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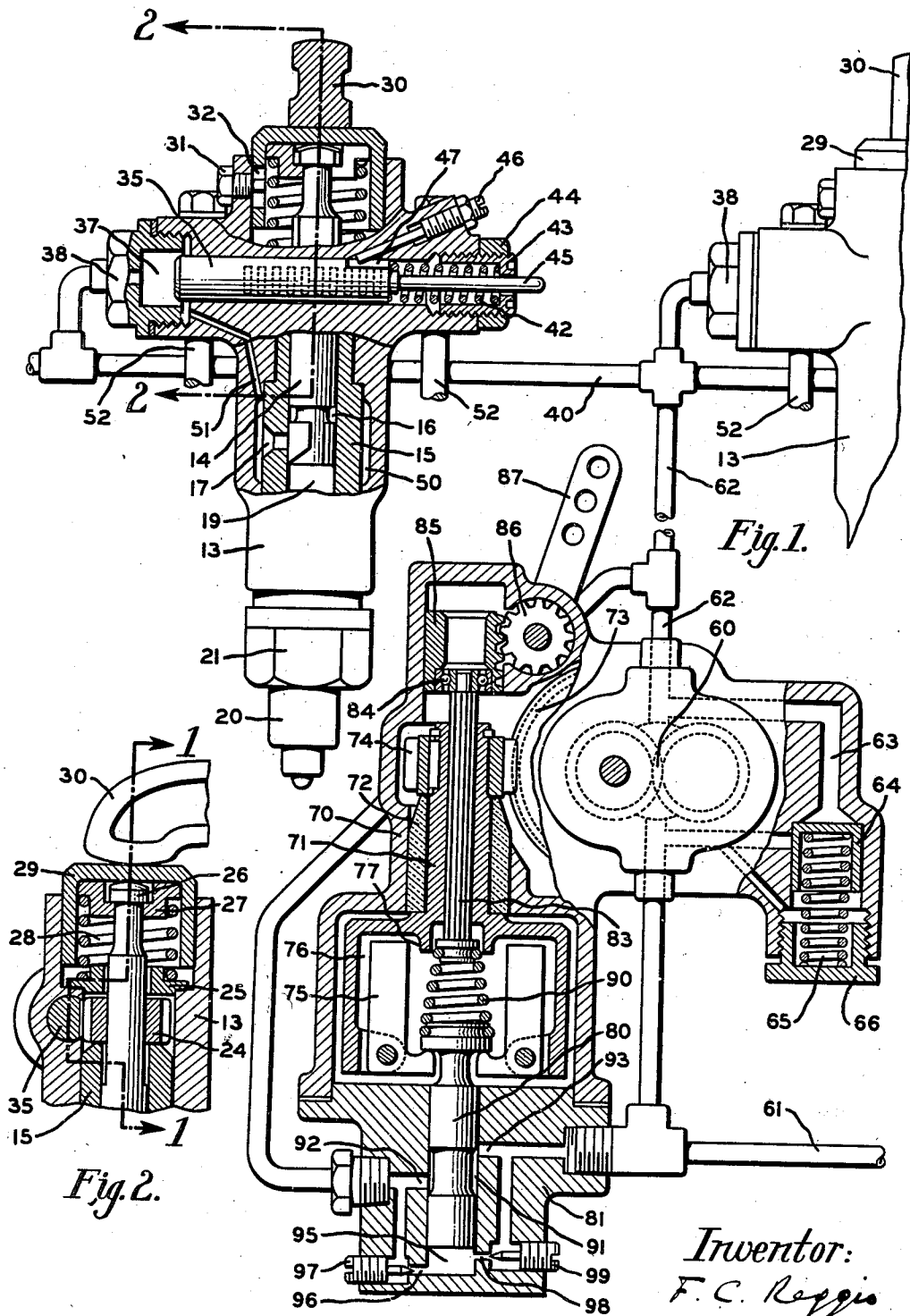
F. C. REGGIO

2,384,340

GOVERNING MEANS

Filed May 6, 1940

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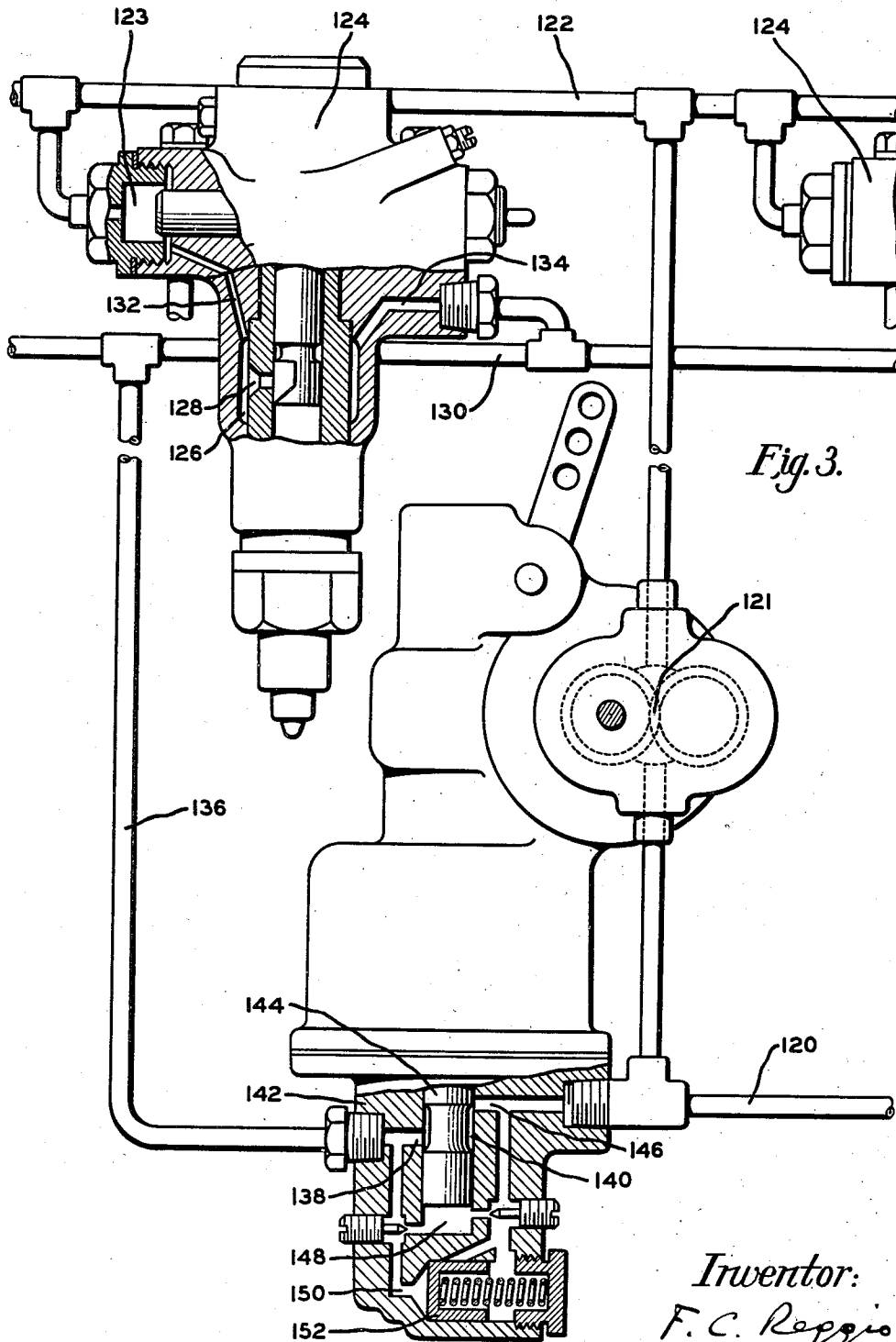
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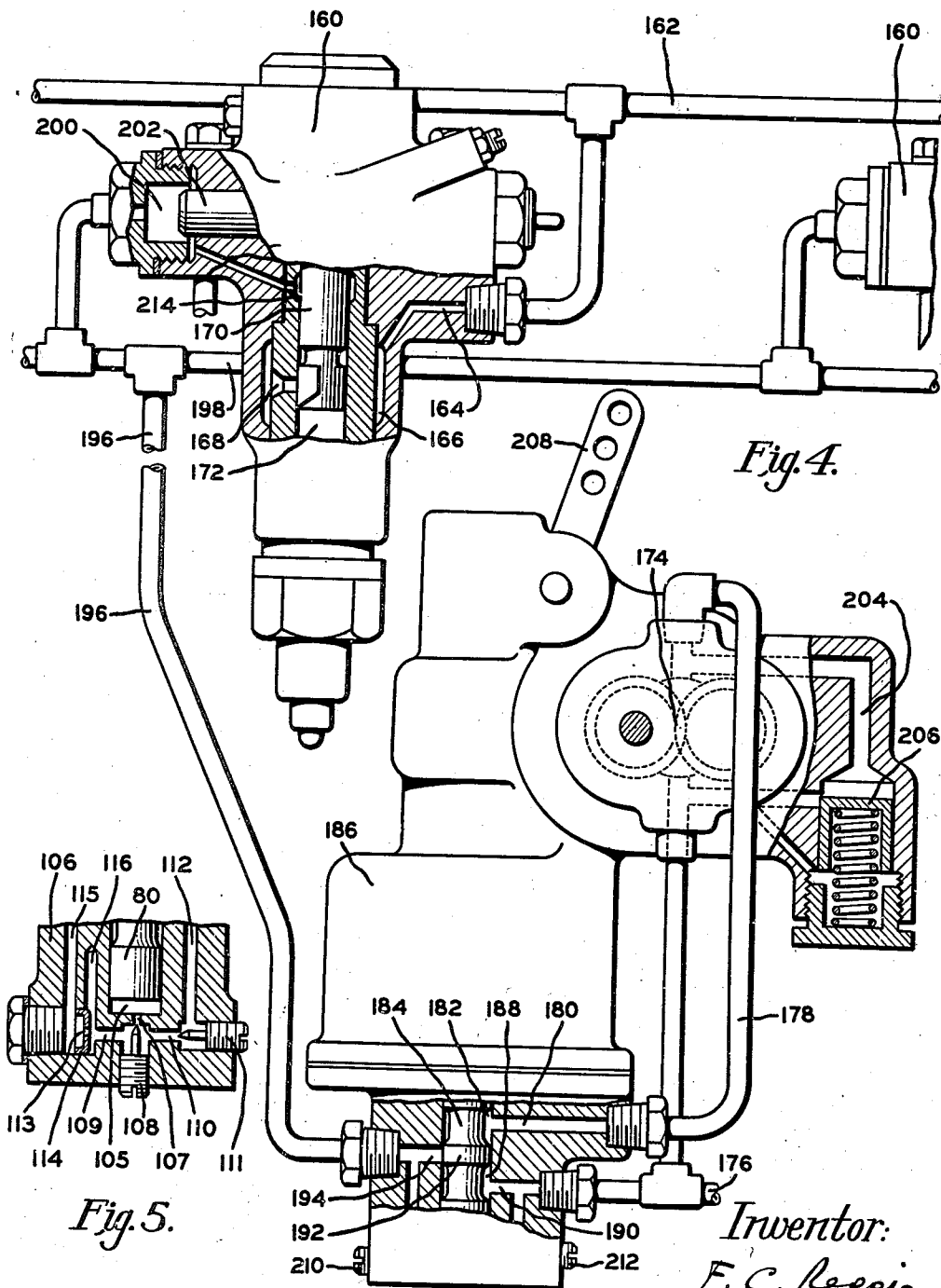


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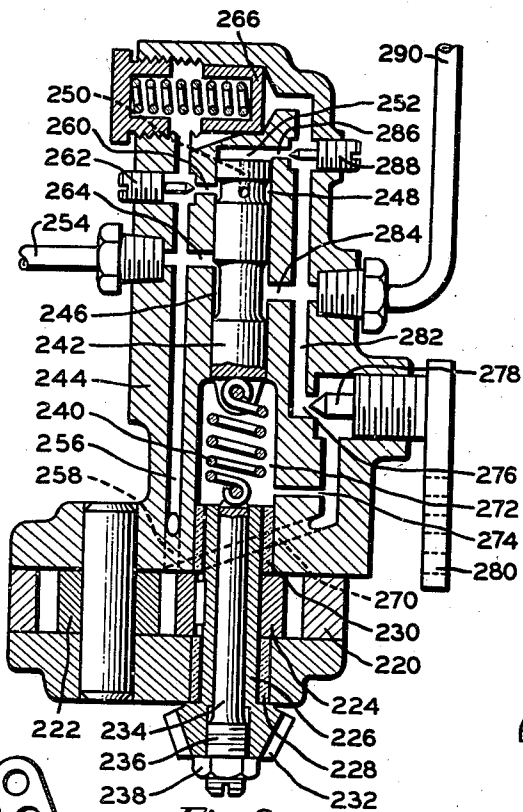


Fig. 6.

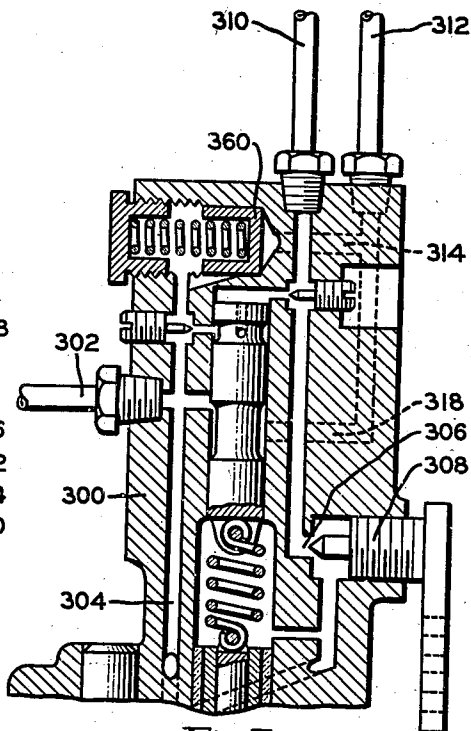


Fig. 7.

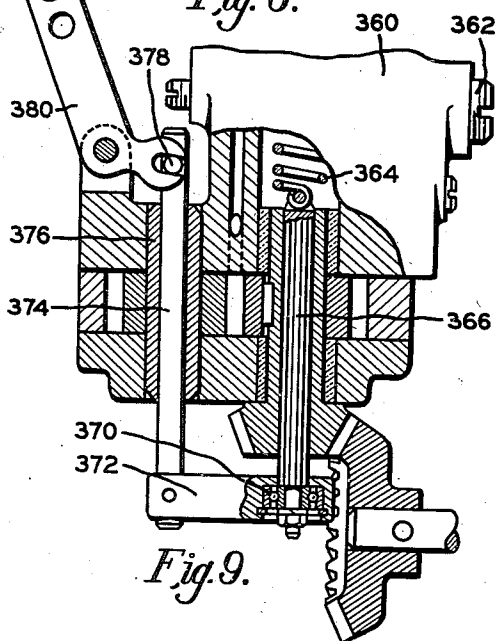


Fig. 9.

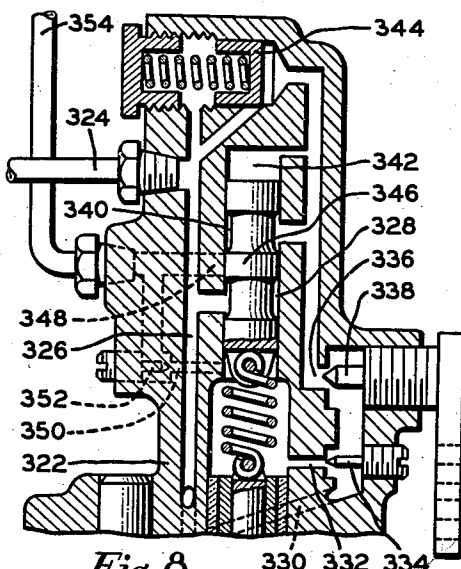


Fig. 8.

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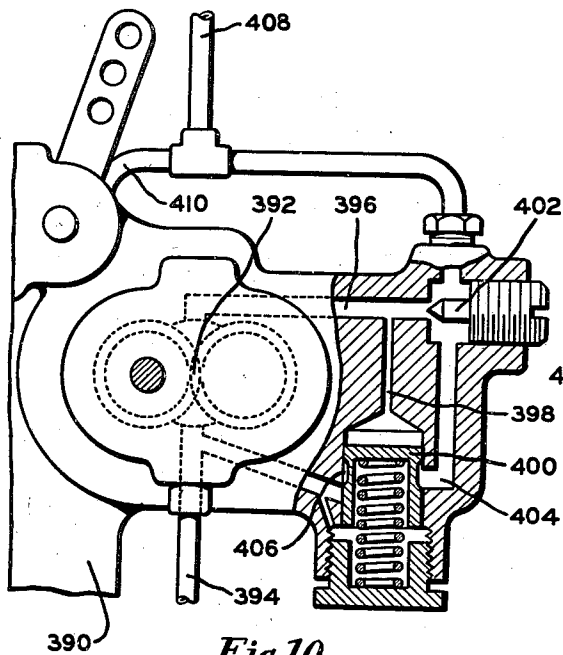


Fig. 10.

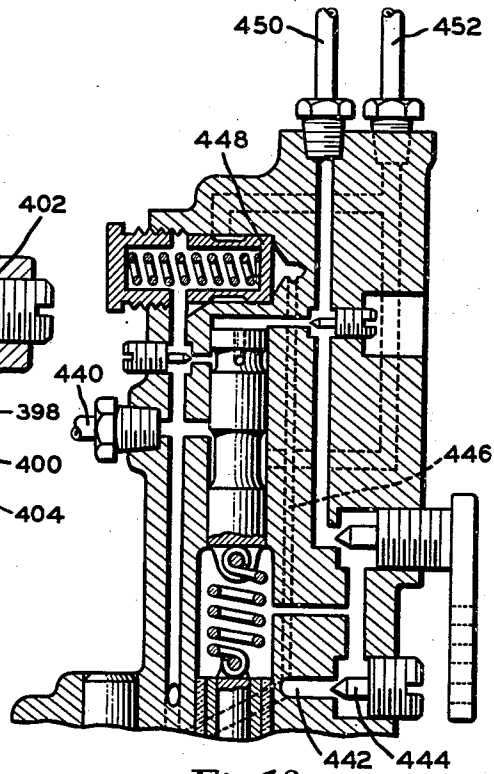


Fig. 12.

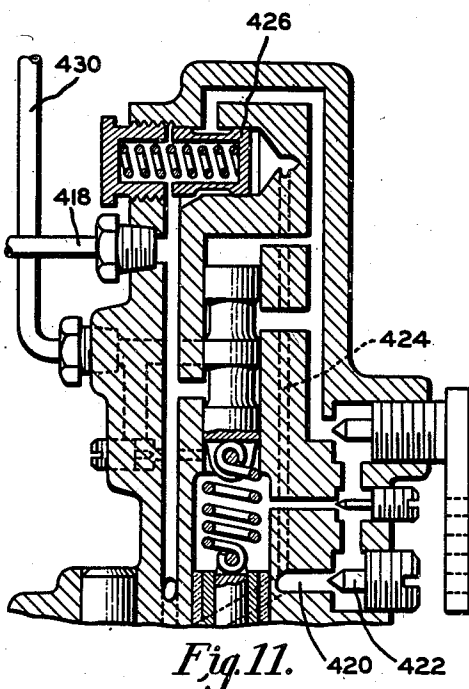


Fig. 11.

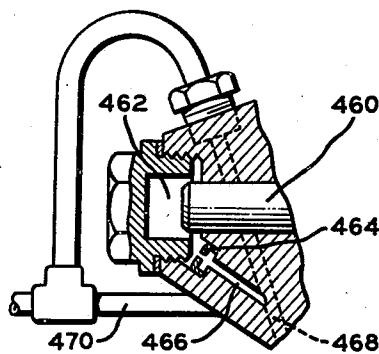


Fig. 13.

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UNITED STATES PATENT OFFICE

2,384,340

GOVERNING MEANS

Ferdinando Carlo Reggio, Harwich Port, Mass.

Application May 6, 1940, Serial No. 333,529

19 Claims. (Cl. 264—3)

This invention relates to governing means for prime movers and to automatic regulating devices, and in so far as the subject matter is common it is a continuation in part of my copending patent applications Serial No. 236,249, filed October 21, 1938, now Patent No. 2,313,264, issued March 9, 1943, and Serial No. 254,355, filed February 3, 1939.

In one of the preferred embodiments of the invention the governor or regulating device includes hydraulic servomotor means controlled by a pilot valve. The latter is actuated by means responsive to an operative condition such as the speed of a prime mover, and is further subject to a fluid pressure load which varies with changes of adjustment of said servomotor means.

An object of the invention is to provide an improved construction and arrangement of governor for prime movers.

Another object is to provide a simple and compact construction and arrangement of hydraulic governor.

A further object is to provide a hydraulic governor for prime mover, having a pilot valve actuated in response to changes of speed and load of said prime mover.

A still further object resides in the provision of resilient means for continuously rotating the plunger pilot valve and transmitting an actuating load thereto.

Another object resides in the provision of simple and accurate means for adjusting the speed difference of the prime mover between no load and full load operation of the latter.

A still further object of the invention resides in the provision of an automatic regulating mechanism having simple and effective anti-hunting arrangement.

It is to be clearly understood that the governor according to the invention may be used in connection with any suitable prime mover or engine, to regulate the supply of energy or fuel thereto or to vary other operative condition upon which the speed thereof is dependent. The governor may thus be employed to actuate an admission valve, a throttle, the control member of a fuel injection pump, or it may be used for regulating the pitch of a variable-pitch aircraft propeller to control the speed of the engine driving said propeller, etc.

One example of embodiment of the invention is disclosed herein, in which the governor is employed to control the fuel supply of an internal combustion engine provided with fuel injection units separately mounted on the cylinders there-

of. In this specific embodiment a further object involves the omission of the mechanical linkage between the governor and the fuel injection pumps and the substitution therefor of a hydraulic column whose pressure may be varied by said governor. The substitution of such hydraulic column in place of a mechanical linkage may be of especial use where the fuel injection elements are far removed from the governor and from one another. Under such circumstances the friction losses incidental to the considerable length of mechanical linkage are avoided. Furthermore the danger that a damaged fuel pump element might interfere with the operation of the governor and the control of the other pump elements is eliminated.

The fuel delivery of the injection pumps is a predetermined function of the pressure in the liquid column interconnecting pump elements and governor. The latter includes a pressure regulating valve designed to control the pressure of said liquid column and thereby the engine fuel supply. The adjustment of said valve depends on the simultaneous values of the forces transmitted thereto by:

(a) Resilient means under the control of the operator for setting the desired engine speed;

(b) Means responsive to the engine speed; and

(c) A hydraulic pressure substantially equal or proportional to the pressure of said liquid column and therefore varying as a predetermined function of the engine fuel supply.

One more specific object of the invention is to provide means whereby the ratio between the hydraulic pressure mentioned in c and the pressure of said liquid column may be adjusted substantially between 1 and the lowest possible value consistent with steady engine operation. Varying this ratio between said values causes the "speed droop" of the engine, or the speed range between full load and no load, to vary between its maximum designed value and its minimum actually possible value.

The above and other objects and advantages of the invention will be apparent as the description proceeds; and while I have illustrated and described some of the preferred embodiments of the invention as they now appear to me, it will be understood that such changes may be made as fall within the scope of the appended claims. It will further be understood that the invention is not limited to governing means in which the pilot valve is actuated in response to changes of speed, but may be used in connection with other automatic regulating devices in which the pilot

valve is actuated in response to changes of one or more operative conditions other than speed, in order to avoid irregular operation thereof such as hunting or throbbing. In the following description and in the claims various details will be identified by specific names for convenience, but they are intended to be as generic in the application as the art will permit.

A few examples of embodiment of the invention are illustrated in the accompanying drawings in which:

Fig. 1 is a longitudinal section showing one arrangement of hydraulic governor and fuel injection pumps;

Fig. 2 is a fragmentary section along line 2—2 of Fig. 1;

Figs. 3 and 4 are in part elevation and in part longitudinal sections of modified arrangements of hydraulic governor and injection pumps;

Fig. 5 is a fragmentary longitudinal section showing an alternative construction of part of the hydraulic governor;

Figs. 6 to 9 are longitudinal sections through hydraulic governors of a different type;

Fig. 10 is a fragmentary section showing a partial modification of the governor of Fig. 1;

Figs. 11 and 12 are sections of other modified types of governors;

Fig. 13 is a fragmentary section of a modified arrangement of fuel injection pump.

A fuel injection pump 13 for internal combustion engine, shown in Fig. 1, has a plunger 14 having a reciprocating motion of constant stroke in a barrel 15 provided with a port 17. At the inner end of the plunger, beyond an annular groove 16, the surface of the plunger contacting the bore of barrel 15 and adapted to intermittently close the port 17 is limited by at least one inclined edge. The effective discharge stroke of the plunger is that portion of the inward displacement of the plunger during which the port 17 is closed and the liquid fuel displaced in pressure chamber 19 is forced through a spring loaded valve and a nozzle into the engine cylinder. Such valve and nozzle, not shown in detail in the drawings, is not a part of the invention, and may be of any suitable type. Owing to the inclined edge at the inner end of plunger 14 the effective stroke and in turn the fuel delivery varies with the angular adjustment of the plunger relative to port 17.

An intermediate portion of plunger 14 is slidably but non-rotatably mounted in a pinion 24 axially located between the barrel 15 and a washer 25 resting against a shoulder of pump housing 13. The outer end of plunger 15 is formed with a head 26 holding a seating collar 27 for the coil spring 28 and is enclosed by a cover 29 slidably mounted in a bore of housing 13. A rocker arm 30 operated from an engine-driven cam actuates the plunger for the pressure stroke, the spring 28 effecting the return or suction stroke. A small screw 31 through the housing 13 engages a slot 32 in the cover 29 to keep the latter in place when the rocker arm 30 is removed.

The plunger 14 may be angularly turned for adjustment of the fuel delivery by a rack cut in the intermediate portion of a plunger 35 and meshing with pinion 24. One end of rack 35 extends into a chamber 37 closed by a cap 38 to which fuel is led from the fuel feed line 40. A calibrated coil spring 42 is mounted between the opposite end of rack 35 and a threaded adjustable cap 43 provided with a lock nut 44. A rod 45 of small diameter secured to rack 35 and

coaxial therewith extends through a central opening of cap 43 and can be manually operated to temporarily modify the axial adjustment of rack 35. The extreme position that the latter can reach toward the right may be adjusted by means of a screw 46 provided with a lock nut and cooperating with a notch 47 formed in the rack 35.

An annular chamber 50 is provided in housing 13 about the barrel 15. Chamber 50 communicates with pressure chamber 19 through port 17, and with inlet chamber 37 through a duct 51. The injection pump 13 is attached to the engine by means of cap screws 52.

In a multicylinder engine all of the fuel pumps 13 are connected with the fuel feed line 40, the inner section of which is sufficiently large to secure a substantially uniform fuel pressure in all points of the line. Two such pumps 13 connected with line 40 are shown in Fig. 1.

The adjustment of rack 35 and in turn the fuel discharge are dependent on the value of the fuel pressure in line 40. An increase in fuel pressure causes a movement of rack 35 toward the right until the increased hydraulic load on the latter is balanced by a correspondingly increased resilient load of spring 42. The preferred arrangement is such that a displacement of rack 35 toward the right causes an increase of fuel delivery of the pump. Fuel pressure in feed line 40 and fuel discharge of pumps 13 will therefore simultaneously increase or decrease according to a predetermined relationship. The springs 42 of the various pumps 13 have the same rate of deflection and their loads are initially adjusted by means of the threaded caps 43 so that with a given pressure in the inlet chamber 37 all of the pumps 13 discharge a determined amount of fuel per plunger stroke. Screws 46 are also set for identical maximum value of fuel discharge per stroke. Pumps 13 are therefore interchangeable and when connected with a feed line 40 adapted to maintain a substantially uniform pressure in all the inlet chambers 37 they will deliver the same amount of fuel.

Control of the engine fuel supply may thus be obtained by governing the fuel pressure in the feed line 40.

To that end a fuel transfer pump 60 shown as a pump of the gear type but that may be of any other suitable type, driven from the engine or by other prime mover, draws fuel at substantially atmospheric pressure or other substantially constant pressure from a supply line 61 and discharges it into lines 62 and 40. Between discharge and inlet ports of pump 60 there is a by-pass 63 which is controlled by a valve 64 loaded by a spring 65 held by an adjustable cap 66 set for a determined maximum value of the fuel pressure in lines 62 and 40. When the pressure in the latter reaches said value, valve 64 opens the by-pass and prevents any further increase in fuel pressure.

A speed governor 70 includes a shaft 71 rotatable in a sleeve 72 and driven from the engine through a gear 73 and a pinion 74. Shaft 71 carries flyballs 75 having extreme outer and inner positions determined by shoulders 76 and 77 formed on shaft 71. Inwardly extending arms of flyballs 75 engage a shoulder of the pilot valve 80, which valve is mounted in a bore of body 81 coaxial with shaft 71 and is adapted to rotate and reciprocate therein. A splined rod 83, slidable in a splined axial hole broached in the shaft 71, carries at its upper end the inner race of a

thrust bearing 84 the outer race of which is secured to a sleeve 85 slidable in a bore of housing 70. A pinion 86 connected with speed control lever 87 engages a rack cut in sleeve 85 and serves to adjust the axial position of the latter.

A compression spring 90 has its upper coil tightly wound about a groove formed at the lower end of rod 83, while its lower coil is solidly seated in a groove machined at the upper end of the pilot valve 80. A groove 91 is machined in the intermediate portion of valve 80 and is permanently connected through a port 92 with the line 62, while it communicates with fuel supply line 61 through a port 93 whose open area is dependent on the axial adjustment of valve 80. Below valve 80 a compensating chamber 95 is connected with port 92 and line 62 through an orifice 96 adjustable by means of a needle valve 97, and is connected with port 93 and line 61 through an orifice 98 adjustable by means of a needle valve 99. When valve 80 is in its lower position communication between compensating chamber 95 and port 93 is interrupted.

When the engine is inoperative the fuel pressure in lines 62 and 40 is the same as in the supply line 61. The injection pumps 13 are adjusted for no delivery. The pilot valve 80 is in its lower position and closes the orifice 98. The valve 64 is also closed, as well as port 93. Assuming the fuel transfer pump 60 to be driven from the engine, as the latter is cranked fuel is discharged into lines 62 and 40 and the pressure therein rises rapidly and reaches its maximum value whereupon valve 64 opens the by-pass 63 and prevents a further increase in pressure. Accordingly the injection pumps 13 are adjusted for maximum fuel delivery and, actuated by the rocker arms 30, inject fuel into the engine cylinders. As firing occurs in the latter and the engine speed increases and approaches the value corresponding to the setting of the speed adjusting lever 87, valve 80 is lifted by the flyballs 75 and uncovers first the orifice 98 and afterwards a portion of the area of port 93. The area of orifices 96 and 98 adjusted by needle valves 97 and 99 respectively being very small, the amount of fuel by-passed through the compensating chamber 95 is very small and has no noticeable effect on the value of the fuel pressure in the feed line 40 whenever the speed of the transfer pump 60 is above that corresponding to low engine cranking speed.

Since the capacity of the transfer pump 60 is in excess of the total maximum discharge of the injection pumps 13, when the engine is operating under the control of the governor, that is when the by-pass 63 is closed by valve 64, the excess fuel is by-passed through the open area of port 93. A downward displacement of valve 80 causes a reduction of said open area and therefore an increase in speed of the fuel flowing through said area and a corresponding increase of pressure in the feed line 40. Conversely, an upward displacement of valve 80 causes an increase of the open area of port 93, a reduction of speed of the fuel by-passed through said port and a corresponding decrease of pressure in line 40.

The adjustment of the pilot valve 80 relative to port 93 is dependent on the simultaneous actions of the load of spring 90, of the centrifugal force acting on the flyballs 75, and of the fuel pressure in the compensating chamber 95. It is therefore dependent on the adjustment of the

speed adjusting lever 87, on the engine speed and on the engine fuel supply.

Any governor in order to be stable must regulate the speed of its engine, at least temporarily, in such a manner that the speed is lower for full load than for no load. If the difference between maximum and minimum speed divided by the average speed, or "speed droop," is high the speed regulation is "soft" and the change from a given amount of engine fuel supply to another caused by a change of load on the engine is not abrupt but gradual. A smaller speed droop causes a more abrupt speed regulation because small speed variations involve large changes in the engine fuel supply. Assuming the orifice 98 to be closed by the needle valve 99 while the orifice 96 is open, the fuel pressure in the compensating chamber 95 is substantially the same as in the feed line 40. The actual engine speed is dependent on the load of the compression spring 90 minus the load applied to the lower end of valve 80 by the fuel pressure in chamber 95. When the engine is operating under steady load the valve 80 is in stable equilibrium; if the engine load increases the engine speed decreases and the centrifugal force acting on the flyballs 75 also decreases causing a downward movement of valve 80 and an increase of fuel pressure in feed line 40. The discharge of the injection pumps 13 increases and the engine resumes speed. The fuel pressure in the compensating chamber 95 also increases while the load of spring 90, due to the extremely small vertical motion of valve 80 is substantially unchanged. The equilibrium of the governor will be reestablished when the engine reaches the speed that corresponds to that value of the centrifugal force of flyballs 75 that balances the load of spring 90 minus the load due to the fuel pressure in chamber 95. The engine speed under the heavier engine load will therefore be lower than the engine speed under lighter engine load.

Conversely when a drop in engine load occurs the engine accelerates and the flyballs 75 lift the valve 80 causing a reduction of fuel pressure in line 40 and in compensating chamber 95. The fuel discharge of injection pumps 13 decreases causing the engine speed to decrease until the centrifugal force of flyballs 75 balances the load of spring 90 minus the load due to the fuel pressure in chamber 95. The decrease in engine load will therefore cause an increase of engine speed. The condition of stability of the governor, that is the ability to cause the engine speed to decrease when the engine load increases, is thus satisfied. The desired nominal engine speed is set by means of lever 87 that may be adjusted by the operator or by automatic devices. Anticlockwise rotation of lever 87 increases the load of spring 90 and in turn the engine speed. A stop may be provided to limit the anticlockwise rotation of lever 87 thereby determining the maximum engine speed, while an adjustable resilient stop may limit the clockwise rotation of the lever to set the engine idling speed. Neither stops are shown in the drawings. Clockwise rotation of lever 87 beyond its position determined by the resilient stop causes valve 80 to uncover port 93 completely thereby stopping the engine.

The stabilizing or anti-hunting effect obtained by the application of an axial variable fluid pressure load to the pilot valve 80 may more clearly and distinctly be described as follows: If the engine decelerates, as it happens when the load

thereon is increased, the spring 90 overcoming the diminishing centrifugal force applied to the flyballs 75 moves the pilot valve 80 downward so as to increase the pressure in the fuel line 40 and displace the hydraulic servomotor member or pressure-actuated control rack 35 toward the right to increase the engine fuel supply. In the absence of any change of pressure in chamber 95 the displacement of the pressure actuated member 35 would eventually be checked as the engine accelerates and the increasing centrifugal force moves the pilot valve 80 back to its normal position, but by this time the engine is still accelerating and an unnecessarily high engine speed is attained, causing overtravel of the pilot valve 80, whereupon the pressure in the fuel line 40 drops to an exceedingly low value and causes a displacement of the pressure actuated rack 35 toward the left and a decrease of engine fuel supply. The engine decelerates, the speed thereof drops to a value lower than normal, and hunting may continue indefinitely.

To correct this condition of instability various compensating or anti-hunting devices have been proposed. The anti-hunting arrangement according to the invention as illustrated in Fig. 1 consists essentially in the provision of a fluid pressure chamber 95 at one end of the pilot valve 80 for exerting on the latter an axial variable pressure load in the following manner. As upon increase of load the engine decelerates, the spring 90 overcomes the diminishing centrifugal load and shifts the pilot valve 80 downward, causing a pressure increase in the fuel line 40 and a displacement of the rack 35 toward the right so as to increase the engine fuel supply. As a result the pressure in chamber 95 also increases and restrains the downward movement of the pilot valve. Although this restraining pressure exerted on the pilot valve is very small as compared to the centrifugal force, it causes the valve 80 to resume its normal adjustment quickly so that overtravel of the valve, overcorrection by the servomotor member 35 and overspeed of the engine do not occur.

When the engine accelerates upon a drop of engine load the action of the governor is the reverse of that just described, causing a pressure drop in chamber 95 and thus restraining the upward displacement of the pilot valve 80 so as to avoid overtravel of the latter and overcorrection on the part of the servomotor member 35. The danger of hunting is thus eliminated.

In the foregoing the assumption has been made that orifice 98 is closed and the fuel pressure in the compensating chamber 95 acting on valve 80 has substantially the same value as the fuel pressure in the feed line 40 and in chambers 37 wherein said pressure actuates the racks 35, the operation of the governor being thus similar to that disclosed in my said copending application Serial No. 236,249, in which the governor has a predetermined non-adjustable amount of speed droop. It is to be clearly understood that according to the invention, where it is not desired that said speed droop be adjustable, the governor of Fig. 1 may be simplified by eliminating the orifice 98 and the needle valve 99. Needle valve 97 may also be eliminated, or used for controlling the dampening effect of orifice 98 on the movements of valve 80. Similar modifications may obviously be introduced in the other forms of governors illustrated in the drawings. If we now assume that orifice 98 is closed by needle valve 97 while the orifice 98 is open, the pressure

in chamber 95 will be constant and the adjustment of valve 80 will be dependent on the load of spring 90 and on the centrifugal force of flyballs 75, and substantially independent of the engine fuel supply. No restraining action is exerted on the pilot valve upon initiation of the correcting movement of the servomotor member 35. Operation of the governor is therefore likely to be unstable.

Obviously when both orifices 96 and 98 are open the fuel pressure in chamber 95 has a value comprised between the pressures in lines 40 and 61. Assuming, for simplicity's sake, that the pressure in line 61 is equal to the surrounding atmospheric pressure, the effective pressure p in the compensating chamber 95 is

$$p = \frac{p_1}{1 + \left(\frac{a_2}{a_1}\right)^2}$$

where p_1 is the effective pressure in line 40 and a_1 and a_2 are the open areas of orifices 96 and 98 respectively. That is, the ratio between the fuel pressures in chamber 95 and in line 40 may be varied between 1 and 0 by properly adjusting the needle valves 97 and 99. The speed droop varies substantially proportionally with said ratio.

Any determined value of speed droop within the designed limits of the governor corresponds to a definite ratio between the open areas of orifices 96 and 98. Yet the open area of either one or the other of said orifices may be arbitrarily chosen so as to obtain a suitable dampening effect, or "dash pot" action, on valve 80.

The only friction that the governor has to overcome is that opposing the axial displacement of valve 80, which friction has a low value due to the fact that valve 80 is non-rotatably connected by means of spring 90 with splined rod 83 and therefore continuously turns in the bore of body 81. Friction opposing the axial displacement of the racks 35 of the injection pumps 13 does not influence the operation of the governor. If one of the pump plungers 14 should stick in its barrel and remain stationary at the end of its discharge stroke with the load of spring 28 insufficient to bring it back against the rocker arm 30, the pump would stop delivering fuel to its cylinder, but it would not interfere with the control of the remaining pumps, while with a conventional mechanical linkage interconnecting all the racks 35 the damaged pump might prevent the regulation of the other pumps thereby putting the engine out of control.

A further advantage of the present injection system is the possibility of checking the operation of each pump while the engine is operating by temporarily increasing or decreasing its fuel delivery by means of rod 48.

The effect of the friction resistance opposing the axial movement of the racks 35 is reduced by the pressure surges caused by the rapid opening and closing of port 17 and transmitted to the inlet chamber 37 through conduit 51. Owing to said surges the hydraulic load applied to the racks 35 has a pulsating character. The range of said high frequency pulsation of the load applied to the racks may be reduced, if it should appear desirable, by partially modifying the injection pump 13 of Fig. 1 as shown in Fig. 13, in which fuel from the feed line 470 is led through conduit 480 to the annular chamber surrounding the pump barrel and its port. Said chamber, where the surges of the fuel pressure originate,

is connected with chamber 462 into which the rack 460 extends by means of a duct 466 provided with a plug 464 having a small orifice adapted to dampen the high speed surges while it does not appreciably interfere with the comparatively slow motion of the rack due to the action of the governor. The spring 42 will be preferably designed so that its load is considerably in excess of the friction resistance opposing the sliding motion of the rack 35, or other load which may be exerted thereon, so that variations in the value of the latter load or friction will be comparatively negligible, and the adjustment of the pressure actuated rack 35 will be substantially determined by the fluid pressure in chamber 37.

Obviously, non-adjustable orifices may be substituted for the needle valves shown in Fig. 1. A partial modification of the latter is indicated in Fig. 5, in which a compensating chamber 105 formed in the governor body 106 at the lower end of valve 80 communicates through an orifice 107 adjustable by means of a needle valve 108 with a duct 109 connected through an orifice 110 adjustable by means of a needle valve 111 with a conduit 112 leading to the fuel supply line indicated by numeral 61 in Fig. 1, and connected through an orifice 113 drilled in a thin wall of plug 114 with conduit 115 communicating with the fuel feed line. A small cushion chamber 116 may be provided at the top of duct 109, filled with air or vapor separating from the fuel. The engine speed droop is adjusted by needle valve 111, while different degrees of dampening of the axial displacement of valve 80 are obtained with various adjustments of needle valve 108. High frequency pressure surges that might be transmitted to the compensating chamber 105 may be suppressed and pressure variations therein may be delayed by the cushion chamber 116. The rate of pressure change in the chamber 105 is dependent upon the effective area of the orifices 113 and 110 as well as on the volume of the air chamber 116. Where the volume of the latter is large and the orifices 113 and 110 are small the pressure variations in chamber 105 occurring upon a certain alteration in the adjustment of the servomotor member 35 or 202 are slower than they would be with a smaller air chamber 116 and/or larger orifices. The effective volume of the air chamber and the orifice areas may thus be dimensioned or adjusted so as to obtain the desired magnitude and lag in the restraining pressure exerted on the pilot valve so as to adapt the governor or regulator to suit systems which respond with various degrees of promptness to the controlling movement of the servomotor. It will be obvious that in substitution for or in addition to the air chamber 116 other equivalent cushioning means may be provided such as resiliently movable wall means, spring loaded plunger, or bellows or diaphragm means; and that non-adjustable orifices may be substituted for needle valves 111 and 108.

In injection systems in which the fuel injection pumps are mounted on the engine cylinders it may be convenient to provide for their continuous cooling and venting by means of an active circulation of fuel therein. Such an arrangement, including various elements already disclosed in detail, is shown in Fig. 3, in which a transfer pump 121 draws fuel at substantially constant pressure from a fuel supply line 120 and discharges it under pressure into the feed line 122. The injection pumps 124 are con-

nected in parallel between the feed line 122 and the return line 130 from which fuel is led to the governor 142 through a line 136. In each pump the fuel flows through inlet chamber 123 into which one end of the rack extends, duct 132, annular chamber 126 surrounding the pump barrel and its port 128, and outlet duct 134. The fuel conduits connecting the feed line 122 with the inlet port 138 of governor 142 are of such large sections that under all operating conditions the fuel pressure in the inlet chambers 123 and pumps 124 and in the inlet port 138 of governor 142 is substantially the same. Between the discharge port of the transfer pump 121 and the fuel feed line 122 a fuel strainer or filter, not shown in the drawings, may be inserted.

In the governor body 142 a by-pass 150 connects the inlet port 138 with port 146 partly covered by the valve 144 and communicating with the fuel supply line 120. The by-pass 150 is controlled by a spring-loaded maximum pressure regulating valve 152 having the same function as valve 64 of Fig. 1. Operation of the governor is similar to that disclosed in connection with Fig. 1 and it is therefore regarded unnecessary to again describe it in detail.

In the foregoing injection systems the hydraulic medium used in the governor and in the hydraulic operating connection between governor and injection pumps is the liquid fuel burnt in the engine cylinders, such as Diesel oil, or gas-oil, or gasoline.

However according to the present invention other hydraulic mediums may be used, for example the engine lubricating oil as it is illustrated in the arrangement shown in Fig. 4 of the drawings, in which the engine injection pumps 160 are supplied with fuel by way of a fuel line 162 connected through a conduit 164 with the annular chamber 166 surrounding the pump barrel and its port 168. The fuel pressure in line 162 is maintained at a suitable value by a conventional transfer pump, or by gravity. When the engine operates, fuel is intermittently admitted to the pressure chamber 172 and forced by plunger 170 through an injection valve, not shown in section, into the engine cylinder.

A pump 174 driven from the engine or other suitable prime mover draws lubricating oil at substantially constant pressure from line 176 connected with the engine oil sump or other suitable reservoir and discharges it under pressure through line 178 and port 180 into an upper groove 182 formed in the pilot valve 184 of the engine-driven governor 186. A lower groove 188 formed in the same valve is connected through port 190 with the oil supply line 176. The portion 192 of valve 184 comprised between grooves 182 and 188 controls a port 194 communicating by way of lines 196 and 198 with the chambers 200 of each of the fuel injection pumps 160, into which chambers one end of racks 202 extends. A by-pass 204 controlled by a spring-loaded valve 206 connects the discharge and inlet ports of pump 174. The diameter of port 194 and the axial length of portion 192 of valve 184 are equal so that when the latter is in its neutral position as shown in Fig. 4 port 194 is closed, and a small axial displacement of valve 184 above or below its neutral position causes port 194 to communicate with groove 188 or groove 182 respectively. The governor operates as follows: a substantially constant pressure is maintained in line 178 and in groove 182. This pressure is higher than the

pressure existing in the lower groove 188. The pilot valve 184 slidably and rotatably mounted in its bore is actuated in the same way as valve 80 of Fig. 1, that is valve 184 is controlled by the simultaneous actions of a spring adjusted by way of lever 208, of flyballs driven from the engine and of the oil pressure in the compensating chamber. The effects of changes of the adjustment of valve 184 are also the same as for the valve 80 of Fig. 1, that is an upward movement of the valve, connecting line 198 with the low pressure groove 188 and line 176, causes a decrease of pressure in chambers 200 and a corresponding reduction in the engine fuel supply, while a downward movement of valve 184, connecting line 198 with the high pressure groove 184, causes an increase of pressure in chambers 200 and a corresponding increase of engine fuel supply. The restraining action exerted on the pilot valve 184 is also the same as that already disclosed in detail in connection with the pilot valve 80. The operation of the governors of Fig. 4 and Fig. 1 is therefore similar. Under steady engine load operation a small amount of lubricating oil flows through port 194 to compensate for the oil by-passed through the compensating chamber.

In injection pumps 160 designed for handling fuels of poor lubricating property a duct 214 may be provided to lead lubricating oil from the chamber 200 to a groove formed in the bore of the pump barrel in order to lubricate the plunger 170. Thus the pressure of the lubricant increases with the load of the injection pump.

The governor illustrated in Fig. 4, for example, includes a fluid pressure actuated plunger 202 controlled by means of the valve 184 and having a rack and pinion connection with the injection pump plunger 170 to control the engine fuel supply. It is to be clearly understood, however, that according to the invention any suitable fluid pressure actuated means may be substituted for the plunger 202. Said fluid pressure actuated means may be, or include, a piston slidable in a cylinder, or a flexible diaphragm or bellows, and as already stated may be used in different manners, for instance to actuate the admission valve of a prime mover, the carburetor throttle of an internal combustion engine, or may be employed to adjust the pitch of a variable-pitch propeller driven from an aircraft engine, in order to control the speed of said prime mover, internal combustion engine, or aircraft engine, respectively. The foregoing obviously applies to any of the governors herein disclosed.

The foregoing governors are of the centrifugal type, that is include engine-driven flyballs transmitting to the governor valve a load proportional to the square of the engine speed. According to the present invention equivalent speed-responsive means may be substituted for said flyballs.

Fig. 6 shows a hydraulic governor designed for use of liquid fuel as hydraulic medium and that may be substituted for the governor of Fig. 1. A fuel transfer pump of the gear type 220 including an idler gear 222 and a driving gear 224 keyed to a shaft 226 supported in bearings 228 and 230 is driven from the engine by way of a pinion 232. A rod 234 tightly mounted in a longitudinal central bore of shaft 226 and axially adjustable by means of a threaded portion 236 and a lock nut 238 is secured to the lower end of a tension coil spring 240. The upper end of said spring is attached to a supply valve 242 slidably and rotatably mounted in a bore coaxial

with shaft 226 formed in the governor body 244. The valve 242 has a central groove 246 and an upper groove 248. The latter is connected with the compensating chamber 252 at the upper end of valve 242 by way of a duct 250. The inlet port 258 of transfer pump 220 is connected by way of duct 256 with the fuel supply line 254, with port 264 controlled by the valve 242, with the low pressure side of a spring-loaded valve 266, and by means of an orifice 260 adjustable by a needle valve 262 with groove 248. The discharge port 270 of the transfer pump 220 communicates by way of port 274 with the spring compartment 272 and, through an orifice 276 adjustable by means of a needle valve 278 secured to the speed lever 280, with duct 282. The latter in turn is connected through port 284 with groove 246, with the high pressure side of valve 266, through orifice 286 adjusted by needle valve 288 with compensating chamber 252, and by way of line 290 with the fuel feed line 40 and the fuel injection pumps 13 of Fig. 1.

The transfer pump 220, drawing fuel from the supply line 254 in which the pressure is kept substantially constant, discharges a volume of fuel proportional to the engine speed through the orifice 276 thus determining on opposite sides of said orifice a difference of pressure that is proportional to the square of the engine speed. Assuming the orifice 260 to be closed by the needle valve 262, owing to the difference of fuel pressure between chambers 272 and 252 a hydraulic load is applied to the valve that tends to lift the latter and has the same character as the centrifugal load transmitted to the valve 80 of Fig. 1 by the flyballs. Said hydraulic load is balanced by the load exerted on valve 242 by the tension spring 240. The compensating device being inoperative and no restraining action being exerted on the pilot valve 242, the governor is unstable. If both orifices 286 and 260 are open the pressure in compensating chamber 252 has a value comprised between the pressures in line 290 and in supply line 254. Assuming again, for simplicity's sake, the fuel pressure in fuel supply line 254 to have the surrounding atmospheric value, for a given adjustment of needle valves 288 and 262 the ratio between the pressures in line 290 and in compensating chamber 252 has a definite value. The difference between said pressures, and the intensity of the compensating effect applied to valve 242, therefore increase or decrease when the fuel pressure in line 290 and the engine fuel load increase or decrease respectively. The movement of the pilot valve 242 determined by speed changes is thus restrained upon initiation of the correcting displacement of the pressure actuated member or servomotor. Increasing the open area of the orifice 260 or decreasing the open area of orifice 286 causes an increase of the speed droop. Said compensating and restraining effect, tending to lift the governor valve 242 against the load of spring 240, is thus equivalent to that described in connection with Fig. 1. Furthermore, axial upward or downward displacements of valve 242 again cause a decrease or increase of pressure in line 290 and a decrease or increase of engine fuel supply respectively. The maximum fuel pressure is determined by valve 266. It is therefore evident that the operation of the governors of Figs. 1 and 6 is similar. However in the governor 244 the speed control lever 280, instead of adjusting the load of the governor spring, controls by way of the needle valve 278 the open area of orifice 276. Changes in the adjustment of

lever 280 cause a modification of the pressure difference between chamber 272 and compensating chamber 252 and therefore a variation of engine speed. The valve 242 is carried along in rotation from shaft 226 through rod 234 and spring 240, so as to reduce the effect of friction resistance tending to oppose the axial displacement of the only movable part of the governor.

While in the previously described governors, for a given adjustment thereof, the speed droop is smaller and the energy is larger at high speed than at low speed, in the governor of Fig. 6, in which the load of spring 240 is substantially independent of the engine speed, the energy and the speed droop are also substantially independent of the engine speed.

A substantially similar governor 300, that may be substituted for the governor of Fig. 3, is shown in Fig. 7 in which an amount of fuel proportional to the engine speed, drawn at constant pressure from the supply line 302 by a transfer pump not shown in the drawings, is discharged through an orifice 306 whose open area may be controlled by way of a needle valve 308 secured to the speed control lever, thus determining a difference of pressure exerting on the governor valve an axial load proportional to the square of the engine speed. At the upper end of the valve a compensating chamber is provided. The total amount of fuel discharged by the transfer pump, minus the negligible amount flowing through the compensating chamber, is led through line 310 to the fuel feed line 122 of Fig. 3, while line 312 is connected with the return line 130 of the same Fig. 3. The fuel injection pumps 124 are connected in parallel between lines 122 and 130. The fuel lines are of sufficiently large section so as not to create objectionable differences of pressure in the various points of the pressure system. The operation of this form of governor is otherwise similar to that of governor 246.

In the governor 322 of Fig. 8, which is interchangeable with governor 186 of Fig. 4, lubricating oil at substantially atmospheric pressure or other constant pressure is drawn through line 324 and duct 326 by a transfer pump, not shown, and discharged in volume proportional to the engine speed through an orifice 336 adjustable by way of a needle valve 338, thus creating a difference of pressure, proportional to the square of the engine speed, at the opposite ends of the pilot valve 346. The oil pressure beyond the orifice 336, at the upper end of valve 346 and in the upper groove 340 of the same is kept at a substantially constant pressure by means of a spring loaded valve 344, while in the lower groove of valve 346 the oil pressure is the same as in supply line 324. When the valve 346 is in its neutral position, as shown in Fig. 8, its intermediate portion defined by said grooves covers port 348 connected by way of line 354 with line 198 of Fig. 4. The oil pressure in line 354 thus controls the engine fuel supply. An upward displacement of valve 346, or a downward displacement of the same, establishing a communication between line 354 and lower groove 322 or upper groove 340 of valve 346, causes a decrease or an increase respectively of the pressure in line 354. At the lower end of valve 346 the spring compartment or compensating chamber is connected through an orifice adjustable by means of needle valve 334 with the discharge port 330 of the oil pump, and through the orifice 350 adjustable by way of needle valve 352 with line 354. An increase or decrease of

pressure in line 354 thus produces an increase or decrease respectively of the pressure in the compensating chamber, resulting in a compensating load applied upwardly to the valve 346 against the action of the governor spring, which load is larger or smaller when the engine load is higher or lower. The operation of governor 322 is therefore similar to that of the previously described governors.

Fig. 9 shows a partial modification of the governors of Figs. 6 to 8 whereby the engine speed may be set by adjusting the load of spring 364 the lower end of which is secured to a splined rod 366 slidable in a splined hole axially broached in the governor shaft. A thrust bearing 370 has its inner race secured to rod 366 while its outer race is carried by a bracket 372 attached to a rod 374 slidable in an axial bore of the idler gear shaft 376. A pin 378 carried by rod 374 engages a fork machined at the lower end of lever 380. The load of spring 364, and in turn the engine speed, may thus be set by way of said lever.

In the foregoing injection systems the adjustment of the injection pumps corresponding to the actual maximum delivery per stroke is determined either by the screw 46 limiting the displacement of rack 35 or by the adjustment of the maximum pressure regulating valve. The actual maximum fuel delivery per stroke of the injection pumps is substantially independent of the engine speed, being only affected by changes of volumetric efficiency versus engine speed. However in many applications it is desirable that the engine torque should be considerably larger at low speed than at high engine speed. To that end the maximum pressure regulating valve 64 of Fig. 1 may be modified as shown in Fig. 10 in which the discharge port 396 of the fuel transfer pump 392 is connected by way of duct 398 with the upper chamber of a spring-loaded maximum pressure regulating valve 400, and through an orifice adjustable by means of a needle valve 402 with a line 408 leading fuel to the feed line 40 of Fig. 1, with a line 410 connected with the governor, and with a by-pass 404 communicating with the inlet port of the transfer pump 392. Said by-pass 404 is controlled by valve 400 in such a way that when the fuel pressure in conduit 396 reaches a predetermined value the annular groove 406 of valve 400 registering with by-pass 404 establishes a communication between discharge and inlet ports of pump 392 thereby preventing a further increase of pressure in conduit 396. The total fuel delivery of pump 392 is forced through the orifice adjustable by needle valve 402 thereby creating a difference of pressure proportional to the square of the engine speed between conduit 396 and line 408. The maximum value that the fuel pressure may attain in line 408, and in turn the maximum value of the discharge per cycle of the injection pumps, is therefore smaller at high engine speed than at low speed. Decreasing or increasing by means of needle valve 402 the area of the orifice causes the difference between the maximum fuel discharge per stroke at low and high engine speed to increase or decrease respectively. By variously adjusting the needle valve 402, the load of the spring of valve 400 and the screw 46 different characters of engine torque versus engine speed may be obtained.

The torque controlling device of Fig. 10 may obviously be introduced in the various forms of governors described above, for example to the governor 322 of Fig. 8, as shown in Fig. 11 in which lubricating oil is drawn from the supply

line 418 and discharged by a pump into the orifice 420 connected by duct 424 with the inner end of the maximum pressure regulating valve 426. The pressure drop beyond the orifice adjusted by needle valve 422 increases with the engine speed and varies with the adjustment of said needle valve. The oil pressure in the upper groove of the pilot valve, and in turn the maximum value of the pressure in line 430 leading oil to the injection pumps and the maximum value of fuel delivery per stroke are therefore smaller at high engine speed than at low engine speed.

Fig. 12 shows the application of the torque controlling device to the governor 300 of Fig. 7. Fuel is drawn from the supply line 440 by a transfer pump and discharged into duct 442 connected with the chamber at the inner end of valve 448. The latter prevents the pressure in duct 442 from rising above a predetermined value. The fuel discharged into duct 442 flows past an orifice adjusted by needle valve 444 thus determining a pressure drop that increases with the engine speed. The maximum value of the fuel pressure in lines 450 and 452, between which the injection pumps are connected in parallel, and in turn the maximum value of the fuel discharge per stroke of the latter are therefore smaller at higher engine speed than at lower engine speed, and the difference therebetween may be varied by modifying the adjustment of needle valve 444.

These embodiments of the invention have been shown merely for purpose of illustration and not as a limitation of the scope of the invention; and while in these embodiments the pilot valve is actuated in response to changes of speed of a prime mover, it will be appreciated that the invention may also be used in connection with regulators actuated in response to one or more operative conditions other than speed, and with automatically controlled servomotors in which it is desired to avoid the danger of hunting. It is therefore 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 other changes, substitutions, additions and omissions may be made in the construction, arrangement and manner of operation of the parts within the limits or scope of the invention as defined in the following claims.

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

Certain features disclosed herein are claimed in my copending patent application Serial No. 533,417, filed April 29, 1944.

What I claim is:

1. In a control system for prime mover having fluid containing means and pressure actuated control means operatively connected with first resilient means and adapted to be linked to said prime mover for increasing or decreasing the supply of actuating substance thereto in response to an increase or decrease respectively of the fluid pressure in said containing means, a slidable piston valve for controlling the pressure of said fluid therein, second resilient means for applying a load to said valve tending to move the latter in a direction to increase the pressure of said fluid, means for applying to said valve a load directed against said resilient load and substantially pro-

portional to the square of the speed of said prime mover, a pressure chamber at one end of said valve for applying a pressure load thereto, and fluid flow communication means between said chamber and said fluid containing means for varying said pressure load applied to said valve upon changes of supply of said actuating substance to correct the speed of said prime mover for load changes, said fluid flow communication means connected with said chamber including flow restricting means for dampening the displacements of said valve.

2. In a fuel control system for internal combustion engine having fluid containing means and a fluid pressure actuated resiliently loaded control means for increasing or decreasing the engine fuel supply upon an increase or decrease respectively of pressure in said containing means, a slidable piston valve for controlling the pressure of said fluid therein, means for applying to said valve an adjustable resilient load tending to move the latter in a direction to increase the pressure of said fluid, engine driven flyweights for applying to said valve a centrifugal force directed against said resilient load, a chamber at one end of and defined in part by said valve, reservoir means vented to the atmospheric pressure, fluid duct means including flow restricting orifices between said fluid containing means, said chamber and said reservoir means for keeping the effective pressure in said chamber substantially proportional to the effective pressure in said containing means, the ratio of proportionality therebetween being dependent on the relative effective areas of said orifices, the pressure in said chamber exerting on said valve a fluid pressure load directed against said resilient load whereby the adjustment of said valve is dependent on said centrifugal force acting against said resilient load minus said fluid pressure load, and upon an increase or decrease of engine fuel supply the equilibrium of said valve is resumed for a decreased or increased value respectively of engine speed, and means for adjusting the effective area of one at least of said orifices to control said ratio of proportionality and thereby the difference of engine speed between no load and full engine load.

3. In a fuel control system for internal combustion engine including fluid containing means and pressure actuated control means for increasing or decreasing the engine fuel supply upon an increase or decrease respectively of the pressure in said containing means, and an engine driven pump for delivering a flow of fluid substantially proportional to the engine speed, conduit means for establishing a fluid flow communication between said pump and said control means, orifice means for restricting said flow to create a pressure differential substantially proportional to the square of the engine speed, an axially slidable plunger valve for controlling the pressure in said containing means, means for applying a resilient load to said valve tending to move the latter in a direction to increase the pressure in said containing means, pressure chambers at the ends of said valve in fluid flow communication with the upstream and downstream sides of said orifice means for applying to said valve a differential pressure load directed against said resilient load, duct means having therein fluid at substantially constant pressure, and fluid flow communication means including flow restricting means between one of said chambers and said duct means whereby an increase or decrease of pressure in said containing means causes a corresponding

variation of pressure in said last mentioned chamber which increases or decreases respectively said differential pressure load applied to said valve, thereby causing the engine speed to be higher at no load than at full engine load.

4. In a fuel control system for internal combustion engine including pressure actuated control means for varying the engine fuel supply as a predetermined function of the pressure of a liquid actuating said control means, and engine driven means for delivering a flow of said liquid substantially proportional to the engine speed, conduit means for providing a hydraulic connection between said engine driven means and said control means, orifice means for restricting said flow to produce a pressure drop substantially proportional to the square of the engine speed, a slidable plunger valve for controlling the pressure of said liquid actuating said control means, means for applying a resilient load to said valve tending to move the latter in a direction to increase the engine fuel supply, duct means containing liquid at substantially constant pressure, pressure chambers at the ends of said valve communicating with opposite sides of said orifice means for applying a differential pressure load to said valve dependent on the engine speed and directed against said resilient load, one of said chambers also communicating with said duct means whereby said differential pressure load is also dependent on the pressure of said liquid actuating said control means and causes the engine speed to be higher for no engine load than for full engine load, and means for adjusting said orifice means to control the engine speed.

5. In a fuel control system for an internal combustion engine including pressure actuated control means for varying the engine fuel supply in response to changes of pressure of a liquid actuating said control means and engine driven pump means for delivering a flow of said liquid substantially proportional to the engine speed, conduit means for providing a hydraulic connection between said pump means and said control means, orifice means for restricting said flow to produce a pressure drop substantially proportional to the square of the engine speed, a valve for controlling the pressure of said liquid actuating said control means, means for applying a resilient load to said valve tending to move the latter in a direction to increase the engine fuel supply, duct means containing liquid at substantially constant pressure, pressure chambers in communication with opposite sides of said orifice means for applying to said valve a differential pressure load dependent on the engine speed and directed against said resilient load, one of said chambers also communicating with said duct means whereby the pressure in said last mentioned chamber and in turn said differential pressure load is also dependent on the pressure of said liquid actuating said control means and thereby causes the engine speed to be higher for no engine load than for full engine load, and means for adjusting said resilient load to control the engine speed.

6. In a fuel control system for internal combustion engine including hydraulically actuated control means for varying the engine fuel supply in response to changes of pressure of a liquid actuating said control means, and engine driven pump means for delivering a flow of said liquid substantially proportional to the engine speed, conduit means for providing a communication

between said pump means and said control means, orifice means in said flow for producing a pressure drop variable with the engine speed, a valve for controlling the pressure of said liquid actuating said control means, means for applying a resilient load to said valve tending to move the latter in a direction to increase the engine fuel supply, duct means containing liquid at substantially constant pressure, and means, including first and second pressure chambers, said first chamber communicating with one side of said orifice means, and said second chamber having connections including flow restricting means with said duct and with the opposite side of said orifice means, for applying to said valve a differential pressure load directed against said resilient load and dependent on the engine speed and on the pressure of said liquid actuating said control means to cause the engine speed to be higher for no load than for full engine load, one at least of said flow restricting means being adjustable to vary the difference of engine speed between no load and full engine load.

7. In a fuel control system for internal combustion engine having hydraulically operated control means for increasing or decreasing the engine fuel supply in response to an increase or decrease respectively of the pressure of a liquid column connected with said control means, a pressure regulating unit therefor including an engine driven pump having a rotating shaft, orifice means for restricting the discharge flow of said pump, a slidable plunger valve coaxial with said shaft for controlling the pressure of said liquid column and in turn the engine fuel supply, resilient means connected with said shaft and with said valve for rotating the latter and applying thereto a resilient load tending to move said valve in a direction to increase the engine fuel supply, duct means containing liquid at substantially constant pressure, pressure chambers at the ends of said valve, and connections including at least two flow restricting orifices between said chambers, the upstream and downstream sides of said first mentioned orifice means and said duct means for applying to said valve a differential pressure load directed against said resilient load and increasing or decreasing with an increase or decrease respectively of the engine speed and fuel supply.

8. In a control system for internal combustion engine having fluid pressure actuated control means for increasing or decreasing the engine fuel supply upon an increase or decrease respectively of the pressure of a fluid actuating said control means, an engine driven pump, and conduit means for connecting said pump with said control means, orifice means in said conduit means between said pump and said control means for producing a fluid pressure drop increasing with the engine speed, resiliently loaded valve means for keeping the pressure upstream of said orifice means substantially constant whereby the pressure downstream of the latter actuating said control means decreases with an increase of engine speed thereby causing the engine fuel supply and the engine torque to be higher at low speed than at high engine speed, and means for varying the resilient load of said valve means to control the engine load.

9. In a control system for internal combustion engine having fluid pressure actuated control means for increasing or decreasing the engine fuel supply in response to an increase or decrease respectively of the pressure of a fluid actuating

said control means, an engine driven pump, and conduit means for connecting said pump with said control means, orifice means in said conduit means between said pump and said control means for restricting the flow of fluid supplied by said pump to produce a drop of pressure increasing with the engine speed, pressure regulating means for limiting the pressure upstream of said orifice means to a substantially constant value whereby the maximum value of the pressure that said fluid downstream of said orifice means actuating said control means may attain decreases with an increase of engine speed thereby causing the maximum engine fuel supply and the maximum engine torque to be higher at low speed than at high engine speed, and valve means connected with resilient means and actuated in response to the engine speed for regulating the pressure of said liquid actuating said control means to govern the engine speed.

10. In a control system for internal combustion engine having fluid pressure actuated control means for increasing or decreasing the engine fuel supply in response to an increase or decrease respectively of the pressure of a fluid actuating said control means, an engine driven pump, and conduit means for connecting said pump with said control means, orifice means in said conduit means between said pump and said control means for restricting the flow of fluid supplied by said pump to produce a drop of pressure increasing with the engine speed, resiliently loaded pressure regulating valve means actuated in response to the pressure upstream of said orifice means for decreasing the maximum value of the pressure which said fluid downstream of said orifice means actuating said control means may attain with an increase of engine speed thereby causing the maximum engine fuel supply and the maximum engine torque to be higher at low speed than at high engine speed.

11. In a control system for internal combustion engine having control means for increasing or decreasing the engine fuel supply in response to an increase or decrease respectively of the pressure of a fluid actuating said control means, an engine driven pump, and conduit means for connecting said pump with said control means, positive stop means cooperating with said control means for determining the absolute maximum engine fuel supply, orifice means in said conduit means for restricting the flow of fluid delivered by said pump to produce a drop of pressure which increases with the engine speed, resiliently loaded pressure regulating valve means actuated in response to the pressure upstream of said orifice means for decreasing the maximum value of the pressure downstream of said orifice means actuating said control means with an increase of engine speed whereby the engine torque is higher at low than at high engine speed and the difference therebetween is dependent on the size of said orifice means, and resiliently loaded means adjustable in dependence of the engine speed and pressure of said liquid actuating said control means for varying the latter pressure to govern the engine speed.

12. A governor including a hydraulic servomotor, a plunger valve controlling said servomotor, a fluid pump having a rotatable but not axially slidable shaft coaxial with said valve, a flow restricting orifice on the discharge side of said pump to determine a fluid pressure drop substantially proportional to the square of the pump speed, fluid pressure chambers at the ends of said valve, one

of said chambers being connected with the upstream side of said orifice and the other with the downstream side thereof, and a spring connected by means of thrust and torque transmitting attachments with said shaft and valve to rotate the latter and resiliently oppose axial motion thereof.

13. A governor as set forth in claim 12 further including a fluid means having therein flow restricting means between said servomotor and one of said chambers whereby the fluid pressure in said chamber varies with changes of adjustment of said servomotor.

14. A regulating device including a servomotor, a slidable member for controlling said servomotor, means for axially moving said member, continuously turning means coaxial with said member, a coil spring having its ends attached by means of thrust and torque transmitting connections to said turning means and to said member, and means for preventing axial motion of the spring end connected with said turning means, whereby said spring maintains said member in continuous rotation and exerts an axial load thereon dependent upon the axial adjustment of said member.

15. A governor having a hydraulic servomotor, a plunger valve controlling said servomotor, a rotatable shaft coaxial with said valve and carrying flyweights for shifting said valve, a member coaxial with said shaft and connected therewith so as to be slidable but not rotatable relatively to said shaft, a coil spring for opposing outward motion of said flyweights, said spring having its ends connected by means of thrust and torque transmitting attachments with said shaft and with said valve respectively, whereby said spring causes rotation of said valve, and control means for altering the axial adjustment of said member to vary the axial resilient load exerted by the spring on said valve.

16. A hydraulic governor having a slidable pilot valve, flyweights for shifting said valve, a continuously rotating member coaxial with said valve, means for controlling the axial adjustment of said member independently of the axial motion of said valve, and a coil spring coaxial with said valve for opposing outward motion of said flyweights, said spring having its ends connected by means of thrust and torque transmitting attachments with said member and valve, respectively.

17. A hydraulic regulating device including a pilot valve element, a continuously rotating member coaxial with said element, a coil spring having its ends connected by means of thrust and torque transmitting attachments with said member and element respectively for causing said element to rotate continuously and exerting thereon an axial load which varies with changes of axial adjustment of said element, and variable pressure fluid containing means defined in part by said element for exerting an operative variable axial pressure load on said element.

18. A fluid pressure actuated regulating device including a fluid pump having a rotatable shaft, a slidable valve coaxial with said shaft, variable pressure fluid containing means defined in part by said valve for exerting an operative axial load thereon, and spring means connected by way of thrust and torque transmitting attachments with said shaft and with said valve for causing rotation of said valve and resiliently opposing axial motion thereof.

19. A regulating device including a fluid pressure actuated servomotor, a slide valve controlling said servomotor, a fluid flow restricting orifice, two

fluid chambers at the opposite ends of said valve connected with the upstream and downstream sides of said orifice respectively, a fluid flow connection between said servomotor and one of said chambers to cause pressure variations in the latter with changes of adjustment of said servomotor, a continuously rotating member coaxial with

said valve but not slidable therewith, and a spring connected by means of thrust and torque transmitting attachments with said member and valve to rotate the latter and resiliently oppose the axial motion thereof.

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