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# NEW CONCEPTS IN RECOIL MECHANISMS

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## I INTRODUCTION

The function of a recoil mechanism is to moderate the firing load on the supporting structure. This moderation is accomplished by prolonging the time of resistance to the reaction force caused by the action of the gun on the propellant gases. If no resistance is offered, the reaction force will be as great as the action force caused by the propellant gas. In other words, if the gun tube is rigidly fixed to the gun mount/carriage, the supporting structure is subjected to the full force of the propellant which, for large guns, may be 2- to 3- million pounds. To withstand such a force, the structure has to be not only strong and heavy but also wide-based to prevent tip over. As the gas pressure propels the projectile toward the muzzle, it exerts an equal and opposite force on the breech, which tends to drive the gun backward. The recoil mechanism suppresses this force gradually and also limits the rearward movement.

All guns, prior to about 1890, were mounted rigidly on a wheeled carriage which rolled backward to dissipate the recoil energy. The US first started to manufacture a gun with a recoil mechanism, the French 75mm field gun, during World War I. Since that time, the design of recoil mechanisms has been actively pursued.

The recoil mechanism is composed of three basic components: a recoil brake, a counterrecoil mechanism, and a buffer. The recoil brake normally consists of a hydraulic cylinder and a piston assembly. The counterrecoil mechanism consists of a recuperator and a counterrecoil cylinder assembly. These two subsystems (recuperator/counter-

recoil cylinder) can be either separate (independent recoil system) or combined (dependent recoil system). Sometimes, the counterrecoil mechanism and recuperator are used synonymously. The recuperator's function is to store part of the recoil energy in order to return the recoiling parts to the in-battery position, whereas the counterrecoil mechanism is used to return the recoiling parts to the in-battery position. The buffer's function is to absorb the counterrecoil energy as the weapon approaches the in-battery position, thereby assuring a smoothly controlled stop. The recoil brake may be either a hydraulic or a hydrospring type, whereas the recuperator may be either a hydropneumatic or a hydrospring type. Some examples of independent recoil mechanisms are the Schneider and Filloux types (1). In dependent recoil mechanisms, such as Puteaux and St. Chamond types (1), only the recoil piston rod is connected to and moves with the other recoiling parts and the fluid in the recoil brake directly influences the counterrecoil mechanism. All such mechanisms are single recoil systems (primary recoil systems). Sometimes, it is advantageous to have a secondary recoil system between the top and bottom carriages of heavy weapons (double recoil system). The Filloux and St. Chamond recoil mechanisms are variable recoil types (to accommodate changes in elevation), whereas the other two are constant recoil types.

Since these systems are all quite complex and contain several moving parts, their reliability is limited. As indicated later, the state of the art in the development of compressible hydraulic fluids is also quite limited. Today's artillery weapons may fire about one-quarter of the time and spend about another quarter of the time in moving about the battlefield. The remaining time is spent resolving RAM-D (reliability, availability, and maintainability-durability) problems. Various systems effectiveness studies indicate that the improvement which could be realized by doubling the RAM-D characteristics is significantly greater than that which could be realized by doubling the capability of weapon subsystem performance. There is nothing which can surmount this payoff potential of RAM-D improvements. In view of the multiple difficulties cited above, it is the object of this paper to discuss various recoil mechanism concepts having as few moving parts as possible, virtually unlimited compressibility, and transmitting the smallest amount of force to the supporting structure. These concepts are discussed in the following section. The development of a generalized mathematical model and the governing equations are given in section 3. The numerical results are presented in section 4 and the conclusions are stated in section 5



454

## II CONCEPTS

A compressible fluid recoil mechanism presently under development uses a Dow Corning silicone fluid (DC 200-10CS). Although this fluid is considered compressible, its bulk modulus at 48°F (8.8°C) and at one atmosphere pressure is 141,200 psi (2). As temperature drops to -60°F and pressure increases to 4,000 psi, this fluid could have a bulk modulus as high as 260,300 psi. The ideal fluid should be highly compressible (low molecular weight), should possess a high flash point (low flammability) and its characteristics should be nearly independent of pressure and temperature. (A fluid's compressibility is the inverse of its bulk modulus). Artillery weapons are expected to function in a temperature range of -65°F to 120°F and with internal pressures ranging from 0 to 4,000 psi. Also, the temperature may rise approximately 1°F for each round fired unless adequate time is allowed for its heat dissipation between rounds or after a series of rounds. A perflourinated fluid (FC 75, 0.8 CS viscosity at 77°F) manufactured by 3M Company, and Freon E3 (1.3 CS viscosity at 77°F), currently no longer being manufactured by Dupont, may be up to 25% more compressible than the Dow Corning silicone fluid, although they still fall far short of the desired compressibility for recoil mechanisms.

It is not desirable to design flexibility into recoil mechanism test fixtures with the intent of boosting the compressibility of the total system beyond that of the fluid because the rate of loading of test fixture is so great (0 to 4,000 psi or higher within a few milliseconds) that the cylinder could buckle. Also, the oil seals between the moving and fixed members may make point contact rather than area contact thus causing leakage and permitting entrainment of the surrounding air. The air solubility for MIL-H-5606 is reported to be 0.109 (volume of air/volume of hydraulic fluid) whereas it is 0.168 to 0.19 for the Dow Corning silicone fluid at atmospheric conditions. Of course, the air content increases as the temperature drops and as the test fixture is actuated and could reach 1.3. Part of the entrained air may be undissolved. The dissolved and undissolved gases, collecting in pockets or foaming, can cause irregular motion and loss of control in recoil mechanisms. Magorien (3) discusses the behavior of air in hydraulic systems. However, the gas content increases the compressibility of the fluid and thus reduces the bulk modulus much beyond that for "liquids".

Consider, as an example, a liquid with a bulk modulus (B) in contact with an inert gas, such as $N_2$, with a specific heat ratio ( $r$ ) of 1.4. The contents of the system may vary from 0 to 100% and may be separated by an elastic or floating member. The bulk modulus of a liquid is defined as



455

$$B = -\frac{v\,dp}{dv} \tag{2.1}$$

where p is the pressure and v is the volume. The bulk modulus is a function of pressure and temperature. However, the temperature variation of the fluid is small (such as one or two degrees) during any round of fire. The pressure, on the other hand, does vary nearly 3,000 psi. Therefore, the bulk modulus may be considered as a quadratic function of pressure ($a + b\,p + c\,p^2$). In this instance, the integration of equation 2.1 yields

$$-\ln v = \frac{2}{\sqrt{q}} \tan^{-1}\left(\frac{2cp+b}{\sqrt{q}}\right) \text{ where } q = 4ac-b^2 \tag{2.2}$$

Sometimes, the pressure range is small and the bulk modulus can be considered as a linear function of pressure, p. Thus, the integration of equation 2.1 yields

$$\frac{v}{v^o} = \left(\frac{a+bp^o}{a+bp}\right)^{1/b} \tag{2.3}$$

where the superscript 'o' indicates initial conditions.

For concept evaluation or design purposes, such as, what will a combination of gas and liquid do as opposed to a liquid alone, or how to go about determining the compressibility of a given liquid, it may be advisable to consider the bulk modulus as independent of pressure at the desired temperature. In this case, the integration of equation 2.1 yields,

$$\frac{v}{v^o} = e^{(p^o-p)/B} \tag{2.4}$$

For example, a liquid ($B = 120{,}000$ psi $p^o = 3{,}000$ psi, $p = 200$ psi) undergoes a compression of about 2.33% because

$$\frac{v}{v^o} = e^{.0233} \simeq 1.0233$$

The equation of state for a gas may be written as

$$\frac{v_g}{v_g^{\,o}} = \left(\frac{p_g^o}{p_g}\right)^{\frac{1}{r}} \tag{2.5}$$



456

If a given liquid ( $\bar{B}$ ) can do the job of a combination of some other liquid ( B ) and a gas, let it be called a super fluid and indicate its properties by a bar over the symbols. In order to make a fair comparison, let the total volumes and pressure ranges remain the same for both systems. By the use of equations (2.3) and (2.4), the super fluid's bulk moduli, respectively, become

$$\bar{B} = \frac{\frac{v_g^o}{v_h^o}\left(\frac{p^o}{p}\right)^{\frac{1}{r}} + \left(\frac{a+bp^o}{a+bp}\right)^{1/b}}{\frac{1}{rp}\frac{v_g^o}{v_h^o}\left(\frac{p^o}{p}\right)^{\frac{1}{r}} + \frac{1}{a+bp}\left(\frac{a+bp^o}{a+bp}\right)^{1/b}} \qquad (2.6)$$

$$\text{and } \bar{B} = \frac{\left(\frac{v_g^o}{v_h^o}\right)\left(\frac{p^o}{p}\right)^{\frac{1}{r}} + e^{-(p-p^o)/B}}{\frac{1}{rp}\frac{v_g^o}{v_h^o}\left(\frac{p^o}{p}\right)^{\frac{1}{r}} + \frac{1}{B}e^{-(p-p^o)/B}} \qquad (2.7)$$

The effective (or super fluid) bulk modulus is calculated at 3,000 psi for MIL-H-5606. This is shown in figure 1 as a function of nitrogen content. The bulk modulus is very sensitive to small changes of gas content when the percentage of gas is small. The bulk modulus of Dow Corning's expensive DC 200 silicone fluid is 168,607 psi at 3,000 psi and at 72°F. However, a better bulk modulus can be attained by combining inexpensive hydraulic fluid and gas.

A system with high RAM-D can be characterized as the one which is simple and contains few moving parts. Consider the M140 recoil mechanism mounted on the M60A1 tank. Although, this mechanism looks simple and has only three moving parts, its failure rate is high. Some of the failures are due to breakage (every few hundred cycles) of its giant mechanical spring, leakage of oil around the seals, failure to return to the battery position, inadequate replenisher, wear and tear of piston and sleeve, hang out of battery, breakage of the inertia valve, and either loosened or broken bolts. Most of these problems can be solved by elimination of the mechanical spring and the replenisher, and the utilization of only one moving part. Such concepts are shown in figures 2 through 5.

The bulk modulus of a fluid system as a function of the amount of nitrogen or air is shown in figure 1. Such a system is quite sensitive to the presence of small amounts of nitrogen and is



457

much less sensitive to larger amounts since the bulk modulus decreases significantly as nitrogen content is increased. The relationship between bulk modulus and temperature is also shown in figure 1. The bulk modulus varies drastically between -65°F and 120°F and drops continuously as temperature increases. If the temperature is low at a particular weapon's location, the bulk modulus increases; however, increasing the gas content of the system will decrease the bulk modulus and thus compensate for the low ambient temperature. If the temperature is higher at another operational location, the bulk modulus is lower; however, the gas content can be lowered to increase the bulk modulus. Thus, the gas-oil mixture concept can accommodate both compressibility and temperature compensation in lieu of the separate mechanisms required in the past.

It was proposed by other investigators to use Dow Corning DC 200 as a compressible fluid for new systems under development. The cost of DC 200 silicone fluid is several times that of the hydraulic fluid used in many weapon systems. These inexpensive hydraulic fluids can be used in the conceptual systems instead of the Dow Corning fluid because the compressibility and temperature effects are compensated for. Of course, the conceptual systems weigh less than a pure liquid system.

With conventional recoil systems, the recoiling parts move rearward upon initiation of the breech force, i.e., upon firing. The rearward motion is resisted by the inertia of the recoiling parts, friction, and the fluid pressure forces exerted inside the recoil mechanism. The fluid pressure forces are a result of the throttling of the fluid and compression due to the differential area of the piston (a moving part) and its motion. However, if the recoiling parts are put in motion in the direction of projectile travel (forward run-up) and the breech force is not applied until these parts have attained a predetermined forward velocity (equal to about half of the recoil energy), lower resistance forces will result. Such a principle is called "soft recoil". The development of this technique began in the mid 1960's. The conventional versus soft recoil principles are illustrated in figure 6. Since the resisting force in a soft recoil mechanism is much smaller than in a conventional recoil mechanism, the time for application of this force is much longer. To date, the soft recoil principle has been applied to the M204, 105mm towed howitzer only.

Since the objective of any recoil mechanism is to design the resisting force to be as small as possible (which can account for all its breech force), and to have as few parts as possible, the

conceptual sketch shown in figure 2 was initially adopted. The initial position of the piston (the moving part) is shown. The hydraulic fluid, or DC 200, is at about 3,000 psi pressure and fills the entire recoil cylinder including the buffers.

The fluid pressure acts on all exposed surfaces. The pressurized fluid acting on the face (end) of the piston forces it to move (slide over the sleeve) in the forward direction, after it is unlatched. When it reaches a predetermined velocity, the firing (or initiation of breech force) takes place. Since this breech force overcomes the forward motion, the piston is brought to rest, momentarily, and then accelerates to the rear. Of course, there is an opposing force of pressurized fluid acting on the face of the piston. This opposing force continues to grow as the piston moves to the rear and compresses the fluid into a smaller volume. Eventually, the piston is brought to rest momentarily with or without the assistance of the rear buffer. (If the predetermined velocity of the piston is properly matched with the breech force, the piston stops without the aid of the rear buffer.) A sensor was developed to determine the velocity (4). Even when the velocity is accurately calculated and the sensor operates correctly, there may be occasions where the breech force cannot be applied at the correct time because of ignition delay or other considerations. In such situations, the actual velocity of the piston may be greater than the predetermined velocity and it may not be able to return to the latched (initial) position. It is desirable to utilize about 40% of the breech force to stop the forward (outward) motion of piston. This will cause the piston to enter approximately 2 inches into the rear buffer before its motion is stopped. The latch is reset by the piston near the end of its rearward motion. The pressurized fluid then forces the piston to move slightly forward to the latched (initial or in-battery) position.

Another purpose of the rear buffer is to check out the cook-off function. (Cook-off is the spontaneous firing of a round due to excessive heat or other cause.) The application of breech force to the piston takes place when the piston is in its initial position. The piston accelerates rearward because of the cook-off breech force. The pressure of the trapped fluid inside the buffer will be much higher than that of fluid outside the buffer. The design of the buffer should be such that the piston's rearward motion ceases before it slams into the rear end of the recoil cylinder and also such that the resisting force does not build to a point permitting weapon tip-over.

The front buffer is intended to control misfires. A misfire, or firing delay, may take place because of a bent firing pin or other malfunction. If no front buffer is provided, the piston may slam into the end of recoil cylinder. In the event of a misfire the piston enters the front buffer and traps the fluid inside the buffer raising its pressure much beyond the pressure inside the recoil cylinder. This sets up an opposing force to stop the forward motion of the piston. The design of the front buffer should be such that the piston velocity is reduced to a negligible value before it reaches the front end of the recoil cylinder and yet not permit cavitation inside either the cylinder or the buffer during any part of the cycle.

Some of the characteristics of this system are discussed in reference 5. However, in this reference the governing differential equations (coupled) were not simultaneously solved. The author solved these equations simultaneously and studied the system's operating characteristics. It was determined that the recoil cylinder would need to be about 8 feet long and cavitation would result if based upon the conceptual sketch of figure 2. The author modified the conceptual sketch as shown in figure 3. The normal run (without misfire and cook-off) characteristics remained the same. However, the handling of cook-off and misfire situations would be improved if cook-off and misfire chambers are as shown in figure 3. The storage chamber, in addition to the buffers, provides a shorter path and a better control of piston motion inside the buffers. The storage areas prevent a rapid rise and fall of buffer pressures and thus reduce the chances of weapon tip-over as well as cavitation inside the buffers.

To make the conceptual model as general as possible, to make the system lighter, to increase the compressibility of the system, to compensate for changes in temperature, and to use an inexpensive and small amount of oil (fluid), a gas, such as nitrogen, is introduced into the central part of the recoil cylinder and separated from the liquid by a polyurethane liner (dotted line). Since the pressures will remain the same on either side of the liner (because of its elasticity) it is expected that the operating pressures won't pose a problem. If the liner is not satisfactory, there is always the technique of the nineteen fifties, that of separation of gas and liquid by a floating piston.

It is recalled that the predetermined velocity chosen (about 50% of breech force) is somewhat less than the ideal because of ignition delay and other considerations. The piston enters the rear buffer approximately 2 inches (because of the slightly less than ideal soft recoil cycle) before it reverses its motion. Since the



4160

piston could pick up significant velocity due to the high pressure (more than 20 inch/sec by the time it reaches the latching position), it may be difficult for a mechanical latch to function. In the event of a cook-off, the velocity of the piston just before latching will exceed this speed. To control the motion of the piston just before it latches into the in-battery position, a combination of the middle buffer and a step on the piston is provided as shown in figure 4. In addition, a gas chamber is utilized as shown by the dotted line in the figure.

The recoil cylinder will be about 6 feet long for a 155mm weapon system. If a one-quarter-inch thick piston is chosen for any of these concepts it may not be strong enough and induced vibration or buckling could result. If a three-eighths-inch thick piston is chosen, the effective area of the piston, due to its concentricity, is about 16 $in^2$. The effective piston area of existing weapons is about 10 $in^2$. However, the effective piston area can be controlled by choosing either a multi-cylinder (nonconcentric) configuration or a piston designed as shown in in figure 5. Thus, the effective area is not a serious design problem.

The concept based on figure 4 is more general and, therefore, it is chosen for further evaluation. However, the concepts based in figures 2 and 3 are special concepts based on figure 4.

## III MATHEMATICAL MODEL OF HYDROPNEUMATIC SOFT RECOIL MECHANISM

The hydropneumatic soft recoil concept is shown in figure 4. This schematic figure is not drawn to a scale. This concept is a concentric recoil mechanism and the piston, its only moving part, is shown in its latched (in-battery) or initial position. First, fluid fill-up takes place. Thus, nitrogen bottles can be attached to the gas chambers for pressurization. The pressure inside the cylinder will not exceed 3,000 psi, when the piston is in its latched position. In a normal run, the piston is moved forward by the pressurized fluid to a distance less than L2, returns past the latch by virtue of breech force, and finally is pushed forward to the latch by fluid pressure.

In figure 4, L indicates length, R indicates radius, and P indicates pressure. The recoil cylinder consists of four chambers. The cook-off chamber is inside the left end of the cylinder and the misfire chamber is inside the right end of the cylinder. The latching chamber is next to the cook-off chamber. A two-ring piston is used to control its motion just before latching. The normal chamber



461

is in front of the misfire chamber.

Mathematical models have been developed for both configurations (figures 3 and 4). However, because of space limitations, only the generalized model will be discussed. The application of Newton's second law yields the equation of motion for the piston:

$$\ddot{x} = (RP - HI * PBF\ (xx))/M_r - g \sin \theta \tag{3.1}$$

$$\dot{x} = \int \ddot{x}\ dt \tag{3.2}$$

$$x = L_3 + \int \dot{x}\ dt \tag{3.3}$$

where x = location of piston, $\dot{x}$ = velocity, $\ddot{x}$ = acceleration, RP = rod pull, PBF (xx) = breech force function, HI=0 or 1 to indicate inaction or action of breech force, $M_r$ = mass of recoiling parts, g = gravity, θ = elevation of gun, and $L_3$ = length of rear (cook-off) chamber. The rod pull can be written as

$$RP = s * (1.345p_r + 2.97p_f) + p_r A_3 + p_m\ (A_7 - A_4) - p_f A_7 \tag{3.4}$$

The first term indicates frictional force due to seals (5) and the subscripts r, m, and f signify the rear, middle or latching, and front chambers, respectively. Also $A_3 = \pi\ (R_3{}^2 - R_5{}^2)$, $A_4 = \pi\ (R_3{}^2 - R_4{}^2)$, $A_5 = \pi\ (R_4{}^2 - R_5{}^2)$, and $A_7 = \pi\ (R_7{}^2 - R_4{}^2)$.

The fluid volume (applicable to the driving force of the piston) at any time can be written as

$$v = v_g^o \left(\frac{p_2{}^o}{p_2}\right)^{1/r} + v^o{}_h\ e^{(p_2{}^o - p_2)/B} - (F v_f + J v_r + M v_m) \tag{3.5}$$

where $p_2$ = pressure in the normal chamber,

$$v_f = v_f{}^o\ e^{(p_f^o - p_f)/B} - f \int_o^t Q_f\ dt \tag{3.6}$$



4462

$$v_r = v_r^o \, e^{(p_2^o - p_r)/B} - j \int_0^t Q_r \, dt \tag{3.7}$$

$$v_m = v_m^o \, e^{(p_2^o - p_m)/B} + j \int_0^t Q_r \, dt - m \int_0^t Q_m \, dt \tag{3.8}$$

$$v_f^o = v_{fs} + (A_7 + A_f) L_1, \; v_{fs} = \pi L_1 \, (R_8^{\,2} - R_1^{\,2}),$$
$$A_f = \pi \, (R_1^{\,2} - R_7^{\,2}) \tag{3.9}$$

$$v_r^o = v_{rs} + (A_3 + A_r) L_3, \; v_{rs} = \pi L_3 \, (R_8^{\,2} - R_2^{\,2}),$$
$$A_r = \pi \, (R_2^{\,2} - R_3^{\,2}) \tag{3.10}$$

$$v_m^o = v_{ms} + L_3 \, (A_7 + A_m), \; v_{ms} = (L_5 - L_{4/2}) \, \pi$$
$$(R_8^{\,2} - R_6^{\,2}), \; A_m = \pi \, (R_6^{\,2} - R_7^{\,2}) \tag{3.11}$$

$$v_2^o = v_{2s} + L_2 \, (A_7 + A_m), \; v_{2s} = (L_2 + L_4/2) \, \pi$$
$$(A^2 - R_6^{\,2}) \tag{3.12}$$

$$Q_r = C_d A_r \left( \frac{2g \, |p_r - p_m|}{\rho} \right)^{1/2}, \quad Q_f = C_d \, A_f \left( \frac{2g \, |p_f - p_2|}{\rho} \right)^{1/2}$$
$$Q_m = C_d \, A_m \left( \frac{2g \, |p_2 - p_m|}{\rho} \right)^{1/2} \tag{3.13}$$

F, J, M, f, j, m, and s are all logical parameters and will be defined later. The symbols A, v, Q are used for area, volume, and flow discharge rates, respectively. The coefficient of discharge for fluid flow through the orifice is denoted as $c_d$. The physical volume based on solid boundaries can be written as

$$v = v_f^o + v_r^o + v_m^o + v_2^o - (Fv_f + Jv_r + Mv_m) + A_5 (x - L_3) \tag{3.14}$$



463

One can obtain the following equation for $p_2$ by equating the rate of change of the two volumes described above:

$$p_2 = \Big\{ -\left[A_5+F(A_7+A_f)-J(A_3+A_r)+M(A_4+A_r-A_m)\right] \dot{x} + F\left[v_f{}^o e^{(p_f{}^o-p_f)/B} \dot{p}_f/B+fQ_f\right] + J\left[v_r{}^o e^{(p_2{}^o-p_r)/B} \dot{p}_r/B+jQ_r\right] + M\left[v_m{}^o e^{(p_2^o-p_m)/B}\dot{p}_m - j\,Q_r+mQ_m\right]\Big\} \Big/ \left[\frac{v_g{}^o}{rp_2}\,(p_2^o/p_2)^{1/r} + e^{(p_2{}^o-p_2)/B}\,v_h{}^o/B\right] \quad (3.15)$$

Similar treatment can be given for the rate of change of pressures inside the rear and front buffers. The volume of oil at any time in either buffer can be written as

$$v = F\,v_f + J\,v_r \quad (3.16)$$

The physical volume based on boundaries is

$$v = F\left[v_{fs}+(A_7+A_f)(L-x)\right] + J\quad v_{rs}+(A_3+A_r)\,x \quad (3.17)$$

where $L = L_3+L_5+L_2+L_1$

The rate of change of pressure in either buffer can be derived by equating the rate of change of volumes and the result is given below.

$$\dot{p}_r \text{ or } \dot{p}_f = \frac{\left[F(A_7+A_f)-J(A_3+A_r)\right]\dot{x}-(FfQ_f+JjQ_r)}{Fv_f{}^o/B\; e^{(p_f{}^o-p_f)/B} + Jv_r{}^o/B\; e^{(p_2{}^o-p_2)/B}} \quad (3.18)$$

Of course, these equations are valid for eithe $0 \leqslant L_3$ or $L_3+L_2 \leqslant L_3+L_2+L_1$.

A similar equation can be derived for pressure in the latching chamber. The volume of oil in the middle buffer is



464

$$v = M\left[v_m{}^o e^{(p_2{}^o - p_m)/B} + j\int_o^t Q_r\, dt - m\int_o^t Q_m\, dt\right] \quad (3.19)$$

The physical volume of the latching chamber can be expressed as

$$v = M\left[v_{ms} + (A_4 + A_r)(L_3 - x) + (A_7 + A_m)\, x\right] \quad (3.20)$$

The rate of change of pressure in the latching chamber can now be expressed as

$$\dot{p}_m = \frac{M\left[(A_4 + A_r - A_7 - A_m)\,\dot{x} + jQ_r - mQ_m\right]}{v_m{}^o/B \quad e^{(p_2{}^o - p_m)/B}} \quad (3.21)$$

The logical parameters, for many of the equations in this section are given below:

$$s = 1 \text{ for } \dot{x} < 0, \quad s = -1 \text{ for } \dot{x} > 0$$

$$F = 1 \text{ for } x > L_2 + L_3;\ F = 0,\ A_f = 0 \text{ for } x < L_2 + L_3$$

$$J = 1 \text{ for } x < L_3;\ J = 0,\ A_r = 0,\ x > L_3$$

$$j = 1,\ f = 1 \text{ for } p_f \text{ or } p_r > p_2;\ j = 1 \text{ for } p_r > p_m$$

$$j = -1,\ f = -1 \text{ for } p_f \text{ or } p_r < p_2;\ j = -1 \text{ for } p_r < p_m$$

$$M = 1 \text{ for } L_3 - L_4 \le x \le L_3 \text{ and } M = 0, \text{ otherwise}$$

$$m = 1 \text{ for } p_m > p_2 \text{ and } m = -1 \text{ for } p_m < p_2 \quad (3.22)$$

The equations derived in this section, represent normal fire, cook-off and misfire conditions. The finite-element calculations as well as the one-dimensional analytical results of figure 2 indicate that the deflections of the recoil cylinder and sleeve are quite small and are within engineering or manufacturing tolerances. The governing equations for deflection of the recoil cylinder and the sleeve are second-order ordinary differential equations. The rate of change of pressure equations (3.15), (3.18), and (3.21) also will be different if deflections are taken into account. Since the deflections are small, the governing equations (with deflections) are not



4605

given here. The equations derived above, represent a system of six first-order ordinary differential equations. These are all coupled to be solved simultaneously.

## IV RESULTS

The hydropneumatic soft recoil conceptual sketch is shown in figure 4. The mathematical model for its operational characteristics is given in section 3. Since standard techniques can be utilized (the governing equations are not complex), the system of coupled ordinary differential equations as well as logical parameters are programmed for the CDC 6500/660 system and solved by Continuous System Simulation Language (CSSL). The breech force (figure 7) is computed for a zone 7 charge, 155mm weapon system. Due to a lack of analytical functions, the tabular form of breech force is provided as input and interpolation techniques are utilized during the course of the calculations.

Following is the input data for a typical computer run:

| | | |
|---|---|---|
| $L_1$ = 20 in. | $R_1$ = 9.2684 in. | $p^o$ = 2,000 psi |
| $L_2$ = 30 in. | $R_2$ = 9.0484 in. | $v_g{}^o/v_h{}^o$ = 0.22 |
| $L_3$ = 20 in. | $R_3$ = 9.0284 in. | $\theta$ = 0 |
| $L_4$ = 8 in. | $R_4$ = 8.5 in. | $M_r$ = 25.9 lbs |
| $L_5$ = 20 in. | $R_5$ = 8.125 in. | A = 11 in. |
| $C_d$ = 0.9 | $R_6$ = 9.2684 in. | X(0) = 20 in. |
| D = 8.125 in. | $R_7$ = 9.2484 in. | B = 150,000 psi |
| $\rho$ = .033942 lbs-sec$^2$/in.$^4$ | n = 1.3 | |

The shape of the buffers are functions of x. These values are omitted due to lack of space. Tabular data is input for these functions and interpolation techniques are utilized.

Most of the firing, at least 99%, takes place in a normal manner. Such firings are termed a "normal run" and the designs are optimized for them. The free-run velocity of the piston before firing is given as 230 in/sec for the breech force shown in figure 7. The velocity of the piston for a normal run is shown in figure 8.



4166

The piston attained a velocity of 230 in/sec after traveling about 25 inches. Firing is initiated at this time. The piston's velocity is reduced to zero within one inch of travel. The piston reverses its motion and reaches a maximum velocity of 227 in/sec after traveling 5.8 inches. The piston continues its rearward coasting movement and attains a velocity of 57 in/sec upon arrival at its initial position. At this time the latch is reset. The piston travels slightly more than an inch into the rear buffer before its rearward velocity is reduced to zero. Finally, the piston is pushed forward by the pressurized fluid and reaches a speed of about 29 in/sec just before engagement of the latch.

The piston location, as a function of time, is also shown in figure 8. Forward travel requires 216 msec and rearward travel requires 177 msec. The piston completed its cycle in 423 msec.

The pressure, $p_2$, in the normal chamber, is the same as that in the misfire chamber and is shown in figure 9. The pressure drop is only 400 psi, although the piston (with a differential area of 19.6 $in^2$) travels about 25 inches due to the presence of gas in the system. The pressure inside the rear buffer is identical to the pressure in the normal chamber except for the short period of time that the piston is inside the buffer. During this period, there is a sharp rise in pressure to 2,504 psi followed by a fall-back to 1,647 psi. There is a similar behavior pattern for the pressure inside the latching chamber. In this instance, the pressure rises to 2,458 psi and falls back to 1,696 psi.

The rod pull is shown in figure 10. This has a low value during the free-run period, starting with a peak of 30,541 lbs and dropping slowly to 24,459 lbs. The initiation of breech force quickly raises the rod pull to 38,260 lbs at the time the piston reverses its direction. Thereafter, the rod pull rises slowly and reaches 47,800 lbs by the time the piston reaches its initial position. Finally, the rapid rise of pressure inside the buffer momentarily increases the rod pull to 78,018 lbs, which then drops back to a nominal value just before the piston is latched.

The following applies to a misfire simulation. The breech force is not applied although the piston velocity is 230 in/sec. The piston continues its forward travel and reaches a peak velocity of 250.15 in/sec just before it enters the front buffer. A braking force is applied by the misfire chamber and the piston's velocity is reduced to less than 1 in/sec by the time it travels 16 inches inside the buffer. At this point the piston begins to oscillate.



467

Whether or not an external or internal spring is used to stop this motion, it is not expected to create any damage to the mechanism because of its very low velocity and friction. The time required for the piston to reach a velocity of 0.6 in/sec, is 472 msec. The pressures in the cook-off, latching, and normal chambers are identical. These pressures vary from an initial pressure of 2,000 psi to 1,366 psi and then drop to a level similar to the normal chamber pressure in a normal run as shown in figure 9. All pressures would have been very small (indicating cavitation and a vacuum) if not for the presence of gas. The pressure inside the misfire chamber is the same as the pressure in the other chambers until the piston enters the misfire chamber. Because the buffer's design is not optimized, the misfire chamber pressure rises to 7,000 psi and then drops to 2,321 psi. (Bear in mind a misfire is not a normal firing condition for a howitzer.)

Following are the cook-off results. The breech force is applied while the piston is at rest at its initial position. As expected, the piston starts rearward and reaches a peak speed of 342.9 in/sec within 12 msec and travels a distance of 2.3 in. Finally, the piston's rearward velocity is reduced to zero within 38.7 msec after traveling a distance of 7.32 in. Obviously, the calculated distance and time are small and a need for redesign of the buffer is evident. The pressure in the cook-off chamber is as high as 10,000 psi. Outside the chamber it is 2,146 psi. Because of these high pressures (the latch is not expected to operate at this time), the piston starts forward reaching a velocity of 49 in/sec and dropping to a low of 5.79 in/sec before hovering at about 20 in/sec near the latching position. During this forward travel the pressure in the cook-off chamber is as low as 1,390 psi, is slightly higher than 2,000 psi in the normal chamber, and is between these extremes in the latching chamber. The elapsed time for the entire cook-off cycle is 380 msec. Cook-off is less likely to occur than misfire. Its operational cycle is less gentle than that for normal fire.

## V SUMMARY

Various existing recoil mechanisms are reviewed and their drawbacks are discussed. The importance of RAM-D is stressed. Several new concepts are introduced having desirable characteristics such as: one moving part, highly compressible system much beyond the state of the art with liquids only, temperature compensating device, lower operating pressures with no cavitation or vaccuum, transmission of minimum forces to the supporting structure, ideal distribution of forces (symmetric and in line with the breech force), simplicity, and packaging of recoil and counterrecoil functions with-



4/18

in a single envelope (includes misfire and cook-off functions). All such characteristics are incorporated into a new concentric hydropneumatic soft recoil mechanism shown in figure 4.

Mathematical models are developed by use of Newton's law of motion, continuity equation, definition of bulk modulus for liquids, perfect gas law, and Bernoulli's equation. Virtually unlimited compressibility is proven and a method of compensating for temperature variations is presented. The breech force data for a 155mm weapon (including all propellant charge zones) is coupled to a system of six, first-order, ordinary differential equations. These are programmed for a digital computer and solved by CSSL. The operating characteristics are obtained not only for normal fire but also for cook-off and misfire. The deflections of the recoil cylinder and sleeve are found to be negligible. The pressure in each chamber, the trunnion force, and the location and velocity of the piston are obtained as a function of time. The concepts may not be feasible as "liquid only" systems. Although a recoil mechanism without a latching chamber would be short and light in weight, it would be very difficult to latch the piston without damaging the latching mechanism. A recoil mechanism having a latching chamber, a liquid-gas working medium, and employing a partial or a complete soft recoil principle shows promise of success. A parametric study is recommended after the buffers are redesigned. The objective of the redesign would be to raise the buffer pressures to the desired level as quickly as possible and holding these pressures constant until all motion ceases. As an alternative, a conventional recoil mechanism, with fluidic control (no moving parts) and sensor devices (currently being pursued), is recommended as a means to obtain constant rod pull for all zones of fire.

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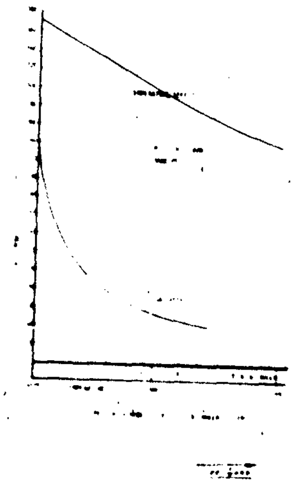
469



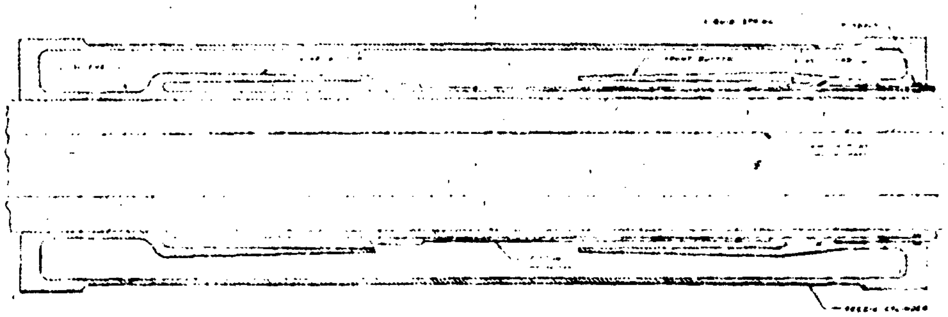


FIGURE 2. COMPRESSIBLE FLUID SOFT RECOIL CONCEPT

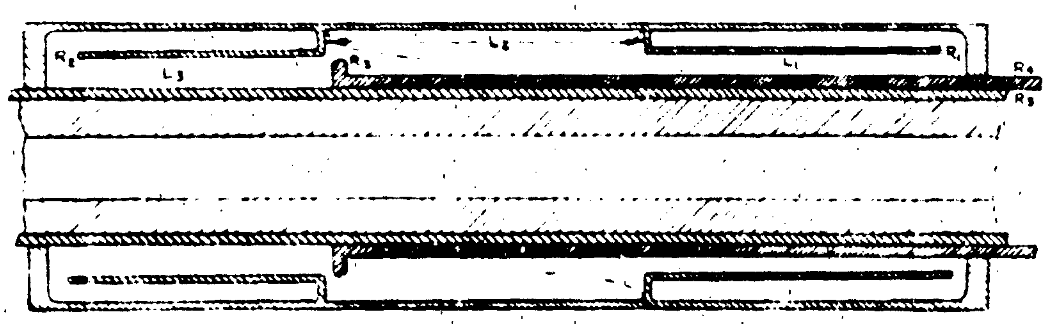



FIGURE 3 REVISED SOFT RECOIL CONCEPT





470

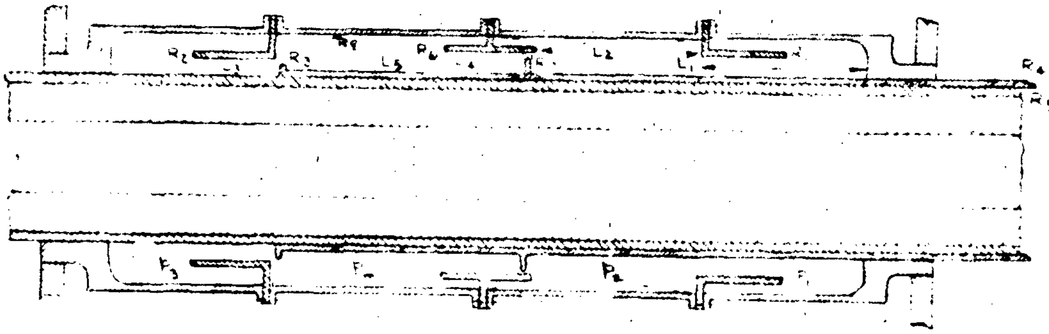


FIGURE 4 HYDROPNEUMATIC SOFT RECOIL CONCEPT

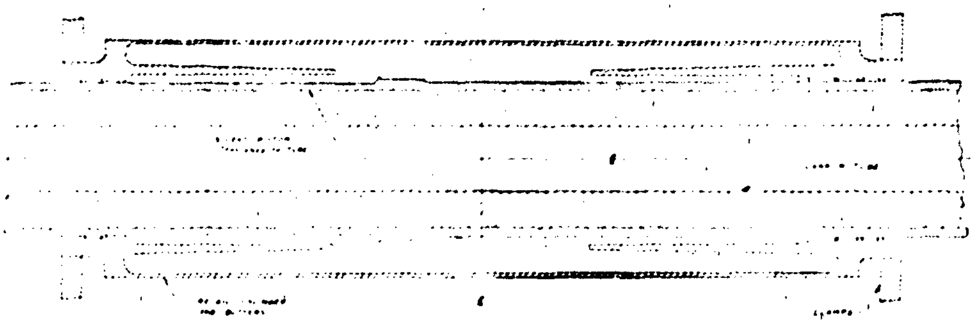
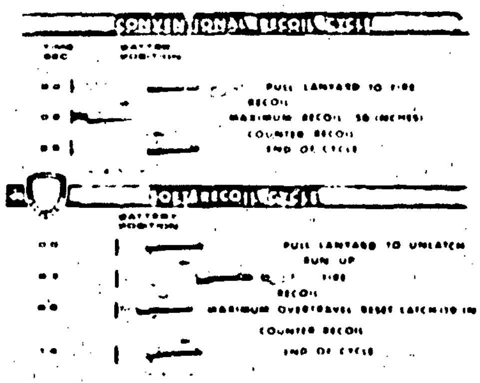


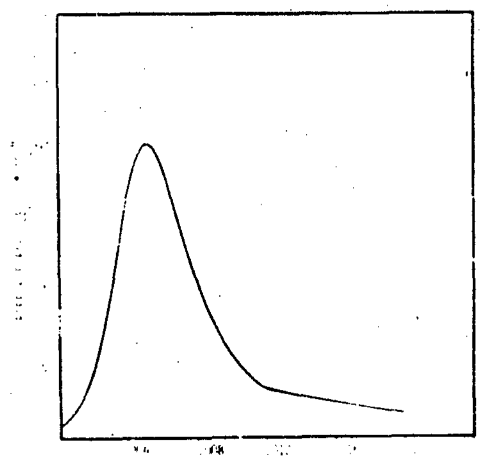
FIGURE 5 ALTERNATIVE PISTON DESIGN

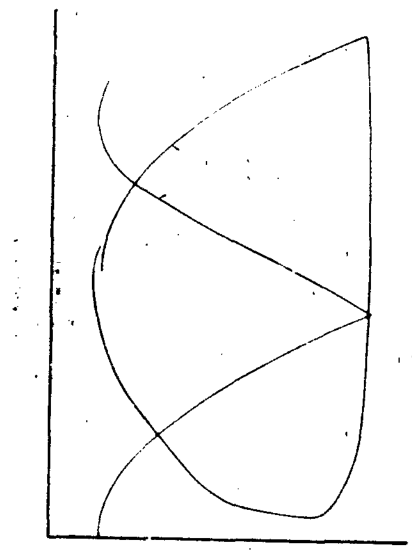


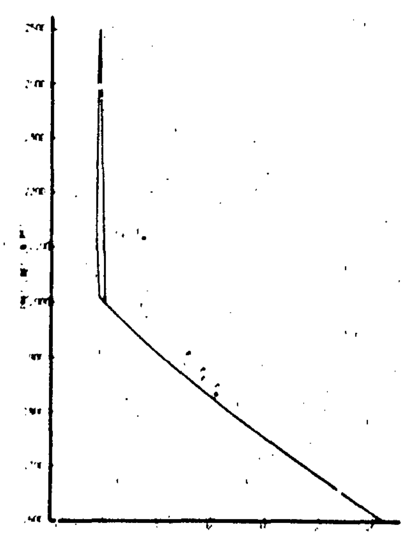


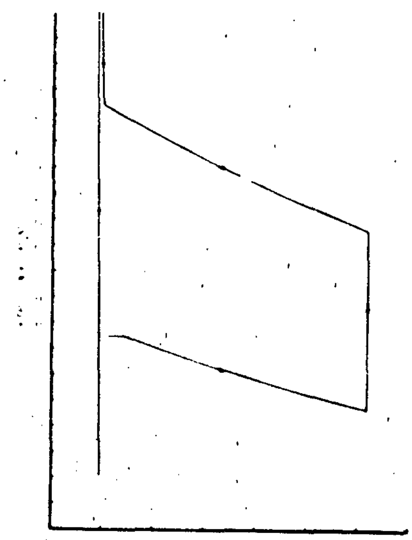



471













472