

July 29, 1941.

H. G. ADLER

2,250,982

GOVERNOR

Filed Feb. 20, 1939

4 Sheets-Sheet 1

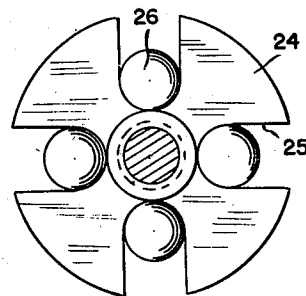
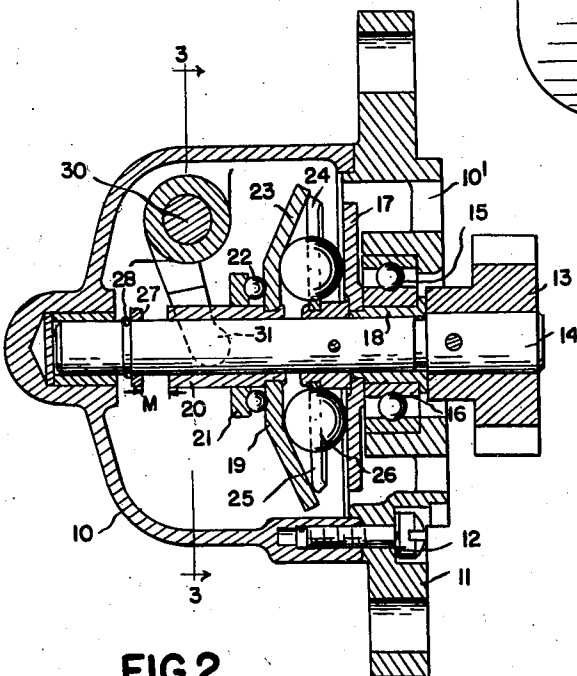
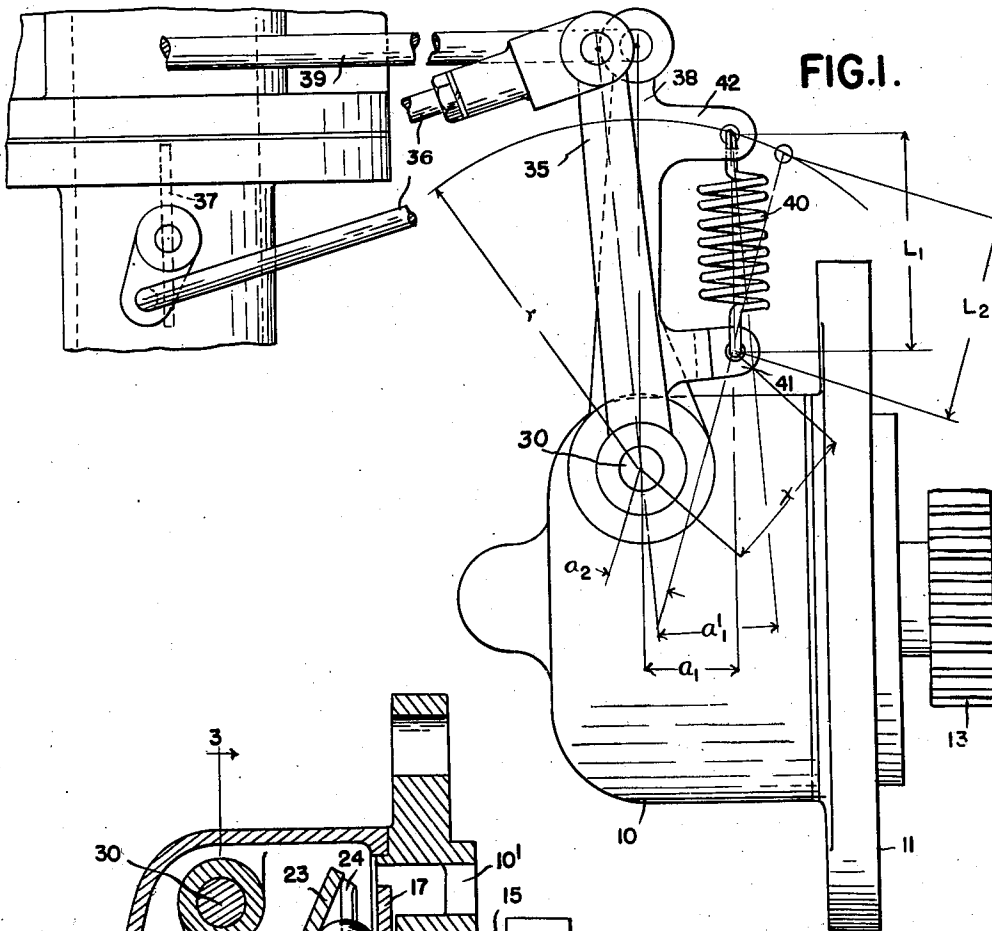


FIG. 2.

FIG. 4.

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FIG. 6.

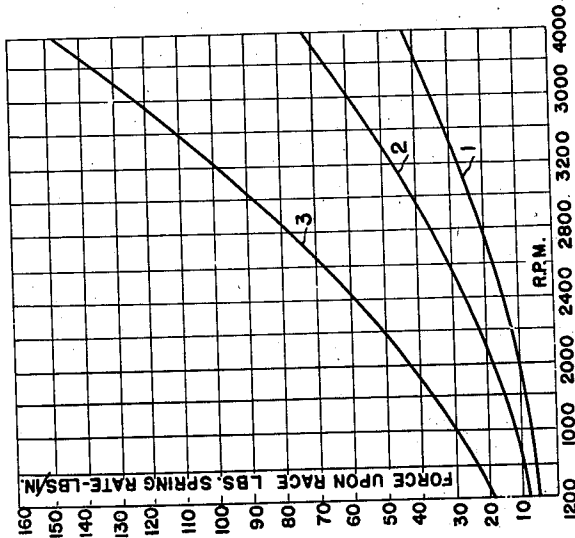
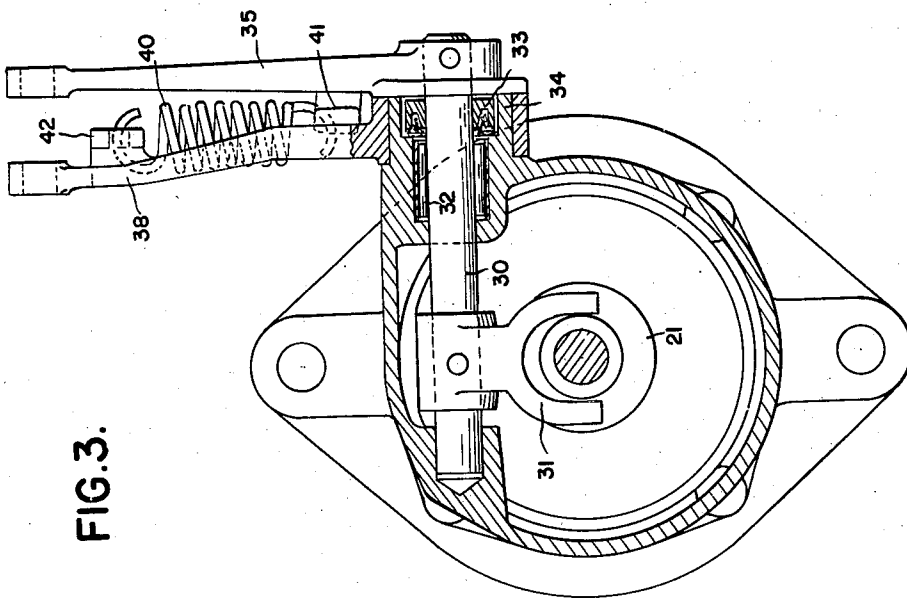


FIG. 3.



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GOVERNOR

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4 Sheets-Sheet 3

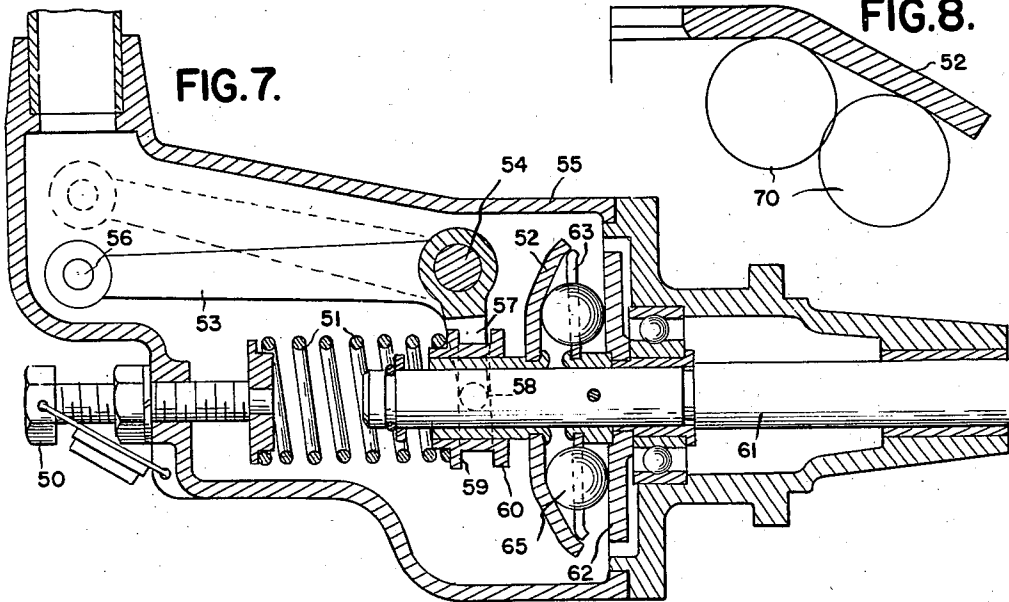


FIG. 5.

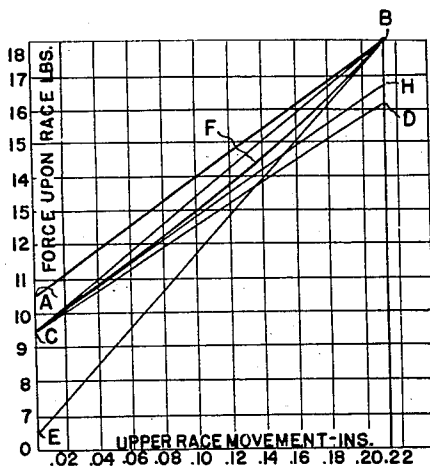
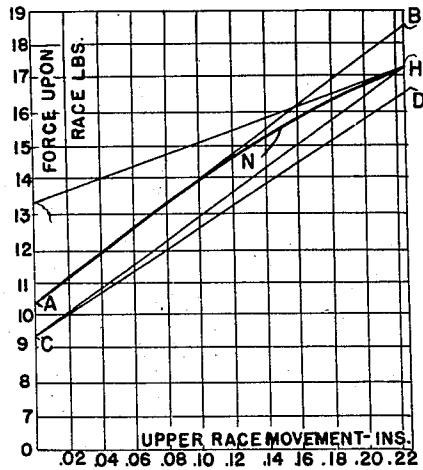


FIG. 9.



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4 Sheets-Sheet 4

FIG. II.

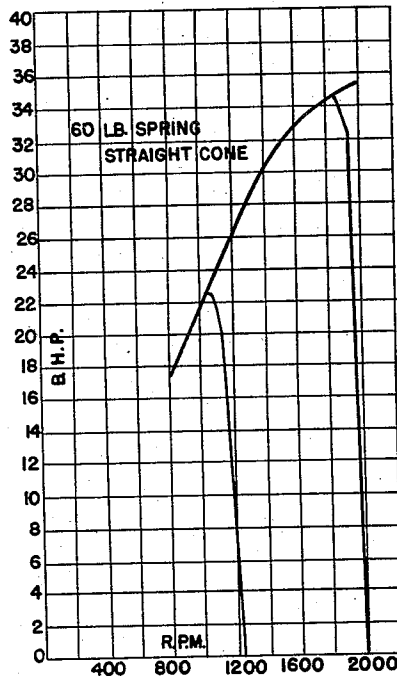


FIG. 12.

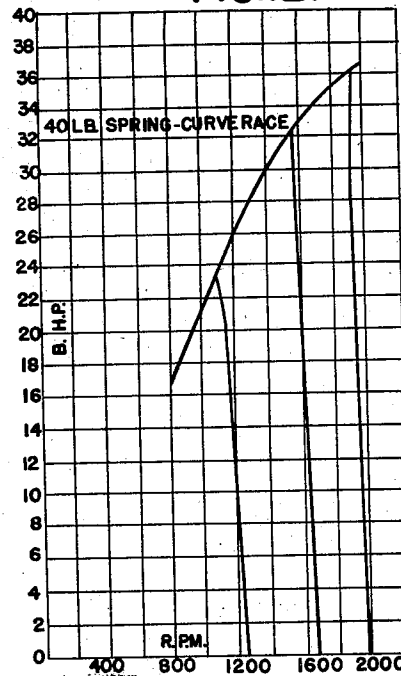


FIG. 13.

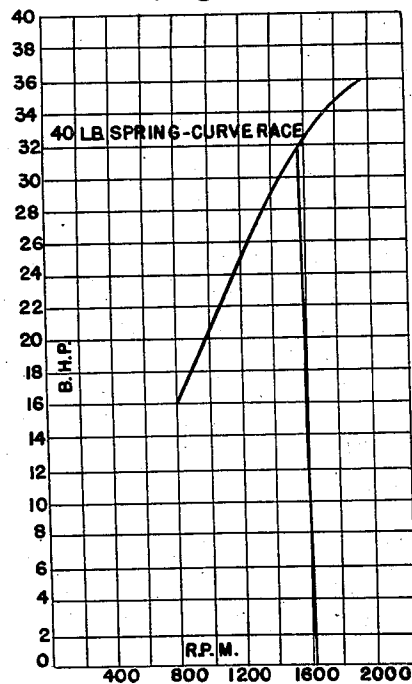
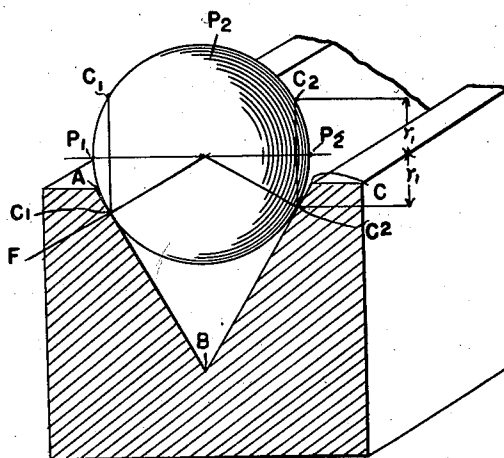


FIG. 10.



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## UNITED STATES PATENT OFFICE

2,250,982

GOVERNOR

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Application February 20, 1939, Serial No. 257,557

20 Claims. (Cl. 264—3)

The present invention relates to a governor for internal combustion engines.

It is an object of the present invention to provide an internal combustion engine governor which, while very compact, is exceedingly efficient and can be manufactured economically and with a high degree of uniformity.

Another object of the present invention is to provide a speed responsive device which is so constructed as to be substantially free from wear.

A further object of the invention is to provide an engine speed governor which is inherently stable.

It is a further object of the present invention to provide an engine speed governor which may be adjusted to operate with substantially the same efficiency over a large range of speeds.

It is a further object of the present invention to provide an engine speed governor which may be adjusted while in actual operation for different speeds.

Other objects of the invention will be apparent as this description proceeds and when taken in conjunction with the accompanying drawings, wherein—

Fig. 1 is a side elevation of my improved governor indicating its connection to the throttle of an internal combustion engine;

Fig. 2 is a vertical section of the governor shown in Fig. 1;

Fig. 3 is a section on the line 3—3, Fig. 2;

Fig. 4 is a detail view of a portion of the mechanism;

Figs. 5 and 6 are charts showing certain governor characteristics;

Fig. 7 is a vertical section showing a somewhat different embodiment of my invention;

Fig. 8 is an enlarged fragmentary detail of the device shown in Fig. 7;

Fig. 9 is a chart showing governor characteristics for a governor of the type shown in Fig. 7;

Fig. 10 is a diagram illustrating why rolling motion of balls obtains; and

Figs. 11 to 13 are charts showing performance of my improved governor.

Referring first to Figures 1 to 4, the governor comprises a housing 10 attached to a base 11 by suitable securing means, such as for example, screws 12. The governor is mounted adjacent the engine to be governed and is adapted to be driven by a gear mating with the gear 13 shown in Figure 2.

Lubrication may take place through openings 16, and if desired a fan or centrifugal element may cooperate therewith.

The gear 13, which may be a straight spur gear or a helical gear, as desired, is keyed or otherwise secured to a shaft 14, suitably journaled in the housing 10.

In order to reduce friction a ball bearing, generally indicated at 15, is provided, the inner race of which, 16, is secured to a disk 17. The inner race 16 is mounted on a bushing 18 which is rotatable relative to the shaft 14. A second plate 19 is rotatably mounted on a bushing 20, which in turn is rotatably mounted on the shaft 14. A washer 21 rotatably mounted on the bushing 20 is formed with a circular groove therein for the reception of balls 22, and a corresponding annular groove is formed in the plate 19. The plate 19 is formed adjacent its outer circumference with a conically shaped portion 23 for a purpose which will presently be described.

Keyed or otherwise secured to the shaft is a driving member 24, shown in detail in Figure 4. This member is provided with a plurality of radially extending slots 25 for the reception of relatively heavy governor balls 26. Rotation of the shaft 14, as will be readily apparent, rotates the driving member 24 and hence the balls 26. Due to centrifugal action against the conical portion 23 of the plate 19, the balls tend to force the plate 19, together with the washer 21, to the left, as seen in Figure 2.

The shaft 14 has a stop washer 27 carried thereby and abutting against a split ring, or the like, 28, received in a suitable groove in the shaft 14.

Suitable means, which will subsequently be described in detail, are provided for transmitting motion of the washer 21 to the engine control means, and in addition for biasing the washer 21 and hence the plate 19 toward the right in Figure 2.

It is desired to point out certain important advantages of the construction as it is thus far described: The centrifugal element of this invention consists of balls confined between freely mounted upper and lower plates and driven by a ball carrier or driver rigidly assembled to the governor drive shaft. As previously described, the plate 17 is assembled to the inner race 16 of the ball bearing 15 so that it can rotate freely therewith. The drive shaft 14 passes through the bushing 18 and can rotate freely therein. The upper plate 19 is rotatably mounted on the shaft 14 by means of the bushing 20. Antifriction balls 22 provide for free rotation between the plate 19 and the washer 21.

When the drive shaft 14 is rotated at a given

speed, each ball develops a centrifugal force which is transmitted to the plate or race 19, and more particularly to the conically shaped portion 23 thereof. This tends to move plate 19 and the washer 21 to the left, as seen in Figure 2. The magnitude of this force upon the plate 17 is determined not only by the angular velocity of the drive shaft and radius of gyration of each ball, but also by the cotangent of the angle formed by the conical portion 23 at the point of contact with the balls 26.

The advantages of the novel arrangement of the elements of this invention as thus far described can now be pointed out. Assume that the pressure upon the upper race 19 due to centrifugal force upon the balls is balanced by the resisting force of this race due to the spring tension, subsequently to be described, in such a way that the balls are at some position intermediate to innermost and outermost positions. It is then evident that at a constant speed, each element of the unit, consisting of the drive shaft 14, plates 17 and 19, the driver 24, and the balls 26, are rotating at the same angular velocity. There will be no relative movement between any of these parts and consequently there will be no wear.

Now if the speed be, say, slightly increased, the centrifugal force upon the balls will increase, causing a small radial motion of the balls with a corresponding axial motion of the conical race 19. However, since the upper race 19 and the lower race 17 are mounted freely upon the drive shaft, these plates, or races, during the period of acceleration will not rotate at the angular velocity of the drive shaft. Further, since the change in speed is accomplished during several revolutions of the drive shaft, each ball 26 will roll easily into its new position causing the upper and lower plates to rotate at such speeds as to automatically permit this free rolling motion. That during acceleration the balls actually roll into their new positions may best be shown by reference to Figure 10, wherein ABC is a section of a V-grooved track of infinite length containing ball P<sub>2</sub>. It will be noted that the ball contacts the inclined sides of the groove at points C<sub>1</sub> and C<sub>2</sub>. If now a force F be applied to the center of the ball, the turning moment F<sub>r1</sub> thereby produced will cause the ball to roll along the track. The rotation will take place about axis P<sub>1</sub>P<sub>2</sub>, which is always perpendicular to the bisector of angle ABC.

In the embodiments shown in Figures 2 and 7, this axis of rotation is therefore inclined to lower plate 17.

Of course, there must always be a small difference in speed between the speed of the driver and the upper and lower races, before rolling motion can take place. Thus, referring to Figures 2 and 4, assume that the driver is accelerating while it rotates in a clockwise direction. In doing so, it would tend to rotate each ball in an anticlockwise direction, and this latter rotation would require that the speed of upper race 23 and lower race 17 be somewhat less than the speed of the driver. However, since, as stated, the driver is accelerating, and since further each race is freely mounted upon shaft 14, it follows that the speed of the races is at each instant less than the speed of the driver, so that pure rolling must take place.

Similarly, when the driver is decelerating, the force F is applied to the opposite side of each ball and they are, therefore, caused to rotate clockwise, which requires that the upper and lower

races rotate faster than the driver. Since however, the driver is decelerating, this condition is present automatically, and pure rolling again is certain to take place.

It is therefore evident that in this combination of elements the balls can never skid. When the speed is constant, there is no relative motion between the balls, driver and races. When the speed changes, the balls freely move into new positions with pure rolling motion. I found that this ideal condition eliminates wear, and permits of extremely close regulation without lag, as shown by results of actual tests appended hereto.

It is evident from the above that while the balls move into new positions the speed of the upper and lower plates lags, or leads, the angular velocity of the drive shaft by but few revolutions per minute. Since this small difference in speeds promotes the free radial motion of the balls, there is again no perceptible wear due to the motion on either the upper or lower plates.

In order to transmit motion of the washer 21 to the control in a manner to regulate the speed of the internal combustion engine, I provide the following associated mechanism: Suitably journaled in the casing 10 and having one end extending outside the casing, I provide a shaft 30. Keyed or otherwise secured to the shaft 30 is a bifurcated member 31 (see Figure 3) adapted to span the member 14 and having portions engaging the washer 21. If desired, suitable bearings, such for example as needle bearings 32, may be provided, and in addition a suitable oil seal 33 may also be provided for the shaft 30. The casing 10 is provided with a circular projection 34 which is centrally apertured to receive the shaft 30. A lever 35 is keyed or otherwise secured to the projecting end of the shaft 30 and, as seen in Figure 1, is adapted to be connected by a link 36 with the throttle or other control 37 of the internal combustion engine. Journaled on the cylindrical portion 34 of the casing is a second lever 38. This lever is therefore angularly adjustable about the axis of the shaft 30. A rod or other adjusting means 39 is connected to the lever 38 and the other end of the rod 39 may be conveniently accessible to the operator of the motor.

In order to provide means for counterbalancing the movement of the washer 21 due to the centrifugal action of the balls 26, I provide a spring 40 interconnecting levers 35 and 38 in a manner now to be described. Lever 35 has projecting therefrom a short arm 41 which preferably is offset so as to be substantially aligned with a corresponding arm 42 projecting from the lever 38. As will be evident, tension of the spring 40 tends to rotate the lever 35 in a counter-clockwise direction as seen in Figure 1, and this force is transmitted through the bifurcated member 31 directly to the washer 21. By adjusting the position of the lever 38, the initial position of the arm 42 may be predetermined. It is contemplated that the lever 38 may be set to desired position, at which time the engine is governed at a desired speed.

Other means are provided for positively closing the throttle of the motor to idle position. This means takes the form of the arm 41 of the lever 35 which, as best seen in Figure 3, is bent so as to be directly engaged by lever 38. Thus when lever 38 is moved clockwise in Figure 1 to limiting position, lever 35 is mechanically moved to set the throttle in idling position.

The axial force upon the upper plate or race

19 varies, directly as the weight of the balls, directly as the square of the drive shaft speed, directly as the cotangent of the base angle of the cone, and directly as the radius of gyration of the balls. Therefore, at constant speed and with a straight cone surface, the relation between movement of the cone and the force upon it is represented by a straight line, as shown by the line AB, Figure 5. In this figure the ordinates represent the axial force upon the upper race, and the abscissae represents the movement of the race. The line AB is taken to represent the force on the upper race of 2000 R. P. M., while CD represents the force upon the upper race at 1900 R. P. M. Consequently, if the engine is to be governed so that at full load it will operate at 1900 R. P. M. and at no load at 2000 R. P. M., the spring resistance upon the upper race must be 9.4 lbs. at 1900 R. P. M. and 18 lbs. at 2000 R. P. M.

In Figure 5 the two speeds referred to were selected without regard to stability of performance, and it is possible that, if a spring resistance as shown by the line CB were applied, the engine would hunt, at least at, or near, no load.

This condition of hunting can generally be remedied by the application of a spring of a greater rate of build up, that is, one having a resistance characteristic such as indicated by the line EB, for example, in Figure 5. However, this would result in inefficient performance, as now the engine would attain a considerably lower full load speed and the per cent regulation would thereby be increased.

It is generally found in tests on engines equipped with governors, that as the spring rate of a stable centrifugal governor is decreased, hunting takes place at, or near, no load, and then as the decrease in spring rate is continued, hunting may take place between no load and 10% load. A further decrease in spring rate may then cause the engine to hunt anywhere between no load and 30% load. A still further reduction in spring rate may cause hunting between no load and 50% load, etc.

It follows from the above, that since in the first case the engine is stable from 100% to 10% load, and in the second case from 100% to 30% load, and in the third case from 100% to 50% load, etc., that in order to obtain stable performance, without undue loss in efficiency, or undue increase in per cent regulation, the effective spring rate must increase as the load upon the engine decreases.

Referring again to Figure 5, if a spring rate represented by the line EB is sufficient to insure stability at no load, a spring resistance characteristic shown by the curve CFB will provide stable performance at all loads, and still maintain the desired per cent regulation.

In Figure 6 is shown the force developed by the balls upon the upper race at various speeds. Curve 1 shows the force developed when the balls are in. Curve 2 shows this force when the balls are out. Curve 3 shows the effective spring rate at each speed, that is, the increase in spring load per inch deflection, if it were applied directly to the upper race 17.

The geometry involved in properly designing a governor of the type disclosed is shown in Figure 1. In this figure  $r$  represents the radius of adjusting movement of arm 42 carried by manual lever 38. The radius of movement of arm 41 carried by the governing lever 35 is represented by  $x$ . In the position shown,  $a_1$  represents the

arm of the spring 40 with the manual lever at a given or assumed setting, and  $a_2$  represents the arm of said spring when manual lever 38 is adjusted to control at a different speed. It is to be understood that movement of lever 38 to various positions causes the governor to control the engine at various corresponding speeds. At any given setting of manual lever 38, control between full load and no load results in swinging of governing lever 35 about its pivot point, which in turn causes the effective spring arm to vary, one such effective spring arm (with manual lever 38 as shown) being indicated at  $a_1'$ .

In order that the governor may perform its function of close regulation at widely different selected speeds, it is necessary to determine  $r$ ,  $x$ ,  $a_1$ ,  $a_2$ , and  $L_2$ . The method of design will now be disclosed.

In the design of the governor, the dimension  $m$  is selected, being the total travel of race 19 desired. Its limiting positions correspond to full open and minimum throttle.

It can be shown that when a spring is applied externally its effective rate as applied to the race 19 varies directly as the square of its lever arm, i. e.

$$R = R_s a^2 \quad (1)$$

where

$R$  = effective spring rate

$R_s$  = actual rate of spring, and

$a$  = spring arm

From Formula 1, the spring arm which, with a given spring rate, will produce a desired effective rate, may be computed, i. e.

$$a = \sqrt{\frac{R}{R_s}} \quad (2)$$

Referring again to Figure 6, it will be noted that the desirable effective spring rate decreases as the speed decreases. In order to maintain good performance over a long range of speeds, the spring rate must be changed so as to provide the proper effective rate for each speed.

However, there are many governor applications where the same engine is required to operate at various governed speeds. Thus, in the case of a tractor, plowing may be done at a certain speed, while when using the power take off another speed is desired, and when driving the tractor from one place to another, still another speed may be required. In order to avoid the troublesome necessity of actually changing springs, I have provided the structure described. The adjusting lever 38 is mounted so that it rotates concentrically with the lever 35. From the known operating requirements the maximum and minimum speeds are known, and from the chart shown in Figure 6 the corresponding spring rate can be established. A spring rate is then selected and the length of arms  $a_1$  and  $a_2$  are computed from Formula 2 above. From this data the load upon the spring and the corresponding deflection may be readily computed and in this manner the values of  $L_1$  and  $L_2$  established. For this purpose, torques about pivot points are balanced, the arm of the force acting axially on the race being conveniently taken as one inch, whence

$$F_g = F_s a_1 \quad (5)$$

where

$F_g$  = axial force on the race

$F_s$  = force of spring

From the above

$$F_s = F_g / a_1 \quad (6)$$

which may be computed and  $L_1$  computed as

$$L_1 = L_0 + d \quad (7)$$

where

$L_0$  is the original length of spring, and

$d$  = the spring deflection necessary to develop the force  $F_1$

The value of  $x$  may then be determined by evaluating and plotting the parametric equations:

$$Y = 2L_1\sqrt{x^2 - a_1^2} + L_1^2 \quad (3)$$

$$Y = 2L_2\sqrt{x^2 - a_2^2} + L_2^2 \quad (4)$$

The point at which the graphs of these equations intersect establishes the value of  $x$ , and since  $L_2$  is known,  $r$  may be determined graphically, or else computed by triangulation.

This completes the computations and the geometry of Figure 1 may now be laid out. Manual lever 38 and arm 42, and governing lever 35 and arm 41 may be drawn. It will be recognized that dimensions of these parts are kept within convenient limits by the selection of a suitable spring, but that the actual spring selected may have a rate between reasonably wide limits. After the actual spring rate is known, the foregoing method of design results in a governor which operates to govern engine speed with a very low per cent regulation, at various selected speeds, and with high stability.

Referring to Figure 1, it will be noted that when the adjusting lever 38 is adjusted so that the engine operates at the desired governed speed, the motion of the operating lever 35 from full load to no load increases the spring arm from  $a_1$  to  $a_1'$ . Since the effective spring rate increases as the square of the length of this arm, the resistance applied to the upper race will have the characteristic shown by the curve CFB in Figure 5. It follows therefore that this simple and efficient combination of levers can be arranged not only to provide uniform regulation over a long range of speeds, but that because of the parabolic character of the spring resistance curve, close regulation may be obtained at any speed without loss in stability.

As stated above, the rod 39 may be connected to an instrument board, or accelerator pedal. Thus the engine speed is controlled through the governor only. For example, in order to idle the engine, the rod 39 is moved to the right in Figure 1 until the speed is reduced to the desired idling speed.

It will be noted that the axial thrust due to the centrifugal force developed by the balls 26 and the spring tension of the spring 40 is applied to the ball bearings 15 and not to the drive shaft 14. The axial thrust upon the drive shaft is therefore nearly zero when a spur gear is used and is relatively small when the drive gear is spiral. This thrust is taken up by the bushing of driver 24 in one direction.

The operating lever 35 is acted on by the centrifugal means including balls 26, and by spring 40. It connects directly to the throttle valve 37. This lever 35 is therefore properly called an operating lever, or a governing lever, and both terms are applied to it herein. Lever 38 is referred to as a manual lever, since as shown it is manually adjustable, and it is also referred to as an adjusting lever. It will be obvious that lever 38 need not be set literally by hand, i. e. "manually," but can be otherwise positioned.

In certain instances, it may be unnecessary to

provide for ready variation in the control speed, and in this case the modification illustrated in Figures 7 and 8 may be employed. This modification differs from that previously described principally in the substitution of a screw 50 for varying tension in the spring 51, and in the substitution of a curved race 52 for the conical race 23, shown in Figure 7. The bifurcated bell crank 53 is pivoted as at 54 within the housing 55, and the free end 56 thereof is adapted to be connected by means of a link or other suitable member with the control means for the internal combustion engine. The other arm 57 of the bell crank 53 is bifurcated and has a pair of opposed lugs, or the like, 58 adapted to be received within a groove 59 in the member 60.

In this modification, as in that previously described, the race 52 is mounted for free rotation relative to the shaft 61, as is also the straight race 62. The driver 63 may be substantially identical with that shown in Figure 4, and the balls 65 are of course the same.

In this modification as in that previously described, the races 52, 62, the driver 63, and the balls 65 rotate at constant angular speed except during variations in speed. During variations in speed there is a relative angular movement between the races 52 and 62, and the balls 65 roll freely to their new position.

In the present embodiment the spring resistance, since it is applied directly to the race 52, is a straight line function. In order to provide stability combined with close regulation, it is therefore necessary to modify the movable race 52.

Referring to Figure 5, assume that the spring curve CFB is known to provide stability at all loads, that is, at all positions of the upper race. A tangent may now be drawn to this curve at the point C. This tangent then represents the characteristic of the spring 51 shown in Figure 7, and is indicated by the line CH in Figure 9. The derivative of the lines EB and AB, Figure 5, are then computed and the difference between them found. This difference, when added to the derivative of the line CH, Figure 9, determines the derivative of the line JH.

Since when a curved race is used its axial movement becomes necessarily slightly greater than that of the straight cone, it becomes necessary to predetermine the amount of this movement before the chart of Figure 9 may be drawn. This new movement may be closely approximated from

$$S_2 = \frac{A+B}{A+H} S_1 \quad (5)$$

where

$S_2$  = movement of curved race

$S_1$  = movement of straight race

A, B, and H = force upon race at corresponding points as shown on chart

From data thus established, the chart of Figure 9 may now be drawn as shown. Curve ANH is drawn so that it is tangent to AB at A, and to JH at H.

The value of the actual angle of the cone at the outermost ball position corresponding to point H in Figure 9, may now be determined from the equation

$$\cot \alpha = \frac{H}{B} \cot \theta \quad (6)$$

where

$\cot \alpha$  = cotangent of angle when balls are in outermost position

$\cot \theta$  = cotangent of angle when balls are in innermost position

H = force upon race at point H

B = force upon race at point B

By the use of the above method, the initial angle, the final angle, and the movement, or lift, of the upper race 52 can thus be determined. A parabolic curve can then be selected so that it is tangent to initial and final angles of the upper race, thus establishing the locus of the center of the ball. The profile of the surface of the race 52 may then be determined graphically, as indicated at Figure 8. This is done by drawing a series of circles, such as 70, with a radius equal to the radius of the ball 65 and with centers located on the above established parabolic path of the ball. A curve tangent to all of the circles establishes the profile of the upper race 52.

I found that in some cases a curved race is of great benefit even when used in connection with embodiment shown in Figure 2. Thus, referring to Figure 11, a 60 lbs. rate spring was required in order to produce stability with a straight race, and this resulted in 7½% regulation at 2000 R. P. M. When a curved race was substituted, it became possible to reduce the spring rate to 40 lbs. and still maintain good stability.

By reference to Figure 12 it will be seen that the governor now regulated the engine much better and that 4% regulation was obtained at 2000 R. P. M.

The excellent performance shown in chart 6 was obtained by increasing  $L_1$  (Figure 1), thereby reducing the spring arm  $a_1$ . Here the total speed variation is not more than 30 R. P. M., or 1½% regulation at 1600 R. P. M.

The foregoing detailed description of structure is given to enable those skilled in the art to practice the invention, the scope of which is indicated by the appended claims.

What I claim as my invention is:

1. A governor comprising a casing having a centrally apertured externally projecting stud, a shaft in said casing projecting from said stud, centrifugal means in said casing tending to rotate said shaft, an operating lever fixed on the projecting end of said shaft, a second lever pivoted to said stud concentrically with said shaft, spring means interconnecting said levers and adapted to oppose said rotation of said shaft.

2. A governor comprising a casing having a centrally apertured externally projecting stud, a shaft in said casing projecting from said stud, centrifugal means in said casing tending to rotate said shaft, an operating lever fixed on the projecting end of said shaft, a second lever pivoted to said stud concentrically with said shaft, said levers having arms projecting therefrom in a direction transverse to the axis of said shaft, one of said arms being located substantially further from the axis of said shaft than said other arm, and a spring interconnecting said arms.

3. A governor comprising a casing having a centrally apertured externally projecting stud, a shaft in said casing projecting from said stud, centrifugal means in said casing tending to rotate said shaft, an operating lever fixed on the projecting end of said shaft, a second lever pivoted to said stud concentrically with said shaft,

spring means interconnecting said levers and adapted to oppose said rotation of said shaft, abutments carried by said levers and adapted to abut whereby said operating lever may be mechanically moved to and retained in one extreme position by said second lever.

4. In a centrifugal governor, an operating lever urged in one direction by centrifugal means, a manual lever concentric with said operating lever, and spring means interconnecting said levers and adapted to resist movement of said operating lever by said centrifugal means in such a way that the torque arm of said spring means with respect to said levers increases upon an increase in speed.

5. In a centrifugal governor, an operating lever urged in one direction by centrifugal means, a manually adjustable lever concentric with said operating lever, each of said levers having an arm projecting therefrom, said arms projecting from said levers in directions transverse to the axis of rotation of said levers, and spring means interconnecting said arms at their free ends, said spring means being connected to said levers so that its torque arm with respect to said operating lever increases substantially upon movement of said lever in response to an increase in speed.

6. In a centrifugal governor, an operating lever urged in one direction by centrifugal means, a manual lever concentric with said operating lever, each of said levers having an arm projecting therefrom in a direction transverse to the axis of rotation of said levers, and spring means interconnecting said arms at their free ends, said spring means being connected to said levers so that its torque arm with respect to said operating lever increases substantially upon movement of said lever in response to an increase in speed, said manual lever being adjustable and adapted to be set in adjusted position to vary the speed at which said governor controls.

7. In a centrifugal governor, an operating lever urged in one direction by centrifugal means, a manual lever concentric with said operating lever, each of said levers having an arm projecting therefrom, in a direction transverse to the axis of rotation of said levers, and spring means interconnecting said arms at their free ends, the line of action of said spring being remote from the axis of rotation of said levers, said spring being connected to said levers so that the effective torque arm of said spring with respect to said operating lever increases substantially upon movement of said operating lever in response to an increase in speed.

8. In a centrifugal governor, an operating lever urged in one direction by centrifugal means, a manual lever concentric with said operating lever, and spring means interconnecting said levers and adapted to resist movement of said operating lever by said centrifugal means, said levers having abutting portions whereby upon limiting movement of said manual lever, said operating lever may be moved to and retained in one extreme position by said manual lever.

9. A speed regulator comprising a casing, a drive shaft in said casing, a slotted driving member movable with said shaft, balls in said slots, a flat plate and a concavely formed plate abutting said balls from opposite sides, both of said plates being freely rotatable on said shaft, one of said plates being movable along said shaft, a second shaft in said casing perpendicular to said drive shaft, a bifurcated arm carried by said second

shaft engaging said longitudinally movable plate, said second shaft projecting from said casing, an operating lever carried by the projecting end of said shaft, a manually adjustable lever carried by said casing coaxial with said operating lever, and a spring interconnecting said levers and adapted to oppose movement of said operating lever by said longitudinally movable plate.

10. A governor for controlling between a predetermined maximum and a predetermined minimum speed having a rotatable element adapted to apply a controlling force variable with the speed being governed, and regulating means opposing said force, comprising: a governing lever, a manual lever mounted concentrically with said governing lever, a spring interconnecting said levers, the position of said spring at any predetermined speed being determined by determining  $x$  from the following equations:

$$Y = 2L_1 \cdot \sqrt{x^2 - a_1^2} + L_1$$

$$Y = 2L_2 \cdot \sqrt{x^2 - a_2^2} + L_2$$

in which  $a_1$  and  $a_2$  are the length of spring arm and  $L_1$  and  $L_2$  are lengths of spring respectively at upper and lower limits of speed range, where  $x$  is the radial distance from the axis of the lever arms to the point of suspension of the spring to the governing lever, where  $L_2$  is the length of said spring at said predetermined minimum speed and  $L_1$  is the length of said spring at said predetermined maximum speed.

11. A governor having a rotatable element adapted to apply a controlling force variable with the speed being governed, and regulating means opposing said force, comprising: a governing lever, a manual lever mounted concentrically with said governing lever, a spring interconnecting said levers, said spring having at a predetermined maximum speed a length  $L_1$  and a spring arm  $a_1$ ,  $a_1$  being determined from the equation  $a_1 = \sqrt{R/R_s}$  in which  $R$  is the selected effective spring rate at said speed, and  $R_s$  is the actual rate of said selected spring, said spring being connected to said levers so that its torque arm on said governing lever increases upon increases in speed.

12. In a governor, a casing having a centrally apertured externally projecting stud, a rotatable shaft in said casing projecting through said apertured stud, centrifugal means in said casing tending to rotate said shaft, an operating lever fixed on the projecting end of said shaft, an adjusting lever pivoted to said stud concentrically with said shaft, a laterally projecting arm on one of said levers, a laterally projecting arm on the other of said levers, said arms extending generally in the same direction, a spring interconnecting said arms, one of said arms being offset so that said spring is substantially perpendicular to the axis of said shaft.

13. In a governor, a casing having a centrally apertured externally projecting stud, a rotatable shaft in said casing projecting through said apertured stud, centrifugal means in said casing tending to rotate said shaft, an operating lever fixed on the projecting end of said shaft, an adjusting lever pivoted to said stud concentrically with said shaft, a laterally projecting arm on one of said levers, a laterally projecting arm on the other of said levers, said arms extending generally in the same direction, a spring interconnecting said arms, one of said arms being offset so that said spring is substantially perpendicular to the axis of said shaft, the offset portion of said

one lever serving as an abutment whereby extreme movement of said control lever mechanically positions said operating lever in one extreme position.

14. A governor adapted to control motor speed at any speed within a wide range, comprising a rotatable shaft, a slotted member keyed to said shaft, a pair of plates mounted on said shaft for free rotation thereon, one of said plates being axially movable on said shaft, said plates defining an outwardly restricted annular chamber, balls in the slots of said slotted member, means resisting relative axial movement of said plates comprising a shaft, a lever carried by said shaft engaging said axially movable plate, an operating lever rigidly carried by said shaft, a manually adjustable lever mounted for relative movement to said shaft about an axis concentric therewith, and a spring interconnecting said operating lever and said manually adjustable lever, said parts being constructed and arranged to employ substantially full movement of said balls to govern speed at any position of adjustment of said manual lever.

15. A governor comprising a rotatable shaft, a slotted driving member rotatable with said shaft, balls in said slots, a pair of plates mounted on said shaft, abutting said balls at opposite sides thereof and defining between said plates an outwardly restricted chamber, means urging said plates toward said balls, said plates being rotatable on said shaft, one of said plates being axially movable on said shaft, and an abutment carried by said shaft limiting axial movement of said last plate on said shaft.

16. In a centrifugal governor, a governing lever urged in one direction by centrifugal means, a manual lever adjustable concentric with the governing lever and extending generally in the same direction, spring means interconnecting said levers having its line of action generally parallel to said levers but offset therefrom so as to have a substantial torque arm when said levers are strictly parallel.

17. In a centrifugal governor, a governing lever urged in one direction by centrifugal means, a manual lever adjustable concentric with the governing lever and extending generally in the same direction, spring means interconnecting said levers having its line of action generally parallel to said levers but offset therefrom so as to have a substantial torque arm when said levers are strictly parallel, said spring being located such that its torque arm with respect to said governing lever increases upon movement of said governing lever in response to a speed increase.

18. In a centrifugal governor, centrifugal means, an operating lever urged in one direction by said centrifugal means, an adjusting lever pivoted concentrically with said operating lever, a governor spring interconnecting said levers and adapted to resist movement of said operating lever by said centrifugal means, the connections of opposite ends of said spring to said levers being at widely different radial distances from the common pivot axis of said levers.

19. In a centrifugal governor, centrifugal means, an operating lever urged in one direction by said centrifugal means, an adjusting lever pivoted concentrically with said operating lever, a governor spring interconnecting said levers and adapted to resist movement of said operating lever by said centrifugal means, the connections of opposite ends of said spring to said levers being at widely different radial distances from the com-

mon pivot axis of said levers, the connection to said adjusting lever being furthest from said pivot axis.

20. A governor comprising a casing having a centrally apertured externally projecting stud, a shaft in said casing projecting from said stud, centrifugal means in said casing tending to rotate said shaft, an operating lever fixed on the projecting end of said shaft, a second lever piv-

5 oted to said stud concentrically with said shaft, spring means interconnecting said levers and adapted to oppose said rotation of said shaft, said operating lever having an enlarged portion surrounding said shaft and engaging a correspondingly shaped portion of said second lever to retain said second lever on said stud.

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