March 12, 1946. H. SCHRECK 2,396,392

HYDRAULIC OPERATION OF MECHANISM

Filed May 8, 1943 4 Sheets-Sheet 1

FIG. 1.

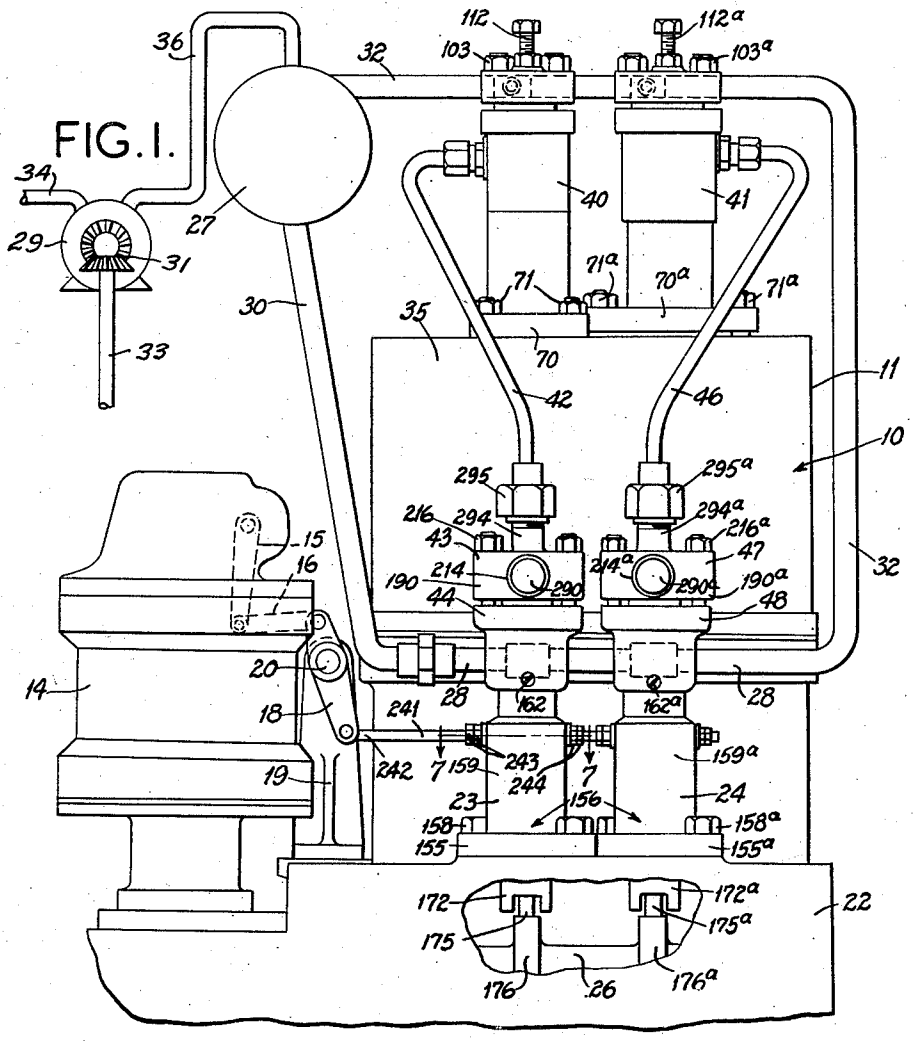
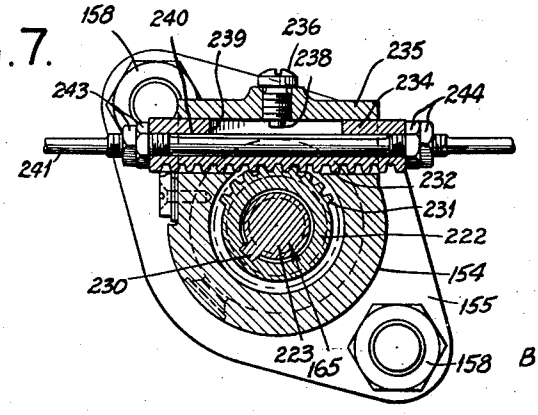


FIG. 7.



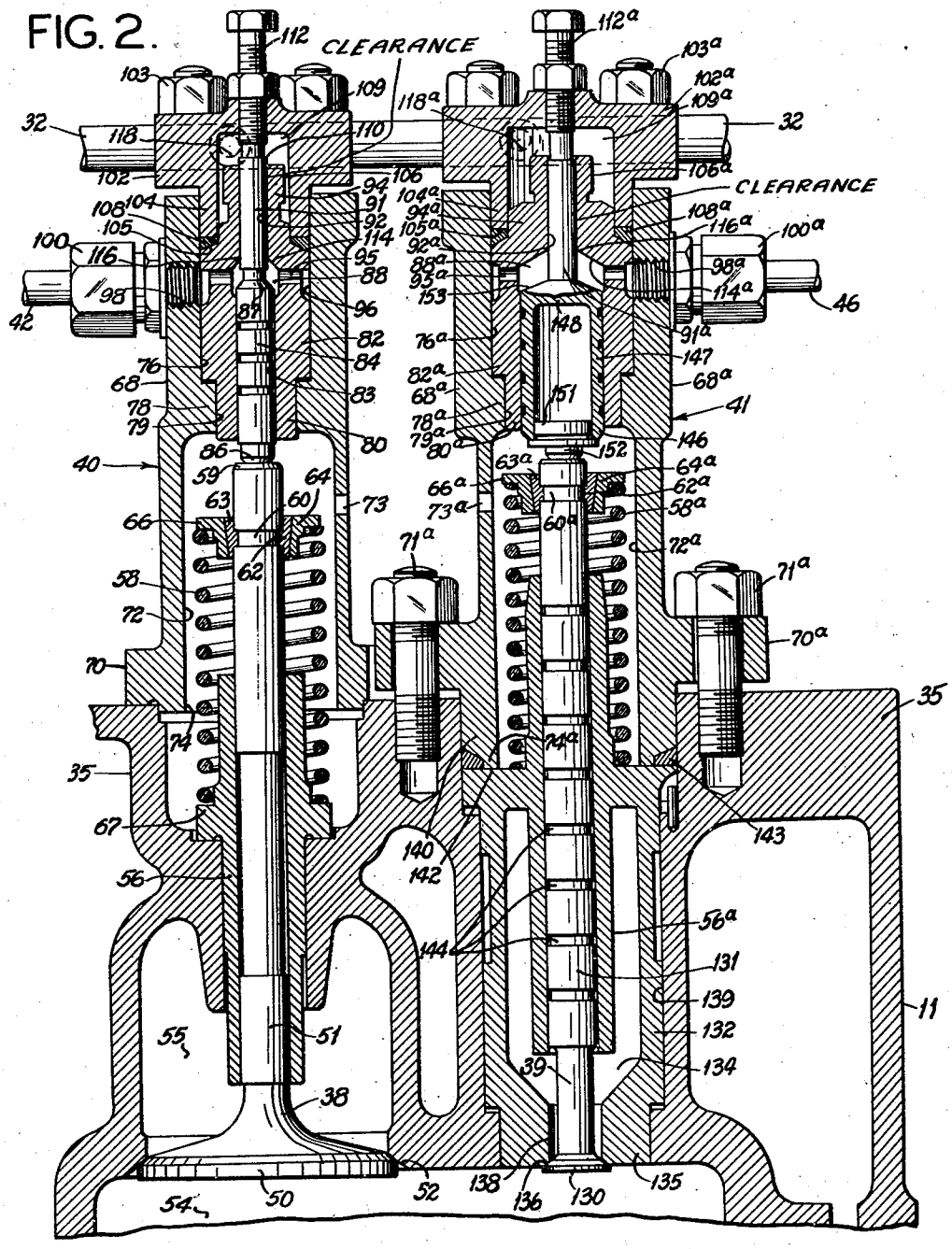
INVENTOR;
HENRY SCHRECK
BY Paul L. Kisher
ATTORNEY

March 12, 1946. H. SCHRECK 2,396,392

HYDRAULIC OPERATION OF MECHANISM

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FIG. 2.



INVENTOR;
HENRY SCHRECK
BY Paul L. Kuehn
ATTORNEY

March 12, 1946. H. SCHRECK 2,396,392

HYDRAULIC OPERATION OF MECHANISM

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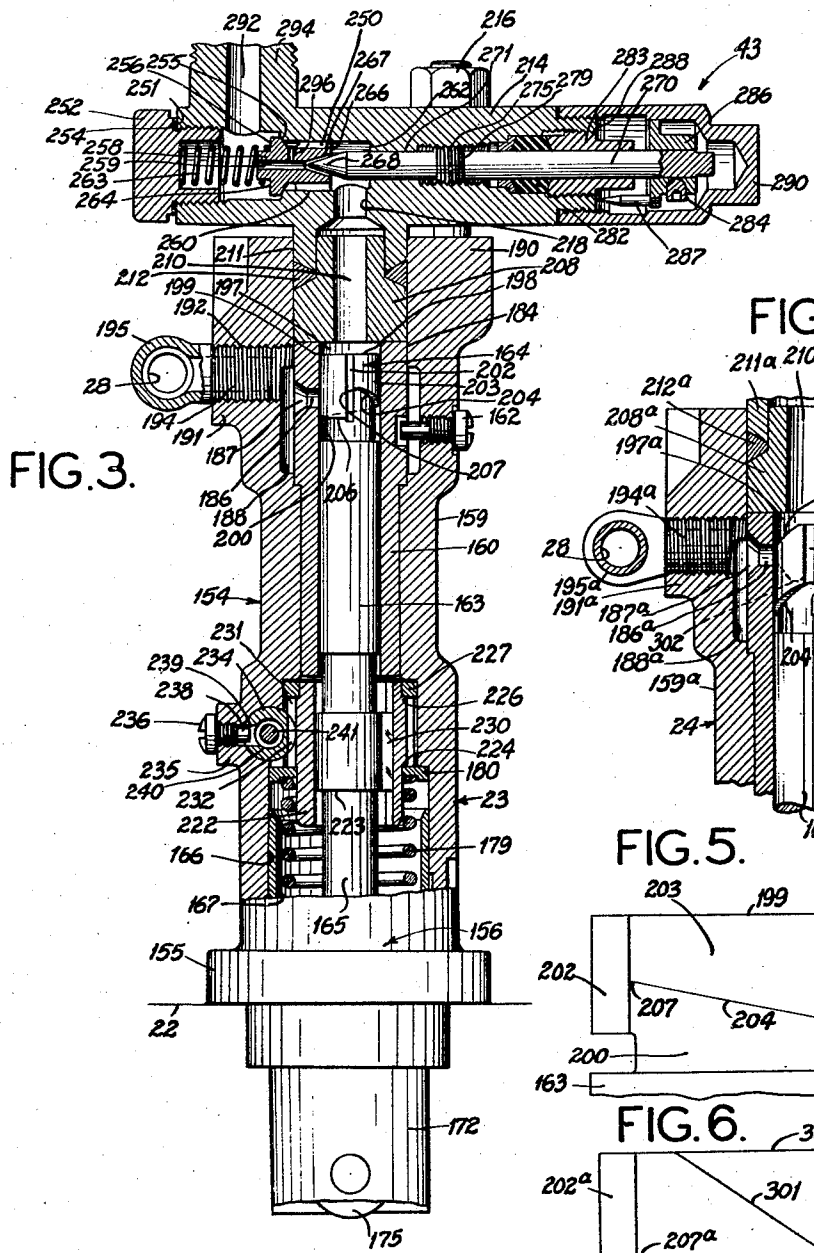


FIG. 3.

FIG. 4.

FIG. 5.

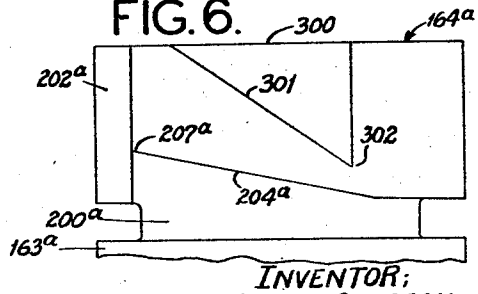


FIG. 6.

INVENTOR;
HENRY SCHRECK
BY Paul L. Kuhn
ATTORNEY

March 12, 1946. H. SCHRECK 2,396,392

HYDRAULIC OPERATION OF MECHANISM

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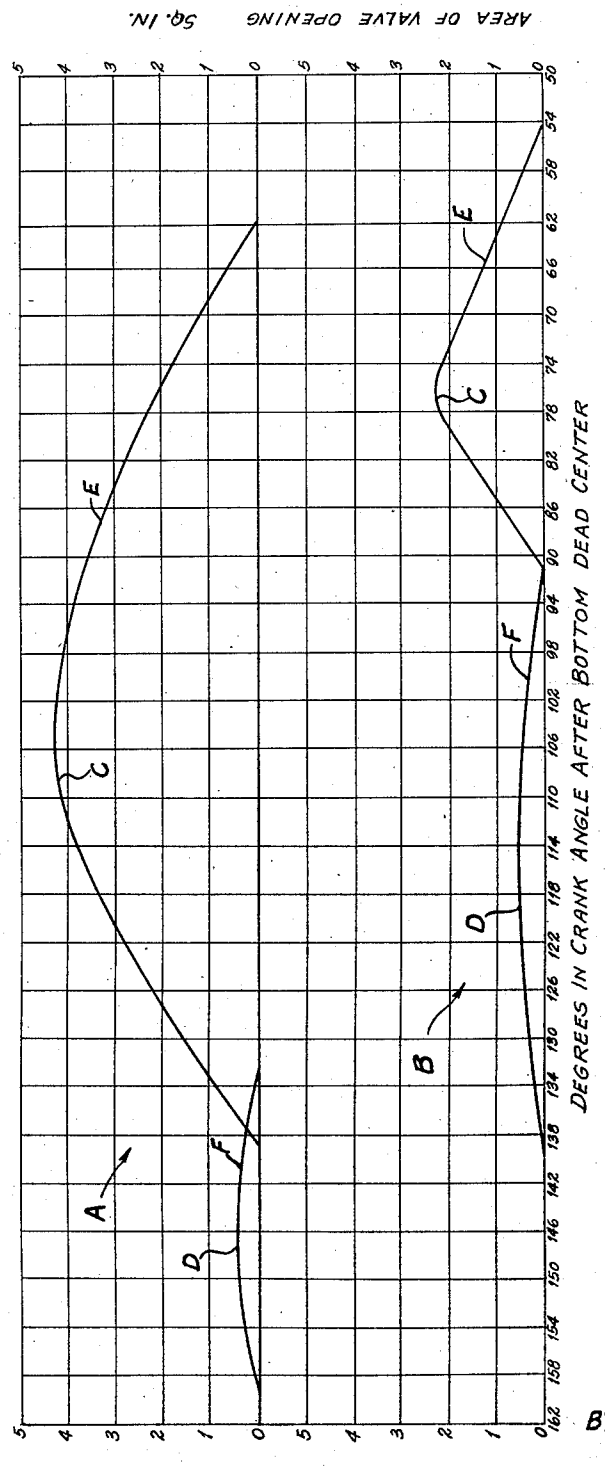


FIG. 8.

INVENTOR;
HENRY SCHRECK
BY Paul L. Kuhn
ATTORNEY

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UNITED STATES PATENT OFFICE

2,396,392

HYDRAULIC OPERATION OF MECHANISM

Henry Schreck, Beloit, Wis., assignor to Fairbanks, Morse & Co., Chicago, Ill., a corporation of Illinois

Application May 8, 1943, Serial No. 486,192

6 Claims. (Cl. 123—90)

This invention relates to hydraulic operation of mechanism, such as used for the control of valves and the like, and has particular reference to an improved system of hydraulic operation of mechanism which although of general application, is especially suitable for the actuation of valves of internal combustion engines.

A particular object of the invention resides in an improved system of fluid actuation of mechanism, such as fluid control valves, which affords smooth, efficient and relatively silent valve actuation, and which when utilized in the operation of engine fuel and air control valves, results in improved engine operation.

A primary objective of the present invention is attained in an improved hydraulic actuator for a reciprocating element, useful as for engine valve operation, and in which provision is made by completely hydraulic control for varying the timing, the extent and the duration of hydraulic impulse; according to one useful application of this system and improved structure thereof the invention objectively provides for variation by means of a single control, in timing, duration and lift of opening of a puppet valve for an internal combustion engine.

Another important objective of the invention is attained in an improved combination of a plurality of hydraulic impulsing assemblies of control means operating entirely hydraulically and in the hydraulic system for readily attaining a differential control of a plurality of valves, for example, responsively to varying load conditions in an internal combustion engine.

A still further important objective of the invention is attained in an improved arrangement in a complete hydraulic actuator assembly, for effecting a convenient manual control of surge conditions in the system, whereby to assure at all times and under all conditions exactitude of timing, elimination of unwanted noise, and general smoothness and efficiency of valve operation.

A further object resides in the provision of a relatively compact, structurally simple, hydraulic actuating system of the character indicated, which includes fluid pump means regulated in respect to fluid pressure output, and hydraulically operated valve means of improved character, which are suitable for association with an internal combustion engine to control the valves thereof.

Other objects and advantages of the presently improved system of hydraulic actuation of valves and the like, will appear readily from the following description of a preferred exemplary embodiment of the invention as illustrated by the accompanying drawings.

While as hereinabove indicated, the presently improved hydraulic actuating system is of general application, it is preferred to illustrate and describe the system herein in association with an internal combustion engine having valves of fluid actuated type, wherein the system is in direct control of the operation of these valves. Accordingly, in the drawings;

Fig. 1 is an assembly view in elevation, of an engine shown in part only, with certain parts broken away for clearness, and incorporating the hydraulic valve actuating system of the present invention;

Fig. 2 is an enlarged vertical section through the fuel and air control valves of the system, illustrating an improved valve and valve-actuating assembly;

Fig. 3 is an enlarged section taken vertically through a fluid pump, and through a manually controlled valve device associated therewith, the pump and valve shown being operatively associated with the engine air control valve of Fig. 2;

Fig. 4 is a fragmentary sectional view of the pump adapted for operative association with the engine fuel control valve as appears in Fig. 2;

Fig. 5 is a planar development of a control element carried by or formed as a part of a special pump or impulsing plunger, such as shown by Fig. 3;

Fig. 6 is a planar development, similar in nature to that of Fig. 5, but showing, in plan, the nature and relation of liquid control margins of the control element carried by the plunger, in the structure of Fig. 4;

Fig. 7 is an enlarged sectional view as taken transversely through a fluid pump, along the line 7—7 in Fig. 1, showing a pump control feature thereof, and

Fig. 8 illustrates graphically by curves, the areas of valve opening attained in the cycle of impulsing valve operation under given conditions of engine operation.

Referring first to Fig. 1 of the drawings, there is illustrated an internal combustion engine generally designated by the numeral 10, including in the portion thereof shown, a cylinder assembly 11 extending above the engine crankcase (not shown) and an engine actuated governor device 14 of suitable type and construction, having an operating arm 15 (shown in broken outline) connected by a link 16 (also in broken outline) to a control arm 18 mounted on a frame 19 for

swinging movement about a pivot 20. The function of the governor assembly in the present valve actuating system will appear hereinafter. Although not necessary to be described in detail, it will be understood that governor 14 is engine driven, as by suitable gearing or chain drive connections. The engine illustrated is of the same general type as that forming the subject matter of application filed July 11, 1941, Serial Number 401,884, by this applicant.

Mounted on the engine camshaft casing shown in its upper portion 22, and adjacent the cylinder 11, are a pair of fluid pumps 23 and 24 of reciprocating plunger type as indicated by Figs. 3 and 4, each of which is cam-actuated preferably from the engine camshaft shown at 26, within the casing 22. Although not shown, the camshaft is engine driven, as by suitable gearing or other connection. The pumps are supplied with a suitable hydraulic operating fluid, and may be supplied from a common source provided as by a tank assembly indicated generally at 27, through a supply header 28 connected to each of the pumps, and at one end to the tank 27 by a feed pipe 30. The opposite end of header 28 is connected to tank 27 by a return pipe 32. There is thus provided a closed fluid system, with fluid circulation therein maintained during engine operation, by the suction and fluid by-pass functions of the several pumps in the instance where the supply source 27 is provided by a tank, as will appear more fully hereinafter, or by the pressure circulation effect of the source 27 when constituted in part by a suitable pump 29. As illustrated in Fig. 1, the pump 29 which may be engine driven through the gearing 31 and drive shaft 33 operatively connected (not shown) with a rotating part of the engine, as for example, the camshaft 26, receives fluid from a suitable source (not shown) through conduit 34, and delivers the fluid under pressure, through conduit 36 leading preferably into the upper end of tank 27. It may be noted here that in the event a tank alone is employed as the fluid source 27 of the system, such tank may and preferably will be somewhat elevated relative to the header 28, so as to provide a desired initial fluid-pressure head in the header.

Within the cylinder head 35 of cylinder 11, are a pair of valves 38 and 39 (Fig. 2) the valve 38 serving for example, as a control for metering air in the cylinder, while the adjacent valve 39 controls fuel admission. By way of preliminary reference to overall function, it may be noted that these valves are actuated respectively, by hydraulic valve-motors 40 and 41 suitably mounted on the cylinder head as shown. Actuating fluid is delivered to the air valve operating motor 40 by the pump 23 through a conduit connection 42 and a valve device 43 hereinafter to be described, and arranged by preference, at the discharge end 44 of the pump. In a similar manner, actuating fluid for the fuel valve operating motor 41, is delivered by pump 24 through a conduit connection 46 and a valve device 47 similar to the valve device 43, at the discharge end 48 of the pump.

Since certain of the valve motors, pumps, valves and related elements are illustrated as vertically mounted, some of the parts are hereinafter sometimes referred to as upper or lower elements, but without restriction as to necessary positions of mounting or usage.

Fig. 2 illustrates in vertical section, the fuel and air valves of the engine and the hydraulic actuating motors therefor, as provided for the cylinder shown. Describing first the air valve 38 and its operating motor 40, the valve 38 which includes a valve head 50 and valve stem 51, is arranged in the cylinder head 35 so that the valve head 50 controls the opening and closing of the cylinder air port 52 between the cylinder chamber shown in its upper portion at 54, and the air chamber 55 formed in the cylinder head, the latter chamber being in communication with an air conduit (not shown). The valve stem 51 is journalled for reciprocable movement, by a sleeve or bushing member 56 fixed in the cylinder head 35, the valve stem projecting therebeyond for operative association with the motor 40 and a valve-closure spring 58. As appears, the valve stem below its upper end 59, is circumferentially recessed at 60, for the reception of the internal flange 62 of a tapered split collar 63. The split collar in turn, is engaged by an internally tapered annular ring 64 having an outstanding circular flange 66. Through the collar 63 and the wedged connection of the ring 64 therewith, the ring is thus fixed to the valve stem. The spring 58 has its upper end bearing against the ring flange 66, and its lower end abutting a flanged portion 67 of the sleeve member 56, being thus arranged to load or bias the valve stem 51 in the upward direction for seating the valve head 50 to close the air port 52.

The valve motor 40 includes a casing 68 having a mounting flange 70 engaging the cylinder head 35 and secured thereto, as by the studs 71 (Fig. 1), the casing providing a lower chamber 72 open at its lower end 74, for receiving the upper or outermost portions of the valve stem 51 and spring 58. The chamber 72 may be open to the atmosphere as at 73, in order to prevent air entrapment in the chamber. Casing 68 further has an upper chamber 76, open at the top and separated from the lower casing chamber 72 by a transverse wall 78. The wall 78 is apertured centrally thereof at 79, to receive therethrough the lower end portion 80 of a cylinder member 82 seated in the upper chamber 76. The member 82 provides a cylinder bore 83 for a vertically movable piston 84 having its lower end 86 normally engaging the upper end 59 of the valve stem 51. The piston has a pressure head 87 exposed in an enlarged cylinder pressure chamber 88 at the upper end of the bore 83, and is provided with a stop element or pin-like member 91 extending beyond the head 87 and through a pin bore 92 formed in the head section 94 of the cylinder member 82. Fluid-pressure admission to the cylinder chamber 88, is effected through radial passages 95 in the cylinder member, communicating with an external annular channel 96 therein, which in the assembly positionment of the cylinder, aligns with a threaded passage 98 through the wall of the casing 68. Communicating with the passage 98 is the fluid conduit 42 leading from the pump unit 23 (Fig. 1), connection of the conduit in fluid flow association with the passage 98 being made in a fluid-tight manner, as by a detachable coupling element 100 threadedly received in passage 98, as shown.

The cylinder 82 is retained in assembly with the casing 68 by a removable cap element 102 suitably secured to the upper end of the casing, as by the studs or bolts 103, the lower end 104 of the cap projecting within the upper casing chamber 76 for assembly abutment with an annular shoulder 105 on the cylinder head portion 94, formed by reducing the cylinder head as shown at 106. A suitable gasket 108 of a compressible character, is by preference, interposed

between the cap and cylinder shoulder 105 to effect a fluid-tight seal at this zone.

The reduced cylinder head portion 106 is thereby adapted for reception in the cap 102, to cooperate therewith in defining a chamber 109 within the cap 102 for a purpose which will appear. The upper end 110 of piston stop pin 91 is exposed in chamber 109 for abutment with an adjusting screw 112 extending through the top of the cap 102. It will now appear that through the screw 112 and pin 91, the upward extent of piston movement and hence of the valve stem 51, may be controlled to any desired degree, as for effecting a condition wherein, when the valve 50 is fully closed, the valve stem 51 and piston 84 will be in substantial engagement as illustrated, whereby to avoid, or at most have but an irreducible minimum of initial lost motion of the piston relative to the valve stem.

In the operation of the valve as thus far described, admission of fluid under pressure into the pressure chamber 88, will effect a downward displacement of piston 84 and a corresponding downward movement of the valve stem 51 and valve head 50, to open the port 52 as for proper control of air for combustion in the engine cylinder chamber 54. Upon release of fluid pressure in motor chamber 88, the valve return spring 58 then restores the valve 50 to port-closing position, and returns the piston 84 and stop element 91 to an initial position wherein the stop pin 91 is substantially in abutment with the adjusting screw 112.

An important feature of the presently improved valve assembly is found in the present provisions for removing air and other gases from the motor actuating fluid upon its admission under pressure to the motor chamber 88. The piston stop pin 91 which extends through cylinder head bore 92, is given a predetermined degree of clearance therein sufficient to permit pressure-flow of gases and a minimum quantity of fluid entrapped in such gases, from the pressure chamber 88, between and along the opposed surfaces of the bore 92 and pin 91 therein, and into the upper chamber 109 heretofore described, for ultimate discharge therefrom. Such clearance longitudinally between pin 91 and bore 92 is indicated in Fig. 2 by the legend "Clearance." In order to assure delivery of gases into the leakage or vent passage through bore 92 as above described, the uppermost or top wall 114 of the pressure chamber 88, is inclined upwardly and convergently toward the vertical axis of the cylinder head bore 92, to terminate in the circular margin 116 defining the lower end of the bore 92. By reason of the predetermined clearance of the pin 91 in bore 92, the margin 116 cooperates with the adjacent opposed surface of the pin to form an inlet port-opening for the leakage passage through the bore 92. Thus as air and other gases in the fluid under pressure in chamber 88, collect in the upper zone of the chamber, such air and gases will be concentrated at the leakage port by reason of the upward convergence of the chamber top wall 114, for pressure discharge into and along the clearance passage in bore 92.

Such air and other gases together with a small amount of fluid, which thus attain and accumulate in the cap chamber 109, may be readily removed therefrom in any suitable manner. As presently preferred, a discharge opening 118 is provided through a wall portion of the cap element 102 near the upper zone of the chamber 109, and is connected to the return conduit 32 of the fluid system heretofore described, by a suitable conduit (not shown). Thus fluid and gases received under pressure in the cap chamber 109, may readily pass into conduit 32 for delivery to the tank or pump unit 27, wherein the gases may be vented to atmosphere in any suitable manner not presently illustrated.

The valve and actuating motor assembly 39, 41 controlling fuel admission to the engine cylinder is in all important respects as to structure and function, broadly similar to the above described air control valve assembly 38, 40. Hence in the following description thereof, those parts which correspond to the similar parts of the air valve assembly, will be designated by the same reference numerals differentiated however, by the letter suffix *a*.

Valve 39 includes a valve head 130 and vertically extending valve stem 131, the latter being journalled in a sleeve 56*a* which is here formed as a part of a cylinder head insert member 132, providing a fuel-receiving chamber 134 supplied with fuel from a suitable source not shown, and having its lower end 135 presented to the engine cylinder chamber 54. A port opening 136 and port passage 138 formed in the lower end portion of member 132, serve to communicate the fuel supply chamber 134 with the cylinder chamber 54, and the valve head 130 as shown, is arranged to control the port 136. The engine cylinder head 35 is recessed as at 139, for receiving the insert member 132, and the latter is retained therein, through the lower projecting end 140 of the valve motor casing 68*a* which engages the upper end 142 of member 132 through a sealing gasket 143, the casing 68*a* in turn being secured to the cylinder head 35 as by the casing flange 70*a* and studs 71*a*. The valve stem 131 vertically movable in sleeve 56*a* is sealed against fuel leakage upwardly therealong by a plurality of annular grooves 144 of suitable character, and supports at its upper end portion 59*a* a split collar 63*a* engaged in the stem groove 60*a*. The collar is tapered on its external surface as shown, and supports thereon the correspondingly tapered ring 64*a*. A flange 66*a* on ring 64*a*, provides a bearing abutment for the upper end of valve-closing spring 58*a*, the spring thus loading the valve stem 131 for urging the same upwardly to seat the valve head 130 at the fuel inlet port 136.

The casing 68*a* for the valve motor 41, provides the lower chamber 72*a* open at its lower end 74*a*, for receiving the upper portions of the valve stem 131 and spring 58*a*, and open to the atmosphere, as through a vent opening 73*a*. An upper chamber 76*a* formed by the casing 68*a* and separated from the lower chamber therein by the transverse wall 78*a*, has mounted therein a cylinder member 82*a* which differs from the cylinder member 82 of valve motor 40 only in that the diameter of the cylinder bore 83*a* is somewhat greater. The lower end 80*a* of cylinder 82*a* is projected downwardly through an opening 79*a* in the casing wall 78*a*, in order that the lower end 146 of a piston 147 may be exposed in the lower casing chamber 72*a*, for abutting engagement through a piston element hereinafter referred to, with the upper end 59*a* of the valve stem 131. The piston 147 being larger in diameter than the piston 84 of valve motor 40, is preferably of hollow construction providing an integral piston head 148, and being open at its lower end 146. The latter open end of the piston is closed by a wall insert 151 which is formed to provide an abutment projection 152 for engaging the valve stem 131, as shown. The piston head

148 has its external pressure surface 153 exposed in an enlarged cylinder pressure chamber 88a, and is provided with a stop pin 91a which projects therebeyond through a bore 92a in the cylinder head 94a. As in the instance of valve motor 40, fluid-pressure communication with the cylinder chamber 88a, is effected through radial passages 95a in cylinder 82a, external annular cylinder wall channel 96a, and casing passage 98a in communication with the fluid conduit 46 leading from pump unit 24 (Fig. 1). Assembly connection of the conduit 46 may be made as by a fluid-tight coupling 100a.

Cylinder 82a is retained in casing 68a by removable cap member 102a which in turn is suitably secured to the upper end of the casing by studs 103a. The lower end 104a of cap 102a projects within the upper casing chamber 76a for assembly engagement with the annular shoulder 105a on the cylinder head 94a, the shoulder being formed by reducing the cylinder head as at 106a. A suitable gasket 108a interposed between the cap end 104a and the cylinder shoulder 105a, serves to effect a fluid-tight seal in this zone. The reduced head 106a of the cylinder cooperates with the cap 102a to form a chamber 109a within the cap, for receiving gases and leakage fluid from the pressure chamber 88a in the manner heretofore described in connection with the valve motor 40. As before, the stop pin 91a is exposed at its upper end, in the cap chamber 109a for engagement with a positioning screw 112a threaded in the cap. For the purposes heretofore described, the pin 91a is given a predetermined clearance in the pin bore 92a, as indicated in Fig. 2 by the legend "Clearance," and the upper wall 114a of pressure chamber 88a is inclined upwardly and convergently toward the axis of the bore 92a, to terminate in the margin 116a defining the lower open end of the pin bore. The margin 116a in cooperation with the adjacent surface of pin 91a, forms the inlet port for the leakage passage through the bore 92a. Discharge of gases and leakage fluid collecting in the head chamber 109a is as before, effected through the return conduit 32.

It is here noted that the structure and operation of valve motor 40 associated with engine valve 38, as well as the similar motor 41 associated with engine valve 39, both as hereinbefore described, are fully disclosed and claimed in a copending application entitled Hydraulic operating units, filed May 8, 1943, by the present applicant, and bearing Serial No. 486,194.

Turning now to the fluid pump and control valve unit 23, 43 illustrated in sectional elevation by Fig. 3, the unit which is associated with the engine air valve motor unit 40 through the conduit connection 42, includes a supporting frame or casing 154 having a mounting flange 155 near its lower open end 156, for assembly securement upon the upper wall of the camshaft casing 22 (Fig. 1), as by mounting studs or bolts 158. Suitably seated within the casing 154 in the upper section 159 thereof, is a cylinder member or liner 160 which may be properly positioned in the casing by a locating set screw 162. Vertically reciprocable in the cylinder liner is a pump plunger 163, by which is carried a special control element or means, preferably constituted by a head attachment 164 presently to be described, and a lower shank portion 165 projecting downwardly through an enlarged lower chamber 166 in the casing 154. Although the control element 164 is usually formed integrally with the plunger 163, it functions in many respects as a separate control means. The casing chamber 166 provides a cylinder for a guide sleeve 167 vertically movable therein. A frame 172 operatively supports a cam-follower roller 175 which engages a cam 176 (Fig. 1) of a predetermined preferably symmetrical contour not here illustrated, fixed upon the engine camshaft 26 referred to hereinbefore. A suitable compression spring 179 having its upper end bearing against a ring element 180 fixed in the casing 154, and its lower end bearing against a suitable abutment (not shown) movable with frame 172, serves to urge the pump plunger, sleeve and frame 172 downwardly, for maintaining cam-following engagement of the roller 175 with the pump actuating cam 176. It may be noted that the guide sleeve 167 assures maintenance of a true vertical movement of the rod 165, roller frame 172 and roller 175, in the cam actuation of the pump plunger thereby. Although not herein illustrated, suitable provision is made to prevent angular movement of the roller frame 172 and guide piston 167 about the plunger axis.

The plunger cylinder 160 near its upper open end 184, is provided with a port opening 186 and a passage 187 laterally through the cylinder wall, the passage opening to an annular channel 188 formed in the casing 154 about the cylinder, as shown. The casing below its upper end 190, is externally bossed at 191, to provide for a tapped aperture 192 therethrough in communication with the casing channel 188, the tapped aperture receiving in threaded assembly therein, the threaded end 194 of a supply and by-pass fitting 195 connected into supply header 28 extending from the fluid supply cource 27 (Fig. 1).

The control element or means carried by plunger head portion 164, is adapted for controlling fluid flow through the cylinder port 186, and to this end, the upper end 198 of the control element is substantially flat or planar as shown, to provide the annular port-cut-off edge 199. Suitably below the face 198, the body of the element is annularly reduced or recessed to provide a channel 200, which has fluid communication with the top surface 198, and hence with the pump displacement chamber 197, through a longitudinal groove 202 formed in the surface 203. The upper margin 204 of the channel 200 is spiralled upwardly from a point 206 at the lower end of the groove 202, to a point 207 immediate the ends of the groove, as clearly appears in Figs. 3 and 5. The spiralled margin 204 provides a port control edge for a purpose to appear, while the surface 203 intervening the control margins 199 and 204, constitutes a port-closing valve surface, cooperating with port 186.

Seated in the casing 154 above the upper open end 184 of the plunger cylinder 160, is a member 208 which is formed with a fluid discharge passage 210 vertically therethrough and in communication with the plunger cylinder to receive fluid displaced from chamber 197, by the plunger upon its upward movement. A cap-like element 211 forming a part of the casing 214 of valve device 43 presently to be described, serves in part through packing means 212 and when the casing 214 is secured, as by bolts 216, to the upper end 190 of the pump casing 154, to retain the member 208 in its assembly position. The cap 211 is centrally apertured at 218 for establishing communication between the discharge passage 210 of member 208 and the interior of the valve assembly 43.

It will appear from the foregoing description

of the pump mechanism, that upon rotation of the cam 176 to bring the follower roller 175 onto the low surface (not shown) of the cam, as effected through the action of the pump spring 179, the control element 164 carried by the pump plunger, and thus lowered in the cylinder 160, will be positioned to open the port 186 above the control edge 199, for the admission of fluid to the displacement chamber 197 above the plunger. Whereupon, further rotation of cam 176 to elevate the plunger in the cylinder through roller-cam engagement, will effect initial fluid cut-off at the port 186 by the cut-off edge 199, and then full port-closure by the valve surface 203. Concurrently, the fluid admitted to the cylinder, will be discharged under pressure through the passage 210 in member 208, as a result of the upward plunger displacement. During upward movement of the plunger, and depending upon the angular position of the plunger, and hence of the control element 164, in the cylinder, the lower control edge 204 thereof will uncover the port 186 to the channel 200, whereby to release fluid pressure on the discharge side of the pump and permit return flow of fluid through the groove 202, channel 200 and port 186, back into the fluid supply and by-pass header 28. The fluid-pressure impulse function of the pump as thus indicated, will be referred to more fully hereinafter.

Angular orientation of the pump plunger 163 in the cylinder 160 so as to position the control element 164 and hence spiral control edge 204 relative to the port 186, serves to uncover the port at a time earlier or later in the upward fluid delivery stroke of the plunger. This may be manually adjusted, but is preferably controlled in accordance with engine operating conditions (for example, load) as reflected in the engine-actuated governor mechanism indicated at 14 in Fig. 1, in order to regulate thereby, the duration of opening of the air valve 38 as will appear more fully hereinafter. As presently preferred, the governor control connection to the pump plunger for determining the angular position of the latter, or more specifically the position of control element 164 in the pump cylinder 160, includes as shown by Figs. 3 and 7, a sleeve or pinion element 222 rotatably arranged in the pump casing 154 adjacently below the pump cylinder 160, and about the intermediate section 223 of the plunger shank 165. The pinion 222 is suitably constrained against longitudinal displacement in the casing, as by a lower external shoulder 224 in engagement with the spring seating ring 180, and by an upper shoulder 226 in engagement with a ring element 227 seated in the casing 154. Connection of the pinion to the plunger shank section 223 is effected by any suitable form of securement permitting relative reciprocal movement, but preventing relative angular displacement; e. g., through one or more splines 230. The external surface of the pinion 222 is formed to provide gear teeth 231 which are engaged by the teeth 232 of a rack member 234 slidably carried in the portion 235 of casing 154 for movement in a direction normal to the pump axis. Rotation of the rack member may be prevented by a set screw 236 having its end 238 extending into a longitudinal slot 239 in the member 234. The rack 234 is by preference, a tubular member, and projected through the bore 240 thereof is a governor-actuated shaft or rod 241 which has one end 242 (Fig. 1) suitably connected to the governor lever 18. As shown by Fig. 7, the rack member is adjustably positionable longitudinally of the rod 241, as by lock-nuts 243 and 244 at the opposite ends of the member and in threaded engagement with the rod 241 as illustrated.

It will thus appear from the foregoing, that during engine operation, the governor 14, through its control arm 18 and the shaft or rod 241, will serve to position the rack 234 according to conditions of engine function, such as speed or load, and through the rack and sleeve or pinion 222, will serve angularly to orient the plunger 163 and element 164 in the cylinder 160, such as to properly relate the spiral control edge 204 with the fluid port 186. The provision for longitudinal adjustment of the rack member 234 on the rod 241, serves a purpose which will be described hereinafter.

The fluid pressure delivery from the pump 23 is directed through the valve device 43 presently to be described, and through conduit 42 (Fig. 1) to the pressure chamber 88 of the air valve motor 40 (Fig. 2), and upon pressure release of the fluid in chamber 88 as effected by the pump when the spiral control edge 204 of the control element 164 opens port 186, the fluid is returned through the pump and port 186 back into the supply header 28, this last fluid movement being effected as a result of the upward movement of the valve motor piston 84 under the bias of motor spring 58. The principal function of the valve device 43 is to check or cushion the return flow of fluid from the valve motor to a predetermined extent, whereby to eliminate or minimize adverse operational effects, otherwise due to violent, undamped oscillatory surging of the fluid column between the pump and valve motor, upon the relatively sudden pressure release by the pump. Otherwise a hydraulic ramming or so-called hammer effect would be produced upon the air valve 38, with attendant disruption of smooth and efficient engine operation by reason of the resulting ineffective valve-closure, even sometimes resulting, without present control features, in an unwanted valve movement. By thus regulating the fluid return flow upon pump pressure release, the fluid pressure on the motor piston 84 is relatively gradually reduced, which thereby allows the valve-closure spring 58 to effect a correspondingly gradual or attenuated closure of the valve 38 in a positive and relatively quiet manner.

As shown by Fig. 3, the valve device 43 comprises a casing 214 heretofore referred to, formed to provide a fluid chamber 250 open at the casing end 251, the open end of the chamber normally being closed by a threaded head or cap element 252 sealed against fluid leakage at the opening by a suitable gasket 254. In the intermediate zone of the chamber 250, the casing is formed to provide an annular bevelled shoulder 255 constituting a valve seat for the valve head 256 of a valve element 258. As controlled by valve element 258, the seat 255 may be said to define the main valve port of the device. The valve element which is of substantially tubular form to provide a central longitudinal bore or passage 259, and is externally flanged or spidered rearwardly from the valve head 256, is movable longitudinally in chamber 250, being guided in such movement by engagement of its flanges with the casing wall portion 260 of reduced diameter near the inner end 262 of the chamber. A suitable compression spring 263 bearing at one end against the cap 252, and at its opposite end against a seat 264 on the valve head 256, serves to bias the valve element inwardly of the chamber 250 to seat the valve head on the chamber seat 255. The valve

passage 259 at one end 266 thereof, is outwardly bevelled to form an aperture or port-opening 267 for cooperation with the pointed end 268 of a needle-valve 270, formed of a thermally-sensitive material as an aluminum alloy, for a purpose to appear, extending axially into the chamber 250 through a guide passage 271 formed in the casing 214. The needle-valve passage 271 is internally threaded as at 275, to provide a threaded seat for an enlarged threaded portion 279 on the needle-valve 270. The threaded portion 279 may be provided by a separate collar suitably secured on the needle-valve 270, or as presently preferred, may be formed as an integral part of the valve. As will appear now, the needle-valve 270 is thus mounted for longitudinal adjustment through its threaded portion 279 in the threaded seat 275, effected upon rotation of the needle-valve element, so as to regulate the position of the port-controlling end 268 thereof relative to the port 267 in the valve 258 when the latter is seated to close the main valve port 255.

Packing means 282 arranged in the casing for preventing fluid leakage along the needle-valve, is retained and suitably compressed therein by a gland nut or collar 283 in threaded engagement with the casing. The needle-valve is projected outwardly beyond the collar 283, and has secured thereto as by set screw 284, a hand control or knob 286 by which the needle-valve may be manually rotated for effecting adjustment thereof. In order to indicate externally of the valve device, the relative position of the valve tip 268 on needle-valve 270, and the port 267 controlled thereby, in any adjusted position of the needle-valve, a pointer 287 may be carried by the knob 286 for cooperation with an appropriate scale (not shown) on the outer end 288 of the casing. Additionally, a readily removable cover or cap 290 may be threaded onto the casing for protecting the manual control end of the needle-valve.

Fluid communication between the valve chamber 250 at the inner end 262 thereof, and the passage 210 in the pump head member 208, is effected by the passage 218 laterally in the valve casing 214 in the zone of the casing element 211. On the opposite side of the valve 258 and near the end 251 of chamber 250, is a fluid passage 292 extending from the chamber through a valve casing extension or lateral boss 294, and communicating with this passage is one end of the conduit 42 extending to air valve motor 40 (Figs. 1 and 2). Connection of the conduit to the casing extension may be effected by a suitable coupling 295 (Fig. 1). In addition, a relief port 296 is provided in a side wall portion of the valve 258, for a purpose which will appear presently.

The pump mechanism 24 and control valve device 47 associated with the fuel valve motor unit 41 through conduit 46, are in all important structural and functional respects, similar to the now described pump assembly 23 and valve 43, with the single exception that the plunger head 164*a* of pump 24, is provided with a port control edge at its upper end, as shown by the fragmentary sectional elevation of Fig. 4. For convenience, the pump elements shown by Fig. 4 and which are substantially identical with the corresponding elements of the pump illustrated by Fig. 3, are given the same reference characters with appropriate letter suffix, so that the description hereinabove, may readily apply to these parts.

It is here noted that the structure and manner of operation of the like control valve devices 43 and 47 are hereinabove indicated, are fully disclosed and claimed in a copending application by the present applicant, entitled Fluid flow control device, filed May 8, 1943, and bearing Serial No. 486,193.

Referring to Fig. 4, the control element 164*a* carried by plunger 163*a* of pump 24, is formed to provide the annular channel 200*a*, lower spiral control margin 204*a*, and a longitudinal groove 202*a* between channel 200*a* and the planar head end portion 300. This portion is cut away or otherwise suitably reduced to provide an upper port-control edge 301 extending from the head surface 300 downwardly in a somewhat spiralled trend, to a point 302 approximately diametrically opposite the high point 207*a* of the lower control edge 204*a*. It is important to note here that the upper and lower control margins 301 and 204*a* extend in the same direction about the plunger head, but the angular degree of spiral provided for the upper control edge 301 differs to a predetermined extent, from that of the lower edge 204*a* (see Fig. 6), for a purpose presently to appear. As may be stated generally at this point, however, the now described arrangement of the plunger control margins provides, through angular orientation of the plunger in the cylinder as effected by the governor 14 in accordance with conditions of engine operation, for regulation of the timing of port closure at port 186*a*, by the upper control edge 301, and timing of port opening by the lower control edge 204*a*, both relative to the upstroke of the plunger which is of constant extent. Actuation of the plunger 163*a* in pump unit 24 may be effected in the manner provided for the pump unit 23, as by the cam 176*a* on camshaft 26 (Fig. 1).

As will be observed in Fig. 1, the governor 14 is utilized to effect simultaneous control of plunger orientation in the pump devices 23—24, the governor-actuated shaft or rod 241 being extended for this purpose, for connection to the pump rack 234 and the similar rack (not shown) of pump unit 24. Each of these racks is secured upon the rod 241 in the manner hereinbefore described and as illustrated by Fig. 7, such that each may be adjustably positioned longitudinally of the rod 241, as for a purpose to appear. In engines of more than one cylinder, the pump units associated with each of the additional cylinders may be similarly associated with the governor 14, as by extending the governor-actuated shaft 241 for connection to the rack members of the several pump assemblies. Or if desired in the case of multi-cylinder engines, more than one governor may be employed to effect the desired control.

Turning now to the function and method of operation of the presently described hydraulic impulse valve actuating system during engine operation, and considering first the fluid pump unit 23, 43 and the associated engine air valve motor unit 40, it will be observed that upon the down stroke of pump plunger 163 to open the pump port 186, fluid under the pressure head afforded by the elevated supply tank 27, will flow into the cylinder 160 above the plunger head and upwardly in the pump discharge passage 210 to the valve chamber 250 through the passage 218. As the engine camshaft through cam 176 thereon, effects upward displacement of the pump plunger 163, the control edge 199 will pass over the port 186 in the initial phase of the plunger upstroke, to close the port against further inlet of fluid, by the valve forming surface 203 closely overlying the port. As this occurs, the fluid ahead of the

plunger will be placed under considerable pressure and will react against the valve 258 to displace the same against the bias of valve spring 263, for establishing full fluid discharge through the main port 255. The fluid under pressure thus passing through port 255 and about valve element 258, and thence through conduit 42 to the motor chamber 88 in valve motor 40, reacts upon piston 84 to effect downward displacement thereof and a corresponding displacement of the air valve stem 51 and valve head 50, whereby to open the air port 52 for communication therethrough, with the engine cylinder 54. Following closure of the pump port 186, and as the plunger 163 continues in its upstroke, the fluid acting on the air valve motor piston 84, will be placed under increasing pressure to continue downward displacement of the piston 84 against the reaction of the valve spring 58, whereby to effect corresponding port-opening displacement of the air valve 38.

During the existence of fluid pressure in the valve motor chamber 88, any air or other gases in the fluid will be relieved or removed from the fluid in the manner heretofore described, as through the leak passage in bore 92, chamber 109 and the relief connection to the fluid-return conduit 32 of the system.

As the pump plunger 163 approaches the limit of its upstroke, the pressure release control edge 204 of the control element 164, will uncover the port 186 to the channel 200, whereby to release the pressure in the conduit 42 and pressure chamber 88 of valve motor 40, for permitting air valve closure at the port 52. The opening of port 186 by control edge 204 will occur earlier or later in the upstroke of the plunger, depending upon the angular position of the latter in the cylinder as determined by the governor 14 in the manner hereinbefore described, whereby to effect pressure release on the air valve motor at a period correspondingly earlier or later in the plunger upstroke. The spiral control edge 204 as oriented in the pump cylinder, thus affords regulation of the extent of air valve opening, or valve lift, through variation in the duration of pressure application upon the valve motor piston 84, as effected by relating the control edge to open the port 186 and hence release the pressure on the valve, at a time earlier or later in the upstroke of the plunger 163. At the same time and by the same means, the duration of valve opening is thus determined, as will be now fully appreciated, and as graphically illustrated by Fig. 5.

Upon pressure release by the pump, the fluid in chamber 88 and conduit 42 will be released for return through valve device 43 and the pump, to the supply header 28, the fluid return flow being effected under the influence of valve piston 84 in its upward or return displacement by the valve closing spring 58. By reason of the restriction or throttling of the fluid return flow by the valve device 43 as will be presently described, sudden closure of the air valve 38 is effectively prevented. If such were permitted, a rapid fluctuation or surging of the fluid column between the valve motor and pump, normally would be present, which would tend to produce a fluid hammer effect upon the motor piston and therethrough, upon the valve 38, causing rapid partial opening and closure of the valve, with attendant disadvantages of abnormal wear of the valve parts and a noisy valve operation.

Upon pump pressure release in the system, the spring 263 of throttling valve 43, displaces valve 258 to port-closing position (Fig. 3) relative to the main port 255. However, when this occurs the return flow of fluid to the pump is established at a greatly reduced or attenuated rate of flow, through the valve passage 259 and the port opening 267. The extent of throttling of fluid flow through port 267 is determined and controlled by the adjustable needle-valve 270 through regulated projection of its port control tip end 268 in the bevelled port opening 267. Although a very small portion of the fluid returned through the pump, normally will be by-passed about the needle-valve controlled port 267 through the small lateral opening 296 in the valve 258, the opening 296 serves as a pressure relief expedient, to prevent pressure build-up in the fluid return flow from the valve motor chamber 88 in the event the needle-valve 270 is projected inwardly inadvertently or otherwise, to an extent such as to close, or approximate closure of the port 267 by the valve tip 268. In normal operation, the small leakage at the opening 296 may be readily compensated for by proper adjustment of the needle-valve. The ports 267 and 296, because small in effective area, exhibit a marked effect in damping out line surges, particularly just after closing of the associated engine valve.

As hereinbefore indicated, the needle-valve 270 is formed from a thermally-sensitive material such as a suitable aluminum alloy, whereby to afford automatic or self-adjustment in the return-flow throttling function of the valve device, as attained responsively to and in accordance with temperature differences in the actuating fluid employed in the system. For example, during initial or starting operation of the engine embodying the present valve-actuating system, given as one example of the utility of the hydraulic impulsing system, the valve-actuating fluid may be relatively cool or at a low temperature and hence subject to a greater flow-inertia than obtains when the fluid is at a higher temperature. The resulting thermal contraction of the needle-valve 270 effected to an extent corresponding to the existing low temperature of the actuating fluid, thus affords an increased opening at the return-flow port 267 by reason of the retraction of needle-valve tip 268 from the port. The increased port-opening thereby compensates for the greater flow-inertia of the fluid throttled in its return-flow through the valve device. Normally and as a result directly or indirectly, of continued engine operation, the temperature of the actuating fluid may rise to an appreciable degree, in consequence of which the needle-valve 270 will undergo thermal expansion in proportion to the increase in the fluid temperature. There is thus effected in corresponding degree, an increase in fluid return-flow restriction at the port 267, by reason of the expansion projection of the needle-valve tip 268 toward and into throttling position in the valve port 267. Accordingly, the needle-valve control is not only manually adjustable for regulating the return-flow throttling function of the valve device 43 in accordance with given conditions of engine operation as hereinbefore indicated, but is self-adjustable in any manually controlled setting of the needle-valve, to compensate for temperature differences in the actuating fluid in the system. The latter as now will appear, serves to equalize the throttled return-flow through the valve device as changes occur in the temperature of the actuating fluid.

During the throttled return flow of fluid from the valve motor chamber 88, a certain portion of the fluid normally will be returned through the

pump and port 186 to the supply header 28, under a pressure determined in part, by the pressure exerted on the fluid in chamber 83 as the valve motor piston 84 moves upwardly in its return displacement by spring 58. That portion of the fluid attaining the header 28 is, however, under sufficient pressure to maintain fluid flow along the header and into the return pipe 32, for return to the fluid source 27.

The foregoing fully illustrates the cycle of hydraulic valve actuation attained by the presently improved system and apparatus, it being noted that this cycle with certain differences as between the air and fuel valve operation, occurs, in the application described, in each of the several pump and valve units. With respect to the hydraulic impulse actuation of the air valve, the extent of valve-lift and duration of valve opening in the cycle of operation, is variable in accordance with varying conditions of engine operation, this being accomplished automatically as described, by the engine governor 14 which serves to regulate the angular position or orientation of the control means on plunger 163 in the pump cylinder, for relating higher or lower zones of the pressure release control edge 204 with the pump port 186.

In regard to the fluid impulse actuation of the fuel valve unit 39—41, angular adjustment of the pump plunger 163*a* to orient the control element thereon, particularly upper control edge 301 relative to the port 186*a*, determines the timing of port closure at port 186*a* by the control edge. Port closure thus will occur earlier or later in the upstroke of the plunger, depending upon the angular position of the plunger and control means thereon, as controlled by the governor 14 in accordance with engine operating conditions. As a result, the time of initial fuel valve opening is thereby determined and controlled in relation to the earlier or later closing of the pump port 186*a* by the control edge 301. The lower control edge 204*a* on the other hand, thereafter controls the extent of valve-lift and duration of valve opening in substantially the same manner as obtains in the air valve operation hereinabove noted, although as now appears, the extent of lift and duration of fuel valve opening may and normally will be somewhat different from the extent and duration of air valve opening, in each cycle of valve operation. Variation in the duration of fuel valve opening, upon alteration of the angular position of the control element 164a, is afforded by reason of the differing degrees of angular spiral obtaining in the control edges 301 and 204*a*, which thus determine the extent of plunger movement necessary to close port 186*a* by the upper control edge 301 and then open the port by the lower control edge 204*a*, in the upstroke of the plunger.

With respect to the governor controlled mechanism for determining angular positions of the pump plungers and control elements thereon, in the several pump assemblies, the provisions herein for adjustment of the rack member 234 and the like rack member (not shown) of pump unit 24, longitudinally of the governor-actuated shaft 241, constitute an important feature of the presently improved hydraulic impulsing system. In the present exemplary application of the impulsing system to the operation of cylinder valves of an internal combustion engine, it will be observed that upon securement of the rack 234 of pump unit 23 to the governor-actuated shaft 241 in one longitudinally adjusted position thereon, the governor 14 as operated to effect a predetermined control position of its lever 18 and hence of the shaft 241, corresponding for example, to a given condition of engine speed and load, will thus position the rack 234 to establish thereby, a predetermined angular position of the plunger control element 164 in the pump cylinder, to accord with the foregoing conditions. Consequently, in the instance of the air valve unit 38—40 associated with the pump unit 23 under present discussion, the extent of valve-lift and duration of valve opening of the air valve 38 in each cycle of pump impulsing operation, are thus determined by the given angular orientation of the plunger control element 164 under the above stated conditions. However, under the same conditions of engine speed and load and governor actuation to the control position corresponding to the given speed and load factors, the extent of air valve lift and duration of valve opening in the cycle of pump operation, may be increased or decreased within limits, by readjustment of the rack member 234 in one direction or the other longitudinally of the governor actuated control shaft 241.

In like manner, the pump unit 24 associated with the fuel valve motor unit 41, may be regulated through positional adjustment of its rack member (not shown), such that for a given condition of engine operation, there will obtain a corresponding desired lift and duration of fuel valve opening. By reason of the particular form of the control element 164*a* on the plunger of pump 24, the beginning of fuel valve opening in this instance, will be correspondingly regulated by the adjustment of the rack, as will be now appreciated.

From the foregoing, it will appear that through the rack adjustment provisions, either or both of the pump units may be regulated relative to the common governor-control therefor, to adjust the impulsing functions of the pumps for given conditions of engine operation, as in respect to speed, load or other factors. In the exemplary embodiment of the impulsing system as herein applied to the operation of engine cylinder valves, one important advantage of the provisions for rack adjustment on the governor shaft 241, is that it affords relative orientation adjustments of the control elements 164 and 164*a* in the pump units 23 and 24, to care for fuels of different characteristics. For example, assume that the engine is operated on a gaseous fuel of a given calorific value. The rack member of the pump 24 associated with the fuel valve motor unit 41, then may be adjusted on the governor shaft 241, to predetermine the beginning of fuel valve opening, valve lift and duration of valve opening for a given condition, say, of engine speed and load, such that only that quantity of fuel of the given calorific value, determined to be required under the foregoing conditions, will be admitted to the engine cylinder in each cycle of pump impulsing operation. At the same time, the rack 234 of pump unit 23 associated with the air valve motor unit 40, may be correspondingly adjusted so as to determine the extent of lift and duration of air valve opening, whereby to condition the air valve 38 for controlling the cylinder air required to maintain efficient combustion of the given fuel. Thus the system may be readily adjusted in this manner, for fuels of different characteristics and calorific values.

With reference now to Fig. 8 which illustrates graphically by the diagrams A and B shown, the

nature of fuel and air valve operation attained by the present hydraulic valve-actuating system, the curve C in each diagram relates to the air valve 38, while the curve D in each instance relates to the fuel valve 39. Each of these curves is plotted in terms of area of valve-opening in the cycle of valve operation, as will appear. Considering the first or upper diagram A, the relative timing of fuel and air valve operation and the extent and duration of valve displacement as therein indicated, represent the events of valve operation in each cycle of hydraulic impulse actuation thereof, obtaining under given conditions of engine function, as in respect to engine speed, load and the character of fuel utilized. The form of the curves C and D and the relative timing of valve opening and valve closure shown, result from initial relative adjustment of the pump rack elements longitudinally of the governor-actuated control shaft 241; relative adjustment of the needle-valve elements in the return-flow control valves 43 and 47, and by reason of the pump impulsing control elements 164 and 164a which as fully disclosed herein, are relatively so arranged and proportioned with respect to each other, and each in respect to its associated hydraulic system, as to permit the graphically indicated regulation in relative displacement and timing of fuel and air valve operation.

The curves of diagram B are those which may obtain say under a considerably different condition of engine speed and load, and are indicative in respect to the curves of diagram A, of the relatively wide range of control in the relative timing and displacement of fuel and air valve operation by the present hydraulic impulsing system.

The downwardly extending portion E of curve C in diagram A, which is representative of the closing displacement of the air valve 38, is determined as to the degree of slope thereof, mainly by regulation of the needle-valve 270 in the return-flow control valve 43 of the system, which thus conditions the valve for restricting the return-flow of fluid from the valve-actuating motor 40 of the air valve 38, such as to retard valve closure to the extent indicated by the slope of the curve portion E. Retardation of valve closure to a greater or lesser degree through appropriate regulation of the needle-valve 270, will result in a correspondingly more gradual or sharper slope of the curve portion E, the latter being illustrated by the relatively sharper slope of the curve portion E in diagram B. The foregoing is applicable in like manner, to the downwardly sloped portion F of the fuel valve curve D in each of the diagrams.

The particularity of the preceding disclosure is given solely by way of illustrating merely one useful application of the invention, regarded in its broader aspects. The remarkably fine regulation possible and facilities for attaining dependable control of periodic reciprocal motion, are applicable generally to fields beyond that of the actuation of puppet valves for internal combustion engines. It will have appeared that the assembly embodies essentially a reciprocal impulsing element or plurality, such as a pump plunger, a hydraulic system and a hydraulic motor together with novel means operating entirely by hydraulic control for varying the timing of either or both the beginning and end of the impulsing movement derived, such timing being in respect to the constant or periodically operating mechanical means energizing the system, as for example, a cam and camshaft. It will further appear that besides timing of start and finish of the impulsing movement, the extent of such movement and its duration may similarly to the other characteristics of motion mentioned, be readily and conveniently controlled through a single control member for an individual or any plurality of related assemblies. It is further felt that as exemplified by the present disclosure, the facility for utilizing say a plurality of conventional mechanical energizing elements such as cams on a camshaft may, by the present improvements, be utilized through a plurality of the hydraulic systems, to effect a differential control of different reciprocally actuated elements, for example, valves. Similarly, by only minor structural differences as found for example, in the different control elements carried by the different plungers of a plurality of the hydraulic units described, not merely a differential control but in fact an even opposite regulation of different units may be readily attained in response to a given unidirectional movement of a control element, such, for example, as the rack structure under the influence of the governor of the engine as described. This feature is thought to have wide application to the control of a plurality of fluids, as for proportioning relatively exact mixtures thereof, for combustion or other purposes as may be desired.

The present invention as now fully described in connection with a preferred exemplary embodiment thereof, attains an efficient, compact and fully adjustable hydraulic impulse actuating mechanism which is particularly suited to the operation of fuel and air valves of internal combustion engines and the like, the mechanism and system of valve operation providing for positive and relatively quiet valve displacement, the latter particularly during valve closing movement.

It is to be understood of course, that while the present disclosure relates in particular to a preferred embodiment of the invention, certain changes or modifications in the several elements of the mechanism and the arrangement thereof, may be effected without departing from the spirit and full intended scope of the invention as defined by the appended claims.

I claim:

1. In a hydraulic actuating assembly for attaining a relatively rapid reciprocal movement, as of an engine valve, a plunger, means for periodically reciprocating the plunger, a reciprocal motor element arranged generally to follow the reciprocal movements of the plunger, a conduit and connections, forming with the plunger and motor element when substantially liquid filled, a hydraulic column of substantially fixed volume at all times, and substantially closed at all times except for periodic communication with a source of hydraulic fluid supply, and a valve element in the hydraulic column, providing alternate paths of fluid movement respectively to permit movement of fluid in the column in opposite directions, and adapted to reduce the rate of fluid movement from the motor element in comparison with rate of fluid flow permitted toward the motor element.

2. In a hydraulic actuating assembly for attaining a relatively rapid reciprocal movement, as of an engine valve, a plunger, means for periodically reciprocating the plunger, a reciprocal motor element arranged generally to follow the reciprocal movements of the plunger, a conduit and connections, forming with the plunger and motor element when substantially liquid filled, a

hydraulic column of substantially fixed volume at all times, and substantially closed at all times except for periodic communication with a source of hydraulic fluid supply, and a valve element in the hydraulic column, providing alternate paths of fluid movement respectively to permit movement of fluid in the column in opposite directions, and adapted to reduce the rate of fluid movement from the motor element in comparison with rate of fluid flow permitted toward the motor element, the valve element being provided with a continuously-open passage therethrough, providing for a trickle flow of liquid in either direction through the valve at any time.

3. The combination with a hydraulic impulse system providing a fluid conduit and a pump of fluid impulse character, for establishing fluid pressure flow in said conduit and releasing fluid pressure in the conduit to permit fluid return to the pump, of a pressure-sensitive valve element in said conduit, operable upon fluid pressure delivery from said pump, for establishing, at times, a full flow communication between the conduit and said pump, and means adjustable relative to said valve element and cooperable therewith upon pump release of fluid pressure in the conduit, to restrict the return flow of fluid through the conduit to said pump.

4. The combination with a hydraulic valve actuating system including a valve, a fluid pressure pump for applying and relieving fluid pressure on said valve, to cause valve opening and valve closure, and a conduit connection between said pump and valve, forming therewith a hydraulic system closed except for fluid replenishment, of pressure-sensitive valve means in said conduit connection, operable responsively to fluid pressure discharge from said pump, for establishing fluid pressure flow from the pump through said conduit connection and system to said valve, and adjustable means cooperable with said valve means for controlling pump release of fluid pressure on said valve.

5. The combination with a hydraulic valve actuating system providing a valve, a fluid pressure pump for applying and relieving fluid pressure on said valve to cause valve opening and valve closure, and a conduit connection between said pump and valve, of fluid flow control means in said conduit connection, said means including a valve element having a restricted opening therein, movable from an initial position for establishing fluid pressure flow from the pump through said conduit connection to said valve, and said valve element in its initial position, serving to restrict the return flow of fluid through said conduit connection upon pump release of fluid pressure on said valve, and adjustable means coacting with said restricted opening in the valve element for regulating the degree of restriction of fluid return flow through said opening.

6. The combination with a hydraulic impulse system providing a fluid conduit and a pump for alternately establishing fluid pressure flow in said conduit and releasing fluid pressure in the conduit to permit fluid return therethrough to the pump, of a pressure-sensitive valve member in said conduit, operable in response to fluid pressure delivery by said pump, to establish fluid flow communication between the conduit and pump, and a control element cooperable with said valve member upon pump release of fluid pressure in the conduit, to restrict return flow of fluid to the pump, said control element being of a thermally-sensitive material and adaped to equalize the return flow of fluid to the pump, in response to temperature differences in the fluid in the system.

HENRY SCHRECK.