June 4, 1963 R. H. THORNER 3,092,084

GOVERNOR DEVICE

Filed Sept. 30, 1957 4 Sheets-Sheet 1

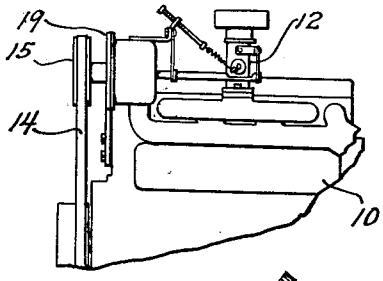


Fig. 1

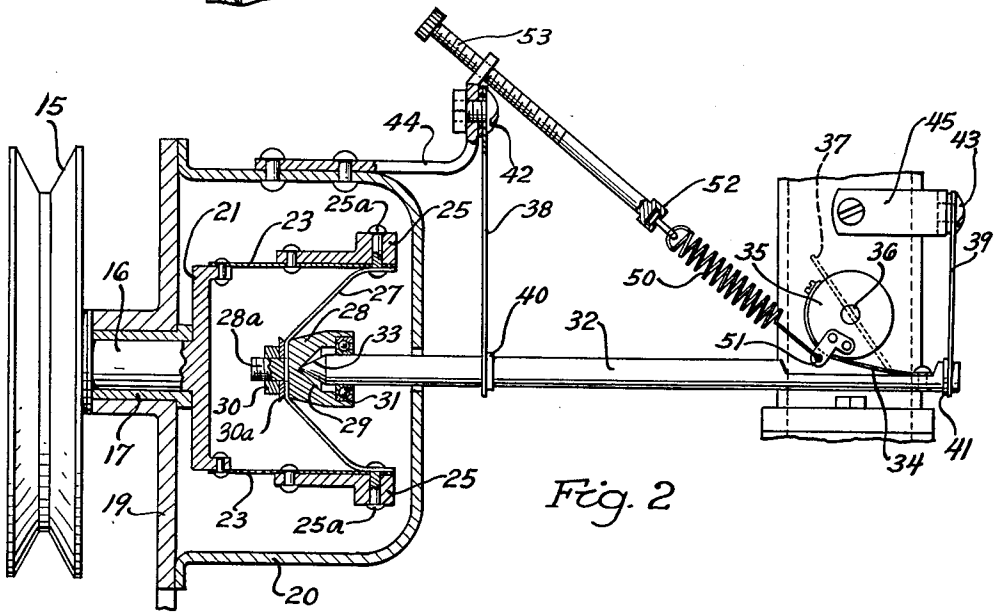


Fig. 2

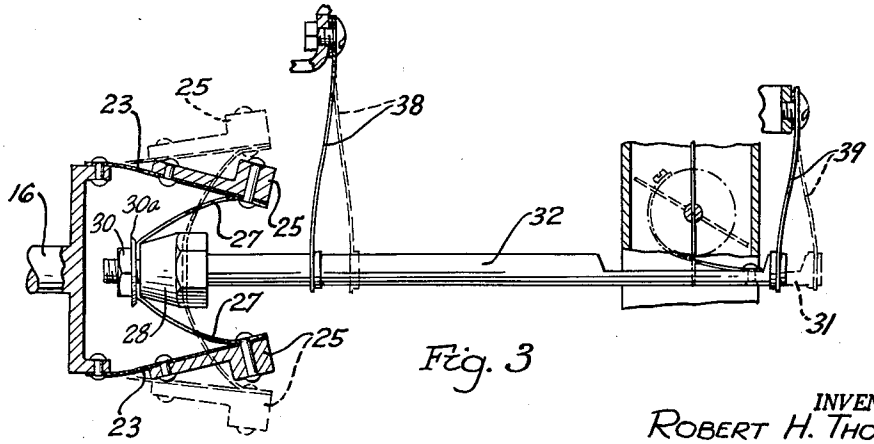


Fig. 3

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GOVERNOR DEVICE

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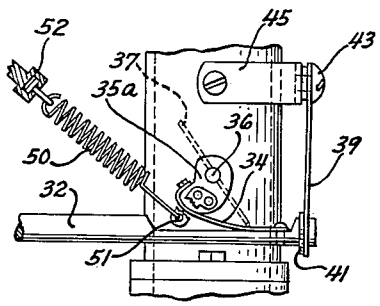


Fig. 4

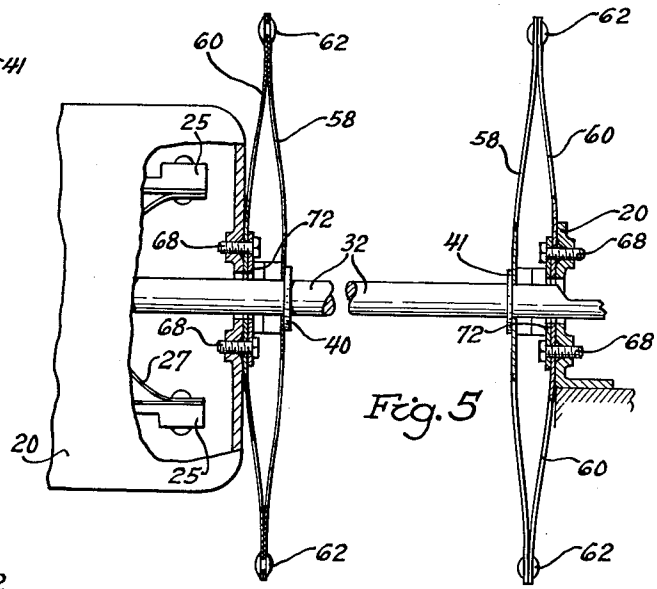


Fig. 5

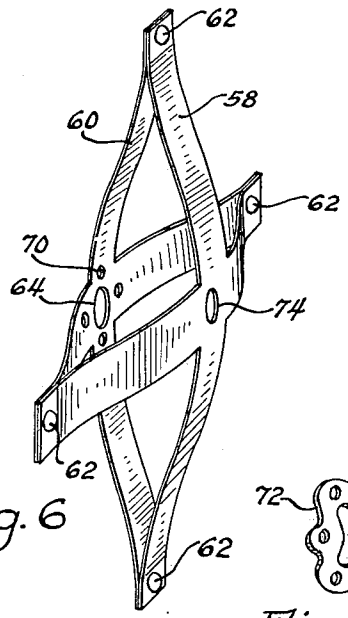


Fig. 6

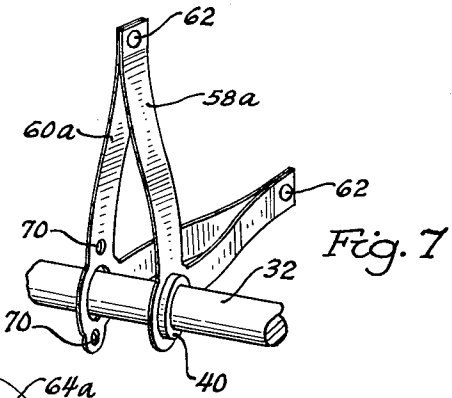


Fig. 7

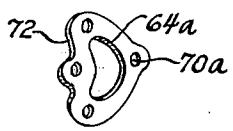


Fig. 6a

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GOVERNOR DEVICE

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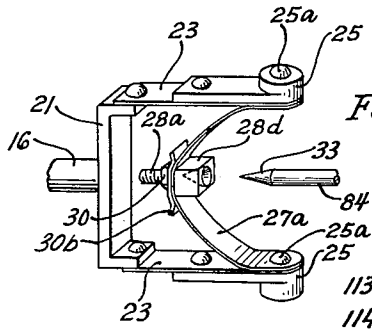


Fig. 10

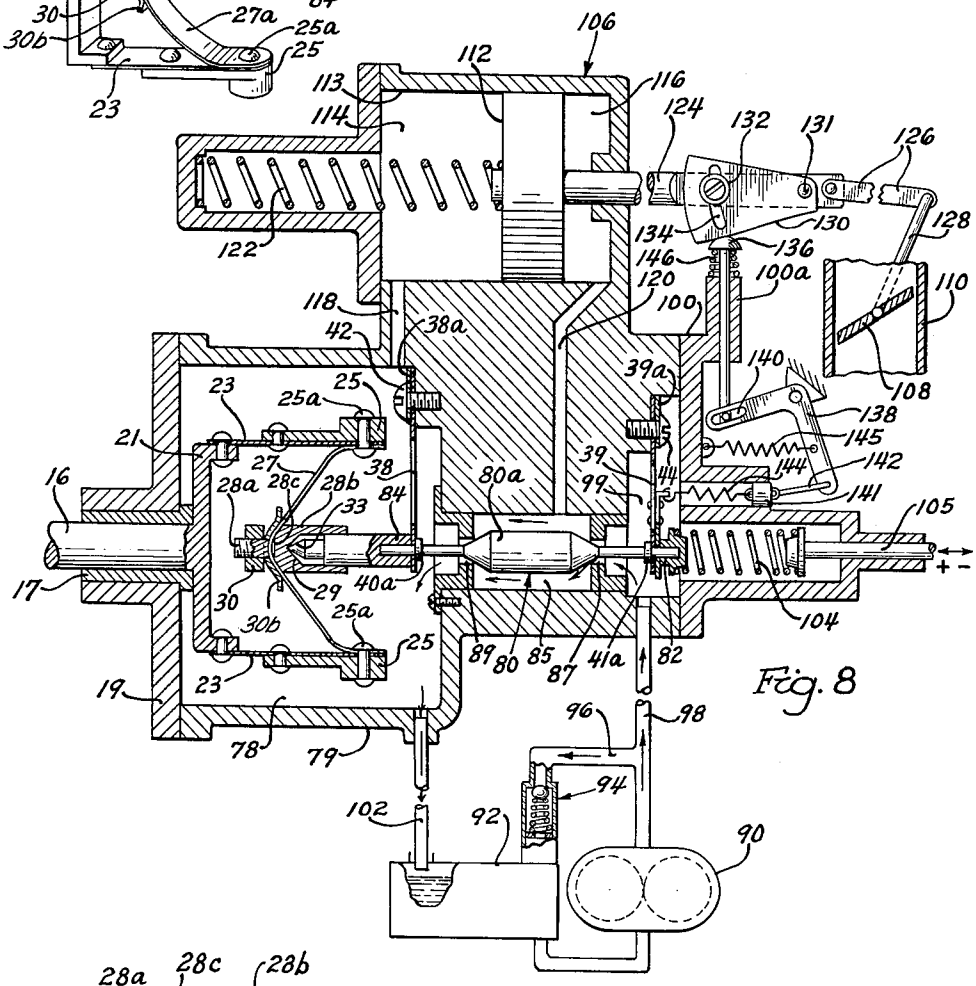


Fig. 8

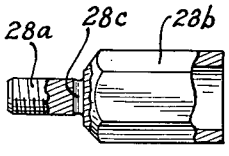


Fig. 9

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GOVERNOR DEVICE

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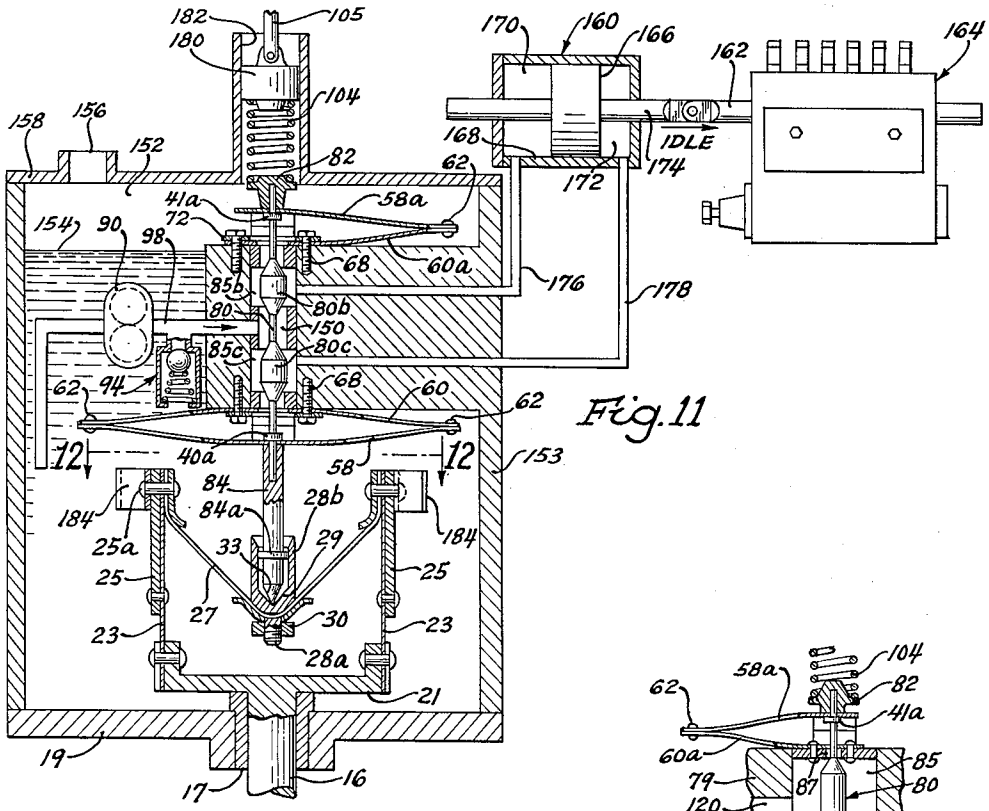


Fig. 11

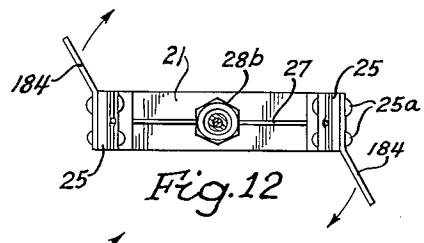


Fig. 12

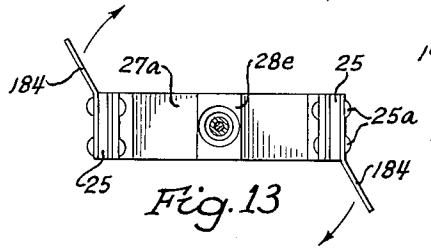


Fig. 13

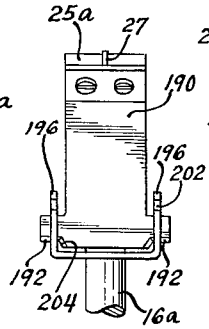


Fig. 15

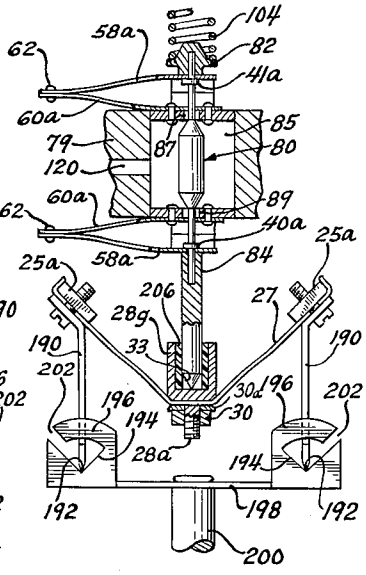


Fig. 14

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United States Patent Office 3,092,084 Patented June 4, 1963

3,092,084

GOVERNOR DEVICE

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Filed Sept. 30, 1957, Ser. No. 687,241
22 Claims. (Cl. 121—42)

This invention relates to governors, and more particularly to governors intended to regulate speed of various machines and devices, or other performance factors thereof, in response to speed. In a number of its aspects the invention relates to an internal combustion engine having improved speed governor or speed regulating means. The present application is a continuation-in-part of my copending application Serial No. 216,822, filed March 21, 1951, for "Governor Device," now Patent No. 2,808,042 and Serial No. 291,381, filed June 3, 1952, for "Fluid Operated Speed Governor."

In constructing governors, particularly for internal combustion engines, the theoretical advantages of centrifugal governors have been well appreciated in the art. Such governors generate or produce centrifugal forces which vary in a certain well known relationship to the speed of the engine, and therefore provide a proper theoretical basis for attaining desired speed regulation, and offer requisite simplicity of operation. However, several practical difficulties encountered in constructing centrifugal governors have greatly hampered application of such governors, particularly for internal combustion engines. As a result thereof, centrifugal type governors have not attained a universal application, and other types of governors are in substantial use in many industrial and automotive applications, particularly with internal combustion engines.

One of the most critical difficulties in constructing centrifugal governors has been found in the presence of friction inherent in the construction of conventional governors of this general nature. The prime necessity for reducing to an absolute minimum all friction or drag in the operation of the governor mechanism has been well understood by those skilled in the art. It should be appreciated that in direct mechanical centrifugal governors, the revolving weights as well as any moving parts connected to and actuated thereby must not only move far enough to sense or indicate the speed change, but the speed differential (difference) must be of such magnitude that the weight force becomes far enough unbalanced from the opposing force (gravity or spring) to let the predominant one actually perform the work of moving the controlled member (such as an engine throttle) against the inherent friction in the governor mechanism.

In governors in which the weights are positively connected to a servo-valve or pilot-valve controlling a fluid circuit which, in turn, operates a pressure responsive member connected to the controlled member, the fluid circuit provides the force which performs the work of moving the controlled member, whereas the weights serve only to produce sensing forces acting on the pilot-valve, which sensing forces are balanced by the forces of suitable resilient means or the like. In such governors, the effects of any inconsistency in the movements in the flyweight mechanism or in a sliding servo-valve created by friction as well as dirt, gum, surface wear, and like causes are amplified at the controlling member because of the amplifying nature of the mechanism itself. From the above, it can be now appreciated that the lower is such friction, the smaller is the required speed differential, and the more sensitive is the governor.

At this point it is important to appreciate the difference between the objectionable results of friction in a governor mechanism, and the action of the forces opposing outward movement of the revolving weights. The latter forces,

usually the force of gravity or a spring force, are consistent in their magnitude. Such a consistent force when balanced may be overcome by the slightest overpowering force, providing there is no frictional resistance. The frictional resistance encountered in conventional direct centrifugal governors or in the pilot valve and flyweights of servo-governors is not consistent in its magnitude due to the difference between the static and dynamic coefficients of friction, dirt, gum, surface wear, close fitting pilot valves, and like causes, and abruptly diminishes after being overcome by a force producing relative movement of parts in frictional contact. Such inconsistency causes delayed response of the governor mechanism to changed conditions, and may produce unstable operation of the governor. Reducing the friction in governors has been done mainly by costly precision machining of the governor parts, use of ball bearings, hardening and grinding or even lapping various sliding surfaces; and for centrifugal servo-governors having sliding "spool" type pilot valves, suitable mechanism is usually provided to rotate the valves. These expedients result in only a reduction of friction and not in the elimination thereof. Such reduction of friction is not only incomplete, but the cost of manufacturing is usually increased to such an extent as to make such governors not practical for many applications.

Another puzzling problem in this art is found in the necessity or great desirability of correlating the operating forces produced by the revolving weights and the forces opposing their outward movement (such as gravity or springs) in such a manner as to produce close regulation at all governed, particularly low, speeds.

One of the main objects of the present invention is to provide an improved speed governor whereby the difficulties and disadvantages of prior governor constructions are overcome and largely eliminated, and a simple, virtually frictionless governor is produced insuring its instantaneously responsive, consistent and dependable performance.

Another object of the invention is to provide an improved governor of the centrifugal type which is more sensitive, gives closer regulation, and is particularly advantageous in automotive and precision governor applications.

A further object of the present invention is to provide an improved governor of the centrifugal type in which the constrained surfaces, or surfaces in sliding contact are virtually eliminated in the suspension of the revolving weights, in the weight forces transmitting mechanism, and in the mechanism actuating the final control member.

A still further object of the present invention is to provide an improved governor of the centrifugal type which is much simpler in construction and is much less expensive to manufacture than conventional governors, and in which governor such advantages inherently result from its construction and not from elimination of any desired mechanism or decreasing the quality thereof.

A still further object of the invention is to provide an improved governor of the centrifugal type which is compact and rugged in construction, is very light, has small number of parts, is not liable to get out of order for a number of years of operation, has no lost motion in its parts, and requires minimum of attention after installation.

A still further object of the present invention is to provide an improved governor of the foregoing nature which inherently reduces in a substantial manner the problem of coordinating in a desired manner the forces produced by the revolving weights and the forces utilized to oppose the same.

A further object of the present invention is to provide an improved servo-governor of the centrifugal type in

which the constrained surfaces, or surfaces in sliding contact are virtually eliminated in the suspension of the revolving weights and in the suspension of the pilot valve which controls the final control member and in the connection of the flyweight device to its actuated member.

It is an added object of the present invention to provide an improved governor of the foregoing nature, particularly but not exclusively for engines, which is simple in construction, safe and dependable in operation, is easy to install and service, and relatively inexpensive to manufacture.

Further objects and advantages of the invention will be apparent from the following description, taken in connection with the appended drawings, in which—

FIG. 1 is a side view of an internal combustion engine including a speed governor embodying the present invention;

FIG. 2 is a view, partly in section, showing the governor of FIG. 1 on an enlarged scale, with governor parts being shown in their respective positions corresponding to the partially open position of the carburetor throttle;

FIG. 3 is a view similar in part to FIG. 2, showing the range of operative mechanical movements of the governor parts and the carburetor throttle;

FIG. 4 is a modification of FIG. 2 in which cam-type means are incorporated to provide an inherent reset or speed-droop-reducing action in a direct mechanical governor;

FIG. 5 shows a modified construction of frictionless suspension of the force-transmitting member or pilot-valve of the governor;

FIG. 6 is a perspective view of the leaf spring support of FIG. 5;

FIG. 6*a* is a perspective view of an anti-torque washer illustrated for use with the leaf spring suspension of FIG. 6 but also applicable for use with the suspension form shown in FIG. 7;

FIG. 7 is a perspective view showing a further modified construction of leaf spring suspension for the force transmitting bar or pilot valve of a fluid servo-governor;

FIG. 8 is a somewhat diagrammatic view showing an adaptation of the invention for a fluid servo-governor having a single-acting "position" type of servo-motor;

FIG. 9 is a perspective view of a detail of the member connecting the flyweight and pilot valve in FIG. 8;

FIG. 10 is a perspective view of a modified form of the flyweight device adaptable in any form of overall governor combination shown herein;

FIG. 11 is a somewhat diagrammatic view showing an adaptation of the invention for a fluid servo-governor having a double-acting type of servo-motor, and optional "anticipating" vanes in the flyweight mechanism;

FIG. 12 is a view along line **12—12** of FIG. 11;

FIG. 13 is a view along line **12—12** of FIG. 11 showing a modified form of flyweight;

FIG. 14 is a side view of a modified flyweight mechanism for use in any of the governor forms shown herein; and

FIG. 15 is an end view of the flyweight mechanism of FIG. 14.

It is to be understood that the invention is not limited in its application to the details of construction and arrangement of parts illustrated in the accompanying drawings, since the invention is capable of other embodiments and of being practiced or carried out in various ways. Also it is to be understood that the phraseology or terminology employed herein is for the purpose of description and not of limitation.

In accordance with the invention I provide an improved governor of the centrifugal type in which the mechanism producing centrifugal or weight forces does not include any surfaces in relatively sliding contact, such as are found in bearings, hinges, sliding collars, and similar expedients wherein friction is inherent and unavoidable. I provide the desired construction preferably by utilizing thin leaf springs of negligible rate or the like, whereby the required change in positions of various members is produced by bending such springs within their elastic limit, and therefore is resisted only by the internal or molecular resistance of such springs to bending; which resistance in thin springs (such as say, .005" thick) is not only negligible but, as is particularly important, is consistent throughout the entire elastic range thereof. I utilize the exceedingly high resistance of such springs to side bending to produce exceedingly rigid constructions yielding only in a single predetermined direction.

I utilize such a construction in the weight suspension mechanism, weight force-transmitting mechanism in some forms of the invention, and the mechanism actuating the final control member, or in any one or two of such mechanisms, as well as in the suspension of the pilot valves of the governors utilizing the same. In the weight force-transmitting mechanism, I preferably use a string-like member, such as music wire, which provides the same frictionless characteristics as a leaf spring, in combination with novel means to transmit the forces from the wire to the actuated member of the flyweight mechanism.

While the present invention is described with reference to an internal combustion engine having a governor regulating the speed thereof, it will be understood that the invention is not limited thereto, and that it is fully applicable to other installations such as steam or gas turbine engines; my improved governor may also be used for regulating other devices in response to speed.

In the drawings, there is shown by way of example governor mechanisms embodying the present invention. Referring specifically to FIGS. 1 to 3, the governor shown therein is illustrated in combination with an internal combustion engine to control the speed thereof. In the embodiment shown, the governor mechanism actuates the carburetor throttle. It will be understood, however, that it may be made to actuate a separate governor throttle operatively mounted within a "sandwich" type governor body, or any other type of engine control member. In FIG. 1, the numeral **10** designates an engine having a carburetor **12**, and a fan belt **14** passed over a pulley **15** mounted on the governor shaft **16** journaled in a bearing **17** provided in a frame plate **19** suitably secured to a governor casing **20**, which is attached to the engine in any suitable manner. The shaft **16** has radial extensions **21** provided thereon to which there are connected with the aid of swingable arms such as leaf springs **23** two centrifugal weights **25**. The weights **25** are connected together with the aid of a generally U- or C-shaped force-transmitting member **27** comprising, in the form shown in FIG. 2, a string-like member such as thin music-wire adapted to convert the radial movements and forces of the weights **25** into axial movements and forces at the central portion of the member **27**. Although the flyweight supports (leaf springs **23**) swing radially toward and away from each other in relation to their parallel positions, they will be considered substantially parallel in all positions for purposes of this disclosure.

A retainer **28**, which in the form shown includes a conical recess **29** for purposes to be described, is secured to the member **27** by suitable means. In the form shown, the retainer **28** is made of hexagonal or square stock and includes a threaded extension **28***a* having a radially-drilled hole therethrough adjacent the left face of the retainer (as will be described in reference to FIG. 9), through which hole the wire **27** is retained. A washer or clamp piece **30***a*, preferably having a semi-cylindrical radial groove to partially fit over the wire **27** (not visible), secures the wire between the washer and the left face of the retainer **28** when a nut **30** is tightened on the threaded extension **28***a*. The other ends of the wire project through radially cross drilled holes in the rivets **25***a* at the leaf springs **23**, so that the wire **27** is positively secured to the leaf springs **23** and to the retainer **28**, as described. The

retainer 28 includes a ball bearing 31 or other anti-friction means for purposes to be described.

The leaf springs 23 as well as other leaf springs hereinafter described are made preferably of SAE spring steel or stainless steel thin sheet stock, or Phosphor-bronze sheet or beryllium copper sheet. Any other sheet material having consistent elasticity may also be used. Strips of such stock have virtually no resistance to bending perpendicularly to their plane but offer very high resistance to bending within their plane, and more important are consistent and frictionless in their action.

Means are provided to transmit the forces produced by the centrifugal weights from the retainer 28 to the carburetor throttle. In the embodiment of the invention illustrated in FIGS. 1–3 said means are exemplified by a bar member 32 having one alignment means comprising a pin with a conical pivot 33 at its end for insertion in another cooperating alignment means comprising the conical recess 29 in the retainer 28 to give a limited area contact or substantially a point contact therewith. The opposite end of the bar 32 is recessed and connected in any suitable manner to one end of a leaf spring or strap or any other suitable band or string-like material 34. The other end of the strap 34 is secured to a crank or guide member 35 mounted on the shaft 36 of the carburetor throttle 37.

The bar 32 is frictionlessly suspended with the aid of two substantially parallel and spaced leaf springs 38 and 39 for limited axial movement and to constrain bar movement in a direction transverse to the bar axis. The lower ends of the springs 38 and 39 are secured to the bar 32 in any suitable manner. In the present embodiment said ends bear against flanges 40 and 41 provided on the bar 32 and are soldered thereto. The upper ends of the springs 38 and 39 are secured by any suitable means as by screws or rivets at 42 and 43 to supporting structures, in the present embodiment to the casing 20 and carburetor body, with the aid of suitable brackets, such as 44 and 45.

The function of the bearing 31 and the pivot 33 of retainer 28 can now be understood. The retainer 28 is maintained in its axial position by the pivot 33 and bar 32 which, in turn, is held in position by the rigidity of the leaf springs 38 and 39 in a direction transverse to the axis of the bar. The pivot 33 is maintained in position at the apex of recess 29 by the opposing force of flyweights 25 and a spring 50 to be discussed. Any slight axial misalignment of the pivot 33 with respect to the axis of the flyweight is readily and frictionlessly accommodated by longitudinal and torsional bending of the wire 27. Such slight misalignment, such as .005 to .010" for example, might occur in production variations in assembly, or due to the swinging action of the leaf springs (which would produce partial torsional bending of the wire when the flyweights are in a rotary position 90° from the position shown).

The anti-friction bearing 31 or other low friction means in cooperation with the frictionless (thrust) pivot 33 is provided for several reasons. First, it avoids torques or twisting forces on the bar 32 which might cause the leaf springs 38 and 39 to buckle. Secondly, it permits a wire 27 of very small diameter (as low as .007 to .014") to be used and still transmit the rotary driving force from the shaft 16 to the retainer 28, which is rotated solely by means of wire 27. Hence, the lower the rotary friction at the retainer 28, the lower is the diameter of wire which may be used, so that slight axial misalignments are accommodated with minimum resistance to wire bending longitudinally and in torsion as the flyweight revolves. The ball bearing may be replaced by a small needle bearing which would positively hold the retainer 28 in axial alignment with the rod 32, which alignment, in the form shown, is produced by the line contact of the balls of bearing 31 and the point contact of pivot 33 held at the apex of recess 29 by spring 50 as will be explained. Or, if desired, an anti-friction sleeve made of hard-chrome plated steel or "Teflon," for example, may be substituted for the ball (or needle) bearing as will be discussed in reference to FIGS. 8, 11 and 14. The inventive concept disclosed herein would be unchanged if the pivot 33 were reversed from the conical recess 29, so that the recess would be carried by the bar 32 and the pivot would be carried by the wire 27.

Means are provided to compensate for the curvature in the travel of the lower ends of the leaf springs 38 and 39, and maintain the pivot and retainer in axial movements. In the present embodiment such means are effected by the provision of spring 39 shorter than the leaf spring 38, and connection of the leaf springs to the bar 32 at a predetermined distance from the pivot end of said bar. Operation of such means can be easily understood from the following explanation: presuming both leaf springs 38 and 39 to be in their middle or vertical positions, the movement of the bar 32 to the extreme right or the left positions would tend to raise the pivot end of the bar because of the arcuate character of the path of the end of the leaf springs. If the leaf springs 38 and 39 were of the same length such undesirable condition would prevail. However since the leaf spring 39 is shorter, the upward movement of the right end of the bar 32 is greater than the upward movement of the bar at leaf spring 38, in consequence whereof the bar rotates for a small angle around its center at the leaf spring 38, moving the pivot 33 downward (as the construction appears on the drawing), thereby compensating for its upward movement. By a proper selection of the respective lengths of the leaf springs 38 and 39 and the distance between them as well as of the distance between the flange 40 and the pivot end of the bar 32, the tendency of the pivot end 33 to deviate from axial movements may be substantially eliminated and substituted by only slight rocking of the retainer 28. The desired axial movement of the pivot 33 may also be obtained by the expedient shown in FIGS. 5, 6 and 7 and hereinafter described. It should be appreciated at this point that frictional resistance at the point of contact of the pivot end 33 of the bar and in the bearing 31 as well as in the bearing 17 of the governor due to its rotation is overcome by the driving force. This frictional resistance is not in the direction of governor controlling or responsive movements, and therefore has no effect whatsoever on the frictionless character of my governor.

A tension spring 50 operatively mounted at the crank 35 is adapted to oppose with predetermined forces the outward radial movements of the weights 25. The force of spring 50 balances the opposing forces developed by the weights 25; and this balance is established at a desired speed of the governor and consequently of the engine. One end of the spring 50 is secured to the circular crank 35 with the aid of a bracket 51, while its other end is connected with the aid of a swivel 52 to an adjustment screw 53 mounted in a suitable supporting structure for convenient reach by the operator. In the present embodiment the screw 53 is mounted in the bracket 44. By turning the screw 53, the spring 50 is stretched or shortened, whereby its tension is adjustably varied. This, in turn, adjustably changes the point at which the balance of the forces mentioned above occurs, and therefore adjustably varies the governed speed of the engine.

It will now be clear in view of the foregoing that in operation of the engine as the engine starts from rest the governor throttle is wide-open, and as the engine is accelerated toward the predetermined governed speed this throttle remains in the wide-open position. When the governed speed is attained the governor throttle is closed sufficiently by the action of the weights until the balance of forces in the governor is established at the governed speed. When such a balance occurs at the desired speed, increase of the engine speed causes further outward movements of the weights 25, axial movement of the retainer

28 and bar 32 to the right, and movement of the throttle 37 in the closing direction, thereby restricting the flow of fuel mixture to the engine until the speed thereof is decreased to its desired value. Decrease in the speed of the engine below the predetermined speed brings about the reverse operation of the governor.

In any speed-regulating mechanism for reasons discussed above, it is highly desirable that the friction of all the elements which move in response to speed changes must be minimized, and at best, entirely eliminated. The total of the elements that move together in response to speed changes might be termed the "sensing-mechanism" and may or may not encompass the entire speed-regulating mechanism depending on its construction or type. In the embodiment shown in FIG. 2, which is a direct-acting centrifugal governor, virtually all elements of the overall governor combination are a part, of the sensing mechanism since they all move together in direct response to speed variations, and friction in any of the moving parts would be highly detrimental to the operation of the governor. It is apparent from the above description that virtually all sliding surfaces and hence all friction in the combination of elements in the present invention are substantially eliminated in its speed-responsive movements; and that such substantial elimination of friction inherently is accompanied by a simplified structure compared to conventional direct-acting speed-regulating mechanisms.

FIG. 4 shows a modification of the governor of FIG. 2 in which the crank or guide member 35 is shaped in the form of a cam 35*a* having a progressively larger effective radius acting on the band or strap 34 as the throttle 37 moves toward its closed position. To understand the effect of this cam-like guide member, it must be appreciated that the governor of FIG. 2 inherently must produce a speed-droop in engine operation since the flyweights must assume a different position to produce a changed throttle position (and at a changed balance force of spring 50). As the speed increases, and bar 32 is moved to the right, as viewed in FIG. 2, the throttle closes until the spring 50 balances the new centrifugal force of the weights. Since this balancing spring force has increased, the governed speed increases slightly (as the load decreases). With the cam-like guide member 35*a* of FIG. 4, as the load decreases and the throttle-opening reduces, the effective radius of member 35*a* increases. Hence, due to the increased torque applied to the shaft 16, the flyweights will move out slightly more to balance the spring 50, which closes the throttle slightly more, and hence reduces the speed droop.

FIGS. 5 and 6 illustrate a modified leaf spring suspension of the bar member which eliminates the effect of the arcuate path of the ends of the strip leaf springs shown in FIG. 2. In this construction two cross-shaped leaf springs 58 and 60 have their corresponding leg-ends secured together by suitable means, as by rivets, as shown at 62. An opening 64 is provided in the central portion of cross-spring 60 for free passage of the bar member 32, and this central portion is secured by suitable means, as by threaded bolts 68, to the governor housing 20 as shown, the bolts being inserted through the holes 70 provided therefor. An anti-torque washer 72 (shown in FIG. 6*a*, which is preferably made of spring steel to provide a lock-washer action, is installed between the leaf spring 60 and the heads of the bolts 68 which are inserted through the holes 70*a* corresponding to the holes 70 of leaf spring 60. The holes 70 are made large enough to accommodate a predetermined amount of leaf-spring movement in all directions of the plane of the leaf spring to facilitate alignment of the pivot 33 (FIG. 2) and bar member 32 in its axial position despite production variations. The holes 70*a* are made with a close fit for the bolts 68, and after the leaf spring 60 is aligned to place the bar member 32 and pivot 33 in proper axial position the bolts are tightened. Then the torque of the bolts is transmitted only to the anti-torque washer (which is held in rotary position by the bolts) whereas the leaf spring 60 receives only thrust, so that the tightening of bolts 68 does not move the leaf spring out of its set position. The central portion of the other cross-spring 58 includes an opening 74 adapted to slip over the bar member 32 and is soldered or otherwise secured to the flange 40 of the bar member.

With the construction above described, each leg of each leaf spring provides rigidity in its plane in a direction transverse to the rigidity provided by the other leg of that leaf spring. Thus movement of the shaft member forces each of the two sets of supporting leaf springs to bend in such manner as to cause the joined leg-ends 62 to move toward the bar member when the leaf springs of each pair spread apart and away from the bar member when the leaf springs of each pair move toward each other. The inherent rigidity due to the arrangement of the leaf springs prevents non-axial movements of the bar member so that its movement is substantially in a straight line.

FIG. 7 shows a construction similar to the supporting leaf springs of FIGS. 5 and 6, except that the leaf springs of FIG. 7 are V-shaped. In FIG. 7, the leaf springs 58*a* and 60*a* comprise in effect two of the adjacent legs of the leaf spring support of FIG. 6 including its central portion. The installation and operation of the form of the leaf spring suspension shown in FIG. 7 is otherwise the same as for the cross-spring support of FIGS. 5 and 6. The form of leaf spring support shown in FIG. 7 requires less space than the form shown in FIGS. 5 and 6, but the cross-spring type of suspension provides more rigidity in preventing non-axial movements. Similar leaf spring supporting means may be made with three legs to be Y-shaped, which would provide rigidity and spring rate between that of the cross-spring and V-spring suspensions.

The governor described above and illustrated in FIGS. 1–3 is of a direct acting type in which the centrifugal unit or device is connected directly to the controlled member, which in the embodiment shown is the carburetor throttle. In a larger number of governor installations where increased forces are required to operate the controlled member, it is necessary to amplify the forces derivable from the centrifugal unit. This may be done in the present invention by introducing a fluid servo-mechanism between the bar member and the engine control member. The pilot valve of the servo-mechanism in this embodiment is part of the bar member or shaft means supported by the leaf springs.

FIG. 8 illustrates a governor of the servo-mechanism type, in which the centrifugal unit is similar to that of the governor of FIGS. 1–3. Accordingly the elements of the flyweight in FIG. 8 are indicated by the same numerals as in FIG. 2. The only difference in the flyweight mechanism from that in FIG. 2 is that the ball bearing has been omitted for reasons to be discussed, and the clamp piece 30*b* is bent to conform to the shape of the retainer 28*b*, shown best in FIG. 9. The flyweight mechanism is encased in a chamber 78 formed by a housing 79 and actuates a pilot valve 80 supported for frictionless axial movements by the leaf springs 38 and 39 in a similar manner to the suspension of the bar member 32 of FIG. 2 by the leaf springs. The leaf springs are secured to the housing 79 by screws 42 and 44 having anti-torque members, such as 38*a* and 39*a*, between the head of the screws and the housing.

The pilot valve 80 includes a valve body member 80*a* and shaft extensions projecting on both sides thereof which includes the flanges 40*a* and 41*a*. The leaf spring 39 is apertured to fit over the extension to the right of the valve body 80*a* and is secured to a spring retainer 82 by suitable means, as by spinning or soldering. The assembly of the retainer 82 and leaf spring 39 is adapted to slip-fit onto the extension to the right of the flange 41*a*. The leaf spring 37 is secured between the flange 40*a*

and a guide member 84, the entire assembly being secured together by suitable means as by soldering and/or pressing member 84 on the shaft extension. The guide member has the pivot 33 at its left end, as viewed in FIG. 8, which turns in the conical recess 29 in retainer 28*b* in the manner described in reference to FIG. 2. The left portion of the guide member is undercut cylindrically so that only the end portion of retainer 28*b* contacts the larger diameter of guide member 84. The guide member is made of any material having high anti-friction properties, such as steel having a hard-chrome plating or a plastic known as "Teflon" (which could be a sleeve pressed onto a steel cylindrical core to form the guide member). With this construction, a bearing is not shown (although it may be used if desired) since much lighter weights 25 can be used, so that the force of the biasing spring 104 would be less. With these lower forces the retainer construction shown in FIG. 8 may be used if desired.

The valve body 80*a* is suspended for operative movements within a chamber 85 and includes a pair of opposite conical faces for cooperating with a fluid inlet orifice 87 and an outlet orifice 89 to form a pair of variable restrictions which vary oppositely and gradually as the valve body 80*a* is moved. Any fluid, such as a liquid or gas, having sufficient pressure (or vacuum) to operate a servo-motor (to be described) may be used to flow through the fluid circuit of the governor. In the form shown in FIG. 8, a liquid pump 90 supplies the working fluid such as oil under pressure from a reservoir 92. The pump may be an integral part of the governor and driven by shaft 16, or it may be a separate pump, or the pressure fluid may be taken from any available source, such as from the oil pump of the engine being controlled for example. In any case it is highly desirable that the fluid pressure at the entrance of the pilot valve at orifice 87 be regulated to substantially a constant value so that pressure variations acting on the pilot valve do not adversely disturb the speed-functionality of the flyweight forces. In the form shown, a conventional spring-loaded ball-type regulator 94 is provided in a by-pass conduit 96 from the pump outlet to the reservoir so that a substantially constant predetermined pressure is supplied to the valve at orifice 87. However any type of pressure regulator may be used, such as the series diaphragm-regulator shown in my co-pending patent application, Serial No. 683,318, filed September 11, 1957, for a "Fluid Pressure-Sensing Governor Mechanism."

In the fluid circuit of the form shown in FIG. 8, oil flows through a conduit 98 into a chamber 99 formed by a cover or end piece 100, and the fluid passes through orifice 87 into chamber 85 and out through orifice 89 to chamber 78 under substantially atmospheric or constant pressure, and out to the reservoir 92 through a drain conduit 102. Due to the modulating characteristics of the valve body in relation to orifices 87 and 89, the pressure in chamber 85 gradually and smoothly varies from the pressure in chamber 99 to the pressure in the drain chamber 78 as the pilot valve and valve body gradually is moved from its extreme left to its extreme rightward position as viewed in FIG. 8. The pilot valve 80 is a particular type in a fluid-bleed principle of pressure-control system in which two restrictions in series are provided and at least one of the restrictions must be varied. This principle provides a pressure in chamber 85 which varies gradually or is "modulated" as a function of the movement of the pilot valve. In the form shown, both the restrictions in series at orifices 87 and 89 are variable oppositely which permits modulation of the pressures from the source value of pressure to the outlet value of pressure. However, if desired, a single variable orifice may be used with the other orifice or restriction being of fixed size, but this arrangement will not provide the full range of pressure modulation as will two variable orifices in series.

A spring 104 acts on the right end of the pilot valve 80 to oppose and balance the forces produced by the flyweight mechanism. The free end of spring 104 is carried for frictionless movements by retainer 82, thereby being supported by the leaf springs 38 and 39. A speed control member 105 is adapted to act on the right end of spring 104 to provide manual adjustment thereof for selecting the governed speed. The construction details of leaf springs 38 and 39 in cooperation with pilot-valve 80 and anti-torque members 39*a* to provide perfect axial alignment with respect to orifices 89 and 87, respectively, are disclosed in detail in my Patent No. 2,737,165 for a "Governor Device," issued March 6, 1956.

A servo-motor 106 is provided to actuate the throttle 108 of a carburetor 110, or any other control member of an engine or machine to be governed. The servo-motor 106 includes a pressure responsive member, such as a piston 112 sliding in a cylinder 113 to form with the cylinder end walls two fluid chambers 114 and 116. Chamber 114 is subjected to drain pressures through a conduit 118, and chamber 116 is subjected to the pressure in chamber 85 through a conduit 120. A spring 122 urges the throttle toward its idle position and opposes the force on piston 112 due to the pressure in chamber 116.

In operation of the construction thus far described, as the engine speed increases, the pilot valve 80 and its valve body 80*a* assume a position rightwardly, as viewed in FIG. 8, until the adjusted force of the spring 104 balances the force produced by the flyweights. The pressure in chamber 85 controlled by the valve body as above-described decreases as the pilot valve moves rightwardly, and this pressure is transmitted to chamber 116 to produce a corresponding position of the piston 112 in balance with spring 122. However, due to the large area of the piston 112 and the relatively high pressure of the fluid available in supply conduit 98, the forces produced by the piston are very large in relation to the initial signal forces produced by the centrifugal flyweights 25, so that any friction in the fuel control means and its linkages is negligible percentagewise. If the load on the engine reduces to increase engine speed, the flyweights expand and the pilot valve 80 moves further rightwardly, as shown, to reduce the pressure in chamber 116, and the spring 122 then moves the throttle in a clockwise direction to restore a stable governed speed when the force of spring 122 again is in balance with the force of piston 112. If the engine load changes to reduce the engine speed, the governor mechanism acts in a reverse manner to restore a stable speed. Any desired governed speed may be selected by manual adjustment of control member 105.

It can be seen that all the cooperative movements of the centrifugal flyweights, the pilot valve 80 and its valve body 80*a*, and the free end of the spring 104 are completely frictionless in responding to minute changes of speed. Hence the governor is extremely sensitive and produces excellent and consistent performance. Furthermore, the central portion of the flyweight assembly at the retainer 28*b* and pivot 33 is constrained for substantially frictionless axial movements by the leaf spring supporting members 38 and 39 while maintaining the frictionless characteristics of the flyweight mechanism. In the servo-type speed-regulating device shown in FIG. 8, the speed "sensing-mechanism" as previously referred to comprises all the elements that must move together directly in response to speed variations, which elements are the flyweight mechanism, the pilot valve 80 with its leaf spring supports 38 and 39, the free end of biasing spring 104, and the connection of the flyweight mechanism and pilot valve at the pivot 33 and retainer 28*b*. Thus, in the form shown in FIG. 8, the entire "sensing-mechanism" is sub-

stantially free of all sliding-surface contact in its speed-responsive movements and hence is frictionless; and these relatively small movements (and forces) of the sensing-mechanism are amplified to any desired amount by the servo-motor which follows precisely the movements of pilot valve **80** due to the aforesaid frictionless characteristics of the entire sensing-mechanism. Such desirable characteristics of the "sensing-mechanism" are accomplished with a structure which is inherently simpler and less critical and costly to manufacture than conventional fluid governors with servo-mechanisms.

The servo-governor mechanism shown in FIG. 8 and described thus far may be used quite satisfactorily in many applications without the added mechanism shown and to be described. The governor as described thus far will produce a speed droop for the same reason and in the same manner as explained in reference to the mechanical governor form shown in FIG. 2. This is true since a small change in speed is required to re-position the pilot valve throughout its small travel (about .040 to .050" for example). This characteristic is due to the small change in force of spring **104** as the pilot valve is moved by the flyweights whose force must change to balance the corresponding changed force of spring **104**, which in turn, can only be done by a corresponding slight change in speed. Similarly, since a variable pressure (with pilot valve travel) is transmitted to chamber **116**, the piston will assume definite and different positions in balancing the variable force of spring **122** as the piston force varies. Thus as the engine loads and the speed tends to reduce, and the weights **25** move inwardly to move pilot valve **80** leftwardly, the force of spring **104** reduces slightly from its previous balanced position, so that a slightly less force from the flyweights is required to balance the reduced spring force, which reduction in flyweight force is only produced by a reduction in speed so that a "speed droop" is produced as the load is increased. However, in the governor described thus far, the speed-droop or regulation can be very low, such for instance as 100 r.p.m., due to the frictionless nature of the device.

In some governor installations, better regulation or less speed-droop, or even isochronous operation is required. The added mechanism shown in FIG. 8 is a "reset" or feedback device to provide less speed-droop or even isochronous operation. Referring to FIG. 8, piston **112** actuates a shaft member **124** for operating throttle **108** through a link **126** and a throttle lever **128**. The shaft **124** carries a cam **130** hinged at a pin **131** for angular adjustment and secured to the shaft by a lock screw **132** projecting through a slot **134**. The cam actuates a cam follower member **136** carried and guided by an extension **100***a* of the end piece **100**. The follower actuates a bell crank **138** by a pin carried on the follower and riding in a slot **140** of the bell crank to actuate a plunger **141** slidable in a cylindrical bore by means of a link **142**. A light reset extension spring **144** connects the plunger to leaf spring **39** or to a thin plate secured thereto by rivets, or the spring can be connected in any other manner to apply a reset force on the pilot valve. The springs **145** and **146** are provided to balance the force of the fluid pressure in chamber **99** acting on the plunger and to insure that the follower rides on the cam.

The operation of the reset mechanism is as follows: When the load decreases tending to increase engine speed which effects a movement of piston **112** rightwardly tending to reduce the opening of throttle **108**, the cam moves the follower and bell-crank **138** to increase the force of reset spring **144**. This additional spring force accompanying movements of the servo-motor piston and throttle tends to move the pilot valve further to the right, as viewed in FIG. 8, thereby reducing the speed-droop slightly from the normal speed-droop that would be obtained without the reset mechanism. The cam may be set angularly to select the amount of "reset" from the normal speed-droop to produce a lower net regulation or even isochronous operation with complete stability. An increase in engine load would produce a reverse operation of the reset mechanism.

FIG. 9 is a perspective view of the retainer **28***b*, wherein the hole **28***c* for the wire **27** is clearly shown. FIG. 10 is a modified flyweight construction wherein the elements thereof which are the same as in the flyweight device of FIG. 8 are so numbered. In the device of FIG. 10, a leaf spring or other band or strap member **27***a* has replaced the wire **27**, and retainer **28***b* of FIG. 8 has been replaced with a retainer **28***d* in FIG. 10. The main structural difference in the two flyweight devices is that the retainer **28***d* in FIG. 10 contacts the pivot **33** only at the apex of the conical recess of the retainer (shown dotted), and the retainer does not include a bushing or bearing extending to contact guide member **84**, since the rigidity of leaf spring **27***a* to cross movements assists the point contact of the pivot. However, with this form of flyweight, the pivot must be aligned more accurately than with the flyweights of FIGS. 2 and 8, because any misalignment when the flyweight is in a rotary position 90 degrees from that shown in FIG. 8 would tend to buckle the leaf spring member **27***a*, whereas the wire member **27** can provide a twisting or torsional spring action at this time. However, if the member **27***a* is made of nylon or other plastic or cloth material, then the retainer of the type shown in FIGS. 2 and 8 may be used.

FIG. 11 is a modified form of the servo-governor of FIG. 8 in which the single-acting servo-motor of FIG. 8 is replaced with a double-acting servo-motor in FIG. 11, so that the pilot valve action is of the "excursion" type as opposed to the "position" type of action in the form shown in FIG. 8. Referring to FIG. 11, the construction of the pilot valve **80** differs from that in FIG. 8 in that two valve body members **80***b* and **80***c* are carried by the pilot valve **80** and disposed for pressure modulating movements within chambers **85***b* and **85***c*, respectively. Each valve body, as in FIG. 8, includes two conical faces for modulating the apertures of corresponding orifices for controlling the flow in and out of the respective chambers for each valve body. Pressure liquid is supplied from pump **90** which is driven by the shaft **16** or separate drive means and the liquid is regulated by a pressure regulator such as the by-pass regulator **94** in the same manner as in the form of FIG. 8. The regulated pressure liquid is directed to a chamber **150** wherein the liquid divides into two branches of the fluid circuit. In one branch of the circuit the liquid flows upwardly, as viewed in FIG. 11, past the inlet orifice into chamber **85***b* and through the outlet orifice into a large reservoir chamber **152** in a casing **153** which is filled by liquid to the level **154** through the filler opening and air vent **156** in a cover **158**. In the second branch of the fluid circuit the liquid flows downwardly, as viewed in FIG. 11, through the inlet valve for chamber **85***c* and through this chamber and then out to the reservoir chamber through the outlet orifice for chamber **85***c*. The leaf spring support illustrated for the lower end of pilot valve **80**, as viewed in FIG. 11, is the form shown in FIGS. 5 and 6 (cross-springs) and the elements thereof are numbered accordingly. The leaf spring support illustrated for the upper end of the pilot valve is the form shown in FIG. 7 (V-spring), and the elements thereof are numbered to correspond to the disclosure of FIG. 7. The centrifugal flyweight device is substantially the same as that shown in FIG. 8 but the unit is mounted vertically and immersed in the liquid in the reservoir. The elements of the flyweights corresponding to similar elements of flyweight mechanisms in FIGS. 2 and 8 are so numbered.

A servo-motor **160** is provided to actuate the engine control member, which in the form shown in FIG. 11 is a rack **162** of a conventional fuel pump **164** of a diesel engine. The servo-motor includes a piston **166** slidable in a cylinder **168** and forming two variable pressure chambers **170** and **172**. The piston carries a shaft **174** extend-

ing through the end walls of the cylinder and suitably connected to the rack for actuation thereof. The chamber **170** communicates with chamber **85***b* by a conduit **176**, and chamber **172** communicates with chamber **85***c* by a conduit **178**.

In operation of the governor shown in FIG. 11, assuming the pilot valve is in a neutral position, as the speed increases due to a load decrease, the weights **25** move outwardly to move the pilot valve upwardly which increases the opening of the outlet orifice for chamber **85***c* and the inlet orifice for chamber **85***b*, while decreasing the opening of the inlet orifice of chamber **85***c* and the outlet orifice of chamber **85***b*. This action gradually increases the pressure in chambers **85***b* and **170** while simultaneously decreasing the pressure in chambers **85***c* and **172**. As a result of the difference in pressures in the servo-motor chambers **170** and **172**, the piston **166** moves the rack **162** in a reduce-speed direction until the speed has returned to the initial speed prior to a change in load. Then at this same speed, the pilot valve and flyweights have been returned to their original positions existing prior to a load change, and the servo-motor stops moving. When the load increases, the governor acts in a reverse manner to restore the set speed. The governed speed is adjusted or selected by manual setting of the member **105** which positions a plunger **180** slidable in a bore **182** to vary the force of spring **104**. Since the pilot valve has returned to its original position after a speed correction following a change in load, the force of balancing spring **104** has not been changed in the final balanced position of the pilot valve following the load change, so that the governed speed is unchanged. Since the servo-motor is double-acting, the piston would continue to move if the pilot valve did not return to its original position. Thus the pilot valve operates by intermittent "excursions" from its mid-position, and since it always returns substantially to the same position with the same spring force, the governor inherently tends to be isochronous. The reset mechanism of FIG. 8 could be added to the governor of FIG. 11 in order to produce a speed-droop, if desired. By contrast, the governor of FIG. 8 is of the "position" type which inherently produces a small speed-droop, and the reset mechanism is provided to achieve isochronous operation, when desired.

The servo-motor of the forms shown in FIGS. 8 and 11 is of the piston type, but any type of pressure responsive member such as a diaphragm or bellows may be used as desired, or the vane-type servo-motor illustrated in my Patent No. 2,661,728 issued December 8, 1953, may be used satisfactorily. Also any type of fluid whether liquid or gas under pressure or vacuum may be used in any of the servo-type governors shown herein.

The cross-type of leaf spring support has been illustrated at the flyweight-end of the pilot valve to provide more rigidity at this end. It is understood that both supports might be cross-springs, or V-springs, or the spring supports shown in FIG. 8. Also in the flyweight of FIG. 11, the guide member **84** includes a flange or land **84***a* to maintain the retainer **28***b* in axial position. This land may be separate from the guide member or a part thereof and is made of anti-friction material, such as "nylon" or "Teflon" for example. Also the rivets securing leaf springs **23** and wire **27** are shown to clamp the springs between suitably contoured guide members to prevent bending beyond the elastic limit.

A novel feature illustrated in the form of governor in FIG. 11 and applicable in any form shown herein, if desired, is the provision of what might be relatively termed "anticipating" means in the flyweight mechanism. Referring to FIGS. 11 and 12, the latter figure being a top view of the flyweight, a pair of vanes **184** are secured to the flyweights as shown and include portions at an angle to the direction of rotation of the vanes. Thus as flyweight and vanes revolve in a clockwise direction, as shown in FIG. 12, any resistance of the oil acting at an angle to the vanes tends to move the vanes outwardly. When the flyweight rotates at a fixed or governed speed, the oil rotated in a toroidal path generated by the vanes tends to revolve at substantially the same velocity as the vanes. But when the speed suddenly increases due to a decrease in load, the flyweight speed is instantly and simultaneously increased mechanically by the drive shaft **16** whereas the inertia of the oil produces a lag thereof. As this very instant when the first speed-change signal is applied to the governor, the lag of the oil increases the oil-impact force on the vanes to produce a temporary and instantaneous additional starting force and movement of the flyweight outwardly (more than produced by centrifugal force) until the oil is "dragged" by the vanes to the new speed. This extra flyweight movement at the very start of the governor correcting action tends to give a fast and extra correction of the control member when it is needed most. Since the vanes produce the temporary extra flyweight force at the immediate start of the correcting cycle, the vane action has been termed "anticipating" with respect to the rest of the cycle. As soon as the oil is revolving at nearly the same speed of the vanes again, the extra force is substantially dissipated. The amount and length of time the extra force is applied depends on the size and angle of the vanes.

Since there is a slight resistance to the vanes even at a uniform speed there will be a very slight temperature effect in the governor, a higher engine speed being produced when the oil viscosity is decreased as the temperature increases. This slight temperature effect can be compensated for by making the leaf springs supporting the pilot valve **80** of thermostatic bimetal, and arranging the springs to apply an upward force on the pilot valve as the oil temperature increases, thereby increasing the pressure in chamber **170** and decreasing the pressure in chamber **172** to move the rack **162** slightly toward the low speed position and compensate for the tendency to increase the speed due to the temperature (viscosity) effect on the vanes. For this purpose, all four of the bimetal leaf spring-pilot valve supports shown in FIG. 11 would be assembled such that their legends would tend to turn in an upward direction, as viewed in FIG. 11. If oil having a small change in viscosity with temperature changes is used, temperature correction may not be necessary.

FIG. 13 is a top view of the flyweight mechanism in FIG. 11 modified to the form shown in FIG. 10 in which the leaf-spring-connecting member **27***a* is illustrated, but the retainer **28***e* is the same as shown in FIG. 11 except that it is made of rectangular stock instead of hexagonal stock in order to provide a better guide surface of the leaf spring **27***a* in bending.

FIG. 14 shows another form of flyweight illustrated actuating the single-acting pilot valve shown in FIG. 8 for which corresponding elements are so numbered, but suspended by the V-springs shown in FIG. 7, although any of the leaf spring supports shown herein may be used. FIG. 15 is an end view of FIG. 14. In FIGS. 14 and 15 the flyweight comprises the rigid metallic (or other firm material) swingable arms **190** having a knife-edge **192** disposed to rock in V-shaped openings **194** in four supports **196** (two for each arm) bent upwards from a driving plate **198** secured to a shaft **200** to be rotated thereby. The slots **202** are provided in each of the four supports to permit insertion of the arms **190** before assembling the entire flyweight together. The knife-edge **192** projects outwardly through the openings **194** as shown best in FIG. 15; and the arms have a downward extension **204** which have angular sides adjacent the supports **196** to prevent movement of the arms in their own planes and to prevent rubbing or other friction in the swingable movements of the arms. The arms **190** are connected together by wire **27** having its ends securely clamped between the flyweights **25***a* and the ends of the arms **190**, which are shown bent at an angle, by suitable

means as by the screws which are shown threaded into weights 25*a*. The central portion of the wire carries the retainer assembly which is substantially the same as described in previous forms of the governor in which the same elements are so indicated by the same numbers as for the other forms shown herein. In this form of retainer, the pivot 33 (which might be of semi-spherical or other similar shape, if desired) abuts against a flat end of the bore of the retainer instead of a conical recess, and a relatively long anti-friction sleeve 206 (such as "Teflon") is pressed into the bore of retainer. The inner bore of the sleeve has ample clearance for the guide member 84 but has sufficient length to align the retainer axially. Any of the other retainer constructions may be used with this flyweight if desired. In operation, as shaft 200 rotates, the flyweights swing outwardly to move the pilot valve upwardly against the speeder spring 194, which spring force holds the knife edges 192 of the arms against the bottom of the V-openings 194. The operation of the entire governor is the same as described in reference to FIG. 8. This flyweight device will include the lowest spring rate in its construction of any of the flyweight means disclosed herein, although any of the flyweight devices illustrated herein may be used in any of the governor forms shown, as well as in other speed-operated devices.

What I claim is:

1. In a rotary-speed-responsive mechanism, rotatable driving means, two diametrically opposite and substantially parallel frictionless swingable supporting members, each of said members including weight means and including frictionless hinge means imparting rigidity in one direction for operatively connecting said weight means to said driving means for rotation therewith and to provide frictionless radial angular movements of said members and said weight means transverse to said first direction, frictionless flexible means operatively connected to said two weight means and rotatable with said driving means and having a substantially central portion movable axially of said driving means in response to said radial movements of said weight means, said flexible means being generally U-shaped and having sufficient length to render said central portion capable of substantial radial movements away from said axial position, means axially movable substantially along the axis of said driving means and contacting said axially movable portion of said flexible means for actuation thereby, frictionless swingable means imparting rigidity in a direction transverse to said axis acting to support said axially movable actuated means and constrain same for frictionless movements thereof in said axial direction and to prevent surface contact of said actuated means during operational movements thereof, first alignment means operatively connected to said flexible means at said substantially central portion thereof for rotation with said driving means, said axially movable actuated means including second alignment means cooperating with said first alignment means, said two cooperating alignment means including means preventing relative movements thereof in a radial direction and utilizing said rigidity of said swingable means to constrain said first alignment means for frictionless axial speed responsive movements with said actuated means during rotation of said first alignment means and to maintain said central portion and said weight means in substantially concentric relation with respect to said axis.

2. The combination of elements defined in claim 1, wherein each of said hinge means for said two supporting members includes a leaf spring member, one end of each of said leaf spring members being operatively secured to said driving means, and the freely swingable ends of said leaf spring members including said weight means.

3. The combination of means defined in claim 1, in which said flexible means includes a wire element, and said means preventing relative movements including cooperative pivot means to transmit axial forces from said wire element to said axially movable actuated means and to reduce the rotary friction of said contact thereof as said first alignment means rotates in relation to said movable actuated means, and one of said two alignment means including a cylindrical bore, the other of said two alignment means including a cylindrical portion rotatable relative to said bore to produce said constraining action.

4. In a rotary-speed-responsive mechanism, rotatable driving means, two diametrically opposite and substantially parallel frictionless flexible leaf spring supporting members imparting rigidity in one direction, one end of each of said leaf spring members being operatively secured to said driving means, and the swingable free ends of each of said leaf spring members including weight means for rotation with said driving means and to provide frictionless radial angular movements of said members and said weight means transverse to said first direction, a wire element operatively connected to said two weight means and rotatable with said driving means and having a substantially central portion movable axially of said driving means in response to said radial movements of said weight means, means axially movable substantially along the axis of said driving means and contacting said axially movable portion of said wire element for actuation thereby, frictionless swingable means imparting rigidity in a direction transverse to said axis acting to support said axially movable actuated means and constrain same for frictionless movements thereof in said axial direction and to prevent surface contact of said actuated means during operational movements thereof, first alignment means operatively connected to said wire element at said substantially central portion thereof for rotation with said element and said driving means, said axially movable actuated means including second alignment means cooperating with said first alignment means, said two cooperating alignment means including means preventing relative movements thereof in a radial direction and utilizing said rigidity of said swingable means to constrain said first alignment means for frictionless axial speed responsive movements with said actuated means during rotation of said first alignment means and to maintain said central portion and said weight means in substantially concentric relation with respect to said axis.

5. In a rotary-speed-responsive mechanism, rotatable driving means, two diametrically opposite and substantially parallel frictionless swingable supporting members, each of said members including weight means and including frictionless hinge means imparting rigidity in one direction for operatively connecting said weight means to said driving means for rotation therewith and to provide frictionless radial angular movements of said members and said weight means transverse to said first direction, frictionless flexible means operatively connected to said two weight means and rotatable with said driving means and having a substantially central portion movable axially of said driving means in response to said radial movements of said weight means, said flexible means being generally U-shaped and having sufficient length to render said central portion capable of substantial radial movements away from said axial position, means axially movable substantially along the axis of said driving means and contacting said axially movable portion of said flexible means for actuation thereby, frictionless swingable means imparting rigidity in a direction transverse to said axis acting to support said axially movable actuated means and constrain same for frictionless movements thereof in said axial direction and to prevent surface contact of said actuated means during operational movements thereof, first alignment means operatively connected to said flexible means at said substantially central portion thereof for rotation with said driving means, said axially movable actuated means including second alignment means co-

operating with said first alignment means, said two cooperating alignment means including means preventing relative movements thereof in a radial direction and utilizing said rigidity of said swingable means to constrain said first alignment means for frictionless axial speed responsive movements with said actuated means during rotation of said first alignment means and to maintain said central portion and said weight means in substantially concentric relation with respect to said axis, and said swingable means including leaf spring means disposed to provide substantially straight-line movements of said guide means.

6. The combination of means defined in claim 5, and said leaf spring means comprising a pair of substantially parallel leaf spring members, one end of each of said leaf spring members being operatively secured to a fixed portion of said mechanism, the freely swingable ends of said leaf spring members being operatively connected to said actuated axially movable means, one of said leaf spring members being shorter than the other of said leaf spring members, the relative length and spacing of said leaf spring members and said axially movable member being such that cooperative axial movements of said first alignment means and said second alignment means is directed along substantially a straight line as said movable member swings on said leaf spring members.

7. The combination of elements defined in claim 5, and said leaf spring means comprising a pair of leaf spring members, each of said spring pair having at least two legs substantially transverse to each other in substantially the same plane, the central portion joining the two legs of one of said spring pair being operatively secured to a stationary portion of said mechanism, and the central portion joining the two legs of the other of said spring pair being fixed to said actuated movable member, the other corresponding two leg-ends of both of said springs being fixed together, and a second similarly arranged spring pair connected to said movable means at a distance from said first spring pair and substantially parallel thereto, whereby the speed responsive movements of said movable actuated means and said second alignment means are constrained for frictionless movements in substantially a straight line.

8. In a rotary-speed-responsive mechanism, rotatable driving means, two diametrically opposite and substantially parallel frictionless swingable supporting members, each of said members including weight means and including knife-edge means imparting rigidity in one direction for operatively connecting said weight means to said driving means for rotation therewith and to provide frictionless radial angular movements of said members and said weight means transverse to said first direction, frictionless flexible means operatively connected to said two weight means and rotatable with said driving means and having a substantially central portion movable axially of said driving means in response to said radial movements of said weight means, said flexible means being generally U-shaped and having sufficient length to render said central portion capable of substantial radial movements away from said axial positions, means axially movable substantially along the axis of said driving means and contacting said axially movable portion of said flexible means for actuation thereby, frictionless swingable means imparting rigidity in a direction transverse to said axis acting to supporting said axially movable actuated means and constrain same for frictionless movements thereof in said axial direction and to prevent surface contact of said actuated means during operational movements thereof, first alignment means operatively connected to said flexible means at said substantially central portion thereof for rotation with said driving means, said axially movable actuated means including second alignment means cooperating with said first alignment means, said two cooperating alignment means including means preventing relative movements thereof in a radial direction and utilizing said rigidity of said swingable means to constrain said first alignment means for frictionless axial speed responsive movements with said actuated means during rotation of said first alignment means and to maintain said central portion and said weight means in substantially concentric relation with respect to said axis.

9. In a rotary-speed-responsive mechanism, rotatable driving means, two diametrically opposite and substantially parallel frictionless swingable supporting members, each of said members including weight means and including knife-edge means imparting rigidity in one direction for operatively connecting said members to said driving means for rotation therewith and to provide frictionless radial angular movements of said members and said weight means transverse to said first direction, a wire element operatively connected to said two weight means and rotatable with said driving means and having a substantially central portion movable axially of said driving means in response to said radial movements of said weight means, means axially movable substantially along the axis of said driving means and contacting said axially movable portion of said wire element for actuation thereby, frictionless swingable means imparting rigidity in a direction transverse to said axis acting to support said axially movable actuated means and constrain same for frictionless movements thereof in said axial direction and to prevent surface contact of said actuated means during operational movements thereof, a first alignment means operatively connected to said wire element at said substantially central portion thereof for rotation with said driving means, said axially movable actuated means including second alignment means cooperating with said first alignment means, said two cooperating alignment means including means preventing relative movements thereof in a radial direction and utilizing said rigidity of said swingable means to constrain said first alignment means for frictionless axial speed responsive movements with said actuated means during rotation of said first alignment means and to maintain said central portion and said weight means in substantially concentric relation with respect to said axis.

10. In a frictionless centrifugal flyweight device, rotatable driving means, two diametrically opposite and substantially parallel frictionless swingable supporting members imparting rigidity in one direction, each of said members including weight means and including frictionless hinge means for operatively connecting said members to said driving means for rotation therewith to provide frictionless radial angular movements of said members and said weight means transverse to said first direction, wire means operatively connected to said two weight means and rotatable with said shaft means and having a portion movable axially of said driving means in response to radial movements of said weight means, a movable element operatively connected to said axially movable portion, means to constrain said movable element and said axially movable portion for said axial movements and to maintain said weight means in substantially concentric relation with respect to said axis.

11. In a rotatable centrifugal flyweight device immersed in a liquid for rotation therein, radially movable centrifugal flyweight means rotating with said device in said liquid to produce forces varying as a function of the rotary speed of said device, said flyweight means including frictionless means to support same and provide frictionless movements thereof in said radial direction and including frictionless means disposed to transmit said forces, said flyweight means including vane means disposed at an angle to the direction of rotation of said flyweight means, said vane means projecting into said liquid sufficiently to cause at least a portion of said liquid to rotate with said vane means, said vane means acting on said liquid at said angle to produce a temporary additional radial movement and force of said flyweight means when the rotary speed of said flyweight means changes and

the inertia of said liquid prevents a corresponding change in the velocity thereof, and said temporary movement and force gradually dissipating as the liquid velocity approaches the changed angular velocity of said flyweight means.

12. The combination of means defined in claim 11, and temperature-sensitive means responsive to the temperature of said liquid disposed to act on said flyweight device to compensate for the effects on said vanes of viscosity changes of said liquid produced by changes of temperature thereof.

13. In a governor for a rotating machine having a control member to regulate automatically the speed of rotation thereof, the combination of means to actuate said control member comprising, rotatable driving means, two diametrically opposite and substantially parallel frictionless swingable supporting members, each of said members including weight means and including frictionless hinge means imparting rigidity in one direction for operatively connecting said members to said driving means for rotation therewith and to provide frictionless radial angular movements of said member and said weight means transverse to said first direction, frictionless flexible means operatively connected to said two weight means and rotatable with said driving means and having a substantially central portion movable axially of said driving means in response to said radial movements of said weight means, said flexible means being generally U-shaped and having sufficient length to render said central portion capable of substantial radial movements away from said axial position, bar means axially movable substantially along the axis of said driving means and contacting said axially movable portion of said flexible means for actuation thereby, frictionless swingable means imparting rigidity in a direction transverse to said axis acting to support said bar means and constrain same for frictionless movements thereof in said axial direction and to prevent surface contact of said bar means during operational movements thereof, first alignment means operatively connected to said flexible means at said substantially central portion thereof for rotation with said driving means, said bar means including second alignment means cooperating with said first alignment means, said two cooperating alignment means including means preventing relative movements thereof in a radial direction and utilizing said rigidity of said swingable means to constrain said first alignment means for frictionless axial speed-responsive movements with said bar means during rotation of said first alignment means and said weight means and to maintain said central portion and said weight means in substantially concentric relation with respect to said axis, an operative connection between said bar means and said control member whereby axial movements of said bar means cause a speed-controlling movement of said control member, and substantially frictionless biasing means opposing the forces produced by said weight means, whereby the speed-responsive movements of said weight means, said flexible means, said alignment means, said bar means, and said biasing means are substantially frictionless and thereby respond to minute changes in the forces produced by said weight means to actuate said control member and maintain within a predetermined variation a desired value of the speed of said machine.

14. A governor for an engine having a throttle rotatably mounted on a shaft, a centrifugal device driven by said engine and including frictionless means to produce forces which vary as a function of the rotary speed thereof, an actuated member operatively connected to said device and responsive to said forces produced thereby, frictionless biasing means to oppose the forces produced by said centrifugal device, a guide member mounted on said throttle shaft and having a cam-like surface of different radii at different angular positions of said throttle shaft, and means including a frictionless flexible tension connecting member connected to said actuated member and wrapped around said guide member surface to translate movement of said actuated member into predetermined rotational movement of said throttle.

15. In a governor device for a rotating machine having control means to regulate the speed of rotation of said machine, the combination of means for effecting movement of said control means comprising, a pressure responsive member adapted to be connected to said control means for actuation thereof, a fluid circuit having a flow of fluid therethrough and communicating with said pressure responsive member, valve means including a movable valve member in said fluid circuit for controlling pressure therein acting on said pressure responsive member for effecting movements thereof, frictionless swingable means imparting rigidity in one direction acting to support said valve member for substantially frictionless movements in a direction transverse to said first-named direction by maintaining said movable valve member suspended within the fluid controlled thereby completely free of surface contact other than fluid contact during operational movements thereof, a centrifugal device driven by said rotating machine and including substantially frictionless flyweight means to produce forces which vary as a function of the rotary speed thereof and acting on said supported valve member to effect speed responsive movements thereof for producing movements of said pressure responsive member with amplified forces for moving said control means, substantially frictionless biasing means opposing the forces produced by said flyweight means, said elements and means being so arranged that the cooperative speed-responsive movements of said supported valve member, said flyweight means, and said biasing means are substantially frictionless and thereby respond substantially instantaneously and consistently to minute changes in said forces produced by said flyweight means caused by changes in the rotary speed of said machine to effect speed-controlling movement of said control means for maintaining within a predetermined variation a desired value of the speed of said machine, and reset means operably movable with said pressure responsive member and said control means and producing forces acting on said supported valve member to control said variation of the desired value of speed.

16. The combination of means defined in claim 15 and variable force second biasing means acting on said pressure responsive member to oppose the forces produced thereon by fluid pressures in said fluid circuit acting on said pressure responsive member, the force of said second biasing means varying as a function of the movement of said pressure responsive member, said valve means controlling pressures on only one side of said pressure responsive member, the configuration of said supported valve member in relation to the said other valve member being such as to produce modulated pressures acting on said pressure responsive member to effect movement thereof as a function of the movement of said supported valve member, and the force of said frictionless biasing means varying as a function of the movement of said supported valve member, whereby the movement of said supported valve member varies as a function of the speed of said machine to effect corresponding movement of said control means also as a function of the speed of the machine.

17. The combination of means defined in claim 15, in which said frictionless biasing means comprises a spring, and said reset means includes a second spring modifying the effective force of said first named spring.

18. In a frictionless centrifugal flyweight device, rotatable driving means, two diametrically opposite and substantially parallel frictionless swingable supporting members imparting rigidity in one direction, each of said members including weight means and including frictionless knife-edge hinge means for operatively connecting said members to said driving means for rotation therewith to provide frictionless radial angular movements

of said members and said weight means transverse to said first direction, wire means operatively connected to said two weight means and rotatable with said shaft means and having a portion movable axially of said driving means in response to radial movements of said weight means, a movable element operatively connected to said axially movable portion, means to constrain said movable element and said axially movable portion for said axial movements and to maintain said weight means in substantially concentric relation with respect to said axis.

19. The combination of elements defined in claim 1, and in which said flexible means includes a wire element, and in which said first alignment means includes means for securing same to said wire element, whereby slight non-axial movements of said movable actuated means produced during said radial angular movements of said weight means are accommodated without friction by torsional and longitudinal frictionless bending of said wire element.

20. The combination of elements defined in claim 10, and pivot means including a pin member and cooperating recess member to effect said operative connection of said movable element with said axially movable portion of said wire means, means to connect one of said last-named two members to said wire portion, whereby slight non-axial movements of said movable actuated means produced during said radial angular movements of said weight means are accommodated without friction by torsional and longitudinal frictionless bending of said wire element, and said means to constrain said movable element including a pair of spaced substantially parallel leaf spring members to support said movable element for frictionless movements in response to said movements of said axially movable portion and said connected member.

21. The combination of means defined in claim 18, and said driving means including an element having two parallel pairs of projecting supports, said pairs being diametrically opposite to each other and projecting substantially at right angles to said element, each of said supports including a generally V-shaped aperture therein comprising one cooperating portion of said knife-edge hinge means and an open slot leading to each said aperture, each of said swingable supporting members including a knife-edge comprising another cooperating portion of said knife-edge hinge means disposed to rock in said V-shaped apertures to provide said frictionless radial angular movements of said swingable member, each of said swingable supporting members including opposite side projections extending through said apertures, respectively, of each of said parallel pairs of said supports and disposed to be inserted into said apertures through said slots.

22. The combination of means defined in claim 1, and a controlled member operated by said mechanism, a pressure responsive member operatively connected to said controlled member to effect actuation thereof, a fluid circuit having a flow of fluid therethrough and communicating with said pressure responsive member, and said axially movable actuated means including valve means supported for frictionless movements by said swingable means in said fluid circuit for controlling pressure therein acting on said pressure responsive member for effecting speed-responsive movements thereof and of said controlled member.

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