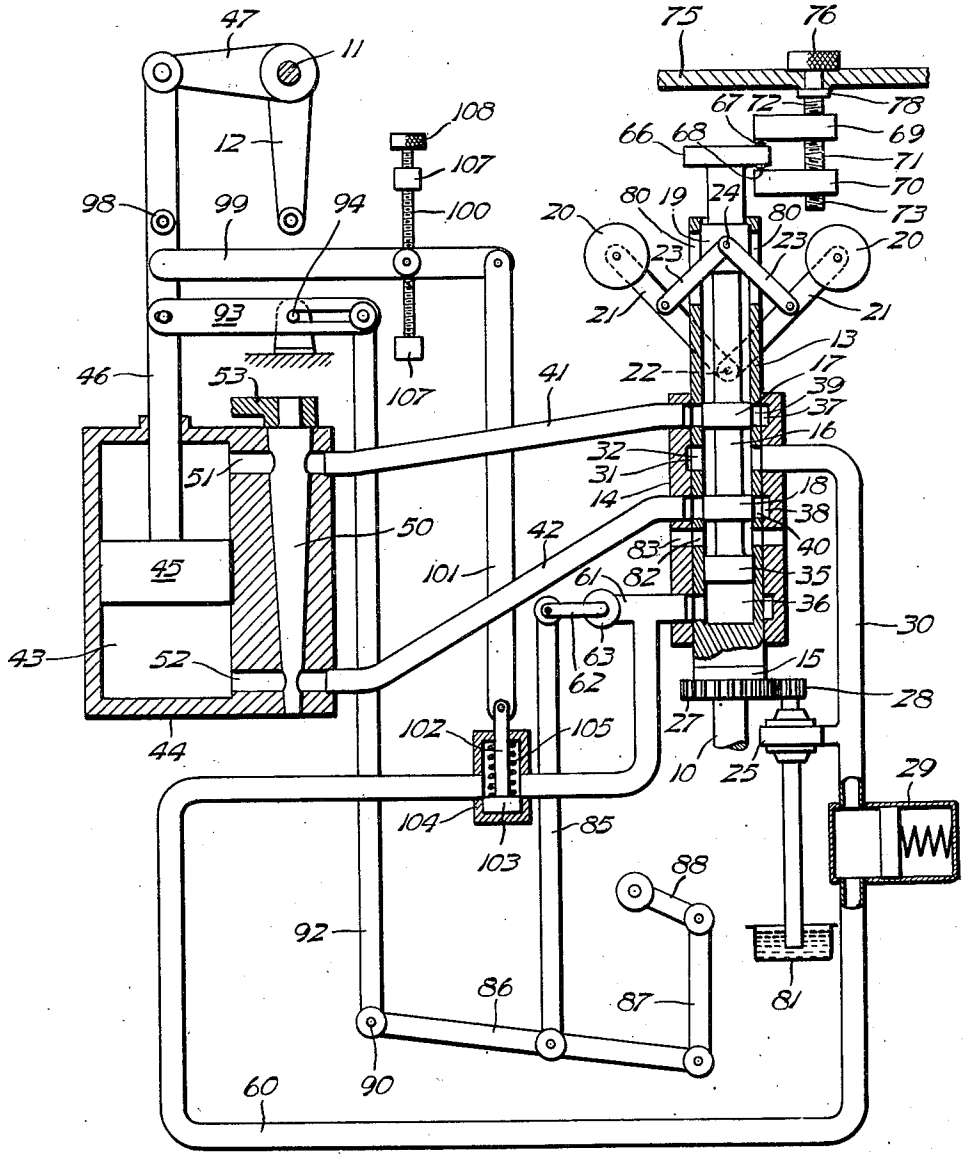
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SELF-COMPENSATING PILOT MEANS FOR HYDRAULIC GOVERNORS

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Fig. 1.



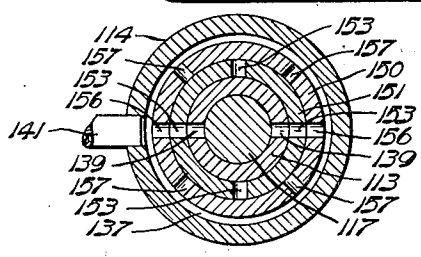


Fig. 2.

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SELF-COMPENSATING PILOT MEANS FOR HYDRAULIC GOVERNORS

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3 Claims. (Cl. 121—43)

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This invention relates to governors for speed control and similar adaptations.

The underlying principles of governor control are adaptable to controlling and regulating various conditions and the principles of the present invention are not limited to any particular field of governor adaptation. However, the widest present field of use of governors is probably in the control of the speed of internal combustion engines and as a convenient way of setting forth the principles of the present invention, reference will be had to such field by way of example. For instance, in Diesel engine installations the governor mechanism is made responsive to changes in the speed of the engine and in response to such speed changes varies the fuel setting of the engine to correct for the speed changes to thus maintain a constant speed or, at least, a predetermined relationship between engine load and speed.

In conventional flyball governors, the centrifugal flyballs are direct acting and not only indicate speed but also serve directly as a source of mechanical energy to change engine fuel setting. In such governors a corrective fuel change requires a substantial speed change to overcome friction in the various parts. This results in a tendency to "over-govern," whereupon the governor makes an adjustment by a series of over-shooting oscillations in the fuel setting change which gradually damp out at the conclusion of the governing adjustment. This undesirable phenomenon is known as "hunting" and various expedients and auxiliary mechanisms have been availed of in an endeavor to reduce "hunting," and its attendant objections and undesirabilities. In some instances the "hunting" oscillations never damp out and "hunting" is continuous.

In view of the undesirabilities of direct acting centrifugal governors, particularly from the standpoint of sensitivity, a considerable field of development in hydraulic governors has evolved. In the usual hydraulic governor, relatively small flyballs are used merely to indicate speed and to move a small pilot valve in a hydraulic system, the valve being arranged to release extraneous hydraulic fluid pressure energy for actually manipulating the fuel control system and other governor adjuncts. The speed change which produces a response in hydraulic governor application can be very much less than in a direct acting centrifugal governor. However, special provision must still be made to eliminate or minimize "hunting" and conventional hydraulic governors employ complicated hydraulic and mechanical systems to effect compensation, that being the term generally given to the function of reducing "hunting" to a minimum, so that a new fuel setting due to governor adjustment

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is attained as nearly as possible by a direct movement from the old to the new setting without objectionable oscillation.

The present invention has to do with the substantial elimination of the complicated auxiliary mechanism ordinarily employed in governors generally to effect compensation. The principles of the present invention are applicable both to mechanical centrifugal governors where the centrifugal force directly effects correction and to hydraulic governors wherein a hydraulic relay effects the ultimate correcting adjustment. In its broadest aspect the present invention involves a forced continuous phyiscal reciprocation or oscillation of the member that is immediately responsive to centrifugal action. Or, even more broadly, the present novel result, achieved by any suitable means to that end, is that adjusting element which is controlled by the governor moves to a new position of adjustment by a series of small steps.

This oscillation is not primarily dependent on the condition of the governor and may be a continuous oscillation of constant frequency occurring at all times while the governor is in operation. However, as a mere matter of practical expediency and since the oscillation serves no useful purpose when the governor is not effecting an adjustment, it is sufficient if means be provided whereby the described oscillation is continuously produced during all times when the governor is making an adjustment incident to a speed change of the prime mover or due to a voluntary change in the speed setting thereof.

In hydraulic governors this oscillation is preferably applied directly to the pilot valve. When the pilot valve is oscillated continuously, even while no speed change is occurring or being corrected for, the ports controlled by the pilot valve are so arranged as not to be affected by pilot valve oscillation when the engine is "on speed." This can be accomplished simply by providing an overlap as between the pilot valve and the ports equal to the amplitude of oscillation of the valve. However, as soon as a speed change occurs, the neutral oscillating center of the pilot valve is displaced and the pilot valve thus controls the ports by a series of short impulses. That is, the ports are alternately opened and closed in relatively rapid sequence and the degree of opening becomes less and less as the speed approaches normal, because the center of oscillation of the pilot valve approaches its neutral position.

It has been found that this mode of controlling the pilot valve of a hydraulic governor reduces over-shooting to practically zero without the necessity for any ancillary compensating mechanism.

Another particularly advantageous use of governors of the kind disclosed in this application

is in controlling the pitch of variable pitch aircraft propellers. In this field the governor of the present invention is particularly desirable because of its simplicity and because its autocompensation operation gives greater stability than any other governor known to the art.

It is to be understood that the broad principle outlined in the foregoing is not restricted in its application excepting as defined in the appended claims. However, by way of example, the present application sets forth a preferred embodiment of the invention. In the embodiment illustrated in the accompanying drawing and described in detail in the following description, means are provided whereby the pilot valve is oscillated only when it is displaced from its normal neutral position. In this embodiment it is not necessary to provide an overlap between the pilot valve and the ports controlled thereby, excepting for the usual overlap of one thousandth of an inch or so provided merely for manufacturing tolerance.

Figure 1 of the drawing is a general schematic view of one form of the governor of the present invention.

Figure 2 is a cross-section through the pilot valve housing taken through one of the pilot valve heads and the shaft ports controlled thereby of another form of the governor of the present invention.

In the drawing the numeral 10 designates a shaft which is connectible with the engine or other device or system to be governed. The purpose and effect of the governor system is to maintain a constant velocity of the shaft 10 despite changes in load, in the case of an engine, or of other conditions tending to vary the velocity of the shaft 10. In the alternative the governor serves to maintain a constant degree of speed droop, that is, a predetermined lessening of velocity with increases in load.

The system of the present invention is readily adjusted to varying degrees of speed droop, as occasion may require, including adjustment to zero speed droop, in which case the velocity is maintained constant regardless of the load, within predetermined maximum load limits. Speed droop and the ability to accurately regulate it are principally desirable when multiple prime movers are operating in parallel against a common load and it is desired that the load be uniformly distributed over the several prime movers.

Adjustment of the engine or other device or system being governed is accomplished by rotating a shaft designated 11 in the drawing and in the case of an engine the shaft 11 may be connected with the throttle valve or other fuel control means, as by means of an arm 12. In engine governor operation the load on the engine is reflected in the angular setting of shaft 11.

The fly-weight head and the hydrodynamic fluid system which cooperates therewith is shown at the right in the drawing and comprises a shaft or sleeve 13 rotatable in a support 14 and having a resilient driving connection 15 with the shaft 10 which is driven by the engine being governed. While the resilient driving connection 15 is conventionally used in this art, it may actually be omitted in the governor arrangement of the present invention, which is found to eliminate hunting under even the worst conditions encountered in operation. The shaft 13 is axially bored to receive a pilot valve 16 which has enlarged axially spaced valve heads 17 and 18 and is formed at its upper end with an enlargement 19 for co-

operating to support a plurality of fly-weights 20. The fly-weights 20 are carried by arms 21 which have a common pivot 22 at the outside of shaft 13. The arms 21 are pivotally engaged by links 23 which are also pivotally connected to the enlargement 19 of pilot valve 16 as at 24.

From the foregoing it will be clear that any tendency of centrifugal force to move the fly-weights outwardly, upon increase in speed of the shafts 10 and 13, tends to straighten the toggles which the arms 21 and the links 23 form, and this results in lowering movement of the valve 16 in the bore of the shaft 13. A decrease in speed, evidenced by a tendency of the fly-weights to move inwardly toward the shaft 13, tends to collapse the toggles, and the links 23 accordingly raise the pilot valve 16 in the shaft 13. The pilot valve 16 rotates with shaft 13 but is free to move axially therein, subject to the forced oscillation which will be described hereinafter.

The necessary hydraulic pressure required for operation of the governor is provided by a pump, indicated at 25, which may be arranged to be driven from shaft 10 by a pair of gears 27 and 28. The pump 25 is preferably of the reversible type wherein the output is uni-directional regardless of the direction of rotation of the pump drive means. The pump communicates with a conventional hydraulic pressure accumulator indicated at 29 by means of a conduit 30, and the conduit 30 extends to the support 14 where it communicates with an annular passage 31 formed in support 14 and extending about shaft 13, the latter having a series of peripheral openings 32 whereby conduit 30 has continuous and uninterrupted communication with the space between valve heads 17 and 18.

The pilot valve 16 is further formed at its lower end with a piston head 35 which cooperates with the bore of shaft 13 to form a fluid pressure chamber 36. The chamber 36 is provided in a manner which will presently appear, with a selectively variable degree of hydrodynamic fluid pressure, and such pressure acts against piston head 35 to oppose the downward thrust which the centrifugal action of the fly-weights 20 imparts to stem 16.

The housing 14 has a pair of annular passages 37 and 38 and the shaft 13 has a series of peripherally spaced openings in communication with each of the annular passages. The openings in shaft 13 are designated 39 and 40, respectively, and are normally axially in registry with the valve heads 17 and 18. The passages 37 and 38 have conduits 41 and 42, respectively, which communicate with the upper and lower side of a cylindrical chamber 43 formed in a housing element 44. A power or actuating piston 45 is disposed in chamber 43 and has a piston rod 46 which connects with an arm 47 secured to shaft 11 whereby movements of piston 45 in chamber 43 directly produce rotative movement of shaft 11 and accordingly directly determine and reflect the load setting of shaft 11, or any other variable condition which shaft 11 may serve to adjust.

In a general way, it will be seen from the foregoing that acceleration of shaft 10 will, by increased centrifugal force on fly-weights 20, lower pilot valve 16 and, by lowering of valve head 18, connect conduit 30 with conduit 42. This directs fluid pressure against the lower side of piston 45 and accordingly rotates shaft 11 in a clockwise direction to decrease the fuel setting,

in instances where an engine is being governed, to accomplish the necessary slowing of shaft 10 to the predetermined speed level. As stated in the preamble hereto, the actuating piston may be used for regulating other governor applications.

In the drawing the numeral 50 designates a tapered plug valve which extends across and controls passages 51 and 52 which connect the conduits 41 and 42 with cylindrical chamber 43. The valve 50 is manually adjustable as by means of a handle 53 and its position of adjustment determines the effective cross sectional areas of passages 51 and 52 and this controls the rapidity with which actuating piston 45 responds to changes in velocity of the input shaft 10 and thus permits selective adjustment of the governor characteristic which is termed "rapidity" by those skilled in the art. "Rapidity" is a term denoting the speed with which the actuating piston moves from one fuel setting to another. Instances where best regulation is attained by free and accurate adjustment of "rapidity" are found in the control of prime movers, notably two-cycle gas engines.

It will be noted that a further conduit 60 leads from accumulator 29 and this conduit supplies fluid pressure which serves as the source of adjustable hydrodynamic fluid pressure in chamber 36 beneath piston 35 of pilot valve 16. The hydrodynamic pressure in conduit 60 may be varied by reason of the fact that conduit 60 has an outlet passage portion 61 whose degree of outlet opening may be selectively varied by pivotal movement of an arm 62 which controls the degree of opening of an outlet valve 63 which may vary in form but in the illustrated instance is a common commercial form of valve known in the art as a cock or petcock.

The novel mode of compensation which is here set forth by way of example will now be described. The upper enlargement 19 of pilot valve 16 has fixed thereto a disc 66 which rotates with pilot valve 16 and fly-weights 20. In the forms shown, the disc 66 has a pair of fixed preferably rounded protuberances 67 and 68 which normally pass just between a pair of relatively fixed fingers 69 and 70 during rotation of the fly-weight mechanism and its related structure. Fingers 69 and 70 are supported by an adjusting screw 71 which is reversely threaded as at 72 and 73.

Screw 71 is supported against axial movement in a wall portion 75 of the governor housing by means of a knurled knob 76 at one end of the screw and a collar 78 which may be fixed to screw 71 in any desired manner. Rotation of fingers 69 and 70 may be prevented in any convenient manner as by having their side edges abut the side wall (not shown) in the governor housing. Fingers 69 to 70 may be simultaneously adjusted by manipulation of knob 76 of screw 71 and are normally spaced in such manner that the protuberances 67 and 68 just clear the fingers when the fly-weight mechanism rotates with the engine or other controlled device "on speed."

In normal operation and with the engine operating at a particular load and at a predetermined velocity setting, the hydrodynamic fluid pressure in chamber 36 below pilot valve 16 is such as to balance the centrifugal force acting upon the fly-weights 20 and tending to urge the pilot valve 16 downwardly. Accordingly, the pilot valve remains in neutral position where the valve heads 17 and 18 close conduits 41 and 42, respectively.

However, upon an increase in speed of the fly-weights 20, their added centrifugal force overcomes the fluid pressure beneath pilot valve 16, as determined by the setting of outlet valve 63, and the pilot valve 16 tends to move downwardly to connect fluid pressure conduit 30 with conduit 42 through passages 31, 32, 40 and 38, thus moving actuating piston 45 upwardly in chamber 43. This tendency is arrested by camming engagement of protuberances 68 of disc 66 against finger 70 which forces pilot valve 16 back to neutral position once during each rotation of the fly-weight mechanism. The pilot valve is thus caused to oscillate between its dislocated position, as established by the preponderance of centrifugal force over hydrodynamic pressure in chamber 36, and a neutral position.

Movement of the pilot valve 16 back to neutral position in the second phase of each cycle of reciprocation thereof checks upward movement of actuating piston 45 so that movement thereof to a new position, reflecting a new fuel setting of shaft 11 reduced in value, is approached by a continuous series of short intermittent increments. This results in the attainment of a new more clockwise and therefore reduced fuel setting of shaft 11 without any perceptible or appreciable degree of over-shooting.

Each time the pilot valve 16 is lowered from neutral position to connect conduit 30 momentarily with conduit 42, the fluid expelled from chamber 43 above piston 45 passes through conduit 41 and discharges within the shaft 13 above valve head 17. Such discharged fluid may emerge from shaft 13 through openings 80 which are formed in shaft 13 to accommodate links 23. The fluid so discharged, which is conventionally a lubricant, lubricates the linkage and other moving parts therebeneath.

It will be understood by those skilled in the art that the entire mechanism of the drawing may be housed in a suitable casing of any form whose bottom may be used as a sump for pump 25, whereby all discharge outlets of the system discharge openly into the casing, where they gravitate to the sump for re-use by the pump. The sump is indicated schematically at 81 in the drawing.

A decrease in centrifugal force resulting from a decrease in speed of shaft 10 and fly-weights 20 results in a reverse manner to open the fuel control. In such case, intermittent raising of pilot valve 16 intermittently connects conduits 42 with discharge passages 82 and 83 formed in the shaft 13 and the housing 14, respectively.

For adjusting the speed at which the governor is set to regulate the prime mover, or to vary other desired operating constants in other governor applications, the outlet valve 63 is manually adjustable to control the size of the orifice leading from chamber 36 beneath pilot valve 16 and thereby control the hydrodynamic pressure urging the valve 16 upwardly against the downward urge imparted thereto by the centrifugal force of the fly-weights 20. The valve arm 62 which pivotally controls valve 63 has a link 85 which connects with a lever 86, the latter having a second link 87 pivotally connected thereto. The link 87 is connected to a manually settable arm 88. For the moment we may consider the third pivot of lever 86, designated 90, as fixed, and it will be seen that with pivot 90 fixed oscillation of arm 88 oscillates valve 63 to change its degree of constriction.

As thus far described the governor system is isochronous, its speed being constant, for any

given speed setting, regardless of the load, within usual operating limits. It is desired, however, to selectively adjust the governor system so as to introduce, when desired, adjustable controlled speed droop; that it, a predetermined percentage of reduction in speed for each increase in load. To this end the pivot 90 of lever 86 is connected to a link 92 whose other end pivotally engages a lever 93. The lever 93 has a fulcrum support 94 which is adjustable toward and away from the pivotal connection with link 92 and the other end of lever 93 pivotally engages connecting rod 46 of actuating piston 45.

In operation, the position of the throttle control shaft 11 directly reflects the prevailing load. Accordingly, the arm 47, connecting rod 46, and the actuator piston 45, by their positions, also directly reflect the prevailing load. As shown in the drawing, an increase in load will cause connecting rod 46 to move downwardly. With the fulcrum support 94 set as shown, and with manual speed setting arm 88 in a given position, the pivotal connection between link 87 and lever 86 will serve as a fulcrum for the latter and downward movement of connecting rod 46 will accordingly move link 85 upward and produce the effect of slowing the speed setting by opening the valve 63 in proportion to the increase in load as evidenced by downward movement of connecting rod 46.

This decrease in hydrodynamic pressure beneath the pilot valve 16 causes downward movement thereof and results in pilot valve 16 resuming a normal neutral position at a lower load setting of actuating piston 45 and shaft 11. By this means the engine speed is reduced in proportion to the increase in load and is accordingly a function varying inversely with respect to the load. While the foregoing description of the speed droop control has, for clearness, been described sequentially, it is to be understood that speed droop adjustment occurs continuously and automatically and may occur with and during a governing and compensating cycle caused by a change in speed of the input shaft 10.

Movement of fulcrum support 94 toward the right as viewed in the drawing lengthens the lever arm of the connecting rod 46 and shortens the lever arm of the link 92. This increases the ratio of the connecting rod lever arm to the link lever arm so that the percentage of speed droop is proportionately reduced. If the fulcrum support 94 is moved so that it is coincident with the pivotal connection of link 92 with lever 93, then movement of connecting rod 46 will have no effect on link 92 and the governor will operate isochronously, that is, with zero speed droop. While rarely required, it is obvious that negative speed droop may be introduced by extending the slot in lever 93 so that fulcrum support 94 may be moved to the right of the pivotal connection between link 92 and lever 93.

While the primary consideration in governor operation is usually accurate speed control, such control is only desired within certain maximum load limits and it is desired that such maximum load limits may be freely established and adjusted. It has previously been indicated that the prevailing load is directly indicated by the vertical position of the connecting rod 46. Accordingly a projection 98 is formed on the connecting rod 46 and lies in the path of one end of a lever 99 which is pivotally supported by a screw 100. The other end of lever 99 connects with a link 101 which in turn connects with a valve rod 102 of a valve 103 which is disposed in a cylinder 104.

The cylinder 104 is disposed in conduit 60 and a compression coil spring 105 normally biases valve 131 to the illustrated position where fluid flow through conduit 60 by way of cylinder 104 is unimpeded. However, when the connecting rod 46 moves to a predetermined lower limit position the projection 98 engages the end of lever 99 and it, through link 101, raises valve 103 so that when the predetermined load limit position is reached, valve 103 blocks conduit 60. Vertical adjustment of the position of pivotal supporting screw 100 obviously varies the point at which projection 98 engages lever 99 and accordingly determines the load limit. Such adjustment is effected by changing the axial position of screw 100 with respect to internally threaded fixed supports 107, which is accomplished by manual rotation of knob 108.

While greatest clearness of disclosure is attained by the schematic mode of representation employed in the drawings, it will be understood by those skilled in the art that all of the elements shown diagrammatically in the drawing will be housed in a suitable unitary casing of any desired form and configuration, as has been suggested previously herein. It will further be clear to those skilled in the art that all of the various adjustable fulcrums and valves will have suitable connections accessible from the outside of such casing and that suitable dials for indicating prevailing speed, load limit and speed droop settings will likewise be provided, as is conventional in the art.

Various alternative constructions and methods may be employed to achieve the general aim and purpose of the present invention, namely, elimination of the usual elaborate compensating arrangements and the attainment of what has previously herein been called auto-compensation. These alternatives may comprise merely alternative methods for producing oscillation of the pilot valve or, as will appear, may comprise methods which produce high-frequency intermittent application of fluid pressure energy to change the fuel setting by continuously repeated intermittent increments without oscillation of the pilot valve.

One alternative method of producing continuous oscillation of the pilot valve permits the elimination of the mechanical means shown in Fig. 1 for oscillating the pilot valve. This alternative method requires no new mechanical structure and therefore requires no further illustration. According to this alternative, the particular proportions or setting of the accumulator spring and the accumulator valve are selected to produce continuous automatic oscillation of the pilot valve at a particular frequency. Since, according to this alternative, oscillation of the pilot valve is continuous even when no adjustment is being made, it is necessary to increase the length of the pilot valve heads 17 and 18 to permit them to overlap the ports 39 and 40 by an amount equal to the amplitude of oscillation, so that fluid communication therethrough is established only when the pilot valve is shifted from neutral. An overlap greater than the amplitude of oscillation would normally be provided in practice to allow for the small overlap usually present in any pilot valve which is not oscillating.

By selecting an accumulator spring of a particular natural vibration frequency and by adjusting the accumulator setting in accordance therewith, the pilot valve will oscillate at the

same periodicity as the vibration frequency of the spring. Small pressure changes in chamber 36 produce changes in the effective area of the relief ports of the accumulator in its attempt to maintain constant pressure, and this produces the phenomenon of rapid pressure changes alternation between a maximum and minimum value at a frequency corresponding to the vibration frequency of the accumulator spring.

An alternative construction which entirely avoids the necessity for oscillating pilot valve 16 is achieved by merely omitting the annular passages 37 and 38 of housing 14. By this means the conduits 41 and 42, when pilot valve 16 is displaced from neutral, will communicate with openings 39 and 40 of shaft 13 only when such openings are actually passing the inlet to conduits 41 and 42 formed in housing 14 during rotation of shaft 13 in housing 14. This results in intermittent or pulsating application of pressure against piston 45 in chamber 43 during governor adjustment.

A more refined form of the alternative outlined in the foregoing paragraph is shown in Fig. 2 which is a cross-section through the pilot valve housing of this embodiment taken through one of the pilot valve heads and the shaft ports controlled thereby.

In Fig. 2, the numeral 114 designates a housing corresponding to housing 14 of Fig. 1 and an annular passage 137 in its inner periphery corresponds to annular passage 37 of Fig. 1. Conduit 141 likewise corresponds to conduit 41 of Fig. 1 and the numeral 117 designates a pilot valve head corresponding to the pilot valve head 17 of Fig. 1. The numeral 113 designates a shaft or sleeve corresponding to shaft 13 of Fig. 1 and is provided with a pair of radial passages 139 which are normally closed by the pilot valve head 117 as in Fig. 1.

In the modification now being described, a pair of intermediate bushings 150 and 151 are disposed between shaft 113 and housing 114. Bushing 150 is fixed relative to bushing housing 114 and bushing 151 may be rotated to either of two positions of angular adjustment, one of them being illustrated in Fig. 2. Adjustable bushing 151 is provided with four equally spaced radial passages 153 and fixed bushing 150 is provided with two sets of radial passages. One set comprises a pair of passages 156 which are diametrically opposed. The other set comprises four equally spaced passages 157.

When adjustable bushing 151 is in the position illustrated in Fig. 2 it connects the two passages 156 of stationary bushing 150 with shaft 113 so that the ports 139 of the latter will communicate with annular passage 137 of housing 114 twice during each rotation of shaft 113. In this position of adjustment the four passages 157 of stationary bushing 150 are blocked off by bushing 151. If adjustable bushing 151 be rotated 45° from the position illustrated, it will block off the two passages 156 of bushing 150 and establish communication with the four passages 157 thereof. This causes the passages 139 of shaft 113 to communicate with annular passage 137 four times during each rotation of shaft 113 and thus doubles the frequency of the intermittent application of pressure in chamber 43 during governor adjustment.

What is claimed is:

1. A governor mechanism comprising a pressure chamber and a regulator movable therein in response to differential pressure, a fluid passage communicating with the pressure chamber, a rotary member and a bearing support therefor, a pilot valve movable axially in said support in response to changes in speed in the rotary member, an opening in said rotary member and means on said pilot valve normally blocking said opening, speed responsive means actuable by said rotary member for controlling axial movements of said pilot valve, said fluid passage leading to said rotary member in axial registry with said opening whereby, upon displacement of the pilot valve to expose said opening fluid pressure is transmitted to the pressure chamber in intermittent jets as said opening passes the fluid passage during rotation of the rotary member.

2. A governor mechanism comprising a pressure chamber and a regulator movable therein in response to differential pressure, a pair of fluid passages communicating with the pressure chamber at opposite sides of said regulator, a rotary member and a bearing support therefor, a pilot valve movable axially in said support in response to changes in speed in the rotary member, a pair of axially spaced openings in said rotary member and axially spaced valve means on said pilot valve normally blocking said openings, means for maintaining fluid pressure between said spaced valve means, speed responsive means actuable by said rotary member for controlling axial movements of said pilot valve, said pair of fluid passages leading to said rotary member in axial registry with said pair of openings whereby, upon displacement of the pilot valve to expose either of said openings to the fluid pressure in the space between the valve means differential pressure is applied to the regulator member in intermittent increments as said openings pass the fluid passages during rotation of the rotary member.

3. A governor mechanism for prime movers comprising a pressure operated actuating piston connectible with the prime mover for controlling the supply of an energizing medium thereto, a pilot valve for controlling pressure to opposite sides of said piston, speed responsive means for moving said pilot valve from neutral position upon a change in speed of the prime mover for moving the actuating piston to regulate the supply of energizing medium to the prime mover, a relatively stationary abutment and means projecting from the pilot valve for intermittent engagement therewith upon rotation of the pilot valve, said abutment being outside the path of movement of the projection when the pilot valve is in neutral position but closely adjacent thereto, whereby when the pilot valve is moved from neutral position said projection is intermittently engaged by said abutment to return the pilot valve intermittently to neutral position against the action of the speed responsive means.

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