Nov. 17, 1953

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HYDRAULIC SERVO MECHANISM FOR
GAS TURBINE FUEL REGULATORS

2,659,349

Filed July 30, 1952

2 Sheets-Sheet 1

Fig. 1.

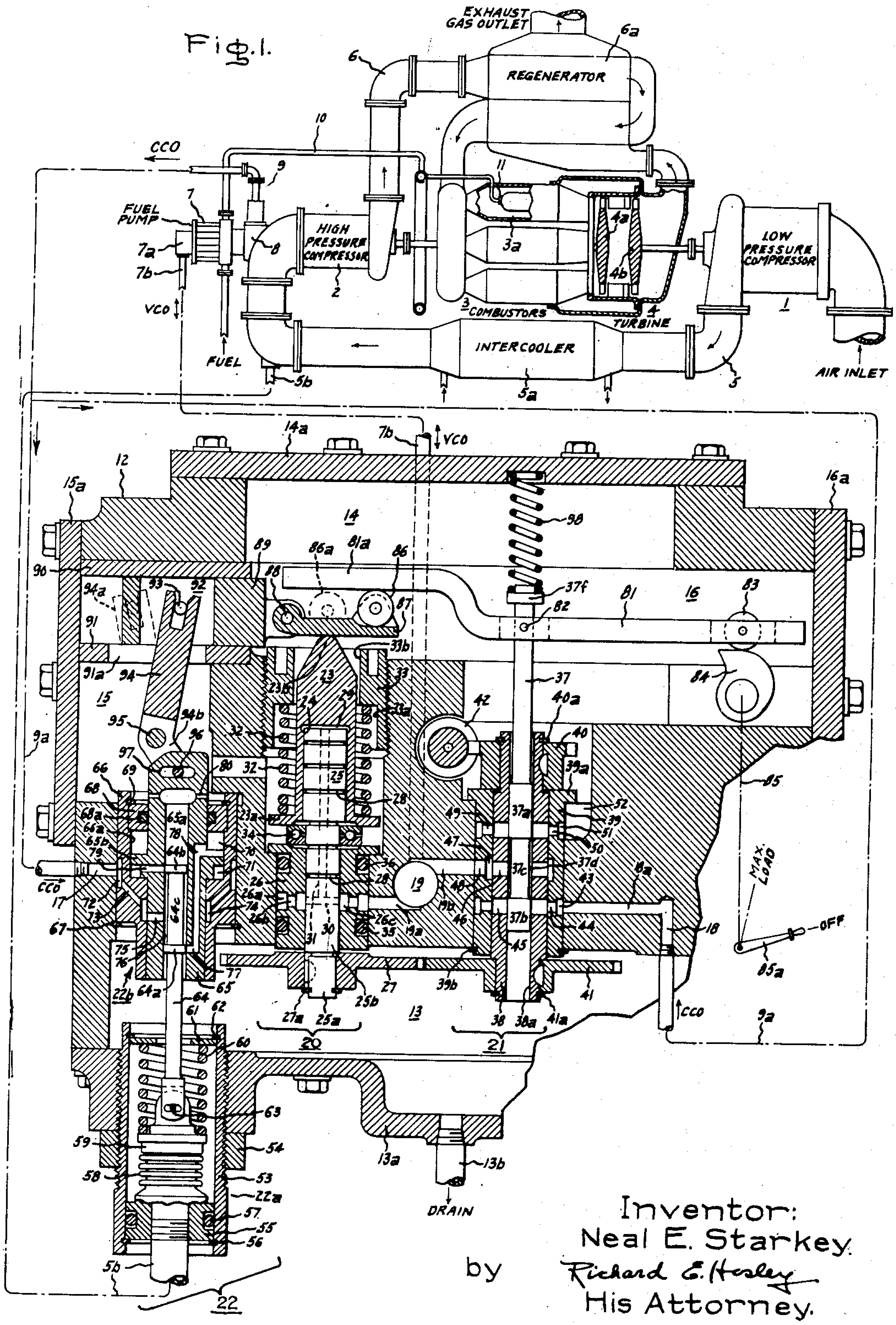

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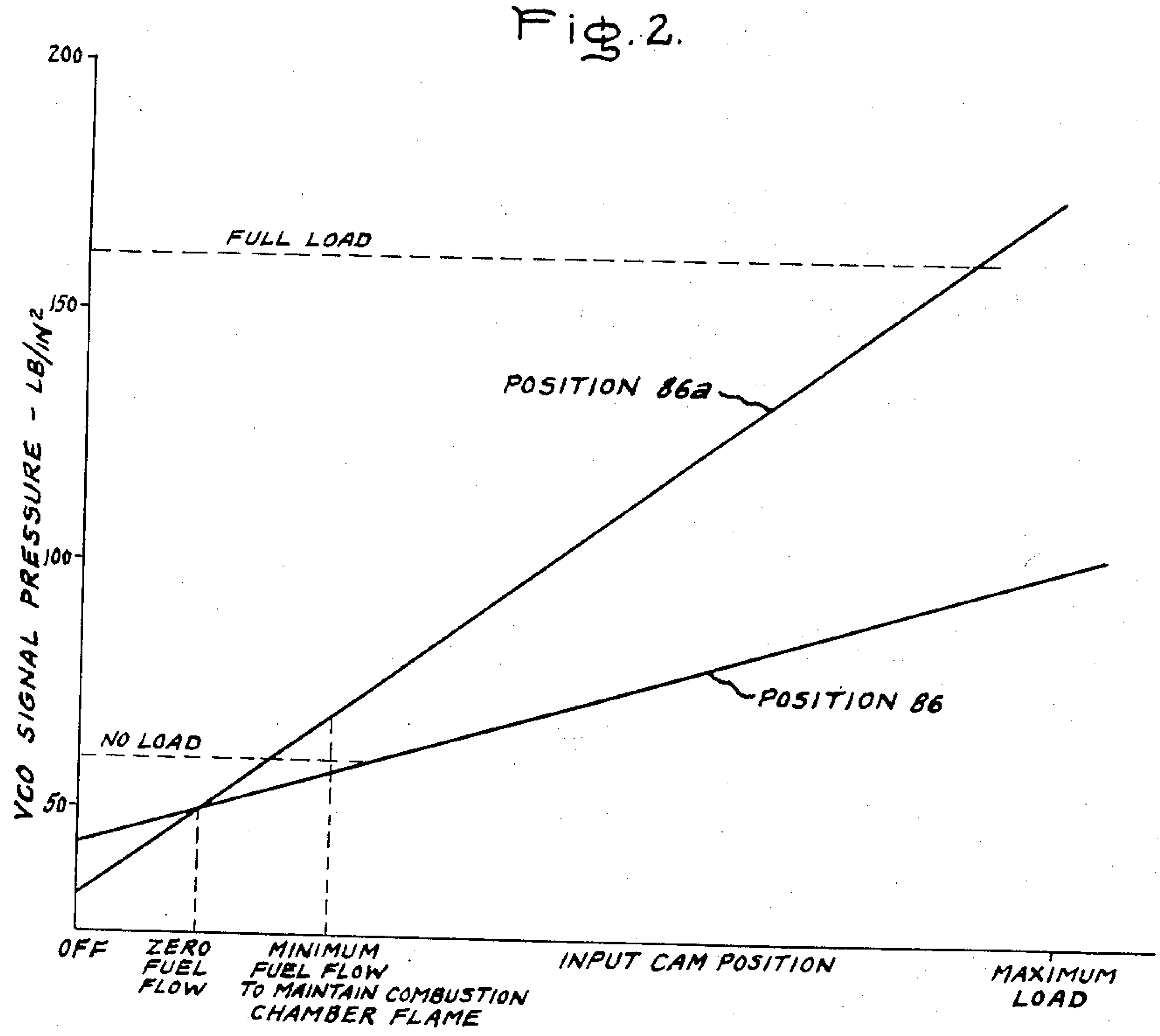
Fig. 2.



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2,659,349

UNITED STATES PATENT OFFICE

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HYDRAULIC SERVO MECHANISM FOR GAS TURBINE FUEL REGULATORS

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Application July 30, 1952, Serial No. 301,676

2 Claims. (Cl. 121—41)

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This invention relates to hydraulic servo-devices, particularly to a combination of hydraulic servo-mechanisms for use in the fuel regulating system of a thermal powerplant, such as a gas turbine.

While not necessarily limited thereto, this invention is particularly adapted for use in a gas turbine fuel system of the general type shown in the copending application of Bruce O. Buckland, Serial No. 183,332, filed September 6, 1950 and assigned to the same assignee as the present application. Generally, such a fuel system comprises a variable displacement pump connected to supply a liquid fuel such as diesel or "Bunker C" oil to a plurality of nozzles in the gas turbine combustion system. The displacement of the fuel pump is automatically varied by a hydraulic servo-device actuated by a variable control oil pressure signal, which is supplied by a regulator containing a complex combination of various condition-responsive servo-devices cooperating to produce the variable control oil pressure. Hereinafter, this variable pressure signal which acts on the fuel pump servo to determine the rate of fuel supply will be referred to as the "VCO pressure."

Another gas turbine fuel system of a type which may profitably incorporate the present invention is illustrated in United States patent to N. E. Starkey et al., 2,558,592, issued June 26, 1951, and assigned to the same assignee as the present application.

The general arrangement of the specific powerplant for which the present invention was developed, and other details of the regulating system therefor are disclosed in the United States patent of George R. Fusner and Chapman J. Walker, No. 2,617,253, issued November 11, 1952, and assigned to the same assignee as the present application. This powerplant is characterized by a two-stage gas turbine with mechanically independent rotors, one of which drives a low pressure compressor in series with a high pressure compressor driven by the other turbine rotor. The fuel pump is also driven from the high pressure compressor rotor. This means that, for a given setting of the fuel pump regulator, the rate of fuel supply will be a function of high pressure compressor shaft speed. As will be appreciated by those skilled in the art, the volume rate of air flow through the high pressure compressor to the combustion system is a function of the rotational speed of the high pressure compressor. But, since the low pressure compressor is driven at a variable speed by its independent turbine rotor, the weight

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rate of air flow through the system is also a function of the density or pressure at the inlet to the high pressure compressor. Therefore, in order to make the fuel regulating system accurately responsive to the weight flow of air, it is necessary that the fuel regulating system incorporate means for sensing the pressure at the inlet to the high pressure compressor.

Accordingly, the object of the present invention is to provide an improved hydraulic servo-mechanism for a gas turbine fuel regulating system of the type described including means for modifying the VCO fuel rate pressure signal in accordance with the inlet pressure of the high pressure compressor.

A further object is to provide an improved hydraulic regulating component of the type described, in which the VCO pressure is balanced directly against a calibrated "main-spring," and the modifying effect of the high pressure compressor inlet pressure is introduced at a point between the VCO piston and the pilot valve which determines the VCO pressure, in order to improve the sensitivity of the mechanism by reducing friction effects in the compressor pressure responsive modifying mechanism.

Other objects and advantages will become apparent from the following description taken in connection with the accompanying drawings, in which Fig. 1 is a diagrammatic representation of a gas turbine powerplant with a fuel system having hydraulic servo-mechanism in accordance with the invention, the servo-mechanism being shown in section in more detail, and Fig. 2 is a graphical representation of the performance of the regulating mechanism to show the effect of the invention.

Generally, the invention is practiced by providing a hydraulic piston maintained in equilibrium condition by the VCO pressure acting against a calibrated spring, with a pilot valve for establishing the VCO pressure, and a compressor pressure responsive bellows connected to produce a modifying effect on the linkage which connects the VCO piston with the pilot valve.

Referring now more particularly to Fig. 1, the invention is shown as applied to a gas turbine powerplant comprising a low pressure compressor **1**, connected in series with a high pressure compressor **2**, a combustion system **3**, and a two-stage turbine **4**. The conduit **5** connecting the compressors contains an intercooler **5***a*, and the conduit **6** connecting the high pressure compressor with the combustion system contains a regenerator **6***a* in which waste heat from the exhaust gas is transferred to the high pressure air on its way to the combustors. It is to be particularly observed that the two-stage turbine 4 comprises a first stage rotor 4a directly coupled to the high pressure compressor rotor, and a mechanically independent second-stage rotor 4b directly coupled to the low pressure compressor rotor.

The variable displacement fuel pump is shown at 7 as connected to an accessory drive pad 8 and driven at a speed proportional to the speed of the high pressure compressor by suitable gearing (not shown). Also driven from the accessory pad 8 is a second positive displacement pump 9, the function of which is to provide hydraulic operating liquid for the fuel regulating system. It will be understood that the pump 9 furnishes control oil at a constant pressure, as determined by suitable pressure regulating valve means (not shown), and this constant pressure control oil will be referred to hereinafter as "constant control oil" (CCO).

The variable pressure signal (VCO) for altering the displacement of the fuel pump 7 is communicated to a hydraulic servo-device indicated diagrammatically at 7a in Fig. 1. The exact mechanism by which this servo determines the stroke of the fuel pump need not be described here, being disclosed more particularly in the above-mentioned application of B. O. Buckland, Serial No. 183,332. In order to understand the present invention, it is necessary only to note that the VCO pressure supplied to the servo-device 7a varies the stroke of the fuel pump 7 so that the fuel supply rate to the combustion system is a joint function of the VCO signal pressure and the high pressure compressor shaft speed. In Fig. 1, the fuel pump is shown connected by a conduit 10 with a spray nozzle 11 in combustor 3a. It will of course be understood that similar fuel supply conduit means are provided for the other combustors. The details of the combustion chambers, fuel nozzles, and the arrangement of piping through which the fuel pump supplies oil to the combustors are not material to this invention.

Referring now to the hydraulic servo-mechanism to which the present invention specifically relates, it will be seen in Fig. 1 that the housing 12 defines a bottom chamber 13 closed by a cover plate 13a, a top chamber 14 closed by a cover plate 14a, a left side chamber 15 with a cover plate 15a, and a right side opening 16 closed by cover plate 16a. The constant pressure operating liquid is communicated by the CCO supply conduit 9a to an inlet port 17 in the left-hand side of the housing, and a second inlet port 18 in the bottom of the housing. The VCO signal pressure, which is the "useful output" of the device, is communicated to the fuel pump servo 7a by the VCO conduit 7b, which communicates as shown in dotted lines with a port 19 located approximately at the geometric center of the housing 12.

The hydraulic servo-device comprises the VCO piston assembly indicated at 20, the VCO pilot 21, and the compressor pressure responsive servo indicated generally at 22. Generally stated, the function of these components is that the pilot 21 controls the supply of CCO operating liquid to the VCO piston 20, and the compressor pressure compensator 22 alters the characteristics of the linkage which connects the VCO piston with the VCO pilot.

Referring now more particularly to the construction of the VCO piston assembly 20, it will be seen that the movable piston member 23 is actually a plunger member defining a cylindrical recess 24 in which is slidably disposed a piston member 25. Piston 25 does not move longitudinally relative to the housing 12, but is rotatably supported in a bushing member 26. The projecting end portion 25a carries a gear 27, the function of which will be seen hereinafter. Piston 25 may be provided with a plurality of circumferential grooves 28 for improving hydraulic balance, in a manner which will be understood by those skilled in the art. The piston 25 and the moving cylinder member 23 cooperate to define a pressure chamber 29 to which the VCO signal pressure is communicated by a transverse drilled hole shown in dotted lines at 30 communicating with a longitudinal drilled hole 31.

The VCO pressure existing in chamber 29 produces an upward force on the VCO piston 23 which is balanced against a carefully selected coil spring 32. This spring surrounds piston 23 with the lower end thereof engaging a radial end flange 23a, the upper end of the spring seating in a cylindrical recess 33a of a threaded bushing member 33. It will be apparent in Fig. 1 that the moving VCO piston 23 passes freely through a cylindrical opening 33b in bushing 33, with generous radial clearance therebetween. Thus, it will be seen that piston 23 is positioned freely by the VCO pressure in chamber 29 balanced against the calibrated spring 32. The force exerted by spring 32 may of course be adjusted somewhat by varying the depth to which the threaded bushing 33 is screwed down into the housing portion receiving it. The lowermost position of piston 23 is determined by engagement of the lower end flange 23a, with a stop member 34, which may be an anti-friction bearing slipped over the reduced diameter piston portion 25b before piston 25 is inserted in the bushing 26 and the gear 27 assembled. It will be apparent in Fig. 1 that gear 27 is keyed to the shaft end portion 25a, and may be retained by any suitable means such as the snap-ring 27a.

It is also to be noted that the VCO liquid is supplied to an annular groove 26a in the outer circumference of bushing 26 by way of a passage 19a communicating with the VCO port 19. From the annular channel 26a, liquid is communicated by a plurality of circumferentially spaced ports 26b to an internal annular groove 26c. It will be obvious that this supply groove 26c is always in free communication with the transverse drilled hole 30 in the piston portion 25b. Leakage from channel 26a through the clearances between bushing 26 and the housing is prevented by a pair of O-ring seals 35, 36, the arrangement of which will be obvious from the drawing.

The condition shown in Fig. 1 represents the minimum VCO pressure, with piston 23 at the bottom of its range of movement against the stop member 34.

The VCO pilot assembly 21 comprises a longitudinally movable pilot spindle 37 having a pair of axially spaced lands 37a, 37b connected by a reduced diameter portion 37c and slidably disposed in a bushing 38, having an axial bore 38a. Bushing 38 is rotatably disposed in a second bushing member 39, which may be retained in place in the housing by an upper end flange 39a and a snap-ring 39b engaging the lower end thereof. Keyed to the upper end of the rotatable bushing **38** is a gear **40**; and a second gear **41** is keyed to the lower end of bushing **38** so as to engage with gear **27** on the rotatable piston member **25**. Gear **40** may be conveniently retained by snap-ring **40a**, and gear **41** may be secured by snap-ring **41a**.

The purpose of these gears is to effect continuous rotation of the pilot bushing **38** and the piston member **25** in order to reduce friction effects between piston **25** and the VCO piston **23**, and between the pilot **37** and its bushing **33**, respectively. The idea of thus rotating one of a pair of cooperating sliding parts to reduce friction effects therebetween is not a part of the present invention. Such rotational movement of bushing **38** and piston **25** is produced by a worm gear **42** carried on a shaft suitably journaled in the housing **12**, the mechanical details of which are not important to the present invention.

Operating liquid is supplied from the CCO inlet port **18** by way of a passage **18a** to an annular groove **43**, communicating by way of a plurality of circumferentially spaced ports **44** with cooperating ports **45** in bushing **38**. The annular space **37d** surrounding the pilot spindle portion **37c** communicates by way of ports **46** with an internal annular groove **47** and a plurality of spaced ports **48** in bushing **39**. Port **48** communicates by way of conduit **19b** in the housing with VCO port **19**.

Bushing **38** also defines a plurality of spaced ports **49** cooperating with ports **50** and an annular drain groove **51**. For effecting free discharge of spent operating liquid from groove **51**, the housing is provided with an axially extending slot shown at **52**. This permits free egress of operating liquid past the end flange **39a** and the generously proportioned recess in which gear **40** is disposed.

The compressor pressure compensator **22** comprises a pressure responsive bellows assembly indicated generally at **22a** and a hydraulic force-amplifying servo indicated generally at **22b**.

The bellows assembly **22a** comprises a cylindrical housing member **53** threadedly received in cover plate **13a** and secured in adjusted position by a lock-nut **54**. The axial bore through the tube member **53** contains an end seal member **55** retained by an internal snap-ring **56**, with fluid leakage between the tube **53** and seal ring **55** prevented by an O-ring seal **57**. Disposed in the tubular housing **53** is a flexible bellows **58** having its lower end sealed to the ring member **55**, high pressure compressor inlet pressure being communicated by conduit **5b** to the interior of bellows **58**. The upper end of the bellows is sealed to a fitting **59** which serves as a seat for a biasing coil spring **60**, the upper end of which abuts a washer **61** secured by a snap-ring **62**. The bellows end fitting **59** also carries a transverse pivot **63** by which the motion of the bellows is communicated to the pilot rod member **64**.

The hydraulic amplifier assembly **22b** comprises the pilot rod **64**, having axially spaced lands **64a**, **64b** separated by a reduced diameter portion **64c**. Pilot **64** is slidably disposed in the axial bore **65a** of a piston member **65** which is in turn slidably disposed in a cylinder member **66**. Cylinder **66** may be conveniently retained in a recess in the housing by an internal snap-ring shown at **67**. The upper portion of cylinder **66** defines a chamber **66a**, the upper end of which

is closed by a seal ring **68**. Ring member **68** may be retained by an internal snap-ring **69** and fluid leakage between ring **68** and the outer surface of piston member **65** is prevented by an O-ring seal **68a**. It will be seen that the piston portion **65b** of the member **65** cooperates with recess **66a** to define an upper pressure chamber **70** and a lower pressure chamber **71**.

The hydraulic passages in the amplifier **22b** are as follows. The constant pressure CCO oil from inlet port **17** is supplied to an annular groove **72** in the outer circumference of cylinder member **66**, whence it passes by way of several drilled holes **73** to an internal annular groove **74**, which communicates with several circumferentially spaced holes **75** in piston member **65**. By way of these passages, CCO oil is supplied continuously to the annular space **76** defined between the spindle portion **64c** and the bore **65a** of the piston **65**. Downward movement of pilot spindle **64** causes the lower land **64a** to uncover another set of ports **77** so that oil is admitted by way of a communicating axial passage **78** to the upper chamber **70** defined above piston **65b**. Such downward movement of pilot spindle **64** also causes the upper land **64b** to uncover a set of drain ports **79**, so that liquid trapped below the piston **65b** leaves the chamber **71** and is discharged through the top portion of bore **65a** and through one or more transverse drilled holes **80**.

Conversely, upward movement of pilot spindle **64** causes operating liquid from the annular space **76** to be supplied by way of the passages **79** to the lower chamber **71**, forcing piston **65b** upwardly, while operating liquid trapped in chamber **70** leaves by way of the axial passage **78**, being discharged downwardly past the land **64a** and out through the generous clearance space defined between spindle **64** and the adjacent wall of bore **65a**.

Those skilled in the hydraulic servo-mechanism art will appreciate that this arrangement is merely a hydraulic power amplifier for producing a displacement of piston **65** proportional to the input movement of the pilot spindle **64** effected by the flexible bellows **58**. This permits use of a comparatively small and light bellows **58**, since the bellows need exert a force only sufficient to position the pilot spindle **64**, hydraulic pressure supplying whatever power is required to effect the compensating adjustment described below.

The mechanical linkage by which the VCO piston **20** is interconnected with the VCO pilot **21** and the compressor pressure compensator **22** produces its modifying effect is as follows.

The principal member of the interconnecting linkage is a floating lever **81** having a middle portion connected by a pivot **82** to the upper end of VCO pilot spindle **37**. The right-hand end of lever **81** carries a cam follower roller **83** engaging a contoured cam **84** carried on a transverse shaft **85** journaled in the housing and arranged to be positioned by external means, the precise nature of which need not be noted here. For purpose of illustration, the cam positioning means is represented by the manual handle **85a**. It is to be noted that the handle **85a** is in the "off" position, with cam **84** rotated clockwise as far as it will go, so as to position cam follower **83** to the uppermost point in its range of movement.

In simpler, uncompensated, hydraulic servo-mechanisms of this general type, the left-hand end of lever **81** bears directly against the rounded upper end 23b of the VCO piston 23. In accordance with the present invention, however, there is interposed between piston end portion 23b and the lever end portion 81a a transversely adjustable roller 86. This adjustable abutment roller is journaled at the right-hand end of a lever 87 which is, at its left-hand end, pivoted at 88 to a shiftable block member 89 slidably disposed between upper and lower guide members 90, 91. The central portion of block 89 is provided with a transverse recess 92 across which extends a pivot member 93 engaging the forked end of a lever 94, which is in turn carried on a pivot 95 secured in the housing. As seen in Fig. 1, lever 94 pivots from the position shown in full lines to that indicated in dotted lines at 94a. The lower guide member 91 is provided with an opening 91a which is of course of sufficient size that motion of the lever 94 will not be obstructed. The lever 94 is actually a sort of bell-crank having a forked lower end portion 94b carrying a transverse pivot 96. Pivot 96 passes through a transversely elongated slot 97 formed in the extreme upper end portion of piston member 65.

It will be apparent from the above description of the mechanism that vertical movement of piston 65 will cause bell-crank lever 94 to rotate about pivot 95 so as to effect transverse sliding movement of block 89 and adjust the abutment roller 86 transversely relative to VCO piston 23. It is also to be noted that the adjustable block 89 is shown in its extreme right-hand position. The extreme left-hand position corresponds to the dotted lever position 94a, in which roller 86 is in the position indicated by dotted lines at 86a as being directly over the end portion 23b of the VCO piston. It will be understood that, when the abutment roller is in this dotted position 86a, the compensating mechanism has no effect on the lever 81, and the VCO piston 23 acts on lever 81 precisely as if the compensating roller 86 were not present. On the other hand, when roller 86 is displaced to the right of the dotted line position 86a, a special compensating effect is introduced into the linkage, the degree of this compensation being of course proportional to that called for by the compressor pressure responsive bellows 58.

In order to maintain lever 81 properly seated against the roller 86 and the cam 84, a comparatively light coil spring 98 is interposed between a socket formed in the top cover plate 14a and the upper end fitting 37f of pilot spindle 37. The downward force of spring 98 is just sufficient to maintain good contact between lever 81 and the roller 86 and cam 84 respectively, this spring force being so small as to have no effect on the positioning of VCO piston 23, etc.

In normal operation, spent operating liquid draining from the hydraulic amplifier 22b and from the VCO pilot 21 completely fills the chambers 13, 14, 15, 16 and returns to the operating liquid supply system through drain conduit 13b. Thus all moving parts operate submerged in lubricating oil.

The integrated operation of this servo-mechanism may be outlined as follows.

Assume first that the powerplant is shut down, which means, of course, that there will be no compressor discharge pressure in the bellows 58, and no hydraulic motive fluid supplied to the CCO supply conduit 9a. Therefore, the flexible bellows 58 will be collapsed by the compression spring 60, so that pilot 64 is in its lowermost position with land 64a below the port 77 and

land 64b below the drain port 79. Without CCO pressure, the VCO piston 23 will also occupy its lowermost position, as shown in Figure 1, by reason of the downward biasing force of the main-spring 32. Finally, the fuel rate selecting cam 84 will be in its maximum clockwise position, with cam follower 83 in maximum elevated position and lever 81 occupying the position shown in the drawing.

If now the powerplant is started and the fuel rate selecting cam 84 is caused to rotate counterclockwise, the cam follower roller 83 will descend, lever 81 will pivot clockwise about the abutment roller 86, with the result that pivot 82 causes VCO pilot spindle 37 to descend, admitting oil to the VCO chamber 19, and by way of passages 30, 31 to the pressure chamber 29 in VCO piston 23. Increasing pressure in chamber 29 causes piston 23 to rise against the bias of spring 32, thus causing lever 87 to pivot counterclockwise about pivot 88, so that roller 86 causes the left-hand end of lever 81 to rise. This motion restores pilot 37 to the aligned condition, in which the lower land 37b blocks off the CCO inlet port 45. It follows that, as cam 84 continues to rotate counterclockwise, the VCO pilot 37 will be positioned to establish an increasing VCO pressure in the chamber 29, piston 23 being progressively positioned upwardly as the VCO pressure increases. Thus it will be seen that there will be a preselected VCO pressure corresponding to a preselected position of piston 23 for each angular position of the input cam 84.

Meanwhile, the CCO pressure supplied to the compensator servo 22b has entered the upper chamber 70 and forced piston 65 downwardly to its lowermost position, corresponding to the completely collapsed condition of pressure responsive bellows 58. This causes the abutment roller 86 to be positioned to its extreme right-hand condition, as shown in full lines at 86, so that the maximum degree of compensating effect is introduced. Then, as the high pressure compressor inlet pressure rises into the normal operating range, increasing pressure in bellows 58 causes the hydraulic amplifier 22b to effect progressive positioning of lever 94 counterclockwise about pivot 95 so that the abutment roller 86 is positioned progressively to the left, meaning that the degree of the compensating effect introduced progressively changes.

Once the machine is operating in its normal range, it will be seen that any change in the high pressure compressor inlet pressure, due to changes in speed of the low pressure compressor 1, will be accompanied by a change in the position of the abutment roller 86 so as to appropriately modify the follow-up action of the VCO piston 23 on lever 81.

The net effect of the above-described method of operation may be seen graphically in Fig. 2. Here the abscissa represents rotational displacement of the input cam 84 from the minimum or "off" position shown in Fig. 1 to its maximum counterclockwise position corresponding to the "maximum" position of handle 85a. The ordinate represents the VCO signal pressure communicated from port 19 by way of conduit 7b to the hydraulic servo 7a of the fuel pump. As noted on the curves in Fig. 2, the lower curve represents the performance of the apparatus with the abutment roller 86 in the full line position 86 shown in Fig. 2, while the upper curve represents the performance with the abutment roller in the dotted line position 86a, directly over the VCO piston **23**. Actually, these "curves" are not curved at all but are straight lines, the contour of the input cam **84** being so chosen that straight-line performance curves will result. This makes it easier to design the fuel system and the fuel pump control servo *7a*.

Thus, it will be seen that shifting the abutment roller **86** a comparatively small distance has a very substantial effect on the operating characteristics of the combination.

The invention makes it possible to so design the regulating mechanism that a preselected fuel-air ratio is obtained for any given position of the input cam **84**, regardless of variations in the operating speed of the low pressure compressor **1**. This means that, for each position of input cam **84**, there will be a preselected rate of fuel supply which is properly proportioned to the weight flow of air to the combustion system, irrespective of any variations which may occur in the speed of the independently driven low pressure compressor **1**.

It will be obvious to those acquainted with the art that many modifications and substitutions of mechanical equivalents may be made. For instance, the hydraulic amplifier **22b** might be replaced by any suitable means for positioning the compensating lever **94** as a function of high pressure compressor inlet pressure. Also, it may be noted that the compressor pressure signal may be derived from the conduit **5** anywhere between the discharge end of the low pressure compressor and the inlet casing of the high pressure compressor. It will be obvious that the mechanical details of the VCO piston, and of the VCO pilot, and the linkage interconnecting them may also take many other forms. It is of course intended to cover by the appended claims all such modifications as fall within the true spirit and scope of the invention.

What I claim as new and desire to secure by Letters Patent of the United States is:

1. In hydraulic servo-mechanism, the combination of a housing, a first reciprocable piston member having a cylindrical chamber in one end thereof and containing a second non-reciprocable piston member defining with the slidable piston a pressure chamber, the second piston having a passageway for communicating liquid to said chamber, a main coil spring surrounding the first piston and having an end portion engaging the piston to exert a force thereon in opposition to

the pressure in said chamber, a source of operating liquid at substantially constant pressure, pilot valve means for supplying liquid from said source to said passageway and including a pilot spindle disposed for longitudinal sliding movement in a direction substantially parallel to the axis of said pistons, linkage means connecting the first piston to position said pilot spindle including a first lever member disposed generally normal to the axis of the piston and pilot and having an intermediate portion connected to position the pilot spindle, an input member engaging the end of the lever remote from the first piston, and variable abutment means interposed between the other end of the lever and the first piston, said abutment means comprising a slider member adapted to be positioned linearly in a direction generally parallel to the lever, and a second lever member pivoted at one end to said slider and having a mid-portion engaged by the end of the first piston, the other end of said second lever having a member engaging the adjacent end portion of the first lever, and means for positioning said slider towards and away from the first piston to vary the position at which said second lever end portion engages the first lever relative to the axis of the piston.

2. Hydraulic servo-mechanism in accordance with claim 1 in which the means for positioning the slider comprises a pressure signal responsive bellows connected to a pilot spindle, the pilot spindle being slidably disposed in a double-acting hydraulic servo-piston member with ports controlled by the pilot spindle and passages for supplying operating liquid to either side of the piston alternately, and lever means connecting said servo-piston with the slider whereby the latter is positioned in accordance with said pressure signal.

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