July 20, 1943. A. KALIN 2,324,516

GOVERNOR

Filed Sept. 10, 1941

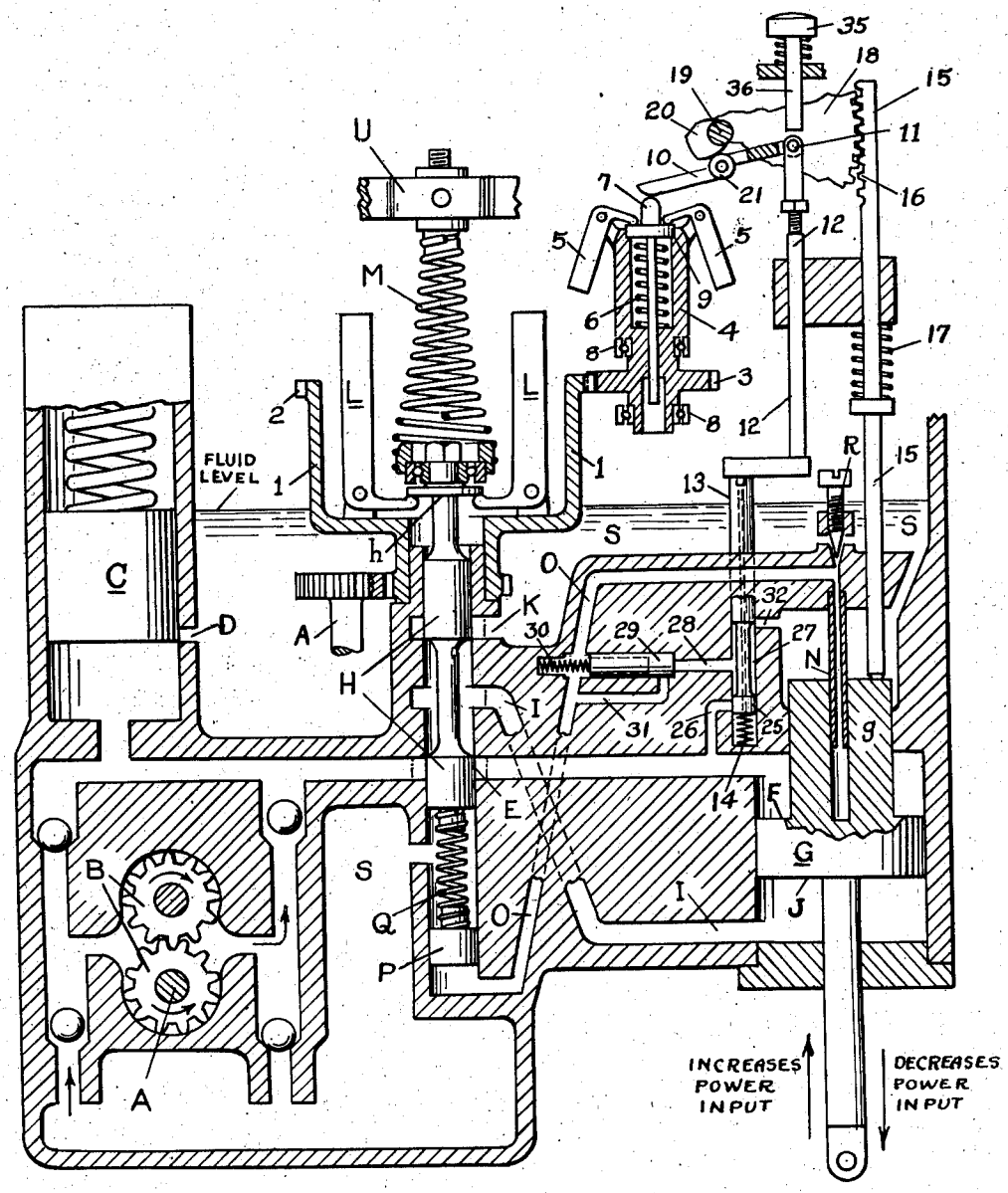


FIG 1

INVENTOR
ALBERT KALIN
BY
George W. Soule
ATTORNEY

Patented July 20, 1943 2,324,516

UNITED STATES PATENT OFFICE

2,324,516

GgovernOR

Albert Kalin, Cleveland, Ohio

Application September 10, 1941, Serial No. 410,283

2 Claims. (Cl. 264—3)

This application is an improvement shown but not claimed in my application Serial No. 385,228 filed March 26, 1941.

The present invention relates to an improved load limiting, speed responsive mechanism in combination with (e. g.) an isochronous governor adapted to perform regulating operations on a prime mover through the intermediary of hydraulically acting means. This indicates one object hereof.

The invention relates more specifically to mechanism in or in combination with a governor for Diesel or other internal combustion engines in which destructive stresses occur or other undesired results obtain when fuel is fed to the engine at too high a rate at low speeds under abnormal load. Said mechanism, in other words, limits the brake-mean-effective-pressure (B. M. E. P.) in the engine cylinders when the speed of the engine drops below a predeterminable low point, indicating a further general object.

A further object is to provide a B. M. E. P. limiting means in connection with a governor operative to regulate speed through the intermediary of a hydraulically acting servo-motor and a speed responsive element controlling a pilot valve for the servo-motor, wherein said means and the pilot valve are always free to operate independently of each other so far as mechanical connections between the two are concerned.

A specific object is to provide a B. M. E. P. controlling or limiting mechanism, for a governor including a pilot valve as outlined above, which mechanism becomes operative at predeterminable low speeds on part of a governed engine to move the valve positively in the direction to cause reduction of rate of feeding of fuel to the engine but hydraulically through the intermediary of yieldable elastic means.

Other objects and features of the invention will become apparent from the following description.

In the accompanying drawing Fig. 1 is a schematic or diagrammatic general assembly view of a governor incorporating a preferred general arrangement of elements embodying the invention, the governor generally being essentially the same as that shown in the parent application and in my Patent 2,219,229 issued October 22, 1940.

Describing first the governor arrangement generally, parts indicated by letters, A is a drive shaft for the governor arranged to be connected to the engine or other prime mover to be governed, the shaft driving a gear pump B. The gear pump B is shown with a well known check valve system and ducts arranged to enable the pump to draw hydraulic fluid from a sump S and deliver it to an accumulator having a spring loaded plunger C and discharge vent D which cooperate to keep the pressure in the accumulator at constant positive value. Fluid from the accumulator is supplied to a governor port E and continuously therebeyond to one face F of a double acting hydraulic servo-motor power piston G. The port E is controlled by a lower land portion of a pilot valve plunger H which, if moved downwardly from the normal position in which shown, opens the port E to a duct I leading to the opposite (effectively larger) face J of the power piston, causing upward movement of the piston. Upward movement of the pilot valve plunger from its normal position causes an upper land portion thereof to open the duct I at a port K, dumping pressure fluid from below the power piston into the sump and causing the piston to descend by reason of the constant pressure maintained at the piston face F.

The pilot valve plunger H is moved vertically in opposite directions as a result of the difference between centrifugal force on a set of fly-balls L on a ball head I rotated by the prime mover (as by shaft A and appropriate gearing connected to the ball-head) and opposing force exerted by an adjustable speeder spring M, so that an increase in speed on part of the prime mover raises the pilot valve plunger and a decrease in speed lowers said plunger. Regulation of speed by the servo-motor incident to change in load is initiated by such movement of the pilot valve plunger H.

Power input to the governed prime mover is increased by raising the power piston G (and decreased by lowering the piston) as indicated on the drawing. Thus, in governing, for example, a Diesel engine, increase in load causes decreased centrifugal force on the fly-balls and lowering of the pilot valve plunger, thus initiating an increase in the rate of supply of fuel to the engine; and decrease in load causes raising of the pilot valve plunger and consequent decrease in the rate of supply of fuel.

An arrangement such as thus far described requires stabilization in order to prevent vacillation in speed or hunting consequent upon load change and subsequent speed-correcting operation of the governor. The stabilizing mechanism includes a hydraulic compensation system supplied with fluid from the sump and actuating and receiving compensation pistons of said system operating as a function of regulating movement of the servo-motor piston G to damp or check

each movement of the servo-motor piston and prevent temporary overregulation of the prime mover thereby.

The actuating compensation piston N works in a trunk portion *g* of the power piston G and displaces hydraulic fluid in a duct O leading to the receiving compensating piston P, slidably mounted in a cylindrical extension of the outer pilot valve element, whenever the power piston G starts to perform a regulating operation in either direction. The space above the trunk portion of the power piston is open to the sump through an upper wall portion of the power cylinder block. The motion of the receiving compensation piston P is always opposite the regulation-initiating movement of the pilot valve plunger H, and places temporary primary yielding compensating restraint on each movement of said valve plunger through the intermediary of a coil spring Q connecting the compensating piston P and valve plunger H. The primary compensating force is removed, after regulation is effected, by relieving the positive or negative pressure in the duct O through a very small aperture in an adjustable secondary compensation leak-off needle valve R communicating the duct O with the sump.

As fully explained in my Patent 2,219,229, the fly-balls cause continuous rotation of the pilot valve plunger in its coacting fixed valve sleeve through contact of the ball-arm fingers with a disc portion *h* of the plunger; and the spring Q is so connected with the plunger and receiving compensation piston P that the piston is likewise rotated in its fixed cylinder. This eliminates the possibility of binding of either the valve plunger or said piston due to foreign matter in the hydraulic fluid and assures that the pilot valve is free from friction and therefore free to respond to very slight changes in speed. The spring Q also allows relative axial movement between the parts connected thereby so that the valve is always free to respond promptly to any change in centrifugal force acting on the fly-balls.

In operation, when the engine load is increased the resultant decrease in speed causes relative inward movement of the fly-balls L; lowering of the pilot valve plunger; opening of the port E; admission of hydraulic fluid through duct I to the lower side of the power piston G, and increase of fuel to the engine. The upward movement of the piston G pumps fluid through duct O; raising the receiving compensation piston P; compressing the spring Q and exerting an upward slight force on the pilot valve plunger sufficient to offset the temporary overbalance of speeder spring force relative to the centrifugal force on the fly-balls, thus closing the port E. The port E is ordinarly closed sufficiently quickly to prevent overtravel of the power piston in the speed correcting direction; and, as the engine returns to normal speed, the compression force of the spring Q is dissipated by reason of leakage of fluid from the duct O into the sump through the secondary compensation leak-off aperture of the needle valve R. During the primary compensating action of the compensating pistons the flow through the needle valve R is negligible due to the relatively small size of the valve aperture.

The mechanism operates the same as described above for a decrease in load except that all the movements are reversed, the compensating spring Q being placed temporarily in tension by suction action of the compensation fluid system, and hydraulic fluid being finally drawn from the sump into the duct O through the aperture of the needle valve R to relieve said tension.

Speed adjusting or changing means is represented by the speeder member U (for instance a lever) at the upper end of the speeder spring M. Additional speed changing means would, of course, be provided, for example as in my application Serial No. 385,228 identified above, or in other ways as well known in the art.

The brake-mean-effective-pressure control or load limiting means hereof employs a known arrangement and principle of operation including the provision of special speed responsive means acting in a manner inverse to the usual speed weighing means of speed governors in that said special speed responsive means functions when the speed of the engine drops below a predeterminable point to limit the rate at which the hydraulically acting regulating means (e. g. power piston G) can cause fuel to be fed to the engine. Thus, for example, if the engine begins to labor heavily under the applied load, the fuel input will be cut down rather than increased as would ordinarily happen on a governed engine when added load reduces speed. One advantage of the present arrangement, however, is that said function of the inversely operating special speed-responsive means (outlined above) cannot normally interfere in any way with the normal function of the main speed weighing means or cause the latter to lag or fail to respond quickly to slight changes in speed within the usual range of deviation from normal consequent upon expected load changes. Also, as herein adapted and arranged the special speed responsive mechanism has only to move a small value thereby to cause the necessary movement of the pilot valve to initiate operation of the fuel regulating servo-motor in the proper direction, whereby the special speed responsive mechanism, being never subjected to substantial load, can be made very sensitive and of small size. The main speed regulating governor mechanism and the B. M. E. P. control governor mechanism "float" independently of each other during load change and ordinarily the operation of neither has any influence on the other. The arrangement as schematically shown is as follows:

The inversely operating speed weighing means, as shown, is a rotatably mounted flyball head 4 turned through intermeshing gears 2 and 3 on respective ball-heads 1 and 4; depending flyballs 5, and a spring 6 within the body of the ball-head 4 and reacting upwardly against a plunger 7 in contact with overhanging ball arm fingers of the flyballs 5. Bearings for the ball head 1 are shown at 8 supported by the governor casing. The relation of flyball weight, length, etc. to the strength of the spring 6 is such that the flyballs 5 normally hold the plunger 7 in the depressed position shown, as against a shoulder 9 on the ball-head body 4. When the speed of the engine drops to a point at which (e. g.) the engine should not be subjected to increased fuel, as explained above, the spring 6 then raises the plunger 7 against the decreased centrifugal force of the flyballs 5. Such would occur were the engine to become overloaded, as will be obvious.

Upward movement of the plunger 7 of the ball-head 4, from the position in which shown, is ineffective when the power piston G is in a certain position intermediate of the upper and lower limits of its stroke, for example somewhat below the mid position, illustrated. In such lower posi-

tions of the power piston the rate of feeding of fuel is under a safe maximum at any possible low speed, even with the engine subjected to abnormally heavy load and tending to stall. But when the power piston is in the upper range of movement (as above the position in which shown) at a time when excessive engine load reduces the speed of the engine until the spring actuated plunger 7 is raised by the spring 6 then said upward movement of the plunger 7 causes the engine to be shut down (or the fuel to be decreased until the excessive load is removed) as follows:

A lever 10 is pivoted at one end, as at 11, to a vertically slidable bar 12, the lever having its free end resting on the plunger 7. When the bar 12 is depressed the lower end of the bar operates a valve plunger 13, pressed upwardly toward the rod 12 as by a spring 14, and arranged and operating as will be described later, to cause the pilot valve H to initiate power-input-decreasing movement of the power piston G.

A vertically slidable rod 15 having a toothed rack portion 16 is arranged to be raised by the power piston G, as through the trunk portion *g* thereof, during the upper range of movement of the power piston. A return spring 17 for the bar 15 moves said bar downwardly, causing it to follow the power piston. The rack teeth on the bar 15 mesh with a pinion or pinion segment 18 on a horizontal shaft 19 suitably supported in fixed position. A fulcrum member 20 on the shaft 19 engages the lever 10, as through a roller 21 on said lever, to enable the upward movement of the special-flyball-actuated plunger 7 to depress the bar 12 increasing distances as the power piston moves through its upper ranges. Thus the special flyball operated plunger 7 can depress the valve-operating rod 12 only when the speed-regulating servo-motor piston G tends to operate to supply fuel to the engine at rates above the predeterminable maximum mentioned above, for otherwise the fulcrum member 20 is rotated clockwise out of the positions in which it can act as a fulcrum for the lever 10.

So long as the governor mechanism, exclusive of the inversely operating flyball mechanism 4, 5 etc. and valve mechanism controlled thereby, described below, is in control of the engine a lower land portion 25 of the valve plunger 13 is in closing relation to a passage 26 leading to a constant pressure space of the pump and accumulator. When the plunger 13 is depressed, constant pressure fluid flows through the passage 26, past a neck portion of the valve plunger 13, through the guide bore 27 in which the plunger 13 slides, and to a passage 28 in which is slidably mounted a cylindrical valve plug normally held in the position in which shown, as by a spring 30. The plug 29, when subjected to constant pressure from said source, moves to the left to open a passage 31 leading to the compensation fluid duct O, so as to subject fluid in said duct O to high pressure.

Except when the hydraulic compensation system is in operation as earlier explained, the pressure in the compensation fluid duct O is negligible; and it will be seen therefore that diversion of high pressure fluid to the duct O will act on the receiving compensation piston P to raise it, placing the spring Q in compression and thereby raising the pilot valve plunger H against the opposing action of the speeder spring M. This causes the power piston to descend, as already explained, because the upward movement of the pilot valve plunger dumps high pressure fluid from below the power piston G.

So long as the valve plunger 13 remains depressed, fluid from the high pressure side of the regulating hydraulic system can be prevented from leaking out of the compensation fluid duct O through the leak-off valve R, possibly causing failure of the B. M. E. P. control mechanism to effect a positive shutdown movement in a relatively open adjustment of the leak-off valve, by so arranging the plug 29 that it blocks the duct O (as at the region occupied by the spring 30) whenever the passage 31 is caused, by leftward movement of the plug 29, to communicate with the passages 26, 27 and 28. Thus the slowing down operation performed by the B. M. E. P. control governor mechanism is positively assured under all circumstances.

When the valve plunger 13 moves to raised position, return movement of the plug 29 to the right, as against a shoulder portion of its guide (returned position shown) is enabled by providing a bleeder passage 32 intercepting the upper end of the guide bore 27 for the valve plunger 13 adjacent an upper land portion of said valve plunger, one end of the passage 32 communicating with the sump in a suitable manner. The bleeder passage 32 is closed by the upper land portion of the valve plunger 13 when the latter is depressed sufficiently to move the lower land 25 into unblocking relation to the high pressure admission passage 26.

The downward pressure of the rod 12 on the valve 13 ceases as soon as the power piston descends to a point such that the fulcrum member 20 is swung out of operative position with respect to the lever 10. The power piston G continues to move downwardly only until the upward movement of the valve plunger 13 unseals the passage 32 for communication with the bore 27 and, thereupon, fluid pressure in the duct O is reduced to sump pressure by flow of operating fluid from the duct O through passage 31, the guide bore for the plug 29 and the passage 28. Any pressure above sump pressure remaining in the duct O after the plug 29 closes the upper end of the passage 31 escapes by way of the needle valve apertures at valve R. Then the regular governor mechanism again resumes control of the engine.

The member 35 shown in alignment with the upper end of the shut-down bar 12 is a hand operable device having a stem 36 poised over the bar 12 or lever 10 to cause shutting down of the engine at any desired time, as in an emergency, independently of the B. M. E. P. control mechanism and main speed responsive means.

I claim:

1. A speed governor combination comprising a main speed responsive governor mechanism including control means and regulating means adapted and arranged to cause increased input to a prime mover upon increase in load and vice versa, an inversely operating speed responsive governor mechanism including a normally closed valve and a fluid supply line controlled by the valve, said line having a portion connected with the valve and another portion into which fluid from the supply line is directed when the valve is opened, and fluid pressure actuatable means operatively associated with the main governor control means and caused to act thereon in a manner to oppose the usual action thereof upon decrease in speed of the governed prime mover, whereby in event excessive load on the prime mover depresses the speed thereof to such an extent that increased input is then undesirable,

said inversely operating mechanism can, by control of said valve, prevent the first governor mechanism from causing further increase of input to the prime mover in said event.

2. In a speed governor for prime movers, a main hydraulic relay governor mechanism,—including speed responsive means, pilot valve means operated thereby and a regulating servomotor controlled by the pilot valve means—, arranged normally to control the speed of the prime mover, a secondary speed responsive mechanism, a source of hydraulic pressure fluid, a hydraulic system operatively separate from the hydraulic relay and including hydraulically displaceable means connected to an element of the pilot valve for movement of said element, a valve associated with said system and normally disconnecting said source and the system, and means adjusted by the joint action of the servomotor and the secondary speed responsive mechanism for operating the valve to connect said source and hydraulic system for operating the pilot valve means independently of the first mentioned speed responsive means.

ALBERT KALIN.