamber wall but the rear portion continues to move, causing the case to stretch plastically. If the bolt does not provide sufficient resistance to prevent the stretching of the case from exceeding the allowable elongation of the case material (about 0.015 inch), the case will separate.

At this point, special mention should be made of what effect excessive head space would have if an attempt were made to use unlubricated ammunition. As pointed out in Phase 1, the motion of the case may not take up all the head space before the chamber pressure builds up to a value high enough to cause the forward portion of the cartridge case to stick. Assuming that some excess head space still remains, the base of the case will be unsupported and the high pressure will cause the case to be stretched. If the excess head space is so large that it is not all taken up before the stretch exceeds the allowable elongation of the case material, separation will occur.

NOTE: It should be realized that the forces produced by the peak chamber pressures are vastly in excess of the strength of the thin brass of the cartridge case. With forces of this magnitude, the case stretches quite easily. This may be illustrated as follows: With the forward portion of the case stuck, the force tending to stretch the case is equal to the pressure multiplied by the area of the inside of the base. Since the maximum chamber pressure for a 20-mm cartridge is 45,000 psi and the inside base area is approximately 0.5 square inch, the stretching force is in the neighborhood of 22,000 pounds. The area of metal in tension for the case wall may be approximately 0.1 square inch and the ultimate strength of the half-hard brass of which the case is made is about 50,000 psi. Therefore, the maximum resistance which the case can offer to stretching and separation is only approximately 5000 pounds. It easily can be seen that this resistance is practically negligible when compared to the stretching force of 22,000 pounds. In fact, forces sufficient to produce separation can occur whenever the chamber pressure is over 10,000 psi, and as shown in fig. 1-1, such pressures exist for almost the entire time the projectile is still in the bore. (Note that the force expended in overcoming friction or in stretching the case reduces the force applied to the bolt. If this effect is excessive, the gun may fail to continue firing because the energy transmitted to the bolt is insufficient to operate the gun mechanisms).

The preceding description applies to the condition in which the case is either entirely unlubricated or insufficiently lubricated. If a fairly thick film of a suitable lubricant is applied between the cartridge case and the chamber wall, an entirely different condition results. It should not be thought that the purpose of the lubricant is merely to make the cartridge case "slippery". Its true purpose is to form a continuous film which remains between the case and the chamber wall and effectively "insulates" their surfaces from metal-to-metal contact, even under extremely high chamber pressures. Since there is no metal-to-metal contact to cause seizing and sticking, the case slides freely in the chamber and the only resistance to its movement (except for the resistance offered by the bolt) is the force required to produce shear in the lubricant film.

According to established laws regarding friction, the frictional resistance for well-lubricated, smooth metallic surfaces is relatively very low and is practically independent of the pressure between the surfaces. The major considerations are the area of the lubricant film in shear, the viscosity of the lubricant, and the speed of the relative movement between the surfaces. Thus, a well-lubricated cartridge case will move almost as freely under high chamber pressure as under low pressure and therefore proper lubrication should completely eliminate seizing of the cartridge case, which is the basic cause of case separation.

Although proper lubrication will eliminate seizing of the cartridge case, other difficulties can occur if the case is permitted to move too far while the chamber pressure is extremely high. Excessive movement will result in a condition where the rear portion of the case is out of chamber and will therefore receive no radial support from the chamber walls. The high internal pressure may then cause the case to swell at the base portion or even to rupture. Furthermore, if in the chamber, the seal between the case and chamber may be broken thus permitting the escape of hot powder gases at the breech. (In most rapid-firing blowback guns some escape of gases at the breech is inevitable, but if this effect is excessive it can cause damage to the breech mechanism or may even be dangerous for operating personnel.)

It is of interest to note a special difficulty which can occur with ammunition having a cartridge case of large diameter when compared to the projectile diameter (bottle-necked or strongly tapered case). With this type of case, the internal pressure tends to produce high tensile stresses in the case walls. (See fig. 1–3.) If the case is bottle-necked, the rearward movement of the case creates a space between the shoulder of the case and the chamber. Since this leaves the forward portion of the case unsupported, the internal pressure tends to deform the case by pushing the shoulder forward to fill the space created by the movement. If the case is of the tapered type, the rearward movement tends to create a gap between the wall of the case and the chamber. Since this leaves the wall of the case unsupported, the internal pressure expands the case to close the gap.

Under either of the conditions described above, the deformation of the case can be of considerable magnitude if the movement of the bolt is excessive and it is quite possible that the deformation will exceed the allowable limit of the brass. This may cause the case to tear or rupture in such a way as to cause a separation or to cause difficulties in extraction or ejection. For this reason, plain cylindrical cases or cases with only a slight taper are to be preferred for use in blowback guns.

Phase 3. The final phase of the cartridge case movement starts when the chamber pressure has decreased to a level which permits the case to contract, thus reducing the friction to a negligible value. After this point, the remaining pressure continues to drive the case to the rear but the forces on the case are low enough so that there is no danger of the case failing by separation or rupture. This phase ends as the gas pressure approaches zero (0.008 or 0.009 second after ignition of the primer). Although at this point the driving force is reduced to zero, the case and bolt continue to move of their own momentum with sufficient kinetic energy to complete the extraction and ejection of the case and to operate the gun mechanisms for the remainder of the automatic cycle.

Conclusions

All of the points discussed in the preceding description of the behavior of the cartridge case in a blowback gun may be summarized in one very important principle which controls all of the characteristics and fundamental design requirements for this type of weapon. This principle may be stated simply as follows:

> THE PRIMARY DIFFICULTIES IN BLOWBACK OPERATION ARE THE DIRECT RESULT OF EXCESSIVE CARTRIDGE CASE MOVEMENT DURING THE PERIOD OF EXTREMELY HIGH CHAMBER PRESSURE AND THESE DIFFICULTIES ARE AGGRAVATED BY INADEQUATE CASE LUBRICATION.

In the design of blowback guns, each factor mentioned in this statement must be considered carefully in order to avoid operational difficulties. This analysis leads to the following general conclusions.

Chamber Pressure. All of the difficulties that arise are the direct result of extremely high chamber pressure. Therefore, it follows that if the chamber pressure could be kept low, these difficulties would disappear immediately. Unfortunately, high chamber pressures are essential in high-powered heavy machine guns (which are the main concern of this publication) and accordingly, the use of low chamber pressure cannot be considered as a solution to the design problem. (The relative ease with which the blowback principle can be applied to low-powered small-caliber ammunition is amply demonstrated by the large number of self-loading pistols, sub-machine guns, and light machine guns that have used this principle successfully.)

Case Movement. The major problem confronting the designer of a high-powered blowback gun is how to limit the movement of the cartridge case during the action of the extremely high pressures resulting from the explosion of the propellant charge. This problem and the design difficulties related to it are the direct result of a number of conflicting requirements.

Since, in blowback weapons, the source of the power for operating the mechanism is derived from the thrust applied to the cartridge case by the powder gases, it is essential for the cartridge case to move while the gas pressure is acting. However, it is this very motion under pressure which causes separations and other damage to the cartridge case. When steps are taken to reduce these difficulties by limiting the movement of the cartridge case, care must be exercised to insure that sufficient energy will be available to operate the gun effectively at the desired rate of fire. Furthermore, if adequate lubrication of the cartridge case is provided so that a fairly high bolt velocity can be tolerated during the period of high chamber pressure, it may be found that the resulting rapid extraction of the case will cause rupture of the case or cause the breech seal to be broken too soon.

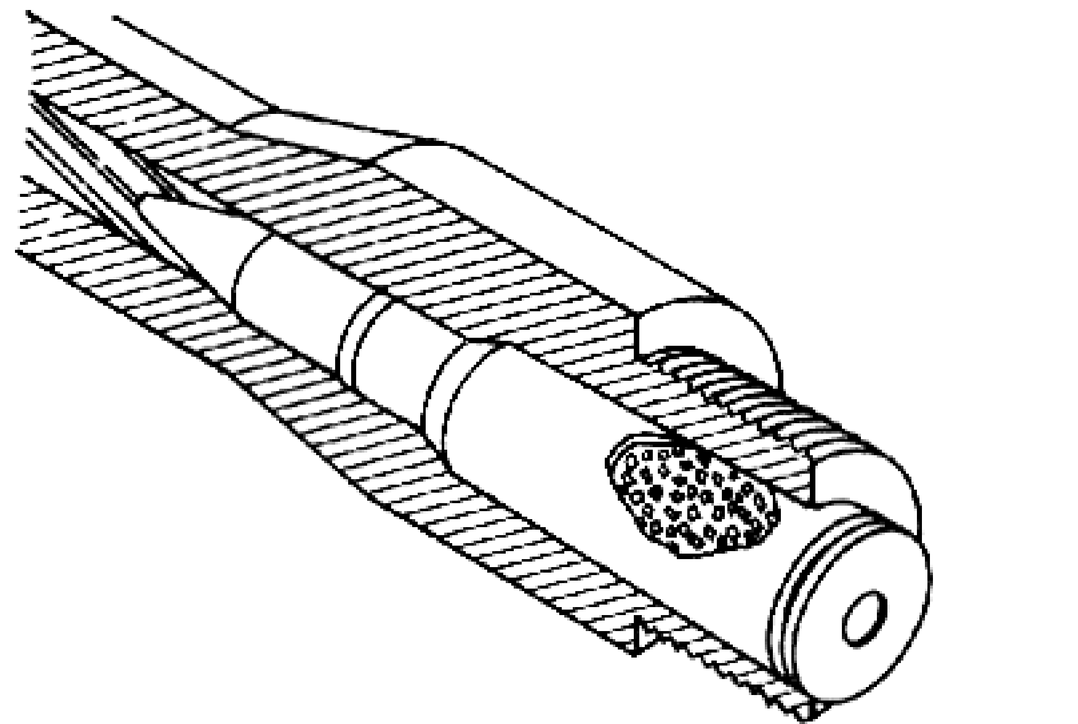
The foregoing considerations and other points which will be discussed later cause the design of a blowback gun to be a problem in balancing various critical factors, one against the other, in order to obtain the required performance characteristics. There are many possible solutions to this design problem and the particular solution employed is what determines the basic features of the gun. A detailed explanation of how these solutions are applied in practice is given later under the heading "Blowback Systems".

LUBRICATION. The importance of providing suitable lubrication for high-powered ammunition used in blowback guns cannot be emphasized too strongly. Experience has shown that without proper lubrication, difficulties such as case separation, chamber seizure, loss of bolt recoil energy and poor extraction seem to make it impossible to attain high performance in a blowback gun. In fact, so essential is lubrication to this type of weapon that the term "oil-omatic" has been suggested as being more suitable than "automatic" for use in referring to blowback machine guns.

It is very important to realize that under practical operating conditions, factors may be encountered which make it very difficult to maintain suitable lubrication for ammunition. For example, guns and ammunition used in arctic operations or carried by aircraft flying at very high altitudes can be subjected to extremely low temperatures that can cause ordinary lubricants to fail completely. It also should be remembered that greasy substances applied to the ammunition or chamber wall easily pick up sand or other contamination that can cause the lubricant film to be broken and cause the cartridge case to seize in the chamber. This problem is particularly serious for any gun which is to be used in the field. All things considered, one of the first things that the designer of a blowback gun must do is find an ammunition lubricant which is suitable both for its intended purpose and for the operating conditions to which the gun will be subjected. If such a lubricant can not be found, it is safe to say that blowback operation will not be practical and should be abandoned in favor of some other system of operation in which cartridge case lubrication is not of such critical importance.

Although the term "lubricant" usually brings to mind an oil or grease, there are many other substances which can qualify (at least from the theoretical point of view) as ammunition lubricants. This is so because the only really important requirements for an ammunition lubricant are that it must form a continuous insulating film and that the force required to produce shear in the film must be smaller than the force required to produce separation of the cartridge case. It can be seen readily that in order to satisfy these requirements, a substance need not have any of the unctuous or "slippery" character usually associated with lubricants used for minimizing friction.

The fact that it is relatively easy to think of almost any number of substances which could satisfy the theoretical requirements might lead one to believe that it would be a small problem to find an ammunition lubricant suitable for any and all conditions. The sad truth of the matter is that such a substance has been sought eagerly but unsuccessfully ever since 1898. In this quest of over half a century, designers, inventors, research groups, and just plain practical men have suggested an amazing number of materials which seemed to have possibilities. The list includes waxes, graphite mixtures, a variety of liquids, and solid coatings, all of which are so numerous that recording them here would take too much space. However, controlled tests and the hard lessons of practical experience have eliminated them all.



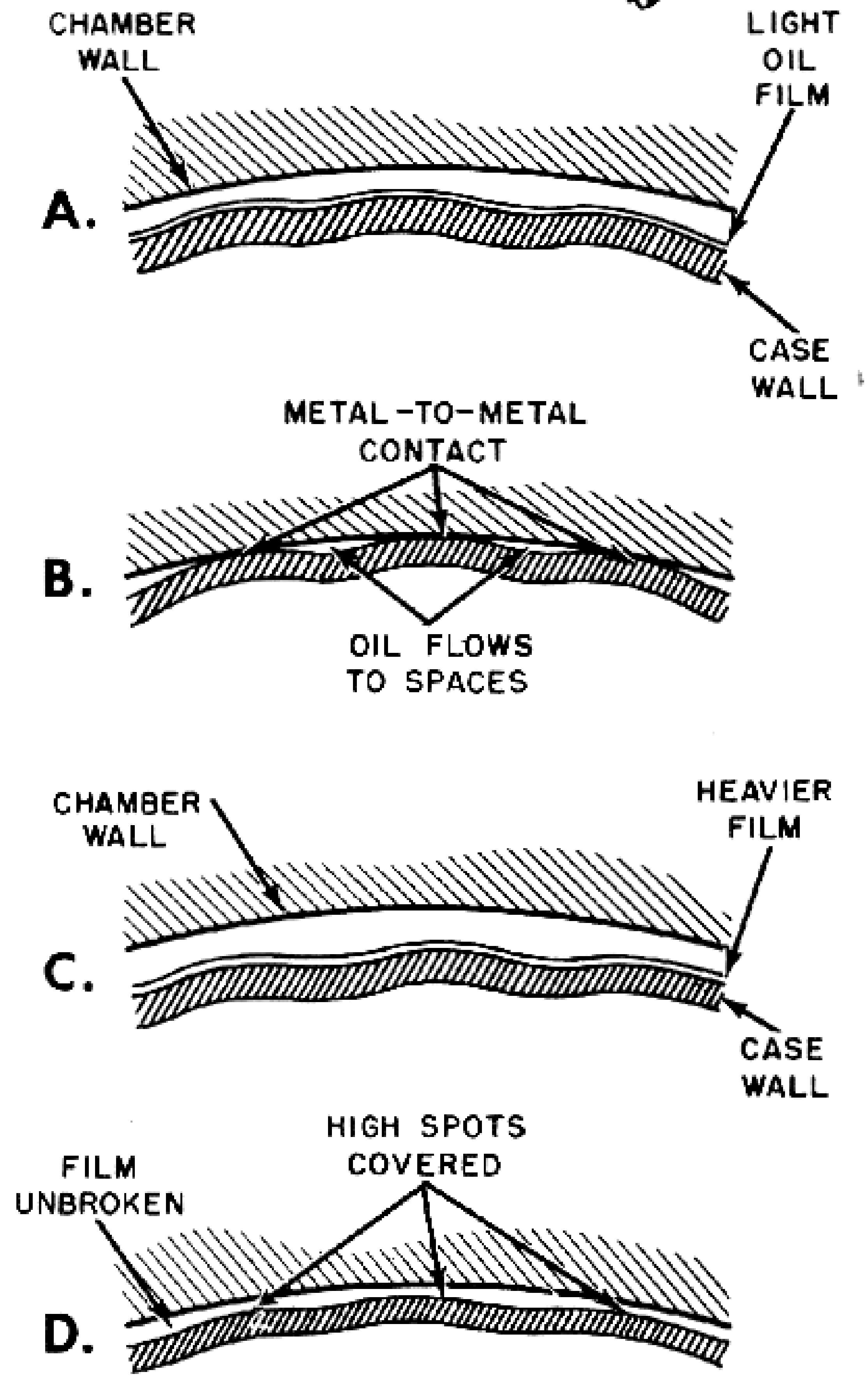


Figure 1–4. Effects of Light and Heavy Lubricant Films.

Other more radical attempts to avoid case seizure through the use of such devices as chromed, fluted, or stepped chambers have met with a similar lack of success. These suggested materials and remedies have failed because they all amounted to stepping out of one difficulty into another. For example, one coating which seemed otherwise satisfactory was scraped off the ammunition by the feeder in the form of small shavings that soon gummed up the breech mechanism. Other substances produced excessive fouling in the chamber, and so on.

The result of all this experimentation is that today, as in 1898, heavy-bodied oil and grease, with all their attendant difficulties, are still the only substances which have been used to lubricate ammunition with any reasonable degree of success. Either of them will produce a good film which will provide the required insulating effect and under many conditions of operation the difficulties encountered in their use are not too serious.

It should be remarked here that light oil, such as sewing machine oil, is not a satisfactory lubricant for ammunition. It may be argued that even light oil can form a film and is just as incompressible as heavier oil or grease and therefore should serve as well to provide the required insulation. However, practical experience has shown that the use of light oil is not effective in preventing case separations. Apparently, the difficulty with light oil is that a sufficiently thick film can not be maintained. As shown in fig. 1–4A, the light oil will form a film which covers the entire cartridge case, but there are bound to be small irregularities in the cartridge case wall. When the case is expanded in the chamber by the pressure of the powder gases, as shown in fig. 1–4B, the light oil will be caused to flow off the high spots into the low spaces and since the film is so thin, there will not be sufficient oil to over-fill these spaces. Therefore, all the oil will be forced off the high spots, thus permitting metal-to-metal contact to occur at these spots. On the other hand, a sufficiently thick film of a heavier bodied lubricant can be maintained so that even though there is some flow into the low spaces, there is still enough lubricant to cover the high spots (figs. 1–4C and 1–4D).

There is another advantage to the use of heavier bodied lubricants. In any gun, the exploding powder gases produce a flame which escapes around the cartridge case during the extremely brief instant required to expand the neck of the case against the chamber. (This effect is especially noticeable when the rotating band of the projectile is larger in diameter than the neck of the case. With this condition, the case neck must be expanded considerably before the seal is made with the result that the escaping flame will be longer in duration.) The escaping flame tends to burn off or dry up the lubricant applied to the case. Light oil is particularly subject to this burning or drying action. The thicker films formed by heavier oils or grease seem not only to stand up better but also to provide a seal which helps to limit the escape of the flame.

THE CARTRIDGE CASE MUST BE THOROUGHLY LUBRICATED FOR THE ANALYSIS GIVEN IN THE PRECEDING PARAGRAPHS TO APPLY. WHEN A THICK FILM OF OIL EXISTS AROUND THE NECK OF THE CARTRIDGE, THE GAS PRESSURE IS NOT SEALED OFF FROM THE OUTSIDE OF THE CASE BUT IS TRANSMITTED HYDRAULICALLY BY THE OIL FILM. THIS CONDITION MAKES THE ENTIRE CARTRIDGE CASE ACT AS A HYDRAULIC PISTON AND PRODUCES A REARWARD FORCE EQUAL TO THE BREECH PRESSURE TIMES THE CROSS-SECTION AREA OF THE CHAMBER AT ITS LARGEST DIAMETER. THEREFORE, WITH THE USE OF A HEAVY-BODIED LUBRICANT, THE INITIAL FORCE DRIVING A NECKED CARTRIDGE CASE TO THE REAR IS CONSIDERABLY GREATER THAN IT WOULD BE FOR AN UNLUBRICATED CASE, MAKING ADEQUATE LUBRICATION ABSOLUTELY NECESSARY IN ORDER TO OBTAIN CONSTANT PERFORMANCE. THIS IS ONE OF THE REASONS THAT THE PRESENCE OF OIL ON THE AMMUNITION IS VITAL FOR WEAPONS EMPLOYING ANY FORM OF BLOWBACK.

CARTRIDGE CASE. In many instances when a gun is designed, the designer is required to use a particular type of available ammunition or at best he finds it possible to choose one of several available types. It rarely happens that a new type of ammunition can be developed to suit the special requirements of a new gun design. In other words, the characteristics of the ammunition are usually not under the control of the gun designer and he has to do the best he can with what ammunition is available to him.

Although the gun designer usually has little or no control of the ammunition he must use, there may be occasions when he can exercise some choice in selecting some of its characteristics. For blowback weapons, there are definite advantages to be gained from selecting ammunition with certain desirable cartridge case characteristics. These characteristics may be outlined as follows:

1. The case should be made as strong as possible in order to minimize the possibility of separations or other damage. One way to accomplish this is to provide a relatively great thickness of metal at the points where the case is likely to fail. Another method is to use a strong material in fabricating the case. To illustrate this point, the half-hard brass from which cases are usually made may have an ultimate strength in tension of approximately 50,000 psi while other materials, depending on the type used, may have an ultimate strength two or three times greater than this value.
2. Other factors influence the choice of the case material. The material should be highly elastic so that it can be expanded considerably by the chamber pressure without undergoing plastic deformation. If the material has this property, the case will more readily shrink back to its original dimensions when the chamber pressure drops to a low value, thus facilitating extraction. In addition, the material should have a large allowable elongation for suddenly applied loads so that the case can be stretched considerably without separating.
3. In order to minimize deformation of the cartridge case, the case should be nearly cylindrical with only a slight taper and little if any bottle neck. The reasons why taper and bottle neck should be avoided have been explained previously under PHASE 2 of the behavior of the cartridge case.
4. The finish and dimensions of the case should be such as to minimize frictional resistance to the movement of the case. The case should be smooth and free of scratches or other irregularities which would tend to impair the efficiency of the lubricant by causing discontinuities in the lubricant film. Also, the case should be as small as possible in order to keep the area of the case walls to a minimum. Then for any given chamber pressure, the friction forces on the case will be minimized.

Evaluation of the preceding characteristics will reveal that the attainment of one desirable characteristic may make it difficult or impossible to attain one or more of the others. Furthermore, it may happen that the attempt to obtain qualities which make the cartridge more suitable for use in a blowback weapon may introduce other problems such as poor ballistic performance or difficulties in manufacture. Nevertheless, all of these characteristics should be considered carefully in the selection of an existing cartridge for use in a blowback weapon or in setting up the requirements for a new cartridge. Close attention paid to the problem of obtaining the most suitable cartridge case may spell the difference between success and failure for a new gun design.

BLOWBACK SYSTEMS

Blowback is defined as a system of operation in which energy used for providing power for the gun mechanism is obtained from motion of the cartridge case as the case is pushed rearward out of the chamber by the pressure created by the explosion of the powder charge. This thrust on the cartridge case is a direct result of the total reaction to the forward thrust applied in moving the projectile and the expansion of the powder gases.

In some guns, all of the energy required for the performance of the automatic cycle is obtained through blowback action while in others only a portion of the required energy is obtained through blowback, the remainder being derived from some other system of operation. In any event, the blowback effect is present, at least to some extent, whenever the bolt of a gun is not locked while there is powder gas pressure in the chamber.

When blowback action occurs, the energy derived from the pressure of the powder gases appears in the form of kinetic energy transferred to the bolt mechanism, or in other words appears in the form of a velocity imparted to the bolt mass. The basic problem involved in blowback operation is in controlling this velocity so that the gun will operate as desired. There are many methods by which the control of the rearward motion of the bolt may be accomplished in guns employing blowback and these various methods are referred to as blowback "systems".

In the following pages the various systems used for controlling the bolt velocity are described and analyzed by considering the sequence of events during the automatic operating cycle. This analysis is concerned only with the functioning of the mechanisms and with the general factors affecting design.

PLAIN BLOWBACK SYSTEM

The problems encountered in controlling the bolt velocity in a blowback gun can be appreciated best by analyzing the so-called "plain blowback system". In this system, blowback provides all of the operating energy and the motion of the cartridge case is limited only by the inertia of the bolt.

NOTE: As will be explained, the plain blowback system is not suitable for application in a high-powered heavy machine gun. It is treated here because it serves to illustrate many of the basic principles involved in other more successful variations of blowback.

Although the actual form of the mechanism for a particular gun of this type may differ considerably from that shown in fig. 1–5, the mechanism shown in the figure is representative of the type from the standpoint of function. The mechanism consists essentially of the bolt (which backs up the cartridge case and is free to slide in the breech casing) and the driving spring (which absorbs the kinetic energy

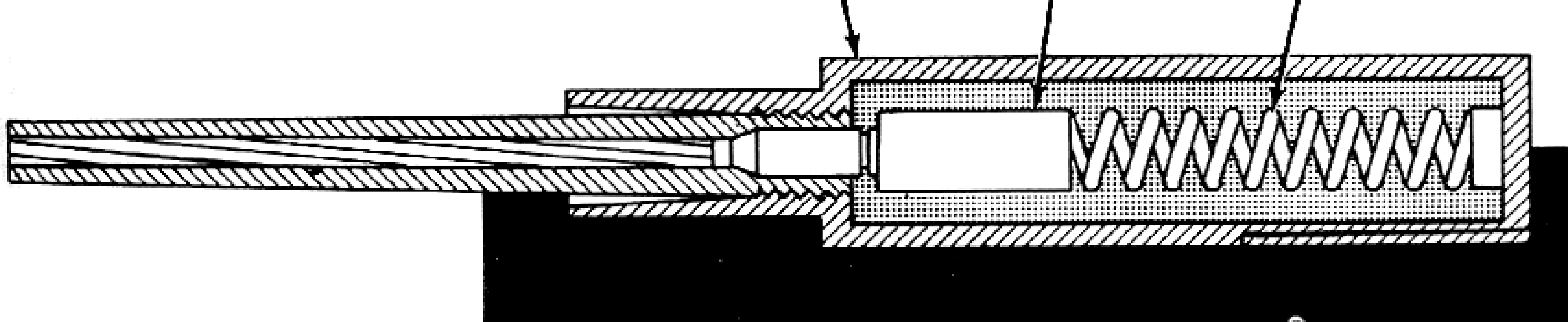


Figure 1–5. Simplified Schematic of Blowback Mechanism.

of the bolt when the bolt is blown back and then drives the bolt back to the firing position).

Cycle of Operation

The automatic cycles of operation for guns employing simple blowback are all more or less the same and occur as follows:

The cycle starts when the bolt is driven against the base of the cartridge in the chamber, firing the round. When the cartridge is fired, the pressure of the powder gases drives the projectile through the barrel of the gun and at the same time immediately drives the cartridge case to the rear against the resistance offered by the bolt. At this point, the only significant resistance to the acceleration of the bolt is due to the inertia of the bolt mass.

The force exerted by the powder gases exists for a relatively very short time. In a typical 20-mm gun with a total barrel length of slightly over five feet, the projectile leaves the muzzle after approximately 0.0023 second and the residual gas pressure continues to act for another 0.008 or 0.009 second. Thus the bolt is subjected to accelerating forces only during the first 0.008 or 0.009 second after ignition of the primer. After this point, the powder gas pressure is zero and the force driving the bolt back is also zero but the case and bolt continue to move of their own momentum. As the bolt moves back, the case is extracted from the chamber and ejected. The bolt slows down as its motion compresses the driving spring until the resistance of the spring combined with the action of the buffer has decreased the bolt velocity to zero. At this point, all of the kinetic energy originally imparted to the bolt (except for losses resulting from extraction, ejection, and friction) is stored in the spring as potential energy.

The driving spring then pushes the bolt forward. As the bolt moves forward, it cocks the firing mechanism, picks up a fresh cartridge from the feed mechanism, and carries this cartridge into the chamber. Just before the bolt reaches its fully forward position, the potential energy stored in the driving spring has been transformed back into kinetic energy of the bolt and cartridge (except for losses occasioned in feeding the cartridge, in cocking the firing mechanism, and in overcoming friction). Therefore the bolt is moving with considerable velocity and the kinetic energy resulting from this velocity is absorbed by impact at the end of the forward travel. As the bolt comes to rest, ignition occurs and a new cycle begins.

Analysis of Plain Blowback

In the preceding description of blowback operation, it was pointed out that the most critical factor affecting the design of a gun employing this system is the movement of the cartridge case during the action of the powder gas pressure. There are two considerations relating to this movement which impose definite limitations in the design.

1. If no lubrication is provided, the high pressures generated in the early part of the explosion will cause the cartridge case to seize in the chamber. Therefore, separation of the case will result unless the movement of the bolt is limited so that the allowable elongation of the case material is not exceeded while the case is stuck. Although the precise elongation limit for a specific cartridge case can be determined only by careful experimentation, a good rule for the brass cases of 20-mm rounds is that the bolt movement should not exceed 0.015 inch during the first 0.0015 second of the propellant explosion. In other words, the average velocity of the bolt during this time should not exceed 10 inches per second or at the most one foot per second.
2. Even if chamber seizure can be avoided by means of adequate lubrication, the cartridge case can not be permitted to move out of the chamber so far that its thin walls do not receive any radial support while the residual pressure is still fairly high. If this were permitted to happen, it could easily result in swelling or bursting of the case near the base. Here again, the exact limit for a particular cartridge case can only be determined experimentally, but a good rule to follow for brass 20-mm cases is that the movement should not exceed 0.250 inch during the first 0.010 second of the propellant explosion. By this time, the residual pressure will be zero or at least so low that further movement of the case can occur without any danger. This means that during this time the average velocity of the bolt should not exceed 25 inches per second or approximately two feet per second.

This limit is based on the use of an ordinary cartridge case which enters the chamber only to the

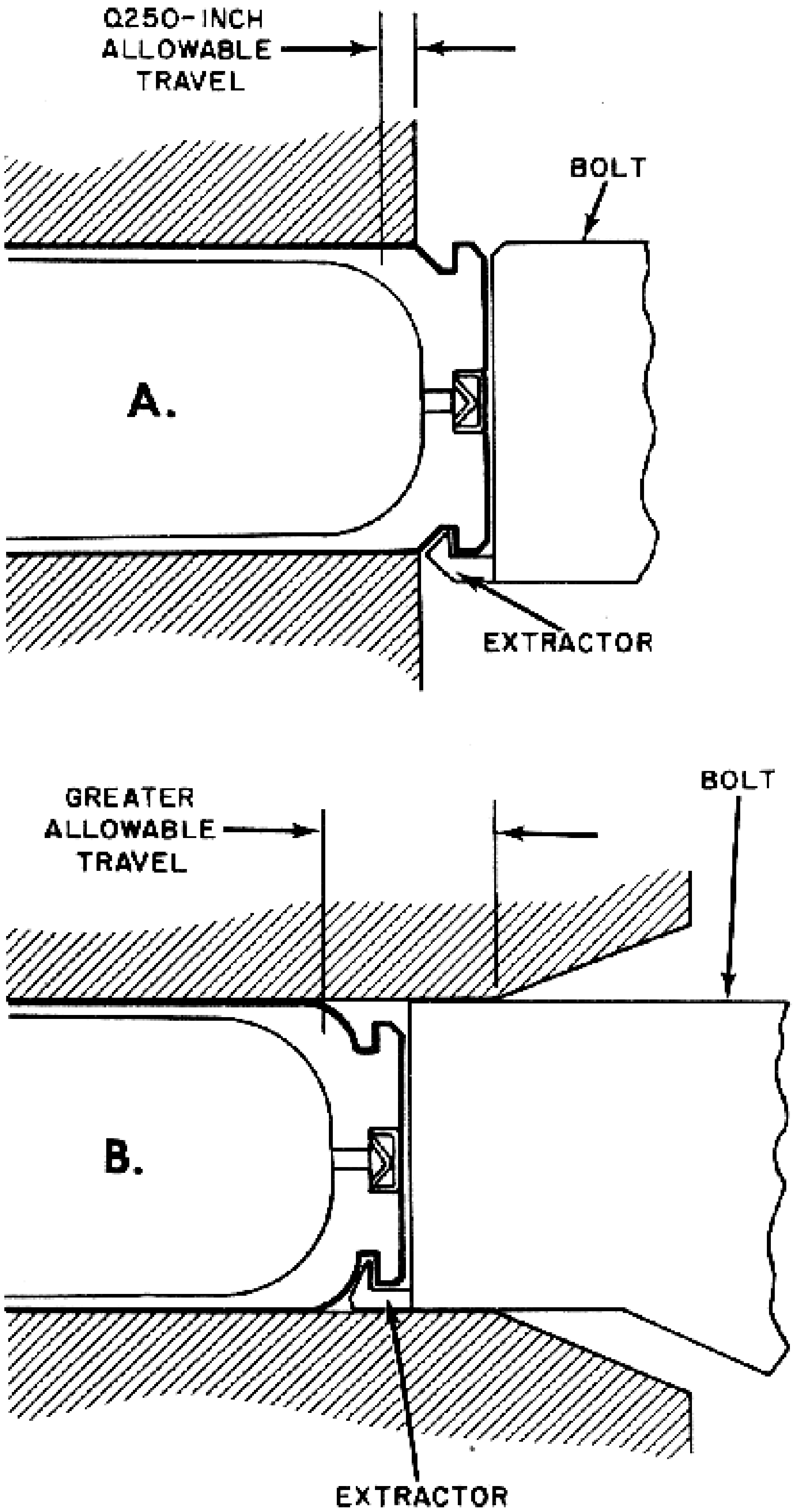


Figure 1–6. Advantage of Using Special Cartridge Base Form for Blowback.

point shown in fig. 1–6A (as is necessary to allow for the presence of the extractor in the extractor groove). If a cartridge case with the special base form shown in fig. 1–6B is used, the cartridge may be pushed further into the chamber and when it is blown back it may therefore travel a greater distance before the walls near the base become unsupported. By this means, the allowable bolt velocity during the first 0.01 second may be increased somewhat but it should be realized that even with this provision, the velocity will still be limited to a relatively low value.

IMPORTANT NOTE: There are two extremely significant points which may be arrived at from further consideration of the foregoing rules:

Fig. 1–7 shows graphically the manner in which the bolt velocity varies in a plain blowback gun during the first 0.010 second of the cycle of operation. The shape of the curve shows that the velocity increases rapidly for the first one or two thousandths of a second and then, as the powder gas pressure falls, it increases much more gradually. (The *shape* of the curve will be the same for any plain blowback gun. Only the vertical *scale* will vary, depending on the particular gun design.) If the gun is designed to permit an average bolt velocity of one foot per second during the first 0.0015 second in order to comply with the first rule, the scale of the graph is then determined so that the average velocity over 0.010 second will be approximately two feet per second, which is the limit required in accordance with the second rule.

In other words, if lubrication is employed to prevent case separation in an attempt thus to escape the limitation of the first rule, the bolt velocity still can not be increased because this will result in exceeding the limitation imposed by the second rule. This means that to gain any significant benefit from lubrication in a plain blowback gun, special steps must be taken to permit a greater average bolt velocity than two feet per second during the first 0.010 second. (As pointed out previously, some moderate improvement in this direction can be achieved by the use of the special type of cartridge case shown in fig. 1–6B.)

The important point involved in these considerations is that, without lubrication, the results of using both of the given rules are so similar that either rule may control the design of the gun. If lubrication is used, the second rule will control, and the average bolt velocity allowable in the first 0.010 second will still be two feet per second unless some special means are employed to increase this limit. In any case, the average bolt velocity must be limited to a very small value, somewhere in the order of

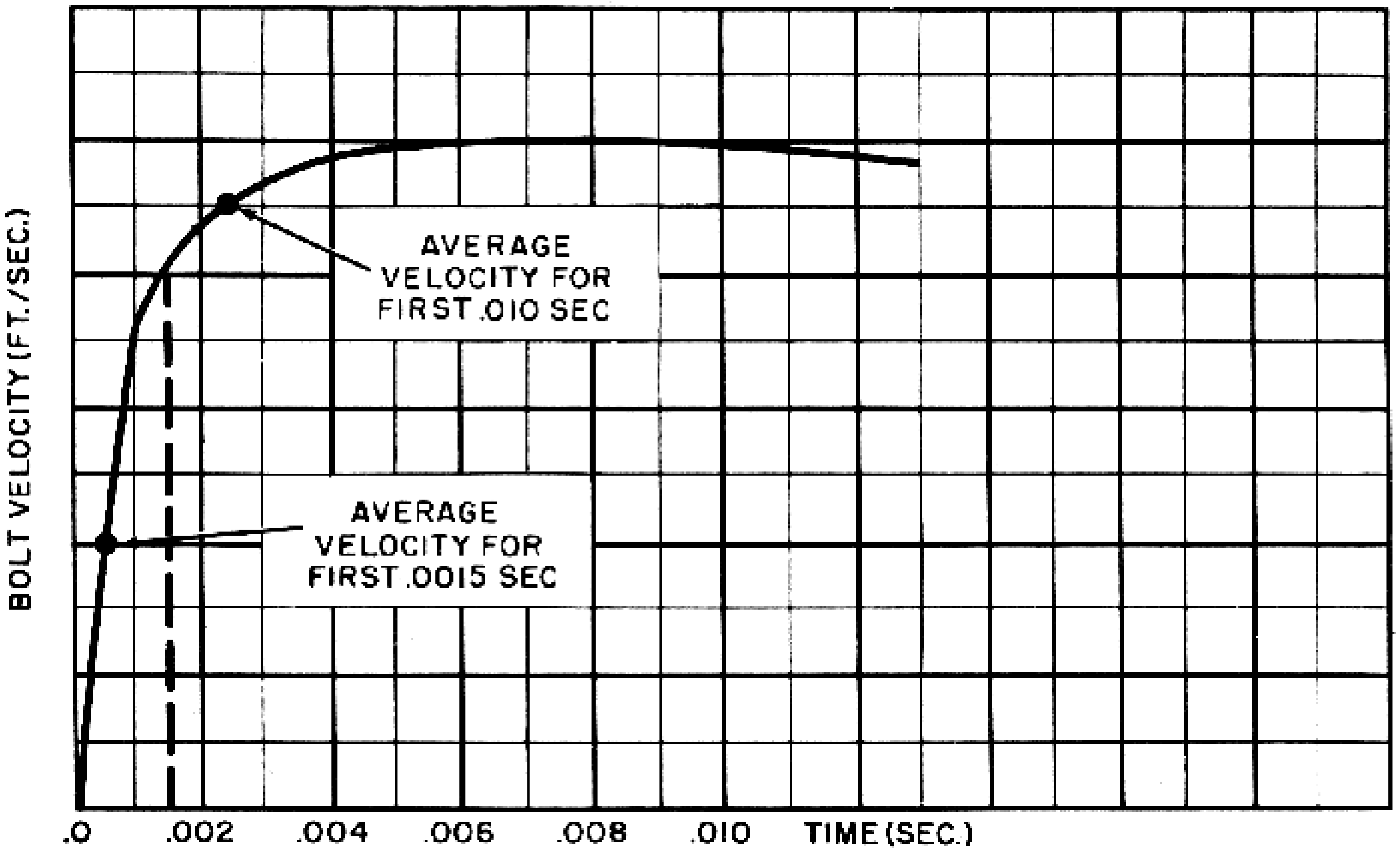


Figure 1–7. Bolt Velocity Versus Time in a Plain Blowback Gun.

a few feet per second. Furthermore, from the shape of the curve in fig. 1–7, it can be seen that even the maximum bolt velocity attained will not be much greater than this value.

The second important point is that with such a low bolt velocity allowable, a high-powered plain blowback machine gun could never attain a reasonable rate of fire. To illustrate this point, the bolt of a 20-mm gun must open about 10 inches in order to permit feeding. Thus, in opening and closing, it must travel a total distance of nearly two feet per cycle, and if it does this at an average velocity of about two feet per second, the firing rate will be the ridiculously low figure of approximately 50 or 60 rounds per minute.

There are other difficulties encountered with a high-powered plain blowback gun and, in fact, these difficulties are so serious that it is difficult to make such a gun function as an automatic weapon. These difficulties are examined further in the following paragraphs. It might seem that such an analysis would amount to an unnecessary preoccupation with an impractical system. However, although this analysis deals with exaggerated conditions, it is made intentionally to disclose and highlight the fundamental concepts involved in the blowback principle and to provide a basis for understanding the other forms of blowback.

Fig. 1–8 shows the condition existing in a blowback gun a few ten-thousandths of a second or so after the ignition of the propellant charge. The pressure of the powder gases is driving the projectile forward and at the same time is driving the cartridge case and bolt to the rear. At this time, the chamber pressure is approximately 45,000 pounds per square inch and the force driving the projectile and cartridge case will be in the neighborhood of 22,000 pounds (for a typical 20-mm round). Of course, some of the driving force is expended in overcoming the friction resisting the motion of the projectile and an additional portion of the driving force is used in imparting rotation to the projectile as it is spun in the barrel by the rifling. At the breech end, similar losses occur in overcoming the frictional resistance to the motion of the cartridge case and bolt and in compressing the driving spring. However, these losses, at the very most, can amount to only two or three thousand pounds. This means that the remaining force of nearly 20,000 pounds is applied entirely in producing the forward acceleration of the projectile mass and the rearward acceleration of the bolt mass. In other words, the only significant factor affecting the motion of these masses is the inertia of the projectile and of the bolt. Therefore, for purposes of simplified analysis, the frictional losses, the losses to projectile rotation, and the resist-

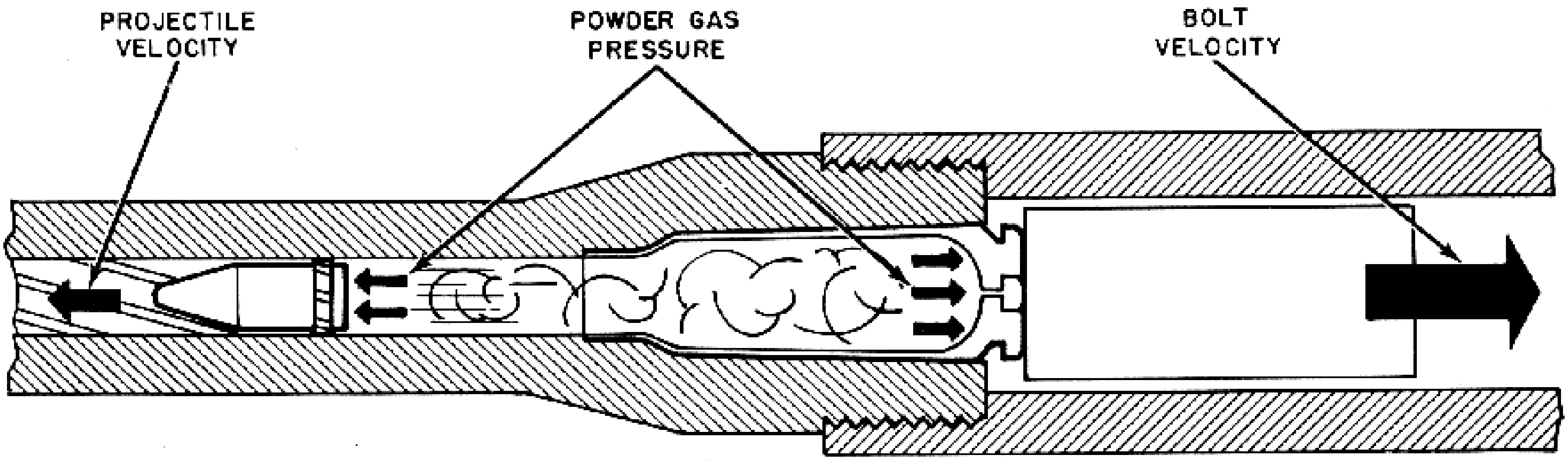


Figure 1–8. Velocities in a Blowback Gun.

ance of the driving spring can be ignored completely.

NOTE: There is one point which requires special clarification at this time. In many descriptions of blowback actions, it is strongly implied that the driving spring contributes a substantial portion of the resistance which limits the acceleration imparted to the bolt by the powder gases. Actually, this is not so. Although it is true that the driving spring absorbs the kinetic energy of the recoiling bolt and thus limits the total distance it moves, the resistance of the spring does not have any real effect in the early phase of the cycle of operation. The bolt acceleration occurs mainly while the powder gas pressures are high and are exerting a force of many thousands of pounds on the bolt. The driving spring, in order to permit the bolt to open enough to allow feeding, must offer a relatively low resistance. Although this resistance is sufficient to absorb the bolt energy over the comparatively great distance through which the bolt moves in recoil, it is not great enough to offer significant opposition to the powder gas pressure until the chamber pressure has dropped to a relatively low level well after the projectile has left the muzzle.

Now consider for a moment what would happen if the bolt had the same weight as the projectile (0.29 pound). If we admit the far-fetched assumption that the cartridge case will remain in one piece, the bolt will be blown back with nearly the same speed as the projectile is driven forward. The four-inch long cartridge case will be blown entirely out of the chamber when the projectile is only four inches down the bore and the hot powder gases, at a pressure of about 45,000 pounds per square inch, will blast out of the breech. Whatever portion of the breech mechanism is not ruined by the blast effect of the explosion will be smashed by the bolt as it flies back at such a high velocity. Even though the bolt has been subjected to only part of the total explosive force, it will have an energy nearly as great as the muzzle energy of a high-powered rifle bullet.)

By increasing the weight of the bolt, we can reduce the velocity at which it recoils and thus eliminate the dangerous condition described in the preceding paragraph. However, as previously explained, the velocity must be limited to a very low value in order to avoid case separation or rupture of the case at its base, and consequently the maximum rate of fire attainable will be somewhere around 60 rounds per minute. It is not necessary at this point to describe the methods used for computing the bolt weight necessary to accomplish this condition, so suffice it to say that it will be about 500 pounds. It is apparent that this weight is much too great to consider in any practical 20-mm gun design.

It has often been remarked that a plain blowback machine gun could be built to fire high-powered 20-mm ammunition providing that there was no objection to the excessive weight necessary in the bolt or to the inherent low rate of fire. The fact of the matter is that such a gun would be entirely impractical for other more cogent reasons. For example, if it is assumed that the initial velocity of the 500-pound bolt mass is about 2.50 feet per second, the energy imparted to the bolt will be only about 50 foot-pounds. Since this energy is supplied to the gun mechanism only 50 times per minute, or at the rate of 2500 foot-pounds per minute, only 8/100 horsepower will be available for doing the work necessary for feeding, firing, ejecting, overcoming friction, and for performing the other functions required of the gun mechanisms. This seems hardly sufficient power to operate a machine in the class of the breech mechanism of a conventional 20-mm gun. In other words, the heavier the bolt, the less energy it will absorb from the propellant explosion and if the bolt is too heavy, the gun may fail to function automatically because insufficient power will be available to operate the mechanism.

Another reason why the gun will fail to operate is because the driving spring would of necessity be a pitifully weak affair. Since the spring must absorb only 50 foot-pounds of energy and must do this over a movement of at least 10 inches in order to permit feeding a 20-mm round, the average force exerted by the spring can be no more than 60 pounds (60 pounds $\times$ 0.833 feet $=$ 50 foot-pounds). It is hardly necessary to point out that a spring as light as this will not work very successfully as the driver for a 500-pound bolt and it is also doubtful whether it could exert enough direct force to operate the feed mechanism or to perform other tasks requiring a fairly powerful thrust. Another serious shortcoming of such a spring would become obvious by the embarrassing way in which the bolt would slide open whenever the gun is elevated a little above the horizontal.

The principal points disclosed in the preceding analysis of a high-powered plain blowback gun may be summarized as follows:

1. To prevent case separation or rupture of the case near its base, the bolt velocity must be limited to an extremely low value.
2. For this system, lubrication of the ammunition does not permit a significantly higher bolt velocity.
3. In order to limit the bolt velocity as required, the weight of the bolt must be excessively great.
4. Because of the necessity for low bolt velocity, the rate of fire is too low for any practical purpose.
5. The driving spring is not a significant factor in limiting bolt velocity.
6. The bolt of a blowback gun is the means whereby energy for operating the mechanism is obtained from the propellant explosion. Because of the excessive bolt mass required in a plain blowback gun, and the low rate of fire, insufficient power is obtained for maintaining automatic operation.
7. The driving spring is of necessity very weak and can not exert sufficient direct force to perform its function satisfactorily.

All of these points demonstrate amply why no successful plain blowback machine gun has ever been developed to fire high-powered ammunition.

Mathematical Analysis of Plain Blowback

The following paragraphs describe a systematic procedure for performing the computations necessary in a basic analysis of a plain blowback gun. For the most part, the methods described are conventional and follow the lines of analysis which have been used for some time in general engineering studies of guns, particularly artillery weapons. The main endeavor in this present analysis was to reorganize the existing methods and to modify and supplement them as necessary to adapt them for convenient application to the specific problems encountered in the design of blowback machine guns.

It must be emphasized at the outset that the principal concern of this analysis is to establish a method which may be used to determine the bolt mass required to limit the bolt recoil velocity to a safe value and, on the basis of this mass, to make preliminary calculations for determining the driving spring design data, rate of fire, bolt motion characteristics, and other useful information.

In this analysis, no attempt will be made to discuss the straightforward machine design methods by which the results are applied in arriving at the particular physical form of a breech mechanism. Also, no detailed computations are made to cover the effects of such factors as friction or the incidental forces imposed on the breech mechanism by the auxiliary mechanism such as the feeder, firing device, or ejector. These effects are entirely negligible in determining the bolt mass, but they will have some influence on the overall bolt motion and rate of fire. In any case, they can be properly taken into account only in the more or less advanced stages of a design when the form of the gun mechanism becomes fairly well established. At this point, the preliminary values determined can easily be modified as required.

The analysis which follows is based on the assumption that a particular cartridge with known characteristics is to be used and that the desired muzzle

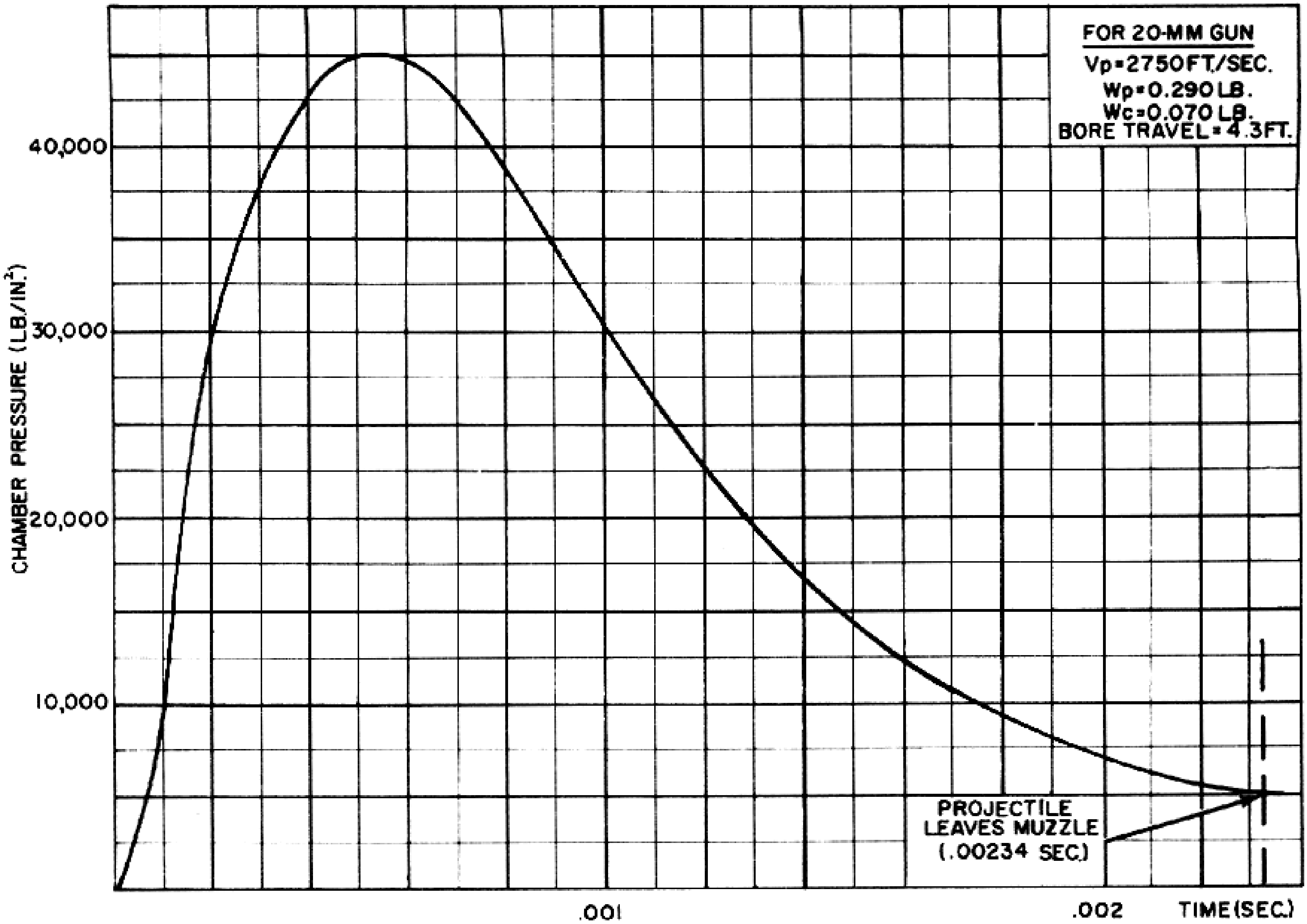


Figure 1–9. Graph of Chamber Pressure Versus Time (20 mm Gun).

velocity and barrel length have been predetermined. It is also assumed that all necessary interior ballistics data are known and that graphs showing the time variation of projectile velocity and chamber pressure are available (figs 1–9, 1–10, and 1–11).

NOTE: For some design problems, all or part of this information may not be available. Analytical methods by which the required data and graphs can be approximated for use in preliminary studies may be determined by conventional interior ballistics computations.

As the analysis progresses, its application will be illustrated by means of sample calculations. Although these calculations and related graphs are for a specific 20-mm cartridge and barrel, the methods are applicable to a gun of any caliber. The calculations cover the following important points:

1. Determination of the bolt weight required to limit the recoil velocity to a safe value.
2. Determination of the data necessary for designing a driving spring which will permit the bolt to open the required distance for feeding.
3. Computation of the rate of fire.
4. Development of graphs showing how bolt velocity and bolt travel vary with time.
5. Computation of power absorbed by the bolt.

In the course of describing these computations, the following fundamental formulas will be developed and explained:

a. Momentum relation for time projectile is in bore.
b. Formula for determining momentum and velocity of free recoil.
c. Expression for duration of residual pressure.
d. Formula for determining initial bolt energy.
e. Formulas for determining spring retardation.

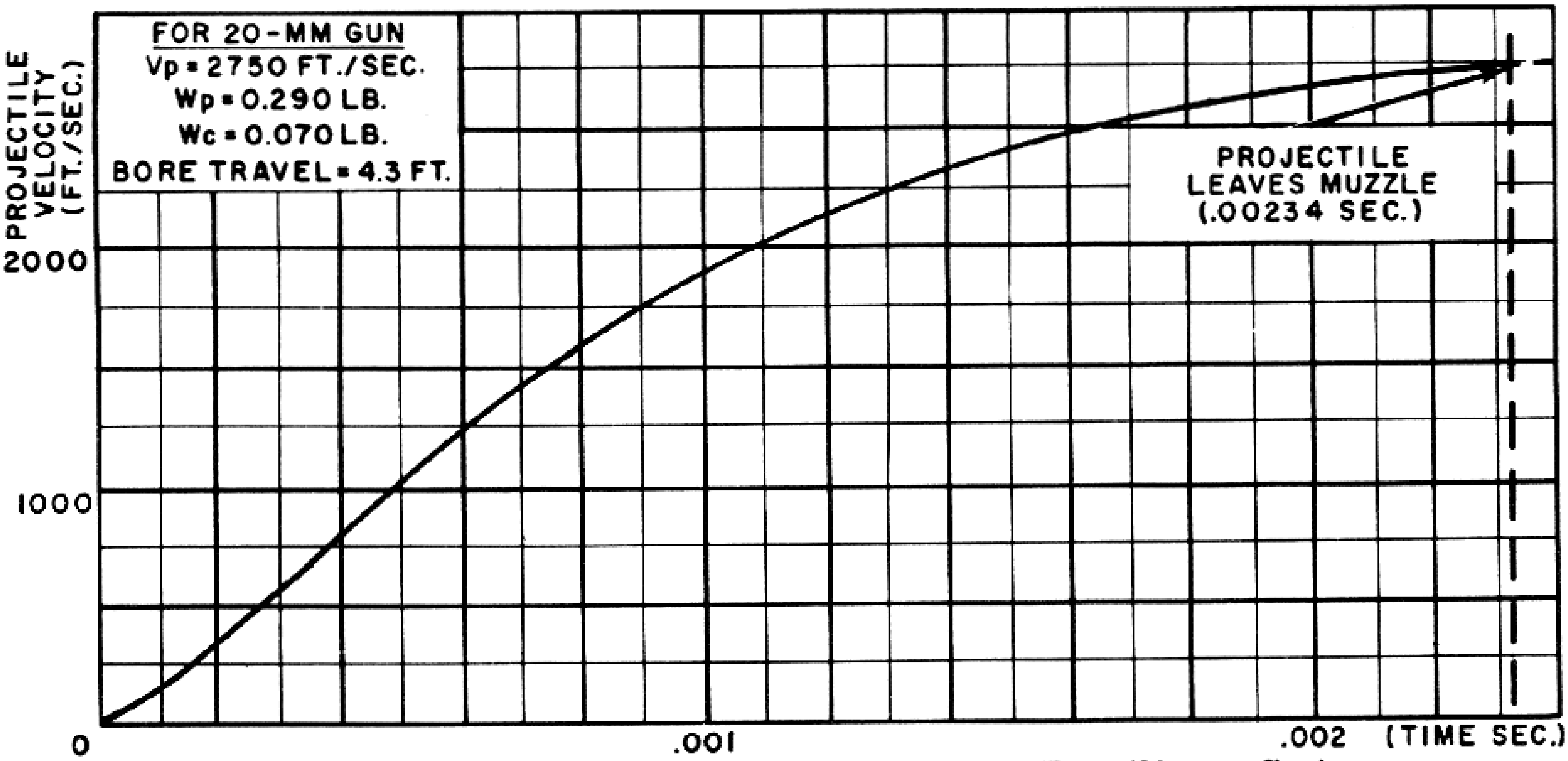


Figure 1–10. Graph of Projectile Bore Travel Versus Time (20 mm. Gun).

f. Energy equation for bolt and driving spring.
g. Formula for determining time to recoil.
h. Expression for computing rate of fire.

1. *Determination of bolt weight*

When the propellant charge is exploded, the projectile and powder gases move forward through the bore while the bolt is driven to the rear. Since the bolt is not connected to the barrel, the forces resisting the motion of the projectile are not equal to the forces resisting the motion of the bolt. However, because such an extremely small percentage of the explosive impulse is required to overcome friction and account for other losses, it may be assumed with very little error that the momentum of the bolt at any time is equal to the combined momentum of the projectile and powder gases. Assuming that the center of mass of the powder gases moves through the bore at one-half the velocity of the projectile, the momentum equality may be expressed as follows:

$$(1-1)\quad M_r v_{r_f} = M_p v_p + M_c \frac{v_p}{2} - \left(M_p + \frac{M_c}{2}\right) v_p$$

The subscript, f, in the symbol for the bolt velocity indicates that this is the *free* or unretarded bolt velocity. In other words, it is assumed for the present that the bolt motion is not subjected to the retarding effects of the driving spring or friction. This assumption will not introduce any significant inaccuracy for the time during which the powder

Symbols Used in Analysis

A	Area of bore cross-section—in.^{2}
C	Arbitrary constant of integration
D	Total recoil travel of bolt—ft.
d	Recoil travel of bolt for time t—ft.
E_r	Initial bolt energy—ft. lb.
F_{av}	Average spring force over distance D—lb.
F_o	Initial compression of spring—lb.
g	Acceleration of gravity—32.2 ft./sec.^{2}
K	Spring rate—lb./ft.
M_c	Mass of powder charge—lb.sec.^{2}/ft.
M_p	Mass of projectile—lb.sec.^{2}/ft.
M_r	Mass of bolt (recoiling parts)—lb.sec.^{2}/ft.
P	Muzzle pressure—lb./in.^{2}
T	Time to recoil—sec.
t	Time—sec.
T_{res}	Time of duration of residual pressure—sec.
V_p	Muzzle velocity of projectile—ft./sec.
v_p	Velocity of projectile in bore at time t—ft./sec.
v_r	Velocity of retarded recoil at time t—ft./sec.
V_{r_f}	Maximum velocity of free recoil—ft./sec.
v_{r_f}	Velocity of free recoil at time t—ft./sec.
$v_{r_{f(all)}}$	Allowable average recoil velocity—ft./sec.
W_c	Weight of powder charge—lb.
W_p	Weight of projectile—lb.
W_r	Weight of bolt (recoiling parts)—lb.

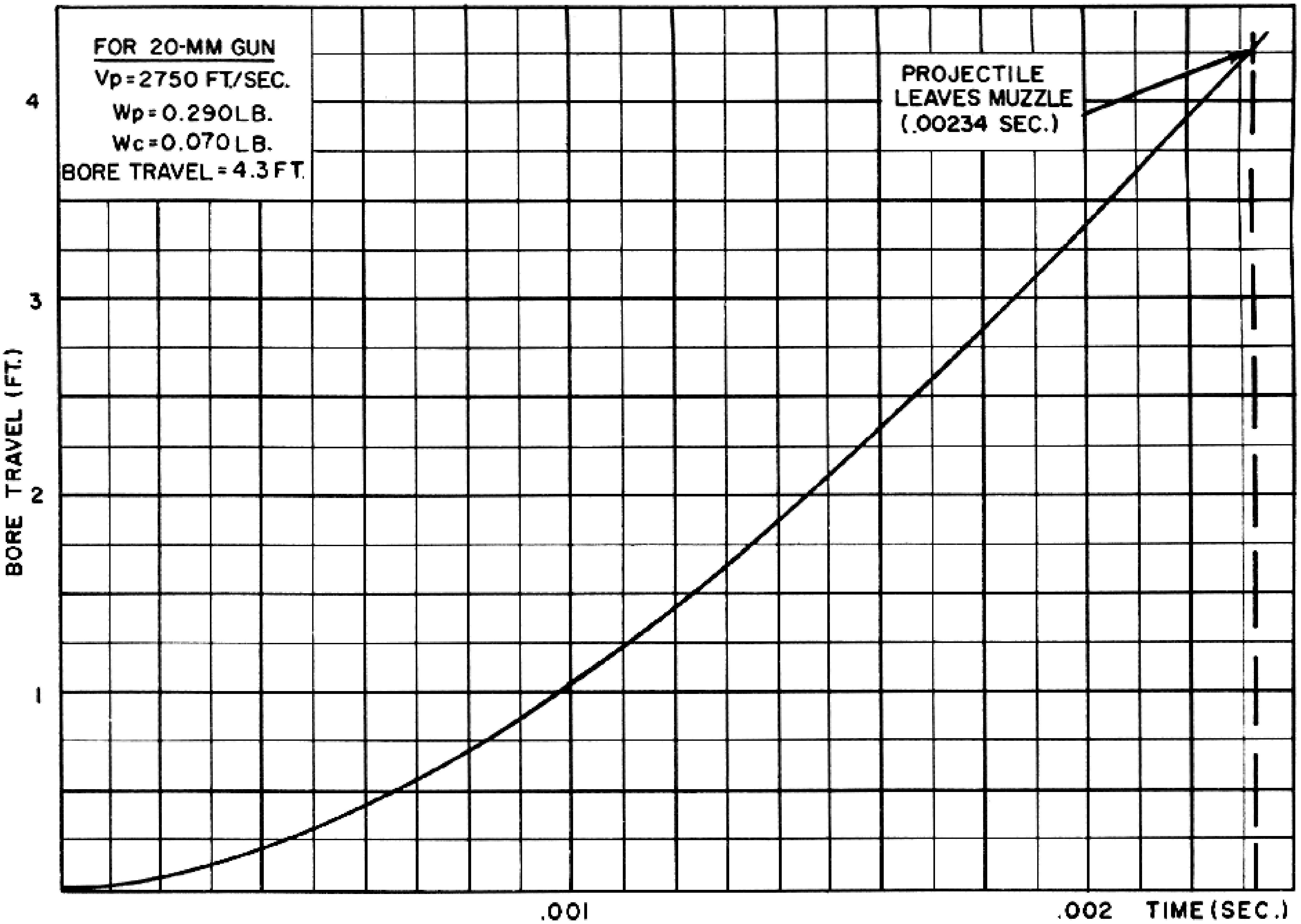


Figure 1–11. Graph of Projectile Bore Travel Versus Time (20 mm Gun).

gas pressures act because the retarding forces are negligible when compared to the forces exerted by the powder gases.

Equation 1–1 can be used to plot a graph of bolt momentum versus time for the period before the projectile leaves the muzzle. The weights of the projectile and powder charge are both known and fig. 1–10 gives the projectile velocity at any time. Therefore, the ordinate of the bolt momentum curve at any time, t, can be determined by multiplying the corresponding ordinate of the projectile velocity curve by the factor

$$\left(M_p+\frac{M_c}{2}\right)$$

For the 20-mm cartridge upon which fig. 1–10 is based:

$$M_p+\frac{M_c}{2}=\frac{1}{g}\left(W_p+\frac{W_c}{2}\right)=\frac{1}{32.2}\left(.29+\frac{.070}{2}\right)=.00101$$

Therefore, before the projectile leaves the muzzle, the bolt momentum is:

$$M_r v_{r_t}=.00101\ v_p\ \text{(lb. sec.)}$$

The curve obtained by using this relation is shown in fig. 1–12. The same curve is also shown in fig. 1–13 as the portion between t=0 and t=.00234 sec. (In this figure, the time axis is compressed in order to permit consideration of what happens after the projectile leaves the muzzle.)

The momentum relations which exist after the projectile leaves the muzzle cannot be formulated simply, but a special method can be employed to extend the curve plotted by using equation 1–1. To draw the remainder of the curve, use is made of the maximum free bolt momentum as determined from the equation:

$$M_r V_{r_f}=M_p V_p+4700\ M_c \qquad (1\text{–}2)$$

This is an empirical relation based on the results of experimental firings of various guns. It amounts to saying that the final momentum imparted to the recoiling parts is equal to the sum of the muzzle

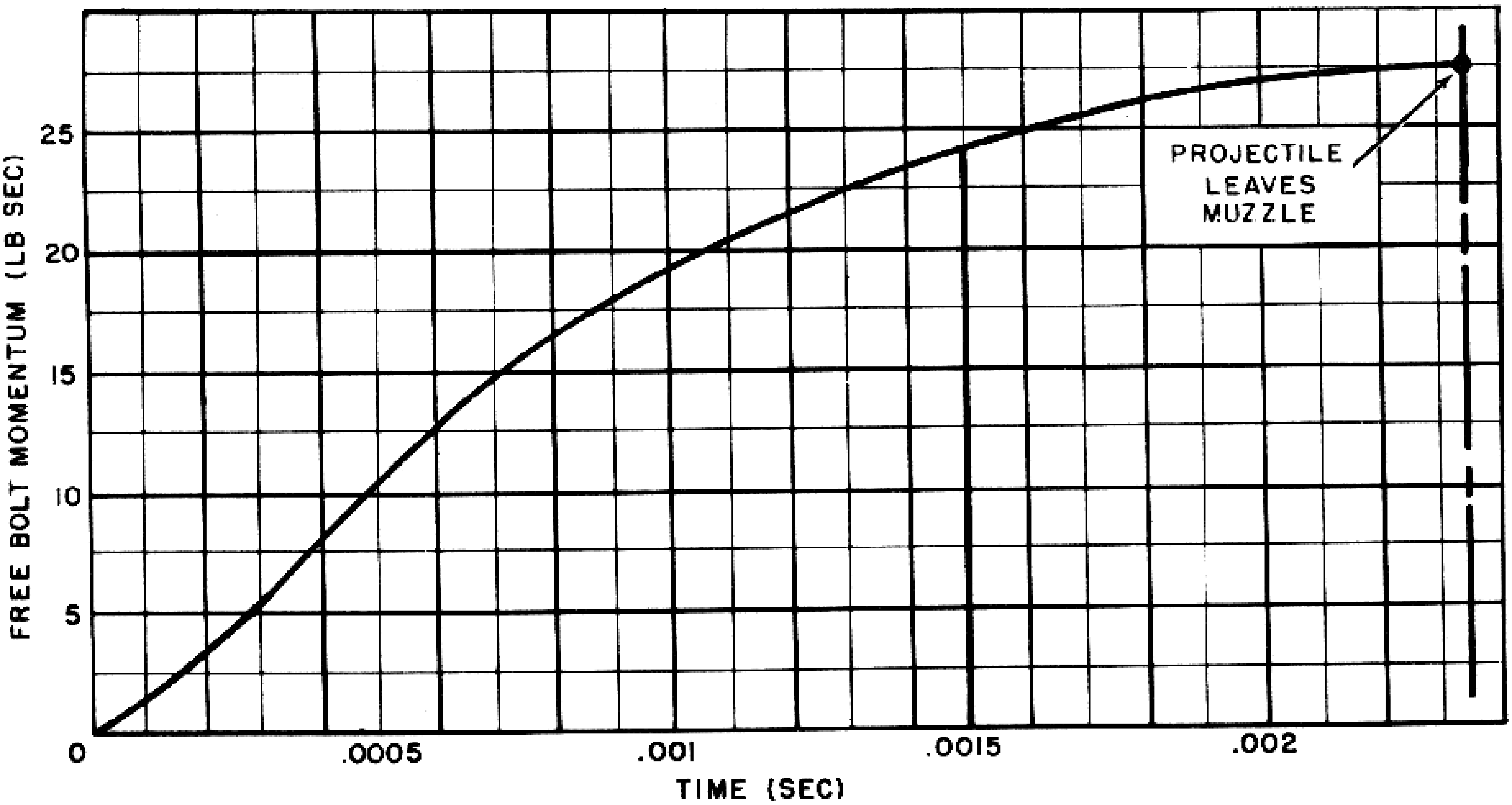


Figure 1–12. Free Bolt Momentum for First 0.0015 Second.

momentum of the projectile and the momentum of the powder gases, assuming that the powder gases leave the gun at an average velocity of 4700 feet per second. For the cartridge and barrel used as an example:

$$M_r V_{r_f} = \frac{.29}{32.2}\,2750 + 4700\,\frac{.070}{32.2} = 35 \text{ (lb. sec.)}$$

A line representing this value of free bolt momentum is drawn on the bolt momentum graph (fig. 1–13) and the curve previously drawn is extrapolated until it becomes tangent to the line. The point at which the curve is tangent represents the time at which the residual pressure becomes zero and therefore imparts no further momentum to the bolt. Although an error in locating the exact point of tangency will not have any serious effect on the accuracy of the results, it may be of some assistance in plotting to determine this point by using Vallier's formula for approximating the duration of the residual pressure:

$$T_{res} = \frac{M_c}{AP}\,(9400 - V_p) \tag{1–3}$$

For the sample cartridge and barrel:

$$T_{res} = \frac{.070}{32.2 \cdot \frac{\pi}{4}(.790)^2 \cdot 5000}\,(9400–2750) = .00592 \text{ (sec.)}$$

To obtain the total time of action of the powder gases, this value is added to the time at which the projectile leaves the muzzle:

$$T_{res} = .00234 + .00592 = .00826 \text{ (sec.)}$$

Extending the original curve until it is tangent at this point gives the complete free bolt momentum curve shown in fig. 1–13.

This curve can be used directly to determine the bolt weight necessary to limit the bolt velocity to a safe value over a given interval of time. By integrating under the curve for the desired time interval and then dividing the result by the time interval, the average bolt momentum for the interval can be found. Dividing this average momentum by the allowable average velocity for the interval will give the required bolt mass. That is to say:

$$M_r = \frac{\int_{t_1}^{t_2} (M_r v_{r_f})\,dt}{(t_2 - t_1)\,v_{r_{f(all)}}} \tag{1–4}$$

A simple and sufficiently accurate method of evaluating the integral in equation 1–4 is to measure the area under the curve by use of the so-called trapezoidal rule or some other method of approximate integration such as Simpson's rule.

Now assume that the cartridge case is not lubricated. As previously explained, this condition re-

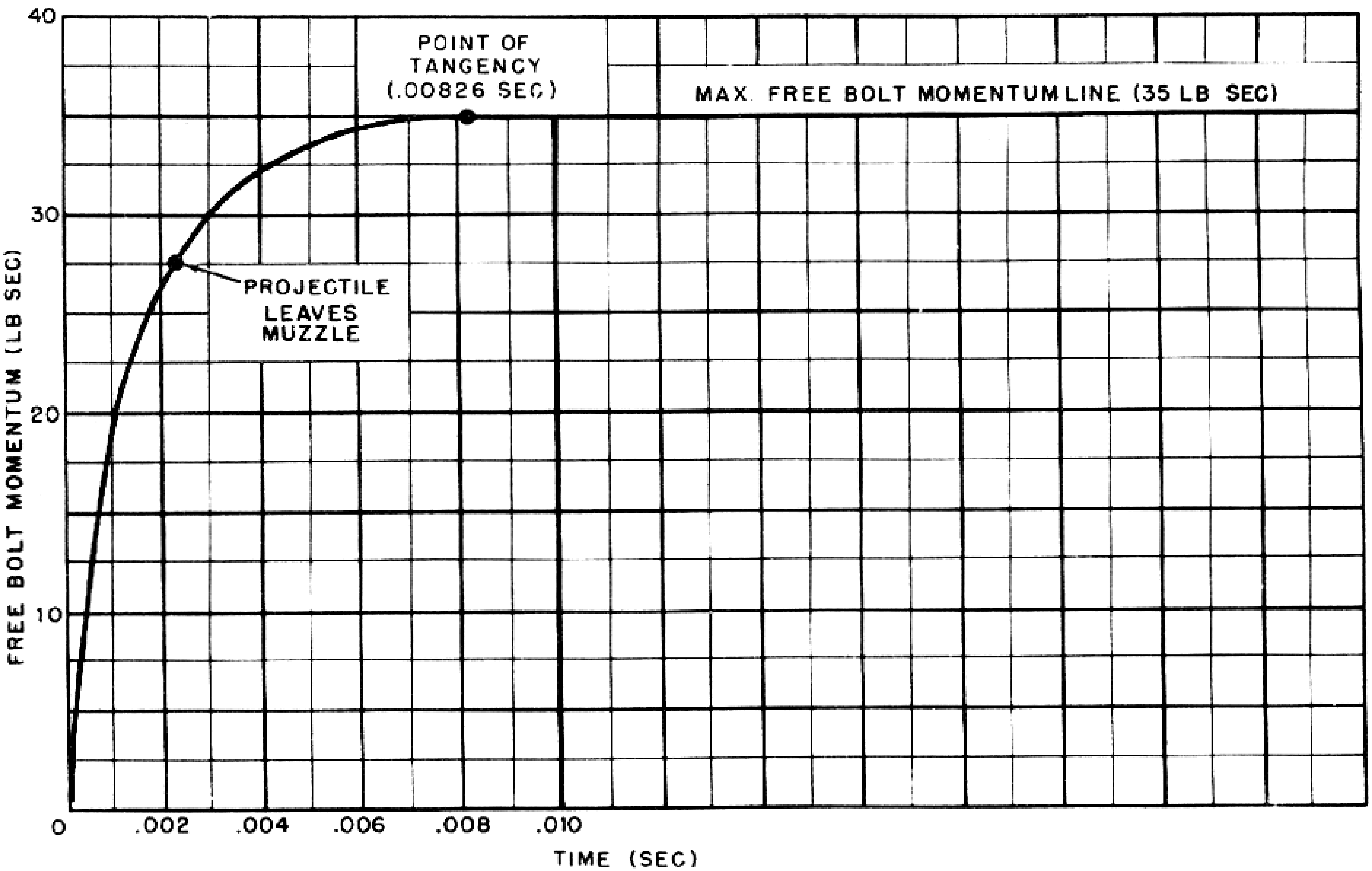


Figure 1–13. Free Bolt Momentum for First 0.010 Second.

quires that the average bolt velocity be limited to one foot per second for the first 0.0015 second in order to prevent case separation. Integrating under the curve shown in fig. 1–12 from t=0 to t=.0015 gives a total area of 0.02298 (lb. sec.^{2}). Using equation 1–4 to evaluate M_r gives:

$$M_r = \frac{.02298}{.0015 \cdot 1} = 15.32 \left(\frac{\text{lb. sec.}^2}{\text{ft.}}\right)$$

Therefore, the required weight is:

$$W_r = gM_r = 32.2 \cdot 15.32 = 493 \text{ (lb.)}$$

It was also pointed out previously that if the cartridge case is of the form shown in fig. 1–6A, the average bolt velocity (even with lubrication) must be limited to two feet per second for the first 0.010 second in order to prevent rupture near the base of the case. Assuming this limitation, integrating under the curve shown in fig. 1–13 from t=0 to t=.010 gives a total area of 0.305 (lb. sec.^{2}). Using equation 1–4 to evaluate M_r gives:

$$M_r = \frac{.305}{.010 \cdot 2} = 15.25 \left(\frac{\text{lb. sec.}^2}{\text{ft.}}\right)$$

Therefore, the required weight is:

$$W_r = gM_r = 32.2 \cdot 15.25 = 492 \text{ (lb.)}$$

Note that this weight is almost exactly the same as obtained by assuming the first limitation. As previously explained, the necessary bolt weight can be decreased somewhat if lubrication is used and some special method can be found by which the allowable average velocity over the first 0.010 second can be increased to more than two feet per second. However, for purposes of continuing the analysis, these special methods will be ignored and the bolt weight will be taken as 500 pounds.

2. *Determination of driving spring design data*

Equation 1–2 gives the maximum free recoil momentum as:

$$M_r V_{r_f} = M_p V_p + 4700 \, M_c$$

Solving this equation for V_{r_f} gives the maximum free recoil velocity as:

$$(1\text{–}5) \quad V_{r_f} = \frac{M_p V_p + 4700 \, M_c}{M_r} = \frac{W_p V_p + 4700 \, W_c}{W_r}$$

For the 500-pound bolt and the conditions of the example:

$$V_{r_f}=\frac{.29\cdot 2750+4700\cdot .070}{500}=2.2\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

Since the total acceleration of the bolt occurs in less than 0.010 second and since the retardation which can be offered during this interval by the driving spring is very small, it may be assumed, when considering the effect of the driving spring, that the initial velocity of the bolt is equal to the maximum free recoil velocity expressed by equation 1 5.

The initial bolt energy is given by the expression:

$$(1\text{-}6) \qquad E_r=\frac{1}{2}M_rV_{r_f}^2=\frac{W_rV_{r_f}^2}{2g} \text{ (ft. lb.)}$$

Evaluating this expression for the condition of the example gives the initial bolt energy as:

$$E_r=\frac{500\cdot 2.25^2}{2\cdot 32.2}=39.4 \text{ (ft. lb.)}$$

In other words, the bolt may be considered to start compressing the driving spring with an initial energy of 39.4 foot pounds. The spring must be proportioned so as to absorb this amount of energy over the entire distance through which it is compressed times the average force required to produce this deflection. That is:

$$(1\text{-}7) \qquad E_r=F_{av}D \text{ or } F_{av}=\frac{E_r}{D}$$

If it is assumed that the bolt in the example must open 10 inches (0.833 feet) in order to permit feeding of a 20-mm cartridge, the average force exerted by the spring must be:

$$F_{av}=\frac{39.4}{.833}=47.2 \text{ (lb.)}$$

It should be noted here that friction between the bolt and its slide will produce an essentially constant retarding force. If it is expected that the force required to overcome the friction will be considerable, this force should be determined and subtracted from the average spring force computed by using equation 1 7. However, for purposes of the present analysis, the friction force will be neglected.

Having the average spring force, the remaining problem is to choose spring characteristics such that this average force will result from compressing the spring through the required recoil distance. The values to be chosen are the initial compression, F_o, and the spring rate, K. For example, if the initial compression is taken as 17.2 pounds, a maximum force of 77.2 pounds will produce the required average force of 47.2 pounds. Since the difference between the maximum force and initial compression is 60 pounds and the recoil distance is 10 inches, the spring rate will be 6 pounds per inch or 72 pounds per foot. Any other combination of F_o and K could be chosen, providing that they produce an average force of 47.2 pounds.

NOTE: Actually, the design of a driving spring for a practical machine gun is a rather complex problem and can not be disposed of so easily. Spring losses, shock loads, forced vibrations set up along the length of the spring, and other factors can cause serious difficulties in operation and may even result in unpredictable breakage. For this reason, the design of springs to be used in a rapidly oscillating mechanism like a machine gun is a specialized art in the field of machine design. Special techniques of analysis are necessary and often a successful design can be arrived at only through careful experimentation. Also, as will be explained later, the choice of F_o and K can have an effect on the rate of fire attainable.

Although the factors mentioned in the preceding note should receive due consideration in an actual design problem, it will be assumed here that the arbitrarily selected combination of $F_o=17.2$ pounds and $K=72$ pounds per foot will result in a satisfactory spring.

The required bolt weight and the characteristics of the driving spring have now been determined and this determination was made in accordance with the stipulated requirements for limiting the bolt motion and for allowing sufficient recoil travel to permit feeding. In other words, the basic design of the gun is now established and the only task remaining is to consider what performance this design will give.

It is obvious that a gun having such a heavy bolt and such a weak driving spring will not be practical. However, the only way to improve the design would be to employ some special method for permitting a higher average bolt velocity. The question of whether or not such a method can be found is not important for the purposes of the present analysis and will therefore be left to the ingenuity of the designer. The main purpose of this analysis is to establish a method of approach to the design of a plain blowback gun and this method can be applied regardless of what particular bolt velocity is allowable.

3. *Derivation of bolt motion equations*

Since the bolt weight and driving spring data have been determined, it is now possible to investigate the performance of the gun. The first step in this investigation will be to derive a number of important equations relating to the motion of the bolt. These derivations will all be based on the fact that, as the bolt moves in recoil, the driving spring absorbs and stores energy. If it is assumed that the losses due to friction and other causes are negligible, the energy remaining in the bolt at any time during recoil is expressed by the equation:

$$(1-8)\qquad \frac{M_r v_r^2}{2} = E_r - \int_0^d (F_o + Kd)\,dd$$

$$= E_r - \left(F_o d + \frac{Kd^2}{2}\right)$$

This equation may be used for deriving the equation expressing the relation between time and bolt motion as follows: Solving for v_r gives:

$$v_r = \sqrt{-\frac{K}{M_r}d^2 - \frac{2F_o}{M_r}d + \frac{2E_r}{M_r}} = \frac{dd}{dt}$$

$$dt = \frac{dd}{\sqrt{-\frac{K}{M_r}d^2 - \frac{2F_o}{M_r}d + \frac{2E_r}{M_r}}}$$

From a table of integrals, this expression is of the form:

$$\int \frac{dd}{\sqrt{ad^2+bd+c}} = \frac{1}{\sqrt{-a}}\sin^{-1}\frac{-2ad-b}{\sqrt{b^2-4ac}} + C$$

where: $a = -\frac{K}{M_r}\ (a<0)$

$b = -\frac{2F_o}{M_r}$

$c = +\frac{2E_r}{M_r}$

Therefore:

$$t = \sqrt{\frac{M_r}{K}}\left[\sin^{-1}\frac{\frac{2K}{M_r}d + \frac{2F_o}{M_r}}{\sqrt{\frac{4F_o^2}{M_r^2} + \frac{8KE_r}{M_r^2}}}\right] + C$$

$$= \sqrt{\frac{M_r}{K}}\left[\sin^{-1}\frac{Kd+F_o}{F_o^2+2KE_r}\right] + C$$

But at the end of the recoil movement, the energy stored in the driving spring is equal to the initial bolt energy:

$$E_r = F_o D + \frac{KD^2}{2}$$

Therefore:

$$F_o^2 + 2KE_r = F_o^2 + 2K\left(F_o D + \frac{KD^2}{2}\right)$$

$$= F_o^2 + 2KF_oD + K^2D^2$$

$$= (F_o + KD)^2$$

Substituting this value in the equation for t gives:

$$t = \sqrt{\frac{M_r}{K}}\left[\sin^{-1}\frac{Kd+F_o}{KD+F_o}\right] + C$$

to evaluate C: When $t=0$, $d=0$ and therefore

$$C = -\sqrt{\frac{M_r}{K}}\sin^{-1}\frac{F_o}{KD+F_o}$$

Substituting this expression for C gives the equation for the time, t, required to recoil any distance, d.

$$(1-9)\qquad t = \sqrt{\frac{M_r}{K}}\left[\sin^{-1}\frac{Kd+F_o}{Kd+F_o} - \sin^{-1}\frac{F_o}{KD+F_o}\right]$$

Solving this equation for d gives the inverse relation expressing the distance recoiled in any time, t.

(1–10)

$$d = \frac{KD+F_o}{K}\sin\left[\sqrt{\frac{K}{M_r}}\,t + \sin^{-1}\frac{F_o}{KD+F_o}\right] - \frac{F_o}{K}$$

Equation 1–9 may also be used to obtain the total time T required for the bolt to move through the entire recoil distance D. Substituting D for d gives:

$$T = \sqrt{\frac{M_r}{K}}\left[\sin^{-1}1 - \sin^{-1}\frac{F_o}{KD+F_o}\right]$$

$$(1-11)\qquad T = \sqrt{\frac{M_r}{K}}\left[\frac{\pi}{2} - \sin^{-1}\frac{F_o}{KD+F_o}\right]$$

$$= \sqrt{\frac{M_r}{K}}\left(\cos^{-1}\frac{F_o}{KD+F_o}\right)$$

If it is assumed that the losses are negligible, although this is impossible, the time for the bolt to return will be equal to the recoil time. That is, the time for a complete cycle of operation will be 2T seconds. On this basis, the rate of fire in rounds per minute will be:

$$N = \frac{60}{2T} = \frac{30}{T}$$

$$(1-12)\qquad N = \frac{30\sqrt{\frac{K}{M_r}}}{\cos^{-1}\frac{F_o}{KD+F_o}}$$

Evaluating N for the conditions of the example gives the rate of fire as:

$$N = \frac{30\sqrt{\frac{72 \cdot 32.2}{500}}}{\cos^{-1}\frac{17.2}{72 \cdot .833 + 17.2}} = 48 \text{ (rounds per minute)}$$

With the rate of fire and bolt energy known, the power absorbed by the bolt can be computed by means of the formula:

$$(1\text{–}13) \qquad HP = \frac{E_r N}{33{,}000}$$

The horsepower absorbed by the bolt in the gun of the example will be:

$$HP = \frac{39.4 \cdot 48}{33{,}000} = .0573$$

4. *Development of theoretical time-travel and time-velocity curves*

Curves which show the bolt travel and bolt velocity with respect to time are very useful in determining what performance can be expected of a design and also provide useful data for the design of the feeder, ejector, firing device, and other auxiliary mechanisms. The formulas derived under the preceding heading may be used to obtain such curves and other data of interest. However, it should be realized that all of these equations were derived under the assumption that the initial bolt energy was transferred to the bolt instantaneously and therefore do not take into full consideration the period of time during which the powder gas pressures act. For a gun having a low rate of fire, the events which occur during this period of time are negligible in considering the total bolt travel but when the rate of fire is high, the powder gas pressures exist for a significant portion of the time required for recoil. For this reason, it is usually desirable to take the effects of the powder gases into consideration when developing bolt motion curves.

Since the effects of the powder gases can not be formulated readily, a special method is employed to plot the curves. The method consists essentially of plotting a curve of free bolt velocity and then subtracting from each ordinate of this curve the loss in velocity resulting from the retarding effect of the spring.

The free bolt velocity curve can be derived directly from the free bolt momentum curve previously plotted (figs. 1–12 and 1–13) by dividing each ordinate of the momentum curve by the mass of the bolt. The resulting curves are shown in figs. 1–14 and 1–15.

To determine the retarding effects of the spring, use is made of the law expressed by the equation:

$$(1\text{–}14) \qquad F\,dt = M\,dv$$

This law states that the change in the momentum of a mass is equal to the applied impulse (the product

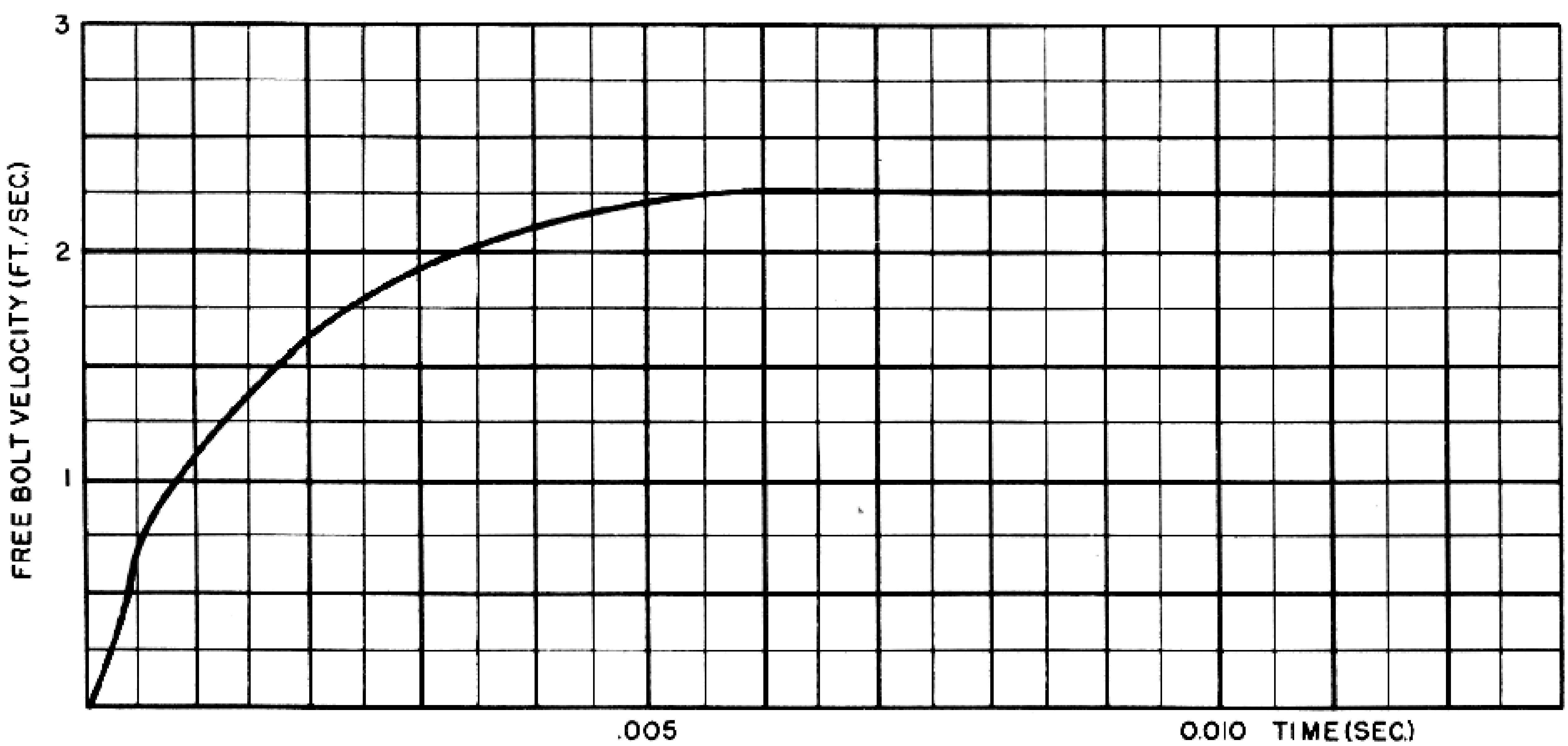


Figure 1–14. Free Recoil Velocity Versus Time (0–0.010 Sec.).

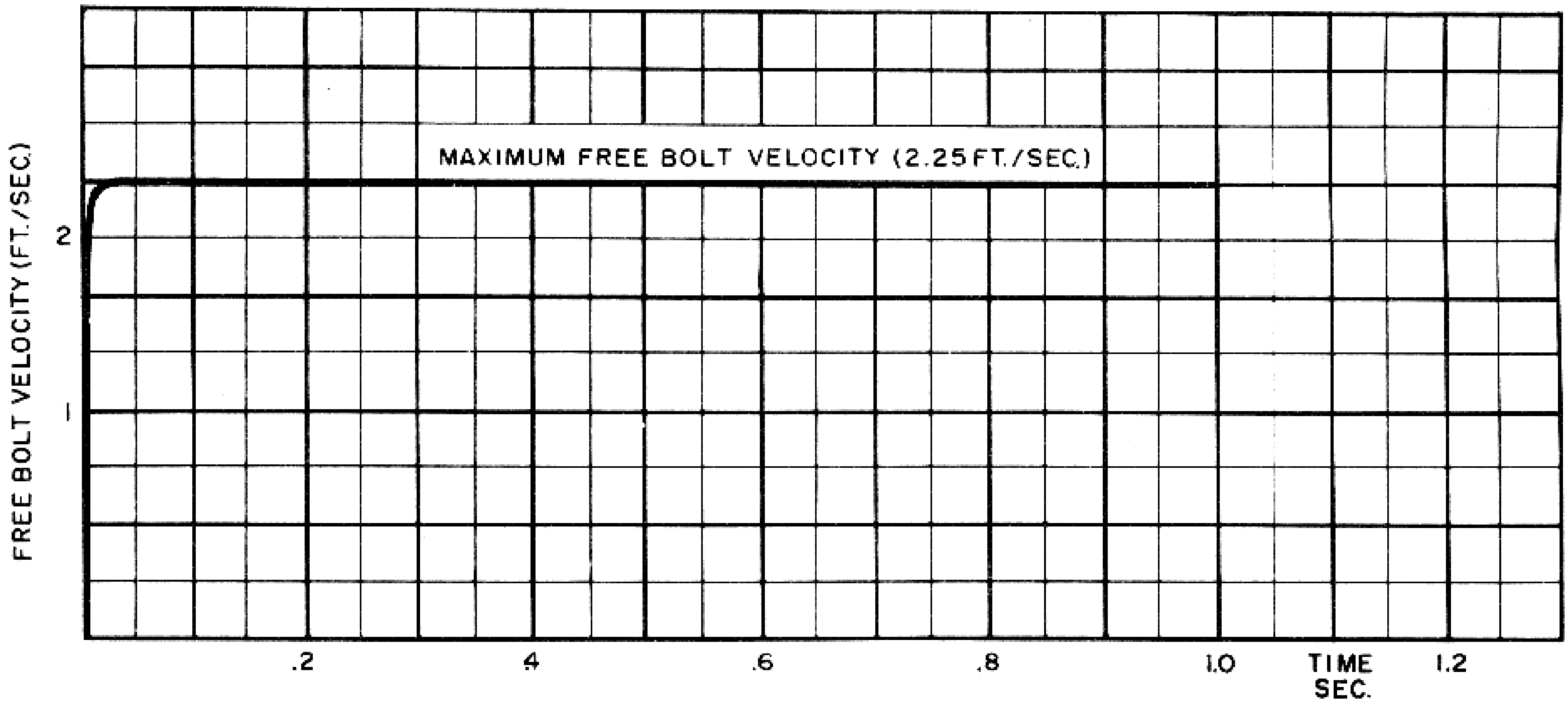


Figure 1–15. Free Recoil Velocity Versus Time (0–1.0 Sec.).

of the force and the time for which it is applied). Solving for dv gives:

$$dv=\frac{Fdt}{M}$$

To obtain the variation of the change in velocity with respect to time, this expression is integrated.

$$v=\int_0^t \frac{Fdt}{M}=\frac{1}{M}\int_0^t Fdt \qquad (1\text{–}15)$$

In accordance with equation 1–15, the retarding effect of a force on a given mass can be determined as follows:

1. Plot a curve showing the variation of the force with respect to time.
2. Measure the area under the curve between $t=0$ and some time t_1.
3. Divide the measured area by the mass. This gives the ordinate of the retardation curve for the time t_1.
4. Repeat 2 and 3 for other values of t and plot the retardation curve.

When this procedure is applied to the driving spring and bolt, the resulting curve shows the loss in bolt velocity resulting from the resistance of the spring. Since the free bolt velocity curve shows the gain in bolt velocity resulting from the thrust of the powder gases, the difference between the curves will be the net bolt velocity, or in other words, the velocity of retarded recoil.

If the retarding force were constant or if its variation with respect to time were known, the application of this method would be very simple. However, when the force varies with bolt displacement as it does in the case of the driving spring, a difficulty is encountered. In order to plot a graph showing the variation of the retarding force with respect to time, it is necessary to have a curve showing the variation of bolt displacement with respect to time and it is this very curve that we are attempting to plot.

This difficulty can be circumvented by considering the problem in two stages. For the first 0.010 second while the powder gas pressures are acting, the retardation offered by the spring will be small and in any case will be almost entirely due to the constant effect of the initial compression. The varying force due to the spring rate during this interval will almost certainly be negligible but, if necessary, it can be approximated very closely. These considerations make it possible to obtain accurate results for the first 0.010 second. For the remainder of the cycle of operation, the powder gas pressures are not acting and the displacement of the bolt can be determined analytically without any trouble.

The procedure for plotting the velocity and displacement curves for the first 0.010 second is as follows:

1. Plot curve of free bolt velocity versus time (fig. 1–14).

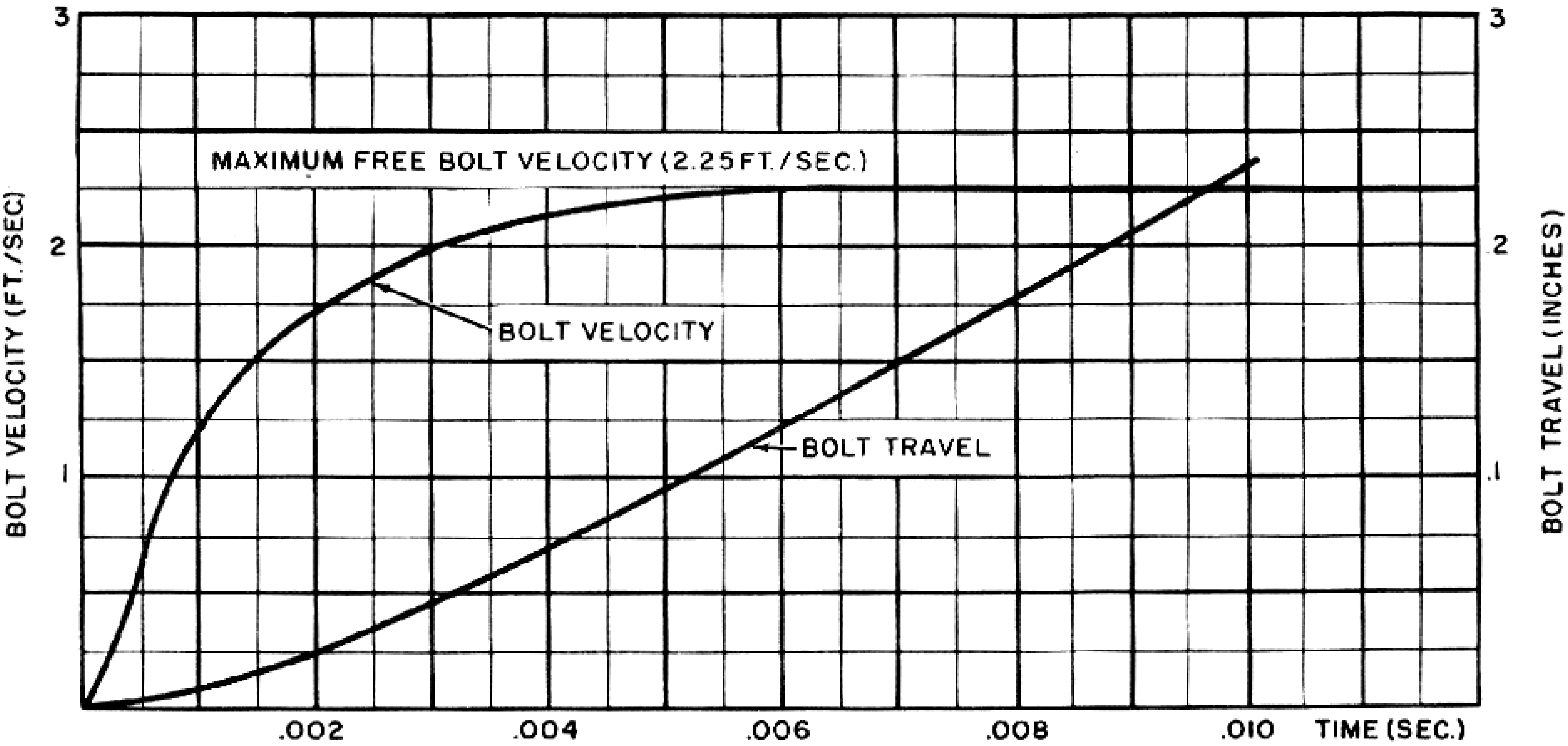


Figure 1–16. Time-Travel and Time-Velocity Curves (0–0.010 Sec.).

2. The loss in velocity due to the initial compression of the driving spring is equal to $F_o t/M_r$. Determine the velocity loss for various values of t, subtract each from the corresponding ordinate of the free bolt velocity curve and draw a curve through the resulting points. If the effect of the spring rate proves to be negligible, this curve is the retarded velocity curve.
3. Integrate under the curve drawn in step 2 to obtain the displacement curve.
4. Assume that the curve drawn in step 3 represents the actual bolt displacement curve and use this curve to determine the retardation due to the spring rate K. Ordinarily, it will be found that this retardation is so small that it will not have any effect worthy of consideration.
5. If the retardation determined in step 4 is sufficient to affect the velocity, use it to modify the curve drawn in step 2, and then integrate under the new curve to obtain a corrected displacement curve.
6. Steps 4 and 5 can be repeated as often as is necessary until no significant change occurs in the displacement curve. Actually, this process of successive approximation should never be necessary and satisfactory results should be achieved in the first three steps or at least in the first five steps.

Fig. 1–16 shows the curves obtained for the gun of the example. In this particular design, the spring is so weak that its retardation effects are entirely negligible during the first 0.010 second. To illustrate this point, the loss in velocity due to F_o over this interval is

$$V=\frac{F_o t}{M_r}=\frac{17.2\cdot .01\cdot 32.2}{500}=.0111\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to K, as determined by the method of step 4 is only about 0.0004 feet per second. Thus, in this gun the retarded velocity curve for the first 0.010 second is practically identical with the free velocity curve.

The remainder of the bolt displacement curve can now be determined analytically by using equation 1–10:

$$d=\frac{KD+F_o}{K}\sin\left[\sqrt{\frac{K}{M_r}}\,t+\sin^{-1}\frac{F_o}{KD+F_o}\right]-\frac{F_o}{K}$$

However, since some bolt travel (d′) occurred during the first 0.010 second, the values of F_o, D, and t must be changed to take this motion into account and d′ must be added to the resulting values obtained for d. The changed values to be used in equation 1–10 are:

$$F_o'=F_o+Kd'$$

$$D'=D-d'$$

$$t'=t-.010$$

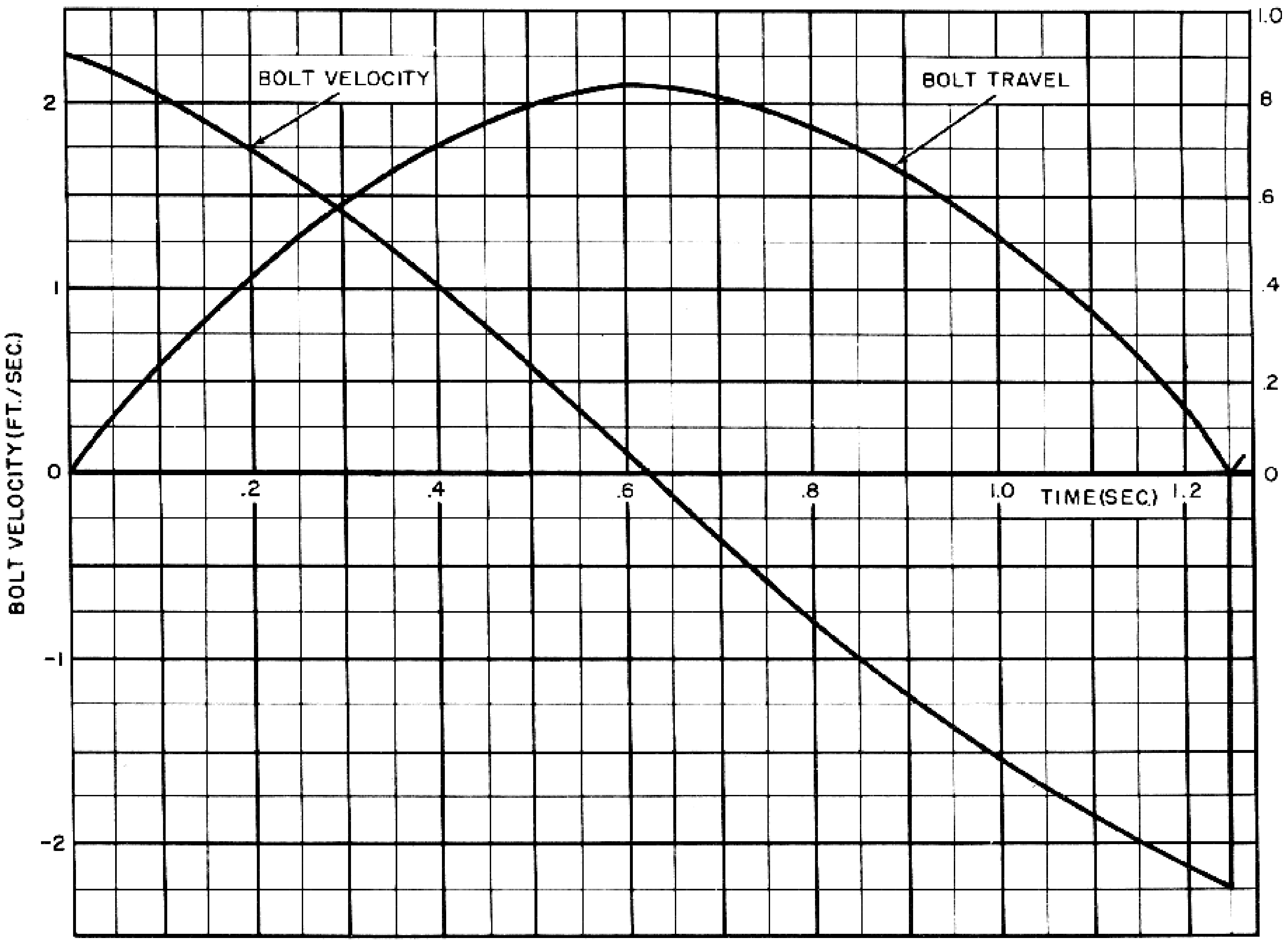


Figure 1–17. Time-Travel and Time-Velocity Curves (Complete Cycle).

The modified form of equation 1–10 is now:

$$d=\frac{K(D-d')+F_o+Kd'}{K}\sin\left[\sqrt{\frac{K}{M_r}}(t-.010)+\sin^{-1}\frac{F_o+Kd'}{K(D-d')+F_o+Kd'}\right]-\frac{F_o+Kd'}{K}$$

$$d=\frac{KD+F_o}{K}\sin\left[\sqrt{\frac{K}{M_r}}(t-.010)+\sin^{-1}\frac{F_o+Kd'}{KD+F_o}\right]-\frac{F_o+Kd'}{K}+d'$$

This equation is employed to complete the bolt displacement curve. The ordinates of the displacement curve are then multiplied by K and increased by F_o to obtain a curve showing the variation of the total spring force with time. Integrating under this curve and dividing by M_r in accordance with equation 1–15 will give a graph of the loss in velocity due to the spring force. Subtracting this curve from the free bolt velocity curve will give the complete graph for the retarded bolt velocity.

Fig. 1–17 shows the displacement and velocity curves obtained by this method for the gun of the example. After the necessary substitutions are made in equation 1–10, the final form of the equation to be used after the first 0.010 second is:

$$d=1.072\sin[2.15t+.223]-.239$$

The spring force curve obtained from the displacement curve of fig. 1–17 is shown in fig. 1–18.

If so desired in the course of a design, the effects of friction and loads incident to operating the gun mechanism may be taken into account in plotting the displacement and velocity curves. These forces are handled in the same way as the spring force. For example, the friction force resisting bolt motion will be essentially constant and therefore can be taken into account by increasing F_o in equation 1–10. A constant or varying load which exists for only a small portion of the cycle (such as the force

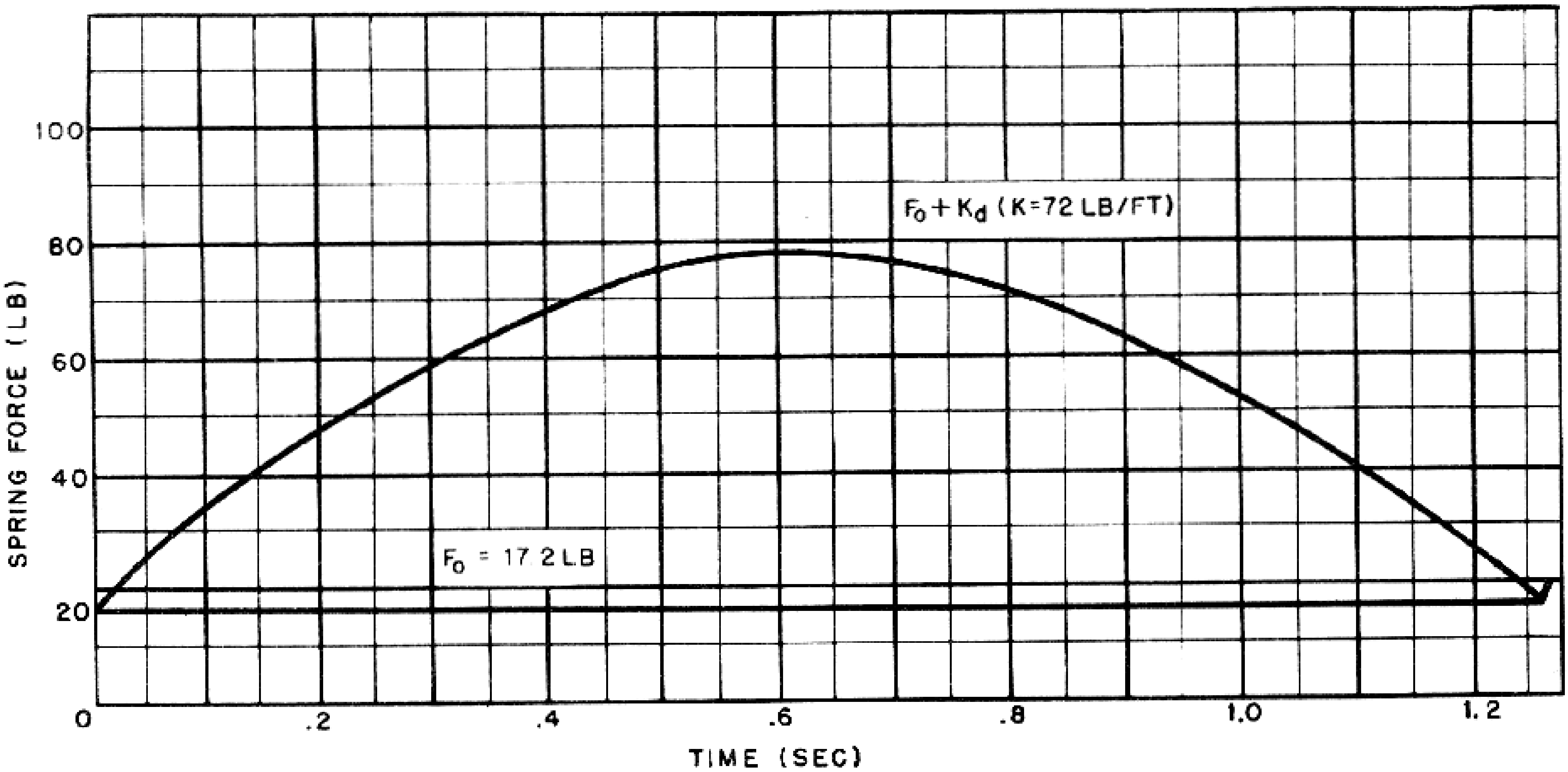


Figure 1–18. Variation of Spring Force With Time.

required to remove a cartridge from the feeder) can be treated in a like manner providing that the problem is considered in stages by methods similar to those described in the preceding paragraphs.

Another useful curve for design and analysis purposes may be obtained by plotting bolt velocity versus displacement. This curve may be drawn easily because the velocity and displacement curves can be used to determine the velocity corresponding to any displacement. Fig. 1–19 shows the velocity-displacement curve for the gun of the example.

5. *Note concerning driving springs*

In paragraph 2, it was pointed out that the selection of F_o and K for a driving spring will have a significant effect on the rate of fire. This may be seen by considering the time to recoil for the extreme conditions in which (a) the initial compression is zero, and (b) the spring constant is zero and only a constant force retards the recoil.

The expression for the time to recoil when the initial compression is zero can be obtained directly from equation 11 by setting $F_o=0$. This gives:

$$T=\sqrt{\frac{M_r}{K}}\frac{\pi}{2} \tag{1–16}$$

The expression for the time to recoil when the spring constant is zero can not be obtained from equation 1–11 because substituting K=0 yields an indeterminate expression. However, the expression can be derived by substituting K=0 in equation 1–8 and solving the resulting differential equation. This gives the relation between recoil distance and time as:

$$T=\sqrt{\frac{2M_r}{F_o}}\left[\sqrt{D}-\sqrt{D-d}\right] \tag{1–17}$$

When t=D, d=D and therefore:

$$T=\sqrt{\frac{2M_rD}{F_o}} \tag{1–18}$$

For the conditions of equation 1–16, $K=2\ F_{av}/D$, where F_{av} is the average spring force over the distance D. For the conditions of equation 1–18 $F_o=F_{av}$. Making these substitutions in equations 1–16 and 1–18 gives the following results:

For $F_o=0$	For $K=0$
$T=\sqrt{\frac{M_rD}{2F_{av}}}\frac{\pi}{2}$	$T=\sqrt{\frac{2M_rD}{F_{av}}}$
$=\sqrt{\frac{M_rD}{F_{av}}}\frac{\pi}{2\sqrt{2}}$	$=\sqrt{\frac{M_rD}{F_{av}}}\sqrt{2}$

Since that average spring force must be the same in both cases, the time to recoil is greater for the condition when K=0 by a factor of

$$\frac{\sqrt{2}}{\pi/2\sqrt{2}}=\frac{4}{\pi}=1.27.$$

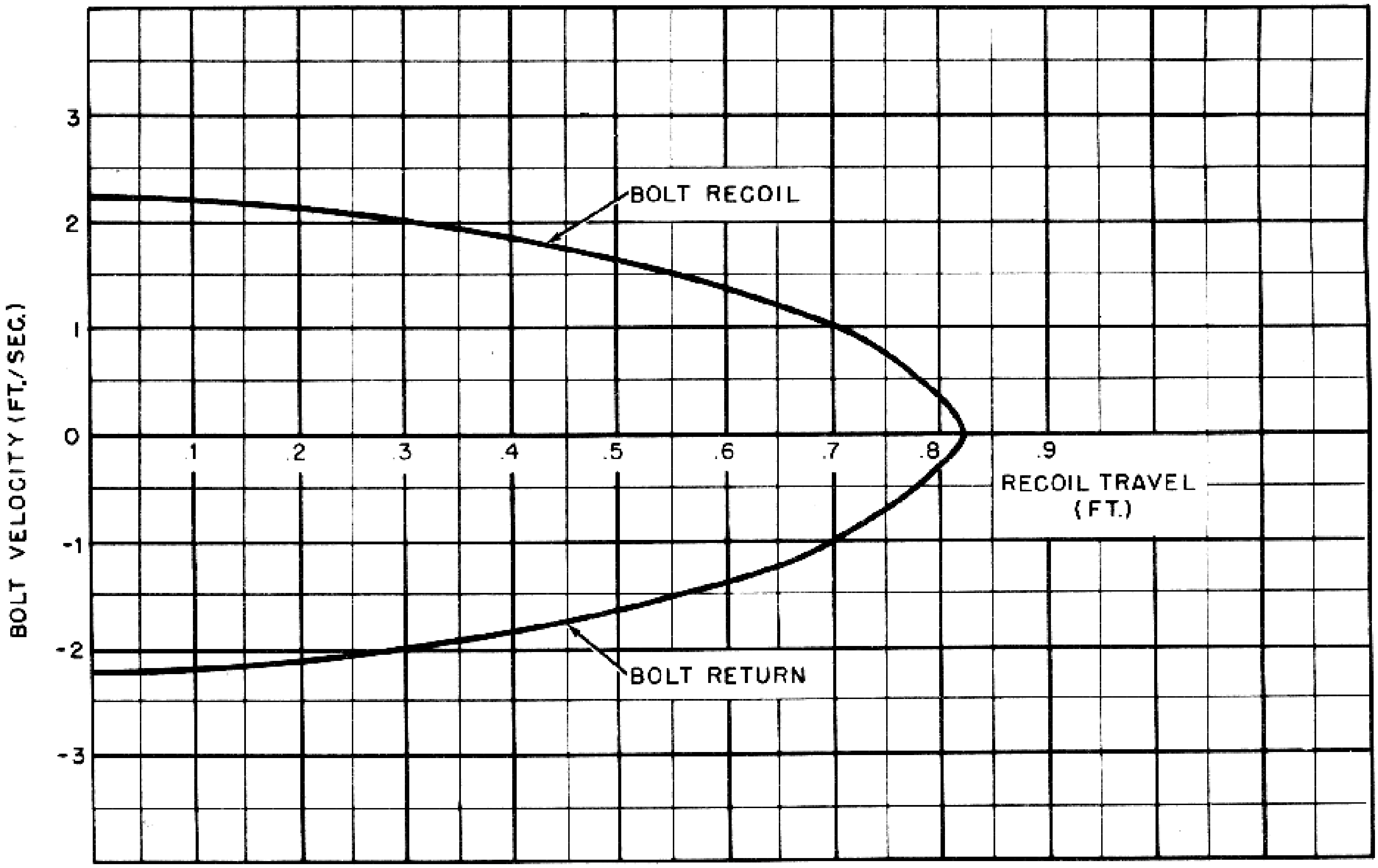


Figure 1–19. Velocity-Displacement Curve.

The condition in which neither K nor F_0 is zero will give a time to recoil somewhere between these two extremes, depending on the combination selected. This indicates that in the design of the driving spring, F_0 should be made as small as possible in order to improve the rate of fire.

BLOWBACK SYSTEM WITH ADVANCED PRIMER IGNITION

In the plain blowback system, the bolt returns to the firing position with relatively low velocity but with considerable kinetic energy and this energy is absorbed by impact before the next cycle starts. Since the bolt is stationary when the new round is fired, all of the explosive force of the round is effective in accelerating the bolt to the rear. As has been explained, this condition requires the use of an extremely heavy bolt in order to keep the bolt velocity within safe limits.

A substantial saving in bolt weight and other advantages can be realized by making use of the kinetic energy of the returning bolt. Instead of permitting this energy to be dissipated by impact before the next round is fired, it is possible to time the ignition so that the new round is fired just before the bolt reaches its fully forward position. In this method of operation, known as "advanced primer ignition," the impulse of the propellant explosion must first slow and stop the returning bolt before it can propel the bolt to the rear. With this action, only a portion of the explosive impulse is effective in blowing back the bolt and the interval of time during which the pressure of the powder gases acts to produce a rearward acceleration of the bolt is also reduced. Both of these effects permit the use of a much lighter bolt and produce a condition in which higher bolt velocities are allowable. Thus, not only can the gun be lighter, but it is also possible to achieve a higher rate of fire.

The form of the mechanism for a blowback gun employing advanced primer ignition (fig. 1–20) is basically the same as for a plain blowback gun except that the firing device shown in the figure

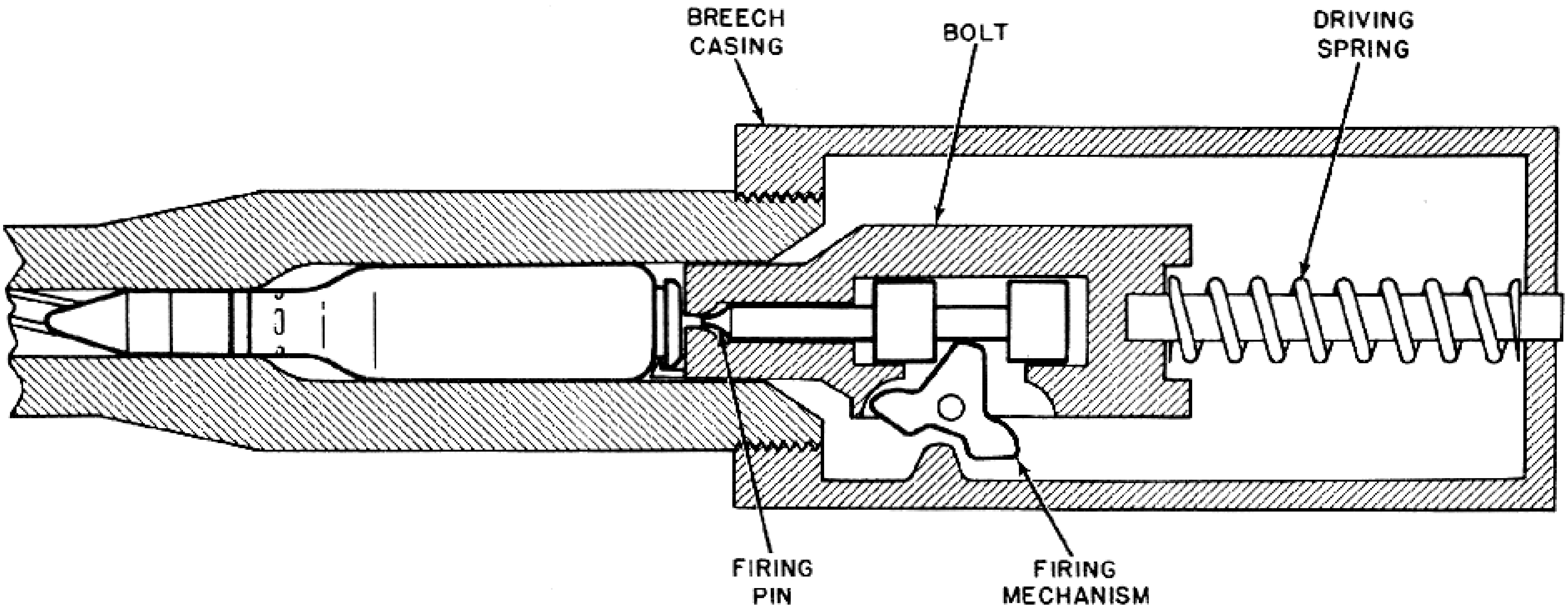


Figure 1–20. Simplified Schematic of Blowback Mechanism With Advanced Primer Ignition.

is arranged to ignite the primer of the fresh cartridge just before the bolt reaches its fully forward position. Although the details of an actual gun of this type may be considerably different in form from the schematic representation shown in fig. 1–20, the basic mechanism shown illustrates the essential mechanical function. The fundamental parts of the mechanism are the bolt (which backs up the cartridge case and is free to slide in the breech casing) and the driving spring (which absorbs the kinetic energy of the bolt when the bolt is blown back and then drives the bolt back to the firing position).

Cycle of Operation

The automatic cycle for a blowback gun with advanced primer ignition occurs as follows:

The cycle starts with the bolt seared to the rear and with the driving spring compressed. When the sear is released, the bolt is thrust forward by the driving spring and as it moves forward, the bolt picks up a fresh cartridge from the feed mechanism and carries this cartridge into the chamber. When the cartridge has entered the chamber sufficiently that the case walls are adequately supported but before the bolt has reached its fully forward position, the firing device is actuated to ignite the primer.

The pressure resulting from the explosion of the propellant charge drives the projectile through the barrel of the gun and at the same time immediately exerts a force which acts through the base of the cartridge case to slow down the forward motion of the bolt, bring the bolt to a stop, and then propel the bolt to the rear. At the time the bolt is driven to the rear, the only significant resistance to the acceleration of the bolt is due to the inertia of the bolt mass.

The force exerted by the pressure of the powder gases lasts for only about 0.008 or 0.009 second after ignition of the primer. After this point, there is no further force driving the bolt to the rear but the recoiling parts continue to move of their own momentum. As the bolt moves back, the spent cartridge case is extracted from the chamber and ejected. The resistance of the driving spring gradually slows the bolt down until it has reduced the bolt velocity to zero. At this point, the cycle of operation is complete and a new cycle begins as the energy stored in the driving spring starts the bolt moving forward.

Analysis of Advanced Primer Ignition

The most important factor to be considered in analyzing the effects of using advanced primer ignition is the manner in which the energy of the returning bolt is utilized. If the relatively small losses to friction and to operating the mechanism are ignored, the driving spring will return the bolt with a final velocity which is almost equal to the initial velocity of recoil. The amount of impulse which must be supplied by the propellant explosion to stop the forward motion of the bolt will then be approximately equal to the amount of impulse required to produce the initial recoil velocity, or in other words, only about one-half the impulse produced by the explosion is effective in causing the recoil velocity. Therefore, when advanced primer ignition is employed, the momentum imparted to the bolt is only one-half of that which would be obtained with plain blowback. This means that from this standpoint alone, the bolt weight can be reduced by a factor of two.

In addition to the advantage mentioned in the preceding paragraph, the use of advanced primer ignition in a blowback gun results in other advantages which are related to the manner in which the cartridge case moves under the pressure of the powder gases. The general nature of this movement is as follows:

Immediately after ignition of the primer, that is, in the very early stage of the propellant explosion, the bolt is still moving forward and is thrusting the case into the chamber. Any frictional force which results from the high pressure inside the case and which tends to resist the forward movement will result in compression stresses in the case walls. These stresses are not troublesome because they do not tend to cause case separation and, in addition, the case is well inside the chamber so that its walls are adequately supported. On the other hand, when the forward motion has been stopped and the case starts to move to the rear, the factors which limit blowback operation become effective and they must be taken into account. These factors give rise to two considerations:

1. If no lubrication is provided, the peak chamber pressures will cause the cartridge case to seize in the chamber and therefore separation of the case will occur unless the bolt movement is limited so that the elongation of the case material is not exceeded while the case is stuck. As explained in the analysis of plain blowback, the precise elongation permissible for a specific cartridge case can be determined only by careful experimentation under actual firing conditions, but a good rule to follow for the brass cases of 20-mm rounds is that the bolt movement to the rear should not exceed 0.015 inch during the first 0.0015 second of the propellant explosion. With advanced primer ignition, the first 0.0009 second (approximately) of the explosion is utilized to stop the forward motion of the bolt so that the critical time during which case separation can occur endures for only 0.0006 second. Since a bolt movement of 0.015 inch is permissible during this time, the average allowable bolt velocity for the same time is 25 inches per second, or about 2 feet per second. This is twice as great as the velocity permissible with plain blowback but is still a very low value.
2. If the ammunition is lubricated in order to avoid chamber seizure, case separation will not occur but the motion of the bolt must still be limited to prevent the case from moving so far out of the chamber that the thin walls near the base are unsupported while there is any considerable residual powder gas pressure. As for plain blowback, this means that for a case of the type shown in fig. 1–6A, the travel in the first 0.010 second of the propellant explosion should not exceed 0.250 inch. Since the rearward motion does not start until 0.0009 second has elapsed, this movement will occur in 0.0091 second and therefore the allowable average velocity is only 27.3 inches per second, or at the most 2.5 feet per second (instead of 2 feet per second as is allowable with plain blowback).

From the foregoing it is apparent that although the use of advanced primer ignition permits slightly higher bolt velocities than are attainable with plain blowback, the advantage in this direction is not very great with the type of ammunition shown in fig. 1–6A. The increase in permissible bolt velocity will produce a somewhat greater rate of fire and will also result in a further decrease in the bolt weight necessary. This saving in bolt weight, coupled with the fact that the bolt weight is reduced by a factor of two because only one-half the explosive impulse is effective in producing recoil, will give a total weight reduction factor of somewhere between 2 and 2.5. In other words, instead of the 500-pound bolt required for the 20-mm gun used as an example in the analysis of plain blowback, the bolt weight for a 20-mm blowback gun with advanced primer ignition would be somewhat over 200 pounds. Although this represents a great improvement over plain blowback, a gun having such a heavy bolt would still not be practical, particularly since the rate of fire will not be much greater than the 48-round-per-minute rate attainable with the plain blowback gun used as an example.

To make full use of the potentialities of advanced primer ignition, it is necessary to use lubricated cartridge cases having the special base form shown in fig. 1–6B. With this form, the diameter of the base is smaller than the maximum case diameter so

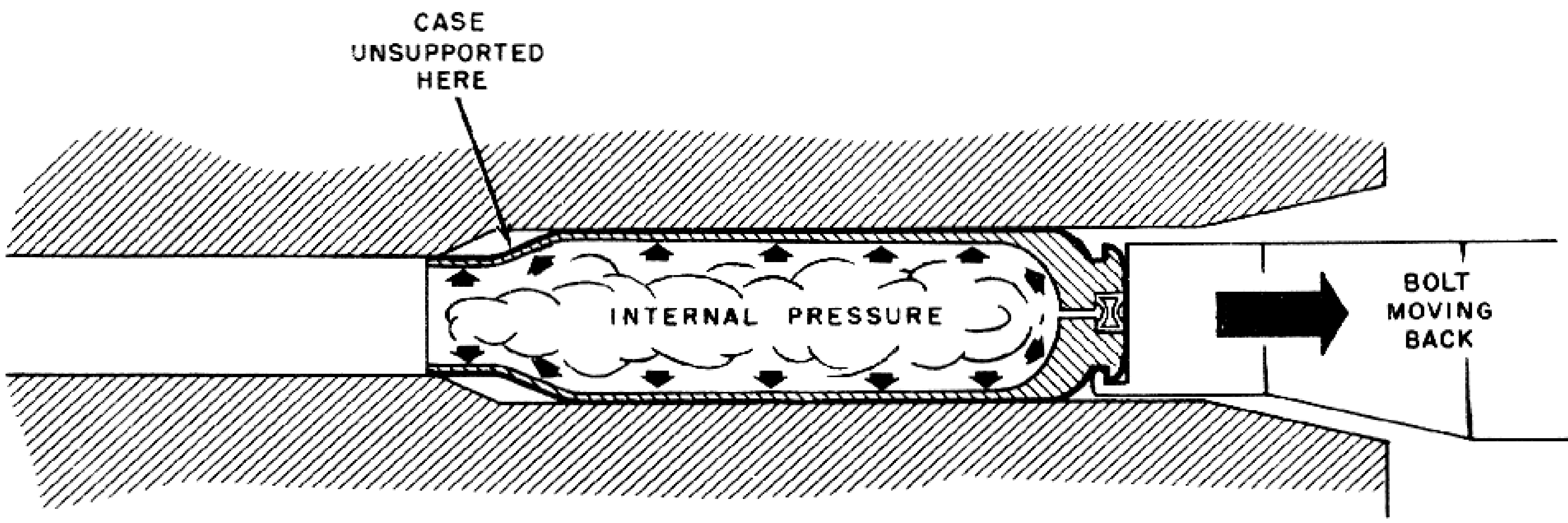


Figure 1–21. Deformation of Bottle-Necked Case.

that the bolt and extractor can follow the case into the chamber, which is made especially long. When the case is blown back, it can travel a considerable distance before the walls near its base become unsupported and therefore a higher bolt velocity is permissible. In an existing 20-mm gun employing advanced primer ignition, the walls near the base of the cartridge case are quite heavy and the case is thrust deeply into the chamber with the result that the case can be blown back nearly two inches in the first 0.010 second without causing rupture near the base. Since the time for this movement is approximately 0.0091 second, the allowable average recoil velocity for this interval is about 220 inches per second or approximately 18 feet per second. This recoil velocity results in a weight of less than 30 pounds for the recoiling parts and in a rate of fire of between 400 and 500 rounds per minute. Although this is not an exceptionally high cyclic rate for a 20-mm gun, a design having the characteristics mentioned is thoroughly sound and practical for some applications.

It is appropriate here to mention some general points concerning the use of ammunition of the type considered in the preceding paragraph. With a cartridge case of the form shown in fig. 1–21, the bottle-necked portion at the front of the case is not supported by the chamber while the case moves under the pressure of the powder gases. Therefore, the internal pressure will tend to push the neck forward and will also tend to expand the mouth of the case. This tendency is so pronounced in some guns that the extracted cases are deformed to the point where the bottle-neck is entirely gone and the cases become practically cylindrical. Because of this extreme deformation, the mouth of the case will sometimes split open. It also should be noted that the use of ammunition which can move well inside the chamber makes the problem of obtaining precisely timed ignition of the primer less critical than it would be for conventional ammunition of the form shown in fig. 1–6A. With a 20-mm cartridge of this conventional type, only a 0.250-inch rearward travel is permissible during the action of the gas pressure and to obtain satisfactory performance the ignition must be timed so that the forward motion of the case will be stopped very close to the position shown in the figure. If ignition is too late, the bolt will strike the rear face of the barrel. If ignition is too early, the case will not enter the chamber far enough and when blown back it will not receive support for the full 0.250 inch. Since the total permissible travel under pressure is only 0.250 inch, a difference of only a few hundredths of an inch in the point at which ignition occurs can have a serious effect on performance and the ignition must therefore be timed very precisely. With ordinary percussion primers and simple firing devices, such precision is difficult to obtain. However, with ammunition of the form shown in fig. 1–6B (for which the permissible movement under pressure can be nearly two inches), extreme precision in timing is not necessary and a variation even as great as $\frac{1}{8}$ inch in the position at which ignition occurs for successive rounds will not have any unfavorable effect on performance.

All things considered, the use of advanced primer ignition is an effective means for obtaining satisfactory performance in an action based on the blow-

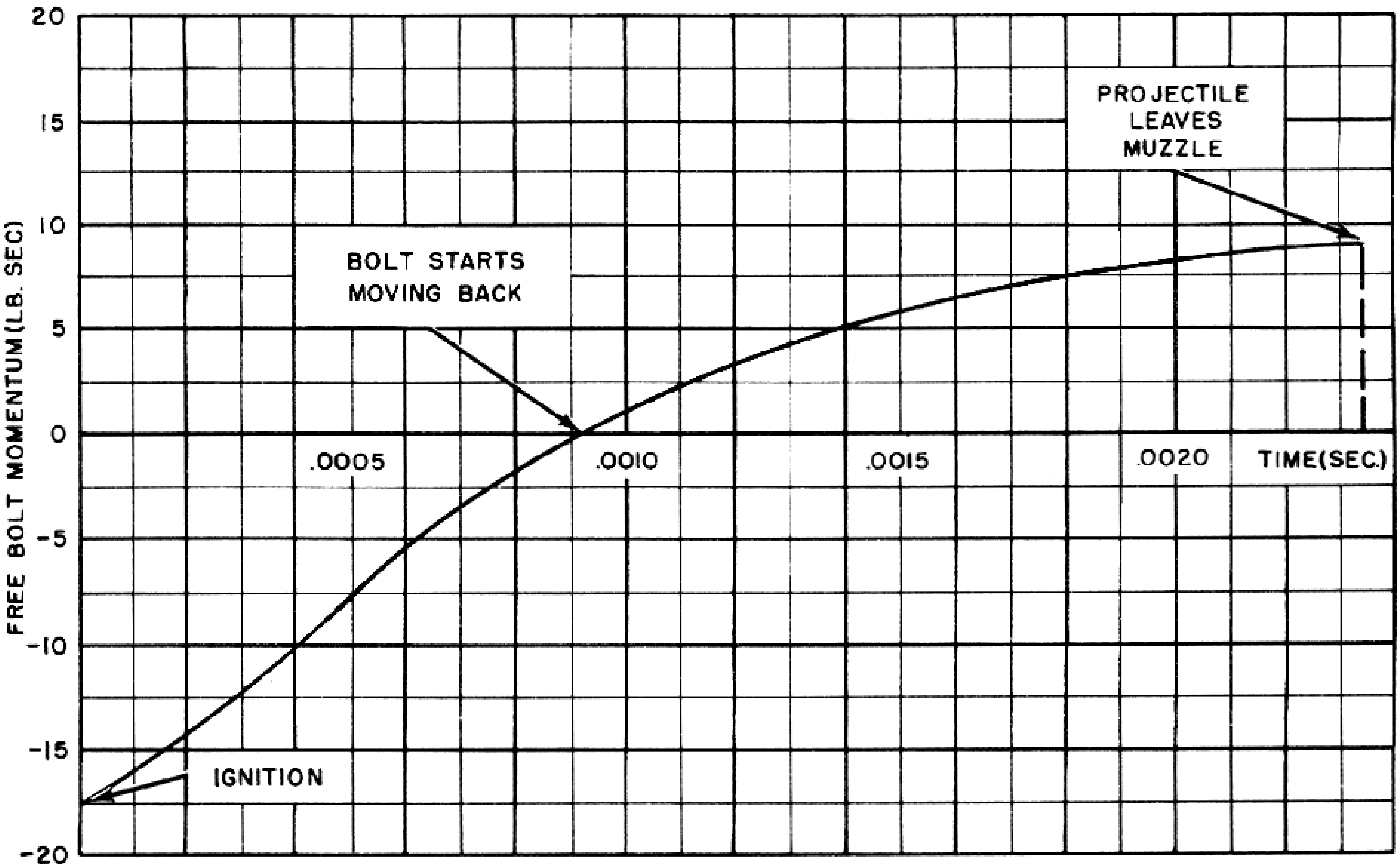


Figure 1–22. Free Bolt Momentum for First 0.00234 Second.

back principle. However, when comparing advanced primer ignition with plain blowback it must be emphasized that the advantages gained by making use of the momentum of the returning bolt are somewhat limited when this factor alone is considered. With all other things being equal, these advantages are as follows: First, the weight of the recoiling parts in an advanced primer ignition gun can be less than half of that in a plain blowback gun. Second, slightly higher bolt velocities are allowable in an advanced primer ignition gun and therefore a somewhat greater rate of fire can be attained than with plain blowback.

It should be realized that these advantages alone are not sufficient to permit the design of a practical gun employing conventional ammunition. Reasonable performance can be obtained only by using ammunition with the special base form previously described. Of course, if this ammunition were used in a plain blowback gun, the bolt weight could be considerably reduced and a higher rate of fire could be obtained but the weight would still be too great and the cyclic rate would still be too low for the design to be practical. The use of advanced primer ignition, however, further reduces the bolt weight and further increases the rate of fire to the point where a practical design can be achieved.

Mathematical Analysis of Advanced Primer Ignition

The following mathematical analysis of advanced primer ignition is based on the same general principles used for the analysis of plain blowback. In this analysis, it is assumed that the same ballistic data used for plain blowback are available. (See figs. 1–9, 1–10, and 1–11.) Since many of the methods and formulas employed in this analysis are the same or are very similar to those used for plain blowback, the derivations of the formulas and the explanation of the procedures will not be repeated here. However, as they arise, any new concepts or new formulas will be explained.

1. *Determination of bolt weight*

The free momentum imparted to the bolt by the impulse created in the explosion of the propellant is

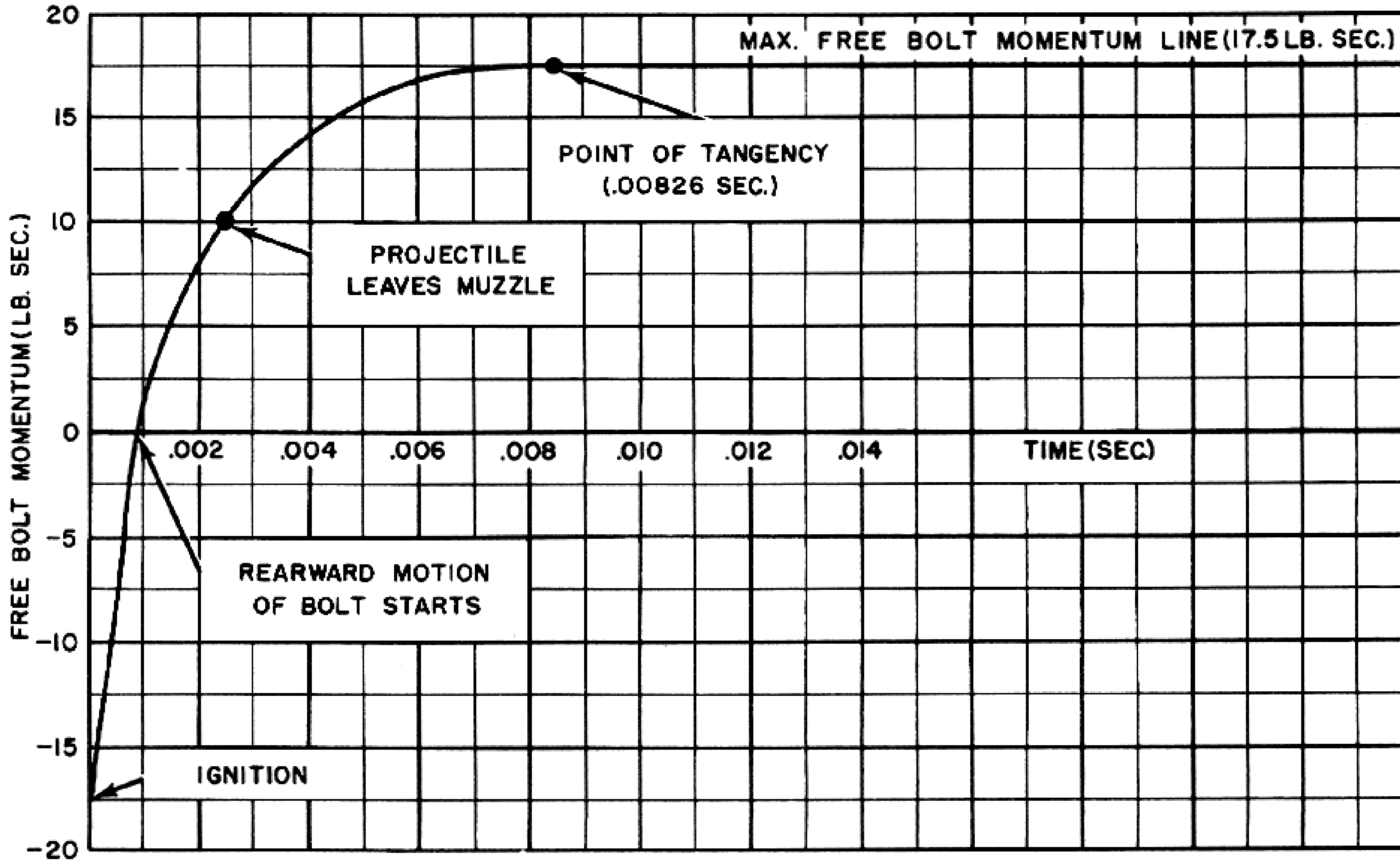


Figure 1–23. Free Bolt Momentum for First 0.010 Second.

exactly the same for advanced primer ignition as for plain blowback. However, it must be remembered that at the instant the propellant charge is ignited, the bolt is moving forward and has a momentum equal to one half the total change in momentum which is produced by the propellant explosion. At any instant after ignition of the propellant, the free momentum of the bolt will be the sum of its initial momentum and the change in momentum produced by the impulse of the propellant explosion.

As has been shown previously, the total change in momentum produced by the propellant explosion is expressed by equation 1–2 as:

$$M_pV_p+4700\ M_c$$

Therefore the initial forward momentum of the bolt is

$$\frac{1}{2}\ (M_pV_p+4700\ M_c)$$

Since this momentum is directed forward, its sign is negative. Before the projectile leaves the muzzle, the momentum change produced by the impulse of the propellant explosion is:

$$\left(M_p+\frac{M_c}{2}\right)v_p$$

Accordingly, the free momentum of the bolt at any instant during this time will be the sum of its initial momentum and the change in momentum resulting from the impulse of the propellant explosion. That is:

$$(1\text{–}19)\quad M_rv_{r_f}=\left(M_p+\frac{M_c}{2}\right)v_p-\frac{1}{2}\ (M_pV_p+4700\ M_c)$$

Equation 1–19 can be used to plot a graph of free bolt momentum versus time for the period before the projectile leaves the muzzle. For the cartridge and barrel on which fig. 1–10 is based, the equation is evaluated as follows:

$$M_rv_{r_f}=\frac{1}{32.2}\left[\left(.29+\frac{.070}{2}\right)v_p-\frac{1}{2}\ (.29\times2750+4700\times.070)\right]$$

$$=.00101\,v_p-17.5$$

The curve obtained by using this relation is shown in fig. 1–22. The same curve is also shown in

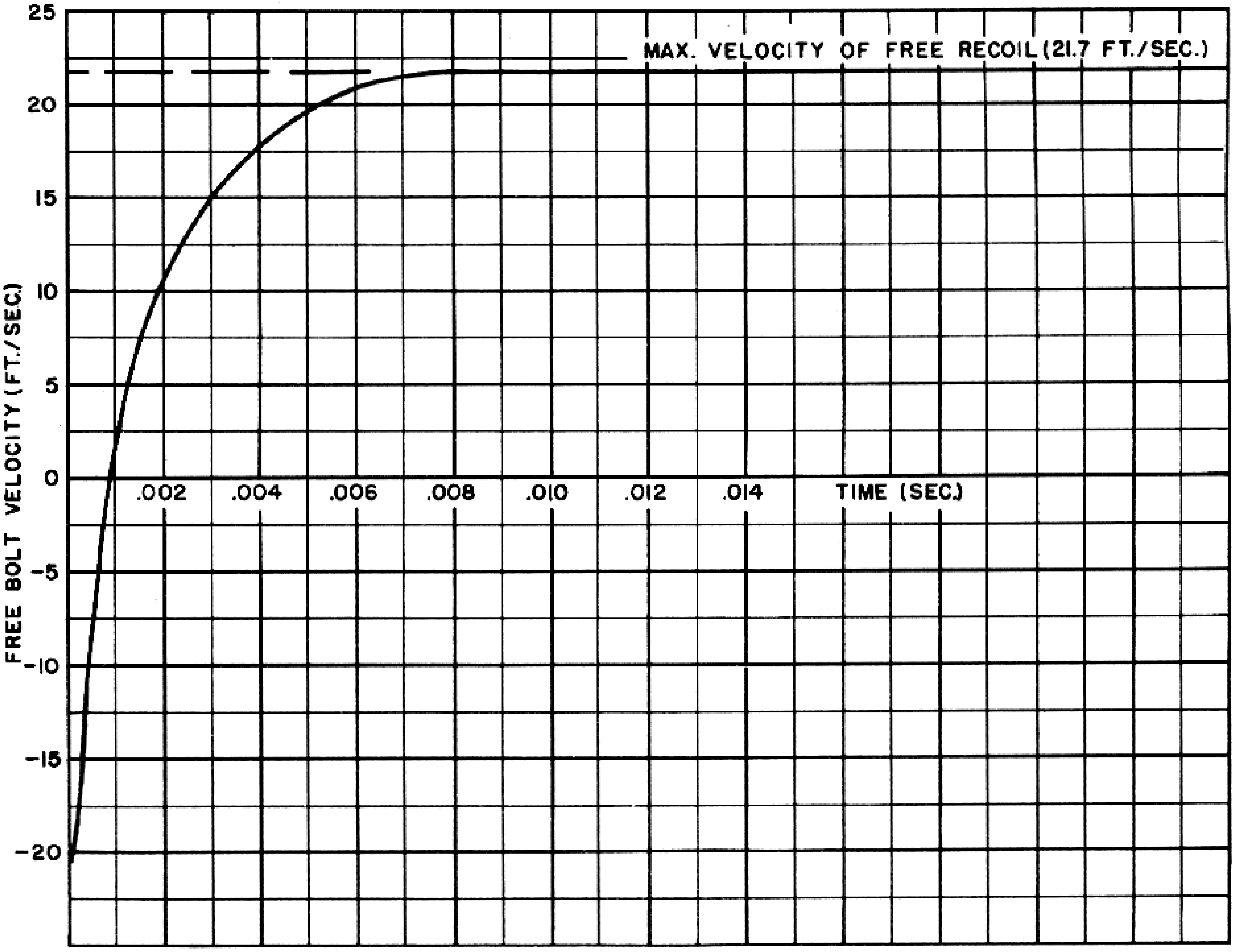


Figure 1–24. Free Bolt Velocity Curve for First 0.014 Second.

fig. 1–23 as the portion between $t=0$ and $t=0.00234$ second. Note that these curves have the same form as the corresponding curves for plain blowback (figs. 1–12 and 1–13) except that the zero axis is shifted upward by 17.5 (lb. sec.) which is half the total change in momentum produced by the impulse of the propellant explosion.

The momentum curve for the time after the projectile leaves the muzzle is completed in the same way as for plain blowback. In this case the maximum free bolt momentum is given by the expression:

(1–20) $$M_r V_{r_f} = \frac{1}{2}\,(M_p V_p + 4700\, M_c)$$

For the conditions of the example:

$$M_r V_{r_f} = \frac{1}{2}\left(\frac{.29}{32.2}\,2750 + 4700\,\frac{.070}{32.2}\right) = 17.5 \text{ (lb. sec.)}$$

Fig. 1–23 is used to determine the bolt weight by using the relation expressed by equation 1–4:

$$M_r = \frac{\int_{t_1}^{t_2} (M_r v_{r_f})\,dt}{(t_2 - t_1)\, v_{r_{f(all)}}}$$

In this case, it will be assumed that the special ammunition shown in fig. 1–6B is to be used and that the allowable movement during the first 0.010 second is two inches. As fig. 1–23 shows, the rearward movement does not start until 0.0009 second has elapsed and therefore the time over which the two-inch movement can occur is 0.0091 second. This gives the allowable average velocity for the interval from $t=0.0009$ to $t=0.0010$ as:

$$v_{r_{f(all)}} = \frac{2}{12 \times .0091} = 18.30 \text{ (ft./sec.)}$$

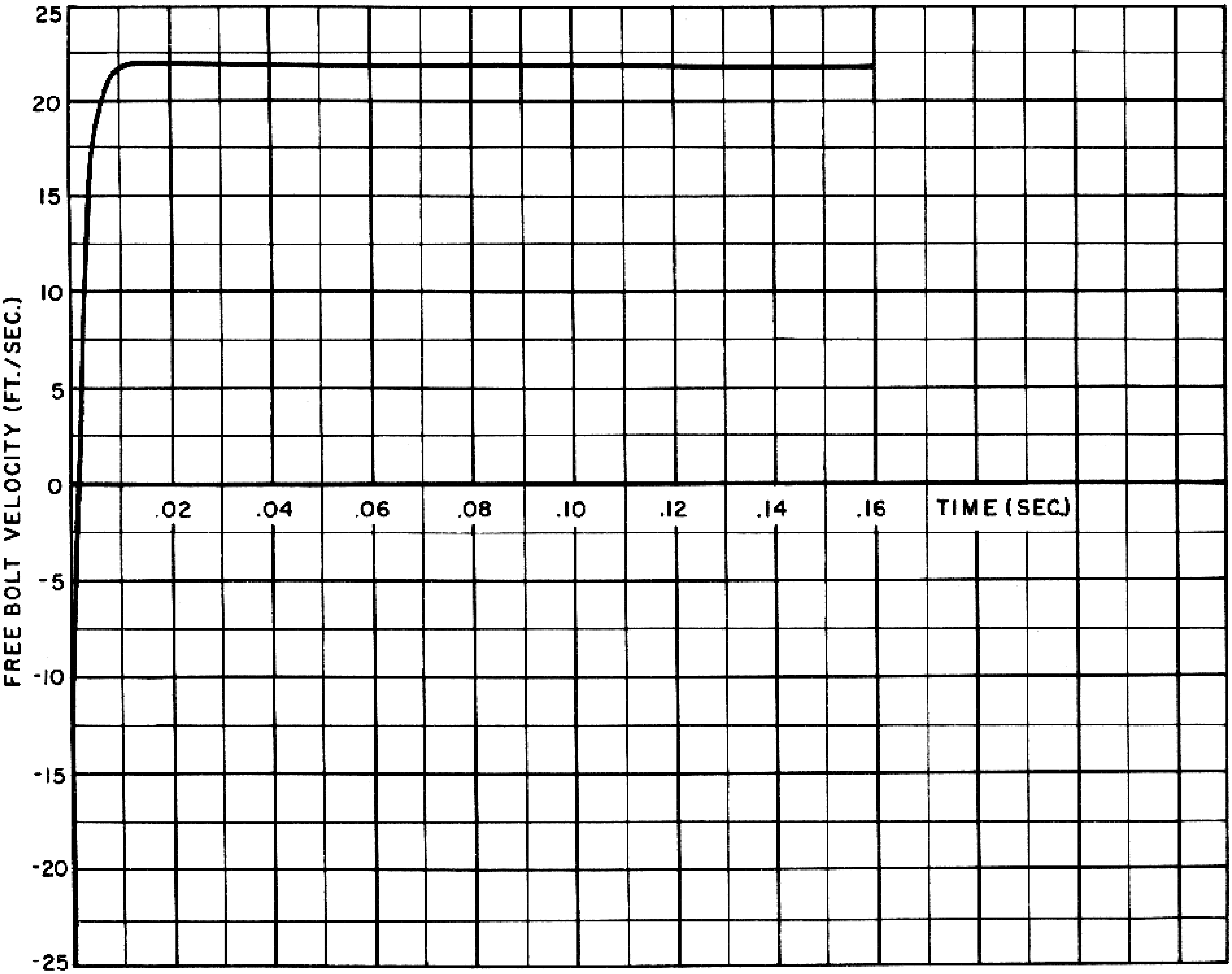


Figure 1–25. Free Bolt Velocity for 0.160 Second.

Integrating under the curve of fig. 1 23 from $t=0.0009$ to $t=0.0091$ gives a total area of 0.1310 (lb.-sec.²). Using equation 1–4 to evaluate M_r gives:

$$M_r=\frac{.1310}{.0091\times18.3}=.789\left(\frac{\text{lb. sec.}^2}{\text{ft.}}\right)$$

Therefore the required weight is:

$$W_r=gM_r=32.2\times.789=25.5 \text{ (lbs.)}$$

Accordingly, the weight of the recoiling parts will be taken as 26 pounds for the remainder of the analysis.

2. *Determination of driving spring design data*

The maximum velocity of free recoil is found in the same way as for plain blowback by using the equation expressing the maximum free recoil momentum to solve for the velocity.

$$\text{(1-21)}\qquad M_rV_{r_f}=\frac{1}{2}\,(M_pV_p+4700\,M_c)$$

$$V_{r_f}=\frac{M_pV_p+4700\,M_c}{2\,M_r}=\frac{W_pV_p+4700\,W_c}{2\,W_r}$$

For the 26-pound bolt and the conditions of the example:

$$V_{r_f}=\frac{.29\times2750+.070\times4700}{2\times26}=21.9\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

Again assuming that this velocity is imparted instantaneously to the bolt, the initial bolt energy is given by equation 1–6:

$$E_r=\frac{1}{2}\,M_rV_{r_f}^2=\frac{W_rV_{r_f}^2}{2g}\ \text{(ft. lb.)}$$

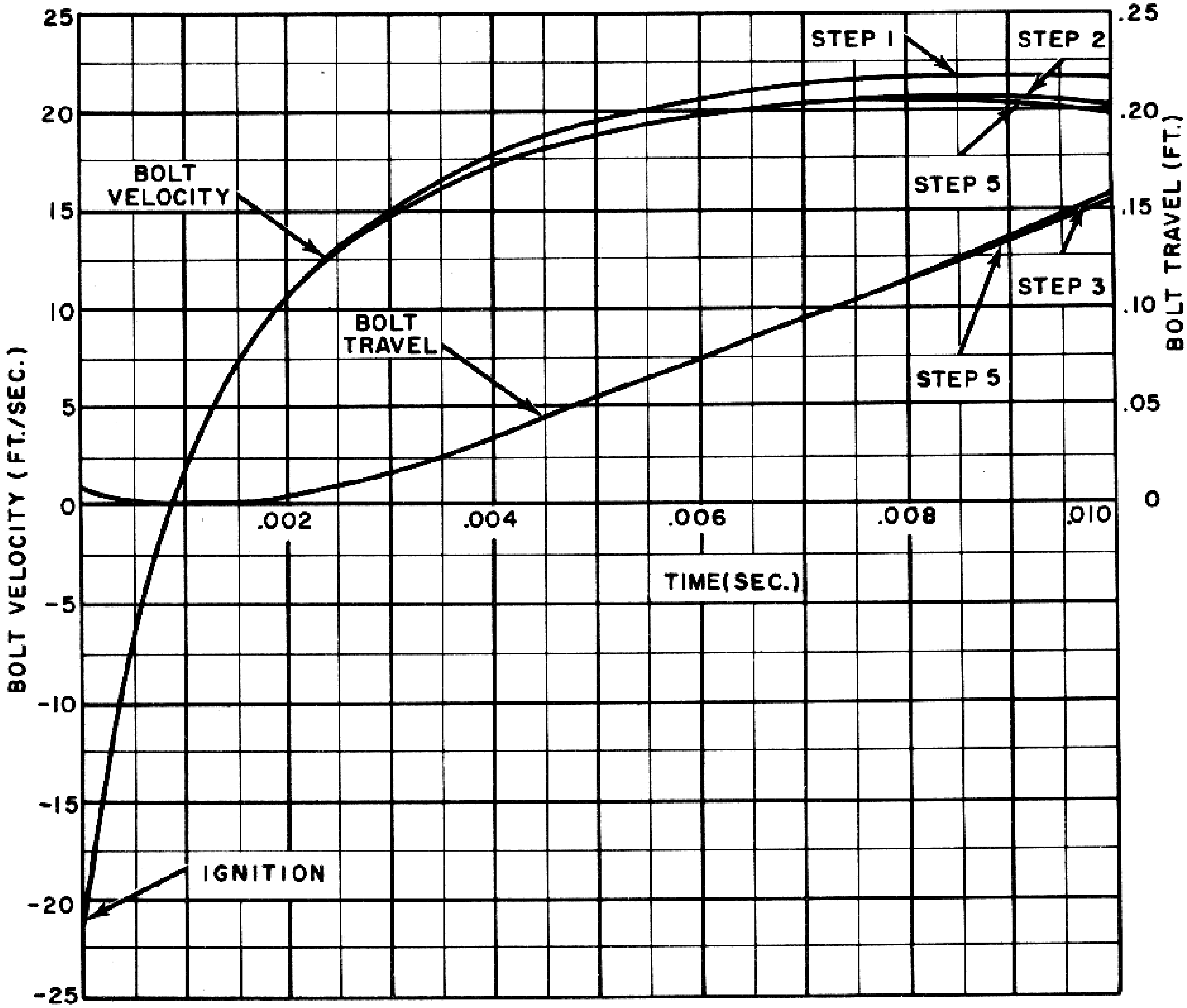


Figure 1–26. Time-Travel and Time-Velocity Curves for First 0.010 Second.

Evaluating this expression for the conditions of the example gives:

$$E_r = \frac{26 \times 21.9^2}{2 \times 32.2} = 194 \text{ (ft. lb.)}$$

The bolt driving spring must be proportioned so that it will absorb this amount of energy over the entire distance through which the bolt moves in recoil. The energy absorbed by a spring is equal to the distance through which it is compressed times the average force required to produce this deflection. That is:

$$E_r = F_{av} D \text{ or } F_{av} = \frac{E_r}{D}$$

If it is assumed that the bolt in the example must open 10 inches (0.833 feet) in order to permit feeding of a 20-mm cartridge, the average force exerted by the spring must be:

$$F_{av} = \frac{194}{.833} = 233 \text{ (lb.)}$$

In designing the spring so that it will produce this average force, the same factors described in the analysis of plain blowback should be considered. However, an arbitrary choice of spring characteristics will suffice for present purposes. If the initial compression is taken as 130 pounds, a maximum force of 336 pounds will produce the required average force of 233 pounds. Since the difference between the maximum force and the initial compres-

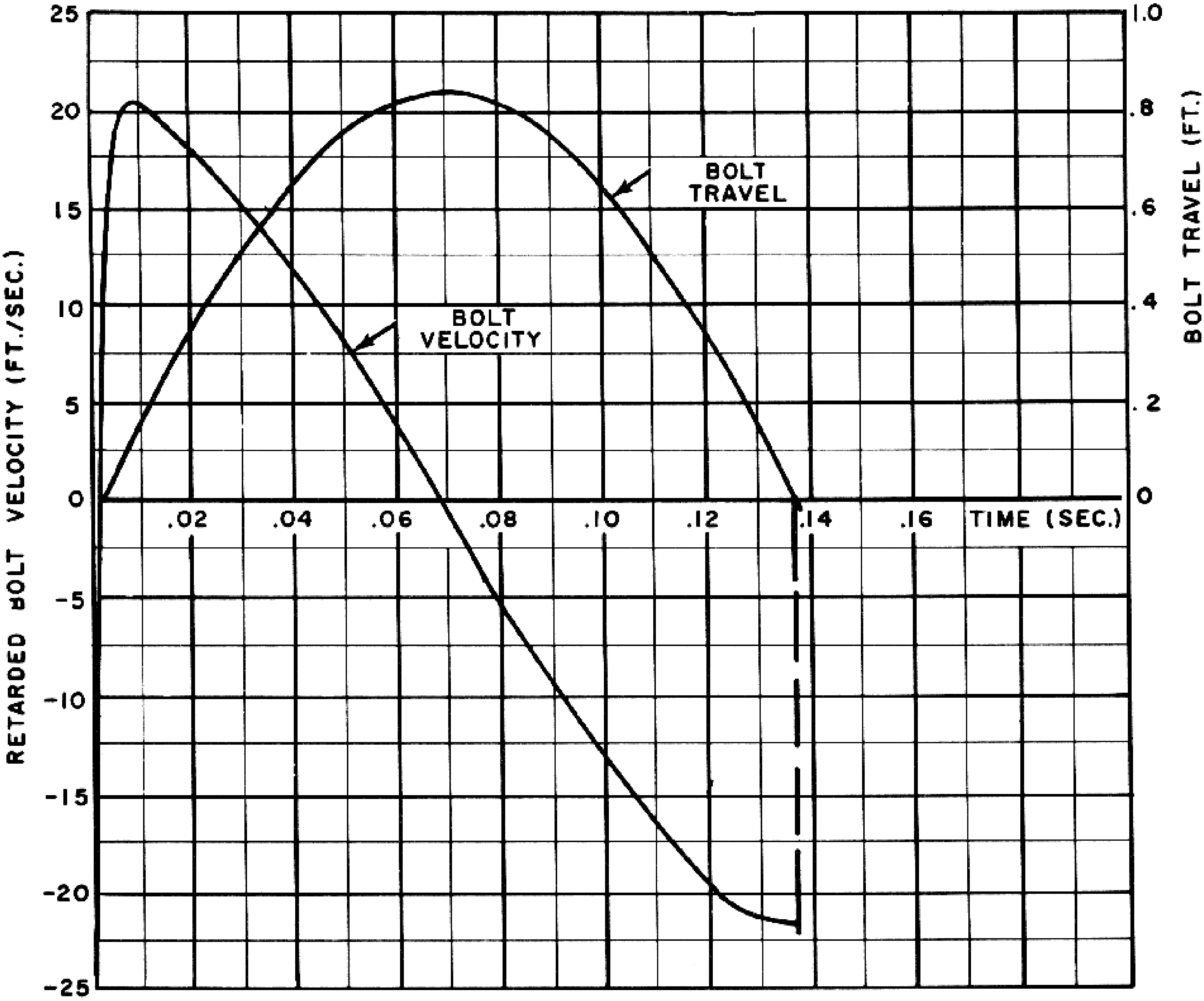


Figure 1–27. Time-Travel and Time-Velocity Curves (Complete Cycle).

sion is 206 pounds and the recoil distance is 10 inches, the spring rate will be 20.6 pounds per inch or 247 pounds per foot. Realizing that this choice is arbitrary, it will be assumed here that the selected values of $F_o=130$ pounds and $K=247$ pounds per foot will result in a satisfactory spring.

Since the bolt weight and the characteristics of the driving spring have been determined, the basic design of the gun is now complete and the remaining task is to consider what performance this design may be expected to give.

3. *Bolt motion equations*

Since the bolt motion equations derived for plain blowback are all based only on the amount of bolt energy and on the characteristics of the driving spring, these equations are not affected by the use of advanced primer ignition. (Of course, the values to be substituted in the equations are different, but the relationships between these values are not changed.) Therefore, the equations will not be derived again here but they are listed for ready reference:

$$(1\text{–}9)\qquad t=\sqrt{\frac{M_r}{K}}\left[\sin^{-1}\frac{Kd+F_o}{KD+F_o}-\sin^{-1}\frac{F_o}{KD+F_o}\right]$$

$$(1\text{–}10)\qquad d=\frac{KD+F_o}{K}\sin\left[\sqrt{\frac{K}{M_r}}\,t+\sin^{-1}\frac{F_o}{KD+F_o}\right]-\frac{F_o}{K}$$

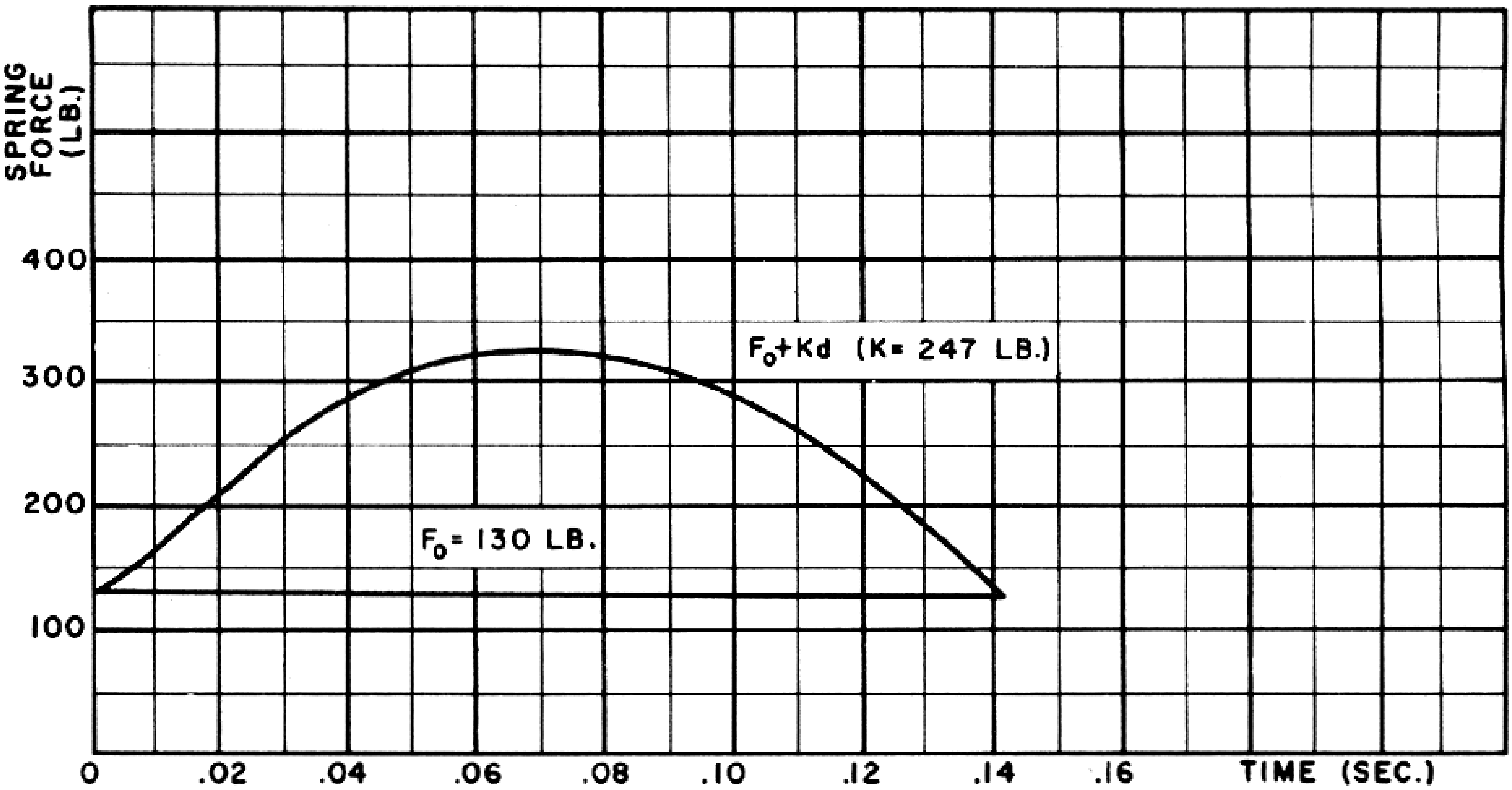


Figure 1–28. Variation of Spring Force With Time.

(1–11) $$T=\sqrt{\frac{M_r}{K}}\left(\cos^{-1}\frac{F_o}{KD+F_o}\right)$$

(1–12) $$N=\frac{30}{T}=\frac{30\sqrt{\frac{K}{M_r}}}{\cos^{-1}\frac{F_o}{KD+F_o}}$$

(It should be noted that these equations are all based on the simplifying assumption that the maximum velocity of free recoil is transferred instantaneously to the bolt. A more thorough analysis of the actual occurrences during the progress of the propellant explosion is given in the following description of the methods used for developing the theoretical time-travel and time-velocity curves.)

On the basis of equation 1–12, the rate of fire for the gun of the example is:

$$N=\frac{30\sqrt{\frac{247\times 32.2}{26}}}{\cos^{-1}\frac{130}{247\times .833+130}}=446\text{(rounds per minute)}$$

With the rate of fire and bolt energy known, the power absorbed by the bolt can be computed by means of the formula:

(1–13) $$HP=\frac{E_rN}{33,000}$$

The horsepower absorbed by the bolt in the gun of the example will be:

$$HP=\frac{194\times 446}{33,000}=2.62$$

This should be adequate power for operating the gun mechanism.

4. *Development of theoretical time-travel and time-velocity curves*

The theoretical curves showing bolt travel and bolt velocity with respect to time for a gun using advanced primer ignition are developed according to the same general principles employed for plain blowback but there are certain differences in the details as a result of the fact that ignition of the primer occurs while the bolt is still moving forward.

The development of the curves starts with the free bolt velocity curves (figs. 1–24 and 1–25) which are derived directly from the free bolt momentum curve previously plotted (fig. 1–23) by dividing each ordinate of the momentum curve by the mass of the bolt. (Bolt weight is 26 pounds.)

Fig. 1–26 shows the time-travel and time-velocity curves obtained for the gun of the example for the period from t=0 to t=0.010 second. These curves

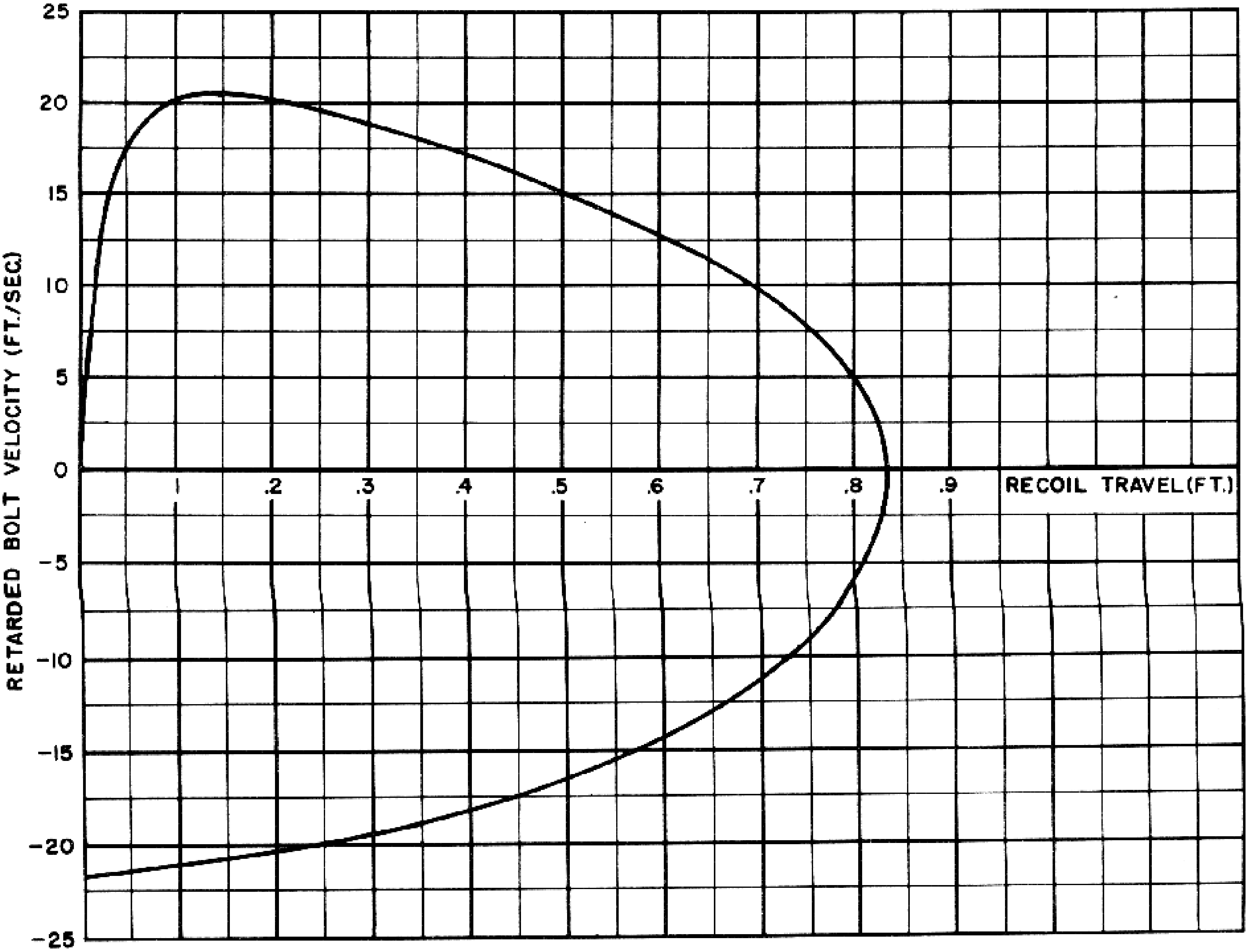


Figure 1–29. Velocity-Displacement Graph.

were derived by the step-by-step system described under plain blowback and the steps referred to in the figure are those in the procedure described for plain blowback. Note that for the first 0.0009 second the bolt travel curve dips downward to zero and then starts to rise as the bolt moves back, indicating that the bolt travel is measured from the most forward position reached during the propellant explosion. In this particular design the total velocity loss resulting from the effect of the initial spring compression during the time interval between $t=0.0009$ to $t=0.010$ is equal to 1.47 feet per second, computed as follows:

$$V=\frac{F_o t}{M_r}=\frac{130\times.0091\times32.2}{26}=1.47\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the spring constant K, as determined by the method of step 4 is only about 0.19 feet per second. This loss is so slight that it is not necessary to continue the process of successive approximation any further than step 5.

The remainder of the bolt displacement curve can now be determined analytically by using equation 1–10, modified as necessary to account for the displacement d' during the first 0.010 second. The changed values to be used in equation 1–10 are the following:

$$F_o'=F_o+Kd'=130+247\times.156=168.5$$

$$D'=D-d'=D-.156$$

$$t'=t-.010$$

Making the required substitutions as explained for plain blowback gives the final form of the equation to be used after the first 0.010 second as:

$$d=1.361\sin[17.49\,t+.351]-.527$$

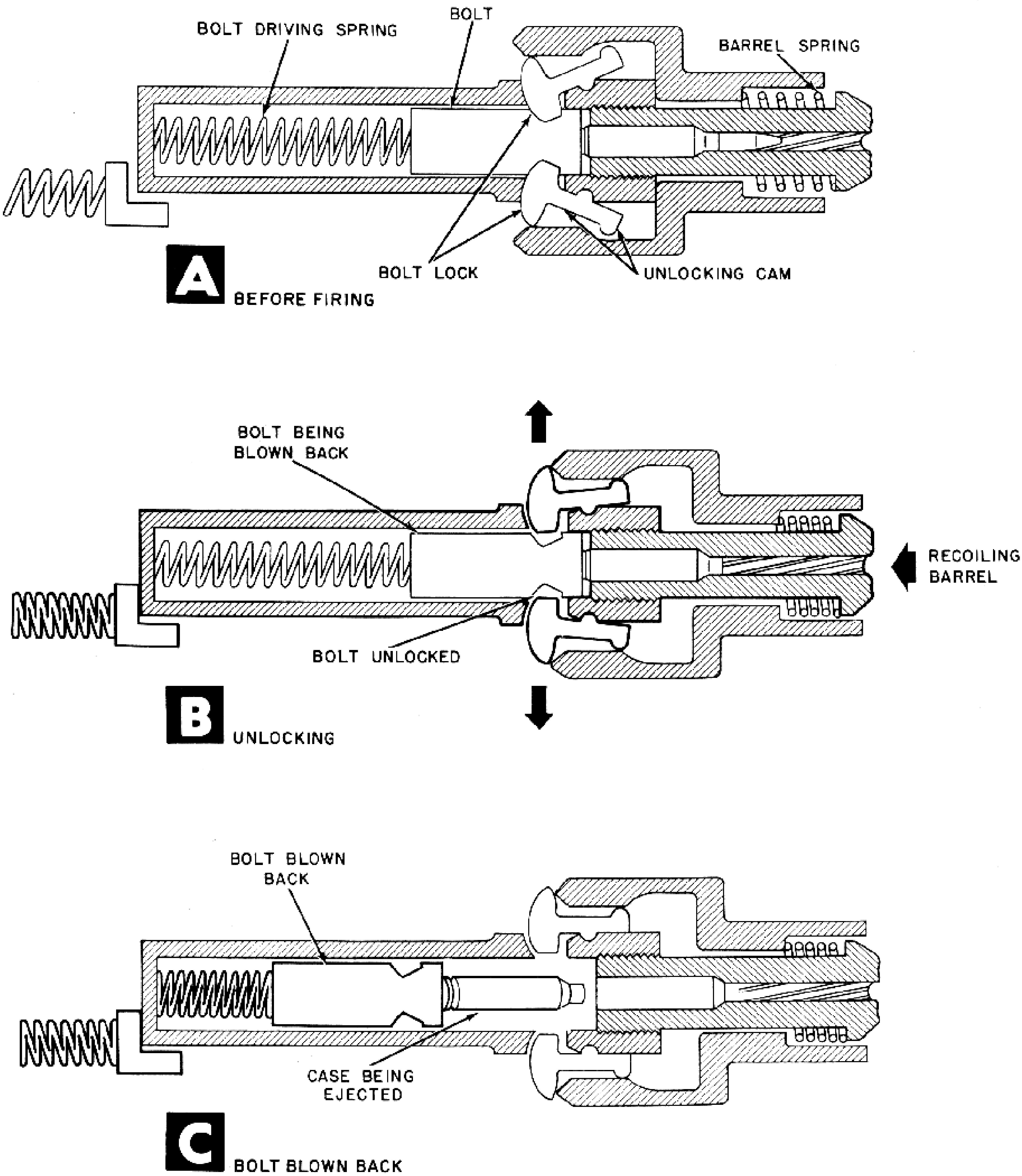


Figure 1–30. Simplified Schematic of Delayed Blowback Mechanism (Recoil-Unlocking).

This equation is used to complete the bolt displacement curve and gives the result shown in fig. 1–27. The ordinates of the displacement curve are then multiplied by K and increased by F_o to obtain a curve showing the variation of the spring force with time (fig. 1–28). Integrating under this curve and dividing the results by M_r in accordance with equation 1 15 will give a graph of the velocity loss due to the spring force. Subtracting this curve from the free bolt velocity curve will give the complete graph for the retarded bolt velocity, which is also shown in fig. 1–27. Fig. 1–29 shows the velocity versus displacement curve for the gun of the example as derived from the curves of fig. 1–27.

DELAYED BLOWBACK SYSTEM

All of the major difficulties associated with the design of blowback machine guns stem from excessive movement of the cartridge case during the time of action of the powder gas pressure. The preceding paragraphs show how the use of advanced primer ignition causes the initial high-pressure phase of the propellant explosion to be expended in stopping the forward motion of the bolt, with the resulting advantages of a lighter bolt and a slightly higher rate of fire than would be permissible with plain blowback. These advantages are attributable to the fact that the rearward motion of the bolt is delayed for approximately one-thousandth of a second, thus halving the impulse which is effective in blowing back the bolt and also reducing the time during which the bolt is accelerated to the rear. Since this time is reduced, the average rearward velocity of the bolt can be higher during the interval without exceeding the allowable movement.

The beneficial effects of the delayed rearward bolt movement obtainable with advanced primer ignition lead to the conclusion that still greater benefits could be achieved by further delaying the movement. Unfortunately, the delay which can be obtained with advanced primer ignition is more or less fixed and amounts to the time required for the propellant explosion to produce approximately one-half of its total impulse. (In other words, for a 20-mm gun such as used in the examples previously discussed, the delay is approximately 0.0009 to 0.0010 second.) To delay the rearward motion of the bolt beyond this time, it is necessary to resort to a special operating system referred to as "delayed blowback".

Delayed blowback may be defined as the system of operation in which the bolt remains locked until the peak powder gas pressures have passed and a safe operating limit is reached after the projectile clears the muzzle. The bolt is then unlocked by some means so that it can be blown back by the residual pressure with sufficient energy to perform the remainder of the cycle of operation. In this system, the time at which the bolt is unlocked can be controlled in the design so that any desired portion of the residual pressure can be utilized. Of course, the bolt must be unlocked while there is still sufficient impulse available from the residual pressure to produce the required operating energy.

Any gun in which the bolt is unlocked while there is still some residual pressure is subject to some blowback and partakes of some of the characteristics of the delayed blowback system. Such guns include certain gas-operated and short-recoil-operated weapons in which the bolt is unlocked almost immediately after the projectile has left the muzzle. However, in guns of this type, the operating energy does not come primarily from blowback but is derived mainly from the action of the gas piston or from the motion of the recoiling parts; therefore these guns will not be described at this time. The present analysis is concerned only with guns in which delayed blowback is the main source of energy.

The methods which have been used to unlock the bolt in delayed blowback guns are very numerous. Some are relatively simple and some quite elaborate but all are intended to keep the breech rigidly locked for a portion of the time of action of the powder gases. In some guns, the barrel and locked bolt are permitted to recoil together for a short distance and this motion is then utilized to perform the operation of unlocking. In other guns, the barrel is tapped so that a portion of the expanding powder gases can be by-passed to operate a piston or lever which actuates a mechanism so arranged that the breech is unlocked shortly after the projectile leaves the muzzle. Another method, known as primer actuation, uses an arrangement whereby the chamber pressure causes the primer of the cartridge to move to the rear in its pocket. As the primer is set back, it impinges on a sliding member and drives this member to the rear. The mechanism is devised so that the sliding member will unlock the bolt after a sufficient delay to permit the projectile to clear the muzzle. Still another method, known as the "booster principle", uses a device at the muzzle to trap the muzzle blast. The muzzle device is moved by the expanding gases and this movement is transferred to the breech mechanism through a mechanical connection which unlocks the bolt. After unlocking occurs, the trapped gas pressure blows back the bolt.

The basic point to remember is that a delayed blowback gun is one in which the bolt remains locked until after the projectile leaves the muzzle and is then unlocked so that the residual pressure can blow the bolt back. Whether the delayed unlocking is accomplished through recoil actuation, gas actuation, setback of the primer, booster actuation, or any other method is not of critical importance in considering what effects the delay has on the blowback action. Accordingly, in the following description and analysis of delayed blowback, it should be understood that, although recoil actuation is used to illustrate the principles, these principles apply regardless of what method of actuation is employed.

Cycle of Operation

Fig. 1–30 shows, in schematic form, a typical recoil-actuated delayed blowback mechanism. The cycle of operation is as follows:

The cycle starts with a cartridge in the chamber and with the bolt locked (fig. 1–30A). When the gun is fired, the reaction to the forward momentum imparted to the projectile and powder gases causes the barrel and locked bolt to recoil together. This recoil movement continues for a short distance until the projectile has cleared the muzzle and the bolt unlocking device is actuated (fig. 1–30B). When the bolt is unlocked, the residual pressure drives the cartridge case to the rear against the resistance offered by the inertia of the bolt.

After unlocking occurs, the barrel is stopped by a buffer and returned to battery by the barrel spring. When the residual pressure has dropped to zero, the bolt continues to move to the rear of its own momentum (fig. 1–30C). As the bolt moves back, the spent cartridge case is extracted and ejected and the bolt driving spring is compressed. When the driving spring and buffer system has absorbed all of the kinetic energy in the bolt and brought the bolt to a stop, the energy stored in the spring drives the bolt forward. The bolt picks up a fresh cartridge from the feeder as it moves forward, loads this cartridge in the chamber, and then relocks to the barrel, thus completing the cycle of operation.

Analysis of Delayed Blowback

In a delayed blowback gun, the blowback action occurs under the relatively low residual pressure which exists after the projectile has left the muzzle and therefore case separation does not present a serious problem. However, it should be realized that the residual pressure can be considered to be relatively "low" only when it is compared with the peak chamber pressure. In a typical 20-mm gun with a barrel length of nearly 60 inches, the initial value of the residual pressure is approximately 5000 pounds per square inch and this pressure may be considered to decrease exponentially with time as shown in fig. 1–31, until it reaches zero approximately 0.008 second after ignition of the propellant charge. Pressures of the magnitudes which exist for the major portion of the time shown in the figure are great enough to cause rupture of the thin walls near the base of the case if the case is permitted to move too far out of the chamber while these pressures are acting. Therefore, it is necessary in a delayed blowback gun to limit the bolt movement so that the cartridge case does not move too quickly out of the chamber.

The exact limit for movement of a specific cartridge case during the action of the residual gas pressure in a given gun can be determined only by experimentation under actual firing conditions. However, a fair estimate for present purposes can be obtained by considering the pressures shown in fig. 1–31 in relation to the strength of a typical 20-mm cartridge case. If it is assumed that after the thin portions of the case walls become unsupported the maximum internal pressure that can be safely allowed is 750 pounds per square inch, the period during which rupture of the case can occur is shown by the shaded portion of fig. 1–31. If it is further assumed that the bolt is unlocked 0.001 second after the projectile leaves the muzzle, the critical time extends from $t=0.00334$ to $t=0.00500$ second or lasts approximately 0.00166 second. Now as previously explained, a cartridge of the conventional type shown in fig. 1–6A should not be permitted to move out of the chamber by more than approximately one-

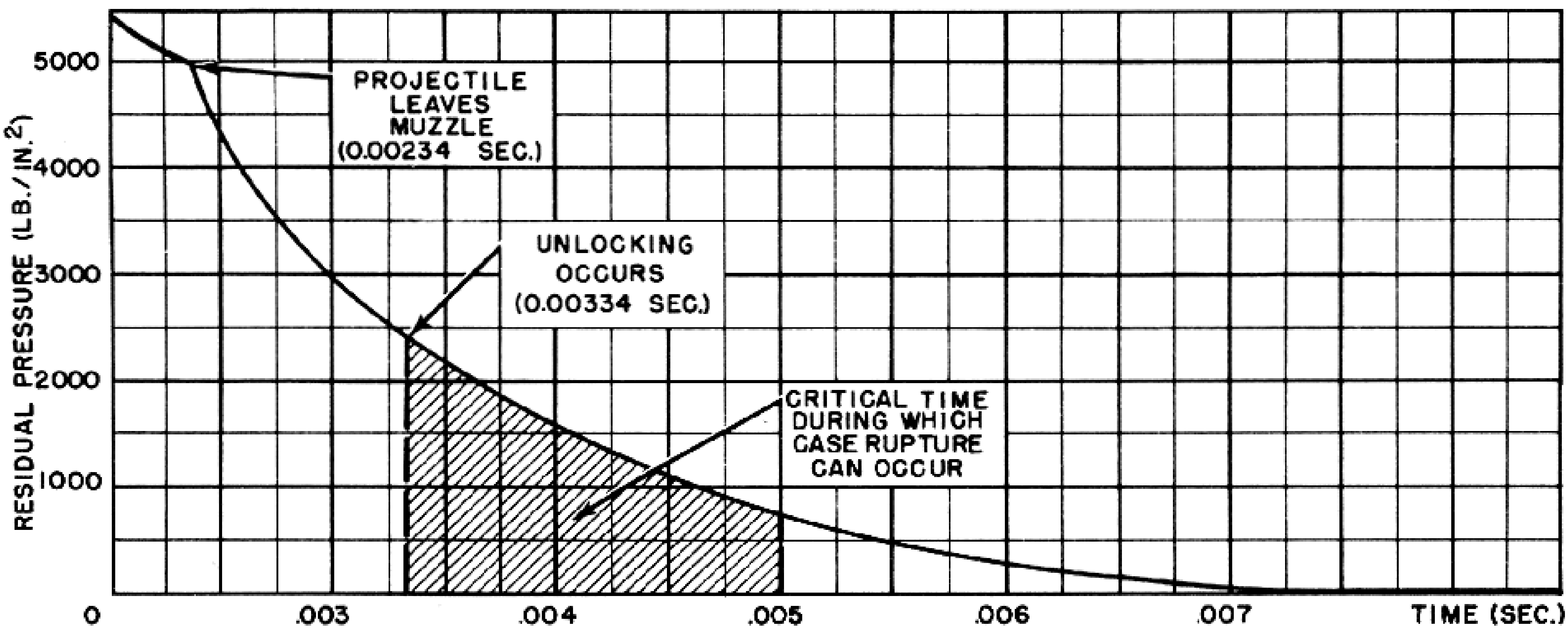


Figure 1–31. Residual Pressure Curve and Critical Period for Case Rupture (20 mm Gun of Example).

quarter inch during the time for which the residual pressure is high enough to cause rupture of the case near its base. Since a one-quarter inch movement is permissible and the time during which this movement can occur is about 0.00166 second, the allowable average bolt velocity during this interval is equal to

$$\frac{0.250}{12\times.00166}=12.5\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

As shown in fig. 1–31, the residual pressure continues to act for some time after 0.00500 second and therefore the final velocity reached by the bolt can be considera ly higher than the 12.5 (ft./sec.) average value allowable for the first 0.00166 second after the bolt is unlocked. The foregoing considerations show that the use of delayed blowback permits the attainment of relatively high bolt velocities with conventional ammunition so that it is not necessary to use the special ammunition described for advanced primer ignition.

When delayed blowback is used, only a very small portion of the total impulse of the propellant explosion is available for operating the gun mechanism. The relative magnitude of the available portion of the total impulse can be visualized by examining fig. 1–32, which shows the variation of the chamber pressure with time. Assuming that the bolt is unlocked 0.001 second after the projectile leaves the muzzle, the shaded area in this figure (when multiplied by the bore cross-section area) would represent the impulse which produces the relative velocity between the bolt and the barrel. The unshaded area under the curve (also multiplied by the bore cross-section area) would represent the impulse applied to the barrel and bolt while these parts are locked together.

The points made in the preceding paragraph indicate that the bolt must be relatively light in order for the small available impulse to produce a high bolt velocity and sufficient bolt energy. Also, since such a great impulse is applied before the barrel and bolt are unlocked, some means must be provided to account for the recoil reactions which occur before unlocking. Of course, in a delayed blowback gun employing recoil unlocking, the impulse before unlocking is utilized to produce the recoil motion of the barrel and bolt. In guns employing gas unlocking, primer actuation, booster actuation, or some other form of unlocking, it is usually also necessary to mount the barrel so that it can recoil to the rear and thus provide a means for controlling the magnitude of the trunnion reaction.

It should be pointed out here that the barrel and bolt, while locked together, can acquire a considerable recoil velocity with respect to the breech casing of the gun. After unlocking occurs, the blowback action drives the bolt to the rear and imparts an additional velocity to the bolt. Therefore, the actual velocity of the bolt measured with respect to the breech casing is the sum of the velocity of the barrel at the instant of unlocking and the velocity imparted by blowback. For this reason the bolt

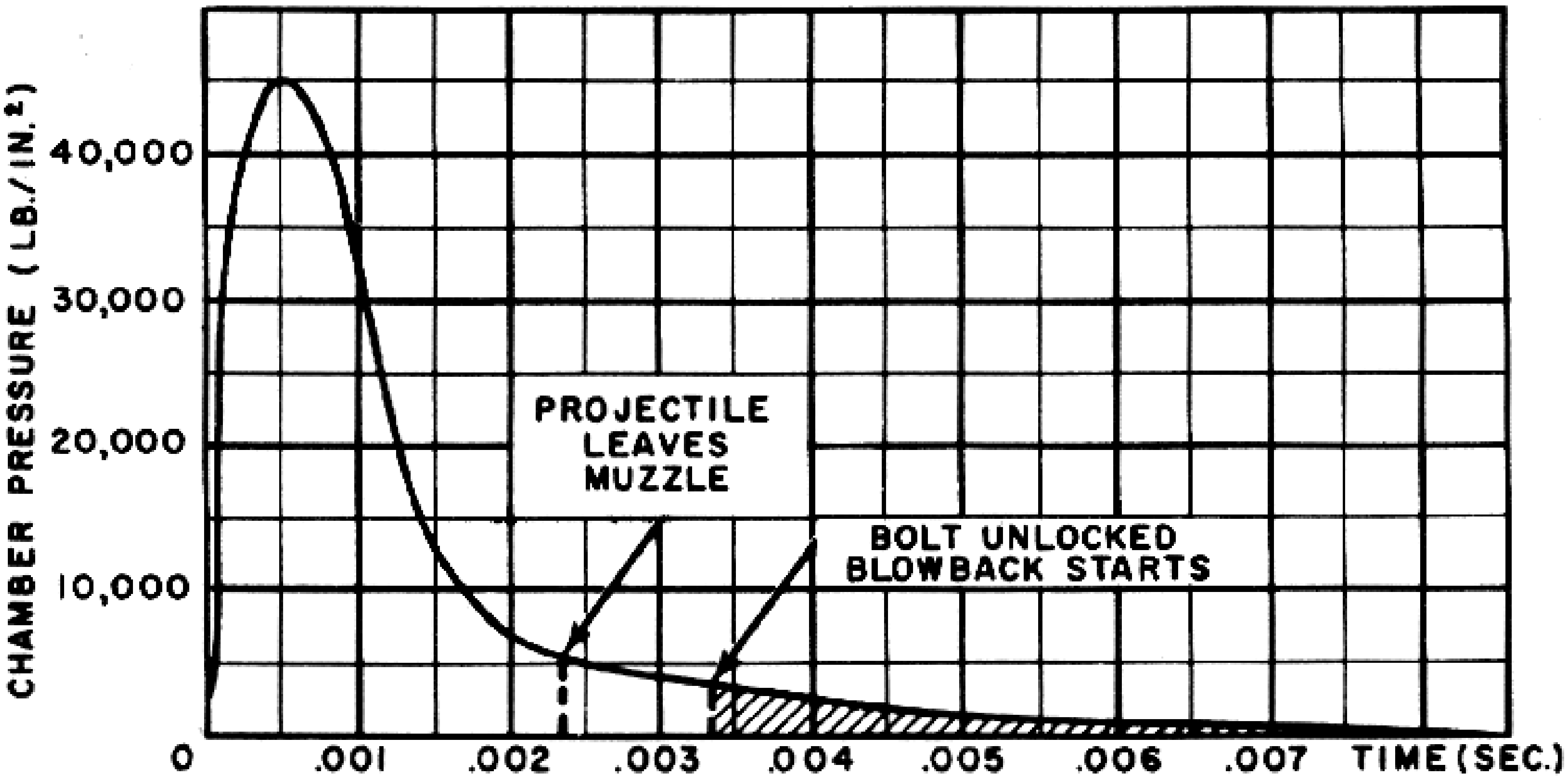


Figure 1–32. Variation of Chamber Pressure With Time Showing Proportion Available for Blowback (Breech Unlocked 0.001 Second After Projectile Leaves Muzzle).

velocities obtained with delayed blowback are fairly great and accordingly guns employing this principle are capable of relatively high rates of fire.

Mathematical Analysis of Delayed Blowback

The following mathematical analysis of delayed blowback is based on the same general principles used for the analysis of plain blowback except that the problem must be considered in two stages. These stages cover the conditions which exist before unlocking and the conditions which exist during and after blowback occurs. In this analysis it will be assumed that the same ballistic data used for plain blowback are available. (See figs. 1–9, 1–10, and 1–11.) Since many of the methods and formulas employed in this analysis are the same as or very similar to those used for plain blowback, the derivations of the formulas and the explanations of the procedures will not be repeated here. However, as they arise, any new concepts or new formulas will be explained.

In order to perform an analysis of delayed blowback, it is necessary to assume certain definite characteristics of the gun to be used as an example. First, it will be assumed that recoil unlocking will be employed. Since both the barrel and bolt move in recoil before unlocking, the weight of these parts must be known in order that their motion characteristics may be determined. In an actual design problem, it would first be necessary to design the barrel and plan the mechanism sufficiently at least to permit making a preliminary estimate of the weight of the recoiling parts. For purposes of the present analysis it will be assumed that the barrel and its related parts weigh 40 pounds and the effective weight of the bolt will be taken as 10 pounds, giving a total weight of 50 pounds for the recoiling parts. It will also be assumed that unlocking occurs 0.001 second after the projectile leaves the muzzle of the gun. After the analysis is completed in an actual problem, it may be desirable to modify the assumed values to adjust the performance of the gun. However, the same method of analysis will apply and the assumed values will serve here to illustrate the method.

1. *Conditions of free recoil before unlocking*

Before unlocking occurs, the combined mass of the barrel and bolt moves in recoil. For the condition of free recoil the momentum relation for the time before the projectile leaves the muzzle is the same as expressed by equation 1–1:

$$M_r v_{r_f} = \left(M_p + \frac{M_c}{2}\right) v_p$$

Solving for v_{r_f}:

$$v_{r_f} = \frac{.29 + \frac{.070}{2}}{50}\, v_p = .00650\, v_p$$

Using the projectile velocities shown in fig. 1–10, this relation can be used to plot the curve of free

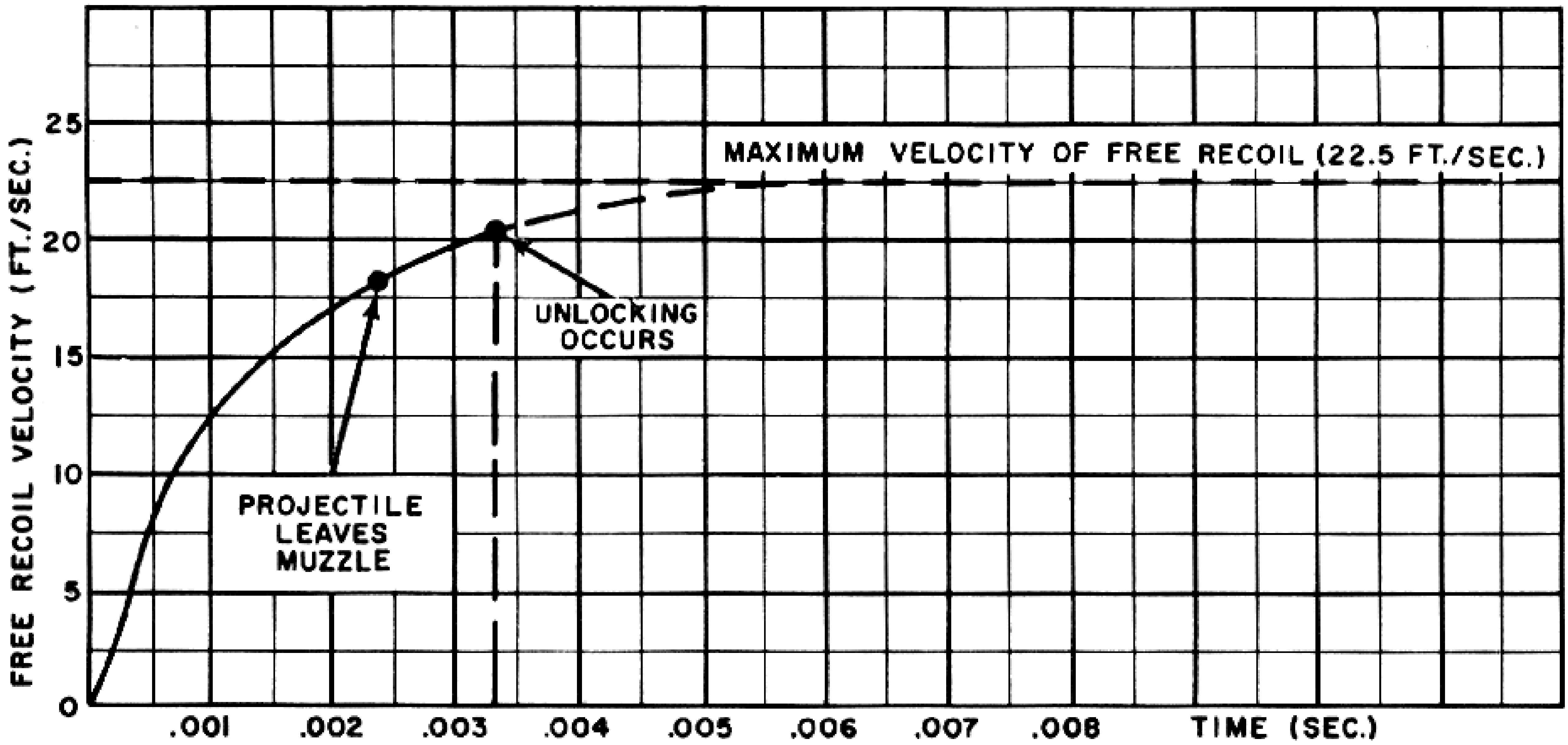


Figure 1–33. Free Recoil Velocity Before Unlocking.

recoil velocity versus time for the period before the projectile leaves the muzzle. The curve so obtained is shown in fig. 1–33 as the portion between t=0 and t=0.00234 second. In order to plot the remainder of the curve, it is necessary to determine the maximum velocity of free recoil and extrapolate the curve by the methods previously described. Actually the extrapolated curve will apply only until unlocking occurs, 0.001 second after the projectile leaves the muzzle. The dotted portion of the curve in fig. 1–33 after t=0.00334 second is used only for purposes of extrapolation and does not relate to the conditions of recoil.

2. *Effect of springs before unlocking*

Immediately after unlocking, the rearward motion of the barrel is stopped by a buffer and the bolt is blown back by the residual pressure. Since the barrel is stopped by the buffer, the only real functions of the barrel spring are to assist the barrel to return to battery and to hold the barrel firmly in the battery position. Therefore the value of the force exerted by the barrel spring is not critical and the characteristics of the spring may be selected arbitrarily. To return the barrel quickly to battery, a fairly strong spring is desirable. As will appear, such a spring will not have any great effect on the recoil motion while the powder gas pressures are acting and therefore, the resistance of the spring can be made quite high. On this basis, the initial compression of the spring will be selected as 400 pounds and the spring constant as 400 pounds per inch.

Since the bolt is light (10 pounds) and receives a relatively low impulse from blowback, the driving spring for the bolt will necessarily be relatively light in order to permit the bolt to open the 10 inches assumed necessary for feeding a 20-mm round. The actual characteristics of the bolt spring can not be determined at this time but it can be safely assumed that the bolt spring will have an initial compression of approximately 120 pounds and a spring constant of about 20 pounds per inch.

3. *Theoretical time-travel and time-velocity curves before unlocking*

The theoretical time-travel and time-velocity curves for the time before unlocking are developed according to the same general principles used for plain blowback. Fig. 1–34 shows the curves obtained for the gun of the example for the period of time from t=0 to t=0.00334 second. These curves were derived by the step-by-step system described under plain blowback and the steps referred to in the figure are those in the procedure described for plain blowback. In drawing these curves, the combined resistances of the barrel and bolt springs were taken into account. In this case, the total

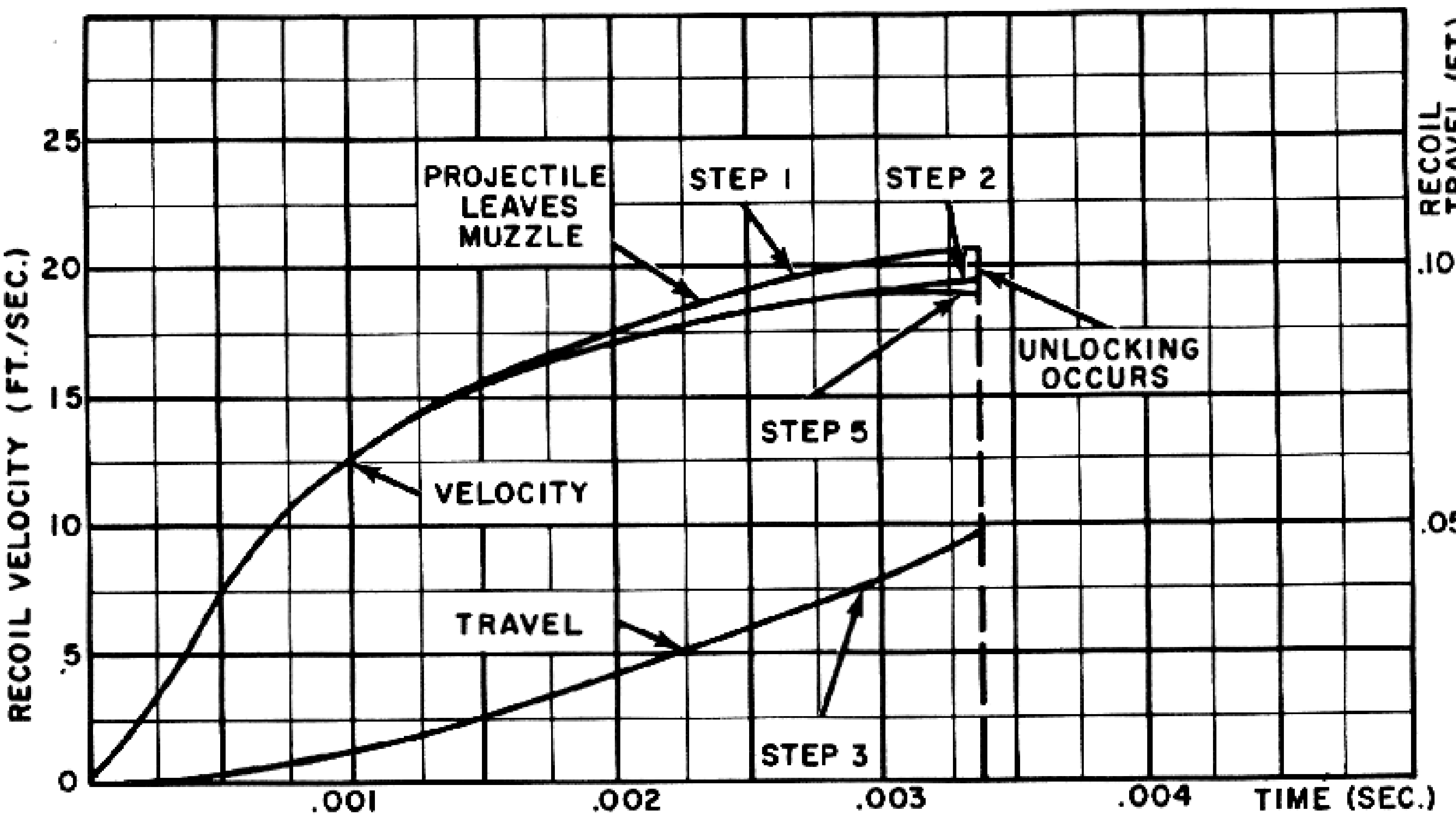


Figure 1–34. Time-Travel and Time-Velocity Curves Before Unlocking.

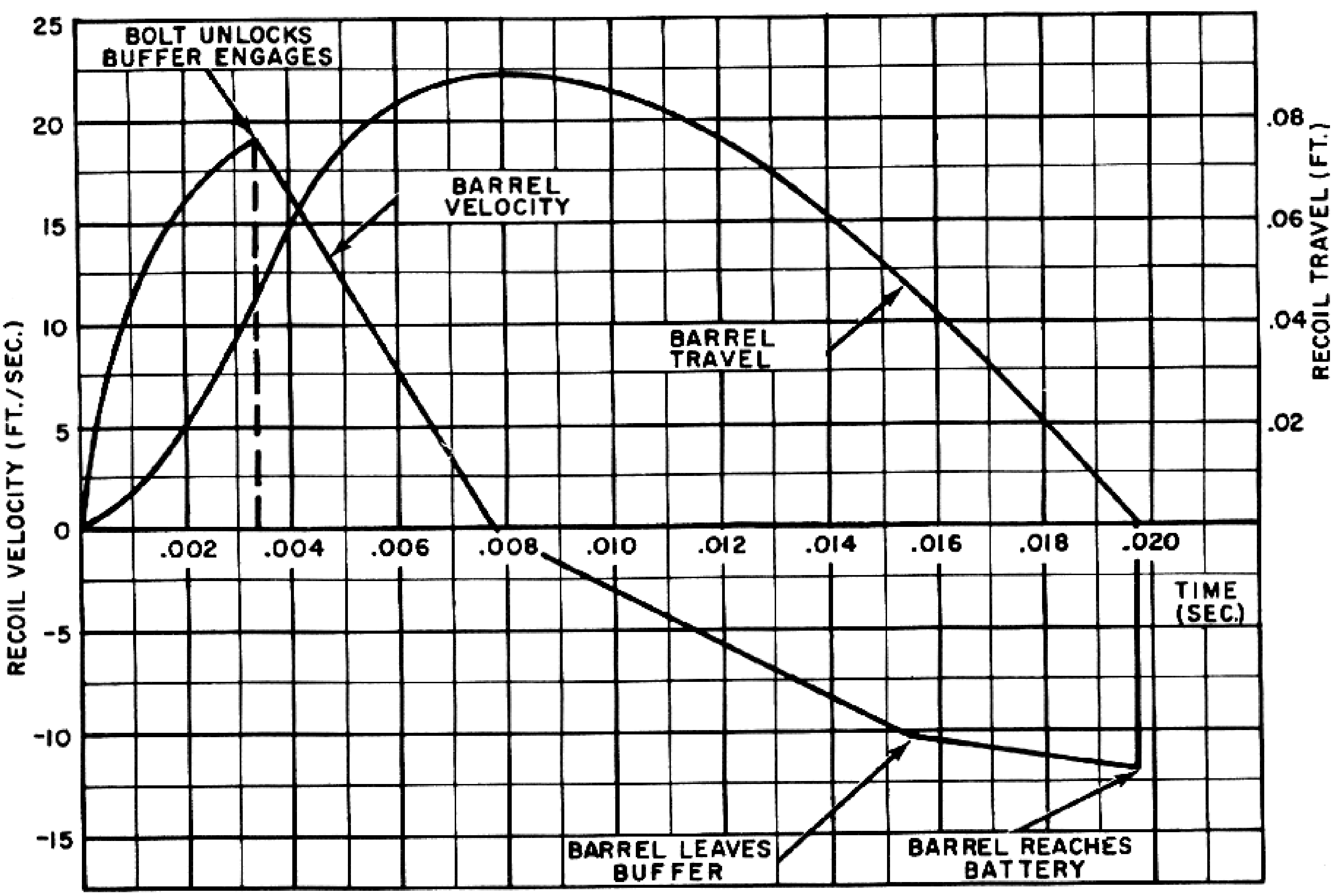


Figure 1–35. Time-Travel and Time-Velocity Curves for Barrel.

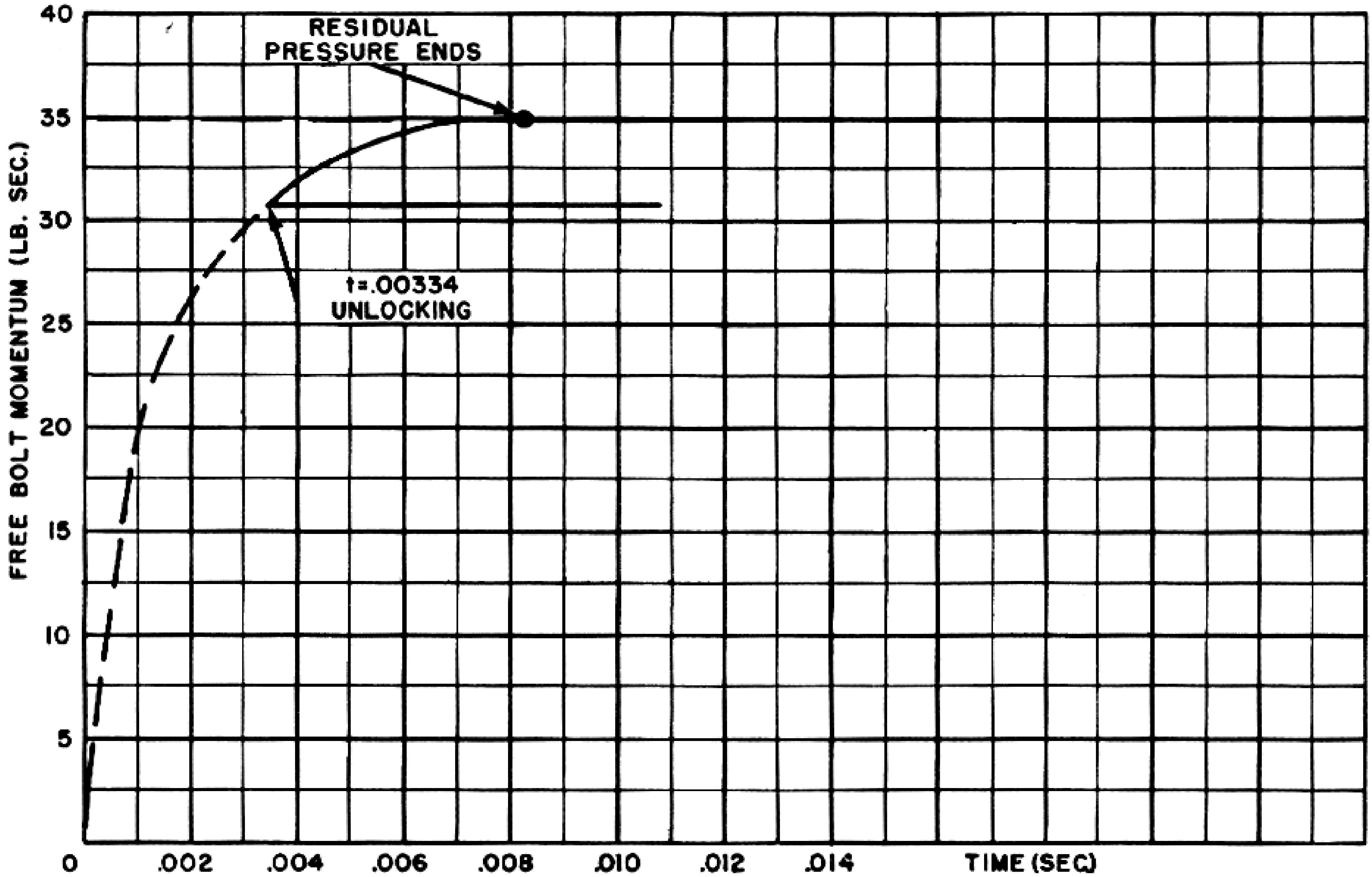


Figure 1–36. Free Bolt Momentum Imparted After Unlocking.

velocity loss resulting from the effect of the combined initial compression of the barrel and bolt springs for the first 0.00334 second is equal to 1.118 feet per second, computed as follows:

$$V=\frac{F_o t}{M_r}=\frac{520\times.00334\times32.2}{50}=1.118\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the combined spring constant K, as determined by the method of step 4 is only about 0.26 foot per second. This loss is so slight that it is not necessary to continue the process of successive approximation any further than step 5. Examination of the curves of fig. 1–34 reveals that the total recoil movement before unlocking is 0.0470 foot (0.565 inch) and the recoil velocity at the instant of unlocking is 19.2 feet per second.

4. *Barrel motion after unlocking*

Immediately after unlocking, the barrel strikes the buffer and its rearward motion is halted. Since the barrel weighs 40 pounds and is moving at a velocity of 19.2 feet per second, its kinetic energy is equal to

$$E_r=\frac{1}{2}MV^2=\frac{1}{2}\frac{40}{32.2}(19.2)^2=229\ (\text{ft. lb.})$$

In order to bring the barrel to a stop, this kinetic energy must be absorbed by a buffer. If it is assumed that the motion of the barrel is stopped within one-half inch after engaging the buffer, the average force exerted on the buffer must be:

$$229\times\frac{12}{.50}=5500\ (\text{lb.})$$

This serves to indicate that the barrel buffer must be constructed very ruggedly. Buffers for applications of this sort are usually constructed intentionally so that there will be a considerable energy loss during the buffing action. This is done in order to damp out undesired motion of the barrel. In a well-designed buffer, the energy loss can be as high as 70 percent; that is, the kinetic energy of the barrel after its motion is reversed will be only 30 per cent of the energy it contained upon first striking the buffer.

The motion of the barrel during the time of action of the buffer will depend largely on the particular characteristics of the buffer used. It should be noted that while the buffer is acting, the effect of the barrel spring is so small by comparison that

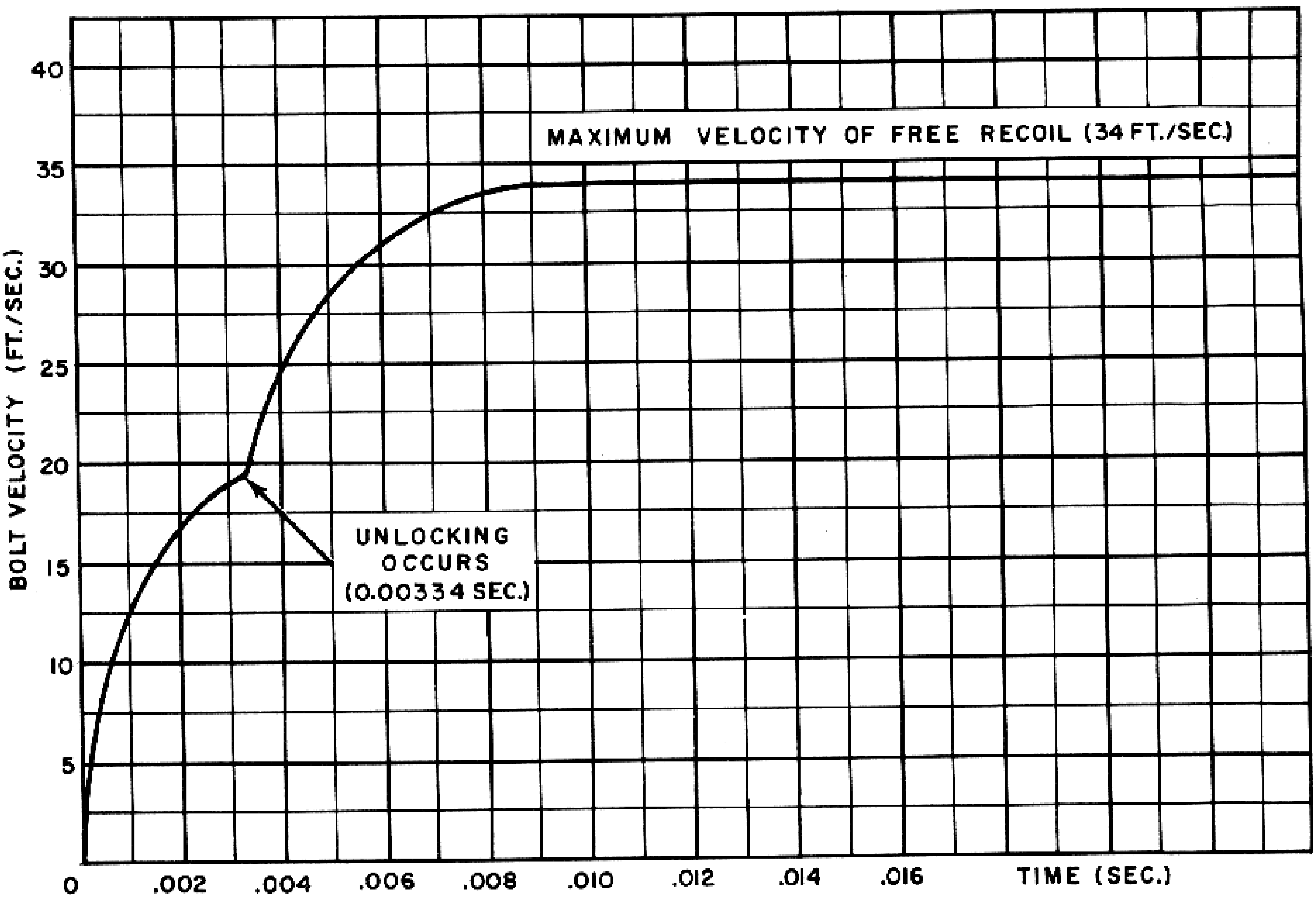


Figure 1–37. Free Bolt Velocity Curve for First 0.010 Second.

it is negligible. For purposes of illustrations, it will be assumed here that the resistance offered by the buffer to compression is constant at 5500 pounds for the full compression of one-half inch and it will also be assumed that the expansion force of the buffer is 30 per cent of this value, or 1,800 pounds. On this basis and using equation 15, the barrel motion is as shown in fig. 1–35. The velocity loss due to the buffer is given as:

$$V=\frac{Ft}{M}=\frac{5500\times 32.2}{40}\,t=4420\ t$$

In this expression t is measured from the time the buffer engages. The forward velocity imparted to the barrel by the buffer is given as:

$$V=\frac{Ft}{M}=\frac{1650\times 32.2}{40}\,t=1330\ t$$

where t is measured from the time the forward motion of the barrel starts. This relation holds until the barrel has moved forward one-half inch. Since the barrel is subjected to a uniform acceleration during this interval, the time required for the one-half inch movement can be determined by solving for t in the expression

$$\frac{.5}{12}=\frac{1}{2}\,1330\ t^2$$

$$t=.00792\ (\text{sec.})$$

After the barrel leaves the buffer it is further accelerated forward by the barrel spring. The movement remaining to the battery position is 0.0470 foot (fig. 1–34) and it may be assumed with little error that the average force exerted by the barrel spring over this distance will be the initial compression (400 pounds) plus approximately one-half the force resulting from the spring constant for the 0.0470-foot deflection

$$\left(\frac{226}{2}=113\ \text{pounds}\right)$$

The gain in velocity due to this force will be:

$$V=\frac{Ft}{M}=\frac{513\times 32.2}{42}\,t=393\ t$$

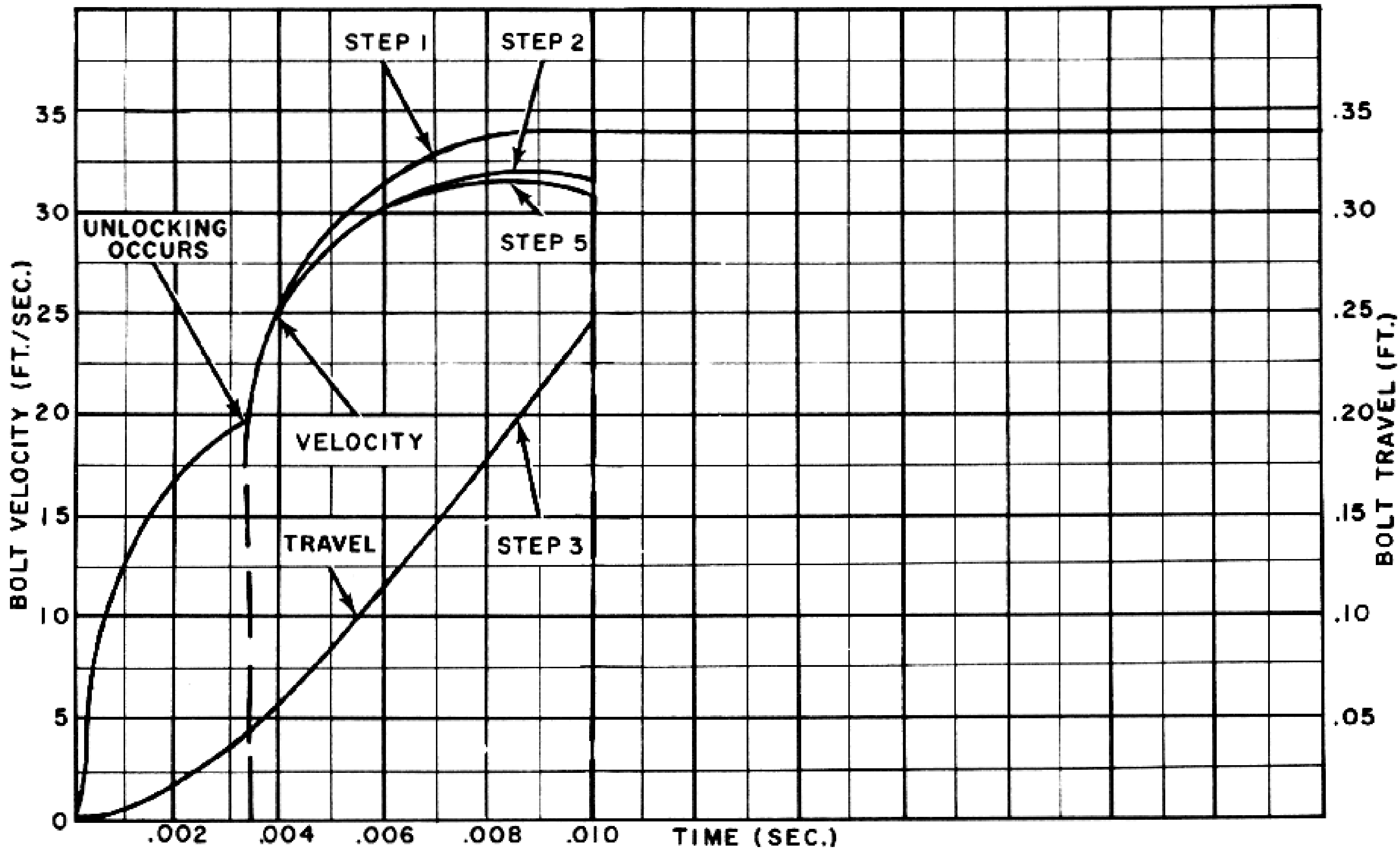


Figure 1–38. Time-Travel and Time-Velocity Curves for Bolt for First 0.010 Second.

Since the velocity of the barrel upon leaving the buffer was equal to 10.52 feet per second, the velocity for the last 0.0470 foot of this return to battery is given by the expression:

$$v = 10.52 + 393\ t$$

The time required to move this distance can be determined by solving the t in the following equation:

$$.0470 = \left[10.52 + \frac{393\ t}{2}\right] t$$

$$t = .00490\ \text{sec.}$$

The curves resulting from the above analysis are shown in fig. 1–35. As shown in the figure, the barrel reaches the battery position with a velocity of 12.0 feet per second at 0.0198 second after the ignition of the propellant. The barrel is brought to rest by a buffer and remains in this position until the bolt returns and a new cycle is started.

5. *Bolt motion after unlocking*

At the instant of unlocking, the barrel and bolt are both moving at the same recoil velocity. After unlocking, the bolt retains this velocity and acquires additional velocity from blowback. The methods used for analyzing the bolt movement after unlocking are generally the same as those used for plain blowback and employ the momentum relation expressed by the curve shown in fig. 1–13. In this case, the applicable portion of the curve is the part after 0.00334 second, at which time unlocking occurs. This portion of the curve is reproduced in fig. 1–36. The values of bolt momentum read above the horizontal line which crosses the curve at 0.00334 second represent the after unlocking. Dividing these values by the mass of the 10-pound bolt for the gun of the example gives the additional velocity imparted by blowback. Fig. 1–37 shows the complete curve of velocity for the bolt. (Up to 0.00334 second, the curve shows the retarded velocity curve from fig. 1–34 as obtained for both the barrel and bolt. After 0.00334 second, the curve shows the additional free velocity imparted by blowback.) The theoretical curves of bolt travel and bolt velocity versus time are then obtained by the same methods used for plain blowback.

Before the time-travel and time-velocity curves can be drawn, it is necessary to determine the char-

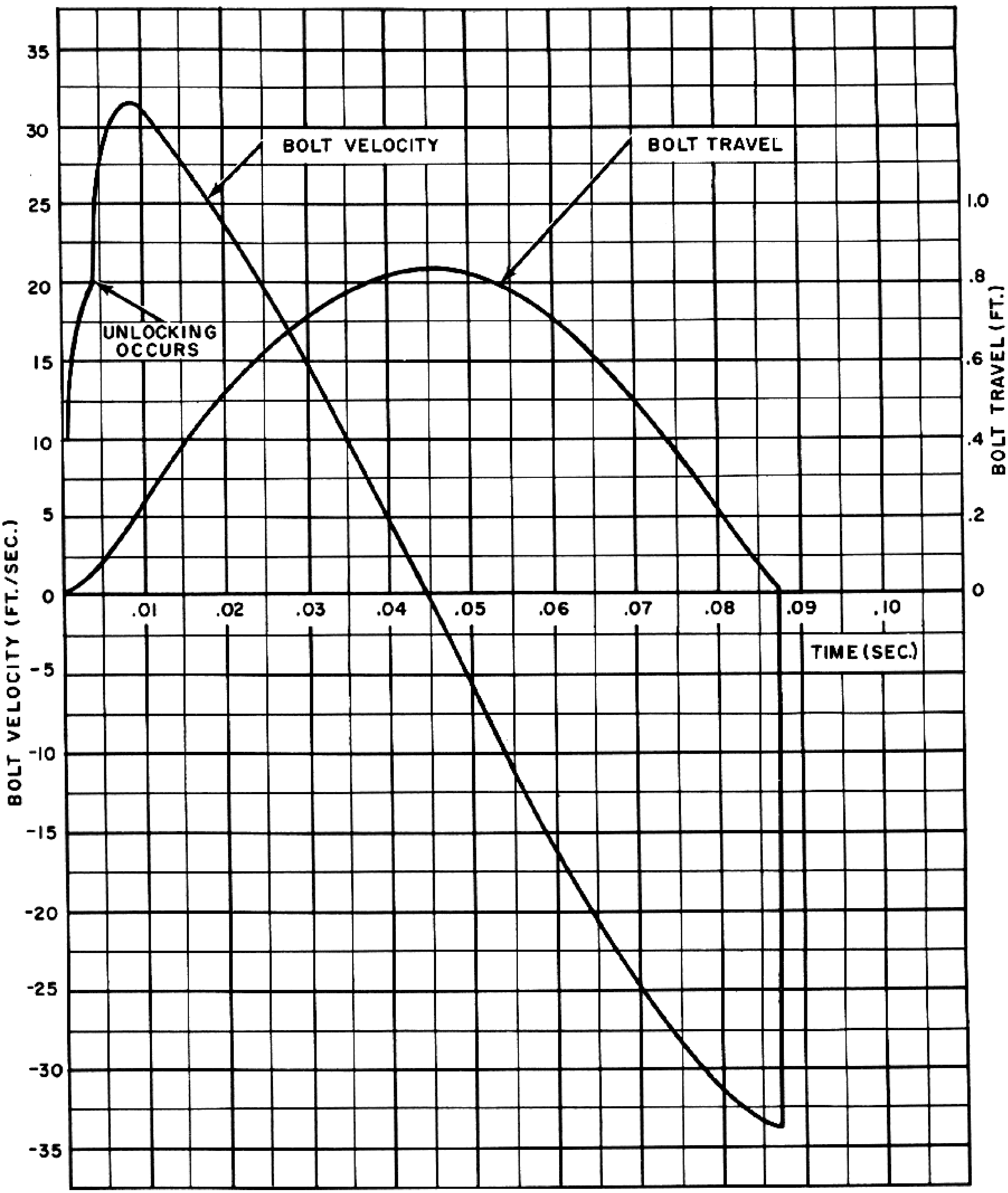


Figure 1–39. Time-Travel and Time-Velocity Curves for Bolt (Complete Cycle).

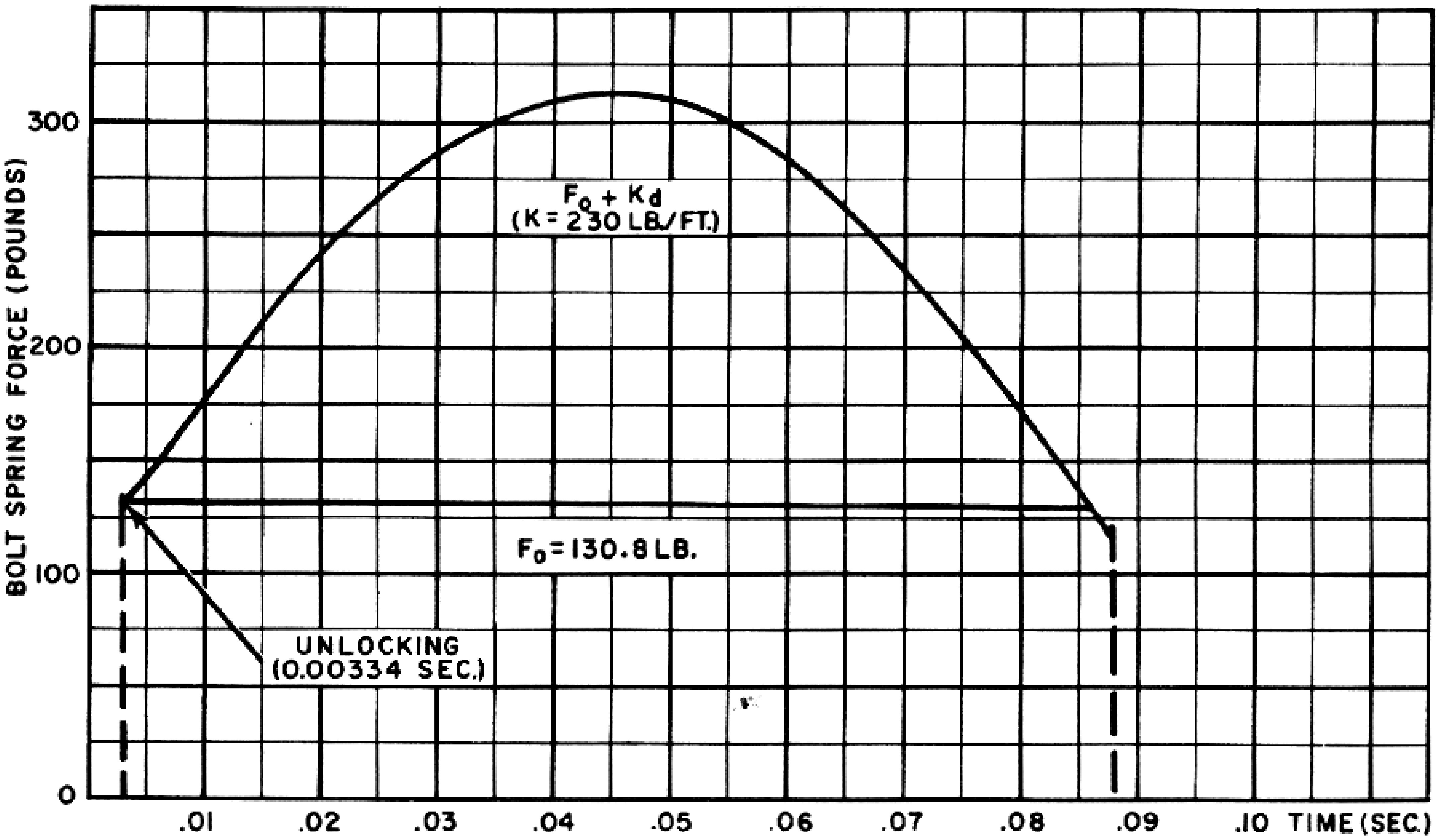


Figure 1–40. Variation of Spring Force With Time (Bolt Spring Only After Unlocking).

acteristics of the bolt driving spring so that the bolt will open the required distance of 10 inches to permit feeding. Fig. 1–37 shows that the maximum free recoil velocity of the bolt is 34 feet per second. Since this velocity is imparted within 0.010 second, it may be assumed that the initial bolt energy is imparted instantaneously. The value of this energy is given by equation 1–6:

$$E_r = \frac{1}{2} M_r V_{r_t}{}^2 = \frac{W_r V_{r_t}{}^2}{2g} \text{ (ft. lb.)}$$

Evaluating this expression for the conditions of the example gives:

$$E_r = \frac{10 \times 34^2}{2 \times 32.2} = 180 \text{ (ft. lb.)}$$

Since the bolt driving spring must absorb this amount of energy over a distance of 10 inches (0.833 feet), the average force exerted by the spring must be:

$$F_{av} = \frac{180}{.833} = 216 \text{ lb.}$$

In designing the spring so that it will produce this average force, the same factors described in the analysis of plain blowback should be considered. However, an arbitrary choice of spring characteristics will suffice for present purposes. If the initial compression is taken as 120 pounds, a maximum force of 312 pounds will produce the required average force of 216 pounds. Since the difference between the maximum force and initial compression is 192 pounds and the recoil distance is 10 inches, the spring constant will be 19.2 pounds per inch or 230 pounds per foot. Realizing that this choice is arbitrary, it will be assumed here that the selected values of F_o=120 pounds and K=230 pounds per foot will result in a satisfactory spring.

Having the characteristics of the bolt driving spring and the free recoil velocity curve of fig. 1–37, the time-travel and time-velocity curves for the bolt can be constructed using the same general approach used for plain blowback. At the instant unlocking occurs, the bolt driving spring has been compressed 0.0470 foot (fig. 1–34). This means that for considering the period of blowback, the initial compression of 120 pounds must be increased by the effect of the spring constant for this deflection. That is:

$$F_o = 120 + .047 \times 230 = 130.8 \text{ (pounds)}$$

Fig. 1–38 shows the time-travel and time-velocity curves obtained for the bolt of the gun used as ex-

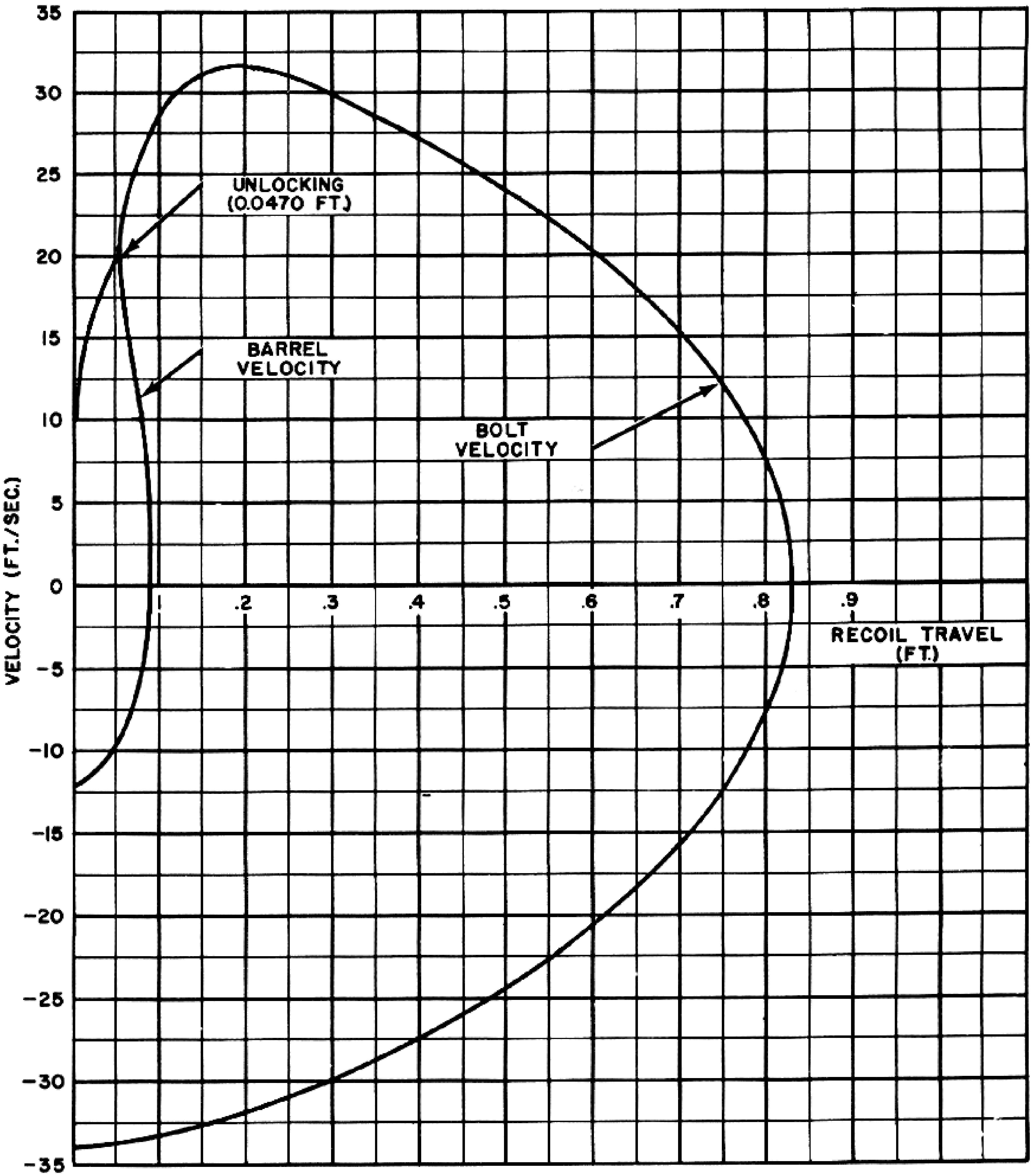


Figure 1–41. Velocity-Displacement Graph (Barrel and Bolt).

ample for the period from $t=0$ to $t=0.010$ second. These curves, for the time after $t=0.00334$ second were obtained by the step-by-step system described under plain blowback and the steps referred to in the figure are those in the procedure described for plain blowback. The velocity loss resulting from the initial compression of the bolt driving spring for the time interval between $t=0.00334$ and $t=0.010$ second is equal to 2.81 feet per second, computed as follows:

$$V=\frac{F_o t}{M_1}=\frac{130.8\times.00666\times32.2}{10}=2.81\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the spring constant K, as determined by the method of step 4 is only about 0.697 foot per second. This loss is so slight that it is not necessary to continue the process of successive approximation any further than step 5.

The remainder of the bolt displacement curve can be determined analytically by using equation 1–10, modified as necessary to account for the displacement d' during the first 0.010 second. The changed values to be used in equation 1–10 are the following:

$$F_o'=F_o+Kd'=120+230\times.245=176.3$$

$$D'=D-d'=D-.245$$

$$t'=t-.010$$

Making the required substitutions as explained for plain blowback gives the final form of the equation to be used after the first 0.010 second as:

$$d=1.353\sin[27.2t+331]-.521$$

This equation is used to complete the bolt displacement curve and gives the result shown in fig. 1–39. The ordinates of the displacement curve are then multiplied by K and increased by F_o to obtain a curve showing the variation of spring force with time (fig. 1–40). (Note that the initial compression used is 130.8 pounds which takes into account the compression resulting from the recoil movement before unlocking.) Integrating under this curve and dividing the results by M_r in accordance with equation 1–15 will give a graph of the velocity loss due to the spring force. (All computations start at the time of unlocking, 0.00334 second.) Subtracting this curve from the free bolt velocity curve of fig. 1–37 will give the complete graph for the retarded bolt velocity, which is also shown in fig. 1–39. Fig. 1–41 shows the velocity versus displacement curves for the gun of the example as derived from the curves of figs. 1–35 and 1–39.

From fig. 1–39, the time for the complete cycle is given as 0.0875 second. This gives the rate of fire for the gun of the example as:

$$N=\frac{60}{.0875}=686 \text{ (rounds per minute)}$$

RETARDED BLOWBACK SYSTEM

One of the principal disadvantages of the plain blowback system is the excessive bolt weight required to limit the movement of the cartridge case to a safe value during the action of the powder gas pressure. This particular difficulty can be circumvented by employing the "retarded" blowback system—a system in which a special type of retarding mechanism is operated by the movement of the bolt. The mechanism itself is composed of relatively light parts and the inertia forces which result when these parts are set in motion by the bolt are therefore relatively small. However, the mechanism is arranged so that the bolt must act through a tremendous mechanical disadvantage to overcome the inertia forces and is accordingly subjected to a very high resistance to motion. In other words, although the bolt and the associated mechanism may be quite light, the effective resistance to bolt acceleration can be made just as great as that which would be obtained by the use of a very heavy bolt.

The mechanism of a retarded blowback gun is similar to that of a plain blowback gun except for the presence of the retarding mechanism. This mechanism can take many forms but the basic principle underlying all of these forms is that the bolt must overcome the inertia forces in the mechanism by acting through a high mechanical disadvantage. One such mechanism is illustrated schematically in fig. 1–42. This mechanism is essentially a toggle joint and when the bolt is closed, the angle "a" indicated in the figure is nearly 180 degrees. Under these conditions, the linkage acts like a crank mechanism which is slightly off dead center and most of the blowback force acting on the linkage is not effective in opening the bolt but is transmitted directly through the links to the breech

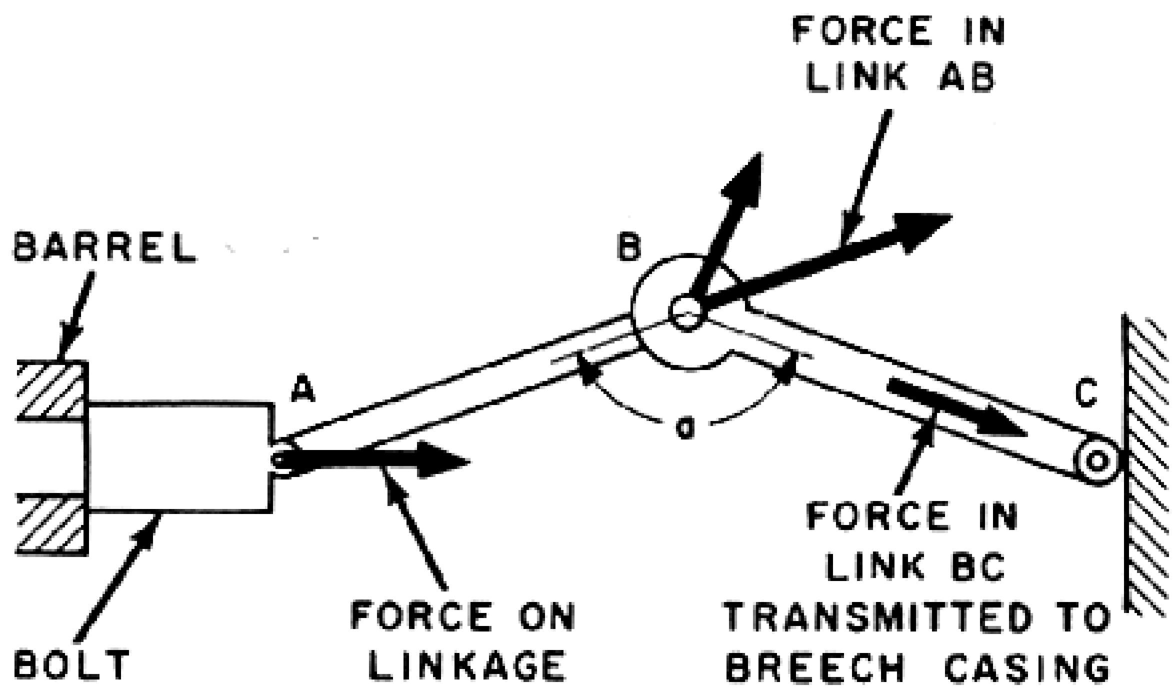


Figure 1–42. Straight Toggle Linkage.

casing. Only a very small part of the blowback force acts on the linkage mass at point B in the direction necessary to overcome its inertial resistance to rotation about point C. The result of this action is that the relatively small inertial resistance of the linkage mass is effectively multiplied so that it produces the same resistance to bolt acceleration as a very heavy mass located at point A. In this mechanism, the retardation offered by the linkage does not remain constant. As the bolt opens and angle "a" decreases, the bolt acts at a smaller disadvantage and the mass multiplying effect of the linkage decreases. The effect of this change is that the more the bolt opens, the less resistance it encounters.

The toggle linkage described in the preceding paragraph can be arranged in many different ways. For example, fig. 1–43 shows a linkage which operates on the same principle but for some applications may be more economical of space. A mechanism of this type, as used in the Schwarzlose machine gun, is shown in fig. 1–44. In some cases, the links may not pivot on hinge pins as indicated schematically in figs. 1–42 and 1–43 but may have cam-shaped ends which make rolling contact with each other as the links move with respect to each other.

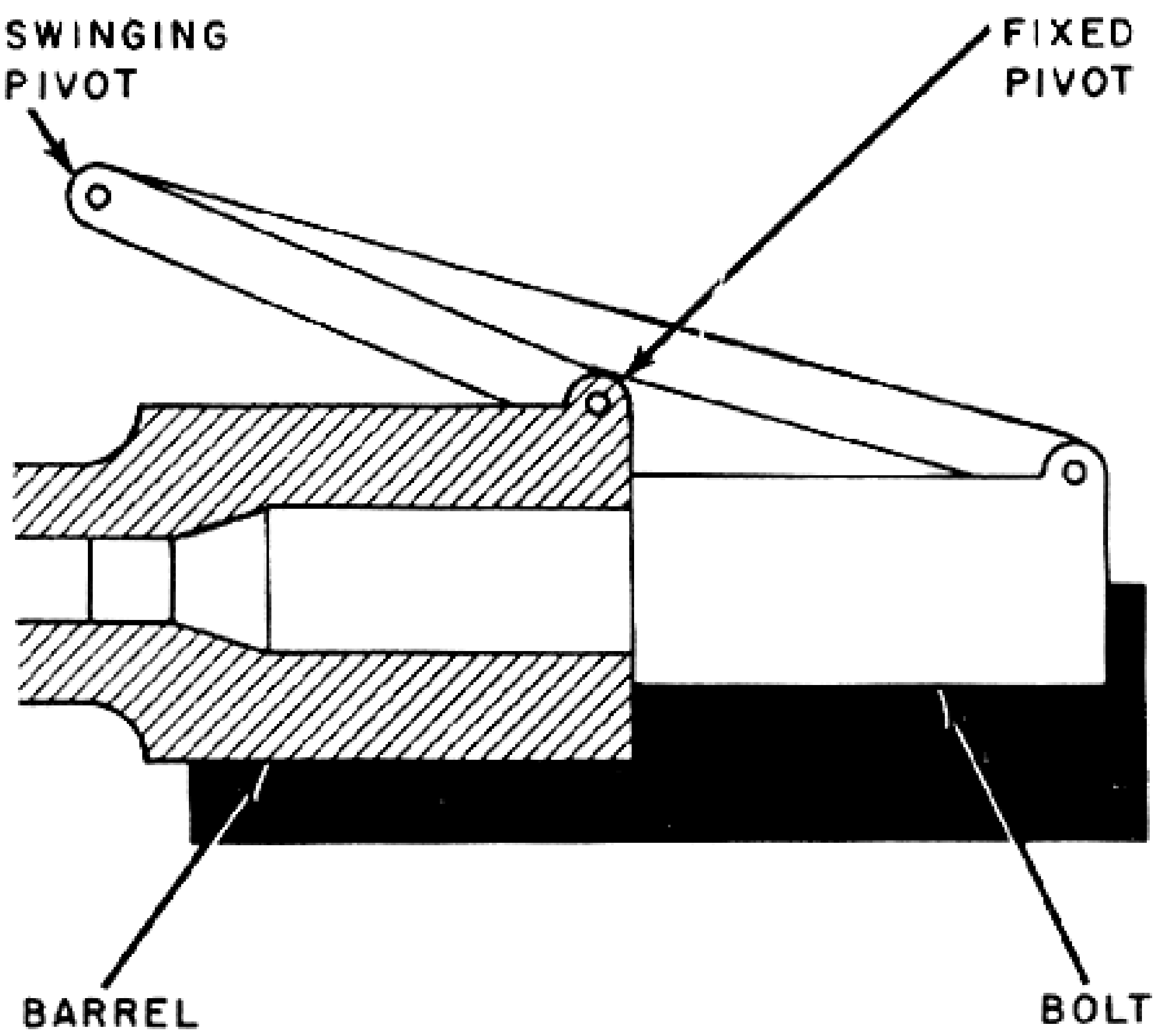


Figure 1–43. Compact Form of Toggle Linkage.

Linkages represent only one of many types of mechanisms that can be used to produce retardation by causing the bolt to operate against a high mechanical disadvantage. A wide variety of inclined surfaces, cams, spirals, wedges, screw threads, and other devices can be employed in such a manner that only a small component of the blowback force is effective in causing sliding or rotating motion of the mechanism. Many such devices are illustrated in part XI of this publication and therefore will not be described here.

Cycle of Operation

The cycle of operation for a retarded blowback gun is essentially the same as for a plain blowback gun. When the cartridge is fired, the pressure of the powder gases drives the cartridge case to the rear against the resistance offered by the bolt and the associated retarding device. After the powder gas pressure has fallen to zero (in less than 0.010 second for a typical 20-mm gun), the bolt continues to move to the rear by virtue of the momentum imparted to the recoiling parts by the explosion. During the rearward motion of the bolt, the spent cartridge case is extracted and ejected and the bolt driving spring is compressed. When the driving spring and buffer have absorbed all of the kinetic energy of the recoiling parts, the bolt is pushed forward to pick up a fresh cartridge from the feed mechanism and load this cartridge into the chamber. After the cartridge is fully chambered, it is fired and a new cycle begins.

Analysis of Retarded Blowback

From the standpoint of performance with high-powered ammunition, the most significant characteristic of a retarded blowback gun is that the rearward motion of the cartridge case starts immediately after ignition of the propellant charge, which is exactly what happens in a plain blowback gun. Thus, if the retardation is constant so that the effect of the retarding mechanism is exactly the

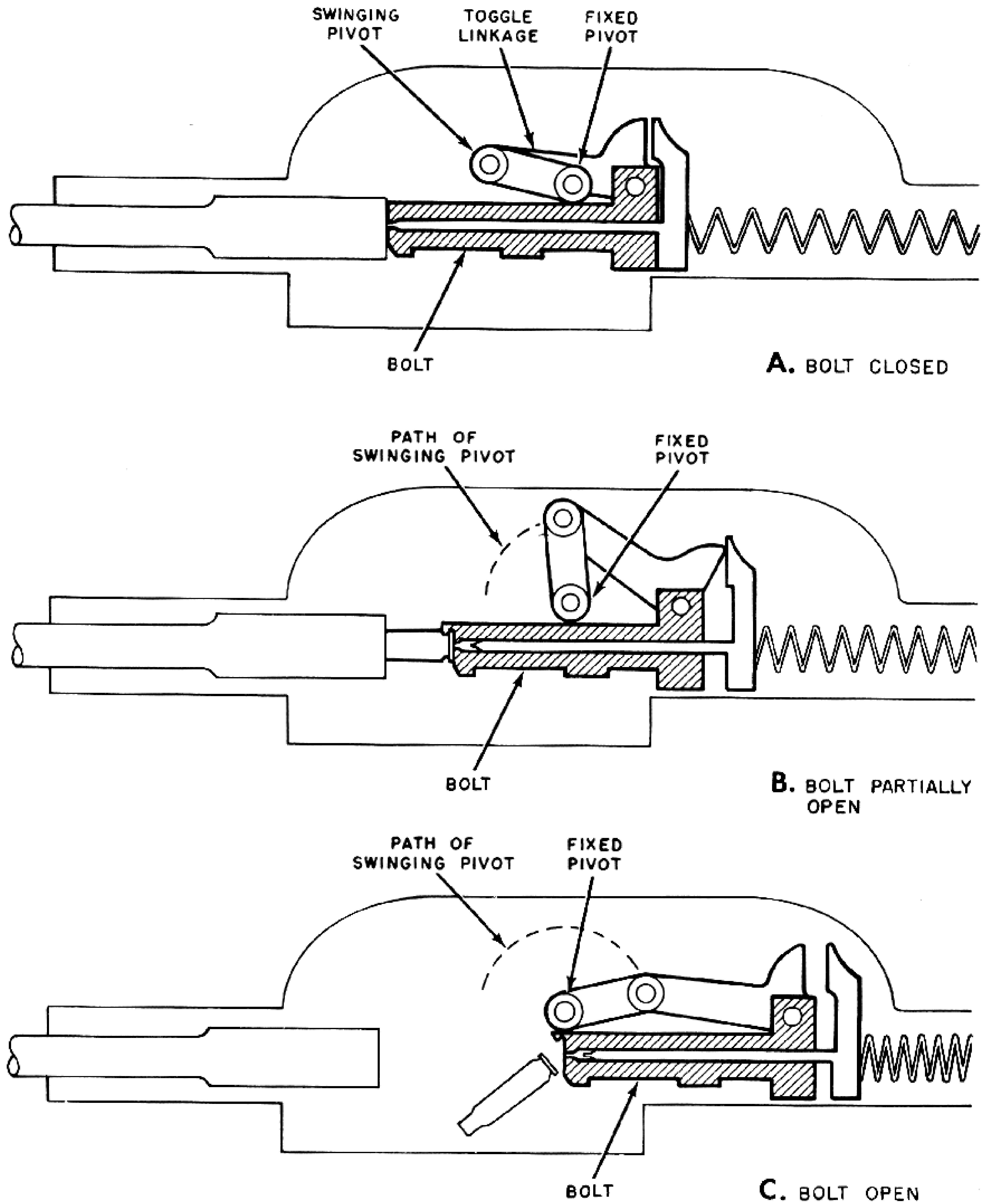


Figure 1–44. Breech Mechanism of Schwarzlose Machine Gun.

same as that of a very heavy bolt, a retarded blowback gun would suffer from all of the disadvantages of a plain blowback gun (except for the disadvantage of excess weight). These disadvantages, which are explained in detail under "Plain blowback system," are principally a very low rate of fire and insufficient bolt energy for operating the gun mechanism. Therefore, it appears that there is little point in using the retarded blowback system merely as a means of saving weight.

The real advantage of using retarded blowback can be obtained only if the retardation characteristics of the mechanism are *not* uniform. The ideal type of retarded blowback mechanism would be one in which a very high resistance to bolt movement is encountered during the period of high chamber pressure which exists until after the projectile has left the muzzle. This high resistance would result in a very low bolt velocity and a very small bolt movement during the period of high pressure. It would then be desirable for the last part of the small bolt movement to cause a change in the characteristics of the retarding mechanism so that the resistance is decreased by a substantial amount a millisecond or so after the projectile has left the muzzle and the residual pressure has dropped to a safe operating limit. The bolt, being subjected to greatly decreased resistance, would then be blown back safely at a relatively high velocity. An examination of the preceding requirements will indicate that an ideal retarding mechanism approximates the effect of a delay mechanism except that the bolt is not rigidly locked at any time. (Cf. "Delayed blowback system".)

It is important to note that effective use of retarded blowback requires a high degree of precision of mechanism and timing of operation. For safety and proper functioning, the change in the resistance offered by the retarding device must occur at exactly the correct time during the propellant explosion. This fact makes barrel length the most critical factor to be taken into consideration. For example, it has been stated that in the Schwarzlose machine gun (which employs a toggle linkage for retardation) the barrel length is critical to an unbelievable degree. If the barrel were slightly longer, the residual pressure would be too high as the linkage rises. Since the retardation would then be too low for the existing pressure, this would create a condition of extremely violent recoil and possible explosive rupture of the cartridge case. If the barrel were slightly shorter, the residual pressure would drop excessively before the linkage has risen sufficiently. Since the retardation would then be too high for the available pressure, there would be insufficient recoil and the gun would fail to function.

The foregoing example serves to indicate that a retarded blowback gun for high-powered ammunition must be designed and developed with great care. The parts of the retarding mechanism must be well

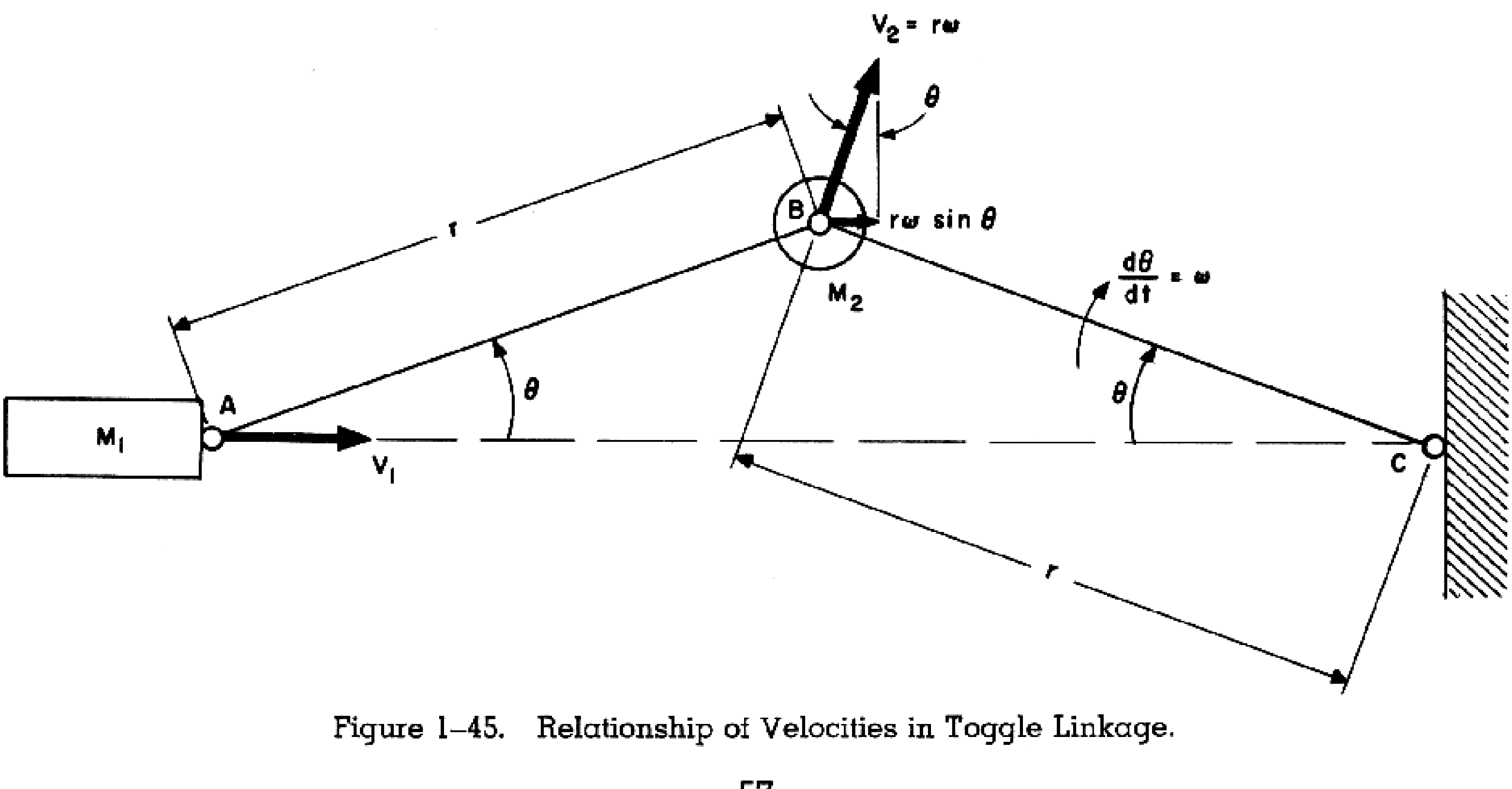


Figure 1–45. Relationship of Velocities in Toggle Linkage.

made and precisely fitted and the ammunition must be uniform in performance. Also, since the high forces resulting from the initial phases of propellant explosion are transmitted directly through the parts of the retarding mechanism, these parts must be of high-strength materials and carefully heat-treated. All things considered, mechanisms of this type are difficult and costly to manufacture and are at best suitable only for guns in which the recoil forces are not excessively great.

Mathematical Analysis of Retarded Blowback

The general character of the mass multiplying effects which can be obtained with a retarding device can be illustrated by considering a typical toggle linkage of the form shown in fig. 1–45. The linkage consists of two arms of equal length, r, pivoted at points A, B, and C. For purposes of analysis, the entire mass of the linkage M_2 is considered to be located at point B. The mass of the bolt itself is designated as M_1. The position of the linkage at any instant of time is defined by the angle θ between arm BC and the path of motion of the bolt.

1. *Initial retardation effect*

As arm BC rotates about point C, mass M_2 moves at a tangential velocity of $V_2 = r\omega$ where ω is the angular velocity of arm BC in radians per second. As indicated in the figure, the angle between the vector V_2 and the vertical is equal to θ and therefore the horizontal component of V_2 (in the direction of the bolt velocity V_1) is equal to $r\omega \sin \theta$. Since arm AB moves at the same angular velocity as arm BC, the bolt velocity V_1 is expressed by the relation:

$$V_1 = 2r\omega \sin \theta$$

Differentiating to obtain acceleration:

$$\frac{dV_1}{dt} = 2r \sin \theta \frac{d\omega}{dt} + 2r \cos \theta \frac{d\theta}{dt}$$

but:

$$\frac{d\omega}{dt} = \frac{d^2\theta}{dt^2} \text{ and, } r \frac{d^2\theta}{dt^2} = \frac{dV_2}{dt}$$

also:

$$\frac{d\theta}{dt} = \omega$$

Therefore:

$$\frac{dV_1}{dt} = 2 \sin \theta \frac{dV_2}{dt} + 2 r \cos \theta \, \omega^2$$

At the instant the bolt motion starts, $\omega = 0$. Hence, the relation between the initial accelerations of the masses reduces to:

$$\frac{dV_1}{dt} = 2 \sin \theta \frac{dV_2}{dt}$$

or

$$\frac{dV_2}{dt} = \frac{1}{2 \sin \theta} \frac{dV_2}{dt}$$

This relation shows that if θ is small, the acceleration of the linkage mass M_2 will be much greater than the acceleration of the bolt mass M_1. For example, if $\theta = 4°$, $\sin \theta = 0.0698$ and the acceleration of mass M_2 will be greater than the acceleration of mass M_1 by a factor of 7.17. In other words, the inertia reaction of a mass located at point B is 7.17 times greater than for the same mass located at point A.

Actually, the mass multiplying effect is further increased by the fact that the bolt acts through the linkage at a great mechanical disadvantage. Fig. 1–46 shows a vector diagram of the forces in the linkage. The inertia reaction of mass M_2 is equal to M_2 (dV_2/dt) directed perpendicular to arm BC. This force is in equilibrium with the indicated forces exerted at point B by the linkage arms. Taking component perpendicular to arm BC gives the force P_2 in arm AB as:

$$P_2 = \frac{M_2 \frac{dV_2}{dt}}{\sin 2\theta}$$

Taking horizontal components at point A gives the relation between P_2 and the force P_1 exerted on arm AB by the bolt

$$P_1 = P_2 \cos \theta$$

Substituting the value of P_2 into this expression gives:

$$P_1 = \frac{M_2 \frac{dV_2}{dt}}{\sin 2\theta} \cos \theta$$

As previously explained, at the instant the bolt motion starts:

$$\frac{dV_2}{dt} = \frac{1}{2 \sin \theta} \frac{dV_1}{dt}$$

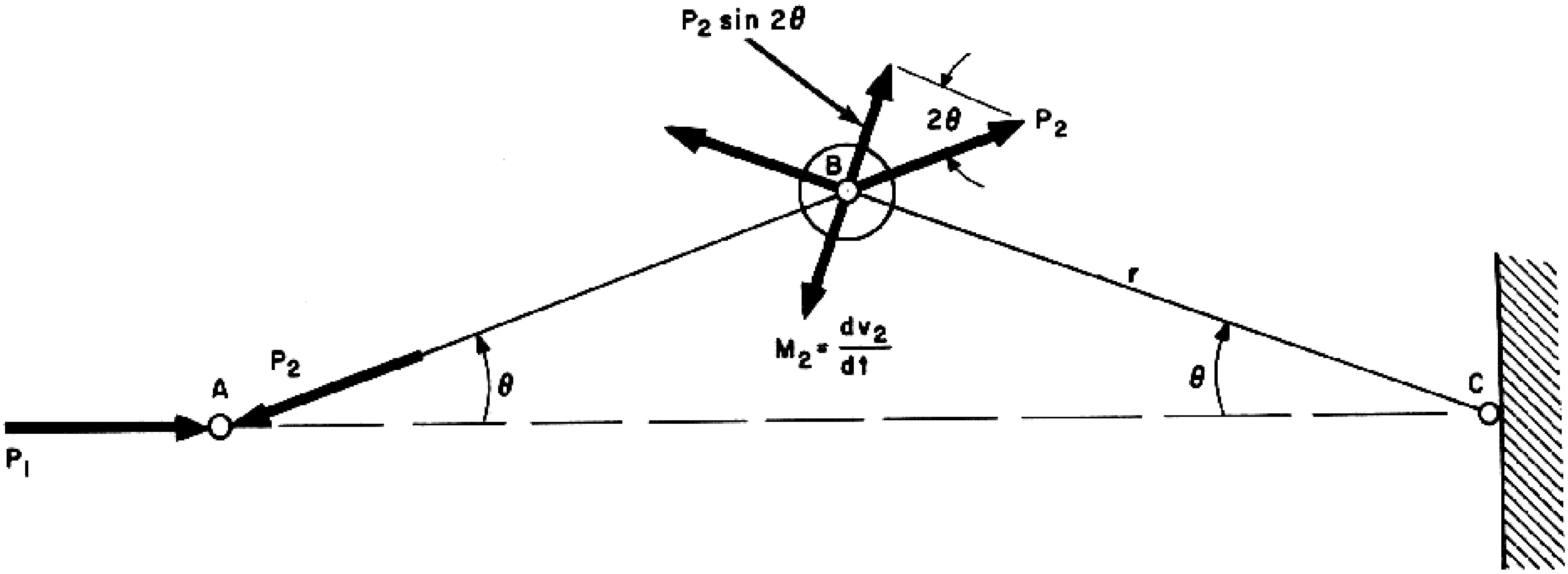


Figure 1–46. Effect of Inertia Reaction of Linkage.

Making this substitution in the expression for P_1 gives

$$P_1=\frac{M_2\dfrac{1}{2\sin\theta}\dfrac{dV_1}{dt}}{\sin 2\theta}\cos\theta$$

$$=\frac{\cos\theta}{2\sin\theta\sin 2\theta}M_2\frac{dV_1}{dt}$$

Since $\sin 2\theta=2\sin\theta\cos\theta$:

$$P_1=\frac{1}{4\sin^2\theta}M_2\frac{dV_1}{dt}$$

The factor, $1/(4\sin^2\theta)$ is the initial mass multiplying effect of the linkage and, for the instant the bolt motion starts, expresses the ratio by which the retardation offered by the linkage mass exceeds that of an equal mass located at point A. For example:

$$\text{If } \theta=4^\circ \quad \frac{1}{4\sin^2\theta}=\frac{1}{4(.0698)^2}=51.7$$

$$\theta=1^\circ \quad \frac{1}{4(.0175)^2}=820$$

$$\theta=\frac{1^\circ}{2} \quad \frac{1}{4(.00873)^2}=3280$$

Note that as θ is made smaller, the mass multiplying effect increases tremendously. (When $\theta=\frac{1}{2}^\circ$, a one-pound mass at point B would have the same initial retarding effect as a 3,280-pound mass located at point A.)

The foregoing analysis indicates that by means of a toggle linkage, the initial retardation can be made to be as great as desired merely by decreasing the initial angle. However, there are practical limitations on how small the initial angle should be.

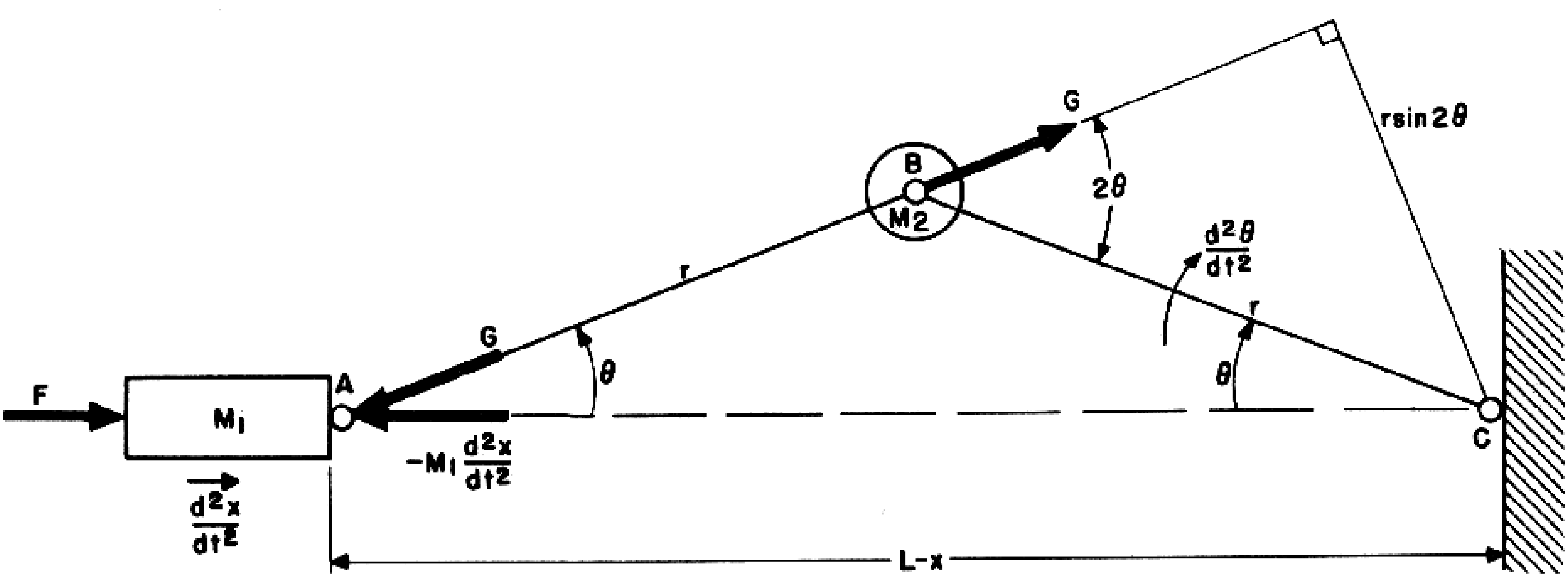


Figure 1–47. Forces in Toggle Linkage.

When angle θ is very small, proper functioning demands an extremely high precision in the dimensions of the parts. However, at small angles, tremendous forces are exerted on the links and the resulting deformation of the linkage will produce a condition in which the linkage will not act as effectively as theoretical analysis would indicate. For practical applications, the initial value of angle θ should not be much smaller than from 5 to 10 degrees.

2. *Analysis of bolt motion*

The analysis of the bolt motion in a retarded blowback gun employing a toggle linkage can not be conducted by the same methods as are described in this publication for other blowback guns. This is so because the mass effect of the linkage is not constant but varies as the parts of the linkage rotate, with the result that the motion can not be interpreted in terms of a simple relationship between impulse and momentum.

The differential equation expressing the free recoil conditions in the linkage may be derived from fig. 1–47. F is the force exerted on the bolt by the powder gas pressure at any instant of time. (F is an empirical function of time which may be derived from a graph of chamber pressure versus time of the type shown in fig. 1–9.) Considering mass M_2 and arm BC at angle θ, the torque resulting from the force G applied to M_2 by arm AB will be equal to the moment of inertia of arm BC times the angular acceleration of this arm about point C. That is:

$$Gr \sin 2\theta = I \frac{d^2\theta}{dt}$$

Since the entire mass of the linkage is assumed to be concentrated at point B, the moment of inertia is given by the relation

$$I = M_2 r^2$$

Making this substitution and solving for G gives:

$$G = \frac{M_2 r}{\sin 2\theta} \frac{d^2\theta}{dt^2}$$

Now considering the equilibrium of the horizontal forces on the bolt at point A gives the relation

$$F = M_1 \frac{d^2x}{dt^2} + G \cos\theta$$

Substituting the expression for G previously obtained gives:

$$F = M_1 \frac{d^2x}{dt^2} + \frac{M_2 r}{2 \sin\theta} \frac{d^2\theta}{dt^2} \cos\theta$$

But $\sin 2\theta = 2 \sin\theta \cos\theta$

Therefore:

$$F - M_1 \frac{d^2x}{dt^2} + \frac{M_2 r}{2 \sin\theta} \frac{d^2\theta}{dt^2}$$

Angle θ may be eliminated from this expression as follows: If L is the initial distance between point A and point C and x is the linear displacement of the bolt along line AC:

$$\theta = \cos^{-1} \frac{L-x}{2r}$$

Differentiating to obtain $d\theta/dt$ gives:

$$\frac{d\theta}{dt} = \frac{\frac{dx}{dt}}{2r\sqrt{1-\left(\frac{L-x}{2r}\right)^2}} = \frac{\frac{dx}{dt}}{\sqrt{4r^2-(L-x)^2}}$$

Differentiating to obtain $d^2\theta/dt^2$ gives:

$$\frac{d^2\theta}{dt^2} = \frac{\sqrt{4r^2-(L-x)^2}\,\frac{d^2x}{dt^2} - \frac{L-x}{\sqrt{4r^2-(L-x)^2}}\left(\frac{dx}{dt}\right)^2}{[4r^2-(L-x)^2]}$$

$$\frac{d^2\theta}{dt^2} = \frac{[4r^2-(L-x)^2]\,\frac{d^2x}{dt^2} - (L-x)\left(\frac{dx}{dt}\right)^2}{[4r^2-(L-x)^2]^{3/2}}$$

Also:

$$\sin\theta = \sqrt{1-\cos^2\theta} = \sqrt{1-\left(\frac{L-x}{2r}\right)^2} = \frac{1}{2r}\sqrt{4r^2-(L-x)^2}$$

Substituting the expressions for $d^2\theta/dt^2$ and $\sin\theta$ in the differential equation gives:

$$F = M_1 \frac{d^2x}{dt^2} + M_2 r^2 \left[\frac{[4r^2-(L-x)^2]\,\frac{d^2x}{dt^2} - (L-x)\left(\frac{dx}{dt}\right)^2}{[4r^2-(L-x)^2]^2}\right]$$

Unfortunately, the solution to this equation can not be expressed in finite form in terms of elementary functions of t. The situation encountered here is similar to that which arises in the mechanics of a simple pendulum. In the case of the simple pen-

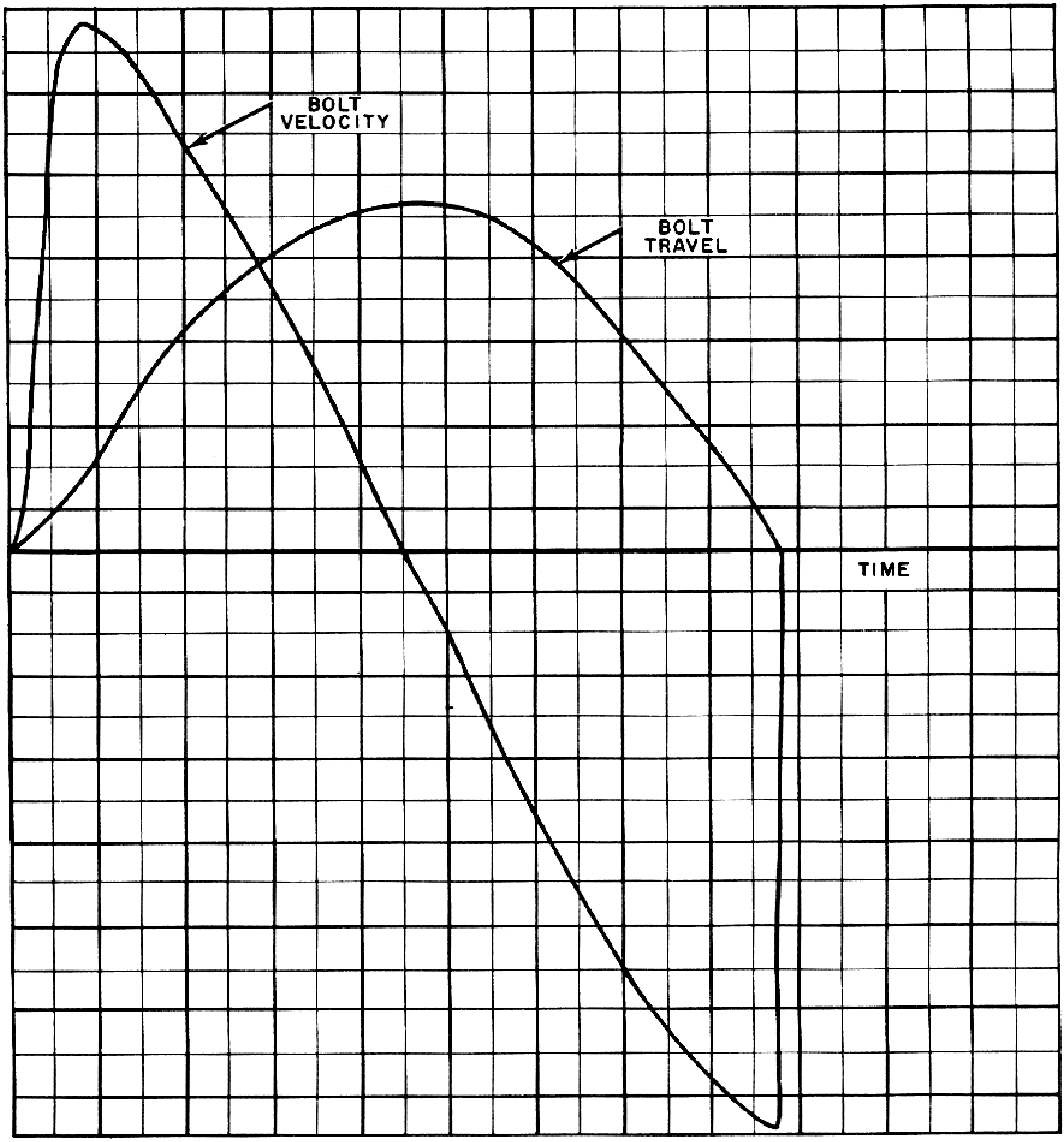


Figure 1–48. Time-Travel and Time-Velocity Curves for Complete Cycle of a Retarded Blowback Gun.

dulum, the relationship between the displacement of the bob and time involves an integral of a special type known as an "elliptic integral" which can be evaluated by the use of infinite series or by means of special tables. However, the equation derived above for the toggle linkage breech mechanism is further complicated by the fact that F is an empirical function of time and any attempt to obtain a solution by direct methods would involve extremely intricate manipulations. Probably the best method of attack would be to assign suitable values to M_1, M_2, r, and L and then (using a curve expressing the variation of F with time) to evaluate the integrals involved in the solution by one of the standard methods for numerical approximation. Because of the form of the differential equation and the fact that F varies with time, even this approach will be very laborious and much too tedious to demonstrate in this publication. Accordingly, the time-travel and time-velocity curves will not be developed here.

The general form of the time-travel and time-velocity curves for the complete cycle of a retarded blowback gun employing a toggle linkage (fig. 1–48) will closely resemble the form of the curves for a delayed blowback gun.

Chapter 2

RECOIL OPERATION

PRINCIPLES OF RECOIL

The principles of recoil can be understood best by considering the forces which result from firing a cartridge in an elementary gun. Such a gun (shown schematically in fig. 2–1) consists of a barrel having a chamber at its rear end for receiving the cartridge and a breech closure in the form of a bolt. The bolt is rigidly locked to the barrel after the cartridge is inserted, thus providing a firm support for the base of the cartridge case so that the case will not be blown out of the chamber by the explosion of the propellant charge.

When the cartridge is fired, the explosion of the propellant results in the rapid generation of extremely high gas pressure in the chamber and the expansion of this high-pressure gas drives the projectile forward through the bore. As the powder gases expand behind the projectile, the center of mass of the gases also moves forward. While the projectile is in the bore, the same pressure which causes the projectile and powder gases to move forward also acts simultaneously at the breech end of the gun to produce an equal and opposite reaction which tends to drive the entire gun to the rear. The force resulting from this reaction is called the "recoil force" and the magnitude of this force at any instant depends on the chamber pressure.

Fig. 2–2 shows how the chamber pressure varies with time in a typical 20-mm gun with a total barrel length of slightly less than five feet. The parts into which the curve is divided show the proportion of the pressure expended in producing the velocity of the projectile and in imparting motion to the powder gases. (A very small proportion of the total pressure is expended in overcoming bore friction and in imparting rotation to the projectile, but this proportion is too small to be shown in the figure.) The pressure producing the recoil force is the total of both parts of the curve. Fig. 2–2 shows that the recoil force does not immediately cease when the projectile leaves the gun after 0.00234 second but continues to act until the residual pressure falls to zero at approximately 0.008 second.

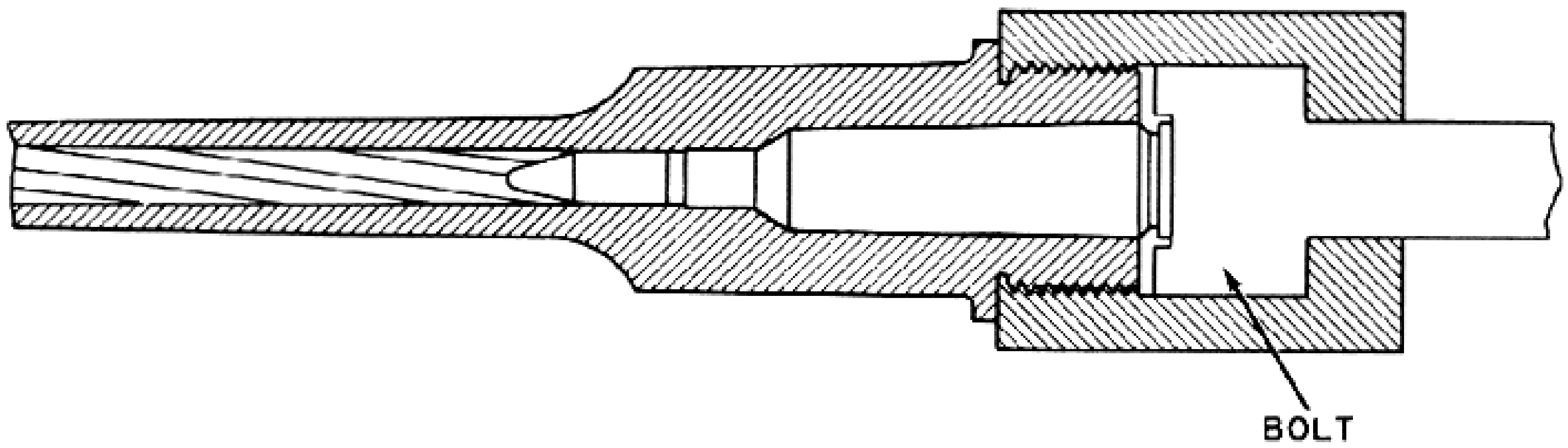


Figure 2–1. Elementary Gun.

The effect of the recoil force on the gun depends entirely on how the gun is mounted. If it is assumed that the gun is mounted so that it can move freely without friction or any other restraint, the impulse of the recoil force (acting over the entire 0.008 second) will impart to the gun a rearward momentum equal to the total forward momentum of the projectile and powder gases. The velocity of the gun will depend on its mass; the smaller the mass, the higher the velocity. When the residual pressure falls to zero at 0.008 second, the gun will have achieved its maximum velocity and since it is assumed that there are no external restraining forces, the gun will continue to move at this velocity. This hypothetical condition is referred to as the condition of "free recoil".

With any practical weapon, the rearward motion of the gun must be controlled and limited by the application of restraining forces, producing the con-

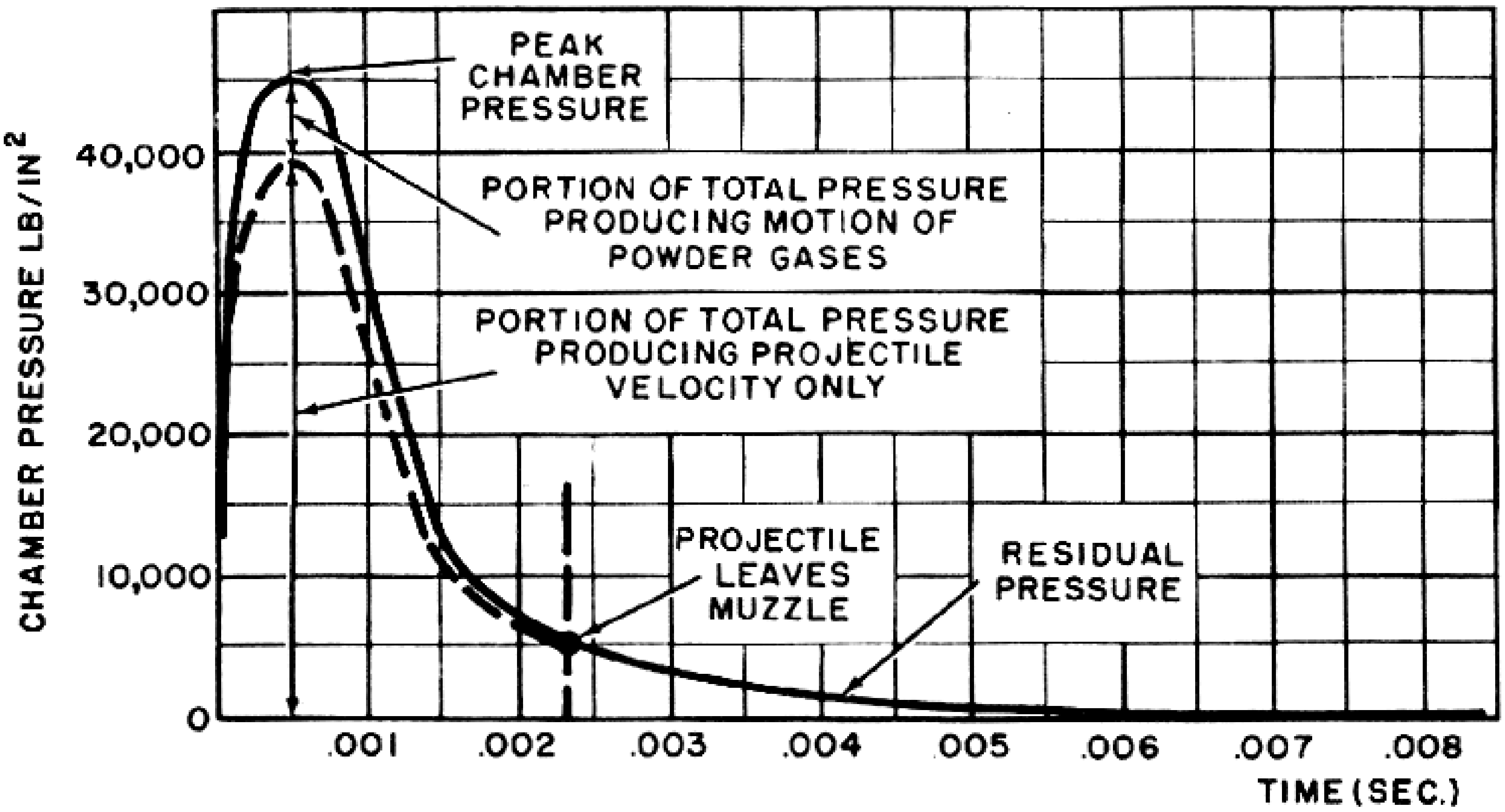


Figure 2–2. Variation of Chamber Pressure With Time.

dition known as "retarded recoil". In many guns, the effect of recoil is merely an inconvenience which must be tolerated since it can not be avoided. The primary source of difficulty in such cases is the tremendous magnitude of the recoil forces resulting from firing high-powered ammunition. (For example, it might be pointed out here as a matter of general interest that maximum recoil force in a 16-inch gun having a maximum chamber pressure of 38,000 pounds per square inch is the stupendous figure of almost 8,000,000 pounds. Even in a 20-mm gun, the maximum recoil force can amount to over 22,000 pounds.) While it might be possible to build a rigid mount capable of directly withstanding forces of this magnitude, such a mount would be entirely too heavy and cumbersome for any practical purpose. Accordingly, it is necessary to permit the gun to move during and after the action of the recoil force so that the momentum of the gun can be cancelled through the application of a smaller retarding force which acts over a considerable interval of time and distance. Since the gun moves in recoil, it acquires kinetic energy which must be absorbed in order to bring the gun to rest. For the above reasons, many guns (particularly those of heavier calibers) are mounted in a slide and are provided with a recoil brake and a recuperator, both of which absorb the recoil energy. Most of the energy is dissipated by the brake in the form of heat and the energy stored in the recuperator is utilized to return the gun to battery.

In hand-held weapons, the recoil momentum is cancelled by a force applied to the weapon through the arm or shoulder of the firer. For example, when a Springfield rifle is fired, the maximum force of recoil is about 3700 pounds. (The maximum chamber pressure of 52,000 psi times the bore cross-section area $(\pi/4)\ (.30)^2$ in., gives the maximum force as $52{,}000 \times .0707 = 3670$ pounds.) Actually, this force is not applied directly to the shoulder of the firer but produces the rearward acceleration of the gun known as "kick." The shoulder of the firer then acts as a recoil brake which absorbs the resulting kinetic energy of recoil over a travel of two or three inches. If it is assumed that the rifle moves three inches in recoil, the average force applied by the shoulder against the stock must be in the order of 60 pounds. If an attempt were made to limit the recoil distance to one-quarter inch, the average force would be 720 pounds. (This indicates why it is not a good idea to fire a high-powered rifle with the shoulder rigidly backed up against a solid support.)

Up to this point, recoil has been considered as undesirable because it creates forces and energies which must be absorbed and dissipated after each shot of an ordinary gun. However, in an automatic weapon, these same forces and energies represent a convenient source of power for operating the gun mechanism. Machine guns which utilize this source of power are said to employ the "recoil system" of operation.

The amount of operating energy available from recoil depends on the particular cartridge used and on the weight of the recoiling parts. The use of a powerful cartridge will result in a high total momentum imparted to the projectile and powder gases and the corresponding reaction will produce a high momentum of the recoiling parts. However, the velocity of recoil will depend on the weight of the recoiling parts; the greater the weight, the lower the velocity. Since the kinetic energy of a weight is directly proportional to its mass and the square of its velocity, the net result is that the energy contained in the recoiling parts will be inversely proportional to their weight. In other words, increasing the weight of the recoiling parts will not change the momentum imparted to them but will cause a decrease in both the recoil velocity and the recoil energy. Thus it is evident that the weight of the recoiling parts has an important effect on the performance of a recoil-operated gun.

All of the kinetic energy imparted to the recoiling parts is transferred to them in the period of less than 0.01 second during which the pressure of the powder gases is active. After this time, the parts continue to move of their own momentum and their energy is transferred to the gun mechanism for performing the various operations required in the automatic cycle. In various types of recoil systems, the energy may be employed in different ways. These systems will be described and analyzed in the following pages.

RECOIL OPERATION

In the recoil system of operation, provision is made for locking the bolt to the barrel and these parts are mounted in the receiver so that they can slide to the rear. The gun is fired with the bolt locked to the barrel and these parts remain locked together as they are thrust back by the pressure resulting from the explosion of the powder gases. In some guns, the energy derived from this motion is used to perform the entire cycle of operation; in other guns the energy derived from recoil may perform only certain functions in the cycle or may merely supplement the energy derived from another system of automatic operation.

The distinguishing characteristic of the recoil system is that energy used for operation is obtained from the recoil movement of the barrel and bolt while these parts are locked together. Any gun in which the bolt is locked to the barrel while there is pressure in the chamber will be subject to some recoil action, but unless the recoil is put to use in operating the gun mechanism, the gun does not employ the recoil system. For example, some gas-operated guns are arranged so that recoil can occur while the barrel and bolt are locked together but this motion is permitted only in order to obtain a reduced trunnion reaction. Since no energy for operating the gun mechanism is obtained from recoil, such guns can not be said to use the recoil system.

In the following description of recoil operation, it will be assumed that the bolt remains locked to the barrel during the entire time of action of the powder gas pressure. Some guns using the recoil system are designed so that the bolt is unlocked when the residual pressure has fallen to a safe operating limit but has not yet reached zero and are therefore subject to some blowback action.

In a gun operated purely by recoil the bolt remains locked to the barrel until the chamber pressure has become zero and therefore, there are no problems (such as are encountered in blowback) resulting from movement of the cartridge case under pressure and lubrication of the ammunition is unnecessary.

> NOTE: Of course, this statement is true only if the gun is adjusted for correct head space. If the head space is excessive, case separation will occur for reasons which are explained in detail under *Blowback*.

Since the use of recoil operation disposes of the limitations imposed by cartridge case movement, the main problem confronting the designer of a recoil weapon is how to make efficient use of the energy available for operating the mechanism.

RECOIL SYSTEMS

The number of different machine guns employing the recoil system of operation is very large and an examination of these weapons will reveal an extreme diversity of mechanical detail and functional arrangement. However, in spite of these dissimilarities, all recoil-operated weapons can be placed in one of two basic sub-classes: long recoil or short recoil.

Long recoil is defined as a system of operation in which energy for operating the gun mechanism is obtained from a recoil movement which is greater than the overall length of the complete cartridge. During this entire movement, the bolt remains locked to the barrel. At the end of the rearward movement, the bolt is unlocked from the barrel and is latched in its rearmost position while the barrel moves forward in counter-recoil, thus pulling the chamber off the empty cartridge case and ejecting the case. When the barrel has moved forward far enough to provide a sufficient opening for feeding, and just before its counter recoil movement is completed, the bolt is unlatched. The bolt then moves forward to perform the function of loading and the cycle is completed as the bolt relocks to the barrel.

In a short-recoil weapon, the bolt remains locked to the barrel for only a portion of the recoil stroke required to permit feeding. After unlocking occurs, the barrel may move a short distance with the bolt but it is then stopped. The bolt, on the other hand, continues to move to the rear and it may complete this movement by virtue of the momentum it had at the time of unlocking or it may receive additional momentum through the action of a mechanical device (known as an "accelerator") which transfers some of the energy of the barrel to the bolt. In either case, the rearward motion of the bolt continues until the opening is sufficient for feeding and the bolt is then moved forward to close and lock the breech. In some guns, the bolt may push the barrel back to the firing position but in others, the return motion of the barrel is accomplished independently before the bolt closes.

LONG RECOIL SYSTEM

In the following paragraphs the long recoil system of operation is described and analyzed by considering the sequence of events which occur in the automatic cycle of operation. As with other systems treated in this publication, this analysis is concerned only with the functioning of the mechanisms and with the general factors affecting design.

Because of its relatively simple functional characteristics, the long recoil system will be analyzed first. Actually, existing examples of guns using this system are far from simple when viewed from the standpoint of mechanical detail because of the complexities involved in actuating the locking and latching devices. However, these mechanical complexities are not directly concerned in an analysis of the basic motions and forces encountered in long recoil operation and therefore will not be considered here.

Fig. 2–3A shows schematically the essential elements of a gun which operates by the long recoil system. These elements consist of the bolt, an arrangement for locking the bolt to the barrel and unlocking it, a bolt driving spring for returning the bolt after recoil, a spring to return the barrel, and a latch to hold the bolt to the rear while the barrel is moving forward. The other portions of fig. 2–3 show different stages during the cycle of operation.

Cycle of Operation

The operating cycle of a typical long recoil gun occurs as follows:

The cycle starts with a cartridge in the chamber and with the bolt locked to the barrel (fig. 2–3A). When the cartridge is fired, the pressure of the powder gases drives the projectile and gases forward and at the same time drives the barrel and locked bolt to the rear in recoil. During the action of the powder gas pressure, the retardation offered by the springs is relatively small and the only really significant factor in limiting the recoil acceleration is the mass of the recoiling parts.

The force exerted by the powder gases exists for a relatively very short time. In a typical 20-mm gun with a barrel length of about five feet, the projectile leaves the muzzle after approximately 0.0023 second and the residual pressure continues to act until 0.008 or 0.009 second after ignition of the primer. After this point, the powder gas pressure is zero and the recoil force is also zero but the barrel and bolt continue to move of their own momentum. As the barrel and bolt move back, the barrel spring and bolt driving spring are compressed and the resistance of these springs gradually slows down the recoil parts until their velocity is zero at the extreme recoil position shown in fig. 2 3B. (The

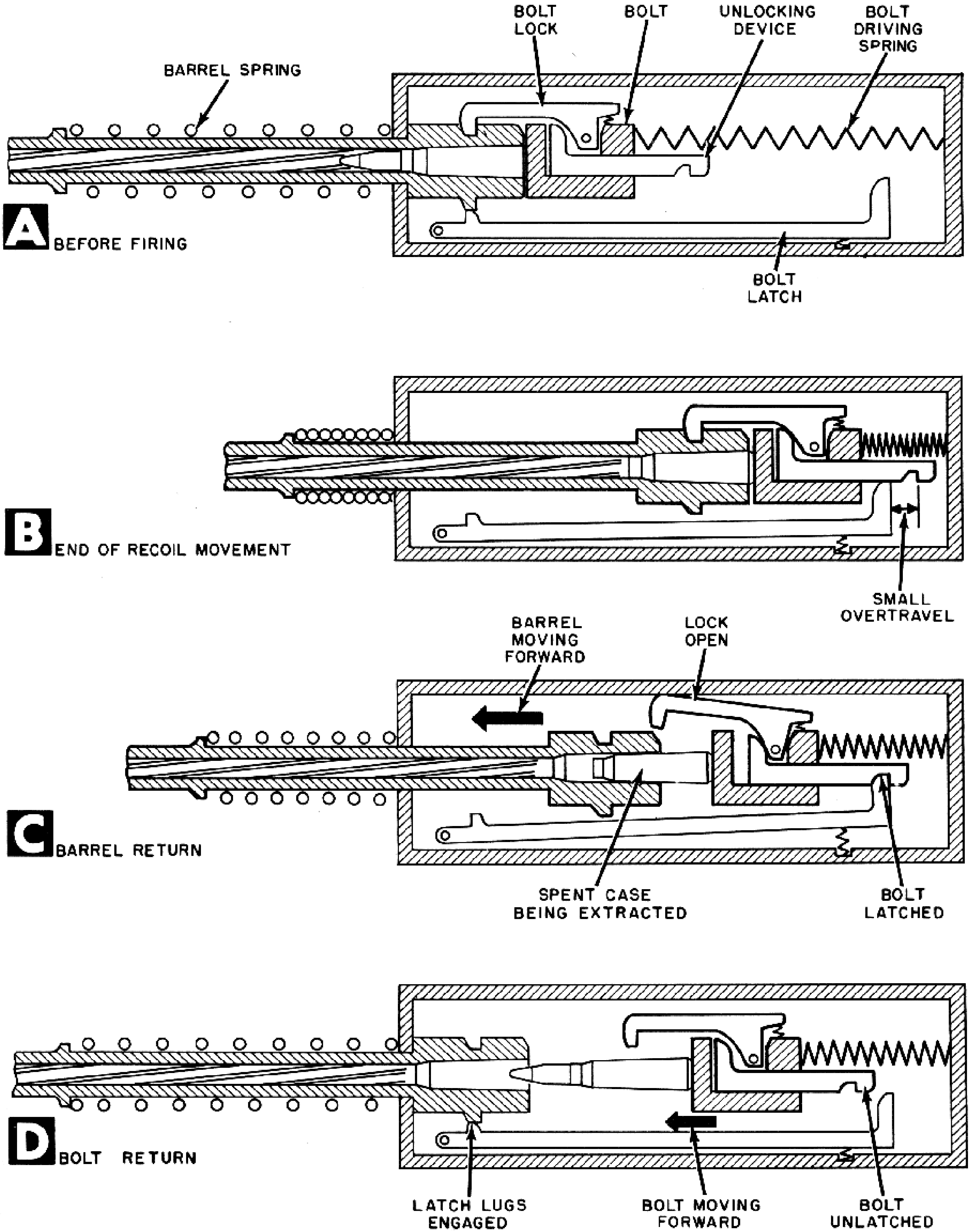


Figure 2–3. Elements of Long Recoil System (Schematic).

small overtravel indicated in the figure is allowed in the design to take care of normal variations in the loading of the cartridges.) Note that up to this point, the bolt is still locked and no automatic operation other than firing has been completed.

After the springs have brought the barrel and bolt to rest, the force of the compressed springs starts moving these parts forward. However, the bolt unlocking device is almost immediately caught and held by the bolt latch (fig. 2–3C) and the forward pull of the barrel causes the bolt lock to open. Therefore, the barrel continues to move forward, causing extraction and ejection of the case. Feeding of a fresh round can not occur until the barrel is at or nearly at the fully forward position, because only then is the bolt open a sufficient amount to permit feeding. Just before the barrel reaches the fully forward position, all of the potential energy stored in the barrel spring during recoil has been transformed back into kinetic energy (except for losses such as are occasioned in unlocking, extraction, ejection, or overcoming friction). Therefore the barrel is moving with considerable velocity and its kinetic energy must be absorbed either by impact or by a buffer.

When the barrel reaches the fully forward position, it actuates the bolt latch to release the bolt (fig. 2–3D). The bolt driving spring then pushes the bolt forward to feed and load a new round, lock the breech, and actuate the firing mechanism to start a new cycle of operation. Just as the bolt locks to the barrel, all of the potential energy stored in the bolt driving spring during recoil has been converted back to kinetic energy (except for the losses due to feeding, loading, locking, and actuating the firing mechanism). This kinetic energy is absorbed by impact as the bolt is brought to rest.

Analysis of Long Recoil

In the preceding descriptions of recoil operation and of the long recoil system, it was pointed out that, since the bolt remains locked until the chamber pressure is zero, there are no special difficulties arising from cartridge case movement under pressure and that the principal problem confronting the designer is how to make efficient use of the available recoil energy. The manner in which the recoil energy is utilized in a long recoil gun is largely dependent upon the power of the cartridge considered in relation to the attainment of a practical rate of fire.

The long recoil system has several important advantages which relate to the problems involved in handling recoil forces and which make it especially suitable for large-caliber guns employing high-powered ammunition. In such guns, the large bore cross-section area coupled with high chamber pressure produces extremely great recoil forces and these forces tend to result in violent action and excessive recoil energy. In a long recoil gun, the rearward thrust of the recoil forces is expended in producing motion of the combined mass of the barrel and bolt and since this mass is quite heavy, the recoil energy will be correspondingly relatively low. Second, since the recoil energy can be absorbed more or less gradually over a relatively long movement in recoil, the forces involved in absorbing the energy can be kept within reasonable limits, thus giving fairly low trunnion reactions as well as reduced stresses in the parts of the gun mechanism.

Although the long recoil system is well adapted to handling large-caliber high-powered ammunition, it has a disadvantage in regard to rate of fire. The basic difficulty with the long recoil system is that the sequences necessary in the automatic cycle of operation do not permit efficient utilization of time. To demonstrate this point, the entire time necessary for the recoil movement is essentially wasted because the bolt must remain locked this entire period, thus delaying the accomplishment of unlocking, extraction, ejection, and the other automatic functions. Furthermore, after unlocking occurs, a substantial portion of the time required for barrel return is expended merely in extracting the spent case. Ejection and feeding must be delayed until the return of the barrel has been practically completed and only then can the bolt start its return movement. Because all of these delays are inherent in the long recoil system, guns based on the long recoil system tend to have a rather low rate of fire.

When considering the expression "rate of fire", it is important to realize that the terms "low" or "high" are only relative and depend entirely on the caliber of the gun. What may be thought of as a very low rate for a 20-mm gun might be a terrifically high rate for a gun of much greater caliber. In a larger sense, the meanings of the terms "low" or "high" as applied to rate of fire are dictated by the type of target for which the gun is intended. For high speed air-to-air combat in which the gun can be brought to bear on target for usually only a split second, a rate of fire of 1000 rounds per minute might be considered none too high. In such combat, the target may be fairly light so that effective hits can be scored with ammunition of relatively small caliber. However, for targets such as buildings, ships, and heavily armored vehicles, it is usually desirable to use high-powered ammunition of large caliber and there probably will be ample time for firing. Under such conditions, a rate of fire of over 1000 rounds per minute is not necessary and it may be preferable to deliver a few devastating blows to the target rather than a large number of lighter hits.

Because of its advantages in regard to handling high recoil forces, the long recoil system is applied extensively for large-caliber weapons and, considering the caliber of these weapons, the rates of fire attained are fairly high. The disadvantage of the long recoil system in regard to rate of fire becomes evident only for smaller caliber guns. In such guns, the distances through which the recoiling parts must move are more or less predetermined by the total length of the cartridge used. Therefore, the only way in which the tendency toward a low cyclic rate can be minimized is to design the gun so that the recoil movement and the return movements of the barrel and bolt occur at the highest velocity attainable on a practical basis. Thus, in an analysis of long recoil as applied to smaller caliber guns, primary attention must be given to those factors which affect the barrel and bolt velocities.

Assuming the use of a particular type of ammunition, the main factor affecting the velocities of the recoiling parts is the weight of these parts. With a given ammunition, the total forward momentum imparted to the projectile and powder gases is some definite amount and the resulting reaction will produce an equal and opposite momentum in the recoiling parts. In other words, the momentum imparted to the recoiling parts is some definite amount which is determined by the cartridge used. The recoil velocity which corresponds to this momentum will be inversely proportional to the weight of the recoiling parts and therefore the maximum recoil velocity will depend on the weight of the recoiling parts. This necessitates that, to obtain a high recoil velocity, the recoiling parts should be made as light as possible.

Unfortunately, there are definite limitations to how small the recoiling parts can be. In order to perform their functions and to withstand the forces to which they are subjected, the barrel, bolt, and other recoiling parts must be ruggedly constructed and will necessarily be fairly massive. In fact, all is based on the propellant charge and the more powerful the ammunition, the heavier these parts will be. Since there is a limit to how light the operating components can be, the maximum attainable recoil velocity is similarly limited.

The extent of this limitation can be illustrated by considering a 20-mm gun. In a gun of this caliber with a maximum chamber pressure of 45,000 pounds per square inch, the barrel and barrel extension assembly alone could hardly weigh much less than 35 pounds, no matter how economically it is designed from the standpoint of weight. Allowing a conservative 6 or 8 pounds for the bolt, locking device and firing mechanism and another 6 or 8 pounds for the effect of the spring masses gives a total minimum weight close to 50 pounds. The recoil momentum produced by the assumed cartridge will be approximately 35 (lb. sec.) and dividing this figure by the mass of the 50-pound recoiling parts gives a maximum free recoil velocity of about 22.5 feet per second. Now it must be realized that this velocity represents (to a fair approximation) the highest value attainable in the assumed weapon because an attempt to increase this velocity by lightening the parts would make the parts too frail.

Thus it appears that the designer of a 20-mm gun operated purely by long recoil is "stuck with" a maximum initial recoil velocity of somewhere near 22.5 feet per second. It will be recalled from the description of the long recoil cycle of operation that the barrel must recoil and counter-recoil the full distance before the bolt starts its return movement. For a 20-mm gun, the recoil distance must be from eight inches to nearly one foot to permit feeding and therefore the barrel must travel a total distance of almost two feet per cycle. Even if it is assumed (although this is impossible) that this entire motion is accomplished at the maximum velocity of 22.5 feet per second and that the bolt return time is ignored completely, this will mean that the time required for each cycle will be $2/22.5 = .089$ second which gives a rate of fire of 675 rounds per minute. Actually the barrel must be stopped and its motion must be reversed at the end of recoil and its average velocity will necessarily be lower than the maximum free recoil velocity. Furthermore, the bolt return time will have a significant effect on the rate of fire under practical conditions. Therefore, the maximum attainable rate of fire will be considerably lower than 675 rounds per minute and the best that can be expected is in the neighborhood of 300 to 500 rounds per minute. In modern terms, this is considered a prohibitively low cyclic rate for any 20-mm machine gun.

From what has been said, it can be seen that to achieve even the relatively low rate of 500 rounds per minute with long recoil operation, the designer must carefully utilize every possible means at his command to take full advantage of the following points:

1. To obtain a high initial recoil velocity, the recoiling parts must be made as light as possible consistent with the practical requirements for strength, rigidity, and durability.
2. The recoil distance should be no greater than the minimum necessary to provide an adequate opening for feeding, as governed by the overall length of the incoming round.
3. The gun mechanism should be arranged to minimize delays by taking advantage of every possible instant of time (rapid unlocking, eject at instant extraction is completed, etc.).
4. In smaller caliber guns, instead of using the barrel spring to absorb most of the recoil energy, this spring could be made as light as possible and be depended upon only to hold the barrel in battery. This would produce a condition of low retardation, thus permitting the barrel and bolt to recoil the entire distance with little decrease in velocity. The reversal of their motion at the end of recoil could then be accomplished by causing the recoiling parts to rebound from an extremely stiff buffer spring. Although this arrangement gives a high average velocity of recoil, it should be noted that the reversing action of the buffer spring will be rather violent, and accompanied by high impact forces. In addition, the impact loss in the buffer may be considerable with the result that the slowness of the long counter-recoil movement of the barrel would reduce or eliminate entirely any advantage gained during recoil. Since the advantage of employing this method is doubtful in a long-recoil gun, the use of a conventional barrel spring will be assumed from this point forward.
5. The barrel spring and bolt spring should be proportioned so that the total time required for returning the barrel and bolt will be minimized. (The calculations necessary to accomplish this are described in the mathematical analysis of long recoil.)
6. It might be mentioned here that some improvement in the rate of fire can be accomplished by the use of a so-called "recoil intensifier" or "muzzle booster". This device utilizes the residual gas pressure by trapping the muzzle blast in such a way as to increase the rearward thrust on the muzzle face of the barrel and thus increases the recoil velocity.

All of the preceding points are concerned with the problem of obtaining the optimum rate of fire from the long recoil system as applied to a smaller caliber gun, but as has been remarked previously, even with these refinements, the cyclic rate will still be relatively low. This limitation, coupled with the design disadvantages arising from the long travel of the heavy barrel and from the inherent complexity of the mechanism, probably accounts for the fact that no successful machine gun using the long recoil system has appeared among modern weapons of 20-mm or smaller caliber.

In the large caliber guns to which the long recoil system has been applied successfully, the high recoil energy involved has made it necessary to ignore many of the points described in the preceding paragraphs. In these guns there is so much excess recoil energy that the problem is one of minimizing and disposing of energy rather than one of taking advantage of every possible means to increase the velocity of recoil and rate of fire. Therefore the approach is exactly opposite to that described for smaller caliber guns. The recoiling parts are deliberately made heavy to minimize the energy imparted to them, and recoil brakes and buffers are utilized to remove energy from the system. The recoil stroke is made as long as practical considerations will permit and muzzle brakes rather than muzzle boosters are sometimes employed. All things considered, the long recoil system is best adapted to handling high recoil forces and is not the best system for use in smaller caliber weapons in which high cyclic rates are desired. If the number of rounds delivered per minute is the primary consideration

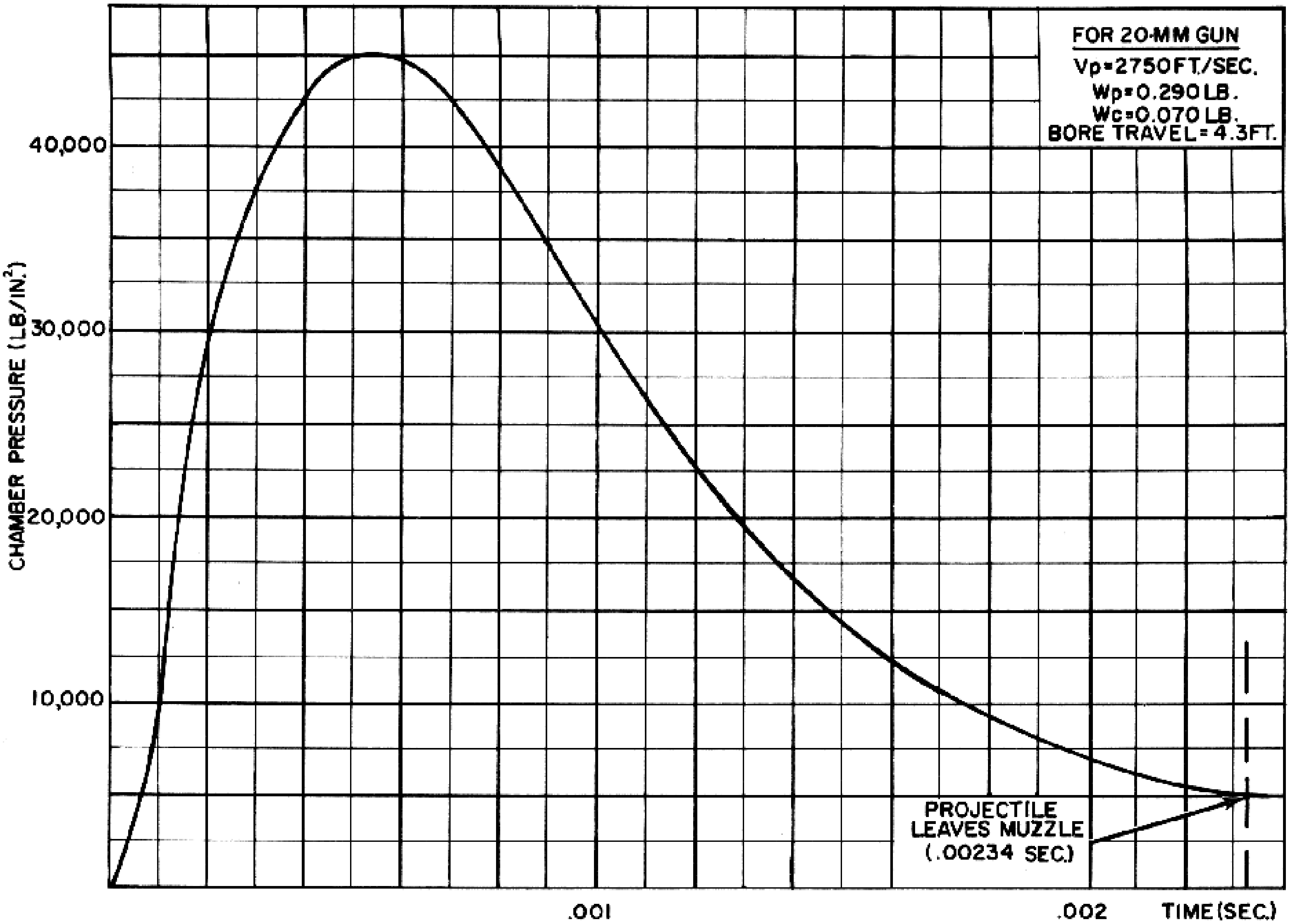


Figure 2–4. Graph of Chamber Pressure Versus Time (20 mm. Gun).

and the recoil forces are not excessive, it is safe to say that some other system than long recoil should be used.

In the following mathematical analysis, the gun used to illustrate the computations will be a 20-mm gun in spite of the fact that long-recoil may not be the ideal system for a gun of this caliber. This is done in order to permit comparison of the results of the analysis with the results obtained from the analyses of the other operating systems described in this publication which also used 20-mm data for purpose of illustration.

Mathematical Analysis of Long Recoil

The following paragraphs describe a systematic procedure for performing the computations necessary in a basic analysis of a gun operated purely by long recoil. This procedure follows the same general lines used for the analysis of the other systems described in this publication with the modifications necessary for considering the specific problems arising in the design of long recoil machine guns. As in the other analyses, no attempt will be made to discuss the straightforward machine design methods used for arriving at the particular physical form of the mechanisms or to make detailed computations for the effects of friction or the relatively minor forces incident to operating the auxiliary mechanisms such as the bolt lock, feeder, firing device, or ejector.

The following analysis is based on the assumption that a particular cartridge with known characteristics is to be used and that the desired muzzle velocity and barrel length have been predetermined. It is also assumed that all necessary interior ballistics data are known and that graphs showing the time variation of projectile velocity, chamber pressure,

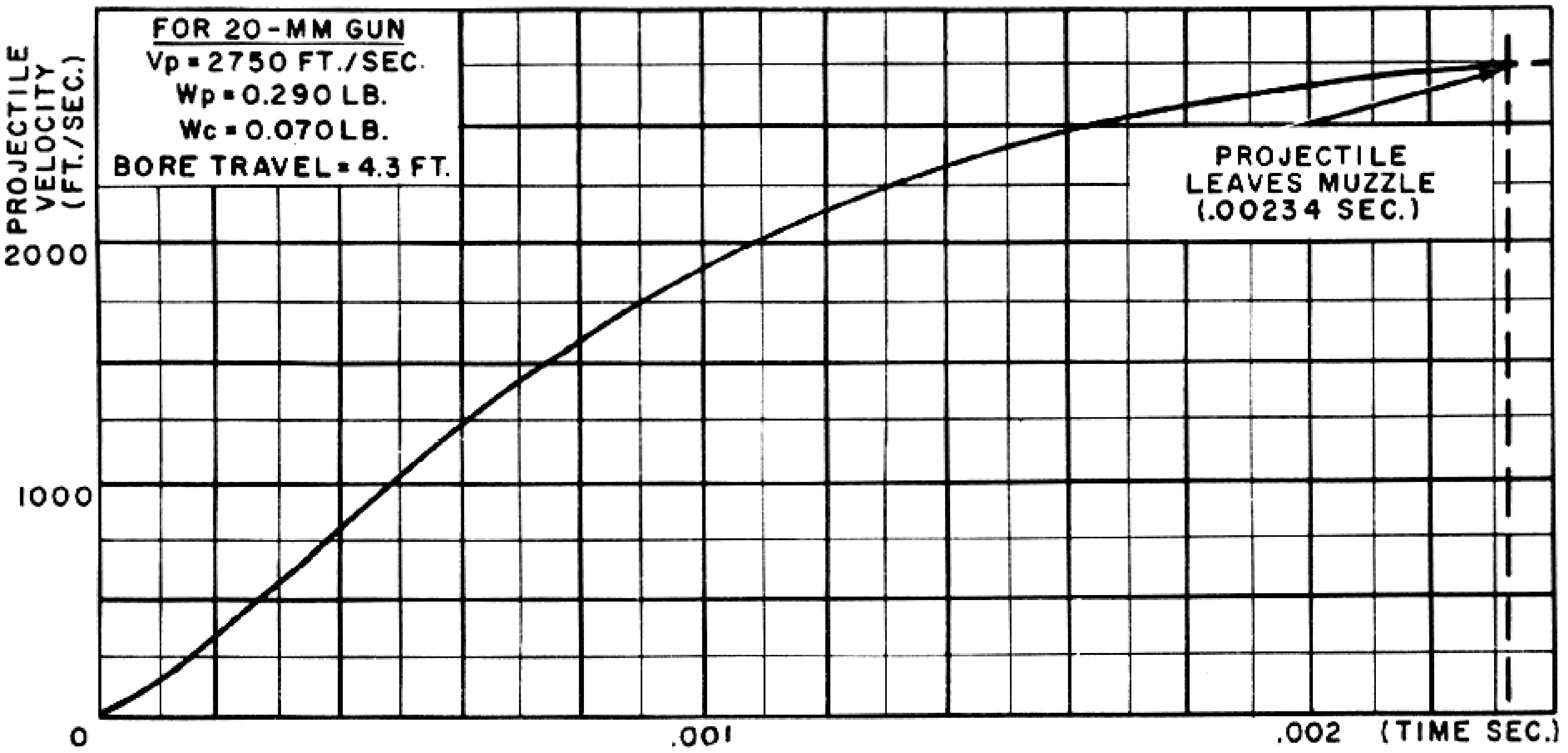


Figure 2–5. Graph of Projectile Velocity Versus Time (20 mm Gun).

and bore travel are available (figs. 2–4, 2–5, and 2–6).

NOTE: For some design problems, all or part of this information may not be available. Analytical methods by which the required data and graphs can be approximated for use in preliminary studies may be determined by conventional interior ballistics computations.

As has been explained, the primary factor affecting the performance of a long-recoil operated gun is the weight of the recoiling parts and in order to obtain a reasonable rate of fire, this weight must be kept to the bare minimum consistent with the requirements for strength, rigidity, and durability. The weight of these parts will be affected not only by strength considerations but also by the particular configuration of the mechanism selected by the designer. (For example, if the designer wishes to have the bolt slide in a long barrel extension, the recoiling parts might be heavier than they would be if the bolt moved on guide rails which were part of the receiver.) Accordingly, in order to determine the weight of the recoiling parts, the barrel must be designed first and the remainder of the mechanism must be planned at least to the extent which will make it possible to obtain a fair preliminary estimate of what weights will be involved. In the process of planning the mechanism it will also be necessary to determine what distances the parts must travel.

Of course, the final dimensions and weights of some of the recoiling parts can not be defined until complete consideration is given to the forces which act on these parts as the result of the accelerations and shocks to which they are subjected in operation and which yet remain to be determined. However, the estimated weight obtained from a carefully made preliminary design and layout should be accurate enough to serve as the starting point for the calculations necessary to determine the operating forces. It is these calculations which are the main concern of the following analysis.

As the analysis progresses, its application will be illustrated by means of sample calculations. Although these calculations and the related graphs are for a specific 20-mm cartridge and barrel and are based on an assumed weight of recoiling parts, the methods are generally applicable to long-recoil guns of any caliber. The calculations cover the following important points:

1. Determination of the conditions of free recoil.
2. Determination of the data necessary for designing barrel spring and bolt driving spring which will permit the recoil distance required for feeding.
3. Computation of the rate of fire.
4. Development of graphs showing how the velocity and travel of the recoiling parts vary with time.

Symbols Used in Analysis

- A Area of bore cross-section—in.^{2}
- B Ratio between average forces of barrel spring and bolt spring.
- C Arbitrary constant of integration (also ratio between masses of barrel and bolt).
- D Total recoil travel of bolt—ft.
- d Recoil travel of bolt for time t—ft.
- d Distance recoiled in first 0.010 second—ft.
- E_r Initial bolt energy—ft. lb.
- F_{av} Average combined force of barrel and bolt spring over distance D—lb.
- F_{av_1} Average force of barrel spring over distance D—lb.
- F_{av_2} Average force of bolt spring over distance D—lb.
- F_o Combined initial compression of barrel spring and bolt spring—lb.
- F_{o_1} Initial compression of barrel spring—lb.
- F_{o_2} Initial compression of bolt spring—lb.
- g Acceleration of gravity—32.2 ft./sec.^{2}
- K Combined spring rate of barrel and bolt springs—lb./ft.
- K_1 Spring rate of barrel spring—lb./ft.
- K_2 Spring rate of bolt spring—lb./ft.
- M_c Mass of powder charge—lb. sec.^{2}/ft.
- M_p Mass of projectile—lb. sec.^{2}/ft.
- M_r Total mass of recoiling parts—lb. sec.^{2}/ft.
- M_1 Mass of barrel—lb. sec.^{2}/ft.
- M_2 Mass of bolt—lb. sec.^{2}/ft.
- P Muzzle pressure—lb./in.^{2}
- T Time to recoil—sec.
- T′ Total cycle time—sec.
- T_1 Time for counter-recoil of barrel—sec.
- T_2 Time for return of bolt—sec.
- t Time—sec.
- t_r Approximate total return time for barrel and bolt—sec.
- T_{res} Time of duration of residual pressure—sec.
- V_p Muzzle velocity of projectile—ft./sec.
- v_p Velocity of projectile in bore at time t—ft./sec.
- v_r Velocity of retarded recoil at time t—ft./sec.
- V_{r_f} Maximum velocity of free recoil—ft./sec.
- v_{r_f} Velocity of free recoil at time t—ft./sec.
- W_c Weight of powder charge—lb.
- W_p Weight of projectile—lb.
- W_r Weight of recoiling parts—lb.

5. Computation of the power absorbed by the recoiling parts.

In the course of describing these calculations, the following fundamental formulas will be developed and explained:

a. Momentum and velocity relation for time projectile is in bore.
b. Formula for determining momentum and velocity of free recoil.
c. Expression for duration of residual pressure.
d. Formula for determining initial energy of the recoiling parts.
e. Formulas for determining spring retardations.
f. Energy equation for recoiling parts and springs.
g. Formula for determining time to recoil.
h. Expression for computing rate of fire.

1. *Condition of free recoil*

Under the heading "Principles of Recoil" it was pointed out that, if a gun is mounted so that it can move freely without friction or any other restraint, the impulse of the recoil force will impart to the gun a rearward momentum equal to the total forward momentum of the projectile and powder gases. For the time the projectile is in the bore, this momentum relationship is expressed by the equation:

$$M_r v_{r_f} = M_p v_p + M_c v_c \tag{2–1}$$

Since the powder gases will be thoroughly mixed by the turbulence created in the explosion it is reasonable to assume that the center of mass of the gases moves forward at one-half the velocity of the projectile. Actually, this is not quite accurate because the presence of the enlargement at the chamber and the fact that the rifling does not extend the full length of the space occupied by the gases creates a condition in which the volume of the space is not uniformly distributed along its length. Nevertheless, the assumption is close enough for present purposes. Therefore equation 2–1 may be rewritten as:

$$M_r v_{r_f} = M_p v_p + M_c \frac{v_p}{2} = \left(M_p + \frac{M_c}{2}\right) v_p \tag{2–2}$$

NOTE: It should be pointed out here that the momentum equality expressed by equation 2–2 is not affected by the internal frictional forces opposing the motion of the projectile and powder gases or by the forces incident to engraving the rifling band and to imparting the rotational velocity of the projectile. Although all of these forces retard the forward motion

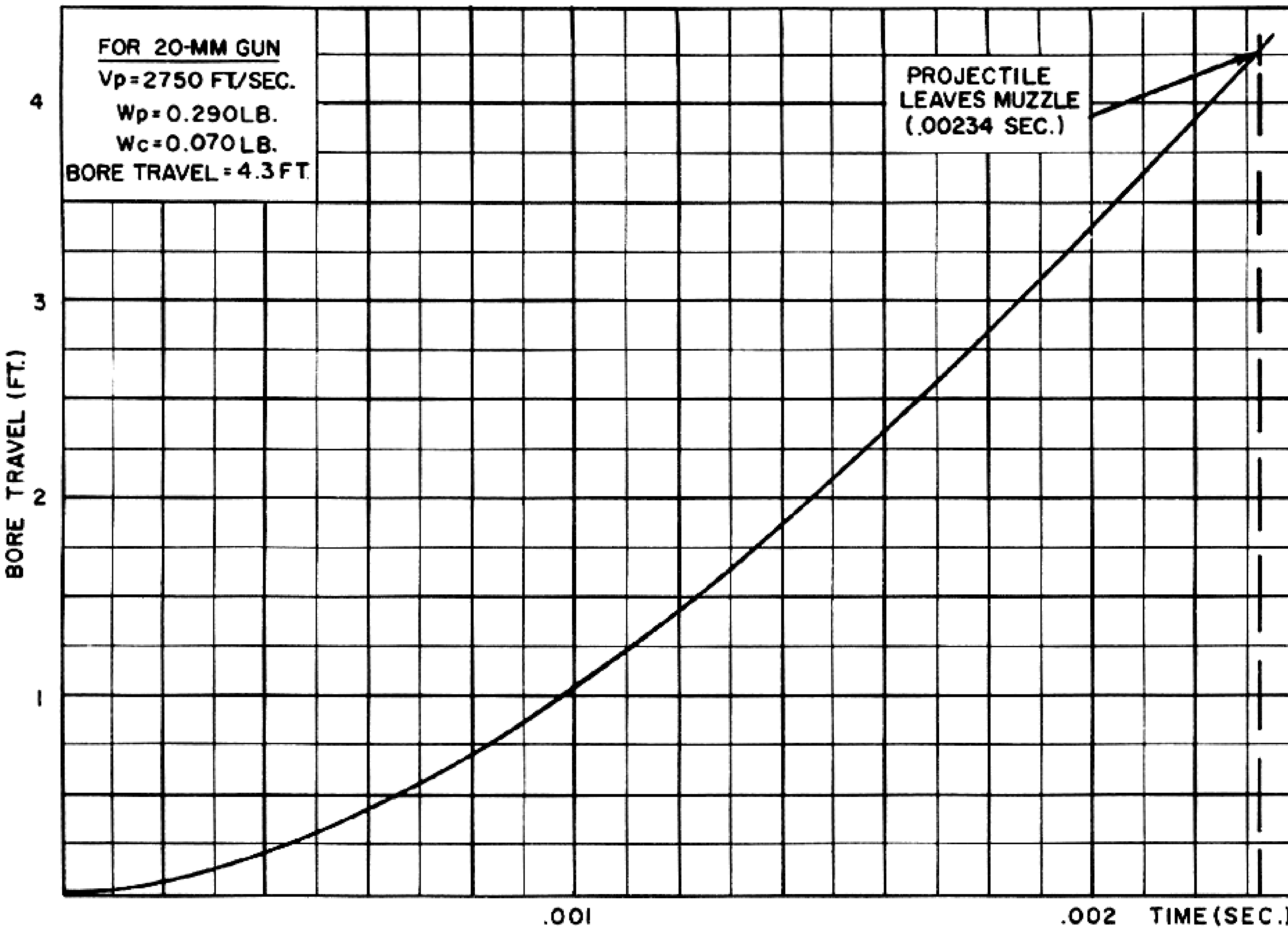


Figure 2–6. Graph of Projectile Bore Travel Versus Time (20 mm. Gun).

of the projectile and powder gases, they produce equal and opposite reactions on the barrel which result in a corresponding retardation of the rearward movement of the gun. In other words, the internal resistances merely decrease the effective impulse producing motion but they do not cause any inequality in the forward and rearward momentums.

Solving equation 2–2 for v_{r_f} gives the velocity of free recoil for the time the projectile is in the bore as:

$$v_{r_f} = \frac{M_p + \frac{M_c}{2}}{M_r} v_p = \frac{W_p + \frac{W_c}{2}}{W_r} v_p \tag{2-3}$$

Equation 2–3 can be used to plot a curve showing the free recoil velocity versus time for the period before the projectile leaves the muzzle. The weights of the projectile and powder charge are both known and it is assumed that the weight of the recoiling parts have been estimated in accordance with a preliminary design plan. Also, the velocity of the projectile at any time is known from the available ballistic data (fig. 2–5). Therefore, the ordinate of the free recoil velocity curve at any time, t, can be found by multiplying the corresponding ordinate of the projectile velocity curve by the factor:

$$\frac{W_p + \frac{W_c}{2}}{W_r}$$

Assuming that in the 20-mm gun to be used as an example the estimated weight of the recoiling parts is 50 pounds and the weights of the projectile and powder charge are as shown in fig. 2–5, the value of the multiplying factor is:

$$\frac{W_p + \frac{W_c}{2}}{W_r} = \frac{.29 + \frac{.070}{2}}{50} = .00650$$

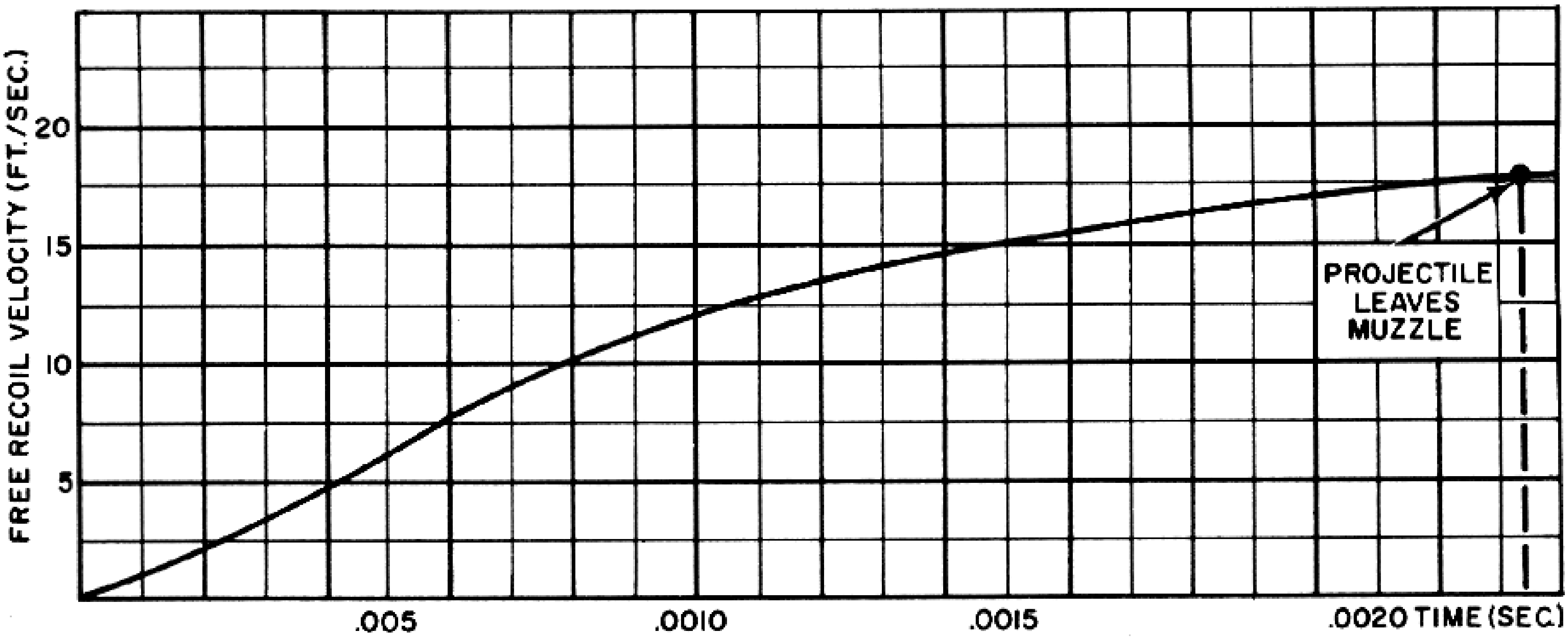


Figure 2–7. Free Recoil Velocity While Projectile Is in Bore.

Therefore, before the projectile leaves the muzzle, the free velocity of the recoiling parts is:

$$v_{r_f} = .00650\ v_p \left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The curve obtained by using this relation is shown in fig. 2–7 and is also shown in fig. 2–8 as the portion between t=0 and t=.00234 second. (In fig. 2–8, the time axis is compressed in order to show how the velocity varies after the projectile leaves the muzzle.)

The manner in which the free recoil velocity varies after the projectile leaves the muzzle can not be determined from equation 2–3 because the projectile is no longer part of the system. Since the effect of the residual pressure can not be expected in simple terms, a special method is used to extend the curve obtained from equation 2–3. This method is based on the fact that the results of experimental firings of various guns show that the maximum velocity of free recoil may be closely approximated as:

$$\text{(2–4)} \qquad V_{r_f} = \frac{W_p V_p + 4700\ W_c}{W_r}$$

This relationship is equivalent to saying that the maximum momentum imparted to the recoiling parts is equal to the sum of the muzzle momentum of the projectile and the momentum of the powder gases, assuming that the powder gases leave the gun at an average velocity of 4700 feet per second. For the gun used as an example:

$$V_{r_f} = \frac{.29 \times 2750 + 4700 \times .070}{50} = 22.5 \left(\frac{\text{ft.}}{\text{sec.}}\right)$$

A line representing this value of the maximum velocity of free recoil is drawn on the velocity graph (fig. 2–8) and the curve previously drawn from equation 2–3 is extrapolated until it becomes tangent to the line. The point at which the curve becomes tangent represents the time at which the residual pressure becomes zero and therefore imparts no further velocity to the recoiling parts. Although an error in locating the exact point of tangency will not have any serious effect on the accuracy of the results, it may be of some assistance in plotting to determine this point by using Vallier's formula for approximating the duration of the residual pressure:

$$\text{(2–5)} \qquad T_{res} = \frac{M_c}{AP}\ (9400 - V_p)$$

For the sample cartridge and barrel:

$$T_{res} = \frac{.070}{32.2 \times \frac{\pi}{4}\ (.790)^2 \times 5000}\ (9400 - 2750)$$

$$= .00592\ (\text{sec.})$$

To obtain the total time of action of the powder gases, this value is added to the time at which the projectile leaves the muzzle:

$$T_{res} = .00234 + .00592 = .00826\ (\text{sec.})$$

Extending the original curve until it is tangent at this point gives the complete free recoil velocity

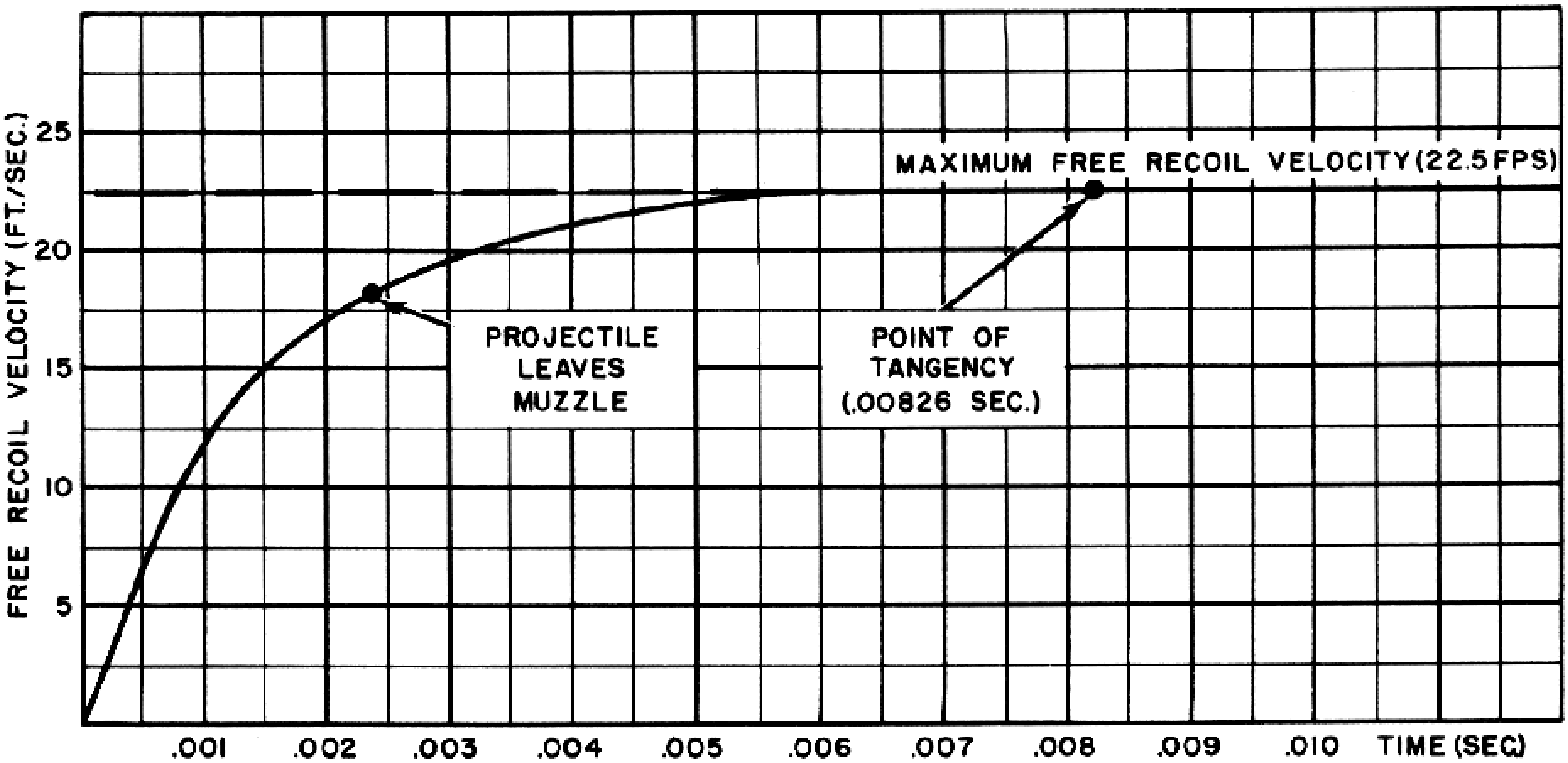


Figure 2–8. Free Recoil Velocity for First 0.010 Second.

curve shown in fig. 2–8. This curve will be used as the basis for the remainder of the calculations in this analysis.

2. *Determination of spring design data*

The calculations made for obtaining the free recoil velocity curve give the maximum velocity of free recoil as 22.5 feet per second. Since the total acceleration of the recoiling parts occurs in less than 0.010 second and since the retardation offered during this interval by the barrel spring and bolt driving spring will be very small, it may be assumed for purposes of considering the effect of the springs that an initial recoil velocity equal to the maximum velocity of free recoil is imparted instantaneously to the recoiling mass. On the basis of this assumption, the initial kinetic energy of the recoiling parts is given by the expression:

$$E_r=\frac{1}{2}M_rV_{r_f}{}^2=\frac{W_rV_{r_f}{}^2}{2g} \text{ (ft. lb.)}$$

Evaluating this expression for the conditions of the example gives the initial bolt energy as:

$$E_r=\frac{50\times 22.5^2}{2\times 32.2}=394 \text{ (ft. lb.)}$$

Since it has been assumed that this kinetic energy is instantaneously transferred to the recoiling parts, the springs must be proportioned to absorb this energy over the entire distance through which the barrel and bolt move in recoil. The energy absorbed by the springs is equal to the distance through which they are compressed times the average force required to produce this deflection. That is:

$$\text{(2–7)} \qquad E_r=F_{av}D \text{ or } F_{av}=\frac{E_r}{D}$$

If it is assumed that the bolt in the example must open 10.5 inches to permit feeding of a 20-mm cartridge (including a slight overtravel allowed to account for minor variations in the loading of the cartridges), the total recoil distance is 10.5 inches (0.875 feet). Therefore the average combined force of the barrel spring and bolt driving spring must be:

$$F_{av}=\frac{394}{.875}=450 \text{ (lb.)}$$

It should be noted here that the friction between the recoiling parts and the slide will produce an essentially constant retarding force. If it is expected that the force required to overcome friction will be considerable, this force should be determined and subtracted from the average spring force computed by using equation 2–7. Ordinarily, however, the friction force should be small when compared to the average spring force of 450 pounds and for purposes of the present analysis, the friction force will be neglected.

Having the average combined spring force, it remains to choose spring characteristics such that this average force will result from compressing both barrel return and bolt springs through the required recoil distance. The first problem that presents itself in this connection is how the spring resistance should be proportioned between the springs. One obvious choice would be to make the spring resistances proportional to the weights of the barrel and bolt respectively but if it is important to obtain the maximum possible rate of fire, the particular spring combination which would produce this condition should be determined before the selection is made.

The effect of the manner in which the springs are proportioned to each other will become apparent during the counter-recoil strokes of the barrel and bolt. During recoil both springs act together but during counter-recoil the barrel spring returns the barrel to battery and then the bolt driving spring returns the bolt. The strength of the springs will determine the time in which these parts are returned and by proper choice of the springs, the total return time can be held to a minimum. The following calculations show how the ratio B between the average forces exerted by the barrel spring and bolt driving spring is determined so that the total return time will be minimum.

At the end of the counter-recoil movement, the kinetic energy in either the barrel or bolt will be equal to the average force exerted by its spring times the counter-recoil distance.

$$E=\frac{1}{2}MV^2=F_{av}D$$

Solving for the terminal velocity V gives:

$$V=\sqrt{\frac{2F_{av}D}{M}}$$

The time required for counter-recoil will be approximately inversely proportional to this velocity:

$$t_r\alpha\sqrt{\frac{M}{2F_{av}D}}$$

Therefore, the total return time for the barrel and bolt may be expressed as

$$t_r\alpha\sqrt{\frac{M_1}{2F_{av_1}D}}+\sqrt{\frac{M_2}{2F_{av_2}D}}$$

(Note that for simplicity, this relation does not consider the effects of the overtravel previously mentioned.)

Now:

$$M_1=CM_2$$

$$F_{av_1}+F_{av_2}=F_{av} \text{ and,}$$

$$F_{av_1}=B\,F_{av_2}$$

Solving the last two equations simultaneously for F_{av_1} and F_{av_2} gives:

$$F_{av_1}=\frac{F_{av}B}{B+1} \text{ and,}$$

$$F_{av_2}=\frac{F_{av}}{B+1}$$

In the equation for the total return time, substituting CM_2 for M_1 and the above values for F_{av_1} and F_{av_2} yields:

$$t_r\alpha\sqrt{\frac{CM_2}{2\frac{F_{av}B}{B+1}D}}+\sqrt{\frac{M_2}{2\frac{F_{av}}{B+1}D}}$$

Simplifying:

$$t_r\alpha\left[\sqrt{\frac{C(B+1)}{B}}+\sqrt{B+1}\right]\sqrt{\frac{M_2}{2F_{av}D}}$$

The value of t_r will be minimum when the factor enclosed in brackets is minimum; that is when

$$\frac{d}{dB}\left[\sqrt{\frac{C(B+1)}{B}}+\sqrt{B+1}\right]=0$$

$$\frac{d}{dB}\left[\sqrt{B+1}\left(\sqrt{\frac{C}{B}}+1\right)\right]=0$$

Differentiating:

$$\sqrt{B+1}\left(\frac{-\frac{C}{B^2}}{2\sqrt{\frac{C}{B}}}\right)+\left(\sqrt{\frac{C}{B}}+1\right)\frac{1}{2\sqrt{B+1}}=0$$

Solving for B:

$$\frac{\sqrt{B+1}}{B}\sqrt{\frac{C}{B}}=\left(\sqrt{\frac{C}{B}}+1\right)\frac{1}{\sqrt{B+1}}$$

$$\frac{B+1}{B}\sqrt{\frac{C}{B}}=\sqrt{\frac{C}{B}}+1$$

$$\left(\frac{B+1}{B}-1\right)\sqrt{\frac{C}{B}}=1$$

$$\frac{1}{B}\sqrt{\frac{C}{B}}=1$$

$$B=\sqrt[3]{C}$$

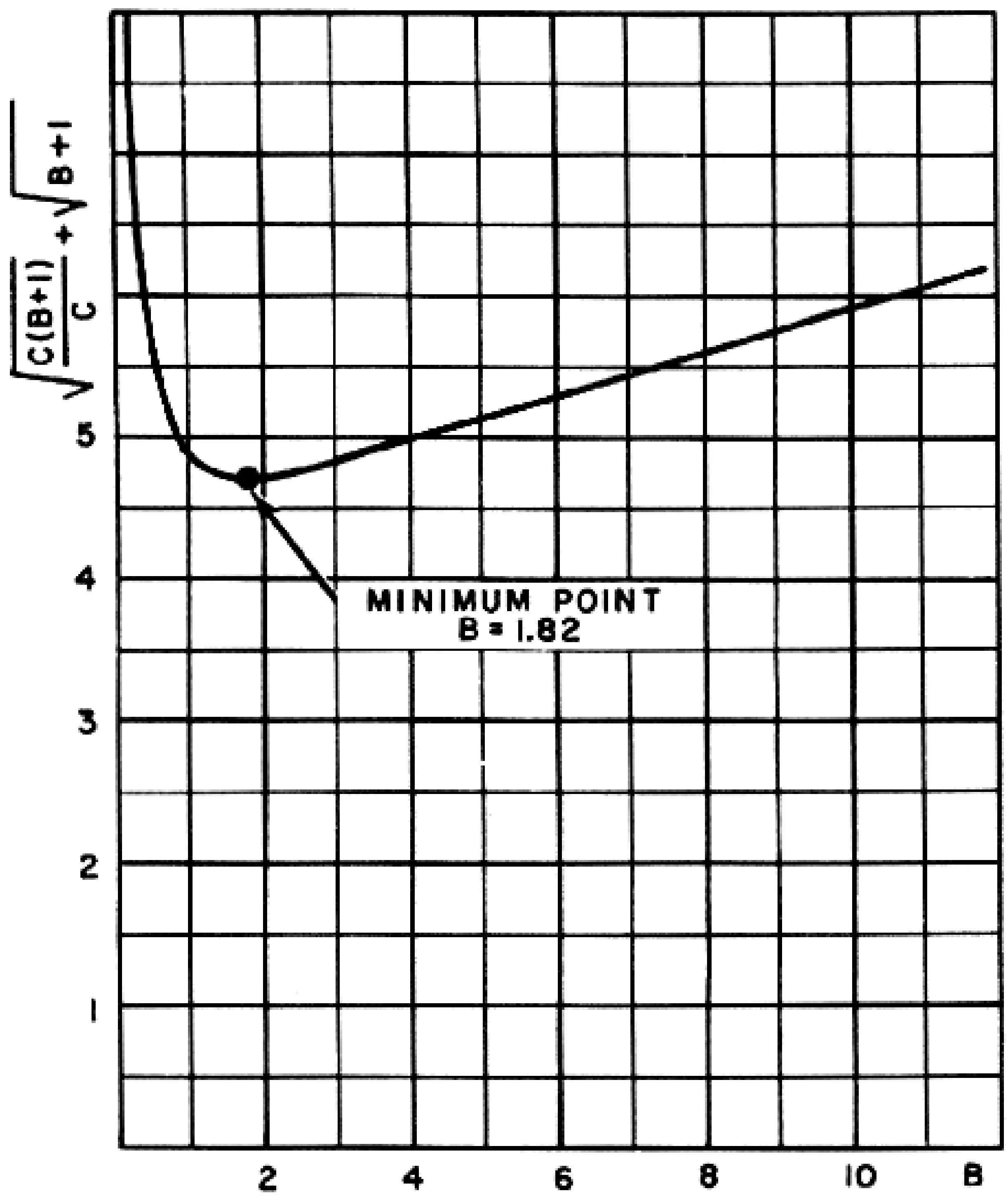


Figure 2–9. Graph Showing Values of Factor $\sqrt{\frac{C(B+1)}{C}} + \sqrt{B+1}$ Versus B When C=6.

Thus, it appears that the total counter-recoil time will be minimum if the barrel spring and bolt driving spring are proportioned to each other so that the ratio between their average forces is equal to the cube root of the ratio between the barrel weight and bolt weight. It will be assumed for purposes of the example that the barrel is six times heavier than the bolt, that is, C=6.

Fig. 2–9 shows the value of the factor

$$\left[\sqrt{B+1}\left(\sqrt{\frac{C}{B}}+1\right)\right]$$

for various values of B when C=6. Note that the curve reaches its minimum point at B=1.82 which is the cube root of 6. If this ratio between the spring forces were used in the example, the total required average force of 450 pounds breaks down so that the bolt spring would have an average force of 160 pounds and the average force of the barrel spring would be 290 pounds. It should be realized that the use of a bolt spring with an average force as high as 160 pounds would cause difficulty in charging the gun manually unless some mechanical aid is provided. For this reason, it may be desirable to increase the ratio B, thus sacrificing some speed of operation for ease in charging. However, if the gun is to be charged pneumatically or by some other means, the computed values are not unreasonable and will therefore be employed for the remainder of the analysis.

The next problem relating to the design of the springs is to choose values for the initial compression F_o and the spring constant K such that the required average force will be obtained when the springs are compressed through the desired recoil distance (10.5 inches in the example). As has been mentioned before, the practical problems involved in designing springs for machine guns require very careful analysis to take into consideration such factors as spring losses, forced vibrations set up along the length of the spring, shock loads, and other complications. If this is not done, serious operational difficulties or even failure of the spring may result. For this reason, the design of springs which are to be subjected to large and rapidly varying forces is a highly specialized art in the field of machine design and often a satisfactory spring for a machine gun can be found only by experimental means. Since it is beyond the scope of the present analysis to attempt a practical spring design at this point, an arbitrary selection of the spring characteristics will be made.

In making the selection of F_o and K for the springs of the gun used as an example, the value of F_o will be kept relatively small in order to gain an advantage in regard to rate of fire. (Cf. paragraph 5 in the mathematical analysis of the plain blowback system.) For the barrel spring, an initial compression of 115 pounds is reasonable and this choice requires that the maximum force be 465 pounds to produce the required average force of 290 pounds. Since the difference between the maximum force and initial compression is 350 pounds and the recoil distance is 10.5 inches (0.875 feet), the spring rate will be 400 pounds per foot or 33.3 pounds per inch. For the bolt spring, if F_o is selected as 55 pounds, the average force of 160 pounds will be obtained if the maximum force is 265 pounds. The difference between the maximum force is then 210 pounds. Since the recoil distance is 0.875 feet, the spring rate is 240 pounds per foot or 20 pounds per inch.

The foregoing determinations complete the basic preliminary design of the gun and the remaining task is to investigate what performance this design may be expected to give.

3. *Derivation of recoil equations*

Having the estimated weights of the recoiling parts and having the characteristics of the barrel spring and bolt spring, the performance of the gun can be determined by considering the energy relations which exist during recoil. These relations are expressed by a number of important equations which will be derived on the basis of the fact that as the recoiling parts move to the rear, the springs absorb and store the kinetic energy imparted by the propellant explosion. If it is assumed that the recoil energy is imparted instantaneously and that the losses due to friction and other causes are negligible, the energy remaining in the recoiling parts at any time during recoil is expressed by the equation:

(2-8)

$$\frac{M_r v_r^2}{2}=E_r-\int_0^d (F_o+Kd)\,dd=E_r-\left(F_o d+\frac{Kd^2}{2}\right)$$

NOTE: In this equation, F_o represents the combined initial compressions of the barrel spring (F_{01}) and of the bolt spring (F_{02}). Similarly K represents the combined spring rate of the barrel spring K_1 and of the bolt spring, K_2.

This equation may be used for deriving the equation expressing the relation between time and the motion of the recoiling parts as follows: Solving for v_r gives:

$$v_r=\sqrt{-\frac{K}{M_r}d^2-\frac{2F_o}{M_r}d+\frac{2E_r}{M_r}}=\frac{dd}{dt}$$

$$dt=\frac{dd}{\sqrt{-\frac{K}{M_r}d^2-\frac{2F_o}{M_r}d+\frac{2E_r}{M_r}}}$$

From a table of integrals, this expression is of the form:

$$\int\frac{dd}{\sqrt{ad^2+bd+c}}=\frac{1}{\sqrt{-a}}\sin^{-1}\frac{-2\,ad-b}{\sqrt{b^2-4\,ac}}+C$$

where:

$$a=-\frac{K}{M_r}\ (a<0)$$

$$b=-\frac{2F_o}{M_r}$$

$$c=+\frac{2E_r}{M_r}$$

Therefore:

$$t=\sqrt{\frac{M_r}{K}}\left[\sin^{-1}\frac{\frac{2K}{M_r}d+\frac{2F_o}{M_r}}{\sqrt{\frac{4F_o^2}{M_r^2}+\frac{8KE_r}{M_r^2}}}\right]+C$$

$$=\sqrt{\frac{M_r}{K}}\left[\sin^{-1}\frac{Kd+F_o}{F_o^2+2KE_r}\right]+C$$

But at the end of the recoil movement, the energy stored in the driving spring is equal to the initial bolt energy. That is:

$$E_r=F_oD+\frac{KD^2}{2}$$

Therefore:

$$F_o^2+2KE_r=F_o^2+2K\left(F_oD+\frac{KD^2}{2}\right)$$

$$=F_o^2+2KF_oD+K^2D^2$$

$$=(F_o+KD)^2$$

Substituting this value in the equation for t gives:

$$t=\sqrt{\frac{M_r}{K}}\sin^{-1}\frac{Kd+F_o}{KD+F_o}+C$$

To evaluate C: when t=0, d=0 and therefore

$$C=-\sqrt{\frac{M_r}{K}}\sin^{-1}\frac{F_o}{KD+F_o}$$

Substituting this expression for C gives the equation for the time, t, required to recoil any distance, d.

(2-9) $$t=\sqrt{\frac{M_r}{K}}\left[\sin^{-1}\frac{Kd+F_o}{KD+F_o}-\sin^{-1}\frac{F_o}{KD+F_o}\right]$$

Solving this equation for d gives the inverse relation expressing the distance recoiled in any time, t.

(2-10) $$d=\frac{KD+F_o}{K}\sin\left[\sqrt{\frac{K}{M_r}}\,t+\sin^{-1}\frac{F_o}{KD+F_o}\right]-\frac{F_o}{K}$$

Equation 2–9 may also be used to obtain the total time, T, required for the recoiling parts to move through the entire recoil distance, D. Substituting D for d gives:

$$(2\text{–}11)\qquad T=\sqrt{\frac{M_r}{K}}\left[\sin^{-1}1-\sin^{-1}\frac{F_o}{KD+F_o}\right]$$

$$T=\sqrt{\frac{M_r}{K}}\left[\frac{\pi}{2}-\sin^{-1}\frac{F_o}{KD+F_o}\right]$$

$$=\sqrt{\frac{M_r}{K}}\left(\cos^{-1}\frac{F_o}{KD+F_o}\right)$$

or:

$$T=\sqrt{\frac{M_{r_1}+M_{r_2}}{K_1+K_2}}\left[\cos^{-1}\frac{F_{o_1}+F_{o_2}}{(K_1+K_2)D+F_{o_1}+F_{o_2}}\right]$$

If it is assumed that the losses are negligible, the times required for the barrel and bolt to return to the firing position may be determined by the use of equation 2–11 providing that the proper values are substituted for M_r, K, F_o, and D. Making these substitutions and adding the times gives the time for a complete cycle as:

$$(2\text{–}12)\qquad T'=\sqrt{\frac{M_{r_1}+M_{r_2}}{K_1+K_2}}\left[\cos^{-1}\frac{F_{o_1}+F_{o_2}}{(K_1+K_2)D+F_{o_1}+F_{o_2}}\right]+$$

$$\sqrt{\frac{M_{r_1}}{K_1}}\left[\cos^{-1}\frac{F_{o_1}}{K_1D+F_{o_1}}\right]+$$

$$\sqrt{\frac{M_{r_2}}{K_2}}\left[\cos^{-1}\frac{F_{o_2}}{K_2D-F_{o_2}}\right]$$

Note that in the last two terms of equation 2–12, no consideration is given to the overtravel which was allowed in the design. (See fig. 2–3.) The resulting error is so slight that there is no need to complicate the equation further by taking it into account.

Since T′ is the total time required for one complete cycle of operation, the rate of fire in rounds per minute will be:

$$(2\text{–}13)\qquad N=\frac{60}{T'}$$

For the conditions of the example, N is evaluated as follows:

The total weight of the recoiling parts is 50 pounds. The effective weight of the barrel and its related parts will be taken as 40 pounds and the effective weight of the bolt and its related parts will be taken as 10 pounds. Therefore:

$$T'=\sqrt{\frac{40+10}{32.2(400+240)}}\left(\cos^{-1}\frac{115+55}{(400+240).875+114+55}\right)+\sqrt{\frac{40}{32.2\times400}}\left(\cos^{-1}\frac{115}{400\times.875+115}\right)+$$

$$\sqrt{\frac{10}{32.2\times240}}\left(\cos^{-1}\frac{55}{240\times.875+55}\right)$$

$$=.0660+.0737+.0487$$

$$=0.1884\ \text{(second)}$$

The rate of fire is then:

$$N=\frac{60}{.1884}=319\ \text{(rounds per minute)}$$

(This rate of fire is based on the use of springs which absorb all of the recoil energy. As pointed out previously, some improvement in the rate might be gained by using lighter springs and permitting the parts to rebound from buffers at the end of recoil but since such a gain would not be very great, no consideration will be given to this method here.)

With the rate of fire and recoil energy known, the horsepower absorbed by the recoiling parts can be computed by means of the formula:

$$(2\text{–}14)\qquad HP=\frac{E_rN}{33{,}000}$$

The horsepower absorbed by the recoiling parts in the gun of the example will be:

$$HP=\frac{394\times319}{33{,}000}=3.81$$

4. *Development of theoretical time-travel and time-velocity curves.*

In the design of the various details of the breech mechanism and of the other gun mechanisms such as the feeder and firing device, it is necessary to have information relating to the motion of the recoiling parts during the progress of the cycle of operation. This information can be presented in convenient form by means of theoretical curves which show the relation between time, the travel of the recoiling parts, and the velocity of the recoiling parts. Such curves may be drawn by using the formulas developed under the preceding heading, but it should be realized that all of these formulas are based on the assumption that the initial energy was transferred instantaneously to the recoiling parts. Therefore, curves plotted in this way would not take account of the detailed effects resulting from the conditions which exist during the time of action of the powder gas pressures. This time is so short that it will be relatively negligible when the overall cycle of operation is considered, particularly if the rate of fire is low. However, for higher rates of fire, the time of action of the powder gas pressure may represent a small but significant portion of the time required for recoil and accordingly should be given due consideration in plotting bolt motion curves. In addition, because of the high accelerations which occur during the propellant explosion, it is highly desirable to determine in detail what motion characteristics may be expected in the initial portion of the recoil stroke.

The effects of the powder gas pressure can not be expressed by simple equations and therefore a special method is employed to account for these effects in plotting the bolt motion curves. The method consists essentially of first plotting a curve of free recoil velocity and then subtracting from each ordinate of this curve the velocity loss resulting from the retarding effects of the springs.

The curves showing the velocity of free recoil versus time were developed previously and are shown in figs. 2–7 and 2–8. These curves will be used to illustrate the following description of the method.

To determine the retarding effects of the spring, use is made of the law expressed by the equation:

(2–15) $$F\,dt = M\,dv$$

This law states that the change in the momentum of a mass is equal to the applied impulse (the product of the force and the time for which it is applied). Solving for dv gives:

$$dv = \frac{F\,dt}{M}$$

To obtain the variation of the change in velocity with respect to time, this expression is integrated.

(2–16) $$v = \int_0^t \frac{F\,dt}{M} = \frac{1}{M}\int_0^t F\,dt$$

In accordance with equation 2–16, the retarding effect of a force on a given mass can be determined as follows:

1. Plot a curve showing the variation of the force with respect to time.
2. Measure the area under the curve between $t=0$ and some time t_1.
3. Divide the measured area by the mass. This gives the ordinate of the retardation curve for the time t_1.
4. Repeat steps 2 and 3 for other values of t and plot the retardation curve.

Applying this procedure using the mass of the recoiling parts and the combined resistance of the barrel spring and bolt spring produces a curve showing the loss in recoil velocity resulting from the action of the springs. Since the free recoil velocity curve shows the gain in velocity resulting from the thrust of the powder gases, the difference between the curves will be the net recoil velocity, or in other words the velocity of retarded recoil.

The foregoing method would be very simple if the retarding force were constant or if the variation of this force with respect to time were known. However, when the force varies with recoil travel as it does with the springs assumed for purposes of analysis, a difficulty is encountered. In order to plot a graph showing the variation of the retarding force with respect to time, it is necessary to have a curve showing the variation of the recoil travel with respect to time, and the latter curve is one of those which yet remain to be determined.

To overcome this difficulty, the problem is considered in two stages. For the first 0.010 second while the powder gas pressures are acting, the loss in velocity resulting from the retarding effect of the springs will be relatively small and will be almost entirely due to the constant effect of the initial compression. The varying force due to the spring constant during this interval of time will almost certainly be negligible but, if necessary, it can be approximated very closely. In this way, accurate results can be obtained for the first 0.010 second and for the remainder of the cycle of operation, while the powder gas pressures are not acting, the recoil travel can be determined analytically without any trouble.

The procedure for plotting the velocity and travel curves during the first 0.010 second is as follows:

1. Plot curve of free recoil velocity versus time (fig. 2–7).

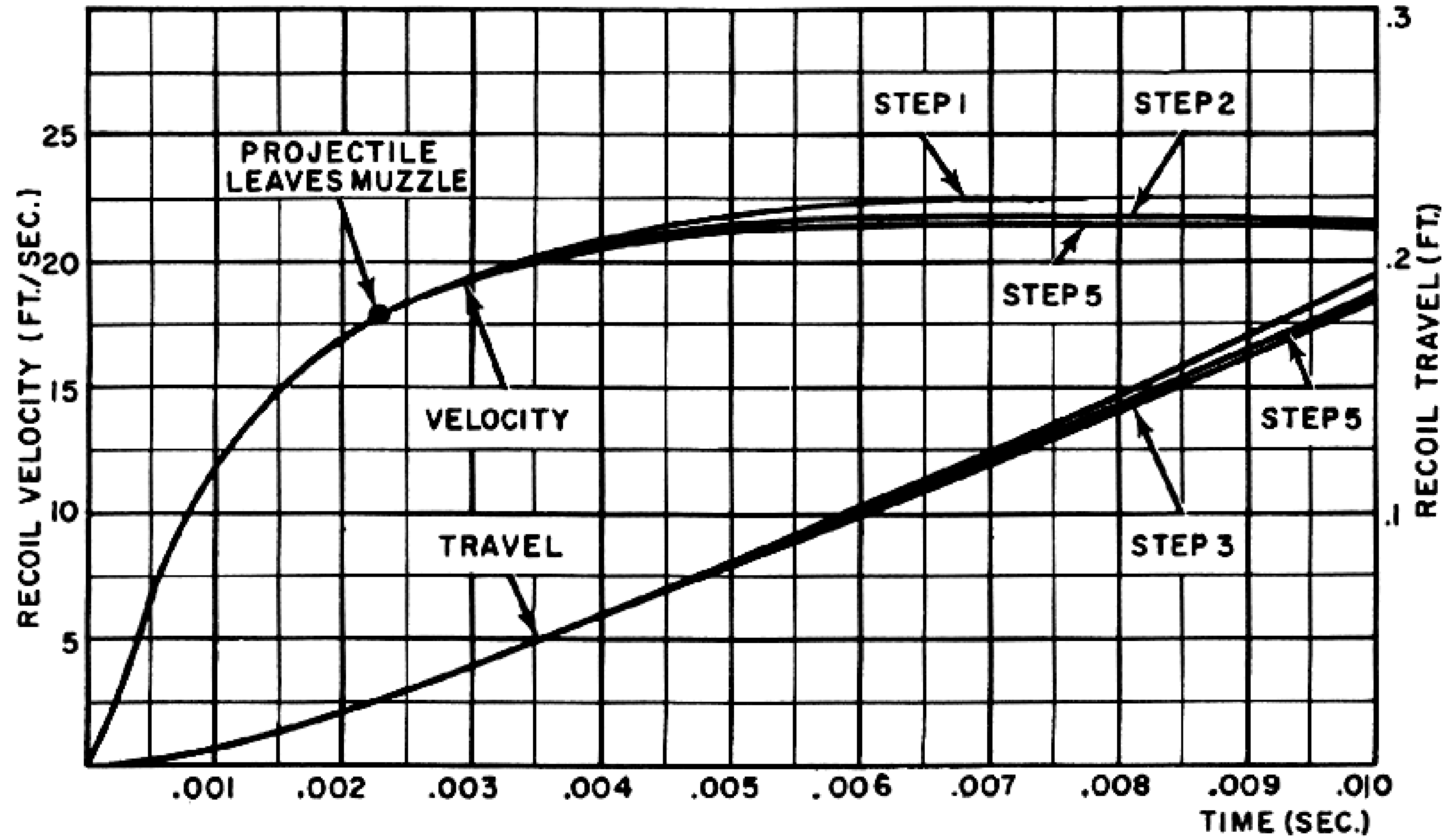


Figure 2–10. Development of Time-Travel Curves for First 0.010 Second.

2. The loss in velocity due to the initial compression of the springs is equal to:

$$\frac{(F_{o_1}+F_{o_2})\,t}{M_r}$$

Determine the velocity loss for various values of t, substract each from the corresponding ordinate of the free recoil velocity curve and draw a curve through the resulting points. If the effect of the spring constant proves to be negligible, this curve is the retarded velocity curve.
3. Integrate under the curve drawn in step 2 to obtain the displacement curve.
4. Assume that the curve drawn in step 3 represents the actual time-travel curve and use this curve to determine the retardation due to the spring constant. (Use the combined spring constant for the barrel spring and bolt spring, K_1+K_2.) Ordinarily, it will be found that this retardation is so small that it will not have any effect worthy of consideration.
5. In the event that the retardation determined in step 4 is sufficient to affect the velocity, use it to modify the curve drawn in step 2 and then integrate under the new curve to obtain a corrected displacement curve.
6. Steps 4 and 5 can be repeated as often as is necessary until no significant change occurs in the displacement curve. Actually, this process of successive approximation should never be necessary and satisfactory results should be achieved in the first three steps or at least in the first five steps.

Fig. 2–10 shows the curves obtained for the gun of the example. The total loss in velocity due to the combined effect of the initial compressions of the springs during the first 0.010 second is:

$$V=\frac{(F_{o_1}+F_{o_2})\,t}{M_r}=\frac{(115+55).01\times32.2}{50}$$

$$=1.09\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the combined effect of the spring constants as determined by the method of step 4 is only about 0.345 foot per second. The final curves shown in fig. 2–10 are the result of performing step 5. Since the velocity loss due to the effect of the spring constant is so small, step 6 need not be taken.

The remainder of the displacement curve for the recoil stroke can now be determined analytically by using equation 2–10:

$$d=\frac{KD+F_o}{K}\sin\left[\sqrt{\frac{K}{M_r}}\,t+\sin^{-1}\frac{F_o}{KD+F_o}\right]-\frac{F_o}{K}$$

However, since some recoil travel (d') occurred during the first 0.010 second, the values of F_o, D, and t must be modified to take this motion into account and d' must be added to the resulting values obtained for d. Also note that the values for K, F_o, D, and M_r must be selected to suit the particular portion of the operating cycle under consideration. During recoil, both springs act on the total mass of the recoiling parts but in counter-recoil the barrel is returned to battery first while the bolt remains latched and then the bolt is returned. Thus it is necessary to consider the motions in three phases: recoil motion, barrel counter-recoil, and bolt return.

To account for the recoil movement during the time of action of the powder gases, the following changed values are used in equation 2–10:

$$F_o'=F_o+Kd' \text{ or } F_o'=F_{o_1}+F_{o_2}+(K_1+K_2)d'$$

$$D'=D-d'$$

$$t'=t-.010$$

Also since the barrel and bolt and their springs act as a unit during the recoil stroke, the following substitutions are also made in equation 2–10:

$$F_o=F_{o_1}+F_{o_2}$$

$$K=K_1+K_2$$

$$M_r=M_{r_1}+M_{r_2}$$

Making these substitutions gives the modified form of equation 2–10 as it applies to the recoil stroke as:

$$d=\frac{(K_1+K_2)(D-d')+F_{o_1}+F_{o_2}+(K_1+K_2)d'}{K_1+K_2}\sin\left[\sqrt{\frac{K_1+K^2}{M_{r_1}+M_{r_2}}}\,(t-.010)+\sin^{-1}\frac{F_{o_1}+F_{o_2}+(K_1+K_2)d'}{(K_1+K_2)(D-d')+F_{o_1}+F_{o_2}+(K_1+K_2)d'}\right]-\frac{F_{o_1}+F_{o_2}+(K_1+K_2)d'}{K_1+K_2}+d'$$

Simplifying:

$$d=\frac{(K_1+K_2)D+F_{o_1}+F_{o_2}}{K_1+K_2}\sin\left[\sqrt{\frac{K_1+K_2}{M_{r_1}+M_{r_2}}}\,(t-.010)+\sin^{-1}\frac{F_{o_1}+F_{o_2}+(K_1+K_2)d'}{(K_1+K_2)D+F_{o_1}+F_{o_2}}\right]-\frac{F_{o_1}+F_{o_2}+(K_1+K_2)d'}{K_1+K_2}+d'$$

This equation is employed to plot the displacement curve from the time $t=0.010$ until the bolt is latched at the rear. Since the design used as example allows for only a very slight overtravel, this overtravel will be neglected and it will be considered that the bolt is latched when the counter-recoil movement is equal to D (0.875 foot).

After the bolt is latched and the bolt lock is opened, the barrel continues its forward movement in counter-recoil and is now driven by the barrel spring alone. The movement of the barrel during this time can be determined by using equation 2–10 in a special way. Since the barrel spring only is acting, the value K_1 is substituted for K, F_{o_1} is substituted for F_o, and M_{r_1} is substituted for M_r. In the equation, the time used for determining the barrel counter-recoil curve must be equal to $t+(T_1-T)$; where T is the time to recoil expressed by equation 11, and T_1 is the time required for the barrel alone to complete its counter-recoil movement. This substitution is necessary because the period of the sine curve expressing the barrel counter-recoil movement is different from that of the curve expressing the recoil movement of the combined mass of the barrel and bolt. Making the necessary substitutions gives the modified form of equation 2–10 as it applies to the barrel counter-recoil movement as:

$$d=\frac{K_1D+F_{o_1}}{K_1}\sin\left[\sqrt{\frac{K_1}{M_{r_1}}}\,(t+T_1-T)+\sin^{-1}\frac{F_{o_1}}{K_1D+F_{o_1}}\right]-\frac{F_{o_1}}{K_1}$$

This equation is applicable from the time $t=T$ to the time $t=T+T_1$ at which time the curve reaches the zero axis.

For purposes of analysis, it will be assumed that the bolt is unlatched just as the barrel reaches the battery position at the time $t=T+T_1$. The movement of the bolt after its release can be determined by using equation 2–10 in the same general way as for the barrel. In this case, since the bolt is driven by the bolt spring alone, the value K_2 is substituted for K, F_{o_2} is substituted for F_o, and M_{r_2} is substi-

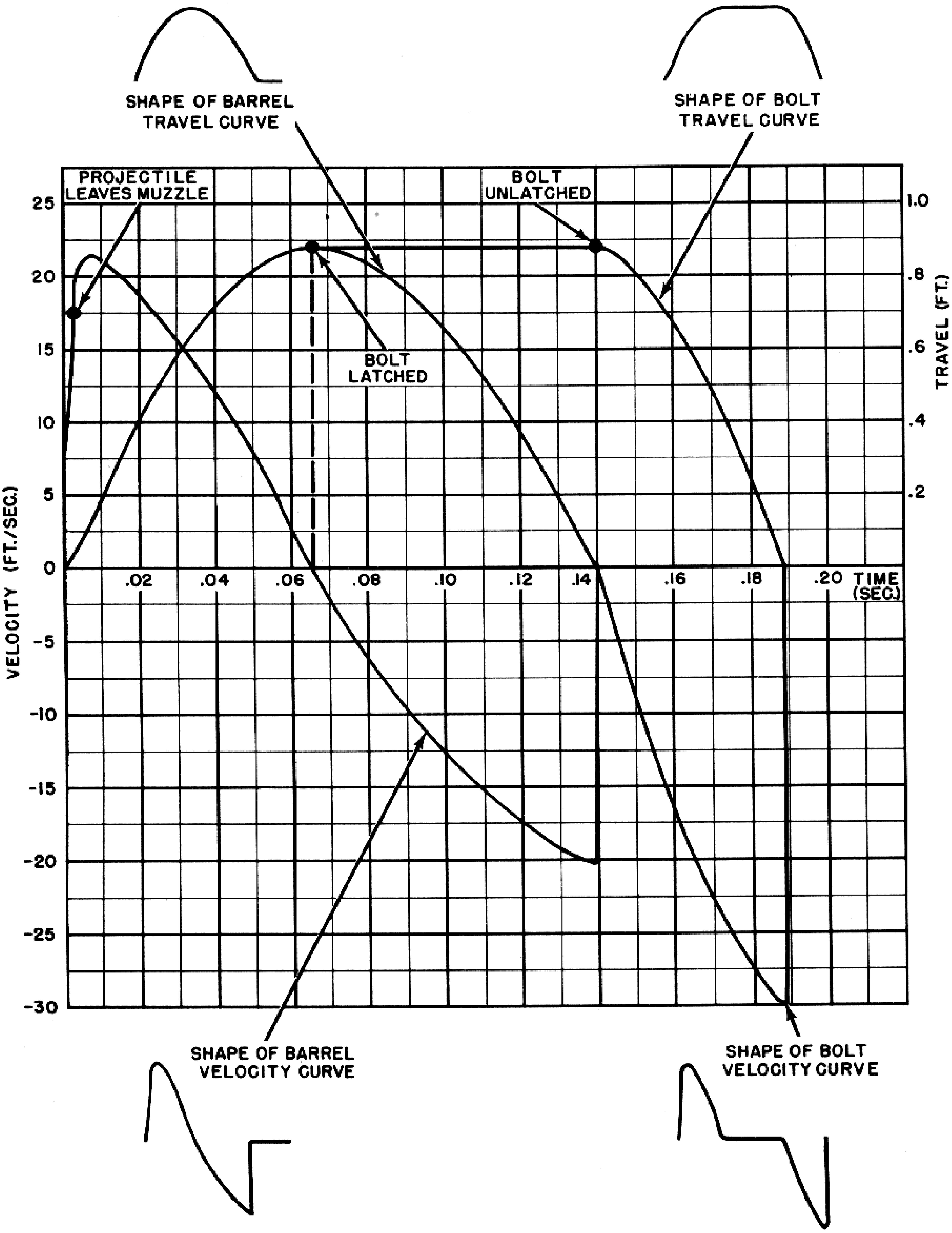


Figure 2–11. Time-Travel and Time-Velocity Curves (Complete Cycle).

tuted for M_r. The time used in the equation must be equal to $t+T_2-(T+T_1)$; where T_2 is the time required for the bolt alone to complete its return movement. This substitution is necessary for the same reason as explained in the preceding paragraph. Making the indicated substitutions gives the modified form of equation 2–10 as it applies to the bolt return movement as:

$$d=\frac{K_2+F_{o_2}}{K_2}\sin\left[\sqrt{\frac{K_2}{M_{r_2}}}(t+T_2-T-T_1)+\sin^{-1}\frac{F_{o_2}}{K_2D+F_{o_2}}\right]-\frac{F_{o_2}}{K_2}$$

This equation is applicable from the time $t=T+T_1$ to the time $t=T+T_1+T_2$, at which time the curve reaches the zero axis.

The three equations described in the preceding paragraph are used to complete the curves showing the recoil and counter-recoil movements of the barrel and bolt. The resulting curves can be used to determine curves showing the variation with time of the forces exerted by the springs during the various parts of the cycle of operation. To obtain the spring force curves for the recoil stroke, the ordinates of the corresponding portion of the displacement curve are multiplied by the factor K_1+K_2 and are increased by $F_{o_1}+F_{o_2}$. For the barrel counter-recoil movement, the spring force curve is determined by multiplying each ordinate of the corresponding portion of the displacement curve by K_1 and increasing the result by F_{o_1}. The spring force curve for the bolt return stroke is determined in the same way using K_2 and F_{o_2}. Integrating under the first portion of the curve and dividing by $M_{r_1}+M_{r_2}$ in accordance with equation 2–16 gives a graph showing the loss in recoil velocity resulting from the combined force of the barrel and bolt springs. Subtracting this curve from the free bolt velocity curve produces the graph showing the retarded recoil velocity. The curve showing the

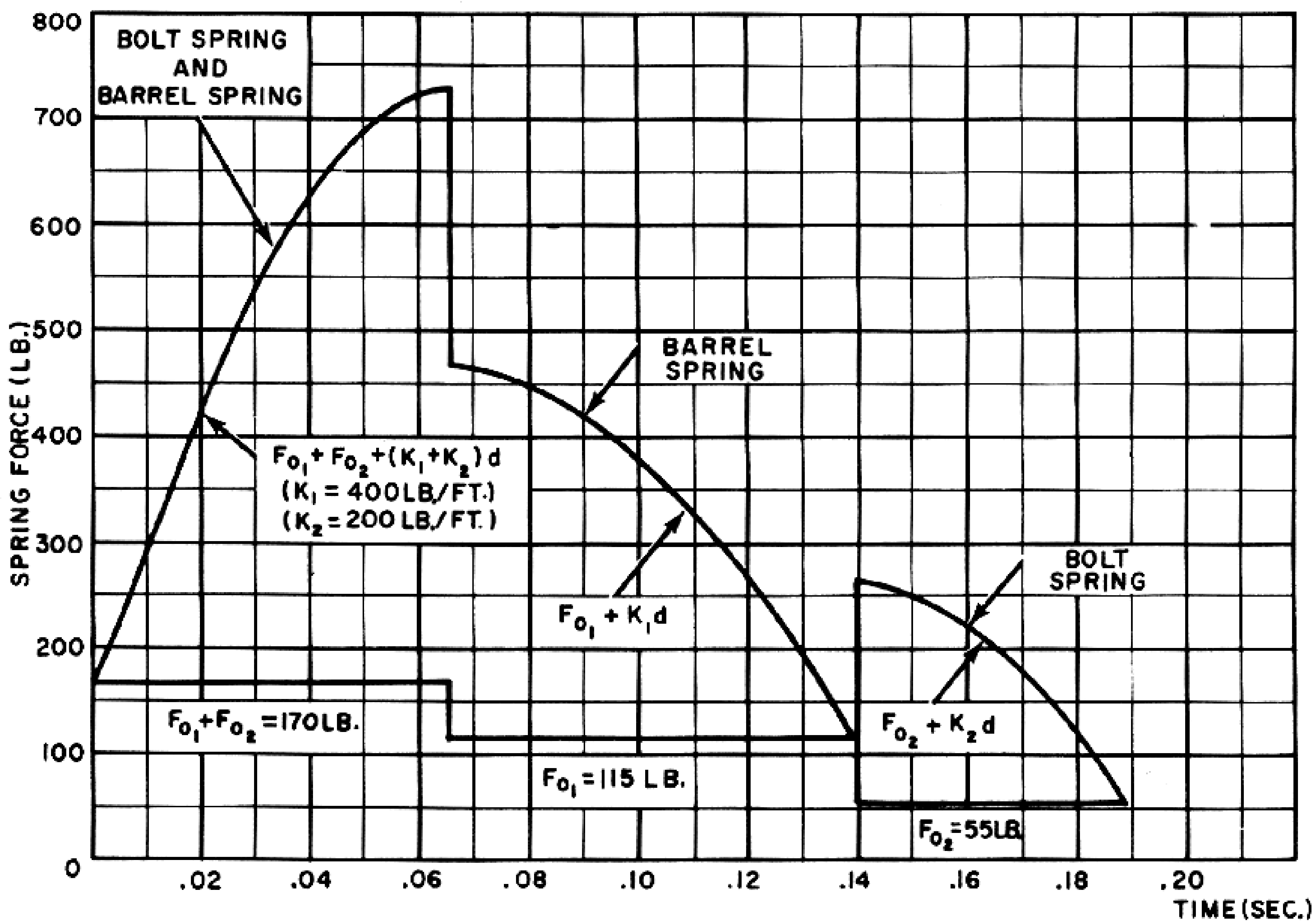


Figure 2–12. Variation of Spring Forces With Time.

counter-recoil velocity of the barrel is obtained by integrating under the second portion of the spring force curve and dividing by M_{r_1}. (The resulting ordinates are plotted as negative values because the barrel is now moving forward.) A similar method is used to plot the curve showing the velocity at which the bolt returns. The third part of the spring force curve is used and the integrals are divided by M_{r_2}.

Fig. 2–11 shows the displacement and velocity curves obtained by the method described in the preceding paragraphs, using the data for the gun of the example. After the necessary substitutions are made, the final forms of the three equations used after the first 0.010 second are:

(t from 0.010 to T)

$$d = 1.14 \sin [20.3t + .204] - .265$$

(t from T to $T + T_1$)

$$d = 1.162 \sin [17.93t + .393] - .287$$

(t from $T + T_1$ to $T + T_1 + T_2$)

$$d = 1.103 \sin [27.8t - 2.31] - .229$$

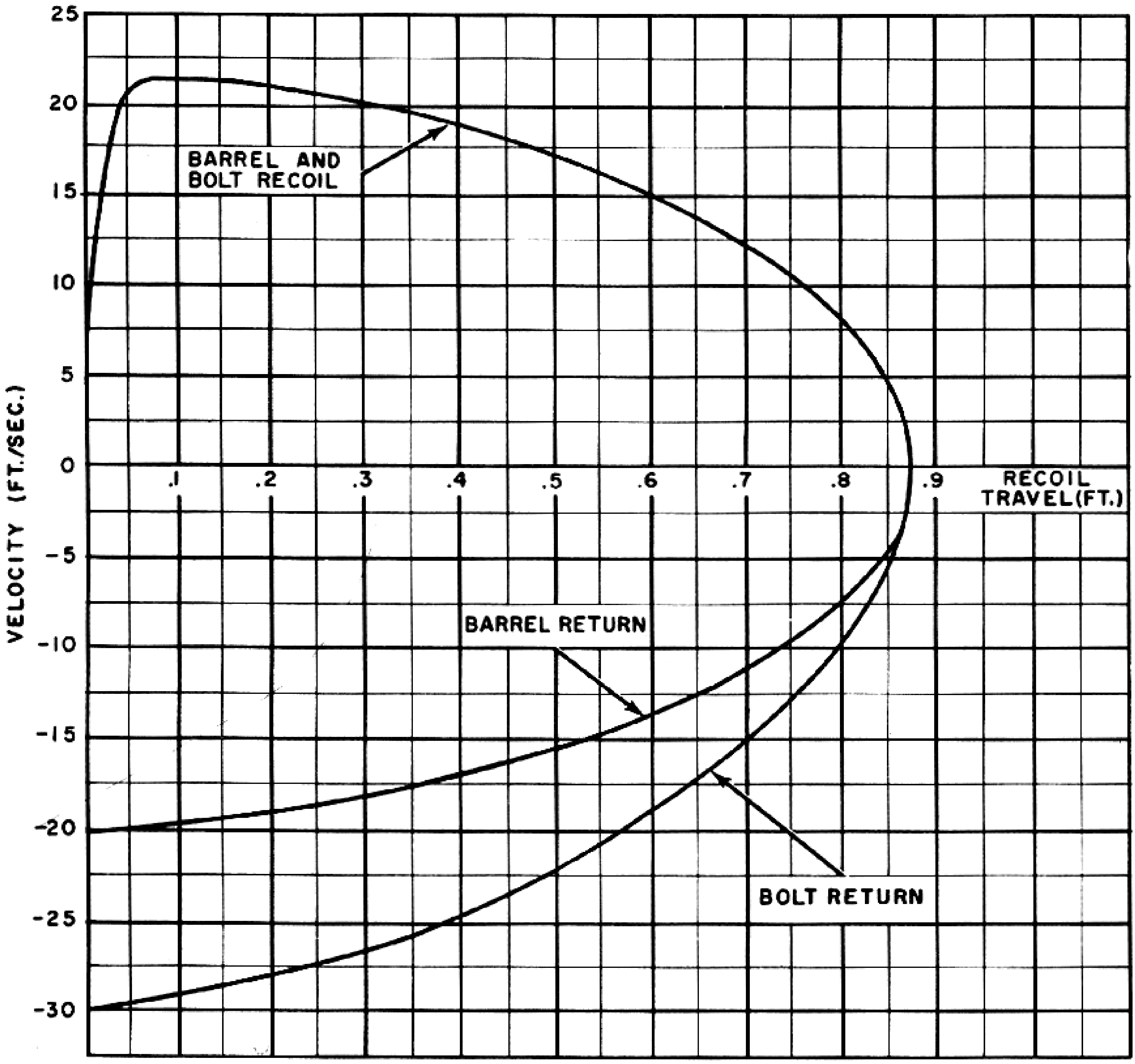


Figure 2–13. Velocity-Displacement Curve.

The spring force curve obtained from the displacement curve of fig. 2–11 is shown in fig. 2–12.

In an actual design problem, it may be desirable to consider the effects produced on the displacement and velocity curves by forces resulting from friction and from operation of the gun mechanism. These forces are treated by methods similar to those employed for the spring forces. For example, the friction force resisting the recoil movement will be essentially constant and therefore can be taken into account by increasing F_0 in equation 2–10. If the force under consideration is a constant or varying load which exists for only a small portion of the operating cycle (such as the force required to strip a cartridge out of the feeder), it can be treated in a similar manner, providing the problem is considered in stages by methods like those described in the preceding paragraphs.

Another useful type of curve for design and analysis purposes may be obtained by plotting the velocities involved in each portion of the cycle against the corresponding values for the displacement. These curves can be drawn easily because the displacement and velocity curves shown in fig. 2–11 can be used to obtain the velocity corresponding to any displacement. Fig. 2–13 shows the velocity versus displacement curves for the barrel and bolt of the gun of the example.

SHORT RECOIL SYSTEM

In the short recoil system of operation, the barrel and bolt remain locked and recoil together for a short distance until the powder gas pressure has dropped to a safe limit. The recoil movement is then utilized to unlock the bolt and after unlocking, the barrel is stopped while the bolt continues to move to the rear until the opening between the barrel and bolt is sufficient to permit feeding. It would be possible for the bolt to complete this movement merely by virtue of the momentum it possesses at the instant of unlocking, but in all short-recoil weapons, in order to speed up operation, the bolt is given additional momentum by means of an accelerating device which transfers energy to the bolt from the barrel during the short time that the barrel is still moving to the rear after unlocking. Also, unlocking usually occurs before the residual pressure reaches zero and therefore the bolt receives an additional impulse from blowback action.

The essential elements of a gun which operates by the short recoil system are shown schematically in fig. 2–14A. These elements consist of the bolt, an arrangement for locking the bolt to the barrel and for unlocking it, an accelerating device, a barrel stop, a backplate buffer, and springs for returning the barrel and bolt after recoil. In the mechanism illustrated, the barrel is latched in the position at which it is stopped after unlocking. The returning bolt unlatches the barrel before the bolt locks to the barrel extension and then the barrel and bolt return to battery together. In other weapons, the barrel may be returned to battery independently before the bolt returns. The other portions of fig. 2–14 show different stages during the cycle of operation.

Cycle of Operation

The operating cycle of a typical short recoil gun occurs as follows:

The cycle starts with a cartridge in the chamber and with the bolt locked to the barrel (fig. 2–14A). When the cartridge is fired, the pressure of the powder gases drives the projectile and gases forward through the bore and at the same time drives the barrel and locked bolt to the rear in recoil. During the action of the powder gas pressure, the retardation offered by the combined action of the bolt driving spring and barrel spring is relatively small so that the only really significant factor in limiting the recoil acceleration is the mass of the recoiling parts.

The large forces exerted by the peak pressure of the powder gases exist for a relatively short time. In a typical 20-mm gun with a barrel length of about five feet the projectile leaves the muzzle 0.008 or 0.009 second after ignition of the primer. However, 0.001 or 0.002 second after the projectile leaves the muzzle, the residual pressure has dropped to a safe limit and the bolt may be unlocked in order to take advantage of the blowback action produced by the residual pressure. Therefore, at this instant, the unlocking device is actuated to free the bolt from the barrel (fig. 2–14B).

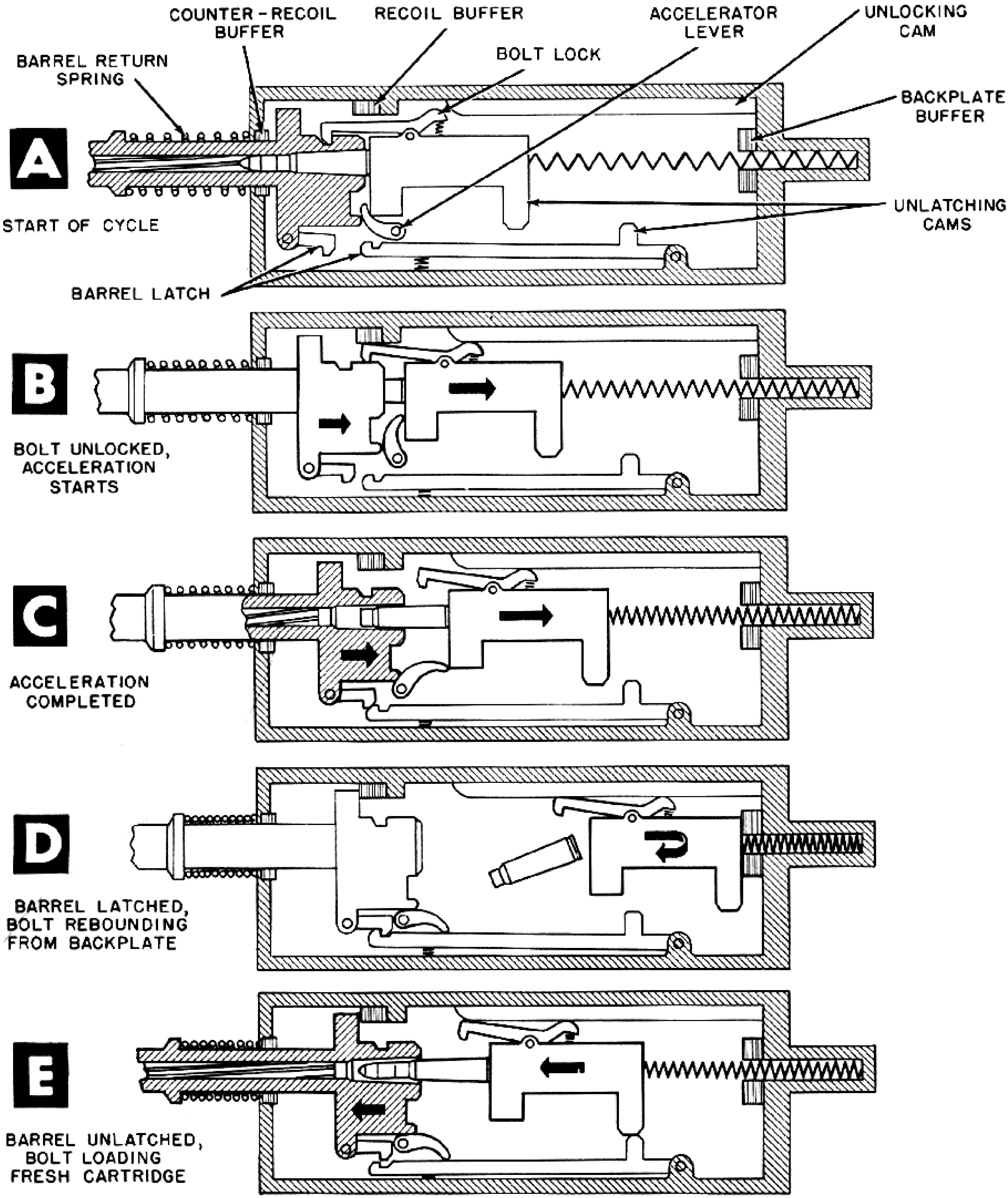


Figure 2–14. Elements of Short-Recoil System (Schematic).

Shortly after unlocking occurs, the barrel (which is still moving to the rear of its own momentum) strikes the accelerating device (fig. 2–14C). This device can take many forms, but in the mechanism illustrated, it is a lever pivoted in the breech casing. Fig. 2–15 shows the action of this lever in detail. In fig. 2–15A, the barrel extension is just in engagement with the lever and in fig. 2–15B the lever has been rotated to where it has started to thrust on the bolt. As the barrel continues moving to the rear, the lever rotates and the point of contact between the barrel extension and the lever moves closer to the lever pivot (fig. 2–15C). This causes the top of the lever to move more rapidly, thus imparting a high acceleration to the bolt.

When the accelerating action is completed, the barrel strikes a buffer stop which absorbs the remaining recoil energy in the barrel and its associated parts. After the barrel has been stopped, it is latched in its rearward position so that it is not immediately driven forward to battery by the compressed barrel return spring.

The combined action of the accelerating device and the blowback produced by the residual pressure imparts a high velocity to the bolt and then the bolt continues to move to the rear of its own momentum until the opening between the barrel and bolt is sufficient to permit feeding. As the bolt moves back, the spent cartridge case is extracted from the chamber and ejected and the bolt driving spring is compressed. This spring is relatively light and its only function is to assist the return motion of the bolt. Therefore, the driving spring does not absorb any great portion of the kinetic energy of the recoiling bolt and the bolt moves through its entire recoil distance at high velocity. The bolt then strikes the backplate buffer and rebounds. The forward velocity of the bolt immediately after leaving the backplate is somewhat less than the velocity at which it strikes the backplate because the impact is not purely elastic and some energy is lost as heat in the exchange.

As the bolt moves forward, its motion is aided by the driving spring. The bolt picks up a fresh cartridge from the feed mechanism and loads this cartridge into the chamber. Just before the bolt locks to the barrel, the barrel is unlatched so that the bolt and barrel move forward into battery while locked together. Shortly before the recoiling parts reach their most forward position, the firing mechanism is actuated and a new cycle begins. Since the cartridge is fired before the counter-recoil motion is completed, the forward velocity of the recoiling parts is first checked by the initial part of the rearward thrust exerted by the exploding propellant charge and the recoiling parts are then driven to the rear. (Timing the firing in this way eliminates the need for a heavy counter-recoil buffer to absorb and dissipate the forward kinetic energy of the recoiling parts.)

Analysis of Short Recoil

The most outstanding feature of the short recoil system of operation is that by proper design, very high cyclic rates can be attained. The bolt is unlocked without unnecessary delay shortly after the projectile has left the muzzle and then the bolt, which is already moving with considerable recoil velocity, is propelled to the rear at even greater velocity by the combined effects of the accelerator and blowback. With this high bolt velocity, the recoil movement of the bolt and its return to battery are accomplished in a very short time.

From the foregoing, it is evident that the bolt velocity in a short recoil gun is determined by three separate factors: (1) the recoil velocity at the time of unlocking, (2) the additional velocity imparted by blowback, and (3) the additional velocity resulting from the action of the accelerating device. In the following paragraphs, each of these factors is analyzed separately.

The recoil velocity of the bolt at the instant of unlocking is the result of the total impulse applied by the powder gases to the combined mass of the barrel and bolt while these parts are locked together. For any particular cartridge, the total forward momentum of the projectile and powder gases will produce an equal and opposite momentum in the recoiling parts. The recoil velocity resulting from this momentum will be inversely proportional to the weight of the recoiling parts; that is, the lighter the recoiling parts are, the higher will be their velocity. Therefore, within reasonable limits, to obtain a high recoil velocity while the barrel and bolt are still

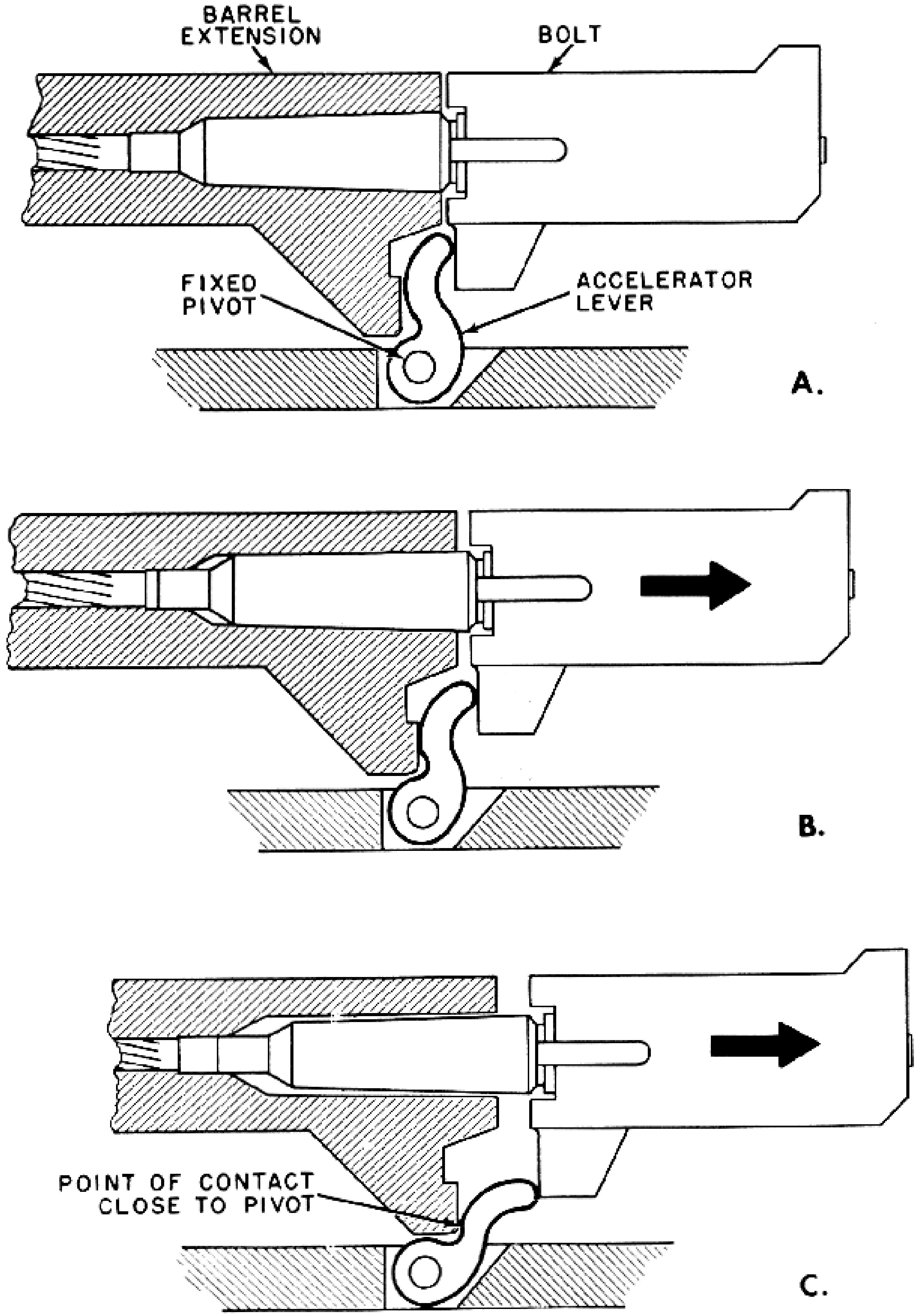


Figure 2-15. Action of Catapult Lever-Type Accelerating Mechanism.

locked together, the total weight of the recoiling parts should be kept to a minimum.

The maximum recoil velocity attainable in a practicable gun is limited because there are limitations on how light the recoiling parts can be. In order to perform their functions and to withstand the forces to which they are subjected, the barrel, bolt, and other recoiling parts must be ruggedly constructed and will necessarily be fairly massive. This is particularly true of the barrel which must be designed for sufficient strength to withstand the peak pressure of the propellant explosion and is also true of the barrel extension, bolt, and bolt locking device, all of which are subjected to the large forces produced by the powder gas pressure.

As has been mentioned previously, the effect of the limitations on the recoil velocity can be illustrated by considering a 20-mm gun. In a gun of this caliber, with a maximum chamber pressure of 45,000 pounds per square inch, the total weight of the recoiling parts including the barrel, bolt, bolt locking device, firing mechanism, springs, and other parts could not be much less than 40 to 50 pounds. The recoil momentum produced by a typical high-powered cartridge of this caliber would be approximately 35 (lb. sec.) and dividing this figure by the mass of the recoiling parts will give a maximum free recoil velocity of from 22 to 28 feet per second. Actually, in a short recoil gun, the bolt will be unlocked before this maximum velocity is attained and the recoil velocity at the instant of unlocking will be closer to 20 feet per second. It should be realized that an attempt to obtain a significantly higher velocity than this in the assumed weapon by lightening the parts would probably make the gun too frail.

The next point to consider is the additional bolt velocity produced by blowback after the bolt is unlocked. This effect in a short recoil gun is very similar to the action in a delayed blowback gun and all of the design factors which apply with delayed blowback are applicable with short recoil. The principal factor affecting the design is the allowable movement of the cartridge case during the action of the residual gas pressure.

Before blowback action is analyzed, it is appropriate to mention an operational feature associated with unlocking which can greatly improve performance during the blowback phase of the cycle. Effective blowback action is largely dependent on the absence of binding or excessive friction between the cartridge case and the walls of the chamber. When lubricated ammunition is used, friction and binding do not present a problem but with unlubricated ammunition, such as is used in short recoil guns, trouble may be encountered with binding. This binding results because the peak chamber pressures and the heat of the explosion expand the cartridge case tightly against the chamber and since unlocking occurs while there is still an appreciable residual pressure, the cartridge case does not have a chance to contract sufficiently to permit it to move freely under blowback. This difficulty can be avoided by employing an operational feature known as "initial extraction." When this feature is incorporated in the unlocking mechanism, the bolt is not unlocked completely at first but is cammed back slightly just sufficiently to cause the taper of the cartridge case to break free of the chamber walls. Immediately thereafter, the bolt is unlocked completely and blowback can occur without binding.

The first consideration relating to the blowback action is that unlocking must not occur too soon. At the instant initial extraction occurs and the bolt is unlocked, the pressure of the powder gases in the chamber will begin to drive the cartridge case and bolt to the rear. The velocity of the bolt motion imparted by this blowback effect will depend on the magnitude of the gas pressures and the mass of the parts subject to the blowback action. Since the bolt of a short recoil gun is of necessity quite light, it is possible in the presence of high gas pressures for the bolt to acquire an extremly high velocity. However, there is a definite limitation on the magnitude of the velocity which can be allowed under practical conditions. If the cartridge case and bolt move too rapidly under high pressure, the rear end of the case will move so far out of the chamber that the thin walls near the base of the cartridge case will not be supported by the chamber and will therefore rupture as the result of the high internal pressure. In other words, the motion of the cartridge case must be limited so that the case does not move too far while the chamber pressure is high enough to cause rupture.

The actual limit on the amount the cartridge case can be permitted to move as it is related to the chamber pressure will of course depend on the specific cartridge case under consideration. A good way to estimate the limit for a given cartridge case is to consider what pressure could be withstood by the case when the case has moved just far enough to the rear so that the thin walls near the base are exposed. (See fig. 2–16.) For an ordinary 20-mm cartridge case this occurs when the case has moved approximately 0.250 inch to the rear. When the case has reached this position, it is reasonable to assume that the internal pressure should not be in excess of 750 pounds per square inch, in order to be sure that the case will not be ruptured. Fig. 2–17, which is a graph showing the residual pressure variation with time for the assumed gun and cartridge, indicates that the pressure does not fall to 750 pounds per square inch until 0.005 second after ignition of the propellant charge.

Since the weight of the bolt in a short recoil gun is kept as low as possible, the only means whereby the movement of the bolt can be limited as desired is by selecting the proper time for unlocking. If the bolt is unlocked too soon, it will receive too great an impulse from the powder gases and its average velocity will be so great that the allowable 0.250-inch movement will be exceeded before the pressure drops to the safe limit of 750 pounds per square inch. If unlocking is delayed too long, the impulse imparted to the bolt will be unnecessarily small and the full benefit of blowback will not be realized. The ideal unlocking time for a given bolt weight is that which will permit the bolt to move the full allowable 0.250 inch and no more by the time that the pressure has dropped to 750 pounds per square inch.

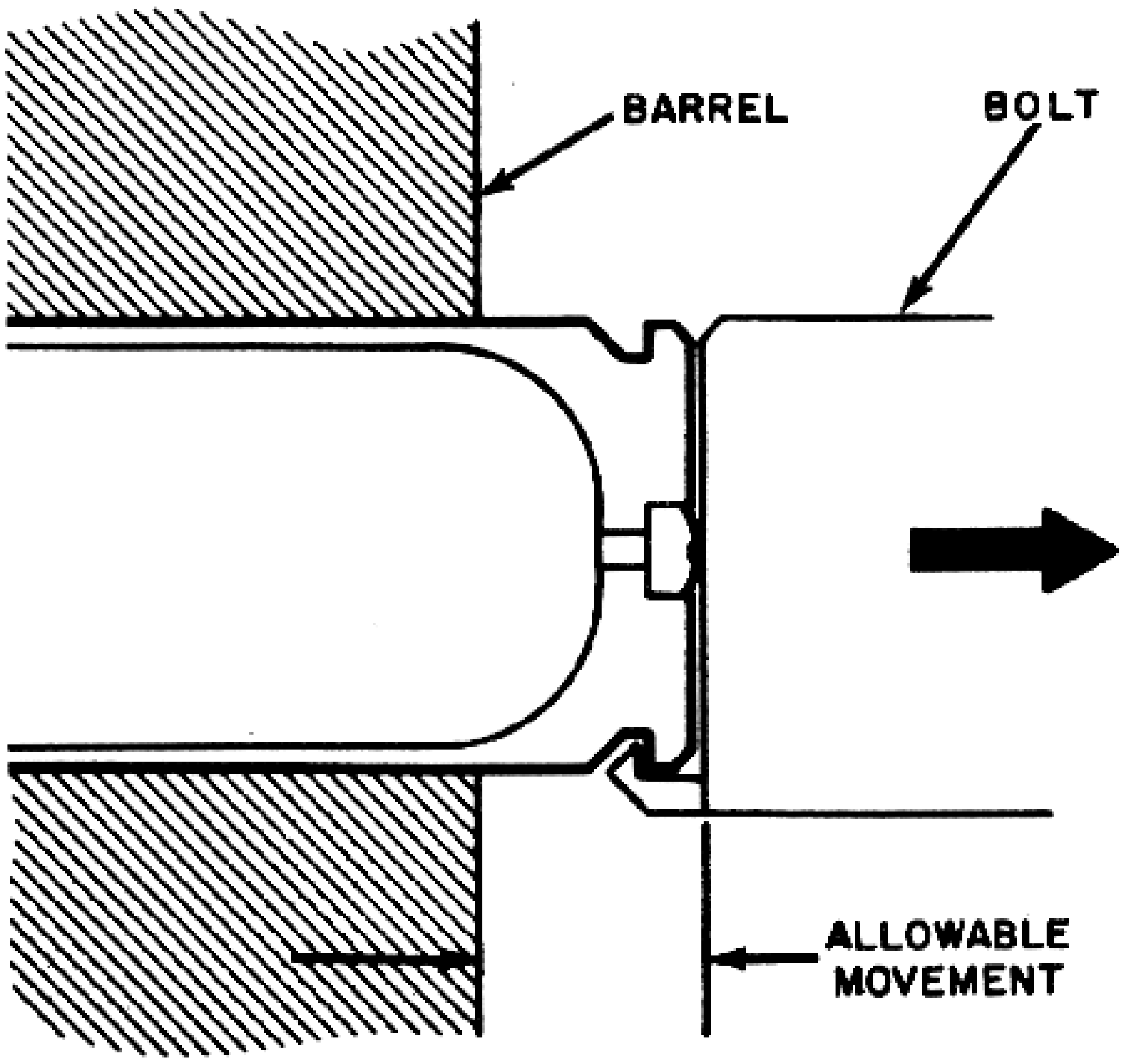


Figure 2–16. Limit of Cartridge Case Movement Rearward Before Residual Pressure Reaches Safe Value.

It is appropriate here to consider the effect of bolt weight as it is related to the blowback action in a short recoil gun. First, the primary purpose in making use of blowback is to obtain an increase in bolt velocity and therefore it is obviously desirable to set up the design so as to make the best possible use of the available blowback action. As has been explained in the preceding paragraphs, it is possible for blowback to produce extremely high bolt velocities but unfortunately it is necessary to limit the bolt motion in order to avoid rupture at the base of the cartridge case. (The limit assumed for the sample conditions is a bolt motion of 0.250 inch up to the time that the residual pressure has dropped to 750 pounds per square inch.) The question now at hand is how to limit the bolt movement as required and yet obtain a high velocity. The answer to this question lies in the selection of bolt weight and unlocking time.

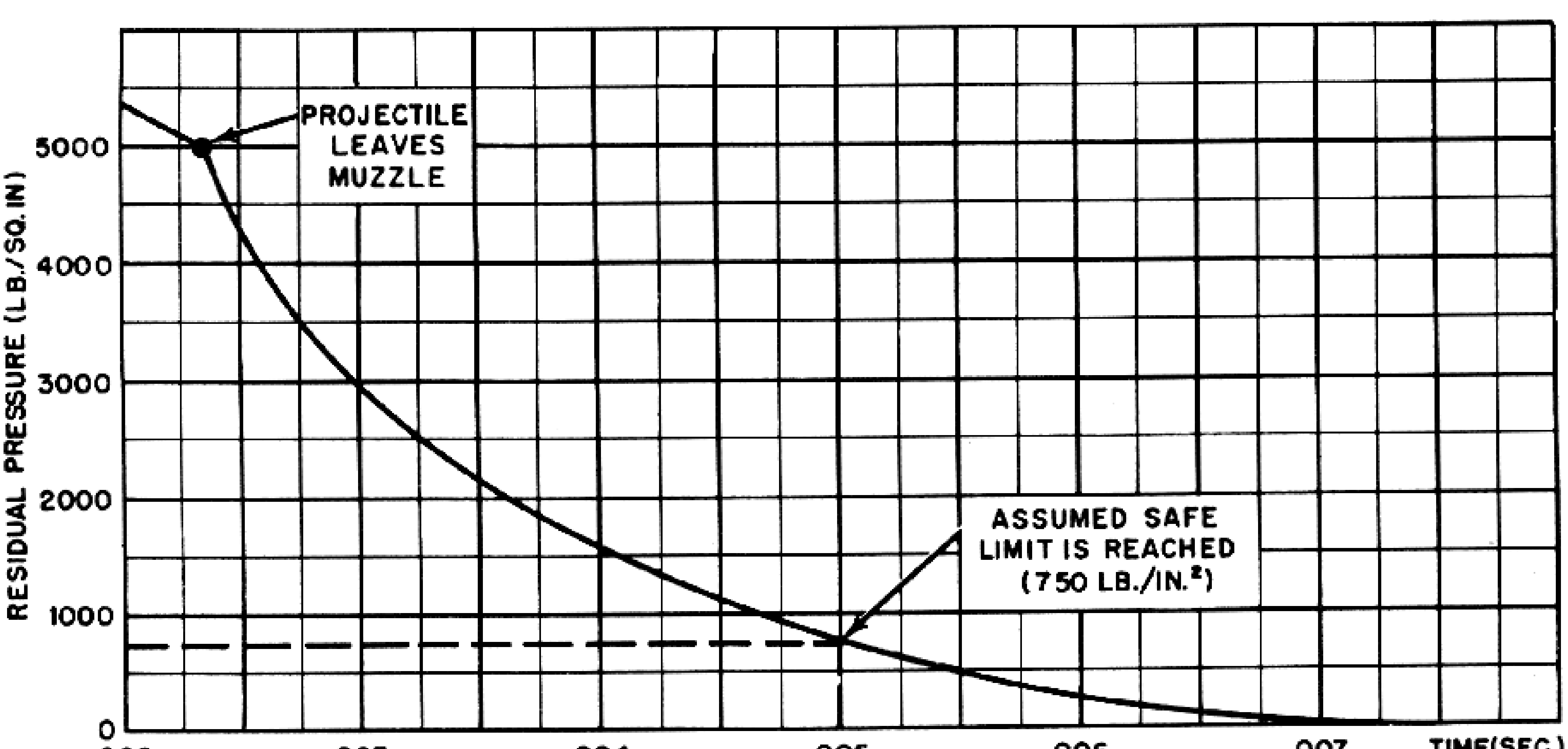


Figure 2–17. Graph of Residual Pressure Versus Time for a Typical 20 mm Gun.

The basic controlling factor is the limit on the allowable movement (0.250 inch). If the time in which this movement is accomplished is long, the average velocity of the movement will necessarily be low. However, if the time for the movement is made very short, the average velocity may be very large. For example, suppose the bolt is unlocked 0.002 second before the safe pressure of 750 pounds per square inch is reached at 0.005 second. The bolt can now travel the 0.250 inch in 0.002 second. That is to say, its average velocity for this interval can be:

$$V_{av}=\frac{D}{t}=\frac{.250}{12}\times\frac{1}{.002}=10.4\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

Now assume that the bolt is unlocked only 0.001 second before the safe pressure is reached. It can then travel the 0.250 inch at an average velocity for the interval of:

$$V_{av}=\frac{D}{t}=\frac{.250}{12}\times\frac{1}{.001}=20.8\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The foregoing indicates that, by shortening the time for which the blowback operates before the safe pressure is reached, increased bolt velocity can be achieved without exceeding the allowable bolt movement.

Of course, it should be realized that shortening the time of blowback action reduces the blowback impulse available for producing the velocity. Thus, in order to gain an increase in allowable average velocity by reducing the time of action, it is necessary to reduce the bolt weight. To illustrate this point, if the velocity of 10.4 feet per second cited in the first example above was obtained with an 8-pound bolt, it would be necessary (with the same cartridge and gun) to reduce the bolt weight by a factor of at least 4, even if it is assumed that the average blowback pressure is the same for both examples. Actually, since the residual pressure decreases with time, the average pressure for the second example would be considerably less than for the first and therefore a further reduction in bolt weight would be required. The actual weight reduction factor would be more nearly in the neighborhood of 6, giving a bolt weight of only 1.3 pounds. (This would probably be much too light for a practical gun.)

Thus, it appears that a substantial gain in average allowable bolt velocity by means of reducing the time of blowback action can be achieved only by a drastic reduction in bolt weight. Although it is not practical to attempt to reduce bolt weight by an excessive amount, it is important to note that efficient utilization of blowback in a short recoil gun can be of great advantage in attaining the high bolt velocity necessary for a high rate of fire. However, this advantage can be gained only through precise timing of unlocking in combination with careful attention to minimizing bolt weight.

The third subject for consideration is the action of the accelerating device. From what has been said in the preceding paragraphs about the effects of blowback, it can be seen that the action of the accelerator should be delayed until after the residual pressure has dropped to the safe limit there specified (750 pounds per square inch for the sample conditions). If this delay is not provided, the effect of the accelerator will be wasted up until the time the pressure has dropped to the safe limit. As has been explained, the velocity of the bolt must be limited so that the bolt movement does not exceed 0.250 inch while the pressure is above 750 pounds per square inch. Since blowback alone can easily impart the velocity required to produce the 0.250-inch movement, assistance from the accelerator is not necessary or desirable. However, once the pressure has fallen below 750 pounds per square inch, rupture of the cartridge case will not occur and the need for limiting the bolt velocity no longer exists. At this point, then, the accelerator can start its action and further increase the velocity of the bolt.

There are several important points concerning the action of the accelerating device which should be considered at this time. First, the device should be designed carefully to act smoothly and positively so that it can transfer velocity to the bolt without excessive shock or friction. This is particularly necessary because even at best the action of the accelerator usually must be completed in a few thousandths of a second and will therefore be quite violent. If the action is not made as smooth as possible, battering of the parts, deformation of the mechanism, and frequent breakages will be unavoidable. Because of the extremely large forces involved in transferring the required energy so rapidly from the barrel mass to the bolt, the parts of the accelerator should be ruggedly dimensioned and properly heat-treated for maximum strength and wearing qualities.

There is no ideal design type for an accelerator and many different types have been used with good results. Accelerators of various types are illustrated in Part XI of this publication and therefore particular design configurations will not be treated in detail here. The present discussion is limited to the design factors affecting the use of acceleration in short recoil guns. The basic design requirements for an accelerating device can be enumerated as follows: The mechanism should, of course, be as simple as possible so that it can be manufactured with the least difficulty and is fundamentally reliable. The form of the mechanism should be such that its parts are compact and rugged and do not require delicate adjustment for effective functioning. If the foregoing requirements are satisfied, the next major consideration is one of "efficiency". In this connection, the term "efficiency" is used in a special sense. Since the object of using the accelerator is to speed up the bolt by making effective use of the kinetic energy in the recoiling barrel, the efficiency of the accelerator can be reckoned in terms of what percentage of the available energy is transferred to the bolt.

To analyze the factors involved in the transfer of energy from the barrel to the bolt, consider the type of accelerator shown in fig. 2–18. (This type of accelerator is known as a "catapult spring" device.) With this type of accelerator, as the barrel recoils, the catapult strikes a latch in the breech housing and the barrel compresses the catapult spring as it moves to the rear in recoil. At the same time, the bolt moves to the rear until a lug on the bottom of the bolt is engaged by a catch on the top of the catapult. The catapult latch is then released so that the compressed catapult spring will drive the bolt to the rear. Without becoming concerned with the complications involved in properly timing the various latching and unlatching operations and other practical considerations, assume that all of the kinetic energy of the recoiling barrel is stored in the catapult spring (that is, the catapult spring brings the barrel to a complete stop) and further assume that all of this energy is transferred to the bolt without loss.

Now take the barrel weight as 45 pounds and the bolt weight as 5 pounds. Assume that the velocity of the barrel before starting to compress the catapult spring is 20 feet per second and that the velocity of the bolt just before the catapult is released is 32 feet per second (allowing for the increase due to blowback). The initial kinetic energy of the barrel (the amount of energy assumed to be stored in the spring) is:

$$KE_1=\frac{1}{2}\frac{W_1}{g}V_1^2=\frac{1}{2}\times\frac{45}{32.2}\times 20^2=280 \text{ (ft. lb.)}$$

The kinetic energy of the bolt before the catapult action starts is:

$$KE_2=\frac{1}{2}\frac{W_2}{g}V_2^2=\frac{1}{2}\times\frac{5}{32.2}\times 32^2=79.5 \text{ (ft. lb.)}$$

Since it is assumed that all of the kinetic energy originally possessed by the barrel is transferred through the spring to the bolt, the final kinetic energy of the bolt will be:

$$KE_3=280+79.5=359.5 \text{ (ft. lb.)}$$

The final velocity of the bolt will then be:

$$V_3=\sqrt{\frac{(KE)\times 2g}{W}}=\sqrt{\frac{359.5\times 64.4}{5}}$$
$$=68.1\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

Thus it appears that even under the ideal condition of 100 per cent efficiency of energy transfer, the factor by which the bolt velocity is increased for the stipulated values of mass and velocity is only approximately 2 times. The factor will vary depending on the ratio between the barrel and bolt masses (increasing with this ratio) and will also vary slightly depending on the values of the initial velocities. Nevertheless, the values assumed in the example are more or less representative of a typical 20-mm gun and it can be seen that there is a definite limit on the velocity gain that can be expected.

Under actual conditions, it is usually not practical to attempt to bring the barrel to a complete stop through the action of the accelerating device

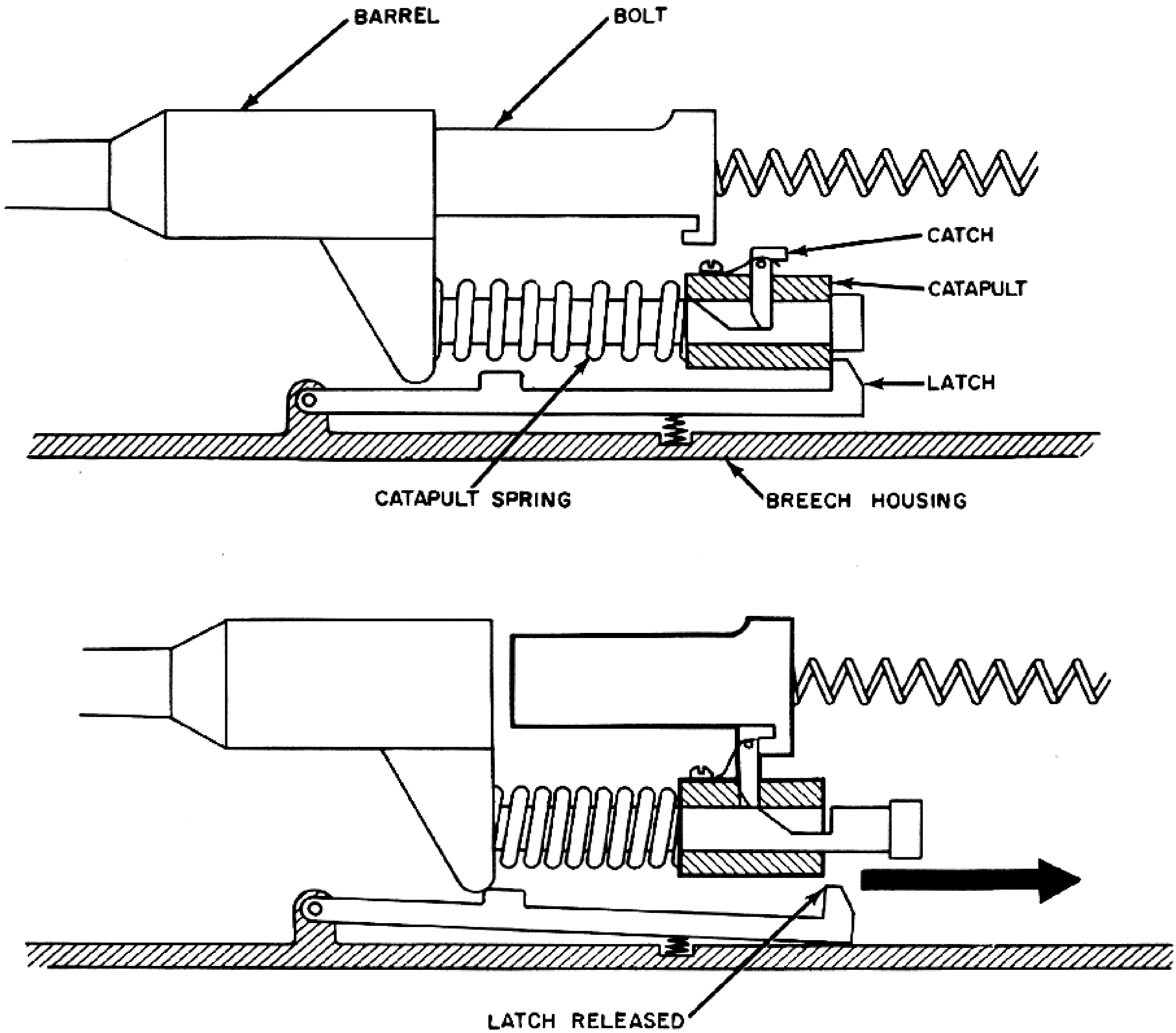


Figure 2–18. Action of a Catapult Spring Accelerating Device.

alone and this by itself reduces to a considerable extent the amount of energy transferred. Furthermore, there will be friction and impact losses and other inefficiencies in the mechanism which also will reduce the velocity gain factor. All things considered, the velocity gain in any practical, well designed mechanism will probably be such that the bolt velocity after acceleration will be approximately 1.5 times its velocity at the start of acceleration.

It is well to mention here that the same general conclusions apply to accelerators of the lever or cam type. Merely increasing the lever ratio in a lever-type accelerator will not result in a higher velocity gain. In the final analysis, there is only a certain amount of energy available in the recoiling barrel and no matter how efficiently this energy is transferred, the change it can produce in the velocity of the bolt is definitely limited. A lever ratio which is too high will result, not in a greatly increased bolt velocity, but in excessive strain and shock on the mechanism.

The preceding analysis has been concerned with the three basic factors which determine the bolt velocity in a short recoil gun. These factors are:

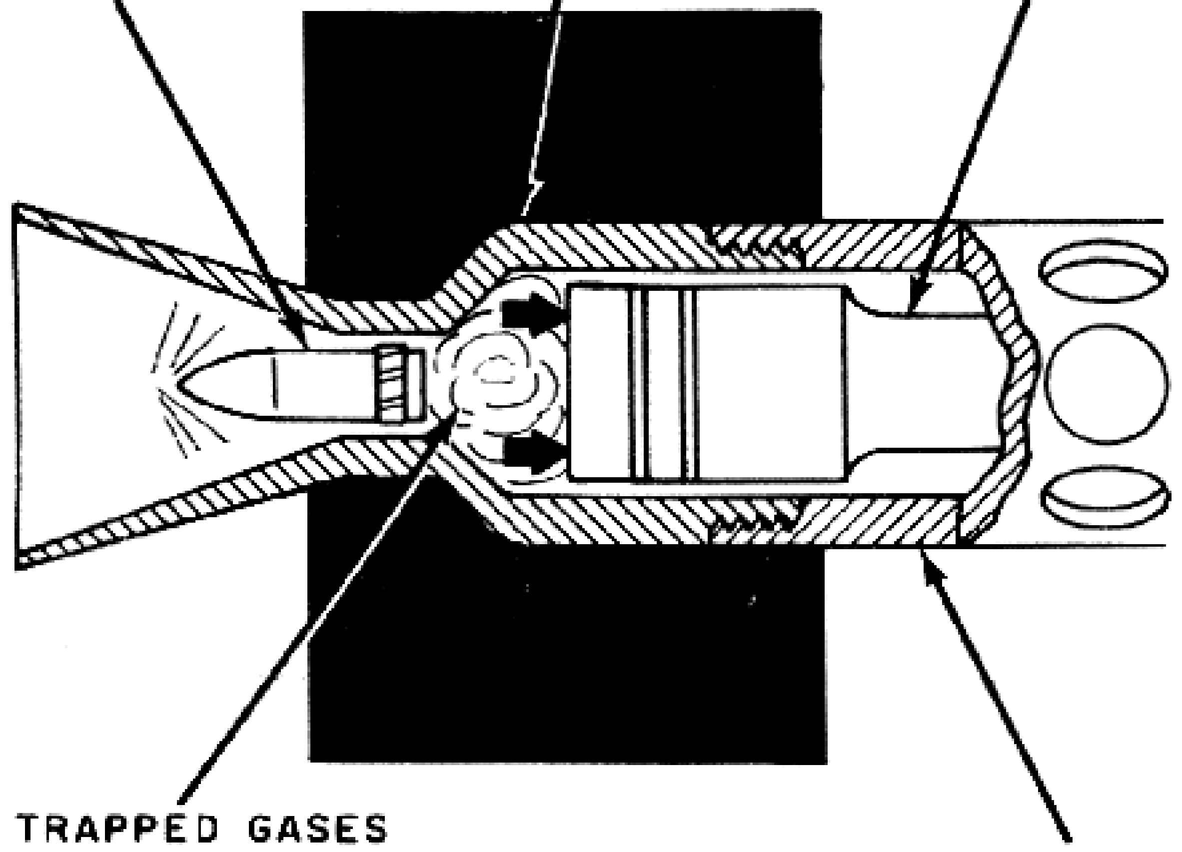


Figure 2–19. Example of a Muzzle Booster.

(1) the recoil velocity imparted to the barrel and bolt while these parts are still locked together, (2) the additional velocity imparted to the bolt by blowback after unlocking, and (3) the gain in bolt velocity produced by the accelerator. Since the principal reasons for using the short recoil system of operation is to obtain a high rate of fire, each of these factors is considered from the standpoint of how to achieve the highest possible bolt velocity consistent with safety and reliable functioning. The methods used to obtain a high bolt velocity may be summarized as follows: The total weight of the recoiling parts should be kept to a minimum so that the recoil velocity before unlocking will be high. Next, the bolt should be as light as possible and should be unlocked at the proper instant so that the blowback effect can produce a large increase in velocity without causing rupture of the cartridge case. Finally, the accelerating device should start to function just at the instant the residual pressure has reached a safe limit and should be designed to produce the maximum possible transfer of energy from the barrel to the bolt.

Another means of increasing the rate of fire in a short recoil gun is to employ a device known as a "muzzle booster" or "recoil intensifier". An example of a device of this type is shown in fig. 2–19 and other forms are illustrated in Part XI of this publication. The booster is installed at the muzzle of the gun and operates by trapping the muzzle blast in such a way as to apply a heavy thrust on the front face of the barrel. This additional thrust causes the recoiling parts to have a higher velocity and hence increases the velocity inherited by the bolt as well as increasing the energy available for acceleration of the bolt. Also in some designs, it is possible for the trapped gas to produce a slightly greater blowback effect than would be obtained without the booster. Although the muzzle booster can be used with good effect to increase the rate of fire of a short recoil gun, it is important not to employ excessive boosting action. If the booster acts too powerfully, extremely violent recoil will result with the consequent danger of severe shocks and pounding.

All of the foregoing analysis is related to the methods which can be used to impart a high initial velocity to the bolt. It is now appropriate to consider the motion of the bolt after this velocity is imparted. The bolt is given its initial velocity early in its rearward travel and then it completes its motion of its own momentum. To permit feeding, the bolt must move to the rear through a distance at least as great as the overall length of the complete cartridge and then its motion must be reversed to load the gun and close the breech. In some guns, this action is accomplished through the use of a relatively powerful driving spring which is compressed as the bolt moves in recoil. The spring absorbs the kinetic energy of the bolt over the full recoil travel, finally stopping the rearward motion of the bolt when all of the kinetic energy of the bolt has been absorbed. The spring is designed so that this occurs when the opening is sufficient to permit feeding. The compressed spring then drives the bolt forward to complete the operating cycle. This type of design has a serious drawback from the standpoint of speed of operation. Since the bolt is gradually slowed down by the spring, its velocity varies from maximum at the beginning of recoil to zero at the end of recoil. (See fig. 2–11, which is a graph showing how the bolt velocity varies with time under these conditions.) The fact that the bolt velocity varies from maximum to zero in the manner illustrated means that its average velocity will only be slightly greater than one-half its maximum velocity. In other words, if this type of action were used in a short recoil gun, in spite of all the pains taken to achieve a high initial bolt velocity, the overall travel of the bolt would be accomplished at a much lower average velocity.

To overcome this disadvantage, the bolt driving spring can be made relatively very light so that it offers a low retardation and will permit the bolt to move its entire recoil distance with little loss in velocity. In this case, the function of the driving spring is merely to provide a positive force which is just sufficient to insure that the bolt will close. Stopping the bolt at the end of its travel and reversing its motion can then be accomplished by causing the bolt to rebound from a so-called "backplate buffer". This device is in effect an extremely stiff spring which absorbs all of the kinetic energy of the bolt over a very short distance and then delivers energy back to the bolt to propel it forward. The reversing action produced by the backplate is so abrupt that the effect may be classified as an elastic impact.

In order to obtain a high rate of fire, it is also important for the bolt return to be accomplished at high velocity. If there were no energy losses involved in the reversing action, the forward velocity of the bolt after leaving the backplate would be equal to the velocity at which it strikes the backplate. This would be the ideal condition. However, in actual practice the coefficient of restitution for the bolt and backplate is usually considerably less than unity and the best that can be expected is a coefficient in the neighborhood of 0.60 or 0.70; that is, the velocity after impact will be 60 or 70 per cent of the velocity before impact. This represents satisfactory performance, but if the coefficient of restitution is too low as the result of poor backplate design, the return of the bolt will be sluggish and the rate of fire will be affected adversely. In this connection, it should be emphasized that the purpose of the backplate buffer is to reverse the motion of the bolt with as little loss of energy as possible. In many instances, the term "buffer" is used to refer to a device which has the primary purpose of dissipating impact energy rather than of conserving energy to produce an efficient rebound action. For this reason it might be better to refer to the backplate buffer as a "bolt deflector" or "rebound plate."

Another important consideration in the operation of a short recoil gun is related to the events which occur at the end of the return motion of the bolt during the transition from the completion of one operating cycle to the start of the next cycle. It is at this point that the timing of the various operations is particularly critical. The return of the bolt, which is moving at high velocity, and the return of the heavy barrel to battery can be attended by heavy shocks and severe vibrations if proper attention is not given to the problem of synchronizing the operations and dissipating the kinetic energy of the moving parts. The problems are multiplied in high rate of fire guns if firing of the fresh cartridge occurs before the vibrations resulting from firing the previous round have settled out. Under these conditions, the action of the gun may be extremely erratic and jerky and stresses may develop which will literally cause the gun to batter itself to pieces.

The value of proper timing is well illustrated by the type of mechanism used at the start of this section to serve as an example of a short recoil gun. In this mechanism, the barrel is latched in its rearward position after recoil and remains in this position while the bolt completes its movement to the rear and returns. Just before the bolt reaches the barrel on the return stroke, the barrel is unlatched and starts moving forward due to the force exerted on it by the barrel spring. Therefore, the barrel is already moving forward when the bolt re-engages with the accelerator lever. Now since the bolt is moving at a higher velocity than the barrel, the accelerator lever acts to slow down the bolt and to increase the velocity of the barrel. In a correctly arranged design, this action will be initiated with little shock and will smoothly decrease the relative velocity between the barrel and bolt to a much lower value than existed before the lever was engaged. Thus, when the bolt strikes the barrel and locks to it, the impact is reduced. Also, since the barrel is not latched and is free to move forward, the severity of the metal-to-metal shock which results when the bolt strikes the breech face is further decreased. Now both the barrel and bolt continue moving forward together and strike a counter-recoil buffer, but before the velocity of counter-recoil is brought to zero, the fresh round which has been loaded in the chamber is fired. The force of the propellant explosion then opposes the forward motion and this force quickly brings the recoiling parts to a stop and then propels them to the rear. Although this reversal of motion occurs with great rapidity, the cushioning effect of the explosion causes it to be accomplished smoothly and without vibration or any heavy metal-to-metal shock.

It should be noted that utilizing a portion of the explosive impulse for reversing the motion of the recoiling parts has the effect of decreasing the recoil velocity to some extent and therefore will tend to reduce the rate of fire. However, the loss is relatively slight and is more than compensated for by the resulting smoothness of operation.

Mathematical Analysis of Short Recoil

The details of the mathematical analysis of a design employing short recoil will depend to some extent on the particular forms of the mechanisms utilized for performing the various operational functions and therefore it is not feasible to set up an analytical method which will apply universally to all short recoil guns. However, the basic functions performed in all guns of this type are sufficiently similar to justify illustration of the general methods by analysis of a gun which is more or less representative of the type.

The following analysis employs methods which are similar to those used in the other portions of this publication with the addition of the special procedures required for handling the specific design problems related to the short recoil system of operation. Again, as for the other systems, the analytical methods treated here are concerned only with the basic bolt and barrel motions and the related forces. No attempt will be made to discuss the straightforward machine design methods by which the results are applied in arriving at the particular physical form of the mechanisms. Also, no detailed computations are made to cover the effects of such factors as friction or the incidental forces imposed on the breech mechanism by the auxiliary mechanisms such as the feeder, firing device, or locking device. These effects will have only a relatively slight influence on the bolt and barrel motions. In any case, they can be properly taken into account only in the advanced stages of a design when the form of the gun mechanism becomes fairly well established. At this point, the preliminary analysis of the bolt and barrel motion can easily be modified as desired.

The analysis which follows is based on the assumption that a particular cartridge with known characteristics is to be used and that the desired muzzle velocity and barrel length have been predetermined. It is also assumed that all necessary interior ballistics data are known and that graphs showing the time variation of projectile velocity and chamber pressure are available (figs. 2–4, 2–5, and 2–6).

> NOTE: For some design problems, all or part of this information may not be available. Analytical methods by which the required data and graphs can be approximated for use in preliminary studies may be determined by conventional interior ballistics computations.

In the preceding description of the factors involved in short recoil operation, considerable emphasis was placed on the importance of keeping the weight of the recoiling parts to a minimum in order to achieve a high rate of fire. As has been mentioned previously in this publication, the weight of the recoiling parts of any gun will be affected not only by requirements for strength, rigidity, and durability but will also depend to a large extent on the particular configuration selected by the designer. (To repeat the example previously cited, if the designer wishes to have the bolt slide in a long barrel extension, the recoiling parts might be heavier than they would be if the bolt moved on guide rails which were part of the receiver.) Therefore, the weight of the recoiling parts can not be determined with any accuracy until the barrel has been designed and the remainder of the mechanism has been laid out at least to the extent which will make it possible to make a fair preliminary estimate of what weights will be involved. In the process of planning the mechanism, it will also be necessary to determine what distances the parts must travel.

Of course, the final dimensions and weights of some of the recoiling parts can not be defined until complete consideration is given to the forces which act on these parts as the result of the accelerations and shocks to which they are subjected in operation and which yet remain to be determined. However, the estimated weights obtained from a carefully made preliminary design and layout should be accurate enough to serve as a starting point for the calculations necessary to determine the operating forces. It is these calculations which are the main concern of the following analysis.

As the analysis progresses, its applications will be illustrated by means of sample calculations. Although these calculations and the related graphs are for a specific 20-mm cartridge and barrel and are based on certain assumed weights and other characteristics, the general approach described is applicable to short recoil guns of any caliber. The calculations cover the following important points:

1. Determination of the conditions of free recoil.
2. Determination of correct time for unlocking.
3. Computation of data required for design of accelerator.
4. Selection of characteristics of barrel return spring and bolt driving spring and determination of data for design of backplate buffer.
5. Development of graphs showing how the velocity and travel of the barrel and bolt vary with time.

In the course of describing these calculations, the following fundamental formula will be developed and explained:

a. Momentum and velocity relation for time projectile is in bore.
b. Formula for determining velocity of free recoil.
c. Expression for duration of residual pressure.
d. Formulas for determining spring retardations.

(Because of the fact that before unlocking occurs there is no difference between the analysis of a short recoil gun and a long recoil gun, some of the explanations and derivations are identical for both systems of operation. In such cases, to avoid the inconvenience and confusion of attempting to refer back to the explanations given under long recoil, the material will be repeated here.)

1. *Conditions of free recoil*

Under the heading "Principles of Recoil" it was pointed out that, if a gun is mounted so that it can move freely without friction or any other restraint, the impulse of the recoil force will impart to the gun a rearward momentum equal to the total forward momentum of the projectile and powder gases. For the time the projectile is in the bore, this momentum relationship is expressed by the equation:

$$\text{(2–17)} \qquad M_r v_{r_f} = M_p v_p + M_e v_e$$

Since the powder gases will be thoroughly mixed by the turbulence created in the explosion, it is reasonable to assume that the center of mass of the gases moves forward at one-half the velocity of the projectile. Actually, this is not quite accurate because the presence of the enlargement at the chamber and the fact that the rifling does not extend the full length of the space occupied by the gases creates a condition in which the volume of the space is not uniformly distributed along its length. Nevertheless, the assumption is close enough for present purposes. Therefore equation 2–17 may be rewritten as:

$$\text{(2–18)} \qquad M_r v_{r_f} = M_p v_p + M_e \frac{v_p}{2} = \left(M_p + \frac{M_e}{2}\right) v_p$$

NOTE: It should be pointed out here that the momentum equality expressed by equation 2–18 is not affected by the internal frictional forces opposing the motion of the projectile and powder gases or by the forces incident to engraving the rifling band and to imparting the rotational velocity of the projectile. Although all of these forces retard the forward motion of the projectile and powder gases, they produce equal and opposite reactions on the barrel which result in a corresponding retardation of the rearward movement of the gun. In other words, the internal resistances merely decrease the effective impulse producing motion but they do not cause any inequality in the forward and rearward momentums.

Solving equation 2–18 for v_{r_f} gives the velocity of free recoil for the time the projectile is in the bore as:

$$\text{(2–19)} \qquad v_{r_f} = \frac{M_p + \frac{M_e}{2}}{M_r} v_p = \frac{W_p + \frac{W_e}{2}}{W_r} v_p$$

Equation 2–19 can be used to plot a curve showing the free recoil velocity versus time for the period before the projectile leaves the muzzle. The weights of the projectile and powder charge are both known and it is assumed that the weight of the recoiling parts has been estimated in accordance with a preliminary design plan. Also, the velocity of the projectile at any time is known from the

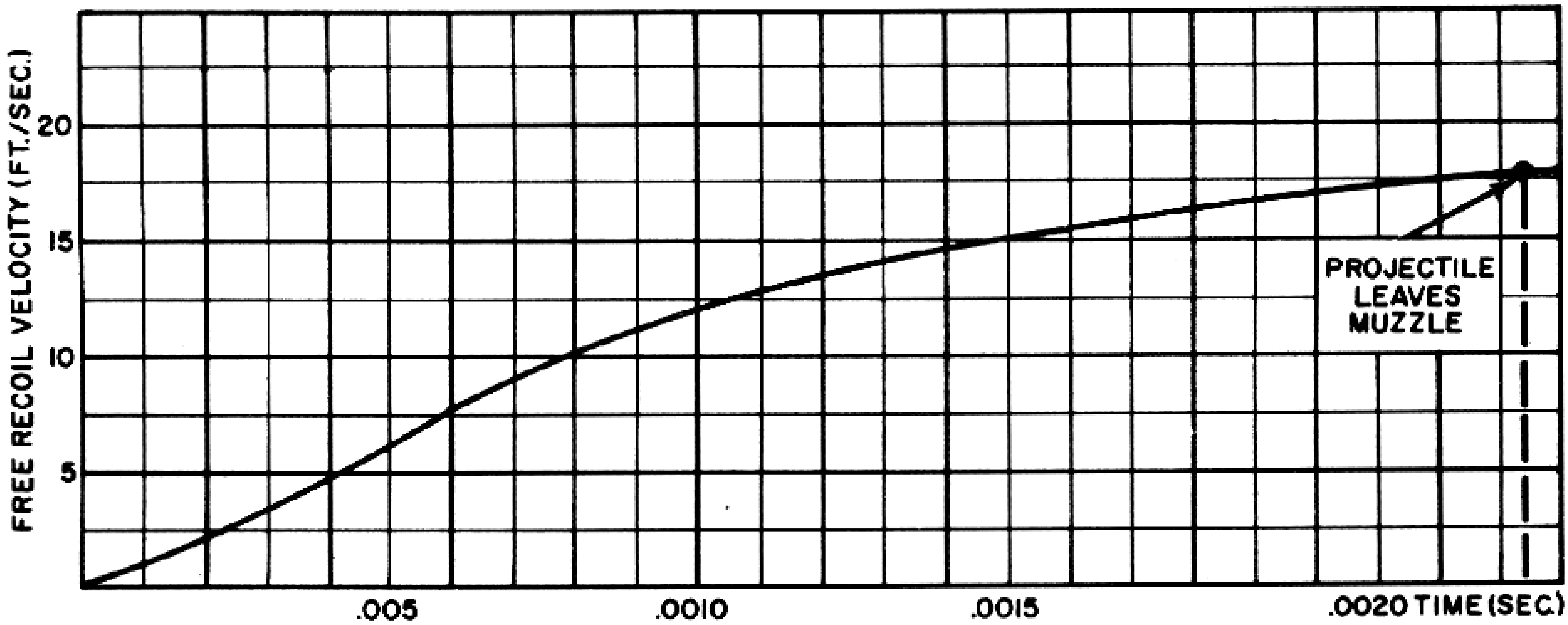


Figure 2–20. Free Recoil Velocity While Projectile Is in Bore.

available ballistic data (fig. 2–5). Therefore, the ordinate of the free recoil velocity curve at any time, t, can be found by multiplying the corresponding ordinate of the projectile velocity curve by the factor:

$$\frac{W_p+\frac{W_c}{2}}{W_r}$$

Assuming that in the 20-mm gun to be used as an example the estimated weight of the recoiling parts is 50 pounds and the weights of the projectile and powder charge are as shown in fig. 2–5, the value of the multiplying factor is:

$$\frac{W_p+\frac{W_c}{2}}{W_r}=\frac{.29+\frac{.070}{2}}{50}=.00650$$

Therefore, before the projectile leaves the muzzle, the free velocity of the recoiling parts is:

$$v_{r_f}=.00650\ v_p\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The curve obtained by using this relation is shown in fig. 2–20 and is also shown in fig. 2–21 as the portion between t=0 and t=.00234 second. (In fig. 2–21, the time axis is compressed in order to show how the velocity varies after the projectile leaves the muzzle.)

The manner in which the free recoil velocity varies after the projectile leaves the muzzle can not be determined from equation 2–19 because the projectile and a portion of the powder gases are no longer part of the system. Since the effect of the residual pressure can not be expressed in simple terms, a special method is used to extend the curve obtained from equation 2–19. This method is based on the fact that the results of experimental firings of various guns show that the maximum velocity of free recoil may be closely approximated as:

$$V_{r_f}=\frac{W_pV_p+4700\ W_c}{W_r} \tag{2–20}$$

This relationship is equivalent to saying that the maximum momentum imparted to the recoiling parts is equal to the sum of the muzzle momentum of the projectile and the momentum of the powder gases, assuming that the powder gases leave the gun at an average velocity of 4700 feet per second. For the gun used as an example:

$$V_{r_f}=\frac{.29\times2750+4700\times.070}{40}=22.5\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

A line representing this value of the maximum velocity of free recoil is drawn on the velocity graph (fig. 2–21) and the curve previously drawn from equation 2–19 is extrapolated until it becomes tangent to the line. The point at which the curve becomes tangent represents the time at which the residual pressure becomes zero and therefore imparts no further velocity to the recoiling parts. Although an error in locating the exact point of tangency will not have any serious effect on the accuracy of the results, it may be of some assistance in plotting to

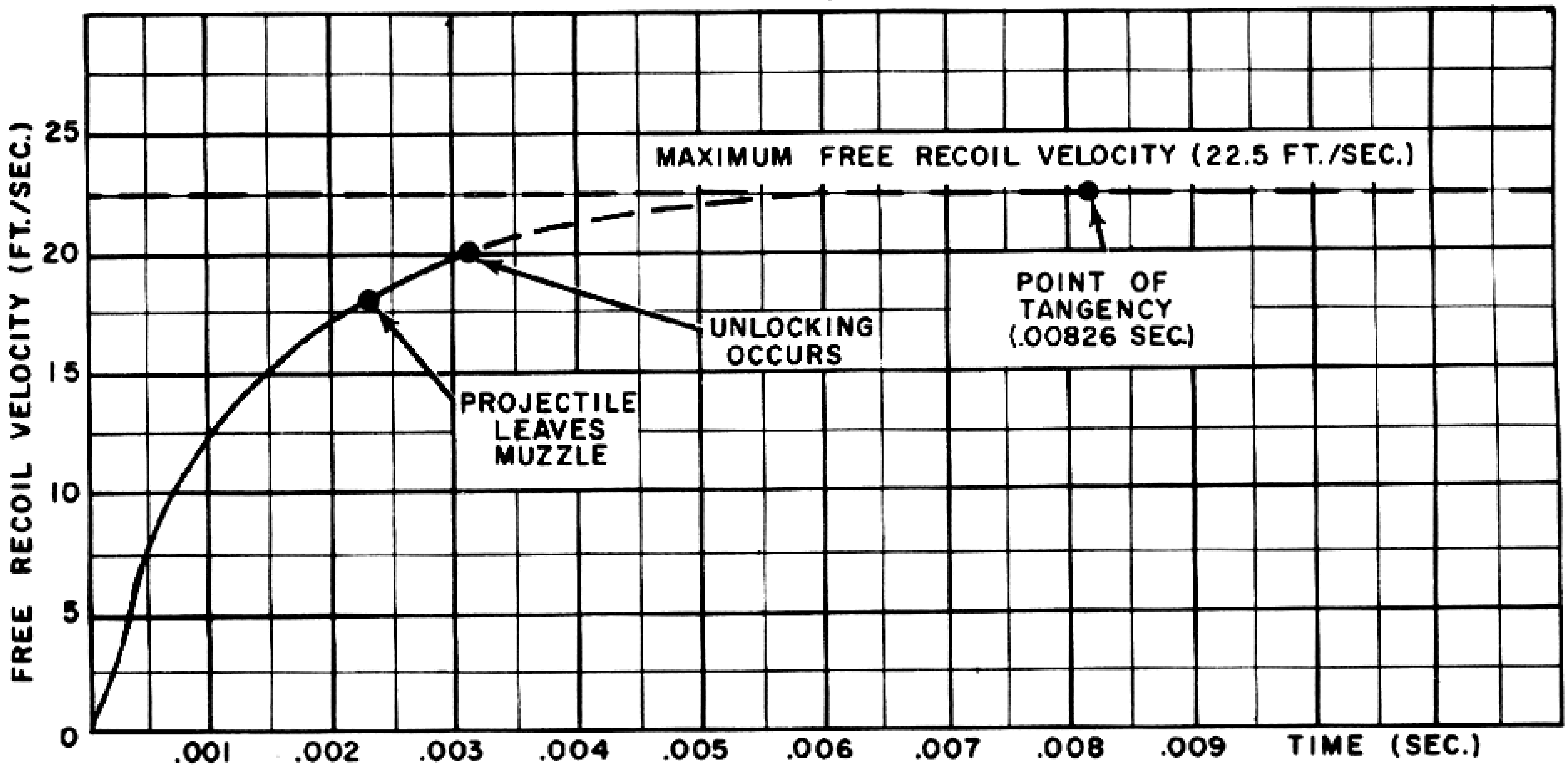


Figure 2-21. Free Recoil Velocity Before Unlocking.

determine this point by using Vallier's formula for approximating the duration of the residual pressure:

$$T_{res}=\frac{M_c}{AP}(9400-V_p) \qquad (2\text{-}21)$$

For the sample cartridge and barrel:

$$T_{res}=\frac{.070}{32.2\times\frac{\pi}{4}(.790)^2\times5000}(9400-2750)$$

$$=.00592 \text{ (sec.)}$$

To obtain the total time of action of the powder gases, this value is added to the time at which the projectile leaves the muzzle:

$$T_{res}=.00234+.00592=.00826 \text{ (sec.)}$$

Extending the original curve until it is tangent at this point gives the complete free recoil velocity curve shown in fig. 2-21. Actually, the entire curve shown in the figure does not apply to the actual recoil conditions in a short recoil gun because unlocking occurs before the residual pressure has become zero.

2. *Effect of blowback before acceleration and computation of unlocking time*

The next point for consideration is the effect on the bolt velocity of the blowback action which occurs between the time that unlocking occurs and the time that the accelerator starts to operate. As pointed out in the analysis of short recoil, the ideal condition for this portion of the blowback action is that the bolt should move 0.250 inch with respect to the barrel by the time that the residual pressure has dropped to the safe limit of 750 pounds per square inch. (These figures are based on assumed safe values for a typical 20-mm cartridge and should be checked experimentally for any specific cartridge.)

For purposes of determining the blowback effect, it is only necessary to consider the velocity of the bolt with respect to the barrel. After the bolt is unlocked, the residual pressure has no further effect on the barrel but merely imparts motion to the bolt with respect to the barrel. It will be assumed here that the bolt weight, as estimated from a preliminary layout of the mechanism is 5 pounds.

Fig. 2-17 which is a graph of the residual pressure versus time for the sample gun, shows that the residual pressure reaches 750 pounds per square inch at 0.005 second. The problem is to decide how long before this point the bolt should be unlocked so that its motion with respect to the barrel will be 0.250 inch at 0.005 second. This problem can be solved using the data in fig. 2-21. If the ordinates of the velocity curve in fig. 2-21 are multiplied by the mass of the recoiling parts, the resulting

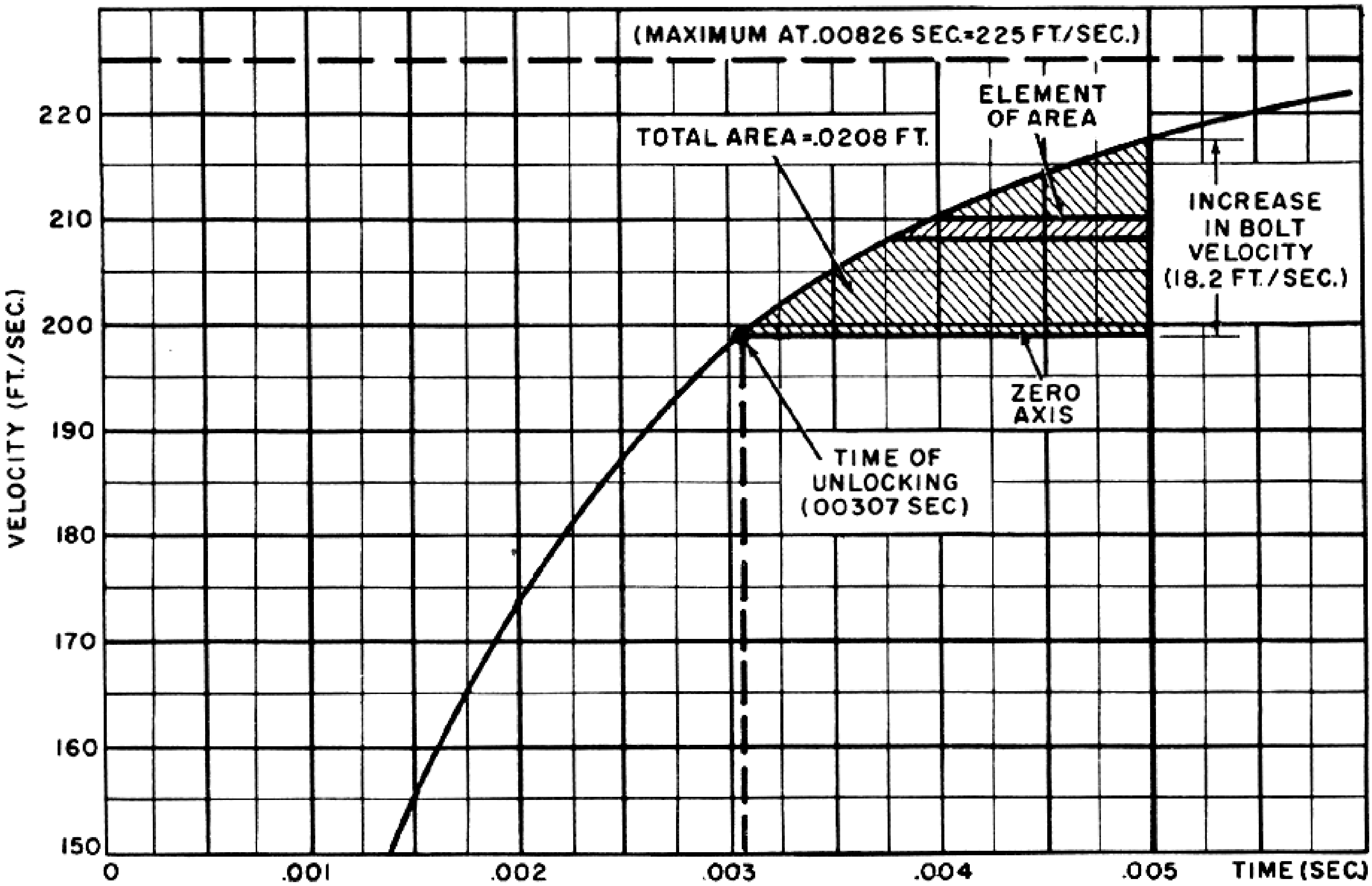


Figure 2–22. Changes in Bolt Velocity Imparted by Blowback and Determination of Time for Unlocking.

curve will show the momentum of these parts at any instant or, since the momentum is equal to the applied impulse, the curve will also show the impulse applied up to any instant. Now, after unlocking occurs, the impulse shown by this curve will be applied in changing the velocity of the bolt and therefore it is possible to divide each ordinate of the impulse curve by the bolt mass to obtain the new curve shown in fig. 2–22. (Only a portion of the vertical scale is shown in order to produce the significant portion of the curve in a large size.) The actual velocity values shown by this curve are meaningless but between any two values of time, the curve does show accurately what *change* in bolt velocity would be produced by the impulse.

Having the curve of fig. 2–22, it is only necessary to determine where to place the zero velocity axis so that the area between this axis and the curve up to 0.005 second is equal to 0.250 inch or 0.0208 foot. (Since this is a velocity-time graph, areas under the curve represent displacement.) The zero axis can be located quite simply by drawing a line along the 0.005-second ordinate and measuring the area between this line and the curve, taking the elements of area as shown in the figure and working downward until the area is equal to 0.0208 foot. The abscissa of the point where the line bounding the lower limit of this area intersects the curve is the required time of unlocking (0.00307 second). Ordinates measured above the lower bounding line are equal to the free recoil velocity, with respect to the barrel, imparted to the bolt by blowback. The curve shows that the gain in free bolt velocity between the time of unlocking and 0.005 second is 18.2 feet per second.

It should be noted that although this computation neglects the retarding effect of the bolt driving spring, the resulting error is extremely small and entirely insignificant.

The data shown in fig. 2–22 are used in the computations for completing the bolt motion curves up to 0.005 second. The manner in which the data are used is explained in paragraph 4 in connection with the plotting of the theoretical time-travel and time-velocity curves before acceleration.

3. *Selection of spring characteristics*

Since the only purpose of the barrel return spring and the bolt driving spring is to assist the barrel and bolt to return to battery and neither spring is required to absorb all of the recoil energy of the parts, the values of the forces exerted by the springs are not critical. For this reason, the characteristics of these springs may be selected more or less arbitrarily.

The barrel assembly is assumed to weigh 45 pounds and to return this relatively heavy mass quickly to battery a fairly strong spring is indicated. As will appear, such a spring will not have any great effect on the recoil motion and therefore the resistance of the spring can be made quite high. On this basis the initial compression of the spring will be selected as 250 pounds and the spring constant as 300 pounds per inch.

The bolt spring will be made very light so that it will not offer a high retardation to the five-pound bolt. An initial compression of 25 pounds and a spring constant of 10 pounds per inch will provide adequate force for assisting the closing of the bolt.

4. *Theoretical time-travel and time-velocity curves before acceleration*

Because of the complexities resulting from the multiplicity of actions during the recoil and counter-recoil movements in a short recoil gun, it is not practical to attempt to derive analytically expressions for the time to recoil and time to counter-recoil. Also, such derivations would be extremely complicated unless it is assumed that the initial kinetic energy is transferred instantaneously to the recoiling parts, ignoring the detailed effects which occur during the action of the powder gas pressures. However, in high-rate-of-fire guns employing the short recoil system, the time of action of the powder gases is extremely significant and must be given due consideration in plotting the bolt motion curves. A detailed analysis of this type is particularly important because most of the critical actions and high accelerations occur during the progress of the propellant explosion and it is therefore highly desirable to determine what motion characteristics may be expected in the initial portion of the recoil stroke.

Since the effects of the powder gas pressure can not be expressed by simple equations, a special method is employed to account for these effects in plotting the bolt motion curves. The method consists essentially of first plotting a curve of free recoil velocity versus time and then subtracting from each ordinate of this curve the velocity loss resulting from the retarding effects of the springs.

The curve showing the velocity of free recoil versus time for the time interval before unlocking was developed previously and is shown in fig. 2–21. This curve will be used to illustrate the following description of the method.

To determine the retarding effects of the springs, use is made of the law expressed by the equation:

$$F\,dt = M\,dv \tag{2–22}$$

This law states that the change in the momentum of a mass is equal to the applied impulse (the product of the force and the time for which it is applied). Solving for dv gives:

$$dv = \frac{F\,dt}{M}$$

To obtain the variation of the change in velocity with respect to time, this expression is integrated.

$$v = \int_0^t \frac{F\,dt}{M} = \frac{1}{M}\int_0^t F\,dt \tag{2–23}$$

In accordance with equation 2–23, the retarding effect of a force on a given mass can be determined as follows:

1. Plot a curve showing the variation of the force with respect to time.
2. Measure the area under the curve between $t=0$ and some time t_1.
3. Divide the measured area by the mass. This gives the ordinate of the retardation curve for the time t_1.
4. Repeat steps 2 and 3 for other values of t and plot the retardation curve.

Applying this procedure using the mass of the recoiling parts and the combined resistance of the barrel return spring and the bolt driving spring produces a curve showing the loss in recoil velocity resulting from the action of the springs up to the time of unlocking. Since the free recoil velocity curve shows the gain in velocity resulting from the thrust of the powder gases, the difference between the curves will be the net recoil velocity, or in other words the velocity of retarded recoil.

The foregoing method would be very simple if the retarding force were constant or if the variation of this force with respect to time were known. However, when the force varies with recoil travel as it does with the springs assumed for purposes of analysis, a difficulty is encountered. In order to plot a graph showing the variation of the retarding force with respect to time, it is necessary to have a curve showing the variation of the recoil travel with respect to time, and the latter curve is one of those which yet remain to be determined.

This difficulty can be overcome by employing a process of successive approximation. While the powder gas pressures are acting, the loss in velocity resulting from the retarding effect of the springs will be relatively small and will be almost entirely due to the constant effect of the initial compression. The varying force due to the spring constant during this interval of time will almost certainly be negligible but, if necessary it can be approximated very closely.

The procedure for plotting the velocity and travel curves for the time before unlocking occurs is as follows:

1. Plot the curve of free recoil velocity versus time (fig. 2–22).
2. The loss in velocity due to the initial compression of the springs is equal to:

$$\frac{(F_{o_1}+F_{o_2})\ t}{M_r}$$

   Determine the velocity loss for various values of t, subtract each from the corresponding ordinate of the free recoil velocity curve and draw a curve through the resulting points. If the effect of the spring constant proves to be negligible, this curve is the retarded velocity curve.
3. Integrate under the curve drawn in step 2 to obtain the displacement curve.
4. Assume that the curve drawn in step 3 represents the actual time-travel curve and use this curve to determine the retardation due to the spring constant. (Use the combined spring constant for the barrel return spring and the bolt driving spring, K_1+K_2.) Ordinarily, it will be found that this retardation is so small that it will not have any effect worthy of consideration.
5. In the event that the retardation determined in step 4 is sufficient to affect the velocity, use it to modify the curve drawn in step 2 and then integrate under the new curve to obtain a corrected displacement curve.
6. Steps 4 and 5 can be repeated as often as is necessary until no significant change occurs in the displacement curve. Actually, this process of successive approximation should never be necessary and satisfactory results should be obtained in the first three steps or at least in the first five steps.

Fig. 2–23 shows the curves obtained for the gun of the example for the interval before unlocking. The total loss in velocity due to the combined effect of the initial compressions of the springs during this interval (0 to 0.00307 second) is:

$$V=\frac{(F_{o_1}+F_{o_2})t}{M_r}=\frac{(250+25).00307\times 32.2}{50}$$

$$=.543\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the combined effect of the spring constants as determined by the method of step 4 is only about 0.115 feet per second. The final curves shown in fig. 2–23 are the result of performing step 3. Since the velocity loss due to the effect of the spring constants is so small, step 5 need not be taken.

After unlocking occurs, the bolt and the barrel are independent of each other and each is affected only by its own spring. Since the bolt is unlocked, the barrel is no longer affected by the pressure of the powder gases and therefore its free recoil characteristic is to continue moving at the same velocity it had at the instant of unlocking. This is indicated in fig. 2–24 by the fact that the free barrel velocity curve after unlocking is a horizontal line.

The retarding effects of the barrel return spring and bolt driving spring up to the time of acceleration (0.005 second) are determined by the same general method as used before unlocking. However, at the instant of unlocking, the springs have been compressed 0.0414 foot (fig. 2–23). This means that in consideration of the period after unlocking, the initial compression of each spring must be increased by the effect of its spring constant for this deflection. That is:

$$F_{o_1}=250+.0414\times 300\times 12=250+149=399\text{ (lb.)}$$

and;

$$F_{o_2}=25+.0414\times 10\times 12=25+5=30\text{ (lb.)}$$

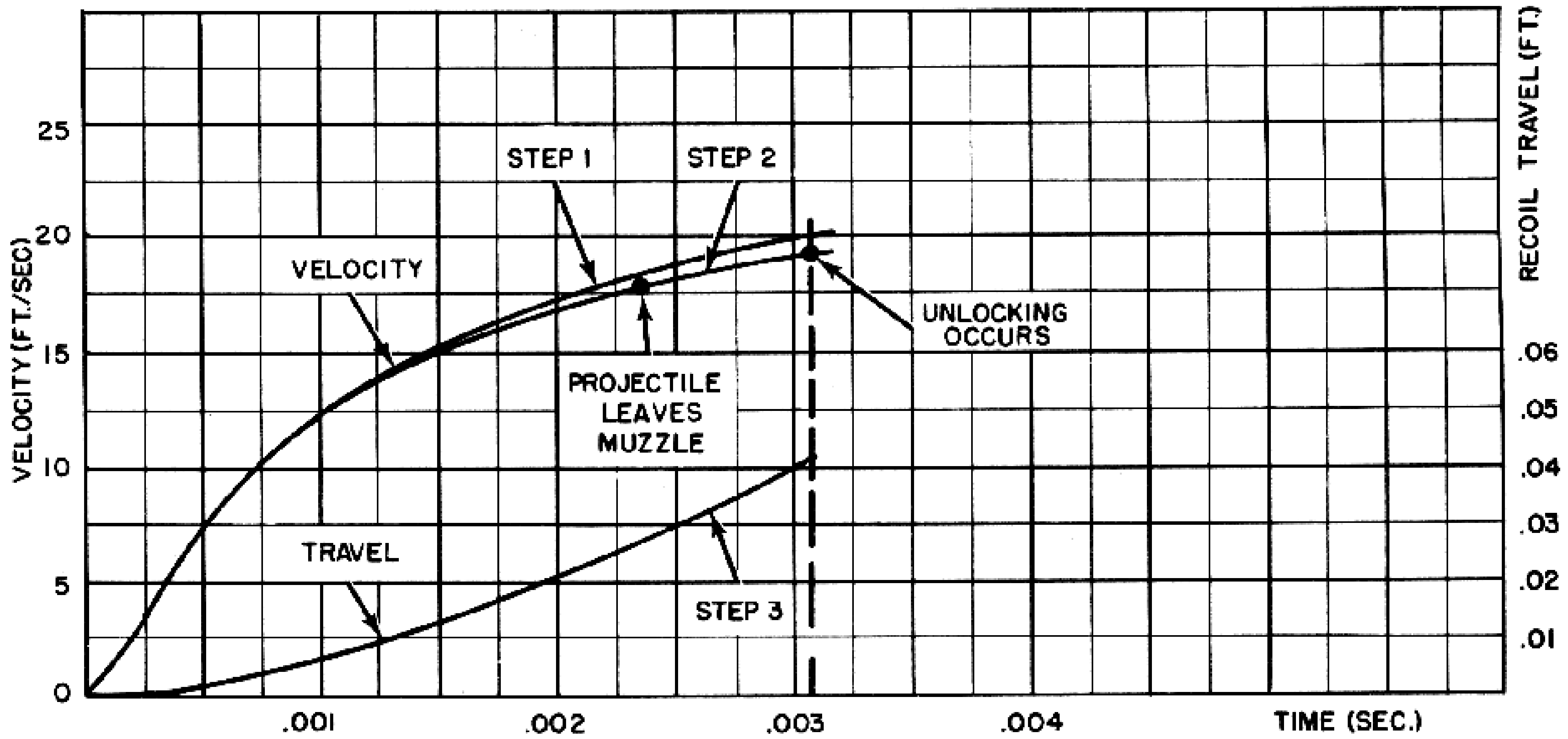


Figure 2–23. Development of Time-Travel and Time-Velocity Curves for Period Before Unlocking.

The total loss in barrel velocity due to the corrected initial compression of the barrel return spring during the interval between unlocking at 0.00307 second and the start of acceleration at 0.005 second (an interval of .005—.00307=0.00193 second) is:

$$V=\frac{F_{o_1}}{M_1}t=\frac{399\times.00193\times32.2}{45}=.550\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the effect of the spring constant as determined by the method of step 4 is only about 0.089 foot per second. The final barrel motion curves shown in fig. 2–24 are the result of performing steps 2 and 3. Since the velocity loss due to the effect of the spring constant is so small, it is not necessary to take step 5.

Having the final barrel motion curve for the interval between unlocking and the start of acceleration, it is now possible to draw the free bolt velocity curve for this interval. This portion of the free bolt velocity curve is plotted from the data shown in fig. 2–22 by adding to the ordinates of the retarded barrel velocity curve the corresponding ordinates of the curve in fig. 2–22, measuring these ordinates above the zero axis. The resulting free bolt velocity curve is shown in fig. 2–24.

The retarding effect of the bolt driving spring is determined by the same methods employed for the barrel motion curves. The loss in bolt velocity due to the corrected initial compression of the bolt driving spring is:

$$V=\frac{F_{o_2}}{M_2}t=\frac{30\times.00193\times32.2}{5}=.373\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the effect of the spring constant as determined by the method of step 4 is, in this case, only about 0.053 foot per second. The final bolt motion curves shown in fig. 2–24 are the result of performing steps 2 and 3. Since the velocity loss due to the effect of the spring constant is so small, it is not necessary to perform step 5. Note that the displacement between the barrel at bolt at 0.005 second (as indicated by the travel curves) is 0.0208 foot or 0.250 inch as required. The retarded velocity curves for the bolt and barrel show that at the instant acceleration starts, the velocity of the bolt is 36.7 feet per second and the velocity of the barrel is 18.5 feet per second.

5. *Motions during period of bolt acceleration*

When the cycle of operation has progressed for 0.005 second as described up to this point, the residual pressure has decreased to the assumed safe operating limit of 750 pounds per square inch and it is now possible to increase the velocity of the bolt without danger of rupture of the cartridge case. It should be noted that at 0.005 second, the residual powder gas pressure has not yet reached zero and

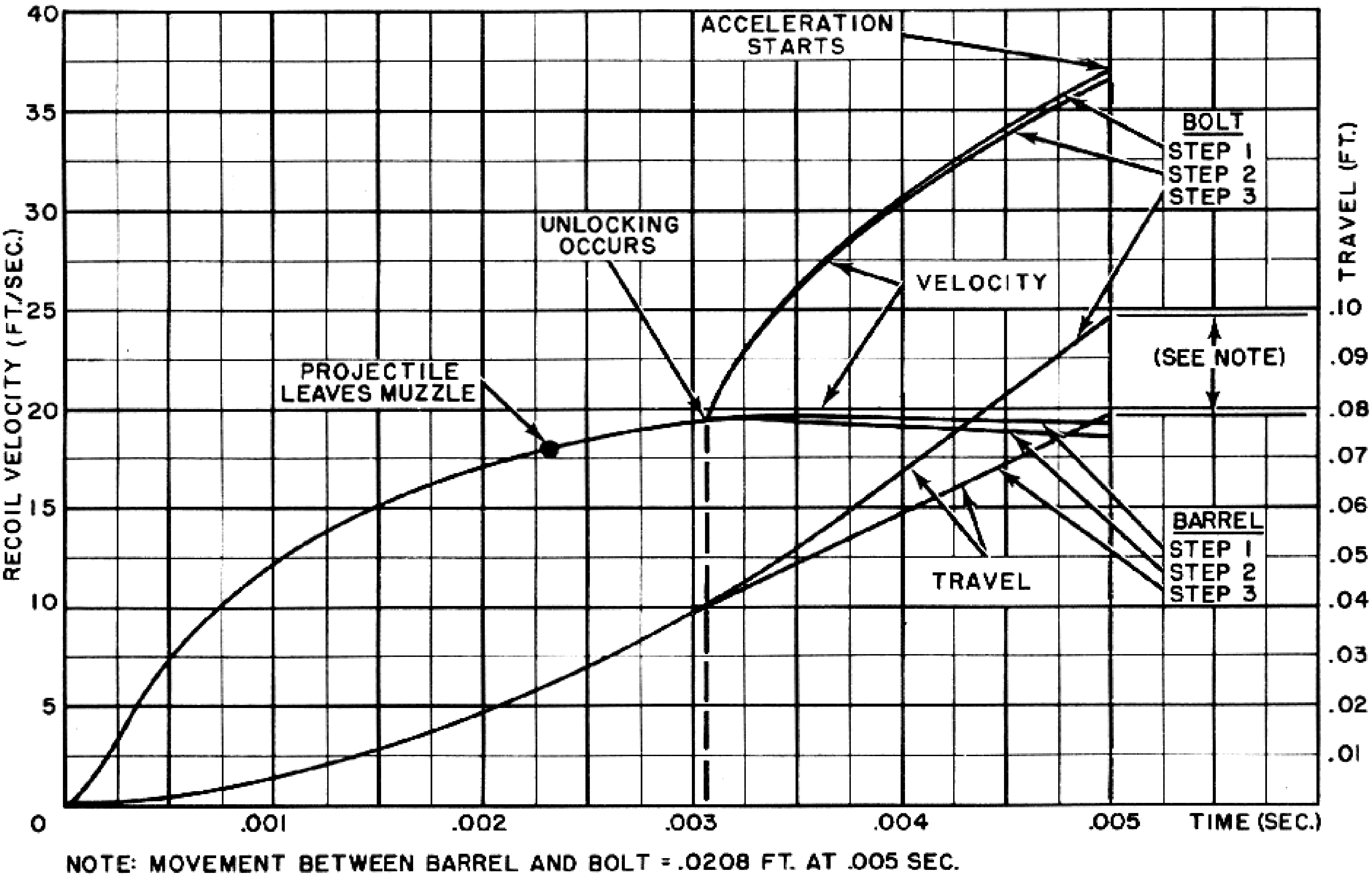


Figure 2–24. Development of Time-Travel and Time-Velocity Curves for Period Between Unlocking and Start of Acceleration.

therefore there will be some small additional blowback effect which will occur simultaneously with the action of the accelerator.

The first point to consider is the amount of kinetic energy available from the moving mass of the barrel. Since the barrel weighs 45 pounds and at 0.005 second is moving at a velocity of 18.5 feet per second, the kinetic energy possessed by the barrel mass is:

$$KE=\frac{1}{2}MV^2=\frac{1}{2}\times\frac{45}{32.2}\times18.5^2$$
$$=239 \text{ (ft. lb.)}$$

These 239 foot pounds of barrel energy do not constitute all of the energy which can be utilized to accelerate the bolt because some work will be done on the bolt as the result of the force exerted by the residual pressure. Although this force can be determined at any instant from the graph of residual pressure versus time (fig. 2–17), it is not possible at this time to determine the work done by this force because the bolt displacement for the interval of acceleration is not yet known. However, the distance moved by the bolt during the 0.003 second for which the residual pressure will continue to act will certainly be quite short and the average force exerted by the powder gas pressure during this movement will also be relatively small. These conditions indicate that the work done on the bolt by the last portion of the residual pressure will be of little consequence when compared with the 239 foot pounds of energy available from the barrel mass. A rough estimate indicates that the work which can be done on the bolt by the remaining blowback action will be in the order of only 10 or 15 foot pounds. Since this amount of work is so small, little error is likely to occur if it is assumed for the present that the energy available for acceleration is 250 foot pounds.

It is of interest to consider what the result would be if the efficiency of energy transfer during acceleration were 100 per cent (that is, if all of the available energy could be transferred to the bolt without any loss). At the instant acceleration starts, the velocity of the bolt is 36.7 feet per second. Since the bolt weighs 5 pounds, the kinetic energy possessed by the bolt is:

$$KE=\frac{1}{2}MV^2=\frac{1}{2}\times\frac{5}{32.5}\times 36.7^2$$
$$=104.5 \text{ (ft. lb.)}$$

Assuming that all of the available 250 foot pounds of energy is transferred to the bolt, the kinetic energy of the bolt after acceleration will be:

$$KE=104.5+250=354.5 \text{ (ft. lbs.)}$$

Having this kinetic energy, the velocity of the bolt after acceleration would be:

$$V=\sqrt{\frac{2(KE)}{M}}=\sqrt{\frac{2\times 354.5\times 32.2}{5}}$$
$$=66.7\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

Since the bolt velocity before acceleration was 36.7 feet per second, this represents a velocity gain factor of 1.815 times. It should be realized that this velocity gain factor is the theoretical maximum based on a 100 per cent efficiency in transferring the available energy from the barrel to the bolt and therefore the bolt velocity in this particular design could never exceed 66.7 feet per second regardless of what type of accelerating device is used.

In actual practice, an energy transfer efficiency of 100 per cent can not be attained because there will always be unavoidable losses which decrease the amount of energy transferred. One of these losses is due to the fact that it is impractical to attempt a design in which the action of the accelerator alone brings the barrel to a complete stop. Therefore there will always be some energy remaining in the barrel after the accelerating function is completed. There will also be losses due to friction, impact, and extraction of the cartridge case and some energy will be absorbed by the barrel return spring and bolt driving spring. Without having a definite mechanical design to work with and in the absence of detailed design data, there is no positive way to arrive at accurate specific values for these losses and accordingly it will be necessary here to select arbitrarily values which are reasonable practical estimates. Let it be assumed that residual barrel velocity after acceleration is 6 feet per second and that the other incidental losses amount to 50 foot pounds.

On this basis, the residual kinetic energy in the barrel after acceleration will be:

$$KE=\frac{1}{2}MV^2=\frac{1}{2}\times\frac{45}{32.2}\times 6^2=25.1 \text{ (ft. lbs.)}$$

Subtracting this energy and the incidental losses of 50 foot pounds from the available 250 foot pounds of energy gives the energy actually transferred as 175 foot pounds. Adding this energy to the initial kinetic energy of the bolt (104.5 foot pounds) produces a final kinetic energy of 280 foot pounds in the bolt. With this kinetic energy the final velocity of the bolt after acceleration will be:

$$V=\sqrt{\frac{2(KE)}{M}}=\sqrt{\frac{2\times 280\times 32.5}{5}}$$
$$=60\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The velocity gain factor is now:

$$\frac{60}{36.7}=1.63$$

The efficiency of energy transfer is:

$$\frac{175}{250}\times 100-70 \text{ percent}$$

This appears to be a value which reasonably could be expected under practical conditions.

Having established the fact the action of the accelerator will speed the bolt up from 36.7 feet per second to 60 feet per second (a velocity gain of 24.3 feet per second) and will slow the barrel down from 18.5 feet per second to 6 feet per second (a velocity loss of 12.5 feet per second), the problem now is to design the accelerator which will produce these changes in velocity. The major consideration in this design is to arrange the accelerator mechanism so that the required transfer of energy will be accomplished without heavy shock or excessive acceleration forces and the velocities of the barrel and bolt will vary smoothly.

The basic characteristics of the motions which occur during the action of the accelerator will depend on whether the accelerator is of the catapult spring type or of the lever or cam type. Either type may be analyzed readily, but for purposes of the present discussion, it will be assumed that the accelerator to be used is of the lever type.

The first point of interest in designing the accelerator is the time which should be allowed for the completion of the transfer of kinetic energy. From the standpoint of attaining a high rate of fire, it would be desirable to accomplish the transfer rapidly for two reasons. First, this would quickly bring the bolt to a high velocity, thus decreasing the time required for the completion of the bolt movement. Second, the barrel velocity would be reduced quickly with the result that the rearward travel of the barrel would be relatively small. This would be advantageous because it would reduce the distance the bolt must move to produce an opening large enough to permit feeding.

On the other hand, quick transfer of kinetic energy requires a high acceleration of the bolt and rapid deceleration of the barrel and the resulting inertia reactions will cause large forces to act on the accelerator mechanism. Increasing the time allowed for the transfer of energy will reduce the acceleration and the forces on the mechanism but will also result in decreasing the rate of fire.

Considering the above factors, the time allowed for energy transfer should be some moderate value; not so short that excessive forces will be produced and not so long that the rate of fire will be reduced seriously. For the gun of the example a value of 0.004 second should be acceptable.

After the selection of the length of time which will be allowed for the action of the accelerating device, the next step is to decide what motion characteristics are desired for this interval. In order for the energy transfer to occur with minimum shock, it would be reasonable to arrange the mechanism so that the forces on the parts will at first be zero and then will increase smoothly to the higher values required to produce rapid acceleration and deceleration. Such a force variation will result in a condition in which the displacements and velocities, as well as the accelerations, will vary in a smooth manner.

Except for the relatively small amount of kinetic energy expended in compressing the bolt driving spring and barrel return spring during the period of acceleration, and except for the small gain in energy due to the remaining blowback action and the slight incidental losses in the operation of the mechanism, the sum of the kinetic energies contained in the recoiling system consisting of the barrel and bolt will remain constant. Therefore, the characteristics of the bolt and barrel movements will depend mainly on the properties of the accelerating mechanism. Since the accelerating mechanism can be designed to establish the relationships between the movements of the recoiling parts, these movements can be controlled as desired. In other words it is possible to approach the design of the accelerator by first specifying the desired variation in barrel velocity. The corresponding velocity variations for the bolt can then be determined by taking into consideration the kinetic energy obtained from the barrel, the energy imparted by the remaining blowback action, and the energy lost in operating the gun mechanism. The variations of bolt velocity and barrel velocity being known, the accelerating device can be designed so that these velocity variations will be produced.

The basic procedure by which the accelerator may be designed is as follows: First, the desired characteristics of the barrel velocity are established by drawing, arbitrarily, a suitable barrel velocity curve for the interval of acceleration. The shape of the curve should be similar to that shown in fig. 2–25 so that the deceleration (and consequently, the force required to produce the deceleration) will start at zero and increase smoothly to the maximum value at the end of the interval. Note that the barrel velocity curve selected for the example indicates a residual velocity of 6 feet per second at the end of the acceleration interval.

The barrel velocity curve is now used to compute, for each instant, the decrease in barrel kinetic energy as indicated by the decrease in barrel velocity. This computation is made on the basis of the relationship:

$$\Delta\ (KE) = (KE)_o - \frac{1}{2}\, Mv^2$$

which states that the change in kinetic energy up to any instant is equal to the initial kinetic energy minus the kinetic energy resulting from the velocity v which exists at that instant. For the barrel of the gun of the example:

$$\begin{aligned}\Delta\ (KE) &= 239 - \frac{1}{2} \times \frac{45}{32.2}\, v^2 \\ &= 239 - .700\ v^2\end{aligned}$$

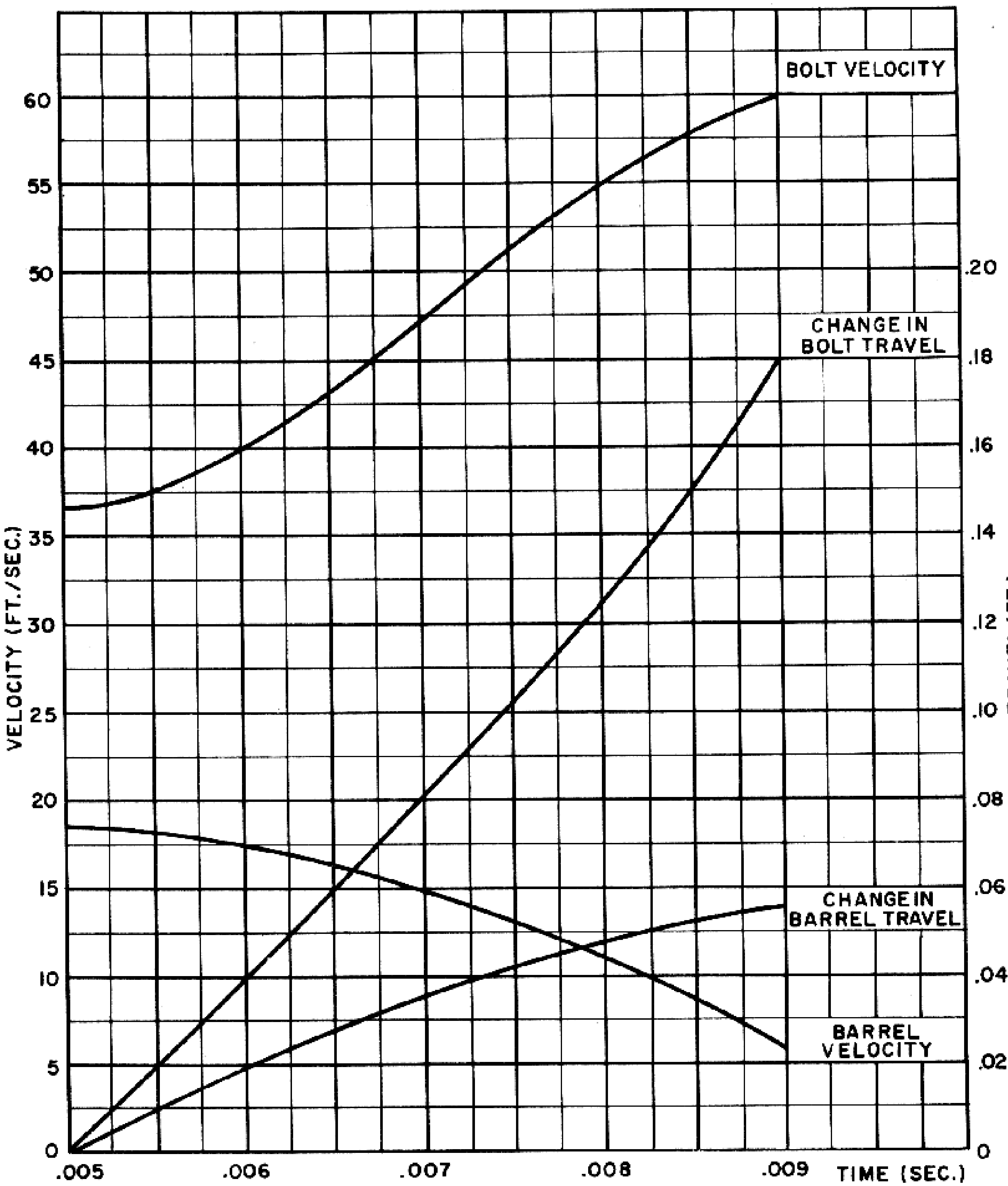


Figure 2–25. Effect of Acceleration on Travel and Velocity of Barrel and Bolt.

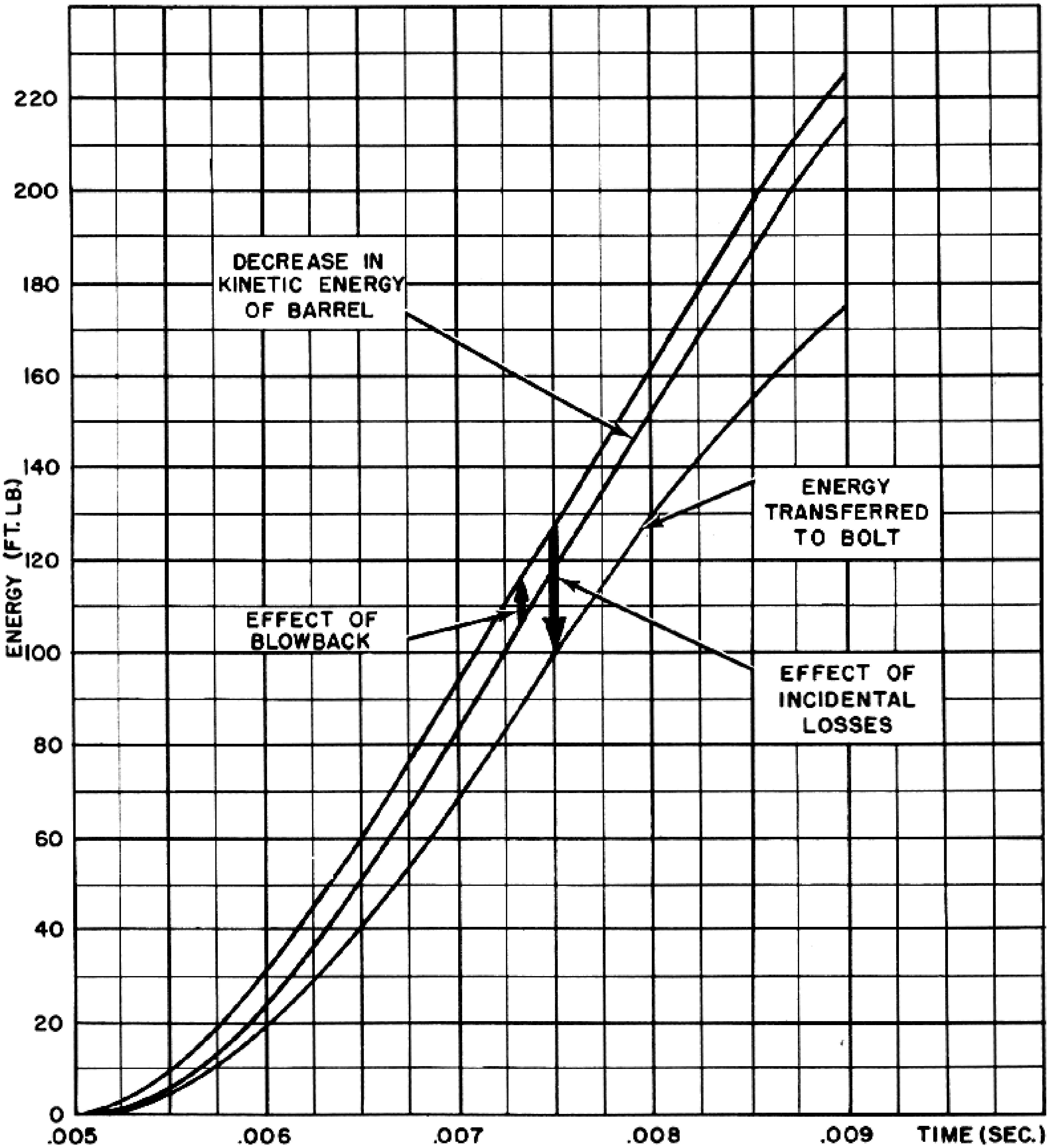


Figure 2–26. Energies Affecting Motion of Bolt During Action of Accelerating Lever.

Fig. 2–26 shows the curve obtained by using this relationship to determine the decrease in the kinetic energy of the barrel. This curve does not represent the energy transferred to the bolt because the remaining blowback action and the incidental losses still must be considered. The effects of the blowback action and of the incidental losses can not be determined precisely at this time because the bolt motion has not yet been determined. However, both effects are not very great and can be estimated with a sufficient degree of accuracy to insure small error in the computations. In an actual design problem, the result obtained by using these estimates could be considered to be a first approximation which can be refined if necessary. It was previously estimated that the effect of blowback up to 0.00826 second (when the residual pressure becomes zero) would perform a total work of 11 foot pounds on the bolt. Fig. 2–26 shows how this energy is assumed to be added to the kinetic energy obtained from the barrel. The incidental losses, which have been estimated as 50 foot pounds, are mainly due to the retardation offered by the barrel spring and bolt driving spring. These losses must be subtracted from the total available energy and for purposes of this analysis, it will be sufficiently accurate to assume that the losses occur at a constant rate over the interval of acceleration. That is, for the sample conditions, since there is a total loss of 50 foot pounds of kinetic energy over the interval of 0.004 second, it may be assumed that energy is lost at the constant rate of 50/.004=12,500 foot pounds per second during the acceleration process. The lowest curve in fig. 2–26 shows the result of this subtraction and gives the energy transferred to the bolt at any instant.

The values of the energy transferred to the bolt, as shown in fig. 2–26, are used to determine the bolt velocity curve shown in fig. 2–25 as follows: The velocity of the bolt at the start of acceleration is 36.7 feet per second and its kinetic energy at that time has been computed to be 104.5 foot pounds. For any instant during acceleration, this value is added to the amount of energy transferred to the bolt up to that instant (from fig. 2–26). This total kinetic energy of the bolt is then used to compute the bolt velocity according to the relationship:

$$V=\sqrt{\frac{2g\,(KE)}{W}}$$

For the conditions of the example, W=5 pounds and therefore:

$$V=\sqrt{\frac{2\times32.2(KE)}{5}}=\sqrt{12.88(KE)}$$

The next step is to integrate under the velocity curves to obtain curves showing the changes in barrel travel and bolt travel during the period of acceleration. These travel curves are also shown in fig. 2–25. The values given by these curves are added to the barrel travel and bolt travel which exist at the beginning of the acceleration in order to extend the time-travel curves to 0.009 second. Fig. 2–27 shows the complete time-travel and time-velocity curves up to this point.

All of the data of interest in designing the accelerator lever may either be found in fig. 2–27 or may be derived directly from this figure. Fig. 2–28 shows data which may be used for the accelerator design. The barrel deceleration and bolt acceleration curves were obtained by measuring the slopes of the velocity curves. The curves showing the relative travel and relative velocity between the barrel and bolt were derived by noting the difference between the heights of the ordinates of the travel and velocity curves. The velocity ratio curve was obtained by dividing the bolt velocity by the barrel velocity.

Having the data shown in figs. 2–27 and 2–28, the shape of the accelerating lever can be plotted by the conventional methods employed for cam layout. In designing the lever, it should be noted that the lever ratio which exists at any instant must be equal to the velocity ratio shown for that instant in fig. 2–28. Also, to minimize wear on the contact surfaces, these surfaces should be designed as far as possible to have a rolling action, and sliding contact at the surfaces should be held to a minimum. Since the details of the layout process will depend largely on the space requirements of a particular design and because it is not the intent in this publication to describe conventional machine design procedures, no attempt will be made here to explain the layout of the lever. The general shape and action of a lever for one particular design is shown in fig. 2–15. However, it should be realized that the shape of the lever for a different gun may differ considerably from that shown in fig. 2–15, depending on the arrangement of the mechanism and the desired motion characteristics.

Before the analysis of the accelerating action is concluded, it is interesting to note the magnitude of the forces which act on the barrel and bolt during acceleration. Fig. 2–28 shows that the maximum deceleration of the barrel mass is equal to 5600 feet per second per second (at 0.009 second). Since the barrel weighs 45 pounds, the force required to produce this deceleration is:

$$F=Ma=\frac{45}{32.2}\times5600$$
$$=7820 \text{ (lbs.)}$$

Although this force is large, it is not excessive.

The maximum acceleration of the bolt occurs at 0.00705 second and is equal to 8000 feet per second per second. Since the bolt weighs 5 pounds, the force required to produce this acceleration is:

$$F=Ma=\frac{5}{32.2}\times8000$$
$$=1245 \text{ (lbs.)}$$

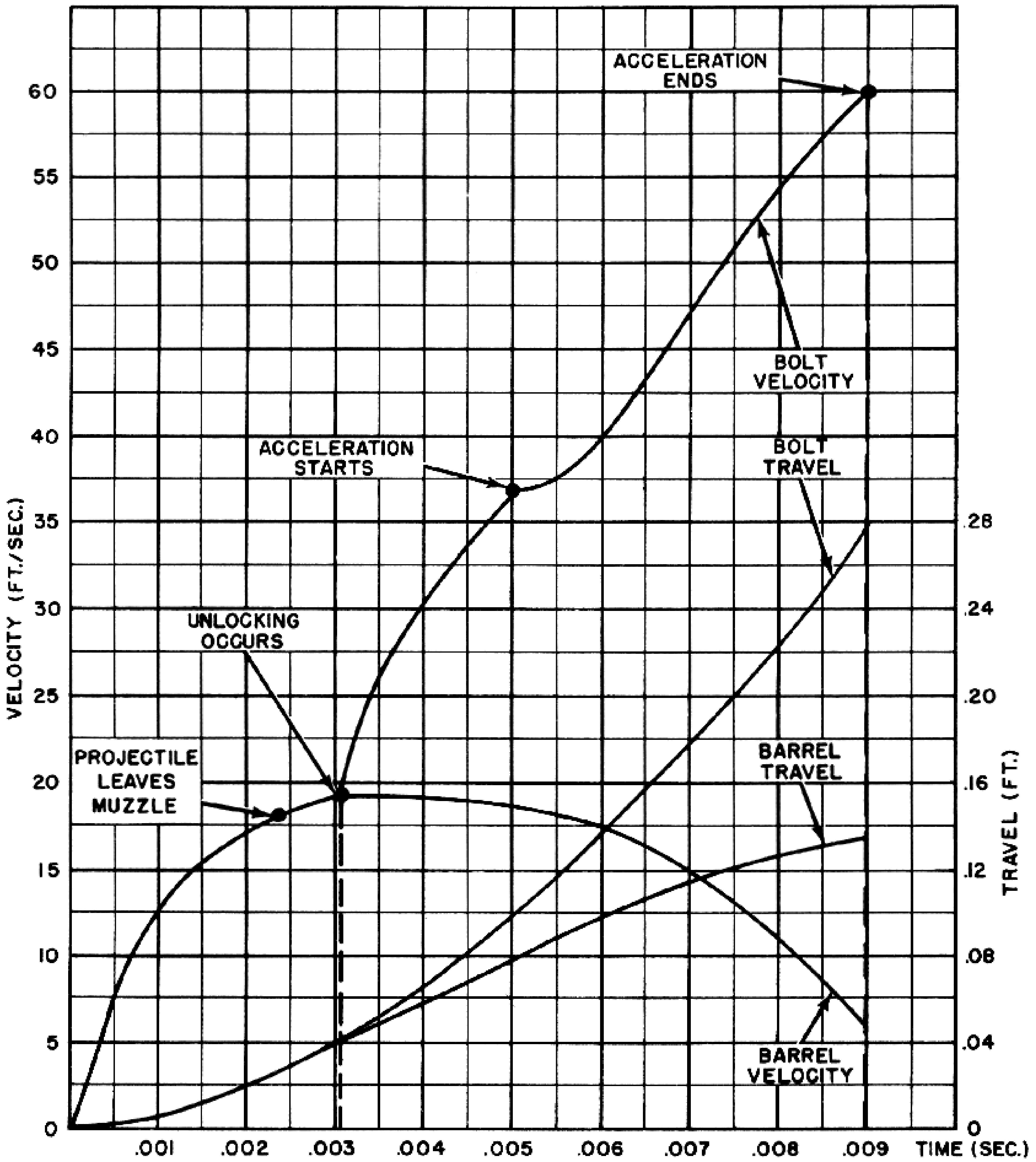


Figure 2–27. Time-Travel and Time-Velocity Curves Up to End of Acceleration Period.

6. *Barrel recoil motion after acceleration*

After the action of the accelerating device has been completed, the barrel must be stopped and latched, and the bolt must continue its rearward travel until the opening between the barrel and bolt is sufficient to permit feeding of a fresh cartridge.

At the end of the acceleration period, the barrel of the gun used as example is moving at the rate of 6 feet per second. Since the accelerating device is no longer acting, the barrel motion will now be impeded only by the barrel return spring. As shown in fig. 2–27, the barrel travel at 0.009 second is 0.135 foot, and at this displacement the force of the barrel spring is:

$$F_{o_1} + K_1 D = 250 + 300 \times .135 \times 12 = 736 \text{ (lb.)}$$

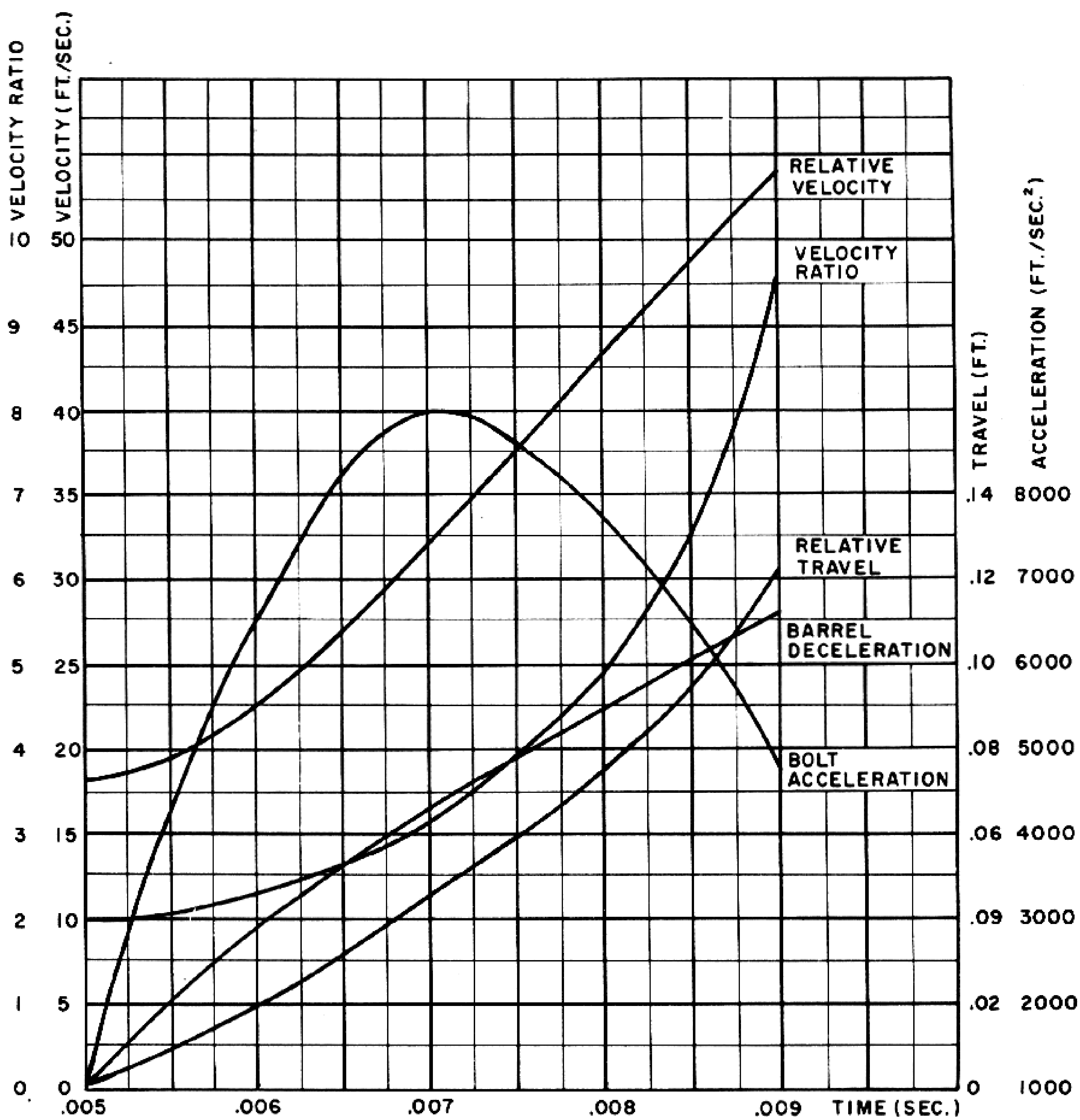


Figure 2–28. Data Pertaining to Action of Accelerating Lever.

For purpose of determining the barrel motion from 0.009 second on, this force of 736 pounds can be considered to be the initial compression of the spring and the retarding effect of the spring can be computed by the same methods used for developing the motion curves for the period before acceleration. The velocity loss due to the initial compression will be:

$$V = \frac{F_0}{M} t = \frac{736 \times 32.2}{45} t$$
$$= 527 t$$

That is, the effect of the initial compression causes the barrel velocity to decrease at the rate of 527 feet per second per second. This loss is shown in fig. 2–29 by the curve designated as step 2. The

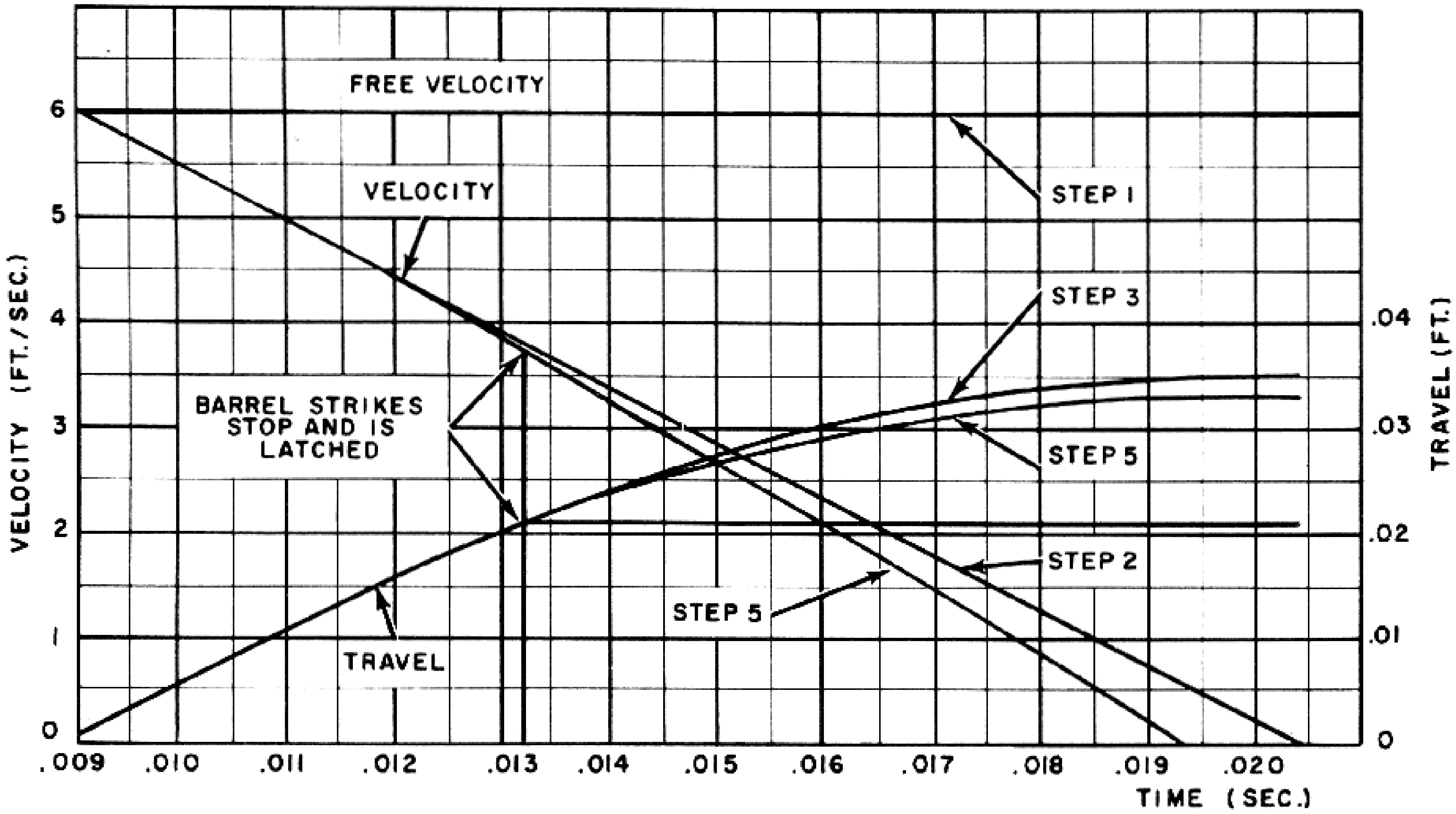


Figure 2–29. Effect of Barrel Return Spring on Barrel Motion After Acceleration.

loss due to the effect of the spring constant is determined by the method of step 4 of the procedure previously described and the modified curves are designated as step 5. Since the performance of step 6 produces a negligible change in the curves, the curves drawn in accordance with step 5 represent the effect of the barrel spring on the barrel motion. The curves show that if the barrel spring alone resists the barrel motion after the acceleration action is completed, the barrel will move an additional 0.033 foot (0.40 inch) and will come to a stop at 0.0193 second. Both this distance and the time are a little too great since it is desirable to stop the barrel as soon as possible. Therefore, a stop will be provided to limit the barrel movement to 0.250 inch (0.0208 foot) after acceleration is completed. As indicated in fig. 2–29, this movement occurs at 0.132 second. At this point the velocity of the barrel is only 3.70 feet per second and therefore has a kinetic energy of only 9.55 foot pounds, indicating that the shock of hitting the stop will be relatively light. After being halted by the stop, the barrel is latched in place so that it remains in its rearmost position. This is shown in fig. 2–29 by the fact that the barrel travel curve is a horizontal line after 0.132 second. The data from fig. 2–29 are used to extend the barrel motion curves on the graph (fig. 2–30) showing the motion curves for the complete cycle.

7. Bolt recoil motion after acceleration

At the end of the acceleration period, the bolt is moving with a velocity of 60 feet per second and its motion from this point on is resisted only by the bolt driving spring. As shown on fig. 2–30, the barrel is latched at a displacement of 0.156 foot (1.875 inches). Since it has been assumed that the opening between the barrel and bolt must be 10 inches in order to permit feeding, the bolt must travel a total of 11.9 inches (0.993 foot) before it is stopped by the backplate buffer. Fig. 2–27 shows that the bolt travel at the end of the acceleration period is 0.280 foot and the bolt must travel an additional 0.173 foot.

The motion of the bolt after acceleration may be determined by the same methods as used for the barrel motion. (Cf. fig. 2–31.) The first step is to draw a horizontal line at 60 feet per second to show the free bolt velocity. At the end of the

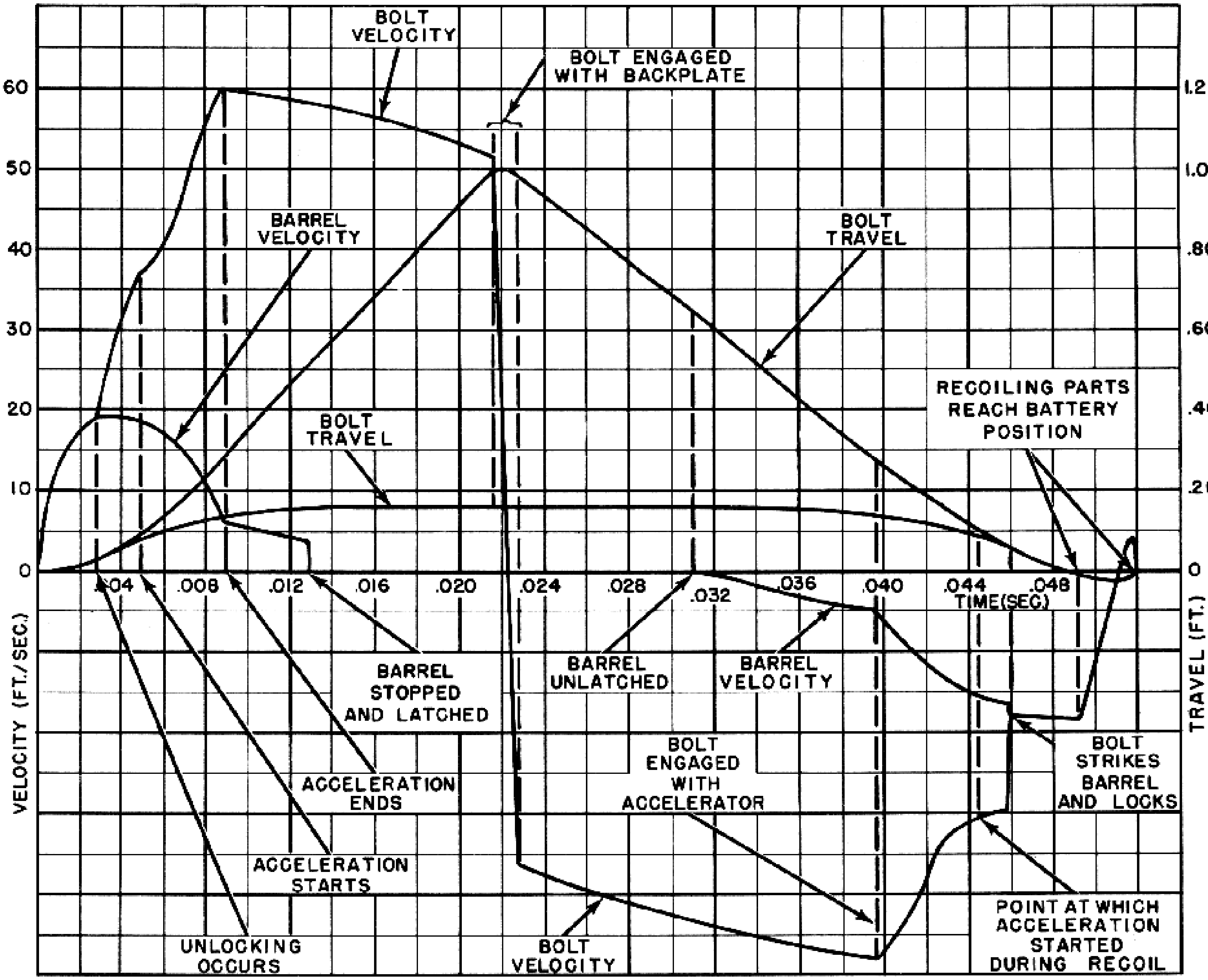


Figure 2–30. Time-Travel and Time-Velocity Curves for Complete Cycle of Operation.

acceleration period, the bolt travel is 0.280 foot and therefore the force on the bolt driving spring must be

$$F_{o_2}+K_2D=25+.280\times10\times12=25+33.6$$
$$=58.6 \text{ (lb.)}$$

Taking this force as the initial compression of the spring for the time after acceleration, the velocity loss due to this force will be

$$V=\frac{F_o}{M}t=\frac{58.6\times32.2}{5}t$$
$$=378\ t$$

The effect of this loss is shown in fig. 2–31 by the curve designated as step 2. The loss due to the effect of the spring constant is determined by the method of step 4 and the modified curves are designated as step 5. The performance of step 6 produces no significant change in the curves and therefore the curves designated as step 5 represent the effect of the bolt driving spring on the bolt motion. The curves show that the bolt reaches the required additional travel of 0.713 foot at 0.216 second and it is at this point that the bolt reaches the backplate buffer. Note that the striking velocity is 51.5 feet per second. The data from fig. 2–31 are used to extend the bolt motion curves in fig. 2–30 up to the time the backplate buffer is struck.

8. *Reversing action of backplate buffer*

The purpose of the backplate buffer is to reverse the bolt motion at the end of the recoil stroke with

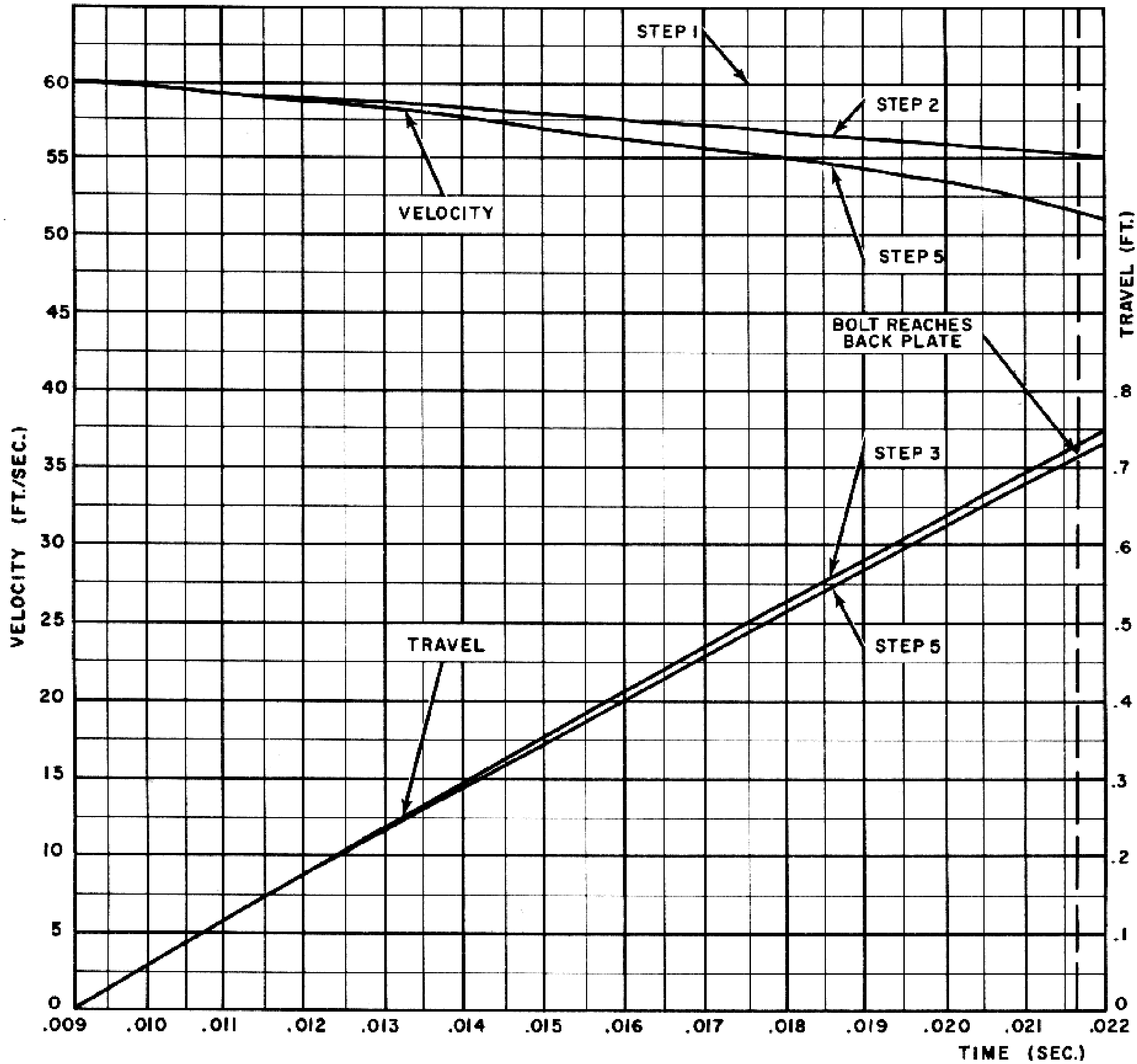


Figure 2–31. Effect of Bolt Driving Spring on Bolt Motion After Acceleration.

the minimum possible loss of time and energy. In the analysis of short recoil it was pointed out that, although the reversing action is accomplished almost instantaneously by causing the bolt to rebound from an extremely stiff elastic member, the impact is accompanied by a loss of energy with the result that the velocity of the bolt after impact will be at best approximately 70 per cent of its striking velocity.

Since the velocity of the bolt before striking the backplate is 51.5 feet per second, the momentum of the bolt is:

$$MV=\frac{5}{32.2}\times 51.5=8.00 \text{ (lb. sec.)}$$

Assuming that the coefficient of restitution of the backplate is .70, the velocity of the bolt after impact will be:

$$V=.70\times 51.5=36.0 \left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The momentum of the bolt will then be:

$$MV=\frac{5}{32.2}\times 36.0=5.77 \text{ (lb. sec.)}$$

Thus, in the reversing action of the backplate, the change in bolt momentum is equal to $8.00+5.77=13.77$ (lb. sec.). If the entire reversing action occurs in 0.001 second, the average force exerted on the backplate must be:

$$\frac{13.77}{.001}=13{,}700 \text{ (lb.)}$$

The striking energy of the bolt is:

$$KE=\frac{1}{2}MV^2=\frac{1}{2}\times\frac{5}{32.2}\times 51.5^2=206 \text{ (ft. lb.)}$$

If it is assumed for purposes of estimation that the backplate offers a constant resistance of 13,700 pounds, the deflection of the elastic member under the impact will be:

$$\frac{206}{13{,}700}=.015 \text{ (ft.) or approximately } 3/16 \text{ (inch)}$$

The data derived in the preceding analysis are used to complete the motion curves of fig. 2–30 for the 0.001 second interval during which the backplate buffer acts.

9. *Motion of bolt after reversal at backplate*

The bolt leaves the backplate with a velocity of 36.0 feet per second, and as it moves in counter-recoil, its motion is aided by the compressed driving spring. This action will continue until the bolt re-engages with the accelerator lever when the bolt displacement from the firing position is equal to 0.28 foot.

The bolt motion under the influence of the driving spring can be determined by essentially the same method previously employed for analyzing the effect of the springs. (Cf. fig. 2–32.) The first step is to draw a horizontal line at minus 36 feet per second to represent the free bolt velocity. In this case, the subsequent procedure must be slightly modified because the action of the spring is aiding the motion of the bolt rather than retarding it. At the start of the return motion of the bolt, the bolt driving spring is compressed 1.008 feet and hence the initial compression of the spring is:

$$F_o+KD=25+10\times 12\times 1.008=25+121$$
$$=146 \text{ (pounds)}$$

The velocity gain due to this force would be:

$$V=\frac{F_o}{M}t=\frac{146\times 32.2}{5}t$$
$$=940\ t$$

The effect of this gain in velocity is shown in fig. 2–32 by the curve designated as step 2. The curve designated as step 3 shows the change in bolt travel that would result from this velocity. Since the bolt is now moving forward, the effect of the spring constant is to decrease the force on the bolt. Therefore, the change in velocity determined by the method of step 4 must be subtracted from the curve obtained in step 2. The modified curves are designated as step 5. Since the change in the travel curve is so slight, it is not necessary to perform step 6 and the curves designated as step 5 represent the effect of the bolt driving spring on the bolt motion.

NOTE: The validity of the foregoing method may be seen by examining the equation expressing the change in velocity of the bolt due to the effect of the driving spring:

$$V=\int_0^t \frac{Fdt}{M}=\int_0^t \frac{F_o+K(D-d)}{M}dt$$

where: D is the total distance the spring is compressed at the start of the forward motion

and d is the forward movement of the bolt from its rearmost position.

$$\Delta V=\frac{1}{2}\int_0^t (F_o+KD)\,dt-\frac{1}{2}\int_0^t (Kd)\,dt$$
$$=\frac{F_o+KD}{M}t-\frac{K}{M}\int_0^t (d)\,dt$$

This is the equation defining the procedure followed in obtaining the curves in fig. 2–32.

Since the bolt leaves the backplate at a displacement of 0.993 foot and the bolt will re-engage the accelerating lever at a displacement of 0.280 foot from the firing position, the bolt must move $.993-.280=.713$ foot up to the time the lever is engaged. Fig. 2–32 shows that the time at which the lever is engaged is 0.0396 second and that the velocity of the bolt at this instant is minus 48.0 feet per second. The data from fig. 2–32 are used to extend

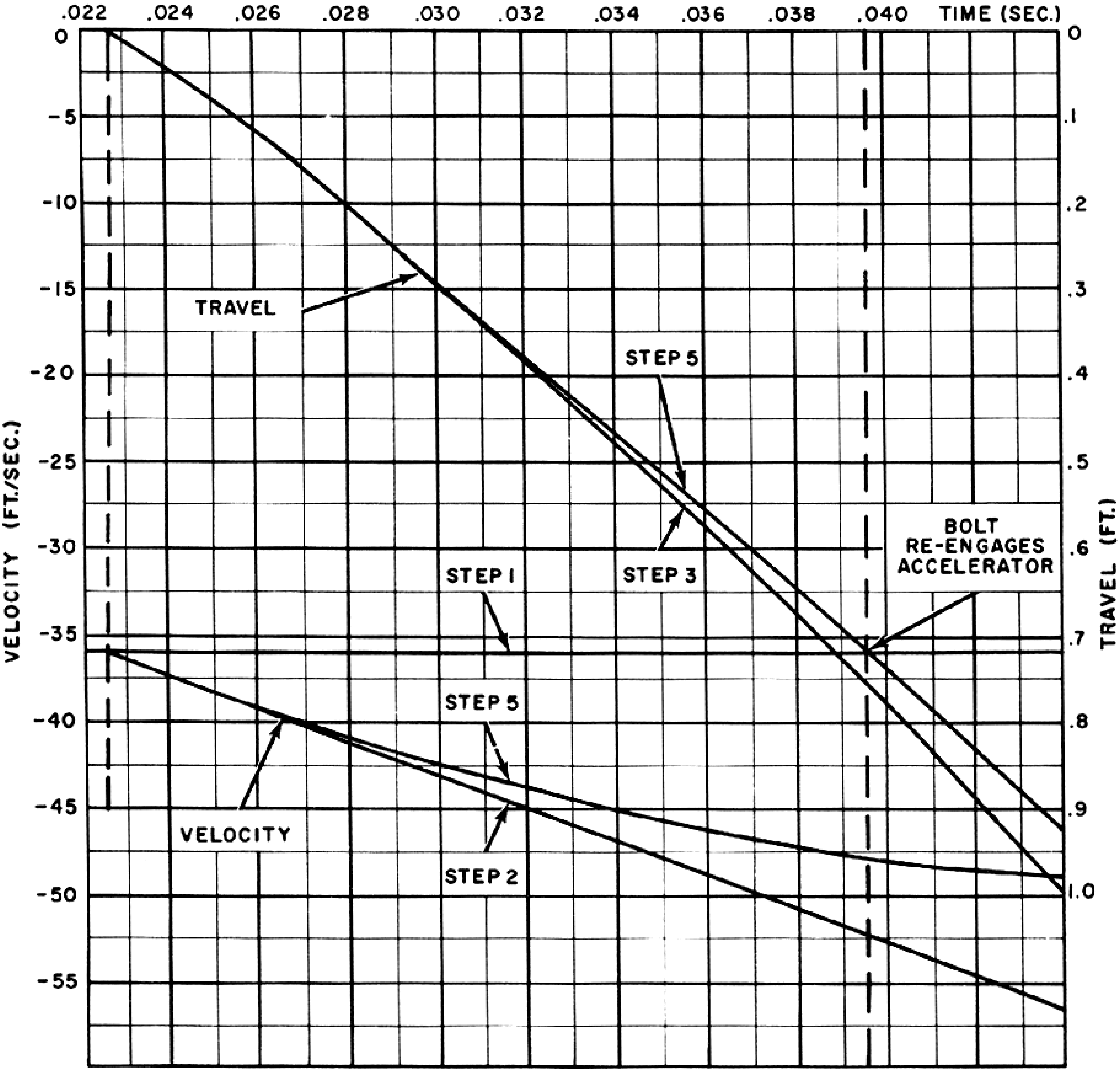


Figure 2–32. Effect of Bolt Driving Spring on Motion of Bolt After Reversal of Motion at Backplate.

the bolt motion curves in fig. 2–30 up to the time that the bolt strikes the accelerating lever.

10. *Motion of barrel before accelerator is reengaged*

In order to give the barrel return spring a chance to start the barrel moving before the bolt reaches the accelerator lever, the barrel should be unlatched while the bolt is still some distance from the point at which the accelerator will be engaged. For smoothest action, the time the barrel is unlatched should be set so that the barrel will move ahead to the position it occupied at the instant that the original acceleration of the bolt was completed. (This position is 0.135 foot from the firing position, as shown in fig. 2–27.) Depending on the configuration of the accelerating lever, it may be necessary to provide a means to insure that the returning bolt will pick up the lever, in order to make certain that the lever will assume its proper position between the bolt and barrel.

At this point in the analysis, the problem is to determine the effect of the barrel return spring on

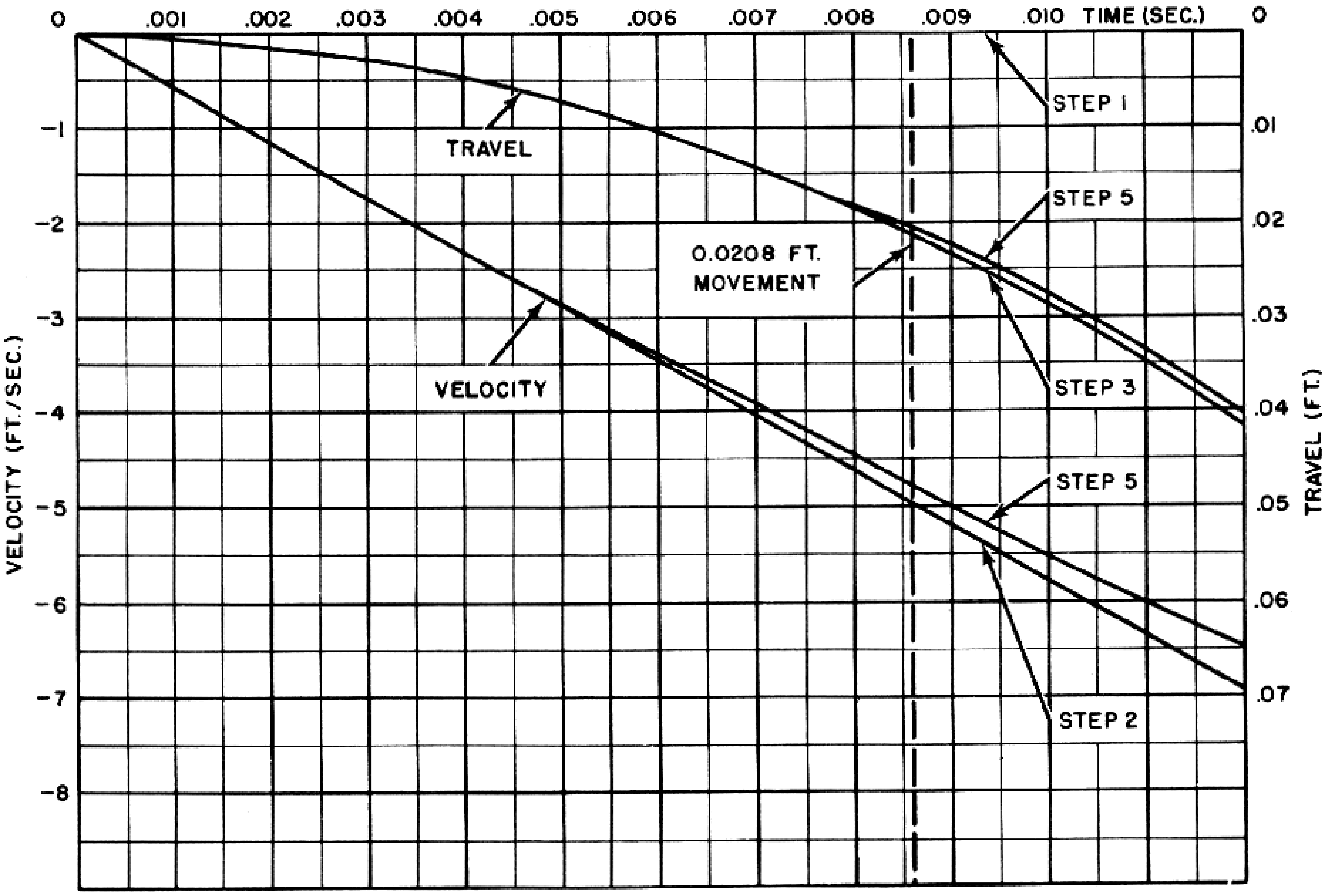


Figure 2–33. Effect of Barrel Spring on Barrel Motion After Unlatching.

the barrel motion and to find the time at which the barrel should be unlatched to satisfy the requirements of the preceding paragraph. The analysis of the barrel return motion is conducted in the same way as for the bolt return motion. (Cf. fig. 2–33.)

In this case, the initial velocity of the barrel is zero and therefore the free-velocity curve coincides with the zero axis. Since the barrel is latched at a displacement from the firing position of 0.160 foot, the initial compression of the barrel return spring is:

$$F_o + KD = 250 + 300 \times 12 \times .156 = 250 + 562$$
$$= 812 \text{ (pounds)}$$

The velocity gain due to this force would be:

$$V = \frac{F_o}{M} t = \frac{8+2\times 32.2}{45} t$$
$$= 580 t$$

This gain in velocity is shown in fig. 2–33 by the curve designated as step 2. The barrel travel which would result from this velocity is shown by the curve designated as step 3. Step 4 is performed by the same method as for the bolt return motion and the modified curves are designated as step 5. The change in the travel curve is so slight that it is not necessary to perform step 6 and therefore the curves designated as step 5 represent the effect of the barrel return spring on the barrel motion.

Since the bolt is latched at a displacement of 0.156 foot and is required to move forward to a displacement of 0.135 foot, the required movement is .156 – .135 = .0208 foot. The curves in fig. 2–33 show that this movement is accomplished in 0.0086 second from the time of unlatching. It has already been specified that the barrel must reach the displacement of 0.135 foot at the instant the bolt reaches a displacement of 0.280 foot, which occurs at 0.0396 second, and therefore the barrel must be unlatched 0.0086 second before this time, or at 0.310 second. The data in fig. 2–33 are used to complete the barrel return motion curves up to 0.0396 second.

11. *Return motion of barrel and bolt during action of accelerating lever*

During the return motion of the barrel and bolt, the effect of the action of the accelerating lever is to transfer kinetic energy from the bolt to the barrel, thus gradually slowing the motion of the bolt and accelerating the motion of the barrel. This action has the beneficial effect of greatly reducing the shock which would occur if the bolt, which is returning at high velocity, merely collided with the rear face of the barrel.

The analysis of the motion of the barrel and bolt during the action of the accelerating lever in counter-recoil is considerably more complicated than for recoil. This is because of the fact that the lever has already been designed and the velocity of the barrel or bolt can no longer be selected as desired but will be determined by the characteristics of the lever.

The analysis during the action of the accelerating lever is conducted on the basis of equating the energies in the system. The energy relationship may be expressed as:

Initial KE+Work done by springs—Energy losses=Barrel KE+Bolt KE.

This relationship, considered together with the fact that the lever causes a known relative motion between the barrel and bolt and also causes a known ratio between their velocities, is used as follows in determining the motions:

At the start of the action of the lever, the velocities of the barrel and bolt are known. That is (from fig. 2–30) $V_1=4.8$ feet per second and $V_2=48$ feet per second. Note that the ratio between these velocities is 10:1 which is the same as the lever ratio of the accelerator at the instant it is engaged. (Cf. fig. 2–28.) This is a desirable condition because it means that the action of the lever will start without a large shock. With the stated velocities, the initial energies in the barrel and bolt will be:

$$\text{For barrel: } E=\frac{1}{2}MV^2=\frac{45}{2\times 32.2}\times 4.8^2$$
$$=16.1 \text{ (ft. lb.)}$$

For bolt:

$$E=\frac{1}{2}MV^2=\frac{5}{2\times 32.2}\times 48$$
$$=178.9 \text{ (ft. lb.)}$$

Therefore, the total initial energy is equal to 16.1+178.9=195 foot pounds.

The energy delivered by the springs can not be determined as a function of time because the relationships between the motions and time have not yet been established. However, it is possible to express the energy delivered by the springs in terms of the distances moved from the start of the accelerating lever action. For a movement from a displacement D_1 to a smaller displacement D_2, the energy delivered by a spring is equal to the average spring force times the displacement.

That is:

$$E=\frac{(F_0+KD_1)+(F_0+KD_2)}{2}(D_1-D_2)$$

(2–24)

$$E=\frac{2F_0+K(D_1+D_2)}{2}(D_1-D_2)$$

During the action of the accelerator, the bolt moves from an initial displacement of 0.280 foot to a final displacement of 0.105 foot and the barrel moves from an initial displacement of 0.135 foot to a final displacement of 0.080 foot. (These values are taken from fig. 2–27.) For the barrel, $F_0=250$ pounds and $K=3600$ pounds per foot. For the bolt, $F_0=25$ pounds and $K=120$ pounds per foot.

Now the bolt travel may be divided into any desired number of increments between 0.280 foot and 0.105 foot and then the corresponding increments of the bolt travel may be found from fig. 2–27 and the results can be tabulated. For later use, the velocity ratio corresponding to each set of increments can be obtained from fig. 2–28 and added to the tabulation. For example, see Table 2–1.

Table 2–1

Increments of Travel and Velocity Ratio

Bolt travel (feet)	Barrel travel (feet)	Velocity Ratio
.280	.135	10.00
.245	.132	6.22
.210	.124	4.30
.175	.112	2.90
.140	.100	2.35
.105	.080	2.00

The values shown in Table 2–1 can be used to determine the values for (D_1+D_2) and (D_1-D_2) in equation 2–8 and the equation can be employed to compute the energy delivered by each spring in moving from the initial displacement to each displacement indicated in Table 2–1. For example, in moving from a displacement of 0.280 foot to 0.245 foot, the bolt spring delivers:

$$E=\frac{2\times25+120\ (.280+.245)}{2}\ (.280-.245)$$

$$=1.98\ \text{(ft. lb.)}$$

The energy delivered by the barrel spring for the corresponding displacement is:

$$E=\frac{2\times250+3600\ (.135+.132)}{2}\ (.135-.132)=2.19$$

The procedure is repeated for the other displacements.

The next point is to consider the energy losses in the action of the lever. To be consistent with the assumption made for the action during recoil, the total loss for the period of action will be taken as 6 foot pounds. This value is so small that it will have little effect, and therefore it can arbitrarily be distributed equally over the period, allowing the loss to increase by 1.2 foot pounds for each of the five increments being used for computation. Since all of the required energy values have now been determined, the velocities of the barrel and bolt for each displacement can be computed as follows:

The energy in the moving parts at any point in their travel is

$$E=\frac{1}{2}\,M_1V_1^2+\frac{1}{2}\,M_2V_2^2$$

But $V_2=RV_1$, where R is the velocity ratio shown in fig. 2–28.

Therefore:

$$E=\frac{1}{2}\,M_1V_1^2+\frac{1}{2}\,M_2\ (RV_1)^2$$

$$E=\frac{1}{2}\ (M_1+M_2R^2)\ V_1^2$$

Solving for V_1:

$$V_1^2=\frac{2}{M_1+M_2R_2}\ E$$

But the total energy in the barrel and bolt at any time must be equal to the initial energy plus the energy delivered by the springs minus the losses.

Therefore:

$$V_1^2=\frac{2}{M_1+M_2R^2}\ (\text{Initial KE}+\text{energy from springs}-\text{losses})$$

This computation will be illustrated by using the values for the first increment in Table 2–1.

$$V_1^2=\frac{2\times32.2}{45+5\times6.22^2}\ (195+1.98+2.19-1.2)=53.5$$

$$V_1=\sqrt{53.5}=7.32\ \text{(ft./sec.)}$$

But

$$V_2=RV_1=6.22\times7.32$$

$$V_2=45.5\ \text{(ft./sec.)}$$

The same procedure is followed for the other values shown in Table 2–1, in each case using the correct velocity ratio, energy from spring, and losses. The resulting values of barrel velocity and bolt velocity are then plotted against the corresponding values of displacement to give the velocity curves shown in figs. 2–34 and 2–35. Since the independent variable in these graphs is travel instead of time, the curves can not be used directly to extend the time-travel and time-velocity curves in fig. 2–30. However since:

$$V=\frac{dD}{dt}\ \text{(where D is travel)},$$

$$dt=\frac{dD}{V}$$

Thus, values of the time required to produce a displacement from D_1 to D_2 can be determined by the relation

$$t=\int_{D_1}^{D_2}\left(\frac{1}{V}\right)dD$$

The first step in determining the time values is to plot a reciprocal velocity curve, as shown in figs. 2–34 and 2–35. Since the barrel and bolt are returning to battery, the displacement is decreasing. Therefore the values of time are determined by integrating under the 1/V curve from right to left.

Having the time versus travel curves, it is now possible to find the travel and velocity corresponding to each value of the time and to use this data to

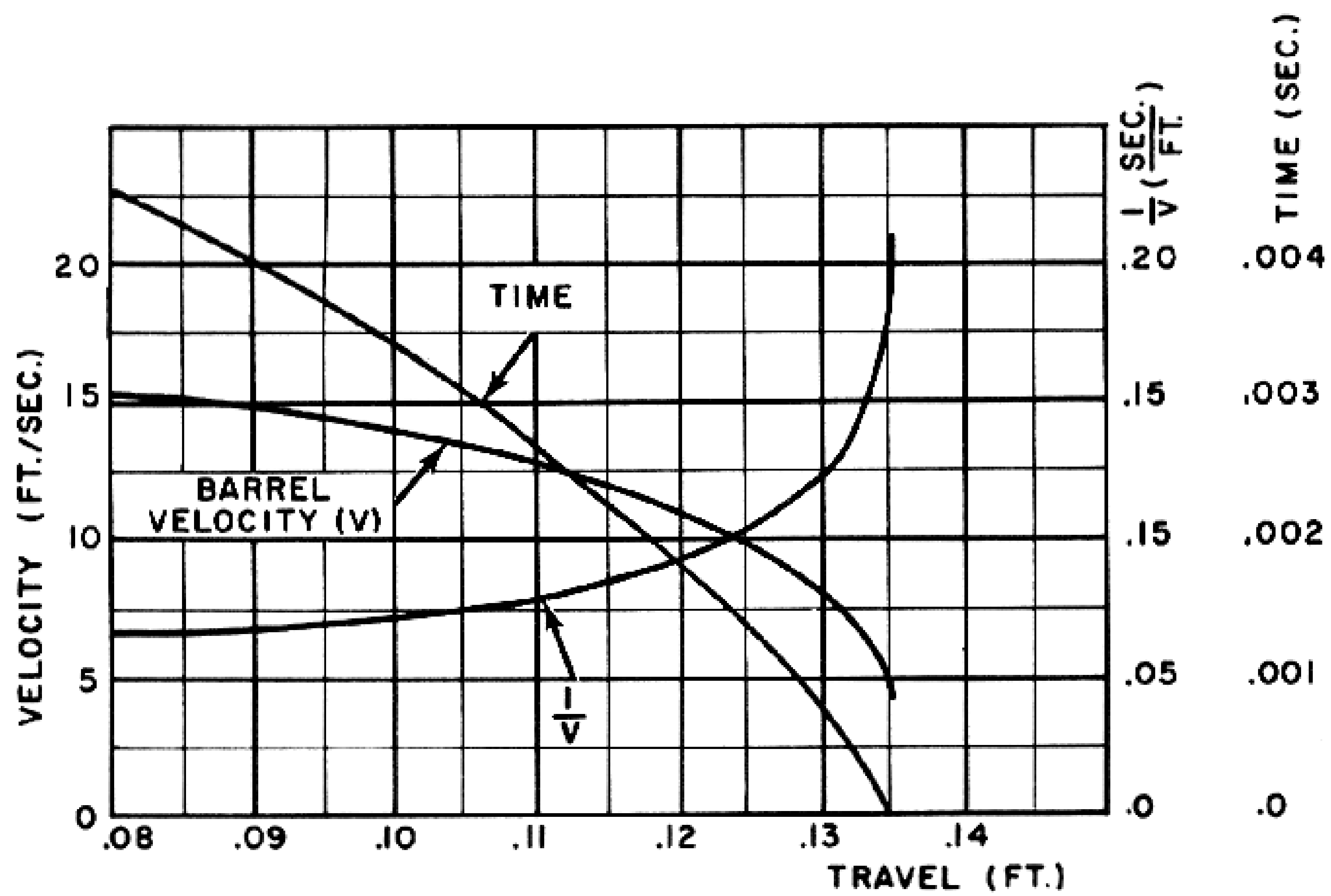


Figure 2–34. Barrel Velocity and Time Versus Barrel Travel During Action of Accelerator in Counter-Recoil.

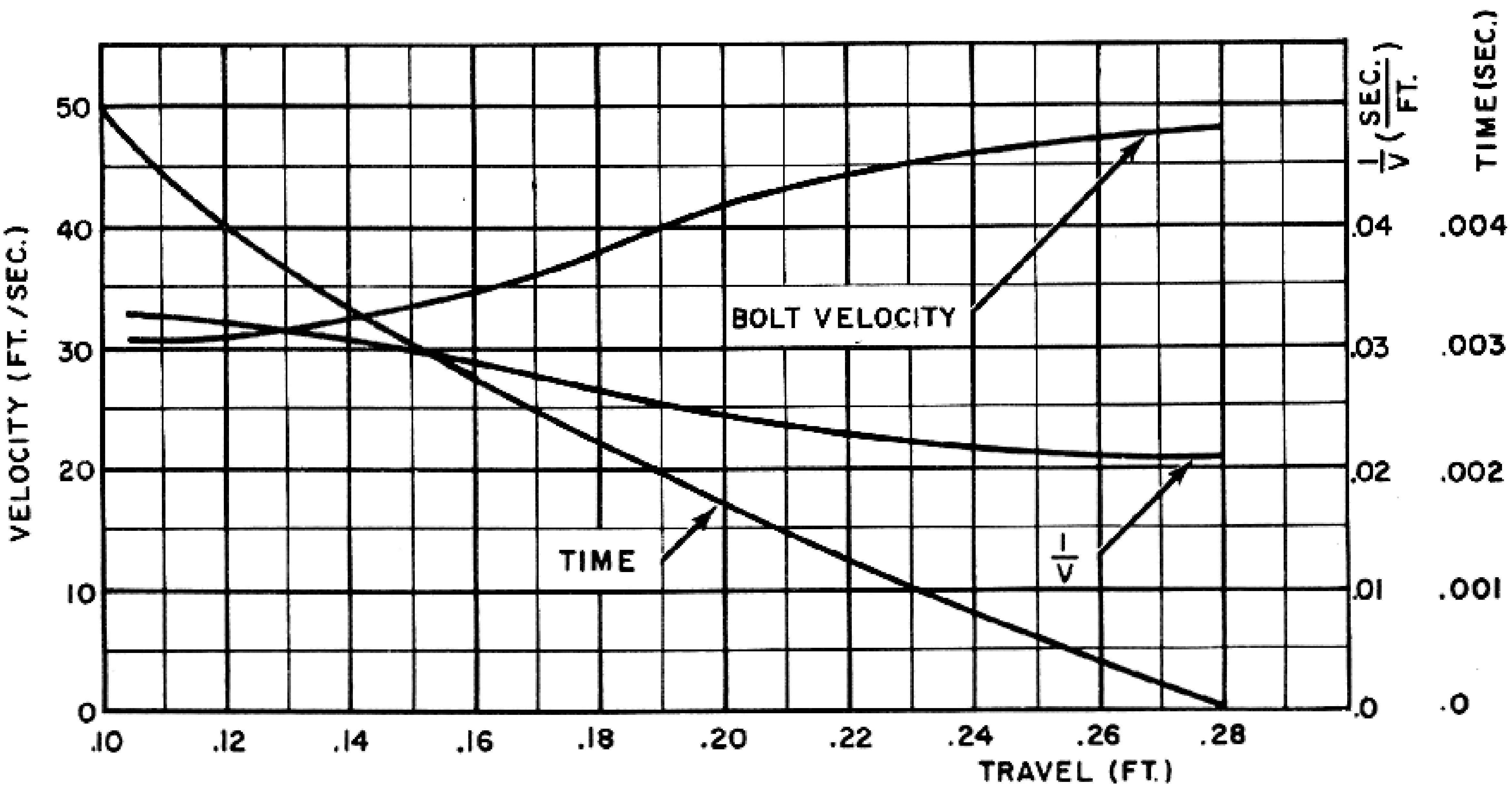


Figure 2–35. Bolt Velocity and Time Versus Bolt Travel During Action of Accelerator in Counter-Recoil.

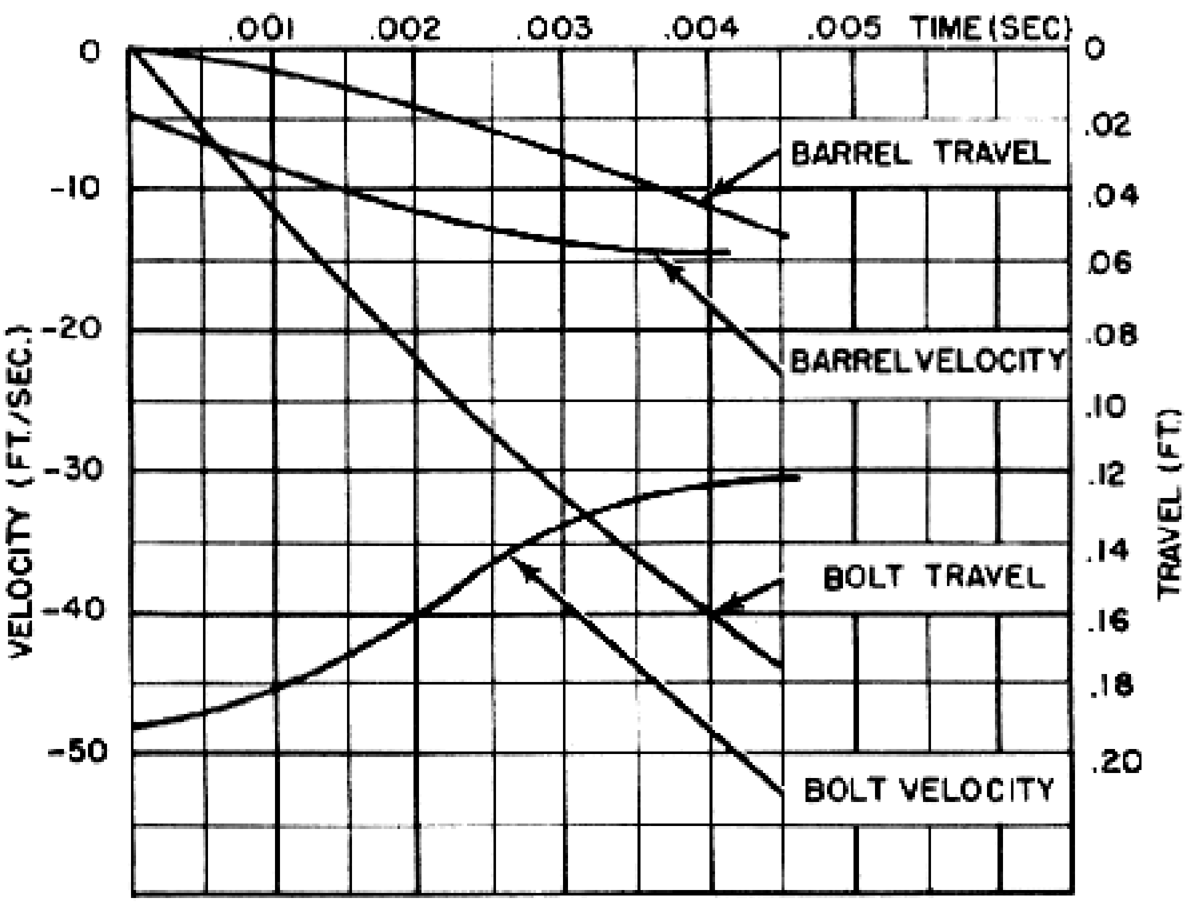


Figure 2-36. Barrel and Bolt Motion Versus Time During Action of Accelerator During Counter-Recoil.

plot curves showing the variation of the velocity and travel with respect to time. These curves are shown in fig. 2–36. The data from these curves can now be used to extend the time-travel and time-velocity curves of fig. 2–30. Note that the action of the accelerating lever increases the barrel velocity from 4.8 feet per second to 15.0 feet per second and decreases the bolt velocity from 48 feet per second to 30 feet per second and that this result is accomplished in 0.0045 second. The total time from the beginning of the cycle up to this point is 0.0441 second.

12. *Barrel and bolt motion after 0.0441 second*

At the point reached in the preceding analysis, the relative displacement between the barrel and bolt is equal to 0.0208 foot (0.250 inch) and the parts are located at the same position in which the acceleration action began during recoil. However, the action of the accelerator does not stop at this point during counter-recoil because the level is still between the barrel and bolt and will continue to act until the bolt reaches the barrel and locks in place. The remaining movement is so small that it will be accomplished in a very short time (approximately 0.0015 second) and therefore the shape of the small portion of the lever which acts during this interval is not critical. On this basis, it is assumed that after 0.0441 second the barrel and bolt velocity curves will follow the same trends as before this time. For the subsequent interval of approximately 0.0015 second, the average bolt velocity as shown in fig. 2–30 will be about 16 feet per second and the average bolt velocity will be about 30 feet per second. This means that the average relative velocity between these parts will be approximately 14 feet per second and the time required for the barrel to strike the bolt will be:

$$t = \frac{D}{V_{av}} = \frac{0.0208}{14} = 0.001485 \text{ second}$$

Thus, the barrel and bolt will meet at 0.0456 second at a displacement from battery of approximately 0.06 foot.

When the bolt and barrel meet, the bolt locks to the barrel. At the instant of impact, the velocity of the bolt is 29.5 feet per second and the velocity of the barrel is 16.5 feet per second. Since the bolt locks to the barrel, the parts will assume some common velocity. The value of the common velocity is determined by the fact that the momentum of the combined mass after impact wil be the same as the total momentum before impact. That is:

$$M_r V_3 = M_1 V_1 + M_2 V_2$$

For the conditions of the example:

$$\frac{50}{32.2} V_3 = \frac{45}{32.2} \times 16.5 + \frac{5}{32.2} \times 29.5$$

$$V_3 = 17.6 \text{ (ft./sec.)}$$

As the barrel and bolt move to battery, this velocity will be increased slightly by the combined force of the barrel return spring and bolt driving spring. Since the remaining motion is so short (0.06 foot), it will be sufficiently accurate to estimate the effect of this force by assuming that the average force will be the force which exists at a displacement of 0.03 foot. This force is:

$$F = F_{o_1} + F_{o_2} + .03\ (K_1 + K_2)$$
$$= 250 + 25 + .03\ (3600 + 120)$$
$$= 387 \text{ (pounds)}$$

The increase in velocity produced by this force will be:

$$V = \frac{F}{M} t$$
$$= \frac{387 + 32.2}{50} t$$
$$= 250\ t$$

This increase in velocity is indicated by sloping the velocity curve after locking occurs by an amount corresponding to a velocity increase of one foot per second in 0.004 second. (See fig. 2–30.) Since the average velocity of the recoiling parts shortly after locking is approximately 18 feet per second, the time required to complete the remaining 0.060-foot movement to battery will be:

$$t=\frac{D}{V_{av}}=\frac{.06}{18}=.0033 \text{ (second)}$$

Therefore, the recoiling parts reach the battery position at 0.0490 second. Note that the velocity of the recoiling parts at the battery position is 18.2 feet per second.

With a velocity of 18.2 feet per second, the kinetic energy of the recoiling parts will be:

$$KE=\frac{1}{2}MV^2=\frac{50}{2\times 32.2}\times 18.2^2=257 \text{ (ft. lb.)}$$

This is a relatively large amount of energy and could cause an extremely severe shock if an attempt were made to stop the recoiling parts merely by permitting the barrel to strike the breech casing. One way to handle this energy would be to provide a heavy buffer which is designed to dissipate a large percentage of the striking energy of the recoiling parts. An idea of the character of this buffer can be obtained by considering what average force it would be required to produce if it is to stop the recoiling parts within 0.250 inch (0.0208 foot). This force would be

$$F_{av}=\frac{KE}{D}=\frac{257}{.0208}=12{,}350 \text{ pounds}$$

It must be emphasized that the buffer should be of a type which dissipates practically all of the energy it absorbs so that the recoiling parts will be braked to a stop and will not tend to rebound from the buffer. If the buffer does not dissipate the striking energy, the residual energy may cause instability of action and may give rise to oscillations which can cause serious damage to the weapon.

As explained in the analysis of the short recoil system of operation, the problem of handling the kinetic energy possessed by the recoiling parts as they return to battery can be greatly simplified by timing the ignition of the next round so that the round is fired while the recoiling parts are still moving forward. In this way, the large forces exerted in the early part of the propellant explosion can be utilized to stop the recoiling parts or at least can be used to aid the buffer.

Although the force of the propellant explosion is capable of stopping and reversing the motion of the recoiling parts without the assistance of a buffer, there is an important point to consider. The final counter-recoil velocity of the gun used as an example is 18.2 feet per second and therefore the momentum of the 50-pound mass is:

$$MV=\frac{50}{32.2}\times 18.2=28.3 \text{ (lb. sec.)}$$

Early in this analysis, it was pointed out that the total impulse which can be produced by the sample cartridge is only 35 pound seconds. Therefore, it appears that if the propellant explosion were used alone to stop the forward motion of the recoiling parts after firing the first round, the impulse remaining to produce recoil would be only 35.0—28.3= 6.7 (pound seconds). Since the impulse producing recoil is now reduced so greatly, the next cycle of operation would be much slower than the first. However, if the gun continued to operate, it would soon settle down to some intermediate rate at which a dynamic equilibrium is established. The exact nature of the process is not important here, but the significant point to be noted is that the rate of fire of the gun will be decreased because of the fact that a considerable portion of the explosive impulse of the propellant is used to stop the forward motion of the recoiling parts. Thus the advantage gained by utilizing the force of the explosion to provide a buffer action can be obtained only by sacrificing speed of operation.

To avoid excessive loss of firing rate, it is generally advisable to stop the forward motion of the counter-recoiling parts by means of the explosive force of the next round. A buffer should also be used so that the momentum of counter-recoil will be partly cancelled before the next round is fired. With this combination of actions, the forward motion of the recoiling parts can be stopped smoothly without the necessity of having an impractically heavy buffer and without the excessive loss of the explosive impulse required for producing a high recoil velocity.

Proper functioning under these conditions will require high precision in timing the firing. Firing too soon will mean that the buffer will not contribute its full share of braking action and too much of the explosive impulse will be utilized to stop the counter-recoil motion. The subsequent recoil will therefore be weak. If, on the other hand, firing occurs too late, the buffer may be overtaxed and too much of the explosive impulse will be available for producing recoil. Recoil will then be too violent. If the effects of small variations in the time of firing prove to be troublesome, these effects can be minimized by utilizing a buffer which is compressed through a greater distance, thus allowing a greater amount of time for the action of the retarding force. For example, it has been shown above that a 50-pound mass moving at 18.2 feet per second will produce a 0.250 inch compression in a buffer exerting an average force of 12,350 pounds. The time for this action can be computed as follows:

$$MV=F_{av}t$$

$$t=\frac{MV}{F_{av}}=\frac{28.3}{12{,}350}=.00229 \text{ (second)}$$

If the compression of the buffer were 0.500 inch instead of 0.250 inch, the necessary average force would be one-half of 12,350 or would be equal to 6,175 pounds. The time of action would then be

$$t=\frac{MV}{F}=\frac{28.3}{6175}=.00458 \text{ (sec.)}$$

Thus, by increasing the length of the stroke of the buffer, the retarding action happens more slowly and a small error in timing will not have as great an effect.

In the computations leading to the time-travel and time-velocity curves shown in fig. 2–30, it was assumed that the recoiling parts were not moving at the instant of firing. Therefore these curves represent the conditions obtained if the buffer alone acted to stop the forward motion of the recoiling parts. On this basis, let it be assumed that the buffer used exerts an average force of 12,350 pounds and that therefore it is compressed 0.0208 foot, bringing the recoiling parts to a stop in 0.00229 second (as previously computed). The recoiling parts are then returned quickly to battery by elastic action as the buffer regains its original dimensions. (As pointed out previously, most of the striking energy of the recoiling parts is dissipated in the buffer, so that there is no tendency for the recoiling parts to rebound.) This action is indicated by the last portions of the curves in fig. 2–30. Note the small overtravel of the recoiling parts past the battery position. The total time to complete the operating cycle is 0.0522 second and therefore the theoretical rate of fire for this particular design is:

$$N=\frac{60}{.0522}=1150 \text{ rounds per minute}$$

If the design were arranged so that a small part of the explosive force of the next round would be utilized to assist in stopping the forward motion of the recoiling parts, the curves would not be quite the same as in fig. 2–30 because there would be some loss in the impulse producing recoil. However except for this small difference, the computations and procedures would be the same as for fig. 2–30. In making the computations for such a design, it would only be necessary to determine in advance what amount of momentum is to be cancelled by the explosion and then to subtract this momentum from the momentum imparted to the recoiling parts at any instant.

CHAPTER 3

GAS OPERATION

PRINCIPLES OF GAS OPERATION

In all machine guns, the fundamental source of operating energy is the high-pressure gas created by the explosion of the propellant charge. This is true, in a general sense, of guns operated by the blowback system, recoil system, or any other system of "true automatic operation" as the phrase is defined in this publication. However, in spite of the fact that the ultimate source of operating energy in all machine guns is the pressure of the powder gases, the term "gas operation" is reserved for a particular type of operating system in which the pressure of the powder gases is employed in a specific way.

In a typical gun which uses the system of gas operation, an opening (or "port") is provided in the side of the barrel as shown in fig. 3–1. When the projectile has passed this opening, some of the high-pressure powder gases behind the projectile are tapped off through the hole and pass through an orifice to act upon a piston or some similar device for converting the pressure of the powder gases to a thrust. This thrust is then utilized through a suitable mechanism to provide the energy necessary for performing the automatic functions required for sustained fire. These functions include unlocking the bolt, retracting the bolt, and operating the other portions of the gun mechanism.

The gas operating mechanism can take very many forms. The most commonly used device consists of a simple gas cylinder and a piston which is driven rearward to transfer its energy to the bolt by direct

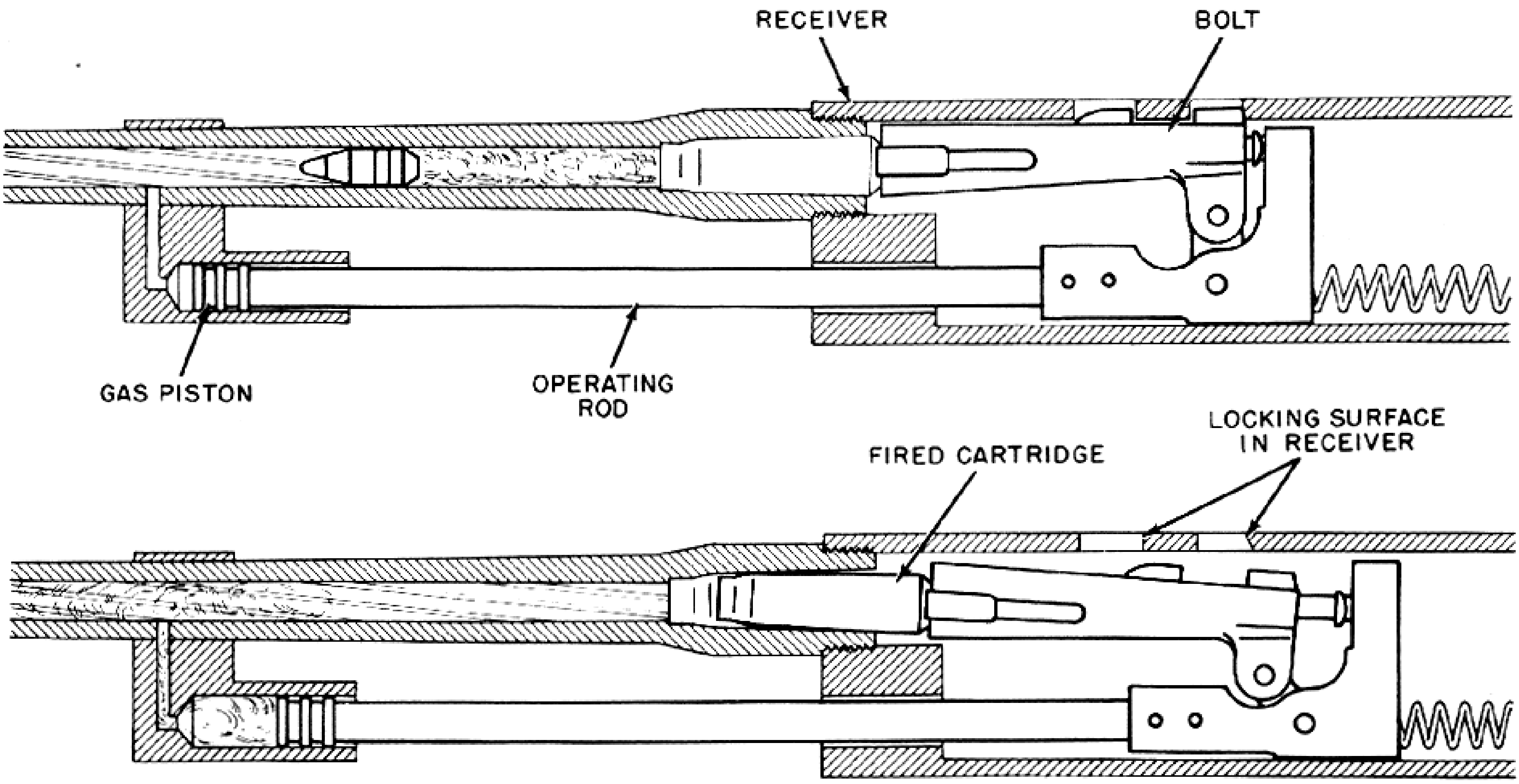


Figure 3–1. Typical Gas Operated Gun.

impact. In some cases, the piston may be driven forward instead of rearward, but this does not involve any significant change in the principle of operation. Even the nature of the member which is acted upon by the gas pressure is subject to great variation. Instead of being a conventional piston, this member can be in the form of a sleeve, slide, or other device arranged to receive an impulse from the gas pressure. In fact, it is interesting to note that the floating chamber (such as is used in the caliber .22 modification of the Browning machine gun) is in reality a special form of gas piston.

The methods used for transferring energy from the piston to the gun operating mechanism are also extremely diverse in form and function. Instead of transferring energy directly to the bolt, the piston itself sometimes moves through a very short stroke and transfers its energy by impinging on an intermediate sliding member or lever. A large number of devices have been designed to minimize the shock involved in the energy transfer through the use of levels, links, or cams. In certain instances, the shock of transfer is reduced by causing the piston to load intermediate springs which subsequently transfer their stored energy to the mechanism.

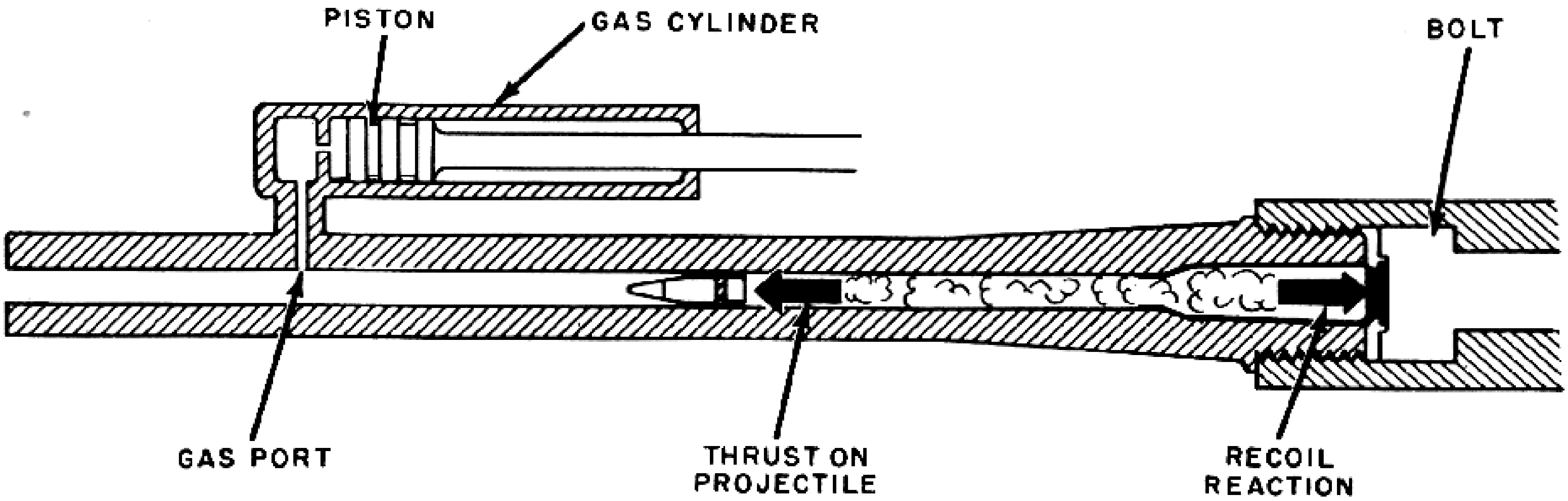
The basic principles involved in gas operation can be outlined by considering the general character of the pressures and forces which result from the firing of the cartridge in an elementary gun provided with a gas port and a piston. Fig. 3–2A shows the condition which exists immediately after the cartridge is fired. The bolt is rigidly locked to the barrel in order to support the base of the cartridge case against the thrust produced by the explosion of the propellant charge. This thrust, acting on the base of the projectile, drives the projectile forward. As the projectile moves through the bore of the gun, the gases expanding behind the projectile also move so that the center of mass of the products of combustion travels forward at a speed nearly equal to one-half the projectile velocity. The same force which moves the projectile and powder gases forward produces an equal and opposite reaction which tends to drive the entire gun to the rear. This reaction is known as the recoil force.

The nature of the recoil force and the manner in which it varies with time are described in considerable detail in Chapter 2. These principles, as stated, are applicable to any gun (including gas-operated weapons) in which the bolt remains locked to the barrel for all or a portion of the time the powder gas pressures act. Therefore, the principles of recoil will not be repeated here.

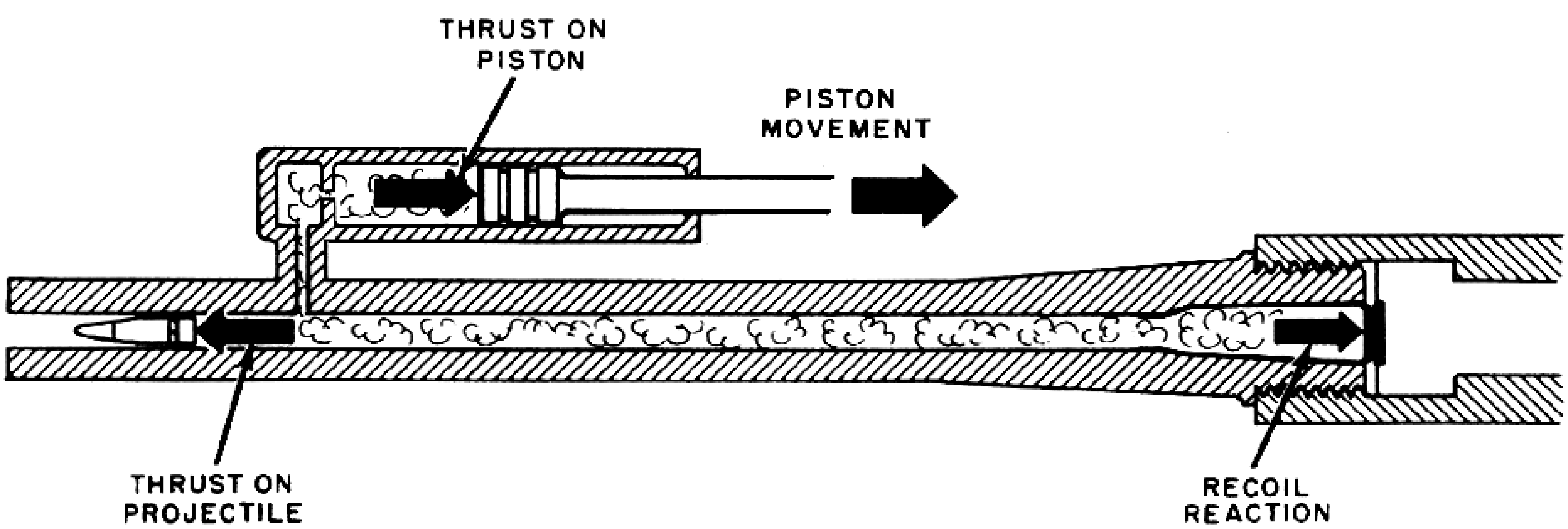
As soon as the projectile has passed the gas port, as shown in figure 3–2B, the high-pressure gases behind the projectile start to flow into the gas cylinder and to build up a pressure against the piston. For any given barrel and cartridge, the rate at which this pressure builds up depends on a number of factors. If the orifice is relatively large, the gases will flow through it freely and easily but if it is small in diameter, it will produce a marked throttling effect and restrict the flow of gas so that the pressure will build up more slowly. The shape of the orifice, as well as its size, will affect the gas flow through it. The rate of pressure increase will also be affected by many other things, such as the pressure difference between the barrel and cylinder, the volume of the cylinder space into which the gases flow, and the rate of change of this volume as it is influenced by the motion of the piston. The force exerted on the piston will be equal to the gas pressure in the cylinder multiplied by the area of the piston. It should be pointed out that ordinarily the amount of gas which flows into the cylinder is relatively very small and will have no significant effect on the muzzle velocity of the projectile.

After the projectile has passed the gas port, and as it moves through the bore and leaves the gun, the pressure within the barrel decreases. The manner in which the pressure decreases for a given propellant charge will be influenced by the length of the barrel. For example fig. 3–3 shows the variations of pressure with time for a typical 20-mm gun with a barrel length of slightly less than five feet and fig. 3–4 shows how the bore travel of the projectile varies with time in the same gun.

Assuming that the gas port in the gun of the example is located 2.5 feet from the point used as the reference for measuring the bore travel of the projectile, the points marked on the curves show the time at which the projectile passes the gas port (0.0016 second after ignition of the propellant charge). Note that the pressure in the barrel at this instant is 11,000 pounds per square inch. At the time the projectile leaves the muzzle at 0.00234 second, the barrel pressure has decreased to 5000 pounds per square inch and the residual pressure continues to act as shown until it falls to zero at about 0.0080 second.



CONDITION BEFORE PROJECTILE PASSES GAS PORT



CONDITION AFTER PROJECTILE PASSES GAS PORT

Figure 3–2. Action of Gas Cylinder and Piston.

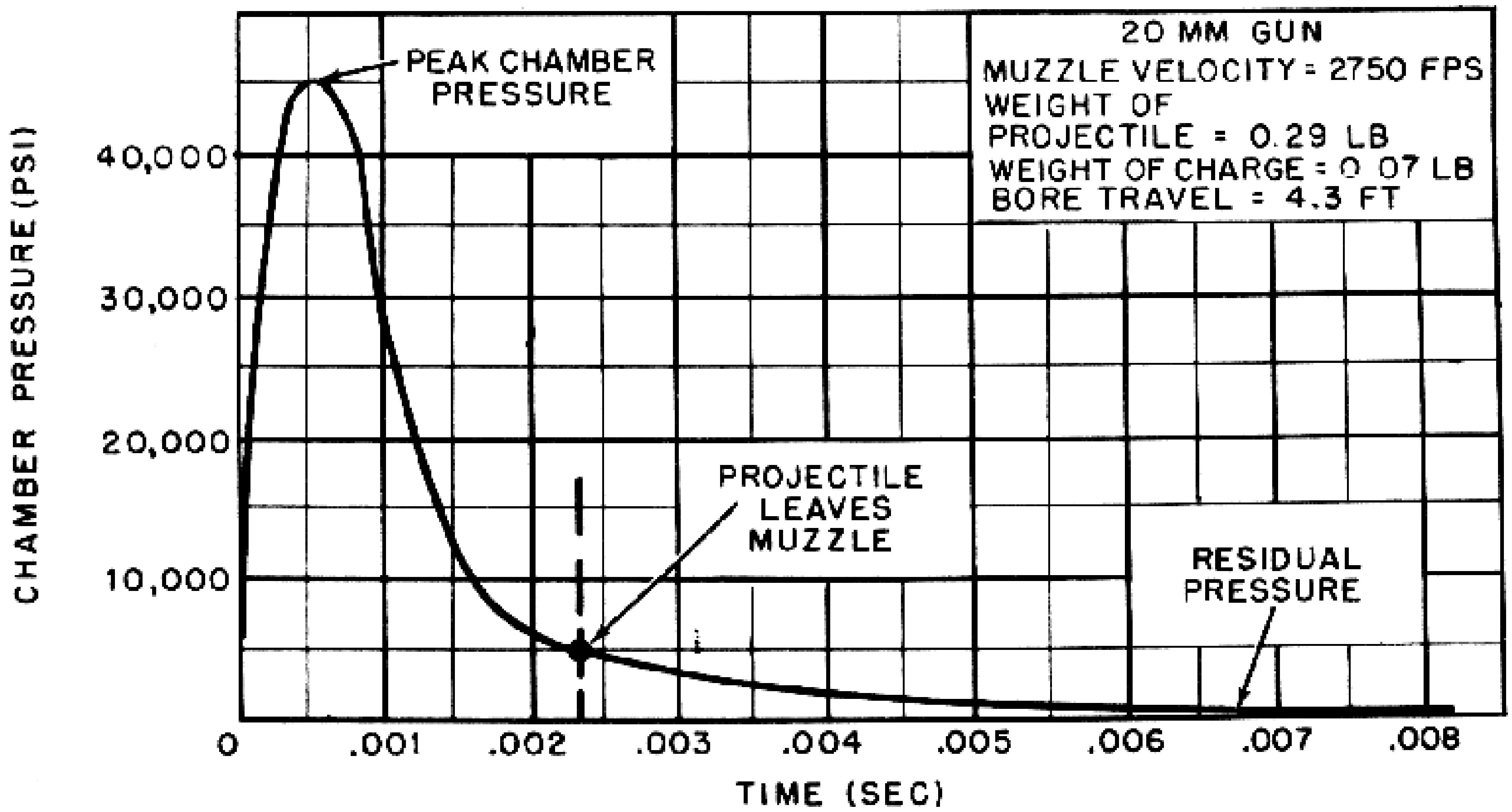


Figure 3–3. Variation of Chamber Pressure With Time.

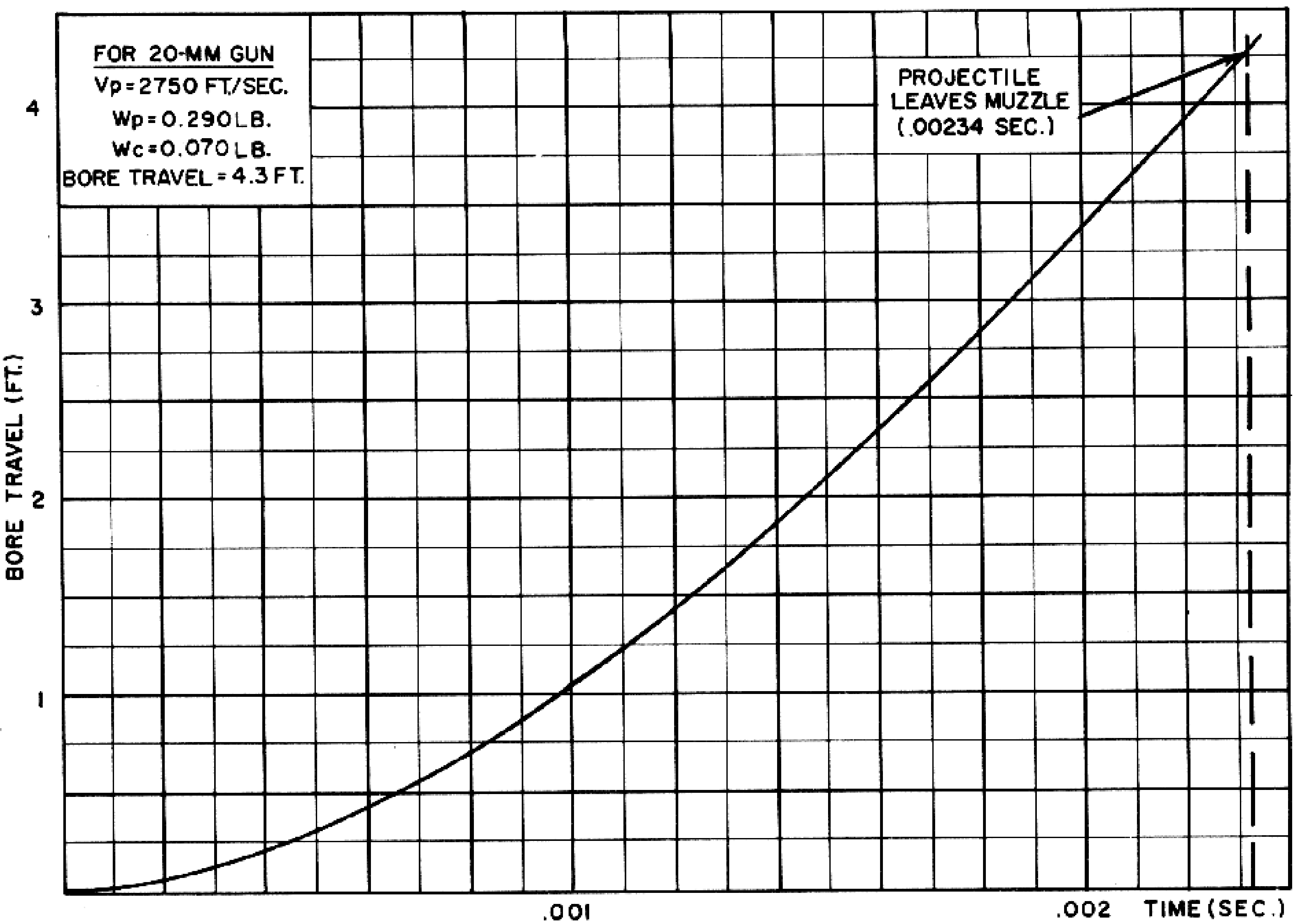


Figure 3–4. Graph of Projectile Bore Travel Versus Time (20 mm Gun).

The barrel pressure curve is reproduced in fig. 3–5. The other curve shown in this figure is the pressure in the gas cylinder. This curve was drawn under the assumption that the amount of gas which flows into the cylinder is too small to have any appreciable effect on the pressure in the barrel and also that the orifice is fairly large so that the pressure in the cylinder builds up rapidly to a value which is practically equal to the barrel pressure. Since the barrel pressure decreases smoothly, the pressure in the cylinder will remain very close to the barrel pressure. If the gas port remains open after the projectile has passed, the pressure in the gas cylinder will become zero at essentially the same time the residual barrel pressure becomes zero. (Ordinarily, however, the orifice is relatively small and produces a throttling effect which causes the pressure in the cylinder to remain considerably lower than the barrel pressure for most of the time of action.)

Examination of fig. 3–5 will show that the gas piston is subjected to a driving force for only a very short interval of time. The pressure starts to act on the piston when the projectile passes the port at 0.00166 second and ceases to act at about 0.008 second when the residual barrel pressure has reached zero. Therefore the total time of action is 0.00800—0.00166=0.00634 second. For the assumed conditions, it is during this brief interval that the piston absorbs the impulse imparted to it by the powder gases. Since the powder gas pressure exists for such a short time, it is not practical to at-

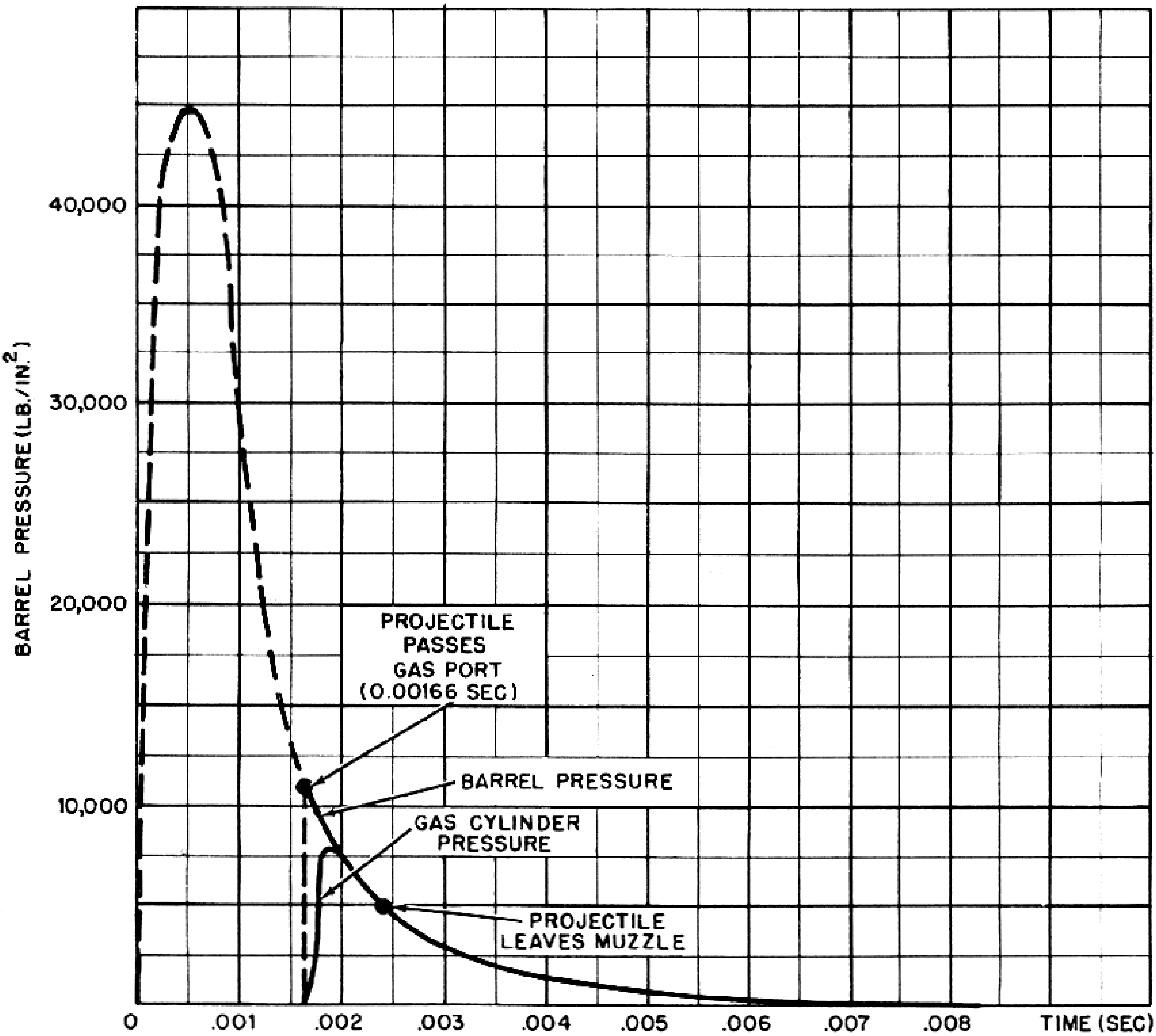


Figure 3–5. Barrel Pressure and Pressure in Gas Cylinder.

tempt to use the force on the piston directly for actuating the gun mechanism. This is particularly true because the bolt must remain locked for a considerable portion of the time during which the powder gas pressures are acting. Therefore, the impulse of the pressure is utilized to impart a velocity to the piston mass and then, by virtue of the kinetic energy thus stored in the piston, the piston can perform the necessary automatic functions even after the powder gas pressure is gone.

Since the force produced on the piston by the gas pressure in the cylinder depends on the area of the piston, the magnitude of the impulse absorbed by the piston will also depend on the piston area; the larger the piston, the greater will be the impulse. The magnitude of the impulse will also depend on the amount of the pressure and the duration of the pressure and therefore will be greatly influenced by the size and location of the orifice. Moving the gas port closer to the chamber and increasing the size of the orifice will cause the impulse to be greater because the pressures applied to the piston will then be higher and these pressures will act for a longer time.

The amount of energy stored in the piston as the result of the applied impulse will be determined by the mass of the piston. The lighter the piston, the greater will be the energy produced by a given impulse. Thus it appears that the conditions of pressure, the location of the gas port, the size and shape of the orifice, the piston area, and the piston mass all have an influence on the amount of energy which can be obtained from the action of the piston. By proper selection and control of these factors, the piston energy can be regulated as desired so that low values or very high values of energy can be achieved. In fact, it is comparatively easy under practical conditions to achieve extremely high values of piston energy in a gas-operated gun. Unless the gas operation is carefully controlled, the action of the piston may be so violent that it can literally smash the breech mechanism.

In the following paragraphs, the gas system of operation is described and analyzed by considering the sequence of events which occur in the automatic cycle of operation. As with the other systems treated in this publication, this analysis is concerned only with the general factors affecting design. (Detailed descriptions of the mechanisms used in actual guns employing the gas system will be referenced where applicable.)

GAS SYSTEM OF OPERATION

As previously mentioned, the gas system of operation can employ many types of gas actuating devices. Although the functional characteristics of a weapon will depend to some extent on the particular type of actuating device employed, all gas-operated guns are basically similar in their operation. Since the gas piston is the most common form of actuating device, it will be used here for purposes of illustration.

Fig. 3–6A shows schematically the essential elements of a gun which operates by the gas system. These elements consist of the bolt, an arrangement for locking the bolt to the barrel and unlocking it, a backplate buffer, a driving spring for returning the bolt to battery, the gas cylinder and piston, a cradle which permits the entire gun mechanism to move in recoil, and a barrel spring which returns the gun to battery. Associated with the barrel spring are buffers which stop the recoil and counter-recoil motions of the gun. The other portions of fig. 3–6 show different stages during the cycle of operation.

Cycle of Operation

The operating cycle of a typical gas-operated gun occurs as follows:

The cycle of operation starts with a cartridge in the chamber and with the bolt locked to the barrel (fig. 3–6A). When the cartridge is fired, the pressure of the powder gases drives the projectile and gases forward through the bore and at the same time drives the entire gun mechanism to the rear in recoil. During the action of the powder gas pressure, the retardation offered by the barrel spring is relatively small so that the only really significant factor in limiting the recoil acceleration is the mass of the recoiling parts.

NOTE: Although this is a gas-operated gun, it is necessary to permit the barrel and locked bolt to move in recoil in order to reduce the force on the gun mounting. In a 20-mm gun with a maximum chamber pressure of 45,000 pounds per square inch, the maximum recoil

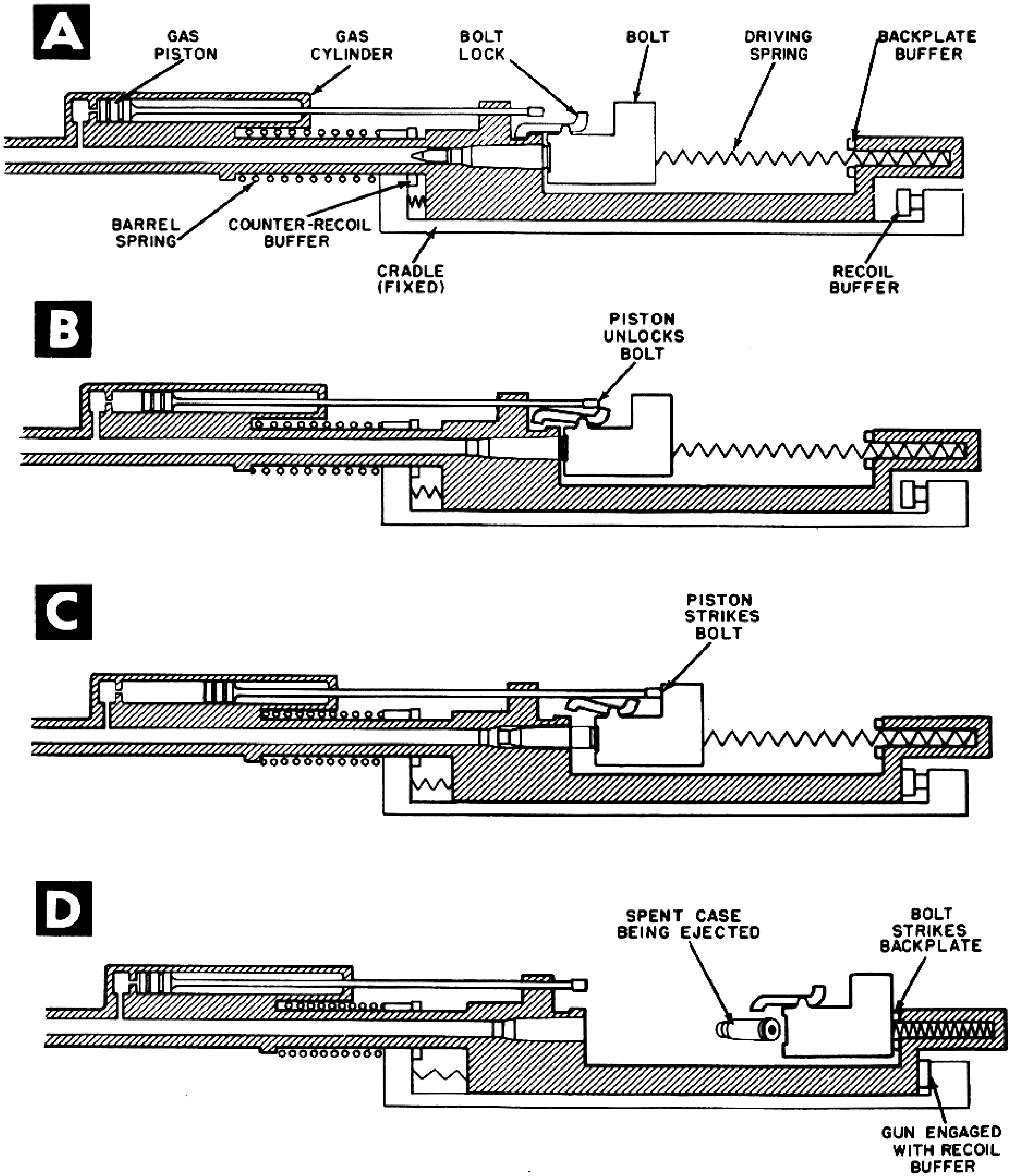


Figure 3–6. Elements of a Gas Operated Gun (Schematic).

force driving the gun to the rear while the bolt is locked is near 22,000 pounds. This force is so large that it would be impractical to attempt to hold the barrel in a rigid mounting.

The time at which the projectile passes the gas port in the barrel will depend on the location of the port. Ordinarily, this will occur from 0.001 to 0.002 second after ignition of the propellant. As soon as the projectile passes the port, the powder gases will start to flow into the cylinder and start to apply a pressure to the piston, thus driving the piston to the rear. The action of the piston is controlled in the design so that the bolt is not unlocked immediately but unlocking is delayed until the pressure in the barrel has dropped to a safe operating limit, 0.001 or 0.002 second after the projectile leaves the muzzle (fig. 3–6B).

After unlocking occurs, the residual pressure (which has still not reached zero) continues to exert a force on the piston and also creates a blowback action on the cartridge case. Shortly after the instant of unlocking, the piston strikes the bolt, so that the kinetic energy stored in the piston is transferred, thus imparting a high acceleration to the bolt (fig. 3–6C). After the piston has acted, the barrel and barrel extension assembly strikes a buffer stop which absorbs whatever recoil energy was not previously absorbed by the barrel return spring. When the barrel and barrel extension assembly has been stopped, it is immediately driven forward by the compressed barrel return spring, and is brought to rest at the battery position by the counter-recoil buffer.

At the instant of unlocking, the bolt already has a considerable rearward velocity because of the recoil movement which occurred before unlocking. This "inherited" bolt velocity is increased by the combined action of blowback and of the gas piston so that a high final bolt velocity is produced. The bolt then continues to move to the rear by its own momentum until the opening between the barrel and bolt is sufficient to permit feeding. As the bolt moves back, the spent cartridge case is extracted from the chamber and ejected and the bolt driving spring is compressed. This spring is relatively light and its only function is to assist the return movement of the bolt. Therefore, the driving spring does not absorb any great portion of the kinetic energy of the recoiling bolt and the bolt moves through its entire recoil distance at high velocity. The bolt then strikes the backplate buffer and rebounds (fig. 3–6D). The forward velocity of the bolt immediately after leaving the backplate buffer is somewhat lower than the velocity at which it strikes the backplate because the impact is not purely elastic and some energy is lost as heat in the exchange.

NOTE: It is important to realize that the velocity with which the bolt strikes the backplate buffer will depend on the condition of motion of the gun at the instant of impact. If the gun happens to be moving forward at this instant, the impact velocity will be higher than if the gun were stationary or moving rearward. It can be seen that if the gun motion is not controlled to insure uniform velocity at the instant of contact, the bolt impact will be entirely unpredictable and will vary widely from shot to shot. This could give very erratic performance and may result in a high incidence of parts breakage.

As the bolt moves forward after rebounding from the backplate buffer, its motion is aided by the driving spring. The bolt picks up a fresh cartridge from the feed mechanism and loads this cartridge into the chamber. The bolt then locks into the barrel. Usually it is desirable to time the operation of the weapon so that the bolt returns and locks to the barrel before the gun has reached the battery position. Then, shortly before the recoiling parts reach their most forward position, the firing mechanism is actuated and a new cycle begins. Since the cartridge is fired before the counter-recoil motion is completed, the forward velocity of the recoiling parts is first checked by the initial part of the rearward thrust exerted by the exploding propellant charge and the recoiling parts are then driven to the rear. (Timing the firing in this way eliminates the need for a heavy counter-recoil buffer to absorb and dissipate the forward kinetic energy of the recoiling parts.)

Analysis of Gas Operation

From the theoretical point of view, the gas system of operation has many features which make it adaptable to a wide variety of weapon designs. This flexibility of application is due primarily to the fact that by proper selection of design values, the functioning of the gas actuating device can be controlled

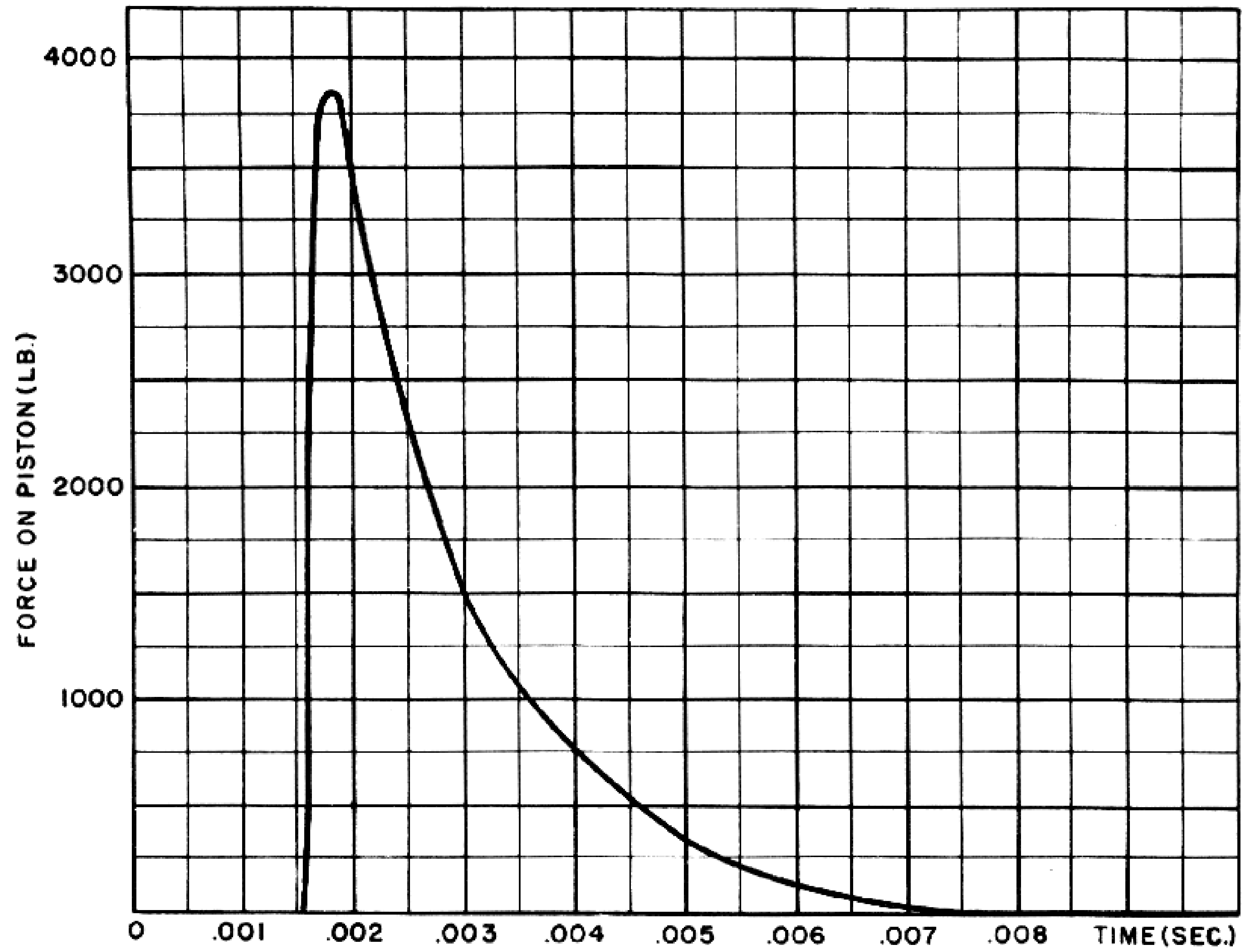


Figure 3–7. Force Exerted on Piston.

to produce small or very large operating forces and the timing of these forces can be regulated as desired. However, there are also certain disadvantages inherent in the gas system of operation and any successful design must take these disadvantages into account and make suitable allowances for them. In the following paragraphs, gas operation is analyzed to evaluate both the advantages and disadvantages of the system.

One of the principal features of the gas system of operation is the large amount of operating energy which can be obtained from the gas actuating device. This energy is available because the gas system makes it possible to tap into the high gas pressure which exists during the propellant explosion and to apply this pressure to a relatively light operating member while the bolt still remains locked. Thus, energy can be accumulated for later use without the necessity for sacrificing rigid support behind the cartridge case during the early phases of the propellant explosion.

NOTE: By way of comparison, it should be pointed out here that energy is also accumulated in the recoiling parts of a gun employing the recoil system of operation. Although this energy is also accumulated during the early phases of the propellant explosion while the bolt is locked, the parts which absorb the explosive impulse in this case are fairly heavy. Therefore, the amount of energy accumulated for a given impulse will be relatively small when compared with the energy imparted by the same impulse to a very light gas piston.

An idea of the magnitude of the energies and velocities which can be achieved by gas operation may be obtained by considering a specific example. In the preceding explanation of the principles of gas operation, a curve (fig. 3–5) was developed to show the variation of pressure with time in the gas cylinder of a sample 20-mm gas-operated gun. (This gun has a barrel length of slightly less than five feet and the gas port is located 2.5 feet from the chamber.

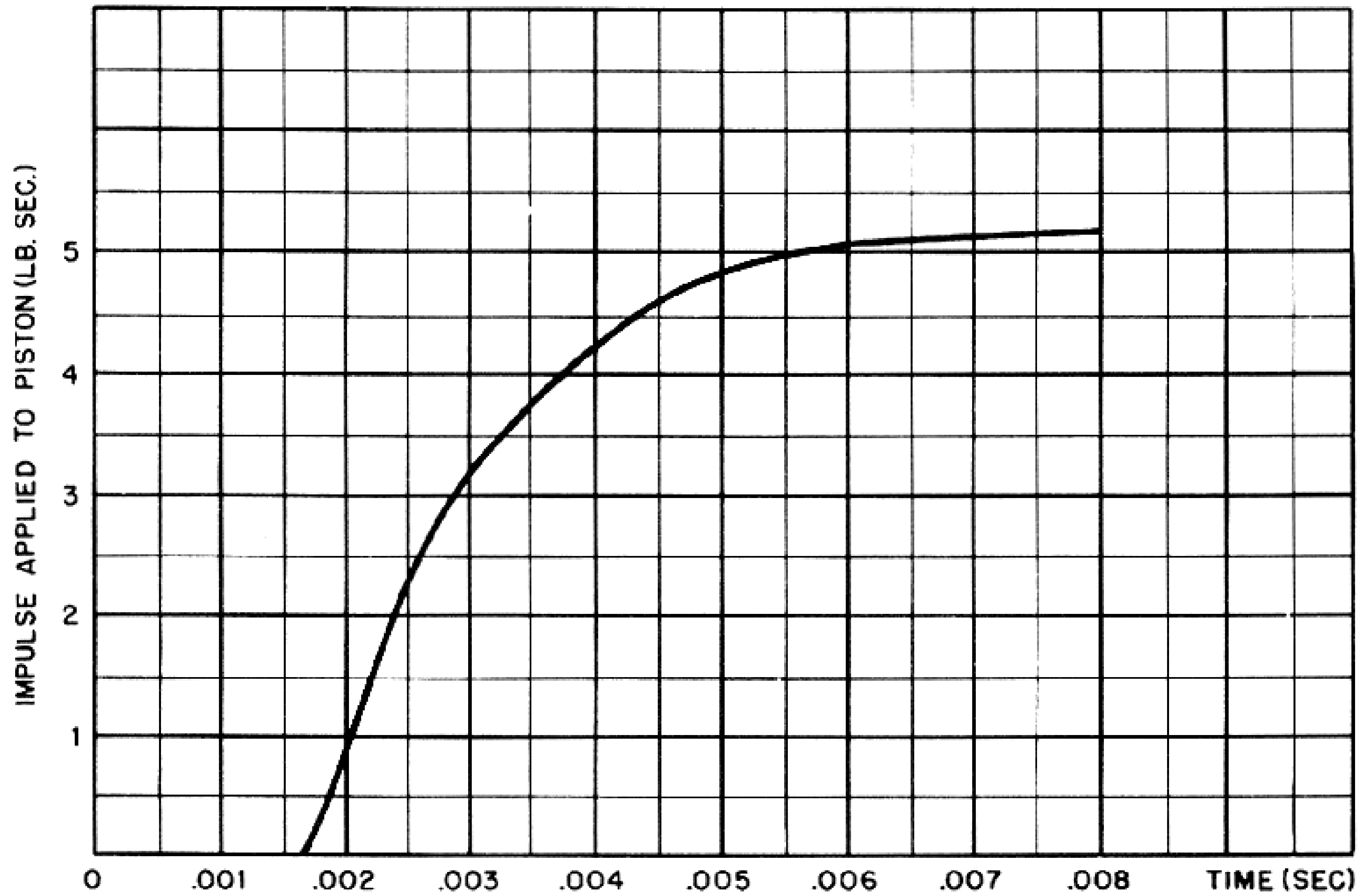


Figure 3–8. Impulse Imparted to Piston.

The projectile passes the gas port 0.00166 second after ignition of the propellant charge. The orifice is assumed to be quite large so that the cylinder pressure quickly becomes practically equal to the barrel pressure.)

In order to determine the impulse applied to the gas piston, let it be assumed that the diameter of the piston in the example gun is equal to the bore diameter, or approximately 0.79 inch. (This is the rule of thumb often used in the design of gas-operated weapons.) The area of the piston will then be approximately 0.5 square inch. The force exerted on the piston at any instant can now be determined by multiplying this area by the pressure for that instant shown in fig. 3–5. The force-time curve which results from plotting these values is shown in fig. 3–7.

If it is considered that the only significant force initially acting on the piston is the pressure of the powder gases, the total impulse applied to the piston up to any instant can be found by measuring the area under the curve of fig. 3–7 up to that instant. The resulting values of impulse can then be used to plot the impulse curve shown in fig. 3–8. Note that the total impulse applied to the piston for the sample conditions is equal to 5.13 pound seconds.

The amount of energy imparted to the piston by this impulse will depend on the mass of the piston. The lighter the piston, the greater will be the energy produced by a given impulse.

NOTE: This can be seen by considering the basic expressions for kinetic energy and impulse:

(3–1) $$KE=\frac{1}{2}MV^2$$

(3–2) $I=MV$ (where I is the impulse)

Solving equation 3–2 for V and substituting the result in equation 3–1 gives:

$$KE=\frac{1}{2}M\left(\frac{I^2}{M}\right)=\frac{1}{2}\frac{I^2}{M}$$

which shows that the kinetic energy produced by a given impulse is inversely proportional to the mass of the object to which the impulse is applied.

If it is assumed that the piston of the sample gun weighs 1.5 pounds, the final velocity imparted to the piston at 0.008 second is determined by dividing the total impulse of 5.13 pound-seconds by the piston mass: That is:

$$V=\frac{I}{M}=\frac{5.13\times32.2}{1.5}=110 \text{ (ft./sec.)}$$

The kinetic energy corresponding to this velocity is:

$$KE=\frac{1}{2}MV^2=\frac{1}{2}\times\frac{1.5}{32.2}\times110^2=282 \text{ (ft. lb.)}$$

These values show that with the assumed conditions of pressure, gas port location, orifice size, piston area, and piston mass, the piston velocity and piston energy obtained are relatively high values. It can also be seen that changing any of these factors can have a marked effect on the values of energy and velocity. For example, these values could be increased tremendously by moving the gas port closer to the chamber, increasing the piston area, and decreasing the piston mass. Conversely, the piston velocity and energy could be drastically reduced by moving the gas port closer to the muzzle, decreasing the size of the orifice to produce a throttling effect, increasing the mass of the piston, and decreasing the piston diameter. Thus it is evident that, theoretically, the design of a gas-operated gun can be set up readily to achieve almost any desired magnitude of piston energy.

It is of interest from the practical standpoint to note that the gas port must be located in the center of one of the rifling grooves. This is important because of the fact that when the rotating band of the projectile is engraved by the rifling lands, fairly large burrs or flashes of copper are formed at the rear of the rotating band. There is a relief recess behind the band to accommodate these burrs while the projectile passes through the bore. (This recess is sometimes erroneously thought to be provided for crimping the case to the projectile.) If the port were in the land of the rifling, the burr formed by that land would be blown into the gas port by the high-pressure gases behind the projectile as the projectile passes the port. After one or several shots, this could cause the gas orifice to become clogged or could interfere with the operation of the piston.

Since the port must be located in a groove of the rifling, it is necessary in the design of the gun to be sure that the angular position of the barrel will be correctly related to the gun mechanism. For this reason, it is practically impossible, without redesign, to change the location of a gas port in a gun which has already been built. This same consideration also precludes the possibility of simply rotating the barrel in its threaded mounting in order to adjust the gun for correct headspace.

In gas-operated guns, the headspace can be controlled only by selective assembly, using parts in which the dimensional variations permitted under the manufacturing tolerances will combine in such a way as to produce the required headspacing. Because of the difficulty of producing and maintaining correct headspacing in a gas-operated gun, it is necessary to use lubricated ammunition in order to avoid cartridge case separations which otherwise would inevitably result. This is a very serious disadvantage which is inherent in the gas system of operation and its importance can not be overemphasized.

Another real difficulty in most gas-operated guns is that the throttling effect produced by the orifice is extremely critical in determining the operating characteristics. The size and shape of the orifice must therefore be selected with great care and must be controlled precisely in manufacture. In order to reduce the maintenance problem, it is also important to take steps in the design to protect the orifice from enlargement by erosion and from clogging as the result of fouling. To simplify maintenance, many guns are provided with orifice plates which can be removed readily for cleaning or replacement. In some instances, the gun is equipped with a built-in arrangement for substituting orifices of slightly different size so that the gun can be adjusted readily to compensate for changes in performance due to bore wear and other causes. In regard to orifice erosion, it should be noted that the problem of erosion increases tremendously as the gas port is placed closer to the chamber. Although the use of orifice inserts of stellite, molybdenum, or similar materials is of some help in reducing erosion, the severe washing effect of the high-temperature, high-velocity gas encountered in the initial part of the propellant explosion still places definite limit on how close the gas port can be to the chamber.

The preceding points are primarily concerned with the amount of energy available from gas operation and with the design factors related to the position of the gas port. Another important consideration in gas operation is concerned with how the action of the piston can be timed so that unlocking and the subsequent transfer of energy to the bolt will occur in the correct time relation to the variation of the chamber pressure. Before entering into the methods whereby this timing is accomplished, it is appropriate to consider first what conditions of timing will produce the greatest functional efficiency.

First, the mechanism should be arranged so that the bolt will be unlocked as soon as the chamber pressure has reached a safe operating limit. This will aid in obtaining a high rate of fire by eliminating any unnecessary delay and will also make it possible to derive useful bolt energy by utilizing the blowback effect available from the residual powder gas pressure. In order to facilitate extraction and to make the best use of blowback, it is important to avoid binding or excessive friction between the cartridge case and chamber wall after unlocking occurs. With lubricated ammunition, friction and binding do not present a problem, but with unlubricated ammunition, the cartridge case will always tend to seize in the chamber. This binding occurs because the peak chamber pressures and the heat of the explosion expand the case tightly against the chamber, and since unlocking takes place while there is still an appreciable residual pressure, the case does not have a chance to contract sufficiently to permit it to move freely under blowback. This difficulty can be avoided by employing an operational feature known as "initial extraction". When this feature is incorporated in the unlocking mechanism, the bolt is unlocked in two stages. In the first stage, the bolt is not unlocked completely but is cammed powerfully back through just a sufficient distance to cause the taper of the cartridge case to break free of the chamber wall. Immediately thereafter, the bolt is unlocked completely and blowback can occur without difficulty.

The utilization of blowback is limited by the fact that the cartridge case can not be permitted to move too far out of the chamber while the residual powder gas pressure is still high enough to cause rupture of the case near the base. The actual limit on the amount by which the cartridge case can be permitted to move as it is related to the chamber pressure will of course depend on the specific cartridge case under consideration. A good way to estimate the limit for a given cartridge case is to consider what pressure could be withstood by the case when the case has moved just far enough to the rear to expose the thin walls near the base. (See fig. 3–9.) For an ordinary 20-mm cartridge case, this occurs when the case has moved approximately 0.250 inch to the rear. When the case has reached this position, it is reasonable to assume that the internal pressure should not be in excess of 750 pounds per square inch, in order to be sure that the case

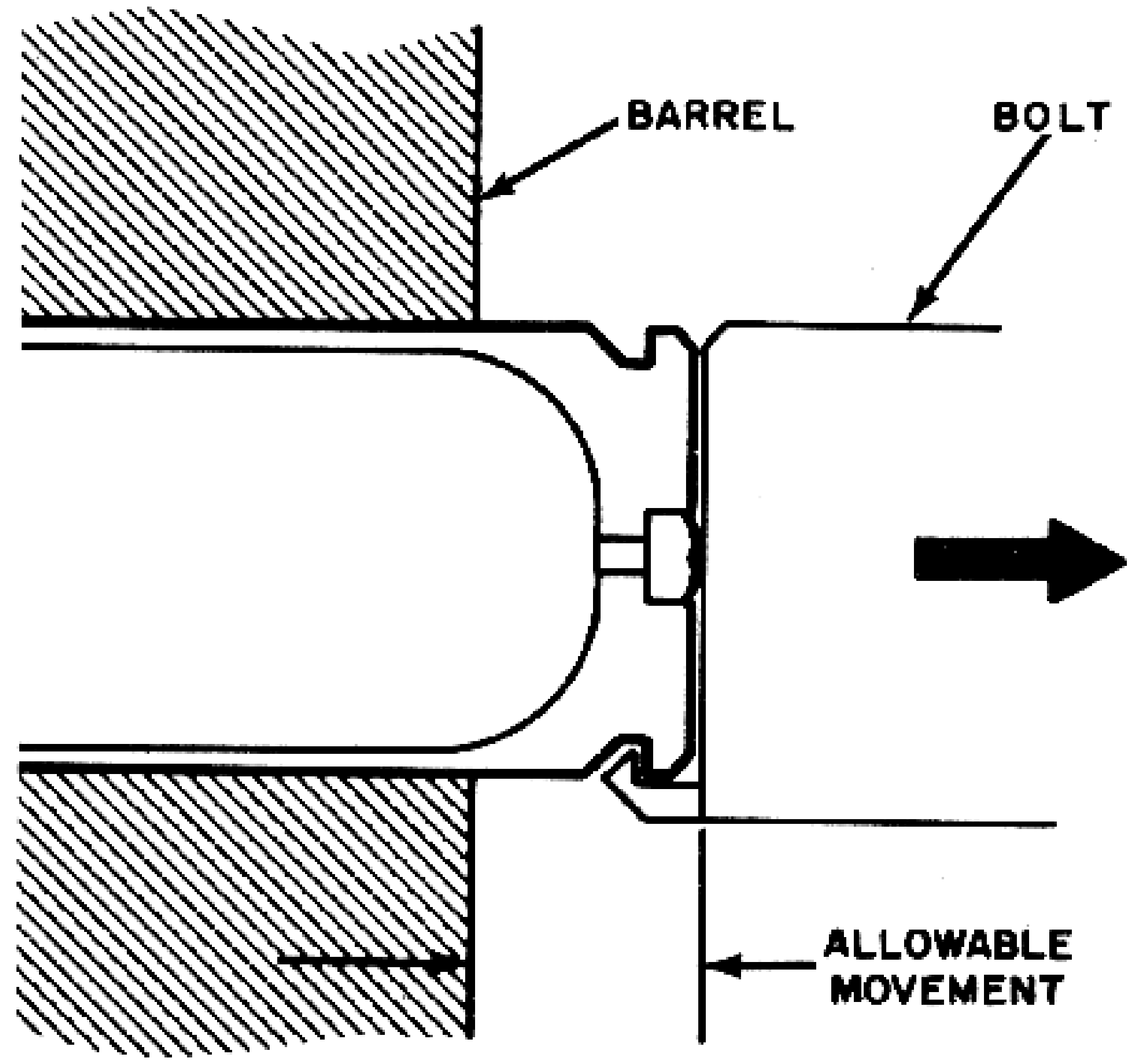


Figure 3–9. Limit of Cartridge Case Movement Rearward Before Residual Pressure Reaches Safe Distance.

will not be ruptured. Fig. 3–10, which is a graph showing the residual pressure variation with time for the assumed gun and cartridge, indicates that the pressure does not fall to 750 pounds per square inch until 0.005 second after ignition of the propellant charge. (Of course, in any practical application, the actual limiting values for a cartridge case should be determined by experiment.)

Having a bolt with a given weight, the bolt movement can be limited as required only by selecting the proper instant for unlocking. If the bolt is unlocked too soon, it will receive too great an impulse from blowback and its average velocity will be so great that the allowable movement will be exceeded before the pressure drops to the safe limit. If unlocking is delayed too long, the impulse imparted to the bolt will be unnecessarily small and the full benefit of blowback will not be realized. The ideal unlocking time for a given bolt weight is that which will permit the bolt to move the full allowable distance and no more by the time that the pressure has dropped to the desired level. After the pressure becomes less than this value, there is no further danger of case rupture and the movement of the case need not be limited. In fact, from this point on, there is no reason for limiting the bolt velocity except to avoid excessive violence of action and exorbitant breakage of parts which would occur

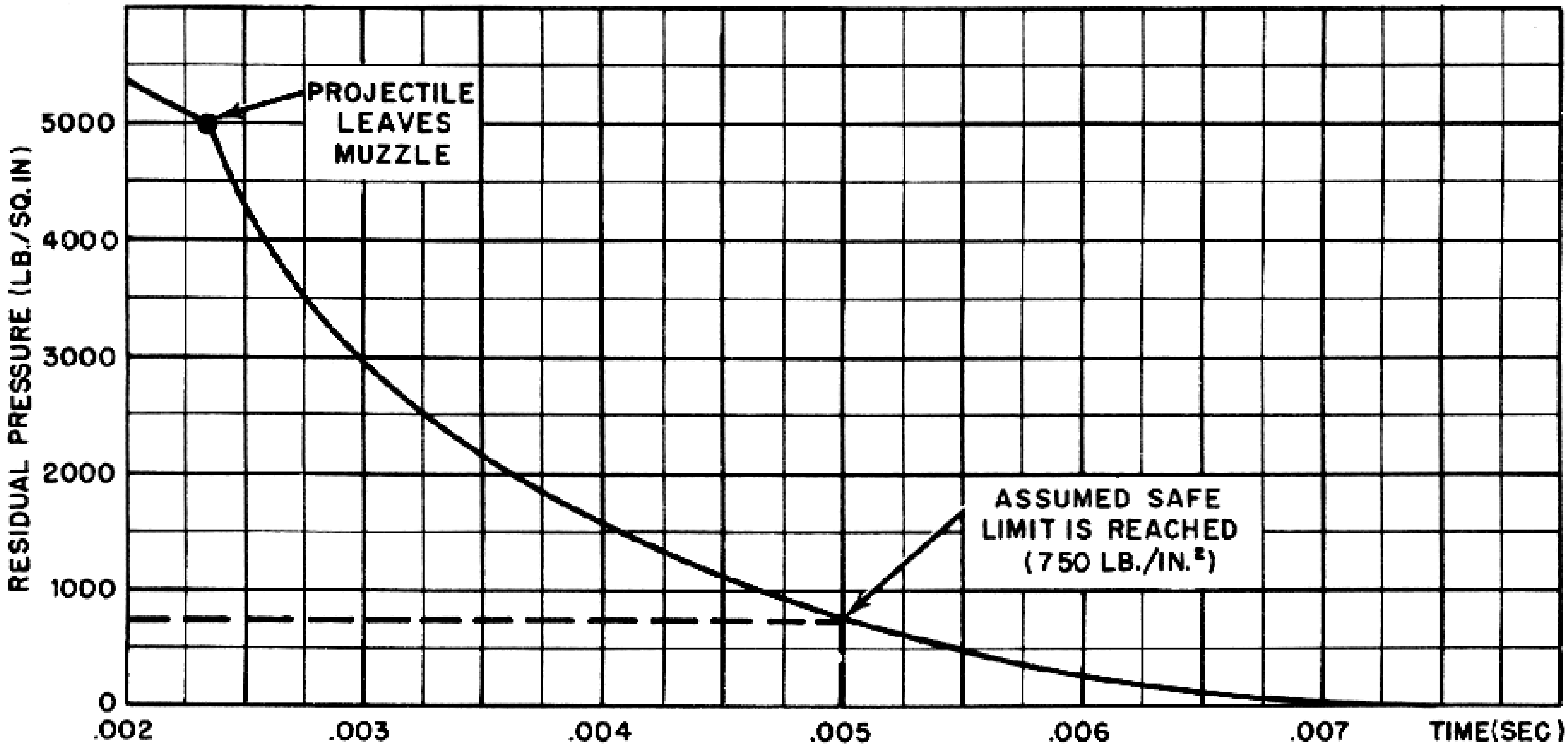


Figure 3–10. Graph of Residual Pressure Versus Time for a Typical 20 mm Gun.

during the reversal of the bolt motion at the end of the recoil stroke.

Since the object of utilizing blowback is to impart velocity to the bolt, it is of interest to determine what conditions lead to the attainment of high blowback velocities. The problem involved in determining these conditions is how to obtain the high velocity and yet not exceed the allowable bolt movement (0.250 inch for the assumed conditions). If the time in which this movement is accomplished is long, the average velocity of the movement will necessarily be low. However, if the time for the movement is made very short, the average velocity may be very large. For example, suppose the bolt is unlocked 0.002 second before the safe pressure is reached. The bolt can now travel the 0.250 inch in 0.002 second. That is to say, its average velocity for this interval must be limited to:

$$V_{av}=\frac{D}{t}=\frac{.250}{12}\times\frac{1}{.002}=10.4\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

Now, assume that the bolt is unlocked only 0.001 second before the safe pressure is reached. It can then travel the 0.250 inch at an average velocity of:

$$V_{av}=\frac{D}{t}=\frac{.250}{12}\times\frac{1}{.002}=20.8\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The preceding examples indicate that, by shortening the time for which blowback operates before the safe pressure is reached, increased bolt velocity can be achieved without exceeding the allowance bolt movement.

Of course, it should be realized that shortening the time of blowback action reduces the blowback impulse available for producing the velocity. Therefore, in order to gain an increase in allowable average velocity by reducing the time of action, it is necessary to reduce the bolt weight to make up for the reduction in the impulse. To illustrate this point, if the velocity of 10.4 feet per second cited in the first of the preceding examples was obtained with an 8-pound bolt, it would be necessary (with the same cartridge and gun) to reduce the bolt weight by a factor of at least 4 to obtain the velocity of the second example, even if it is assumed that the average blowback pressure is the same for both examples. Actually, since the residual pressure decreases with time, the average pressure for the second example would be considerably less than for the first and therefore a further reduction in bolt weight would be required. The actual weight reduction factor would be more nearly in the neighborhood of 6, giving a bolt weight of only 1.3 pounds. (This would probably be much too light for a practical 20-mm gun.)

Thus it appears that a substantial gain in average bolt velocity can be achieved by reducing the time of the blowback action, but this can be done only if the bolt weight is reduced. Although it is not practical to attempt to reduce bolt weight by an excessive amount, it is important to realize that efficient utilization of blowback in a gas-operated gun can be of great advantage in attaining the high bolt velocity necessary for a high rate of fire. However, this advantage can be gained only through precise timing of unlocking in combination with careful attention to minimizing bolt weight.

Having analyzed the factors to be taken into account in determining the time for unlocking, the next point to consider is the timing of the transfer of energy from the gas piston to the bolt. In establishing the timing for this transfer, it is desirable to keep in mind the requirements for effective utilization of the available blowback action. As previously pointed out, it is possible to choose the time of unlocking so that the maximum obtainable impulse is imparted by the residual pressure, this impulse being limited only by the requirement that the bolt movement must not exceed a definite distance (0.250 inch in the example) before the residual pressure has reached a safe limit (750 pounds per square inch at 0.005 second for the assumed conditions). Now, assuming that the time of unlocking is selected so as to take full advantage of this maximum impulse, it can be seen that permitting the piston to act on the bolt before the safe pressure is reached will impart an additional bolt velocity which will cause the allowable movement to be exceeded while the pressure is still dangerously high. The only way to correct this difficulty would be to delay further the time of unlocking in order to reduce the blowback impulse. However, this would mean that full advantage is not being taken of the available blowback impulse and the energy thus lost must be obtained from the piston. From the standpoint of energy alone, this is perhaps not of critical importance because the gas actuating device can be designed to produce almost any desired amount of energy. Nevertheless, it is important to realize that blowback impulse is, in a manner of speaking, "free of charge", while obtaining additional energy from the piston can be accomplished only by paying the full price in terms of increased difficulty in the design of the entire gas mechanism.

From the foregoing, it is apparent that the timing of the piston action should be arranged so that unlocking and initial extraction occur at the correct instant to permit the maximum use of the available blowback action. Then, after a sufficient delay to allow the chamber pressure to fall to the safe operating limit, the piston should start to transfer energy to the bolt. Timing the piston action in this manner will result in the most efficient use of available energy and in the least strain on the operating parts of the gas actuating device. The actual timing of the piston action is controlled by the distance the piston travels in performing its functions. Fig. 3–11 illustrates schematically the basic principles involved in this timing. As soon as the projectile passes the gas port, pressure builds up rapidly in the gas cylinder and drives the piston to the rear. The free piston travel from point A to point B is proportioned so as to allow sufficient time for the projectile to leave the muzzle and for the residual pressure to decrease to the point at which blowback should start. As indicated schematically in the figure, when the piston strikes the locking lever at point B, the bolt is first cammed powerfully back to provide initial extraction and then it is completely unlocked. After unlocking occurs, the free piston travel from point B to point C allows enough time for the blowback action to progress until the residual pressure reaches the safe limit. At this point, the piston strikes the bolt and the resulting impact causes the bolt to be driven rapidly to the rear.

Since the timing of the entire piston action related to unlocking is based on piston travel, it is evident that any factor that affects the manner in which the piston velocity varies with respect to time will have a direct effect on the timing. As has been explained previously, the piston velocity is influenced by various factors such as the conditions of barrel pressure, gas port location, orifice size, piston area, piston mass, and others. To obtain correct timing, it is essential for all of these factors to be controlled closely in the design and to be kept constant during the life of the gun by careful maintenance.

As a matter of practical interest, it should be noted that the problem of obtaining satisfactory timing in a gas-operated gun is often greatly complicated by the fact that the timing is critically dependent on such a large number of factors. Also, since the piston energy is controlled by the same factors which affect the timing, considerable difficulty can be encountered in attempts to improve the performance of a gas-operated weapon merely by making

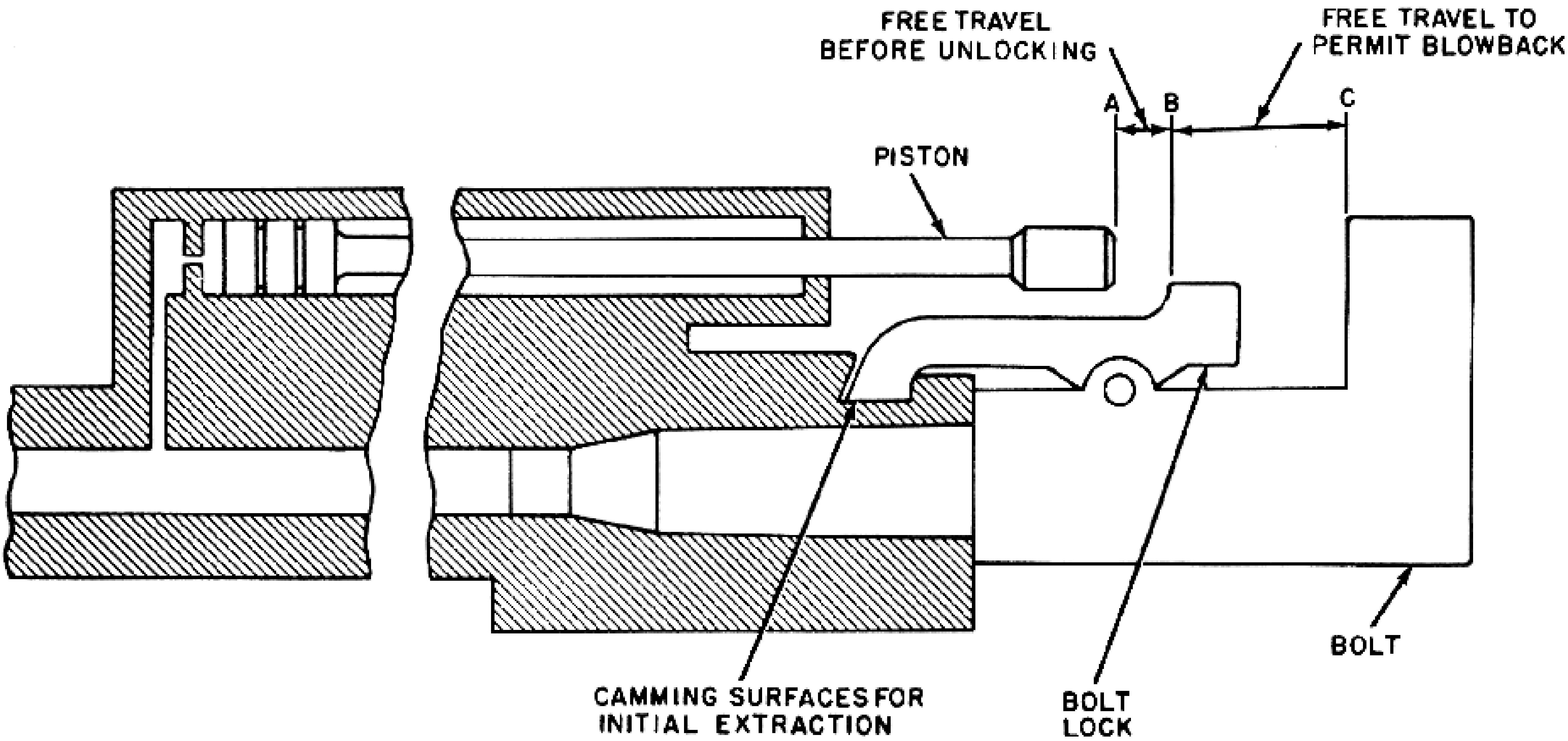


Figure 3–11. Principle of Timing Piston Action.

minor experimental changes to the gas orifice or other individual parts. It will usually be found that to accomplish any real improvement, a fairly extensive redesign of the entire gun will be required.

The analysis up to this point has been concerned with the basic factors affecting the amount of piston energy obtainable and the timing of the piston travel. The next subject for consideration is related to the conditions under which the piston transfers recoil energy to the bolt in order to speed the bolt rearward. There are a number of things which affect bolt velocity. First, since the entire gun is permitted to recoil for the purpose of reducing the forces on the gun mounting, the bolt is unlocked while all of the recoiling parts are in motion. Although it is sometimes desirable to think of the subsequent bolt velocity and piston velocity as being measured relative to the barrel parts, it is always well to remember that the absolute velocity of the barrel parts changes with time.

The recoil velocity attained before unlocking is dependent on the total mass of the recoiling parts and on the magnitude of the impulse imparted to these parts by the propellant explosion up to the time of unlocking. (The retardation of the barrel spring will have such a small effect up to the time of unlocking that the action of this spring can be ignored for the present.) The magnitude of the impulse imparted by the propellant explosion is determined by the interior ballistics properties of the particular cartridge-and-barrel combination employed. Generally speaking, the more powerful the cartridge and the longer the barrel, the greater will be the impulse. The recoil velocity produced by the impulse is inversely proportional to the weight of the recoiling parts; that is, the lighter the recoiling parts, the higher will be their velocity. Thus, with a powerful cartridge, long barrel, and light recoiling parts, the recoil velocity and recoil energy will be correspondingly high.

In this connection, it is important to realize that the velocity and energy of the recoiling parts play an important part in the design of a gas-operated gun. The tendency in the design of modern automatic cannon is toward higher powered cartridges, greater rates of fire, and reduced weight. All of these factors result in greater recoil energies and aggravate the problems involved in controlling the gun motions. In a gas-operated gun, the recoil energy imparted to the gun itself is not used and especially heavy buffers must be provided to dissipate and control this excess energy. Although it is true that by proper timing of the cycle of operation it is possible to make some advantageous use of the recoil velocity inherited by the bolt, this advantage is more than offset by the difficulties resulting from the excess energy in the other recoiling parts. Ideally, it would be desirable for a gas-operated gun not to recoil at all or at least to recoil with a low velocity in order to minimize the problem of energy dissipation. In such a case, entire dependence for the production of bolt velocity could be placed on the piston and blowback. Unfortunately, the requirements of modern gun design as stated above, make a high recoil energy unavoidable, and the designer of the gas-operated gun is required to make the best of the situation. (From the standpoint of efficient design, it is somewhat paradoxical that, when the gas system is employed, a considerable amount of energy is accumulated, this energy is then discarded without being used, and finally energy for operating the gun mechanism is obtained from another source.)

After unlocking occurs and just before the piston strikes the bolt, the gun itself, the gas piston, and the bolt are all moving to the rear. The piston is moving at much higher velocity than the gun because of the action of the gas pressure on the relatively light piston mass. The bolt is also moving at a higher velocity than the gun because of the effect of blowback. When the piston strikes the bolt, the impact causes the piston velocity to decrease. The factors involved in this action can be seen clearly by considering the fundamental mathematical relationships affecting the conditions of impact. These relationships are expressed as follows:

$$M_1V_1+M_2V_2=M_1V'_1+M_2V'_2 \tag{3-3}$$

$$V'_2-V'_1=e(V_1-V_2) \tag{3-4}$$

Equation 3–3 is based on the assumption that the spring forces, friction, or other forces acting during the impact are negligible when compared to the impact forces. Therefore the net impulse received during the period of impact is zero and the total momentum of the system remains the same. M_1 and M_2 are the masses of the piston and bolt, V_1 and V_2 their velocities before impact, and V'_1 and V'_2 their velocities after impact. Equation 3–4 states that the relative velocity of separation of two bodies after impact is directly proportional to their relative velocity of approach before impact. The constant of proportionality is the coefficient of restitution, e, and the value of this constant depends on the materials (approximately 0.55 for steel).

Solving equation 3–4 for V'_1 gives:

$$V'_1=V'_2-e(V_1-V_2)$$

Substituting the result in equation 3–3 gives:

$$M_1V_1+M_2V_2=M_1[V'_2-e(V_1-V_2)]+M_2V'_2$$

Solving for V'_2:

$$V=\frac{M_1V_1+M_2V_2+e\,M_1(V_1-V_2)}{M_1+M_2} \tag{3-5}$$

This equation may be rewritten in the following form:

$$V=\frac{V_1(1+e)}{1+\frac{M_2}{M_1}}+\frac{V_2}{1+\frac{M_1}{M_2}}-\frac{eV_2}{1+\frac{M_2}{M_1}} \tag{3-6}$$

Now if V_2 is zero or is very small when compared with V_1, this equation reduces to:

$$V'_2=\frac{V_1(1+e)}{1+\frac{M_2}{M_1}} \tag{3-7}$$

Examination of equation 3–7 alone might lead to the erroneous conclusion that the piston mass M_1 should be heavier than the bolt mass M_2 in order to produce a high bolt velocity V'_2, or even to the conclusion that the final bolt velocity depends only on the ratio between the masses. However, it must be remembered that the piston velocity V_1 imparted by the gas pressure will be inversely proportional to the piston mass. This factor can be taken into account by setting V_1 equal to I/M_1, where I is the impulse of the powder gases on the piston. It is best to make this substitution in the complete equation 3–6 giving:

$$V'_2=\frac{I(1+e)}{M_1+M_2}+\frac{V_2}{1+\frac{M_1}{M_2}}-\frac{eV_2}{1+\frac{M_2}{M_1}} \tag{3-8}$$

Now it can be seen that to obtain a high bolt velocity starting with a given impulse on the piston, both the piston mass and the bolt mass should be held to a minimum and furthermore the last two terms of the equation show that it is actually better from the overall viewpoint to have the bolt heavier than the piston. This is the reverse of the conclusion which might have been drawn from equation 3–7 alone.

In order to illustrate the conditions involved in the transfer of energy, the following example is given:

Let $W_1=1.5$ (lb.) $V_1=80$ (ft./sec.)

$W_2=5.0$ (lb.) $V_2=20$ (ft./sec.)

$e=0.55$ (for steel)

$$V'_2-V'_1=e(V_1-V_2)=.55\ (80-20)=33.0$$

Therefore $V'_1=V'_2-33.0$

Now, $M_1V_1+M_2V_2=M_1V'_1+M_2V'_2$

or $W_1V_1+W_2V_2=W_1V'_1+W_2V'_2$

$$1.5\times80+5\times20=1.5V'_1+5V'_2=1.5(V'_2-33)+5V'_2$$

$$6.5V'_2=269.5$$

$$V'_2=41.4\ \text{(ft./sec.)}$$

$$V'_1=8.4\ \text{(ft./sec.)}$$

Thus, it appears that for the sample condition, the impact between the piston and the bolt causes the bolt velocity to increase from 33.0 feet per second to 41.4 feet per second and causes the piston velocity to decrease from 80 to 8.4 feet per second.

It is also desirable to consider the energy effects of the impact. The kinetic energy lost by the piston is equal to:

$$E=\frac{1}{2}M_1(V_1^2-V'^2_1)=\frac{1}{2}\times\frac{1.5}{32.2}(80^2-8.4^2)$$

$$=147.4\ \text{(ft. lb.)}$$

The kinetic energy gained by the bolt is equal to:

$$E=\frac{1}{2}M_2(V'^2_2-V_2^2)=\frac{1}{2}\times\frac{5}{32.2}(41.4^2-20^2)$$

$$=102.0\ \text{(ft. lb.)}$$

It should be noted that the loss in piston energy is greater than the gain in bolt energy by 45.4 foot-pounds. This amount of energy is the impact loss and is dissipated in the form of heat. The final piston energy is equal to:

$$E=\frac{1}{2}M_1V'^2_1=\frac{1}{2}\times\frac{1.5}{32.2}8.4^2$$

$$=1.64\ \text{ft. lb.}$$

This remaining energy is also lost because it is not used for accomplishing any useful work. The initial kinetic energy of the piston is equal to:

$$E=\frac{1}{2}M_1V_1^2=\frac{1}{2}\times\frac{1.5}{32.2}\times80^2=149.1$$

Since 122.4 foot-pounds are transferred to the bolt, the efficiency of the piston action is:

$$\frac{102.0}{149.1}\times100=68.4\ \text{per cent}$$

The preceding analysis is primarily concerned with demonstrating the relationships among the various factors affecting the transfer of energy from the piston to the bolt. For the purpose of this demonstration it was assumed that the initial velocities and masses of the piston and bolt were known values. However, an actual design problem usually must be approached from a different point of view. Ordinarily the weight of the bolt will be known from a preliminary design layout, its initial velocity will be known from a motion study, and its velocity after impact will be selected on the basis of the rate of fire requirements. The problem is then to determine what piston weight and velocity are required to produce the desired final bolt velocity. These values can then be used to arrive at the design of the gas actuating mechanism. The procedure to be followed in this case is illustrated by the following example:

Let the following data be known:

Bolt weight, W_2	5 (lb.)
Initial bolt velocity, V_2	30 (ft./sec.)
Desired final bolt velocity, V'_2	60 (ft./sec.)
Coefficient of restitution, e	0.55

(3–9) $V'_2-V'_1=e(V_1-V_2)$

$$60-V'_1=.55V_1-.55\times30$$

(3–10) $V'_1=76.5-.55V_1$

(3–11) $W_1V_1+W_2V_2=W_1V'_1+W_2V'_2$

$$W_1V_1+5\times30=W_1(76.5-.55V_1)+5\times60$$

Solving for V_1:

(3–12) $$V_1=\frac{96.7}{W_1}+49.3$$

Thus, by substituting the known values and solving equations 3–9 and 3–11 simultaneously, an equation is obtained which expresses the necessary relationship between the initial piston velocity V_1 and the piston weight W_1. This equation can be satisfied by an infinite number of pairs of values for W_1 and V_1 and the problem is to select the particular pair of values which best suits the practical requirements of the gas mechanism design. To aid in making the selection it is advisable to tabulate the

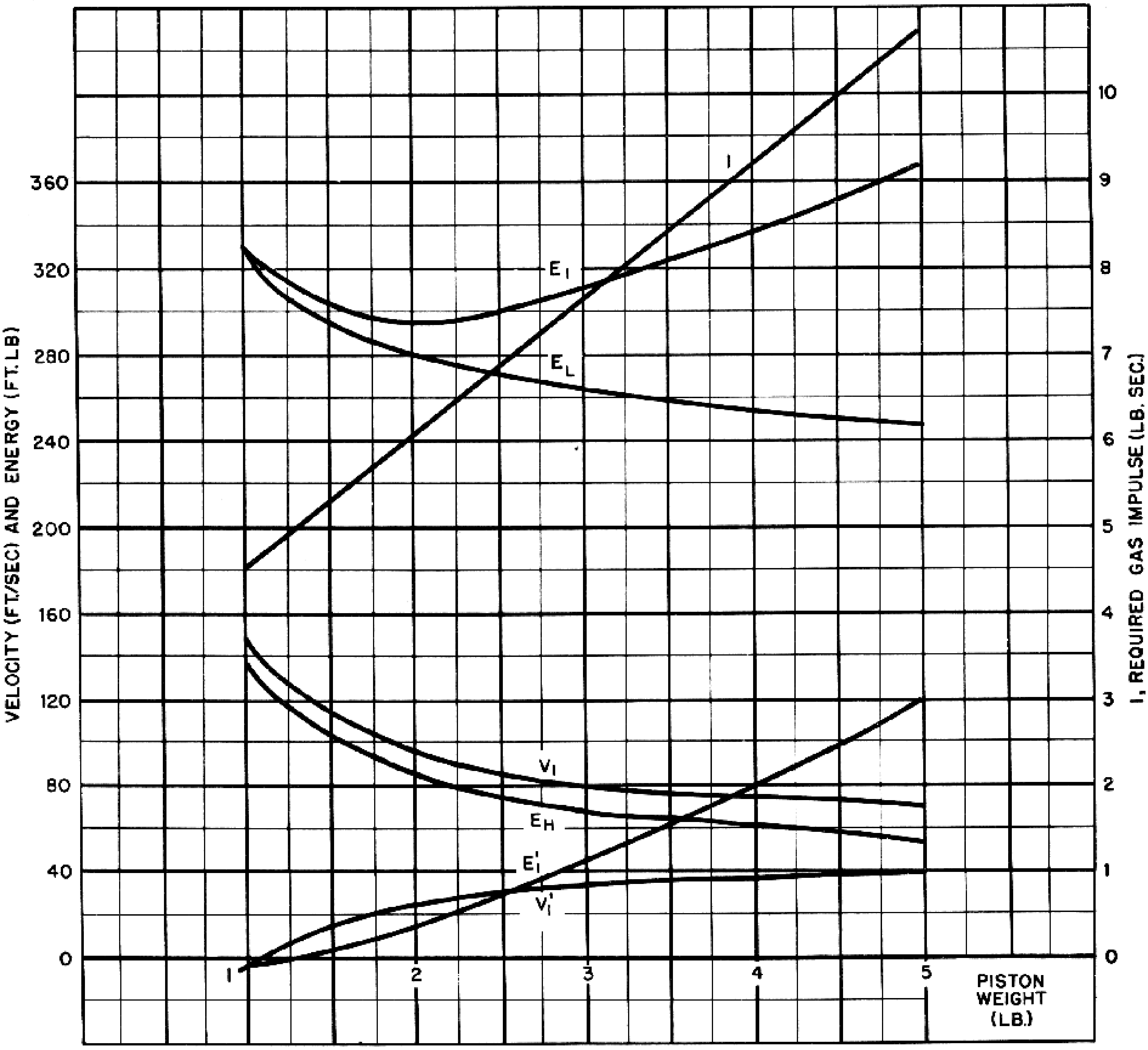


Figure 3–12. Graph of Factors Related to Selection of Piston Weight and Velocity.

corresponding values of W_1 and V_1 along with other related values which influence the design. This has been done in Table 3–1 and the results are shown graphically in figs. 3 12 and 3–13. The other values in the table and graph were found using the following formulas:

Velocity of piston after impact:

$$V'_1 = 76.5 - .55V_1 \text{ (ft./sec.)}$$

Energy lost by piston in impact:

$$E_L = \frac{W_1}{2g}(V_1^2 - V'^2_1) \text{ (ft./lb.)}$$

Initial piston energy:

$$E'_1 = \frac{W_1}{2g} V'^2_1 \text{ (ft. lb.)}$$

Final piston energy:

$$E'_1 - \frac{W_1}{2g} V'^2_1 \text{ (ft. lb.)}$$

Impact loss to heat:

$E_H = E_L - E_G$ (E_G is energy gained by bolt)

Required gas impulse:

$$I = M_1V_1 \text{ (lb. sec.)}$$

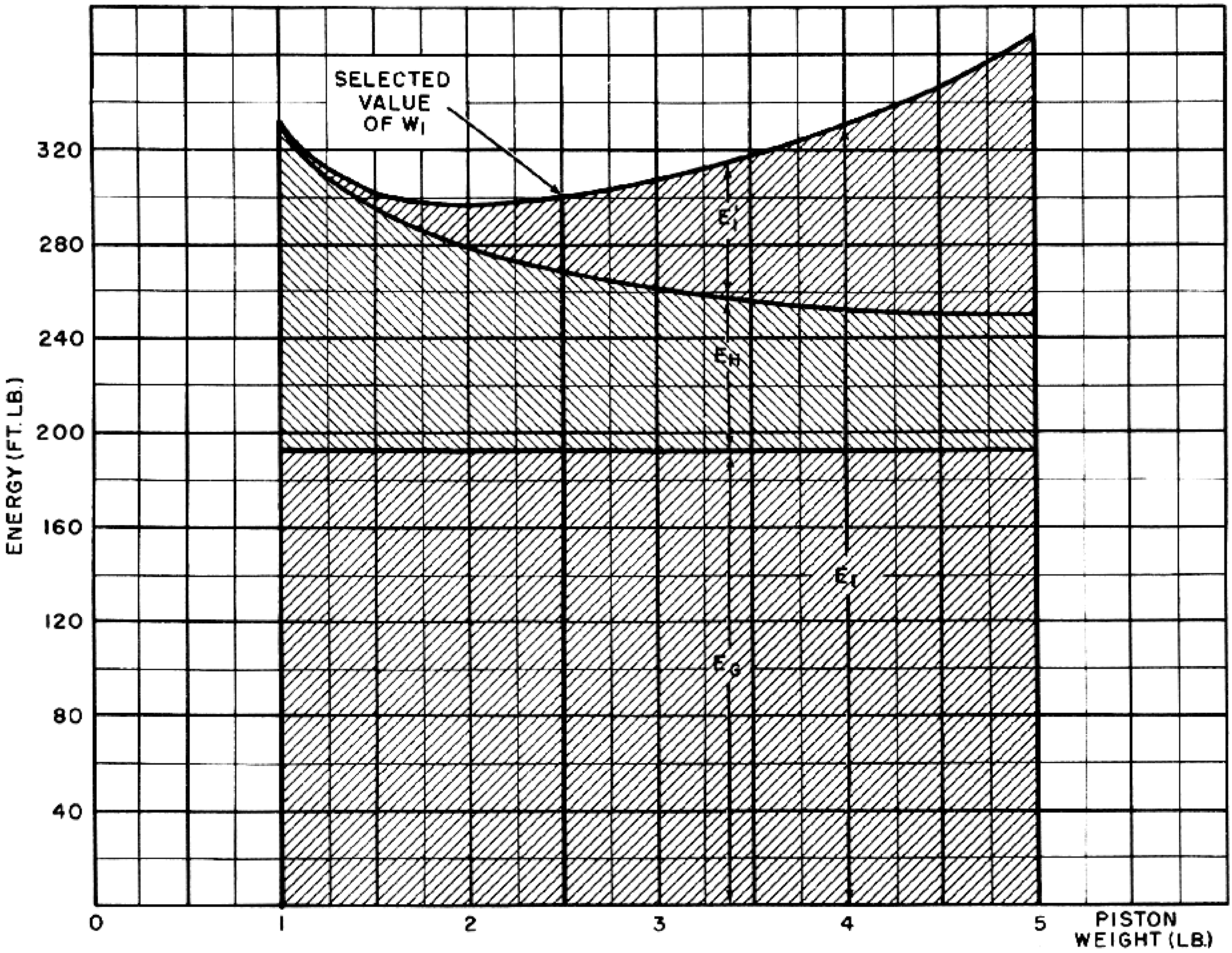


Figure 3–13. Distribution of Piston Energy for Various Weights.

The energy gained by the bolt is the same for all entries and is computed as follows:

$$E_G = \frac{W_2}{2g}(V'^2_2 - V^2_2) = \frac{5}{64.4}(60^2 - 30^2) = 194 \text{ (ft. lb.)}$$

The three items of major interest in Table 3–1 and figs. 3–12 and 3–13 are the energy E_L lost by the piston in the impact, the energy E'_1 remaining in the piston after impact, and the impulse I which must be obtained from the powder gases. The value of E_L is of prime importance because it is this value which indicates the severity of the impact. For the sample conditions, the impact must necessarily be heavy because 194 foot-pounds of energy must be imparted to the bolt, but the total energy involved in the impact must be greater than this value to allow for the impact loss to heat. Note from fig. 3–9 that the value of E_L, which is the sum of the energy transferred to the bolt E_G and the impact loss E_H, increases rather slowly as the bolt weight decreases, but finally begins to increase rapidly when W_1 is less than approximately 1.5 pounds. This would seem to indicate that the piston should be made quite heavy. However, note that, as the piston weight increases, the energy E'_1 remaining in the piston after impact becomes quite large. It is also important to note that the required gas impulse increases rapidly with increase in piston weight. Both of these factors tend to indicate the desirability of choosing a light piston. To make the best of the conflicting requirements, it seems reasonable to choose a piston weight of approximately 2.5 pounds. With this choice, E_L is 270 foot-pounds, only 20 foot-pounds more than for the 5-pound piston. The required gas impulse is only 6.8 pound-

Table 3–1. Values Related to Selection of Piston Weight

W_1	V_1	V_1	E_1	E_1	E_1	E_H	I
1.0	146.0	−3.7	332	332	−0.2	138	4.53
1.5	113.8	14.0	297	302	4.6	103	5.29
2.0	97.6	22.8	280	296	16.15	86	6.06
3.0	81.6	31.6	264	310	46.5	70	7.60
4.0	73.5	36.1	254	335	81.0	60	9.13
5.0	68.7	39.0	249	367	118.1	55	10.68

seconds instead of 10.7 pound-seconds as would be required for the 5-pound piston. This value of 6.8 pound-seconds should be possible of attainment with relative ease, while a value as high as 10.7 pound-seconds might present difficulties. Also note that the piston velocity V'_1 and piston energy E'_1 remaining after impact are reasonable values, as are the heat loss E_H and initial piston velocity V_1.

Having analyzed the factors which are involved in imparting velocity to the bolt, it remains only to consider the motions of the gun mechanism during the completion of the automatic cycle of operation. After the piston has acted on the bolt, the bolt moves to the rear at high velocity, travelling of its own momentum. There is a slight blowback which still acts on the bolt after the piston impact, and this action produces a small increase in the velocity of the bolt motion. The bolt must continue its rearward motion until the opening between it and the barrel is great enough to permit feeding a fresh cartridge. The motion of the bolt must then be reversed to load the fresh cartridge into the chamber and lock the breech. This action of the bolt in a gas-operated gun is similar in some ways to the corresponding action in guns employing other systems of operation and, although the action has been explained previously in this publication, the explanation will be repeated here to avoid the inconvenience of cross-referencing. In some guns, the reversal of the bolt motion is accomplished entirely through the use of a relatively powerful driving spring which is compressed as the bolt moves in recoil. The buffer and spring absorb the kinetic energy of the bolt over the full recoil travel, finally stopping the rearward motion of the bolt when all of the kinetic energy has been absorbed. The spring is designed so that this occurs when the opening is sufficient to permit feeding. The compressed spring then drives the bolt forward to complete the operating cycle. This type of design has a serious drawback from the standpoint of speed of operation. Since the bolt is gradually slowed down by the spring, its velocity varies from maximum at the beginning of recoil to zero at the end of recoil. (See fig. 3–14, a graph showing how the bolt velocity varies with time under these conditions.) The fact that the bolt velocity varies from maximum to zero in the manner illustrated means that its average velocity will only be slightly greater than one-half its maximum velocity. In other words, if this type of action were used in a gas-operated gun, in spite of all the pains taken to achieve a high initial bolt velocity, the overall travel of the bolt would be accomplished at a much lower average velocity.

To overcome this disadvantage, the bolt driving spring can be made relatively very light so that it offers a low retardation and will permit the bolt to move its entire recoil distance with little loss in velocity. In this case, the function of the driving spring is merely to provide a positive force which is just sufficient to insure that the bolt will close. Stopping the bolt at the end of its travel and reversing its motion can be accomplished by causing the bolt to rebound from a so-called "backplate buffer". This device is in effect an extremely stiff spring which absorbs all of the kinetic energy of the bolt over a very short distance and then delivers energy back to the bolt to propel it forward. The reversing action produced by the backplate is so abrupt that the effect may be classified as an elastic impact.

In order to obtain a high rate of fire, it is also important for the bolt return to be accomplished at high velocity. If there were no energy losses involved in the reversing action, the forward velocity

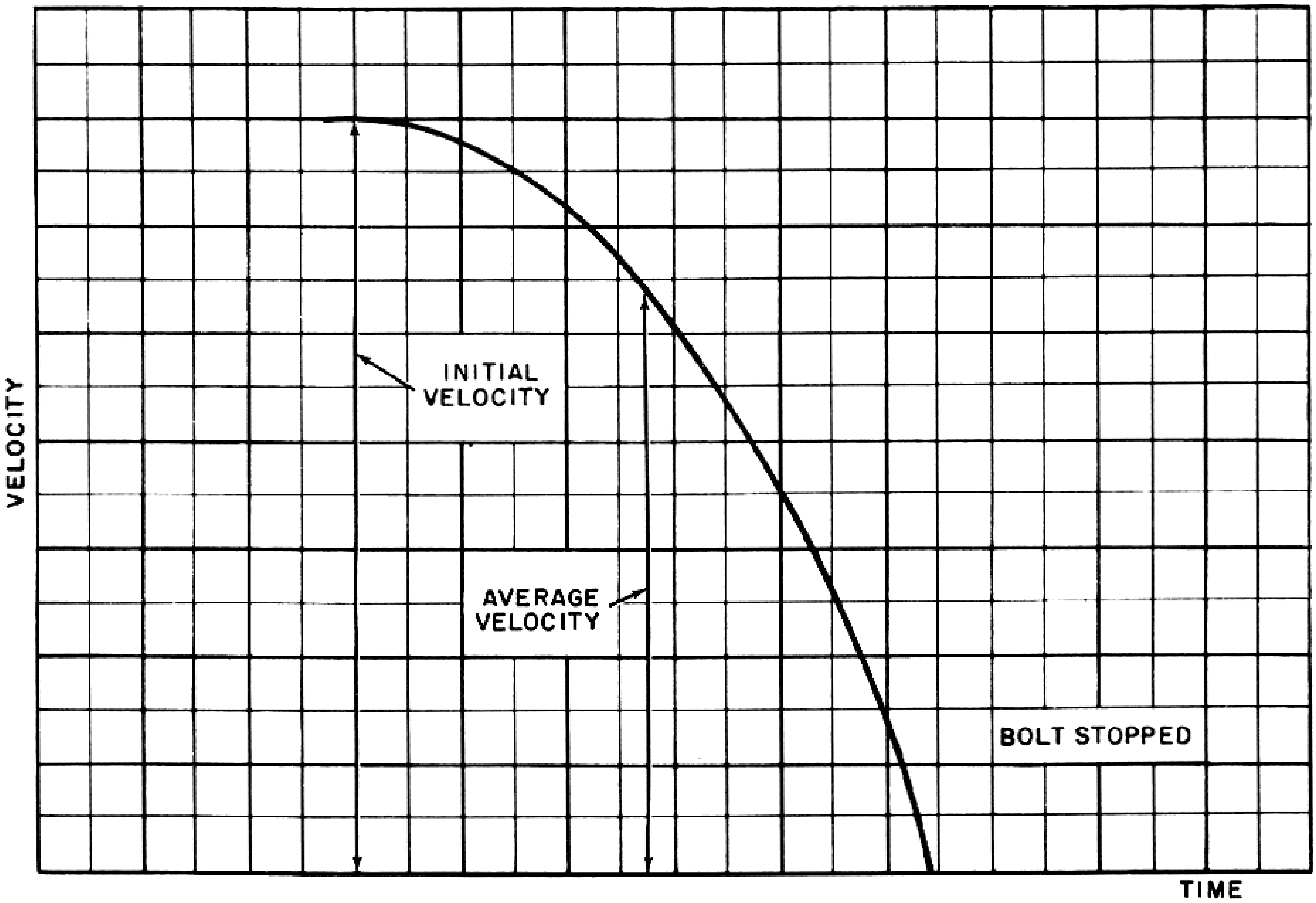


Figure 3–14. Variation of Bolt Velocity With Time When Driving Spring Absorbs All of Bolt Energy.

of the bolt after leaving the backplate would be equal to the velocity at which it strikes the backplate. This would be the ideal condition. However, in actual practice the coefficient of restitution for the bolt and backplate is usually considerably less than unity and the best that can be expected is a coefficient in the neighborhood of 0.60 or 0.70; that is, the velocity after impact will be 60 or 70 per cent of the velocity before impact. This represents satisfactory performance, but if the coefficient of restitution is too low as the result of poor backplate design, the return of the bolt will be sluggish and the rate of fire will be affected adversely. In this connection, it should be emphasized that the purpose of the backplate buffer is to reverse the motion of the bolt with as little loss of energy as possible. In many instances, the term "buffer" is used to refer to a device which has the primary purpose of dissipating impact energy rather than of conserving energy to produce an efficient rebound action. For this reason it might be better to refer to the backplate buffer as a "bolt deflector" or "rebound plate".

The conditions under which the bolt rebounds from the backplate in a gas-operated gun are extremely important and must be considered carefully in the design. In dealing with these conditions, it is essential to make proper allowances for the fact that the entire gun moves in recoil and to time correctly the motions of the various parts. The necessity for careful timing arises principally from the fact that the backplate buffer is mounted on the gun and accordingly moves with the gun in recoil. Since the position of the backplate is not fixed, it is possible to have several different conditions of relative motion and impact, depending on the particular timing used in the design.

The timing involved in this instance is primarily dependent on the motion of the gun in its cradle and is therefore mainly controlled by the action of the barrel return spring and the recoil buffer. If the spring and buffer stop the recoil motion of the gun and start to return the gun to battery before the bolt strikes the backplate, the velocity with which the bolt strikes will be the sum of the bolt velocity measured with respect to the cradle and the forward velocity possessed by the gun at the instant of impact. It also should be realized that motion of the gun in the cradle affects the distance through which the bolt must move with respect to the cradle in order to establish the bolt opening required to permit feeding.

The opposite situation is encountered when the barrel spring and buffer do not stop the recoil motion of the gun until after the bolt strikes the backplate. In this case, the gun is still moving back at the instant of contact and the impact velocity is the difference between the velocity of the bolt measured with respect to the cradle and the rearward velocity possessed by the gun at the instant of impact. Here again, proper allowance must be made for the fact that the motion of the gun in the cradle affects the distance through which the bolt must move with respect to the cradle.

The velocities and movements involved in the preceding timing arrangements would not cause any particular difficulty if the conditions of motion remained the same from shot to shot. However, under practical conditions, some variation in the recoil motion is unavoidable and therefore the relative velocity with which the bolt strikes the backplate will not remain constant and the position of the parts at the instant of impact will also vary. In a high-rate-of-fire gun, each cycle of operation has some effect on the following cycle and under the conditions of timing described above there is a strong tendency for any variation to produce even greater variations. This will cause the gun to quickly "fall out of step", with the associated symptoms of stuttering and extremely erratic action. With the gun operating in this manner, the parts can be subjected to abnormally heavy shocks and excessive parts breakage is certain to result.

A way to avoid malfunctions resulting from variations in the gun recoil movement would be to set up the timing so that the bolt strikes the backplate just at the same instant the recoil motion of the gun is reversed. If this could be arranged, the impact would occur while the gun is not moving and is at a definite position, with the result that the variations described in the preceding paragraph could have no effect. Unfortunately, when an ordinary ring spring buffer is used, it happens that this timing arrangement causes an even more critical condition to exist. With a ring spring type buffer, the velocity of the gun is reduced rapidly to zero and then the action of the compressed buffer immediately drives the gun forward, although with reduced velocity because of the fact that the buffer absorbs a considerable portion of the recoil kinetic energy. Fig. 3–15 is a graphical representation of how the gun velocity varies with respect to time during the action of the buffer. Note that on either side of the instant at which the velocity is zero, the velocity is changing quite rapidly. From this curve, it is evident that even a very slight variation in the instant that the bolt strikes the backplate can have a very great effect on the relative velocity of impact. It appears, then, that if the timing is such that the bolt strikes the backplate during the action of a ring spring buffer, the tendency toward erratic performances is especially pronounced.

The difficulty described above can be avoided by modifying the characteristics of the buffer. The major cause of the trouble is that the ring spring buffer has the steep time-velocity characteristic shown by fig. 3–15. If the buffer is designed to act over a greater interval of time, some improvement will be gained because the change in velocity with respect to time will then not be so abrupt. However, the best results will be obtained if the time-velocity characteristic of the buffer is as shown in fig. 3–16. One way to obtain this type of action is with a hydraulic buffer designed so that its resistance is proportional to the gun velocity, thus producing a viscous damping effect. The slight step in the curve of fig. 3–10 immediately adjacent to the point of zero velocity, is due to the fact that the buffer action terminates in such a manner as to provide the effect of a fixed positive stop. Note that for a considerable interval on either side of the instant of zero velocity, the velocity is nearly zero and changes very slowly. Hence, even if the instant of bolt impact varies slightly over this interval, there will still be very little change in the relative impact velocity. The effect of using this type of buffer is to cause a hesitation which stops the gun and backplate at the same place for each shot and which maintains the gun velocity very close to zero for an adequate time interval. This effect is equivalent to providing the gun with a fixed receiver, thus permitting the conditions of the bolt impact to be kept under precise control.

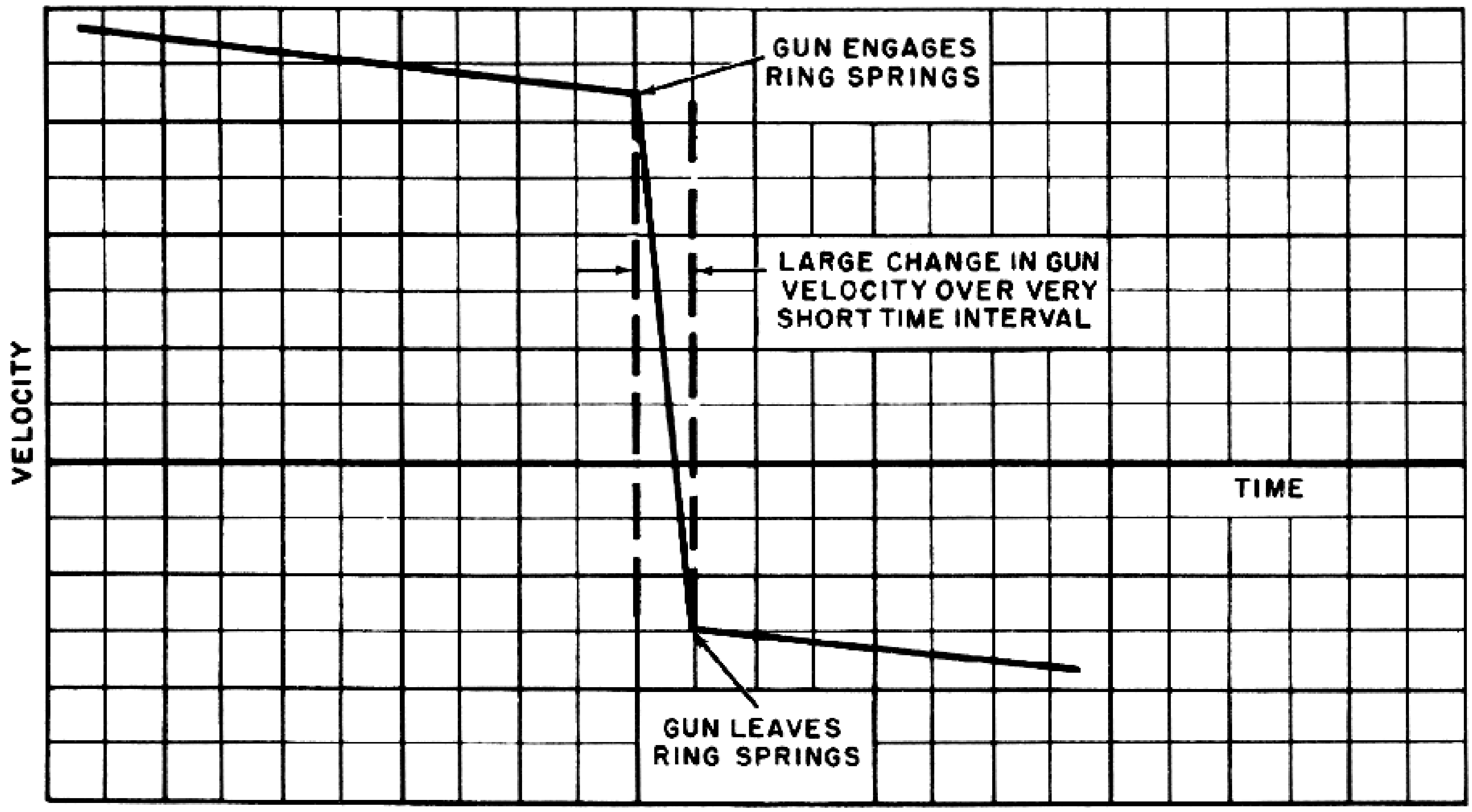


Figure 3–15. Time-Velocity Characteristic of Ring Spring Buffer.

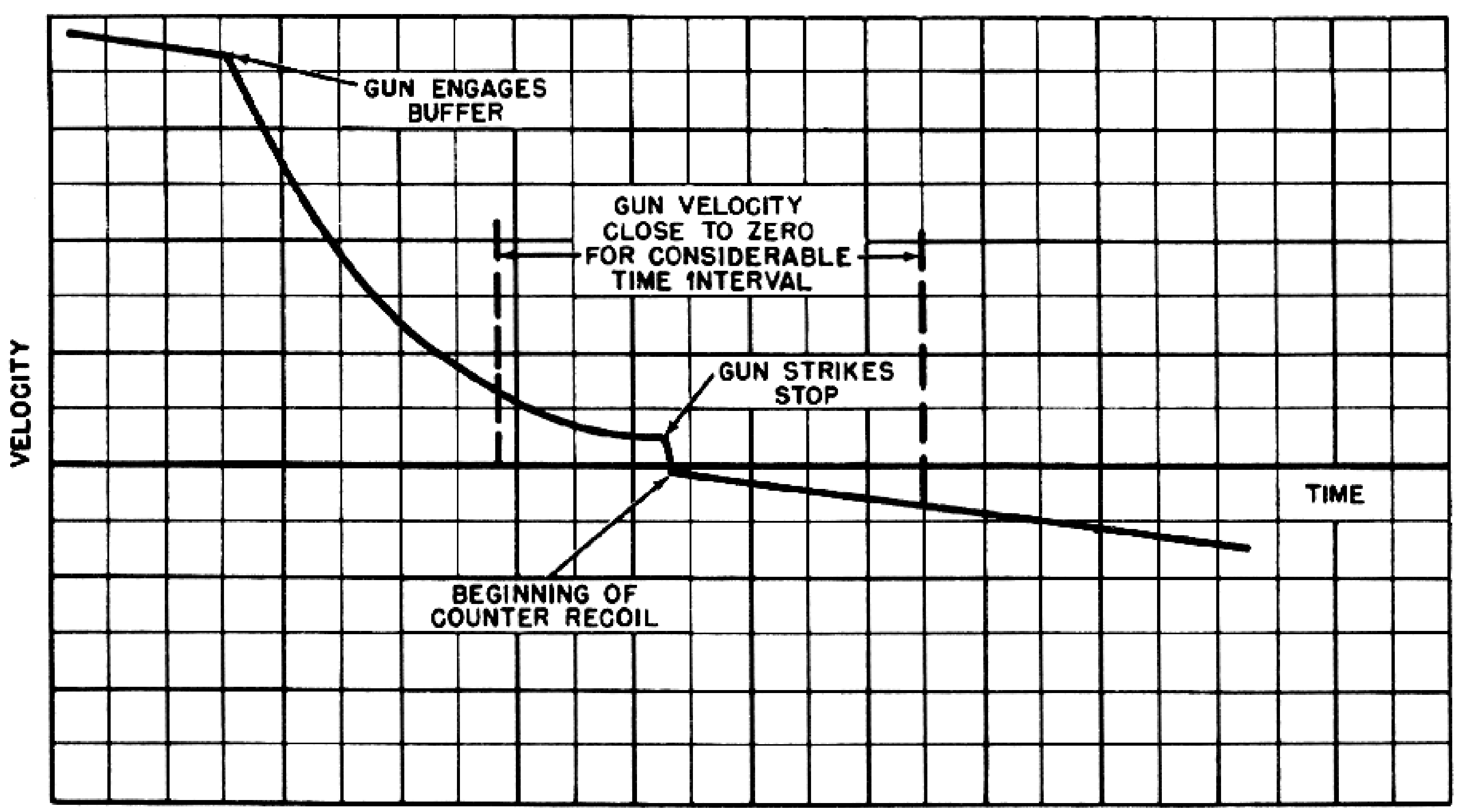


Figure 3–16. Desirable Time-Velocity Characteristic for Recoil Buffer.

The last point to consider in analyzing the action of a gas-operated gun is the forward motion of the gun parts. After the bolt rebounds from the back-plate, both the gun itself and the bolt move toward the battery position, the gun being driven by the barrel spring and the bolt traveling of its own momentum assisted by the action of the bolt driving spring. During the return motion, both the gun and the bolt reach rather high velocities and consequently acquire a large amount of kinetic energy. Unless this energy is properly handled and dissipated, the return of the gun and bolt can result in very damaging shocks and can produce extremely violent oscillations which will interfere with smooth operation at high rates of fire.

To minimize the shock produced when the bolt strikes the barrel, it is desirable to arrange the design so that the bolt will lock to the barrel before the gun has completed its forward movement. With this arrangement, the relative velocity of impact is lower than it would be if the gun were stationary and also the gun is free to yield to the impact.

Now, with the bolt locked to the barrel and with the entire gun still moving forward, the remaining problem is how to handle the kinetic energy contained in the gun as it reaches the battery position. Since the gun will possess a very large amount of kinetic energy, it would not be feasible to stop the forward motion by means of a metal-to-metal impact against a part of the cradle. To avoid severe shock and a violent rebound, it is necessary to provide a buffer action to absorb and dispose of the excess energy in the counter-recoiling gun. This action can be obtained through the use of a heavy-duty buffer which is designed to dissipate practically all of the energy it absorbs.

A very effective method of stopping the forward motion of the gun is to fire the chambered round just before the gun completes its counter-recoil travel. When this is done, the recoil forces generated by the explosion of the propellant charge first stop the gun and then propel it to the rear in recoil. The action of the explosion produces a rapid but very smooth reversal of motion and provides an excellent means of disposing of the excess counter-recoil energy without applying any shock to the gun mounting. Another important advantage of utilizing the propellant explosion in this manner in a gas-operated gun is that it greatly decreases the intensity of recoil. This is true because a fairly large portion of the explosive impulse is expended in stopping the forward motion of the gun and therefore the amount of impulse remaining to cause recoil is reduced. Of course, even if the explosion is employed to provide the counter-recoil buffer action, it is still necessary in guns of large caliber to include a physical buffer to prevent damage to the gun in the event a round fails to fire. Since the presence of the physical buffer is necessary, it may be desirable to time the firing so that this buffer assists in the reversing action.

Mathematical Analysis of Gas Operation

Most of the problems encountered in the design of gas-operated automatic gun can be handled in a straightforward manner by the same general analytical methods used for the other systems of operation described in this publication. These methods lend themselves readily to the analysis of the recoil forces, blowback effect, energy transfers, and spring data and can also be applied directly in developing the theoretical time-travel and time-velocity diagrams. However, there is one problem for which no satisfactory method of analysis is available. This problem concerns the design considerations related to the flow of gases and to the build-up of pressure against the piston or other actuating member. If it were possible, it would be highly desirable to be able to compute exactly what size and shape of orifice should be used, or to know in advance how the configuration of the passage leading from the barrel to the gas cylinder will affect the variation of the cylinder pressure with respect to time. Unfortunately, accurate solutions to these problems and to problems of a similar nature can not be obtained analytically because of the complexities involved in predicting the flow of the turbulent and high velocity gases produced by the propellant explosion.

Although the fact that the problems described in the preceding paragraph can not be solved by mathematical analysis causes a real difficulty for the designer of a gas-operated gun, this difficulty is by no means insurmountable. By making the necessary allowances in the analysis and by the judicious use of available empirical data, it is possible to produce a preliminary design which is safe and yet is arranged to permit convenient modification on the basis of experimental firing. It must be emphasized that this process, when properly carried out, does not amount to "fumbling in the dark" or to making haphazard changes. In the design analysis, the impulse required of the gas actuating device and the required timing for this impulse are carefully determined so as to produce the desired operating characteristics. The object of the experimental firing is then to adjust the factors which affect the impulse and the timing of the impulse so that the required values will be obtained.

The following pages describe a systematic approach to the design analysis of a gas-operated gun. The characteristics chosen for the gun to be used as an example do not represent any existing design, but have been selected to illustrate the principles described in the analysis of gas operation. It is assumed that the gun will be cradle-mounted and will employ a conventional gas piston which transfers energy to the bolt by direct impact. The desired rate of fire will be taken as approximately 1200 rounds per minute. To illustrate the design procedures involved when the propellant explosion is utilized to provide a counter-recoil buffer action, it will be assumed that the gun will be fired just before reaching the battery position. The assumption of these characteristics and of other specific properties of the gun will of course have some influence on the details of the analysis and as a result, the particular methods employed may not be directly applicable in their entirety to other gas-operated gun mechanisms having a different arrangement. Nevertheless, gas-operated guns all function according to the same basic principles and the gun selected as an example should serve to illustrate the approach to a typical design.

The methods which will be used in the analysis follow the same general lines employed throughout this publication with the necessary modifications to adapt the procedure to the gas system of operation. Primary attention will be given only to the factors controlling the motions of the principal operating parts and to the major forces effecting these parts. Here, as in the other parts of this publication, no attempt will be made to cover the conventional methods of machine design by means of which the results of the analysis are applied in arriving at the particular physical form of the mechanisms. Also, no detailed computations are made to cover the effects of such factors as friction or the incidental forces imposed on the actuating device by the auxiliary mechanisms such as the feeder, firing device or locking device. These effects will have only a relatively slight influence on the motions of the main operating parts and, in any case, they can be properly taken into account in the advanced stages of a design when the form of the gun mechanism becomes fairly well established. At this point, the results of the preliminary analysis can easily be modified as desired.

In the design of a gas-operated gun, the weights of the recoiling parts will have a considerable influence on performance characteristics and therefore it is necessary to have certain weight data before the analysis can be started. The weights of the barrel and its associated parts and the weight of the bolt are determined largely by the particular configuration selected by the designer and by the requirements for strength, rigidity, and durability. For this reason, the first step in the design process should be to design the barrel to withstand the forces produced by the selected cartridge and then make a preliminary layout of the entire mechanism. In making this layout, experience and good judgment will aid in arriving at a mechanism of practical proportions and will permit the design to be brought to the point where it is feasible to obtain a fair estimate of what weights will be involved and of what distance the parts will be required to travel. On the basis of these data, it will be possible to perform a preliminary design analysis from which a good approximation of the operating forces can be obtained. Then, if necessary, these forces can be taken into consideration in making adjustments to the design to insure that all parts will have adequate strength. A knowledge of the operating force will also assist in refining the design to eliminate excess weight, particularly in such parts as the bolt and piston where weight has a great influence on the performance of the gun.

The following analysis is based on the assumption that a particular cartridge with known characteristics is to be used and that the desired muzzle velocity and barrel length have been predetermined. It is also assumed that all necessary interior ballistics data are known and that graphs showing the time variation of chamber pressure, projectile velocity, and projectile bore travel are available (figs. 3–17, 3–18, and 3–19).

NOTE: For some design problems, all or part of this information may not be available. Analytical methods by which to approximate the required data and graphs for use in pre-

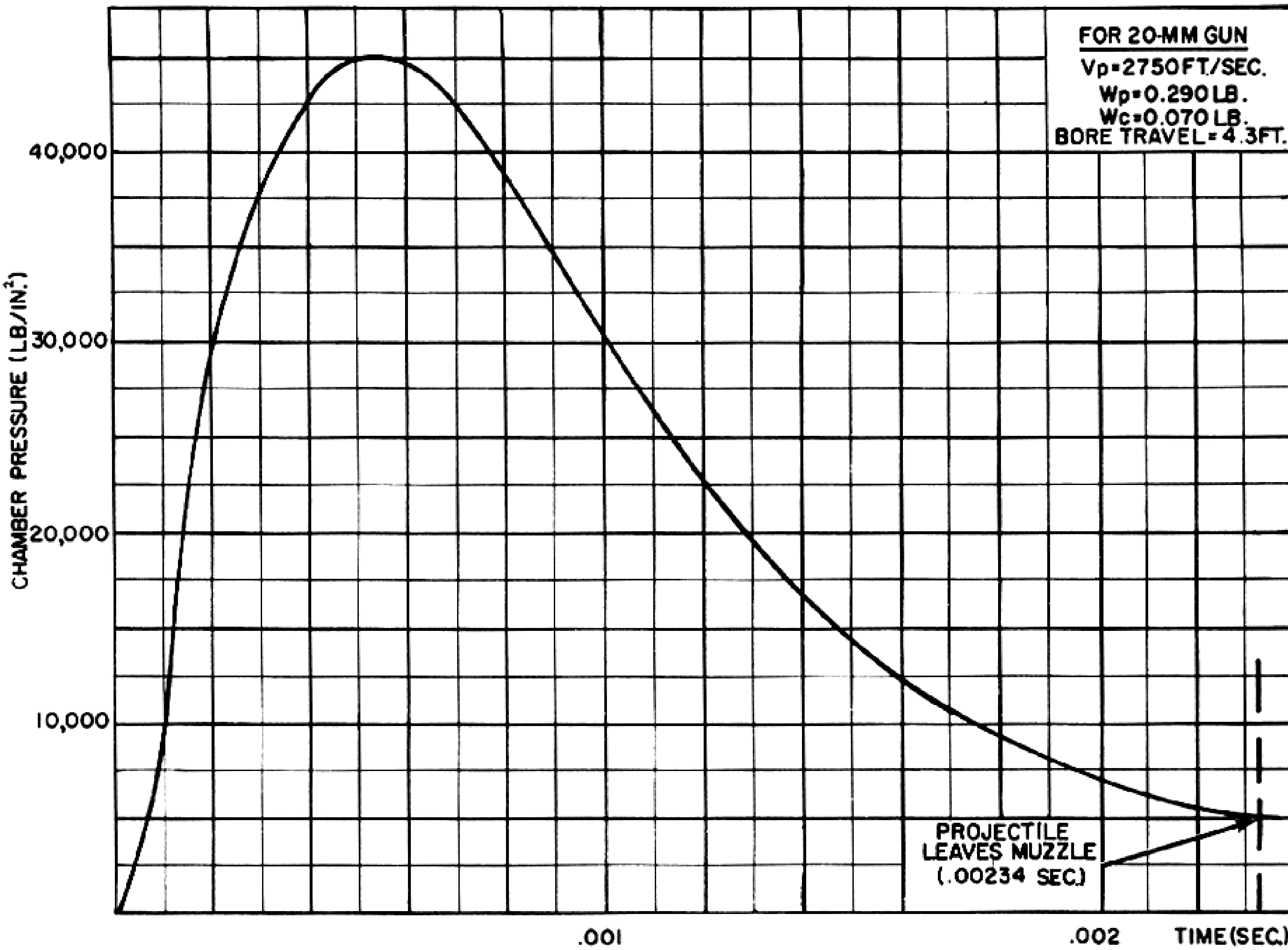


Figure 3–17. Graph of Chamber Pressure Versus Time (20 mm Gun).

liminary studies may be determined by conventional interior ballistics computations.

As the analysis progresses, its applications will be illustrated by means of sample calculations. Although these calculations and the related graphs are for a specific 20-mm cartridge and barrel and are based on certain assumed weights and other characteristics, the general approach described is applicable to gas operated guns of any caliber. The calculations cover the following important points:

1. Determination of the conditions of free recoil.
2. Determination of the correct time for unlocking.
3. Computation of data required for design of the gas actuating device.
4. Determination of data required for timing the operation of the piston.
5. Selection of characteristics of barrel springs and bolt driving spring and determination of data for backplate buffer and buffers associated with barrel spring.
6. Development of graphs show how the velocity and travel of the gun, piston, and bolt vary with respect to time.

In the course of describing these calculations, the following fundamental formulas will be developed and explained.

a. Momentum and velocity relations for time projectile is in bore
b. Formula for determining velocity of free recoil
c. Expression for duration of residual pressure
d. Formulas for determining spring retardations.

(Before proceeding, it should be mentioned that the action of a gas-operated gun up to the time the piston strikes the bolt is very similar from the analytical point of view to the operation of a short-

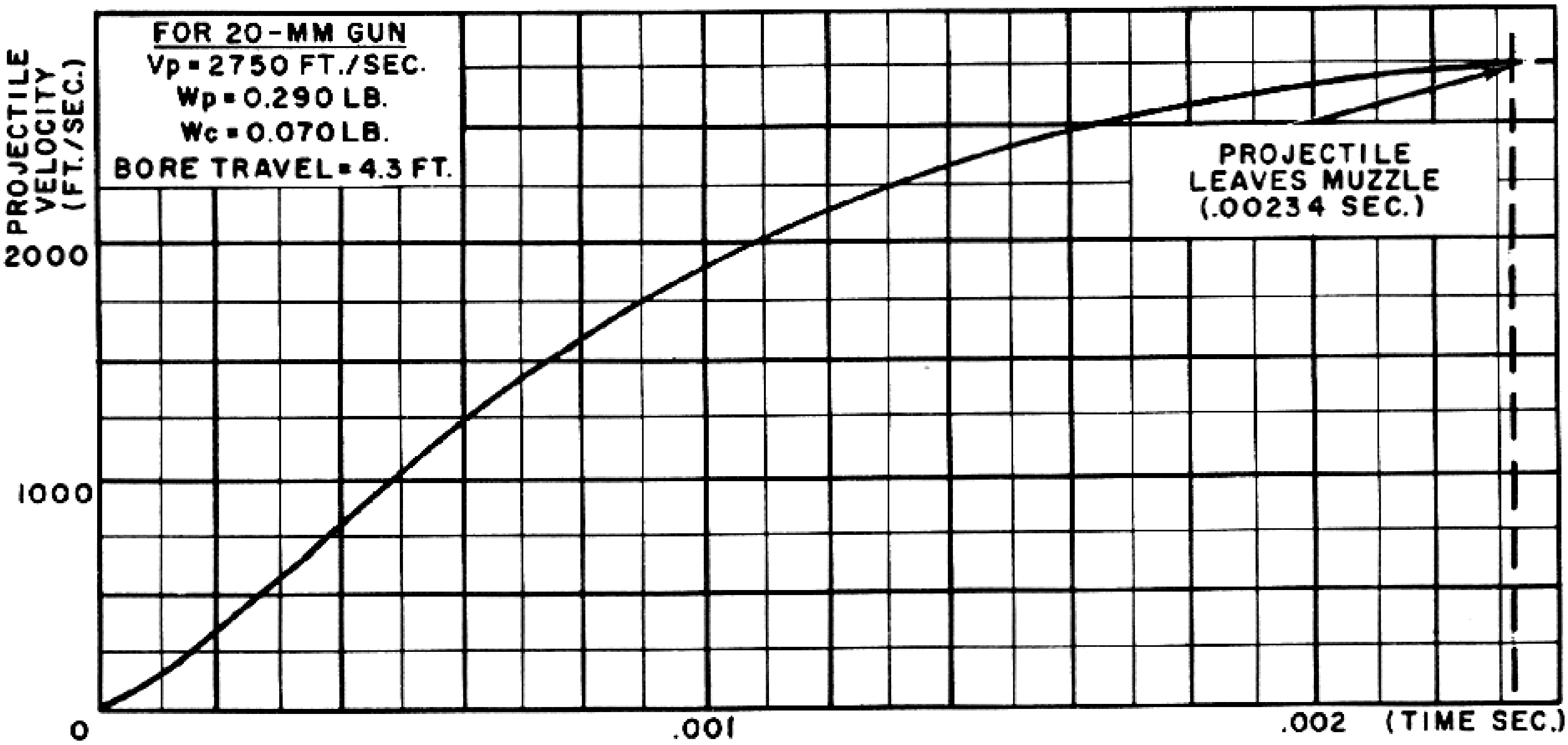


Figure 3–18. Graph of Projectile Velocity Versus Time (20 mm Gun).

recoil gun up to the time the accelerator begins its action. There are certain differences, however, which result from the influence of the gas actuating device. Since these differences make its impractical to refer simply to the applicable portion of the short-recoil analysis, a complete analysis of the entire operating cycle for gas operation will be given here. Because of the close similarity of the two systems at some points in the cycle, a certain amount of repetition of the material covered under short recoil is unavoidable.)

1. *Conditions of free recoil*

Before considering the actual motions of the recoiling parts under the restraint of the barrel spring and bolt driving spring, and before analyzing the effect of firing while the gun is still moving forward, it is necessary in the following method to determine first how the recoiling parts would move under the conditions of free recoil. (For determining the free recoil motion, it is assumed that the gun is fired while stationary and is mounted so that it can move to the rear without friction or any other restraint.) Under these conditions, the impulse of the recoil force will impart to the gun a rearward momentum equal to the forward momentum of the projectile and powder gases. Until the instant the projectile passes the gas port, this momentum relation is expressed by the equation:

(3-13) $$M_r V_{r_f} = M_p V_p + M_c V_c$$

Since the powder gases will be thoroughly mixed by the turbulence created in the propellant explosion, it is reasonable to assume that the center of mass of the gases moves forward at one-half the velocity of the projectile. Actually, this is not quite accurate because the presence of the enlargement at the chamber and the fact that the rifling does not extend the full length of the space occupied by the gases creates a condition in which the volume of the space is not uniformly distributed along its length. Nevertheless, the assumption is close enough for present purposes. Therefore, equation 3–13 may be rewritten as:

(3–14) $$M_r v_{r_f} = M_p v_p + M_c \frac{v_p}{2} = \left(M_p + \frac{M_c}{2}\right) v_p$$

NOTE: It should be pointed out here that the momentum equality expressed by equation 3–14 is not affected by the internal frictional forces opposing the motion of the projectile and powder gases or by the force incident to engraving the rifling band and to imparting the rotational velocity to the projectile. Although all of these forces retard the forward motion of the projectile and powder gases, they produce equal and opposite reactions on the barrel which result in a corre-

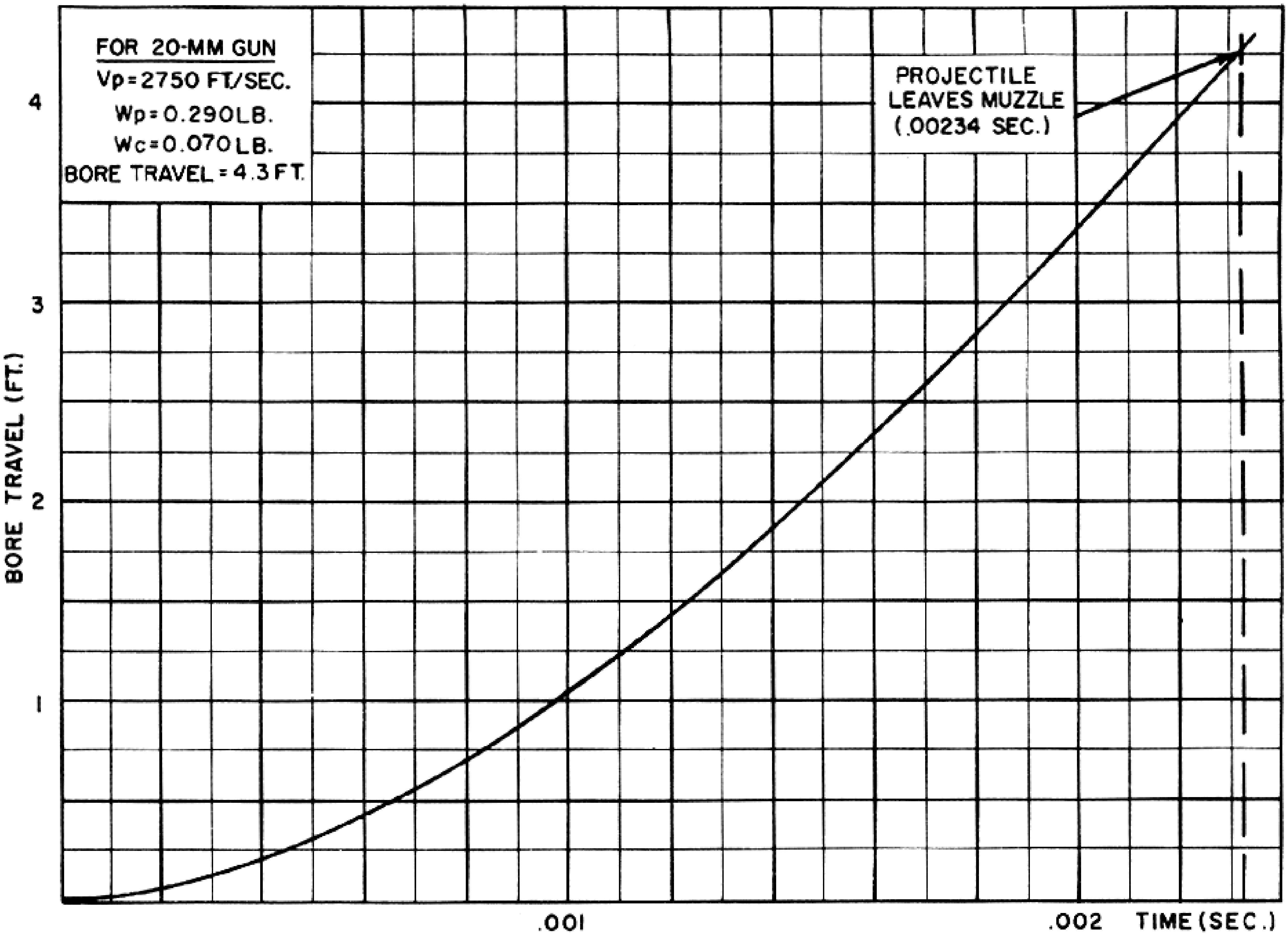


Figure 3–19. Graph of Projectile Bore Travel Versus Time (20 mm Gun).

sponding retardation of the rearward movement of the gun. In other words, the internal resistances merely decrease the effective impulse producing motion but they do not cause any inequality in the forward and rearward momentums.

Solving equation 3–14 for v_{r_f} gives the velocity of free recoil for the time before the projectile passes the port as:

$$v_{r_f} = \frac{M_p + \frac{M_c}{2}}{M_r} v_p = \frac{W_p + \frac{W_c}{2}}{W_r} v_p \qquad (3\text{–}15)$$

Equation 3–15 can be used to plot a curve showing the free recoil velocity versus time for the period before the projectile passes the gas port. The weights of the projectile and powder charge are both known and it is assumed that the weight of the recoiling parts has been estimated from a preliminary design layout. Also, the velocity of the projectile is known from the available ballistic data (fig. 3–18). Therefore, the ordinate of the free recoil velocity curve, t, can be found by multiplying the corresponding ordinate of the projectile velocity curve by the factor:

$$\frac{W_p + \frac{W_c}{2}}{W_r}$$

Assuming that in the 20-mm gun to be used as an example the estimated weight of the recoiling parts is 60 pounds and the weights of the projectile and powder charge are as shown in fig. 3–15, the value of the multiplying factor is:

$$\frac{W_p + \frac{W_c}{2}}{W_r} = \frac{.29 + \frac{.070}{2}}{60} = .00542$$

Therefore, before the projectile passes the gas port, the free velocity of the recoiling parts is:

$$v_{r_f} = .00542\ v_p \left(\frac{\text{ft.}}{\text{sec.}}\right) \tag{3–16}$$

If it were not for the presence of the gas port, this same equation would apply until the instant that the projectile leaves the muzzle. However, after the projectile passes the gas port, the pressure in the gas cylinder rapidly rises and starts to drive the piston to the rear. This pressure acts both rearward on the piston and forward on the front projection of the cylinder bore. Since the piston at this time does not exert any force on the gun (except perhaps through a relatively weak piston return spring), the rearward pressure on the piston does not have any effect on the recoiling gun mass. On the other hand, forward pressure on the front cylinder face is transmitted directly to the gun and acts to oppose the recoil motion. In other words, after the projectile passes the gas port, the pressure in the gas cylinder produces a retarding impulse on the gun which is equal to the impulse imparted to the gas piston.

The actual magnitude of the retarding impulse and the manner in which it develops with respect to time will depend on the operational characteristics of the gas cylinder and on the location of the gas port. At this point in the design, these characteristics are not yet established, so it will be necessary to make a reasonable estimate which can be corrected later if necessary. Experience with weapons of this caliber indicates that the impulse which must be applied to the piston will vary with time approximately as shown in fig. 3–8. Assuming for the present that this curve is applicable to the gun of the example, the free recoil curve for the interval before the projectile leaves the muzzle can be drawn as follows:

Using equation 3–16, a curve is plotted for the interval from t=0 to t=.00234 second as shown in fig. 3–20. This curve is shown dotted after t=.0016 second, at which time the assumed curve of fig. 3–8 indicates that the projectile passes the gas port. For the interval from t=.0016 to t=.00234 second, the changes in recoil velocity produced by the piston impulse are now computed, dividing the ordinates of the curve in fig. 3–8 by the mass of the recoiling parts (assuming that the weight of the piston is negligible when compared to the entire weight of the recoiling parts). That is:

$$\Delta V_{r_f} = \frac{I}{M_r} = \frac{I_g}{W_r} = \frac{32.2I}{60} = .520I$$

This calculation is performed for selected ordinates in the interval and the values obtained are subtracted from the corresponding ordinates of the curve plotted from equation 3–16. The resulting curve is shown in fig. 3–20 and is also shown in fig. 3–21. (In fig. 3–21, the time axis is compressed to show how the velocity varies after the projectile leaves the muzzle.)

The manner in which the free recoil velocity varies after the projectile leaves the muzzle can not be determined from equation 3–16 because the projectile

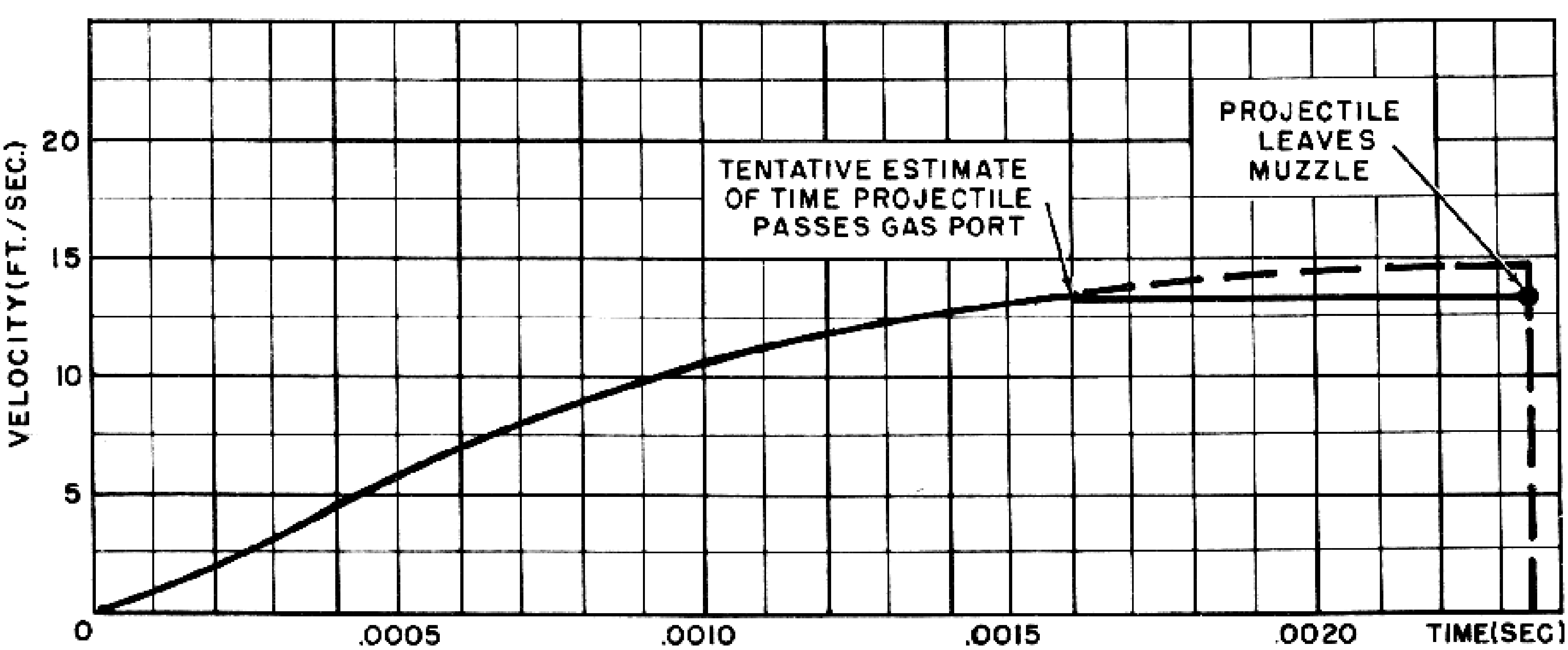


Figure 3–20. Free Recoil Velocity While Projectile Is in Bore.

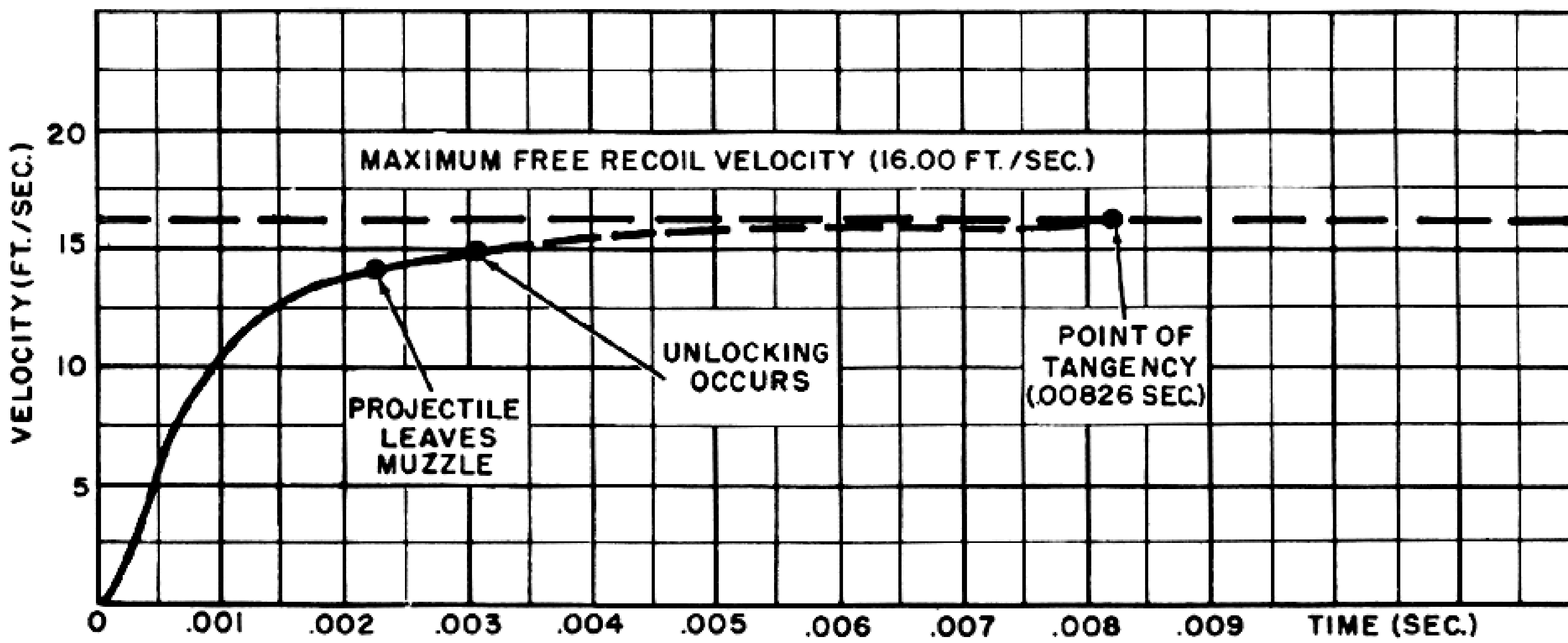


Figure 3–21. Free Recoil Velocity Before Unlocking.

and a portion of the powder gases then are no longer part of the recoiling system. Since the effect of the residual pressure can not be expressed in simple terms, a special method is used to extend the curve obtained by using equation 3–16 and fig. 3–8. This method is based on the fact that the results of experimental firings of various guns show that the maximum velocity of free recoil may be closely approximated as:

$$V_{r_f}=\frac{W_pV_p+4700\ W_c}{W_r} \tag{3–17}$$

This relationship is equivalent to saying that the maximum momentum imparted to the recoiling parts is equal to the sum of the muzzle momentum of the projectile and the momentum of the powder gases, assuming that the powder gases leave the gun at an average velocity of 4700 feet per second.

Although equation 3–17 gives a good approximation of the maximum free recoil velocity for most ordinary guns, it does not take into consideration the retarding effect of the total impulse produced by the particular gas mechanism assumed for this design. It is therefore necessary to subtract the velocity change produced by this impulse from the right side of the equation as follows:

$$V_{r_f}=\frac{W_pV_p+4700\ W_c}{W_r}-\frac{I}{M_r} \tag{3–18}$$

$$=\frac{W_pV_p+4700\ W_c-I_g}{W_r}$$

where it is assumed that the piston mass is negligible when compared with the total weight of the recoiling parts. The assumed curve of fig. 3–8 shows that the total piston impulse is equal to 5.13 pound seconds. Evaluating equation 3–18 for the gun of the example gives:

$$V_{r_f}=\frac{.29\times2750\times4700\times.070-5.13\times32.2}{60}$$

$$=16.00\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

A line representing this value of the maximum velocity of free recoil is drawn on the velocity graph (fig. 3–21) and the curve previously drawn from equation 3–16 and fig. 3–8 is extrapolated until it becomes tangent to the line. The point at which the curve becomes tangent represents the time at which the residual pressure becomes zero and therefore imparts no further velocity to the recoiling parts. Although an error in locating the exact point of tangency will not have any serious effect on the accuracy of the results, it may be of some assistance in drawing the curve to determine this point by using Vallier's formula for approximating the duration of the residual pressure:

$$T_{res}=\frac{M_c}{AP}\,(9400-V_p) \tag{3–19}$$

For the sample cartridge and barrel:

$$T_{res}=\frac{.070}{32.2\times\frac{\pi}{4}\left(.790\right)^2\times5000}$$

$$(9400-2750)=.00592\ (\text{sec.})$$

To obtain the total time of action of the powder gases, this value is added to the time at which the projectile leaves the muzzle:

$$T_{res} = .00234 + .00592 = .00826 \text{ (sec.)}$$

Extending the original curve until it is tangent to the maximum free recoil velocity line at this point gives the complete free recoil velocity curve shown in fig. 3–21. Actually only a portion of the curve shown in the figure applies to the recoil conditions in a gas-operated gun because unlocking occurs before the residual pressure has become zero. It also must be remembered that the curve obtained by the preceding method must be checked after the actual piston impulse curve is obtained.

2. *Effect of blowback before piston strikes bolt and computation of unlocking time*

The next point for consideration is the effect on the bolt velocity of the blowback action which occurs between the time that the bolt is unlocked and the time that the piston strikes the bolt to speed it rearward. As pointed out in the analysis of gas operation, the ideal condition for this portion of the blowback action is that the bolt should move 0.250 inch with respect to the barrel by the time that the residual pressure has dropped to the safe limit of 750 pounds per square inch. (These figures are based on assumed safe values for a typical 20-mm cartridge and should be checked experimentally for any specific cartridge.)

For purposes of determining the blowback effect, it is only necessary to consider the velocity of the bolt with respect to the gun. For this determination, it is necessary to know the bolt weight. It will be assumed here that the bolt weight, as estimated from the preliminary design layout, is equal to 5 pounds. After the bolt is unlocked, the residual pressure continues to act on the bolt, but since the bolt is now free of the gun, the recoil force exerted on the gun by the residual pressure is reduced to a negligible value. (The gases expanding at the muzzle do exert some force on the muzzle face of the barrel, but in the absence of a gas trap, such as exists when a muzzle booster is used, the impulse applied to the muzzle face represents a very small portion of the total impulse resulting from the residual pressure.)

Fig. 3–10, which is a graph of the residual pressure versus time for the sample gun, shows that the residual pressure reaches 750 pounds per square inch at 0.005 second. The problem is to decide how long before this point the bolt should be unlocked so that its motion with respect to the barrel will be 0.250 inch at 0.005 second. This problem can be solved using the data in fig. 3–22. This curve is plotted by the same method used for the curve in fig. 3–21 except that the effect of the piston impulse is not included. In other words, this curve represents the free recoil velocity that would be imparted if there were no gas cylinder. If the ordinates of the velocity curve in fig. 3–22 are multiplied by the mass of the recoiling parts, the resulting curve (fig. 3–23) will show the im-

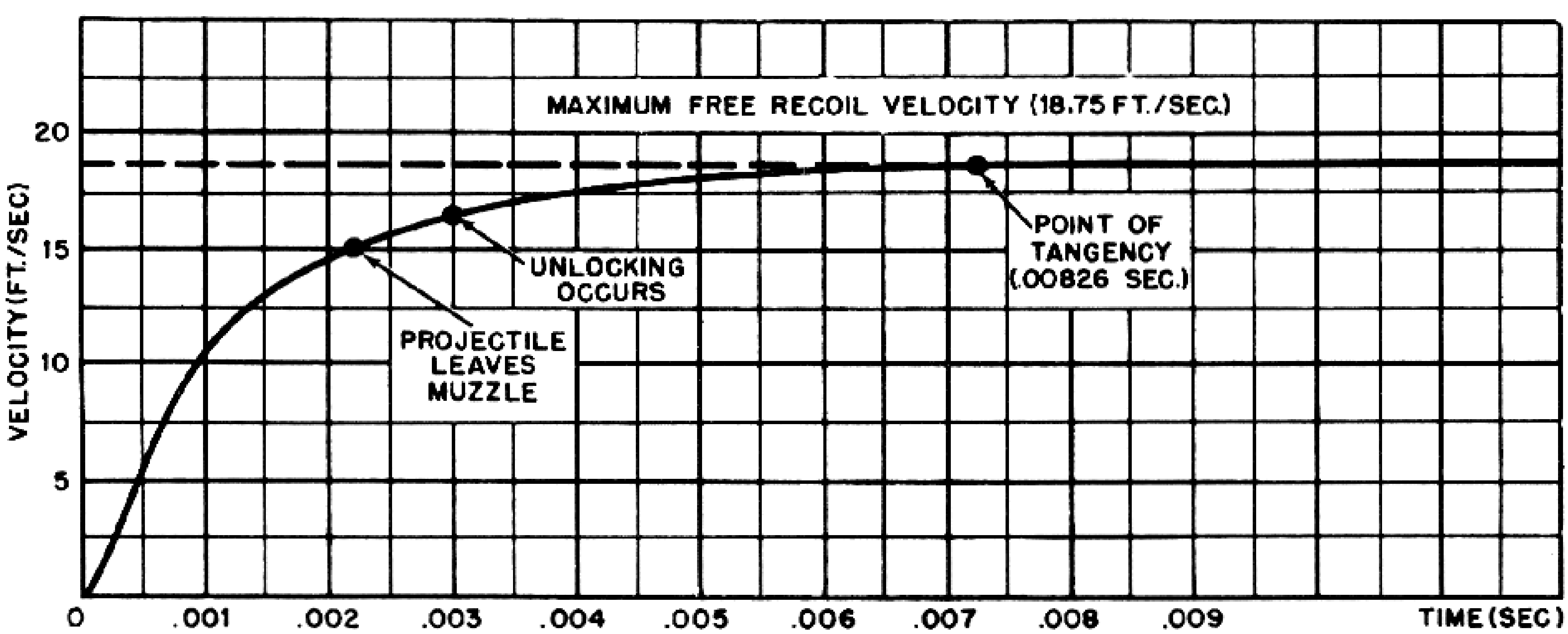


Figure 3–22. Free Recoil Velocity (With No Gas Cylinder).

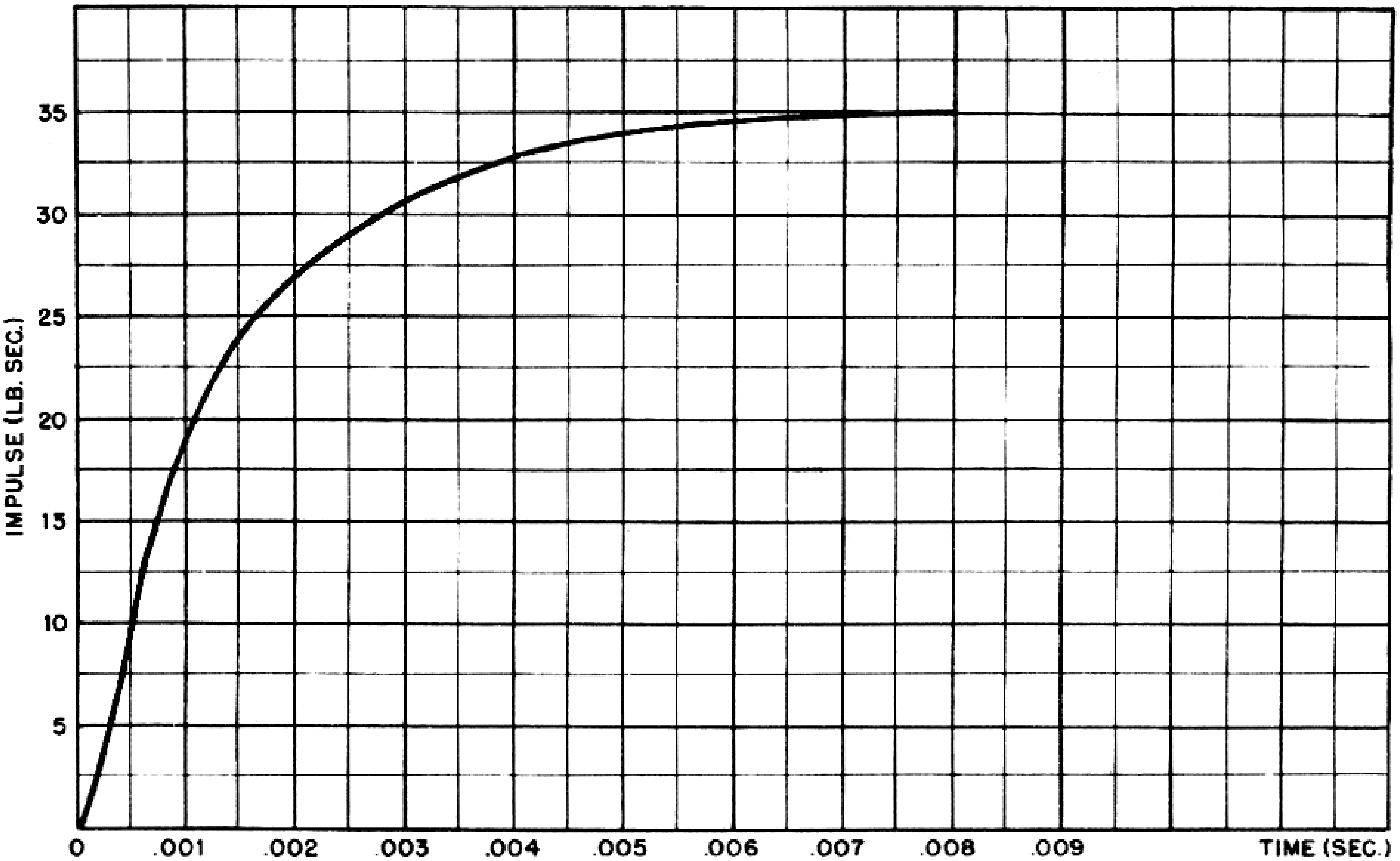


Figure 3–23. Impulse of Chamber Pressure.

pulse imparted up to any instant by the action of the chamber pressure. (That this is true may be seen by recalling the basic relation, I=MV.) The reason for using the special curve of fig. 3–19 to determine the impulse is that only the chamber pressure acts on the bolt. Since the piston impulse is applied only to the gun, it should not enter into the calculations for the blowback action.

Now, after unlocking occurs, the impulse shown by fig. 3–23 will be applied almost entirely in changing the velocity of the bolt. Therefore it is possible to divide each ordinate of the impulse curve by the bolt mass to obtain the new curve shown in fig. 3–24. (Only a portion of the vertical scale is shown in order to produce the significant portion of the curve in a large size.) The actual velocity values shown by this curve are meaningless but between any two values of time the curve does show what *change* in bolt velocity would be produced by the impulse.

Having the curve of fig. 3–24, it is only necessary to determine where to place the zero velocity axis so that the area between this axis and the curve up to 0.005 second is equal to 0.250 inch or 0.0208 foot. (Since this is a velocity-time graph, areas under the curve represent displacement.) The zero axis can be located quite simply by drawing a line along the 0.005-second ordinate and measuring the area between the line and the curve, taking the elements of area as shown in the figure and working downward until the area is 0.0208 foot. The abscissa of the point where the line bounding the lower limit of this curve intersects the curve is the required time of unlocking (0.00307 second). Ordinates measured above this line are equal to the free recoil velocity with respect to the barrel imparted to the bolt by blowback. The curve shows that the gain in free bolt velocity between the time of unlocking and 0.005 second is 18.2 feet per second.

It should be noted that although this computation neglects the effect of the bolt driving spring on the 0.250-inch travel, the resulting error is extremely small and entirely insignificant.

The data shown in fig. 3–24 are used later in the computations for completing the bolt motion curves up to 0.005 second. The manner in which the data

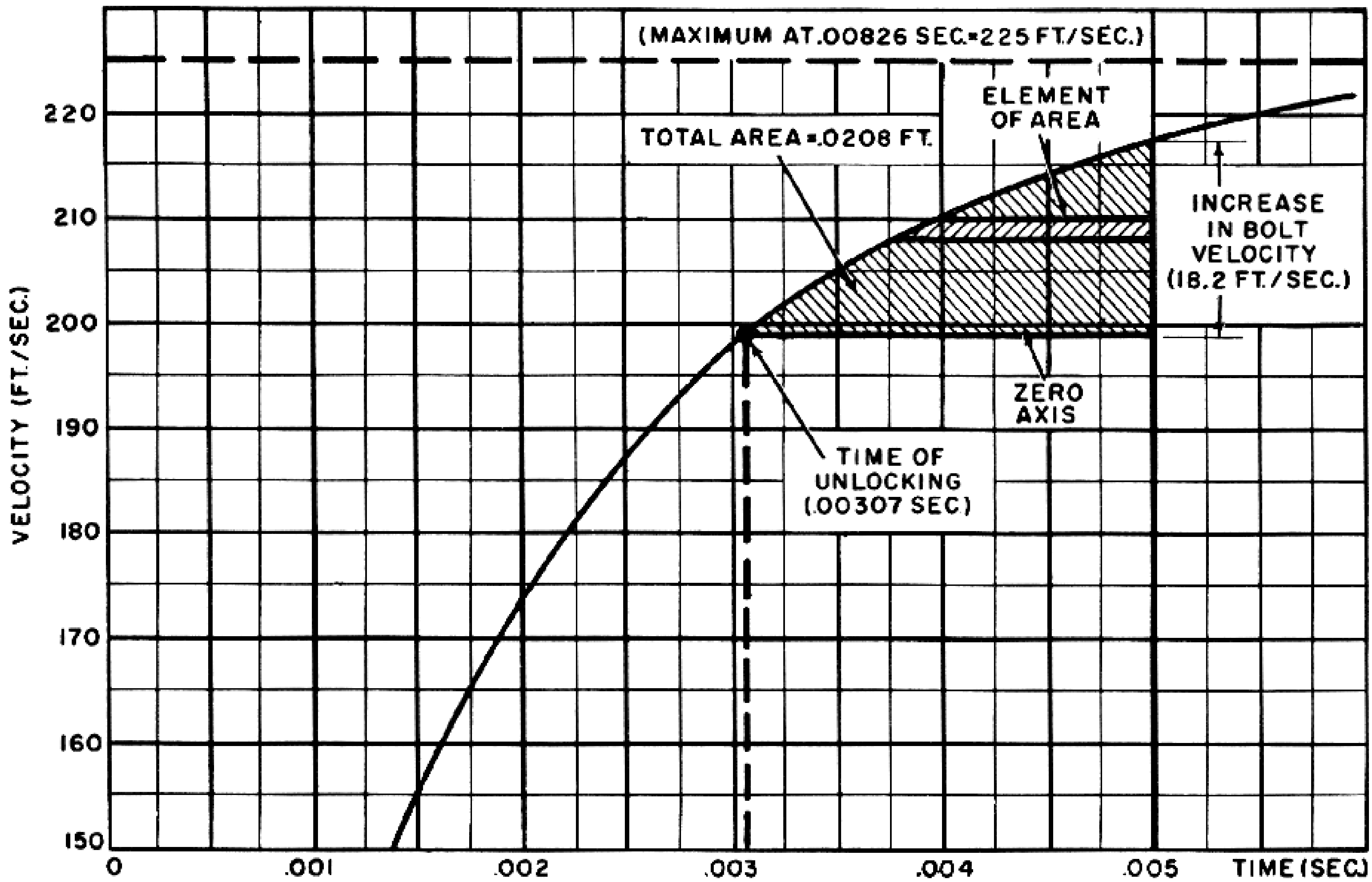


Figure 3–24. Changes in Bolt Velocity Imparted by Blowback and Determination of Time for Unlocking.

are used is explained later in connection with plotting the theoretical time-travel and time-velocity curves for the interval before the piston strikes the bolt.

3. *Selection of barrel spring characteristics and determination of counter-recoil velocity*

Having determined the free recoil condition, the effect of blowback, and the unlocking time, the next step in the analysis is to select the characteristics of the barrel spring and to determine the counter-recoil velocity possessed by the gun at the instant it is fired just before reaching the battery position (advanced primer ignition).

In order to make use of advanced primer ignition it is necessary to select the barrel spring characteristics carefully to insure proper timing of the counter-recoil motion of the gun. If the spring is too strong, the gun will reach the firing position too early and will be stopped by the mechanical buffer before the bolt can return, with the result that the advanced primer ignition effect will be lost. If the spring is too weak, the barrel will return too slowly, thus reducing the rate of fire. Within reasonable limits, however, the choice of the spring characteristics is not absolutely critical because the timing can be modified slightly by adjusting the length of the recoil movement.

The required spring characteristics are determined as follows: A reasonable recoil travel for the gun is approximately 1.5 inches or 0.125 foot. Allowing for the fact that the gun is fired a short distance before it reaches battery, the counter-recoil travel before firing will be taken as 0.100 foot. At a rate of fire of 1200 rounds per minute, the time available for the gun to move through this distance in counter-recoil is somewhat more than half the cycle time or approximately 0.030 second. Therefore the average velocity of counter-recoil must be:

$$V_{av}=\frac{D}{t}=\frac{.100}{.030}=3.33\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

Under the action of the spring, the counter-recoil velocity will not change at exactly a constant rate and hence the average velocity of counter-recoil will be slightly greater than half the maximum velocity (say 60 per cent):

$$V_{max}=\frac{V_{av}}{.60}=\frac{3.33}{.60}=5.54\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

This means that the final kinetic energy of the gun is:

$$KE=\frac{1}{2}MV^2=\frac{1}{2}\times\frac{55}{32.2}\times 5.54^2=26.2\text{ (ft. lb.)}$$

Practically all of this energy must be supplied by the expansion of the barrel spring. To produce this amount of energy by expanding over a distance of 0.100 foot, the spring must exert an average force of:

$$F_{av}=\frac{KE}{d}=\frac{26.2}{.100}=262\text{ (lb.)}$$

Actually, if the spring losses were taken into account, the average force required to compress the spring would be slightly higher than this value. For purposes of this analysis, the effect of the losses will be taken into account by adding 30 pounds (approximately 10 per cent) and it will be assumed that 292 pounds is the average force for compression of the spring. To produce this average force, the initial compression may be taken as 200 pounds and the force at a 0.125-foot deflection as 384 pounds. This requires a spring constant of:

$$K=\frac{384-200}{.125}=1472\left(\frac{\text{lb.}}{\text{ft.}}\right),\text{ or }\frac{1472}{12}=122.5\left(\frac{\text{lb.}}{\text{in.}}\right)$$

In the preceding calculations used for arriving at the spring characteristics, it was determined that the velocity with which the gun reaches the firing position is 5.54 feet per second. However, when the bolt strikes the barrel and locks to it the resulting impact will cause this velocity to increase. Since no information is yet available concerning the actual velocities of the gun and bolt at the instant of impact, it will be estimated for the present that the velocity increase is one foot per second and therefore that the gun will reach the firing position with a velocity of 6.54 feet per second. When the cartridge in the chamber is fired, the impulse exerted by the propellant explosion must first cancel this velocity before the gun will start to move to the rear in recoil.

4. *Effect of using advanced primer ignition on free recoil before unlocking occurs*

The free recoil velocity curve given in fig. 3–21 shows the velocity which would be produced in the recoiling parts if the gun were not moving at the instant of firing. Another way of looking at this curve is that, starting with zero velocity, it shows the change in velocity produced by the impulse of the propellant explosion. This same *change* in velocity would be produced, regardless of what velocity the gun possesses at the instant of firing. Since it has been determined that the gun is moving at a forward (negative) velocity of 6.54 feet per second when it is fired, its velocity at any instant after firing can be determined by simply drawing the same curve, starting at a velocity of —6.54 feet per second instead of from zero velocity. This has been done to produce the curve shown in fig. 3–25. Note that the negative velocity of the gun decreases to zero at 0.00058 second indicating that at this instant the forward motion of the gun is halted. The gun is then driven to the rear (velocity positive).

It is important to point out that the final free recoil velocity attained at the instant of unlocking is now much less than the final velocity that would have been attained if the gun were not moving when fired. This effect of using advanced primer ignition is highly advantageous because the reduction in recoil velocity results in a greatly reduced energy in the recoiling parts, thus simplifying the design of the recoil mechanism and permitting a lower trunnion reaction.

5. *Theoretical time-travel and time-velocity curves before unlocking occurs*

Because of the complexities resulting from the multiplicity of actions during the recoil and counter-recoil movements in a gas-operated gun, it is not practical to attempt to derive analytically, expressions for the time to recoil and time to counter-recoil. Also, such derivations would be extremely complicated unless it is assumed that the initial kinetic energy is transferred instantaneously to the recoiling parts, ignoring the detailed effects which occur during the action of the powder gas pressures. However, in high-rate-of-fire guns employing the gas system of operation, the time of action of the powder gas pressures is extremely significant and must be given due consideration in plotting the bolt motion

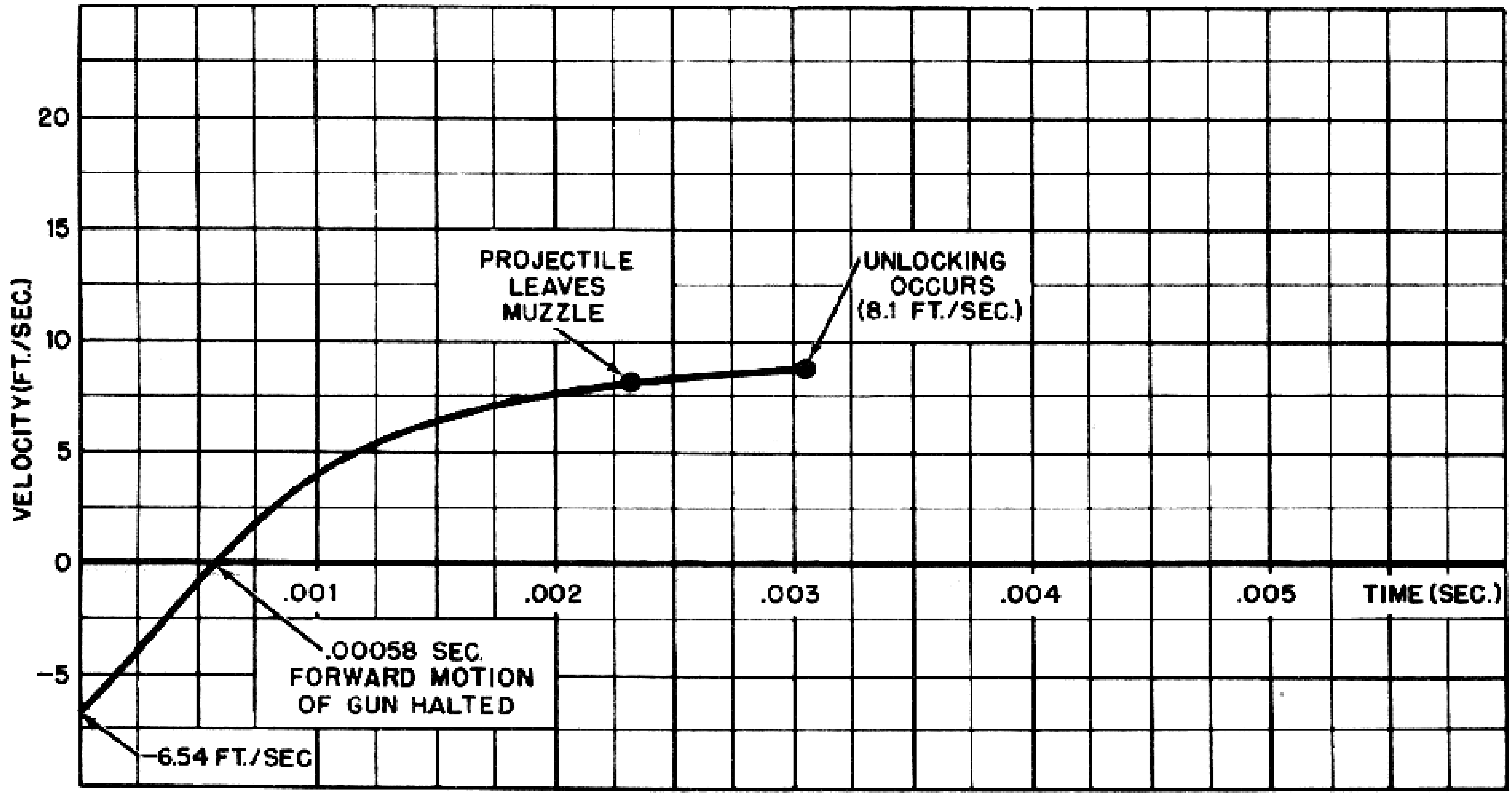


Figure 3-25. Free Recoil Velocity Before Unlocking (With Advanced Primer Ignition).

curves. A detailed analysis of this type is particularly important because most of the critical actions and high accelerations occur during the progress of the propellant explosion and it is therefore highly desirable to determine what motion characteristics may be expected in the initial portion of the operating cycle.

Since the effects of the powder gases can not be expressed by simple equations, a special method is employed to account for these effects in plotting the bolt motion curves. The method consists essentially of first plotting a curve of free recoil velocity against time and then subtracting from each ordinate of this curve the velocity loss resulting from the retarding effects of the springs.

The curve showing the velocity of free recoil versus time for the time interval before unlocking was developed previously and is shown in fig. 3–25. This curve will be used to illustrate the following description of the method.

To determine the retarding effects of the springs, use is made of the law expressed by the equation:

(3–20) $$Fdt = Mdv$$

This law states that the change in the momentum of a mass is equal to the applied impulse (the product of the force and the time for which it is applied). Solving for dv gives:

$$dv = \frac{Fdt}{M}$$

To obtain the variation of the change in velocity with respect to time, this expression is integrated.

(3–21) $$v = \int_0^t \frac{Fdt}{M} = \frac{1}{M}\int_0^t Fdt$$

In accordance with equation 3–21, the retarding effect of a force on a given mass can be determined as follows:

1. Plot a curve showing the variation of the force with respect to time.
2. Measure the area under the curve between $t=0$ and some time t_1.
3. Divide the measured area by the mass. This gives the ordinate of the retardation curve for the time t_1.
4. Repeat steps 2 and 3 for other values of t and plot the retardation curve.

Applying this procedure using the mass of the recoiling parts and the resistance of the barrel spring produces a curve showing the loss in recoil velocity resulting from the action of the spring up to the time of unlocking. Since the free recoil velocity curve shows the gain in velocity resulting from the thrust of the powder gases, the difference between the curves will be the net recoil velocity, or in other words the velocity of retarded recoil.

The foregoing method would be very simple if the retarding force were constant or if the variation of this force with respect to time were known. However, when the force varies with recoil travel as it does with the type of spring assumed for purposes of this analysis, a difficulty is encountered. In order to plot a graph showing the variation of the retarding force with respect to time, it is necessary to have a curve showing the variation of the recoil travel wtih respect to time, and the latter curve is one of those which yet remain to be determined.

This difficulty can be overcome by employing a process of successive approximation. While the powder gas pressures are acting, the loss in velocity resulting from the retarding effect of the spring will be relatively small and will be almost entirely due to the constant effect of the initial compression. The varying force due to the spring constant during this interval of time will almost certainly be negligible but, if necessary, it can be approximated very closely.

The procedure for plotting the velocity and travel curves for the time before unlocking occurs is as follows:

1. Plot the curve of free recoil velocity versus time (fig. 3–26).
2. The loss in velocity due to the initial compression of the barrel spring is equal to:

$$\frac{F_{o_1}t}{M_r}$$

   Determine the velocity loss for various values of t, subtract each from the corresponding ordinate of the free recoil velocity curve and draw a curve through the resulting points. If the effect of the spring constant proves to be negligible, this curve is the retarded velocity curve.
3. Integrate under the curve drawn in step 2 to obtain the displacement curve.
4. Assume that the curve drawn in step 3 represents the actual time-travel curve and use this curve to determine the retardation due to the spring constant. Ordinarily, it will be found that this retardation is so small that it will not have any effect worthy of consideration for the interval before unlocking.
5. In the event that the retardation determined in step 4 is sufficient to affect the velocity, use it to modify the curve drawn in step 2 and then integrate under the new curve to obtain a corrected displacement curve.
6. Steps 4 and 5 can be repeated as often as is necessary until no significant change occurs in the displacement curve. Actually, this process of successive approximation should never be necessary and satisfactory result should be obtained in the first three steps or at least in the first five steps.

Fig. 3–26 shows the curves obtained for the gun of the example for the interval before unlocking. The total loss in velocity due to the initial compression of the barrel spring during this interval (0.00058 to 0.00307 second) is:

$$v=\frac{F_{o_1}t}{M_r}=\frac{200\times.00249\times32.2}{60}$$
$$=.268\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the effect of the spring constant as determined by the method of step 4 is only about 0.013 foot per second. The final curves shown in fig. 3–26 are the result of performing step 3. Since the velocity loss due to the effect of the spring constant is so small, step 5 need not be taken. The curves show that the velocity at the instant of unlocking is 7.9 feet per second and that at this instant the gun has travelled 0.01492 foot in recoil (0.179 inch).

6. *Theoretical time-travel and time-velocity curves after unlocking and selection of bolt driving spring characteristics*

After unlocking occurs, the gun and bolt are essentially independent of each other, except for the relatively small interaction between them through the bolt driving spring. Since the bolt is unlocked, the gun itself is no longer affected appreciably by the pressure of the powder gases and therefore its free recoil characteristic is to continue moving at the same velocity it had at the instant of unlocking. This is indicated in fig. 3–27 by the fact that the free gun velocity curve after unlocking is a horizontal line. The free bolt velocity curve of fig. 3–27 is obtained from the data shown in fig. 3–24 by adding to the ordinates of the free gun velocity curve, the

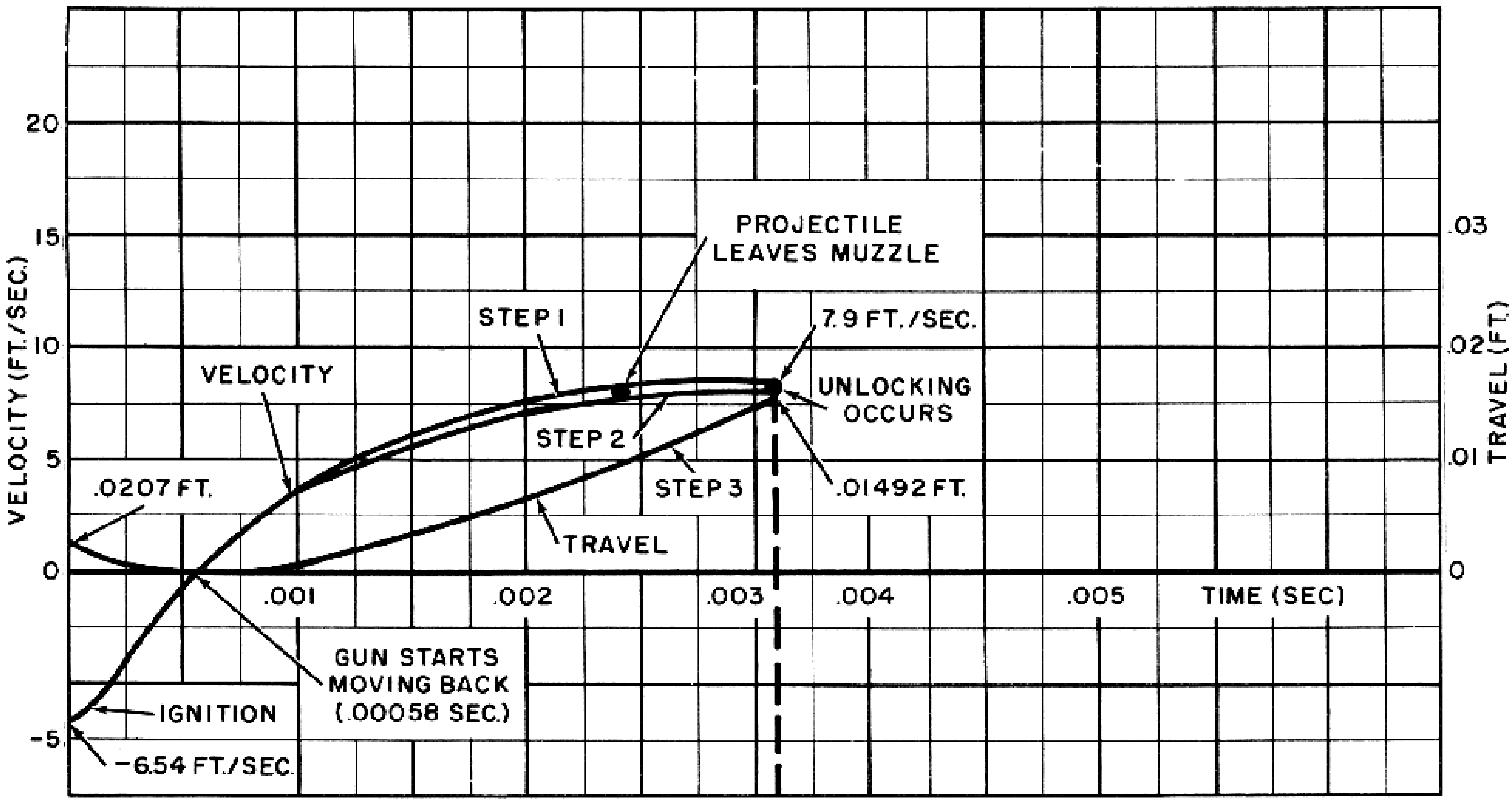


Figure 3–26. Development of Time-Travel and Time-Velocity Curves for Period Before Unlocking.

corresponding ordinates of the curve in fig. 3–24, measuring these ordinates above the zero axis.

Before proceeding, it is necessary to select the characteristics of the bolt driving spring. Since the only purpose of the bolt driving spring is to assist the return of the bolt to the battery position, and since this spring is not required to absorb all of the bolt recoil energy, the magnitude of the force exerted by the driving spring is not critical in the design. For this reason, the characteristics of the spring may be selected more or less arbitrarily. In order to permit a high rate of fire, the spring should be made relatively light so that it will not offer a high retardation to the recoil movement of the five-pound bolt, but on the other hand, the spring should be heavy enough to provide adequate force for assisting the closing of the bolt. Taking both of these requirements into consideration, it appears reasonable that an initial compression of 25 pounds and a spring constant of 10 pounds per inch should produce the desired action. If it is assumed that the bolt must open 10 inches to permit feeding, the maximum force exerted by the spring will be 125 pounds. This is a reasonable value and is not high enough to cause excessive difficulty in charging the weapon.

In the analysis of the motions after unlocking. two factors must be taken into consideration. First at the instant of unlocking, the barrel spring has been compressed 0.01492 foot (fig. 3–26). This means that from the time of unlocking on, the initial compression of the spring must be increased by the effect of its spring constant for this deflection. That is:

$$F_{o_1} = 200 + .01492 \times 122.5 \times 12 = 200 + 22 = 222 \text{(lb.)}$$

Second, it should be realized that after the bolt is unlocked, the force of the bolt driving spring exerts a rearward thrust on the gun. Although this force is relatively small at first, it should be taken into account by subtracting the initial compression of the driving spring from that of the barrel spring and subtracting the effect of the spring constant of the driving spring from that of the barrel spring. It should also be remembered that the movement which must be considered in determining the force produced by the bolt driving spring is the relative travel between the bolt and gun.

Except for the differences noted in the preceding paragraph, the retarding effects of the barrel spring and bolt driving spring are determined by the same

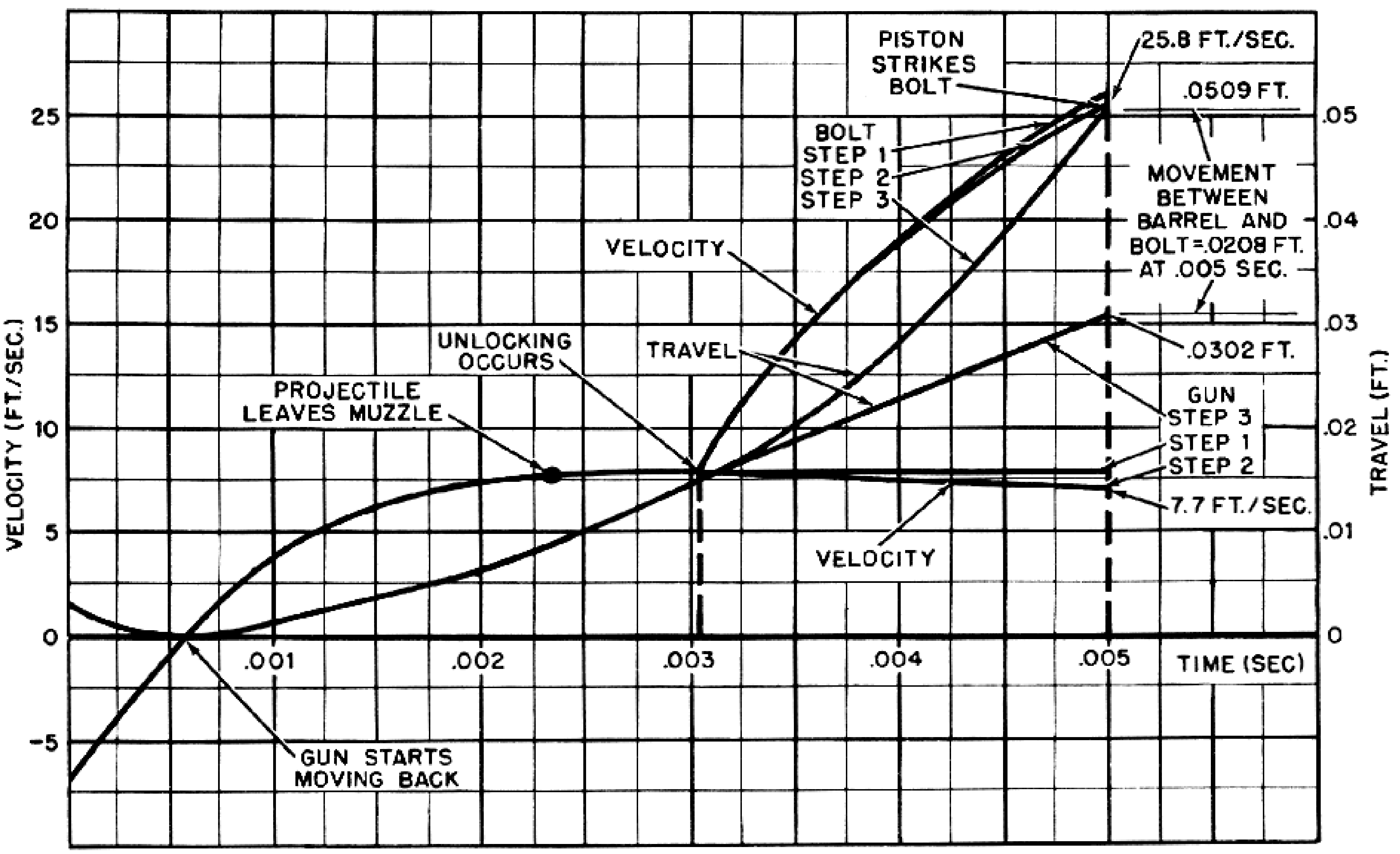


Figure 3–27. Development of Time-Travel and Time-Velocity Curves for Period Between Unlocking and Impact of Piston on Bolt.

general method as used before unlocking. The total loss in gun velocity due to the corrected initial compression of the barrel spring during the interval between unlocking at 0.00307 second and the impact of the piston on the bolt at 0.005 second (an interval of .005—.00307=0.00193 second) is:

$$V=\frac{F_o}{M_1}t=\frac{(222-25)\times 32.2\times .00193}{55}=.223\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The loss due to the combined effects of the spring constants as determined by the methods of step 4 modified to account for the difference described in the preceding paragraph is only about 0.0446 foot per second. The final gun motion curves shown in fig. 3–27 are the result of performing steps 2 and 3. Since the velocity loss due to the effect of the spring constants is so small, it is not necessary to take step 5.

The retarding effect of the bolt driving spring is found as follows: Step 2 is performed to determine the bolt velocity loss due to the effect of the initial compression. The total loss between the instant of unlocking and the instant the piston strikes the bolt is:

$$V=\frac{F_{o_2}}{M_2}t=\frac{25\times 32.2\times .00193}{5}=.310\left(\frac{\text{ft.}}{\text{lb.}}\right)$$

Next step 3 is performed and then the relative movement between the gun and bolt is determined by subtracting the gun travel curve from the bolt travel curve. The relative movement curve can now be used to perform step 4. The loss found in this way is only about 0.012 foot per second. Since the velocity loss due to the effect of the spring constant is so small, it is not necessary to take step 5.

The final curves shown in fig. 3–27 indicate that at the instant the piston strikes the bolt, the bolt velocity is 25.8 feet per second and the velocity of the gun is 7.7 feet per second. The bolt has moved 0.0507 foot to the rear and the gun has moved 0.0302 foot. The relative movement is 0.0207 foot, or very close to the allowable movement of 0.0208 foot (0.250 inch).

7. Impact of piston on bolt

When the cycle of operation has progressed for 0.005 second, as described up to this point, the residual pressure has decreased to the assumed safe operating limit of 750 pounds per square inch and it is now possible to increase the velocity of the bolt without danger of rupture of the cartridge case. It should be noted that, at 0.005 second, the residual powder gas pressure has not yet reached zero and therefore some blowback action will occur during and after the impact of the piston.

The first point to consider is what bolt velocity is desired after impact in order to obtain the required rate of fire. For the specified rate of fire of 1200 rounds per minute, the time available for the bolt to complete its rearward travel is about 0.022 second (slightly less than half the cycle time). The required bolt travel with respect to the gun to provide an opening sufficient for feeding will be taken as 10 inches (0.833 foot). Since the gun recoils 1.5 inches or 0.125 foot, the total bolt travel is $.833+.125=.958$ foot. At 0.005 second the bolt has already moved 0.0507 foot and therefore the remaining travel of 0.907 foot must be accomplished in 0.017 second. This means that the bolt must move this distance at an average velocity of:

$$V_{av}=\frac{D}{t}=\frac{.907}{.017}=53.3\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The effect of the bolt driving spring on the bolt velocity can be estimated as follows: The initial compression of the spring at the instant of piston impact is equal to:

$$F_o'{}_2=F_{o2}+DK_2=25+.0207\times10\times12$$
$$=25+2.48=27.5\ (\text{lb.})$$

The force exerted by the spring at a 10-inch deflection is

$$F=25+10\times10=125\ (\text{lb.})$$

Therefore, the average force of the spring (taken with respect to time) over the 0.018 second required for completion of the bolt travel may be estimated roughly as:

$$\frac{125+27.5}{2}=81.2\ (\text{lb.})$$

(This estimate is not exact because the velocity of the movement will not be constant over the interval, but it is close enough for present purposes.) The velocity loss caused by the action of the driving spring can now be estimated by using the formula:

$$V=\frac{F_{av}xt}{M}=\frac{81.2\times.0170\times32.2}{5}=8.89\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

It will be recalled that there will still be some blowback action after the bolt impact. As shown in fig. 3–24, the action of blowback after 0.005 second will increase the bolt velocity by approximately 7.5 feet per second. Therefore, the net loss of bolt velocity after impact will be the difference between the loss produced by the spring and the gain produced by the remaining blowback action or $8.89-7.5=1.4$ feet per second. Since the loss is approximately 1.4 feet per second and the desired average velocity is 53.3 feet per second, the velocity after the piston impact should be $53.3+1.4/2=54$ feet per second. To allow for the time of action of the backplate, this velocity will be taken as 55 feet per second.

The following data are now available for determining the conditions of piston impact:

Bolt weight, $W_2=5$ (lb.)

Initial bolt velocity, $V_2=25.8$ (ft./sec.)

Desired final bolt velocity $V'_2=55$ (ft./sec.)

Since the parts will be of steel, the coefficient of restitution, e, will be taken as 0.55. Using these data, computations are made by the same methods described in the analysis of gas operation. The equation expressing the relationship between the piston velocity V_1 and the piston weight W_1 is found as follows:

$$V'_2-V'_1=e(V_1-V_2)$$
$$55-V'_1=.55(V_1-25.8)$$
$$V'_1=69.2-.55V_1$$
$$W_1V_1+W_2V_2=W_1V'_1+W_2V'_2$$
$$W_1V_1+5\times25.8=W_1(69.2-.55V_1)+5\times55$$
$$V_1=\frac{94.2}{W_1}+44.6$$

Various values of V_1 and W_1 which satisfy this equation are listed in Table 3–2 and are also shown graphically in figs. 3–28 and 3–29. The other values shown in the table and graphs were found using the following formulas:

Velocity of piston after impact:

$$V'_1=69.2-.55V_1\ (\text{ft./sec.})$$

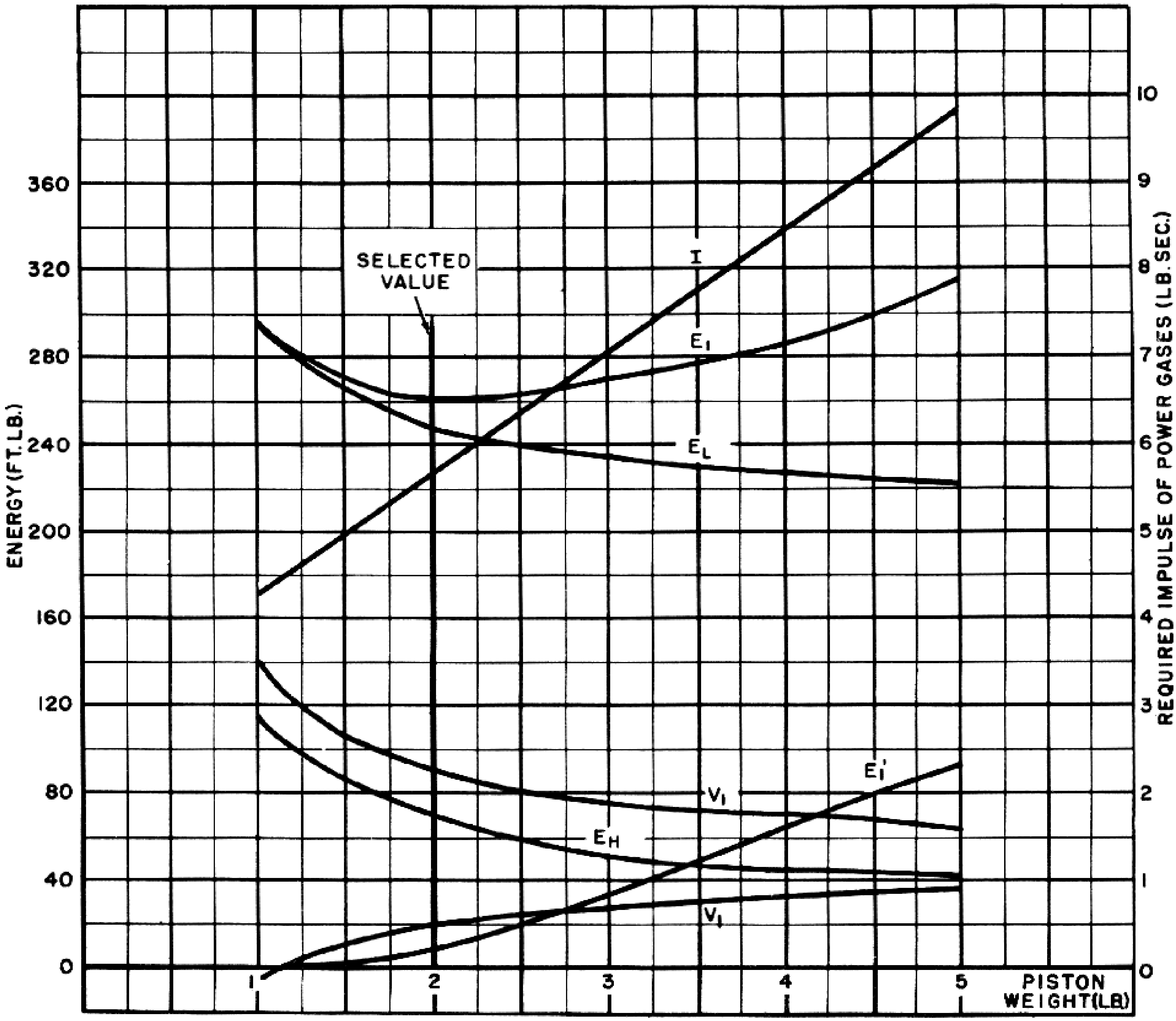


Figure 3–28. Factors Related to Selection of Piston Weight and Velocity in Gun of Example.

Energy lost by piston in impact:

$$E_L=\frac{W_1}{2g}(V_1^2-V'^2_1)\quad \text{(ft. lb.)}$$

Initial piston energy:

$$E_1=\frac{W_1}{2g}V_1^2\quad \text{(ft. lb.)}$$

Final piston energy:

$$E'_1=\frac{W_1}{2g}V'^2_1\quad \text{(ft. lb)}$$

Impact loss (to heat):

$$E_H=E_L-E_G\quad \text{(ft. lb.)}$$

Required impulse:

$$I=M_1V_1\quad \text{(lb./sec.)}$$

The energy gained by the bolt is the same for all entries and is computed as follows:

$$E_G=\frac{W_2}{2g}(V'^2_1-V_1^2)=\frac{5}{64.4}(55^2-25.2^2)=182\ \text{(ft. lb.)}$$

Table 3–2. Values Related to Selection of Piston Weight

W_1	V_1	V'_1	E_L	E_1	E'_1	E_H	I
1.0	138.8	−7.1	298	299	0.76	116	4.31
1.5	107.4	10.1	267	269	2.37	85	5.00
2.0	91.7	18.2	251	261	10.3	69	5.70
2.5	82.3	23.9	241	263	22.1	59	6.38
3.0	76.0	27.4	234	269	35.0	52	7.07
4.0	68.1	31.7	226	289	62.4	44	8.45
5.0	63.4	34.3	221	312	91.3	39	9.85

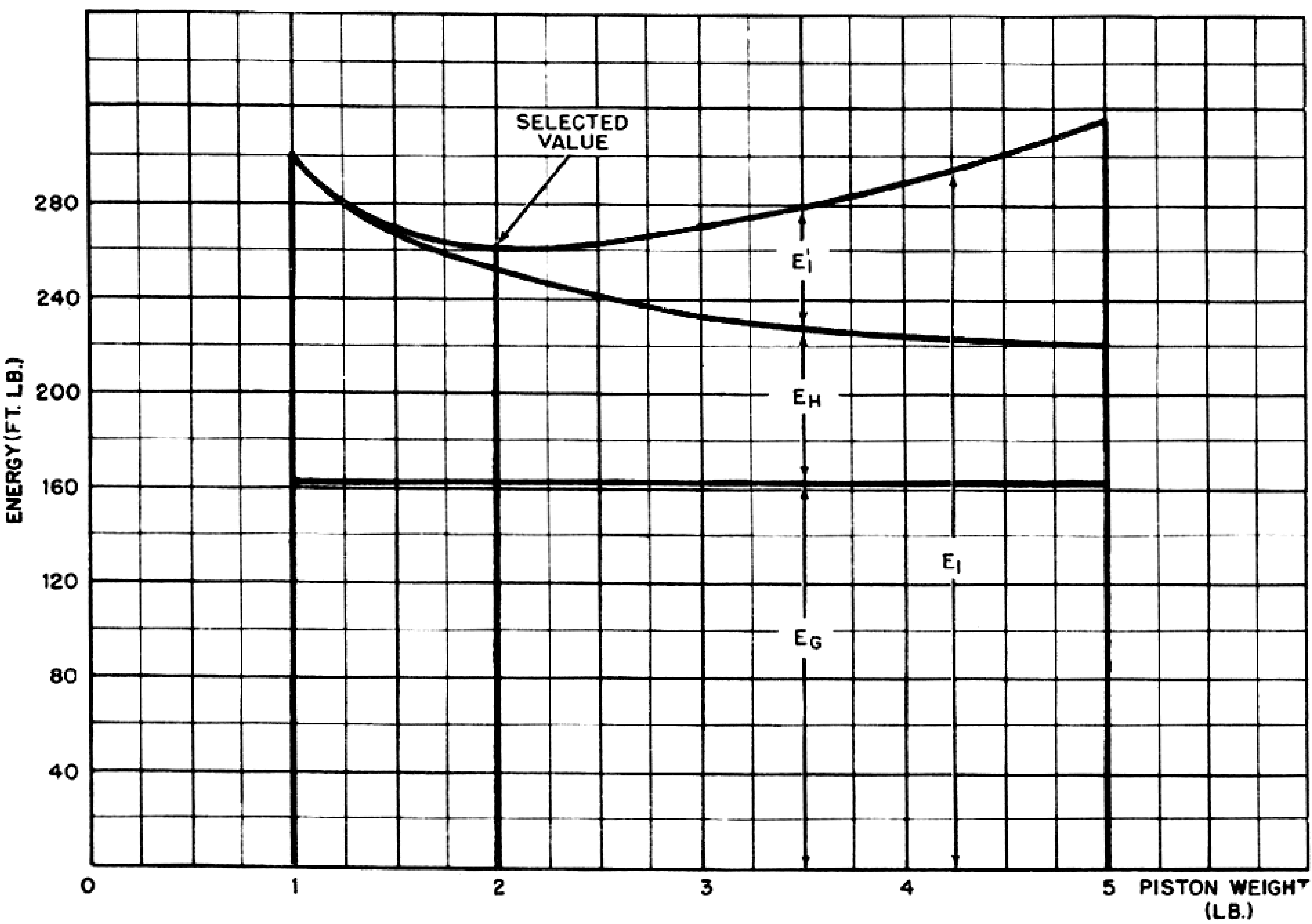


Figure 3–29. Distribution of Piston Energy for Various Piston Weights in Gun of Example.

Consideration of the values in Table 3–2 and of the graphs of figs. 3–28 and 3–29 indicate that a piston weight of 2.0 pounds is a reasonable choice. This choice is based on the same factors explained previously in the analysis of gas operation.

8. *Design of gas mechanism and analysis of piston motion*

Having determined the desired piston weight (2.0 pounds) and velocity (91.7 feet per second), the gas mechanism must be designed to produce this velocity. The design process consists essentially of determining the location of the gas port, selecting the orifice size, and establishing the piston area so that the impulse applied to the piston by the gas pressure will be of the correct value to produce the desired velocity. In the design, due allowance should be made for friction losses and for losses in piston energy resulting from operation of the unlocking device, but these losses will be neglected for purposes of this analysis.

As pointed out at the beginning of this mathematical analysis, the factors controlling the flow of the powder gases through the gas port and orifice into the gas cylinder can not be handled in a simple manner by direct computation. However, it is possible to analyze the piston motion and determine the necessary impulse on the basis of a conservative assumption of how the pressure will rise in the gas cylinder in the presence of an orifice with a reasonable throttling action. When the gun is actually developed, the actual orifice size required to produce the necessary impulse can be determined by experimental firings; starting first with a very small orifice and gradually increasing its size until the desired action is obtained.

As a starting point for the theoretical design, let the distance of the gas port be set so that the projectile will pass the port after a bore travel of 2.5 feet. As shown in fig. 3–19, this will occur at 0.00166 second. At this instant, the gun has acquired a recoil velocity of 6.4 feet per second (fig. 3–26). Since the piston moves in recoil with the gun, the piston has this initial velocity at the time the powder gas pressure starts to act on it. Fig. 3–30 shows a portion of the variation of the barrel pressure with respect to time and also shows a reasonable assumption of how the throttling effect of the orifice will affect the rise of pressure in the cylinder. The design must be arranged so that the pressure, acting between 0.00166 second and 0.005 second, will increase the piston velocity from its initial value of 6.4 feet per second to a final value of 91.7 feet per second.

The impulse of the gas cylinder pressure on a piston of unit area (one square inch) can be found by integrating under the pressure curve between limits of 0.00166 second and 0.005 second. This impulse is approximately 6.0 pound seconds. Now to increase the velocity of the piston (weight 2 pounds) from 6.4 to 91.7 feet per second requires an impulse of:

$$I = M(V'_1 - V_1) = \frac{2}{32.2}(91.7 - 6.4) = 5.3 \quad \text{(lb./sec.)}$$

This will require a piston area of:

$$A = \frac{5.3}{6.0} \times 1 = .884 \text{ (sq. in.)}$$

The piston diameter to produce this area is:

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times .884}{\pi}} = 1.06 \text{ (in.)}$$

This piston diameter is of reasonable size (only slightly larger than the gun bore diameter of 0.79 inch). It should also be noted that the impulse produced by the gas cylinder (5.3 pound-seconds) is sufficiently close to the impulse assumed tentatively in paragraph 1 (5.13 pound-seconds) that it will not be necessary to revise fig. 3–21.

On the basis of the pressure curve of fig. 3–30, it is now possible to determine the piston motion. Since the gun velocity changes during the interval between the start of the piston movement and the impact of the piston on the bolt, it is advisable to consider the absolute movement of the piston (with respect to the cradle). Multiplying each ordinate of the gas cylinder pressure curve by the area of the piston (0.884 square inch) would give the force on the piston at any instant. Dividing by the piston mass and integrating under the curve will give the change in piston velocity produced by the pressure. That is:

$$\Delta V = \int_{t_1}^{t_2} \frac{PA}{M} \, dt$$

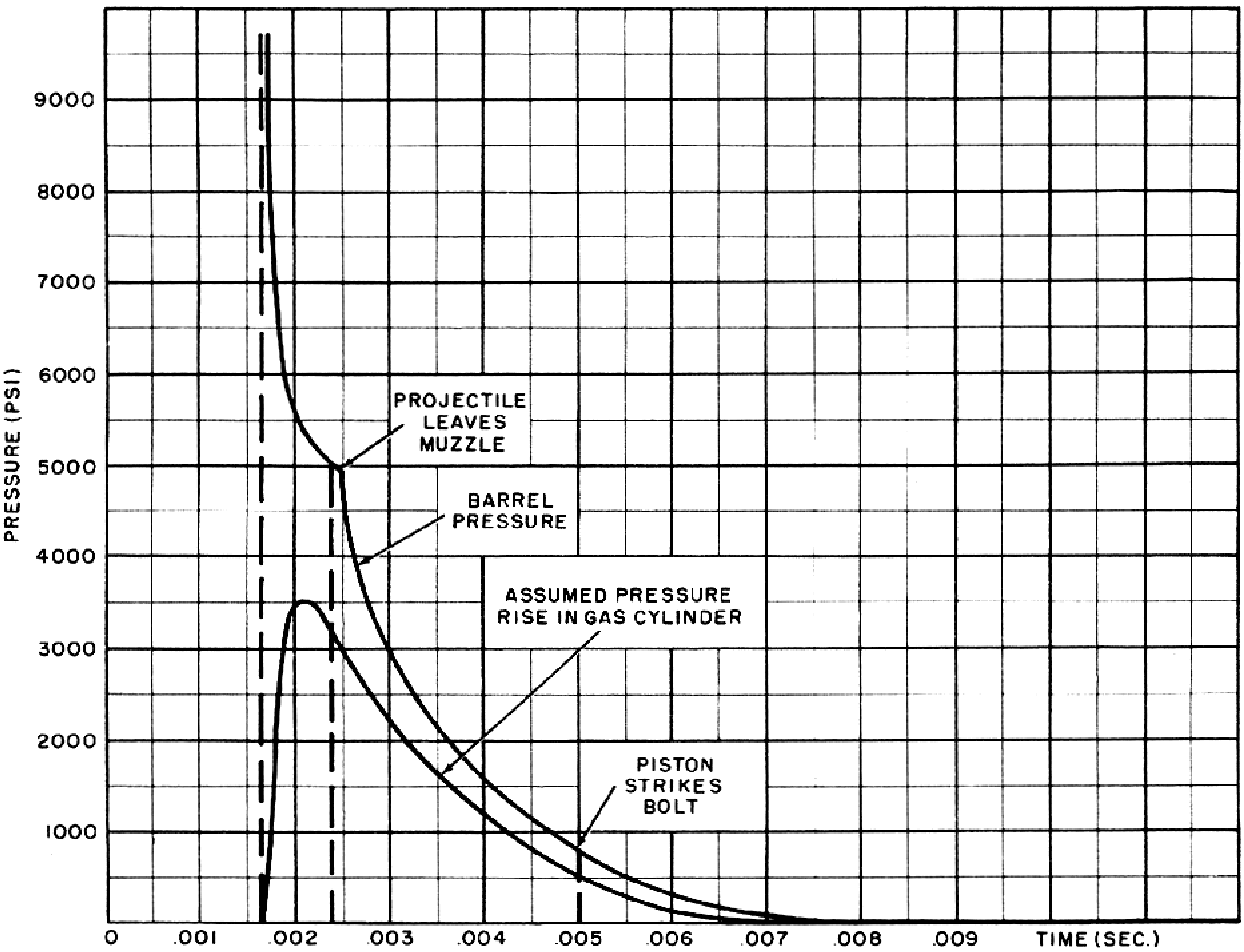


Figure 3–30. Assumed Pressure Rise in Gas Cylinder.

The velocity curve shown in fig. 3–31 is the result of performing this operation and adding the velocity changes so obtained to the initial velocity of 6.4 feet per second. The piston travel curve was obtained by integrating under the velocity curve. For convenient reference, the piston travel and bolt travel curves (obtained from fig. 3–27) are both drawn on the field of fig. 3–31. Comparing the relative movements shows that the free travel of the piston before actuating the unlocking device must be 0.036 foot or 0.432 inch (distance AB in fig. 3–11). The piston must then have an additional free travel of 0.132 foot or 1.504 inches before it strikes the bolt (distance BC in fig. 3–11). This free travel permits sufficient time for blowback to occur while the residual pressure is decreasing to a safe operating limit.

9. *Theoretical time-travel and time-velocity curves after piston impact*

In order to determine the theoretical time-travel and time-velocity curves after the impact of the piston, it is necessary to consider the gun motion and bolt motion simultaneously. As shown in fig. 3–27, the velocity of the gun at 0.005 second is 7.7 feet per second and from this instant on the free recoil velocity may be represented as a horizontal line. (See line designated as step 1 in fig. 3–32.) At 0.005 second the travel of the gun is 0.0302 foot and at this displacement the force of the barrel spring is:

$$F_{o_1}+K_1D=200+122.5\times 12\times .0302=244.5 \text{ (lb.)}$$

For purposes of determining the gun motion from 0.005 second on, this force of 244.5 pounds can be

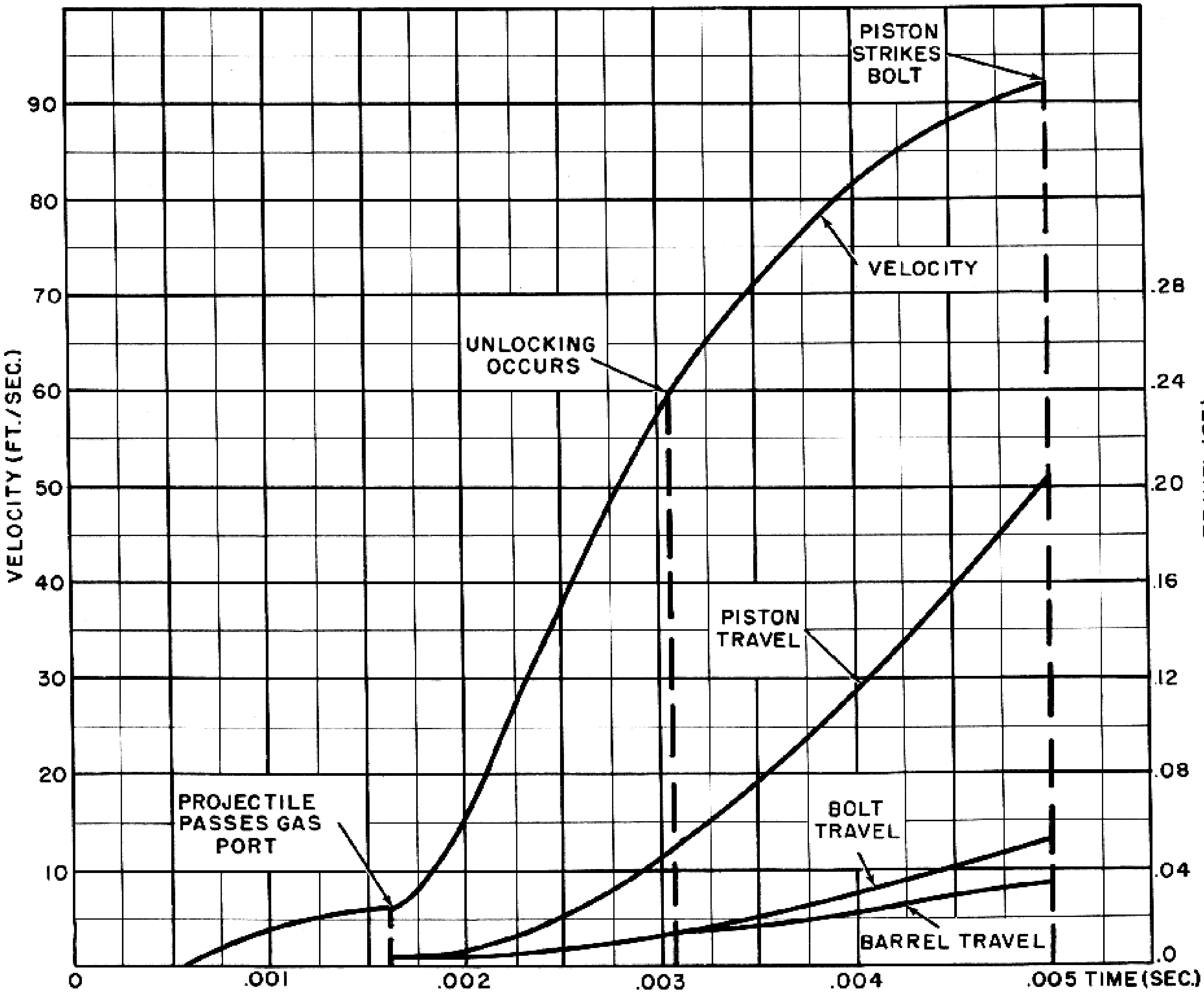


Figure 3–31. Time-Travel and Time-Velocity Curves for Piston Before Impact With Bolt.

considered to be the initial compression of the spring. However, since the bolt is free of the gun, the force of the bolt spring acts rearward on the gun and must be subtracted from the barrel spring force. At the instant of impact the bolt opening is 0.0208 foot. At this deflection, the force of the driving spring is:

$$F_{o2}+K_2D=25+10\times 12\times .0208=27.5$$

Subtracting this force from the corrected initial compression of the barrel spring gives an effective initial compression of 217 pounds. The retarding effect of this force on the gun can be computed by the same methods used for developing the motion curves before piston impact. The velocity loss due to the initial compression will be:

$$V=\frac{F_o}{M}\,t=\frac{21.7\times 32.2}{55}\,t=127\,t$$

That is, the effect of the initial compression causes the gun recoil velocity to decrease at the rate of 127 feet per second. This loss is shown in fig. 3–32 by the curve designated as step 2 for the gun.

When the piston strikes the bolt, the bolt velocity will be increased from 25.8 feet per second to 55 feet per second. This action will occur practically instantaneously as shown in fig. 3–32 by the vertical line at 0.005 second. After the impact occurs, the

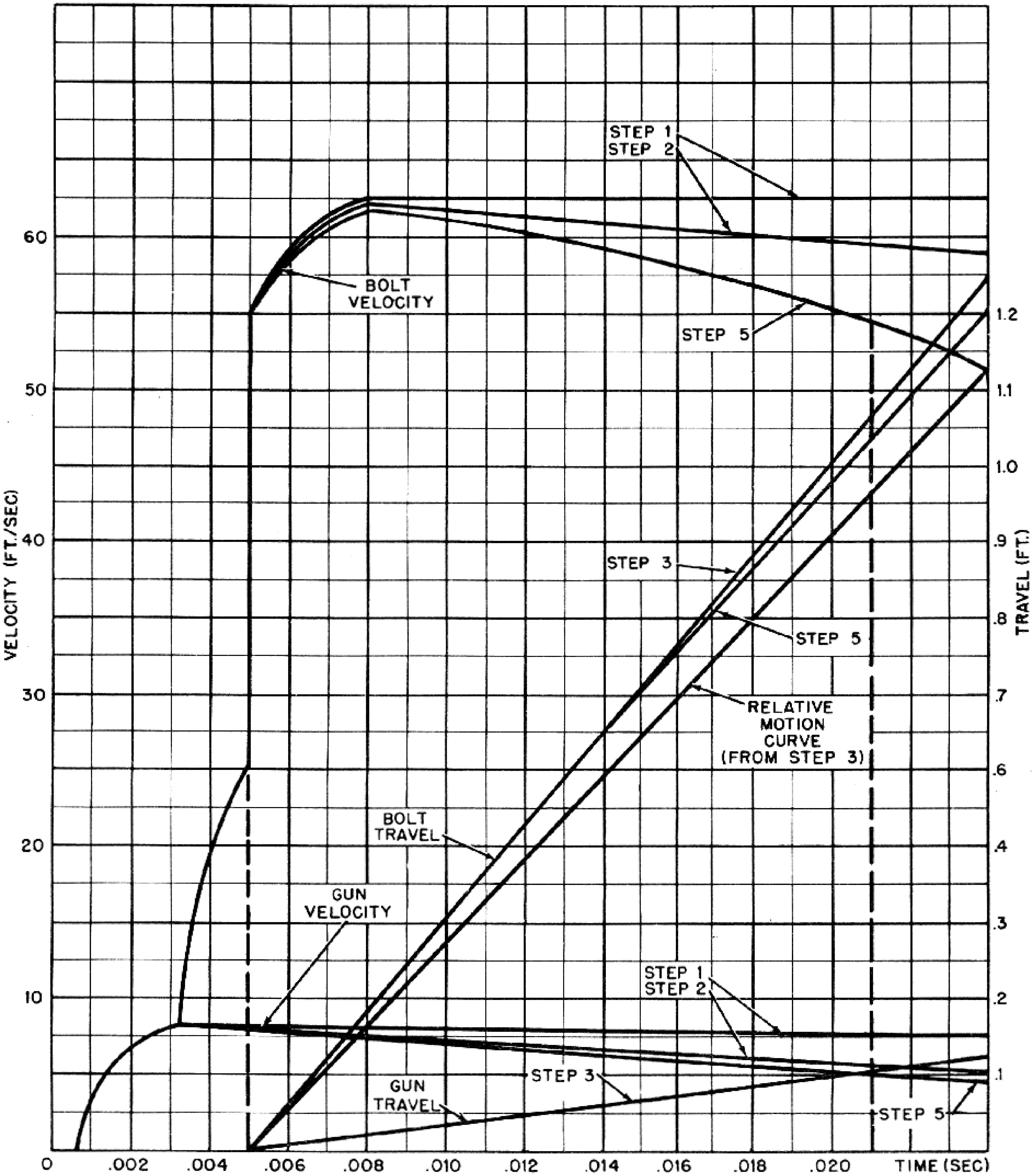


Figure 3–32. Effect of Bolt Driving Spring and Barrel Spring Up to Instant Bolt Strikes Backplate.

residual pressure still continues to act until 0.00826 second and imparts an additional free bolt velocity of 7.5 feet per second. (Cf. fig. 3–24.) This increase is shown by the curve designated as step 1 in fig. 3–33. Note that after the residual pressure has decreased to zero, the free recoil curve is a horizontal line, indicating that the bolt tends to continue moving of its own momentum at the maximum velocity of 62.5 feet per second.

At the instant of impact, the bolt has moved 0.0208 foot with respect to the gun and at this displacement, the force of the bolt driving spring is:

$$F_{o_2}+K_2D=25+10\times12\times.0208-27.5 \text{ (lb.)}$$

For developing the bolt motion curves after the impact, this force of 27.5 pounds can be considered to be the initial compression of the spring and the retarding effect of the spring can be determined by methods similar to those used before the impact. The velocity loss due to the initial compression will be:

$$V-\frac{F_o}{M}t=\frac{27.5\times32.2}{5}t=177\ t$$

That is, the effect of the initial compression causes the bolt velocity to decrease at the rate of 177 feet per second. This loss is shown in fig. 3–32 by the curve designated as step 2 for the bolt.

To determine the effect of the spring constants, allowance must be made for the time that the bolt is moving. The force exerted on the bolt and gun by the driving spring will therefore depend on the relative movement between the bolt and gun. The first step is to use the curves designated as step 2 to obtain a first approximation of the gun and bolt travels (curves designated as step 3 in fig. 3–32). The first approximation of the relative motion between the gun and bolt is obtained by subtracting the gun travel curve of step 3 from the bolt travel curve of step 3.

The effect of the springs on the gun is obtained by performing step 4 in a special way. First, the method of step 4 is applied using the spring constant for the barrel spring, the gun mass, and the barrel travel curve of step 3. This gives the loss in gun velocity due to the barrel spring. Second, the procedure is employed again using the spring constant for the bolt driving spring, the gun mass, and the relative travel curve. This gives the gain in gun velocity due to the

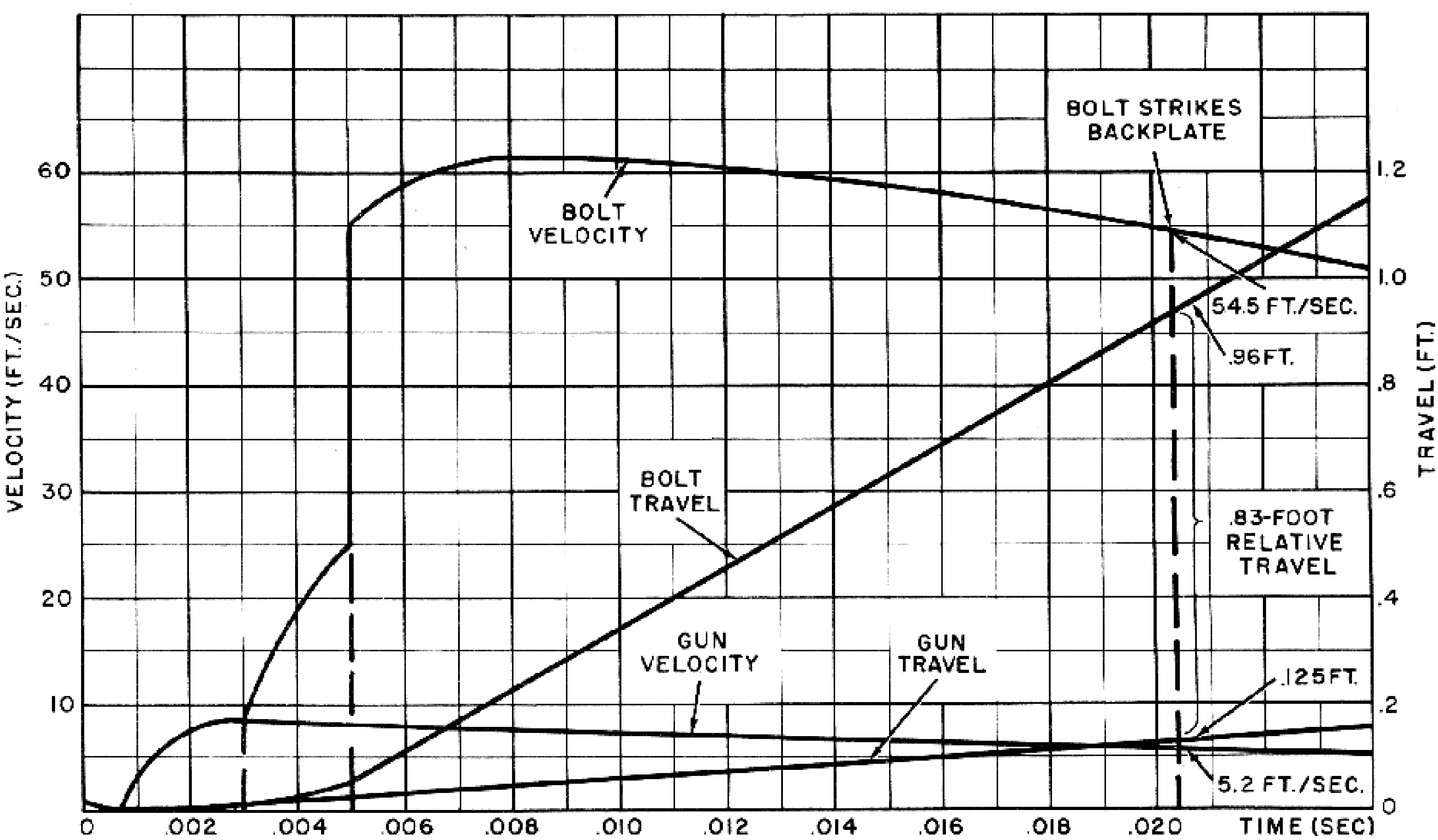


Figure 3–33. Time-Travel and Time-Velocity Curves for Gun and Bolt Up to Instant Bolt Strikes Backplate.

force of the bolt driving spring. The actual loss in gun velocity is obtained by subtracting the gain due to the bolt driving spring from the loss due to the barrel spring. This loss is subtracted from the gun velocity curve designated as step 3 to give the final velocity curve designated as step 5. Since the change in the velocity curve is so slight, it is not necessary to modify the gun travel curve and the curve designated as step 3 can be used as the final gun travel curve.

The effect of the spring constant of the bolt driving spring is obtained by using the relative travel curve in performing step 4 to obtain the bolt velocity curve indicated as step 5 in fig. 3–32. The final travel curve (also designated as step 5) is found by integrating under the velocity curve. Since the performance of step 6 produces a negligible change in the curves, the curves drawn in accordance with step 5 represent the effect of the driving spring on the bolt motion.

Fig. 3–33 shows the velocity curves for the bolt and gun and shows the travel curves obtained by adding the travels shown in fig. 3–32 to the bolt travel and gun travel at the instant of bolt impact (0.0509 for the bolt and 0.0305 for the gun). Having the final travel curves it is now possible to determine at what instant the bolt will reach the backplate. To strike the backplate, the bolt must move a distance of 10 inches (0.833 foot) with respect to the gun. Fig. 3–23 shows that this relative movement occurs at 0.021 second (which is close enough to the value of 0.022 second previously estimated). At the instant of contact, the recoil travel of the gun as shown in the figure is equal to 0.125 foot (which is equal to the 0.125-foot displacement estimated near the beginning of the analysis). Because of the good agreement obtained between the estimated and computed values, it will not be necessary to adjust any of the computations made up to this point in order to obtain the desired rate of fire and the required timing of the movements. Note that the velocity with which the bolt strikes the backplate is 54.5 feet per second and that the velocity of the gun at full recoil is 5.2 feet per second.

10. *Analysis of events at end of recoil*

The purpose of the backplate buffer is to reverse the bolt motion at the end of the recoil stroke with the minimum possible loss of time and energy. In the analysis of gas operation, it was pointed out that although the reversing action is accomplished almost instantaneously by causing the bolt to rebound from an extremely stiff elastic member, the impact is accompanied by a loss of energy with the result that the velocity of the bolt after impact will be at best approximately 60 per cent of its striking velocity.

Since the velocity of the bolt before striking the backplate is 54.5 feet per second, the momentum of the bolt is:

$$MV=\frac{5}{32.2}\times 54.5=8.45 \text{ (lb. sec.)}$$

Assuming that the coefficient of restitution of the backplate is 0.60, the velocity of the bolt after impact will be:

$$V=.60\times 54.5=32.7\left(\frac{\text{ft.}}{\text{sec.}}\right)$$

The momentum of the bolt will then be:

$$MV=\frac{5}{32.2}\times 32.7=5.09 \text{ (lb. sec.)}$$

Thus, in the reversing action of the backplate, the change in bolt momentum is equal to $8.45+5.09=13.54$ (lb. sec.). If the entire reversing action occurs in 0.001 second, the average force exerted on the backplate must be:

$$\frac{13.54}{.001}=13{,}540 \text{ (lb.)}$$

The striking energy of the bolt is:

$$KE=\frac{1}{2}MV^2=\frac{1}{2}\times\frac{5}{32.2}\times 54.5^2=230 \text{ (ft. lb.)}$$

If it is assumed for purposes of estimation that the backplate offers a constant resistance of 13,540 pounds, the deflection of the elastic member under the impact will be:

$$\frac{230}{13{,}540}=.0170 \text{ (ft.) or approximately } \tfrac{3}{16} \text{ (inch)}$$

The data derived above are used to complete the bolt motion curves for the 0.001-second interval assumed for the action of the backplate. The curves

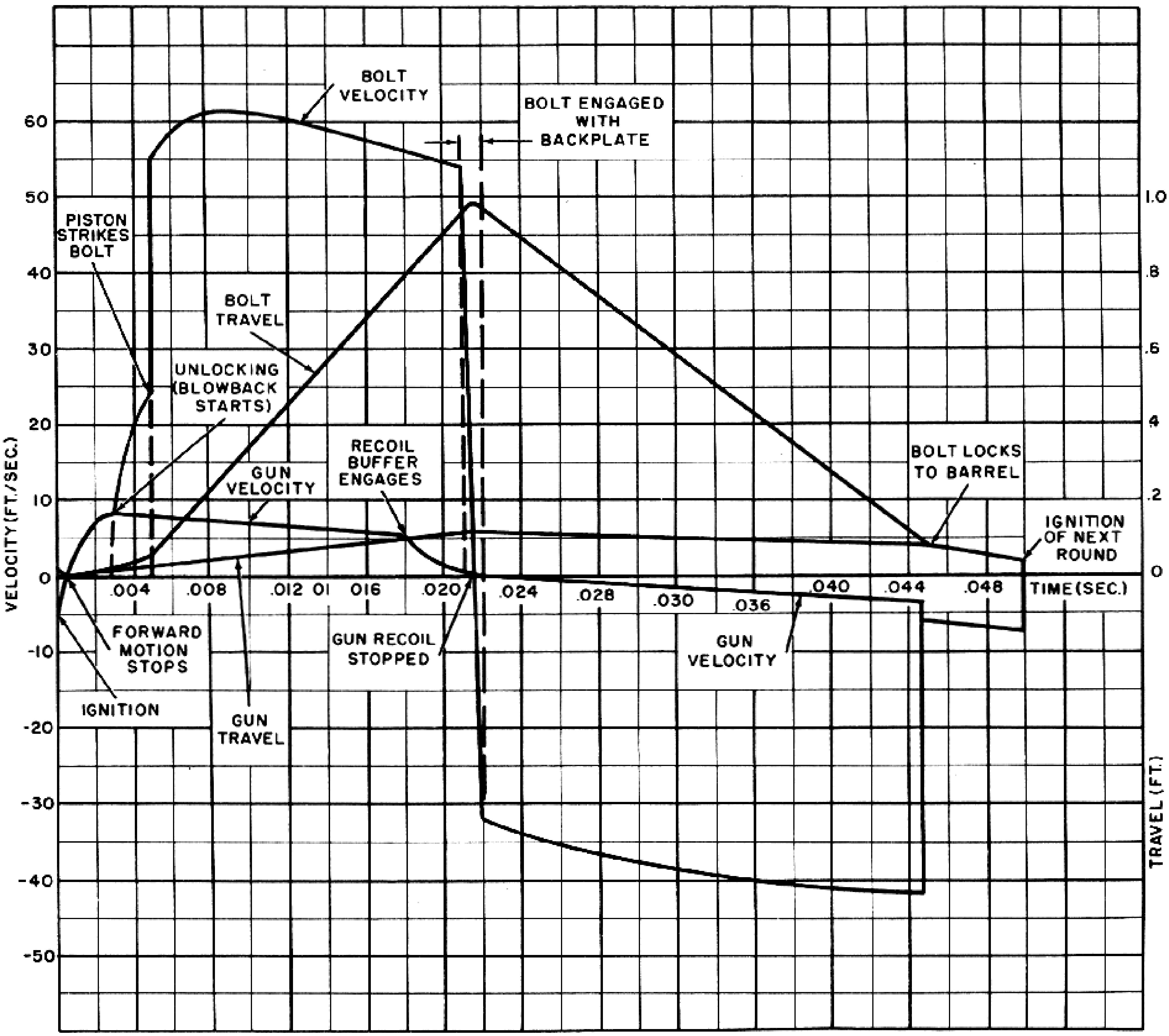


Figure 3–34. Complete Time-Travel and Time-Velocity Curves for Gas-Operated Gun.

are shown in fig. 3–34 which gives the theoretical time-travel and time-velocity curves for the complete cycle.

As explained in the analysis of gas operation, the gun motion must be controlled so that the gun recoil movement is halted just as the bolt strikes the backplate and proper buffer action must be provided so that the gun velocity will be kept low for a reasonable amount of time before and after the bolt impact. This arrangement is necessary in order to insure that slight variation in the instant at which impact occurs will not cause erratic rebound action. No attempt will be made here to design an actual buffer but it will be assumed that a hydraulic buffer is used to produce the velocity characteristic shown in fig. 3–34. Instead of permitting the gun to recoil the entire distance opposed only by the barrel spring as shown in fig. 3–33, the hydraulic buffer starts to act at 0.18 second and reduces the gun recoil velocity from 5.8 feet per second to zero at 0.0210 second. At this point a positive stop is engaged and therefore the impact of the bolt on the backplate does not impart any rearward velocity to the gun mass. (Note that for a considerable time interval on either side of the instant of bolt impact, the gun velocity is in the order of only one or two feet per second, which is very close to zero when compared to the bolt striking velocity of 54.5 feet per second.)

11. *Counter-recoil motions of bolt and gun*

While the bolt and gun are moving forward, the bolt motion is influenced by the force of the bolt driving spring, but the gun motion is influenced by both the force of the barrel spring and the force of the bolt driving spring. (Since the bolt is free, the force of its spring opposes the gun motion.) The counter-recoil motions are determined by essentially the same method previously employed for analyzing the effects of the springs.

The first step is to draw the free bolt velocity curve which is a horizontal line at minus 32.7 feet per second and the free gun velocity curve which is a horizontal line at zero feet per second (fig. 3–35). The subsequent procedure must be modified slightly because the springs are aiding the motions rather than retarding them. At the start of the counter-recoil movement the barrel spring is compressed 0.125 foot and hence, if the spring losses were ignored, the initial compression of the spring for purpose of computing the counter-recoil motion would be:

$$F_{o_1}+K_1D=200+122.5\times12\times.125=383 \text{ (lb.)}$$

However, if 10 per cent is allowed to account for the spring losses, the force exerted by the barrel spring at the start of counter-recoil will be 345 pounds. The bolt driving spring is compressed 0.833 feet and, ignoring spring losses, its force would be:

$$F_{o_2}+K_2D=25+(10\times12\times.833)=125 \text{ (lb.)}$$

Again allowing 10 per cent for the losses, the force exerted by the bolt driving spring at the start of counter-recoil will be 112.5 pounds.

Since the bolt is free of the gun, the force of the bolt spring must be subtracted from the barrel spring force to give the effective force acting on the gun as 345 − 112.5 = 232.5 pounds. The velocity gain due to this force would be:

$$V=\frac{F_o}{M}\,t=\frac{232.5\times32.2}{55}\,t=136\,t$$

The effect of this gain in velocity is shown in fig. 3–35 by the line designated as step 2 for the gun. The gain in bolt velocity due to the initial force of the bolt driving spring would be:

$$V=\frac{F_o}{M}\,t=\frac{112.5\times32.2}{5}=700\,t$$

The effect of this gain in velocity is shown in fig. 3–35 by the line designated as step 2 for the bolt.

The effect of the spring constants is determined by using the curves designated as step 2 to obtain a first approximation of the gun and bolt travels (curves designated as steps 3 in fig. 3–32). The first approximation of the relative motion between the gun and bolt is obtained by subtracting the gun travel curve of step 3 from the bolt travel curve of step 3. This relative motion curve is used during recoil to determine the effect on the gun of the spring constant for the bolt driving spring. Combining this effect with the effect of the barrel spring on the gun gives the gun velocity curve designated as step 5 in fig. 3–35. Since the change in the velocity curve is so slight, it is not necessary to modify the gun travel curve obtained in step 3.

NOTE: Since the gun is moving forward, the effects of the spring constants must be applied opposite to the way they were applied during recoil. (Here, the effect of the spring constant for the barrel spring must be subtracted from the curve obtained in step 2 and the effect of the bolt spring constant must be added.) The validity of this procedure can be demonstrated by examining the equation expressing the change in gun velocity due to the effect of the barrel spring alone.

$$\Delta V=\int_{t_1}^{t_2}\frac{F\,dt}{M}=\int_{t_1}^{t_2}\frac{F_o+K\,(D-d)}{M}\,dt$$

where: D is the total distance the spring is compressed at the start of the forward motion;

and, d is the forward movement of the gun from its rearmost position.

$$\Delta V=\frac{1}{M}\int_{t_1}^{t_2}(F_o+KD)\,dt-\frac{1}{M}\int_{t_1}^{t_2}(Kd)\,dt$$

$$=\frac{F_o+Kd}{M}\,t-\frac{K}{M}\int_{t_1}^{t_2}(d)\,dt$$

which is the equation defining the procedure described above for handling the effect of the barrel spring.

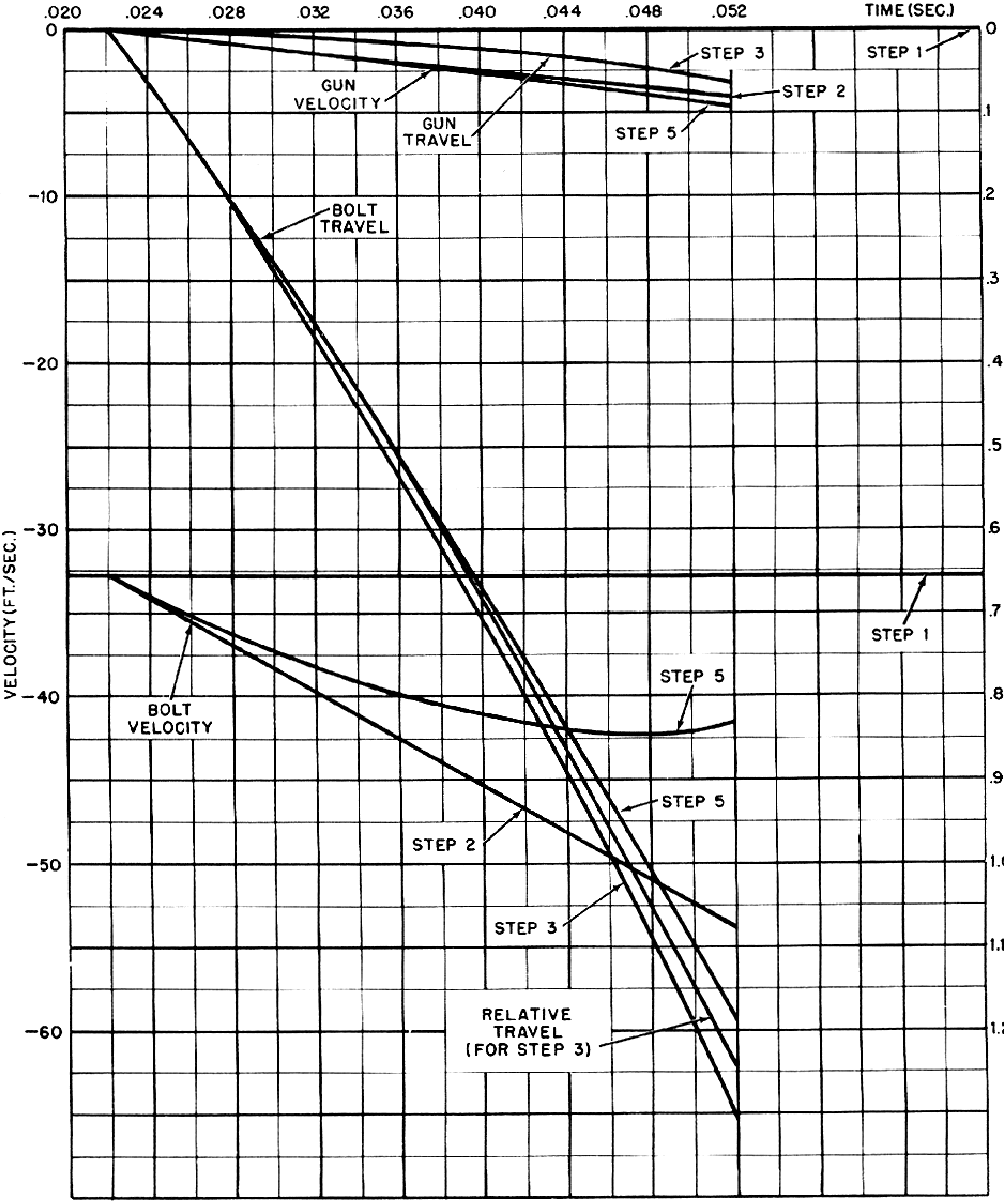


Figure 3–35. Effect of Bolt Driving Spring and Barrel Spring During Counter-Recoil.

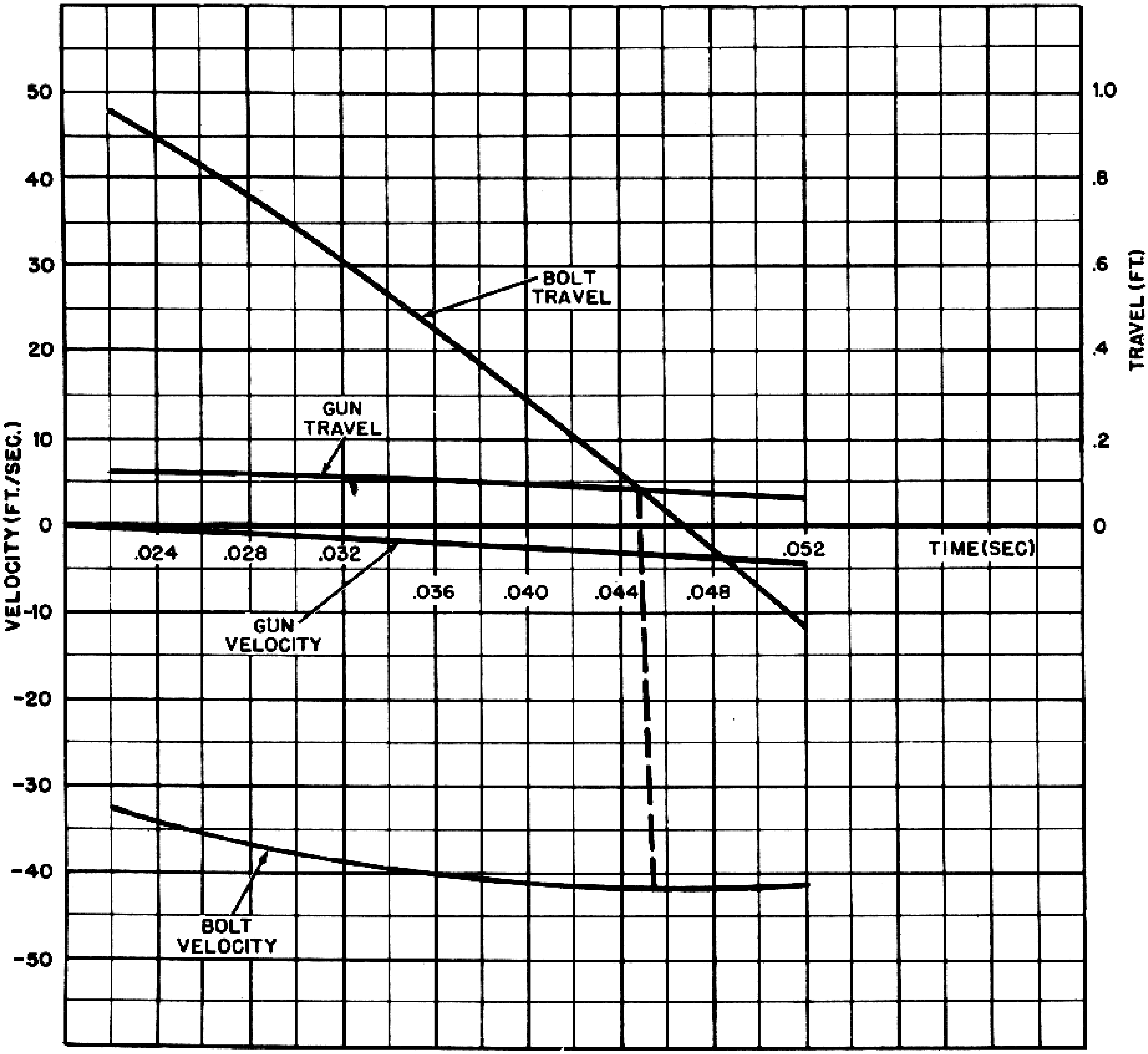


Figure 3–36. Extension of Time-Travel and Time-Velocity Curves for Interval Before Bolt Relocks to Barrel.

The effect on the bolt of the spring constant for the bolt driving spring is determined by using the relative motion curve and following the procedure described in the preceding note. The resulting velocity curve is designated as step 5 in fig. 3–35. Integrating under this curve gives the bolt travel curve designated as step 5.

Fig. 3–36 shows the travel and velocity curves obtained by using the data in fig. 3–35 to extend the curves previously constructed. Note that the bolt travel values obtained from fig. 3–35 are subtracted from 0.96 foot, which is the displacement at which the bolt leaves the backplate. Similarly, the gun travel values are subtracted from 0.125 foot. The travel curves show that the bolt meets the barrel at 0.448 second. At this instant the bolt is moving at 42 feet per second and the gun velocity is 3.5 feet per second. The displacement from the firing position at which the impact occurs is 0.085 foot.

12. *Gun motion after bolt locks to barrel*

When the bolt locks to the barrel, the gun velocity is 3.5 feet per second and the bolt velocity is 42 feet per second. Since the gun and bolt are locked, the parts will assume some common velocity which is determined by the fact that the momentum of the combined mass after impact will be the same as the total momentum before impact. That is:

$$M_rV_3=M_1V_1+M_2V_2$$

For the conditions of the example:

$$\frac{60}{32.2}V_3=\frac{55}{32.2}\times3.5+\frac{5}{32.2}\times42$$

$$V_3=6.7\ (\text{ft./sec.})$$

In the construction of the curves showing the motions up to the time of unlocking (fig. 3–27) it was shown that the gun is fired during counter-recoil when it is still 0.0207 foot from its most forward position. As shown in fig. 3–34, the gun is 0.0805 foot from the firing position when the bolt locks to the barrel. Therefore the remaining travel to the firing position is .085—.0207=.0643 foot. Since the velocity of the gun is 6.7 feet per second at this instant, the time required for the gun to travel 0.0643 foot would be approximately 0.009 second. This time is actually too long for efficient operation, since the locking action can be completed and the effects of the shock of locking can settle out easily within 0.002 or 0.003 second. For this reason, the gun should be fired at least by 0.050 second as shown in fig. 3–34. However, at 0.050 second the gun will not have reached the 0.0207-foot displacement shown in fig. 3–27.

This situation arises because it was necessary at the outset of this analysis to estimate the required barrel spring characteristics before sufficient data were available to determine the timing accurately or to take into account such detailed factors as the effect of the impact of the returning bolt. The discrepancy resulting from the fact that the estimate was not exact is slight and for present purposes, the curves shown in fig. 3–34 show good enough agreement to be used for preliminary design purposes. In an actual design problem, better agreement could be attained by recomputing the curves for a slightly stronger barrel spring. Having the data shown in fig. 3–34, a better estimate can be made of the factors controlling the required strength of the spring.

The effect of the barrel spring after the locking occurs is determined as follows for completing the curves of fig. 3–34. The gun velocity at this instant has been shown to be 6.7 feet per second. As the gun moves forward, this velocity will be increased slightly by the force of the barrel spring. Since the time remaining until the gun is fired is so short, it will be sufficiently accurate to estimate the effect of the barrel spring force by assuming that the average force will be that which exists at a displacement of 0.04 foot. This force is:

$$F_{o_1}+K_1D=200+122.5\times12\times.04=259\ (\text{lb.})$$

The increase in velocity produced by this force will be:

$$V=\frac{F}{M}t=\frac{259\times32.2}{60}t$$

$$=139\ t$$

This increase in velocity is indicated by sloping the velocity curve after locking occurs by an amount corresponding to a velocity increase of 139 feet per second. This curve shows that the average velocity for a considerable interval after locking can be taken as approximately 7.0 feet per second. Using this average velocity, the distance traveled by the gun from the instant of locking at 0.0448 second to the desired instant of firing at 0.050 second will be:

$$D=V_{av}\ t=7.0\ (.050-.0448)$$

$$=7.0\times.0052=.0364\ (\text{ft.})$$

Thus, the position reached by the gun when it is fired at 0.050 second is 0.8050—.0364=.0441 foot from its most forward position. The velocity with which it reaches this position is approximately 7.3 feet per second. As previously explained, these values differ slightly from the original estimates but are close enough for present purposes. With the cycle time of 0.050 second, the rate of fire is 1200 rounds per minute as required.

In conclusion, it should be noted that the curves shown in fig. 3–34 represent the motions which occur after a burst is in progress. The conditions for the first shot in a weapon which uses advance primer ignition require special consideration. Since the action in this case is based on a forward velocity of the gun at the instant of firing, the charging mechanism must be arranged so that the effect of advanced primer ignition is obtained for the first shot. If this is not done, the first recoil will be excessively violent.

Chapter 4

ROTARY CHAMBER MECHANISMS

The preceding chapters of this publication are concerned with the operating and design principles of automatic weapons which employ a reciprocating bolt and derive the energy required for full automatic operation from blowback, long recoil, short recoil, or gas sources. There is another class of weapon, known as the revolver cannon, which is sufficiently different from the weapons described up to this point to merit individual and careful consideration. The principal difference between the revolver cannon and the reciprocating cannon does not lie in the source of operating energy because all machine guns of the full automatic type described in this publication use a portion of the energy released by the explosion of the propellant charge of each round to clear the weapon and to load and fire the next round in such a manner as to produce sustained or repetitive fire. Furthermore, the difference does not occur in the system whereby the energy is derived and applied since revolver cannons of the full automatic type obtain the power for their operation from recoil, gas, or even residual pressure in accordance with the same principles which are applicable to guns having a reciprocating bolt type of action. The real difference between the reciprocating cannon and the revolver cannon lies in the arrangement of the mechanical functions constituting the automatic cycle of operation and in the mechanisms associated with these functions.

Thus it appears that the classification of machine guns into the two groups, reciprocating weapons and rotary action weapons, represents a different type of classification than has been used in the preceding chapters of this publication. In these chapters, primary attention is given to the *systems* by means of which energy is derived from the explosive pressures developed in the combustion of the propellant charge and is applied to produce the mechanical motions necessary for sustained automatic fire. In treating each of these systems (blowback, long recoil, short recoil, and gas), the basic mechanical functions used to illustrate the application of the operating energy are the functions associated with reciprocating type actions. Actually, since these same systems are applied in a very similar manner to the actuation of rotary action weapons, it would have been entirely possible to cover the application of the recoil, gas, and blowback systems to rotary action mechanisms at the same time that their application to reciprocating mechanisms was described. However, the problems arising in the design of rotary action weapons are so special that it is practically mandatory to consider them separately.

HISTORICAL BACKGROUND OF THE REVOLVER

In order to provide a background for the description and analysis of the revolver cannon, a brief review of the history of this type of weapon will be given to clarify the factors which led to its adoption and which affect its relationship with the reciprocating type of machine gun.

Ever since the first recorded use of firearms early in the fourteenth century, there has been a need and a demand for multiple firing weapons. It is interesting to observe that many attempts to create such weapons were made almost immediately after the application of gun powder to the propelling of missiles and that multiple fire was originally achieved by the simple and rather obvious method of stacking a number of barrels side by side with an arrangement for firing the barrels simultaneously or in rapid succession. Although they were remarkable weapons in their day, the "orgues des bombardes", or battery guns, of the fourteenth century were extremely heavy and clumsily mounted and were only moderately successful because of the difficulties encountered in loading the barrels and in igniting the charges as desired.

During the fifteenth and sixteenth centuries, little progress was made in the field of multiple firing but during this time, attempts were made to place a number of barrels in a circular mounting rather than in the flat mounting previously used. This expedient made for a more compact and manageable weapon but still left much to be desired. Although the first guns employing circular mounting were in effect merely several independently functioning guns assembled as one unit, arrangements were soon developed to rotate the barrels about a common longitudinal axis so that each barrel could be brought successively into position to be fired by a single ignition device. During the sixteenth, seventeenth, and eighteenth centuries, many such arrangements were attempted using the match lock, wheel lock, and flintlock methods of ignition. Although none of these arrangements were successful enough to enjoy any wide acceptance, they did establish the basic idea of the revolver-type weapon. The upper drawing in fig. 4–1 shows the general appearance of a three-barrel weapon of the match lock era in which the barrels are revolved by hand. The second gun shown in fig. 4–1 is a seven-barrel rifle employing flintlock ignition. Each barrel was fired as it revolved into alignment with a fixed flintlock firing system.

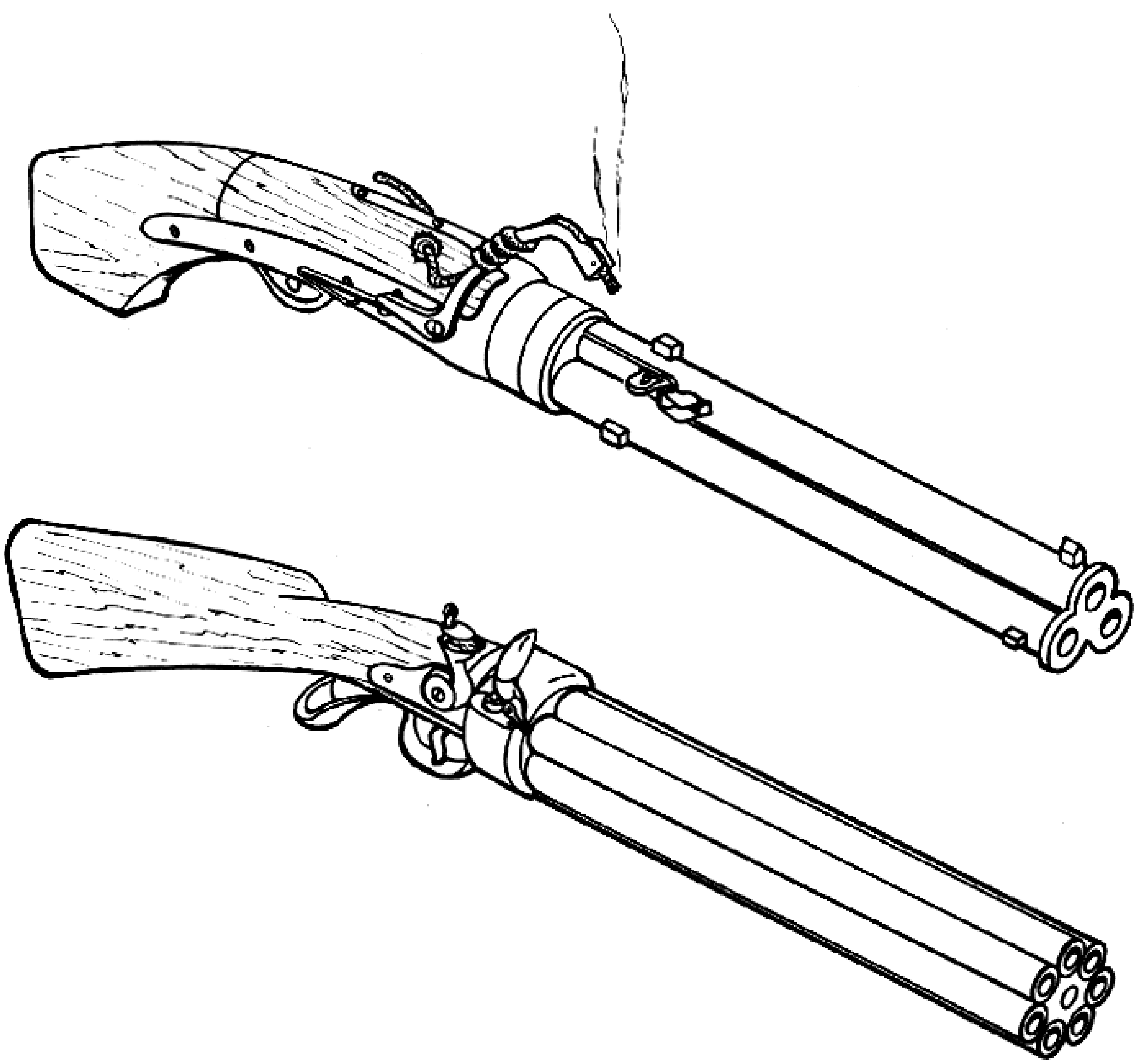


Figure 4–1. Matchlock Revolving Gun and Revolving Flintlock Rifle.

At various times throughout the period described above, many inventors proposed a type of arrangement that was representative of a very important principle in weapon construction. In order to avoid the heavy and clumsy assembly that resulted from placing a number of complete gun barrels in a single circular mounting, it appears to be natural and convenient to use but one barrel and to provide the rear end of the weapon with a cylinder into which several chambers were bored. These chambers could be loaded separately and then the cylinder could be rotated so that each charge could be brought successively into alignment with the barrel and fired. This principle is the same as that used in modern rotary action weapons. Its importance lies in the fact that it provides a rapid and convenient means for achieving repetitive firing and at the same time permits the construction of a weapon which need only be slightly more bulky than an ordinary single-shot arm.

In the course of the eighteenth century, there were a number of notable advances in the development of revolver weapons, particularly the introduction of mechanical means for rotating the cylinder and of a method for providing a gas-tight joint between the cylinder and barrel. An American inventor, Elisha Collier, was particularly active in promoting the rotating cylinder weapon and was responsible for many of the design improvements made in this field. One of his guns gained considerable favor and was even used in India by the English army. However the flintlock method of ignition then in use was not at all suited for use in repeating weapons. It was not until the early part of the nineteenth century, with the advent of percussion ignition, that any real progress was made toward achieving the development of practical and effective multifiring arms. The use of the percussion cap at last freed inventors of the limitations and almost hopeless complications imposed by the older methods of ignition. In 1830, Samuel Colt invented one of the first practical revolver mechanisms employing percussion ignition in conjunction with automatic revolution and locking of the cylinder by the act of cocking the hammer. This invention not only led to a long line of very popular and efficient firearms but it also initiated a veritable flood of inventions relating to the improvement of the revolver-type weapon. Indeed, the four or five decades following Colt's invention may be said to represent the "Golden Age" of the revolver. Soon after the beginning of this relatively brief period, in spite of the inconveniences attendant upon the use of the percussion cap and muzzle loading, the magnificent gun craftsmen and inventors of these times had produced almost every conceivable form of revolver mechanism. Each patent granted on a revolver mechanism or on an improvement in existing weapons was apparently the signal for a new flurry of inventions directed toward circumventing that patent or toward accomplishing still greater improvement. A study of the patents of this period forcibly demonstrates the ingenuity and vision which seemed to be so prevalent in the field of gun development. Even with percussion cap ignition, the revolver hand gun was developed and refined until it reached a high stage of compactness, ruggedness, and reliability. In addition to this steady process of refinement, many more radical ideas were advanced such as methods for sealing the gap between the cylinder and the barrel, arrangements for employing two or more cylinders in tandem so as to increase the number of shots which could be fired before reloading, and a host of other ideas for minimizing the disadvantages of the revolver weapon and overcoming its limitations.

Until the 1860's, the application of revolver principles was confined almost exclusively to hand guns, except for a short-lived adoption by the United States Army of the Colt Revolving Rifle, Model 1855, which did not prove successful under service conditions. However, during the Civil War, both the Union and Confederate forces became interested in multifiring weapons and particularly in adequate machine gun mechanisms capable of producing sustained fire. Beginning in 1861, various machine guns of several types were made available to the armies of both the South and the North but the full advantages of the machine gun in battle were not realized by the military experts of the day and the guns received only limited use, largely for fixed defensive installations.

Of the multifiring guns produced as a result of the stimulus of the Civil War, some were battery guns, some used the reciprocating principle, and some were based on the rotating mechanism principle. However, by far the most remarkable weapon to be conceived during this period was the machine gun invented by Richard Gatling. This was a hand-cranked multi-barreled rotary action weapon which was basically so sound in principle and so amenable to further improvement that it enjoyed world-wide fame and employment for the remainder of the nineteenth century. In fact, this gun was not declared obsolete by the United States Army until 1911.

The advent of the Gatling gun and the development of cartridge ammunition ushered in a new period in gun development. The machine gun had at last come into its own and inventors all over the world began a new wave of development to create competition for the Gatling gun. Many of the weapons produced at this time were of the rotary action type and many had reciprocating mechanisms, but in all the power for operation was supplied through a hand crank or lever turned by the operator. Although these guns were considered at the time to have reached a stage where no further improvement was possible, the era of the hand-operated machine gun came to an end with the invention of the first full automatic machine gun mechanism by Hiram Maxim in 1883. Of course, the full automatic gun did not immediately sweep the world market and instantly eliminated the hand-operated machine gun. Nevertheless, by 1888 most of the major powers of the world had ordered some of Maxim's guns for trial and long before the outbreak of World War I, the Maxim-type gun had been adopted as a first line weapon. The hand-operated machine gun was a thing of the past.

After Maxim's invention and successful development of his full automatic weapon, every significant new invention in the field of machine gun development was directed toward exploiting the principles of full automatic operation. The Maxim gun derived the energy for its operation from the recoil forces produced by the explosion of the propellant charge. The so-called Skoda machine gun patented in 1888 operated by the retarded blowback system and in 1890, John M. Browning produced a successful design for the first automatic gas-operated machine gun. These basic systems (recoil, blowback, and gas operation), all invented before the last ten years of the nineteenth century, have been used ever since to provide the power required for producing sustained fire.

It is particularly important to note that the first full automatic machine guns, whatever system was used to obtain operating energy, were all of the reciprocating type. The reciprocating mechanism was apparently well-adapted to the systems used for obtaining operating energy and, for the most part, the weapons produced even in the early days of full automatic operation were amazingly simple and efficient. According to reliable records of very exhaustive and demanding tests, the reciprocating machine gun could easily produce a rate of fire which was far in excess of that imposed by the military requirements of the years preceding World War II. At the same time, it was clear that the reciprocating principle could be applied to produce weapons which were reliable in performance, easy to maintain, and simple to manufacture. During the years following Maxim's invention, the gun industries of the world produced hundreds of new machine guns and thousands of patents related to the advancement of machine gun mechanisms. However, all of the successful designs adopted for military service were based on the reciprocating principle of operation and the rotary action-type mechanisms, so popular in the day of the manually-operated machine gun, were practically abandoned. The entire inventive effort in the field of machine guns was directed toward refining every detail, exploring every possible type of mechanism, and finding every conceivable combination and permutation of the principles relating to reciprocating actions.

Out of this maze of new weapons, new inventions, and re-invention of old inventions, there finally evolved a relatively small number of basic actions which had been proved by experience, both in test and in battle, to be sound enough and reliable enough to warrant their continued use. Once these actions were settled on, their development and refinement continued in an effort to eliminate "bugs" and to improve performance. Indeed, many of the weapons resulting from this intensive effort left little to be desired. They were powerful, efficient, relatively simple to maintain, and capable of mass production when needed. In a word, they were thoroughly practical and satisfactory weapons for the needs of their time, but again, as has happened so many times in the history of warfare, a new need developed.

From the latter days of World War I and particularly in the early stages of World War II, the rapid strides being made in the improvement of military aircraft confronted the gun industries of the world with an urgent and difficult problem. The new conditions of air combat made it imperative to have guns with greater muzzle velocity, larger caliber, and higher rates of fire. In response to this demand, the machine gun was advanced in tremendous strides until it reached calibers that formerly were considered in the domain of artillery and achieved rates of fire that were comparable with those formerly obtained with rifle caliber guns. In spite of these successes, the ever-increasing demands for even greater power and higher rates of fire continued and it became apparent at last that each succeeding improvement was accomplished with much greater difficulty than the last. In fact, it is generally conceded by many prominent gun designers and ordnance experts that, considering the state of modern metallurgy and technology, the conventional reciprocating gun may have either already reached or may be very close to the limit of its development potentialities. This does not mean that there is no room for further improvements affecting reliability, ease of maintenance, or simplicity in manufacture but merely that the basic physical and dynamic characteristics of the reciprocating mechanism probably do not lend themselves to permitting rates of fire to be increased very far beyond present-day values.

The design problems associated with the reciprocating mechanism were fully appreciated early in World War II and the German Luftwaffe had urgent need, late in 1942, for a high-rate-of-fire, high-muzzle-velocity 20-mm aircraft machine gun. In an attempt to avoid the difficulties experienced with the development of high-performance reciprocating-type guns, the Germans resorted to the rotary chamber mechanism principle and undertook the development of a 20-mm gas-operated rotary-action-type weapon. Under the pressure of the emergency Germany was facing, the development proceeded rapidly and by 1943 the first successful pilot model of the rotary action weapon was constructed. This was the first automatic rotary action cannon in which the cylinder is reloaded during firing. By the time Germany was overrun by the Allied forces, only a few prototypes of the weapon had been constructed. One of these guns was captured at the end of the war and was shipped to the United States. This captured weapon has since formed the basis for an extensive rotary chamber mechanism development program in this country. As a result of the renewed interest in the rotary action principle, this type of weapon once again enjoys equal footing with the reciprocating gun and many ordnance designers feel it holds great promise for future improvement.

It is interesting to note that the German program started in 1942 for the development of a rotary action machine cannon revived the use of principles which had been applied originally many years before but which had been neglected for a long time prior to World War II. One of the first recorded applications of the rotary chamber principle occurred as early as the year 1718 in a patent issued to James Puckle of England. Again, in the era of the American Civil War, a hand-operated machine cannon was developed which was similar in many striking ways to recently developed full automatic rotary action cannon. This was the DeBrame gun patented in 1862. One of the most amazing predecessors of the rotary action machine gun developed by the Germans is a weapon patented in the United States by an inventor named C. M. Clarke in 1905. This gun is a gas-operated, full automatic rotary chamber mechanism in which the mechanism is powered by an operating slide driven by a gas piston. It employs belt feeding, power ramming, and almost every other basic feature that distinguishes the modern family of rotary action weapons inspired by the German design. In fact, this gun is so representative of the rotating mechanism class that it is used later in this chapter to illustrate the basic functioning of these weapons. As to why this very advanced design lay dormant for almost 40 years before being rediscovered, it is probable that its inventor could not have chosen any worse time than 1905 for being inspired to conceive of this particular kind of machine gun. It was just at this time that the Maxim gun, Browning gun, and other reciprocating weapons were nearing the peak of a tremendous wave of popularity. The entire gun world was so preoccupied with the success of these new weapons that the Clarke gun simply failed to be noticed.

From the foregoing brief history of the place of the rotary action weapon in the development of guns, it is evident that there has been a definite pattern underlying the manner in which the rotary action principle has been utilized. The very first rudimentary attempts at achieving a multiple firing weapon quickly led to the basic and obvious device of grouping a number of individual barrels but this idea could not be advanced very far because of the lack of a suitable method for igniting the propellant charges in the barrels and because of the great bulk of the assemblies. As soon as an even slightly better method of ignition evolved, great improvements in the construction of multiple firing weapons occurred almost immediately as is witnessed by the fairly advanced revolving arms of the matchlock, wheel lock, and flintlock eras. However, these weapons, too, were not sufficiently effective and convenient to compensate for their relatively great unwieldiness and complexity and therefore never were extensively used to replace the simpler and more reliable single-shot weapons of the times. Again, in the early part of the nineteenth century, the invention of the percussion cap led quickly to a truly remarkable period of revolver development. The introduction of cartridge ammunition in the latter part of the same century produced even greater improvements in revolver mechanisms, and finally brought the sustained-fire weapon into the realm of practicality in the form of the Gatling gun. Although this weapon and other sustained-fire revolver arms of the period were very effective, they still were rather cumbersome and inconvenient and were soon abandoned after the full automatic reciprocating mechanism came into being. Now, with the new problems facing the gun designers of today, the rotary chamber mechanism principle is again being exploited.

Thus, it seems from the evidence of history that the rotary action principle has several times alternately risen to popularity because of its advantages and then fallen into disuse because of its shortcomings. The basic advantages of the rotary chamber mechanism are many. The very simplicity of the fundamental operating concept is in itself one of the rotary chamber mechanism's most important advantages. The mechanism is essentially rugged and durable and provides for very positive control in the handling of the ammunition. Unfortunately, these and the other advantages of the rotary chamber mechanism are accompanied by an almost equal number of disadvantages, some of which are relatively minor and some which are more serious. The following analysis will consider the rotary action principle from the general standpoint and then will cover the individual physical and operational characteristics of the mechanisms employing this principle in an attempt to throw some light on the basic problems encountered in the design of full automatic rotary chamber cannon.

PRINCIPLES OF MULTIPLE CHAMBER WEAPONS

The term "revolver" as applied with reference to firearms usually first calls to mind a type of mechanism which is best represented by the modern revolver hand gun. The fundamental elements of the mechanism are a single barrel and a cylinder which is bored with chambers for a number of cartridges. The cylinder is mounted so that it can be rotated to bring each chamber successively into alignment with the barrel. The revolver mechanism, of course, receives its name from the fact that the cylinder is rotated in order to achieve repetitive fire. However, in a larger sense, the true principles of the rotary action mechanism can be best understood and the properties of the mechanism can be more fully appreciated by reducing the concept to an even more elementary level. When all the nonessential factors are eliminated, the one remaining characteristic is that in the operation of the mechanism, each chamber is moved laterally with respect to the position at which it is fired. It is this lateral movement which holds the key to all of the characteristics peculiar to weapons of the multiple chamber type and also the key to the characteristics of rotary action guns which represent but one form of multiple chamber weapons.

Because the concept of the multiple chamber weapon forms the basis of all rotary action weapons, it is appropriate here to examine briefly the general nature of mechanisms which fall into the multiple chamber class. The most elementary form of the multiple chamber weapon is the battery gun or "orgue des bombarde" mentioned previously in this chapter. This gun, which consists merely of an assembly containing a number of individual complete barrels can hardly be considered as a mechanism since the only function required in its use is to obtain simultaneous or nearly simultaneous ignition of the propellant charges in all the barrels. A slightly more advanced type of weapon is exemplified by the conventional double-barrel shotgun, drillings, and other firearms which amount to nothing but two or more separate guns compactly assembled into a single stock. Although the elementary battery gun and firearms similar to the double-barrel shotgun are so simple in principle that no further discussion of them is called for here, consideration of the other forms of multiple chamber weapons brings to light a number of interesting and significant points.

One form of multiple chamber weapon, which was conceived very early in the history of firearms and has since reappeared at intervals with varying degrees of success, is the type of repeating weapon which employs a group of complete barrels. The salient feature of these weapons is that repetitive fire is achieved with the use of only one operating mechanism which serves a number of barrels. This means that for all intents and purposes, additional capacity can be built into the weapon merely by adding to the number of barrels without the necessity for greatly increasing the complexity of the operating mechanism or the bulk of the supporting structures. The particular form of a weapon in this class depends entirely on the manner in which the various parts are arranged in order to permit the barrels to be served by the operating mechanism. The fundamental arrangement may be such that the barrels remain fixed while the operating mechanism is moved from one barrel to the next, but the more commonly used method is for the barrels to move with respect to a fixed position of the operating mechanism. In some weapons, the barrels are placed parallel to each other in the same plane and the assembly slides laterally to move the barrels successively into the firing position. These so-called "harmonica" guns have not been used extensively because of the rather obvious disadvantages of such a mechanism for most applications. A much more suitable and practical arrangement is obtained by placing the barrels in a circular group so that the entire group can be rotated about an axis parallel to the bores. This type of mounting was employed in the revolving weapons of the matchlock and flintlock eras, in the popular "pepperbox" pocket pistols of the last century, and in the Gatling gun and other guns of similar construction. Even in recent years, this same principle of construction has been applied in the development of high-rate-of-fire machine guns. Although the multiple barrel principle is sound and its use eliminates many of the difficulties encountered with other forms of revolver cannon, weapons constructed on this principle suffer a severe handicap from the viewpoint of weight and bulk as well as other disadvantages which must be accounted for.

The remaining type of multiple chamber weapon is the important type in which the chambers are separate from the forward portion of the barrel. Weapons in this class usually have one fixed barrel (although there may be more than one barrel) and the chambers are moved successively into alignment

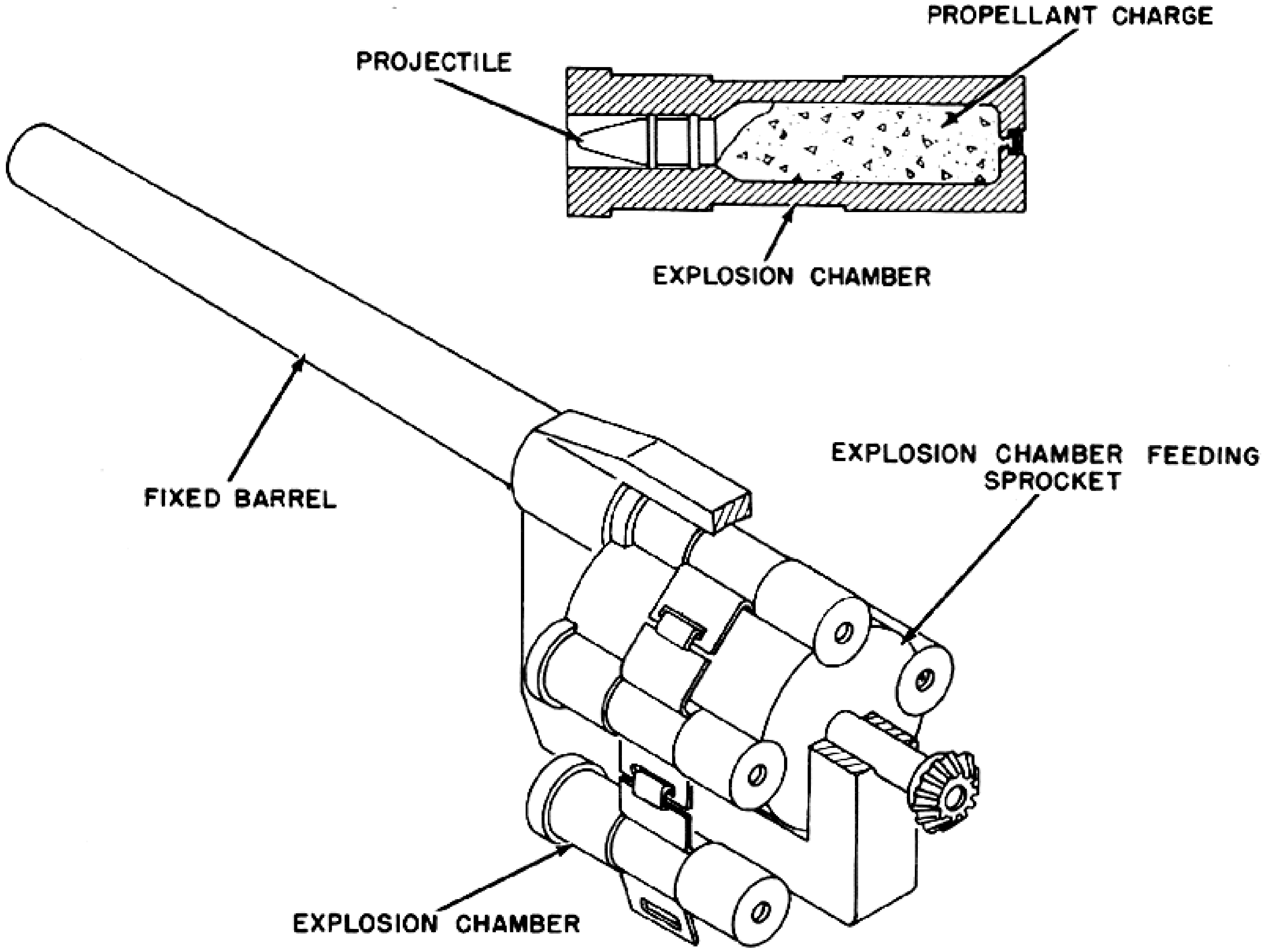


Figure 4–2. Principle of the "Coffee Mill" Type Gun.

with the barrel. Thus, each time a chamber is brought into position behind the barrel, the gun tube is completed and the weapon is in readiness for firing. This arrangement has the advantage of permitting repetitive fire without the necessity for having the weight and bulk of a large number of complete barrels. As mentioned previously in this chapter, this idea was conceived at a very early date in the history of firearms and, in fact, one of the first patented manually operated revolving type machine guns employed the multiple chamber principle. This remarkable weapon, invented in 1718 by James Puckle of England, had a number of separate explosion chambers which could be preloaded and were mounted in a circular array so that they could be successively revolved into alignment with a single barrel. The rotation was accomplished by means of a hand crank and the propellant charges were ignited by the slow-match method.

Over the years, the same general idea of having a single barrel with a number of explosion chambers appeared in a multitude of forms. In many cases, the individual explosion chambers were all bored into a single cylinder in a manner similar to that employed in the typical modern revolver hand gun. This arrangement, which is simple, easily mechanized, and effective, has been so prevalently used that in the popular mind it has become practically synonymous with the idea of the multiple chamber weapon. However, it should be realized that the revolving cylinder mechanism represents but one of the many possible devices which can be used to make use of many chambers. It may be said that the archetype of the multiple chamber weapon is found in a machine gun developed in the United States near the middle of the nineteenth century. This highly interesting weapon, the Ager "Coffee Mill" gun, which was used by the Union forces during the Civil War, was designed so that each charge of powder and ball was loaded into a separate steel container. The steel container forms the explosion chamber for the round and a percussion cap is placed on a nipple screwed into the rear end of the container. In the operation of the gun, a number of loaded containers (explosion chambers) are placed in a hopper at the top of the gun and when the operating hand crank is turned, one explosion chamber at a time rolls down into a recess at the rear of the gun barrel. The gun shown in fig. 4–2 is a simplified version of this type of weapon. A system of cogged rollers operated by the hand crank guides the explosion chamber into place and allows the chamber to be shoved forward to form a prolongation of the barrel and to be locked from behind by a wedge. As the hand crank is turned, the round is fired and the rotation of the cogged rollers causes the empty chamber to be ejected from the weapon at the side. As the fired chamber is ejected, a loaded chamber is brought into place to start a new cycle of operation.

Another unique arrangement for handling explosion chambers which are separate from the barrel is found in a hand gun developed later in the nineteenth century. Although the mechanism happened to be applied in a small caliber, the same device could have been used just as easily in a larger weapon. This gun, which has the surprisingly large capacity of 20 shots, fires cartridge ammunition and has an individual chamber for each cartridge. The separate chambers are linked together with metal plates so as to form an endless loop similar in appearance to an ordinary bicycle chain as shown in fig. 4–3. As the gun is fired, a ratchet-type cylinder within the frame pulls the chain of chambers through the frame laterally in such a way as to bring the chambers successively into alignment with the barrel. Before each cartridge is fired, the gun mechanism securely locks the chamber in place. Of course, one of the prime requirements for a hand gun is that it be compact and easy to handle. It is relatively easy to imagine what embarrassment might be experienced by a man who, in defense of his person, is required to extract from his pocket a gun with a foot or so of loose chain attached. Nevertheless, although the gun no doubt does not represent the most convenient hand arm, the basic idea is sound from the mechanical viewpoint and might even have proved to be useful in a machine gun. At any rate, it serves to demonstrate that very few stones were left unturned in the search for the ideal form of the multiple chamber mechanism.

From the preceding discussion of the type of multiple chamber weapon in which the explosion chambers are separate from the barrel, it is apparent that the basic consideration is that the chambers must be preloaded and then successively moved into place behind the barrel in order to complete the gun tube. For the present, the particular mechanism used to position the chamber, whether this be a rotary action cylinder or some other device, need not be considered and primary attention will be directed to the fundamental problems which arise from the very fact that the chambers are separate from the remainder of the barrel. In the following paragraphs, these problems are discussed in some length from the general point of view, first to outline the broad principles involved in weapons having the chambers separate from the barrel and types of ammunition are not included in the scope of this publication.)

The essential elements of a gun having the explosion chambers separate from the barrel are shown schematically in fig. 4–4. The explosion chamber has its bore shaped to receive and fit the cartridge and is of sufficient strength to withstand the dis-

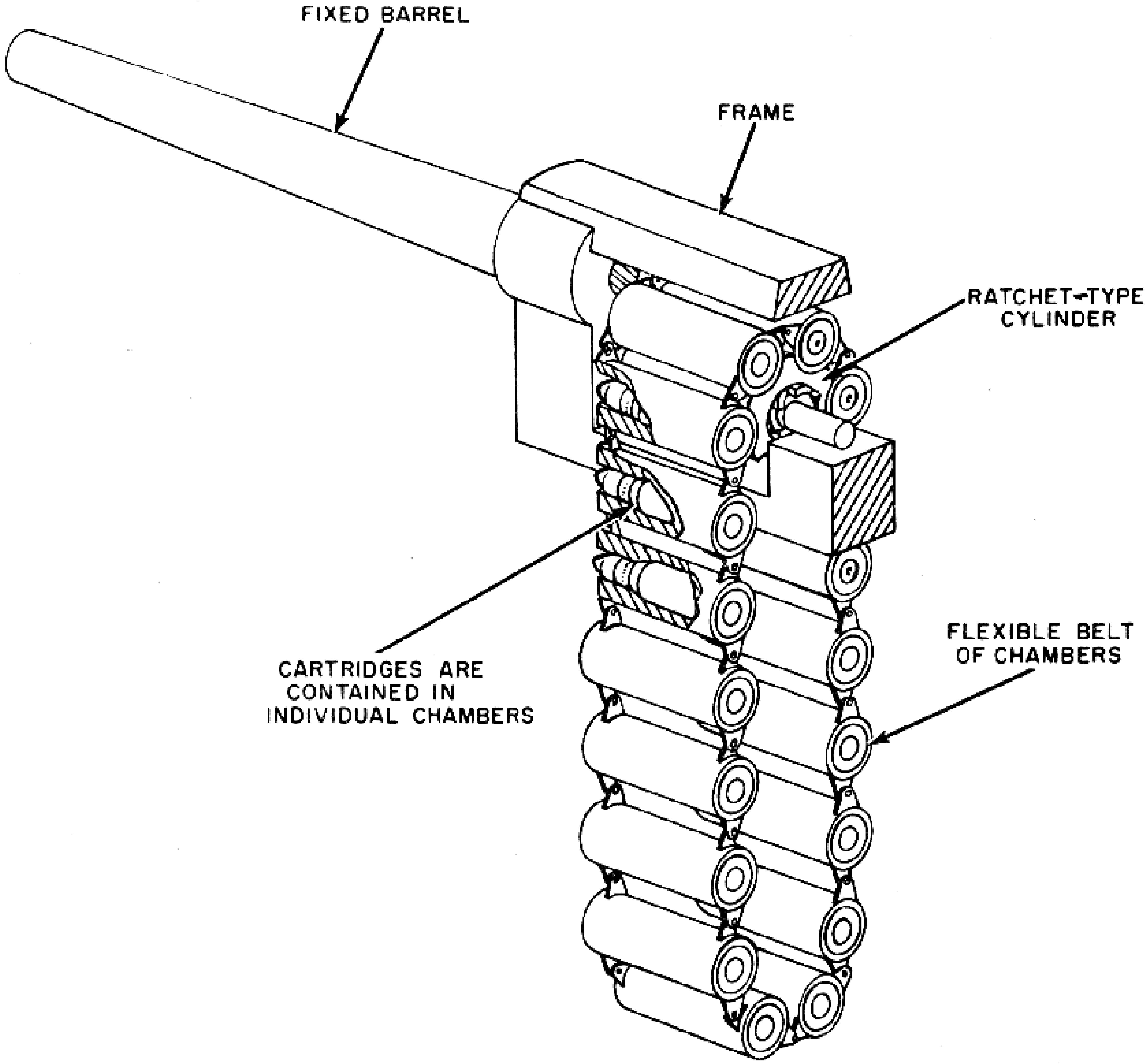


Figure 4–3. Mechanism of "Bicycle Chain" Gun.

second to prepare the way for a detailed analysis of the factors to be considered in the design of rotary chamber mechanisms in particular. (All of this discussion will be concerned primarily with guns of the full automatic type which employ conventional cartridge-type ammunition. Power-driven machine guns and guns which employ other

ruptive forces produced by the explosion of the propellant charge. The barrel of the gun is fixed to a supporting frame which is fashioned to permit the explosion chamber to be held behind the barrel so as to form a prolongation of the bore. Of course, in an actual gun, a suitable mechanism must be provided to move the explosion chambers successively into place behind the barrel and some form of firing device is necessary to ignite the propellant charge of the cartridge. It is important to note that since the explosion chambers must move with respect to the barrel, some operating clearance, or chamber gap, between the rear of the barrel and the front of the chamber is unavoidable.

From a careful consideration of the basic elements shown in fig. 4–4, all of the fundamental problems which arise in the design of machine guns employing separate chambers and barrel will be apparent. First, there is the problem of positioning the explosion chambers. In order for the weapon to be safe and to function properly, the explosion chamber must be placed directly in line with the bore of the barrel and must be held rigidly in this position when the gun is fired. It can be seen that if the chamber its position behind the barrel and thus may result in jamming the action of the gun. If the clearance between the base of the cartridge and the rear face of the frame is too great, another difficulty occurs. When the propellant charge is ignited, the forces produced by the explosion act in all directions on the interior of cartridge case and these forces cause the cartridge case both to be expanded and driven to the rear. The progress of the explosion is so rapid that the expanding walls of the cartridge case are pressed tightly against the supporting walls of the chamber before the case has moved very far to the rear. The pressure of the contact between the thin metal walls of the cartridge case and the walls of the chamber is so tremendous that the resulting frictional forces cause the forward portion of the case to seize firmly to the chamber. If at this point,

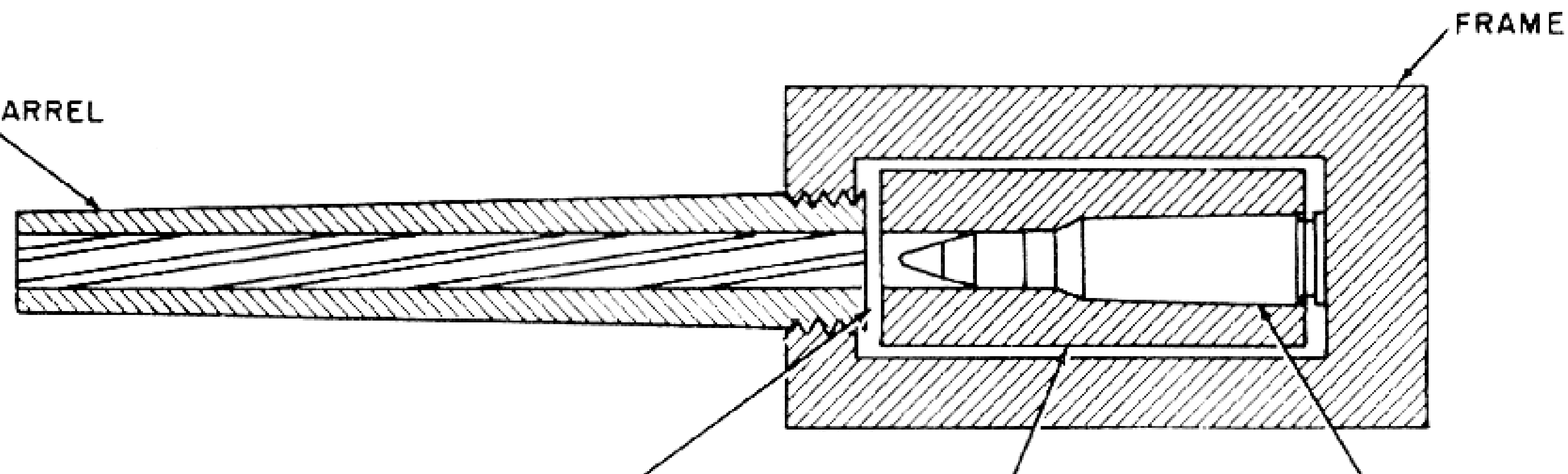


Figure 4–4. Elementary Form of Gun Having Chamber Separate From Barrel.

opening and the barrel bore are not in good alignment at the instant the projectile passes from the chamber into the barrel, the projectile will encounter an interference in its movement which could cause serious damage to the gun or projectile or at least cause shocks which would upset the accuracy of fire. (This type of defect is what causes the "shaving of lead" so commonly experienced in old or badly constructed revolver hand guns.)

In addition to accurate alignment between the chamber opening and barrel bore, it is also important for the explosion chamber to be positioned accurately so that the correct spacing exists between the rear face of the frame and the base of the cartridge when the cartridge is fully seated in the chamber. If the base of the cartridge projects too far out of the rear of the explosion chamber, it may interfere with the movement of the chamber into

the base of the cartridge case is not rigidly supported by the frame of the weapon, the high pressure within the case continues to act against the base and creates tension on the case walls. Since the chamber pressure may be 50,000 pounds per square inch or more and since this pressure acts on a considerable area at the base of the case, the force exerted on the base is so tremendous that any resistance to stretching offered by the tensile strength of the thin walls of the case is entirely negligible. Therefore, the case material stretches quite readily and the base continues to move to the rear while the thin forward portion of the case remains stuck to the chamber wall. If the space between the base of the cartridge case and the supporting surface of the frame was originally too great, the case will stretch beyond the ultimate limit of the strength of the material from which it is made and will be torn in two. When this occurs, the gun will become inoperative because the device used for extraction can not remove the portion of the cartridge case which remains stuck in the chamber. Malfunctions due to cartridge case separation can only be avoided by making certain that the space between the base of the case and the supporting surface of the frame can never be great enough to permit the case material to be stretched beyond the point at which it will fail by tearing. The clearance must be set up accurately by correctly establishing the cartridge head space when the weapon is constructed and by insuring that the explosion chamber is mounted rigidly enough so that it can not move forward under the force produced by the propellant explosion. Also, it is evident that the frame itself must be made sufficiently rugged so that it will not be subjected to an excessive amount of elastic deformation when a round is fired. The reason for this requirement is that a lightly constructed frame, although it may be strong enough to absorb the explosion pressures without permanent deformation or failure by fracture, may be stretched elastically by a sufficient amount to result in a case separation. (Stretching of the frame under pressure from the base of the round has the same effect as excessive head space.)

To sum up the points made in the preceding paragraphs concerning the positioning of the explosion chamber, the chamber must be held accurately in alignment with the barrel bore at the instant of firing so that neither the projectile nor the gun will be damaged when the projectile passes from the chamber to the bore. Also, the body of the explosion chamber must be carefully adjusted to provide the correct cartridge head space and must be held in a strong and rigid mounting so that the stresses due to firing will not move the chamber body forward or stretch the frame elastically through a great enough deflection to result in excessive head space.

The next major factor for consideration is the point at which the barrel bore and the opening at the front of the chamber are contiguous. This particular factor is very important and gives great difficulty in the design of any high-powered gun in which the explosion chamber is a separate physical part from the remainder of the barrel. If the explosion chambers are simply moved laterally in order to position them successively in alignment with the barrel, it is evident that to permit the movement to occur there must be some operating clearance. Because of the necessity for this clearance, the barrel tube formed by the fixed barrel with the addition of the explosion chamber will not be a continuous tube but will be broken by a chamber gap as is indicated in fig. 4–4. Of course, the gap shown in fig. 4–4 is greatly exaggerated but even in a well-fitted revolver there will be some considerable space between the chamber and barrel. Even if the gap seems to be very narrow, it must be realized that it runs around the entire circumference of the gun bore and therefore really represents a leakage opening of significant size. In some weapons (for example, a well-built revolver hand gun), the presence of the chamber gap, although not to be desired, does not give an excessive amount of trouble. In such a gun, as in all hand guns, the chamber pressures are relatively very low and the escape of propellant gases at the chamber gap is not great enough to cause any damage or to result in a serious drop in the muzzle velocity of the projectile which is itself relatively low. However, the condition in a high-powered high-velocity gun is entirely different. In such guns, the variation of the chamber pressure during the passage of the projectile through the bore is such that the projectile will pass the chamber gap at practically the same instant that the chamber pressure reaches its peak. This peak pressure may be well over 50,000 pounds per square inch and at such a pressure the presence of even a fairly small chamber gap would result in a terrific flash at the juncture of the chamber and barrel and would cause the loss of a considerable amount of the propellant gases with a consequent drop in the muzzle velocity of the projectile. In addition, since the chamber gap is so close to the chamber space, the gases which escape through the gap would be extremely hot and would be moving at a very high velocity. Under such conditions, the erosive action of the escaping gas would be so severe that the weapon would quickly be rendered useless.

The critical nature of the problem of gas leakage at the chamber gap naturally has spurred inventors and designers of guns employing separate chambers and barrel to produce methods and devices whereby this problem could be rendered less serious or even eliminated entirely. Even in the day of the first revolving flintlock guns, attempts were made to provide gas-tight joints between the barrel and chambers by camming the cylinder forward before firing. To aid the sealing action, the chambers were counter-bored at the forward end so as to mate tightly with the rear end of the barrel. This device has been used in a considerable number of rotary chamber mechanism designs, even up to fairly recent times. Another method which has been used with some success is to arrange the mechanism so that, when the gun is ready for firing, the forward end of the cartridge case covers the chamber gap and enters the barrel for a short distance. Since the chamber gap is relatively very narrow, the cartridge case material is supported well enough to withstand the peak chamber pressure and is not blown through the gap. Although this method is highly effective in producing a sealing action, the fact that a portion of the round projects into the fixed barrel makes it necessary to cam the explosion chambers or the round itself back and forth in order to permit the chambers to be moved successively into position for firing. Still another method, which has been employed almost universally in modern high-powered weapons, is to use a gas piston seal or obturating sleeve at the forward end of the chamber opening. The piston is in the form of a hollow cylinder to permit the projectile to pass through its center and it is free to slide forward in the chamber opening. Piston rings are provided so that no gas can leak around the outer surface of the piston. After the projectile passes through the center of the piston, the high pressure powder gases behind the projectile act on the rear face of the piston and immediately drive the piston forward. Before the projectile reaches the chamber gap, the action of the high chamber pressure thrusts the piston against the rear face of the barrel with tremendous force, thus spanning the cylinder gap and creating a gas-tight seal.

Another point of interest which affects the problems encountered in sealing the chamber gap is the length of the moving part containing the explosion chamber. Fig. 4–5 is a graph which shows how the chamber pressure varies with the position of the projectile in the bore of a typical high-powered aircraft cannon of 20-mm caliber. Note that at the instant the projectile has moved only approximately four inches (0.33 foot) down the bore, the chamber pressure has reached its maximum value of 45,000 pounds per square inch. When the projectile has moved approximately eight inches down the bore, the pressure is considerably lower having decreased to about 38,000 pounds per square inch. At a propectile bore travel of 12 inches the pressure is still lower (approximately 30,000 pounds per square inch) and by the time the projectile has moved 18 inches, the pressure has dropped to approximately 21,000 pounds per square inch or less than half of the peak chamber pressure. From the preceding numerical values, it is evident from the standpoint of sealing alone that great benefit can be attained by intentionally making the moving part containing the explosion chamber as long as is practical so that the projectile must travel a considerable distance before passing across the chamber gap. By this apparently simple expedient, the gas pressure exerted at the gap can be reduced substantially with a corresponding reduction in the difficulty of obtaining adequate sealing. However, although each additional inch of bore travel makes sealing proportionately easier, the weight problem places a definite limit on the permissible length of the moving part containing the chambers. For example in a six-chambered revolver, increasing the length of the cylinder by one inch would mean that the weight of the gun would be increased by an amount at least as great as the weight of a piece of barrel five or six inches long. Since weight is such an important factor in most weapons, it is more than probable that any advantage gained by lengthening the cylinder would be insignificant when compared with the disadvantages of greatly increased weight and bulk.

In a thorough consideration of the elementary guns shown in fig. 4–4, there are several other points which require some mention. A gun having the basic parts shown in the figure could certainly be made to fire one round of ammunition but, in order for the gun to be capable of repetitive fire and to function as a full automatic machine gun, the factors involved in moving the explosion chambers and in loading and clearing them must be given careful consideration. The first of these factors is the source of power used for operating the gun mechanism. As has been pointed out previously in this publication, a gun placed in the category of full automatic weapons must derive the power required for its operation from the energies released by the explosion of the propellant charge of the ammunition. Actually, few modern high-rate-of-fire machine guns of the type designated as aircraft cannon derive all of their operating energy exclusively from the propellant explosion. In many cases, various auxiliary sources of power are employed. Although the major task of imparting the required motions to the heavier operating parts is accomplished through the medium of energy derived from the powder gases, some of the functions of the weapon are performed by mechanisms which receive power from external sources. For example, in many modern weapons, the feed mechanism which keeps the weapon supplied with ammunition may be driven either by an electric motor or by a pneumatic device operating off a source of high pressure air. In most cases, this is done in order to simplify the gun mechanism by eliminating the devices which would be required to obtain the energy for feeding from the gun itself. Sometimes the use of external energy for feeding is resorted to so that excessive operating energy will not be extracted from the moving parts of the gun, thereby permitting these parts to move at maximum velocity in order that the gun may attain the highest possible rate of fire. In most modern aircraft cannon, the energy for firing the propellant charge usually is not obtained by the impact on the primer of a sliding part of the gun mechanism but firing is accomplished by the application of a voltage obtained from an external source of electrical energy. The charging of the weapon is another function which is performed by means of energy provided from outside of the gun itself. The charging device may be powered manually or pneumatically and in the type of device called a percussion charger or cartridge charger, the power for charging is supplied by the discharge of a blank cartridge of small caliber.

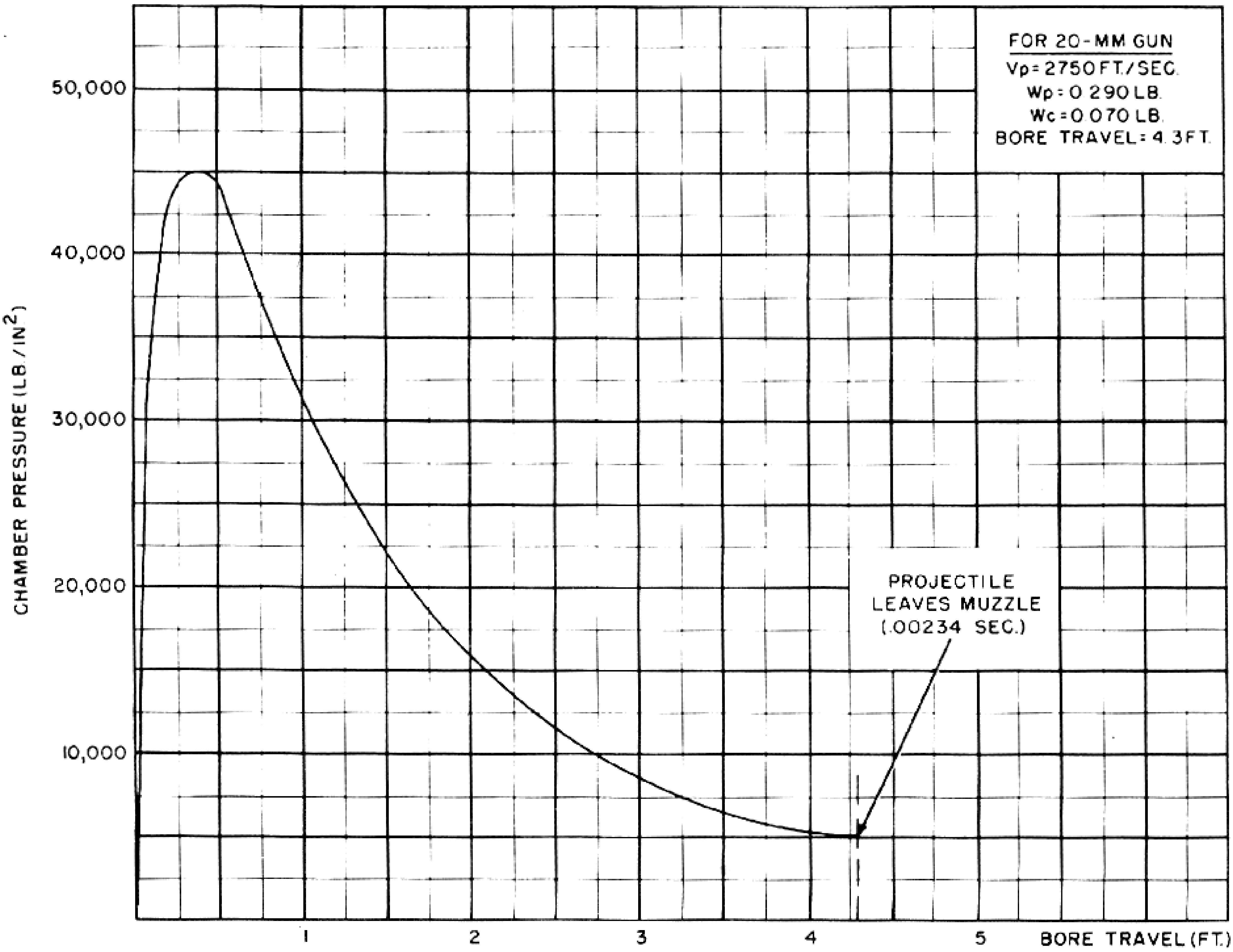


Figure 4-5. Graph Showing Variation of Chamber Pressure With Bore Travel of Projectile.

In spite of the fact that auxiliary power may be employed in the operation of a machine gun, the gun is considered to be in the full automatic classification as long as the major functioning assembly of the gun mechanism itself receives its power from the explosion of the propellant charge. In the case of a weapon having the chamber separate from the remainder of the barrel as shown in fig. 4–4, this means that the propellant explosion must at least be used to impart the required motion to the component which contains the movable explosion chambers, whether these chambers are all bored into a cylinder or are separate units of the various forms previously mentioned in this chapter. If this minimum requirement is not met, the weapon must be classed as a power-driven machine gun of a type which is not within the scope of this publication.

The various systems which can be used to derive and handle the energy provided by the explosion of the propellant charge are the same for multiple chamber weapons as for any other type of full automatic machine gun. All of the successful full automatic multiple chamber machine guns developed up to the present date have employed either recoil actuation or gas actuation since these operating systems seem to be particularly well adapted to the requirements of this type of weapon. By its very nature of involving a linear movement to the rear, the blowback system does not appear to be as well suited for application in a conventional rotary action cannon or other multiple chamber weapon. However, there have been special forms of rotary action weapons in which a linear motion of the cartridge case to the rear after firing is used to great advantage. This motion is utilized not only for the purpose of deriving operating energy, but also to provide for the actuation of a very effective system for sealing the chamber gap between the rear end of the fixed portion of the barrel and the front end of the explosion chamber.

Thus, it is evident that, in theory at least, all of the available systems for deriving operating energy from the explosion of the propellant charge can be used with good effect in multiple chamber weapons as well as in weapons of the reciprocating type. For the present, the particular manner in which these systems are employed and the form of the mechanisms used to apply the derived energy are not important. The main point is that in order to produce a full automatic weapon, one system or a combination of systems must be used and, as a minimum, must be employed to impart the required motions to the explosion chambers.

After a means has been provided for imparting the required motions to the explosion chambers, the next important factor for consideration in producing a full automatic multiple chamber gun is the problem of reloading during firing. In the Ager "Coffee Mill" gun previously mentioned in this chapter, the solution to this problem was achieved by the simple expedient of providing so many chambers that reloading during a burst of any reasonable duration was unnecessary. That is, the individual explosion chambers were loaded in advance with powder and ball and each was provided with a percussion cap. The chambers were than handled like modern-day cartridges except that it was only necessary for the gun mechanism to place them successively behind the barrel, fire them, and then eject them. From many standpoints, this is an excellent and convenient system except for the very obvious disadvantage as regards the weight and bulk of the individual explosion chambers. In the era of cap-and-ball weapons, the time required for reloading chambers apparently provided ample justification for tolerating excessive weight, as is evidenced by the ponderous construction of all of the multiple firing weapons of those days which had a capacity for any considerable number of shots. However, it is readily apparent that for any modern high-rate-of-fire, high-powered machine cannon, the idea of using an unlimited number of individual explosion chambers is entirely impractical. For a heavy caliber machine cannon firing over 1000 rounds per minute, the mere weight and bulk of the required number of chambers would certainly be prohibitive even if the difficulty of providing a sufficient belt pull to move the chambers rapidly through the weapon is ignored. It is hardly surprising, therefore, that weapons employing a large number of separate explosion chambers fell into almost complete disuse immediately after the introduction of cartridge ammunition. The use of metallic cartridge ammunition, in which the projectile, powder charge, and percussion primer are all neatly and ruggedly assembled into one compact, easily handled unit, made it possible to load an entire round into a chamber with great rapidity and also put breech loading on a practical basis because of the sealing function performed by the cartridge case.

Although these advantages eliminated many of the shortcomings previously experienced with powder and ball, the use of metallic cartridge ammunition introduced the new problem of removing the spent cartridge case from the chamber after firing to permit the loading of a fresh cartridge. In repeating and semi-automatic weapons and in most machine guns, the problems of loading fresh cartridges into the gun and of extracting the fired cases from the chamber do not present any very great difficulties if the loading and extraction functions are given proper consideration in the design of the weapon. However, in guns having very high rates of fire and particularly in guns of the multiple chamber type, a special loading and extraction problem arises. It is not the mere process of inserting the loaded cartridge or of removing the cartridge case from the chamber that produces the problem but it is the small time that is allowable for the process to occur. For example, consider an automatic weapon which fires at a cyclic rate of 1500 rounds per minute. With a firing rate of this magnitude, only 0.04 second is allowable for the entire chain of events which must occur during each firing cycle. The particular series of events which is necessary for the automatic functioning of the weapon will of course depend on the particular design characteristics of the mechanism but it can be seen that with such an extremely brief time interval available, the problems attendant upon any mechanical function can be greatly multiplied. For this reason, the means which are used to load the chambers of a multiple chamber weapon and to extract the fired cartridge cases must be given very careful consideration in the design. The consideration must cover, not only the mechanisms which accomplish the functions and the power sources used to drive the mechanisms, but must also take into account the timing of the functions, so that they can occur with no interference from the remainder of the moving parts of the weapon.

Another important factor in the design of a multiple chamber weapon is the control and dissipation of the recoil forces resulting from the firing of the cartridges. If the weapon is recoil-operated, a considerable portion of the recoil energy may be put to use in providing power for driving the gun mechanisms, but in a gas-operated gun, the forces of recoil do not contribute to the action and must be absorbed entirely by the mounting of the weapon. In a multiple chamber gun, the principles relating to recoil are the same as for any other gun. The ignition of the propellant charge produces an explosive combustion which causes the rapid generation of an extremely high gas pressure in the chamber of the gun. The expansion of these high-pressure gases drives the projectile forward through the bore of the gun and since the gases move down the bore after the projectile, the center of mass of the gases also moves forward. (Because of the fact that the gases are not moving forward as a single body but rather are expanding in the bore, the forward velocity of the center of mass of the gases is only approximately one-half of the projectile velocity.) As long as the projectile is still moving down the gun bore, the same pressure which acts to produce the forward motion of the projectile and powder gases also acts at the chamber end of the gun to produce an equal and opposite reaction which tends to drive the entire gun rearward. The force resulting from this reaction is called the "recoil force" and its magnitude at any instant depends on the chamber pressure which exists at that same instant.

After the projectile leaves the muzzle of the gun, the recoil force does not cease to act immediately but continues in existence as the reaction to the forward movement of the powder gases as they pass out of the gun muzzle. The recoil force does not actually fall to zero until the so-called residual powder gas pressure in the barrel has become equal to the atmospheric pressure. (In a representative 20-mm gun this does not occur until approximately eight milliseconds after the ignition of the propellant charge.)

The combined effect of the recoil force which exists before the projectile leaves the muzzle and the recoil force produced by the residual pressure is to impart a rearward velocity to the gun and to create a kinetic energy in the gun mass. The magnitudes of the velocity and of the kinetic energy depend on the power of the cartridge, the mass of the parts subjected to the recoil action, and the manner in which the gun is mounted. In an analysis of the conditions of recoil, it is usually desirable to consider first what velocity and energy would be obtained if the gun is assumed to be mounted so that it is not subjected to retardation from friction or any other restraint. This hypothetical condition is referred to as the condition of "free recoil". With no restraint of any kind imposed upon the gun, the impulse of the recoil force (acting over the entire time of action of the powder gas pressure, including the duration of the residual pressure) will impart a rearward momentum to the gun which is equal to the total effective forward momentum imparted to the projectile and powder gases. The velocity corresponding to this value of momentum will depend on the mass of the recoiling parts; the smaller the mass, the higher will be the velocity. When the residual pressure has reached zero, the gun will have achieved its maximum rearward velocity. Since it is assumed that there are no external restraining forces in the condition of free recoil, the gun will continue to move at this velocity indefinitely.

Although the assumption of free recoil is useful for analytical purposes, this condition is of course never encountered in practice. Under actual conditions of use, any gun must have its recoil motion controlled and limited so that the recoil travel is held to some reasonable value. This must be accomplished by the application of restraining forces which produce the condition known as "retarded recoil". With small-caliber, low-powered weapons it is often possible to hold the parts subjected to the recoil forces in a rigid mounting in such a manner that the forces are directly absorbed by the mounting with no appreciable rearward movement of the gun. However such a mounting would be entirely out of the question in the case of a larger caliber, high-powered weapon. For example, the maximum recoil force in a 20-mm gun can amount to from 20,000 to 30,000 pounds or more and with forces of this magnitude it is essential to provide some sort of shock-absorbing mount. Such a mount must permit some motion of the gun during and after the action of the recoil force so that the momentum of the recoiling parts can be cancelled through the application of a relatively small retarding force which acts on the gun through a considerable interval of time and distance, thus gradually reducing the recoil velocity to zero.

The amount of energy which must be absorbed by the mount and the magnitude of the required restraining force depends on several factors. Assuming a cartridge of given power, the most important single factor is the weight of the recoiling parts. The transfer of kinetic energy to the gun occurs mainly during the early stages of the propellant explosion when the recoil forces are extremely high. Since whatever restraining forces are applied to the recoiling parts must of necessity be relatively small (in the order of a few thousand pounds at the most) and the maximum recoil forces will be in excess of 20,000 or 30,000 pounds (for a 20-mm gun), it can be seen that for the short period of action of the recoil forces, the restraining forces can have but little effect. Accordingly, for the duration of the high powder gas pressure, the recoil forces are almost unopposed in imparting velocity to the gun. Therefore, during the greater part of the action of the powder gas pressure, the only significant factor limiting the velocity produced by the impulse of the propellant explosion is the mass of the recoiling parts. With this in mind, it is evident that a heavy gun will recoil with low velocity and low kinetic energy while a light gun will recoil more violently and with greater energy.

Having a given cartridge and having recoiling parts of some definite mass, these factors more or less determine the general order of magnitude of the recoil energy and velocity. The remaining point to consider is how the energy and velocity of recoil can be dissipated by the recoil system of the weapon. Since the problem is concerned with the absorption of the kinetic energy of recoil, the answer lies in methods whereby this energy can be expended in the performance of mechanical work. If the gun is to be recoil operated, a considerable portion of the energy can be converted to useful purposes in operating the gun mechanism. Making use of a portion of the energy in this manner means that the task of the recoil system of the gun mounting will be lightened and the problem of disposing of excess recoil energy will be minimized. Indeed it is often the case that the energy requirements of a recoil-operated gun mechanism are so great that it becomes necessary to augment the recoil energy through the use of muzzle boosters and other such devices in order to obtain effective operation. In such situations, the problem is often how to find special means of increasing recoil rather than to minimize it.

On the other hand, in a weapon such as a gas-operated machine gun, the recoil energy is not put to any useful purpose and is only a source of inconvenience. Since the energy is not dissipated in operating the gun mechanism, a recoil system must be provided in the gun mounting to absorb and dispose of it by causing it to be expended in doing other mechanical work. This means simply that the recoiling parts must be restrained by a force that acts over a distance, which constitutes the definition of work. Accordingly, a high-powered gas-operated machine gun is mounted in a slide so that it can move to the rear under the action of the recoil forces. The mounting is equipped with some sort of restraining device which disposes of the greater part of the recoil energy and also is provided with some means for storing a portion of the energy to be utilized for returning the gun to the battery position. In a medium-caliber aircraft machine cannon, the recoil energy is usually dissipated as heat in a nest of ring springs or in a hydraulic buffer while the portion of the recoil energy used for driving the gun back to battery is stored in a spring. Larger guns are usually provided with a hydraulic recoil brake and a pneumatic device called a recuperator, both of which absorb the recoil energy. Most of the energy is expended in the brake in the form of heat and the relatively small amount of energy stored in the recuperator is utilized for returning the gun to the battery position.

The design characteristics of the recoil system of a machine gun depend to a large extent on the requirements of the installation and also are affected strongly by the desired rate of fire. From the standpoint of simplifying design problems, it is of course desirable to subject the recoil system to the smallest possible load; that is, to employ the minimum restraining force. However, since the amount of work which must be done in order to expend the recoil energy is more or less fixed and predetermined, it follows that the use of a smaller restraining force will require a longer recoil travel. In most cases, the requirements of the installation do not permit any great amount of recoil travel and even if a long travel were allowable, an excessive movement would create other serious design difficulties. Faced with these conflicting desires for a small restraining force and a short recoil travel, the designer is usually forced to favor a short travel and to accept the difficulties which arise from a high restraining force and the resulting high trunnion reaction of the mount.

Another very pressing reason why it is necessary to maintain a short recoil stroke in high-rate-of-fire guns can be demonstrated by a simple analysis of the time available for the completion of the recoil and counter-recoil travels of a sample machine gun. As the sample, consider a gun having a rate of fire of approximately 1500 rounds per minute and assume that the weight of the gun and the power of the cartridge are such that the initial velocity of the recoiling parts is approximately 10 feet per second. (This is a reasonable value based on an average of conventional multiple-chamber guns in the 20-mm to 30-mm class.) Now, at a rate of fire of 1500 rounds per minute, the recoiling parts must complete their recoil travel and their counter-recoil travel back to the firing position within 1500/50=.04 second. This is the maximum time allowable for these motions because the operation of the gun will not be stable if the firing cycle takes less time than the combined time for recoil and counter-recoil. Since the recoiling parts must reverse their motion at each end of their travel and since this reversal can not occur instantly, it is safe to estimate that their average velocity of movement for a complete cycle of operation will be at the most approximately half their maximum velocity or about 5 feet per second. Multiplying this average velocity by the available time of 0.04 second shows that the total distance travelled can be only $5\times12\times.04=2.4$ inches. This is the combined recoil and counter-recoil movement and therefore the recoil movement can be only half of this distance or 1.2 inches. From the foregoing analytical estimates it can be seen that the recoil travel of a high-rate-of-fire gun is definitely limited to a rather short distance simply because the average recoil velocity is necessarily fairly low and the time available for the complete recoil cycle is so short. As a matter of fact, the estimated value of 1.2 inches is somewhat on the high side. In existing 20-mm multiple-chamber aircraft cannon, the recoil distance is more in the order of from 0.25 to 0.50-inch. This shorter travel is necessary because of the fact that a considerable portion of the recoil energy is dissipated and therefore the average velocity of counter-recoil is much lower than the average velocity of recoil.

With such short recoil travels, the average restraining force imposed on the gun by the recoil system must be fairly high. Assume for instance that the recoiling parts of the gun used as an example weigh approximately 100 pounds. The stipulated initial recoil velocity of about 10 feet per second results in a kinetic energy computed as follows:

$$\text{K.E.}=\frac{1}{2}MV^2=\frac{1}{2}\frac{100}{32.2}10^2$$

$$=155.3 \text{ (foot-pounds)}$$

If this entire amount of energy is to be absorbed over a travel of only 0.25 inch, the average force exerted over that distance must be:

$$F=\frac{K.E.}{D}=\frac{155.3}{.25}\times 12=7450 \text{ (pounds)}$$

In an actual gun, a portion of this force would be provided by a spring which stores sufficient energy to return the gun to battery and the remainder of the restraining force would be supplied by a device like a ring spring buffer which converts a large percentage of the energy it receives into heat. In any case, the entire restraining force of 7450 pounds is transmitted to the gun mounting and appears as the trunnion reaction of the gun. This approximate analysis shows that the conditions of recoil in a high-rate-of-fire multiple-chamber weapon are such that a short recoil travel and a high trunnion reaction can not be avoided. For this reason, the design problems associated with the recoil system of such a gun are rather critical and must be treated with great care.

The last point for consideration in an analysis of an elementary multiple-chamber weapon is the question of timing. Although the timing involved in a multiple-chamber weapon is from the general standpoint much less of a problem than in a reciprocating weapon, there are certain factors which require some attention. Of particular interest are the limitations which affect the rate of fire theoretically attainable. These limitations can be examined best by considering the sequence of events which must occur during a complete automatic firing cycle. To start, assume that a loaded chamber has been positioned accurately in alignment with the barrel bore and that the gun is fully prepared for firing. From the instant that the firing device is actuated to ignite the primer until the instant that the chamber pressure has decreased to a low enough value to permit the chamber to be moved may require an elapsed time somewhere in the order of 10 milliseconds. After the chamber is fired, it must be moved out from behind the barrel while the next loaded chamber is moved into firing position. The time required for this movement is not fixed definitely but depends entirely on the design of the gun mechanism and the amount of power utilized to operate it. It should be realized that the mechanism used to move the chambers must of necessity be rather heavy and have a correspondingly great inertia. In order to move a loaded chamber into firing position, this heavy mechanism must first be unlocked, then accelerated to some high velocity, and finally decelerated until it comes to rest at the firing position, where it is again locked. Even if the explosion chambers are spaced very close together, the distance through which the mechanism must move is considerable and the separate actions of unlocking, acceleration, deceleration, and locking each require some small but significant amount of time. At a very high rate of fire, the time allowable for an entire firing cycle is extremely short and the time which can be permitted for any of the events during the cycle is much shorter. This means that the mechanism must be subjected to tremendous accelerations in order for it to accomplish its required movements within the allowable time.

To demonstrate the order of the linear accelerations which can be expected in a high-rate-of-fire multiple-chamber gun, consider a 20-mm weapon which fires at the rate of 1500 rounds per minute. At this rate of fire, the time allowable for each complete firing cycle is only 0.04 second. Of this total time, an interval of about 0.010 second is required for ignition of the propellant, the combustion of the powder charge, and the decrease of the chamber pressure to a level low enough to permit the chambers to be moved. Accordingly the time remaining during which the chambers can be moved is now only 0.03 second. Of this time, it is reasonable to allow a total interval of at least five milliseconds for the action of the device which unlocks the mechanism before the movement of the chambers starts and again locks the mechanism at the completion of the movement. Now the time remaining for the movement to occur is only 0.025 second. During this extremely brief interval, the heavy gun mechanism must be accelerated to some fairly high velocity and then decelerated until it comes to rest. For the purpose of approximate analysis, let it be assumed that the linear distance separating the chambers is approximately two inches. Let it be assumed further that the operating cams are formed to produce the entire movement in equal periods of uniform acceleration and uniform deceleration.

On the basis of the foregoing data, the time available for the acceleration phase of the movement is half of 0.025 second or 0.0125 second. During this interval, the mechanism must move half of the total two-inch travel, or one inch (.0833 foot).

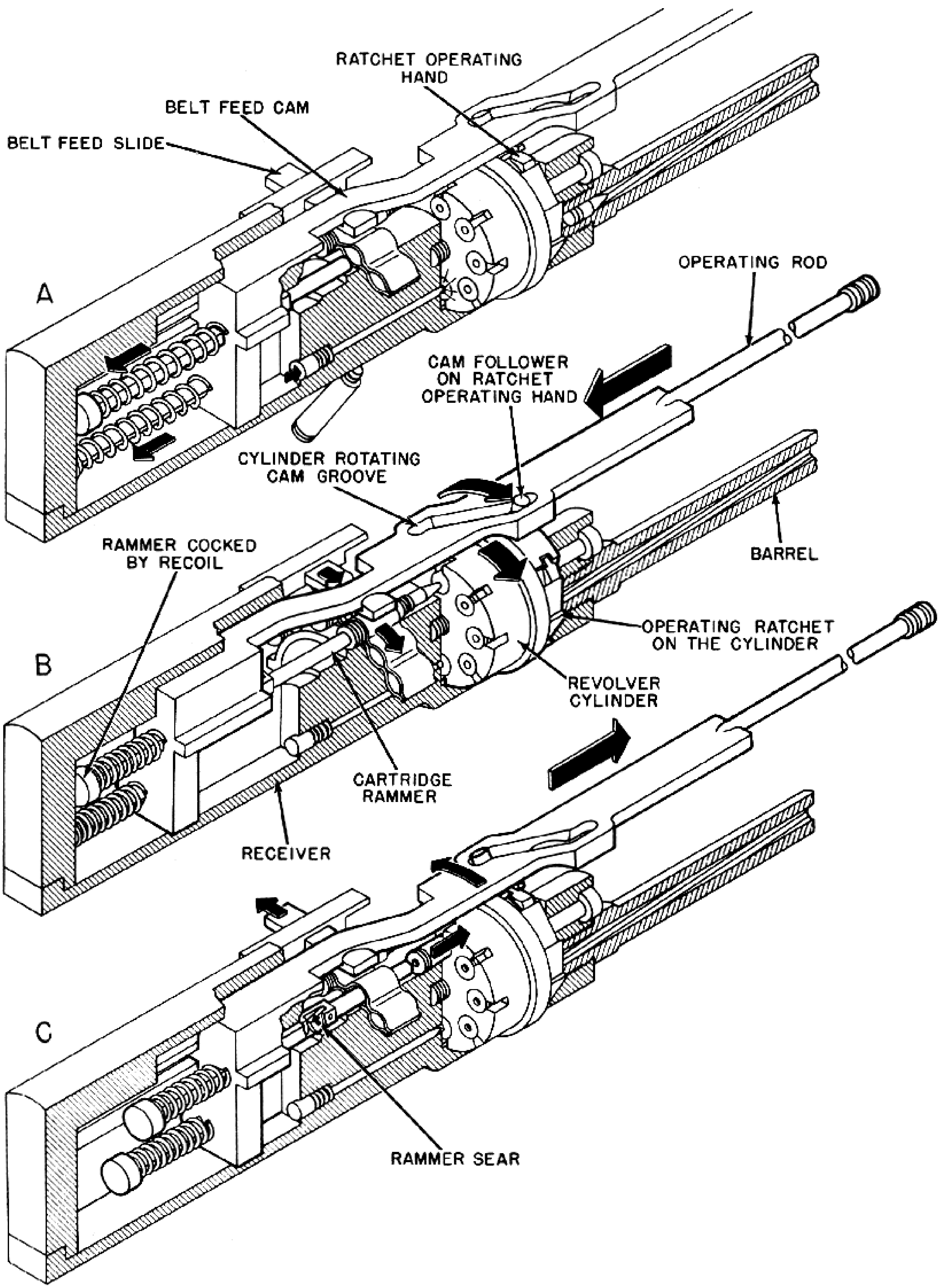


Figure 4–6. Simplified Schematic of a Typical Gas-Operated Rotary Action Cannon.

The required uniform acceleration can be found by using the standard relation between distance, acceleration, and time as follows:

$$S=\frac{1}{2}\,at\,2$$

Solving for "a" gives:

$$a=\frac{2S}{t^2}$$

Making the necessary substitutions and evaluating "a":

$$a=\frac{2\times.0833}{(.0125)^2}$$

$$a=1066\ (\text{ft./sec.}^2)$$

Since it is assumed that the deceleration period is the same as the acceleration period, the value of the deceleration will also be 1066 feet per second per second. This is a high value and is equivalent to approximately 33 times the normal acceleration of gravity. An idea of the magnitude of the force required to produce this acceleration can be had by assuming that the equivalent mass subjected to the acceleration is 50 pounds in weight.

$$F=Ma=\frac{W}{g}\,a$$

$$=\frac{50}{32.2}\times1066$$

$$=1655\ (\text{pounds})$$

Of course, the preceding calculations are based on highly simplified conditions but they do give a fair indication of the general order of the time intervals, accelerations, and forces which can be expected in a high-rate-of-fire multiple chamber gun. Although the analysis is primarily concerned with the movement of the chambers, it can be seen readily that the entire design problem will be similarly affected by the extremely short time available for each weapon function. In order to obtain smooth high-speed operation, every function must be timed with great precision. Ignition of the propellant charge must occur at exactly the correct instant, unlocking must be neither premature nor late, the application of the force which moves the chambers must take place at the instant the unlocking action is completed, and locking must occur just as the loaded chamber reaches the firing position. Even a slight delay at any of these critical points can seriously cut into the time available for the remainder of the operation. A similar necessity for precise timing and extremely rapid action exists in all phases of the operation of a high-rate-of-fire multiple-chamber gun. The loading operation, the ejection of spent cartridge cases, and the conditions of recoil are all subject to the same stringent requirements. In fact, it has been facetiously remarked that the designer of these weapons is an unfortunate individual who is continuously snatching at the coat-tails of the fleeting and infinitesimal instant.

PRINCIPLES OF ROTARY CHAMBER MECHANISMS

The preceding analysis deals primarily with the general operating and design characteristics which are inherent in any full automatic multiple-chamber weapon in which the chambers are separate from the barrel. In the analysis, consideration was given to those basic features which are significant regardless of what particular type of mechanism is employed. However, there is one type of multiple-chamber mechanism which has had a very prominent place in modern gun developments. This mechanism, the rotary action, will be treated in detail in the following paragraphs of this chapter.

The elements of a rotary action are shown schematically in fig. 4–6A. The essential parts are the cylinder (sometimes called the cartridge drum), the barrel, and the frame. A complete revolver cannon must also have some means for utilizing the recoil or a gas piston for automatically indexing the cylinder and must be provided with a recoil system, firing device, and feeder. The particular mechanism shown in fig. 4–6 is taken from the Clarke patent previously mentioned in this chapter. This is a gas-operated weapon and the illustration shows all of the principal parts except for the recoil system and extractor. Fig. 4–6A shows the condition of the mechanism immediately after firing and fig. 4–6B shows the condition with the piston slide driven fully to the rear.

Cycle of Operation

The operating cycle of a typical rotary action weapon is relatively very simple and occurs as follows:

The cycle starts with a cartridge in the chamber at the 180-degree position. This particular weapon has eight chambers bored in the cylinder and at the start of the cycle, the chambers at the zero degree, 45-degree, 90-degree, and 135-degree position are loaded. The chambers on the left side of the cylinder are unloaded. When the cartridge in the chamber aligned with the barrel is fired, the pressure of the powder gases drives the projectile and the gases forward through the bore and at the same time drives the entire gun to the rear in recoil. (The force of the recoil is absorbed by the recoil system.) When the projectile passes the gas port in the barrel, the powder gases flow into the gas cylinder and exert a rearward thrust on the gas piston. The time at which this occurs will depend on the location of the port and ordinarily will be from 0.001 to 0.002 second after ignition of the propellant charge. The piston action and the cylinder rotating cam groove in the piston slide are controlled in the design so that the cylinder is not rotated immediately but rotation is delayed until the pressure in the barrel has dropped to a safe operating limit a few milliseconds after the projectile has left the muzzle. As the piston slide moves to the rear, the cam follower in the cylinder rotating cam groove is driven to the right, causing the operating hand to engage the ratchet on the cylinder and to rotate the cylinder. The hand rotates the cylinder sufficiently to bring the next loaded chamber into alignment with the barrel. The rearward movement of the piston slide also extracts the empty case from the fired chamber and causes the belt feed cam to move the belt feed slide so as to position a fresh cartridge in front of the rammer. As the piston slide moves forward again (fig. 4–6C), the rammer pushes the cartridge out of the belt into the empty chamber now at the zero-degree position. At the same time, the belt feed slide is retracted and the ratchet operating hand is moved back to pick up the next notch in the operating ratchet on the cylinder. This completes the operating cycle and the next cycle starts when the round now in the 180-degree position is fired.

Analysis of Rotary Action Features

As mentioned earlier in this chapter, the revolver is merely one special form of a multiple-chamber weapon in which the chambers are separate from the barrel. As a starting point for an analysis of the important features of the revolver cannon, it is of interest to consider why the rotating cylinder is preferred over other forms of mechanism for moving chambers successively into position. The advantage of the rotating cylinder can be appreciated most readily by comparing a revolver with a so-called "harmonica" gun. In a harmonica gun, the chambers are bored into the edge of a flat plate and to bring the chambers successively into alignment with the barrel, the plate slides laterally across the back of the gun. For a limited number of shots, this type of mechanism can function very well, but it has several obvious and very serious drawbacks. First, the flat array is wasteful of space and awkward. Also, the overall shape and balance of the weapon changes as the chamber plate slides laterally during firing. Of even greater importance is the fact that, to fire a continuous burst of any great duration, the motion of the plate must be reversed during firing. This means that immediately after the reversal, the next chamber to come into line is the same one which was fired just an instant before. A condition of this sort obviously makes it practically impossible to fire continuously at a high rate.

The rotating cylinder eliminates all of the disadvantages of the harmonica gun. The mechanism is compact and relatively easy to control. The shape of the weapon remains symmetrical throughout its operation and the balance remains practically unchanged. The most significant advantage is that the cylindrical arrangement of the chambers gives what is in effect an endless succession of chambers because after a chamber is fired, there is time available for extraction and reloading while the other chambers are being fired.

Although the revolver is very simple from the functional point of view, there are a number of problems connected with the design of revolver cannon which have defied complete solution for over 80 years. In revolver hand guns, all of which use comparatively low-powered ammunition and are fired slowly, these problems either do not exist to any serious extent or have been overcome adequately through good design and careful manufacture. Nevertheless, the operating conditions in high-powered, high-rate-of-fire revolver cannon are so radically different that expedients that have been fairly successful in hand guns fail completely when applied in aircraft cannon.

One of the most difficult problems in a revolver cannon is the leakage of high pressure powder gases which occurs at the joint between the forward end of the cylinder and the rear end of the barrel. Another serious difficulty that occurs with long bursts at high rates of fire is the tremendous amount of heat absorbed by the cylinder. Also there is a critical requirement for high precision in the cylinder indexing mechanism to insure that the chambers will be accurately aligned with the barrel bore. Finally, the fact that the projectile must have a rather considerable free run before it engages the rifling in the barrel causes an additional source of trouble. These four problems, although they may appear at first glance to be simple enough, have been proved by experience to be the stumbling blocks which have stood in the way of rotary chamber mechanism development throughout the entire history of modern machine guns. Because of the importance of these problems, they will be covered in some detail in the following paragraphs and the devices which have been so far applied for their solution will be analyzed in order to give an indication of what progress has been made and what work yet remains to be done.

First consider the problem of gas leakage. That this difficulty has existed and has been recognized for many centuries is very clear from the history of revolver weapons. In the very early days of firearms, when the tools and techniques for constructing accurately fitted mechanisms were not available, all revolving firearms that enjoyed any success were built on the "pepperbox" principle. That is, they employed a number of complete barrels rather than a single barrel with a separate revolving cylinder containing the individual chambers. Although the concept of using a separate cylinder was conceived at a very early date, it was not until the eighteenth century that the art of machining had advanced sufficiently to permit the actual construction of practical revolving cylinder weapons. It is highly significant that almost immediately after these guns appeared, there were many attempts to provide a positive means of sealing the joint between the cylinder and barrel. These attempts have continued, intermittently, from that day on.

Some of the sealing methods used in the hand guns of the past were fairly effective, but none of them have stood the test of time well enough to survive. In the modern revolver hand gun, the only precaution taken to limit leakage is the provision of a close fit between the cylinder face and the rear of the barrel. This does not by any means eliminate the leakage, even in a well-constructed weapon. In any such gun that has been fired several times, a considerable amount of powder fowling will be evident around the front of the cylinder and when the gun is fired at night, a substantial flash will be observed at the chamber gap. In this connection, it is interesting to note that a silencer, which works very effectively on a single-shot arm, is useless on a conventional revolver because the blast at the chamber gap produces just as much noise as the muzzle report. Thus it appears that, for hand guns at least, experience has proved it more practical to ignore the problem of leakage than to complicate the weapon with sealing devices. With pistol ammunition, which is all low-powered, the leakage problem is just not severe enough to warrant the use of remedial gadgets or special and expensive mechanisms.

Unfortunately, it is not possible to treat the gas leakage problem so lightly in a revolver weapon which employs high-powered ammunition. Leakage effects which are minor in pistol calibers become major problems at the extremely high chamber pressures produced by modern aircraft cannon ammunition. Earlier in this chapter, the problem of leakage was treated in general terms under the heading, "Principles of Multiple Chamber Weapons." At this point, it is appropriate to review the facts which are applicable to revolver weapons in particular and to consider in more detail the various devices used to prevent leakage in these weapons.

When a rotating mechanism weapon is fired, the pressure of the powder gases builds up with extreme rapidity and drives the projectile out of the cylinder into the barrel. It happens in most revolvers that the projectile passes into the barrel at practically the same instant that the chamber pressure reaches its peak value. In a modern aircraft cannon, this peak pressure may be in the vicinity of from 50,000 to 60,000 pounds per square inch, and as the projectile passes into the barrel, this tremendous pressure is suddenly brought to bear on the joint between the barrel and cylinder. If the joint is not tightly sealed, a very undesirable situation exists. Any considerable gap will result in a violent and destructive blast which could easily ruin the gun and anything near it. Even the presence of a small opening can cause serious difficulties because the high-pressure gases are at such an extremely high temperature and pass through the opening at such a great velocity that there will be a very rapid erosion at the joint. After a few shots, the damage caused by the erosion could be so great that the gun may fail to function or at least may become unsafe for further use. Blast and erosion are not the only difficulties which can result from ineffective sealing at the joint between the cylinder and barrel. It must be realized that the powder gas pressure continues to act at the joint for the entire time the projectile is being driven through the gun bore. If the leak is bad enough to permit the escape of any significant amount of the powder gases during this time, there may be a considerable loss in the muzzle velocity of the projectile.

There are three basic methods which have been used over the years for the purpose of sealing the joint between the cylinder and barrel to prevent the escape of the powder gases. The oldest of these methods has been used ever since the day of the flintlock and cap-and-ball revolvers and consists of a device for camming or wedging the cylinder forward before the gun is fired. This action closes the chamber gap and presses the forward face of the cylinder tightly against the rear face of the barrel. Another arrangement, which was conceived as soon as metallic cartridge ammunition came into general use, is to move the cartridge forward before the gun is fired so that the neck of the cartridge case is moved over the chamber gap and enters the barrel for a short distance. When the cartridge is fired, the neck is expanded and tightly seals the gap. The third method is that used in most of the present-day revolver cannons. This method uses a sealing piston or sleeve at the front end of each chamber opening in the cylinder. As the projectile passes through the sleeve, the force of the explosion drives the sleeve forward and holds it tightly against the rear face of the barrel until the pressure within the bore decreases to a relatively low value. Thus, during the action of the powder gas pressure, the sealing sleeve spans the chamber gap and holds the leakage to a minimum. In the following paragraphs, the three sealing methods mentioned here are analyzed to indicate the nature of the design problems involved in each of them and to explain further their functioning.

The method of camming the cylinder forward to effect the seal may take several forms. In its simplest form, the rear of the barrel and the front of the cylinder are merely machined to a smooth flat surface so that they will mate closely when the cylinder is cammed forward. Although this method has the advantage of ease of manufacture, it is not very effective because of the virtual impossibility of insuring a tight metal-to-metal contact around the entire periphery of the joint. Any slight defect of the surfaces, or the presence of particles of dirt or fouling in the joint, will adversely affect the sealing action. Even a small inaccuracy in the squareness of the mating surfaces or in the axial alignment of the mechanism will cause a gap to exist over a portion of the joint. If there is an imperfect seal for any of these reasons, the enormous pressure of the propellant explosion will force high-speed jets of extremely hot gas through the gaps. The resulting erosion will quickly ruin the mating surfaces and make further sealing impossible. Because of the sensitivity of this method to slight defects, it is not suitable for use in high-powered weapons.

A great improvement in the sealing action can be obtained by machining a conical surface at the rear of the barrel to mate with a conical counterbore at the forward face of the cylinder. When the cylinder is cammed forward, the surfaces have an action similar to that of an automobile valve. In moving together, the tapered surfaces have somewhat of a self-aligning action and under the high pressure of the camming mechanism, a metal-to-metal contact over a considerable area is more probable than with flat surfaces. However, with steel mating surfaces, the sealing action is still seriously affected by the presence of dirt or surface defects.

Still better results can be achieved by the use of soft metal sealing rings, two forms of which are shown in fig. 4–7. In the first type shown, a copper ring is expanded into a counterbore at the front face of the cylinder and has a conical mating surface of the type just described. When the cylinder is moved forward, the soft copper conforms quite readily to the surface at the rear of the barrel and produces a much better seal than would be obtained with both surfaces made of steel. A superior method of using sealing rings is shown in fig. 4–7B. With the arrangement shown in fig. 4–7A, the gas pressure acts directly on the joint and tends to force through at any point where a slight opening might exist or the contact pressure may be light. The only factor tending to prevent leakage is the force produced on the mating surfaces by the

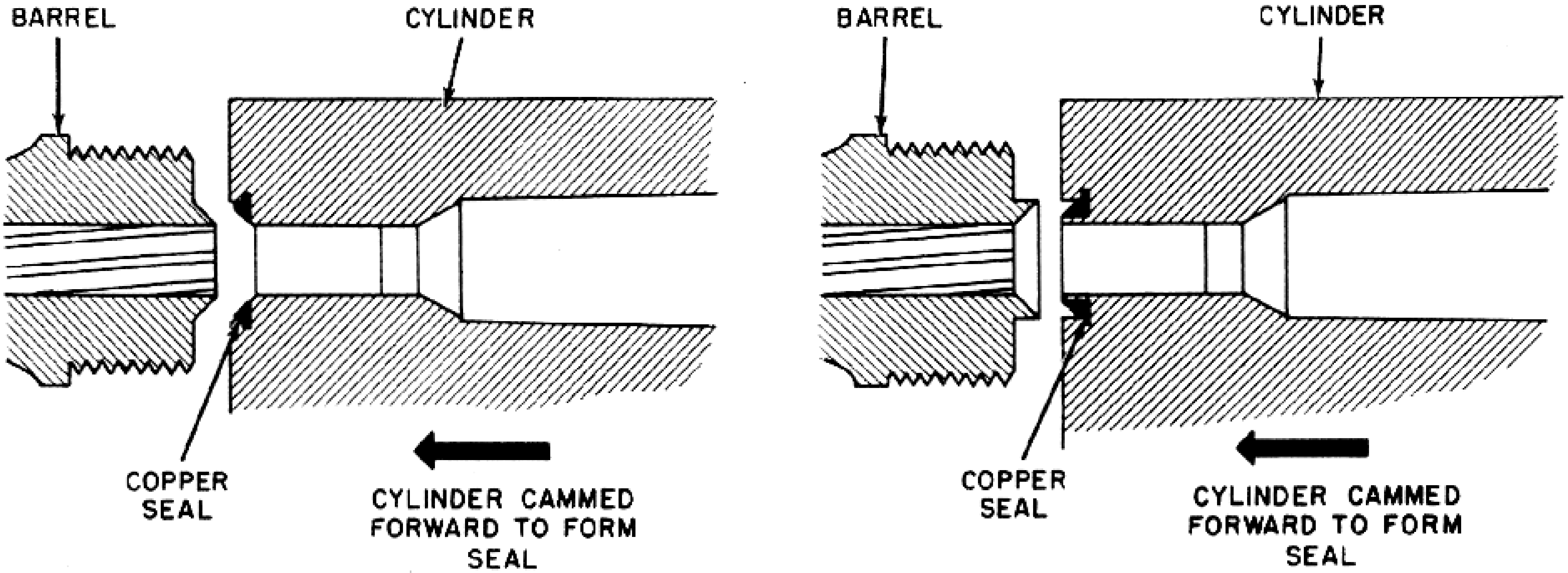


Figure 4–7. Use of Sealing Rings at Chamber Gap.

camming mechanism. On the other hand, a seal of the form shown in fig. 4–7B will tend to be expanded by the explosion of the propellant charge. Therefore the mating surfaces are forced together, not only by the thrust of the camming mechanism, but also by the extremely high pressure of the powder gases. Under the combined actions of these forces, a very tight seal is effected and leakage is practically nonexistent. It is interesting to note that this excellent sealing method can be found in percussion cap revolvers which date from more than 100 years ago.

Although good sealing can be obtained in a revolver by camming the cylinder forward and using sealing rings, the method is accompanied by certain difficulties. The necessity for the sliding motion of the cylinder complicates the cylinder mounting and increases the problems involved in the design of an accurate cylinder indexing mechanism. Also, the reciprocating motion which must be imparted to the cylinder requires additional mechanism to be built into the gun and this mechanism must be ruggedly constructed in order to handle the large forces it must absorb. Furthermore, the forward motion of the cylinder must occur before a round is fired and then, after the chamber pressure has decreased to a safe value, the cylinder must be withdrawn before it can be rotated to place the next round in the firing position. Because the cylinder is of necessity a fairly heavy part and therefore has considerable inertia, both the forward and rearward motions will require some small but significant amount of time which will cut into the time available for the other events of the firing cycle. In a high-rate-of-fire gun where the firing cycle time is in the order of only a few hundredths of a second, the time required for moving the cylinder in and out can easily have a tendency to slow down the rate of fire. Another important point is that the sealing rings must be quite soft to function properly and hence can be dented or deformed easily. If such damage to a ring occurs either through some malfunction of the mechanism or as the result of careless handling during cleaning or maintenance, it is likely that the high-pressure powder gases will quickly blow the seal out entirely, thus producing a very dangerous condition.

The next sealing method for consideration is that in which the cartridge case is pushed forward so that the neck of the case enters the barrel for a short distance. Here, again there are several possible ways of arranging the mechanism. In one method, as shown in fig. 4–8, a so-called "buried" projectile is used, that is, the neck of the cartridge case extends entirely over the projectile. When the cartridge is seated in the chamber opening, a small portion of the neck of the case projects beyond the front face of the cylinder. After the chamber has been positioned behind the barrel, the entire cylinder is moved forward to close the gap between the cylinder and barrel and the projecting neck of the cartridge case enters the barrel. The neck of the case thus spans the small seam where the cylinder joins the barrel. The explosion of the propellant charge causes the cartridge case material to be expanded tightly against the inside of the barrel so that the gas pressure

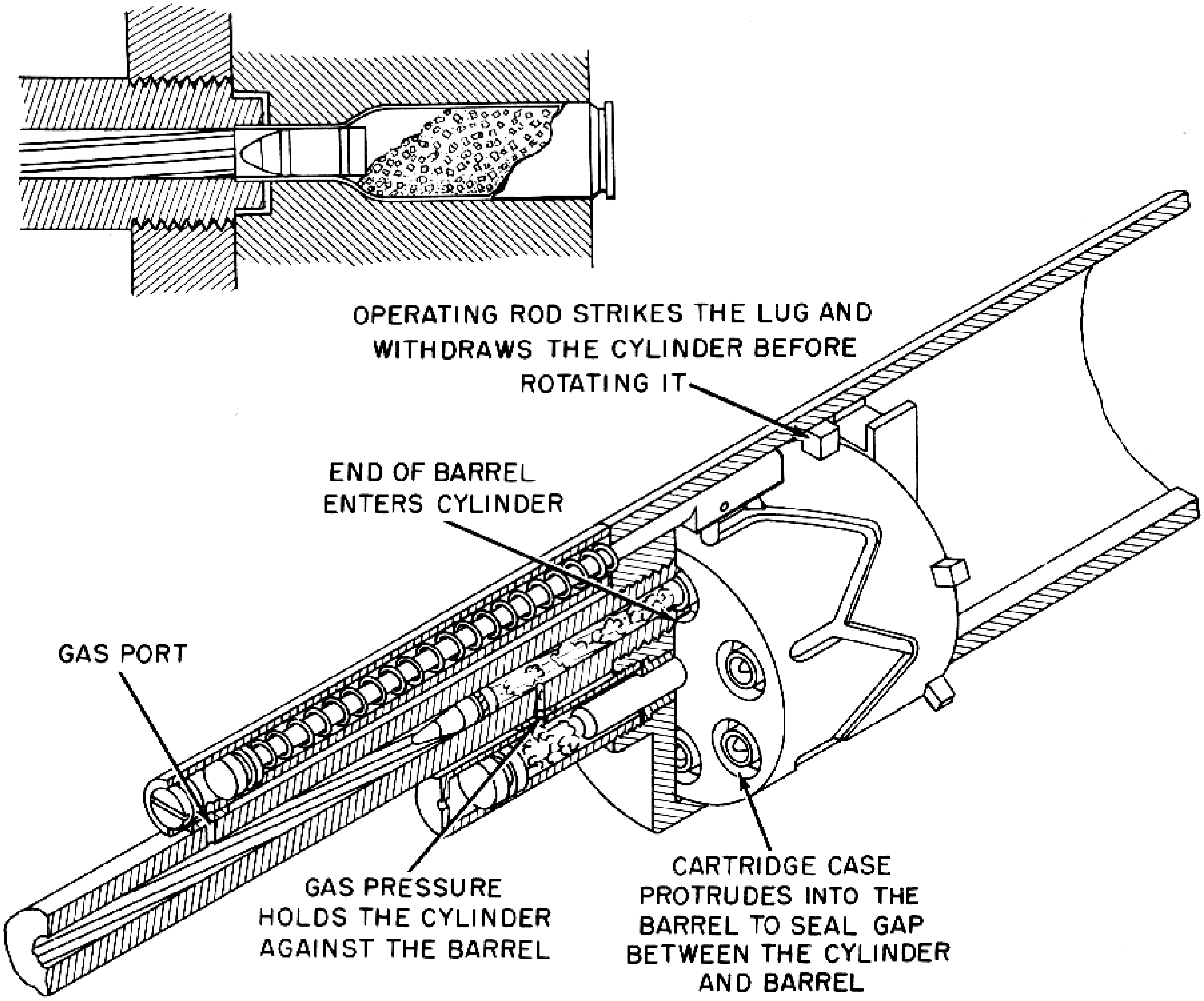


Figure 4–8. Method of Sealing With Neck of Cartridge Case (With "Buried" Projectile Ammunition).

is entirely sealed off from the seam. In spite of the fact that the cartridge case material is quite thin at the neck, there is no tendency for the case to be blown out at the seam, providing of course that the gap is not excessively large. The permissible width of the gap depends entirely on the thickness of the material at the neck of the case. For the thickness found in typical 20-mm cartridge cases, experience has shown that a gap of over $\frac{1}{32}$ inch can be tolerated without failures. In a well-constructed weapon there should be no trouble experienced in keeping the final width of the gap well below this figure and consequently there will be little danger of case blowouts.

The objections to this method of sealing are similar in some ways to those cited for the method previously described because its use requires the cylinder to be cammed forward to effect the seal. Hence it has the same mechanical difficulties and the same tendency to slow down the rate of fire. The form of the ammunition also presents problems, particularly in the larger calibers where it is necessary for the projectiles to be provided with rotating bands. With ordinary ammunition, the cartridge case is crimped to the projectile behind the rotating band but with the "buried" projectile type of cartridge, the case must cover the entire projectile. Such a round is not only difficult to assemble properly, but the projectile is also required to move through the neck of the case for a considerable distance when the gun is fired and therefore may not receive proper support. In addition, the round is of an awkward shape for rapid feeding and the thin, unsupported front portion of the case can easily be damaged in handling or in the ramming phase of the loading cycle.

If ammunition of conventional form is to be used, it is not practical to use the method of sealing described above because it would be necessary for the cartridge to protrude too far beyond the front face of the cylinder. The exposed portion of the cartridge would have to include the entire length of the projectile forward of the crimp as well as the part of the case neck which is to enter the barrel. This means that the cylinder would have to be cammed forward and back again through this whole distance each time the gun is fired. Such an excessive reciprocating motion of the cylinder would be entirely out of the question in any high-rate-of-fire rotary action mechanism.

Dating from about the turn of the century, there have been several instances of a rather clever type of rotary chamber mechanism which makes it possible to use conventional ammunition in such a way that the principle of neck sealing can be employed without the necessity for imparting a reciprocating motion to the cylinder itself. In this mechanism, the cylinder is mounted and rotated in the conventional manner but until a chamber reaches the firing position, the round in that chamber is not pushed all the way home (fig. 4–9). When the chamber does reach the firing position, a sliding bolt drives the

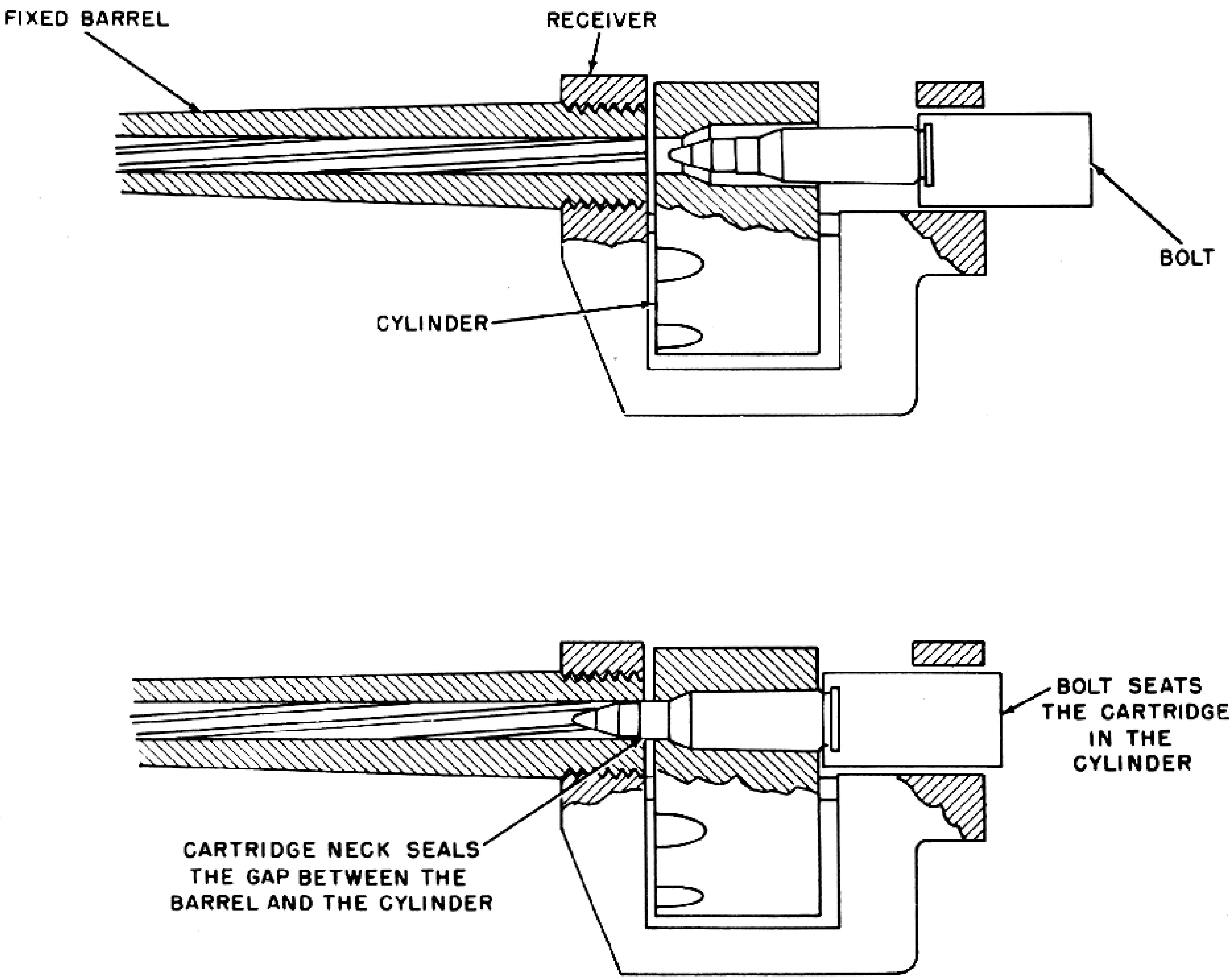


Figure 4–9. Method of Sealing With Neck of Cartridge Case Using Conventional Ammunition.

cartridge forward to seat it fully in the chamber and the bolt is then locked in place. The distance through which the cartridge moves is sufficient to cause the projectile and a small portion of the cartridge case neck to enter the barrel as is required to produce the sealing action. After the cartridge is fired and the chamber pressure decreases to a safe operating limit, the bolt is unlocked and moves to the rear to extract the spent cartridge case.

From the preceding description, it is evident that this mechanism combines the functions of both a revolver action and a reciprocating bolt action. The revolution of the cylinder can be accomplished by the same methods as are employed in a conventional revolver and the bolt can be operated in essentially the same way as in an ordinary reciprocating action. However, there are a number of points concerning the reciprocating portion of the mechanism which require further consideration.

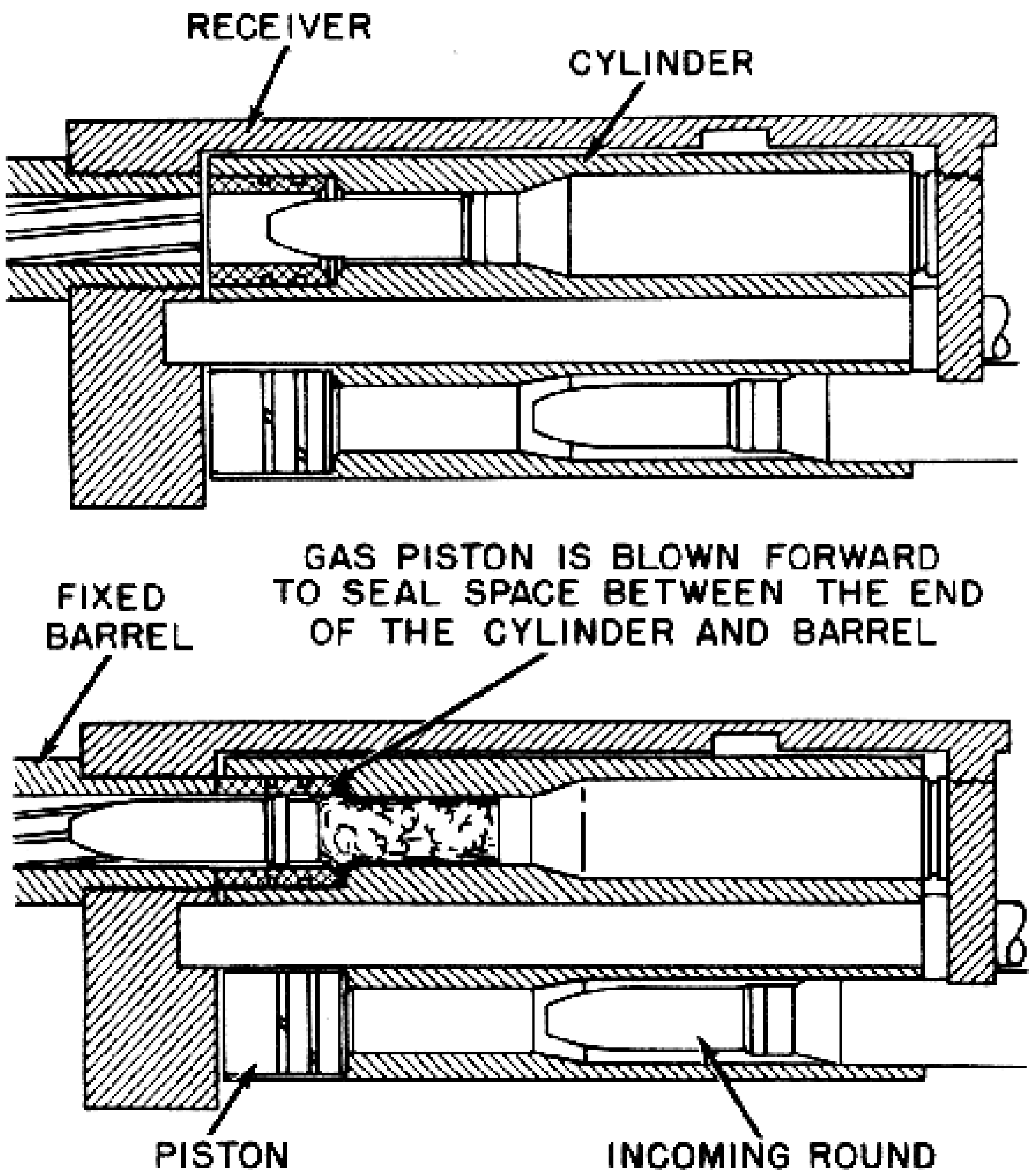


Figure 4–10. Action of Gas Piston Seal.

The characteristics of the bolt mechanism will depend to a large extent on the manner in which extraction is accomplished. In the preferred method of operation, the bolt moves back only through a sufficient distance to be in a position to engage the next round. This means that the bolt only partially extracts the cartridge case and the completion of extraction and ejection is delayed until the rotation of the cylinder moves the cartridge case off to one side of the bolt mechanism. This same movement of the cylinder will bring a fresh cartridge into the firing position and the bolt is then released to push this cartridge home. With this type of operation, it can be seen that the length of the bolt stroke is very short when compared with the bolt stroke of an ordinary reciprocating gun. Since the distance moved by the bolt need only be as great as the length of the exposed portion of the projectile plus a small part of the cartridge case neck, the bolt stroke will only be from one-third to one-half the length of the complete round. On the other hand, the bolt stroke in a conventional reciprocating weapon must be at least as long as the entire round and usually is considerably longer. The advantage of the very short bolt stroke required in the combination action is that the bolt motion can occur at a relatively low velocity. On account of this low velocity, the accelerations and shocks to which the bolt is subjected during firing are well below the critical and limiting values experienced in ordinary high-rate-of-fire reciprocating actions. Therefore, the mechanism is capable of producing very high cyclic rates without the difficulties usually associated with sliding bolt mechanisms.

Except for the fact that the action of the bolt must be synchronized with the indexing of the cylinder, the design of the bolt mechanism of a combination gun is governed by essentially the same principles used for conventional reciprocating guns. The energy for operating the mechanism can be obtained conveniently from recoil or gas actuation and the blowback action produced by the residual pressure can also be utilized with good effect. Because the principles of the blowback, recoil, and gas systems are covered in the preceding chapters of this publication, they will not be repeated here.

The third method of sealing to be covered in this analysis is that which makes use of sealing pistons mounted in the cylinder at the forward end of each chamber opening as shown in fig. 4–10. The piston is in the form of a sleeve with an internal diameter just large enough to permit the projectile to pass through it. At the instant the rotating band of the projectile passes the rear face of the piston, the high-pressure powder gases expanding behind the projectile get behind the piston and drive it forward with great force. The piston moves forward so quickly that it moves across the chamber gap and its front surface becomes tightly pressed against the barrel face before the rear part of the projectile has entered the barrel. As the projectile moves through the barrel, the gas pressure acting on the rear of the piston maintains a tight metal-to-metal seal at the face of the barrel and the piston rings prevent the escape of gas around the outside of the sleeve. When the residual pressure in the bore falls to a relatively low value, the forward force on the piston decreases to the point where the cylinder can be rotated to bring the next round into position for firing.

The gas piston method of sealing the chamber gap is very simple from the functional viewpoint and does not add greatly to the cost and complexity of the gun. In addition, when it is properly used, it produces a quite effective sealing action. However, like all of the other sealing devices described up to this point, the use of piston seals involves a number of important problems which must be given careful consideration in design.

The sealing action produced by the piston is similar in nature to that produced by the first method described in this analysis. That is, the main sealing action results from a high-pressure metal-to-metal contact between two flat surfaces. When the surfaces are in good condition, accurately aligned, and free of dirt particles, they can bear tightly against each other over a sufficient contact area to effect an excellent seal. However, if there is any defect which prevents good contact, the sealing action will be imperfect and the consequent leakage of the extremely hot high-velocity powder gases will quickly erode the surfaces. The seal will then quickly deteriorate and soon become useless.

Another very serious problem arises from the tremendous force applied to the piston by the peak chamber pressure. As mentioned previously, the peak pressure produced by modern high-powered aircraft cannon ammunition can range anywhere from 45,000 to 65,000 pounds per square inch. In a 30-mm gun, the sealing pistons will have an internal diameter of approximately 1.2 inches and a reasonable wall thickness for the piston is about 0.3 inch. This means that the area exposed to the forward acting pressure of the powder gases will be nearly 1.5 square inches. If it is assumed that the peak chamber pressure is 65,000 pounds per square inch, the force which drives the piston forward will be in the order of 98,000 pounds. Since the piston is relatively light in weight, it will be driven forward with great violence and will strike the barrel face with a very sharp impact. Under these conditions, the rear face of the barrel and the front surface of the piston are subjected to a severe hammering and the entire body of the piston must absorb an intense shock. To add to these difficulties, the temperature of the piston rises rapidly during a burst and may quickly reach the point where it tends to weaken the piston material. All of these factors make it extremely difficult to make effective use of piston seals in rotary action cannons which employ very high-powered ammunition. Even when special alloy steels are used, the pistons tend to break down quickly, either from battering at the sealing surface or from actual crushing of the piston body itself.

The use of piston seals creates other problems which are not directly related to the sealing action but do affect the design of the gun itself. One of these problems can be appreciated readily by those familiar with the Williams floating chamber device which is used to cause a .22 caliber cartridge to produce the same recoil as the round used in a .45 caliber automatic pistol. In a revolver weapon, the piston seal produces the same effect of magnifying force, although the presence of the increased force is not evidenced in the same way. The forces resulting from the action of the piston seal and the effect of these forces can be seen in fig. 4–11.

If it were not for the presence of the piston seal, the only rearward force produced by the pressure of the powder gases would be the force due to the pressure on a portion of the base area equal to the bore cross-section area. The rearward pressure on the remainder of the base area is cancelled by an equal and opposite pressure which acts forward on the tapered portion and shoulder of the cartridge case. The effective pressure is transmitted through the base of the cartridge case to the frame of the weapon. The force produced by the pressure creates a tension in the frame and also is the force which causes the weapon to move in recoil.

The action of the piston seal produces an additional force within the weapon. As indicated in fig. 4–11, the powder gas pressure thrusts the piston seal forward and at the same time acts against the surface at the bottom of the counterbore in which the piston slides. The forward acting force is transmitted through the piston to the barrel and thence to the frame while the rearward force is also transmitted to the frame through the cylinder and cartridge case. These forces are equal and opposite and therefore do not intensify the recoil but they do act to cause an additional tension in the frame. As previously shown, this additional force can be almost 98,000 pounds (for a 30-mm gun with a peak chamber pressure of about 65,000 pounds per square inch). Since the effective force on the base of the cartridge case due to the pressure acting on the bore cross-section area is only approximately 74,000 pounds, it is evident that the use of the piston seal can more than double the tension in the frame. With a total tension of 172,000 pounds instead of only 74,000 pounds, it will be necessary to use a much stronger and heavier frame structure than if the piston seals were not used.

Depending on the form of the ammunition fired by the gun, it is possible that the presence of the piston seals may result in a further design disadvantage. In designing the cylinder of a revolver machine gun, it is desirable to space the chambers as closely as possible by keeping the walls between the chambers at the minimum safe thickness. A compact arrangement, such as is shown in fig. 4–12A, makes it possible to hold the cylinder to a small diameter and to minimize its weight and inertia. This not only makes for a less bulky weapon, but also decreases the power required for operation and lessens the strain on the rotating mechanism.

In order to withstand the forces to which they are subjected, piston seals must have a substantial wall thickness. If an attempt is made to counterbore the chambers for receiving suitable seals, a situation like that shown in fig. 4–12B may result. The counterbores might actually overlap or the walls remaining between them might be entirely too thin for safety. To provide sufficient room for the counterbores, it is necessary to respace the chambers as shown in fig. 4–12C. Therefore, the cylinder will be larger and heavier and the design will suffer from all of the consequent disadvantages. Of course, these remarks apply only when the ammunition is nearly cylindrical in form, that is, when the cartridge case is not much larger in diameter than the projectile. When the cartridge case is strongly bottle-necked, there may be ample metal for accommodating the seals at the forward end of the cylinder even when the chambers are spaced as closely as possible at the rear.

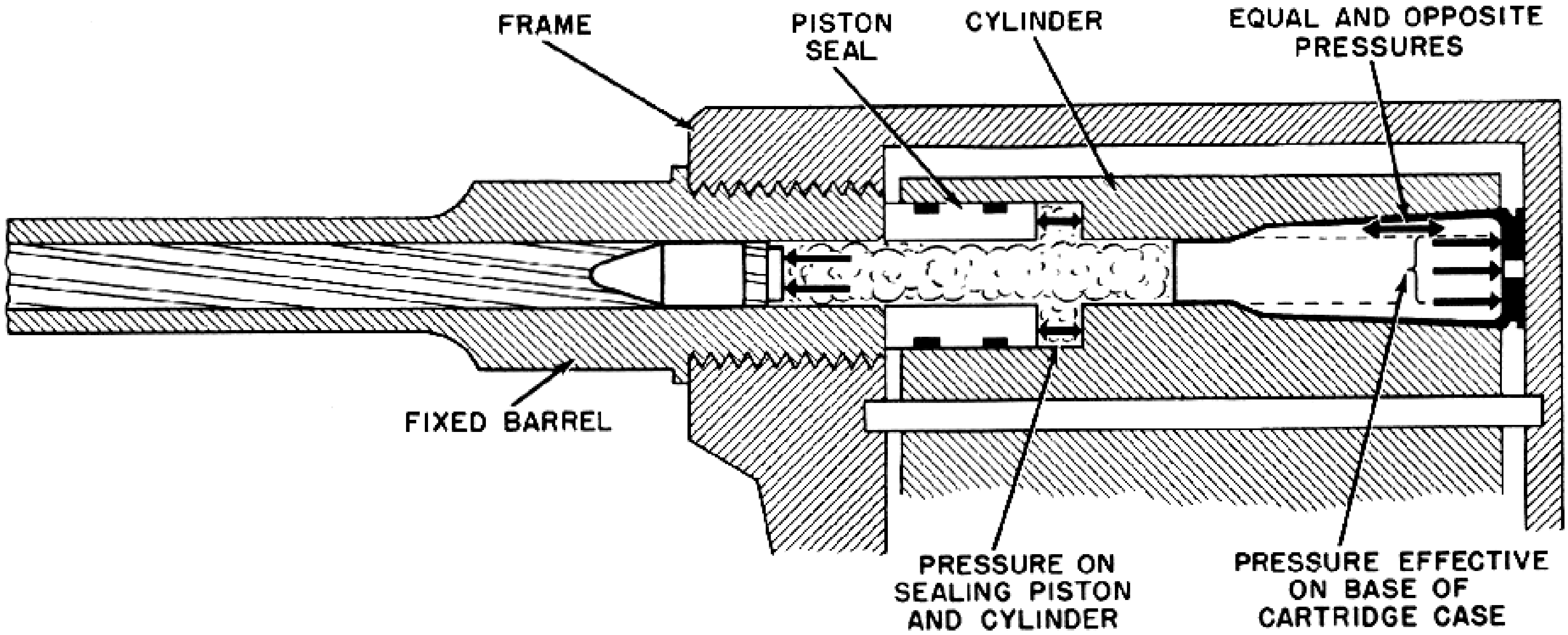


Figure 4–11. Effect of Piston Seal on Force Applied to Revolver Frame.

The preceding paragraphs are concerned with the problem of gas leakage in full automatic rotary action machine guns. The next important problem for consideration is caused by the great amount of heat absorbed by the cylinder during long bursts at the extremely high rates of fire being attempted with modern rotary action cannon. The magnitude of this problem can be illustrated readily by one rather startling fact. The ammunition used in one modern 30-mm revolver gun is loaded with a 1060-grain propellant charge and the gun is designed to fire at a rate of 1200 rounds per minute. Since there are 7000 grains to the pound the amount of powder burned per minute is:

$$\frac{1060}{7000} \times 1200 = 181 \text{ (pounds)}$$

It is truly amazing to realize that this amount of smokeless powder would more than fill two large feed-sacks and that it is burned at such a rate that it all would be consumed in one minute inside a steel cylinder which weighs only about 50 pounds and is less than one foot long. The rate at which thermal energy is released in firing a weapon of this type is tremendous, particularly when the relatively small physical dimensions of the cylinder are taken into account.

During a sustained burst, the cylinder of a high-rate-of-fire rotary action mechanism absorbs enough heat to cause its temperature to rise quite rapidly. Even with bursts of moderate length fired at intervals, the heat will be retained in the cylinder between bursts and can build the temperature up to the danger point in a relatively short period of time. When the temperature of the cylinder reaches a level of from 600° to 800° F., it is possible for a "cook-off" to occur. This condition is particularly dangerous in a rotary action weapon because of the fact that the gun may stop firing with live cartridges in chambers which are not aligned with the barrel. If these cartridges should cook off, the results would be disastrous.

The heat problem in high-rate-of-fire rotary chamber mechanisms is so critical that it either places a severe limitation on the number of rounds which can be fired over a given period, or requires the development of an effective method of cooling the cylinder. Cooling by normal air flow around the cylinder is not likely to be successful, particularly because a large portion of the cylinder is usually covered by the other parts of the gun. The fact that the cylinder revolves will probably make the use of jacketing and liquid coolant impractical. One method of cooling which can be used is to spray a refrigerant gas through the chambers after the fired cartridge is ejected. Further heat removal could be accomplished by using the same means to cool the outside of the cylinder. In any case, the cooling system used for the cylinder will be the major factor affecting the ability of the gun to deliver a high volume of fire.

The third problem which has proved to be a stumbling block in the development of high-rate-of-fire rotary action cannon is the requirement for high precision in the cylinder indexing mechanism. This problem arises because of the necessity for maintaining a very accurate alignment of the chamber with the barrel bore at the instant the gun is fired. In revolver hand guns, this same alignment requirement exists but is much less critical. With low velocity bullets made of soft lead, a small misalignment is not too serious because the worst that can happen is that the gun will shave some lead and the accuracy of fire may be upset. However, in aircraft cannon, the effect of misalignment can be very dangerous. If the chamber and barrel bore are not concentric within 0.005 inch, it is likely that the rotating band of the projectile will be sheared off. This will not only cause the flight of the projectile to be very

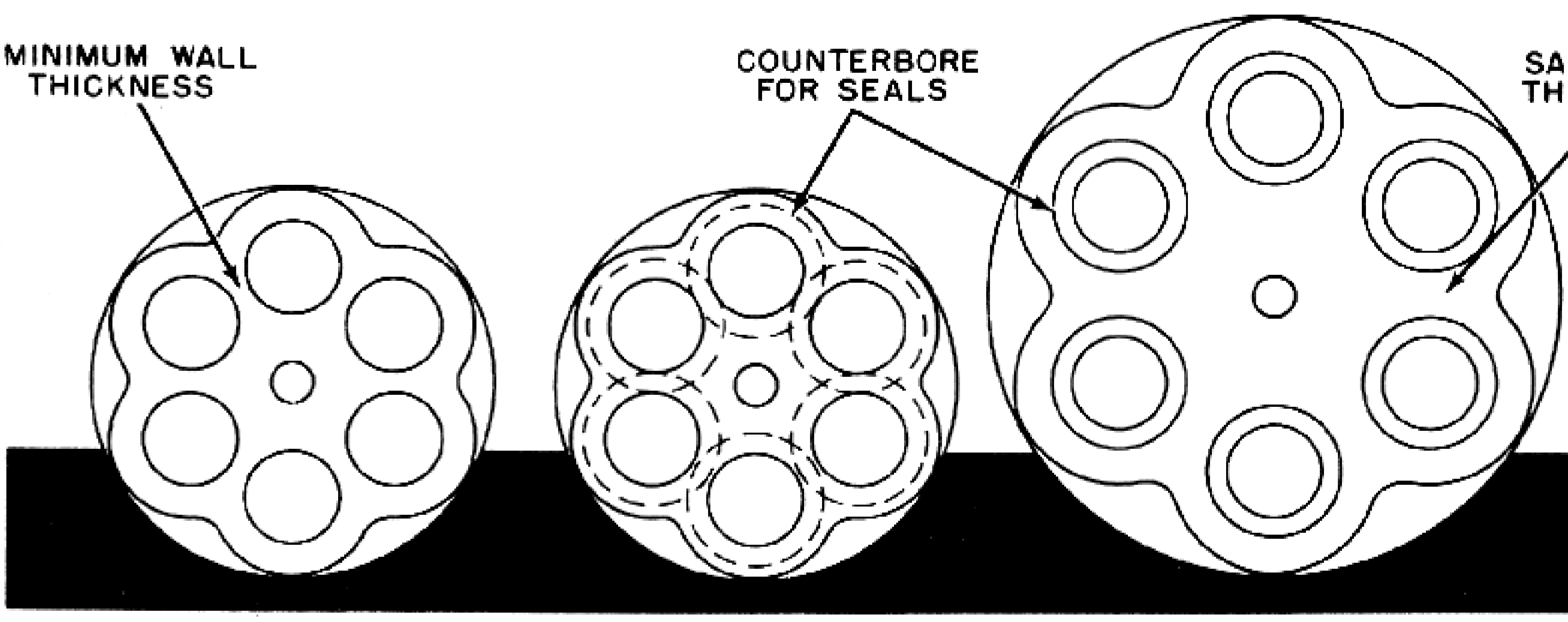


Figure 4–12. Effect of Space Required for Piston Seals.

erratic but the sheared-off band may jam the mechanism or may become caught in the barrel in such a way as to create a dangerous condition. A slightly greater eccentricity can be really disastrous. The projectile enters the barrel at a very high velocity and if it encounters any positive interference as it does so, it may tend to twist and batter itself within the barrel. The possible consequences of such an occurrence, particularly with H. E. ammunition and a chamber pressure in the order of 60,000 pounds per square inch, are not pleasant to contemplate.

The problem of insuring that the chamber and barrel bore will always be concentric within 0.005 inch at the instant of firing is extremely difficult in a high-rate-of-fire revolver. The mere accumulation of dimensional tolerances creates a terrific production problem. The following are some of the factors which must be considered: First, the chambers must be bored in the cylinder so that their axes are accurately parallel and all lie precisely on a true circle of a definite diameter. The angular spacing of the chambers around this circle must also be held accurately. The cylinder must be mounted on an axle which must also be concentric with the circle and the bearings in which the axle turns must permit no side play or runout. The bearings must be located in the frame so that the chambers will line up accurately with barrel bore. If the barrel is screwed into the frame, the concentricity of the bore and the threads must also be held. The fact that the tolerances on all of these dimensions are directly cumulative makes it very difficult to achieve the required alignment between the barrel and cylinder.

The foregoing points are only concerned with the machining and mounting of the rotary chamber mechanism in relation to the barrel. An even more complicated chain of dimensions may be involved in the indexing mechanism and some of the tolerances on these dimensions will accumulate with the tolerances on the dimensions previously mentioned. Although modern manufacturing methods permit very accurate control of dimensions, extremely close control is expensive. It can be seen that even with very small tolerances, the specified 0.005-inch concentricity between the chambers and the barrel is really an exacting requirement.

No matter how accurately a revolver machine gun may be built, the problem of alignment does not end when the weapon is at last completely assembled. Even if the dimensions have been controlled so that the cylinder can be indexed to bring each chamber into alignment with the barrel within the tolerance, there is no assurance that this will still occur when the weapon is burst fired at high cyclic rates. In a high-powered weapon fired at a high rate, the entire gun structure and the mechanism are subjected to very large forces and dynamic disturbances which can cause considerable elastic deformations in the parts. Vibrations, thermal expansion, and other influences can all contribute to dimensional changes which can upset the alignment achieved with the gun in the static condition. To make matters worse, the occurrence of even a small amount of normal wear at certain critical points can quickly destroy the usefulness of the weapon.

Thus it is evident that the proper functioning and reliability of a rotary action cannon is critically dependent on the simple factor of alignment between the chambers and the barrel. This alignment is so difficult to achieve and maintain that the development of rotary chamber mechanisms has been slowed down very seriously. It is possible that this problem may be solved by the development of some device which will provide a direct and positive means for bringing the chamber into alignment with the barrel. It also may be that the fundamental difficulty should be sidestepped completely by using a mechanism which moves the round forward so that the projectile enters the barrel before firing.

The last of the four problems which have bottlenecked rotary action development for over 80 years is concerned with the free run of the projectile before it enters the barrel. In an ordinary rotary chamber mechanism, the projectile must travel for a distance approximately equal to its own length in passing from the cylinder into the barrel. In moving through this distance, the projectile is driven by very high chamber pressures so that it picks up speed very rapidly. By the time the rotating band engages the rifling in the barrel, the projectile is moving forward at high velocity but has not yet started to spin. When the rotating band meets the rifling, the band is immediately engraved and the twist of the rifling causes a torque to be exerted on the projectile. With standard rifling, the applied torque is very large because the rifling attempts to produce instantaneously the rotation speed called for by the twist of the rifling and the high forward velocity of the projectile. Naturally, the projectile has a considerable inertial resistance to rotation and the force required to overcome this resistance to rapid angular acceleration may be greater than the shear strength of the rotating band. If this is the case, the surface of the band may be stripped off completely before the projectile has acquired any considerable spin velocity and the flight of the projectile after it leaves the muzzle will therefore be unstable. A condition of this sort is certain to make the gun hopelessly inaccurate.

Some improvement of this condition may be possible with the use of progressive twist rifling, particularly if the rifling starts at zero twist and increases gradually to full twist. Rifling of this type would tend to accelerate the spin of the projectile more smoothly so that there would be a much better chance of not stripping the rotating band. However, if the projectile is moving too rapidly when it engages even this type of rifling, the rotating band may still be stripped. Although the acceleration would be smooth, it would have to be quite high and it is possible that the rotating band could not stand the required force.

Another solution to this problem would be to employ the combination type of action in which the round is moved forward so that the projectile enters the barrel before firing. If this is done, there will be no free travel of the projectile and the rifling will function in the normal manner with no danger of stripping the rotating band.