

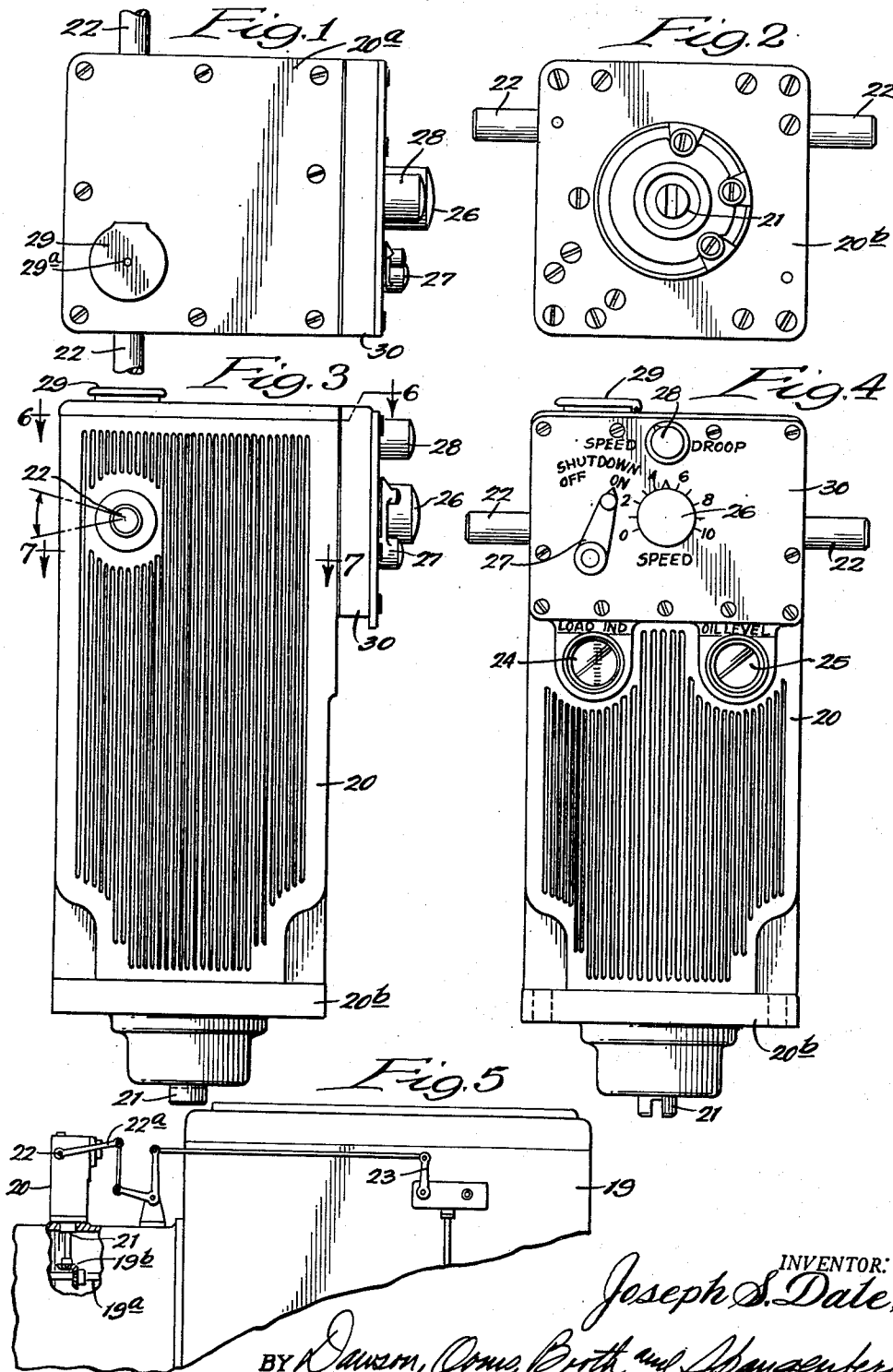
Jan. 26, 1954

J. S. DALE
HYDRAULIC SPEED GOVERNOR

2,667,345

Filed Jan. 17, 1950

6 Sheets-Sheet 1



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2,667,345

HYDRAULIC SPEED GOVERNOR

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Fig. 6

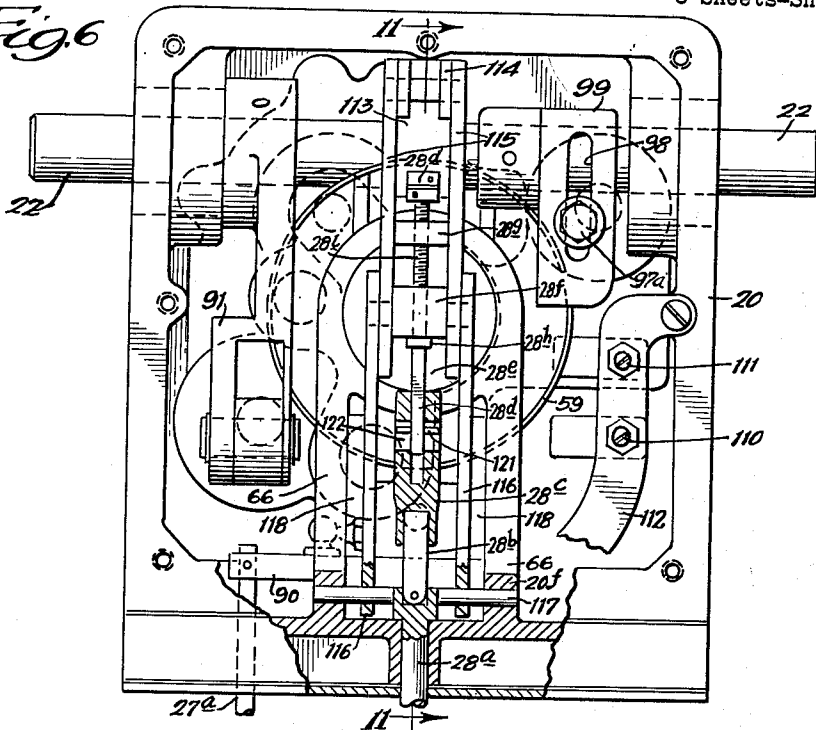
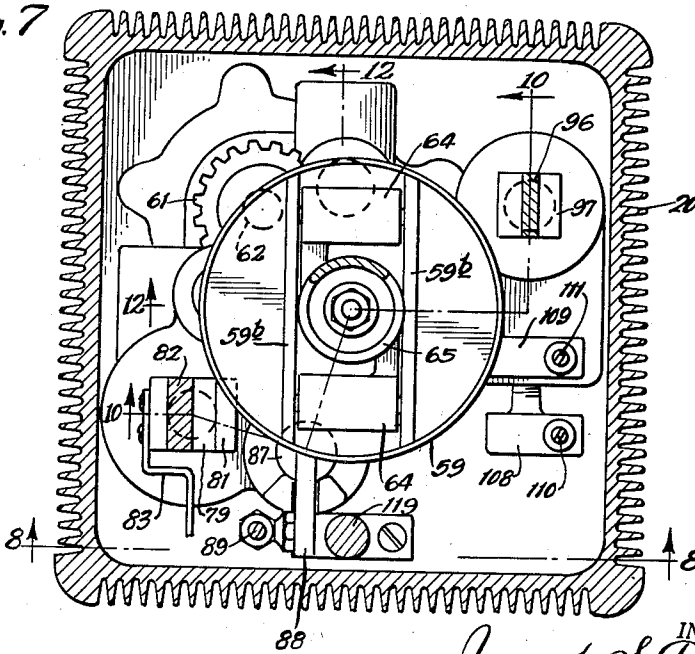


Fig. 7



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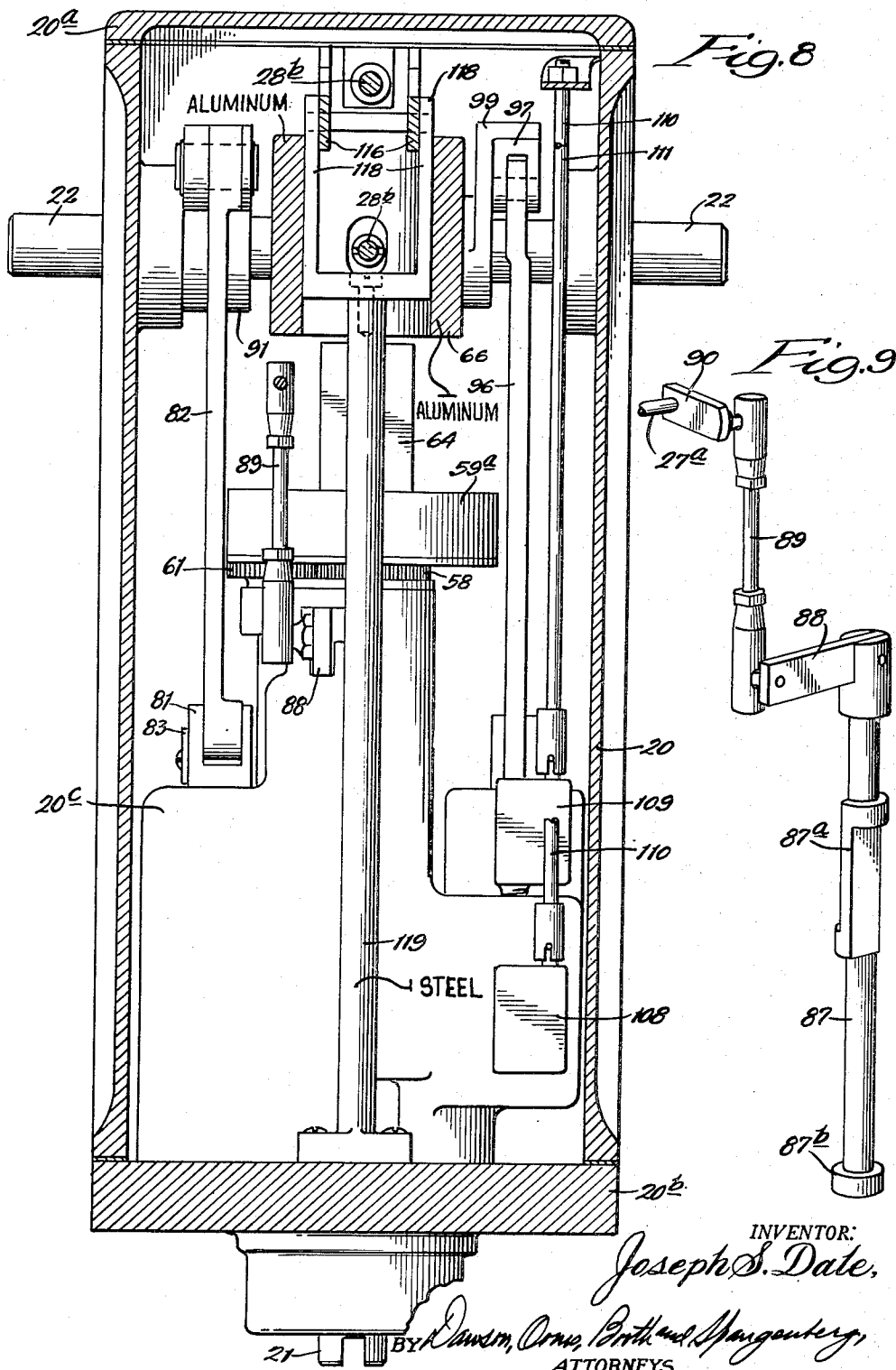
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HYDRAULIC SPEED GOVERNOR

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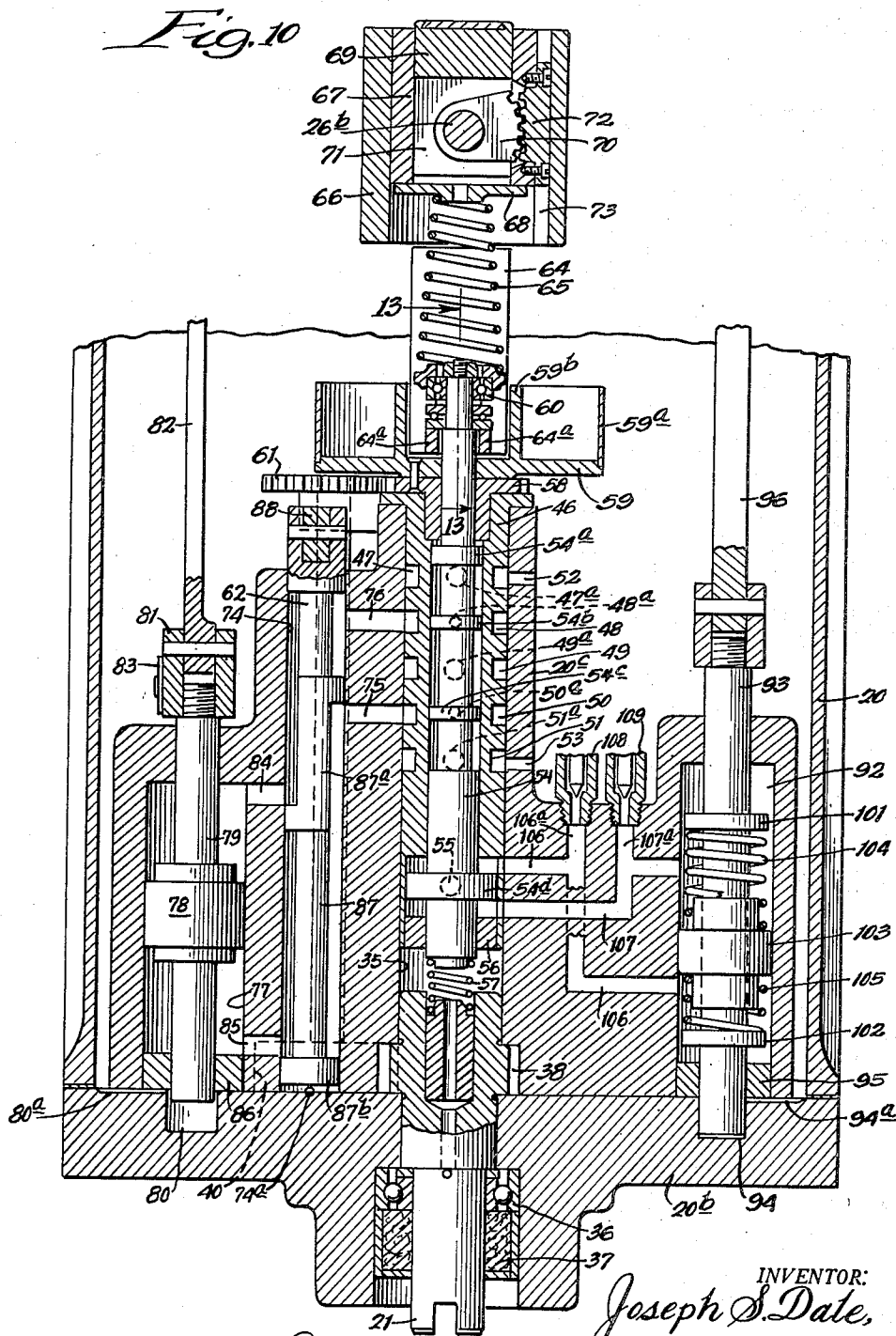
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HYDRAULIC SPEED GOVERNOR

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Jan. 26, 1954

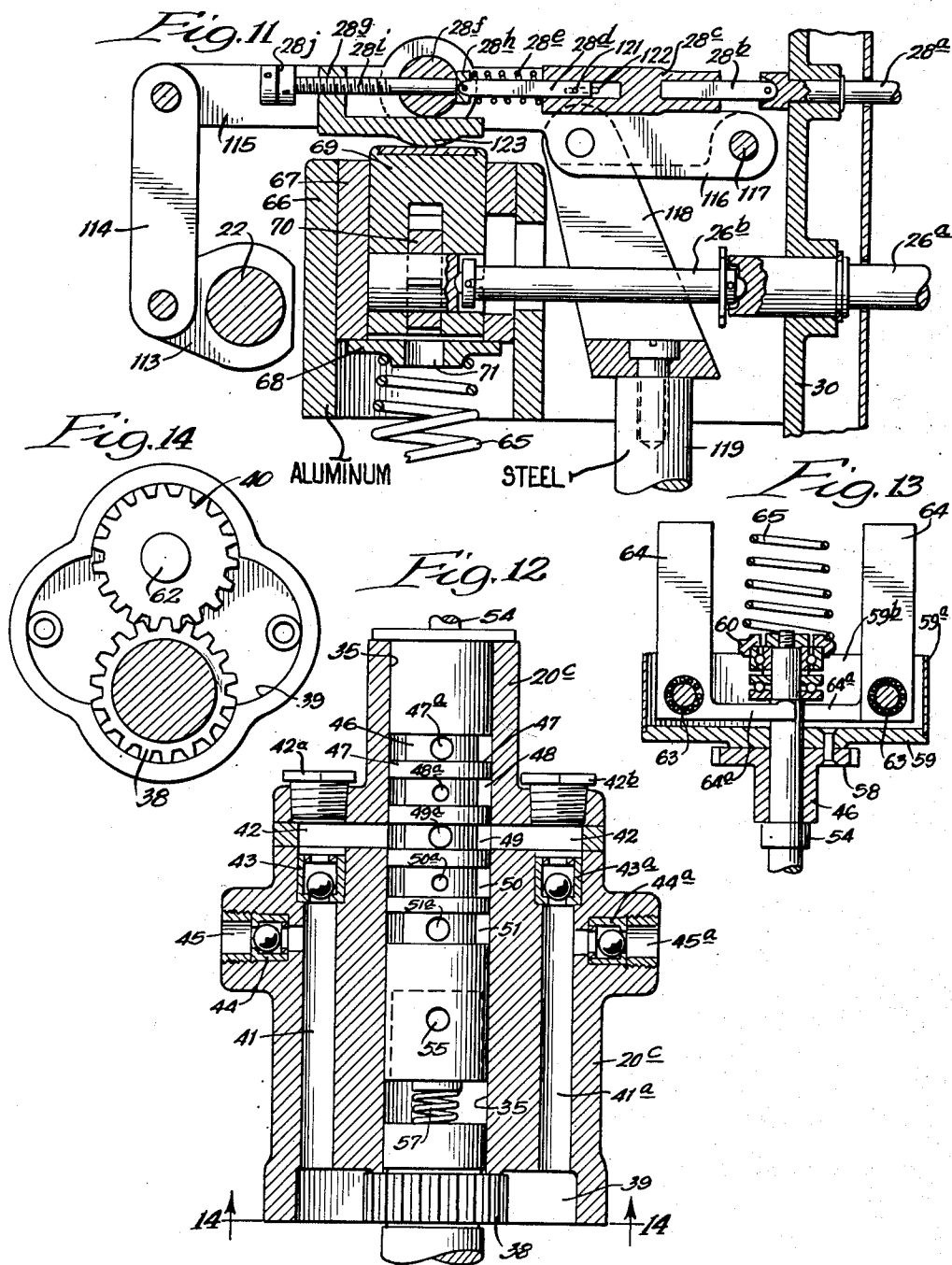
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HYDRAULIC SPEED GOVERNOR

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6 Sheets-Sheet 5



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UNITED STATES PATENT OFFICE

2,667,345

HYDRAULIC SPEED GOVERNOR

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Application January 17, 1950, Serial No. 139,076

20 Claims. (Cl. 264—3)

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This invention relates to hydraulic speed governors; in particular, it concerns novel hydraulically actuated apparatus for controlling the speed of an internal-combustion engine or other engine whose speed and torque are subject to continuous control by adjustment of a fuel-control member.

Speed governor apparatus may be employed for various purposes, but the commonest function of such equipment is to hold an engine at constant speed notwithstanding large variations in load. While my invention herein disclosed, and the novel structural features embodied therein, have broader application than merely in maintaining an engine speed constant, their principal application is in that field.

All constant-speed governors, whether hydraulic, mechanical, or electrical, operate broadly on the principle of negative feedback; that is, in all such devices a variation of speed is caused to produce in the fuel-control apparatus a change tending to restore the speed to its former value. Each change of state of the controlled engine thus sets into motion a train of events which tends to overcome or cancel such change of state.

While the broad governor principle defined in the preceding paragraph has been well known for many years, its practical employment to achieve truly constant speed notwithstanding substantial load variations has led to endless difficulties and only limited success.

The greatest practical difficulty which has faced designers of constant-speed governor apparatus has been the elimination of "hunting." This effect arises when the counter-action taken in response to a change of state of the engine is so great as to cause the engine speed to "over-shoot" the desired speed. In other words, hunting will occur when, in response to an increase in load, the governor increases the engine fuel supply to such an extent that the speed increases, notwithstanding the added load, to a value beyond the desired value. This over-shoot will in turn react upon the governor and cause the fuel supply to be reduced to slow down the engine. Thereupon the fuel reduction may cause another over-shoot in the direction of slower-than-desired speed. If the engine is hunting on both sides of the desired speed, several cycles of over-shoots may occur before the engine settles down to a constant speed again after its normal operation has been disturbed by a change in load.

Means have in the past been devised for compensating governors to avoid over-shoots and thus eliminate hunting. In the broad sense, these compensating devices have all permitted corrective governor actions to be initiated rapidly but, near the end of a given corrective action, have delayed and opposed such action. If the compensation be correctly adjusted in quantity and timing, a compensated governor should be

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able to restore engine speed to normal quite rapidly after a change in load, without either over-shoots or hunting.

Prior-art speed governors for internal-combustion engines have, however, had a substantial operating defect notwithstanding the use of compensation. This defect has resulted from the fact that in internal-combustion engines engine speed is not a linear function of either fuel supply or load. As a result, when a governor-controlled engine is "hunting," the speed over-shoots above and below the desired speed are almost never of equal magnitude. The recovery time on acceleration will be two or three times the recovery time on deceleration, or vice versa. As a result, an engine governor provided with prior-art compensating means could never be adjusted to an optimum setting in which the governor gave rapid restoration of normal speed without hunting on both acceleration or deceleration. Complete elimination of hunting in prior-art devices has in general been accompanied by sluggish response in at least on recovery direction. Since rapid restoration of normal speed is usually important, practical engine governors, therefore, have normally been compensated on a compromise basis. In other words, the compensation has usually been set to that value which seemed to give most nearly stable operation consistent with rapid response to load changes.

One of the principal objects of this invention is to provide a speed governor for internal-combustion engines wherein the compensation can be separately adjusted for accelerative and decelerative effects, with no mutual interaction between the adjustments. The attainment of this object represents a very important advance in the art, since it makes possible the construction of an engine governor which will maintain an engine at constant speed within very wide load limits without any over-shoots or hunting effects whatever.

Another object of my present invention is to provide a hydraulic speed governor for a prime mover which works entirely on super-atmospheric pressures and wherein, therefore, there is no chance whatever of developing a vacuum in the apparatus.

A further object of my present invention is to provide a hydraulic speed governor for internal-combustion engines wherein a novel means is provided for closing down or shutting off the engine without necessity for adjustment of or tampering with the delicately balanced pilot valve which constitutes the sensitive control member. In my structure, the pilot valve, on a shut-down operation, performs its usual regulating function, and the course of flow of the hydraulic fluid within the governor is altered so as to shut off completely the fuel supply rather than to hold the engine at constant speed.

A still further object of my invention is to pro-

vide a hydraulic governor wherein a simplified bimetallic element built into the governor frame structure affords automatic compensation for loss of force of the speeder spring with increased temperature and thus eliminates or greatly reduces the falling off of speed ordinarily occurring as the speeder spring warms up following a cold start.

Still another object of my invention is to provide a means for achieving maximum pilot valve sensitivity which involves placing a non-positive coupling between a rapidly rotating shaft and the body member of the pilot valve. This coupling causes the pilot valve member to rotate and thus maintain at a minimum the frictional resistance offered by the pilot valve to axial movement.

Still another object of my invention is to provide, in a hydraulic governor for internal-combustion engines, a novel and simplified variable "speed droop" mechanism which is dependable, finely variable and inexpensive to construct.

Still another object of my invention is to provide a fly-weight governor for internal-combustion engines which can regulate the speed of the engine down to any desired low value no matter how slow. This desirable object is achieved in my invention by novel construction of the pilot valve and speeder spring effective to maintain contact and tension between the speeder spring and pilot valve even at zero speed.

Another object of my invention is to provide, in a hydraulic governor, an automatically actuated hydraulic pump which is energized by the controlled engine and in which automatic control means are provided for maintaining the maximum pressure at the optimum value for control of the particular engine with which the governor is being used.

Another object of my invention is to provide, in a hydraulic governor, means for adjusting the speed of response of the servo mechanism to shifts in pilot valve position to any value within wide limits, thus permitting the governor characteristics to be "tailored" to suit the characteristics of the controlled engine.

Other objects and advantages of my invention will appear as the specification proceeds.

I have shown a typical embodiment of my invention in the accompanying drawings. Figure 1 is a plan view of my illustrative governor, taken with the casing in place as for normal use. Fig. 2 is a bottom view of the Fig. 1 structure. Fig. 3 is a view in side elevation of the illustrative speed governor, the view being taken from the left as the observer faces the governor. Fig. 4 is a front elevation view of the illustrative speed governor, showing in detail the control and indicator panel. Fig. 5 is a diagrammatic view illustrating the manner in which my speed governor might be connected to a large internal-combustion engine for maintaining the speed thereof at a constant value despite variations in load. Fig. 6 is a top view of my governor apparatus as it appears with the top casing plate removed; alternatively, it might be described as a sectional view taken along the line 6-6 of Fig. 3. Fig. 7 is a plan sectional view of my speed governor taken along the line 7-7 of Fig. 3. Fig. 8 is a sectional side elevation view showing my illustrative governor as it would appear from the side with one side wall of the casing broken away. Alternatively, Fig. 8 might be described as a sectional view taken along the line 8-8 of Fig. 7. Fig. 9 is a perspective view of a mechanical linkage structure which is employed in

the novel shut-down control which is one of the features of my invention. Fig. 10 is a semi-diagrammatic view of my governor which is substantially a sectional view in the vertical plane along the irregular line 10-10 of Fig. 7. To permit full showing in a single figure, however, of certain of the passageways and valves, the relative positions of such components are slightly altered in Fig. 10 as compared with the other figures which show their actual position in the representative embodiment of my invention. Fig. 11 is a vertical sectional view of certain apparatus located in the upper portion of the casing, the section being taken along the line 11-11 in Fig. 6. Fig. 12 is a sectional view taken through the central casting which carries the bore for the pilot valve; the plane of the section is indicated by the line 12-12 in Fig. 7. Fig. 13 is a sectional detail view of the fly-weight structure shown in Fig. 10, the section of Fig. 13 being in a plane at right angles to that of Fig. 10, as indicated by the line 13-13 in Fig. 10. Fig. 14 is a view, partly in section, along the line 14-14 of Fig. 12, showing the structural details of the gears which function as an oil pump and are also part of the gear train which drives the fly-weight mechanism. Fig. 15 is a sectional view showing a modified form of governor in which the peak hydraulic pressure is controlled as a function of engine speed; and Figs. 16 and 17 are perspective and sectional views respectively of another modification of my governor in which fine control is afforded over the rate of flow of hydraulic fluid from the pilot valve to the servo cylinder.

As particularly shown in Figs. 1-4, the illustrated governor is enclosed within a cast housing 20, preferably formed of aluminum. The casing 20 is provided with a top plate 20a and a bottom plate 20b, respectively secured to the main casing 20 with screws and effective to provide a fully enclosed, liquid-tight housing for the governor apparatus.

The interior of the housing, in addition to carrying the operative elements of the governor, functions as a sump or reservoir, at atmospheric pressure, for the light oil or other liquid employed as hydraulic fluid. If desired, a suitable cap aperture 29 may be provided in the top plate 20a as a convenient means of replenishing the hydraulic fluid when necessary. To insure that the gas pressure within the casing is at atmospheric value, an air vent 29a may be provided in the cap of oil inlet 29.

The oil level within the casing, in normal operation of the governor, should be slightly more than half way up the side of the casing. In the particular embodiment illustrated in the drawings, I have shown a transparent glass or plastic window 25 in the front of the casing 20 to provide a convenient means of observing the oil level.

Connection between the shaft of the controlled engine and the governor may be made through drive shaft 21, vertically disposed and projecting downward from the bottom of the casing 20. The output or control shaft 22 is horizontally oriented and mounted in suitable bearings in the sides of casing 20; for convenience in installation with a variety of engines, shaft 22 is extended beyond the casing at both ends. Shaft 22 is secured against axial movement, its rotation being governed by the control action of the governor. In use, the shaft 22 may be connected by suitable linkage to the fuel control of the engine.

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Fig. 5 shows, diagrammatically, a typical installation of my governor for the control of an internal-combustion engine. Engine 19 is provided with an output shaft 19a, coupled by bevel gears 19b to shaft 21 of my governor which, as shown, is mounted at a convenient location adjacent the engine. Control shaft 22 is provided with a lever arm 22a keyed thereto, and arm 22a is coupled by conventional lever linkage to fuel control 23 of engine 19. As connected in Fig. 5, my governor will control fuel control 23 in response to the rotation of shaft 19a so as to maintain the speed of the engine at that constant level set by manual adjustment of my governor.

As may be best seen from Figs. 3 and 4, the upper portion of the face of casing 20 is provided with an instrument panel 30 which carries the manual controls with which my governor is adjusted and regulated. A manually operable knob 26 serves as a speed control, affording to the operator free choice of the speed at which the governor will hold the engine being controlled.

A shut-down control 27 is the manual control which actuates a novel mechanism which is one of the special features of my invention. As will be more fully described hereinafter, movement of the shut-down control to the "off" position causes the governor to stop the engine promptly by cutting off the fuel supply without in any way interfering with the normal operation of the sensitive pilot valve.

Manual control 28, on the instrument panel immediately above the speed control, is the "speed droop" control member. As is familiar to those versed in the engine-governor art, speed droop is what might be called intentional under-regulation. That is, when speed droop has been introduced into a governor, it will, in response to load increases, supply additional fuel but not quite enough to compensate for the added load. Speed droop is used when a number of engines are being operated in parallel, as in electric generating stations, etc. In such stations, it is usually desired that a single engine take all variations in load, while the other engines are expected to operate under substantially constant load at all times. This effect can be achieved by introducing speed droop into the regulation of all of the engines except the one which is to take the load variations. The apparatus herein disclosed for providing speed droop is another of the distinctive features of my invention and it will be fully described hereinafter.

Referring particularly now to Figs. 10, 12, and 14, it may be seen that bottom plate 20b carries on its upper surface a central casting 20c, which serves to support a considerable portion of the governor apparatus and which contains cylindrical bores for the pilot valve, the servo piston, and other important parts of the apparatus.

A central bore 35 in casting 20c extends vertically through the entire casting. Casting 20c should be made of the same metal as is employed for the casing proper; aluminum or other light metal is preferable for this purpose.

Shaft 21 is supported in the lower end of bottom plate 20b by anti-friction bearings 36 and stuffing box 37. The upper end of shaft 21 is keyed to a gear member 38 which is carried in a recess 39 machined into the lower end of casting 20c. The respective hub portions of gear member 38 are supported in a vertical bore in bottom plate 20b and the lower end of bore 35 in casting 20c.

Gear 38 is positioned to mesh tightly with a second gear 40 of identical size. Gears 38 and

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40, together with recess 39, form the principal operative parts of a gear pump which performs the function of supplying hydraulic fluid under pressure to the pilot valve. This gear pump is shown in detail in Fig. 12. As may be seen from that figure, casting 20c contains a pair of vertical bores 41 and 41a which communicate at their lower ends with recess 39. A transverse bore 42 connects the bores 41 and 41a at their upper ends. A ball-check valve 43 is mounted in vertical bore 41 immediately below transverse bore 42, while a similar ball-check valve 43a is mounted in vertical bore 41a below transverse bore 42. A ball-check valve 44, mounted in a branch bore 45, connects bore 41 to the outside of casting 20c; that is, to the sump or oil reservoir. A similar ball-check valve 44a, mounted in a branch bore 45a, provides a connection to sump for vertical bore 41a. Transverse bore 42 passes through the center of the main vertical bore 35 in casting 20c. A steel sleeve 46, shrink-fitted into bore 35, is provided with five annular recesses, each of which is provided with a port connecting the annular recess to the interior of the sleeve. On the drawings the annular recesses are denoted 47-51 respectively, taken in order from the top of sleeve 46 to the bottom, while the ports associated with the respective recesses are denoted 47a, 48a, etc. Ports 47a, 49a, and 51a are relatively large in size and, as will become clear hereinafter with the description of the pilot valve, their dimensions are not critical. Ports 48a and 50a, as shall hereinafter be stated in detail, are control ports, and their dimensions relative to lands 54b and 54c are of critical importance. A transverse bore 52 through the wall of casting 20c connects annular recess 47 with sump. A similar bore 53 connects annular recess 51 with sump. The orientation of sleeve 46 within bore 35 places both branches of transverse bore 42 in communication with annular recess 49.

As may be seen from a study of Fig. 12, rotation of gears 38 and 40 in either direction will result in the transmittal of hydraulic fluid under pressure into annular recess 49 and thence through port 49a. If gear 38 is rotating in one direction, oil will be drawn from sump through ball-check valve 44, around the periphery of recess 39 between the teeth of gears 38 and 40, up vertical bore 41a, and through ball-check valve 43a into transverse bore 42. Should the direction of rotation of gear 38 be reversed, oil is taken from sump through ball-check valve 44a, carried by the gears around the periphery of recess 39 into vertical bore 41, and thence to transverse bore 42 through ball-check valve 43. Thus liquid under pressure is supplied to the interior of sleeve 46 through port 49a any time gear 38 is rotated by shaft 21, regardless of the direction of rotation.

The interior of sleeve 46 is machined with the greatest care and accuracy, since it serves as the cylinder which co-operates with the pilot valve spool 54. This spool is provided with an upper land 54a which fits snugly the inner surface of sleeve 46 and serves to limit the upward movement of the valve member 54. The next land on valve member 54, counting downward from its upper end, is land 54b which in normal operation co-operates with port 48a. The thickness of land 54b is accurately machined to a thickness slightly less than the diameter of port 48a, the difference being of the order of a few thousandths of an inch. The next lower land on valve member 54 is land 54c which co-operates with port 50a.

Ports 48a and 50a are drilled to the same diameter, and land 54c is structurally like land 54b, their axial separation being identical to the center-to-center separation of ports 48a and 50a. Thus the lands and ports just described form a balanced by-pass system carrying the output of pump 38, 40 to sump and maintaining equal pressures in annular recesses 48 and 50 when the lands 54b and 54c are positioned symmetrically with respect to ports 48a and 50a respectively.

Below port 51a, valve member 54 has a relatively lengthy enlarged-diameter portion which fits snugly with the inner surface of sleeve 46. At its lowermost end, sleeve 46 is bored out to an enlarged diameter, and a sump port 55 extends through the wall of such enlarged diameter portion and thence through casting 20c to sump. A land 54d on valve member 54 fits snugly the inner surface of sleeve 46 at the altitude of port 55 and is machined to a thickness equal, within approximately one-thousandth of an inch, to the diameter of port 55. The enlarged-diameter portion of sleeve 46 is closed off at the bottom by a seal ring 56 which is shrink-fitted into sleeve 46. Member 56 serves as a lower bearing for valve member 54.

Valve member 54 rests on a coil spring 57 which is carried by a spring seat formed in the upper surface of gear member 38. The bottom surface of valve member 54 is modified to receive the upper end of spring 57, and the net effect of the spring is to transmit appreciable torque to valve member 54 when shaft 21 is in rotation. Spring 57 does not form a positive coupling, but the limited amount of torque which is transmitted by it to valve member 54 serves to maintain member 54 in relative rotary motion within sleeve 46, thereby maintaining a continuous oil film around valve member 54 and greatly reducing the force necessary to move valve member 54 in the axial direction. This increases the sensitivity of the governor. Spring 57 also performs the useful function of maintaining a continuous upward force on valve member 54, a very useful effect since it keeps the upper end of valve member 54 always in contact under tension with the speeder spring presently to be described, thus permitting effective regulation of the engine down to zero speed.

A gear 58 provided with a carefully machined hub member on one side is fitted into a co-operating bearing machined into the upper end of sleeve 46. Gear 58 is keyed to a coaxially disposed cup member 59 which serves as support for the fly-weight mechanism. Valve member 54 extends upward through suitable apertures in the centers of members 58 and 59, terminating in anti-friction bearing structure 60. Gear member 58 meshes with a gear 61 supported in a stationary bearing which may be carried by casting member 20c and driven by shaft 62 which extends vertically to gear 40 at the lower end of the casing. All of the gears 38, 40, 61, and 58 are identical in size, and accordingly cup member 59 rotates at the same speed and in the same direction as shaft 21. Cup member 59 has a raised rim 59a around its periphery, the function of which is primarily to keep sump oil from getting into contact with and thus affecting the operation of the centrifugal fly-weights. The normal oil level in the casing is lower than the altitude of cup member 59, but the rim 59a serves as an added precaution against the oil's

interfering with the free movement of the fly-weights.

Cup member 59 is formed with a pair of parallel cross walls 59b which carry the bearings that support the fly-weights. Anti-friction bearings 63 support fly-weights 64. Each of the fly-weights is provided with a laterally extending finger 64a (see Fig. 13) the inner end of which presses on the under side of anti-friction bearing structure 60. Thus, when the fly-weights move outward due to centrifugal force of rotation, the fingers 64a press on the under side of bearing structure 60 and thus raise valve member 54. Conversely, when fly-weights 64 slow down, they permit valve member 54 to lower its position.

The upward movement of bearing structure 60, and thus of valve member 54, is opposed by speeder spring 55. Spring 55 is a coil spring which seats at its lower end in a spring seat formed on the upper surface of bearing structure 60.

The speed at which the controlled engine is held is governed by the tension of speeder spring 55. The apparatus which accomplishes control of that tension is supported by a bracket 66 which extends outward in a horizontal plane from the front face of panel 30, terminating at its inner end in a hollow cylindrical portion. The inner surface of the cylindrical portion of bracket 66 is accurately machined for smoothness and constant curvature. Within said inner surface a sleeve 67 is slidably carried. A spring seat 68 rests upon the upper end of speeder spring 55, and sleeve 67 in turn rests upon the upper surface of spring seat 68 (see Figs. 10 and 11). A plunger 69 is slidably carried within sleeve 67; the relative position of plunger 69 and sleeve 67 is governed by the position of a rack-and-pinion structure about to be described. A shaft 26a, which carries the speed control knob 26, is journaled in the front portion of panel 30 and coupled by a flexible coupling to another shaft 26b. Shaft 26b passes through a slot in bracket 66 and a corresponding slot in sleeve 67, terminating in a pivot coupling to a sector pinion 70 which is mounted within a recess 71 in plunger 69. The teeth in sector pinion 70 mesh with the teeth in a rack 72 which is bolted to the side of sleeve 67 and slidably carried in a vertical slot 73 in bracket 66. Rack 72, by thus riding in slot 73, secures sleeve 67 against rotation in bracket 66, while allowing it full freedom to move in the axial direction. As may be seen from Figs. 10 and 11, rotation of shaft 26a accomplished by manual turning of knob 26 will shift the relative position of pinion 70 and rack 72, thus altering the relative axial positions of plunger 69 and sleeve 67. It should be borne in mind that, due to the flexible couplings at each end of shaft 26b, the plunger 69 and sleeve 67 are free to move axially within bracket 66, although their positions relative to one another are governed by the position of knob 26. The upward movement of plunger 69 is limited by cam member 28g, which will be described, with its associated structure, in a later paragraph hereof. For the present, it will suffice to state that since the upward movement of plunger 69 is limited, adjustment of knob 26 in one direction results in lowering sleeve 67 and spring seat 68, thus producing increased tension in speeder spring 55. Adjustment of knob 26 in the opposite direction reduces the axial separation of members 67 and 69, thus raising spring seat 68 and reducing the ten-

sion on speeder spring 65. Thus by appropriate adjustment of speed control knob 26 the amount of centrifugal force necessary to raise pilot valve 54 a given distance against speeder spring 65 can be accurately controlled. In this manner the speed at which the governor will maintain the controlled engine may be accurately regulated.

I shall now describe the structure by which shifts in the position of the pilot valve responsive to movement of the fly-weights results in rotation of control shaft 22.

A vertical bore 74, shown in Fig. 10, passes entirely through casting 20c, parallel to and adjacent the main central bore 35. A transverse bore 75 connects annular recess 50 in sleeve 46 with bore 74; a similar transverse bore 76 connects annular recess 48 in sleeve 46 to bore 74.

A relatively large vertical bore 77 in casting 20c forms the cylinder which carries servo piston 78. Servo piston 78 is mounted on piston rod 79, which extends both above and below piston 78. The lower end of rod 79 terminates in a recess 80 in bottom plate 20b, functioning as a stop member to limit the downward movement of servo piston 78. The upper end of rod 79 passes through a bearing drilled in casting 20c and is attached by a pivot coupling 81 to link 82. An indicating pointer 83 is bolted to coupling member 81 immediately above its junction with rod 79. Pointer 83 extends forward to a point immediately behind the transparent window 24 in the front of casing 20, shown in Fig. 4 and marked "load indicator." Window 24 may be provided with graduated markings. Since the position of the servo piston, and consequent position of the engine fuel control, are a function of the engine load, the position of pointer 83 provides an accurate visual indication of the degree of load on the engine being controlled.

A transverse bore 84 joins the upper end of cylinder 77 with bore 74, while a similar transverse bore 85 joins the lower end of cylinder 77 to the lower end of bore 74. A packing ring 86 closes the bottom portion of cylinder 77 while allowing the lowermost end of rod 79 to pass through into recess 80, already described. A vent 80a connects recess 80 to sump to avoid trapping in recess 80 any oil which might escape past packing ring 86. A similar vent 74a prevents liquid being trapped in the lowermost portion of bore 74.

A spool valve member 87 is carried in bore 74. As may be best seen in Fig. 9, valve member 87 has a principal valve portion 87a which is generally cylindrical in shape but provided with two flat cutaway portions, one of which begins intermediate the ends of the cylinder and extends to the top while the other begins on the opposite side of the cylinder near the upper end and extends to the bottom. This valve member 87a is oriented on the main spool shaft 87 in a position such, as shown in Fig. 10, that the flat portion extending to the top of member 87a provides communication between transverse bore 84 and transverse bore 76 when the valve member 87 is in its normal, fully lowered position. In that position the other cutaway portion of member 87a affords communication between transverse bore 75 and transverse bore 85. At the lower end of spool shaft 87 a cylindrical land 87b fits snugly within bore 74, thus normally blocking off transverse bore 85 from vent 74a.

At its upper end spool member 87 is rigidly coupled to an arm 88 extending outward at right

angles to member 87. At its outer end, arm 88 is joined by a universal joint to a vertical member or link 89 extending upward. At its upper end member 89 is coupled by another universal joint to a lever 90 which is keyed to a shaft 27a. Shaft 27a is journaled in panel 30 and carries knob 27 on instrument panel 30. As may be noted from Figs. 4, 9, and 10, shifting of the manual member 27 to the "off" position will raise spool valve member 87. In the "off" position, spool valve 87 cuts off communication between transverse bore 76 and bore 84, providing instead communication between transverse bore 75 and bore 84. At the same time, land 87b cuts off communication between transverse bore 85 and bore 75, instead connecting transverse bore 85 to sump vent 74a.

At its upper end connecting rod 82 is pivoted to a crank member 91 which is rigidly keyed to control shaft 22. Accordingly, shifts in the vertical position of servo piston 78 are reflected as rotational movements of shaft 22. Shaft 22, in turn, may be coupled by any suitable mechanical linkage to the fuel control of the engine being regulated.

I shall now describe the novel structure by which compensation is introduced in my speed governor to stabilize its operation and to avoid hunting.

A vertical bore 92 in casting 20c functions as compensation cylinder. It contains a central bore at the top which receives a vertical rod 93 extending entirely through bore 92 and seating in a recess 94 in bottom plate 20b. Recess 94 is provided with a sump vent 94a. A packing ring 95, shrink-fitted into the lowermost portion of bore 92, serves to hold rod 93 in alignment and to permit it free axial movement without binding. At its upper end, rod 93 is pivoted to a link 96 which extends upward to a point near the top of the casing, at which it is pivotally secured to a member 97 which is, in turn, carried in a transverse slot 98 in a crank member 99, keyed to shaft 22. By a set screw 97a, member 97 can be locked in any desired position in slot 98, within a substantial range of movement.

Rod 93 carries an upper spring seat 101 and a lower spring seat 102. These spring seats are machined to fit relatively loosely within bore 92. Floating on rod 93 is a plunger 103 which is machined to slide freely in bore 92. It is provided on both its upper and lower faces with annular flanges which serve as spring seats. A coil spring 104 is seated on the upper face of plunger 103, where it presses upward against spring seat 101. Correspondingly, a second coil spring 105 is seated around rod 93 between the lower face of plunger 103 and spring seat 102.

A transverse bore 106 connects the uppermost part of the enlarged-diameter portion of sleeve 46 with the lower portion of bore 92; a transverse bore 107 connects the lowermost portion of the enlarged-diameter part of sleeve 46 to the upper portion of bore 92. A branch bore 106a connects bore 106 to a needle valve 108 communicating with sump. Similarly, a needle valve 109 is connected by a branch bore 107a to bore 107, valve 109 exhausting also to sump.

The needle valves 108 and 109 are made manually adjustable by rod links 110 and 111 respectively which extend upward toward the top of the casing, being secured at their upper ends to a bracket 112 which extends inwardly from casing 20.

As has been previously mentioned in the de-

scription of Fig. 10, the needle valves 103 and 109 are therein shown in order to make clear their function relative to their associated structure, although, as may be seen from Fig. 7 or Fig. 8, the actual position of those valves in the illustrated embodiment of my invention is not on the section line 10-10 of Fig. 7.

I shall now describe the structure by which I achieve controllable speed droop in my governor apparatus.

As may be seen best from Figs. 6 and 11, shaft 22 has keyed to it at its midpoint a crank member 113, to which is pivoted a link 114. The upper end of link 114 is pivoted to a bifurcated lever 115 which extends forward approximately half way across the casing. At the precise midpoint of the casing, directly over the center of plunger 69, the lever member 115 is pivoted to a second bifurcated lever member 116. Lever 115 has two points of support; at its extreme forward end it is pivoted by a pivot rod 117 to a pair of brackets which extend backward from the panel 39. Intermediate its ends, lever 116 is pivoted also to a bracket 118 which is rigidly bolted to a steel rod 119. As may be best observed from Fig. 11, bifurcated lever member 115 is curved downward at the pivot point with bracket 118 so that the pivots which join member 115 to bracket 118 do not interfere with the linkage which joins shaft 28a to cam member 28g.

In the illustrative embodiment, as stated above, shaft 119 is made of steel, while the casing and other structural supports of the governor apparatus are formed of aluminum. I do not desire to limit myself to that particular combination of metals, but it is important in respect to one aspect of my invention, as will be more fully stated hereinafter, that shaft 119 be formed of a metal having a lower coefficient of expansion than the metal of which the casing and other structural parts are formed.

A pivot 28j, which joins lever members 115 and 116, has an enlarged-diameter portion which fills the space between the respective forks of lever member 115; member 28f is provided with a transverse bore, perpendicular to its axis and spaced midway between its ends. Through the aforesaid transverse bore, a bolt member 28i is carried, oriented with its head 28h nearest the front of the casing. The head 28h is pivoted to a link member 28d. The opposite end of link member 28d slides in an axial aperture in coupling member 28c, being secured therein by a pin 121. A slot 122 in coupling 28c affords a limited range of sliding movement to link member 28d. The forward end of link member 28c is provided with a keyed-in extension 28b pivoted at its forward end to shaft 28a, which in turn is journaled into a bearing in panel 30.

The rear portion of bolt 28i is threaded, and it carries, threaded thereto, a cam member 28g which is shaped generally like the letter L. The shorter leg of member 28g is the portion threaded into bolt 28i; the other leg extends forward under pivot member 28f, the dimensions of cam 28g being chosen so that its horizontal surface engages the lower surface of member 28f when bolt 28i is in a horizontal position.

The under side of cam 28g is provided with an arcuate downward extension 123 which is carefully machined to form a sector of a circular cylinder. The open end of bolt 28i is provided with a keyed-on stop member 28j which limits the outward movement of cam member 28g. As may be seen from Fig. 11, rotation of shaft 28a by

manual manipulation of knob 25 will result in forward or backward movement of cam 28g, with consequent shift in the line of contact between cam 28g and plunger 69 which, forced upward by spring 65, presses against cam 28g on its under side. As cam 28g moves across the plunger 69 the relative position of arcuate extension 123 and pivot 28j is altered.

OPERATION

It will be assumed for purposes of this discussion of "operation" that my invention has been suitably mounted adjacent an engine, that shaft 21 has been mechanically coupled to the engine drive shaft, that control shaft 22 has been coupled to the engine fuel control, and that the governor casing has been filled with oil or other suitable hydraulic fluid approximately up to the level of the top of central casting 20c, all passageways, cylinders, and other open spaces therebelow being filled also.

Initially the shut-down control 27 will be placed in the "on" position and the engine started. Thereupon the shaft 21 will be set into rotation and the fly-weight structure mounted on gear 58 will rotate in exact synchronism with shaft 21. (Meanwhile the pilot valve member 54 will also commence to rotate, due to torque supplied by spring 57.)

As the fly-weights 54 move outward, pilot valve member 54 is raised until it occupies its normal operating position wherein lands 54b and 54c are centered with respect to ports 48a and 50a, respectively. When that condition is reached, the engine will stabilize its speed. Any subsequent departure from said stable speed will cause fly-weights 54 to shift either inward or outward. Should the engine speed up as a result of a lightened load, fly-weights 54 will move outward, causing pilot valve 54 to rise. This will result in increased flow of hydraulic fluid out of port 48a into transverse bore 76. (It should be borne in mind that as soon as rotation of shaft 21 commences, the gear pump formed by gears 35 and 40 creates and maintains a substantial fluid pressure in the chamber within sleeve 46 between lands 54b and 54c.) With increased flow of hydraulic fluid out of port 48a into bore 76, the increased fluid pressure is transmitted through bores 76 and 84 into the upper end of chamber 77. As oil flows into chamber 77, servo piston 78 is forced downward, the oil below the piston being free to escape to sump through bores 85, 74, and 75 and ports 50a and 50a. Lowering of servo piston 78 causes rotation of shaft 22 and consequent reduction of the quantity of fuel supplied to the engine. This of course slows down the engine and restores the speed to normal.

If, as a result of heavier load, the engine speed tends to decrease, an exactly similar series of events, initiated when pilot valve member 54 drops, leads to upward movement of servo piston 78 and consequent increase in the engine fuel supply.

Stabilization of the controlling action is accomplished by means of the compensating apparatus heretofore described in detail. Its operation is briefly as follows: When, in response to an added load, pilot valve 54 drops, the lowering of land 54d at once creates a super-atmospheric pressure in bore 107 and the upper end of chamber 92. This reduces the tension on spring 104, since the tendency of the added pressure is to drive plunger 103 downward and compress spring 105. Liquid under plunger 103 is free to escape, since lower-

ing of land 54d results in partial uncovering of port 55 which leads to sump. When, in response to the lowering of the pilot valve 54, the servo piston 78 rises, it causes link 96 to rise also, thereby increasing the compression on spring 105 and thus further decreasing the compression on spring 104. This increases the fluid pressure within bore 107 and the chamber below land 54d. Meanwhile, however, hydraulic fluid has been at liberty to leak off slowly through needle valve 109, so that the hydraulic pressure below land 54d which tends to oppose the downward movement of valve member 54 diminishes gradually with time as liquid escapes through valve 109. The magnitude of movement of link 96 resulting from rotation of shaft 22 may be controlled within wide limits by adjusting the member 97 within slot 98, since the effective length of the crank member 99 may be thereby varied.

By careful adjustment of the position of member 97 in slot 98 and the bleed rate of valve 109, the magnitude and duration of the counter-forces set up by the compensating mechanism can be made to eliminate hunting entirely and achieve a clean, rapid return to normal speed following deceleration resulting from added load. The object of the adjustment should be to avoid overshoot and obtain precisely that degree of damping which accomplishes stabilization without overshoot. That adjustment will provide the most rapid return to normal speed following the imposition of added load on the engine.

Should the engine rise in speed as a result of lightened load, pilot valve 54 will rise, port 55 will be opened to bore 107, the hydraulic pressure in bore 106 will become higher than normal, and plunger 103 will be forced upward, thus compressing spring 104. The shift in the position of shaft 22 following the lowering of servo piston 78 will cause link 96 to be lowered, thus increasing still further the compression of spring 104 and the hydraulic pressure in bore 106. As before, this super-normal pressure is transient, since liquid is free to flow out to sump through needle valve 108 at a controlled rate. Adjustment of needle valve 108 will thus permit stable operation of the governor to restore the engine to normal speed with maximum rapidity following acceleration due to lightened load.

It is important to observe that the adjustments of valves 108 and 109 are entirely independent of one another, since any time one of the valves is functioning the other valve is effectively bypassed to sump by port 55, the appropriate side of which is uncovered by any movement of pilot valve 54. Thus, with the structure I have described, entirely independent compensation adjustment on both acceleration and deceleration is possible. Instead of seeking an adjustment which is at best a poor compromise between the two conditions, as prior-art structures required, my invention permits accurate and independent adjustment of compensation for both acceleration and deceleration.

The magnitude of centrifugal force necessary to cause fly-weights 64 to attain normal operating position depends, of course, on the tension of spring 65. (The tension of spring 57, being constant, need not be considered in this connection.) Adjustment of the tension of spring 65 may be accomplished by any shift in the vertical orientation of spring seat 68. Various factors can accomplish movement of spring seat 68; circumstances under which these factors operate will not be described in detail.

First, spring seat 68 may be raised or lowered by manual adjustment of knob 26 on the instrument panel. This causes pinion 70 and rack 72 to shift their relative positions, with the result that sleeve 67 moves up or down, thus raising or lowering spring seat 68. This adjustment is employed when the operator desires manually to change the speed of the controlled engine.

As the governor "warms up" after a cold start, the temperature of all the components, including speeder spring 65, increases. It is a well-known property of springs that the force exerted is a function of temperature and, if the temperature of a spring is raised, greater compression is required to achieve a given force. Since, in the present invention, the casing, central casting, and other structural parts are made of a metal having approximately double the thermal coefficient of expansion as the material of which post 119 is formed, heating of the governor after a cold start results in raising the frame members substantially more than the pivot by which bracket 118 is anchored to lever 116. This results in gradually increased pressure on plunger 69 by cam 28g as the operating temperature rises. I have found that by using metals having a ratio of thermal coefficients of approximately two to one, such as aluminum and steel, almost perfect compensation for thermal changes can be achieved in this manner. In other words, with the present invention, there is substantially no change in controlled engine speed between a cold start and the stable temperature at which the governor settles down after continued operation. This represents a great improvement over the operation of conventional governors, since they usually require almost continual readjustment for a long period after a cold start.

Normally, when no speed droop is desired, knob 28 is set at the position whereat arcuate portion 123 of cam 28g is in contact with the physical center of the upper surface of plunger 69. Since surface 123 is a sector of a circular cylinder, and since its center of rotation is pivot member 28f, centered directly above the midpoint of plunger 69, crank 113 can shift its position in response to movements of shaft 22 without affecting to any extent the position of plunger 69. If, however, manual knob 28 is rotated in either direction from the neutral point, cam 28g will be moved to an off-center position relative to pivot 28f, and thereafter any rotation of shaft 22 will cause a shift in the position of plunger 69 with resulting variation in the tension of speeder spring 65. Thus, by appropriate adjustment of knob 28, my governor can be made either to maintain constant speed under wide variations in load, drop slightly in speed in response to added load, or increase slightly in speed in response to added load. In other words, my invention may be caused to over-compensate or under-compensate for load changes, should such result be for any reason desired. (As heretofore stated, the usual reason for employing speed droop is to permit control of load distribution between a number of engines being operated in parallel.) While speed droop is not of itself new, the novel physical structure with which it is accomplished in the illustrated embodiment gives a very precise and accurate control over the degree of speed droop without in any way interfering with the normal functions of the governor and is accordingly considered by me to be an important and novel aspect of my invention.

In prior-art structures, shut down of the

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engine has normally been accomplished by some device which forced the pilot valve in the direction opposite to that in which it would normally move responsively to deceleration. Such arrangements are unsatisfactory in that they interfere with the delicate balance of the most critical element in the entire governor—the pilot valve member. I consider that forcing that member to a given position by imposition of a force overriding the influence of the fly-weights is exceedingly bad practice and one which is likely to result in lowered sensitivity and accuracy of control.

In my present invention, I have provided a novel and much more elegant shut-down means. In my invention, when the manual shut-down control 27 is shifted to the "off" position, spool member 87 is raised, with the result that transverse bore 75, normally communicating with bore 65, is placed in communication with bore 84, while bore 65 is vented to sump. As a result, the slightest downward movement of the pilot valve immediately produces downward movement of the servo-piston 78 which in turn causes further downward movement of the pilot valve 54. This process continues until the engine fuel supply is entirely shut off and the engine is stopped. Thus, by my invention, the pilot valve is caused to turn off the engine without being forced from its normal operation in any way; on the contrary, it does exactly what it normally does in response to deceleration, but the effect of that action is to stop the engine.

The Figure 15 modification

Fig. 15 shows a modified form of my governor in which provision has been made for controlling, as a function of the speed of the controlled engine, the maximum hydraulic pressure developed by the gear pump 38, 40. When the controlled engine is being operated at a very high speed, the hydraulic pressure developed by gear pump 38, 40 can reach an excessive value, with the result that hydraulic fluid is driven into the servo-cylinder, and the servo is caused to function more rapidly than the engine can respond to changes in the rate of fuel feed. This state of affairs can, on occasion, cause jerky, unstable operation of the controlled engine.

The modification of Fig. 15 discloses means whereby the pressure of the hydraulic fluid fed into annular recess 49 by gear pump 38, 40 can be held at a reasonable value even when the controlled engine is operating at a very high speed.

As may be seen from the portion of Fig. 15 which is broken away to show internal structure, I have provided a relief valve apparatus which communicates with passageway 42 by way of passageway 151 and 151a. Passage 151a is drilled in a plug 155 which is screwed into the upper end of bore 41 in lieu of plug 42a of the embodiment shown in Figs. 1-14. Conduit 151 intersects a vertical passage 152, drilled in a valve housing 153 which is mechanically supported by plug 155. The lower end of passage 152 is open to sump, although it is normally closed off by a poppet valve 154 mounted at the extreme lower end of a valve rod 156 carried in passage 152 and machined to slide snugly therein. The lowermost end of rod 156, immediately above poppet valve member 154, is machined to form a tapered bleed valve 156a which will permit flowing of hydraulic fluid from passage 152 to sump when poppet member 154 is open; obvi-

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ously, the rate of flow permitted will be dependent on the position of rod 156, the rate of flow increasing as rod 156 is lowered.

About midway of its length, rod 156 is provided with an annular shoulder 157 which forms the upper seat for a coil spring 158, the lower spring seat 159 for spring 158 being formed on the upper end of member 153. Spring 158 holds poppet 154 normally seated over the lower end of passage 152.

Bracket 66 has, in this embodiment, been provided with a slot 66a cut part way down from its upper end to accommodate an arm 160. That arm has a vertical body with an extension at each end, each extension being at right angles to the body and extending away from the body portion in opposite directions. The upper extension, denoted 160a, is firmly secured to sleeve 67; thus arm 160 is free to rise and fall with sleeve 67 since slot 66a affords vertical freedom to arm extension 160a. The lower extension of arm 160, denoted 160b, is drilled and threaded to receive a screw 161; screw 161, in the illustrated embodiment, has a large flat head at its lower end, and its upper end is slotted to permit adjustment by a screw driver. A lock-nut 162 provides a means of locking screw 161 in any desired position.

In the operation of this form of my invention, the screw 161 can be manually adjusted so that the head of screw 161 comes in contact with the upper end of valve rod 156 at the position of sleeve 67 which corresponds to optimum hydraulic pressure from gear pump 38, 40. That is, the speed of the controlled engine is governed by the position of sleeve 67; the hydraulic pressure generated by the gear pump 38, 40 is directly responsive to engine speed. Screw 161 is adjusted to engage valve rod 156 at that setting of sleeve 67 which gives an engine speed such that optimum hydraulic pressure is generated by the gear pump. What this optimum pressure is will vary from installation to installation, since it will depend on the rapidity with which the engine speed responds to changes in fuel supply. If the engine is very rapid in response, relatively large hydraulic pressure can be developed to advantage by the gear pump 38, 40. If, on the other hand, the engine is sluggish in its response to changes in fuel supply, then relatively lower pressure is desirable for operation of the servo mechanism of the governor.

Once the proper setting for screw 161 has been determined, it is locked in place with lock-nut 162. Thereafter, whenever sleeve 67 is pressed downward sufficiently to raise the speed of the controlled engine above that value at which the hydraulic pressure is at optimum value, the head of screw 161 engages valve rod 156 and opens poppet 154, permitting hydraulic fluid to bleed out of the lower end of aperture 152 around tapered valve member 156a. This results in maintenance of substantially optimum pressure within passage 42.

The modification of Figures 16-17

Figures 16 and 17 show another modified form of my invention which makes possible fine control of the rate of flow of hydraulic fluid from the pilot valve to the servo mechanism.

The result of slowing down the rate of flow of fluid into the servo cylinder is to reduce the rapidity with which the fuel supply to the engine is altered in response to a change in load. Con-

trol over that reaction time is desirable to insure smooth, jerk-free, and stable operation of the controlled engine. In other words, the objectives achieved by the modification about to be described are quite similar to those accomplished by the Fig. 15 modification, although the modes of operation of the two structures are different, and they are not exact equivalents. The Fig. 15 modification merely maintains control over the hydraulic pressure developed by the gear pump in the governor to insure that it does not exceed a reasonable value at times when the gear pump is being driven by the controlled engine at high speeds. The structure about to be described, on the other hand, is not limited in its effects to high-speed engine operation, but, on the contrary, permits "tailoring" of the governor response-time to fit the behavior pattern of the particular engine to which the governor is connected, whether it be operated at low speed or high speed.

The structure about to be described is an added feature incorporated with the shut-down mechanism (already described) which utilizes spool valve member 87. It will be recalled that, when the engine is running, valve member 87 in the embodiment of Figs. 1-14 normally occupies a lowered position effective to open fully port 85 to communication with port 75 and to open fully port 84 to communication with port 76.

As may be seen in Figs. 16 and 17, I have provided as an additional feature, to be used if desired, an adjustment means whereby the spool member 87, when in its "on" position, may be caused to close off any desired part of the ports 84 and 85, thus giving a fine manual control over the rate at which hydraulic fluid can flow into and out of the respective chambers of servo cylinder 77. As may be noted from Fig. 16, I have provided, on arm 88, a small projection 165 extending outwardly therefrom at right angles. Projection 165 rests on the upper surface of a cam 166, mounted adjacent projection 165 and carried on a vertical rod 167 the top end of which is supported in a suitable bearing 168 attached to any conveniently located frame bracket. The upper end of rod 167 is slotted (for screw driver adjustment) and threaded to receive a lock-nut 169. As may be best seen from Fig. 17, the surface of cam 166 is cut to rise gradually from a minimum level to a substantially higher level, and the portion of cam 166 engaged by pin 165 depends on the position which rod 167 is occupying. When pin 165 is resting on the lowest portion of cam 166, spool valve member 87 is in the same position that it occupies, when in the "on" position, in the embodiment of Figs. 1-14. If, however, cam 166 be turned, by rotation of rod 167, pin 165 gradually climbs up the curved edge of cam 166 and the "on" position of spool member 87 is raised so that lands 87a and 87b gradually cover, and thus reduce the effective apertures of, ports 84 and 85. By appropriate adjustment of rod 167, the effective size of ports 84 and 85 can be set to any desired value. The operation of spool member 85 as a "shut-down" mechanism is entirely unaffected by the supplemental apparatus just described, since movement of manual control member 27 to the "off" position raises arm 88 and spool member 87 as before without any interference from cam 166 or its associated parts.

While I have in this specification described for illustrative purposes a single embodiment of my invention, it will be understood that many varia-

tions thereon and departures in matters of detail can be made by those skilled in the art without departing from the spirit of my invention. It is accordingly my desire that the scope of my invention be not limited to the precise embodiment shown but that such scope be determined by reference to the appended claims.

I claim:

1. In a hydraulic governor wherein movement of a pilot valve regulates the speed-control mechanism of an engine, anti-hunt mechanism comprising means operative on movement of the pilot valve to set up a transient hydraulic pressure opposing such movement, said means comprising an adjustable valve operable to control the duration and instantaneous magnitude of such opposing force when the pilot valve has moved in one direction, a second adjustable valve similarly operable when the movement of the pilot valve has been in the opposite direction, and means for rendering inoperative each of the valves during periods when the other is functioning.

2. In a hydraulic governor wherein movement of a pilot valve regulates the speed-control mechanism of an engine, anti-hunt means operative responsively to movement of the pilot valve to create a transient hydraulic pressure opposing such movement, a pair of independently adjustable valve means operative respectively to control the instantaneous magnitude and duration of such opposing pressure on acceleration and deceleration, and means effective to render inoperative one of said adjustable means following acceleration of the controlled engine and the other of such adjustable means during deceleration of the controlled engine.

3. In a speed-control governor for engines, said governor being of the type wherein the movement of a pilot valve regulates the speed-control mechanism of the engine and said pilot valve is connected in a hydraulic circuit having a pressure source and a sump, the improvement comprising anti-hunt means operative responsively to movement of the pilot valve to set up a transient hydraulic pressure opposing such movement, a manually adjustable needle valve discharging to sump and operative to control the duration and instantaneous magnitude of such force following acceleration of the engine, and a second manually adjustable needle valve discharging to sump and operative to control instantaneous magnitude and duration of such opposing force following deceleration of the engine.

4. Apparatus according to claim 3 wherein means including a land carried by the pilot valve and cooperating with a sump port operates to by-pass to sump the first needle valve following deceleration of the engine and the second needle valve following acceleration of the engine.

5. In a hydraulic speed-control governor for engines of the class described, a servo piston adapted for connection to the speed-control mechanism of an engine, and a hydraulic circuit comprising a pressure source and a sump connecting the pilot valve and the servo piston, anti-hunt apparatus comprising a closed chamber containing a port which discharges to sump, a piston in the chamber coupled to the pilot valve and operative to move axially in step therewith, a land on said piston member operative to divide said closed chamber into upper and lower portions and normally operative to close off said sump port, a pair of compensating pressure chambers, means coupled to the servo piston op-

erative to produce in one of said chambers a super-atmospheric hydraulic pressure, a pair of conduit means respectively connecting one of said pressure chambers to the upper portion of said closed chamber and the other of said pressure chambers to the lower portion of said closed chamber, and an adjustable needle valve for each of said conduit means, said valves exhausting to sump and being operative respectively to control the duration and instantaneous magnitude of the hydraulic pressure within said conduit means.

6. An engine governor having a drive shaft, fly-weights turned by rotation of the drive shaft, a pilot valve actuated by the fly-weights, and a speeder spring opposing the action of the fly-weights on the pilot valve, said drive shaft, fly-weights, pilot valve, and speeder spring being mounted coaxially, and a slip coupling connecting the drive shaft to the pilot valve operative to transmit a relatively small torque to the pilot valve during rotation of the drive shaft.

7. An engine governor comprising a drive shaft, fly-weights turned by rotation of the drive shaft, a pilot valve actuated by the fly-weights, and a speeder spring opposing the action of the fly-weights on the pilot valve, said drive shaft, fly-weights, pilot valve, and speeder spring being mounted coaxially, and a spring member coupling the drive shaft to the pilot valve operative to transmit torque to the pilot valve from the drive shaft and operative also to impose axial force on the pilot valve in opposition to the speeder spring.

8. An engine governor comprising a casing and structural parts therewithin formed of one metal, fly-weights, a speeder spring mounted to oppose the force exerted by the fly-weights during rotation, a movable spring seat for the speeder spring adapted when moved to vary the tension of the speeder spring, said fly-weights and speeder spring being connected to and supported by said casing and structural parts, an elongated support member mounted on the casing and formed of a second metal having substantially lower thermal expansion coefficient than said first-mentioned metal, and means supported by said support member operative to press on said movable spring seat operative as the temperature of the casing increases to increase the tension of the speeder spring.

9. Apparatus according to claim 8 wherein the casing and structural parts are formed of aluminum and the elongated support member is formed of steel.

10. Apparatus according to claim 8 wherein the support member is formed of metal having a thermal expansion coefficient substantially one-half that of the metal of which the casing and structural parts are formed.

11. In a governor for engines, comprising a control element, fly-weights adapted to move the control element responsively to rotation, a speeder spring imposing force on the control element in opposition to the forces exerted by the fly-weights, and a speed control member adapted for connection to an engine speed-control mechanism, speed droop apparatus comprising a movable spring seat for the speeder spring operative upon movement to vary the tension thereof, a cam mounted for rotation adjacent said spring seat and having an arcuate cam surface pressing on said spring seat, said arcuate surface being circular in section, adjustment means for sliding said cam across the surface of said spring

seat to move the center of curvature of said cam face into coincidence with the axis of rotation of said cam and selectively to either side, a manually operable control mounted externally of said governor, mechanical linkage flexibly coupling said external control to said adjustment means operable responsively to external manual operation of said control means to actuate said adjustment means continuously through its range of movement, and means coupled to the speed-control member operative to rotate the cam about its axis of rotation responsively to movement of the speed-control member.

12. In a governor for engines, comprising a control element, fly-weights adapted to move the control element responsively to rotation, a speeder spring imposing force on the control element in opposition to the forces exerted by the fly weights, and a speed-control member adapted for connection to an engine-control mechanism, speed-droop apparatus comprising a movable spring seat for the speeder spring operative upon movement to vary the tension thereof, a cam mounted for rotation adjacent said spring seat and having an arcuate cam surface pressing on the same, said arcuate surface being circular in section, said cam being supported at at least one point by a support member formed of material having a thermal expansion coefficient substantially less than that of the remainder of the governor structure, means for sliding said cam across the surface of said spring seat to move the center of curvature of said cam face into and out of coincidence with the axis of rotation of said cam, and means coupled to the speed-control member operative to rotate the cam responsively to movement of the speed-control member.

13. In an engine governor having a drive shaft coupled to a governed engine and being rotatably driven thereby, a hydraulic circuit and a pressure source energized by rotation of the drive shaft, and means for changing the speed-control setting of the governor, an adjustable relief valve in the hydraulic circuit operable to limit the maximum pressure in said circuit when the controlled engine exceeds a predetermined speed, and linkage arranged with said valve and said means to automatically actuate said valve when the speed setting of said means exceeds a predetermined value.

14. The structure of claim 13 wherein said linkage is mechanical and progressively opens said valve as the speed-control setting of said means exceeds progressively a predetermined value.

15. An engine governor having a drive shaft, fly-weights turned by rotation of the drive shaft, a pilot valve actuated by the fly weights, a speeder spring opposing the action of the fly-weights on the pilot valve, a servo piston, a hydraulic pressure source energized by rotation of the drive shaft, a hydraulic circuit comprising the pressure source and a sump connecting the pilot valve and the servo piston, an adjustable relief valve connected to the hydraulic circuit at substantially the point of maximum pressure, and mechanical linkage inter-relating the relief valve and the speeder spring operative to open slightly the relief valve when the compression of the speeder spring exceeds a predetermined value and to open the relief valve further upon the imposition of additional compression to the speeder spring.

16. A hydraulic governor for engines, comprising a drive shaft, fly-weights turned by rotation

of the drive shaft, a pilot valve actuated by the fly-weights, a servo cylinder, a servo piston mounted in and reciprocable within said cylinder, a source of hydraulic pressure, a hydraulic circuit comprising the pressure source, the pilot valve, the opposite ends of the servo cylinder, and a sump, and a movable valve member having two extreme positions, operative in the first extreme position to supply hydraulic fluid to one end of the servo cylinder responsively to movement of the pilot valve induced by deceleration of the engine and operative in the other extreme position to supply hydraulic fluid under pressure to the other end of the servo cylinder under like conditions, said valve member being operative also to provide a path to sump for the low-pressure end of the cylinder in either extreme position, and manually adjustable means operative when the valve is in the first extreme position to vary the position of the valve member within narrow limits and thereby to regulate the effective rate of flow of hydraulic liquid into and out of the respective ends of the servo cylinder.

17. In a hydraulic governor of the type having a speeder spring, a sump and a pilot valve slidably mounted in a sleeve and being movable in said sleeve in response to loadings of a speeder spring to regulate the speed of a prime mover, anti-hunt apparatus operable in response to movement of said pilot valve, comprising a land fixedly carried by said pilot valve and being slidably movable in a chamber provided by said sleeve, said sleeve providing also a port communicating with said chamber at substantially the midpoint thereof and with the sump of the governor and having a diameter slightly smaller than the width of said land whereby when said land is aligned with said port it blocks the flow of fluid therethrough, a pair of passages communicating respectively with said chamber adjacent each end thereof, said passages being separated from said port when said land is aligned with the port, and means communicating with said passages for applying transient pressures to said passages selectively and in opposition to the direction of movement of said land and pilot valve, whereby when said pilot valve is moved in one direction in response to a change in the loading of the speeder spring, said land is moved from alignment with said port, opening communication between one of said passages and said port, while a transient pressure in the other of said passages acts against said compensating land and opposes the movement of said pilot valve.

18. In a hydraulic governor for engines, a sleeve, a pilot valve axially movable in said sleeve, a speeder spring for moving said pilot valve in response to changes in loading of the spring, control means for changing the speed setting of the controlled engine, means for actuating said control means in response to axial movements of said pilot valve, a sump, and anti-hunt means comprising a double-acting valve equipped with a valve shaft coupled to said control means and movable in response to movement thereof, said sleeve providing a chamber and a passage communicating with each end of the chamber, said passages communicating at their other ends with the opposite sides of said valve, said sleeve having a port therein communicating with said chamber at substantially the midpoint thereof, said port being connected to the sump of the governor, a land carried by said pilot valve and movable in said chamber, and means in each of said passages

for slowly reducing a pressure increase therein to the sump pressure, said double-acting valve and passages being interrelated with said pilot valve and control means to provide a pressure on one side of said land resisting the movement of said pilot valve while at the same time the chamber portion on the other side of said pilot valve is connected to the governor sump through said port.

19. In a hydraulic governor for a prime mover, a casing providing a sump, pump means in said casing for supplying fluid under pressure to said governor, a sleeve mounted in said casing and providing a pressure port, sump ports and control ports all spaced apart, a servo-motor operable in either direction by the pressure of a fluid exerted thereon, said governor providing passages connecting each of said control ports respectively to one side of the servo-motor, spaced-apart lands carried by said pilot valve and having a width slightly less than the diameter of said control ports, means connecting said servo-motor with the speed-control adjustment of the engine, and a speeder spring and fly weights for positioning said pilot valve with the lands thereof in alignment with said control ports when the speed of the controlled engine is at the speed setting of the governor, said speeder spring and fly weights being responsive to speed changes of the controlled engine to move the pilot valve so that the lands carried thereby are moved out of alignment with said control ports.

20. In a governor for a prime mover having a casing and structural parts therein formed of material having substantially the same coefficient of thermal expansion, a speeder spring adapted to be adjustably tensioned to establish the speed-control setting of the governor, a movable spring seat for the speeder spring adapted when moved to vary the tension of the speeder spring, said speeder spring and spring seat being supported by said casing and structural parts, a support member mounted on said casing and being formed of a material having a substantially lower coefficient of thermal expansion than the material from which said casing and structural parts are formed, and means supported by said support member operative to press on said movable spring seat operative as the temperature of the casing increases to increase the tension of the speeder spring.

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Number	Country	Date
83,091	Austria	Mar. 10, 1921