

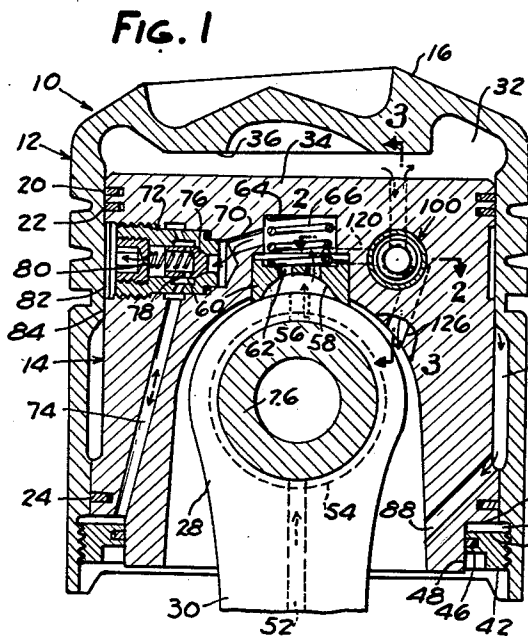
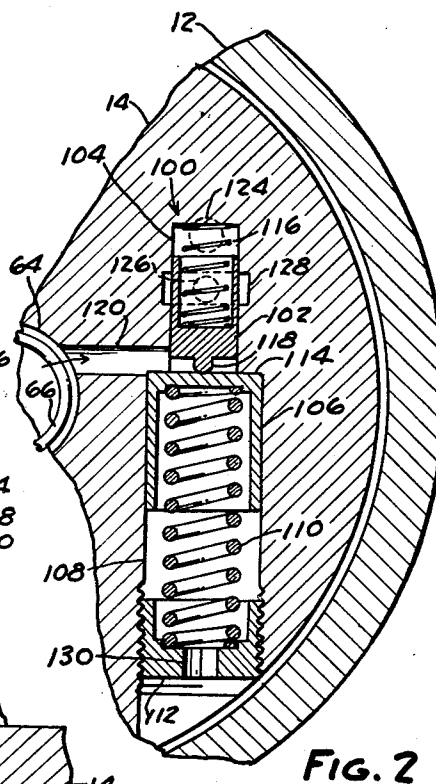
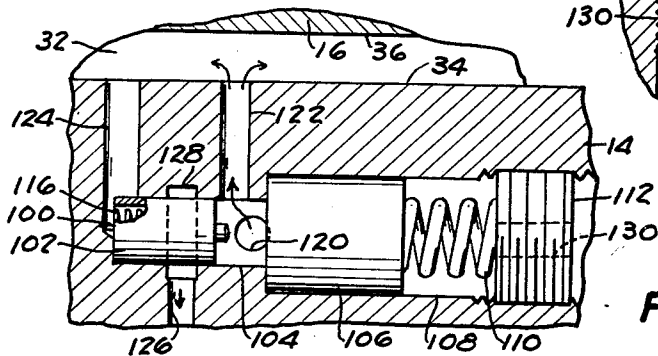
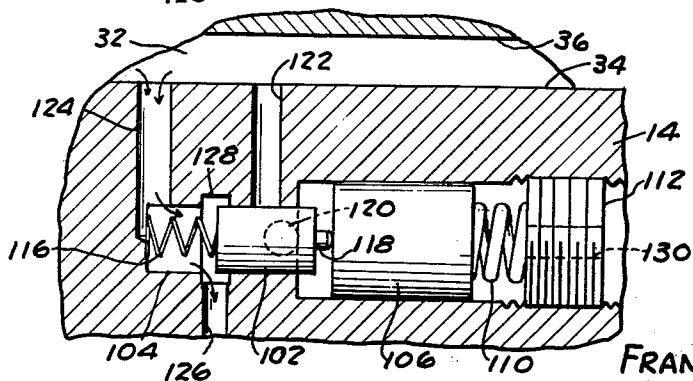
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PRESSURE REGULATING PISTON AND VALVE

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3,185,138

PRESSURE REGULATING PISTON AND VALVE
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 7 Claims. (Cl. 123—48)

This invention relates to improvements in pressure regulating valves and pistons, and more particularly to an improved combination supply and discharge valve for a reciprocating piston of the variable compression ratio (VCR) type disclosed in U.S. Patents Nos. 2,742,027, 3,014,468 and 3,038,458, all issued in the name of Wilfred P. Mansfield.

The aforesaid Mansfield patents disclose various forms of VCR pistons which may be employed in the cylinder or cylinders of an internal combustion engine for varying the compression ratio thereof by varying the clearance volume of the cylinder. By maintaining more uniform combustion chamber pressures, such pistons increase the efficiency of the engine and/or reduce the weight and structural cost of the engine per unit horsepower, in a simple, efficient and economical manner. In one form of the Mansfield VCR piston, an inner piston is connected in the usual manner to a connecting rod and carries an outer piston which is adapted to move axially to a limited extent relative to the inner piston. Internal clearance spaces are provided between the top and bottom ends of the inner and outer pistons which form upper and lower variable volume chambers adapted to contain an incompressible fluid such as oil. By controlling the flow of oil to and from these chambers, the movement of the outer piston relative to the inner piston in response to piston reciprocation and combustion chamber pressure is controlled for varying the clearance volume of the cylinder in which the piston reciprocates.

An object of the present invention is to improve the performance and reliability of a VCR piston.

Another object is to provide an improved combination supply and discharge valve for the upper chamber of the piston which responds rapidly to cyclic variations in the upper chamber oil pressure and which is speed compensated to thereby regulate fluid pressure in the upper chamber in a manner which insures more uniform maintenance of the maximum desired combustion chamber pressure by the piston.

Further objects of the invention are to simplify the design and assembly of the VCR piston and valve system therefor, to increase the structural integrity of the inner piston and/or to reduce cost.

Other objects, features and advantages of the present invention will become apparent from the following description taken in conjunction with the accompanying drawing wherein:

FIG. 1 is a vertical section through the axis of an improved VCR piston constructed in accordance with the present invention.

FIG. 2 is an enlarged fragmentary horizontal section taken on the line 2—2 of FIG. 1.

FIG. 3 is an enlarged vertical section taken on the line 3—3 in FIG. 1.

FIG. 4 is a vertical section corresponding to that of FIG. 3 but illustrating a different operative position of the valve assembly of the invention.

Referring to FIG. 1, there is shown by way of example a preferred embodiment of a VCR piston 10 adapted for reciprocation in the bore of the cylinder of a four-stroke cycle internal combustion engine. Piston 10 comprises an outer piston 12 which is carried on an inner piston 14. Outer piston 12 has a crown 16 which serves

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as the head of the piston 10 and forms the movable wall of the cylinder combustion chamber. Inner piston 14 is slidable within and axially of outer piston 12 and carries rings 20, 22 and 24 which have a fluid sealing, sliding engagement with the outer piston. Inner piston 14 is attached in a conventional manner by a wrist pin 26 to the upper end 28 of a connecting rod 30. An upper, variable volume chamber 32 for containing control fluid is formed between the upper surface 34 of inner piston 14 and the interior surface 36 of crown 16. A lower, variable volume chamber 38 for containing control fluid is defined by the annular space between the upper surface 40 of a retaining ring 42 threaded in outer piston 12 and a shoulder 44 of inner piston 14. Ring 42 carries a sealing ring 46 which has a fluid sealing, sliding engagement with a reduced diameter portion 48 of inner piston 14.

Due to the clearance spaces provided by chambers 32 and 38, outer piston 12 can move downwardly (as viewed in FIG. 1) relative to inner piston 14 until the underside 36 of crown 16 abuts upper surface 34 of the inner piston, and outer piston 12 can move upwardly relative to inner piston 14 until ring 42 abuts shoulder 44. This movement of the outer piston is controlled by automatically regulating the flow of an incompressible fluid into and out of chambers 32 and 38. The control fluid preferably comprises oil supplied to piston 10 from the usual pressurized lubricating oil supply of the engine via an oil passage 52 in rod 30. Passage 52 communicates at its lower end with a crankshaft oil supply passage (not shown) and runs upwardly to the upper end 28 of the rod where it communicates with an annular groove 54 encircling wrist pin 26 and leading to an outlet 56 in the rod. Oil under engine lubricating pressure flows upwardly from the rod passages into a cavity 58 of a cap 60 and then through an axial hole 62 in the cap into a spring chamber 64 in the inner piston. A spring 66 disposed in this chamber biases cap 60 downwardly into sliding sealed engagement with the upper end of rod 30. The flow of oil branches from chamber 64 into separate flow paths leading to lower chamber 38 and upper chamber 32, respectively.

The particular supply circuit to the lower chamber 38 illustrated herein is the subject of a copending application of William A. Wallace and Robert F. Pecha, Serial No. 290,706, filed June 26, 1963, assigned to the assignee of the present invention. In this lower chamber circuit, oil flows from chamber 64 through a passage 70 to a combination inlet and discharge valve assembly 72 which controls the supply of oil to and from a passage 74 running downwardly in inner piston 14 from valve 72 to chamber 38. When the volume of chamber 38 is increased by downward movement of outer piston 12 relative to the inner piston, a spring biased valve member 76 opens in response to the pressure differential between chambers 64 and 38 and permits oil to flow via passages 70 and 74 to chamber 38. When outer piston 12 moves upwardly relative to the inner piston, the resulting reversal of pressure closes valve 76 and oil is forced from chamber 38 via passage 74 to a port 78 in valve member 76 which leads to the hollow interior thereof, port 78 being open at all times.

The oil then flows at a predetermined restricted rate from the spring chamber behind valve member 76 via a restricted orifice 80, calculated to provide the necessary flow restriction, to an annular cooling groove 82. The oil then leaks downwardly to the crankcase via a clearance space 84 between the outer and inner pistons, another annular chamber 86 and a drain passage 88. Valve assembly 72 is thus designed to permit rapid downward movement of the outer piston relative to the inner piston,

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but controls movement in the opposite direction so that it occurs in small increments over several cycles.

In accordance with the present invention, the supply of oil from chamber 64 to upper chamber 32 of the VCR piston, and the release of oil from the upper chamber, is controlled by a combination admission and discharge spool valve assembly 100 disposed within inner piston 14 as illustrated generally in FIG. 1 and in detail in FIGS. 2 through 4. Valve assembly 100 includes a valve spool 102 which has a close sliding fit in a blind bore 104, and a follower spool 106 having a close sliding fit in a counterbore 108. Spool 106 is cup-shaped and receives a heavy spring 110 which abuts at one end against spool 106 and at the other end against an end plug 112 threadably secured in counterbore 108. Spring 110 normally urges spool 106 against a shoulder 114 at the inner end of counterbore 108. Valve spool 102 is also cup-shaped and receives a light spring 116 which abuts at one end against spool 102 and at the other end against the blind end of bore 104. Spring 116 biases spool 102 towards follower 106 so that a nipple 118 on spool 102 normally abuts the end face of follower 106.

The oil supply path to upper chamber 32 includes a radial passage 120 (FIGS. 1 and 2) running from chamber 64 to bore 104. The outlet end of passage 120 intersects counterbore 108 so that communication exists between chamber 64 and counterbore 108 in all positions of spool 106. A vertical passage 122 (FIGS. 3 and 4) runs from bore 104 to upper chamber 32. Communication between passages 120 and 122 via bore 104 is closed by valve 102 when it is in the positions shown in FIGS. 2 and 4, and axial movement of valve 102 to the position illustrated in FIG. 3 opens this communication.

The oil discharge path from upper chamber 32 includes a passage 124 (FIGS. 3 and 4) an inner piston 14 which runs from chamber 32 down to the blind end of bore 104. Passage 124 is offset horizontally slightly beyond the end of bore 104 so that communication exists between chamber 32 and the blind end of bore 104 in all positions of valve 102. An outlet passage 126 runs from an annular groove 128, which encircles bore 104 about midway between passages 122 and 124, down to the underside of inner piston 14 (FIG. 1). Communication between passages 124 and 126 via bore 104 is likewise controlled by valve 102, this communication being closed when valve 102 is in the positions of FIGS. 2 and 3 and being opened when it is in the position shown in FIG. 4.

Follower 106 is moved axially from the position of FIG. 3 to that of FIG. 4 by valve 102, and by the pressure oil from chamber 64 and passage 120 acting on the valve end of the follower. Since follower 106 and valve 102 have only a simple contact connection, the oil pressure in counterbore 108 between the spools reduces the effective force exerted by valve 102 on follower 106 but the full effect of this pressure acts on the follower. The space behind follower 106 is vented to cooling groove 82 by an Allen wrench socket 130 in end plug 112.

Valve operation

Assume that the chambers and passages are flooded with oil and that valve 102 and follower 106 are biased into abutment by their respective springs as shown in FIG. 2. As shown in FIGS. 1 and 2, supply oil from connecting rod 30 passes through chamber 64 and passage 120 into bore 104 between valve 102 and follower 106. If the upper chamber oil pressure acting on the spring end of valve 102 is less than the supply oil pressure, the supply oil pressure acting on the nipple face of valve 102 will move the valve against the force of spring 116 and until the edge of the spool valve uncovers the inlet to passage 122 (FIG. 3). Oil is then supplied via passages 120, bore 104 and passage 122 to the chamber 32. Filling of the upper chamber will continue until the pressure in chamber 32 equals the supply oil pressure. At this time,

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spring 116 moves the valve 102 to the closed position of FIG. 2.

When the upper chamber oil pressure exceeds the pre-load of the discharge regulating spring 110, this pressure acting via passage 124 against the spool valve moves it and follower 102 until the inlet 128 to discharge passage 126 is uncovered by the spring end of valve 102 (FIG. 4). The oil is discharged from chamber 32 via passage 124, bore 104 and passage 126 to the crankcase until the upper chamber oil pressure drops to the pre-load of spring 110. This sequence will be repeated according to the pressure cycle in chamber 32.

Piston operation

Assume that combustion chamber pressure is below a predetermined maximum value which VCR piston 10 is designed to maintain, and that chambers 32 and 38 are both filled with oil. As inner piston 14 decelerates in approaching top dead center position at the end of the exhaust stroke and then accelerates in the opposite direction on the intake stroke, the momentum of outer piston 12 tends to force it upwardly relative to inner piston 14, thereby raising the oil pressure in lower chamber 38 above that existing in spring chamber 64 and passage 70. The maximum lower chamber pressure at this time may range somewhere between 500 and 1000 p.s.i., as compared to a pressure of 200 to 250 lbs. p.s.i. in supply chamber 64, depending upon engine speed.

The oil pressure in lower chamber 38 is transmitted via passage 74 and valve port 78 to the body of oil contained in the chamber behind valve 76. Thus, as ring 42 is forced upwardly by outer piston momentum, oil pressure behind valve 76 builds up until it, plus the pressure of its spring, forces valve 76 closed. Thereafter, as lower chamber pressure peaks, a predetermined amount of oil is forced out of chamber 38 via passage 74, ports 78 to the interior of valve 76 and then through restricted orifice 80 to cooling chamber 82 from which it leaks as previously described.

The controlled leakage of a predetermined amount of oil from lower chamber 38 permits outer piston 12 to move a very small distance (a few thousandths of an inch) upwardly relative to inner piston 14. This in turn increases the volume of, and consequently reduces fluid pressure in, upper chamber 32. When oil pressure in passage 120 exceeds that in chamber 32, valve 102 opens (FIG. 3) to admit oil from supply chamber 64 to chamber 32. This added oil is trapped in chamber 32 by valve 102 when it closes (FIG. 2) due to a pressure reversal caused by the reversal of momentum forces as piston 10 approaches and passes through the bottom dead center position at the end of the intake stroke and at the beginning of the compression stroke. This prevents outer piston 14 from moving back downwardly relative to the inner piston until such time as the oil pressure in chamber 32 exceeds the predetermined pressure at which regulating valve assembly 100 is set to open against the pressure of spring 110. Hence, upward movement of outer piston 12 relative to inner piston 14 may occur for several cycles, terminating when the cylinder clearance volume has been reduced to the point where combustion chamber pressure reaches the predetermined maximum value which produces the valve-opening pressure in chamber 32. Thereafter, a state of relative equilibrium exists wherein the outer piston 12 moves up and down very slightly relative to the inner piston in each cycle, its means relative position being that producing the maximum combustion chamber pressure as predetermined by the setting of valve assembly 100.

If the combustion chamber pressure is suddenly increased, as by opening the engine throttle or increasing the load on the engine, valve assembly 100 is designed to rapidly discharge oil from upper chamber 32 so that outer piston 12 can move rapidly downwardly relative to inner piston 14, thereby increasing the cylinder clear-

ance volume and thus reducing combustion chamber pressure to the desired maximum value within a relatively few cycles of piston 10. Valve assembly 100 is designed to permit this downward relative movement of outer piston 12 more rapidly than restricted orifice 80 permits it to move upwardly in each cycle. This insures rapid relief of excessive combustion chamber pressure and limits oil pumping losses since the outer piston must gradually "creep" back up to regain its original position relative to the inner piston as compared to its more rapid pressure relieving movement.

This downward movement of outer piston 12 relative to inner piston 14 increases the volume of lower chamber 38, causing the oil pressure therein to drop to some value between 150 p.s.i. and zero p.s.i., or even to a negative pressure. As lower chamber pressure drops, valve 76 is forced open by pressure oil from chamber 64 to thereby supply lower chamber 38 with the required volume of oil to satisfy the pressure conditions.

The above operation assumes that VCR piston 10 is reciprocating at a constant speed. However, it has been found that if a conventional spring loaded pressure regulating valve rather than valve assembly 100 is used to control pressure in chamber 32, combustion chamber pressure tends to creep above the desired maximum limit as the speed of the piston increases. This undesirable run-out results at least in part from the effect of piston shell inertia upon the gas force in the cylinder combustion chamber and also upon the upper chamber pressure oil. Unlike a conventional piston which is rigidly attached to the connection rod by means of the piston pin, outer piston 12 is free to float through the limits of the designed-in stops 34 and 44, restrained only by a balance of forces between the upper chamber pressure oil and the gas pressure force in the cylinder. On the upward compression stroke, for example, the inertia of piston 12 resulting from the G-loading of its weight increases the gas pressure (force) in proportion to its physical mass and the speed of operation. This inertia effect tends to increase the controlled cylinder pressure with increasing speed.

However, valve assembly 100 of the present invention prevents this undesirable increase in the cylinder gas pressure due to its ability to compensate for the piston shell inertia effect, thereby resulting in more constant regulation of the cylinder pressure over the speed range. This speed compensating feature is obtained by the constant exposure of the end face of follower 106 to supply oil pressure in chamber 64. Since follower 106 has a larger working area than valve 102, the effective difference in these areas, e.g., the cross-sectional area of follower 106 minus that of valve 102, produces a net compensating force on valve assembly 100 opposite to the valve-closing force exerted by spring 110. This compensating force is always less than the spring force and is derived from the velocity-related variation in oil pressure in chamber 64. The supply oil experiences a proportional inertia loading in the same fashion as piston 12 due to the momentum forces exerted by the column of oil contained in connecting rod passages 52, 54 and 56. The resulting supply oil pressure variation is proportional to the square of engine r.p.m., and is additive to the lubricating system pressure which varies directly with engine r.p.m. Hence, the space between valve 102 and follower 106 is supplied with oil under a pressure which varies with engine r.p.m., increasing at higher speeds and decreasing at lower speeds, as well as with the cyclical variation in the position of piston. The latter variation occurs at double the frequency of, but substantially in phase with, the cyclical variation in combustion chamber pressure, both pressures peaking when piston 10 is in the vicinity of top dead center. Since follower 106 provides the force transmitting connection between spring 110 and valve 102, the oil pressure acting on follower 106 reduces the net force urging valve 102 from the upper

chamber discharge position of FIG. 4 to the closed position of FIG. 2. Hence, the force required to open valve assembly 100 is reduced as engine r.p.m. increases. This speed compensating action has been found effective to overcome the problem of non-uniform regulation of oil pressure in upper chamber 30 which in turn overcomes the problem of non-uniform regulation of the combustion chamber pressure by the VCR piston.

In addition to the aforementioned speed compensating feature which relates to the discharge function of valve assembly 100, the manner in which valve assembly 100 performs its supply function also improves the performance of the VCR piston by providing greater sensitivity and more rapid response to cyclic pressure variations in the upper chamber oil pressure. This is achieved by using a balanced force spool valve which provides large valve area openings for small valve motions and presents minimal resistance to pressure differential changes, thereby permitting more rapid filling as well as discharging of the upper chamber. Second, with one side of valve 102 exposed to the supply oil carried in the connecting rod, further advantage is taken of the inertia effects on this column of oil. At the beginning of the intake stroke when the cylinder pressure is lowest, the residual upward inertia of the supply oil in the connecting rod (reflected as a momentary increase in supply oil pressure) will rapidly force the valve 102 to the filling position of FIG. 3. Thus, the time necessary to open this valve is reduced, thereby increasing filling time for the upper chamber at the beginning of the cycle. At the beginning of the compression stroke, the residual downward inertia in the supply oil (reflected as a momentary decrease in supply oil pressure) will cause the spool valve to snap shut more quickly (to the position of FIG. 2), thereby trapping more oil in the upper chamber. This rapid response to pressure fluctuations coupled with the positive capillary fluid sealing provided by the closely fitted spool valve 102 also helps reduce the tendency of peak cylinder pressure to vary with speed. Since spool valve 102 permits large valve area openings for small valve travel, spring rate effects on cylinder pressure control are also reduced.

Valve assembly 100 also provides improved reliability due to decrease susceptibility to sticking and wear problems. First, valve 102 increases the tolerance of the valve system to dirt, chips and other foreign material. Inasmuch as sealing with spool valve 102 is accomplished by minimal clearances at the periphery of the valve, only those particles smaller than the valve clearance can cause sticking or sealing problems. Since diameter clearances in the neighborhood of .00012/.00016-inch are readily obtainable by using conventional production operations, the sensitivity of spool valve 102 to foreign materials carried in the supply oil is many times less than that of a poppet valve. Secondly, since the spool valve is forced balanced, it does not have to sustain physical loads such as those imposed on a poppet valve. Hence, valve seat pounding and wear are not a problem.

Because of the fewer number of pieces and compactness of valve assembly 100, fewer and smaller holes need to be bored into inner piston 14. This reduction in material removed from the inner piston plus the elimination of high assembly pre-stresses reduces the working stress level of inner piston 14, thereby increasing its structural integrity and potential life.

As is evident in the drawing, only ordinary machining practices are necessary with this design. There are no critical tolerances other than control of spool valve clearance. Bore 104 can be reamed and honed to size and valve 102 centerless ground with only a minor final lapping operation to achieve the necessary clearance. The two-piece design of valve 102 and follower 106 eliminates concentricity problems between bore 104 and counterbore 108. Ordinary tolerances are adequate to control the location of inlet passage 122 and outlet passage 126. Plug 112 permits adjustment of spring 110 to obtain the proper

pressure relief setting. Control of seating torque is not a problem since plug 112 does not form part of a pressure gasket sealing arrangement.

All of the features noted above combine to represent a significant reduction in initial manufacturing and assembly costs.

I claim:

1. In an internal combustion engine piston having first and second parts movable relative to one another in response to reciprocation of the piston and a pressure fluid containing chamber within said piston which varies in internal volume in response to said relative movement and to variations in the quantity of pressure fluid therein, the combination therewith of a supply and discharge valve assembly for said chamber including a bore in one of said parts, inlet and outlet passages communicating with said chamber and said bore at axially spaced points therealong, a spool valve movable in said bore to open said inlet and close said outlet passages and vice versa for controlling flow of fluid therethrough to thereby vary the quantity of fluid in said chamber, means for yieldably biasing said valve toward closed position to thereby regulate the pressure of the fluid in said chamber, speed compensating means including a supply passage communicating with said bore and with a source of pressure fluid which varies in pressure with the velocity of said piston and a movable follower exposed to pressure fluid in said supply passage and operably connected to said valve and biasing means for translating a velocity-related pressure variation in said source into a compensating force for said valve so that it is operable to provide substantially uniform regulation of the fluid pressure in said chamber regardless of the velocity of said piston.

2. The combination set forth in claim 1 including a counterbore in said one port communicating with said supply passage and said bore, said follower being disposed in said counterbore to form a movable wall thereof and having a simple contact connection with said valve, said counterbore having a stop therein for limiting movement of said follower so that said valve and follower move together during opening and closing of said outlet passage and said valve moves independently of said follower during opening and closing of said inlet passage.

3. The combination set forth in claim 1 wherein said follower is interposed between said biasing means and said valve to provide a force transmitting connection therebetween, means limiting said connection to the control of said outlet passage by said valve, said follower having a surface exposed to pressure fluid in said supply passage for developing said compensating force.

4. The combination set forth in claim 1 wherein said follower is disposed between and in abutting contacting relation with said valve and biasing means to thereby provide the operative force transmitting connection between said valve and biasing means, said valve and follower having their adjacent ends exposed to pressure fluid in said supply passage, said end of said follower having a larger effective working surface exposed to the supply passage pressure fluid than said end of said valve.

5. The combination set forth in claim 1 including a connecting rod operably connected to said piston for converting reciprocating movement thereof into rotary motion in the engine, said source of pressure fluid comprising an oil passage in said rod extending lengthwise thereof for supplying oil from a lubrication system of the engine to said piston, said supply passage communicating with said rod passage near the piston end thereof.

6. A variable compression ratio composite piston comprising outer and inner pistons movable axially relative to one another in response to reciprocation of the composite piston, said outer and inner pistons defining a pressure fluid chamber therebetween which varies in internal volume in response to said relative movement and to variations in the quantity of pressure fluid therein, said composite piston having inlet and outlet fluid flow passages communicating with said chamber, a valve movable to open and close said passages for controlling flow of fluid therethrough to thereby vary the quantity of fluid in said chamber, means for yieldably biasing said valve towards closed position relative to said outlet passage to thereby regulate the pressure of the fluid in said chamber, said composite piston having a compensating passage communicating with a source of pressure fluid in said composite piston which varies in pressure with the velocity of said composite piston and a follower exposed to pressure fluid in said compensating passage and having an operable connection to said valve during movement thereof in closing and opening said outlet passage for translating a velocity-related pressure variation in said source into a compensating force acting on said valve so that it is operable to provide substantially uniform regulation of the fluid pressure in said chamber regardless of the velocity of said composite piston, said valve having a pressure surface exposed to pressure fluid in said compensating passage such that opening and closing movement of said valve relative to said inlet is responsive to the pressure variations of said source.

7. A combined supply check valve and pressure regulating discharge valve comprising a body having an inlet passage for supplying fluid to a chamber and an outlet passage for releasing fluid from the chamber, a valve member movable in said body between a first position closing said outlet passage and opening said inlet passage, a second position closing both of said passages and a third position opening said outlet passage and closing said inlet passage, said valve member having a working area always exposed to fluid in the chamber via said outlet passage for biasing said valve member from said first towards said third position, a follower movable in said body in contact with said valve member between said second and third positions thereof, a regulating spring biasing said follower against said valve member for yieldably urging said valve member from said third to said second position thereof to regulate the discharge of fluid via said outlet passage and means limiting movement of said follower such that said valve member moves independently of and out of contact with said follower between said first and second positions, said follower having a working area and said valve member having a second working area, both of said last-mentioned areas always being exposed to fluid entering said inlet passage to develop forces acting oppositely on and tending to separate said valve member and follower, said follower area having a greater effective area than said second area of said valve member.

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