Feb. 20, 1934. E. S. DENNISON 1,948,098

HYDRAULIC GOVERNING SYSTEM

Filed Nov. 28, 1930

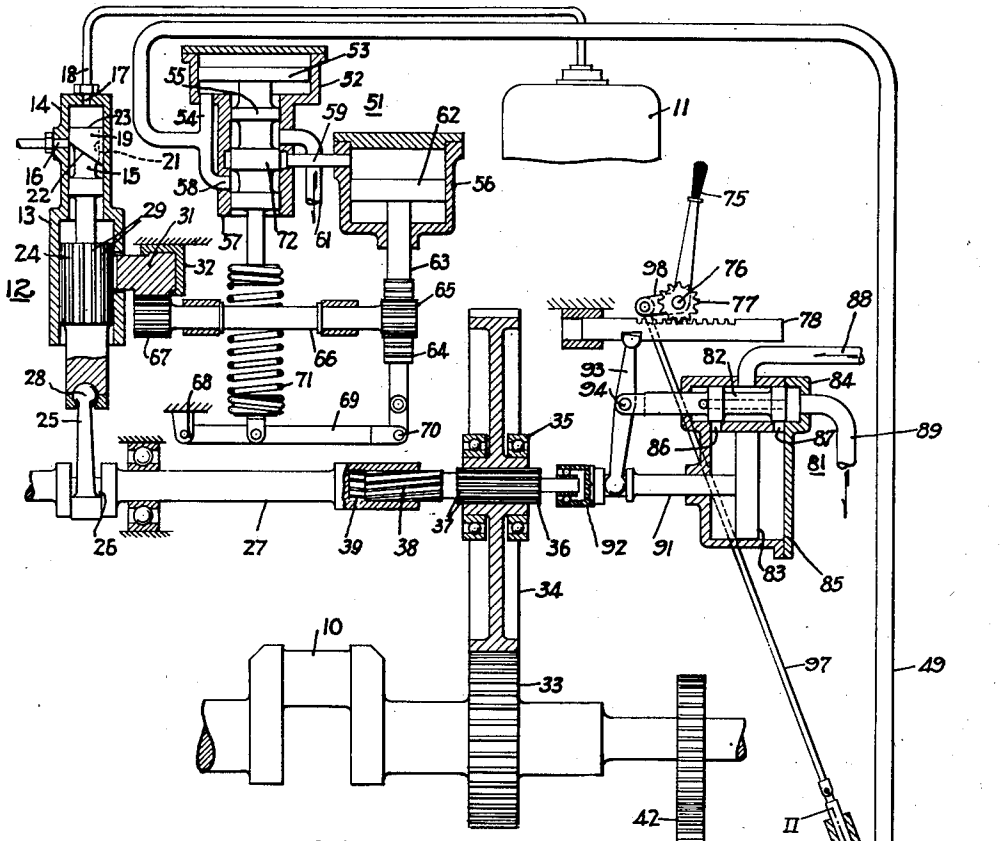


FIG. 1.

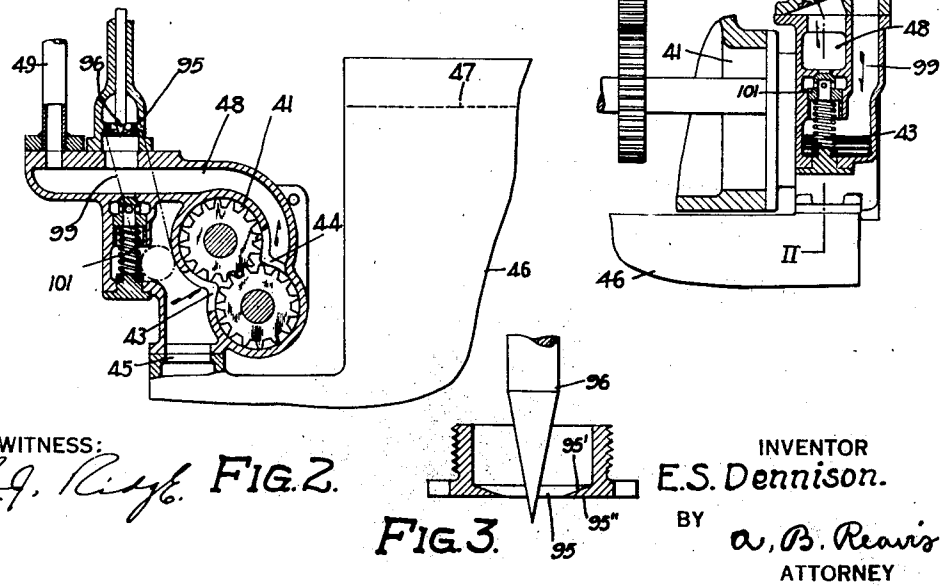


FIG. 2.

FIG. 3.

WITNESS:

INVENTOR

E. S. Dennison.

BY

A. B. Reavis

ATTORNEY

Patented Feb. 20, 1934 1,948,098

UNITED STATES PATENT OFFICE

1,948,098

HYDRAULIC GOVERNING SYSTEM

Edward S. Dennison, Swarthmore, Pa., assignor to Westinghouse Electric & Manufacturing Company, a corporation of Pennsylvania

Application November 28, 1930
Serial No. 498,831

2 Claims. (Cl. 264—14)

My invention relates to governing systems for prime movers and particularly to governing systems of the hydraulic type and still more particularly to governing systems of the hydraulic type as applied to variable speed internal combustion engines and it has for an object to provide a system of the character designated which shall be capable of reliably and accurately maintaining the desired speed of the prime mover under various load conditions.

It has for a further object to provide a governing system of the hydraulic type which shall be so arranged that its effectiveness and accuracy are not impaired or affected by substantial changes in the temperature of the atmosphere in which the prime mover is being operated.

These and other objects are effected by my invention as will be apparent from the following description and claims, taken in connection with the accompanying drawing, forming a part of this application, in which:

Fig. 1 is a diagrammatic view of one form of governing system arranged in accordance with my invention;

Fig. 2 is a partial sectional view of the governor pump and is taken on the line II—II of Fig. 1; and

Fig. 3 is a detail view of a preferred form of orifice and the orifice needle associated therewith.

In United States Letters Patent No. 1,794,518 granted March 3, 1931 to H. T. Herr for an Engine governor, and assigned to the Westinghouse Electric and Manufacturing Company, there is disclosed and claimed a form of governing system of the hydraulic type which is especially applicable to internal combustion engines such as, for example, internal combustion engines of the Diesel type. Engines of this character are frequently employed in railway work to drive locomotives or railway cars, which locomotives or railway cars may be compelled to operate under widely varying atmospheric temperature conditions. It has been found that material changes in the atmospheric temperature may affect the viscosity of the fluid used in the governing system with the result that the action of the governing system may be impaired to some extent.

It has, therefore, been proposed to apply to governing systems of this character some device which shall be responsive to the viscosity of the governing fluid. Such devices act to maintain the characteristic of the governing system constant under variable temperature operating conditions. However, the provision of such devices materially complicates the governing system and, furthermore, such devices are generally unsuited for use in such rugged services as railway transportation work, etc.

I have, therefore, conceived the idea of equipping the pump utilized in such governing systems to develop the fluid pressure with a relatively short recirculating passageway, whereby the same fluid is repeatedly recirculated through the pump at short intervals. In this way, the action of the governing pump is utilized to increase the temperature of the governing fluid to the desired point in a relatively short period of time and to, thereafter, maintain the temperature of the governing fluid at the desired constant. By utilizing such an arrangement, the viscosity of the governing fluid is not affected by varying atmospheric temperature conditions and, as a consequence, the governing system operates at all times with maximum effectiveness.

Referring now to the drawing for a more detailed description of my invention, I show in Fig. 1 an engine crank shaft 10 and an engine working cylinder 11. In the present embodiment, the engine illustrated is of the Diesel type and fuel is periodically supplied to the engine cylinder by a suitable fuel pump, indicated at 12. As shown, the fuel pump 12 embodies a housing 13 having superposed thereon a pump cylinder 14. The latter is provided with a fuel admission port 16 and a fuel discharge port 17, while, disposed within the cylinder, is a pump plunger 15. The discharge port 17 is connected, by means of a suitable conduit 18, to the working cylinder of the engine.

The plunger 15 is provided at its upper end with a slide valve portion 19 arranged to cover and uncover the fuel admission port 16. The slide valve portion 19 is so formed that, upon upward movement of the plunger 15, fuel is displaced from the cylinder through the fuel admission port 16 until such time as the slide valve portion 19 covers the fuel admission port 16, whereupon the fuel in the upper end of the cylinder is trapped and is discharged at a relatively high pressure through the fuel discharge port 17. As soon as the slide valve portion 19 has travelled past the fuel admission port 16, the pressure is released and the fuel is again free to pass through an axially-extending passageway 21 in the slide valve to the lower side of the latter and thence outwardly through the then uncovered port 16. As is apparent, the arrangement is such that, during an intermediate portion of each working stroke of the plunger 15, the slide valve 19 serves to close the admission port 16 and, dur-

ing the time when the latter port is closed, the fuel trapped in the pump cylinder is discharged through the outlet port 17 to the engine cylinder.

Preferably, the slide valve portion 19 has its lower margin or cut-off edge 22 disposed angularly with respect to the leading margin or cut-off edge 23 so that the effective length of the valve portion 19 or, the length of time when the admission port is closed by the valve 19, may be varied by angular adjustment of the plunger 15. The plunger 15 is driven by a crosshead 24 suitably guided for reciprocating movement in the pump body. The crosshead 24 is, in turn, connected through a suitable connecting rod 25 to a crank pin 26 of a pump drive shaft structure, indicated at 27. In order that the crosshead 24 and the pump plunger 15 may be adjusted angularly in the pump housing, the connecting rod 24 is connected to the crosshead by some means such as, for example, a ball and socket joint indicated at 28.

In order to effect angular adjustments of the pump plunger, the crosshead 24 is provided with a plurality of circumferentially-spaced, longitudinally-extending teeth 29 which mesh with corresponding teeth provided on a transversely-extending rack 31. The arrangement is such that the crosshead 24 is free to reciprocate within the pump body while, at the same time, adjustments of the rack 31 effect corresponding angular adjustments of the pump plunger in the pump body. The rack 31 is carried in a suitable guide 32. The pump drive shaft structure 27 is actuated from the crank shaft 10 through a suitable pinion 33, provided on the crank shaft, and a suitable gear 34 provided on the pump drive shaft structure. In the present embodiment, an engine of the four-cycle type is illustrated by way of example and, as a consequence, the pinion 33 and gear 34 are so proportioned that the pump drive shaft structure 27 rotates at one-half crank shaft speed.

The gear 34 is rotatably supported in a suitable bearing 35 arranged to prevent axial displacement of the gear. The pump drive shaft structure 27 embodies a spline shaft 36. The latter is provided with a plurality of axially-extending teeth 37 which, while preventing relative angular movement between the gear 34 and the spline shaft 36, permit relative axial movement therebetween. Also provided on the spline shaft 36 are a plurality of spline teeth 38 inclined with respect to the axis of the shaft, the spline teeth 38 fitting within corresponding internal teeth 39 provided in the shaft section connecting directly with the fuel pump.

The arrangement is such that the pump drive shaft structure 27 is driven in timed relation with the engine crank shaft 10. However, should it be desired to change the timing of the fuel pump relative to the crank shaft, this may be accomplished by moving the spline shaft 36 in an axial direction relative to both the gear 34 and the internal teeth 39, whereupon the teeth 39 and their connected drive shaft section are displaced, angularly, relative to the gear 34 and the crank shaft 10 an amount depending upon the extent of displacement of the spline shaft. By means of such an arrangement, timing of the fuel pump relative to the engine crank shaft may be varied so that the period of injection of fuel into the engine cylinder may be made relatively early or relatively late in the engine working cycle.

Adjustment of both the fuel rack 31 as well as the spline shaft 36 for varying the quantity of fuel injected into the engine cylinder as well as the time of such injection, may be accomplished by means of my improved governing system, preferably arranged in the manner now to be described.

As shown in both Figs. 1 and 2, I provide a pump 41 of the positive displacement type, such as, for example, a gear pump. This pump I term the governor pump and it may be driven from the engine crank shaft through suitable gearing, indicated at 42. The pump is provided with an inlet 43 and an outlet 44. The inlet 43 is connected by means of a passageway 45 with a reservoir, indicated at 46, for governing fluid, for example, oil or lubricant. As is customary, the reservoir 46 is disposed below the crank shaft 10 and the governor pump 41 is also disposed below the crank shaft 10. Also, as is customary, the governing fluid is supplied to the inlet of the pump under a positive pressure. I accomplish this in the present instance by maintaining the level of the fluid in the reservoir 46 at a higher point than the inlet 43 so that the fluid will flow from the reservoir 46 to the inlet 43 by gravity. The normal level of the fluid is indicated at 47 in Fig. 2. Embodied in the housing of the pump 41 is a discharge passage 48 which connects, through a suitable conduit 49, with a fluid pressure relay mechanism, indicated generally at 51.

The fluid pressure relay mechanism 51 embodies a pressure-responsive cylinder 52 containing a pressure-responsive piston 53. The conduit 49 connects, through a branch 54, with the lower end of the cylinder 52 so that the fluid pressure developed by the pump 41 is imposed upon the lower side of the piston 53. The latter is connected to, and actuates, a pilot valve 55, which controls the supply of motive fluid to and the exhaust of motive fluid from a power cylinder 56. The pilot valve 55 is disposed in a suitable cylinder 57 having an inlet 58 connecting with the conduit 49. A passageway 59 affords communication between the pilot valve cylinder 57 and the power cylinder 56, while fluid is exhausted from the relay cylinder 57 through a suitable passageway 61 spaced, axially, from the passageway 59.

The power cylinder 56 embodies a power piston 62 having a piston rod 63 provided with a rack 64. The rack 64 engages a pinion 65 provided on a shaft 66, the latter shaft having another pinion 67 provided at its other end and engaging teeth provided on the fuel rack 31. The arrangement is such that, when the piston 62 is moved in the cylinder 56, the shaft 66 and its pinions 65 and 67 are rotated, effecting linear adjustment of the fuel rack 31. Located at 68 is a fixed fulcrum to which is pivotally secured a follow-up lever 69. The matter is pivotally secured to the piston rod 63, as at 70. Connected between the follow-up lever 69 and the pilot valve 55 is a tension spring 71.

The arrangement of the fluid pressure relay mechanism 51 is such that, when the pressure in the conduit 49 increases, the piston 53 moves upwardly, moving the valve 55 and its valve face 72 so as to uncover the passage 59 and thus permit fluid admitted through the inlet 58 of the relay cylinder to act on the upper face of the power piston 62. Rotation of the shaft 66 and adjustment of the fuel rack 31 to shorten the range of fuel injection is thereby effected. This downward movement of the piston rod 63 continues until the increase in tension or force of the spring 71, the consequence of such down-

ward movement, acts to bring the pilot valve 55 and the piston 53 to their mean equilibrium or cut-off position, that is, the positions illustrated. At such times, no more fluid is admitted to the cylinder 56, and the relay mechanism remains stationary. Upon a further increase in pressure in the conduit 49, the piston 62 is moved further down in the cylinder until such time as the mechanism has reached its limit of travel. On the other hand, should the pressure in the conduit 49 reduce, the spring 71 is then effective to overcome the fluid pressure prevailing on the underside of the piston 53 and the latter is drawn downwardly permitting the pilot valve 55 to uncover the passage 59 and to place the same in communication with the exhaust passage 61. At such times, fluid is discharged from the upper face of the piston 62 and the latter moves upwardly in the cylinder until such time as the tension on the spring 71 has been relieved sufficiently to permit the reduced fluid pressure prevailing on the lower side of the piston 53 to return the latter to its mean or static position. From the foregoing, it will be apparent that an increase in pressure in the conduit 49 is effective to move the fuel rack in one direction and a decrease in fluid pressure in the conduit 49 is effective to move the fuel rack in the opposite direction.

In order to effect changes in the operating speed of the engine, I provide a speed changer lever, 75. The latter is journaled upon a shaft 76 provided with a pinion 77. The latter engages a rack 78 movement of which is effective to shift the spline shaft 36 to change the timing of the fuel pump relative to the engine crank shaft. In order to effect such movements of the spline shaft, I provide a fluid pressure relay indicated generally, at 81, and embodying a pilot valve 82 and a power piston 83. The pilot valve 82 is housed in a suitable cylinder 84 and the piston 83 is housed in a suitable cylinder 85. The pilot valve cylinder 84 communicates with the power piston cylinder 85 through suitable ports 86 and 87 communicating with the respective ends of the power cylinder. Motive fluid from some suitable source is admitted to the cylinder of the pilot valve through an inlet 88 and it is exhausted therefrom through an outlet 89. The power piston 83 is connected by a piston rod 91 and a coupling 92 to the spline shaft 36. The coupling 92 is so formed as to be capable of moving the spline shaft 36 in an axial direction co-extensively with the piston rod 91 while, at the same time, permitting the spline shaft 36 to freely rotate while the piston rod 91 remains in fixed angular position. Pivotally connected between the rack 78 and the piston rod 91 is a follow-up lever 93, which also is connected, at an intermediate point 94, to the relay valve 82.

The arrangement of the fluid relay mechanism 81 is such that when the speed changer lever 75 is adjusted, the rack 78 is moved, moving the pilot valve 82 and permitting motive fluid derived from the inlet 88 to be supplied to one or the other end of the power cylinder 85. At the same time, motive fluid is exhausted from the remaining end of the power cylinder through the outlet 89. The admission and exhaust of motive fluid to and from the power cylinder 85 displaces the piston 83, thereby shifting the spline shaft 36. From the foregoing, it will be apparent that adjustment of the lever 75 results in altering timing of the fuel pump relative to the crank shaft.

Adjustments of the speed-changer lever 75 not only change the timing of the fuel pump relative to the engine crank shaft, but also establishes a new operating speed for the engine. This is accomplished in the most effective manner by providing, in the pump discharge passageway 48, an orifice 95, the flow-area of which may be altered by a suitable orifice needle 96.

As shown particularly in Fig. 3, the orifice 95 is preferably so formed as to embody straight, converging annular surfaces 95′ and 95″ so as to form a relatively sharp-edged restriction through which the tapered orifice needle 96 projects. I have found that a sharp-edged orifice of this character gives better results in a governing system of this type than round-edged or other forms of orifices wherein the orifice edge is relatively blunt because, assuming a fixed setting of the orifice needle, the volume of fluid passing through a sharp-edged orifice is less affected by changes in the viscosity of the fluid than in cases where a blunt-edged orifice is used.

The orifice needle 96 is connected by means of an operating rod 97 and a lever 98 with the shaft 76 so that adjustments of the handle 75 are effective to move the orifice needle 96 to increase or decrease the flow-area of the orifice 95. Fluid discharged through the orifice 95 is directed, by means of a re-circulating passageway 99, directly back to the pump inlet so that the fluid discharged through the orifice is repeatedly recirculated through the pump 41 at short intervals. Also provided in the discharge passageway 48 is a relief valve 101, so that, should the fluid pressure in the passageway 48 become excessive, as may happen momentarily when the orifice needle 96 is rapidly shifted toward a more restricted position, the excess pressure is relieved back to the inlet 43 of the pump. The passageways 95, 48, and 99 are preferably, although not necessarily, formed in the pump casting to minimize the length of the recirculating path from the outlet to the inlet of the pump in order to reduce to a minimum the volume of fluid recirculated and thereby warm up quickly the fluid acted upon by the pump.

From the foregoing description, the operation of my invention will be apparent. It will be noted that, in normal operation, the governor pump 41 supplies fluid, preferably lubricating oil, at a constant pressure to the pressure-responsive cylinder 52 as long as the engine is running at constant speed; and, should there be any tendency for the engine to change its speed for any cause such as a change in load, the pressure developed by the governor pump 41 changes momentarily due to the change in speed of the engine and the relay 51 acts to adjust the fuel rack 31 to compensate for the same. During operation, there is very little flow in the conduit 49, only such fluid being supplied to the conduit 49 as will compensate for leakage and the fluid used in actuating the fluid pressure relay mechanism 51. Hence, substantially all of the fluid discharged by the pump passes through the orifice 95 and the recirculating passageway 99 and returns directly to the inlet 43 of the pump. The same fluid is, therefore, repeatedly recirculated through the pump at very short intervals and, as a consequence, the circulating fluid is maintained at a relatively warm temperature because of friction of circulation and heat absorbed from the pump. In other words, when the engine is operating under relatively low atmospheric temperature conditions, the pump acts to maintain the lubricant at a substantially higher temperature than that of the atmosphere and, conse-

quently, the viscosity of the governing fluid is always maintained such as will insure proper action of the governing mechanism. In starting the engine, if the governing fluid is cold, the governor pump 41, because of the provision of the recirculating passageway 99, heats up the fluid forming a part of the governing system in a very short period of time so that, very soon after the engine is started, the governor system functions with maximum effectiveness. As the temperature of the governing fluid increases, leakage of the lubricant increases until such times as the amount of heat carried away by the leakage equals the heat transmitted to the fluid by the pump after which the temperature of the fluid remains substantially constant. Any make-up fluid required by the system is supplied to the inlet 43 of the pump by the reservoir 46.

During the foregoing operation, it has been assumed that the engine is operating at constant speed. During such times, the handle 75 remains in fixed position as does the relay mechanism 81, the spline shaft 36 and the orifice needle 96. However, the operating speed of the engine may be increased or diminished by adjustment of the lever 75, which adjustment causes the relay mechanism 81 to move the spline shaft 36 and which adjustment also moves the orifice needle 96. If, for example, an increase in the operating speed of the engine is desired, the operating lever 75 is moved to the right, moving the rack 78 to the left. This displaces the pilot valve to the left, causing motive fluid to be admitted to the cylinder 85 through the port 86 and motive fluid to be exhausted from the same cylinder through the port 87. As a consequence, the piston 83 moves to the right, displacing the spline shaft 36 and changing the relative angular position of the fuel pump drive shaft structure 27 relative to the crank shaft 10 so that injection of fuel into the engine cylinder 11 commences at an earlier period in the engine cycle.

Movement of the lever 75 to the right also results in slightly withdrawing the orifice needle 96 from the orifice 95, thereby increasing the flow-area of the orifice 95. As a result, the pressure developed by the pump 41 drops and, this drop in pressure being imposed on the piston 53, the fluid pressure relay mechanism 51 acts to move the fuel rack 31 to increase the amount of fuel injected into the engine cylinder. This adjustment continues until such time as the crank shaft 10 has assumed a sufficient increase in speed so that the pump 41 is capable of again developing, with the enlarged orifice opening, its normal discharge pressure. When the latter occurs, the fluid relay mechanism 51 returns to its mean or static position and the engine continues thereafter to operate at the desired increased speed. If it be desired to decrease the operating speed of the engine, this may be readily accomplished by moving the lever 75 to the left.

It will be noted that, by locating the orifice 95 and the needle valve 96 close to the pump 41, that the fluid discharged through the orifice has a very short path of travel back to the inlet of the pump. As stated heretofore, this permits the pump to act repeatedly upon the lubricant and to maintain the latter at the desired viscosity despite changes in the surrounding temperature.

From the foregoing, it will be apparent that I have devised a form of hydraulic governing system which is capable of operating effectively under widely varying temperature conditions. It will be noted that, in the present embodiment, the recirculating passageway 99 is disposed exteriorly of and apart from the reservoir 46 and that the reservoir 46 supplies to the pump only such fluid as the latter circulates through the conduit 49 as required to compensate for leakage and to motivate the relay mechanism 51. In this way, the pump acts repeatedly upon a relatively small body of liquid and hence the pump is capable of maintaining the liquid at the desired temperature and resultant viscosity. Also, where passageways, including the recirculating passageway, are formed in the pump housing, as shown, not only is a better and more economical structure produced but the length of the recirculating passage is minimized and the structure as a whole is better able to warm the governing fluid. The provision of a recirculating passageway separate and apart from the reservoir 46 is preferably in many instances because of the large capacity of the reservoirs ordinarily provided and the fact that if such a large reservoir forms a part of the recirculating passageway, the fluid circuated through the pump at such relatively long intervals that the pump has very little influence upon the temperature of the lubricant. However, it is to be understood that my invention is not confined to providing a recirculating passageway separate and apart from the reservoir as it contemplates, in its broader aspects, the provision of a recirculating passageway which may or may not include the reservoir and which shall be of such limited volume that the same lubricant is many times recirculated through the pump at relatively short intervals in order that the pump may control the viscosity of the lubricant.

While I have shown my invention in but one form, it will be obvious to those skilled in the art that it is not so limited, but is susceptible of various changes and modifications without departing from the spirit thereof, and I desire, therefore, that only such limitations shall be placed thereupon as are imposed by the prior art or as are specifically set forth in the appended claims.

What I claim is:

1. In a hydraulic governing system for a variable-speed prime mover having a motive fluid admission means and pressure-responsive means for regulating the admission of motive fluid thereto, the combination of a pump for providing a governing fluid pressure varying as a function of the speed of the prime mover and including an inlet and an outlet, conduit means connecting the outlet of the pump and the pressure responsive means, a relatively short and continuous recirculating conduit connecting the outlet of the pump directly to the inlet thereof for conveying fluid as it comes from the outlet of the pump directly back to the inlet and thereby maintain the viscosity of the governing fluid at a low value, speed-changer mechanism including adjustable means for imposing a variable resistance to the flow of fluid through the recirculating conduit so as to vary the pressure of the fluid in the conduit means which is connected to said pressure responsive means, a source of governing fluid, and conduit means connecting the source and the inlet of the pump for supplying only make-up fluid to the system.

2. A hydraulic governing system as claimed in claim 1 wherein the adjustable means for varying the flow of fluid through the recirculating conduit comprises a sharp-edged orifice and an adjustable needle valve movable within the orifice.

EDWARD S. DENNISON.