VARIABLE COMPRESSION RATIO PISTON

Filed Nov. 29, 1963 3 Sheets-Sheet 1

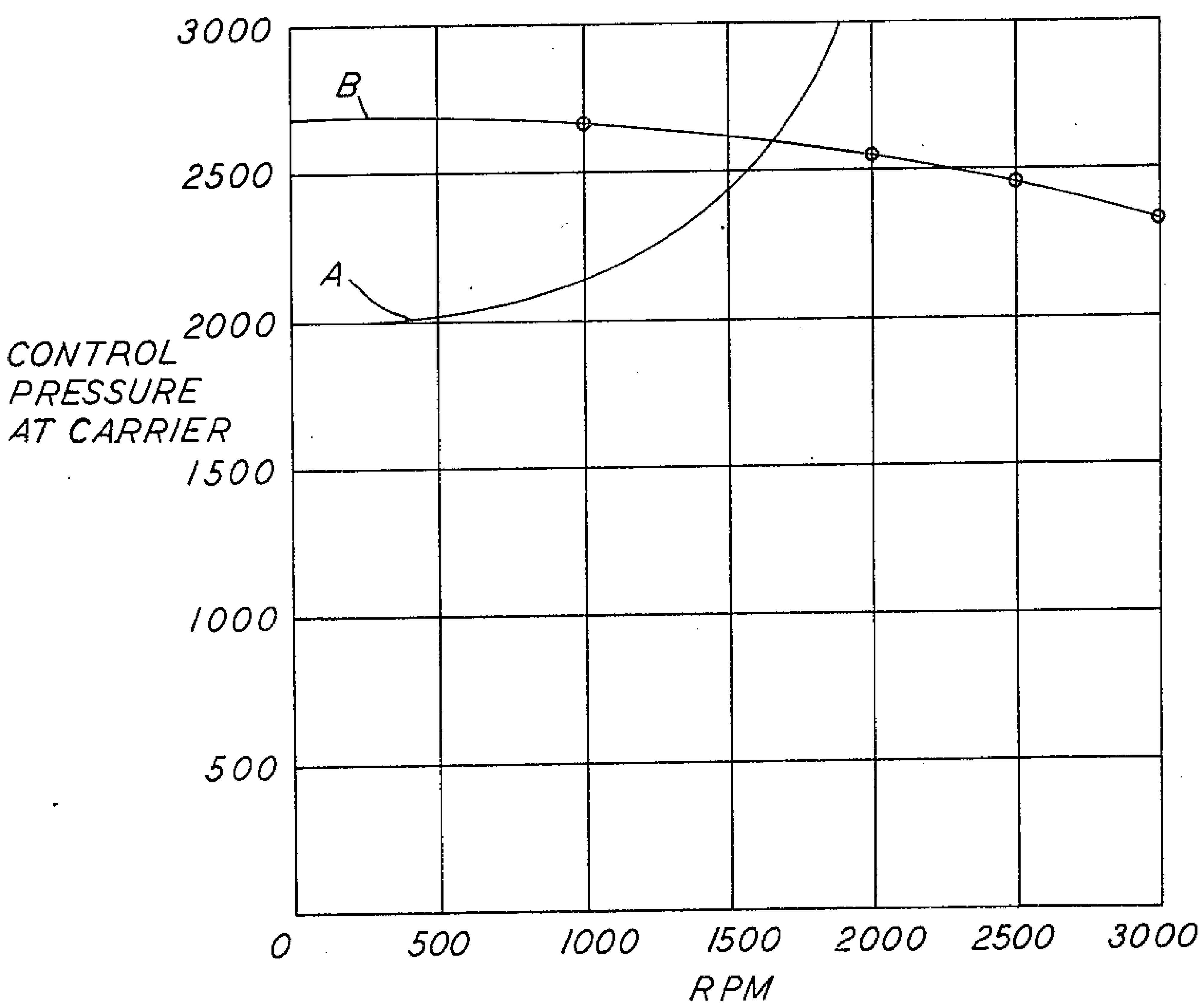

Fig. 1

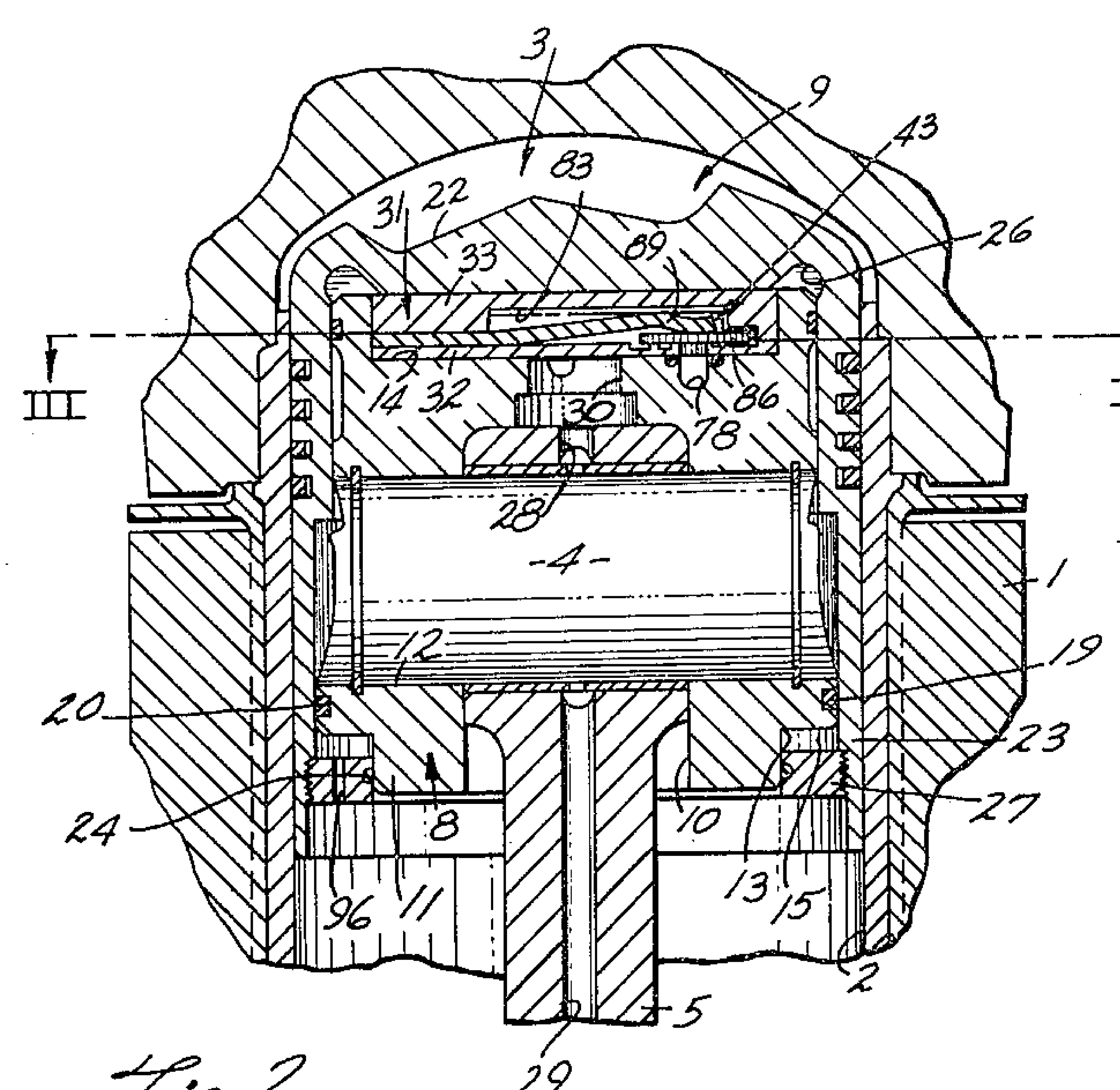

Fig. 2

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Sept. 14, 1965 S. TIMOUR ETAL 3,205,878

VARIABLE COMPRESSION RATIO PISTON

Filed Nov. 29, 1963 3 Sheets-Sheet 2

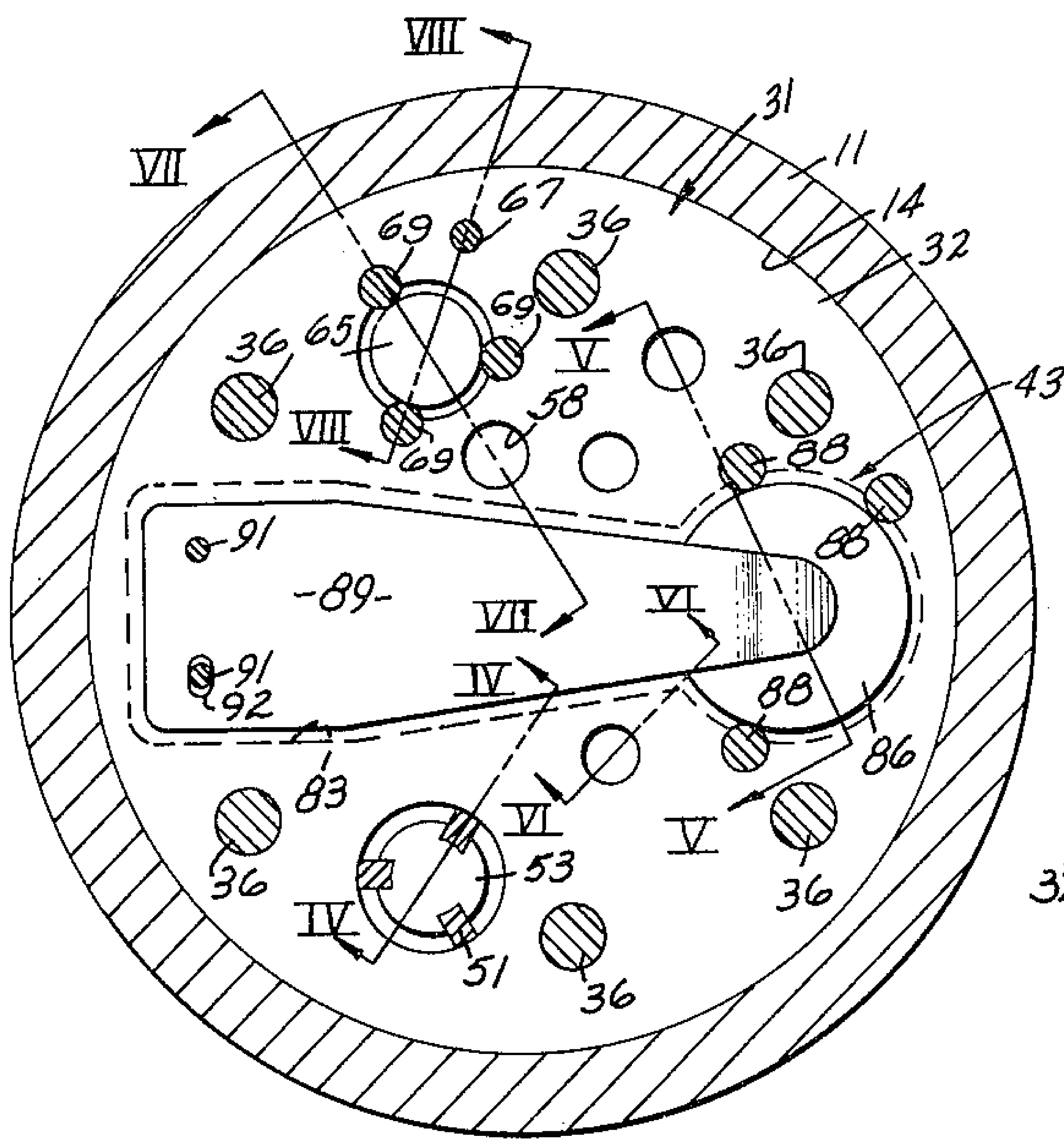

Fig. 3

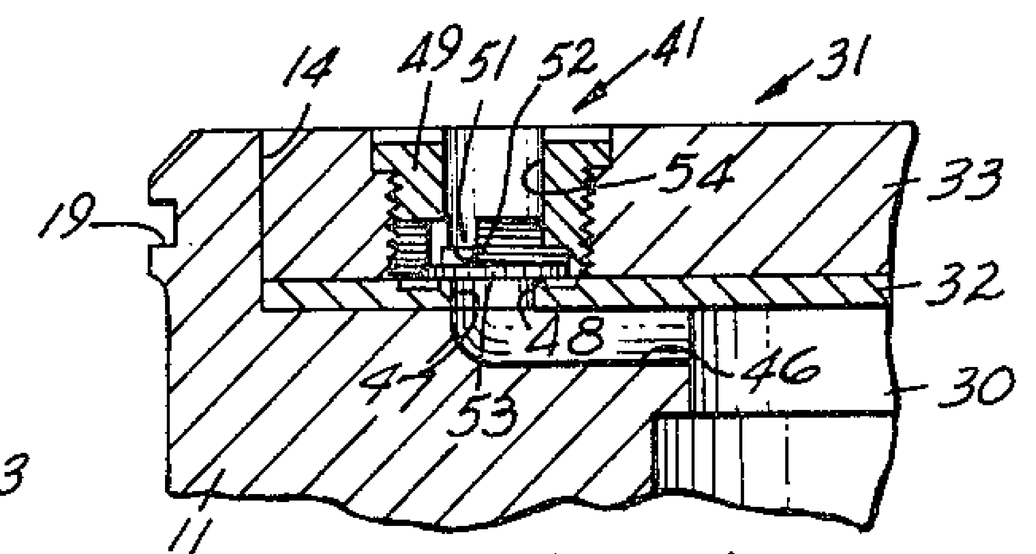

Fig. 4

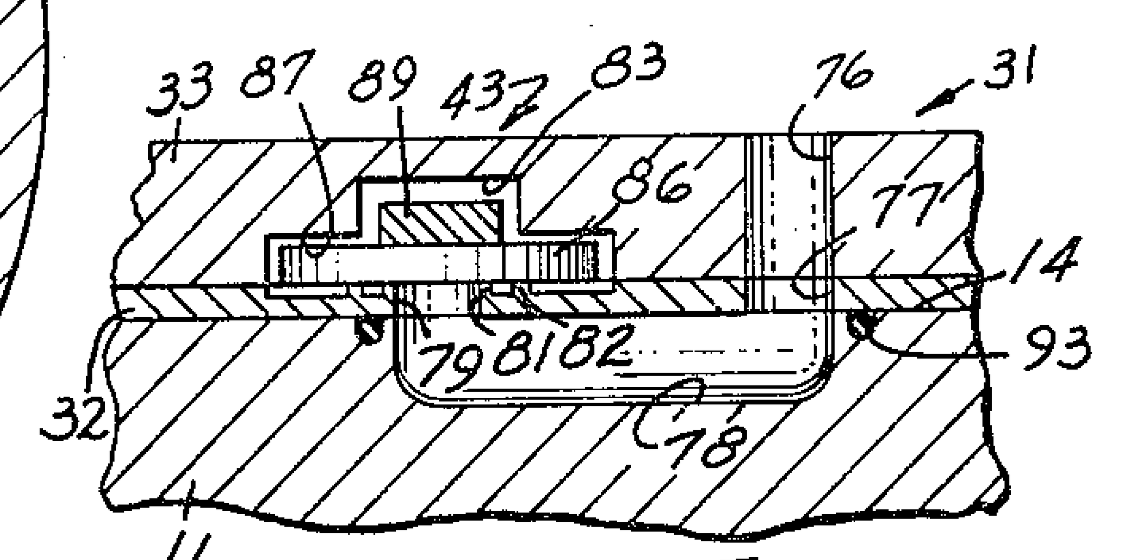

Fig. 5

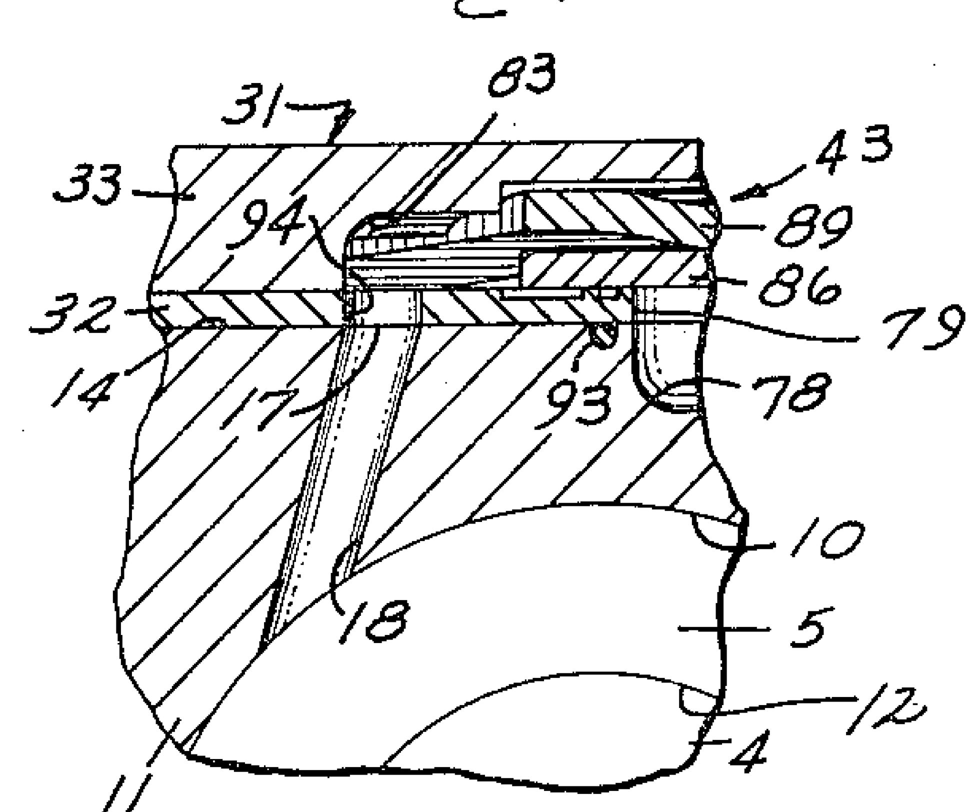

Fig. 6

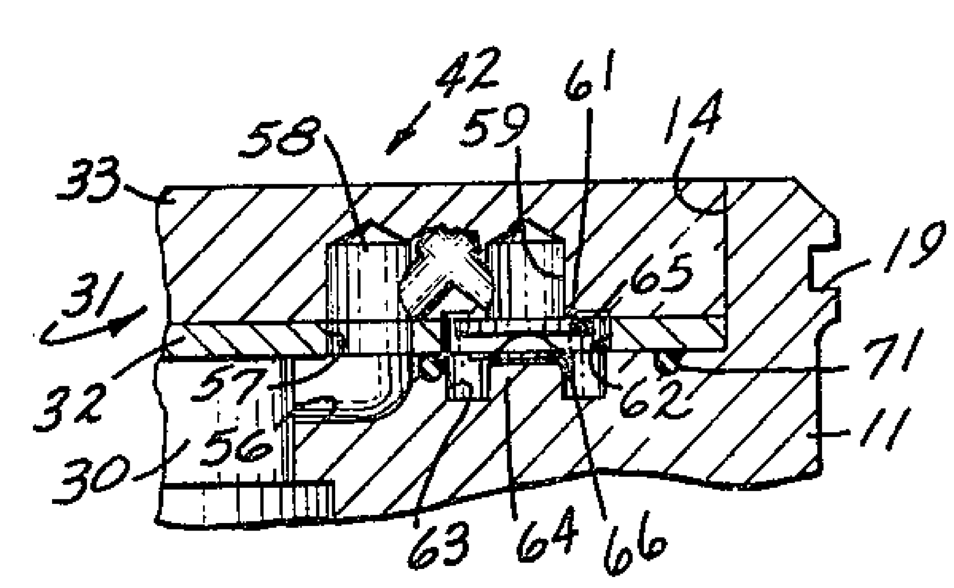

Fig. 7

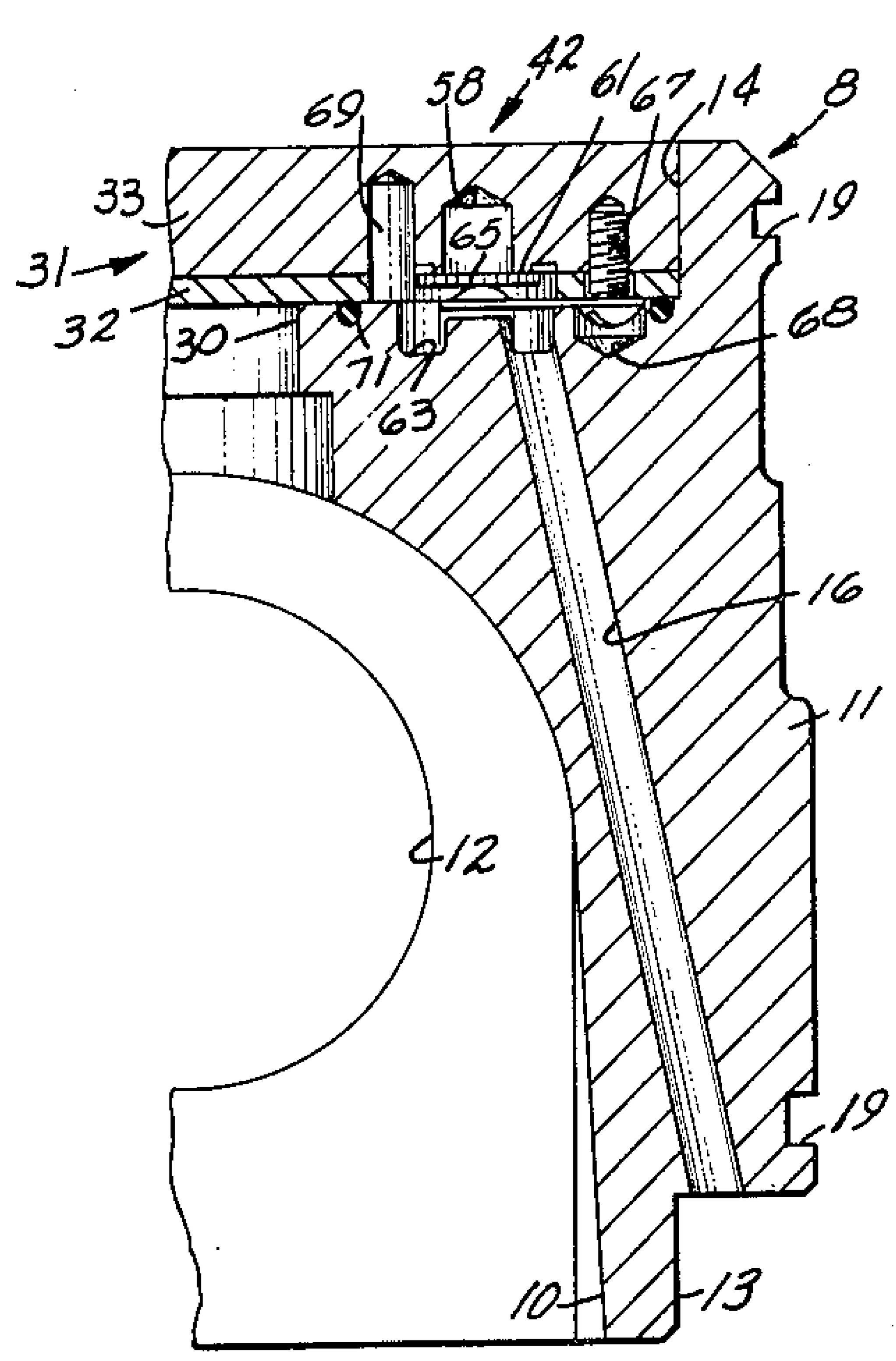

Fig. 8

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VARIABLE COMPRESSION RATIO PISTON

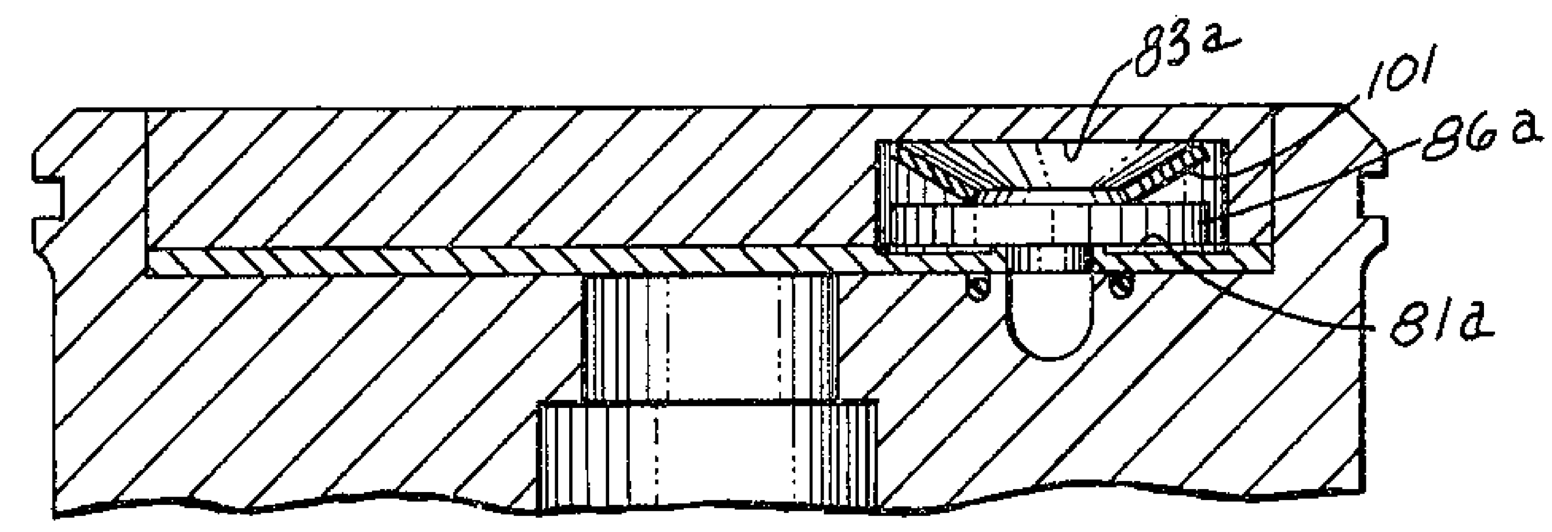

Fig. 9

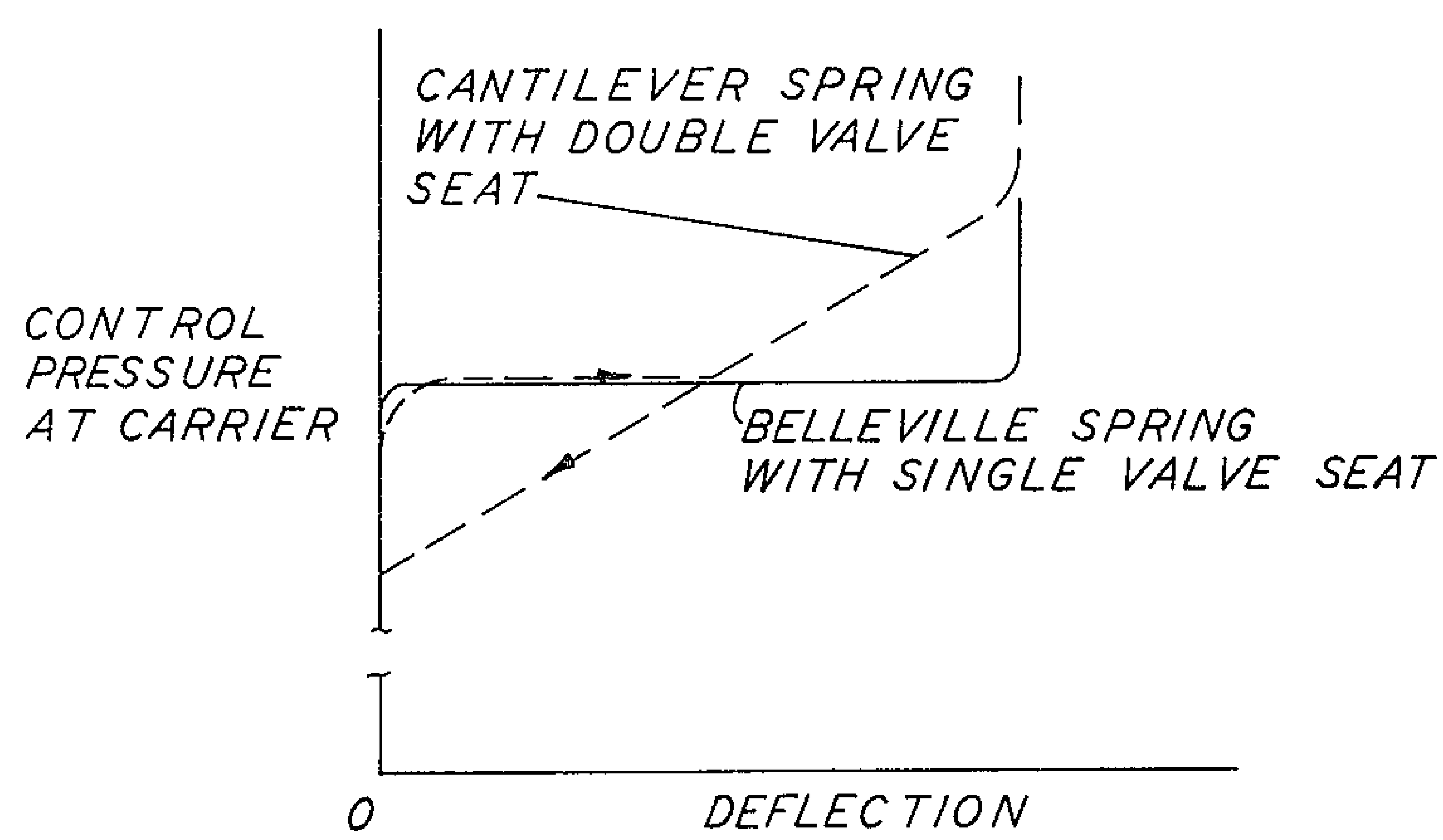

Fig. 10

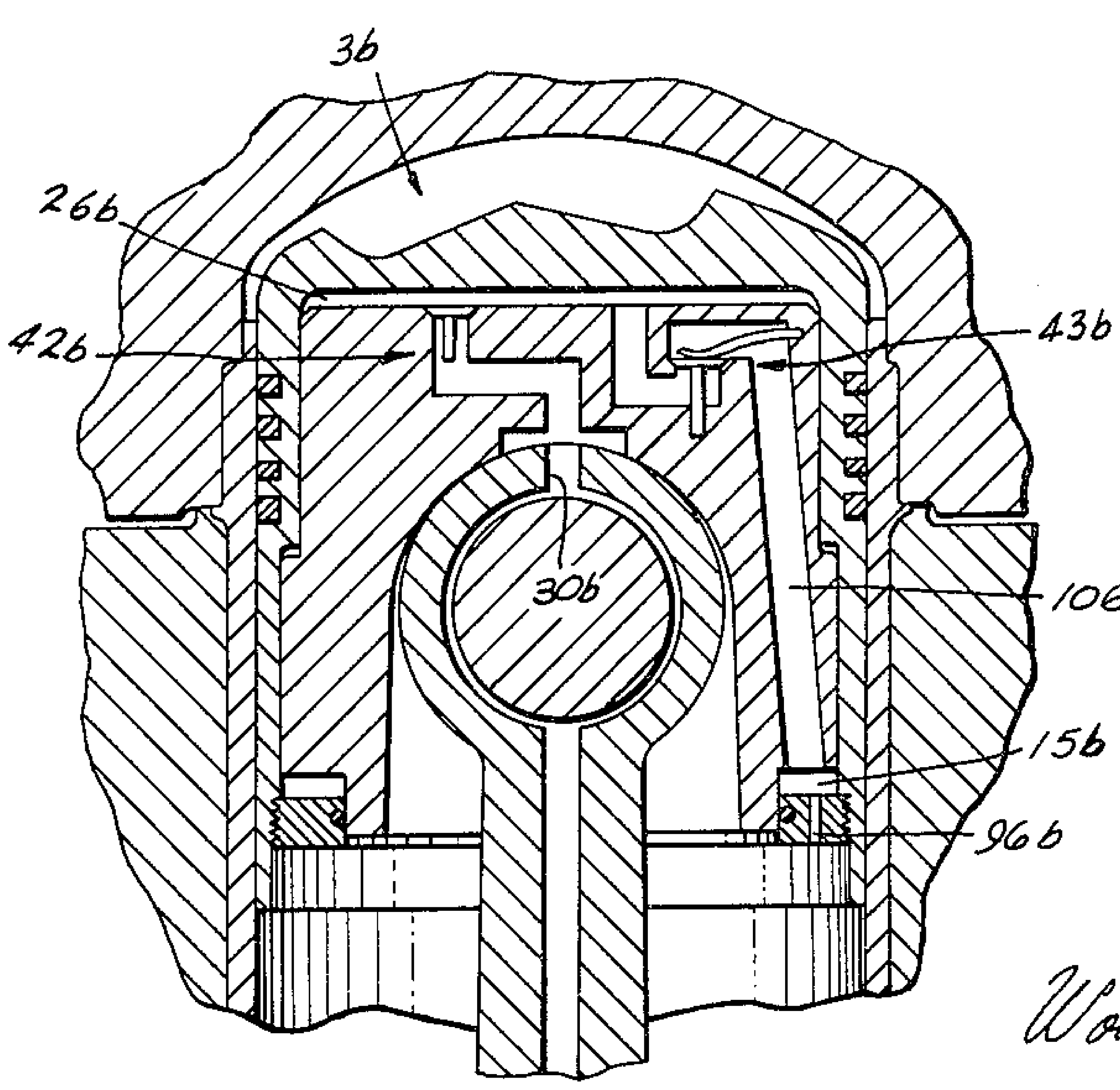

Fig. 11

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VARIABLE COMPRESSION RATIO PISTON

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Filed Nov. 29, 1963, Ser. No. 326,882
16 Claims. (Cl. 123—48)

This invention relates to a piston construction for an internal combustion engine and particularly to a type thereof which is sensitive both to cylinder pressure and to engine speed for varying the compression ratio of said engine.

Pistons of this general type have been previously known and have been exemplified in a number of previous patents, such as the United States Patent to Mansfield, No. 2,742,027. In these constructions, a piston core or carrier is provided with a telescoping piston hood or cover, said hood being so related to said carrier that telescoping movement of the hood with respect to the carrier will vary the compression ratio within the engine cylinder. Pressure fluid chambers are provided between the hood and the carrier in such a fashion that the emptying of one of the chambers of pressure fluid will allow the movement of the piston in one direction with respect to the carrier and the emptying of the other of the chambers of pressure fluid will allow the movement of the piston oppositely with respect to the carrier. A hydraulic system including internal valving receives oil from the engine lubrication system and conducts it to one or the other of said pressure fluid chambers to replenish same.

While this system is basically good and has a number of presently understood advantages which recommend it highly to industry, it also has as previously practiced certain disadvantages which have in the past prevented its adoption as a practical matter. Primary among these disadvantages is the fact that a pumping force for driving the oil from that chamber by which the compression ratio is decreased is opposed by the inertia force tending to move the fluid and hood toward the upper end of the cylinder as same as in the region of top dead-center. This opposing inertia force obviously increases with the square of the engine speed. However, the control force determining the point at which such pressure fluid is released to decrease the compression ratio is a substantially constant force provided by a spring. Thus, with the pumping force increasing with engine speed and a control force remaining constant, the mechanism operates to increase the peak firing pressure within the engine as the engine speed increases. In addition, the time for the fluid to be expelled from the chamber varies inversely with speed, aggravating the problem further. In one actual embodiment, the firing pressure was increased to such an extent that the maximum pressure during combustion of the engine increased 500 p.s.i. in the speed range of 1600 r.p.m. to 2000 r.p.m. This increased the pressures within the cylinder to an excessive level and rendered the apparatus unacceptable.

Further, in certain previous practice, such as that exemplified by the above-mentioned Mansfield patent, the valves, or some of them, were placed in such a manner within the piston that their direction of travel was substantially perpendicular to the direction of reciprocation of the piston. Thus, an increase in acceleration of the piston would cause the valve and/or its supporting stem to bear with increased force against one side or the other of whatever means were provided for guiding same. This increased the frictional relationship between said valve and such guiding means and thus a progressively increasing force was necessary to move such valve as engine speed increased. This, where acting on the relief valve, tended to hinder the compression ratio decrease within the cylinder as engine speed increased, thus further aggravating a problem which was already serious as above set forth. A decrease in velocity of the piston acted conversely.

Accordingly, while the basic approach as known to the prior art is advantageous and suggests the availability of much improved engine performance and fuel economy, the specific devices known to prior art by which the basic idea has been carried out have in their simpler forms been relatively ineffective or unreliable and in their complex forms too costly in either or both original manufacture or maintenance to be acceptable.

Therefore, the objects of the invention include:

1. To provide an automatically expansible and contractible piston capable of responding to pressure developed within a cylinder of an internal combustion engine and therefore to adjust automatically the compression ratio within said cylinder.

2. To provide a device, as aforesaid, which will be sensitive to the pressure developed on the combustion stroke of the engine for adjusting the compression ratio, as aforesaid, but wherein such sensitivity will remain constant or operate in a predetermined manner at all speeds of the engine.

3. To provide a device, as aforesaid, wherein inertia forces acting upon the movable portion of the piston is utilized to pump a pressure fluid which in turn modifies the size of the piston sufficiently to bring about the desired adjustment in compression ratio, but wherein the action of the pumping mechanism is itself modified in response to the speed of the engine as to substantially cancel any tendency of said pumping action to increase with the increased inertia force resulting from increased engine speed, whereby said pumping action will remain substantially constant at all speeds of the engine.

4. To provide a piston, as aforesaid, wherein the same inertia force which acts on the piston to tend to increase its pumping action is also utilized to modify the pumping mechanism whereby the tendency for increased pumping action is cancelled and said pumping action remains essentially constant for all speeds of the engine.

5. To provide a piston, as aforesaid, which will accomplish the objectives set forth above without materially increasing the complexity of the apparatus.

6. To provide apparatus, as aforesaid, which will accomplish the various objects set forth above and will provide a simplified mechanism for so doing.

7. To provide apparatus, as aforesaid, in which the frictional relationships of the valves with respect to the means in contact with which they operate are unaffected by the speed of the engine.

8. To provide apparatus, as aforesaid, in which a single supply of valve pressure fluid may be utilized to feed both the pump-up and the pump-down pressure fluid chambers.

9. To provide a piston, as aforesaid, establishing a differential between the pressure at which the relief valve opens and the pressure at which said relief valve closes whereby to permit escape of pressure fluid through a larger arc of rotation of the engine than has been known to previous practices and whereby to reduce the size of the valve needed to accomplish a given amount of control.

10. To provide a piston, as aforesaid, having a valve unit which can be bench-tested independently of the piston and then assembled into the piston as a separate unit, the same then being easily removable and equally easily replaceable by another unit as necessary for maintenance purposes.

11. To provide a piston, as aforesaid, of sufficiently simple construction that it can be made to a high degree of durability and of consequent reliability, while remain-

ing within the confines of conventional piston construction.

Other objects and purposes of the invention will be apparent to persons acquainted with devices of this general type upon reading the following specification and upon inspection of the accompanying drawings.

In the drawings:

FIGURE 1 is a pressure-versus-speed curve illustrating the pressure relationships in an internal combustion engine with respect to speed according to prior art and according to the present invention.

FIGURE 2 is a central longitudinal sectional view of the piston and cylinder arrangement embodying a preferred form of the present invention.

FIGURE 3 is a sectional view taken on the line III—III of FIGURE 2.

FIGURE 4 is a sectional view taken on the line IV—IV of FIGURE 3.

FIGURE 5 is a sectional view taken on the line V—V of FIGURE 3.

FIGURE 6 is a sectional view taken on the line VI—VI of FIGURE 3.

FIGURE 7 is a sectional view taken on the line VII—VII of FIGURE 3.

FIGURE 8 is a sectional view taken on the line VIII—VIII of FIGURE 3.

FIGURE 9 is an enlarged fragment of FIGURE 2 and showing a modified valve structure.

FIGURE 10 is a pressure-versus-deflection graph including curves for the discharge valve embodiments of FIGURES 2 and 9.

FIGURE 11 is a schematic central cross-sectional view of a further modified piston.

General description

In general, the objectives of the invention are accomplished by the recessing of the upper end of the piston carrier and providing a valve plate to be received within such recess. Within said valve plate are provided suitable disk valves movable between open and closed positions in a direction parallel to the direction of reciprocation of said piston. Passageways are provided within said valve plate communicating either directly or through other means to the respective chambers controlling the relative position between the piston carrier and the piston hood carried thereby.

Detailed description

Referring to FIGURE 2, a cylinder block **1** of any conventional form is provided with a cylinder **2** having therein a reciprocable piston **3** associated in a conventional manner through a wrist pin **4** with a connecting rod **5**. The piston embodying the invention is comprised of a carrier **8** and a hood **9**, said hood **9** being telescopically arranged with respect to the carrier **8** for movement in a direction parallel to their common axis of reciprocation.

Referring first to the carrier **8**, there is provided a body **11** having a central transversely arranged opening **12** in a conventional manner for the reception of said wrist pin **4**. This may be provided in any convenient manner and forms no part of the invention excepting to note that same lies entirely within the hereinafter-described skirt portion of the hood **9** and hence is protected from rubbing against the cylinder walls. A central opening **10** is provided in the usual manner within said piston carrier **8** to provide access by the connecting rod **5** to the wrist pin **4** and to enable the wrist pin **4** to be placed in the usual manner at a point substantially spaced upwardly from the lower end of piston carrier **8**. Said carrier **8** has an annular groove **13** at its lower end which in part defines the lower chamber **15** hereinafter further described. Said carrier **8** is also provided with a recess **14** in its upper end for reception of the valve plate hereinafter further described. A passageway **28** extends from a source of fluid under pressure, in this case from a point communicating with the lubrication passageway **29** for the wrist pin **4**, and communicates through the opening **30** with a suitable portion of the recess **14** for supplying fluid under pressure thereto, in this instance substantially centrally of said recess. A passageway **16** (FIGURE 8) extends from said groove **13** to the recess **14** for conducting pressure fluid, usually engine lubricating oil under pressure, from said valve plate recess **14** to said groove **13**, that is, to said lower chamber **15**. An opening **17** (FIGURE 6), and in this embodiment a pair thereof, in said valve plate recess **14** is provided for receiving the oil being released from the hereinafter-described upper chamber **26**, said opening **17** communicating through a passageway **18** to any convenient point by which said oil can escape back to the low pressure side of said system, here the engine crank case. In this instance, the oil escapes into the wrist pin opening **12**.. Grooves **19** are placed circumferentially around the piston carrier **8** for the reception of appropriate packing rings **20** for sealing between said hood **9** and carrier **8**.

The piston hood **9** consists of a top portion **22** and a skirt portion **23**. Said skirt portion **23** cooperates with the packing rings **20** of said carrier **8** for establishing a fluid-tight relationship therebetween. Said hood **9** defines with the carrier **8** an upper chamber **26** hereinafter further mentioned which upon introduction of pressure fluid such as lubricating oil thereinto under pressure provides an upward force on said hood **9** with respect to the carrier **8** thereby tending to increase the compression ratio of the engine cylinder **2**. At the lower end of said skirt **23** there is attached thereto, such as by threaded relationship, a ring **27** which cooperates with the annular groove **13** for defining the lower chamber **15**. A seal **24** prevents fluid passage from the lower chamber **15** between the ring **27** and the adjacent surface of the body **11**. Said lower chamber **15**, upon action of the inertia forces and the above-mentioned pressure forces has a very high fluid pressure induced thereinto. Leakage of this fluid through orifice **96** controls, substantially independent of speed, the amount the hood **9** extends upwardly from the carrier **8** during each cycle of operation.

Referring now to the valve plate assembly **31** (FIGURES 2 and 3), the same is of generally disk shape and dimensioned to fit snugly within said recess **14**. Basically the valve plate **31** functions to allow oil under pressure from the opening **30** to flow through an upper chamber intake valve generally indicated at **41** (FIGURE 4) into the afore-mentioned upper chamber **26** (FIGURE 2) whereby to raise the hood **9** and help increase the compression ratio of the engine. Further, the valve plate **31** functions to allow oil under pressure from the opening **30** (FIGURES 7 and 8) to flow into a lower chamber intake valve generally indicated at **42** which conducts same through the passageway **16** and into said lower chamber **15** (FIGURE 2). The valve plate **31** further contains an upper chamber discharge or relief valve generally indicated at **43** (FIGURES 2, 5 and 6) which when the upper chamber **26** is overpressurized opens to allow pressure fluid therein to flow through the opening **17** and passageway **18** into the low pressure space **10**. A calibrated orifice **96** is provided through the ring **27** to control discharge of the lower chamber **15** thereby to control the rise of the hood **9** with respect to the carrier **8** by which the compression ratio of the engine cylinder tends to increase by the same amount each engine cycle.

Considering the valve plate **31** in more detail, same comprises two main parts, namely, a lower plate **32** and an upper plate **33** lying on top thereof. Some form of gasket (not shown) is preferably clamped between the two plates to act as a seal. The top surface of the valve plate assembly **31** and hence of the upper plate **33** is preferably even with the top of the body **11** of the carrier **8** for defining therewith an essentially flat surface. The upper plate **33** and lower plate **32** are held against angular movement within the recess **14** by any convenient means such as the bolts generally indicated at 36 (FIGURE 3) which, for example, may be threaded into the body 11 of the carrier 8.

Considering now the upper chamber intake valve 41 (FIGURE 4), a groove 46 connects at one end thereof with the opening 30 which is fed with lubricating oil under pressure from the line 29 lubricating the wrist pin 4. The groove 46 lies in the bottom face of the recess 14 and communicates through an opening 47 in the lower plate 32 (FIGURE 4) with the input side of the aforementioned upper intake valve 41. The lower plate 32 has an annular groove in the upper face thereof coaxial with the opening 47 for defining therewith a valve seat 48. A cylindrical plug 49 threadedly engages and extends through the upper plate 33 and has a plurality, here three, of spaced downwardly extending fingers 51, the radially inner face of each of which has a downwardly facing shoulder 52. A flat, preferably circular valve disk 53 lies atop and in sealing contact with the valve seat 48 when the valve 41 is closed. The valve 41 is held closed by pressure on the upper face of the valve disk 53 and thus prevents the flow of pressure fluid from the opening 30 therepast. When, however, the pressure on the upper face of the valve disk 53 drops below that of the pressure fluid from the opening 30, said pressure fluid will raise the valve disk 53 upwardly from the valve seat 48 and will flow therepast through the central opening 54 of the plug 49 and into the upper chamber 26. Upward movement of the valve disk 53 is bounded by contact of said valve disk 53 with the shoulders 52 of the plug 49. Eccentric movement of the valve disk 53 within its own plane is prevented by the depending fingers 51 on the plug 49. Hence, the valve disk 53 may move only vertically for a small distance in a direction dictated by the difference of pressures of pressure fluid on the opposite sides thereof.

Considering now the lower chamber intake valve 42 (FIGURES 7 and 8) a groove 56 in the bottom face of the recess 14 communicates from the opening 30, aforementioned, through an opening 57 in the lower plate 32 to an enclosed passage 58 in the upper plate 33. Said passage 58 opens at its opposite end through the lower face of said upper plate 33 at a place 59 spaced from the opening 57. In the particular embodiment shown, the enclosed passage 58 comprises a pair of spaced, downwardly opening, blind holes connected above the lower face of the upper plate 33 by a pair of cross-drilled holes. The downwardly opening outlet end 59 of the enclosed passage 58 is coaxial with an annular groove in the bottom face of the upper plate 33 and defines therewith a preferably circular valve seat 61. A relatively large coaxial opening 62 is provided therebelow through the lower plate 32. An upwardly facing well 63 is provided in the lower face of the recess 14 coaxially with the opening 62 and has an upstanding post 64 in the center thereof, the top of which can be slightly below the bottom surface of the recess 14.

The well 63 communicates with the afore-mentioned passage 16 and, hence, therethrough with the lower chamber 15. A valve disk 65 is here shown for purposes of illustration as being biased toward the valve seat 61 by resilient means including a small, cantilevered spring 66 lying therebelow and backed up by the post 64. The opposite end of the spring 66 is secured by means of a screw 67 upwardly threaded through the lower plate 32 and into the upper plate 33 (FIGURE 8). The head of the screw 67 is received into an opening 68 in the lower face of the recess 14. Said spring 66 is, however, used primarily to hold the valve disk 65 in place prior to assembly of the valve plate 31 into the recess 14 and may be omitted if desired. Said opening 68 has a small extension through which the middle portion of the spring 66 extends into the well 63. A plurality, here three, of upstanding pins 69 extend upwardly from the bottom of the recess 14 through suitable openings in the lower plate 32 and upper plate 33. The pins 69 are preferably evenly spaced from each other and from the center of the valve seat 61 at a sufficient distance to closely surround and vertically guide the valve disk 65. Thus, the valve disk 65 is constrained to move only vertically and only between the valve seat 61 and a point spaced somewhat therebelow determined by the interference of the post 64 with the other side of the cantilevered spring 66.

The valve 42 is biased closed by the relatively small upward force of the cantilevered spring 66, where same is used, which in effect adds a small increment to the force derived from fluid pressure in the passage 16 whereby the pressure within the passage 58, hence, the opening 30, must exceed the pressure within the passage 16 plus said small increment due to the spring 66 before the valve disk 65 will move from the valve seat 61 to open the valve 42. This increment is, however, small and may usually be ignored. A resilient sealing ring such as the O-ring 71 lies in a suitable groove in the lower surface of the recess 14 and laterally surrounds the screw 67, well 63 and pins 69. The O-ring 71 bears tightly against the adjacent lower face of the lower plate 32 whereby to seal the interface between the bottom surface of the recess 14 and lower face of the lower plate 32 within the lateral confines of the O-ring 71 against the pressure fluid in the opening 30. Other suitable sealing methods may be utilized, however, if desired.

It will be apparent that the screw 67 and pins 69 may be used in aligning the upper plate 33 with the lower plate 32 during the assembly of the valve plate assembly 31 and that said screw 67 may be utilized to hold the valve plate assembly 31 together prior to its installation as a packaged unit in the recess 14.

Turning now to the upper chamber discharge valve 43 (FIGURES 2, 3, 5 and 6), same is fed through coaxial openings 76 and 77 in the upper plate 33 and lower plate 32, respectively. The opening 76 communicates with the upper chamber 26 above the valve plate assembly 31. The opening 77 communicates through an upwardly opening groove 78 in the lower face of the recess 14 with a further opening 79 through the lower plate 32 and spaced from said opening 77. The upper face of the lower plate 32 has an annular groove therein concentric with and radially outwardly spaced from the opening 79 to form a valve seat 81. The lower face of the upper plate 33 is suitably recessed at 83 to provide room for a relatively large valve disk 86 which can extend radially beyond the valve seat 81 and which is preferably quite massive relative to the hereinabove-described disks 53 and 65. The recess 83 has a shoulder portion 87 which with the valve seat 81 defines the opposite limit of vertical travel of the valve disk 86. A suitable plurality, here three, of circumferentially spaced pins 88 closely surround the valve disk 86 for preventing lateral movement thereof. Hence, the disk 86 may move vertically over a bounded path toward and away from the valve seat 81. The recess 83 further contains a large cantilevered spring 89 which is relatively stiff and which bears at one end thereof (rightward as seen in FIGURES 2 and 3) upon the upper face of the valve disk 86 for strongly urging same into contact with the valve seat 81 whereby the valve 43 is closed. The other end of the spring 89 is prevented from vertical movement by being held between the lower surface of the recess 83 and the adjacent upper surface of the lower plate 32. Said spring 89 is prevented from movement transverse to the direction of piston motion by a pair of pins 91 extending therethrough into the upper plate 33 (FIGURE 3). One of said pins 91 passes through a hole 92 in said spring 89 which is elongated in alignment with the other of said pins 91 to allow for mislocation of holes in said spring 89.

Gasket means such as the O-ring 93 surrounds the groove 78 in the body 11 of the carrier 8 and bears against the adjacent lower face of the lower plate 32 whereby to prevent entrance thereinto of pressure fluid from the opening 30. The afore-mentioned recess 83 in the lower side of the upper plate 33 communicates through an opening 94 in the lower plate 32 with the opening 17 and passage 18 in the body 11 of the carrier 8. Thus, closure of the valve disk 86 against the seat 81 prevents discharge of pressure fluid from the upper chamber 26, into the low pressure area 10 and opening of said valve disk allows said discharge. Since the upper side of the valve disk 86 is subjected to the lowest pressure, the pressure in the crank case of the engine in the system, a relatively strong spring 89 is required to maintain said valve disk 86 in a closed condition against at least the lower range of pressures encountered in the upper chamber 26. The mass of the disk 86 is sufficient to influence the opening or closing of the valve 43 when subjected to high accelerations, as in an operating engine wherein the piston 3 reciprocates rapidly. Hence, the relatively large mass of the valve disk 86 makes the opening and closing of the valve 43 partially dependent upon the rotational speed of the engine.

In certain cases, it will be desired to include a second valve seat 82 annular of and radially spaced outwardly of the seat 81, the valve disk 86 overlying both said seats as shown in FIGURES 2 and 5. The provision of a pair of spaced annular seats 81 and 82 further modifies the operation of the valve 43. For example, as the pressure within the groove 78 builds up to a critical level, the valve disk 86 will begin to lift from the valve seat 81. Thus, the pressure drop across the seat 81 decreases, the pressure in the space between the seats 81 and 82 increases and the pressure drop across the seat 82 also increases. Thus, the average upward force (pressure multiplied by area) on the lower face of the disk 86 increases because of the additional valve area exposed as the discharge valve lifts from its seat 81. Near the critical point, therefore, a relatively small increase in pressure sufficiently overcomes the restoring force of the cantilevered spring 89 and said disk 86 is lifted to open the valve 43 a relatively large amount. The valve then stays open until the pressure in the groove 78 drops to a point significantly below that at which the valve opened. Therefore, for a given opening pressure load, the valve 43 stays open longer than it would given a constant valve disk area, whereby to give the upper chamber 26 additional draining time. Alternately, however, the annular groove between seats 81 and 82 may be omitted and the single valve seat 81 used. A small bleed (not shown) can be installed from between seats 81 and 82 to the passageway 18 to insure that the pressure difference between the seats 81 and 82 is relieved.

The opening 96 through the ring 27 allows a controlled discharge of oil from the lower chamber 15 to control the pump-up of the hood substantially independently of engine speed.

Operation

Briefly, the device of the invention acts to maintain the combustion chamber pressure below a preselected pressure level during the firing of a charge in said combustion chamber. The device of the invention performs this act in a new way and particularly achieves an essentially constant combustion chamber pressure despite wide vartions in engine speed and in throttle opening, i.e., amount of fuel per charge supplied to the combustion chamber.

Assuming a four-stroke cycle engine operating at constant speed and throttle, let us further assume that the piston 3 is about to begin its compression stroke and is at bottom dead center of the crank. In this position, the oil pressure in the lower chamber 15 is lower. The hood 9 is in position with regard to the carrier 8 whereby to provide a high compression ratio. As the piston 3 reaches some point near, typically slightly before, top dead center of the compression stroke, the fuel-air mixture compressed thereby will be ignited by any convenient means (not shown) such as by compression in a diesel engine thereby greatly increasing the combustion chamber pressure. As said combustion chamber pressure goes over a preselected limit, the discharge valve 43 will open whereby oil will flow from the upper chamber 26 past the valve disk 86 and through the openings 94 and 17 to the low pressure area 10 or crank case of the engine. As the excessive combustion chamber pressure drives the oil out of the upper chamber 26, the volume of same will decrease, the hood 9 will move downwardly over the carrier 8 to increase the volume of the lower chamber 15 and, hence, lower the pressure thereof sufficiently that the lower chamber inlet valve 42 will open and the lower chamber 15 will fill past the disk 65 from the opening 30 and through the passageway 16. As the piston 3 moves through the firing stroke toward bottom dead center of the crank, the combustion chamber volume increases and hence after firing is completed the pressure within said combustion chamber drops sufficiently to allow the closure of the upper chamber discharge valve 43.

As the engine crank passes bottom dead center and moves upwardly to begin the exhaust stroke, the exhaust valve opens and the pressure upon the piston 3 is minimized, being only that of the back pressure of the exhaust system. Hence, the upper chamber 26 is at a low pressure and the upper chamber intake valve 41 opens to admit oil to said upper chamber 26. Because the upper chamber 26 is now at a lower pressure than is the lower chamber 15 the pressure drop across the lower chamber inlet valve 42 falls when the upper chamber valve 41 opens whereby said lower chamber inlet valve closes and feeding of the lower chamber 15 stops. The inertia force of the hood 9 induces a high oil pressure in the lower chamber 15. The lower chamber 15 is constantly emptying in a controlled manner through its outlet opening 96. Pressure in the upper chamber 26 also helps the inertia force to move the hood 9 upwardly with respect to the carrier 8 to increase the compression ratio of the cylinder. The rate of increase in the compression ratio is thus controlled by the rate of discharge of pressure fluid through the lower chamber discharge opening 96. Under the assumed constant speed and throttle conditions, the hood 9 preferably reciprocates once on the carrier 8 during each four-stroke cycle between some compression ratio position and one of somewhat lower, but not usually the minimum obtainable, compression ratio. Previous compression ratio adjusting pistons have operated reasonably successfully at constant but low engine speeds with constant throttle settings and low or moderate manifold pressures. However, said previous designs have generally been unsuccessful at open or variable throttle settings and high manifold pressures, at high engine speeds or some combination of said conditions. At open throttle settings, an increased amount of fuel is introduced into the combustion chamber which, upon burning, raises the pressures in said combustion chamber considerably over those encountered during constant low speed and throttle operation. Under sudden full throttle operation at a constant speed it is thus the function of the adjustable piston to reduce the compression ratio from a normal high level to a desired low level in a given number of cycles to compensate for the afore-mentioned sudden open throttle pressure increase. For example, in a particular embodiment of the present invention, the compression ratio of the engine drops from 20 to 1 to 12 to 1 in a minimum period of 20 engine cycles when the throttle is suddenly opened wide. This requires a net amount of pump-down over several cycles and thus increases the volume of oil to be passed through the discharge valve 43 in a given period of time. In said particular embodiment, the amount of oil required to be discharged through the discharge valve 43 during maximum pump-down conditions is approximately four times the oil handled during steady-state operation at the same engine speed. The difference in required oil discharge rates between steady-state operation at low speeds and open throttle operation at high speeds differs because of the speed difference. Thus, in some previous designs, the achievement of the afore-mentioned rate of net pump down has not been possible at medium-to-high engine speeds because of fluid velocity losses due to fluid friction. Friction losses due to high pressure fluid velocity are minimized in the device of the invention by maintaining the discharge valve **43** open for an extended period of time. This increased period of time is allowed, as aforesaid, by the differential between the opening and closing pressures required by the valve **43**.

Thus, the compression ratio adjusts relatively quickly to an increase or decrease in throttle opening whereby the engine is maintained at or near the ratio at which it is most efficient.

At high engine speeds with the throttle set for constant speed, additional factors degrade the performance of previous designs. For example, high engine speeds result in high piston accelerations whereby piston inertia effects adjustable piston performance. Thus, as the piston moves past top dead center, the hood **9** due to its own inertia, tends to move upwardly away from the carrier **8**. Thus, on the firing stroke, for example, the pressure in the upper chamber **26** is made lower than normal. In previous designs, this would result in late opening of the discharge valve and would result in said valve being opened for a shorter amount of time. Furthermore, previous designs have often utilized relatively heavy or large discharge valves so oriented that the inertia of the valve tends to make same remain closed at this position of the cycle thus further compounding the problem. Thus, at high speeds, the upper chamber does not empty as quickly as it ought with the result that the peak combustion pressure of the engine is allowed to rise with engine speed in previous designs as indicated by curve A of FIGURE 1, which departs from the desired ideal of a compression ratio which is constant regardless of engine speed or which drops slightly as engine speed increases as indicated by curve B in FIGURE 1.

The device of the invention, on the other hand, has its discharge valve **43** so arranged that increased inertia forces upon the hood **9** will be compensated for by the similarly increased inertia forces on the valve disk **86**. More precisely, both the valve disk **86** and hood **9** will be urged upwardly with respect to the carrier **8** near top dead center of the firing stroke at high engine speeds. Hence, the valve **43**, at high engine speeds, opens sooner and closes later due to the inertia of said disk **86**, the period during which said valve **43** is open being controlled by carefully controlling the mass of said disk **86**. Thus, the valve **43** opens and closes in response to lower pressures in the upper chamber **26**. Hence, the problem of evacuating the chamber **26** when it is at a pressure reduced by the inertia of the piston hood **9** is compensated for and the resulting relationship of engine speed to pressure required to open the discharge valve **43** is plotted at B in FIGURE 1.

It will be noted that the valves **41** and **42** being relatively light will be relatively unaffected by inertia forces and more importantly, being aligned to traverse in a direction parallel to the axis of reciprocation of the piston, will undergo no frictional inhibition of operation due to a change in engine speed. Thus, the opening and closing of the valves **41** and **42** will be relatively unaffected by any increase in the speed of the engine.

Modification

FIGURE 9 discloses a modified fragment of FIGURE 2 wherein the cantilever spring **89** of FIGURE 2 is replaced by a Belleville spring **101** which essentially comprises a truncated, open ended, cone of spring material such as spring steel having its axial small end bearing upon the valve disk **86a** and its large axial end bearing upon the top wall of the recess **83a**. The Belleville spring **101** may be used when it would be undesirable to extend the open period of the discharge valve **43** by use of the double valve seat and cantilever spring construction as discussed hereinabove with respect to the preferred embodiment of the invention. When a late closure of the valve **43** is not desired, the Belleville spring **101** is used with a single valve seat **81a**. A Belleville spring can have the unusual characteristic of remaining practically undeflected by an increasing load until said load reaches a critical value at which point the Belleville spring deflects through its full deflection path. Thus, the graph of FIGURE 10 shows that an increase in pressure on the lower face of the valve disk **86a** to a particular critical level causes the Belleville spring to immediately deflect to fully open the valve **43a**. The valve **43a** remains in its fully opened position as long as the pressure on the lower face of the valve disk **86a** is above said critical level. When the pressure falls below the critical level the Belleville spring **101** immediately closes the valve **43a**. Thus, the length of time during which the valve **43a** is fully opened is increased over the fully open time with just a cantilever spring. It is possible, however, that this embodiment would allow less fluid flow than the preferred embodiment under maximum pump-down conditions. The graph of FIGURE 10 also shows a possible characteristic curve of said preferred embodiment for purposes of comparison.

FIGURE 11 discloses, schematically, a modified valving system for the piston **3b** which eliminates the lower chamber intake valve **42** (FIGURES 7 and 8) described hereinabove. In FIGURE 11, the upper chamber inlet valve **42b** supplies pressure fluid from the opening **30a** to the upper chamber **26b** in the manner described hereinabove with respect to the preferred embodiment of the invention. The upper chamber discharge valve **43b** is arranged to empty the upper chamber **26b** in the same manner as described hereinabove with respect to the preferred embodiment of the invention except that the outlet of the discharge valve **43** now communicates through a passage **106** with the lower chamber **15b**. The lower chamber **15b** still includes an outlet opening **96b**. The operation of the device in FIGURE 11 is essentially similar to that of the preferred embodiment of the invention with the following few differences. Upon ignition of the combustible mixture in the combustion chamber above the piston **3b**, and consequent opening of the upper chamber discharge valve **43b**, the oil from the upper chamber **26b** travels through said discharge valve passage **106** into the lower chamber **15b**. It will be seen, therefore, that the lower chamber **15b** of the modified structure of FIGURE 11 must be sufficiently large to accommodate the oil emerging through the discharge valve **43b** without increasing the pressure on said fluid. Thus, it is preferred that the cross-sectional area (the area in a radial plane of the piston) of modified lower chamber **15b** be equal to the cross-sectional area of the upper chamber **26**. No similar requirement is made of the primary embodiment of the invention hereinabove described.

Although particular embodiments of the invention have been disclosed hereinabove for purposes of illustration, modifications or variations thereof lying within the scope of the appended claims are fully contemplated.

What is claimed is:

1. A piston including compression ratio adjusting means for use in an internal combustion engine cylinder having a combustion chamber and including:

a carrier and means for pivotally associating said carrier to a connecting rod;

a hood telescoped over said carrier for adjustment with respect thereto in a compression ratio adjusting manner, said hood defining with said carrier first and second pressure chambers so arranged that pressurizing said first chamber will move said hood to increase the compression ratio and pressurizing said second chamber will move said hood to decrease the compression ratio;

a source of fluid under pressure; and

a valve and conduit system for controlling the quantity of pressure fluid introduced into said respective chambers, the improvement in said valve and conduit system comprising:

inlet conduit means and check valve means for connecting said source with one of said chambers and means for supplying the other of said chambers with pressure fluid;

means for connecting said second chamber with a low pressure point of the fluid pressure system;

an outlet fluid conducting system including a one-way relief valve connecting said first chamber to a low pressure point of the fluid pressure system, said relief valve being movable axially of said cylinder toward said combustion chamber for opening, whereby the frictional relationships between said relief valve and the means supporting same will be independent of the speed of reciprocation of said piston, while the inertia of said one-way relief valve will be responsive to the speed of reciprocation of said piston in a manner to diminish the opening pressure of the relief valve with increasing piston speed.

2. A piston including compression ratio adjusting means for reciprocation in an internal combustion engine cylinder and including:

a carrier and means for pivotally associating said carrier to a connecting rod;

a hood having at least one closed end reciprocable in said cylinder and telescoped over said carrier for adjustment axially with respect thereto in a compression ratio adjusting manner, said hood defining with said carrier first and second pressure chambers so arranged that pressurizing said first chamber will move said hood to increase the compression ratio and pressurizing the second chamber will move said hood to decrease the compression ratio;

a source of fluid under pressure; and

a valve and conduit system for controlling the quantity of pressure fluid introduced into said respective chambers, the improvement in said valve and conduit system comprising:

inlet conduit means including first and second axially movable one-way inlet valves sensed oppositely to each other for connecting said source with said first and second chambers;

opening means for connecting said second chamber with a low pressure point of the fluid pressure system;

an outlet fluid conducting system including a one-way relief valve connecting said first chamber to a low pressure point of the fluid pressure system, said relief valve being movable toward said closed end of said hood for opening, whereby the frictional forces on said valves will be substantially independent of the speed of reciprocation of said piston.

3. The device defined in claim 1 wherein there is provided a recess in the end of said carrier facing the engine combustion chamber and said valves are located in a plate fitting snugly within said recess.

4. The device defined in claim 1 including resilient means for constantly but yieldably urging said valve away from said combustion chambers.

5. The device defined in claim 1 wherein a cantilever spring is mounted at one end fixedly with respect to said carrier for acting at its other end to urge said valve in a closing direction.

6. The device defined in claim 2 wherein said inlet conduit means comprises one conduit including one of said inlet valves and extending from said source to one of said chambers and includes also another conduit including the other of said inlet valves extending from said source to the other of said chambers.

7. The device defined in claim 1 wherein said relief valve includes a disk having a surface normally exposed to pressure fluid from said first chamber and an opposite surface exposed to said low pressure point, an annular first seat for engaging said surface and an annular second seat surrounding said first seat, said disk being of sufficient diameter to engage both of said seats simultaneously, means for resiliently urging said disk against said seats to close said relief valve and an annular groove between said seats whereby pressure from said first chamber will be effective against a larger area of said disk valve when said relief valve is opened than when same is closed.

8. A piston having a central axis including compression ratio adjusting means for reciprocation in an internal combustion engine cylinder and including:

a carrier and means for pivotally associating said carrier to a connecting rod;

a hood telescoped to said carrier for adjustment with respect thereto in a compression ratio adjusting manner, said hood defining with said carrier first and second pressure chambers so arranged that pressurizing said first chamber will move said hood to increase the compression ratio and pressurizing the second chamber will move said hood to decrease the compression ratio;

a source of fluid under pressure; and

a valve and conduit system for controlling the quantity of pressure fluid introduced into said respective chambers, the improvement in said valve and conduit system comprising:

a first fluid conducting system including a one-way first inlet valve connecting said source and said first chamber;

a second fluid conducting system including a one-way second inlet valve connecting said source and said second chamber;

said inlet valves being oppositely sensed with respect to each other and each being movable for opening and closing only in a direction parallel to the central axis of the piston;

opening means continuously connecting said second chamber with a low pressure point of the fluid pressure system;

a third fluid conducting system including a one-way relief valve connecting said first chamber to a low pressure point of the fluid pressure system, said relief valve being movable with respect to said piston for opening substantially in the direction in which said piston moves to compress gases in said cylinder;

whereby the frictional relationships between said valves and the means supporting same will be independent of the speed of reciprocation of said piston.

9. The device defined in claim 8 wherein said first inlet valve includes a surface positioned for being subject to pressure within said first chamber and an opposite surface exposed to the pressure from said source, and a seat for engaging said opposite surface.

10. A piston including compression ratio adjusting means for use in an internal combustion engine and including:

a carrier and means for pivotally associating said carrier to a connecting rod;

a hood closed on one end telescoped to said carrier for adjustment with respect thereto in a compression ratio adjusting manner, said hood defining with said carrier first and second pressure chambers so arranged that pressurizing said first chamber will move said hood to increase the compression ratio and pressurizing the second chamber will move said hood to decrease the compression ratio;

a source of fluid under pressure; and

a valve and conduit system for controlling the quantity of pressure fluid introduced into said respective chambers, the improvement in said valve and conduit system comprising:

inlet conduit means including a one-way inlet valve for connecting said source to said first chamber;

opening means connecting said second chamber with a low pressure point in said system;

outlet conduit means including a one-way relief valve for connecting said first chamber to said second chamber, said relief valve being movable toward said closed end of said hood for opening said relief valve on a path parallel to the central axis of the piston;

whereby said first chamber is pressurized from said source and said second chamber is pressurized from said first chamber.

11. The device defined in claim 10 wherein the maximum area of said second chamber is substantially similar to the area of said first chamber whereby said first chamber may empty into said second chamber.

12. The device defined in claim 1 wherein said relief valve includes a Belleville spring urging said relief valve toward a closed position;

whereby, upon the pressure in said first chamber reaching a sufficiently high level, said Belleville spring will allow said relief valve to pass from a fully closed condition to a fully open condition virtually instantly.

13. The device defined in claim 1 wherein said relief valve includes a valve disk, a seat against which said valve disk rests when said relief valve is in a closed condition and resilient means urging said valve disk toward said seat, said valve disk being movable away from said seat in the same direction that said piston moves in to achieve its top dead center position, said valve disk being sufficiently massive as to be urged away from said seat when said piston is near said top dead center position during high speed running of said engine;

whereby the tendency of said hood to lessen the pressure in said first chamber during high speed running of said engine and thereby reduce the amount of pressure fluid released from said first chamber through said relief valve is essentially cancelled.

14. In a compression ratio adjusting piston reciprocable in an internal combustion engine cylinder having a combustion chamber, the combination comprising:

a source of fluid under pressure;

means defining a first pressure chamber pressurizable from said source to increase the compression ratio of said cylinder and a second chamber pressurizable to decrease the compression ratio of said cylinder;

means for supplying pressure fluid to said second chamber;

check valve means for connecting said source with said first chamber;

means communicating with said second chamber for lowering the pressure thereof;

relief valve means connected to said first chamber and openable for allowing removal of pressure fluid therefrom, said relief valve means including an inertia member movable with respect to said piston substantially toward said combustion chamber for opening said relief valve means and substantially away from said combustion chamber for closing said relief valve means;

whereby inertia forces on said inertia member assist opening of said relief valve means to an increased extent with increasing piston speed.

15. The device defined in claim 2 in which:

said first inlet valve comprises a seat facing toward

said closed end of said hood and a disk movable away from said seat toward said closed end for opening said one inlet valve in response to a drop in pressure from said source to said first chamber thereacross;

said second inlet valve comprises a further seat facing away from said closed end, a further disk movable axially away from said seat for opening said second valve and resilient means for urging said further disk against said further seat, said second valve opening in response to a drop of pressure in said second chamber by a predetermined amount below the pressure of said source.

16. In a compression ratio adjusting piston reciprocable in an internal combustion engine cylinder having a combustion chamber, the combination comprising:

a source of oil under pressure;

a piston body including a first pressure chamber pressurizable from said source to increase the compression ratio of said cylinder;

means defining a second chamber pressurizable to decrease the compression ratio of said cylinder;

means defining a first passage between said source and said first chamber;

a first inlet valve disposed in said first passage for opening and closing same, said first valve having a valve disk of relatively light weight arranged for sliding motion axially of said piston, and a valve seat disposed between said source and said disk and closable by said disk to close said valve, said disk being disposed on the combustion chamber side of said seat;

means defining a second passage between said source and said second chamber;

a second inlet valve disposed in said second passage for opening and closing same, said second valve comprising a second valve disk of relatively light weight axially reciprocable with respect to said piston, a seat axially aligned with said second valve disk on the source and combustion chamber side thereof and resilient means for urging said second disk into a closed position;

an aperture from said second chamber for allowing escape of oil therefrom at a controlled, relatively slow rate;

means defining a third passage from said first chamber for venting same and a relief valve in said third passage for opening and closing same, said relief valve comprising a third valve disk axially reciprocable with respect to said piston, a valve seat comprising a pair of radially spaced annular ridges engageable by said third disk for closing said valve so that a greater pressure is required to open said relief valve than to close same and a cantilever spring fixed at one end with respect to said piston and engaging said third disk at the other end thereof for urging same toward said seat, said third disk being disposed on the combustion chamber side of said seat and having a preselected weight so that inertia thereof allows said relief valve to be opened by lower pressures in said first chamber as the piston speed rises.

References Cited by the Examiner

UNITED STATES PATENTS

2,742,027 4/56 Mansfield -------------- 123—78

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