Feb. 13, 1962 E. W. HAILSTON ETAL 3,020,807

CONTROL DEVICE FOR GAS OPERATED FIREARM

Filed April 4, 1958 4 Sheets-Sheet 1

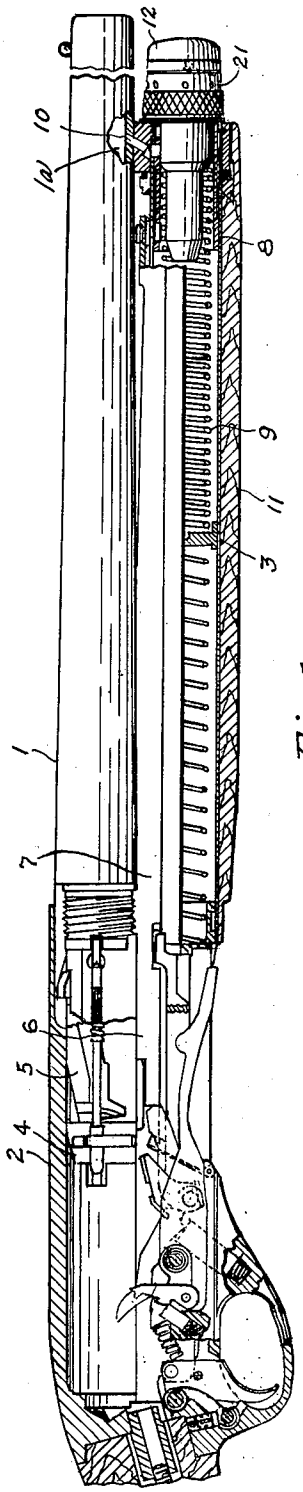


Fig. 1

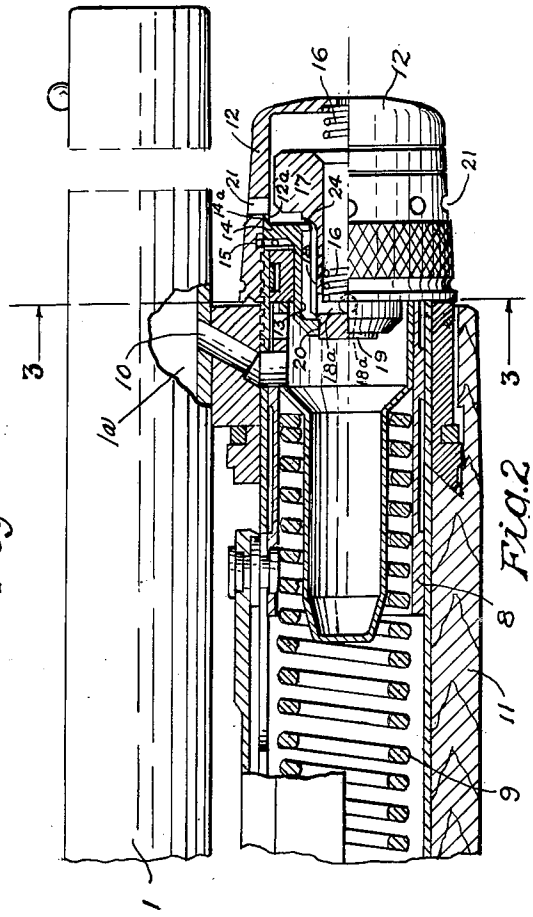


Fig. 2

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CONTROL DEVICE FOR GAS OPERATED FIREARM

Filed April 4, 1958 4 Sheets-Sheet 2

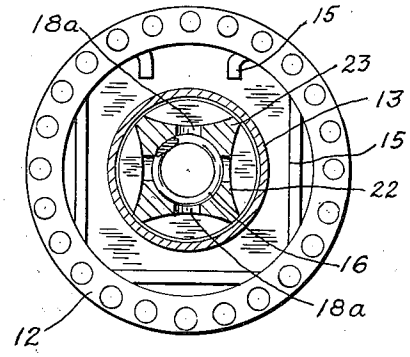


Fig 3

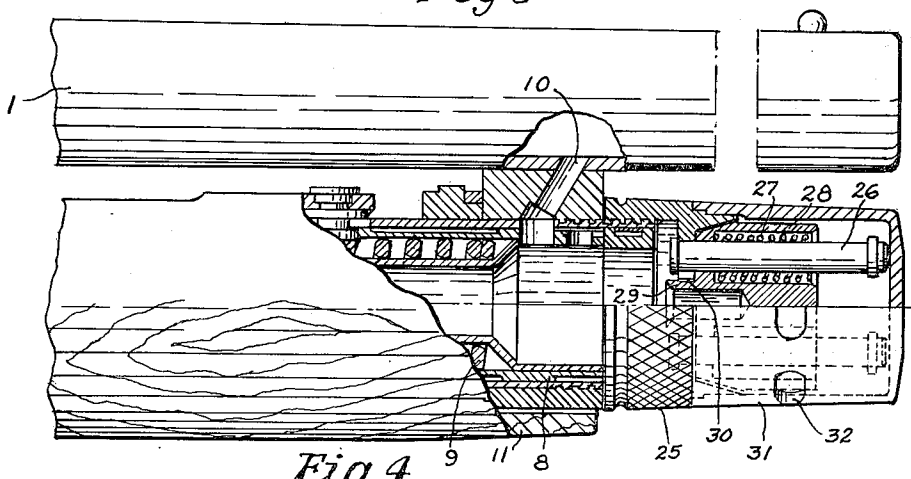


Fig. 4

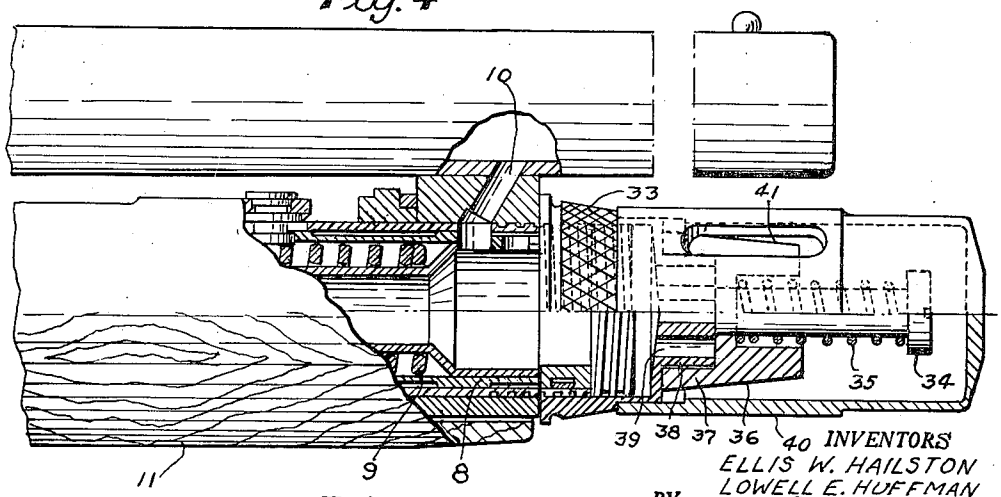


Fig. 5

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CONTROL DEVICE FOR GAS OPERATED FIREARM

Filed April 4, 1958

4 Sheets-Sheet 3

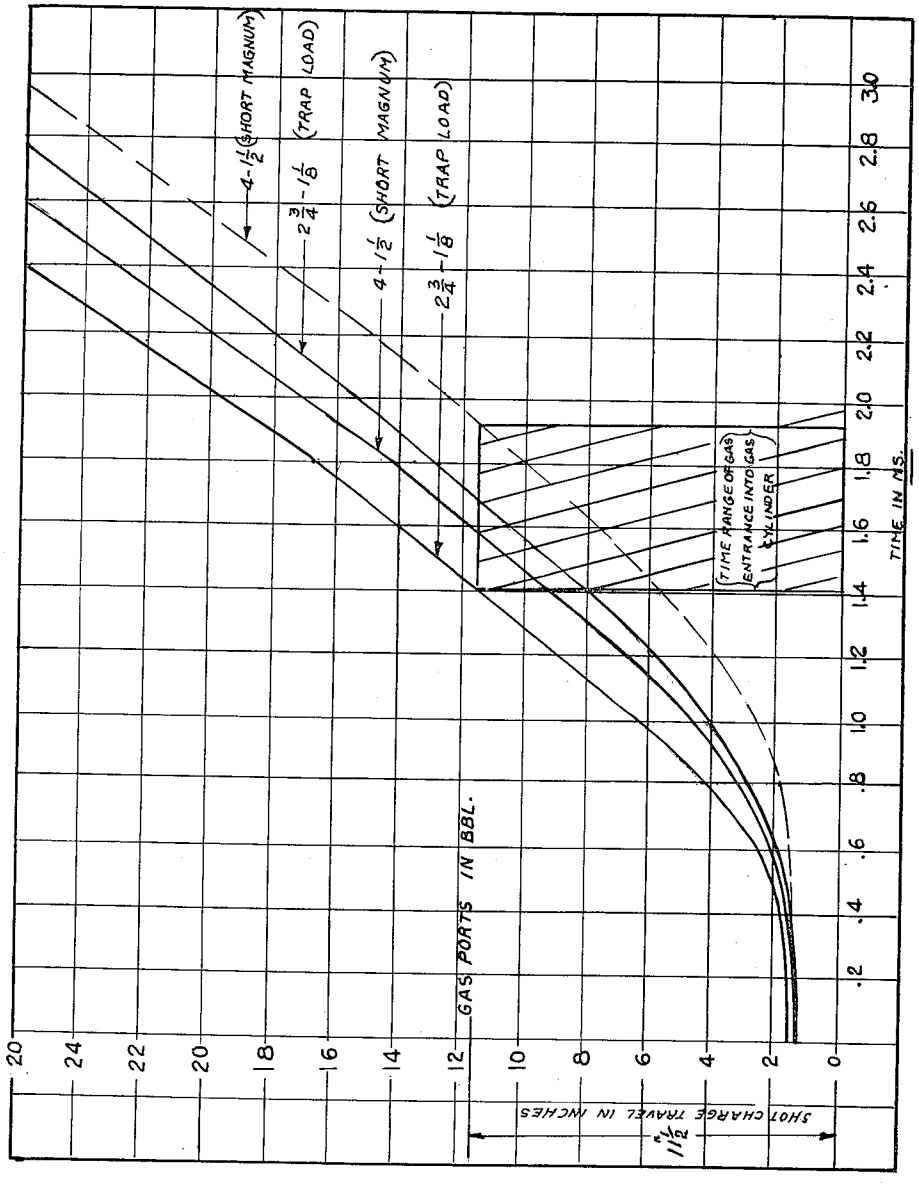


Fig. 6

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CONTROL DEVICE FOR GAS OPERATED FIREARM

Filed April 4, 1958 4 Sheets-Sheet 4

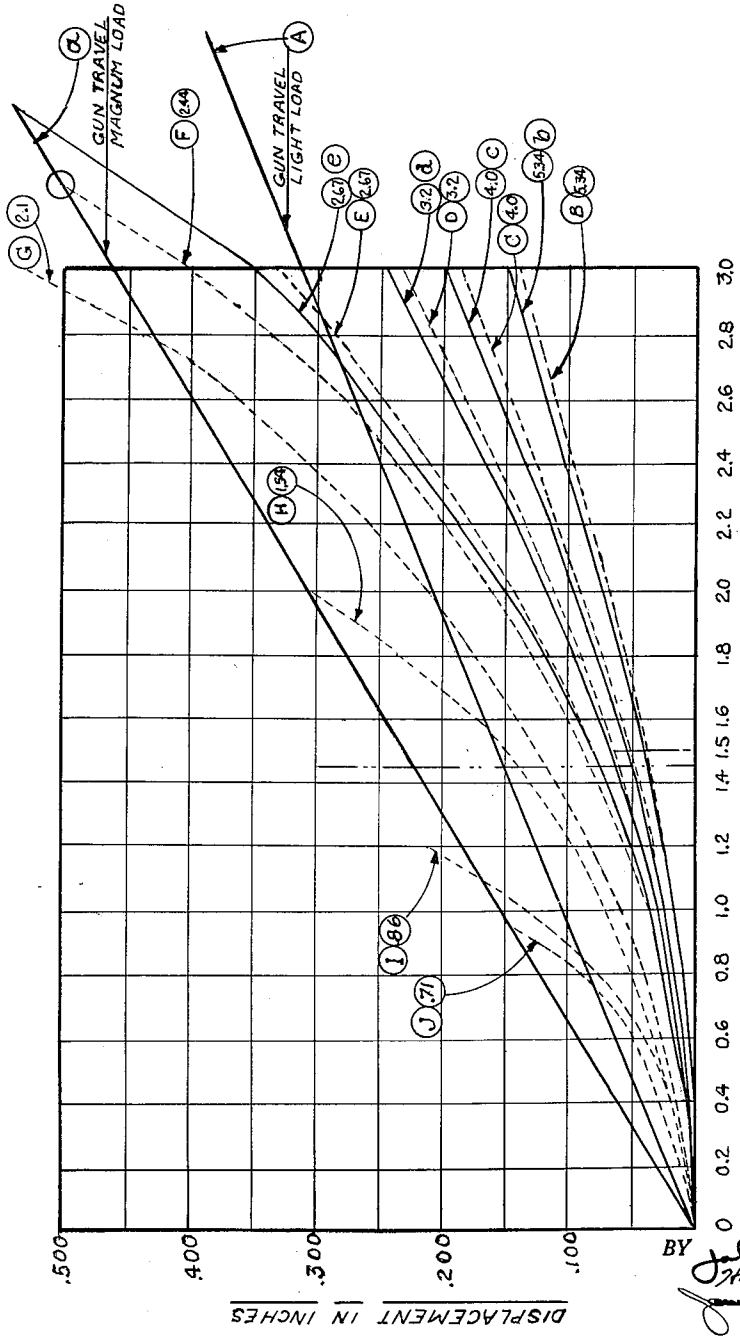


Fig. 7

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United States Patent Office 3,020,807 Patented Feb. 13, 1962

3,020,807

CONTROL DEVICE FOR GAS OPERATED FIREARM

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Filed Apr. 4, 1958, Ser. No. 726,518
6 Claims. (Cl. 89—193)

This invention relates to gas-operated autoloading firearms and has particular reference to a shotgun of that type.

In the utilization of autoloading firearms, it is frequently desirable to have available for use a selection of loads of various power. Thus, shotgun shells are commonly provided for trap and skeet shooting with comparatively light charges of powder and shot which function with minimum recoil and disturbance to the shooter. This permits a shooter to fire several hundred rounds in a single competitive program without fatigue and without developing a flinch due to excessive recoil. On the other hand, for such long range uses as pass shooting of ducks or the hunting of geese, the hunter expects to do comparatively little shooting, is usually heavily clothed, and recoil is not much of a problem. For this purpose, power is the significant factor and the practical limit on charges of shot and powder is the ability of the firearm to withstand the pressure and operating stresses. With manually operated firearms these divergent types of ammunition have only required the designer to provide sufficient strength to handle the maximum loads and he could be assured of adequate performance with light loads.

With autoloading weapons and particularly those which are operated by gas pressure, a serious problem is created, for the very force which is relied upon to produce the greater power is that which operates the action of the firearm. It follows that light loads produce a low value of operating force and that if the firearm is designed to function dependably with light loads, it will receive excessive operating force from heavy loads, perhaps to the point of causing parts breakage and certainly to the detriment of functioning in respect to such exactly timed operations as the feeding of shells from magazine to chamber.

Attempts have been made to alleviate this problem by the provision of spring loaded pressure relief valves which opened to release gas when excessive pressure was developed in the operating cylinder. In these attempts, however, proper account was not taken of the pressure time relationships existing in the ballistic system.

Particularly in shotguns and in some other firearms as well, the pressure developed in the barrel rises fairly abruptly and also drops off abruptly, necessitating that gas for operating the action be tapped off comparatively close to the breech and providing only a short interval of time in which gas may be so utilized. In some gas-operated designs and particularly in that to which this improvement is applied, the gas-operating cylinder is opened to atmosphere after a very short travel of the operating piston. Considering both of these factors, it will be apparent that the duration of pressure application to the operating system is very short and can be almost described as a single impulse.

It should be apparent that, if in such a system we depend upon a pressure relief valve in the gas cylinder to reduce pressures for heavy loads, the relief of pressure will come too slowly to be of much, if any, value in reducing the force applied to the piston. This is true because the impulsive force will be applied at substantially the same time to both the operating piston and the pressure relief valve. Although in theory the greater inertia of the operating piston would seem to delay its action long

enough to permit the relief valve to open, in practice this does not seem to be so.

Accordingly, if effective compensation for the effect of varying loads is to be obtained, some other means must be provided to operate the compensating device. We have found that one of the inevitable results, which accompanies the use of a larger shot charge, larger powder charge, or other means of increasing the power of a load, was described by Newton in his fundamental laws of motion. If the charge is expelled with greater force, the force of recoil is greater and since this reaction is a consequence of motion, it follows that the gun will commence to recoil as soon as the shot charge is moved and will have recoiled a substantial distance by the time the shot charge passes the gas port and allows gas to pass into the operating cylinder.

We contemplate that the best method of controlling the operating force applied to such a system is to utilize the force of inertia resulting from such recoil to control a compensating valve. Accordingly, we provide an inertia member, which may be integral with the compensating valve, and which tends to remain fixed in space as the gun recoils relative thereto. The mass of this member and the distance which the gun must move relative thereto to open the compensating valve are so related that the valve will not open under the recoil forces associated with light loads and will be completely open before the shot charge of a heavy load reaches or passes the gas port. A return spring is provided to act against the inertia member and, in effect, to weigh the recoil force acting on the system. Preferably, the valve should be so arranged that it is balanced or has so little area exposed to gas pressure as to be comparatively independent of directly exerted gas pressure effects.

By utilizing these principles we can provide a system in which the compensating valve is substantially fully open, when a heavy load is fired, before gas is admitted to the operating cylinder and in which the peak pressure and impulse imparted to the operating piston are substantially reduced. With light loads, however, the compensating valve will remain closed, and the full available force will be applied to the operating piston.

The exact nature of the invention as well as other objects and advantages thereof will be more fully set forth in the following specification referring to the attached drawings in which:

FIG. 1 is a longitudinal sectional view showing a shotgun to which our invention has been applied.

FIG. 2 is an enlarged fragment of FIG. 1 illustrating the preferred embodiment of our invention.

FIG. 3 is a partial cross-sectional view on the line 3—3 of FIG. 2.

FIG. 4 is a view similar to FIG. 2, showing a differently proportioned embodiment of our invention.

FIG. 5 is another view similar to FIG. 2, showing a still differently proportioned embodiment of our invention.

FIG. 6 is a graphical diagram illustrating the time versus displacement of the shot charge relative to the breech of the gun for a variety of commercially available shot shells from which the time of passing of the shot charge beyond the gas ports can be derived.

FIG. 7 is a graphical diagram on which the travel of the gun in recoil and the travel of the compensating valve piston, both with relation to a fixed point in space, have been plotted against time for a variety of conditions.

Referring to the drawings, there is shown in FIG. 1 a gas-operated autoloading shotgun conforming to that shown in the copending United States application of Crittendon, Hailston, Haskell, Kelly, and Leek, Serial Number 582,153, filed May 2, 1956, entitled "Autoloading Firearm" now Patent No. 2,941,450, issued June 21, 1960, entitled "Gas Operating Mechanism for an Autoloading

Firearm," and also in the copending United States divisional application of Crittendon et al., Serial Number 685,304, filed September 20, 1957, entitled "Autoloading Firearm" now Patent No. 2,891,341, issued June 23, 1959, entitled "Securing Means for Fore-end of Autoloading Firearm," except for the application thereto of our improved device for compensating for the power of different loads. Specifically, this shotgun comprises a barrel **1** and a receiver **2**. Secured to the receiver and extending forwardly therefrom in parallelism with the barrel is a combined magazine tube and operating cylinder **3**. Housed within the receiver and more fully described in the above-identified application are a breech bolt **4**, locking block **5**, and breech bolt carrier **6**. This carrier **6** is coupled by action bars **7** to a gas piston **8** which is slidably disposed in the operating cylinder **3** and which acts against a breech closing spring **9**. Gas ports **10** in the barrel communicate between the bore **1***a* of the barrel **1** and the operating cylinder **3** and serve when they have been passed by the shot charge and over powder wads to admit gas to the forward end of the operating cylinder. In the application above referred to and in the patent to Simmons, No. 2,814,972, issued December 3, 1957, a manually adjustable valve device is provided in the cap closing the forward end of the operating cylinder to permit the manual selection of a desired size of vent bleeding gas to atmosphere from within the operating cylinder as a means of compensating for loads of different power. The usual wooden fore-end grip **11** serves to enclose the reciprocating action bars and the magazine tube. The construction thus far described is that of the copending applications above referred to, and it is functionally adequate, when the proper adjustments are made. The construction of the prior application does, however, require a specific manual adjustment when a change is made from light loads to heavy loads and vice versa, and one of the principal objects of our invention is to avoid having to make any manual adjustments.

To that end, we have added as a substitute for the manually adjustable member of the applications and the patent referred to above an assembly shown in detail in FIGS. 2 and 3 comprising a cap **12** threadably secured to the forward end of the combined magazine tube and operating cylinder. This cap **12** serves as the basic framework for supporting the automatically compensating valve assembly. A sleeve **13** is fixedly supported in the cap **12** and a flange **14** thereon serves to partition the cap, being retained in place during handling by a snap ring **15** and in use by confinement between the end of the operating cylinder **3** and a shoulder **12***a* in the cap. Gas sealing at this junction is improved by a lip **14***a* on the flange **14** engaging an undercut shoulder **12***a* on the cap as shown in FIG. 2. Supported within the cap and sleeve and urged toward the breech of the gun by a spring **16** is a valve piston. The valve piston comprises, preferably in an integral unit, an inertia flange **17**, a shank **18**, and a valve stem **19**, the latter being received in a vent **20** in the sleeve **13**. A plurality of ports **21** are provided in the cap to permit the escape of gas admitted to the interior of the cap by the action of the valve. The shank **18** is generally of square cross-section but is machined with concave sides as at **22** to provide gas escape passages of increased area. Holes **18***a* in the shank permit gas to enter the center of the hollow shank **18** and further increase the effective size of the gas escape passages. The corners **23** of the shank provide bearing surfaces of reduced area sliding on the interior of the sleeve **13** which maintain freedom of action for the piston. A radius **24** at the junction between shank **18** and flange **17** directs the gas in smooth flow to the ports **21**.

The spring **16** acts to keep the valve closed. When the gun recoils from the firing of a shot, the valve piston tends to remain fixed in space and the valve stem tends to withdraw from the vent **20**. If the recoil is sufficiently severe, as weighed against the spring, the stem will be completely withdrawn from the vent **20** and gas will freely pass from the interior of the operating cylinder to the interior of the cap and thence through the ports **21** to atmosphere. As will be later discussed in more detail, there is a relationship existing between the mass of the valve piston, the force of the spring, and the length of the valve stem which should be observed for optimum performance.

In FIG. 4 we have illustrated a modified embodiment wherein the functionally similar components are differently proportioned and assembled. In this construction the cap **25** serves as the mounting for a pair of guide posts **26** upon which the springs **27** are confined and which guide the valve piston **28**. The valve piston is again a member having substantial inertia and is provided with a stem **29** coacting with a vent **30** in the cap **25** to provide the valving action. A housing **31** provided with ports **32** completes the assembly. As in the case of the other modification, the valve piston tends to remain fixed in space as the gun recoils and to open the vent **30** in those instances where the recoil is of a certain severity, as measured against the springs **27**.

In FIG. 5 we have presented another embodiment wherein the cap **33** is provided with a single post **34** on which a spring **35** is confined and which guides a valve piston **36**. In this form the valve piston is formed to define a rearwardly extending skirt **37** which encloses a forwardly extending stem **38** on the cap **33**. Vents **39** in the cap **33** do not communicate with the interior of the cover **40** until the gun and cap **33** have recoiled a sufficient distance relative to the valve piston **36** to permit the skirt **37** of the piston to clear the stem **38**. Ports **41** in the cover permit the gas so released to escape to atmosphere.

Although all three units are governed by the same operational principles, we prefer the embodiment shown in FIGS. 1 to 3 because of its more compact and pleasing appearance and because the gas flow through the system avoids direct impingement or reversal of gas flow upon any surface which might tend to develop deposits of lead or powder fouling and interfere with free operation of the compensating valve. Flow directly parallel to a surface or tangentially deflected therefrom does not tend to build up deposits of sufficient magnitude to interfere with free operation.

In FIG. 6 the travel of the shot charge in inches for each of several representative commercially available 12 gauge shot charges has been plotted with respect to time. From inspection of these plots, it can be noted that the one ounce shot charge of one typical light trap load passes the point 11½ inches down the barrel where the gas ports **10** are located 1.42 milliseconds after firing. For one typical "Magnum" duck load the 1½ ounce shot charge passes the gas port **10** about 1.92 milliseconds after firing. Obviously, prior to the expiration of this period of time there can be no admission of gas to the gas operating system. It is also known that the gas pressure of a shotgun shell falls off rapidly after reaching its peak value and that in this particular system the gas is free to exhaust to atmosphere after a short travel of the operating piston. Thus, the force imparted to the operating piston comes almost as a single impulse and if any compensating device is to be effective it must have set itself to perform its compensating function by or prior to the time gas is admitted to the system. A compensating arrangement which functions solely on the basis of pressure in the operating cylinder is doomed to be ineffective since the operating piston receives the same impulse which acts on the compensating member and before the latter can function to effectively diminish the pressure in the cylinder substantially the full impulsive force will have been applied to the piston.

An inertia operated member is not subject to the delays in operation which apply to a gas operated member for inertia forces start to operate as soon as the shot

charge moves and are not delayed until the shot charge has moved nearly 12 inches. As is inevitable in any firearm, the reaction to the movement of the shot charge moves the gun and all parts rigidly connected thereto in the direction opposite to that of the shot charge. In this design these recoil forces are applied through the cap **12** to the spring **16** and by the spring are applied to the valve piston. Since the valve piston has, particularly in the flange **17**, considerable mass, it tends to remain fixed in space while the gun recoils and consequently lags behind the gun in recoil. As the load on the spring is increased the valve piston is accelerated to travel at substantially the same rate as the gun and may tend to overtake the gun as the spring recovers from its initial compression.

The condition which should be achieved is one in which the valve will not be opened by the recoil forces associated with a light load but will be consistently and fully opened before the shot charge of a heavy load passes the gas port. The design variables which can be manipulated to reach this result are the mass of the valve piston, the pre-load and rate of the spring acting thereon, and the length of the stem **19** which closes the vent **20**.

FIG. 7 illustrates the effect of some of these design variables. In this figure the curves "*a*" and "A," respectively, show gun displacement in recoil in inches plotted against time in milliseconds for a representative light trap load and another representative "Magnum" duck load, which can be identified in FIG. 6 as the one in which the shot charge passes the gas port at 1.6 milliseconds after firing. Although not strictly correct, within the very short time interval considered between firing and 3.0 milliseconds, the gun may be thought of as being in virtually free recoil compressing the shooter's clothes and the softer flesh of his shoulder, and the gun recoil curves can be considered as straight lines at a rate of 109 inches per second for the light load and 156 inches per second for the "Magnum" load. The dotted line curves identified by the uppercase letters show valve piston travel versus time for the following combinations of mass and spring preload when light loads are fired.

Curve	Valve Piston Mass, grams	Spring Preload, lbs.	Mass/ Preload
B	80	15	5.34
C	80	20	4.0
D	80	25	3.2
E	40	15	2.67
F	34	14	2.44
G	40	19	2.1
H	40	26	1.54
I	11.2	13	.86
J	5.68	8	.71

The full line curves identified by lower case letters *b* through *e* show valve piston travel versus time for some of the same combinations of mass and spring preload when the "Magnum" load is fired.

Curve	Valve Piston Mass, grams	Spring Preload, lbs.
b	80	15
c	80	20
d	80	25
e	40	15

Considering, first, the firing of a light trap load, a vertical dot-dash line on FIG. 7 at 1.45 milliseconds shows that in 1.45 milliseconds after firing a light trap load the gun had recoiled .15 inch (intersection with curve A). In the same period, a valve piston of 40 grams mass engaged by a spring preloaded at 15 pounds had recoiled .075 inch (intersection with curve E), or a relative separation of .075 inch. With this combination, it requires a length for the stem **19** of at least .075 inch to avoid opening of the compensating valve when the light trap load is fired. Locating the intersection of curve B with this same vertical line, we can establish that doubling the mass of the valve piston to 80 grams, other conditions remaining the same, results in the valve piston lagging further behind the recoil of the gun. With this combination the valve piston moved only .036 inch and there was a relative separation of .114 inch. In this situation, the stem **19** would have to be at least .114 inch in length to avoid opening of the compensating valve when a light trap load is fired. Increasing the mass of the piston still further increases the relative movement and the length of the stem while increasing the preload on the spring or decreasing the mass of the piston decreases the relative movement and results in permitting the use of a shorter length for the stem **19**.

Considering next the firing of a "Magnum" load, reference may be made to the vertical line on FIG. 7 at 16 milliseconds. In this period the gun in which the "Magnum" load was fired had recoiled .245 inch (intersection with curve *a*) and, as seen by the intersection with curve *e*, the 40 gram valve piston had moved .095 inch. There had been therefore a relative movement of .150 inch and if the length of stem **19** had been .075 inch, as suggested above, the compensating valve would have been open by .075 inch at the time the shot charge passed the gas ports in the barrel and gas began to feed into the operating cylinder. This amount of opening of the valve is sufficient to materially reduce the force applied to the operating piston and avoids the application of unnecessary shock loads to the breech locking mechanism, etc. The same effects may be noted here for increasing the mass of the valve piston and/or varying the preload on the spring. Although increasing the mass does increase the degree of relative movement, it requires an increase in the length of the stem **19** if the valve is to remain closed on the light loads and the net increase in opening for a "Magnum" load is not of much significance. For example, doubling the mass to 80 grams results in reducing the valve piston travel to .048 inch (intersection with curve *b*) and the relative movement is increased to .197 inch. However, when this mass was used with the light load it was noted above that a stem length of .114 inch was required, and with this amount deducted from .197 inch of relative movement, it will be seen that the valve will have been opened by only .083 inch a relatively insignificant increase in the opening of .075 inch which can be achieved with 40 gram mass and which has been shown to be adequate compensation.

It must be understood, however, that mass and spring force alone do not provide adequate control over the operation of this system but must be used in combination with a valve member having a stem of such length that, although a light load does produce movement of the valve member, it will not open a passage therethrough. Similarly, the stem should be of such length that the valve will be definitely opened by the recoil incidental to firing a heavy load.

To insure that the valve will stay closed with the light loads, we prefer to utilize a small excess in stem length. Thus, where .075 inch is an indicated minimum, with the 40 gram mass of the example above, we have found it desirable to use about .090 inch stem length.

Consideration of these curves should establish the advantage in terms of speed of response of the lighter valve pistons and at the same time establish that there are practical lower limits for both mass and preload in the spring. The mass of the valve piston in grams divided by the spring load in pounds furnishes an arbitrary numerical index by which various combinations can be compared. With a valve stem length of .090 inch, it is indicated that a minimum value for this ratio is about 2.1 and with a stem of substantially zero length, the minimum ratio is indicated to be about 1.5. These indications are, however, subject

to the qualification that, although a similar ratio may be arrived at by concurrently reducing both the piston mass and the spring preload, operation is not dependable unless the spring preload applied to the piston is sufficient to insure that the valve will not pop open as a pressure relief valve when used with low velocity loads. The limiting condition is that the force due to gas pressure of the light or low pressure loads in the gas cylinder (force equals maximum gas cylinder pressure in p.s.i. multiplied by the cross-sectional area of the valve stem) must be less than the force due to the spring preload in pounds which opposes the opening of the valve. For practical purposes, we regard a mass of about 25 grams as the lower useful limit, with a spring preload selected accordingly. If the valve stem diameter is in excess of .340 inch, there is danger with some loads of blowing the valve open as a pressure relief valve, leading to erratic functioning. If this diameter is less than about .280 inch, inadequate gas venting will be provided when the valve is open. We regard a diameter of about .330 inch as optimum.

With loads having recoil properties between those of the light trap loads and the "Magnum" loads which have been used as examples, the balancing of recoil force against the spring, controls the degree to which the valve will be open when gas is applied thereto. It cannot, however, be argued that this action will provide complete compensation between two relatively heavy loads having different recoil properties. Once the valve has been opened sufficiently to permit gas to flow therethrough, it will be noted that the gas can act upon the increased cross-sectional area provided by the heavy flanged section **17** of the valve piston. As a result, when the valve opens at all there is a tendency for it to open more completely which, to some extent, reduces the degree of compensation as between two heavy loads of different recoil properties.

Attention was previously directed to the concavity **22** in the shank **18** of the valve piston, to the radius **24** at the juncture between the shank **18** and the flange **17**, and to the narrow guiding surfaces **23**. The concavity **22** is desirable as a means of increasing the effective area of gas escape in a fashion roughly analogous to a steam nozzle, thereby preventing restriction in the flow of gas through the compensating valve. The radius **24** is desirable since thereby the path of the escaping gas may be directed outwardly without impingement on any perpendicular surface, such impingement causing a build-up of lead and powder residue which impairs operation. The narrow guiding surfaces **23** are guides for the movement of the valve piston. Wide surfaces which contacted a substantial area would tend to foul with residue and to impede operation, while narrow guiding surfaces approaching knife edges tend to be self-cleaning.

The considerations set forth above are directed principally to the preferred embodiment but are applicable to either of the other modifications in all respects except those relating to the particular shape of the valve member. Although only three embodiments have been shown, it should be noted that the design principles set forth will apply equally well to other geometrical forms and arrangements. Accordingly, it should not be understood that the invention is limited to the forms illustrated but reference should be had to the appended claims for a definition of the limits upon the scope of our invention.

We claim.

1. An automatic compensating valve for a firearm having a barrel, a gas operating cylinder in communication with the bore of the barrel and an operating piston reciprocable in said gas operating cylinder, said compensating valve comprising a cap for the end of said gas operating cylinder remote from the breech of the barrel, a partition member separating said cap into a first portion in continuous communication with said operating cylinder and a second portion in continuous communication with the atmosphere, a gas vent extending through said partition member and communicating between said first and second portion of the cap, an inertia member supported in said second portion of the cap for reciprocating movement therein along an axis substantially parallel to the axis of the bore of the barrel between a position nearest to the breech end of the barrel and a position more remote therefrom, a valve piston integral with said inertia member and disposed to close said vent when said inertia member is in its position nearest the breech end of the barrel and to open said vent only when the inertia member has been moved a substantial distance toward said more remote position, and spring means in said second portion of the cap and arranged to act between the cap and said inertia member urging said inertia member toward its position nearest the barrel, the force of said spring being so related to the mass of the inertia member that the recoil incidental to the firing of a light load in the barrel of the firearm will be insufficient to produce movement of the inertia member great enough to open the vent, while the recoil incidental to the firing of a heavy load will be sufficient to produce more movement of the inertia member than required to open said vent, the cross-sectional area of the vent opening and of the valve piston closing same being so related to the force of the spring that the valve will not blow open as a pressure relief valve when a light load is fired.

2. An automatic compensating valve, as defined in claim 1, said integral valve piston and inertia member having a mass not substantially less than 25 grams, said spring member applying a load to the inertia member of such value that the spring load expressed in pounds divided into the mass of the integral valve piston and inertia member expressed in grams yields a quotient of not less than 1.5 and the area of said vent being equivalent to that of a circle having a diameter between 0.280 inch and 0.340 inch.

3. An automatic compensating valve, as defined in claim 1, said valve piston being formed with a longitudinally extending stem which must be withdrawn completely from said vent by longitudinal movement of the integral valve piston and inertia member before the vent is opened to permit gas to escape, and said partition member being formed to define, within the cap for the end of the gas operating cylinder, a sleeve loosely surrounding that portion of the longitudinally extending stem within said second portion of the cap and leading to said vent.

4. An automatic compensating valve as described in claim 3, said combined valve piston and inertia member being guided for reciprocation in said cap for the end of the operating cylinder by a series of longitudinally extending circumferentially evenly spaced edges formed on a portion of said stem not extending into said vent and in parallelism with the axis of reciprocation of said member, said edges having line engagement with the inner surface of the sleeve formed in said partition member.

5. An automatic compensating valve as described in claim 4, the portions of said stem between said edges being indented in a concave conformation increasing the available space for gas flow between said stem and said sleeve in the cap structure.

6. An automatic compensating valve as described in claim 5, said inertia member providing in effect a head of increased diameter on said stem, said head and said stem blending into each other on a filleted surface serving to redirect laterally outwardly gas flowing along said stem from said vent.

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