United States Patent [19]

Masuda

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[54] **FLUID CONTROLLING APPARATUS**

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Feb. 29, 1972 Japan 47-21325

[52] **U.S. Cl.** **137/596.13,** 91/433, 91/436, 91/446, 137/596.1

[51] **Int. Cl.** **F15b 13/02**

[58] **Field of Search** 137/116.3, 596.1, 596.12, 137/596.13; 91/433, 436, 446

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Primary Examiner—Alan Cohan
Assistant Examiner—Gerald A. Michalsky
Attorney, Agent, or Firm—Cushman, Darby & Cushman

[57] **ABSTRACT**

An apparatus for controlling the flow of fluid, comprising a directional control valve unit for controlling the operating direction and operating speed of a fluid motor, and a pressure-compensating valve unit for controlling the flow of fluid passing in said directional control valve such that it will be proportional to the displacement of a valve spool of said directional control valve unit.

7 Claims, 7 Drawing Figures

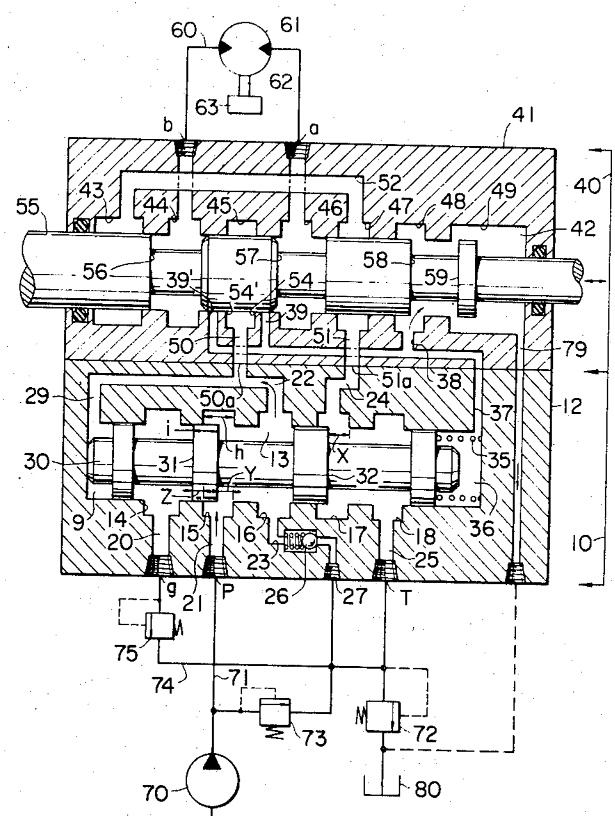

FIG. 1

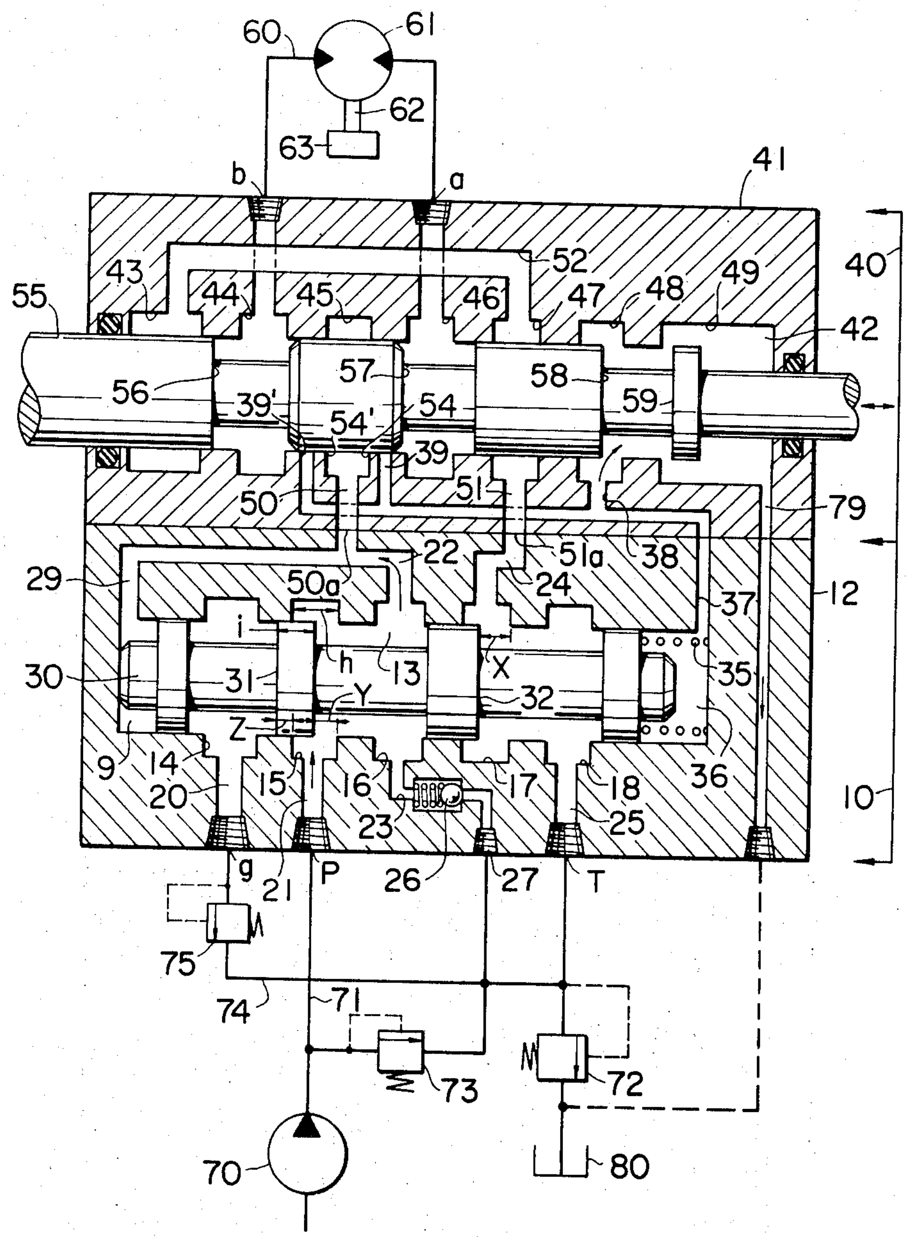
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FIG. 2

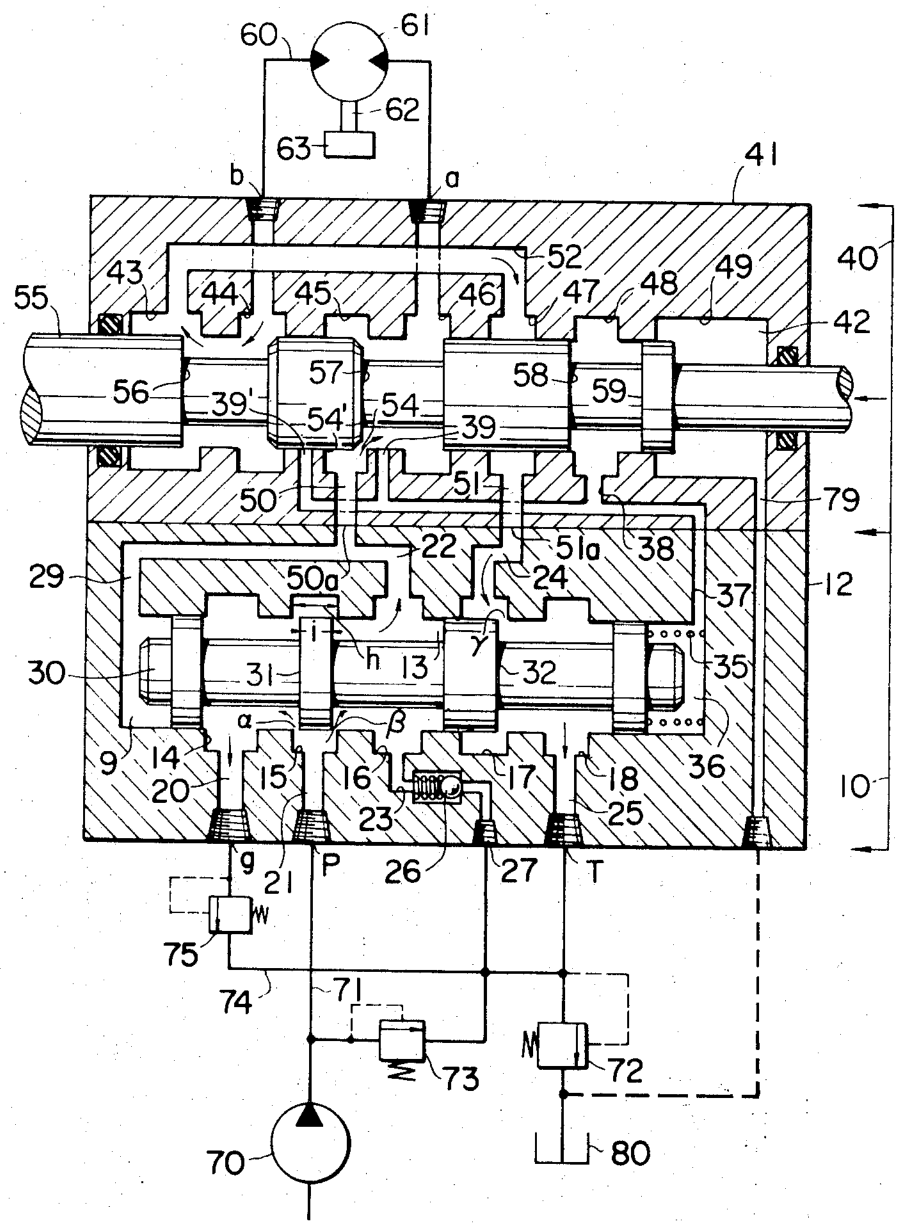
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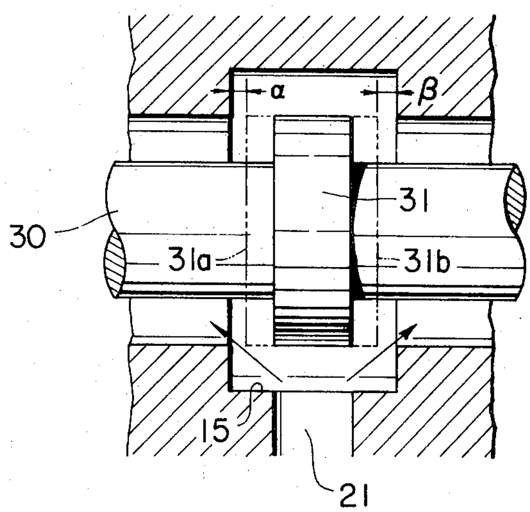
FIG. 3



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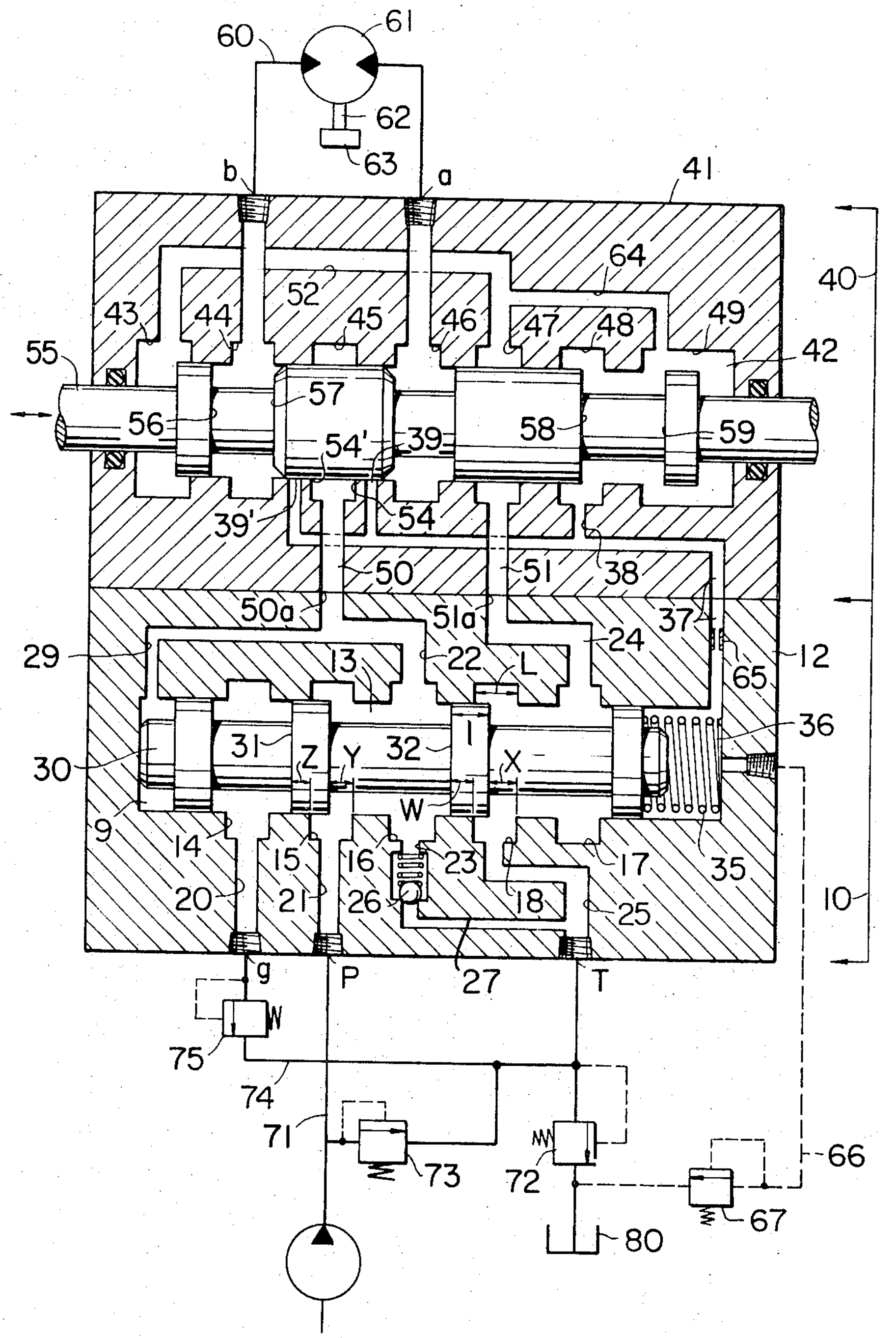
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FIG. 5

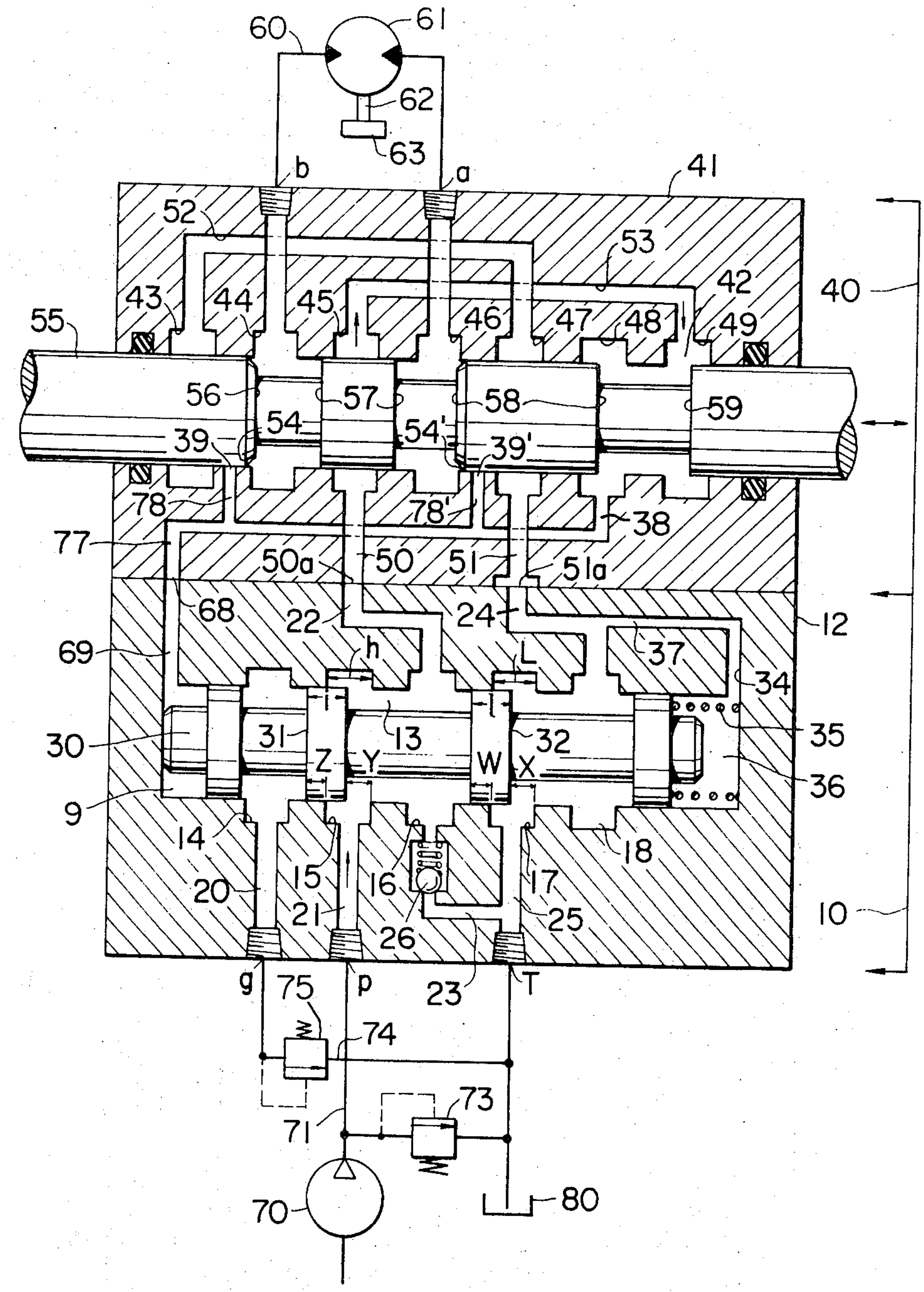
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FIG. 6

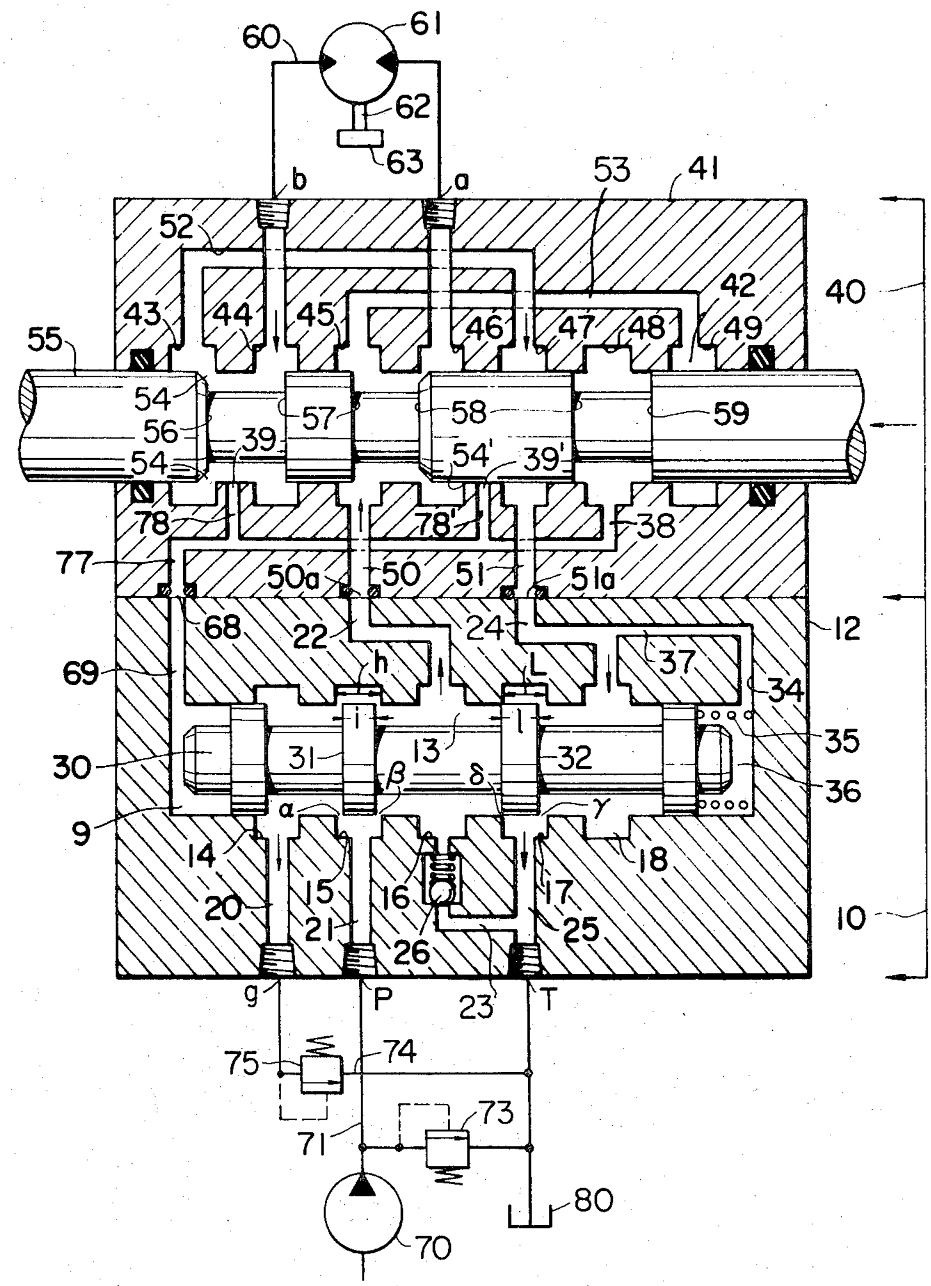
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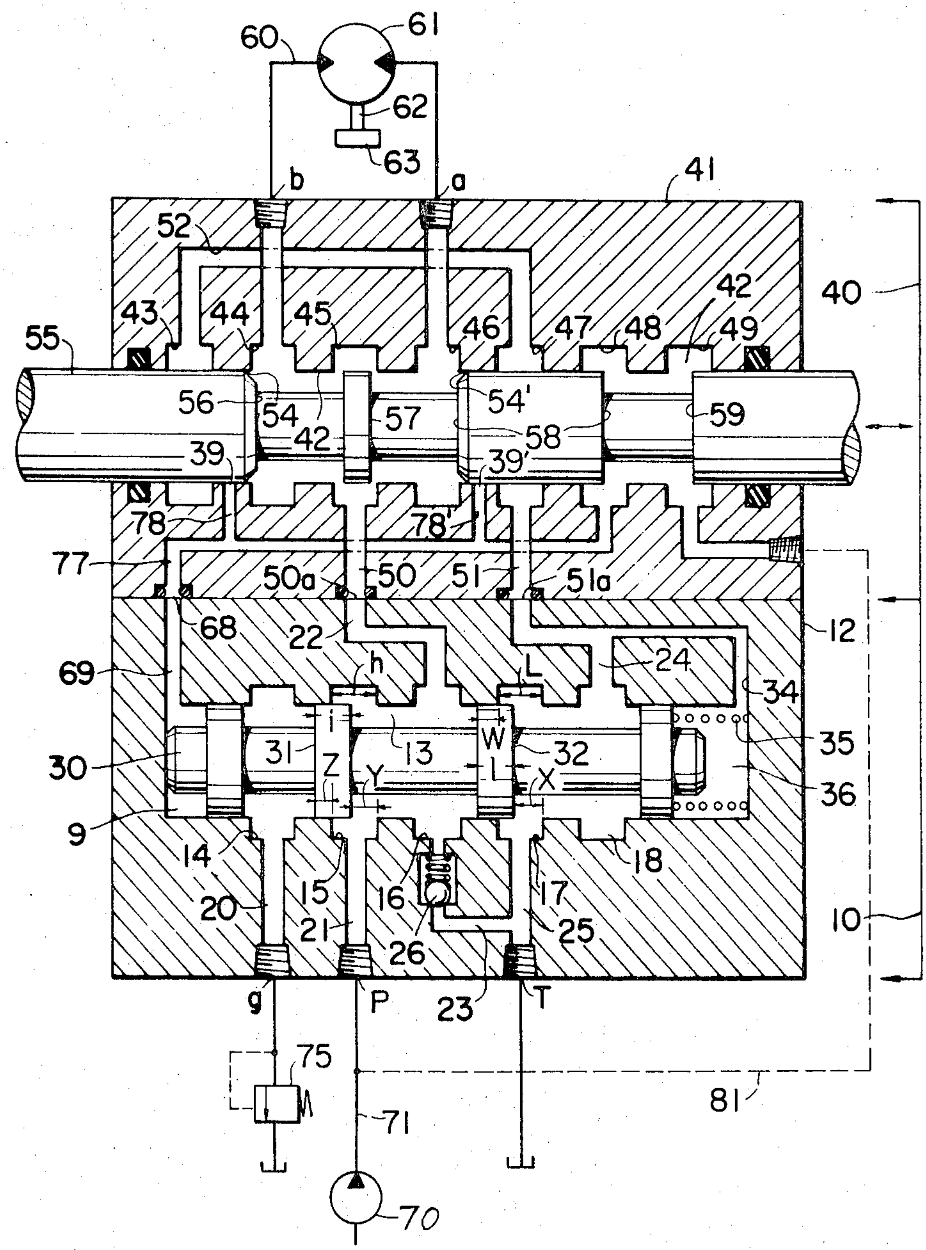
FIG. 7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FLUID CONTROLLING APPARATUS

Various fluid controlling apparatus have been proposed heretofore, which control the operating direction and operating speed of an actuator, e.g. a fluid motor, by varying the displacement of a single valve spool wherein the operating direction and speed of the actuator is controlled in proportion to the displacement of said valve spool. Such prior art apparatus, however, have the shortcoming of having a limiting range of application because they might be highly satisfactory as one to achieve one function but do not concurrently achieve other functions. For instance, the fluid controlling apparatus disclosed in U.S. Pat. No. 3,488,953 and U.S. Pat. No. 3,602,104 are excellent in that a loss of driving force can be minimized by holding pump pressure as low as possible, both when the actuator for operating a working machine is in operation and when the same is not in operation. The apparatus of the latter patent has, in addition to such advantage, the further advantage that a plurality of actuators can be operated concurrently with a single pressure source. However, the aforementioned apparatus have the disadvantage that, while a satisfactory operation may be obtained when the direction of the load exerted on the actuator is opposite to the direction of operation of said actuator, in the event when the direction of the load exerted on the actuator is the same as the direction of operation of the actuator, and hence, the load aids the operation of said actuator, the apparatus do not perform the intended function. Even if the apparatus were arranged so as to perform the intended function when the direction of the load is the same as the operating direction of the actuator, an abnormal pressure would occur in the circuit, causing breakage of the entire circuit.

As a practical example of the case wherein a load is exerted on an actuator in the same direction as the operating direction of the actuator and aids the operation of said actuator, reference may be made to a vehicle (working machine) in which the wheel driving force is obtained from a fluid motor (actuator). In case of such vehicle, the direction of the load and the operating direction of the fluid motor are opposite to each other when the vehicle is running on the normal flat surface, but when the vehicle is running on a descending slope, the wheels rotate freely due to the gravitational downward movement of the vehicle, so that the fluid motor does not operate as motor but operates as pump. Thus, the load is exerted on the actuator in the same direction as the operating direction of the actuator and increasingly aids the operation (running) of the vehicle. Such a condition also occurs also in winches, elevators and other various fluid-driven apparatus which utilize an actuator.

A primary object of the present invention is to provide a novel and improved fluid controlling apparatus which obviates the above-described disadvantage and which is so designed that, when a load exerted on an actuator is acting in a direction to aid the operation of said actuator, fluid passage between a pressure supply line and the inlet side of the actuator is blocked and the fluid is controlled between a return line and the outlet side of the actuator. Further the return line and the inlet side of the actuator are interconnected to form a short circuit therebetween, whereby the pressure of the actuator appears only in said short circuit and thus, the occurrence of an abnormal pressure which has occurred in the prior art apparatus is avoided.

Another primary object of the invention is to provide a fluid controlling apparatus which uniquely is capable of meeting any and all demands including such demands for which a plurality of different types of apparatus have been required in the past, and which is capable of mass production.

Still another object of the invention is to provide a fluid control apparatus which is capable of operating a plurality of actuators with a single pressure source.

A further object of the invention is to provide a fluid control apparatus which, when used for operating a plurality of actuators, is capable of supplying the flow of fluid, preferentially to a primary actuator, without being influenced by the states of loads exerted on the other actuators.

An additional object of the invention is to provide a fluid control apparatus which, when used for operating a plurality of actuators, is capable of maintaining the pump pressure at a level equal to or slightly higher than the largest one of the loads exerted on the actuators, whereby the rotational force required for driving a pressure generating source is reduced.

Another additional object of the invention is to provide a fluid control apparatus in which a fluid inlet passage connected to a fluid pressure source when a valve spool of a directional control valve member is in its neutral position can be formed so as to enable in a desired fluid controlling operation to be achieved, and which can be provided in a compact form as a whole.

Other objects and advantages of the invention will become apparent from the following description of the embodiments thereof shown in the accompanying drawings.

FIG. 1 is a sectional view of a first embodiment of the fluid control apparatus according to the present invention, in which a flow regulating portion is provided on the meter-in side;

FIG. 2 is a sectional view of the fluid controlling apparatus of FIG. 1, in which the spool is displaced to the left;

FIG. 3 is a fragmentary enlarged view of the plunger shown in FIG. 2;

FIG. 4 is a second embodiment of the fluid control apparatus according to the invention, in which a flow regulating portion is provided on the meter-in side;

FIG. 5 is a sectional view of a third embodiment of the fluid control apparatus of the invention, in which a flow regulating portion is provided on the meter-out side;

FIG. 6 is a sectional view of the apparatus of FIG. 5, in which the spool is displaced to the left; and

FIG. 7 is a sectional view of a fourth embodiment of the fluid control apparatus of the invention, in which a flow regulating portion is provided on the meter-out side.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

The present invention will be described here-under with reference to the drawings showing diagrammatically the most preferred embodiments thereof. Since the constructions and modes of operation of these embodiments are considerably different depending upon whether a flow regulating portion provided on a directional control valve is in a so-called meter-in circuit communicating a pump with the inlet side of a fluid motor, or in a so-called meter-outer circuit communicating the pump with the outlet side of the fluid motor with a tank port, the case wherein the flow regulating portion is provided in the meter-in circuit, will be described at first.

I. The case wherein the flow regulating portion is in the meter-in circuit:

The fluid controlling apparatus shown in FIG. 1 comprises two units including a directional control valve unit generally indicated at **40** which operates an actuator **61** in a desired direction at a desired speed, and a pressure-compensating unit generally indicated at **10** which controls the pressure acting on the directional control valve unit **40.** While the pressure-compensating unit **10** and the directional control valve unit **40** are shown as being provided separately and coupled together, they may alternatively be provided integrally as a monoblock, or another same directional control valve unit or units may be mounted in turn on the directional control valve unit **40** in stacked relation.

The directional control valve unit **40** will be described hereunder:

This directional control valve unit **40** includes a main body **41** defining therein a spool slide cavity **42.** The spool slide cavity **42** has seven annular grooves **43, 44, 45, 46, 47, 48** and **49** formed in the wall thereof. Of these seven annular grooves, the four annular grooves **44, 45, 46** and **47** are open in the outside of the main body **41** at four ports *b*, **50***a*, *a* and **51***a* respectively. The leftmost annular groove **43** is communicated with the annular groove **47** by a passage **52** and, therefore, is communicated with a return passage **51** through said annular groove **47.** A closed circuit **60** is connected between the motor ports *a*, *b* (hereinafter referred to simply as ports *a* and *b* respectively) communicating with the annular grooves **46, 44** respectively, and a fluid motor **61** is provided in said closed circuit **60.** The fluid motor **60** is capable of operating a load **63** (working machine) by its drive shaft **62** which is rotated in one or the other direction by the fluid passing in the closed circuit **60** in the right or left direction. The rightmost annular groove **49** is connected with a tank **80** through a passage **79.**

In the spool slide cavity **42** is slidably disposed a valve spool **55** having four lands **56, 57, 58** and **59.** The flow of fluid is controlled as described below by the sliding movement of the spool **55.**

Thus, the leftmost land **56** serves to sever communication between the annular grooves **43** and **44** when the valve spool **55** is in neutral position (the position shown in FIG. **1**), and to establish communication between the annular grooves **43** and **44** when said spool valve **55** is displaced to the left to the position shown in FIG. **2.** The second land **57** from the left closes off an inlet passage **50** and pilot ports **39, 39′** provided on both sides of said passage **50** when the valve spool **55** is in its neutral position as shown in FIG. **1,** and opens the pilot port **39** and forms a variable inlet orifice between the right end face of the land **57** and the annular groove **45,** or a flow regulating portion **54** in a passage communicating the inlet passage **50** with the port *a*, when said spool valve **55** is displaced to the left as shown in FIG. **2.** Conversely, when the valve spool **55** is displaced to the right, the land **57** opens the pilot port **39′** and forms a flow regulating portion **54′** in a passage communicating the inlet passage **50** with the port *b*. The third land **58** from the left serves to establish or sever communication between the annular grooves **46** and **47** by its left end face, according to the position of the valve spool **55.** Thus, it severs the communication when the valve spool is in its neutral and leftward position, and establishes the communication when the valve spool is in its rightward position. The right end face of the land **58** and the rightmost land **59** serve to open or close off a vent passage of the pressure-compensating unit **10** to be described later, according to the position of the valve spool **55.**

Now, the pressure-compensating unit **10** will be described. This pressure-compensating unit **10** includes a main body **12** defining therein a plunger slide cavity **13.** The plunger slide cavity **13** has five annular grooves **14, 15, 16, 17** and **18** formed in the wall thereof. The leftmost annular groove **14** is communicated with a port *g*, formed in the main body **12,** through a bypass passage **20.** The second annular groove **15** from the left is communicated with a port P through an inlet passage **21.** The third annular groove **16** from the left is provided with an outlet passage **22** and a passage **23.** The outlet passage **22** is communicated with the inlet port **50***a* of the directional control valve unit **40** and the passage **23** is communicated with a port **27** through a check valve **26.** The fourth annular groove **17** from the left is communicated with the return port **51***a* of the directional control valve unit **40** through a tank passage **24.** The fifth annular groove **18** is communicated with a port T through an outlet passage **25.**

In the plunger slide cavity **13** of the main body **12** is slidably disposed a pressure-compensating plunger **30.** The plunger **30** is held in its leftward position by a spring **35** disposed in a back pressure chamber **36** formed on the right side of said plunger **30,** whereby the inlet passage **21** and the bypass passage **20** are isolated from each other. To the back pressure chamber **36** is connected one end of a passage **37,** and the other end of said passage **37** is branched into three passages in the main body of the directional control valve unit **40.** One of the branch passage **38** is communicated with the annular groove **48** and the other two branch passages form the aforesaid pilot ports **39, 39′** respectively which are open in the spool slide cavity **42** on both sides of the annular groove **45.** A reference pressure chamber **9** formed on the left side of the plunger **30** is communicated with the outlet passage **22** through the passage **29** on the upstream side of the flow regulating portions **54, 54′**, for receiving a reference pressure to operate the plunger **30.** The pressure-compensating plunger **30** has two lands **31, 32** at about mid portion thereof. The width of the left hand land **31** is determined in relation with the width of the annular groove **15** and the displacement and diameter of the plunger **30.** Namely, in FIG. **1,** the plunger **30** is urged to the left by the spring **35** and in this position the land **31** severs communication between the passage **21** and the bypass passage **20,** but establishes communication between the inlet passage **21** and the outlet passage **22.** When high pressure fluid is introduced into the reference pressure chamber **9** and the plunger **30** is displaced to the right, the inlet passage **21** is communicated with the bypass passage **20** at first. The width of the land **31** is made smaller than that of the annular groove **15** so that the inlet passage **21** is communicated also with the outlet passage **22** in this case. This state is shown in FIG. 3 in an enlarged scale. In general, when fluid is passed in a conduit provided with an orifice, a pressure differential occurs between the fluid pressure upstream of said orifice and the fluid pressure downstream of the same when the opening area of said orifice is smaller than a certain value. Based on this knowledge, in the present invention the land 31 is formed so as to form between the left end face thereof and the annular groove 15 an orifice to create a pressure differential (hereinafter called divided flow controlling portion (α)) during the period before said left end face reaches the position 31*a* shown in FIG. 3. When the left end face of the land 31 is in the divided flow controlling position, the right end face thereof is not in the pressure controlling position to be described later. As the plunger 30 is displaced further to the right and when the left end face of the land 31 is located rightwardly of the position 31*a*, the divided flow control is released. As the plunger 30 is displaced further to the right and when the right end face thereof is located in the position 31*b* shown in FIG. 3, an orifice to create a pressure differential (hereinafter called first pressure controlling portion (β)) is formed between the right end face of said land 31 and the annular groove 15 conversely to the preceding case. In summation, the land 31 and the annular groove 15 are arranged in the relation of Z<Y in FIG. 1 (with the plunger 30 being in the leftmost end of its stroke), so that, during the rightward stroke of the plunger 30, the divided flow controlling state may be obtained at first upon displacement of said plunger over a certain distance, then said divided flow controlling state be released upon further displacement of the plunger over a predetermined distance, and the first pressure controlling state be obtained upon further displacement of the plunger. The other land 32 is given the same consideration as that given to the land 31. Namely, the land 32 is arranged so as to form an orifice to create a pressure differential (hereinafter called second pressure controlling portion (γ)) between the right end face thereof and the annular groove 17 when the plunger 30 is displaced further to the right to a point to sever communication between the inlet passage 21 and the outlet passage 22. That is to say that the lands 31, 32 are formed in the relation of Z<Y<X in FIG. 1.

The port P mentioned above is connected to a pressure supply line 71 which is provided therein with a fixed type fluid pump 70, and a port T is connected to the tank 80 through a back pressure valve 72. A relief valve 73 is connected to the pressure supply line 71 on the outlet side of the fluid pump 70 and the secondary side of said relief valve 73 is connected to the port 27 and the primary side of the back pressure valve 72. Further, a divided flow line 74 connected at one end to the port *g* is connected to the primary side of the back pressure valve 72, and an actuator is provided in said divided flow line 74. In FIG. 1, this actuator is represented by a pressure regulating valve 75 for the sake of convenience. Where it is necessary to distinguish clearly the actuator 75 from the fluid motor 61, the former will be referred to as second actuator and the latter as first actuator.

The function and effect of this embodiment of the fluid controlling apparatus according to the invention will be described hereinafter. In the apparatus, the relation between a load exerted on the first actuator 61 and a load exerted on the second actuator 75 brings about substantial differences in function and effect. Therefore, the case wherein the load exerted on the second actuator 75 is larger than the sum of the load exerted on the first actuator 61 and the force of the spring 35 and the case wherein the former is smaller than the latter, will be described in sequence.

a. The case wherein the load exerted on the second actuator 75 is larger than the sum of the load exerted on the first actuator 61 and the force of the spring 35.

In the directional control valve unit 40, when the valve spool 55 is in its neutral position as shown in FIG. 1, the port *a*, the port *b*, the inlet passage 50 communicating with the port P and the port T are blocked respectively by the valve spool 55, whereby the fluid motor 61 is positively held inoperative. However, the pressure supply line 71 is communicated with the reference pressure chamber 9 on the left side of the plunger 30 through the port P, the inlet passage 21, the outlet passage 22 and the passage 29, while the back pressure chamber 36 is communicated with the tank 80 through the passage 37, the branch passage 38, the annular grooves 48, 49 and the passage 79 and maintained at the atmospheric pressure.

Therefore, when the fluid pump 70 is driven from a prime mover (not shown) under such condition, the fluid discharged from said fluid pump 70 flows into the reference pressure chamber 9 on the left side of the plunger 30 through the pressure supply line and starts to act on the left end face of said plunger 30. At the point when the reference pressure acting on the left end face of the plunger 30 has become larger than the biasing force of the spring 35, the plunger 30 starts to move to the right, opening the divided flow controlling portion (α) at first and then closing the first pressure controlling portion (β). Thus, the entire fluid being discharged from the fluid pump 70 is bypassed to the tank 80 through the divided flow controlling portion (α) and the divided flow line 74, with a pressure which is the sum of the set pressures of the pressure regulating valve 75 and the back pressure valve 72. As such, this embodiment has the capacity of bypassing the fluid discharged from the pump while the valve spool 55 is in its neutral position.

When the valve spool 55 is displaced to the left to the position shown in FIG. 2 thereafter, communication between the annular grooves 48 and 49 is severed by the rightmost land 59 at first, and successively thereafter the pilot port 39 is opened by the valve spool 55 and then the flow regulating portion 54 is opened in a desired degree. As a result, the pressure in the reference pressure chamber 9, by which the plunger 30 is held in its rightward position, becomes temporarily the same as the pressure in the back pressure chamber 36, allowing said plunger 30 to make a return movement to the left under the biasing force of the spring 35. Concurrently, the fluid in the pressure supply line 71 flows to the fluid motor 61 via the port *a* and returns to the tank 80 via a return line constituted by the port *b*, the annular grooves 44, 43, the passage 52, the tank passage 24, the second pressure controlling portion (γ), the outlet passage 25, the port T and the back pressure valve 72. Thus, the fluid motor 61 starts rotating at a speed corresponding to the flow of fluid passing at the flow regulating portion 54 formed on the inlet side of said motor and concurrently a pressure differential is created by the resistance to flow of said flow regulating portion 54, by which pressure differential a circuit is formed to lead the pressure on the upstream side of the flow regulating portion **54** or in the pressure supply line to the reference pressure chamber **9** on the left side of the plunger **30** through the passage **29**, and a circuit is formed to lead the pressure on the downstream side of said flow regulating portion **54** to the back pressure chamber **36** through the pilot port **39** and the passage **37.** The plunger **30** again starts to move to the right against the biasing force of the spring **35**, due to the pressure differential between both sides thereof, and is held stationary in the position where the pressures acting on both ends of said plunger **30** are balanced as a result of the force of spring **35** increasing.

The discharge pressure of the pump in this case is the sum of the pressure in the second actuator **75** and the set pressure of the back pressure valve **72.** The pressure differential between both sides of the flow regulating portion **54** becomes larger as the opening at the first pressure controlling portion (β) is larger, with the result that the fluid passes in the fluid motor **61** at a rate higher than actually required. However, the plunger moves to the right under the effect of said pressure differential, diverging the fluid in excess of that required for operating the fluid motor **61** from the flow in the inlet passage **21** through the divided flow controlling portion (α) and reducing the fluid pressure in the outlet passage **22** to thereby maintain the pressure of fluid at the portion from said first pressure controlling portion (β) to the flow regulating portion **54** at a value which is the sum of the pressure created by the load **63**, that is, the load pressure, the pressure created by the resistance to flow at the flow regulating portion **54** and the resistance to flow at the back pressure valve **72.** Thus, it will be understood that, in the event that the supply of fluid to the fluid motor **61** is larger than necessary, the pressure differential between both sides of the flow regulating portion **54** becomes large, displacing the plunger **30** further to the right according to the interrelation between said pressure differential and the biasing force of the spring **35,** whereby the pressure reducing effect at the first pressure controlling portion (β) is increased. Conversely, in the event when the supply of fluid to the fluid motor **61** is smaller than necessary, the pressure differential between both sides of the flow regulating portion **54** becomes small, so that the plunger **30** displaces to the left according to the interrelation between said pressure differential and the biasing force of spring **35,** whereby the opening at the first pressure controlling portion (β) becomes large and the supply of fluid to the fluid motor **61** increases. Thus, the supply of fluid to the fluid motor **61** is maintained constant. In other words, the pressure differential between both sides of the flow regulating portion **54** is maintained constant by the spring **35.** Consequently, the flow of fluid at the flow regulating portion **54** is maintained always constant with respect to a certain preset opening of said flow regulating portion **54,** whereby the operating speed of the fluid motor **61** is maintained constant.

By changing the operation of the flow regulating portion **54** thereafter, upon operating the valve spool **55,** a flow of fluid corresponding to the changed opening is obtained. Namely, the displacement of the valve spool **55** and the flow of fluid can be controlled in proportional relation.

The operating speed of the fluid motor **61** is thereafter maintained constant in the following manner, with the opening of the flow regulating portion **54** being constant, even when the load **63** exerted to the fluid motor **61** has, for instance, increased.

Thus, when the load **63** has increased, the pressure between the first pressure controlling portion (β) and the fluid motor **61** increases, with the result that the pressure differential between both sides of said first pressure controlling portion (β) becomes small and a larger portion of the fluid discharged from the fluid pump **70** is diverged to the divided flow controlling portion (α) where the resistance to flow is relatively small, so that the supply of fluid to the fluid motor **61** decreases and the pressure differential between both sides of the flow controlling portion **54** becomes small. As a result, the plunger **30** displaces to the left according to the interrelation between said pressure differential and the biasing force of spring **35,** making large the opening at the first pressure controlling portion (β) and increasing the supply of fluid to the fluid motor **61.** In other words, the predetermined pressure differential set by the spring **35** is restored between both sides of the flow regulating portion **54,** with which the operating speed of the fluid motor **61** is maintained constant.

On the contrary, when the load **63** exerted on the fluid motor **61** has decreased, the operating speed of said fluid motor **61** is controlled to be constant in the following manner:

Thus, with the load **63** decreasing, the supply of fluid to the fluid motor **61** increases, with the result that the pressure differential between both sides of the flow regulating portion **54** becomes large. The plunger **30** displaces to the right according to the interrelation between said pressure differential and the biasing force of spring **35** to decrease the opening of the first pressure controlling portion (β). Thus, an excess pressure is removed at the first pressure controlling portion (β) and the supply of fluid to the fluid motor **61** is decreased concurrently, whereby the pressure differential between both sides of the flow regulating portion **54** is maintained constant irrespective of the varying load.

Accordingly, the operating speed of the fluid motor **61** is maintained always constant corresponding to the set value of the flow regulating portion **54,** irrespective of pressure variation in the fluid circuit.

Although the foregoing description has been made with reference to the case wherein the operating direction of the actuator **61** and the direction of load exerted thereon are opposite to each other, it will be understood from the following description that the operating speed of the fluid motor is controlled to be constant also in the case when said both direction are the same:

Thus, in such case the fluid motor **61** operates as a pump, sucking the fluid on the inlet side and discharging the same to the outlet side or into the return line unrestrictedly. As a result, the pressure of fluid on the inlet side of the fluid motor **61** falls rapidly and the pressure in the back pressure chamber **36** also falls rapidly, so that the pressure differential between both sides of the flow regulating portion **54** increases, causing the plunger **30** to move further to the right to close off the first pressure controlling portion (β) at first. With the first pressure controlling portion (β) being thus closed off, the pressure of fluid in the passage from said first pressure controlling portion to the fluid motor **61** tends to become negative. However, since the return fluid discharged from the fluid motor **61** and the fluid discharged from the pump and returning to the tank **80** through the divided flow controlling portion (γ) undergo the resistance created by the back pressure valve **72**, a portion of these return fluid flows to the inlet side of the fluid motor **61** with the pressure set in the back pressure valve **72** while forcibly opening a check valve **26**, and enables said fluid motor to continue its rotation. Thus, the return line is communicated with the inlet side of the fluid motor **61** by a passage **23** and the check valve **26**. In this case, the pressure on the upstream side of the flow regulating portion **54** is equal to the pressure value set in the back pressure valve **72** and the pressure on the downstream side of the same is lower than said pressure value just by a value corresponding to the resistance to flow created at the flow regulating portion **54**. The plunger **30** continues its rightward movement by this pressure differential. When the opening of the second pressure controlling portion (γ) is decreased to smaller than necessary by the displacement of the plunger **30**, the pressure on the outlet side of the fluid motor **61** becomes extremely high, decreasing the rate of rotation of said fluid motor **61**. Consequently, the rate of suction by the fluid motor **61** decreases and the resistance to flow at the flow regulating portion **54** becomes small, so that the pressure differential between both sides of said flow regulating portion becomes small. The plunger **30** moves to the left, expanding the opening of the second pressure regulating portion (γ). Conversely, when the opening of the second pressure regulating portion (γ) is excessively large, the pressure on the outlet side of the fluid motor **61** falls, increasing the rate of rotation of said fluid motor. Therefore, the pressure on the inlet side of the fluid motor **61** tends to be negative and the resistance to flow at the flow regulating portion **54** increases. Consequently, the pressure differential between both sides of the flow regulating portion **54** becomes large, causing the plunger **30** to move to the right to decrease the opening of the second pressure controlling portion (γ). Thus, the plunger **30** is held immovably in a position where the sum of the pressure on the downstream side of the flow regulating portion **54** and the biasing force of the spring **35** is balanced with the pressure on the upstream side of said flow regulating portion **54** or the set pressure of the back pressure valve **72** owing to the pressure controlling function of the second pressure controlling portion (γ) of said plunger. Thus, it will be understood that the pressure differential between both sides of the flow regulating portion **54** is maintained constant by the spring **35**, even when the fluid motor **61** is running by itself.

The most important feature of the apparatus of this invention is that exertion of an excessively large pressure on the outlet side of the fluid motor **61** can be avoided in such state, as may be understood from the following description: when the fluid motor **61** starts running by itself as described above, the plunger **30** moves to the right to a position closing off the first pressure controlling portion (β) but leaving the divided flow controlling portion (α) fully opened. Therefore, the discharge pressure of the fluid pump **70** is equal to the sum of the set pressures of the pressure regulating valve **75** and back pressure valve **72**. However, the outlet passage **22** and tank passage **25** are communicated with each other through the check valve **26** and the pump discharge pressure is not supplied to the fluid motor **61** as the first pressure controlling portion (β) is closed off, so that the pressure occurring in the short circuit is only that created by the self-running fluid motor **61**.

For rotating the fluid motor **61** in the opposite direction, this can be achieved by displacing the valve spool **55** rightward from its neutral position. By the displacement, the flow regulating portion **54** and the pilot port **39** on the right side of the annular groove **45** are kept closed off, and the pilot port **39'** and the flow regulating portion **54'** on the left side of said annular groove are opened by the valve spool **55**. In this case, a bent unload passage leading from the back pressure chamber **36** to the tank is closed off by the land **58** and a circuit symmetrical with that described above is formed, and the flow of fluid can be controlled during the reverse rotation of the fluid motor **61** by the same flow controlling function as described above.

Summarizing the embodiment described hereinbefore, the directional control valve unit is of a four-way, all ports block, neutral-unloading, and flow regulating portion on meter-in type and the pressure-compensating valve unit is of a type having a first pressure controlling portion (β) and a second pressure controlling portion (γ) for fluid control. The main relief valve **73** operates only when the fluid motor **61** stops its rotation under excessive load.

The pressure differential between both sides of the flow regulating portion **54** is maintained constant corresponding to the biasing force of the spring **35** in either case. Therefore, by operating the valve spool **55**, it is possible to start, accelerate, decelerate and stop the fluid motor **61**, as well as to proportion the displacement of the valve spool **55** with the speed of the fluid motor **61** indpendently of fluctuation of the circuit pressure. It is further possible to achieve a proportional control without the occurrence of excessively large pressure even when the load direction and motor operating direction are the same.

Although the present invention has been described in detail hereinbefore with reference to FIGS. **1, 2** and **3**, the embodiment of the apparatus described above is adapted for use in the case wherein the load exerted on the other hydraulic means (second actuator) connected to the divided flow line is larger than the sum of the load exerted on the first actuator **61** and the biasing force of the spring **35**, and the divided flow controlling portion (α) serves merely to bypass or divide excess fluid. It should be noted, however, that the divided flow controlling portion (α) has a pressure compensating function per se which has not been described. It should also be understood that the divided flow controlling portion (α) is not necessary when the other hydraulic means is not connected to the divided flow line, and the same function and effect may be achieved by arranging the apparatus, for example, such that the divided flow controlling portion (α) is held closed by blocking the port *g* either by extending the width *i* of the land **31** in the leftward direction or by means of a plug, and the relief valve **73** is connected to the outlet side of the fluid pump **70** for discharging the bypass and divided flow therethrough. Thus, a completely different apparatus can be submitted simply by plugging the port *g*.

However, with the other hydraulic means connected to the divided flow line, the load exerted on said means changes frequently from time to time during the operation of an actuator of said means, as is well known, and it is not always the case that the changing load is larger than the sum of the load exerted on the actuator **61** and the biasing force of the spring **35.**

Thus, it becomes necessary to arrange the apparatus such that the same function and effect as stated above may be obtained even if the load exerted on the second actuator happens to be smaller than the sum of the load exerted on the actuator **61** and the biasing force of the spring **35** (including the case wherein no load is exerted on the second actuator).

b. Now, the operation of the apparatus in the event when the load on the second actuator is smaller than the sum of the load on the actuator **61** and the biasing force of the spring **35** will be described in detail hereinafter with reference to FIGS. **1** and **3:**

In the event when the load on the other hydraulic means, that is, the load created by the resistance of the second actuator **75** is larger than the sum of the load **63** on the actuator **61** and the biasing force of the spring **35,** during normal operation of the actuator, the fluid discharged from the fluid pump **70** tends to flow toward the actuator **61** from the pressure supply line, and the plunger **30** is displaced corresponding to the load **63** on the actuator so as to control the flow of said fluid at the first pressure controlling portion (β) (pressure reducing function). When the load on the other hydraulic means, i.e. the load on the second actuator **75,** becomes smaller than the sum of the load **63** on the first actuator and the biasing force of the spring **35** (including the case wherein no load is exerted on the second actuator) under such condition, the fluid discharged from the fluid pump **70** is conversely bypassed into the port g through the divided flow controlling portion (α). In order to avoid this, the plunger **30** moves to the left to control the flow bypassing into the divided flow line.

Thus, the plunger **30** displaces to the left to a position in which the left end face of the land **31** is located in the position indicated by the phantom line **31***a* illustrated in FIG. **3,** whereby the divided flow controlling portion (α) comes into effect, with the first pressure controlling portion (β) being ineffective.

With the divided flow controlling portion (α) being in effect, the pump discharge pressure takes a value which is the sum of the pressure created by the load **63,** the flow regulating portion **54** and the back pressure valve **72** respectively, and the pressure of the fluid in the passage from the divided flow controlling portion (α) to the pressure regulating valve **75** is maintained at a value which is the sum of the set pressures of said pressure regulating valve **75** and back pressure valve **72.**

Thereafter, when the load **63** comes to have a changing tendency, the flow of fluid is controlled by the changing opening of the divided flow controlling portion only, with the first pressure controlling portion (β) being held open, in a manner to maintain the pressure differential between both sides of the flow regulating portion **54** constant, contrary to the controlling function of said first pressure controlling portion (β) mentioned above. Since the excess flow of fluid discharged from the pump can be bypassed into the divided flow line and this bypass flow can be utilized for driving the other hydraulic means, during both the normal operation and self running of the actuator **61,** effective use of the driving force of the pump **70** becomes possible.

Further, since the fluid can be supplied to the actuator **61** at a constant rate proportional to the displacement of the valve spool **55,** in spite of a fluctuation of load exerted on the other hydraulic means, it is possible to supply fluid preferentially to the first actuator **61** to ensure satisfactory operation thereof, without being influenced by the operating condition of the other hydraulic means. The apparatus of the invention has the following advantageous feature when the load on the pressure regulating valve **75,** during normal operation of the actuator is smaller than the sum of the load **63** and the biasing force of the spring **35.**

In the conventional valve apparatus the pump discharge pressure is influenced by the relief valve **73** but in the present invention it is influenced by the load **63** on the actuator **61,** and therefore, the output of the pump can be smaller than in the conventional apparatus an amount corresponding to the difference between the load on the relief valve and the load on the actuator. Thus, the outpt loss of the pump can be minimized.

The output loss of the pump can be minimized also because the fluid discharged from the pump is bypassed in its entirety into the divided flow line during the self-running operation of the actuator.

It will be obvious that, when the pressure regulating valve **75** is not provided and the divided flow line **74** is arranged to open directly into the tank, a functional effect equivalent to that obtainable when the load on the pressure regulating valve **75** is smaller than the sum of the load **63** on the actuator **61** and the biasing force of the spring **35,** can be achieved and further self-running control is possible.

In the present invention, as described above, the pressure compensation is achieved by the effect of the first pressure controlling portion (β) when the resistance of the pressure regulating valve **75** is larger than the sum of the load **63** and the biasing force of the spring **35** during normal operation, and by the effect of the divided flow controlling portion (α) when the former is smaller than the latter. Since the pressure compensation is effected independently of the sizes of the two loads, pressure compensation can be achieved satisfactorily regardless of the fluctuation of the load exerted on the second actuator provided in the divided flow line (the line leading to the port g). This is true whether the load on said second actuator becomes larger or smaller than the sum of the load **63** during normal operation of the first actuator and the biasing force of the spring **35.**

Hereinbefore, an embodiment of the invention has been described in detail with reference to FIGS. **1, 2** and **3,** and now other embodiments will be described briefly.

The embodiment shown in FIG. **4** differs from that of FIG. **1** in respect of the path of fluid when the fluid motor **61** is in self-running operation. It differs from the latter also in that it is superior in respect to the stability of operation when the load **63** on the fluid motor **61** is larger than a predetermined value, and in the bent unload type. These differences will be described in detail hereunder:

First of all, the difference in respect of the flow path of fluid during self-running operation of the fluid motor will be described. In this embodiment, the positions of the annular grooves **17**, **18** in the pressure-compensating unit **10** are swapped as shown in FIG. **4**, whereby, while in the embodiment of FIG. **1** the fluid passing at the second pressure controlling portion (γ) flows from the tank pressure **24** into the plunger slide chamber **13** to form a so-called converging flow, in the embodiment of FIG. **4** the fluid passing at the same controlling portion (γ) flows from the plunger slide chamber **13** into the outlet passage **25** to form a so-called expanding flow. Further, in this embodiment the width L of the annular groove **18** and the width of the land **32** are selected in the relation of $l<L$ and Z, Y, W, X in the relation of $Z<Y \leqq W<X$. With such an arrangement, when the fluid motor **61** starts self-running and the first pressure controlling portion (β) is closed off by the plunger **30** which is displaced to the right by virtue of the pressure differential beween both sides of the flow regulating portion **54**, a flow direction controlling portion (δ) is opened fully, either concurrently or immediately thereafter, owing to the relation of $Y \leqq W$ mentioned above, whereby the tank passage **24** and outlet passage **22** are communicated directly with each other. The pressure set by the back pressure valve **72**, during this period, is led into the upstream side of the flow regulating portion **54** through the check valve which is forcibly opened by said pressure, and maintains the pressure in said portion at a superatmospheric pressure.

Thus, the fluid discharged from the self-running fluid motor **61** cycles in the circuit formed by the port *b*, the annular grooves **44, 43,** the passages **52, 24,** the second pressure controlling portion (γ), the flow direction controlling portion (δ), the passage **22** and the flow regulating portion **54,** and maintained at its pressure by the back pressure valve **72.** Thus, in the embodiment of FIG. **4** the fluid sucked by the fluid motor **61** during the self-running operation of said motor is supplied by the recycled return fluid through the flow direction controlling portion (δ), which in the embodiment of FIG. **1,** is supplied through the check valve **26** in its entirety, and therefore, the check valve **26** can be reduced in size.

The embodiment of FIG. **4** is distinguishable over that of FIG. **1** also in that safety of the circuit can be secured more positively.

Namely, in this embodiment a pilot relief valve **67** is provided in the line **66** upon branching said line **66** in a direction from the back pressure chamber **36** towards the tank **80** and a restriction **65** is provided in the passage **37,** while in the directional control valve unit **40** the rightmost annular groove **49** and the passage **52** are communicated with each other through a passage **64.** Such arrangement enables one to enhance the safety of circuit in the event when the load **63** on the fluid motor **61** has increased above a predetermined value. Thus, when the load **63** on the fluid motor **61** has increased above the predetermined value and the pressure in the back pressure chamber **36** has risen above the set pressure of the pilot relief valve **67** during the period in which the fluid motor **61** is operating the load **63** in the normal condition and the load on the second actuator is larger than the sum of the load on the first actuator and the biasing force of spring **35,** said pilot relief valve **67** is opened to discharge the fluid. At the same time, a pressure differential is produced between both sides of the restriction **65** due to the resistance to flow of said restriction and the plunger **30** is displaced to the right, with the result that the pressure downstream of the first pressure controlling portion (β) is limited to a predetermined value set by said pilot relief valve **67,** by the pressure reducing effect of said first pressure controlling portion (β). It is to be noted that under such condition, the rate of rotation of the fluid motor **61** is not constant and is lower than the set value or the motor stops running.

Further, the divided flow controlling portion (α) or flow direction controlling portion (δ) of the plunger **30** respectively plays the role of relief valve, in the size relation of the load **63** on the first actuator **61,** the load on the second actuator **75** and the set pressure of the pilot relief valve **67.**

The apparatus of the invention wherein a flow regulating portion is provided in a meter-in circuit (case I)) have been described hereinbefore with reference to FIGS. **1, 2, 3, 4** but, as another embodiment, the inlet side of the fluid motor **61** may be communicated directly with the tank **80** through a check valve to form a hydraulic circuit during self-running of the actuator.

The directional control valve unit **40** in each of the embodiments described above is of a type having four ports or two flow regulating portions **54, 54'**. This is to meet the requirement that the operating direction of the actuator **61** is changed between two directions and the flow rate and pressure of fluid are controlled during the operation of the actuator in each direction. However, in the event when the flow rate and pressure of fluid are required to be controlled in the operation of the actuator only in one direction, such control can be achieved easily with only one flow regulating portion. Thus, the apparatus of the invention may be provided in various forms, without deviating from the scope of the invention set forth in the appended claims, by optionally combining the above-described embodiments with respect to the symbols of directional control valve, such as the numbers of direction and position.

In each of the embodiments described above, the flow resistance in each passage is ignored but it should be understood that each passage has a cross sectional area large enough not to impair smooth operation of the apparatus. Descriptions of the detailed portions which are well known in the art are also omitted.

(II) The foregoing description has been made with respect to the embodiments of case (I) of the invention in which the flow regulating portions **54, 54'** are provided in the meter-in circuit communicating the pump **70** with the inlet side of the fluid motor **61.** Now, embodiments of the invention in which the flow regulating portions **54, 54'** are provided in the meter-out circuit communicating the outlet side of the fluid motor **61** with the tank **80,** will be described with reference to FIGS. **5, 6** and **7.**

A distinct difference between the apparatus shown in FIGS. **5, 6, 7** and those shown in FIGS. **1, 2, 3, 4** is that, while in the latter it is impossible to obtain a special spool function such as a tank port block and the back pressure valve **72** is required, in the former it is possible to obtain a special spool function and the back pressure valve **72** is not required so that the apparatus can be provided in a compact form as a whole.

1. First of all, how the special spool function is obtained will be described with reference to FIG. 5. The embodiment of FIG. 5 differs from that of FIG. 1 in that the valve spool 55 is so constructed as to form the flow regulating portion 54 between the right end face of the land 56 and the annular groove 43 when said spool is displaced leftward to the position shown in FIG. 6 and to form the flow regulating portion 54' between the left end face of the land 58 and the annular groove 47 when said valve spool is displaced conversely rightward. Thus, the flow regulating portions 54, 54' are located in the meter-out circuit communicating the outlet side of the fluid motor 61 with the tank 80 regardless of which direction the valve spool is displaced. Further, as the consequence of changing the locations of the flow regulating portions 54, 54', the pilot port 39 is open between the annular grooves 43 and 44, and the pilot port 39' between the annular grooves 46 and 47, and said pilot ports 39, 39' are communicated with the reference pressure chamber 9 through passages 77, 69 and an opening 68 respectively.

With the flow regulating portions 54, 54' provided in the meter-out circuit as described above, a special spool function can be obtained in which the inlet passage connected to the port P can be communicated with the ports *a* and/or *b* when the valve spool 55 is in its neutral position.

If, in the apparatus of FIG. 1 having the flow regulating portion 54, 54' provided in the meter-in circuit, an arrangement is made to establish communication between the inlet passage 50 and the ports *a* and/or *b* in the neutral position of the valve spool, for obtaining a tank port block type spool function, and when the valve spool 55 is displaced to the left thereafter so as to establish communication between the inlet passage 50 and the port *a* and between the port *b* and the port T, control of fluid in proportion to the displacement of the valve spool from said position would only be possible and a flow rate of fluid below a predetermined flow rate corresponding to the initial opening cannot be obtained, as the port *a* has already opened when the valve spool was in neutral, passing a predetermined quantity of fluid therethrough. In the event that the flow rate is low, therefore, it would be impossible to operate the actuator 61 in proportion to the displacement of the valve spool 55, and thus the apparatus cannot be used for other than special applications. Thus, with the flow regulating portions 54, 54' provided in the meter-in circuit as in the apparatus shown in FIG. 1, general proportional control cannot be achieved when the inlet passage 50 communicating with the port P is communicated with the port *a* and/or *b* in the neutral position of the valve spool 55.

As contrasted, with the flow regulating portions 54, 54' provided in the meter-out circuit as in the apparatus of FIG. 5, the inlet passage 50 communicating with the port P may be in communication with the port *a* or *b* in the neutral position of the valve spool 55.

Thus, the flow rate of fluid to the fluid motor 61 can be controlled in proportion to the displacement of the valve spool 55, even if communication between the inlet passage 50 and the port *a* or *b* is previously established in the neutral position of the valve spool 55, since the flow regulating portions 54, 54' are provided in the meter-out circuit.

In the apparatus shown, for instance, in FIG. 5, similar to that shown in FIG. 1, the spool function of the valve spool in the neutral position is of the all port block type wherein all ports are blocked. However, the apparatus shown in FIG. 7 is of the so-called tank port block type wherein the inlet passage 50 is communicated with the ports *a* and *b* and the port T is blocked when the valve spool is in its neutral position.

In the apparatus of such spool function type, the inlet and outlet sides of the fluid motor 61 are communicated with each other through the annular groove 45 in the neutral position of the valve spool, so that said fluid motor 61 can be driven easily by an external force such as a external mechanical force.

Reference is made, as other practical example of establishing communication between the inlet passage 50 in communication with the port P and the ports *a* and *b*, to the case of achieving an inching operation of the fluid motor 61 by providing notches on both sides of the land 57 in FIG. 6.

2. Now, the manner in which the entire appartus can be provided in a compact form due to the absence of the back pressure valve 72 will be described.

The omission of the back pressure valve 72 in the apparatus of FIG. 1 renders the apparatus completely unserviceable during self-running operation of the fluid motor 61. Thus, in the self-running state of the fluid motor 61, the return line communicating the outlet side of said fluid motor 61 with the tank 80 and the pump line communicating the outlet passage 22 with the inlet side of said fluid motor 61 are communicated directly with each other, and the fluid motor is acting not as motor but as pump. Therefore, the fluid pressure in the pump line is zero and the pressure differential between both sides of each of the flow regulating portions 54, 54' is also zero. The fluid pressure acting on the opposite ends of the pressure compensating plunger 30 is also zero, so that said plunger 30 is returned to its leftward position under the biasing force of the spring 35. In other words, the apparatus loses its fluid controlling function completely during the self-running operation of the fluid motor 61.

If the back pressure valve 72 is eliminated from the apparatus having the flow regulating portions 54, 54' provided in the meter-in circuit, like that shown in FIG. 1, the apparatus will not perform its intended function in the self-running state of the fluid motor 61 as stated above, and hence, will be usable only with specific working machines.

In order for such apparatus to be usable with any and all working machines, therefore, it is necessary to provide the back pressure valve 72 in cooperation with the short circuit as shown in FIG. 1, to forcibly pressurize the pressure line and thereby to develop a pressure differential between both sides of each of the flow regulating portions 54, 54', necessary to operate the plunger 30.

However, the back pressure valve 72 is unnecessary where the flow regulating portions 54, 54' are provided in the meter-out circuit as in the apparatus shown in FIGS. 5, 6 and 7.

In this case, since the flow regulating portions 54, 54' are provided between the fluid motor 61 and the tank 80, the fluid pressurized by the fluid motor 61 passes at the flow regulating portions 54, 54' even when said fluid motor is self-running and acting as pump, and a pressure differential occurs without fail between both sides of each of said flow regulating portions 54, 54' in such state of the fluid motor **61**. This is why the back pressure valve **72** is not required.

The elimination of the back pressure valve **72** apparently enables the apparatus of FIG. **5** to be provided in a much more compact from than the apparatus shown in FIG. **1**.

This is because, in the apparatus shown in FIG. **1**, the back pressure valve **72** needs to be essentially the same size as the directional control valve unit **40** and pressure-compensating unit **10**, since the fluid discharged from the pump **70** passes therethrough in its entirety.

The apparatus shown in FIG. **5**, because of the absence of the back pressure valve **72**, can be provided, as a whole, in a size two thirds of that of FIG. **1**, or in a very compact form.

3. Although the most distinct differences between the apparatus of FIG. **5** and that of FIG. **1** have been described above, both apparatus are different also in respect of the bent unload type in neutral.

More specifically, in the apparatus of FIG. **5** the inlet passage **50** and the reference pressure chamber **9** are communicated with each other through the annular groove **45**, a passage **53**, the annular groove **49** and passages **77, 69**. With such arrangement, the pump is unloaded in the neutral state of apparatus, as in the apparatus of FIG. **1**.

4. The other portions are identical with those of the apparatus shown in FIG. **1** and will not be described herein.

Hereinbefore, the embodiments of the invention have been described with reference to the drawings upon categorizing them, for the sake of convenience in description, into the case (I) wherein the flow regulating portions **54, 54′** are provided in the meter-in circuit and the case (II) wherein the same are provided in the meter-out circuit. In short, the apparatus of the invention have the following meritorious features:

A. With the apparatus of the invention, the operating speed of a working machine is in proportional relation with the displacement of the valve spool in said apparatus, no matter how the load exerted on said working machine changes, because the pressure of fluid in the directional control valve unit **40** is compensated such that the pressure differential between both sides of each of the flow regulating portions **54, 54′** be maintained constant, by the pressure-compensating unit **10** having the divided flow controlling portion (α), the first pressure controlling portion (β) and the second pressure controlling portion (δ). Further, in the apparatus of the invention the occurrence of an abnormal pressure in the hydraulic circuit, in the self-running state of the fluid motor **61**, can be completely avoided especially because the second pressure controlling portion (δ) is provided and the return line and pressure line are in communication with each other when the first pressure controlling portion (β) is closed off.

B. The discharge of the pump can be effectively utilized because, the second actuator **75** when provided in the bypass line **74** can be operated by a flow in excess of that necessary for operating the first actuator **61**. Furthermore, the flow rate necessary for operating the first actuator **61** may always be maintained at a desired value for reason that the divided flow rate through the divided flow regulating portion (α) may be reduced to an extent by narrowing the open area thereof when the divided flow rate led through the divided flow regulating portion (α) to the second actuator **75** tends to increase owing to that the load exerted on the second actuator is smaller than that on the first actuator, and that the flow rate led into the first actuator **61** through the first pressure controlling portion (β) may be reduced to a extent by narrowing the open area thereof when the flow rate led into the first actuator through the first pressure controlling portion tends to increase owing to that the load exerted on the first actuator is maller. Namely, the fluid may be led preferentially to the first actuator **61** so as to enable the operating speed of the first actuator **61** to be maintained at a desired constant value predetermind by the displacement of the spool **55**. It is also to be noted that the driving force necessary for driving the pump **70** can be reduced because the pump pressure is maintained at a level equal to or slightly higher than the largest one of the loads exerted on the actuators connected to the circuit and is not maintained constant by a relief pressure as is in the conventional apparatus.

What is claimed is:

1. A fluid controlling apparatus for controlling the operation of a hydraulic actuator, comprising

a. a directional control valve unit (**40**) having a spool slide cavity (**42**), and an inlet passage (**50**), actuator ports (*a,b*) and a return passage (**51**) respectively arranged in crossing relation to said spool slide cavity (**42**), and including a valve spool (**55**) disposed in said spool slide cavity (**42**) for sliding movement therein to provide variable dimension flow regulating portions and to establish or sever communication between said inlet passage (**50**) and the inlet port (*a* or *b*) of the actuator or between the outlet port (*b* or *a*) of the actuator and said return passage (**51**);

b. a pressure-compensating unit (**10**) having a plunger slide cavity (**13**), and an inlet passage (**21**), a bypass passage (**20**), an outlet passage (**25**), an outlet passage (**22**) and a tank passage (**24**) respectively communicating directly with the inlet passage (**50**) and the return passage (**51**) of said directional control valve unit (**40**), all of which are arranged in crossing relation to said plunger slide cavity (**13**), and including a pressure-compensating plunger (**30**) disposed in said plunger slide cavity (**13**) to define a reference pressure chamber (**9**) and a back pressure chamber (**36**) on both sides thereof respectively and being slidable in said plunger slide cavity (**13**) to control the flow of fluid between said inlet passage (**21**) and said bypass passage (**20**), between said inlet passage (**21**) and said outlet passage (**22**) or between said tank passage (**24**) and said outlet passage (**25**), and further having surfaces respectively for receiving the fluid pressure in said reference pressure chamber (**9**) and the fluid pressure in said back pressure chamber (**36**) applied thereto in squarely opposite directions against each other, and a spring (**35**) disposed in said back pressure chamber (**36**) to urge said plunger (**30**) to move up to a position to sever communication between said inlet passage (**21**) and said bypass passage (**20**); and

c. passage means for communicating said outlet passage (**25**) directly with said motor port (a) or (b) while said pressure-compensating plunger (**30**) is in the position to sever communication between said inlet passage (**21**) and said outlet passage (**22**).

2. A fluid controlling apparatus for controlling the operation of a hydraulic actuator, according to claim **1**, wherein said directional control valve unit (**40**) is adapted to form therein the variable flow regulating portion between the inlet passage (**50**) and the inlet port (a or b) of the actuator, and there are provided a passage (**29**) for establishing communication between the inlet passage (**50**) and the reference pressure chamber (**9**) and a passage (**37**) for establishing communication between the motor port (a) or (b) and the back pressure chamber (**36**).

3. A fluid controlling apparatus according to claim **2**, wherein a back pressure valve (**72**) is provided between the outlet passage (**25**) of the pressure-compensating unit (**10**) and a fluid storage tank (**80**).

4. A fluid controlling apparatus according to claim **2**, wherein pilot ports (**39, 39'**) are provided between the inlet passage (**50**) of the directional control valve unit (**40**) and the actuator ports (*a, b*) respectively and are communicated with the passage (**37**) communicating with the back pressure chamber (**36**).

5. A fluid controlling apparatus according to claim **1**, wherein said directional control valve unit (**40**) is adapted to form therein the flow regulating portion between the return passage (**51**) connected with a fluid storage tank (**80**) and the outlet port (*b* or *a*) of the actuator, and there are provided a passage (**34**) for communicating the return passage (**51**) with the back pressure chamber (**36**) and a passage (**69**) for communicating the actuator port (*a*) or (*b*) on the outlet side with reference pressure chamber (**9**).

6. A fluid controlling apparatus according to claim **5**, wherein the inlet passage (**50**) communicating with a fluid pressure source is communicated with the actuator port (*a*) and/or (*b*) while the spool valve (**55**) is in its neutral position.

7. A fluid controlling apparatus according to claim **5**, wherein pilot valves (**39, 39'**) are provided between a passage of the directional control valve unit (**40**) connected with the fluid storage tank (**80**) and the outlet ports (*a*) and (*b*) of the actuator respectively and are adapted to be communicated with the passage (**69**) communicating with the reference pressure chamber (**9**).

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