

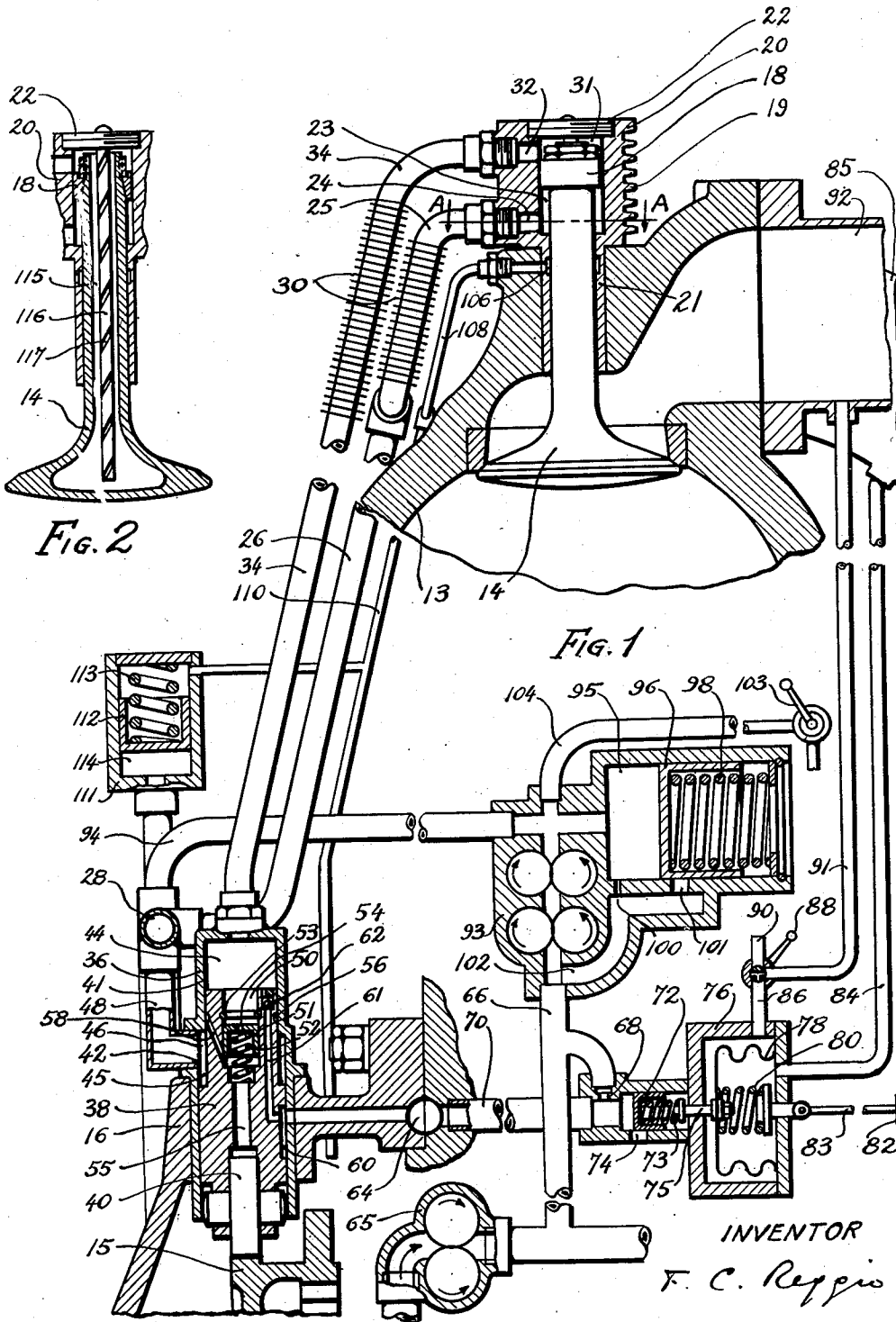
Dec. 16, 1952

F. C. REGGIO
HYDRAULIC VALVE-OPERATING SYSTEM OPERABLE
TO VARY VALVE LIFT AND TIMING

2,621,640

Filed Jan. 20, 1943

4 Sheets-Sheet 1



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4 Sheets-Sheet 2

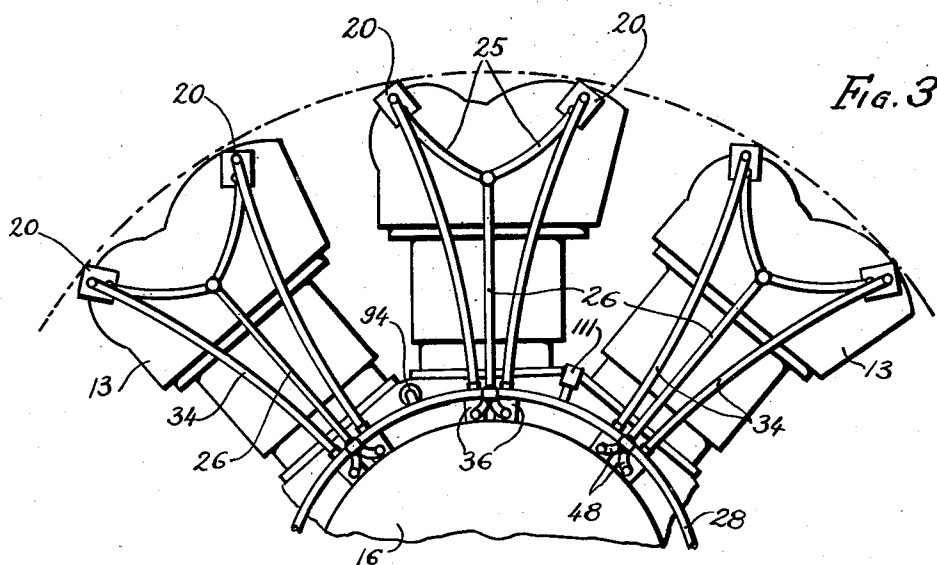


Fig. 3

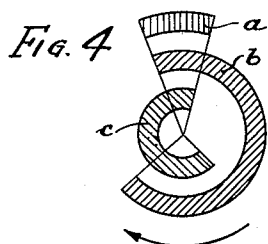


Fig. 4

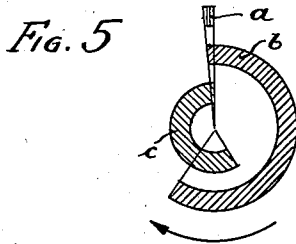


Fig. 5

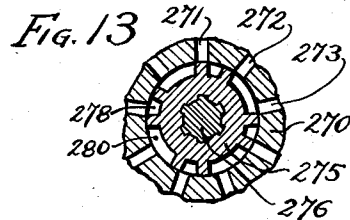


Fig. 13

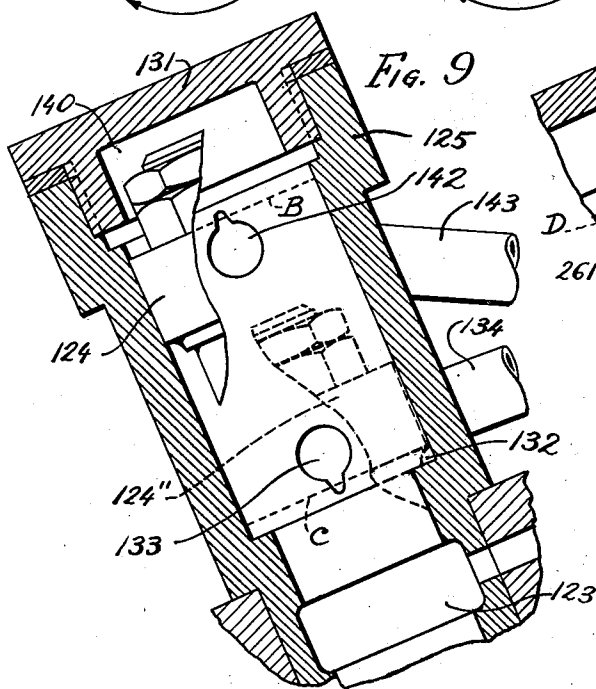


Fig. 9

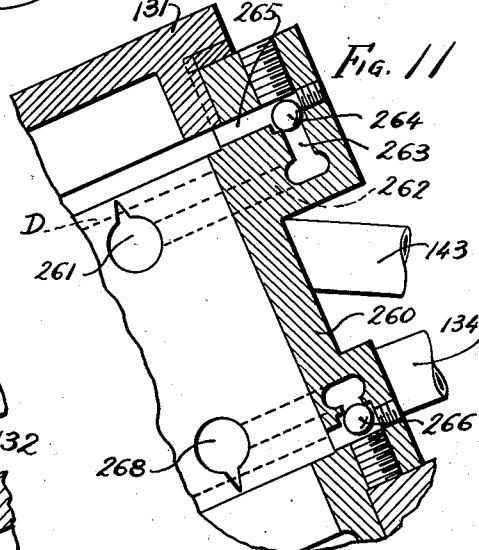


Fig. 11

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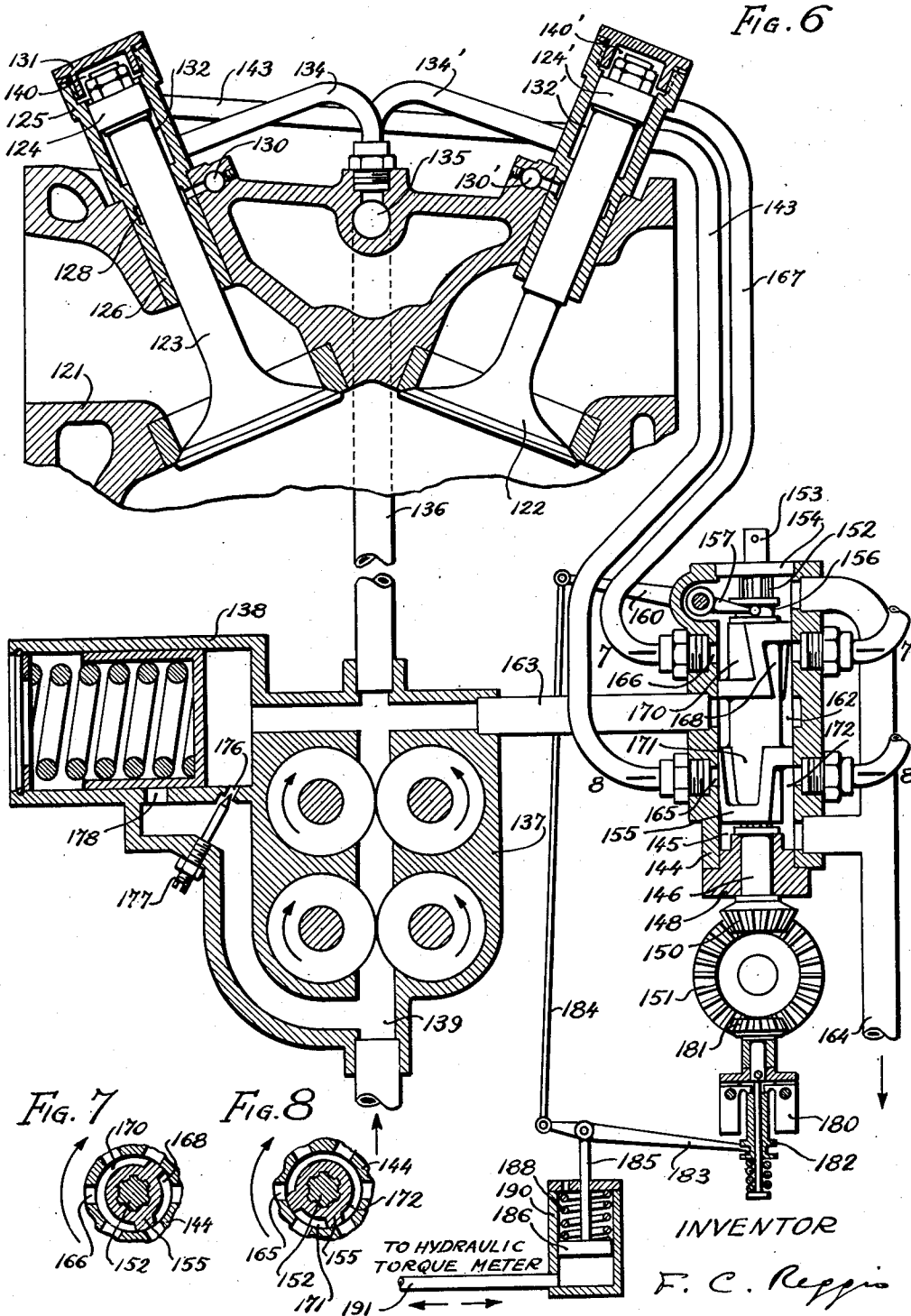
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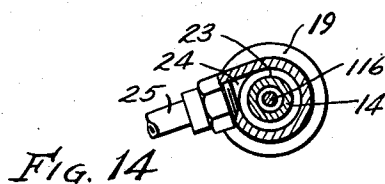
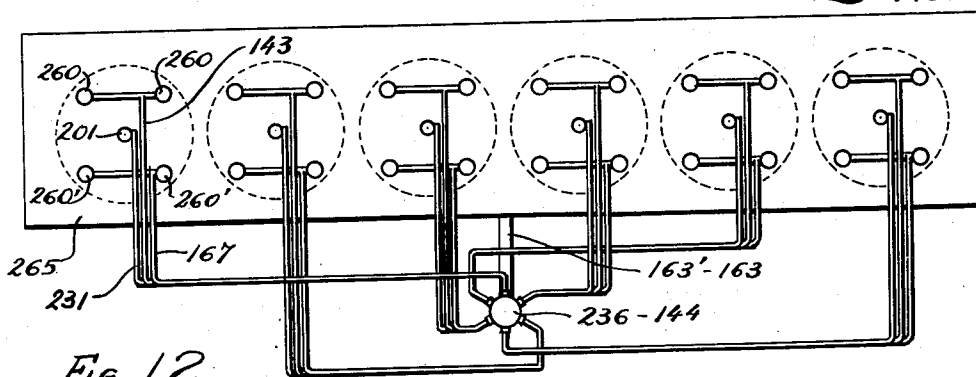
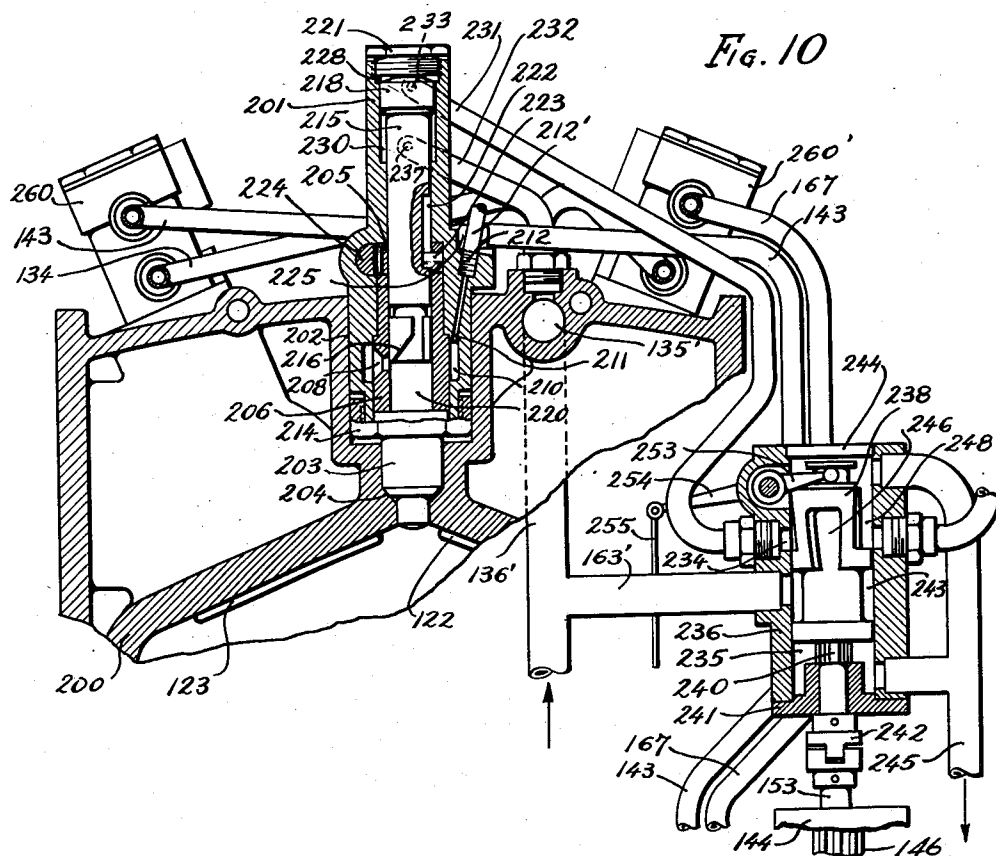
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4 Sheets-Sheet 4



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UNITED STATES PATENT OFFICE

2,621,640

HYDRAULIC VALVE-OPERATING SYSTEM
OPERABLE TO VARY VALVE LIFT AND
TIMING

Ferdinando Carlo Reggio, Buffalo, N. Y.

Application January 20, 1943, Serial No. 472,947

13 Claims. (Cl. 123—90)

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This invention relates to the fluid pressure actuation of reciprocating members and more particularly to mechanisms for hydraulically operating the valves of internal combustion engines and the like, and the plungers of engine fuel injection pumps.

One object of the invention is to provide a light, compact and simple valve mechanism.

Another object is to provide a hydraulic mechanism permitting a considerable increase in the valve velocity.

A further object resides in the provision of a valve mechanism utilizing lubricating oil under high pressure for actuating and cooling the engine valves.

Still another object is to provide a hydraulic valve mechanism which permits a considerable reduction in the weight and size of the engine.

A further object is to provide, in substitution for the valve springs, means for applying a fluid pressure load to the valves tending to close the latter.

A still further object is to provide means for automatically altering the valve timing with changes of engine operative conditions.

Another object resides in the provision of a valve mechanism wherein the number and weight of the reciprocating parts is considerably reduced.

A further object resides in the hydraulic actuation of the plunger of a fuel injection pump.

Other objects and advantages of the invention will become apparent from the following description taken in connection with the accompanying drawings, in which:

Fig. 1 is a fragmentary vertical section showing one of the preferred embodiments of the invention applied to a radial air-cooled aircraft engine.

Fig. 2 is a fragmentary section through the valve of Fig. 1.

Fig. 3 is a fragmentary front view of the engine of Fig. 1.

Figs. 4 and 5 are diagrams indicating the valve timing under different engine operative conditions.

Fig. 6 is a section illustrating a second preferred embodiment of the invention applied to a liquid-cooled engine.

Figs. 7 and 8 are sections along the lines 7—7 and 8—8 of Fig. 6, respectively.

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Fig. 9 shows a detail of Fig. 6 in larger scale.

Fig. 10 is a section illustrating a third preferred embodiment of valve and fuel pump actuating mechanism according to the invention, applied to a liquid-cooled engine.

Fig. 11 is a section of a detail of Fig. 10 in larger scale.

Fig. 12 is a diagrammatic view of the mechanism of Fig. 10 when applied to a six cylinder in-line engine.

Fig. 13 is a fragmentary section through a rotary control valve for a nine cylinder radial engine.

Fig. 14 is a section along line A—A of Figure 1.

The illustrative example of embodiment shown in Fig. 1 includes conventional parts of a radial air cooled aircraft engine which will easily be recognized by those skilled in the art, namely a portion of cylinder head 13, a valve 14, and a cam 15 mounted within the engine housing 16 for actuating the valves. The valve 14 is slidably mounted in a guide 21 above which there is provided a cylinder 20 having a bore coaxial with the guide 21, and cooling fins 19. The cylinder 20, preferably carried by or attached to the guide 21, may be secured to the cylinder head 13 by screws, not shown. A piston 18, closely fitting the cylinder 20 and slidable therein, is attached to the stem of the valve 14 by means of a nut and lock nut. The annular chamber 23 below the piston 18 is connected by means of port 24 and pipes 25, 26 with an annular conduit 23, while the cylinder chamber 31 above piston 18, closed by a cover 22, is connected by means of port 22 and pipe 34 with the upper chamber 44 of a differential cylinder 36 having slidable therein a differential plunger 38 actuated through a roller 40 from the cam 15. The plunger has an upper portion of smaller diameter slidable in the bore 41 of the cylinder 36, and a coaxial lower portion of larger diameter slidable in the bore 42, thus providing an annular chamber 45 connected by means of port 45 and duct 48 with the annular conduit 28.

The plunger 38 is provided with an axial bore 50 wherein there is slidably mounted a piston 51 upwardly urged by a calibrated spring 52. A ring 53 seated in a groove formed in the bore 50 limits the upward displacement of piston 51. The space below the latter communicates by way of a drain conduit 55 with the engine crankcase.

An annular groove 56 formed in the plunger 38 between chambers 44 and 45 is connected by way of duct 59 with the conduit 55. A small orifice 54 may be provided in the piston 51 to establish flow communication between chamber 44 and the drain conduit 55. An axial groove 60 formed in the lower cylindrical portion of the plunger 38 communicates by way of a duct 61 formed in the latter with the chamber 44 through an orifice controlled by a ball or check valve 62 permitting oil flow from the groove 60 to the chamber 44 but preventing the return flow from said chamber back to the groove 60. The latter also communicates with an annular duct 64 which receives lubricating oil under pressure from an engine pump 65 through a pipe 66, a flow-restricting calibrated orifice 63 and a pipe 70. Connected with the latter there is provided a pressure regulating plunger valve 72 controlling a discharge orifice 74 and loaded by a calibrated spring 75 mounted between said valve and a spring seat carried by a slidable rod 75 coaxial with the spring 73, whereby the load of the latter and in turn the oil pressure in the duct 64 are determined by the axial adjustment of said rod 75.

The rod 75 extends through the wall of a housing 76 and is attached to the movable wall of a bellows 78 contained therein and connected with a spring 80 which may be adjusted by the operator by means of a manual control 82 and a suitable linkage 83. A pipe 84 connects the engine induction manifold 85 with the space within the bellows 78; and the space comprised between the housing 76 and the bellows 78 may be connected by means of a conduit 86 and a three-way cock 83 either with the pipe 90 open to the atmosphere, or with pipe 91 connected with the engine exhaust manifold 92. With the cock 83 as shown in Fig. 1 the bellows 78 is surrounded by the exhaust pressure; and if the cock is rotated 90° clockwise the bellows 78 will be surrounded by atmospheric pressure.

An engine driven pump, preferably of the high-pressure type having two or more stages and diagrammatically represented at 93, receives oil from a suitable reservoir, dump, or other source, for example from the discharge line 66 of the engine driven lubricating oil pump 65, and through pipe 94 supplies high pressure oil to the annular conduit 28 and to the oil accumulator and pressure regulator cylinder 95 having slidable therein a piston 96 loaded by a spring 98. The piston controls axially spaced orifices 100 and 101 through which the excess oil is discharged into a low-pressure return conduit 102 which, in the example shown in Fig. 1, is connected with the intake side of pump 93. A hand pump 103 may be provided to supply pressure oil to the accumulator 95, and thence to the annular duct 64, through pipe 104, and prime the hydraulic system before starting the engine.

The oil discharge of the volumetric engine-driven pump 93 is substantially proportional to the engine speed. When the engine is operating within a predetermined speed range the larger orifice 101 is covered by the piston 96 and said oil discharge minus the various leakages of oil which are negligible as compared with the output of pump 93, is forced through the small calibrated orifice 100. The oil velocity through this orifice is thus substantially proportional to the engine speed, and the corresponding drop of pressure between upstream and downstream sides of orifice 100 is therefore proportional to the square of the engine speed, thus maintaining the oil

pressure in the annular conduit 28 at a value which is also substantially proportional to the square of the engine speed. If said speed decreases below a predetermined lower value, the piston 96 covers part of the orifice 100 and prevents the oil pressure from dropping below a corresponding minimum pressure limit. Conversely, if the engine speed increases beyond the maximum rated speed the piston 96 uncovers the orifice 101 and prevents the oil pressure in the conduit 28 from increasing above a preselected corresponding maximum pressure limit.

In the preferred embodiment of the invention, the orifices 100 and 101 are so spaced and the spring 98 is so designed as to obtain the following operation. When the engine is stationary, the piston 96 is in its extreme left position and covers both orifices 100 and 101. As soon as the engine is cranked, the pump 93 discharges oil into the various lines and in the cylinder 95. Since no oil outlet is provided, the oil pressure builds up and moves the piston 96 to the right until the orifice 100 is partly opened or uncovered. As the capacity of the pump 93 is considerably larger than the various oil leakages on the discharge side of the pump even at very low engine cranking speed, as soon as the engine crankshaft is turned, the piston 96 assumes such a position as to partially open the discharge orifice 100. Under these conditions, the specific pressure of the oil on the discharge side of the pump 93 will of course be equal to the load of spring 98 (corresponding to said position of piston 96) divided by the piston area. It follows that even at the lowest cranking speed of the engine, enough oil pressure is maintained to properly operate the hydraulic valve gear, as set forth hereinafter.

As the cranking speed of the engine increases, the piston 96 moves further to the right to increase the open area of orifice 100, until the latter is completely uncovered. In the preferred embodiment of the invention this happens just when the engine operates at idling speed. It is therefore apparent that for all values of engine speed from a few R. P. M. up to idling speed the position of the piston 96 varies merely by an amount equal to the diameter of the small orifice 100. As a result, the load of spring 98, and the oil pressure which is proportional to the spring load, are approximately constant for all engine speeds up to idling speed. Within such a speed range the spring-loaded piston 96 and the orifice 100 operate like a conventional pressure regulating valve, uncovering more or less of the area of orifice 100 to accommodate the variations in the delivery of the volumetric pump 93, which delivery is proportional to engine speed, while the oil pressure upstream of the orifice and the velocity of flow of the oil through the uncovered portion of the orifice 100 remain approximately constant.

As already stated, at idling speed the position of piston 96 is such as to fully uncover the orifice 100. On the other hand, the larger orifice 101 is so located as to be uncovered by the piston 96 only when the engine speed exceeds the maximum designed engine speed. Thus for all values of engine speed comprised between idling and maximum speed, that is to say, throughout the normal operating range of engine speeds, the orifice 100 is fully open, while the larger orifice 101 is completely covered by the piston 96; and all of the oil delivered by the volumetric engine-driven pump 93 is discharged through the fully uncovered orifice 100. Thus for all engine speeds within said operating range the velocity of oil flow

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through orifice 100 is proportional to the engine speed, and it follows of course that the oil pressure upstream from the orifice 100, in the cylinder 95, is proportional to the square of the engine speed.

The annular chamber 23 in the valve actuating cylinder 20, connected with the conduit 28, serves to apply to the piston 18 an upward oil pressure load tending to close the valve 14. Within the preselected engine speed range said load is proportional to the square of the engine speed and therefore proportional to the inertia of the valve. The conventional spring or springs connected with the engine valve may thus be eliminated. In similar manner the annular chamber 45, also connected with the conduit 28, exerts on the plunger 38 a downward oil pressure load which acts against the inertia of said plunger to maintain at all times the roller 40 thereof in contact with the cam 15.

As the engine operates, oil is displaced at each cycle to and from the small annular chambers 23 and 45, and oil pressure pulsations occur which keep the piston 96 of the pressure accumulator 95 in continuous motion. In an engine having a large number of valves at different phases these pressure surges are very considerably reduced. In order further to weaken the intensity of these surges, pressure cushioning means such as chambers containing a compressible fluid or chambers having yielding wall means may be provided in connection with any suitable part of the hydraulic system, for example near the valve cylinders 20 or the differential plungers 38, or the conduit 28, as it will be obvious to those skilled in the art. An example of such cushioning means is illustrated in Fig. 1 in the form of a cylinder 111 having a chamber 114 therein connected with the conduit 28 and defined in part by a slidable piston 112 loaded by a spring 113. The space at the opposite side of the piston 112 is connected by means of a duct with the drain pipe 110.

Lubricating oil leaking from the high-pressure chamber 23 between the valve stem and the guide 21 is trapped in the groove 106 and led back to the engine sump by way of the drain pipes 108 and 110. Oil upwardly leaking from the chamber 45 is trapped in the groove 56 and returned to the sump by means of the ducts 58 and 55.

Fig. 1 shows the valve 14 in closed position, the plunger 38 in its lowermost adjustment, and the small piston 51 in intermediate adjustment between its uppermost and lowermost positions relative to the plunger 38, the load of the piston spring 52 being balanced by the oil pressure in chamber 44. As the plunger 38 is lifted by the rotating cam 15 the oil pressure in said chamber increases and displaces the piston 51 downwardly against the load of spring 52, whereupon oil is forced through pipe 34 into the chamber 31 and lifts the valve 14. The deceleration and return stroke of the latter are determined by the upward load exerted on the valve piston 18 by the high pressure oil contained in the annular chamber 23. When the valve 14 closes the plunger 38 is still in its return stroke, the oil pressure in chamber 44 drops to a low value, and the spring 52 expands and lifts the piston 51. However during this cycle some oil has escaped through the small orifice 54 and various other oil leakages have occurred, and the piston 51 will therefore attain a higher position relative to the plunger 38 than it had at the beginning of the cycle. The oil pressure in chamber 44 will accordingly be lower than in the duct 64, causing the check valve 62

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to open and admit a certain amount of oil into chamber 44.

It is thus apparent that the displacement of the piston 51 has the effect of rendering a portion of the stroke of plunger 38 inoperative, causing the valve 14 to open later and close earlier. If the pressure in the annular duct 64 is sufficiently low the piston 51 will attain its upper most position and rest against the ring 53 when the plunger 38 is at rest between lifts, and the valve 14 will remain open during the minimum designed duration. Conversely, if the pressure in conduit 64 is maintained at sufficiently high value so as to keep the piston 51 constantly in its lowermost position, the latter becomes ineffective and the period during which the valve 14 remains open reaches its maximum designed value. By this arrangement the valve timing may be controlled while the engine is in operation as shown diagrammatically in Figs. 4 and 5, wherein *b* and *c* indicate the intake and exhaust phases respectively, and *a* represents the scavenging phase during which both intake and exhaust valves are open. The former figure, corresponding to the higher value of the oil pressure in the annular duct 64, obtained when the rod 75 is in its extreme left adjustment, shows a considerable scavenging phase or valve overlap, during which the cylinder may be swept by the air or mixture from the supercharger. Conversely, if the rod 75 is adjusted in its extreme right position, the oil pressure in duct 64 is kept at the lower designed limit, and the scavenging phase is reduced to zero or to a negligible value as shown in Fig. 5.

The bellows 78 cooperating with the spring 89 actuates the rod 75 to increase or decrease the duration of the scavenging phase or valve overlap as the pressure in the engine induction manifold 85 becomes higher or lower than that in the exhaust pipe 92. With the three-way cock 88 rotated 90° clockwise, the bellows 78 becomes responsive to the difference between the engine manifold pressure and the surrounding atmospheric pressure, and operates the rod 75 to reduce the valve overlap to a negligible value when said pressure difference becomes negative. The return flow through the engine cylinder at the end of the exhaust stroke from the exhaust pipe 92 to the induction manifold 85 when the pressure in the latter is low, as when the engine is idling, may thus be avoided. The valve timing may also be adjusted by the operator while in flight by means of the control member 82.

In Fig. 2 there is indicated a valve 14 having a hollow stem or cavity 115. A rod 116 carried by the cover 22 extends therein leaving an annular interspace through which during engine operation oil flows back and forth at high velocity and thus effectively cools the valve by transferring heat by conduction and convection from the valve head to the stem portion thereof, in substantially the same way as in the sodium-cooled valves which are extensively used in aircraft engines. A helical groove 117 is provided in the rod 116, and ports 24 and 32 are preferably directed tangentially with respect to the cylinder 20 as shown in Figure 14 to impart rotary motion to the oil therein and further improve the cooling of the valve 14. Oil cooling devices, such as fins 19 and 30, are provided in connection with the cylinder 20 and the oil conduits 24 and 25.

Fig. 3 diagrammatically illustrates the various hydraulic connections in the valve mechanism of a radial aircraft engine, wherein the cylindrical chamber 31 of each valve cylinder 20 is connect-

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ed with the corresponding chamber 44 of a plunger housing 36 by means of a pipe 34, while the annular chambers 23 and 45 are all connected by means of pipes 25—26 and 48, respectively, with the annular conduit 28 which is provided with one or more pressure cushion chambers 111 and receives high pressure oil through the line 94.

The mass of valve 14 with piston 18 attached thereto is only a fraction of the total mass of a conventional valve with the reciprocating mechanism connected therewith, and accordingly the load exerted on piston 18 by the oil pressure in the annular chamber 23 need be only a small fraction of the load of the conventional valve springs. If high pressure oil is employed, the bore of cylinder 20 may be made only slightly larger than the stem of the valve 14. Furthermore, owing to the fact that only axial load is applied to the valve and that the latter is cooled by means of lubricating oil, the valve guide 21 may be shortened. The cylinders 20 may therefore be made very small as compared with the relatively bulky conventional valve rocker housings, and the frontal area and the weight of the engine may be considerably decreased. In addition, the cylinder head becomes more simple, and more space in the immediate vicinity of the combustion chamber becomes available for cooling fins.

In the above described mechanism each valve is actuated from a corresponding cam-actuated plunger by means of a substantially incompressible liquid column, the valve lift being determined by the cam profile. Fig. 6 illustrates another preferred embodiment of the invention wherein an engine-driven rotary control valve is substituted for said cam and plungers. The cylinder head 121 of a liquid-cooled, valve-in-head engine is provided with intake and exhaust valves 122 and 123. The latter valve has attached thereto a piston 124 slidable in a cylinder 125 attached to and coaxial with the guide 126 wherein there is formed a groove 128 connected with a drain duct 130. The upper end of the cylinder 125 is closed by a threaded cover 131 and the chambers 140 and 132 on opposite sides of the piston 124 communicate by means of ports 142 and 133, shown in Fig. 9, with the pipes 143 and 134 respectively. Similarly, the intake valve 122 carries a piston 124' slidable in a cylinder having upper and lower chambers 140' and 132' communicating with pipes 167 and 134' respectively. Said pipes 134 and 134' are connected with a liquid cooled conduit 135 communicating by means of a duct 136 with the discharge side of a high pressure engine-driven oil pump 127 diagrammatically indicated as a two-stage gear pump, and with an oil accumulator and pressure regulator 138 similar to the unit 95 and having similar functions, namely to return the excess oil to the inlet side 139 of the pump 127, to maintain the oil pressure substantially proportional to the square of the engine speed between predetermined maximum and minimum values thereof, and to cushion the pressure surges in the high-pressure system. The oil pressure in chambers 132 and 132' constantly exerts on the valves 123 and 122 a load tending to close the latter.

The pipes 167 and 143 are connected with ports 166 and 165 opening in the upper and lower parts of a cylindrical bore 145 formed in the housing 144 of a rotary control valve having a rotor 155 therein slidably mounted on a splined portion 152 of a shaft 146 journaled in the lower valve cover

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148 and carrying a pinion 150 meshing with an engine-driven gear 151. The upper portion 153 of the shaft 146 is journaled in and extends through the upper cover 154 of the valve. In this valve arrangement, designed for a four cycle, in-line engine, the shaft 146 revolves at half crankshaft speed. The rotor 155 is provided with a groove 156 engaging a lever 157 connected with an external lever 160 whereby, by moving the latter, the axial adjustment of the rotor 155 relative to the ports 166 and 165 may be varied. A groove formed in the central portion of the rotor provides an annular chamber 162 which by means of a conduit 163 is kept in constant communication with the discharge side of the pump 137. The upper and lower portions of the bore 145 are connected with a drain conduit 164 for leading low pressure oil back to the engine sump or other suitable reservoir.

The rotor 155 in its upper portion is provided with two grooves 168 and 170, the former communicating with the groove 162 and the latter with the low pressure drain conduit 164. As the rotor 155 revolves these two grooves 168 and 170 successively register with the port 166 thereby successively connecting the chamber 140' with the high pressure side of the pump 137 and with the drain conduit. In similar manner the lower portion of the rotor 155 is provided with two grooves 171 and 172 adapted successively to register with port 165, the former groove communicating with the groove 162 and the latter with the drain conduit 164, so that in operation the cylinder chamber 140 is also successively connected with the high and low pressure oil lines 163 and 164. The controlling edges defining said four grooves 168 and 170 to 172 are, in part at least, not parallel to the axis of the rotor but formed in such manner as to vary the duration of the intervals during which the ports 165 and 166 are connected with the high and low pressure oil lines when the axial adjustment of the rotor 155 is changed.

With the rotor 155 in the angular position shown in Figs. 6 to 8 the cylinder chambers 140 and 140' are connected by means of lines 143 and 167, ports 165 and 166, and grooves 170 and 172, respectively, with the low pressure drain conduit 154, and the valves 123 and 122 are kept closed by means of the high pressure oil contained in the annular chambers 132 and 132', respectively. As the rotor 155 revolves in the direction indicated by the arrows, the connection between port 165 and groove 172 is interrupted and connection between said port and the groove 171 is established, and high pressure oil from the pump 137 and the accumulator 138 is admitted through the pipe 143 to the cylinder chamber 140, thus determining downward acceleration of the differential valve piston 124. As the resultant hydraulic load applied to said piston is approximately constant during the greater part of the lift, the valve acceleration will accordingly be approximately constant.

As shown in Fig. 9, at the beginning of the valve lift only the small portion of port 142 above the dotted line B is open, the remaining portion thereof being covered by the piston 124. However the valve velocity is low at the very beginning of the lift, and even the small effective area of the port is sufficient for admitting the necessary volume of oil without causing any appreciable drop of pressure in chamber 140. As the lift and the velocity of the valve 123 increase, the piston 124 uncovers the port 142 and the ef-

fective area thereof attains its maximum designed value. At the same time oil is displaced from the annular chamber 132 through the port 133; and as toward the end of the downward stroke of the valve the piston 124 approaches and attains its lowermost position indicated in dotted line at 124'' and covers the portion of said port above the dotted line C, the effective open area of the latter decreases, determining a rapid increase of oil pressure in chamber 132 and causing quick deceleration of the valve so as to reduce the impact speed of the piston 124 against the end wall of the cylinder 125 to a suitable value. The high pressure in chamber 140 keeps the valve in open position until the communication between the port 165 and the high pressure groove 171 is interrupted and the connection between said port and the low pressure groove 172 is established; whereupon the pressure in chamber 140 drops and the high pressure constantly existing in the annular chamber 132 causes upward displacement of the valve 123. Toward the end of this stroke, owing to the decrease in the effective area of port 142, the oil pressure in chamber 140 rises and causes rapid deceleration of the valve 123 so as to reduce the speed of impact of the latter against the valve seat to an appropriate value. Thereafter the valve is kept closed by the oil pressure in chamber 132 until high pressure oil is again admitted to the chamber 140 by the revolving rotor 155.

The intake valve 122 is actuated by means of the upper part of the rotor 155, controlling the port 166, in similar manner, and it is therefore regarded as unnecessary to again describe its operation in detail.

The oil delivery of the engine-driven pump 137 is proportional to the engine speed and exceeds the total oil displacement of the valve cylinders 125 which is also proportional to the engine speed. The excess oil, which therefore is also proportional to the engine speed, returns to the inlet side 139 of the pump through an orifice 176 which may initially be adjusted by means of a needle valve 177. It follows that the oil velocity through the orifice 176 is proportional to the engine speed, and the corresponding pressure differential between upstream and downstream sides of orifice 176 is proportional to the square of the engine speed; that is to say, the oil in the high pressure system, including chambers 132 and conduits 135, 136 and 163, is maintained at a pressure proportional to the square of the engine speed whenever the engine is operating within a predetermined speed range. The upper limit of said range is attained when, under abnormally high engine speed, the resiliently loaded plunger which is slidably mounted in cylinder 138 uncovers a large port 178 and thereby prevents further increase of oil pressure. The lower limit of said range is attained at a predetermined low engine speed, for example engine idling speed, as the plunger covers the orifice 176 in part so as to reduce the effective area thereof and thereby prevents further drop of oil pressure. The actuating hydraulic load applied to the valves by means of this pressure and the positive acceleration thereof are therefore proportional to the square of the engine speed, and the valve velocity is directly proportional to the engine speed. Thus the curve representing the valve lift versus the corresponding angular position of the crankshaft is independent of the engine speed, exactly as if the

valves were reciprocated from a cam. In the present case, however, the valve velocity may be much higher than in the conventional valve mechanism, as the valve deceleration and inertia are not limited by the load of the valve spring. Furthermore the mass of the valve and piston 124, the only reciprocating parts, and of the oil column movable therewith, is only a fraction of the mass of the conventional valve with the reciprocating mechanism connected therewith.

With the rotor 155 having high and low pressure grooves 168 and 170 to 172 as illustrated in Fig. 6 the valve timing and the valve overlap or scavenging phase may be varied by altering the angular position of lever 160 and thereby the axial adjustment of the rotor 155 relative to the ports 165 and 166. A speed responsive device including centrifugal flyballs 180 driven from the engine by means of gears 151 and 181 actuates a resiliently loaded sleeve 182 connected with a floating lever 183 and displaces the right end of the latter downwardly or upwardly upon an increase or decrease of engine speed, respectively. An intermediate point of lever 183 is pivoted to a rod 185 secured to a piston 186 loaded by a spring 18 and slidably in a cylinder 190 connected by means of an oil conduit 191 with a hydraulic torque meter, not shown in the drawing, whereby the piston 186 is displaced upwardly or downwardly upon an increase or decrease of engine torque, respectively. The left end of lever 183 is connected by means of a rod 184 with the lever 160 so as to vary the engine valve timing in dependence upon changes of engine speed and torque, and in particular to reduce the valve overlap as the engine speed, or torque, or both decrease.

If desired, the oil pressure in the high pressure system may be maintained constant regardless of the engine speed by closing the orifice 176 by means of the needle valve 177.

The valve mechanism according to the invention may be used for actuating reciprocating valves of compressors, pumps and the like, as will be obvious to those skilled in the art, and also for actuating other reciprocating members such for example as the plungers of fuel injection pumps as illustrated in Fig. 10, wherein the cylinder head 200 of an engine is provided with a passage 202 in which there is mounted a fuel injector 201 having a nozzle port 203 clamped on a seat 204 and adapted to intermittently discharge fuel under pressure into the cylinder. Mounted in coaxial bores formed in the injector housing 201 there are a rotatable pinion 205 provided with external gear teeth meshing with a fuel control rack 224 transversely slidable with respect to the axis of said pinion, and a sleeve 206 having at least one port 208 connected with a groove 210 to which fuel is led by means of a duct 211 from a port 212 adapted for connection with a fuel line 212'. The nozzle part 203 and the sleeve 206 are tightly clamped by means of a threaded annular retaining member 214. A plunger 215, slidably in the cylindrical cavity provided in the sleeve 206, pinion 205 and injector housing 201, has an inclined or helical scroll 216 for controlling the port 208 so as to vary the fuel discharge of the injector upon changes of angular adjustment of said plunger. A piston 218 of larger diameter than plunger 215, formed integral with or secured to the upper end of the latter, slidably fits in a cylinder machined in the upper portion of the housing 201 and closed by a threaded cover 221, thus providing a chamber

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228 and an annular chamber 233 above and below the piston 218, respectively.

The plunger 215 is shown in its uppermost position, with the piston 218 in contact with the cover 221, in which the fuel pressure chamber 220 is in communication with the annular fuel groove 210. As the plunger 215 is forced downwardly, it closes the port 208 and forces fuel from the chamber 220 through the nozzle 203 into the engine cylinder. The lowermost position of plunger 215 is attained when the piston 218 reaches the lower end of the chamber 230. A longitudinal slot 222 formed in the intermediate portion of the plunger 215 is engaged by the inner end of a pin 223 forced into a radial bore formed in pinion 205 through a suitable opening provided in the housing 201, not shown in the drawing, whereby the angular adjustment of the plunger and in turn the fuel discharge of the injector are determined by the axial adjustment of the control rack 224, as it is well known in the art. The injector 201 is secured to the engine by means of fastening means such as screws 225.

The chambers 228 and 230 are connected by means of ports 233 and 237 which may be similar to the ports 142 and 133 of Fig. 9, with the pipes 231 and 232 respectively. The latter pipe is connected with the high pressure oil conduit 135', similar to the conduit 135 of Fig. 6, whereby an upwardly directed pressure load is constantly exerted on the piston 218. The former pipe 231 is connected with a port 234 opening in the bore 235 of a rotary control valve 236 including a rotor 238 slidably mounted on a splined shaft 240 journaled in the lower cover 241 and driven from the upper end 153 of the shaft 146 (shown in Fig. 6) of the control valve 144 by means of a coupling 242. The rotor 238 has a groove 243 constantly communicating with the high pressure oil conduit 163', similar to conduit 163 of Fig. 6. A cover 244 closes the bore 235 at the upper end thereof, and a low pressure drain conduit 245 is connected with the upper and lower ends of bore 235. The rotor 238 has two grooves 246 and 248 communicating with the high and low pressure lines 163' and 245 respectively, and as the rotor 238 revolves said two grooves successively register with the port 234 and cause the plunger 215 to reciprocate in substantially the same manner as has been described in detail in connection with the valves 123 and 122. As the controlling edges of the grooves 246 and 248 are not parallel to the axis of the rotor 238, the injection timing may be varied by altering the axial adjustment of said rotor relative to the port 234. To that end the rotor 238 is provided with a groove engaged by a lever 253 connected with an external lever 254 actuated by way of the rod 255.

The valves 122 and 123 in Fig. 10 are the same as in Fig. 6, while the valve actuating cylinder 260' or 260, the latter of which is shown in section in Fig. 11, has an upper port 261 including a main circular portion and a comparatively very small triangular portion so designed that when the upper edge of the valve piston 124 approaches its uppermost position indicated by the upper dotted line D, the effective open area of the orifice 261 becomes sufficiently small to cause quick deceleration of the valve before the latter reaches its seat, thus rendering possible the adoption of valve velocities many times as great as those employed in conventional high-speed engines. However with such

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form of port the initial part of the valve lift may be slowed down by the throttling effect of the small initial effective area of the port. To obtain a high value of deceleration prior to the seating of the valve as well as unrestrained initial valve lift acceleration, a second communication is provided between the pipe 143 and the upper part of the cylinder chamber 140 by means of conduits 262, 263 and 265, including a check valve 264 permitting flow of oil from the pipe 143 into the upper chamber but preventing flow of oil in the opposite direction. A similar arrangement including a check valve 265 is provided at the lower end of the annular chamber in connection with the lower port 268, whereby a high value of deceleration may be obtained at the end of the valve lift as well as unrestrained acceleration at the beginning of the valve closing stroke.

The sections of the rotary control valve 144 in Figs. 7 and 8 indicate six equally spaced radial ports in a plane perpendicular to the axis of the rotor 155. The latter rotates in the direction of the arrows at half crankshaft speed. This arrangement is intended for a six cylinder four-cycle engine. Fig. 12 diagrammatically indicates the hydraulic connections, by means of pipes 167 and 143, between the rotary control valve 144 (the upper fragment of which is shown in Figure 10) and the intake and exhaust valves 260' and 260, respectively, as well as the connections, by means of pipes 231, between the rotary control valve 236 and the fuel injectors 201 of a six-cylinder engine 265 wherein the ignition order is 1-4-2-6-3-5. It will be noted that each cylinder is provided with two exhaust valves 260 connected with the rotary valve 144 by means of a common pipe 143, and with two intake valves 260' connected with said rotary valve by way of a single pipe 167.

It will be obvious to those skilled in the art that a valve mechanism as shown in Fig. 1, wherein the valves are operated by means of cam-actuated plungers, may be used in connection with an in-line engine as well as with a radial engine, and conversely the valve mechanisms illustrated in Figs. 6 and 9 to 11, employing a rotary control valve, may be applied to a radial engine as well as to an in-line engine. Fig. 13 indicates a transversal section through the upper portion of a rotary control valve 270 for a nine-cylinder air-cooled radial aircraft engine, having nine equally spaced radial ports 271, 272, 273, etc., connected with the intake valves of the engine cylinders. Said ports are controlled by a rotor 275 driven by a splined shaft 276 at one-eighth crankshaft speed in anticlockwise direction. The rotor 275 is provided with four equally spaced grooves 278 connected with the high-pressure oil system, alternating with four grooves 280 communicating with the oil drain line. The rotary valve is otherwise substantially similar to valve 144, and it is therefore regarded as unnecessary again to describe it in detail.

The various cooling arrangements and oil pressure regulating and pressure surge cushioning devices disclosed in connection with any of the above embodiments may obviously be employed in connection with the others. Furthermore any suitable fluid may be used as hydraulic medium in substitution for the lubricating oil. And it is to be expressly understood that the invention is not limited to the specific embodiments shown.

but may be used in various other ways, and various modifications may be made to suit different requirements, and that changes, substitutions, additions and omissions may be made in the construction, arrangement and manner of adjustment and operation within the limits or scope of the invention as defined in the following claims. Certain features disclosed herein are claimed in application Serial No. 319,577 filed November 8, 1952 by the same applicant.

Where claims are directed to less than all the elements of the complete system disclosed, they are intended to cover possible uses of the recited elements in installations which may lack the non-recited elements.

What I claim is:

1. A reciprocable engine distribution member which moves at every engine cycle, means including an engine-driven fluid pump and a calibrated orifice on the discharge side of the pump for exerting on said member a fluid pressure load in one direction proportional to the square of the engine speed between preselected upper and lower limits of said speed, and means for applying to said member an intermittent actuating load directed against said fluid pressure load.

2. A reciprocable valve member having piston means, first fluid containing means for continuously exerting thereon a fluid pressure load in one direction, means including cooperating calibrated orifice means and a volumetric fluid pump to vary said fluid pressure load substantially in proportion to the square of the number of reciprocations per unit time of said member whereby said load varies substantially in proportion to the inertia of said member, second fluid containing means for intermittently applying a larger fluid pressure load in the opposite direction to said member for reciprocating the latter, a low pressure fluid system, and valve means for admitting fluid from said system to said second containing means to compensate for fluid leakage therefrom.

3. A reciprocable engine distribution member which moves at every engine cycle; means to actuate said member in one direction; first wall means movable with said member; fluid conduit means to exert variable fluid pressure on said wall means in the opposite direction; a calibrated orifice; an engine driven volumetric pump discharging fluid through said orifice to maintain in said fluid conduit means a pressure substantially proportional to the square of engine speed; second movable wall means subject to the fluid pressure in said fluid conduit means; and means for exerting on the second wall means a resilient load directed against said fluid pressure to decrease the intensity of pressure variations in said fluid conduit means upon motion of said member.

4. An engine valve having first wall means movable therewith to actuate the valve; a fluid pressure chamber on one side of the wall means to exert on the latter a pressure load tending to close said valve; second movable wall means capable of a volumetric displacement comparable to that of said first wall means and subjected to the pressure of the fluid in said chamber; means for applying to said second wall means a resilient load directed against said fluid pressure; and means for substantially increasing or decreasing the fluid pressure in said chamber upon increase or decrease of engine speed, respectively.

5. A reciprocable engine valve, a cylinder, piston means slidable in said cylinder for actuating said valve, high pressure fluid containing means

in constant communication with said cylinder for exerting on said piston means a fluid pressure load tending to close said valve, and means including an engine-driven fluid pump and a calibrated orifice respectively upstream and downstream from said fluid containing means for increasing or decreasing the pressure in said fluid containing means with an increase or decrease of engine speed respectively.

6. Engine valve mechanism including means for varying the valve timing, and engine induction and exhaust pressure responsive means for actuating said first mentioned means.

7. Engine valve actuating mechanism having means for controlling the valve overlap, and means for actuating said first mentioned means to decrease said overlap as the difference between the engine manifold pressure and the exhaust pressure decreases.

8. Reciprocable tappet means actuated from a rotative cam, means including a source of pressure fluid whose flow increases with the speed of said cam for continuously exerting a high pressure fluid load on said tappet means to force the latter against said cam and means including fixed-area orifice means in the path of said fluid flow for increasing said pressure fluid load with increase of speed of said cam.

9. A hydraulic valve mechanism including a reciprocable valve, a rotative cam, a plunger actuated from said cam for hydraulically lifting said valve, means including a source of pressure fluid whose flow increases with the speed of said cam for continuously exerting a pressure fluid load on said valve and plunger tending to close the former and to force the latter against said cam, and calibrated orifice means in the path of the flow of said fluid to increase the fluid pressure and thereby the load applied to said valve and plunger substantially in proportion to the square of the cam speed.

10. An engine valve, pressure responsive means exerting a continuous load on said valve in the direction to close said valve, means for cyclically lifting said valve, a volumetric fluid pump driven from said engine, a fluid flow restricting orifice having fixed flow area on the discharge side of said pump to determine therein a pressure proportional to the square of the engine speed within preselected speed limits, said pressure responsive means being subject to the fluid pressure on the discharge side of said pump, whereby the continuous load exerted on said valve varies proportionally to the square of the engine speed and thereby in direct proportion to the inertia of said valve.

11. A slidable valve having a closed position, a mechanism including rotating means to open said valve periodically, fluid pressure responsive means for constantly applying a load to said valve tending to close the latter, and means including a fluid pump delivering a flow of fluid substantially proportional to the speed of said rotating means and fluid flow restricting orifice means having calibrated flow area in the path of said fluid flow to vary the fluid pressure operating on said pressure responsive means in proportion to the square of the speed of said rotating means, whereby the valve closing load varies in proportion to the valve inertia.

12. Engine valve mechanism including valve timing control means, and engine exhaust pressure responsive means actuating said control means.

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13. Engine valve mechanism including valve timing control means, and engine exhaust pressure responsive means and manual control means actuating said timing control means.

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REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date	
885,459	Engler	Apr. 21, 1908	
1,316,977	Ricardo	Sept. 23, 1919	
1,474,794	Shephard	Nov. 20, 1923	15
1,529,201	Meredith	Mar. 10, 1925	
1,696,984	Trbojevich	Jan. 1, 1929	
1,787,120	Noble	Dec. 30, 1930	
1,798,938	Hallett	Mar. 31, 1931	
1,841,337	Roessler	Jan. 12, 1932	

Number
1,953,611
1,994,223
2,011,864
2,027,360
2,069,744
2,072,437
2,121,560
2,146,550
2,159,017
2,191,459
2,223,590
2,326,883
2,329,662

16

Name	Date
Jardine	Apr. 3, 1934
Leveque	Mar. 12, 1935
Lundh	Aug. 20, 1935
Alden	Jan. 14, 1936
Alden	Feb. 9, 1937
Wurtele	Mar. 2, 1937
Duncan	June 21, 1938
Pierce	Feb. 7, 1939
Duncan	May 23, 1939
Duncan	Feb. 27, 1940
Alden	Dec. 3, 1940
Pierce et al.	Aug. 17, 1943
Steiner	Sept. 14, 1943

FOREIGN PATENTS

Number	Country	Date
9,266	Great Britain	of 1903
105,373	Great Britain	Apr. 5, 1917
573,596	France	June 26, 1924