

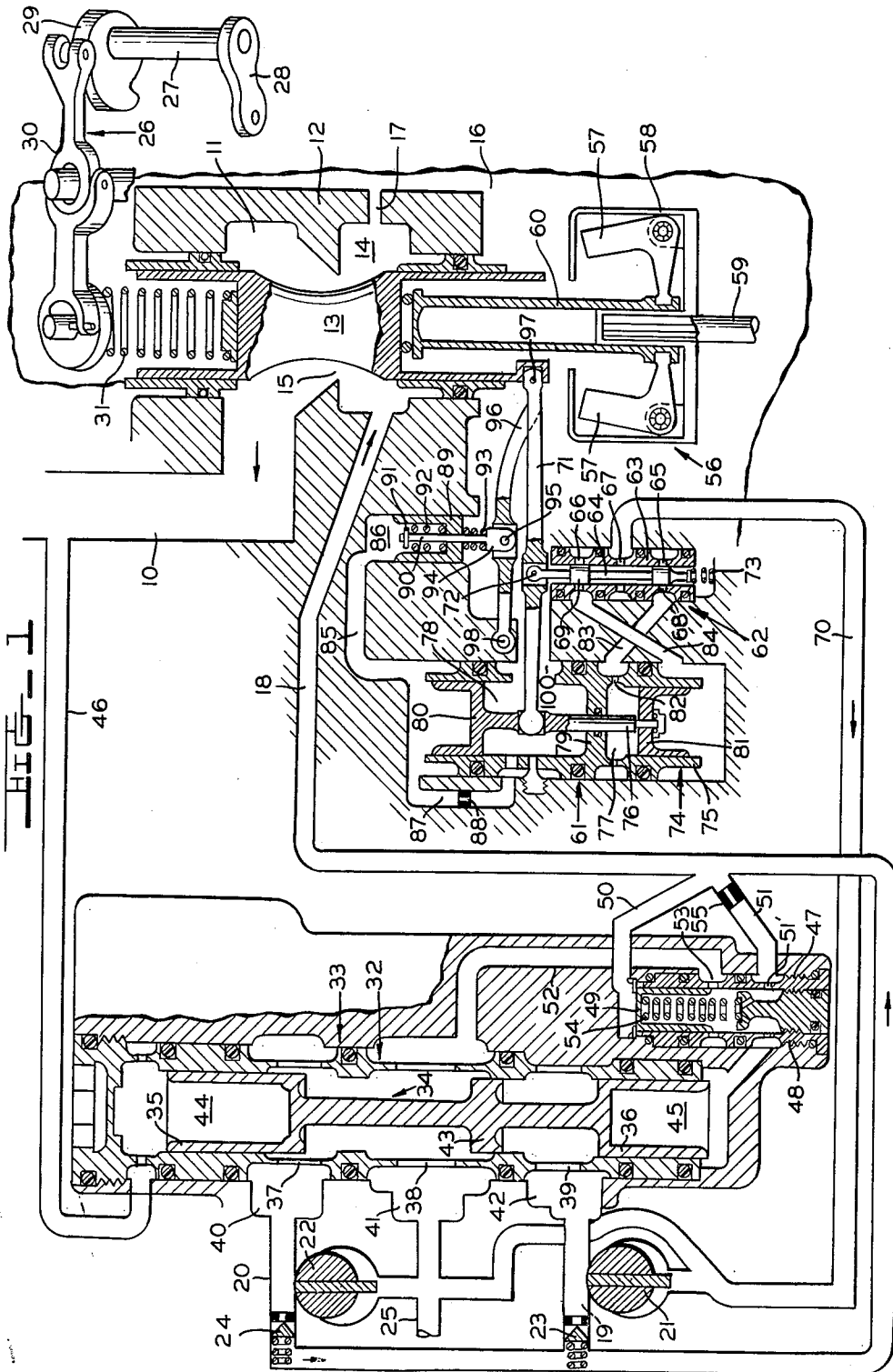
April 25, 1961

W. H. COWLES ET AL

2,981,271

GAS TURBINE ISOCHRONOUS GOVERNOR

Filed June 1, 1954



1

2,981,271

GAS TURBINE ISOCHRONOUS GOVERNOR

Warren H. Cowles, Detroit, James D. Pembleton, Oak Park, and John O'Shaughnessey, Detroit, Mich., assignors to Holley Carburetor Company, Detroit, Mich., a corporation of Michigan

Filed June 1, 1954, Ser. No. 433,676

5 Claims. (Cl. 137—26)

This invention relates generally to fuel regulating systems for fuel burning engines and refers more particularly to an improved governor controlled fuel system.

It is an object of this invention to provide a fuel controlled system having means for automatically regulating the flow of fuel to the engine in a manner such that a predetermined selected engine speed is substantially maintained regardless of changes in the normal operating conditions. In accordance with this invention, the flow of fuel to the engine is automatically regulated independently of the throttle control setting in proportion to changes in the rotative speed of the engine and also in proportion to the rate of change in the engine speed.

The foregoing as well as other objects will be made more apparent as this description proceeds especially when considered in connection with the accompanying drawing wherein the single figure illustrates a fuel regulating system embodying the features of this invention.

For the purpose of illustration, the present invention is shown herein in connection with a turbo-jet type engine having a fuel supply passage 10 connected at the outer end to the usual engine jets, not shown. The inner end of the passage 10 is connected to an annular chamber 11 formed in a body 12 and surrounding a slide-type governor valve 13 having the opposite ends mounted in the body 12 for sliding movement. Also formed in the body 12 adjacent the annular chamber 11 is a second annular chamber 14 communicating with the chamber 11 through an orifice 15 and also communicating with a chamber 16 through a port 17 in the body 12.

Connected at the inner end to the annular chamber 14 is a pressure passage 18 having the outer end respectively communicating with passages 19 and 20. The passage 19 is connected to the discharge side of a fuel pump 21 and the passage 20 is connected to the discharge side of a second fuel pump 22. Check valves 23 and 24 are respectively arranged in the passages 19 and 20 to prevent reverse flow of fuel through the latter passages. The intake sides of the two fuel pumps are connected to a reservoir or fuel supply tank by a passage 25.

With the construction thus far described, it will be noted that fuel from the discharge sides of the two pumps 21, 22 may flow past the check valves into the pressure passage 18. The fuel under pressure in the passage 18 flows into the chamber 16 through the port 17, maintaining pressure in chamber 16 at substantially that of pressure passage 18 and also flows into the supply passage 10 through the orifice 15. The effective cross sectional area of the orifice 15 varies in relation to the axial position of the valve 13 relative to the orifice and, hence, the amount of fuel entering the supply passage 10 from the pressure passage 18 depends on the position of the valve 13 with respect to the orifice 15. The valve 13 may be manually operated to control the amount of fuel introduced to the supply passage 10 by any suitable mechanism indicated generally in the drawing by the numeral 26. Briefly, the mechanism 26 comprises a rock

2

shaft 27, an arm 28 secured to the rock shaft for operating the latter, a cam 29 rotatable as a unit with the rock shaft, linkage 30 operatively associated with the cam 29 and connected through a spring 31 to the outer end of the valve 13 in a manner such that the latter may be variably biased relative to the orifice 15 by the rock shaft 27.

The pressure differential between the pressure in the passages 10 and 18 is held constant by a by-pass or unloading valve 32 comprising a cylinder 33 and a valve member 34 having enlargements 35 and 36 at opposite ends respectively slidably supported within the cylinder 33. The cylinder 33 has three ports 37, 38 and 39 spaced from one another axially of the cylinder. The port 37 communicates with an annular chamber 40 surrounding the cylinder 33 and connected to the passage 20. The port 38 is connected to the intake sides of the pumps 21 and 22 by an annular chamber 41 surrounding the cylinder 33 and communicating with the passage 25. The port 39 communicates with an annular chamber 42 surrounding the cylinder and connected to the pump outlet by the passage 19.

The flow of fuel from the pressure side of the pump 22 into the cylinder 33 and through the port 38 to the intake sides of the pumps is regulated by the enlargement 35 on the valve member 34. The flow of fuel from the discharge side of the pump 21 to the interior of the cylinder 33 and through the port 38 to the intake sides of the pumps is regulated by an enlargement 43 formed on the valve member 34 intermediate the enlargements 35 and 36.

Assuming that the valve 32 is in the position shown in the drawing, the engine fuel requirements are supplied by the pump 21, and the output of the pump 22 is by-passed. Thus, the load on the pump 22 is reduced to a minimum when the capacity of the pump 21 is ample to satisfy engine fuel requirements. On the other hand, should the fuel requirements of the engine increase, the valve member 34 is moved toward its fully closed position wherein the area of the port 37 is further reduced by the enlargement 35 so that less fuel from the pump 22 is by-passed and a greater quantity of the fuel is directed into the pressure passage 18 past the check valve 24.

The valve member 34 is moved in the cylinder 33 by changes in pressure in the spaces 44 and 45 in the cylinder 33 at opposite ends of the valve member 34. The space 44 is connected by a passage 46 to the supply passage 10 so that the pressure in the space 44 is equal to the pressure in the supply passage. The space 45 is connected to the passage 18 through a valve 47 comprising a cylinder 48 and a piston or piston-like pilot valve 49 slidably mounted in the cylinder 48. A passage 50 connects the pressure passage 18 to the cylinder 48 at the inner end of the piston 49 and a control passage 51 is connected to the cylinder 48 at the outer end of the piston 49 through a port 51' which is also connected to the space 45. A third relief passage 52 communicates with the interior of the cylinder 48 through a port 53 positioned to be uncovered by the piston 49 when the latter is in its innermost position in the cylinder 48, as shown in the drawing. The passage 52 is connected to the intake sides of the pumps 21 and 22 through the annular chamber 41. The piston 49 is normally held in its innermost position wherein the port 53 is open by a coil spring 54, and the flow of fuel through the passage 51 is restricted by a jet 55.

In the present instance, the pressure drop across the jet 55 is equal to the pressure drop across the orifice 15 and is held constant by the action of the valve 47. In detail, the pressure acting on the inner end of the valve piston 49 is equal to the pressure in the passage 18 and when the

force applied to the piston 49 by this pressure exceeds the force exerted on the outer side of the piston 49 by the spring 54 and pressure in the cylinder 48 at the outer side of the piston 49, the latter moves outwardly to close the port 53. On the other hand, when the pressure of the fluid at the inner side of the piston 49 decreases below a predetermined value, the piston 49 is moved inwardly to bleed some of the fluid to the intake sides of the pumps 21 and 22. The arrangement is such that under normal conditions of operation the pressure in the space 45 equals the pressure in the space 44 and the bypass valve 32 is balanced.

However, with the above structure it will be noted that should the pressure in the passage 18 increase for any reason, the port 53 is closed by the piston 49 and the pressure in the space 45 increases so that the valve member 34 moves toward its open position to bypass a greater quantity of fuel directly from the pressure sides of the pumps 21, 22 to the intake sides of said pumps thus reducing the quantity of fuel passing through orifice 15, thereby reducing the pressure in passage 18. This reduces the pressure in chamber 45 to the same level as that in chamber 44. On the other hand, if the pressure in the passage 18 falls below the desired value, the port 53 is opened proportionately and the valve member 34 is moved toward its closed position to reduce the quantity of fuel bypassed by the valve 32. Thus, the valve 32 maintains a constant pressure drop across the valve 13.

Tilting of the unit or displacing valve 34 by its own weight would cause valve 49 to bias at some point such that pressure in chamber 45 would increase or decrease to a value just sufficient to hold the valve stationary. This bias would be negligible.

If the small weight of the valve 34 is neglected, pressure in chamber 44 is equal to the pressure in the valve 45 and the pressure in passage 18 is applied at one end of the jet or restriction 55. The pressure at the other side of the jet or restriction 55 is equal to the pressure in chamber 45. Therefore, under steady conditions, the pressure drop across the orifice 15 is the same as the pressure drop across jet 55.

The fuel control valve 13 is also actuated in response to the speed of the engine drive shaft by a governor diagrammatically shown in the drawing by the numeral 56. The governor 56 comprises centrifugal weights 57 which are pivotally mounted on a carrier 58 for outward swinging movement relative to the carrier. The carrier 58 is secured to a shaft 59 driven by the engine and the weights 57 are connected to a tubular shaft 60 splined on the driven shaft 59 for sliding movement relative to the latter. The connections between the weights 57 and the shaft 60 are of a nature to shift the shaft 60 in opposite directions relative to the driven shaft 59 in response to variations in speed of rotation of the latter shaft. The inner end of the shaft 60 is engageable with the outer end of the valve 13 in a manner such that the valve 13 moves axially as a unit with the shaft 60. Thus, the effective area of the orifice 15 is automatically varied in response to changes in speed of the engine.

In accordance with the present invention, a speed derivative function is obtained and for accomplishing this result the arrangement indicated generally by the reference numeral 61 is provided. In detail, the numeral 62 indicates a servo valve comprising a cylinder 63 and a valve member 64 slidably mounted in the cylinder 63. The cylinder 63 is open at opposite ends to the chamber 16 and is provided with axially spaced ports 65 and 66. In addition, the cylinder is provided with a port 67 intermediate the ports 65 and 66. The valve member 64 has axially spaced enlargements 68 and 69 of the same diameter and arranged at opposite sides of the intermediate port 67 in positions to respectively control the ports 65 and 66.

The intermediate port 67 is connected by a passage 70

to the pump inlet passage 25 and permits alternately connecting the ports 65 and 66 to the low pressure passage 25. The inner end of the valve member 64 is connected to a lever 71 intermediate the ends of the latter by a pivot pin 72 and the outer end of the valve member is engaged by a coil spring 73 which urges the valve member in an inward direction.

The arrangement 61 also comprises a slave unit 74 having a cylinder 75 and a piston 76 slidably mounted within the cylinder 75. The cylinder 75 is divided into two chambers 77 and 78 by a partition 79 centrally apertured for receiving the piston 76. The piston 76 has an enlargement in the form of a head 80 at the inner end and slidably mounted in the chamber 78 and has an enlargement 81 of the same diameter as the enlargement 80 slidably mounted in the chamber 77.

The chamber 77 has a port 82 at the inner side of the enlargement 81 connected to the port 65 in the cylinder 63 of the servo valve 62 by a passage 83. The space in the chamber 77 at the outer side of the enlargement 81 is connected to the port 66 in the servo valve cylinder 63 by a passage 84. The space in the chamber 78 at the inner side of the enlargement 80 on the piston 76 is connected by a passage 85 to the inner end of a cylinder 86 and is also connected to the space in the chamber 78 at the outer side of the enlargement 80 by a passage 87 having a restriction 88 therein.

A piston 89 is slidably supported in the cylinder 86 and is centrally apertured to receive a rod 90 having a retainer 91 secured to the outer end thereof. A coil spring 92 encircles the rod 90 between the retainer 91 and the inner side of the piston 89. A second coil spring 93 encircles the rod 90 between the outer side of the piston 89 and an abutment 94 formed on the inner end of the rod 90. The abutment 94 is pivoted by a pin 95 to a lever 96 intermediate the ends of the latter and one end of the lever 96 is pivoted by a pin 97 to the slide valve 13. The other end of the lever 96 is pivoted by a pin 98 to a fixed part. One end of the lever 71 is also pivoted to the slide valve by the pin 97 and its other end is pivoted to the piston 76 of the slave unit 74.

It has been stated above that the space in the chamber 78 at the inner side of the enlargement 80 is connected to the space in the chamber 78 at the outer side of the enlargement 80 by the passage 87 having the restriction 88 therein. In this connection, it will be noted that the space in the chamber 78 at the outer side of the enlargement 80 communicates with the interior of the chamber 16 through an opening 100 which also provides clearance for the lever 71. The purpose of the restriction 88 is to provide an operating pressure differential in the spaces at opposite sides of the enlargement 80 during movement of the valve member 76.

When, for some reason, the speed of the engine driven shaft 59 is increased sufficiently to cause the weights 57 to swing outwardly, the valve 13 is shifted in an inward direction and the effective area of the orifice 15 is thereby reduced, or in other words, the amount of fuel admitted to the supply passage 10 is reduced. Inward movement of the valve 13 by the centrifugal weights 57 imparts a corresponding movement to the pin 97 and causes the lever 71 to swing in an inward direction about its pivotal connection with the valve member 76. Inasmuch as one end of the lever 96 is also pivoted on the valve 13 by the pin 97, this lever is swung inwardly about the pin 98 upon inward sliding movement of the valve 13. Such swinging movement of the lever 96 relative to the piston 89 is permitted by the springs 92 and 93. As a result of inward swinging movement of the lever 71, the servo valve member 64 is moved inwardly to connect the passage 84 with the pressure return line 70 and to connect the passage 83 with the interior of the chamber 16. Thus, the piston 76 of the unit 74 is moved outwardly and the pressure in the cylinder 86 at the inner side of the piston 89 is reduced momentarily to enable inward

movement of the piston 89 by the spring 93 and the pressure in the chamber 16 acting on the outer end of the piston 89. Thus, temporary forces are developed tending to swing the lever 96 inwardly about the pin 98 and thus to swing the lever 71 in a corresponding direction about the pivot pin 72. The arrangement is such that an amplified force is applied temporarily to the valve 13 to assist in moving the same inwardly to reduce the quantity of fuel admitted to the supply passage 10. Differential fluid pressure at opposite sides of piston 89 is only temporary and is dissipated continuously when present by bleed flow through restriction 88. The springs 92 and 93 impart an additive force to the valve spring 31 which varies as a function of the speed derivative contribution obtained by the above structure.

It is apparent from the above that when the centrifugal weights 57 swing radially inwardly as a result of a drop in speed of the engine driven shaft 59, the slide valve 13 is moved outwardly to increase the quantity of fuel admitted to the supply passage 10 by increasing the effective area of the orifice 15. It will of course be apparent that the pivot pin 97 moves outwardly as a unit with the slide valve 13 and that the rate of movement of the slide valve is increased by a reversal of the operation described in the preceding paragraph. Hence, the speed derivative contribution resulting from the foregoing provides improved engine control stability and minimizes the tendency for the control valve 13 to move excessively during rotative engine speed accelerations.

What we claim as our invention is:

1. A fuel control system for a fuel burning engine comprising a spring loaded governor valve, speed responsive means connected to said valve so as to apply a valve closing force thereto dependent on engine speed, resilient biasing means connected to said valve, servo means directly operated by movement of said valve to apply force to operate said biasing means in a sense which assists said valve movement, and means continually dissipating the force applying effect of said servo means so that the biasing effect is a function of the rate of change of engine speed.

2. A fuel control system for a fuel burning engine comprising a spring loaded governor valve, speed responsive means connected to said valve so as to apply a valve closing force thereto dependent on engine speed, hydraulic servo mechanism comprising a pilot valve operatively connected to said governor valve, a piston device movable in accordance with movement of said pilot valve, a control piston, a cylinder in which said control piston is movable, spring means connecting said control piston and governor valve, means operable by said piston device to force hydraulic fluid into said cylinder to move said control piston in a direction to apply a spring force to said governor valve tending to move it in the direction of its initial movement, and a calibrated bleed restriction to provide for escape of fluid from said cylinder.

3. In a turbine engine fuel control, means for manually selecting a desired engine speed, variably positioned valve means operative to control fuel flow to said engine, speed responsive means operatively connected to said engine for positioning said valve in accordance with said selection, a fluid chamber having a first movable wall, resilient means connecting said first movable wall to said valve, means for applying the same fluid pressure to opposite sides of said first movable wall including a re-

stricted passage leading to said chamber and unrestricted passage means for applying pressure to the outer side of said wall, said chamber having a second movable wall, means responsive to the rate of movement of said valve effective to move said second movable wall in a direction to bias the resilient means connecting said first movable wall and said valve in a direction to assist the aforesaid movement of said valve.

4. A fuel control system for a fuel burning engine comprising a spring loaded governor valve, speed responsive means connected to said valve to apply a valve closing force thereto dependent on engine speed, resilient biasing means connected to said valve, said biasing means comprising hydraulically actuated means including a member movable in opposite directions to biasing position by the flow of hydraulic fluid to and from said hydraulically actuated means, servo means directly operated by movement of said valve to control the supply of fluid to said hydraulically actuated means in a sense to cause said biasing means to assist said valve movement, and bleed means operable to continuously dissipate the fluid supply controlled by said servo means so that the biasing effect is a function of rate of change of speed.

5. A fuel control system for a fuel burning engine comprising a spring loaded governor valve, speed responsive means connected to said valve to apply a valve closing force thereto dependent on engine speed, resilient biasing means connected to said valve, said biasing means comprising hydraulically actuated means including a member movable in opposite directions to biasing position by the flow of hydraulic fluid to and from said hydraulically actuated means, said biasing means including opposed springs connecting said member to said valve to bias said valve in a direction corresponding to the direction of movement of said member, servo means directly operated by movement of said valve to control the supply of fluid to said hydraulically actuated means in a sense to cause said biasing means to assist said valve movement, and bleed means operable to continuously dissipate the fluid supply controlled by said servo means so that the biasing effect is a function of rate of change of speed.

References Cited in the file of this patent

UNITED STATES PATENTS

1,777,458	Allen	Oct. 7, 1930
2,168,155	Caughey	Aug. 1, 1939
2,308,165	Fields	Jan. 12, 1943
2,409,975	Curtis	Oct. 22, 1946
2,515,074	Bobier	July 11, 1950
2,533,231	Drake et al.	Dec. 12, 1950
2,545,856	Orr	Mar. 20, 1951
2,549,897	Eurell	Apr. 24, 1951
2,595,618	Vogt et al.	May 6, 1952
2,643,513	Lee	June 30, 1953
2,657,529	Lawrence	Nov. 3, 1953
2,689,606	Mock	Sept. 21, 1954
2,691,382	Frick	Oct. 12, 1954
2,756,810	Simmons	July 31, 1956
2,761,495	Greenland	Sept. 4, 1956
2,765,800	Drake	Oct. 9, 1956
2,780,172	Coar	Feb. 5, 1957
2,855,029	Eastman	Oct. 7, 1958
2,874,764	Booth et al.	Feb. 24, 1959