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NORMAL FUEL CONTROL WITH ACCELERATION OVERRIDE CONTROL FOR GAS TURBINE ENGINE

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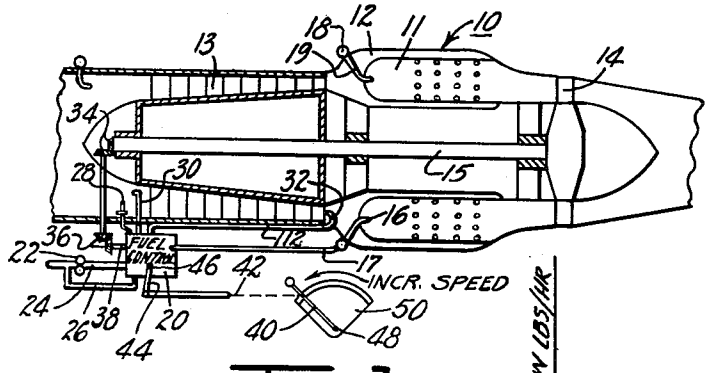


FIG. 1

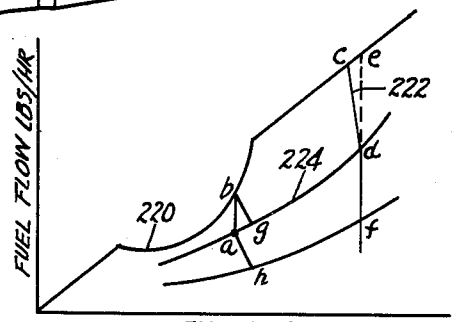


FIG. 3

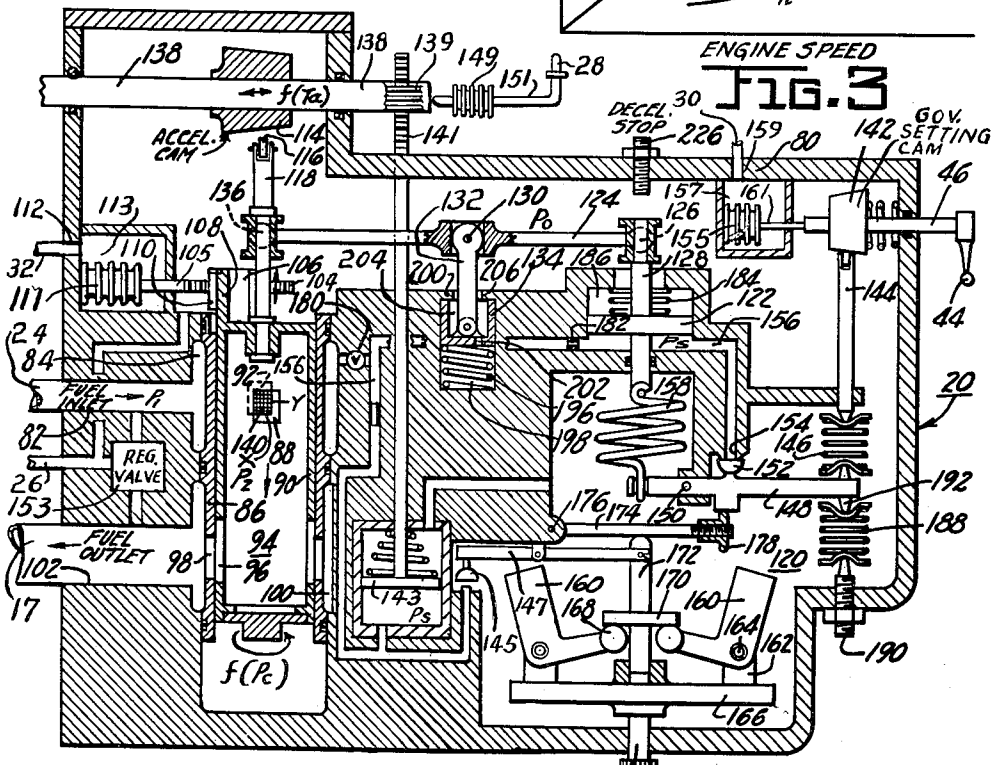


FIG. 2

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NORMAL FUEL CONTROL WITH ACCELERATION OVERRIDE CONTROL FOR GAS TURBINE ENGINE

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This invention relates to a fuel feeding device for combustion engines and more particularly to speed governing mechanism therefor; it has been found to be most particularly adapted for use with combustion engines of the gas turbine type.

It is common practice to utilize centrifugal governing mechanism of the mechanical fly-weight type to control a fuel metering or throttle valve in fuel control systems for gas turbine engines, the governing mechanism including a governor spring which is normally manually adjustable by the pilot to accelerate and decelerate the engine to a selected speed.

The most widely used type of governing mechanism for gas turbine engines has been the so-called proportional type, wherein any deviation in engine speed from an equilibrium setting or operating condition results in a fuel flow correcting opening or closing movement of the throttle valve which is proportional to the speed deviation. One of the primary advantages of this type of governor as compared, for example, with an isochronous or non-speed deviating type, relates to the fact that the above noted proportional action results in engine operating stability, since the valve does not travel to a fully open or fully closed position whenever a slight deviation in engine speed occurs, with subsequent hunting for an equilibrium location.

Heretofore, proportional type governors for gas turbine engines have been classifiable in two main groups, viz. the direct acting kind in which the fly weights and governor spring act directly on the throttle valve, and the servo actuated type in which the governing mechanism functions through a power boost travel type servo device to control the position of the valve. The former type is well illustrated in the Mock Patent No. 2,689,606, filed December 13, 1946, and the latter type in the copending application of Andrew A. Kuzmitz Serial No. 446,335, filed July 28, 1954 (common assignee), now abandoned.

One of the disadvantages of both of the above mentioned classes of proportional type governors is that maximum throttle valve travel is limited by the amount of displacement which can be imparted thereto by the movement of the centrifugal weights, either directly, as in the Mock patent, supra, or through servo mechanism, as in the Kuzmitz application, supra.

My invention overcomes this disadvantage by providing a servo powered proportional type governor in which the position of the throttle valve is almost wholly independent of centrifugal weight position. This has been accomplished by utilizing what may be termed a force type servo governor mechanism in which centrifugal weight force alone is converted into valve travel, as distinguished from a travel type servo governor mechanism in which centrifugal weight travel, resulting from a change in force output, is converted into valve travel.

This invention is illustrated in FIGURES 1 and 2 of the drawings in a fuel control device of the so-called three-dimensional cam type, which type of control is fully disclosed in my copending application Serial No. 248,402, filed September 26, 1951. One of the problems encountered in the use of such controls relates to excessive wear, over a period of time, of the contoured three-dimensional acceleration cam which results from excessive spring

loads and the like imposed on said cam by the cam follower during engine accelerations. My invention provides mechanism associated with the governor for relieving any such excessive load on the cam, thereby greatly increasing the useful life thereof.

One of the primary objects of this invention is to provide an improved engine speed governor wherein a speed selecting force output is directly converted into a fuel flow controlling travel.

Another object of this invention is to provide a force type servo governor for engines which is not subject to certain inherent limitations heretofore existent in direct acting and servo actuated proportional type governors.

A further object of this invention is to provide cam load limiting means in a combustion engine fuel control device.

Additional objects and advantages of this invention will become apparent to those skilled in the art in view of the following description taken in conjunction with the drawings, wherein;

FIGURE 1 is a sectional schematic view of a gas turbine engine and a diagrammatically illustrated fuel control system connected thereto;

FIGURE 2 is a sectional schematic view of an engine governor and associated mechanism adapted to be used in the fuel control device of FIGURE 1, in accordance with the invention; and

FIGURE 3 is a curve chart, illustrating the operational characteristics of the governor shown in FIGURE 2.

Referring now to FIGURE 1, a gas turbine engine is generally indicated at **10** and includes a series of annularly disposed combustion chambers **11** mounted in a casing having a header or air intake section **12**, and a compressor **13**, shown as of the axial flow type, which is driven by means of a turbine **14** through a shaft **15**. Each of the combustion chambers is provided with a burner nozzle **16** to which metered fuel is supplied under pressure by way of a conduit **17**, fuel manifold **18** and individual fuel lines **19**. The conduit **17** receives metered fuel from a fuel control device, generally indicated at **20** in FIGURE 1, which includes the applicant's governing mechanism as shown in FIGURE 2. A pump **22** supplies fuel under pressure to fuel control **20** through a conduit **24**, a portion of which fuel may be by-passed back to the pump inlet through a conduit **26**.

The fuel control **20** contains mechanism adapted to respond to compressor inlet temperature (T_a), as sensed by a temperature bulb **28**, compressor inlet pressure (P_a), picked up by a impact pressure pick up tube **30**, compressor discharge pressure (P_c), picked up at a second impact pressure pick up tube **32**, engine speed (N), which is transmitted to said control by means of a bevel gearing arrangement **34** and **36** and a governor drive shaft **38**, and to the position of a pilot's control lever **40**, which is mounted as a rotatable shaft **48** in a control quadrant **50**, and which is connected to the fuel control device **20** by means of a link **42**, a lever **44** and a shaft **46**.

Referring now to FIGURE 2, the governing mechanism is shown contained within a housing **80** of the fuel control device **20**, which receives fuel at a pressure P_1 from conduit **24** is an inlet passage **82** and an annular chamber **84** formed between the housing and a fixed cylindrical sleeve member **86**, and which discharges fuel to the metered fuel conduit **17** at a pressure P_2 by way of a port **88**, formed in the surface of a reciprocable and rotatable hollow cylindrical metering valve member **90** and adapted to variably register with a port **92** formed in sleeve **86**, a valve chamber **94**, valve ports **96**, sleeve ports **98**, an annular chamber **100** and a fuel outlet passage **102**.

The angular position of metering valve **90** is at all times controlled as a function of compressor discharge

pressure by a geared rack member **104** and a gear sector **106** fixed on an extension **108** of valve **90** and having splined gear teeth **110** in mesh with the gear teeth of rack **104**. The rack **104** is formed on a rod **105** which is fixedly secured to the movable end of an evacuated bellows **111**. The bellows **111** is mounted in a sealed chamber **113** which receives compressor discharge pressure P_c from the impact pressure pick up tube **32** via a conduit **112**. The bellows **111** responds to the compressor discharge pressure P_c in chamber **113** and rotates valve **90** through rack **104** and sector **106** such that a given angular position of valve **90** is maintained for each value of compressor discharge pressure P_c.

The axial position of metering valve **90** within sleeve **86** is controlled either by a contoured three-dimensional acceleration cam **114**, which is adapted to cooperate with a cam follower and valve rod **116** and **118** during an acceleration of the engine, or by all-speed governing mechanism contained within housing chamber **120** and connected to the metering valve by a power servo piston **122** and a walking beam **124**. The walking beam **124** is adapted to be variously fulcrumed at a ball joint connection **126** on a servo piston rod **128**, at a pivot connection **130** to a rod **132** of a cam load limiting piston **134**, or at a ball joint connection **136** on the cam follower rod **118**, in a manner to be described. The axial position of valve **90** is at all times determined by that mechanism which demands the least quantity of fuel, i.e. the acceleration cam and the governing mechanism are arranged in mutually overriding relation, as hereinafter described, so that that one which tends to fix the smallest area at metering ports **88**, **92** controls.

The acceleration cam **114** is mounted on a shaft **138** which is rotatably and axially actuable as a function of certain engine operating parameters such as engine speed and compressor inlet temperature, respectively. The shaft **138** carries a pinion **139** fixed secured thereto, said pinion being engaged with and rotated by a rack **141** formed on a rod fixedly secured to a pitson **143**. The piston **143** is responsive to a servo pressure P_s which is controlled by a half ball servo valve **145**. The half ball servo valve **145** is positioned as a function of engine speed by a pivot lever **147** operatively connected between said half ball servo valve and the centrifugal weights **160**. The shaft **138** is positioned axially by a bellows **149** which has an operative connection with the temperature bulb **28** by means of a tube **151**, the bellows and tube being filled with suitable fluid or material responsive to temperature changes registered by the bulb **28**. The structural details and mode of operation of the above mentioned mechanism for axially and rotatably actuating the shaft **138** is conventional and will be easily understood by those skilled in the art. The acceleration fuel flow schedule may be controlled by the cam **114** and rack **104** to vary as a predetermined function of P_c, T_a and N, such that maximum allowable turbine inlet temperature is substantially maintained throughout a predetermined portion of the acceleration schedule, and the phenomenon known as compressor surge or stall is not encountered. The angular and axial position controls for valve **90** at all times cooperate to define a metering area **140** formed at the portion of registry between ports **88** and **92**. For example, as compressor discharge pressure increases, rack **104** actuates gear sector **106** to move valve **90** in a clockwise direction, as viewed from the top thereof, increasing dimension *x* of the metering port in proportion to the increase in said pressure, and variation in any engine operating parameter which effects a change in the axial position of metering valve **90** varies dimension *y* of port **140**.

Preferably, a constant fuel pressure differential is at all times maintained across metering port **140** by a regulator valve means generally shown at **153** and preferably of the type disclosed in U.S. Patent No. 2,689,606, granted September 21, 1954, to Frank C. Mock and assigned to the present assignee, whereby the flow through said port is always a predetermined function of only those engine operating parameters which control the angular and axial positions of valve **90**.

The governor mechanism contained within chamber **120** connects the pilot controlled levers **40** and **44** to the power servo piston **122** by way of the shaft **46**, a governor setting cam **142** mounted on shaft **46** for rotation therewith and adapted to be axially actuable by compressor inlet pressure and/or temperature responsive mechanism, a cam follower rod **144**, a speed selecting governor spring **146** mounted between spring retainers which abut the rod **144** and a servo valve lever **148**, fulcrumed at **150**, a half-ball type servo valve **152** connected to the lever **148** and cooperating with a discharge orifice **154** in a servo pressure passage **156**, a governor feedback tension spring **158** resiliently connecting the servo piston rod **128** with the left end of servo lever **148**, and a pair of rotatable centrifugal weights **160** mounted on brackets **162** at pivots **164** for rotation with a mounting plate **166** which is keyed to the governor drive shaft **38**, said centrifugal weights having foot members **168** in abutment with a flange **170** of a force transmitting rod **172**, which in turn abuts a lever **174** pivoted at **176** for transmitting a moment of the force output of weights **160** to servo lever **148** at a governor slope adjusting nut **178**. The compressor inlet pressure responsive mechanism may take the form of an evacuated bellows **155** suitably mounted in a sealed chamber **157** which receives compressor inlet pressure from impact pressure pick up tube **30** via a conduit **159**. A rod **161** extends from the movable end of bellows **155** into engagement with the shaft **46** such that a given axial position of the cam **142** is maintained for each value of the compressor inlet pressure. The bellows **155** may be replaced by temperature sensing mechanism similar to the afore-mentioned temperature sensing bulb **28**, tube **151** and bellows **149** in which case the shaft **46** could be actuated as a function of compressor inlet temperature T_a. The compressor inlet pressure and temperature responsive mechanism may be combined to form a conventional density sensitive apparatus, not shown, such that the shaft **46** is positioned as a function of compressor inlet pressure and temperature.

The servo pressure (P_s) passage **156** connects the P_1 pressure chamber **84** to chamber **120**, which is connected to pump by-pass conduit **26** by a passage, not shown, at pump inlet pressure P_0. A discharge pressure regulator valve **180** in passage **156** is adapted to maintain a constant pressure in said passage upstream of a calibrated restriction **182**, the pressure P_s beneath power piston **122** therefore varying solely as a function of the area ratio between restriction **182** and orifice **154**. A power return spring **184** and fuel at pump inlet pressure in a chamber **186** opposes P_s pressure and spring **158** across piston **122**.

A servo balance spring **188**, adjustable by means of a screw **190**, is utilized to act on lever **148** and **192** for the purpose of compensating for variations in manufacturing tolerances between the designed rates of springs **146** and **158**, and for insuring that said latter springs will sustain a pre-load even when the engine is at rest; this latter function insures that neither spring **146** nor **158** will ever be extended to a free height or no-load condition. If either of said springs were ever to reach a no-load condition they might become disconnected from their respective retaining means. The effective force output of the balance spring **188** on the servo system remains substantially constant, irrespective of variations in engine operating conditions, as the result of an arrangement which effects a total range of movement of servo valve **152** amounting to only a very few thousandths of an inch.

During all conditions of engine governing and equilib-

rium operation the fulcrum 130 of walking beam 124 is maintained in the position shown by a cam load limiting spring 196, which is contained in a cylindrical chamber 198 formed on one side of piston 134. The spring 196 normally maintains piston 134 in abutment with a stop 200 formed in the housing 80, and the chamber 198 is vented to P_0 pressure in chamber 120 by an opening 202 in piston 134, a chamber 204, and an aperture 206 in stop 200.

Operation

Referring now to FIGURE 3, sea level acceleration, governing, and steady state engine characteristics are qualitatively illustrated by the curves 220, 222 and 224, respectively, on the curve chart which is plotted on the coordinates of fuel flow in pounds per hour versus engine speed in r.p.m. The basic contour of the characteristic seal level acceleration curve 220 is fixed by the contour of the acceleration cam 114, which is actuable as a function of certain engine operating parameters, as hereinbefore described whereas the height or level of said curve above the engine speed coordinate is primarily controlled by the compressor discharge pressure responsive mechanism. The governor fuel cut-off curve 222 illustrates the operation of my force servo type proportional governor from the acceleration curve to a selected speed point on the engine steady state operating curve. The steady state curve 224 illustrates engine fuel flow demand at all available engine speeds during sea level operation.

Assume that the engine has been started and accelerated to the steady state operating point *a* in the mid-speed range. In this condition of operation the pilot's control lever 40 will be positioned approximately mid-way between the ends of quadrant 50, in which position link and lever 42 and 44 position the governor setting cam 142 on shaft 46 so as to impose a speed selecting load on governor spring 146 through cam follower rod 144 which demands that engine operating speed existent at point *a*. In this condition of operation the half-ball servo valve 152, being maintained in fixed position by an existent force balance on lever 148, controls a servo pressure P_s in passage 156 which imposes a force on servo piston 122; this force is balanced by the opposing forces of the governor feedback spring 158, the power return spring 184, and the pump inlet pressure in chamber 186, whereby the position of walking beam 124 is fixed thereby controlling the axial position of metering valve 90 such that the area of metering port 140 controls that quantity of fuel flow to the nozzle 16 which is necessary to maintain engine speed as set at point *a*.

The forces acting on servo lever 148 in a direction which tends to move servo valve 152 in an opening direction, i.e. in a direction which would result in an accelerating fuel flow to the engine, consist of the moment forces of governor spring 146, feedback spring 158, and the fuel pressure force on servo valve 152 at orifice 154, all acting about fulcrum 150. These moment forces are opposed and balanced by the moment force of the centrifugal weights 160, which varies as the square of engine speed, and the substantially constant moment force of the servo balance spring 188. At any given speed, the engine driven centrifugal weights 160 generate an effective force output on lever 148 which is equal to the weight force output times the ratio of the lever arm between rod 172 and fulcrum 176 to the lever arm between nut 178 and fulcrum 176. The force output of weights 160 is therefore diminished at lever 148 by the lever ratio factor, said factor being adjustable by nut 178 to adjust the governor cut-off slope as will be hereinafter described.

The design of the area of servo piston 122, of the lever ratio of walking beam 124 across fulcrum 130 between pivots 126 and 136, of the area relation between restriction 182 and orifice 154, and of the rate of feedback spring 158, with a given configuration of metering ports 88, 92 enables full governor cut-off action to be effected at any speed with extremely small movement of servo

valve 152, the maximum movement of which will not exceed .004 or .005 inch. This arrangement enables the governing mechanism to actuate metering valve 90 through its total range of movement with substantially no change in the radius of rotation of governor weights 160, whereby the governor mechanism inherently lends itself to a large degree of design versatility with respect to meeting the operating characteristics of engines having widely different fuel requirements, inasmuch as metering valve 90 may be controlled throughout its axial travel range with a negligible change in position of the weights 160. From this it is also apparent that a substantially constant force is imposed on lever 148 by the balance spring 188, irrespective of variations in loading on governor spring 146 at different selected speeds.

Assume now that the pilot actuates control lever 40 in a counterclockwise direction to select, say, a maximum operating speed for the engine, as denoted by point *d* on curve 224, and as illustrated by the position of lever 40 in FIGURE 1. As the pilot rotates lever 40 to the indicated maximum speed position, the fuel control unit reacts substantially instantaneously as follows: Governor setting cam 142 is rotated on shaft 46 to a position at which the cam rise imposes maximum compression on governor spring 146 to select that speed at which the governor weight force and other forces acting on servo lever 148 will be in equilibrium, as illustrated at point *d*; servo lever 148, being no longer in balance, rotates a minute amount in a clockwise direction to increase the area ratio between orifice 154 and restriction 182, thereby instantaneously decreasing servo pressure P_s and unbalancing the forces across servo piston 122; the force differential across piston 122 actuates the piston downwardly, and the walking beam 124 is rotated about fulcrum 130 in a clockwise direction until cam follower 116 comes into contact with acceleration cam 114, in which position the metering area 140 has increased to effect an increase in fuel flow from point *a* on curve 224 to point *b* on curve 220.

The amount of compression imparted to governor spring 146 which is in excess of that necessary to effect an acceleration to point *g* on curve 224 results in a continued downward movement of piston 122, following contact of cam follower 116 with acceleration cam 114, about a new fulcrum 136, as pivot 130 and piston 134 move downwardly from the normally fixed position thereof against spring 196 until piston 122 reaches a position at which the decreased loadings on springs 184 and 158 plus the low pressure in chamber 186 again balances the force of pressure P_s; the loading force imposed on the acceleration cam through follower 116 is thereby held to a relatively small amount, which results only from the degree of compression imparted to the light spring 196.

The increase of fuel flow from point *a* to point *b* results in an excess over that required to run the engine at the existing speed, and acceleration proceeds along curve 220, as determined by the contour of cam 114 which effects an opening movement of metering valve 90 in an axial direction; at the same time compressor discharge pressure responsive rack 104 effects an opening rotational movement of valve 90. The combined axial and rotational movement of valve 90 results in a fuel flow versus speed characteristic such as is illustrated by the curve segment *bc*. As acceleration of the engine proceeds from point *b*, and the rate of opening movement of valve 90 is determined by the rotating cam 114, it is apparent that the fulcrum of walking beam 124 will shift from pivot 136 to pivot 126, as spring 196 actuates piston 134 and pivot 130 upwardly until the piston is again in contact with the stop 200. During such upward movement of piston 134, beam 124 is rotated in a clockwise direction about pivot 126, contact between follower 116 and the decreasing rise of cam 114 being thereby positively maintained, while the engine is accelerating, with a cam load-

ing force which varies within chosen design limits as a function of the rate of spring **196**. It is apparent that selection of a low rate cam loading spring will insure against exceesive loads being placed on the cam **114** during acceleration of the engine, with resultant long cam life.

At the speed indicated at point *c*, the centrifugal weights **160** have generated a force output which is sufficient to begin to overcome the force moments of governor spring **146** and feedback spring **158**, said spring **158** being, at this time, in a relatively relaxed condition. Servo valve **152** therefore moves slightly upwardly to reduce the area ratio between orifice **154** and restriction **182**, thereby producing an immediate increase in pressure P_s which unbalances servo piston **122** in an upward direction, resulting in a counterclockwise rotation of walking beam **124** about fulcrum **130** and a movement of follower **116** away from acceleration cam **114** to reduce the dimension *y* of metering port **140**. As piston **122** moves upwardly, feedback spring **158** elongates and produces an increasing moment of force on servo lever **148** in the same direction as that produced by governor spring **146**. This action results in continued acceleration of the engine as fuel flow decreases along the governor break curve **222**, inasmuch as the increasing force output of spring **158** necessitates a proportional increase in engine speed, so that the effective force output of weights **160** plus the substantially constant force output of balance spring **188** may balance the effective force outputs of spring **146** and **158** at equilibrium operation. In other words, neglecting the substantially constant forces of springs **146** and **188**, and also the small hydraulic force acting on valve **152**, the increasing load on feedback spring **158** feeds into the governor mechanism a demand increase in engine speed which is proportional to the decrease in fuel flow along the governor break curve **222**. This proportional governing characteristic imparts a negative slope to the fuel cut-off curve which is a function of the rate of the feedback spring, and permits the engine to reach a stable equilibrium point of operation.

An unstable condition which is generally known as hunting in the governor art, would result if the feedback spring were not included in the governor mechanism; i.e. without spring **158** a substantially vertical or isochronous governing characteristic, as illustrated by curve *edf*, would result due to the relatively constant force output of governor spring **146** at any given selected speed. Again, without spring **158** a slight variation in engine speed above or below that selected by the pilot would result in movement of servo valve **152** to cause potentially maximum movement of servo piston **122** and metering valve **90** in seeking a condition of equilibrium.

When the fuel flow required to run the engine is being metered at point *d*, no accelerating engine torque exists, and the governing mechanism is in a state of equilibrium; i.e. the moments of forces acting on lever **148** in a clockwise direction exactly balances the opposing counterclockwise moments, and the positions of servo piston **122** and metering valve **90** are fixed. If, for any reason, the engine should tend to increase or decrease speed from point *d*, as with a decrease or increase in the density of the surrounding atmosphere, respectively, fuel flow ould tend to vary along curve **222** to reestablish equilibrium on the new engine operating curve.

With any given fixed design rate of spring **158**, the slope of the governor break curve may be manually adjusted as desired by nut **178**, inward adjustment on lever **174** effecting an increase in the slope of the governor cut-off curve as a result of the increased effective force output of weights **160** with a given increase in engine r.p.m., and outward movement of nut **178** on lever **174** effecting a decreased slope of said curve.

The relative arrangement of parts, as shown in FIGURES 1 and 2, approximates that which would exist if an engine were being controlled by my device to operate at point *d*. It should be remembered that the fulcrum **130** of the walking beam **124** maintains a fixed position, as shown, throughout governor cut-off and equilibrium operation at any selected speed, the fulcrum of beam **124** shifting to pivot **136** only during downward motion of piston **134** against spring **196** following initial contact of cam follower **116** with the acceleration cam, and that said fulcrum shifts from pivot **136** to pivot **126** only during upward movement of piston **134** to the position indicated during an acceleration of the engine to the beginning of governor cut-off action.

A deceleration of the engine is effected by relaxing the loading of spring **146** by counterclockwise movement of pilot's lever **40**, which unbalances the forces acting on lever **148** and effects an upward movement of piston **122** until piston rod **128** contacts a deceleration stop **226**; as beam **124** rotates about fulcrum **130** in a counterclockwise direction to the position determined by stop **226**, the axial position of metering valve **90** and the *y* dimension of metering area **140** are fixed. Fuel flow immediately decreases along the curve *df*, and the engine then begins to decelerate along the curve *fh* as metering area dimension *x* decreases with decreasing compressor discharge pressure. At point *h* the force output of weights **160** has decreased to such an extent that spring **146** begins to overcome said weight force output and move servo valve **152** slightly open to cause an opening movement of the metering valve **90**, which increases fuel flow as the engine speed decreases along the proportional governor break curve *ha*. At point *a* the control again reaches a condition of equilibrium.

From the above description, it is now apparent that I have provided a force type governor servo mechanism having a built in proportional governor break characteristic with means for adjusting the slope thereof as desired, plus means for minimizing the loading to which an acceleration cam is subject during an acceleration of the engine.

It will be apparent to those skilled in the art that various changes in the structure and relative arrangement of parts may be made without departing from the scope of my invention.

I claim:

1. In a fuel feed and power control system for a gas turbine engine having a burner, a fuel conduit for conducting fuel to the burner, valve means for regulating the flow of fuel through said conduit, a movable maximum flow stop operatively connected to said valve means for limiting the rate of valve opening movement during an acceleration of the engine, and a force type servo governor mechanism operatively connected to said valve means for controlling the flow regulating function of said valve means as an inverse function of engine speed including servo valve means, resilient means operatively connected to said servo valve means for imposing an engine speed selecting force thereon, the force output of said resilient means being substantially constant during governor cut-off action to any given selected engine speed, an engine speed responsive element operatively connected to said servo valve means for transmitting a force thereto which opposes said speed selecting force and which varies as a function of existing engine speed, the force output of said speed responsive element varying substantially only with variations in engine speed during governor cut-off action to a selected speed, and a yielding means operatively connecting said servo valve means to said governor valve means for transmitting a force to said servo valve means which varies as a function of the position of said governor valve means during governor cut-off action to a selected engine speed.

2. In a fuel feed and power control system for a gas turbine engine having a burner, a fuel conduit for conducting fuel to the burner, valve means for regulating the flow of fuel through said conduit, a movable maximum

flow stop for limiting the rate of valve opening movement during an acceleration of the engine, and engine speed governor mechanism for controlling the flow regulating position of said valve means during equilibrium operation of the engine including a walking beam, first pivot means connecting said valve means to said beam, second pivot means connecting resilient means to said beam, and third pivot means connecting engine speed responsive means to said beam, said second pivot means being normally adapted to function as a fulcrum for said beam, and said fulcrum being shiftable to said first pivot means following the initiation of control of said valve means by said movable stop.

3. A fuel feed and power control system as claimed in claim 2 wherein said fulcrum is shiftable from said first pivot means to said third pivot means during valve controlling movement of said stop.

4. A fuel feed and power control system as claimed in claim 3 wherein said fulcrum is shiftable from said third pivot means to said second pivot means during actuation of said valve means away from said stop by said engine speed responsive means.

5. In a fuel feed and power control system for a gas turbine engine having a burner, a fuel conduit for conducting fuel to the burner, valve means for regulating the flow of fuel through said conduit, valve stop means operatively connected to said valve means during an acceleration of the engine, and engine speed governing means for controlling the flow regulating position of said valve means as an inverse function of engine speed during governor cut-off operation following an acceleration of the engine including a walking beam, first pivot means connecting said valve means to said beam, second pivot means operatively connecting yielding means to said beam, and third pivot means operatively connecting engine speed responsive means to said beam, said second pivot means being adapted to function as a beam fulcrum during governor cut-off operation, and means for adjusting said engine speed responsive means to select an operating speed for the engine, said fulcrum being shiftable to said first pivot means following selection of a new engine operating speed by said adjusting means.

6. A fuel feed and power control system as claimed in claim 5 wherein said fulcrum is shiftable from said first pivot means to said third pivot means during an acceleration of the engine.

7. In a fuel feed and power control system for a gas turbine engine having a burner, a fuel conduit for conducting fuel to the burner, valve means for regulating the flow of fuel through said conduit, a contoured acceleration cam adapted to be operatively connected to said valve means during an acceleration of the engine, and an engine speed responsive governor mechanism operatively connected to said valve means for controlling the flow regulating position of said valve means during governor cut-off and equilibrium operation of the engine, said last mentioned operative connection including lever means having a movable fulcrum, said fulcrum being movable from a first to a second position along said lever means following actuation of said valve means into an operative connection with said cam means.

8. A fuel feed and power control system as claimed in claim 7 wherein said fulcrum moves from said second position to a third position whenever the position of said valve means varies with the contour of said cam.

9. An engine speed governor comprising governor valve means for controlling the flow of motive fluid to the engine, engine speed responsive means operatively connected to said valve means for controlling the flow regulating position thereof as an inverse function of engine speed, and means operatively connected to said speed responsive means for selecting an operating speed for the engine, said first mentioned operative connection including a walking beam having a fulcrum, first pivot means operatively connected to walking beam and said valve means, second pivot means operatively connected to said walking beam and said engine speed responsive means, and third pivot means operatively connected to said walking beam intermediate said first and second pivot means and supported by resilient means, said fulcrum being shiftable between said first, second and third pivot means during an acceleration of the engine from a first to a second selected engine speed.

10. In a fuel fed and power control system for a gas turbine engine having a burner, a fuel conduit for conducting fuel to the burner, valve means for regulating the flow of fuel through said conduit, a maximum flow stop adapted to be operatively connected to said valve means during an acceleration of the engine, engine speed responsive means operatively connected to said valve means for controlling the flow regulating function thereof as an inverse function of engine speed, said operative connection including lever means connected to said engine speed responsive means, means operatively connected to said speed responsive means for selecting an operating speed for the engine, said lever means having a fulcrum, and resilient means operatively connected to said fulcrum for maintaining a fixed position thereof during control of said valve means by said governor means, said resilient means being adapted to permit a change in the position of said fulcrum during control of said valve means by said maximum flow stop, following the initiation of an acceleration of the engine by said speed selecting means for limiting the force loading on said stop.

References Cited in the file of this patent

UNITED STATES PATENTS

2,253,963	Van Nest	Aug. 26, 1941
2,576,352	Neal	Nov. 27, 1951
2,604,756	Greenland	July 29, 1952
2,638,992	Lindquist	May 19, 1953
2,668,414	Lee	Feb. 9, 1954
2,675,674	Lee	Apr. 20, 1954
2,712,219	Warne	July 5, 1955
2,720,752	Chandler et al.	Oct. 18, 1955
2,759,549	Best	Aug. 21, 1956
2,764,868	Watson et al.	Oct. 2, 1956