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Mueller

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[54] **FUEL PUMP CONTROL SYSTEM AND METHOD**

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[51] **Int. Cl.** **F04b 49/00**

[58] **Field of Search** .... 417/282, 286, 288; 137/115; 60/39.28 R

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[57] **ABSTRACT**

An internal control system for parallel pump fluid pumping systems. The control system acts to open a low-pressure pathway receiving the discharge from at least one of the pumps in response to internally sensed outlet requirements. The open pathway results in re-circulation of low-pressure pumped fluid through a portion of the system while the outlet is supplied from another of the parallel pumps. The control system operates by sensing pressure from various points within the fluid system.

15 Claims, 3 Drawing Figures

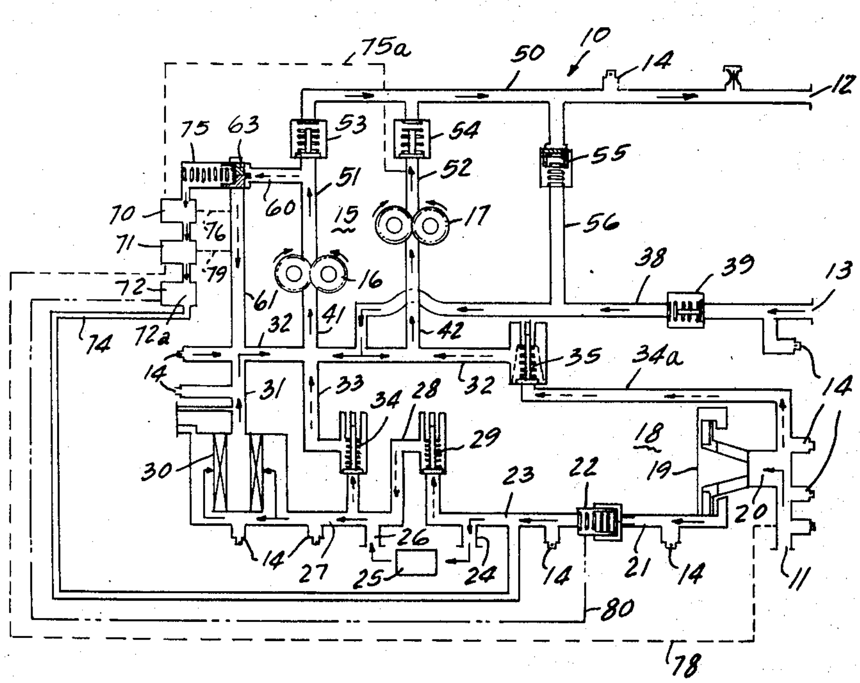

Fig. 1

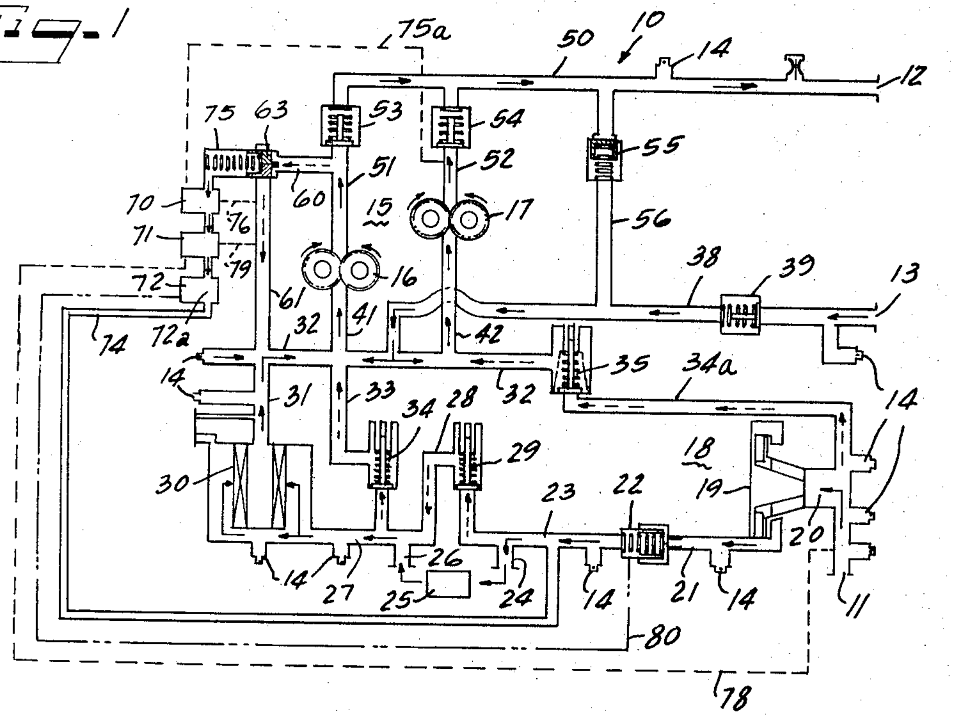
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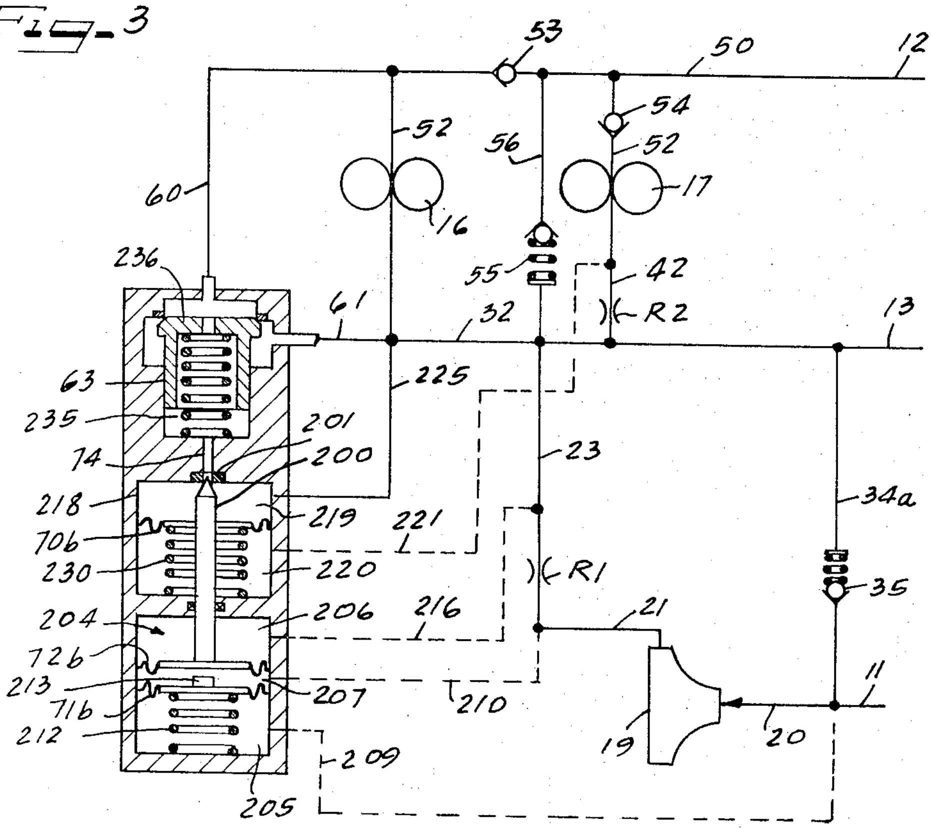
Fig. 3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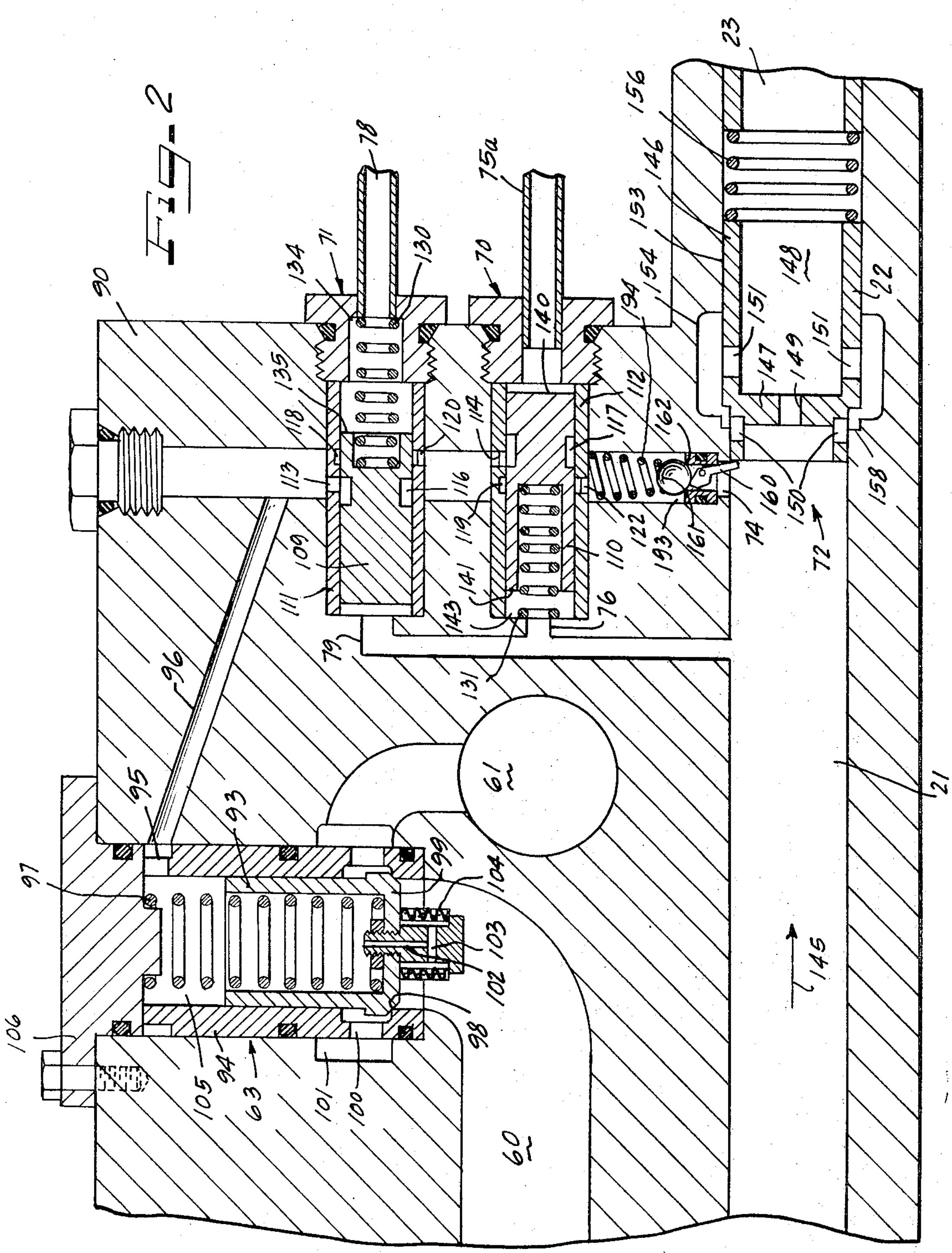
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Fig. 2

FUEL PUMP CONTROL SYSTEM AND METHOD

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to fluid control systems and more particularly to a control system for a multi-pump fluid flow system.

2. Prior Art

Fluid pumping systems, and particularly those systems used to pump fuel in modern aircraft, often require a pumping system capable of delivering the pumped fluid over a wide range of flows and operating pressures. In order to accommodate this, particularly in aircraft systems, the pumping systems normally comprise two or more pumps arranged in parallel, which function together as a main pumping system. The systems generally also include a booster pump located at or near the fuel tank to provide a pressured flow of fuel to an interstage which feeds to the parallel ranked pumps.

Because of the necessity of delivering the output from the main pumping system at a wide range of flows, the system at times will be required to operate near maximum capacity, while at other times, such as when engine demand reduces the need for large quantities of fuel, the pumping system will have considerable overcapacity.

In standard systems, the booster pump is engine driven, and is normally a centrifugal pump whose rotational speed is directly proportional to the speed of the engine. However, the main pumping system pumps are normally positive displacement gear or vane pumps which operate at a constant speed. Because of this, the displacement available in the main pumping system is generally constant. Thus, when engine demand for fuel slackens, the system is doing unnecessary work. This work is a function, not only of the displacement of the pumps, but also of the pressure at which the pump outlet delivers the pumped fluid.

It has, heretofore, been suggested to accommodate the differing flow demands by providing high pressure relief valves downstream of the ranked pumps, which relief valves recognize the existence of abnormal high pressures downstream of the pumps when engine demand for fuel is lessened, and open a bypass shunting a portion of the pumped fuel to an area upstream of the pumps. While such systems accommodate, to a limited extent, the necessary fuel flow variations within the system, they do so at the expense of adding excess work in that the outlet from the main pumping system must, of necessity, still be at high pressure.

The extra work performed on the fluid being pumped has the disadvantage of raising the temperature of the pumped fluid. Since it is common to use the pumped fluid or fuel as a coolant for the engine oil, by shunting it through engine oil coolers, any unnecessary rise in the temperature of the fuel reduces the effectiveness of the system.

Other attempted solutions have relied upon external controls to shut down one or more of the main stage pumps. Such systems are undesirable, first, because they require external controls to the pumping system, which controls normally require electric source connections in the highly dangerous fuel handling system; secondly, because the use of systems capable of shutting down one or more of the pumps allows failure of the system by failure of the pumps to re-start when demand increases. This can be extremely hazardous.

Much of the problem can be eliminated, if a low-pressure escape path is provided for the pumped fuel from one or more of the pumps. By allowing a low-pressure recirculation, the amount of work performed is lessened. However, heretofore, any attempt at a low-pressure escape, such as by shunting at a low pressure the output from one or more of the pumps to an area upstream of the pumps, has required external controlling to open a valve. Such external controlling is undesirable and, further, generally requires the use of hazardous electrical connections and sensing equipment.

SUMMARY

My invention overcomes the disadvantages of the prior art and provides a low-pressure escape path for fluid pumped by one or more of a ranked system of parallel pumps. The low pressure escape path is controlled by a valve which in turn is controlled by a sensing system operating entirely interior of the pumping system. The sensing system of my invention does not require any electrical connections or external input.

For descriptive purposes, the system will be described in connection with a two-pump main stage system, however, the control system is adaptable for use in any multi-parallel pump system. The main pumps are fed from a common interstage and discharge into conduits to a common output line. The conduit from one of the pumps is connected with a bypass conduit for receiving the output from the one pump and channeling it back to the interstage. A pressure balanced valve is interposed in the bypass conduit controlling flow therethrough. The pressure balanced valve is cyclable to an open position by means of a controlled discharge from one section of the pressure balanced valve. The discharge path is open or closed in response to a number of sensing valves positioned in series in the discharge line. The valves sense pressures internally within the fluid flow system and are open or closed in relation to those pressures.

Thus, when conditions within the system are right, that is when engine demand for fuel is lessened to a point when the non-bypassed or noncontrolled pump alone can supply sufficient fuel, the sensing valves will open, allowing the bypass conduit valve to become unbalanced, thereby opening the bypass. Because the bypass valve opens fully, and maintains its open state against a low pressure, the bypass is a low-pressure bypass, thereby reducing the amount of work performed by the one pump.

In the embodiments illustrated, the conditions sensed by the sensing valves include engine speed as a function of booster pump input-output, pressure differential, internal fluid flow to the main stage pumps, and output from the non-bypassed pump.

In a preferred embodiment, axially moving spool pilot valves are used for some of the sensing valves while an axially moving sleeve valve is mechanically coupled with a check valve for another of the sensing valves. In another preferred embodiment, diaphragms attached to a valve closure member are used, the diaphragms sensing the pressure changes and moving the valve closure member in relation thereto.

It is therefore an object of this invention to provide a pumped fluid control system.

It is a further object of this invention to provide a pumped fluid control system for parallel fuel pumping systems.

It is another object of this invention to provide a pumped fluid control system for parallel pump fuel systems wherein the output from one or more of the parallel pumps is bypassed at low pressure to an interstage feeding the pumps.

It is another object of this invention to provide a control system for a pumped fluid system allowing a low-pressure bypass to open downstream of a pump in response to a sensing of internal conditions.

It is yet another object of this invention to provide a control system for pumped fluid, the control system being independent of any external stimuli and responsive to internal conditions sensed within the pumping system.

It is another object of this invention to provide a pumped fluid control system for parallel pump pumping systems, the control system opening a bypass downstream of one or more pumps, the control system actuated by sensing internal conditions within the fluid flow system.

It is a more specific object of this invention to provide a control system for an aircraft fuel pumping system, the pumping system including two or more parallel main pumps, the control system opening a bypass from at least one of the pumps, the bypass being controlled by a valve which in turn is actuated by a series of sensing devices, the sensing devices responsive only to internal conditions within the pumping system.

It is a specific object of this invention to provide a control system for aircraft fuel pumping systems which opens a bypass in response to conditions sensed within the fuel flow system, the control system sensing internal pressures from at least three different portions of the fuel flow system.

It is a general object of this invention to provide a method of bypassing the output of one or more pumps in an aircraft fuel supply system at low pressure in response to lessened engine demand.

BRIEF DESCRIPTION OF THE DRAWINGS

Other objects, features and advantages of the invention will be readily apparent from the following description of certain preferred embodiments thereof, taken in conjunction with the accompanying drawings, although variations and modifications may be effected without departing from the spirit and scope of the novel concepts of the disclosure, and in which:

FIG. 1 is a schematic drawing illustrating a fuel flow system for an aircraft utilizing the control system of this invention.

FIG. 2 is a cross-sectional view of a pump housing casing equipped with the control system of this invention.

FIG. 3 on page 1 of the drawings is a schematic view illustrating a modified form of the control system of this invention with the control system shown in section in a housing.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. 1 schematically illustrates a system flow diagram for a two-pump main pumping system aircraft fuel supply system. The diagram illustrates the bypass and control system of this invention. Although only two main pumps are illustrated in parallel, it is to be understood that this system can be used with a multi-pump ranked parallel pumping system.

In the diagram illustrated in FIG. 1, the pumping system **10** includes an inlet **11** such as may connect with a fuel storage tank and an outlet **12** such as may discharge to an aircraft engine or the like, and/or to hydraulic actuators, oil coolers, and other utilizers of pressurized fuel. A secondary inlet **13** may return fuel from these sources. Numerous pressure and temperature taps, ports and valves **14** are also illustrated in the system.

The pumping system includes a main pumping stage **15** comprising parallel pumps **16** and **17**, and a preliminary or boost pump stage **18** including a booster pump **19**. The booster pump **19** is in communication with the inlet **11** through a conduit opening **20**. The booster pump discharges to a conduit **21** which is communicated through a sleeve orifice valve **22** to a conduit **23** which discharges in turn to a conduit **24** communicating with an external device **25**, such as a heater or the like. The heater **25** is illustrated as communicating with a conduit **26** which in turn communicates with a prefilter conduit **27**. A bypass **28** controlled by a pressure check valve **29** communicates the conduit **23** with the conduit **27**, bypassing the external device **25**, should that device become clogged or closed off. A filter unit **30** is positioned between the conduit **27** and a conduit **31** which in turn communicates to the interstage **32**. A bypass conduit **33** communicates the conduit **27** to the interstage **32** and is controlled by a pressure check valve **34** to bypass the filter **30** in case the filter **30** becomes clogged.

In case the boost pump **19** is inoperative or any of the conduits **21** through **31** becomes clogged, a bypass **34*a*** communicates the inlet **11** through a pressure check valve **35** to the interstage **32**. A conduit **38** may also communicate the interstage **32** to the auxiliary inlet **13** through a pressure check valve **39**. The valves **29, 34, 35** and **39** are all normally one-way valves and are common to prior art duel stage pump flow systems.

It can be seen from the above that in the normal course, all of the fluid supplied to the system through the inlet is normally fed to the boost pump **19** and thence to the interstage **32**. The interstage **32** is in common communication with conduits **41** and **42** which in turn communicate with the intakes of pumps **16** and **17**. Pumps **16** and **17**, in aircraft embodiments, are normally positive displacement gear or vane pumps which operate at constant speed. The booster pump **19** in the boost stage **18** is normally an engine-driven centrifugal pump. Therefore, the output from the boost stage is a function of engine speed. While the two main stage pumps **16** and **17** are illustrated as being substantially equal in size, it is to be understood that one of them, for example, **16**, could be a larger pump in which case it would normally be considered the main pump while the other and smaller pump such as pump **17** would be an auxiliary pump. The main pumping stage **15** is sized to provide sufficient flow for maximum demands. Thus, when the engine is throttled down, the pumping stage will supply to conduit **50**, which is in communication with the outlet **12**, a greater-than-necessary flow of fuel. The outlets from the pumps **16** and **17** are respectively in communication with conduits **51** and **52** which communicate through one-way check valves **53** and **54** to the outlet conduit **50**. The check valves **53** and **54** are normally spring-biased and require a positive pressure to open to communicate the conduits **51** and **52** with the conduits **50**. At the same time, they serve to prevent backflow in those situations where the pressure in the conduit **50** is greater than in either of the conduits **51** and **52**.

In order to provide relief within the conduit **50**, when engine demand is slackened, a high-pressure relief valve **55** is provided communicating the conduit **50** with a pre-pump conduit such as the conduit **38** leading to the interstage **32**. This communication is illustrated as being by means of conduit **56**. The high-pressure relief valve **55** is also a one-way valve providing flow only from the conduit **50** to the conduit **38**.

It will be appreciated that what has been described prior to this is a normal fuel flow diagram, with the exception of valve **22**, commonly found in aircraft fuel supply systems. My invention, hereinafter described, is illustrated as being used with this diagram; however, it is adaptable for use with most multi-pump, parallel ranked pumping systems.

In the system above described, when engine demand slacks, excess pumping capacity in the main pumping stage **15** results either in a greater flow through the system and return via the auxiliary inlet **13** or, in most cases, in a greater pressure in the outlet conduit **50**. The greater pressure in the conduit **50** opens the valve **55**, providing a return flow at high pressure. Thus, the pressure in the conduit **50** is maintained relatively high in order to maintain the valve **55** open. In such circumstances, the parallel pumps **16** and **17** are doing considerable excess work. This will elevate the temperature of the fuel beyond the desired point.

In order to eliminate this problem, I have provided a bypass consisting of conduits **60** and **61** which bypass the output from the pump **16** back to the interstage **32**. Conduit **60** is in communication with conduit **51** at the outlet end of pump **16** and with conduit **61** which in turn communicates to interstage conduit **32**. Interposed between conduits **60** and **61** is valve **63** which controls the bypass. The valve **63** normally closes the connection between the conduits **60** and **61**, so that the output from the pump **16** passes through the valve **53** to the conduit **50**, and thence through the outlet **12**.

The valve **63** of this invention is a pressure-balanced valve which is capable of remaining closed against high pressure, while at the same time capable of remaining open against a low pressure. Because the valve stays open when faced with a low pressure, the bypass conduits **60, 61** will have a pressure approximating that of the interstage **32**. Thus, the bypass of the output from the pump **16** will be at a low pressure, causing the valve **53** to close, assuring that the entirety of the output from the pump **16** is bypassed. In this way, the amount of useless work done by the pump **16** is minimized and the temperature rise of the fuel is kept minimal.

The valve **63** is controlled by sensing devices **70, 71** and **72** which function together to open a drain conduit **74** communicating with one side **75** of the pressure-balanced valve **63**. When the drainage conduit **74** is communicated with the side **75** of the valve enclosure, the pressure balance of the valve **63** will be eliminated and the valve will open. When the sensing devices **70, 71, 72** are closed, the portion **75** of the valve enclosure for the valve **63** will be closed, and the valve **63** will close.

The sensing devices **70, 71** and **72** sense only conditions within the pumping system itself intermediate the inlet **11** and the outlet **12**. No external stimuli act upon the sensing devices.

In a preferred embodiment, each of the sensing devices is a pressure controlled valve having a portion interposed between the portion **75** of the valve enclosure and the drainage conduit **74**. The pressures acting on the valve portion cause a passageway to open through each of the sensing devices. When all three sensing devices are open, the portion **75** of the valve enclosure will be in communication with the drainage conduit **74**.

As illustrated in FIG. **1**, the sensing devices may sense engine speed, pumping system internal fluid flow and auxiliary pump discharge pressure.

For example, the sensing device **70** senses the auxiliary pump **17** discharge pressure via a pressure sensing passage **75*a*** extending from the conduit **52** downstream of the pump **17** to the sensing device **70**. This pressure is compared with the pressure in the conduit **61** via a pressure sensing conduit **76**. With the valve **63** closed, the pressure in the conduit **61** will be the same as in the interstage **32**. As long as the discharge pressure from the auxiliary stage pump **17** is maintained sufficiently high above the interstage pressure, typically 300 or 400 psi, the sensing device **70** will remain open to allow communication between the portion **75** of the valve enclosure and the next sensing device **71**.

The sensing device **71** senses engine speed by comparing the pressures between the inlet **11** via the pressure sensing passage **78** and the outlet or discharge from the boost pump as by the pressure sensing passage **79**. The sensing passage **78** senses the pressure from adjacent the inlet upstream of the booster pump **19**. The passage **79** senses the pressure at a point downstream of the booster pump but upstream of the main pumping stage. This is indicated in FIG. **1** as being a sensing passage between the sensing device **71** and the conduit **61**, which is normally at a pressure equivalent to the interstage **32**. Because the boost pump **19** is an engine-driven pump, the pressure output therefrom is proportional to the engine speed and a comparison of the pressure between the inlet to the pump and the outlet therefrom indicates the engine speed. When the engine speed is in the correct range, the sensing device **71** will open to provide a passageway from the sensing device **70** to the sensing device **72**.

The sensing device **72**, illustrated in FIG. **1**, comprises a drainage path portion **72*a*** and the sleeve orifice valve **22**, with a mechanical connection indicated by the broken line **80** therebetween. The sleeve orifice valve **22** senses internal flow between the boost pump stage and the main pump stage. When the flow rate is low enough to indicate a lessened demand, the sensing device **72** will open a passage between the sensing device **71** and the drainage conduit **74**. Thus, when all three sensing devices are open, the pressure balanced valve **63** will be unbalanced and open to communicate the conduit **60** to the conduit **61**, resulting in a bypass of the output from the pump **16** at low pressure.

FIGS. **2** and **3** illustrate two embodiments of the valve **63** and sensing controls **70, 71** and **72**. In the first embodiment, illustrated in FIG. **2**, the sensing devices **70** and **71** comprise spool valves or pilot valves and the sensing device **72** comprises a ball check valve opened by the mechanical connection with the sleeve orifice valve **22**.

In the embodiment illustrated in FIG. **3**, the sensing devices **70**, **71** and **72** comprise movable diaphragms operating a needle stopper which in turn opens and closes a drainage path for the pressure-balanced valve **63**. It is to be understood that these embodiments are only illustrative of an internal bypass control and that a greater or lesser number of sensing devices could be used which can sense other desired parameters.

Considering first the embodiment illustrated in FIG. **2**, the figure illustrates a cross-sectional view of a portion of the pump housing which includes the bypass valve and sensing controls therefor.

The pump housing **90** includes portions of the conduit **21** from the boost pump outlet and the conduit **23** with the sleeve orifice valve **22** interposed therebetween.

The housing **90** also includes a portion of the conduit **60** in communication with the bypassed or controlled pump discharge and the bypass conduit **61**, with the pressure-balanced valve **63** controlling communication between the conduits **60** and **61**. The sensing controls **70** and **71** are illustrated as being axially movable spool valves while the control unit **72** consists of the sleeve orifice valve **22** and a ball check valve **193** positioned to be acted upon by the sleeve orifice valve **22**.

As illustrated, the valve **63** has a cup-shaped valve body member **93** received in a tubular housing member **94**, which has a passageway **95** at one end which passageway communicates with a drainage passage **96**. The cup-shaped body member **93** is positioned interiorly of the tube-shaped member **94** and is biased by a spring **97** against a valve seat **98** formed on the inner diameter of the tubular member **94**. The bottom wall **99** of the cup-shaped member **93** closes against the valve seat **98** at the juncture with the conduit **60** to block the conduit **60**. Passageways **100** in the tubular body portion **94** open to a conduit **101** in communication with the bypass conduit **61**, downstream of the valve seat **98**. Therefore, in order for the conduit **60** to communicate to the bypass conduit **61**, the valve body **93** must be moved away from the valve seat **98** against the pressure of the spring **97**.

The end wall **99** of the cup-shaped body member **93** has an orifice or passageway **102** communicating with passageways **103** which open to the conduit **60** through a filter **104** which prevents clogging of the narrow passageways **103** and orifice **102**. Because the orifice opens to the conduit, the interior **105** of the tubular body member **94** which is closed at its outer end by a closure plate **106** against which the spring **97** backs and at its other end by the end wall **99** of the valve body **93**, will have an interior pressure substantially equal to the pressure in the conduit **60** due to the bleed-through from the orifice **102**. So long as the drainage opening **96** is blocked by the sensing controls **70**, **71** and **72**, the pressure in the area **105** will be equal to the pressure in the conduit **60**. Therefore, the spring **97** will maintain the valve body **93** against the valve seat **98** to prevent communication between the conduit **60** and the bypass conduit **61**. Communication between the area **105** and the bypass conduit **61** is prevented due to the sealing seat of the valve body **93** against the inner diameter of the tubular portion **94**. However, whenever the drainage path **96** is open to a lower pressure area, in this embodiment to the conduit **23** through the drainage conduit **96** which communicates as at **74** to the conduit **23** downstream of the three sensing devices **70**, **71** and **72**, the pressure within the area **105** will approximate the pressure within the conduit **23** which is upstream of the pump **16**. Therefore, the pressure in the conduit **60** will be sufficiently greater than the pressure in the area **105** to overcome the spring bias and open the valve **63**, allowing communication between the conduit **60** and the bypass conduit **61**. As long as the three sensing controls are open, this condition will continue inasmuch as the drainage openings **95**, **96** and **74** are of a greater size than the orifice **102**.

As illustrated, the sensing devices **70** and **71** are spool valves which are interposed in the drainage line **96**. The valves each have a cylindrical spool **109**, **110** received within a tubular housing **111**, **112**, the tubular housing having openings **113**, **114** therethrough communicating the drainage passage **96** upstream of the spool **109**, **110** to the interior of the tube. The spool valves have circumferential grooves **116**, **117** therein and the tubular housings have inner diameter grooves **118**, **119** communicating to discharge openings **120**, **122** downstream of the spool, the discharge openings **120**, **122** opening to the portion of the drainage conduit **96** downstream of the individual sensing control. When the spool **109**, **110** is moved axially so that the inlet openings **113**, **114** are communicated to the grooves **118**, **119** through the grooves **116**, **117**, the drainage conduit **96** will be open through the sensing controls. Both spool valves are moved by pressure acting against the axial ends of the spools **109**, **110**, which pressure is offset by springs **130**, **131** acting against one of the axial ends of the spools. In the sensing control **71** which senses engine speed as a function of the difference between the boost pump intake pressure and outlet pressure, the axial ends of the tube **111** are communicated, at one end, with the conduit **79** which in turn is communicated with the conduit **21**, thereby sensing pressure downstream of the booster pump, the other end of the spool being communicated via conduit **78** with the inlet, thereby sensing inlet pressure. In this instance, the spring **130** is interposed between a housing portion **134** and the axial end **135** of the spool open to the conduit **76**. Therefore, the pressure downstream of the boost pump sensed by the conduit **79** must be sufficiently greater than the pressure prior to the boost pump sensed by the conduit **78** to overcome the bias of the spring **130** and move the spool axially to a point where the groove **116** communicates with both the sensing device inlet **113** and the sensing device outlet **118–120**. The spool **109** is capable of moving within the tube **111** past that point to a point where the groove **116**, while remaining in communication with the groove **118**, is no longer in communication with the inlet **113**. At this point, the conduit **96** will again be blocked by the sensing control **71**. Thus, the sensing control **71** is open only for an intermediate stage of pressures. When the pressure of the boost pump discharge is low, indicating low engine speed, the spool valve **71** will be closed due to the action of the spring **130** and the inlet pressure **78** acting against the end **135** of the spool to push it to a point where the groove **116** is not in communication with the groove **118**. As the boost pump pressure increases, the spring urging will be overcome and the groove **116** will enter into communication with the groove **118** while remaining in communication with the inlet opening **113**. At this point, the discharge conduit **96** will be open through the control **71**. As the engine speed continues to increase and the boost pump discharge pressure continues to increase, the spool **109** will be forced beyond the point of communication with the opening **113**, thereby again closing the conduit **96** through the sensing control **71**. By dimensioning of the grooves and spools and their spacing from one another, and by control of the tension of the spring, it is possible to set the control **71** to function only in a desired range of pressures, indicating a desired engine speed. For example, it is possible to set the spool valve so that it will remain closed at all speeds below 3,000 to 3,500 r.p.m. and at all speeds above 4,500 to 5,000 r.p.m. In this way, the control **71** will prevent bypass valve bleed flow except in engine speed ranges of approximately 3,500 to 4,500 r.p.m. Thus, only in those speed ranges can the bypass valve **63** open. Alternatively, of course, the valve can be designed so as to remain open below a given speed and to remain closed above that speed, as sensed as a function of boost pump discharge pressure.

The control **70** senses the output of the non-controlled pump **17**. In order to do this, one end of the spool **110** is open via conduit **76** to a prepumped passage such as the passage or conduit **21**, thereby sensing pressure prior to the non-controlled pump **17** and control pump **16**. The other end **140** of the spool **110** is open to conduit **75*a*** which communicates with a post auxiliary pump conduit or non-controlled pump conduit such as the conduit **52** of FIG. **1**. In this control, the spring **131** is interposed between the end **141** of the spool and the housing portion communicating to the conduit **76**. Therefore, the pressure in the conduit **75*a*** must be sufficiently greater than the pressure in the conduit **76** to overcome the bias of the spring **131** to move the spool to a point where the control inlet **114** is communicated with the control outlet **122** by an overlapping of the grooves **117, 119**. The spool end **141** is dimensioned with respect to the housing end **143** adjacent the conduit opening **76**, so that when the pressure in the conduit **75*a*** is sufficiently great to overcome the spring **131** and pressure in the conduit **76**, the valve will remain open by bottoming of the end **141** against the housing portion **143** to prevent further opening movement of the spool. Thus, the sensing control **70**, unlike the illustrated sensing control **71**, has only one open position and one closed position. The spool is prevented from moving axially a sufficient distance to close off communication between the inlet **114** and the groove **117**. Thus, for all non-controlled pump discharges sufficiently large, the control **70** will be open. However, whenever the discharge pressures from the non-controlled pump falls below a given level of pressure rise over the pressure at the non-control pump inlet, the sensing control **70** will close. This could occur due to failure of the non-control pump **17** or for other reasons such as greater engine usage of fuel creating a high flow, low-pressure discharge from the non-controlled pump.

The sensing control **72** senses fuel flow within the system, in the illustrated embodiment prior to the main pumping section. The flow rate is sensed by a sleeve orifice valve **22**. In the illustrated embodiment shown in FIG. **2**, the flow through conduit **21** is in the direction of arrow **145** against the sleeve orifice valve **22**. The sleeve orifice valve is a substantially H-shaped cross section valve body **146** having a cross member **147** blocking off the tubular interior **148** and preventing flow therethrough except through an orifice **149** in the cross member **147**. Upstream of the cross member **147**, radial openings **150** are provided and downstream of the cross member **147**, radial openings **151** are provided. The outer surface of the valve body **146** is in sealing engagement with the wall **153** of the conduit in which it is received, except for a groove **154** in the surface of the conduit. The groove **154** is axially long enough to communicate the radial openings **150** to the radial openings **151** when the valve body is in a given position. The valve body **146** is axially movable in the conduit to close off communication with the groove **154** and the upstream openings **150**. When this occurs, the only flow through the conduit **21-23** occurs through the orifice **149**. A spring **156** acts against the valve to urge it in a direction to cut off communication between the openings **150** and the groove **154**. The spring force is overcome by the pressure of the fluid in the conduit **21** acting against the cross member **147**. Whenever the flow of fluid through the conduit **21** is sufficiently great, the valve **22** will remain open against the spring **156**. As the flow of fluid decreases, the valve will close and all flow will continue through the orifice **149**. The spring **156** is sized to overcome the pressure drop caused by the cross member **147** and orifice **149** at low flow conditions to urge the valve body **22** against the seat **158**, closing off communication between the openings **150** and the groove **154**. As flow increases, the pressure drop across the cross member **147** through the orifice **149** will increase, and the spring pressure will be overcome, causing the valve **22** to open so that the openings **150** are in communication with the groove **154**. By sizing the orifice **149** and the spring rate of the spring **156**, the valve can be designed to remain closed at all flows below a given point, for example, **5** gallons per minute, and to remain open at all flows above that point. Preferably, the orifice **149** will be sized to accept all flows below that point with a very small pressure drop across the cross member **147**. Thus, the spring **156** can have a low spring rate, requiring small pressures, for example on the order of **4** to **5** psi, to open. Thus, the valve does not prevent full flow conditions, and further, interferes very little with flow during low flow conditions.

The ball check valve **193** is positioned with respect to the valve **22** so that the lever arm **160** of the ball check is contacted by the valve body **146** whenever the valve **22** is in a closed position. In such situations, the lever **160** will lift the ball **161** off of the valve seat **162** against the pressure in passage **96** and spring **194**, thereby communicating the drainage passage **96** with the passage **74**. When the valve **22** is in its full flow open position, the mechanical contact between the lever **160** and the valve body **146** cannot occur and the ball will remain seated against the valve closure seat **162**, blocking off communication of the drainage passage **96**. Thus, the sensing control **72**, consisting of the sleeve orifice valve **22** and the ball check **93**, which the passage **96** only during low flow conditions, such as would be encountered when the engine demand for fuel lessens.

As can be seen from the above description, in the embodiment illustrated in FIG. **2**, the bypass control valve **63** can open only when the sensing control **70** senses that the engine speed is within a desired range; when the sensing control **71** senses that the auxiliary or non-controlled pump discharge is sufficiently great to supply the needs of the engine; and, when the sensing control 72 senses that the engine fuel demand is lessened, resulting in a low fuel flow rate through the conduit 21.

FIG. 3 schematically illustrates another embodiment of this invention wherein the bypass valve 63 is controlled by a drainage conduit 74 blocked by a valve 200 against a valve seat 201. The valve 200 is controlled by three pressure-sensing diaphragms 70*b*, 71*b* and 72*b*.

The diaphragms 71*b* and 72*b* are received in a housing chamber 204 and divide the chamber into three areas, an area 205 below the diaphragm 71*b*, an area 206 above the diaphragm 72*b* and an area 207 between the two diaphragms. The area 205 below the diaphragm 71*b* is ported via a conduit 209 to the inlet 11 prior to the boost pump 19 so that the pressure in the area 205 is equal to the inlet pressure. The area 207 is ported via a conduit 210 to the conduit 21 downstream of the boost pump 19 so that the area 207 has a pressure equal to the pressure of the boost pump output. Therefore, the diaphragm 71*b* senses engine speed as a function of boost pump pressure rise. The spring 212 is disposed in the area 205 to ct on the diaphragm 71*b* which has an abutment nib 213, and urges the diaphragm 71*b* in an upward direction until the abutment nib 213 contacts the underside of diaphragm 72*b*. When the pressure rise across the boost pump 19 is sufficiently great, the pressure in the area 207 will be sufficiently larger than the pressure in the area 205 to counteract the force of the spring 212 and remove the influence of the abutment nib 213 from the diaphragm 72*b* by forcing the diaphragm 71*b* downwardly.

The diaphragm 72*b* has the valve 200 attached thereto for movement therewith so that as the diaphragm 72*b* moves upwardly and downwardly in the chamber 204, the diaphragm 72*b* moves the valve 200 between a seating and unseating relationship with the valve seat 201 to open and close the drain passage 74. The diaphragm 72*b* is operated as a function of engine fuel demand. Engine fuel demand is determined as a function of total new fuel input to the pumping system by sensing the pressure drop across a restriction R1 in the conduit 23 downstream of the boost pump and downstream of the connection between the conduits 21 and 210. The pressure downstream of the restriction R1 is ported via conduit line 216 to area 206. As engine fuel demands lessen, the pressure differential across the restriction R1 will change. As the pressure sensed by the line 216 increases, the resistance to downward movement of the valve 200 decreases.

Diaphragm 70*b* is received in housing chamber 218 and is attached to the valve 200, and movable therewith. The diaphragm 70*b* divides the chamber 218 into areas 219 and 220. The area 220 senses pressure via a conduit 221 which is connected with conduit 42 downstream of a restriction R2. Conduit 42 is in turn connected to the intake to the non-controlled pump 17. Area 219 senses pressure in the interstage 32 via a conduit 225, which also forms part of the drain path 74 when the valve 200 is open. The diaphragm 70*b* then senses the flow generated by the positive displacement non-controlled pump 17, as a function of the pressure drop across the restriction R2 which pressure drop is the difference between the pressure at the interstage 32 sensed by a conduit 225 and the inlet pressure to the pump 17 which pressure is sensed by the conduit 221. A spring 230 acts against diaphragm 70*b* to force it and the valve 200 upwardly.

Therefore, it can be seen that the seating of the valve 200 in the valve seat 201 is determined by the ratio of engine flow demand as measured by the differential pressure drop across the restriction R1, and non-controlled pump output as measured by the differential pressure drop across the restriction R2 acting on the diaphragms 72*b* and 70*b* respectively. In any case, where the pressure drop across R1 is larger than the pressure drop across R2, diaphragm 72*b* forces will overcome diaphragm 70*b* forces, and in conjunction with the spring 230, act to seat the valve 200 in the valve seat 201. This will always be the case when the engine fuel flow demand is sufficiently large relative to the auxiliary pump 17 flow capacity. However, when the engine flow demand drops and a portion of the discharge flow from both the pumps 16 and 17 is bypassed through the relief valve 55 and the conduit 56 at high pressure back to the interstage 32, the pressures acting to retain the valve 200 seated against the valve seat 201 will lessen. As engine flow demand drops still further, the pressure drop ratio through R1, which measures total engine flow, or actual pump throughput, and R2, which measures the flow generated by the positive displacement auxiliary pump 17, will change until the pressure drop across R2 exceeds the drop across R1 by a predetermined amount. At this point, with a greater pressure in area 219 than in area 220 and with a small pressure differential between areas 206 and 207, the pressure differential caused by the restriction R2 will act downwardly on the diaphragm 70*b* and will overcome the pressure differential across R1 acting upwardly on diaphragm 72*b* and the force of the spring 230 also acting upwardly and cause the valve 200 to lift off the seat 201. When this occurs, a drainage path is opened from the area 235 behind the end wall 236 of the bypass control valve 63. This area will then be allowed to drain through the drainage path 74 and the conduit 225. Therefore, the relatively large pressure in the conduit 60 will cause the valve 63 to open, communicating the conduit 60 to the bypass conduit 61. At this point, the pressure downstream of the check valve 53 will lessen to a point where the check valve 53 will close, thereby resulting in a recirculation of the output from the main pump 16 at a low pressure through the bypass conduit 60, 61.

It can be seen from the above description that the embodiment illustrated in FIG. 3 therefore senses engine speed as a function of the pressure differential between areas 205 and 207, senses total fuel input as a function of the pressure differential between areas 207, 206, and senses auxiliary or noncontrolled pump pressure as a function of the pressure difference between areas 219, 220. Thus, it is only when the pressures in the areas 205, 206, 207, 219 and 220 are within preset ranges that the bypass valve can open.

It is to be appreciated that this system provides safety advantages, substantially the same as the safety advantages provided in the embodiment illustrated in FIG. 2. That is to say that if, for example, the boost pump 19 would fail, the control system would lose the pressure differential between the areas 207 and 205 so that the spring 212 would maintain the valve 200 closed. Additionally, should the auxiliary or non-controlled pump 17 fail, the pressure drop across R2 would cease, and the spring 230 would maintain the valve closed. Additionally, other safety features are inherent in the designs illustrated.

It will be noted that in the embodiment illustrated in FIG. 3, the non-controlled pump discharge pressure is not sensed as in the system described in FIG. 2. This is because the direct sensing of the auxiliary or non-controlled pump 17 flow generating capacity as a function of the pressure drop across R2 effectively senses the working of that pump.

It should be appreciated that although both of the preferred embodiments discussed herein sense three separate conditions, i.e., engine speed, non-controlled pump working, and engine fuel demand or flow rate, that if desired, a lesser number of functions can be sensed or, additional functions could be sensed, resulting in the addition of other valves or diaphragms controlling the bypass valve.

Although the teachings of my invention have herein been discussed with reference to specific theories and embodiments and although illustrative means for accomplishing explained results have been described, it is to be understood that these are by way of illustration only and that others may wish to utilize my invention in different designs or applications.

I claim as my invention:

1. A pump fluid control system comprising: a fluid pumping system including an inlet for receiving fluid, a booster pump in communication with said inlet and having an outlet in communication with an interstage, a main pump and at least one other pump connected in parallel between said interstage system, a control means for said pumping system for selectively diverting the main pump discharge to a bypass pathway and thereby remove the main pump discharge from the flow to the common outlet, said control means including a valve means for controlling flow in the bypass pathway, said valve means including means for balancing the pressure acting on opposite side of a valve member thereof, said control means including means for sensing only fluid pressure from at least three different areas of said pump system and means for unbalancing the pressure as one side of the valve member in response to the pressures sensed by said sensing means so that unbalancing the pressures as the valve member of the valve means causes the valve means to open and provide a low pressure bypass for the main pump.

2. A pump fluid control system according to claim **1**, wherein said means for unbalancing including a drainage conduit ported to said one side of the valve member and second valve means for controlling flow in said conduit, said second valve means being actuated between a closed and open position by the application of pressure differentials from the pressures sensed by said sensing means.

3. A pump fluid control system according to claim **2**, wherein said second valve means includes at least one spool valve which is axially shifted between an opened and closed position in response to pressure acting on an axial end thereof.

4. A pump fluid control system for bypassing the output of at least one controlled pump in a series of at least two parallel pumps, said pumps normally discharging to a common outlet which comprises: a bypass pathway, said bypass pathway open to the output of the controlled pump, said bypass pathway being effective to provide a low pressure bypass for the output of the controlled pump other than through the normal outlet, said bypass pathway controlled by a valve effective to open and close the pathway, said valve having means for balancing the pressure acting on opposite sides of a valve member, a valve control means having means for unbalancing the pressures acting on the valve member and having means for sensing pressures from at least three areas of a fluid flow system which supplies fluid to the pumps and received the discharge from the pumps, said means for unbalancing being actuated in response to the pressures sensed at the three areas.

5. A pump fluid control system comprising: a fluid flow system having an inlet and an outlet, at least two parallel pumps interposed between said inlet and said outlet working on fluid passing through said fluid flow system, a control system for bypassing the output of at least one of said pumps whereby the output of that pump is not discharged to the outlet when said system is open and whereby the output from that pump is discharged to the outlet when said system is closed, said control system including a bypass pathway, a valve having an enclosure disposed in said pathway and having a valve member coacting with a valve seat to control flow through said pathway, said valve having means for balancing the pressure acting on opposite sides of the valve member, means for bleeding pressure acting on one side of said valve member, means for sensing pressures from at least three areas of said fluid flow system and valve means for controlling said bleed means in response to the pressures sensed by said sensing means.

6. In an aircraft fuel supply system having an inlet from a fuel source and an outlet to a fuel utilizer, the system having a variable speed booster pump open to the inlet and at least two constant speed pumps acting in parallel on the fuel within the system fed by the booster pump and discharging to the outlet, the improvement of means to bypass the output of at least one of said constant speed pumps, said means including a bypass pathway receiving at least a portion of the output of said one of said constant speed pumps, said pathway discharging to an area upstream of said constant speed pumps, valve means for controlling flow through said pathway, said valve means having means for balancing the fluid pressure acting on opposite sides of a valve member so that the valve means normally closes said pathway, control system for said valve means including means for unbalancing one side of said valve member so that the valve means opens said pathway said unbalancing means being actuated in response to pressure differentials between fuel pressures sensed from at least three separate areas intermediate the inlet and outlet of said supply system.

7. A supply system of claim **6** wherein the pressure differentials indicate the speed of said booster pump and the operating condition of at least one of the non-bypassed constant speed pumps.

8. A supply system of claim **7** wherein the pressure differentials indicate the fuel utilization rate of the system as a function of fuel flow through the fuel supply system.

9. A system of claim **7** wherein booster pump speed is indicated as a function of pressure differential between the inlet to the booster pump and the outlet therefrom.

10. A system of claim **8** wherein fuel flow is determined as a function of pressure drop across a restriction intermediate the booster pump and the constant speed pumps.

11. A system of claim **7** wherein the operation of the non-controlled pump is determined as a function of pressure differential across the non-controlled pump.

12. A system of claim **7** wherein the operation of the non-controlled pump is determined as a function of pressure drop across a restriction in a conduit supplying fuel only to the non-controlled pump.

13. A system according to claim **6**, wherein said balancing means comprises an orifice extending through the valve member, which is received in an enclosed chamber.

14. A system according to claim **13**, wherein said unbalancing means includes a drainage path connected to said enclosure and a plurality of pressure responsive members controlling flow in said drainage path.

15. A system of claim **14** wherein at least one of the said pressure responsive members is an axially movable spool valve interposed in said drainage path for the said enclosed chamber, the spool valve axially movable from open to closed positions in response to pressures acting on the axial ends of the spool valve.

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