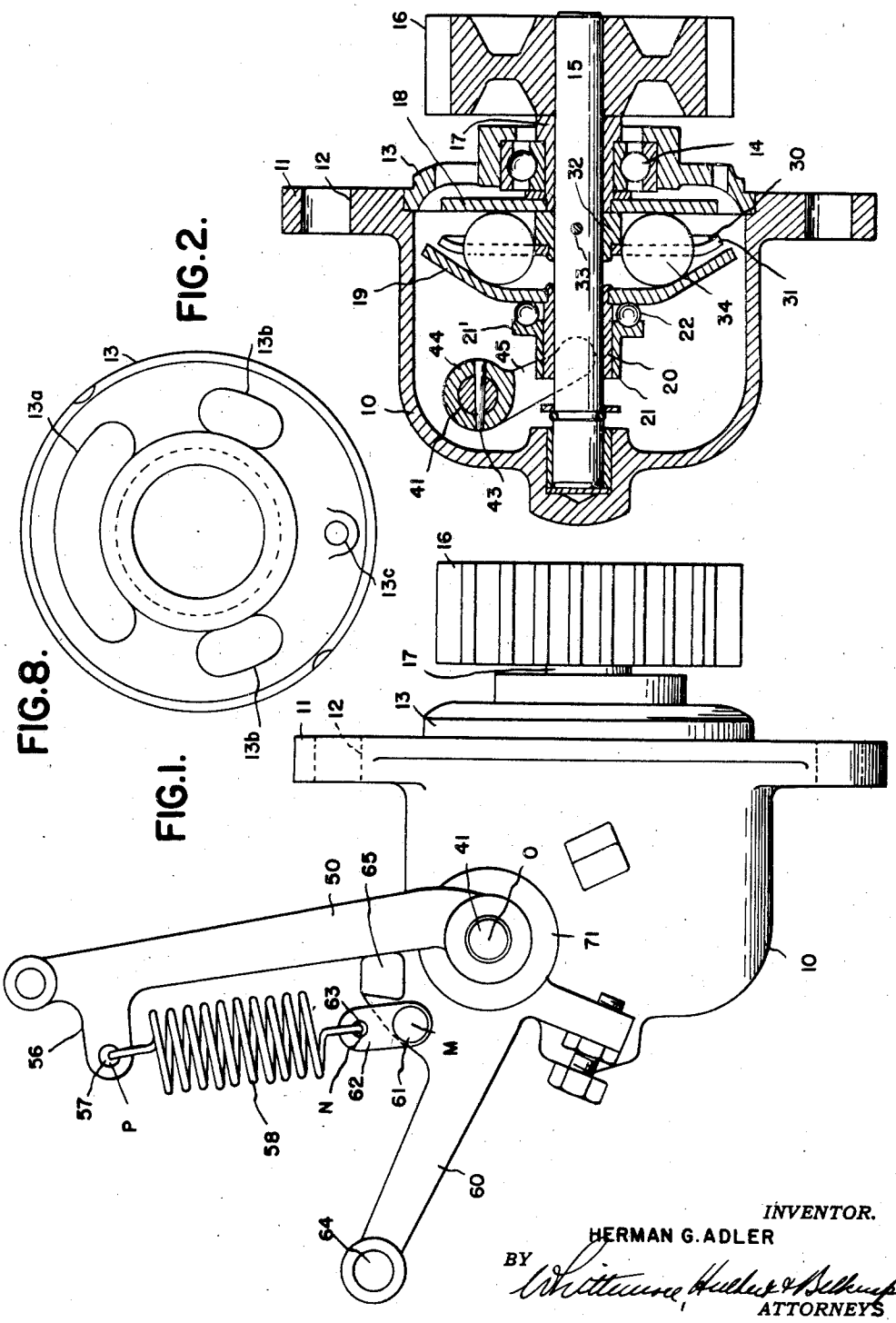
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GOVERNOR

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GOVERNOR

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FIG.6.

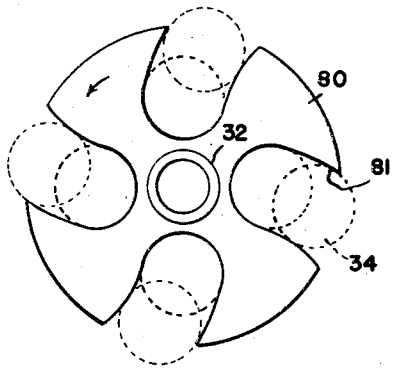


FIG.7.

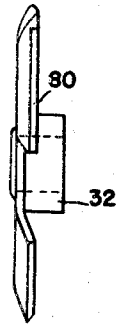


FIG.3.

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GOVERNOR

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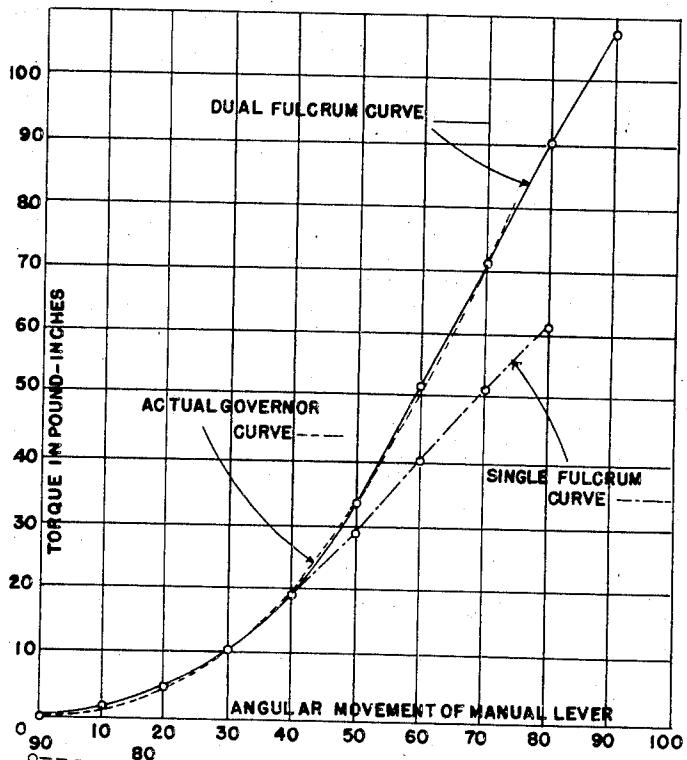


FIG.5.

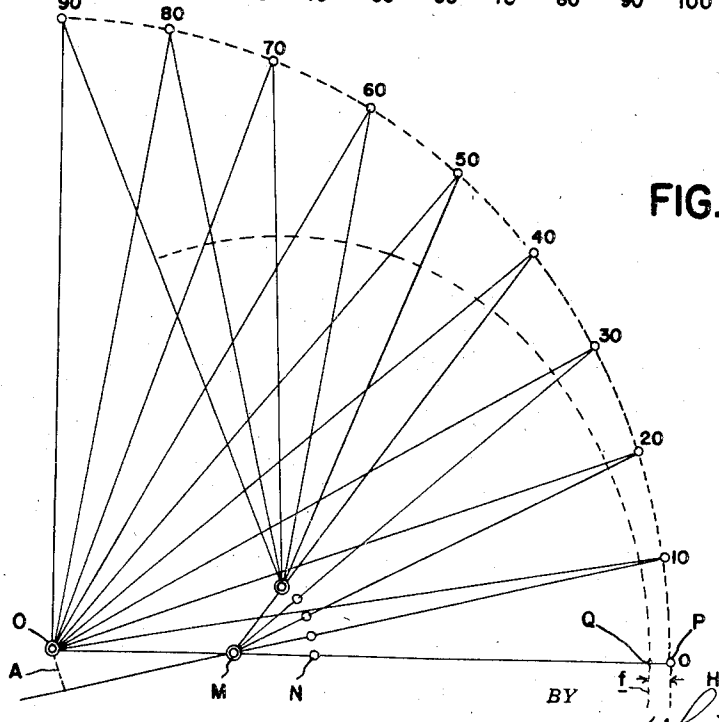


FIG.4.

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Patented June 30, 1942 2,288,382

UNITED STATES PATENT OFFICE

2,288,382

GOVERNOR

Herman G. Adler, Detroit, Mich., assignor to Novi Equipment Company, Novi, Mich., a corporation of Michigan

Application March 31, 1941, Serial No. 386,137

8 Claims. (Cl. 264—17)

The present invention relates to a governor of new design, and is an improvement over the governor shown in my prior copending application, Serial No. 257,557, filed February 20, 1939, now matured as Patent No. 2,250,982. The present application is a continuation in part of my prior copending application, Serial No. 333,655, filed May 6, 1940, now matured as Patent No. 2,250,983.

It is an object of the present invention to provide a governor of simplified design adapted to operate between wide limits of speed regulation.

It is a further object of the present invention to provide in a centrifugal governor, balls of novel characteristics; and more specifically, to provide a novel cooperation between balls and races, particularly by chromium plating either the balls or races or both, but preferably the balls.

It is a further object of the present invention to provide centrifugal balls between inclined races, in combination with a slotted driver characterized by non-radial, and preferably curved driving slots.

It is a further object of the present invention to provide a novel lubricating arrangement which insures ample lubrication of the governor during operation, and which drains the same when idle, and which prevents accumulation of sludge therein.

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

Figure 1 is a side elevation of the governor;

Figure 2 is a vertical section through the governor;

Figure 3 is an end elevation of the governor with parts in section;

Figure 4 is a diagram indicating the geometry of the dual fulcrum spring employed;

Figure 5 is a chart illustrating the torque developed by the dual fulcrum spring;

Figure 6 is a detail elevation of the novel driver;

Figure 7 is a side elvation of the driver shown in Figure 6; and

Figure 8 is a front elevation of the housing closure plate.

The governor comprising the present invention includes a housing 10, having a flange 11 provided with apertures 12 by means of which the housing is bolted or otherwise secured to appropriate supporting structure. The housing 10 is open at one end and is adapted to be closed by a closure plate 13, appropriately secured in place at the open end of the housing. The closure plate 13 is provided with bearings 14, herein indicated as ball bearings, adapted to support a drive shaft 15. The drive shaft 15 has keyed or otherwise secured thereto a driving gear 16 adapted to be driven from an appropriate member associated with the engine to be governed.

A bushing 17 is mounted for free rotation on the shaft 15 within the inner race of the bearing 14, and supports an annular plate 18 for free rotation on the shaft 15. A second cooperating plate 19 is also carried by the shaft 15 for free rotation and axial movement, and is supported thereon by means of an axially movable bushing 20. A thrust collar 21 having a flange 21' is loosely mounted on bushing 20, and ball bearing elements 22 are received between the flange 21' and the plate 19. Both the flange 21' and the plate 19 are provided with shallow grooves for receiving and retaining the balls 22 in position, thus forming a race for the balls.

The plate 19 is indicated as dished in a manner to provide a space between the plates 18 and 19, which is outwardly constricted for a purpose which will presently appear.

Keyed or otherwise secured to the shaft 15 is a driver 30 having generally radial ball receiving slots indicated at 31. I have indicated a supporting sleeve 32 keyed as indicated at 33 to the shaft 15 for supporting the driver 30. Received within the slots 31 and between the plates 18 and 19 are a plurality of balls 34 which, as will be evident, will be driven in rotation by rotation of the shaft 15.

Rotation of the shaft 15 and corresponding rotation of the balls 34, urges the balls outwardly due to the centrifugal force developed. Outward movement of the balls moves the axially movable plate 19 and the associated thrust collar 21 to the left, as seen in Figure 2. This motion is resisted by suitable balancing mechanism, which will now be described in detail.

The housing 10 is provided interiorly with a recess 40 for receiving one end of a rotatably mounted shaft 41, the shaft extending exteriorly of the casing 10 through an outwardly extending apertured boss 42. Keyed or otherwise secured to the shaft 41, as indicated at 43, is a yoke element 44 having depending arms 45 adapted to engage the flange 21' on the thrust collar. Secured to the externally projecting end of the shaft 41 is a control arm 50, having adjacent its end a ball element 51 adapted to be connected to a link 52, which in turn is con-

nected to means for controlling the position of the throttle (not shown). In Figure 3 I have indicated the link 52 as provided with a hollow sheet metal member 53 shaped to partly embrace the ball element 51, and having a spring-pressed plunger 54 therein, urged by a spring 55 to retain the link 52 connected to the ball 51.

The boss 42 has a cylindrically finished surface on which is mounted, coaxially with the shaft 41, an adjusting lever 60. Tension means are provided inter-connecting the levers 50 and 60, and for this purpose the lever 50 is provided adjacent its free end with a laterally extending arm 56, having an eye 57 therein for the reception of a loop of a coil tension spring 58. The lever 60 is provided with a projecting stud 61 on which is pivoted a rigid link element 62, having an eye 63 for the reception of the opposite end of the coil spring 58. Spring 58 and link 62 together make up a spring connection between levers 50 and 60.

The adjusting lever 60 is provided at its free end with a ball 64, similar in all respects to the ball 51 carried by the lever 50, previously described. Suitable manual adjusting means (not shown) are adapted to be connected to the ball 64 for swinging the adjusting arm 60 into predetermined position and for retaining it in this position. As will be appreciated, adjustment of the arm 60 varies the tension and the instantaneous effectiveness of the torque arm of the spring 58 with respect to the control lever 50, and thereby serves as a speed control setting for the governor.

The arms 50 and 60 are axially spaced from each other, but means are provided on one of the arms which is engageable with the other arm, whereby upon swinging of the adjusting arm 60 to limiting position in one direction the control arm 50 will be simultaneously positively moved to a position corresponding to idle position of the throttle. This means takes the form of a projection 65 extending from the arm 60 into the plane of movement of the arm 50 and adapted to engage the arm 50 when the angularity between the arms is reduced to a predetermined amount. The projection 65 serves another important function, as will now be described. As indicated in Figure 1, the projection 65 serves also as an abutment for the rigid link 62, and is adapted when the angularity between the arms 50 and 60 increases to a predetermined amount, to prevent further swinging of the link 62 about the stud 61. This will modify the effectiveness of the spring in a manner which will now be pointed out.

As stated at the outset, the present governor is designed to operate through an extraordinarily wide speed range, and accordingly the means for balancing the torque developed by the centrifugal means must correspond to the developed torque throughout a wide range. As indicated in Figure 5, the governor of the type illustrated may have an actual governor curve, as indicated by dotted lines in this figure. If the link 62 and associated cooperating abutment 65 were omitted so that the spring had a single fulcrum, the curve drawn with the broken line and designated "single fulcrum curve" would be obtained. As is apparent, this curve deviates sharply from the actual governor curve, particularly in the high torque range. It would, of course, be possible to modify the spring characteristics and design so as to cause this curve to correspond closely to the actual governor curve in the high torque range, but if this were done the curve would deviate widely from the actual governor curve in the low torque range. By employing the present construction, that is, resilient means comprising the tension spring and pivoted link, together with an abutment to prevent swinging movement of the pivot link upon attainment of a predetermined angularity between the levers, the curve shown in full line and designated "dual fulcrum curve" is obtained. As will be evident, this curve follows closely the actual governor curve throughout the entire range, with the result that close regulation may be obtained throughout an exceedingly wide speed range.

In Figure 4 I have illustrated the geometry involved in what I term the dual fulcrum construction, and in this figure the axis of the shaft 41 is designated at O; the axis of the stud 61 is designated M; the eye 63 in the link 62 is designated at N; and the eye 57 of the arm 56 is designated P. At Q I have indicated the normal or untensioned length of the spring 58 so that the distance between the curves P and Q indicates the elongation of the spring 58.

The diagram in Figure 4 is graduated in degrees from zero to 90°, and a zero degree condition exists when the points O, M, N and P are aligned. In this condition, as will be obvious, the spring is stretched by an amount measured by the distance between the curves Q and P at the zero point, but no torque is exerted for the reason that the force of the spring is aligned with the pivot center O. In this figure it is assumed that the abutment 65 is so positioned that it engages the link 62 when the arms 50 and 60 have been separated by an angle of 40°. Therefore, throughout the first 40° of relative angular movement between the arms 50 and 60 the tension means, including the spring 58 and link 62, operate in the usual manner, and the torque developed may be computed if the spring rate is known. Thus, for example, when the angular separation between the arms is 10°, the elongation of the spring may be determined by the separation between the curves Q and P at that point, and the effective torque arm developed is indicated at A, the normal distance between the fulcrum center O and the line of action of the spring.

Throughout the first 40° of relative angular movement the torque increases in the usual manner, there being a continuous increase in elongation in the spring and a continuous increase in the effective lever arm of the spring. As soon as the link 62 engages the abutment 54, conditions are changed, and further separation of the levers results in a more rapidly increasing torque. This will be apparent from Figure 4 in which it will be observed that length of the spring as measured between the points N and P will increase more rapidly, due to the fact that the point N is no longer permitted to swing about the point M as the center. In addition the lever arm A will increase much more rapidly, due to the fact that the line of action of the spring now passes through a fixed point at the upper end of the curve N.

In order that this developed torque curve may be fully understood, I have included below in tabular form an illustrative set of figures. In the figures θ represents the angle included between the arms 50 and 60; A represents the effective torque arm or the normal distance from the point O to the line of action of the spring; f represents the elongation of the spring; and RAf

represents the torque where R is the assumed or selected spring rate. As will be apparent, this torque, as indicated in the column RA*f*, is that plotted on the chart of Figure 5.

θ	A	f	RAf
10	.242	.120	1.68
20	.460	.180	4.82
30	.654	.275	10.43
40	.812	.405	19.0
50	1.028	.560	33.4
60	1.182	.755	51.8
70	1.267	.972	71.4
80	1.302	1.192	90.0
90	1.298	1.420	107.0

The depending arms **45** of the yoke **44**, which engage the flange **21'** of the thrust collar **21**, retain the collar **21** against rotation on the bushing **20**. As previously stated, the plates **18** and **19** are mounted for free rotation on the shaft **15**, and during normal operation they will rotate at the same speed as the balls **34**. This is a desirable condition. Since the flange **21'** does not rotate, even though balls **22** are provided intermediate this flange and the plate **19**, in some cases it is found that the plate **19** does not rotate. This gives rise to unsatisfactory operating conditions and causes considerable noise in the governor. It is accordingly desirable to insure that during normal operation of the governor the plate **19** will rotate.

I have found that this aim and other substantially improved results can be obtained by providing a definite relationship between balls **34** and plates or races **18** and **19**.

In the first place the balls must be of a material which resists corrosion. Secondly, they must have a hard wearing surface. Finally, the surface contact between the balls and plates must be such that the plates will rotate in unison with the balls during constant speed operation.

For this last purpose I have found that very satisfactory results are obtained when the surfaces of the balls are slightly roughened. The amount of roughness is very small, being perceptible primarily under a microscope. At present I am aware of no machining operation which would provide the requisite roughness, but I have found that balls which are plated with chromium in the ordinary commercial manner, and are left unpolished, are perfectly satisfactory.

Results might be improved somewhat if the plates on which the balls run were also chromium plated. but in practice I find that this is not necessary, and the balls keep the race surfaces of the plates clean and in satisfactory condition. It is recognized however that similar operating conditions would prevail if the plates were chromium plated, and the balls merely made of a corrosion-resistant material.

The type of chromium plating referred to herein is that described in "Metals Handbook," 1939 edition, published by "American Society for Metals," 7016 Euclid Avenue, Cleveland, Ohio. In this handbook, on page 1102, under the title "Industrial, or 'hard' chromium plating," appear the following remarks:

"*Definition.*—The chromium in 'hard' chromium plating, as distinguished from decorative chromium plating is usually deposited in appreciable thicknesses directly on the base metal. This type of plate is finding wide industrial application. Ordinarily, the base is fully or semihardened steel. In some cases, however, successful applications have been made on soft steel, cast iron, and many nonferrous alloys. Fairly heavy deposits must be made on the softer base materials if heavy pressures are encountered in service.

"The plate depth usually is in the range of 0.0001 in. or more. The heavier deposits are used chiefly for salvage purposes.

"*Hardness.*—The outstanding property of electrochemically deposited chromium is its resistance to wear. Schneidewind reports that the average deposit will show the following hardness values: Vickers-Brinell 625, Mohs 8, and scratch width with Bierbaum Microcharacter test of 2 microns. In employing these values, however, it must be borne in mind that the plating conditions have a profound effect on the hardness of plate. These figures must therefore be regarded as approximate.

"Chromium plate manifests a file hard reaction to a temper testing file. Wear resistance far superior to that found with fully hardened tool or die steels is obtained.

"*Chemical resistivity.*—Chromium plate is not affected by most organic chemicals, alkalies, sulphur, and sulphur compounds. Nitric acid will not dissolve it. The plate is readily attacked by hydrochloric acid and slowly by sulphuric acid.

"*Corrosion resistance.*—Chromium plate does not tarnish or corrode in ordinary atmospheres. However, fairly thin deposits are apt to contact cracks and pinholes which permit corrosive elements to come in contact with the base metal. Proper control of plating conditions minimizes this tendency."

In practice the balls are of carburized and heat treated steel. The chromium plate has a minimum thickness of .0003 inch. After plating, the balls are neither buffed nor tumbled and have the inherent roughness which is herein defined as "unpolished."

Oil sludge as is well known may contain traces of nitric or sulphuric acid and the unpolished chromium plated balls are substantially unaffected by these impurities. While nickel-chrome or other steel alloys could be used due to their resistance to acid and corrosion, I have so far been unable to discover a method of machining which will impart to balls made of these materials the requisite surface characteristics. Furthermore, the cost of nickel-chrome steel balls is double the cost of the chromium plated balls disclosed herein. It will be understood however that the present invention encompasses the use of such alloy balls if roughened to have the surface characteristics of unpolished chromium plated balls.

At present I am not aware of other plating metals than chromium which are satisfactory for the present purpose, but if such are later discovered, their use to produce balls having the specific characteristics covered herein will of course be merely a substitution of equivalents.

Extensive field use has proved that entirely satisfactory results are obtained when the plates **18** and **19** are blanked from cold rolled steel, carburized and hardened. The plates are preferably coined to the desired shape. I have noted that while the plates may be corroded at points not engaged by the balls, the areas where the balls roll are clean and bright. I have therefore concluded that chromium plated plates are unnecessary.

The chromium plating used is the hard plating described in the Metals Handbook, and due to

the "file-hard" surface resulting, wear on the balls is negligible.

Referring now to Figure 3, I have illustrated a compression spring 70 surrounding the shaft 41 and abutting against the yoke 44. The yoke 44 is keyed to the shaft 41, as previously described, with the result that the spring 70 urges the shaft 41 to the left as seen in Figure 3. Exteriorly of the boss 42 I provide a friction and sealing plate 71 which is secured against rotation to the shaft 41. The plate 71 may for example be pressed to the shaft 41 for rotation therewith. The end surface of the boss 42 against which the plates abut, may be finished smoothly, and the plate 71 may be hardened and ground. By selecting a spring 70 of sufficient strength, any desired amount of friction within reasonable limits may be introduced.

The governor as a whole is designed so as to reduce friction of the parts to a minimum, with the result that the friction introduced by the friction plate 71 may be substantially all of the friction present in the device. As a result of this, it is possible to control within accurate limits the over-all friction encountered in the operation of the device. It is desirable to introduce friction so as to slow the response of the governor under certain operating conditions. For example, when the governor is applied to a tractor, and where the tractor is pulling a farming implement, such for example as a disk, across a plowed field, there is a rapid succession of impulses. As a result, the governor in many cases will be found to cause undesirable surges in the motor in synchronism with the impulses applied. Under these circumstances, by providing a predetermined amount of frictional resistance to response, the response may be delayed so that these undesirable surges are avoided. As will be evident, of course, the provision of friction in a device will not seriously affect the speed range or the closeness of regulation of the governor.

Instead of employing a hardened and ground plate 71, it may in some instances be preferable to provide a plate of non-metallic material, such for example as a fibrous material. In either case it is found that where this frictional resistance is introduced in the manner specified, it is unnecessary to supply a separate oil seal for preventing leakage of oil from the interior of the governor around the shaft 41. The friction plate is found to perform this oil sealing function in an entirely satisfactory manner.

Referring now to Figures 6 and 7, I have illustrated a modified form of driver 39, corresponding generally to driver 30 shown in Figure 2. In this case responsiveness of the governor is modified by providing the driver 39 with curved, inclined slots 81 for the reception of balls 34, shown in this figure in dotted lines. Driver 39 is secured to sleeve 32 for rotation with drive shaft 15.

The slots 81 are preferably curved so as to form equal angles with radial lines, or in other words formed as a logarithmic spiral. In practice however this curve may be closely approximated by a circular arc whose center is remote from the axis of the driver.

During constant speed operation, with all parts in equilibrium, the curvature and inclination of the slots do not affect the operation. During acceleration or deceleration, however, the inclination of the slots affects the sensitivity of the governor. Thus for example with the slots inclined rearwardly from the direction of rotation as shown by the arrow in Figure 6, the inclination of the slots aids outward movement of the balls during acceleration, and aids inward movement of the balls during deceleration. As will be obvious, this renders the governor more responsive to speed changes. If the slots were oppositely inclined, the governor's responsiveness would be retarded, thus serving a purpose similar to friction member 71 previously described.

By curving the slots at a substantially uniform angle to the radial lines, the effectiveness of the slots for the purpose set forth is kept substantially constant for all positions of the balls.

It is further within the scope of the invention to provide the slots so as to make different angles with radial lines so as to affect the sensitivity in a predetermined manner for different positions of the balls between balls-in and balls-out position.

Referring now more particularly to Figures 2 and 8, I have provided for novel lubrication of my improved governor. As seen in Figure 2, the housing 10 is provided with a closure plate 13. As best seen in Figure 8, the closure plate 13 has formed therein a plurality of openings, an elongated upper opening 13*a*, a pair of intermediate smaller openings 13*b*, and a bottom opening or drain hole 13*c*.

During operation of the governor, the driving gear 16 and the gear meshed therewith are lubricated by means of an oil supply, at least one of the gears being partly immersed in oil contained at the bottom of the casing. As a result, the gear 16, and the gear or gears associated therewith are adequately lubricated, and in addition oil is splashed and thrown in substantially all directions within the casing.

The closure plate 13 is provided with the upper elongated opening 13*a* and a very substantial quantity of oil passes into the interior of the governor housing 10 through the opening 13*a*. It is found in practice that more oil will be admitted through the opening 13*a* during normal operation than will run out through the lower opening or drain hole 13*c*. In order to maintain the oil level within the housing 10 at a desired height, the pair of intermediate relatively smaller openings 13*b* are provided in the closure plate 13. Openings 13*b* will determine the level of the oil in the governor housing 10. While there will be a tendency for some oil to be drawn into the housing through the opening 13*b*, it will be appreciated that if the oil level in the housing 10 reaches the openings 13*b*, this oil will run out through these openings and prevent the oil level from raising substantially above the lower portions of the openings 13*b*. At the same time there will be a continuous flow of oil out the lower opening or drain hole 13*c*.

When the governor comes to rest and oil ceases to be thrown in through the openings 13*a* and/or the openings 13*b*, this oil will drain out through the lower opening 13*c*. During operation of the governor there will be a continuous flow of oil out the lower opening 13*c*, and I have found in practice that this continuous flow of oil out the lower opening 13*c* prevents the accumulation of sludge in the governor housing 10. Apparently sludge will be washed out through the lower opening or drain hole 13*c*.

As a result of the construction just described, the governor during normal operation will run in a bath of oil, and the lower portion of the plates 18 and 19 will be immersed in a reservoir

of oil to a depth determined by the location of the intermediate openings 135. This insures uniformity of lubrication and contributes to the efficiency of operation and long life of my improved governor.

While I have illustrated and described a specific embodiment of my governor, it will be apparent that this has been done solely 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. In a centrifugal governor, a rotary driving member having outwardly extending slots, means for rotating said member, a pair of plates at opposite sides of said member, said plates being formed to provide an outwardly reduced space therebetween, said plates being mounted for substantially free rotation and for relative axial movement, and balls in said slots engaged by annular surfaces of said plates, whereby the entire surfaces of said balls and annular surfaces of said plates are bearing surfaces, at least one of said surfaces being provided with commercial, hard, unpolished chromium plate.

2. In a centrifugal governor, a rotary driving member having outwardly extending slots, means for rotating said member, a pair of plates at opposite sides of said member, said plates being formed to provide an outwardly reduced space therebetween, said plates being mounted for substantially free rotation and for relative axial movement, and balls in said slots engaged by annular surfaces of said plates, whereby the entire surfaces of said balls and annular surfaces of said plates are bearing surfaces, at least one of said surfaces being provided with commercial, hard, unpolished chromium plate of a thickness of at least .0003 inch.

3. In a centrifugal governor, a rotary driving member having outwardly extending slots, means for rotating said member, a pair of plates at opposite sides of said member, said plates being formed to provide an outwardly reduced space therebetween, said plates being mounted for substantially free rotation and for relative axial movement, and balls in said slots engaged by annular surfaces of said plates, whereby the entire surfaces of said balls and annular surfaces of said plates are bearing surfaces, the surfaces of said balls being provided with commercial, hard, unpolished chromium plate of a thickness of at least .0003 inch.

4. In a centrifugal governor, a rotary driving member having outwardly extending slots, means for rotating said member, a pair of plates at opposite sides of said member, said plates being formed to provide an outwardly reduced space therebetween, said plates being mounted for substantially free rotation and for relative axial movement, and balls in said slots engaged by annular surfaces of said plates, whereby the entire surfaces of said balls and annular surfaces of said plates are bearing surfaces, at least the surfaces of said balls being commercial, hard, unpolished chromium plate.

5. In a centrifugal governor, a rotary driving member having outwardly extending slots, means for rotating said member, a pair of plates at opposite sides of said member, said plates being formed to provide an outwardly reduced space therebetween, said plates being mounted for substantially free rotation and for relative axial movement, and balls in said slots engaged by said plates, said plates being formed of hardened steel, said balls being formed of hardened steel having unpolished surfaces of hard, commercial chromium plating.

6. In a centrifugal governor, a rotary driving member having outwardly extending slots, means for rotating said member, a pair of plates at opposite sides of said member, said plates being formed to provide an outwardly reduced space therebetween, said plates being mounted for substantially free rotation and for relative axial movement, and balls in said slots engaged by annular surfaces of said plates, whereby the entire surfaces of said balls and annular surfaces of said plates are bearing surfaces, said plates being formed of hardened steel, at least the surfaces of said balls being commercial, hard, unpolished chromium plate.

7. In a centrifugal governor, a rotary driving member having outwardly extending slots, means for rotating said member, a pair of plates at opposite sides of said member, said plates being formed to provide an outwardly reduced space therebetween, said plates being mounted for substantially free rotation and for relative axial movement, and balls in said slots engaged by annular surfaces of said plates, whereby the entire surfaces of said balls and annular surfaces of said plates are bearing surfaces, said plates being formed of hardened steel, said balls being formed of hardened steel, having surfaces of commercial, hard, unpolished chromium plate of a thickness of at least .0003 inch.

8. In a centrifugal governor, a rotary driving member having outwardly extending slots, means for rotating said member, a pair of plates at opposite sides of said member, said plates being formed to provide an outwardly reduced space therebetween, said plates being mounted for substantially free rotation and for relative axial movement, and balls in said slots engaged by annular surfaces of said plates, whereby the entire surfaces of said balls and annular surfaces of said plates are bearing surfaces, said plates being formed of hardened steel, said balls being formed of hardened steel, at least one of the surfaces aforesaid being provided with commercial, hard, unpolished chromium plate.

HERMAN G. ADLER.