May 27, 1958 E. LAAS 2,836,670

GOVERNOR WITH SWITCH CONTROL

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FIG. 1.

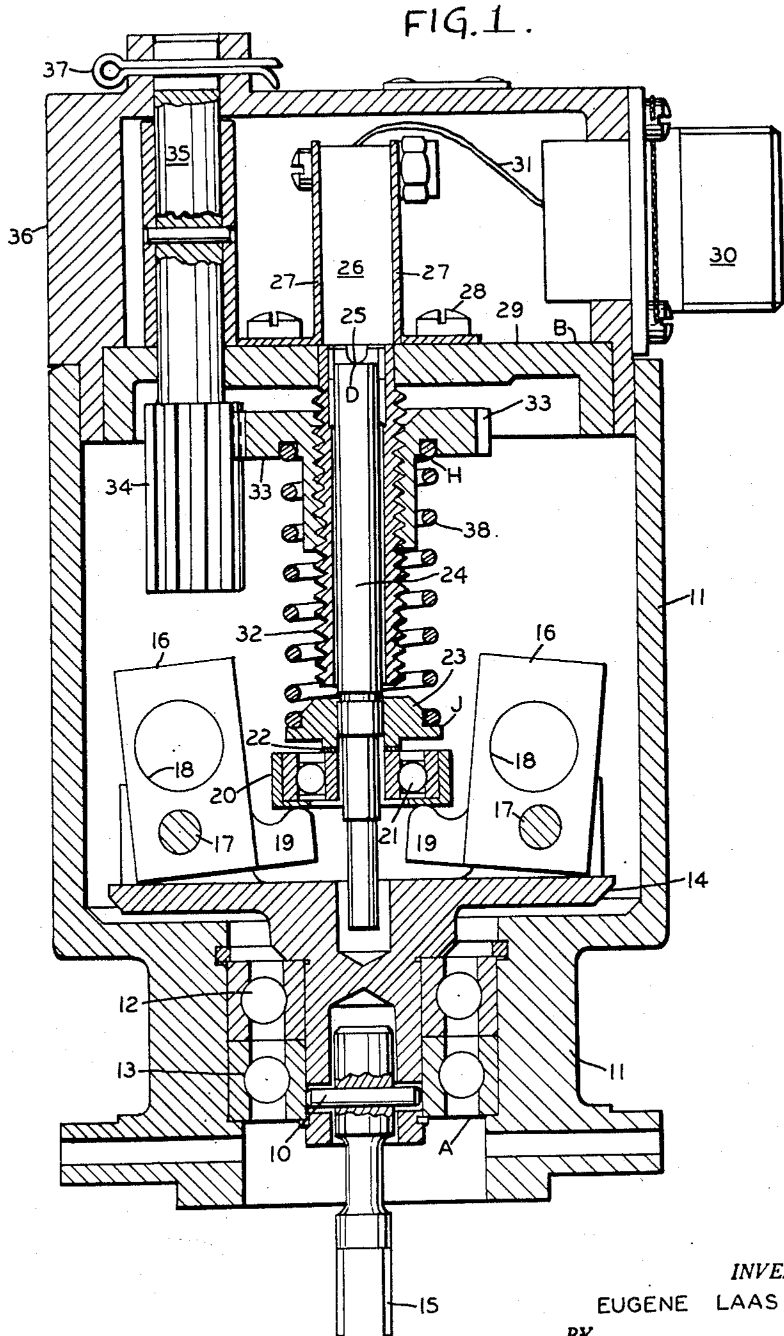

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FIG. 2

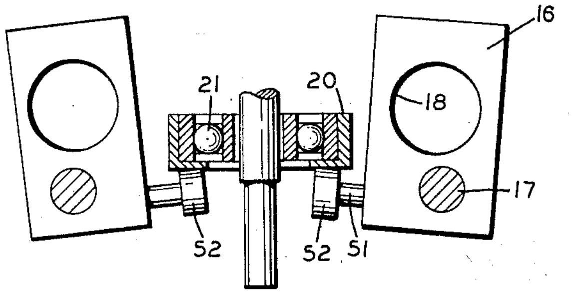


FIG. 3

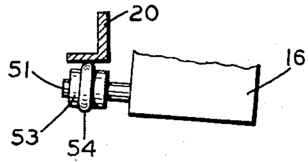


FIG. 4

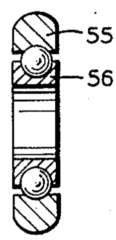
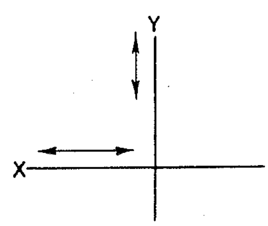


FIG. 5



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GOVERNOR WITH SWITCH CONTROL

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Application April 26, 1957, Serial No. 655,276

6 Claims. (Cl. 200—80)

This invention relates to governors with circuit controlling means adapted to open, or close, or transfer circuits at predetermined speeds of rotation of an engine or other rotating element. The invention is an improvement of copending application for U. S. Letters Patent entitled "Thermally Compensated Governor With Switch Control," Serial No. 607,800, filed September 4, 1956, now U. S. Patent No. 2,810,032, issued October 15, 1957. This invention furthermore, has particular reference to a governor construction in which the speed differential necessary for switch actuation, i. e., the speed difference between the switch "on" and "off" condition is reduced to a minimum value.

The general construction of governor mechanisms which produce a straight line movement of the control member in response to increasing or decreasing speeds of rotation in combination with the operation of a switch or circuit control means is well known in the art. The use of modern high speed combustion engines, particularly aircraft engines, requires exceedingly accurate and precise control of speeds. The most important considerations are (1) thermal stability, i. e. little or no drift of switch actuation with change in temperature, and (2) as small a speed differential for causing a change in switch condition as possible.

The instant design incorporates features which minimize both of the foregoing factors thereby obtaining a speed responsive switch which is usable in many new applications and which is adapted to control speeds with a higher degree of accuracy and precision.

One of the objects of this invention is the provision of a governor with electrical circuit control means which avoids one or more of the disadvantages of prior art devices.

Another object of this invention is the provision of a governor in which the speed differential necessary to cause a change in condition is of minimum value.

Another object of this invention is the provision of a switch in which the frictional losses are greatly reduced.

Another object of this invention is to provide a governor which is substantially unaffected by the temperature rise normally experienced in and around combustion engines.

A further object of this invention is the provision of a governor type switch in which switch actuation is achieved at a predetermined setting which will remain substantially unaffected by temperature and repetitive actuation.

Additional objects, advantages and features of the present invention will be apparent from the following description taken in combination with the accompanying drawings, in which:

Figure 1 is a vertical section of the governor with circuit controlling means embodying features of temperature compensation;

Figure 2 is a partial view of Figure 1, but incorporates features to reduce the frictional losses thereby increasing accuracy of the governor;

Figure 3 is a portion of Figures 1 and 2 respectively, showing a further modification of one of the elements;

Figure 4 shows another modification of one of the elements, and

Figure 5 is a diagram illustrating the direction of certain frictional forces.

Referring now to Figure 1 and numeral **11** in particular, a governor housing is identified which supports nearly all of the components to be described hereinafter. It may be pointed out that in the following description only the more important elements are described in view of the fact that the general design and operation of this type of governor mechanism is well known and clearly described in the prior art. Rotatably mounted within the governor housing **11** by means of ball bearings **12** and **13**, there will be found a spider **14** which is engaged by a driven shaft **15** secured to the spider by coupling pin **10**. Spider **14** supports a set of fly weights **16**, each mounted for pivotal motion about a pin **17**. It will be found that fly weights **16** are equipped with a large aperture **18** in order to reduce the inertia of the fly weights and the mechanism in general. A radial extension **19** on each fly weight **16** is adapted to engage a sleeve cap **20** attached to a ball bearing **21**. The inner race of the ball bearing **21** engages a washer **22** supporting a spring seat **23**. Spring seat **23** is press-fitted to a vertical plunger **24**, the upper end thereof being adapted to engage the actuating mechanism **25** of an electrical circuit switch **26** which, by means of angle brackets **27** and screws **28**, is fastened to a support **29**. A connector **30** and wires **31** establish electrical circuit connection from the exterior of the governor to circuit switch **26**.

Plunger **24** is surrounded by an externally threaded bushing **32** the thread of which is engaged by an internally threaded hub of spur gear **33**. The teeth of gear **33** are engaged by a pinion gear **34** which is fastened to a stud **35** protruding through an upper casing **36** and cotter pin **37** rotatably secures the stud in the casing. A circular recess in the underside of gear **33** serves as upper support for helical compression spring **38**, the lower end of which is supported on spring seat **23**.

It will be noted that by turning stud **35** and consequently pinion **34**, spur gear **33** is moved vertically with respect to spring seat **23** thereby achieving a selectable force adjustment for helical spring **38**.

The operation of the instant device may be visualized as follows:

When shaft **15** is driven, spider **14** rotates in unison therewith and fly weights **16** swing radially outwardly about pins **17** in response to the rotation. As the fly weights swing outwardly, projections **19** of the fly weights tend to lift sleeve cap **20** which in turn tends to raise washer **22**, spring seat **23** and plunger **24** fastened thereto to cause actuation of element **25** of circuit switch **26** which is the condition depicted in the figure. Helical spring **38** by virtue of one of its ends resting upon spring seat **23** is used as a biasing means to counteract the force of the fly weights. As explained above this force is adjustable by means of stud **35** and the elements in engagement therewith. The actuating speed of switch **26** therefore is the result of the force exerted by the fly weights and the counter force exerted by spring **38**. Plunger **24** is designed so that it moves freely in axial direction in order to obtain a precise control. The design described thus far is fairly conventional, but as will be explained hereinafter, provisions have been made to obtain stability of operation irrespective of the influences of temperature and thermal expansion.

An ordinary spring, for instance steel, will introduce large speed errors due to changes in temperature. In order to achieve stability of operation a spring is employed which has a substantially constant modulus of elasticity. In this type of spring the increase of wire length due to thermal expansion is combined with an adjusted decrease in modulus of elasticity to give a combined thermo-elastic coefficient of substantially zero. As an example, a nickel-chromium-iron-titanium alloy marketed under the trade name of "Ni-Span-C" manufactured by the H. A. Wilson Company of Union, New Jersey, having a nominal composition of 42% Ni, 2.5% Ti, 5.5% Chr, and 0.03% C, may be used advantageously to obtain a substantially constant modulus of elasticity. In this manner the force of the spring can be kept substantially constant over temperature ranges from −50 to +150 degrees F.

The next problem is the maintaining of the working space of the spring, that is the space between the lower end of the spring hereinafter referred to as "J" and the upper end of the spring, hereinafter referred to as "H." The following calculations used for illustrative purposes are based upon a temperature rise of 230 degrees F., materials and governor dimensions as found on the actual embodiment of the present invention and depicted in the figure.

Considering the underside of bearing **13** as a base, hereinafter referred to as "A," then spring seat J rises 1.56 inches$\times 65\times 10^{-7}\times$230 degrees F., that is; length A to J$\times$temperature coefficient for steel$\times$temperature rise, or 0.0023 inch.

At the actuating point of circuit switch **26**, the elongation of the plunger **24** from point "D" (upper end) to the lower end of spring ("J") will have the opposite effect, tending to lower point "J" by 0.0022 inch, as the dimension A to J is substantially equal to the dimension J to D.

It will be apparent, that the elongation of the aluminum housing **11** has no effect upon the working space of the spring. The length from the top of the support **29** (point "B") to the top of the spring ("H") will expand with temperature, tending to decrease the spring working space. For ½ inch length of steel this will be: ½ inch$\times 65\times 10^{-7}\times$230 degrees F.=0.00075 inch.

The total effect on the spring working space, assuming spring compressive force positive, is then: 0.0023 minus 0.0022 inch plus 0 plus 0.00075 equals 0.00085 inch. Using a spring rate of 20 pounds per inch, the net change in spring force is 0.017 lb. Using a force level of 7 pounds at 4500 R. P. M., the net shift in spring force is plus 0.017/7$\times$100 equal 0.24% or an increase in speed of 0.12% or 5.4 R. P. M. out of the 4500 R. P. M. nominal speed setting.

The above, as it will be apparent, produces almost a perfect score since errors of this magnitude are zero and negligible for all practical purposes. The compensation shown above illustrates that the aluminum housing expansion has no effect upon spring length and that distances A to J, D to J, and B to H can be selected in such a manner as to be self-compensating thereby maintaining an almost constant working space for the spring.

It can readily be shown by calculation that the shift of the fly weights due to thermal expansion and the elongation of the switch actuating mechanism do not substantially affect the results developed above and produce an additional shift of about one R. P. M., but in a direction so as to lower the speed deviation from 5.4 R. P. M. to 4.5 R. P. M.

In comparison to the above favorable results it may be well to note that if the constant modulus of elasticity spring is replaced by an ordinary steel spring, a shift of 60 R. P. M. minus is obtained for the same condition, namely 230 degrees F. temperature rise and 4500 R. P. M. It will furthermore be readily apparent that by proportioning distances A to J, D to J and B to H so that the self-compensating thermal expansions are lost, this shift of speed will greatly increase beyond the value of 60 R. P. M.

Although the instant switch is stabilized as far as temperature is concerned, there still is a speed differential of 20 R. P. M. or more between the point of contact closing and opening or vice versa in switch **26**. Extensive testing has shown that this speed differential is caused primarily by the frictional forces of the engaging surface of fly weight extension **19** on the underside of sleeve cap **20**. As the speed of the spider **14** increases or decreases, the contact making surface of extension **19** is urged to move radially along the underside of sleeve cap **20**. The pressure exerted by the fly weights is so great and the radial motion so small, that frictional forces effectively block this radial motion. At a first glance, it would appear that this problem could be solved by providing antifriction bearings, such as roller, ball, or needle bearings on the fly weights in such a manner, that the bearings roll radially along the underside of the cap. This, however, does not produce the desired result. It is far more advantageous to shift the bearing mounting by ninety degrees as shown in Figure 2.

The fly weights have been changed to have a radially extending stud **51** upon which are mounted antifriction bearings **52**, the outer races of the bearings engaging the underside of the sleeve cap **20**, and the bearings mounted being oriented in such a manner that their own rotational axis intersects the axis of rotation of the spider **14**.

Assuming now, for example, a speed of 4,500 R. P. M. for the spider, plunger **24** and inner race of bearing **21** do not turn, being held stationary by friction of the spring **38**. If outer race of bearing **21** turns at 4,500 R. P. M., then the antifriction bearings **52** will not turn on their own axis of rotation, because there is zero relative surface speed between bearings **21** and **52**. In this case one would expect normal frictional sliding resistance in the X-direction (radial, Figure 5) between the outer race of bearing **52** and the underside of sleeve cap **20**. The movement in the X-direction is probably in the order of 1/100,000 inch.

However, the bearing **21** turns at less than 4,500 R. P. M. due to its own internal friction and drag. As a consequence, bearings **52** must turn about their own axis at some speed which is a function of this slip and the dimensions of both sets of bearings. As soon as bearing **52** rolls on the sleeve cap **20**, the friction in the X-direction is reduced to a negligible value thus eliminating rubbing or sliding motion. Actually, for small movements of the plunger **24** (Y-direction) between the "on" and "off" conditions of the switch, bearings **52** will tend to move on a spiral track while they are attempting to move in the X-direction.

It will be apparent that a predetermined line contact of bearing **52** with the sleeve cap is desired in order to maintain accuracy and stability. To achieve this, bearings **52** in Figure 2 are inclined with respect to the underside of the sleeve cap, causing the edge of the bearings to establish contact. In Figure 3, as an alternate, bearin **53** has an outer race which is equipped with an annular raised rim, and in Figure 4, the bearing has a conventional inner race **56** but the outer race **55** is of convex shape.

Tests have proven conclusively that by virtue of the foregoing arrangement the speed differential can be reduced to such an amount that the friction of the governor mechanism is negligible and the speed differential is determined only by the force vs. motion condition of the electrical switch assembly itself.

While there have been described and illustrated certain embodiments of the present invention, it will be apparent to those skilled in the art, that various modifications and variations may be made therein without departing from the spirit and field of the invention which should be limited only by the scope of the appended claims.

What is claimed is:

1. The combination of a governor housing; a driven shaft mounted therein; said shaft engaging a spider which

is driven thereby; a plurality of fly weights pivotally fastened to said spider to swing radially outwardly in response to rotation of said shaft and spider; each of said fly weights equipped with an antifriction bearing mounted so that the axes of rotation of said bearings intersect the axis of rotation of said spider; a sleeve cap mounted for rotation about the rotational axis of said shaft and spider and being engaged by said antifriction bearings which are in rolling contact with said sleeve cap when said spider rotates; one end of an axially floating plunger extending through said cap and being urged in axial movement in response to the rotation of said fly weights; the other end of said plunger adapted to engage the actuating mechanism of an electrical circuit switch and causing operation thereof as a result of the plunger's axial movement; a helical compression spring surrounding a portion of said plunger and supported to oppose the axial motion of said plunger toward said switch and thereby opposing also the outward motion of said fly weights.

2. The combination as set forth in claim 1 wherein said antifriction bearings in contact with said sleeve cap comprise ball bearings.

3. The combination as set forth in claim 1 wherein said antifriction bearings in contact with said sleeve cap comprise roller bearings.

4. The combination as set forth in claim 1 wherein said antifriction bearings in contact with said sleeve cap comprise needle bearings.

5. The combination as set forth in claim 1 wherein each of said antifriction bearings in contact with said sleeve cap comprises an inner race and an outer race and said outer race is substantially in line contact with said sleeve cap.

6. The combination of a governor housing; a driven shaft mounted therein; said shaft engaging a spider which is driven thereby; a plurality of fly weights pivotally fastened to said spider to swing radially outwardly in response to rotation of said shaft and spider; each of said fly weights equipped with an antifriction bearing mounted so that the axes of rotation of said bearings intersect the axis of rotation of said spider; a sleeve cap mounted for rotation about the rotational axis of said shaft and spider and being engaged by said antifriction bearings which are in rolling contact with said sleeve cap when said spider rotates; one end of an axially floating plunger extending through said cap and being urged in axial movement in response to the rotation of said fly weights; the other end of said plunger adapted to engage the actuating mechanism of an electrical circuit control means and causing operation thereof as a result of the plunger's axial movement; a helical compression spring surrounding a portion of said plunger and supported to oppose the axial motion of said plunger toward said control means and thereby opposing also the outward motion of said fly weights; said spring having a substantially constant modulus of elasticity between minus 50 and plus 150 degrees F., and said plunger and spider being proportioned to maintain the working space of said spring substantially constant in response to thermal expansion.

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