

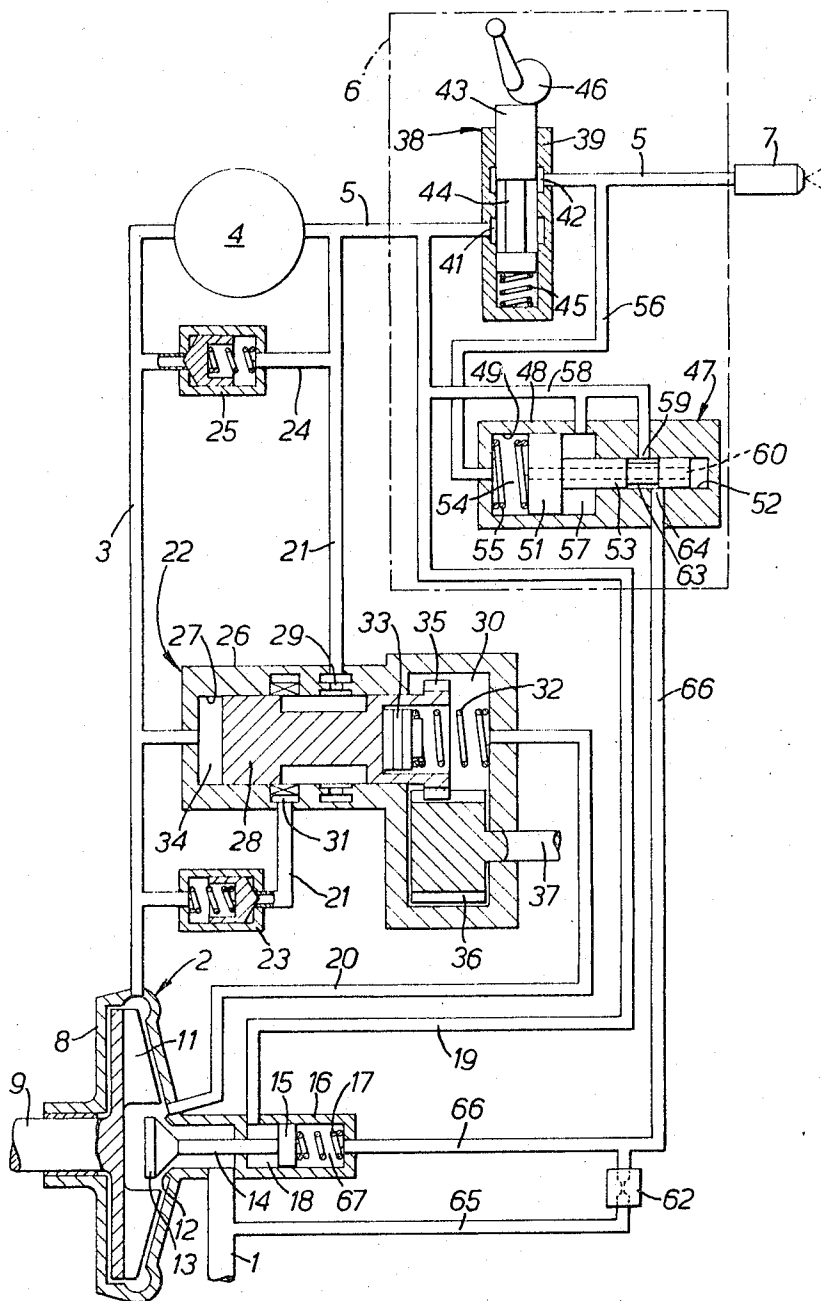
March 21, 1967

S. R. TYLER

3,309,995

LIQUID PUMPING APPARATUS

Filed Feb. 25, 1965



INVENTOR
STANLEY R. TYLER

BY

Arland M. Christensen
ATTORNEY

1

3,309,995

LIQUID PUMPING APPARATUS

Stanley R. Tyler, Cheltenham, England, assignor to Dowty Fuel Systems Limited, Cheltenham, England, a British company

Filed Feb. 25, 1965, Ser. No. 435,198

Claims priority, application Great Britain, Mar. 9, 1964, 9,868/64

6 Claims. (Cl. 103—11)

This invention relates to liquid pumping apparatus and more particularly to pumping apparatus for the supply of liquid fuel to a gas turbine engine.

In accordance with the present invention a pumping apparatus comprises a centrifugal pump having an inlet arranged to receive liquid from a source, a flow control device arranged to regulate the flow of liquid from the source through the inlet of the centrifugal pump, the centrifugal pump being such that under the regulation of flow at its inlet the liquid within the pump may form an annulus whose radial depth controls the pump outlet pressure, a positive displacement pump assembly connected to the outlet of the centrifugal pump and having an outlet for connection with a load at pressure, regulating means operable to vary the effective delivery of the positive displacement pump assembly when the maximum delivery of which this assembly is capable at a given pump speed exceeds the regulated flow through the centrifugal pump, said regulating means being responsive to the pressure rise across the liquid annulus in the centrifugal pump and being operative to maintain said pressure rise at a substantially constant value by varying the effective delivery of the positive displacement assembly.

An embodiment of the invention suitable for the supply of liquid fuel to the burners of an aircraft gas turbine engine will be described with reference to the accompanying diagrammatic drawing.

A pipe 1 connected to a fuel source forms the inlet to an engine driven centrifugal pump 2, while a delivery pipe 3 from the latter forms the inlet to a positive displacement pump 4 of fixed volumetric capacity which may be driven by the engine at a fixed speed ratio with respect to the centrifugal pump. An outlet pipe 5 from the pump 4 is connected through a flow-controlling device, shown within the broken line rectangle 6, to a spray nozzle 7 which represents a number of such nozzles supplying fuel to the combustion chambers of a gas turbine engine. The spray nozzles constitute a load which maintains pressure in the outlet pipe 5.

The centrifugal pump 2 has a casing 8 in which an engine-driven shaft 9 and rotor 11 are mounted, while the supply of fuel to the eye of the rotor 11 is regulated by an inlet valve which comprises a valve seat 12 and a movable valve member 13. The valve member 13 is connected by a stem 14 to a piston 15 which is movable in a cylinder 16 under the opposing influences of a spring 17, acting to open the inlet valve, and of fluid pressures acting differentially on the piston 15 as will be described. The centrifugal pump 2 is of a kind as described in United States specification Ser. No. 3,128,822 wherein the inlet valve controls the rate of liquid flow such that the liquid within the pump forms an annulus whose radial depth controls the pump delivery pressure.

The pumping system includes valve devices to meet the case when the delivery from the centrifugal pump 2 in both less than and more than the delivery from the positive displacement pump 4. In the first case, the excess delivery from the pump 4 is returned to the pipe 3 through a by-pass passage 21 having regulating means formed by a variable throttled by-pass valve 22, and a non-return valve 23 in series therewith. In the second case, the excess delivery from the pump 2 is fed to the outlet pipe

2

5 through a by-pass passage 24 across the pump 4, having a non-return valve 25 therein.

The variable throttle valve 22 comprises a casing 26 having a valve bore 27 in which a waisted piston valve member 28 is axially slidable. The valve bore 27 has an inlet port 29 in open communication with the waisted portion of the valve member 28, and an outlet port 31 which is variably throttled by the valve member 28 at one end of the waisted portion. The valve member 28 is urged in the valve-opening direction by a spring 32 which acts through an axial thrust bearing 33, and in the valve-closing direction by fluid pressure from the delivery pipe 3 which is admitted to a chamber 34 at the end of the valve member 28 which is remote from the spring 32. The spring-loaded end of the valve member 28 extends into a chamber 30 which is connected by a pipe 20 to the interior of the centrifugal pump casing at the eye of the rotor 11. To overcome static friction, the valve member 28 is formed with a spur gear 35 meshing with a spur gear 36 which is carried by a continuously rotated shaft 37. The latter spur gear 36 is elongated to maintain driving connection throughout axial movement of the valve member 28.

A simple form of flow-controlling device is shown within the rectangle 6 to facilitate understanding of the invention. The device includes a flow control valve 38 having a cylinder 39 with an inlet port 41 receiving liquid from the pump 4 and an outlet port 42 delivering liquid to the nozzle 7. A movable valve member 43 has a waisted portion 44 providing at all times an unrestricted flow through the inlet port 41 while the valve member at one end of the waisted portion 44 variably restricts the outlet port 42. The valve member 43 is urged towards the valve-open position by a spring 45 and is actuated in the valve-closing direction by a manually operable cam 46.

The flow control valve 38 acts through a servo valve 47 to control the piston 15 which regulates the inlet valve 12, 13 of the centrifugal pump 2. The servo valve comprises a cylinder 48 having a large bore 49 slidably receiving a piston 51, and a small bore 52 slidably receiving a stem 53 which extends from one end of the piston 51. The large area face of the piston 51 and the cylinder 48 together define a chamber 54 which houses a compression spring 55 and which is connected by a pipe 56 to the outlet pipe 5 on the downstream side of the control valve 38. The annular face of the piston 51 and the cylinder 48 together define a chamber 57 which is connected by a pipe 58 to the outlet pipe 5 on the upstream side of the control valve 38. The end of the stem 53 is connected by a central bore 60 to the chamber 54 whereby the effective pressure loaded area of the piston 51 in the chamber 54 is equal to the annular area in the chamber 57.

The small bore 52 has a port 59 connected to the pipe 58, the port 59 being continuously in free communication with a waisted portion 63 of the stem 54. A second port 64 spaced axially from the port 59 is variably restricted by the stem 53 at one end of the waisted portion 63. The port 64 is connected by a pipe 66 to a chamber 67 in the cylinder 16 on the spring-loaded side of the piston 15. The pipe 66 is connected through a fixed restrictor to a pipe 65 which leads to the pump inlet pipe 1. A chamber 18 in the cylinder 16 on the stem side of the piston 15 is maintained at pump delivery pressure in the pipe 5 by a connecting pipe 19.

The operation of the flow-controlling device will now be described. Liquid flowing through the throttled outlet port 42 produces a pressure drop acting through the pipes 56 and 58 and the chambers 54 and 57 on opposite faces of the piston 51. The resulting force on the piston 51 is balanced by the load of the spring 55. A small movement of the piston 51 will move the stem 53 from a position completely closing the port

3

64 to one fully opening this port, over which movement the spring 55 is arranged to exert a substantially constant force. When there is a constant pressure drop across the outlet port 42, the piston 51 will remain in a position at which the stem 53 partially closes the port 64. The fluid pressure in the chamber 67 is determined by the relative throttling of the port 64 and the fixed restrictor 62, so that an increase in the port opening 64 will cause the piston 15 to move in the opening direction of the valve member 13, while a decrease in the port opening will cause the piston 15 to move in the valve closing direction. If a reduced flow rate through the pipe 5 is required, the cam 46 is turned to lower the valve member 43 whereby the outlet port 42 is increasingly throttled, causing an increased pressure drop which acts through the pipe 58, 56 to move the piston 51 against spring load. The stem 53 increases the throttling of the port 64 to decrease the liquid flow there-through. The same decreased flow through the fixed restrictor 62 causes a pressure reduction in the chamber 67 whereby the piston 15 moves to reduce the opening of the inlet valve 12, 13 of the pump 2. The rate of flow into the pump 2 is accordingly reduced to the level at which the pressure drop across the port 42 is restored to a substantially constant value. Equilibrium of the piston 51 and of the piston 15 in a new position is established.

By similar reasoning it will follow that opposite movement of the cam 46 to raise the valve member 43 will bring about controlled increase in the rate of flow into the pump 2.

It is to be understood that the flow-controlling device 6 may be modified and elaborated to be additionally responsive to further controlling parameters such as maximum speed and air to fuel ratio, by the inclusion of known mechanisms.

Having shown that the flow of fluid through the pumping apparatus can be substantially controlled by regulating the inlet to the centrifugal pump, the operation of the pumping apparatus will now be described under condition of different rates of flow.

At low flow rates, for example when starting up the engines, the centrifugal pump 2 may be incapable of generating the necessary pressure for satisfactory fuel atomization by the nozzle 7, and the positive displacement pump 4 accordingly generates the required pressure. While the centrifugal pump 2 is regulated as described to supply the required rates of flow to the nozzles, the rate may be less than the displacement of the pump 4. Cavitation within the pump 4 is avoided by permitting recirculation of part of the flow through the by-pass passage 21 and the variable throttle valve 22 to the pipe 3. When this recirculation takes place, the movable member 28 of the throttle valve 22 will automatically move to a position causing a degree of closure of the port 31 which passes a recirculated flow equal to the difference between the flow through the pump 4 and the controlled flow through the centrifugal pump 2. This can be explained by considering the effect of a position error of the movable member 28. The position of the movable member is determined by the pressure difference between the outer annulus and the eye of the centrifugal pump 2, as communicated through the pipes 3 and 20 to opposite ends of the member 28, and by the spring 32. This pressure difference is determined, for a given speed of the rotor 11, by the radial depth of the liquid annulus in the pump 2.

If a position error of the movable member increases the opening of the ports 31, the recirculated flow will increase since the flow resistance load provided by the burners 7 remains the same. There will be a corresponding reduction in the liquid flow from the pump 2, causing the radial depth of the liquid annulus to increase. This increases the pressure difference in the pipes 3 and 20, causing the valve member 28 to move against the load of

4

the spring 32 and reduce the opening of the port 31. By similar reasoning it can be shown that a position error of the movable member which decreases the opening of the port 31, will cause a reduction of pressure across the pipes 3 and 20 to allow the valve member to move in the valve-closing direction under spring load.

Thus, the movable member 28 will automatically settle in an equilibrium position of partial closure of the port 31, at all times when the flow through the pump 4 exceeds the controlled flow through the pump 2. The pump 4 and the by-pass valve 22 together form in effect a positive displacement pump assembly whose effective delivery at a given pump speed is automatically variable.

When liquid flow demand rises in excess of the delivery of the pump 4, for example when the engine is running at or near maximum speed at low altitude, the centrifugal pump 2 supplies the required flow at high pressure through the pipe 3. The non-return valve 23 and the by-pass valve 22 close under this high pressure. The excess of liquid flow over that displaced by the pump 4 is by-passed through the non-return valve 25 and the passage 24 to the outlet pipe 5.

The positive displacement pump 4 is preferably designed to deliver fuel somewhat in excess of the demand at the burner ignition condition of the engine, at which condition the centrifugal pump does not provide sufficient pressure to give the required fuel delivery from the nozzles 7.

At high engine speeds, during take-off and climb of the aircraft, the centrifugal pump 2 generates the required pressure and it is capable of operating satisfactorily at high fuel temperatures and high ambient temperatures.

By designing the positive displacement pump 4 to cater for the starting condition, and the centrifugal pump 2 to cater for the high flow required at maximum engine power, it is possible to achieve an economy in weight of the pumping apparatus.

Whilst the described embodiment uses a positive displacement pump 4 of fixed volumetric capacity together with a by-pass to form a variable flow pumping assembly, it is within the scope of the present invention to employ a variable delivery pumping assembly comprising a positive displacement pump of variable volumetric capacity in which the capacity is varied by a device responsive to the pressure difference across the centrifugal pump 2. When high flow rates are required greater than the maximum displacement of the variable capacity pump, the excess delivery from the centrifugal pump will by-pass the variable capacity pump through the non-return valve 25 as described.

Although the positive displacement pump 4 has been described as being engine driven, it may alternatively be driven by a source of auxiliary power such as hydraulic or electric power. Moreover, means may be provided to discontinue the drive to the pump 4 when the engine is running at such a speed that the centrifugal pump develops sufficient pressure. Under these conditions all the fuel flow to the burner nozzles 7 may pass through the non-return valve 25.

I claim as my invention:

1. Pumping apparatus comprising a centrifugal pump having an inlet arranged for connection with a source of liquid, a valve in the inlet, flow control means operable upon said inlet valve to control the flow of liquid into the centrifugal pump such that the liquid therein forms, when the pump is driven, an annulus whose radial depth controls the pump outlet pressure, positive displacement pump means connected to the outlet of the centrifugal pump and having an outlet for connection with a load at pressure, said pump means including flow-regulating means operable to vary the effective delivery of said pump means when the maximum delivery of which said pump means is capable at a given pump speed exceeds the controlled liquid flow through the centrifugal pump and a movable control member of said flow-regulating means

5

responsive to the fluid pressure across the liquid annulus in the centrifugal pump.

2. Pumping apparatus according to claim 1, wherein the positive displacement pump means comprises a pump of fixed volumetric capacity and a by-pass valve forming said flow-regulating means, and passage means connecting the by-pass valve between the outlet and the inlet of said fixed volumetric capacity pump.

3. Pumping apparatus according to claim 2, including a non-return valve arranged in said passage means to open under liquid flow from the outlet to the inlet of the fixed volumetric capacity pump.

4. Pumping apparatus according to claim 1, including a non-return valve connected by a by-pass passage between the outlet of the centrifugal pump and the outlet of the positive displacement pump means, said non-return valve being arranged to pass excess flow from the centrifugal pump when the controlled flow from the latter exceeds the maximum delivery of the positive displacement pump means at the speed at which the latter is driven.

5. Pumping apparatus according to claim 1, wherein the flow control means is connected between the positive displacement pump means and the load at pressure, said flow control means including valve means connected to a fluid pressure actuator which is operatively connected to the valve at the inlet of the centrifugal pump.

6. Pumping apparatus comprising a centrifugal pump having an inlet arranged for connection with a source of liquid, a valve including a fluid pressure actuator therefor disposed in the inlet and operable to control the flow of liquid into the centrifugal pump whereby the liquid forms, when the pump is driven, an annulus whose radial depth controls the outlet pressure of the pump with reference to the pressure downstream of the inlet valve, flow control

6

means including a control member means responsive to the delivery of the centrifugal pump and valve means operatively controlled by said control member and said delivery responsive means, said valve means controlling the fluid pressure actuator whereby the inlet valve of the centrifugal pump regulates the delivery therefrom to a required value, a positive displacement pump connected to the outlet of the centrifugal pump and having an outlet for connection with a load at pressure, a first by-pass connected between the outlet of the positive displacement pump and the outlet of the centrifugal pump, a throttle valve in said first by-pass including a movable member which is responsive to the pressure developed by the centrifugal pump between the outlet thereof and the inlet on the downstream side of the inlet valve, said movable member being operable by said pressure against the force of loading means to restrict variably said throttle valve in the sense that an increase in pressure acting on said member causes an increase in the restriction of said throttle valve, and causes closure of said throttle valve when the controlled delivery of the centrifugal pump exceeds the delivery of the positive displacement pump, a second by-pass connected between the outlet of the centrifugal pump and the outlet of the positive displacement pump, and a non-return valve in said second by-pass arranged to pass excess flow from the centrifugal pump when the controlled delivery therefrom exceeds the delivery of the positive displacement pump

References Cited by the Examiner

UNITED STATES PATENTS

3,043,367 7/1962 Abraham ----- 158—36.4

LAURENCE V. EFNER, *Primary Examiner.*