United States Patent [19]
Olsen

[11] **3,811,795**
[45] **May 21, 1974**

[54] **HIGH PRESSURE FLUID INTENSIFIER AND METHOD**

[75] Inventor: **John H. Olsen**, Vashon, Wash.

[73] Assignee: **Flow Research, Inc.**, Kent, Wash.

[22] Filed: **Jan. 12, 1973**

[21] Appl. No.: **322,956**

[52] **U.S. Cl.** **417/53**, 91/306, 91/313, 417/397
[51] **Int. Cl.** **F04b 35/00**, F01l 25/04
[58] **Field of Search** 91/305, 306, 307, 313; 417/397, 403, 404, 390, 53

[56] **References Cited**

UNITED STATES PATENTS

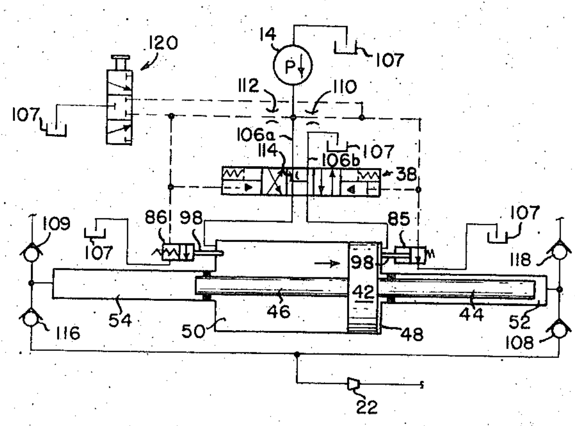
3,465,686	9/1969	Nugier	417/403
2,942,584	6/1960	Rethmeier	91/464
3,540,348	11/1970	Penther	91/306

Primary Examiner—Carlton R. Croyle
Assistant Examiner—Richard Sher
Attorney, Agent, or Firm—Graybeal, Barnard, Uhlir & Hughes

[57] **ABSTRACT**

A pressure intensifying apparatus to deliver a very high pressure stream of water through a nozzle. There is a single working piston having two pressure surfaces of a relatively large area, the working piston being connected to two high pressure pistons each having a pressure surface of a relatively small area. A control valve delivers a high pressure working fluid alternately to opposite sides of the working piston to cause it to reciprocate so that the pressure pistons alternately deliver water at high pressure to the nozzle. In shifting between its two end positions, the control valve passes through an intermediate position at which a restricted flow passage is provided for the working fluid, this restricted flow passage having an effective cross sectional area relative to the effective area of the discharge nozzle such that the back pressure of the restricted passage of the control valve matches the back pressure exerted on the working fluid so that a substantially constant back pressure is imposed on the high pressure source of working fluid. Further, there is a valve shifting mechanism comprising two shifting valves, each of which is responsive not only to physical contact by the working piston, but also to pressurization of its related working chamber to cause rapid shifting of the control valve.

16 Claims, 15 Drawing Figures



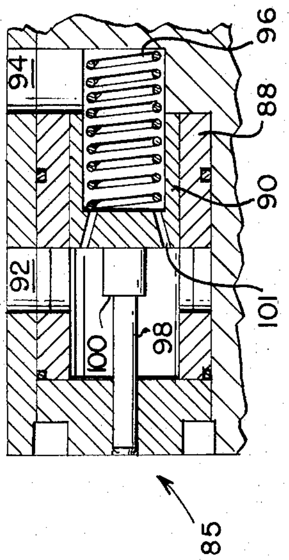


FIG. 3

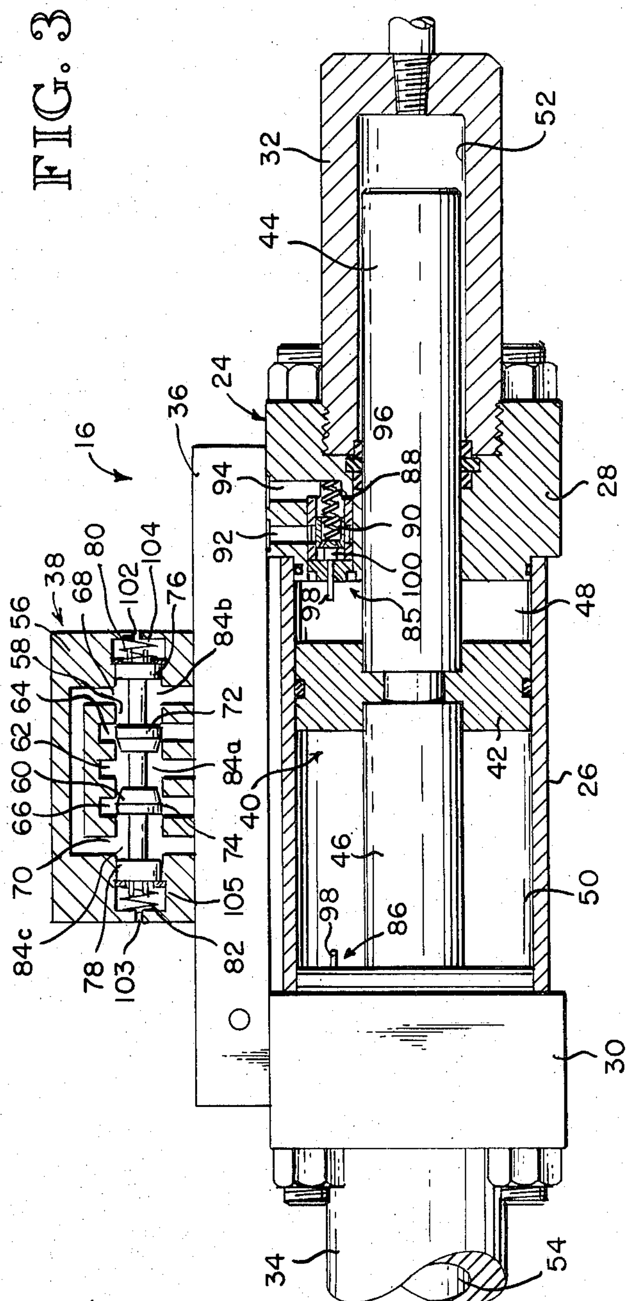


FIG. 2

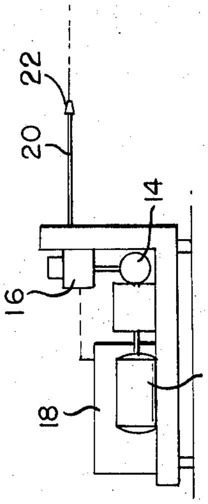


FIG. 1

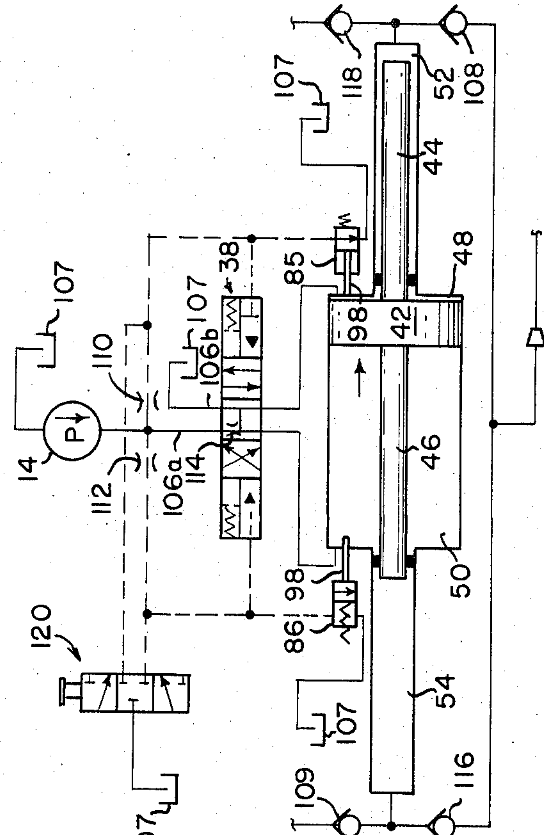


FIG. 5

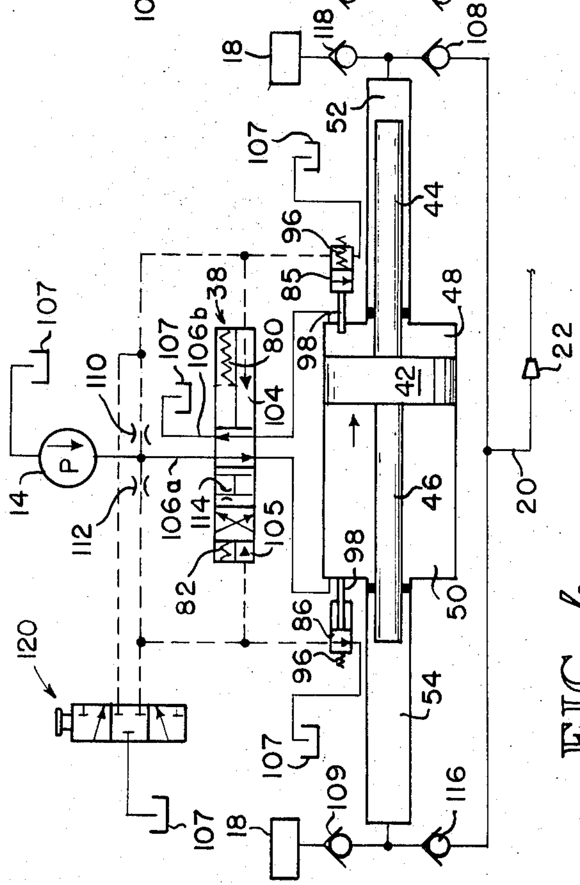


FIG. 4

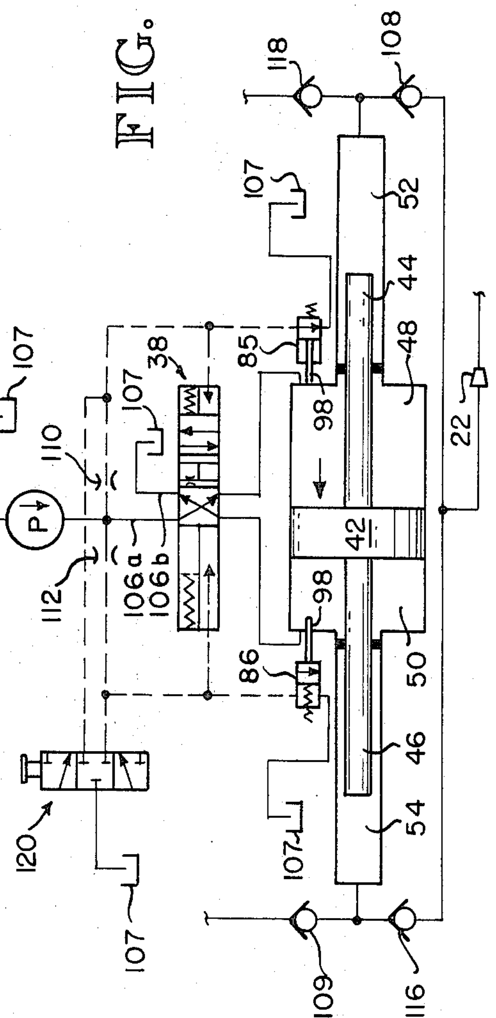


FIG. 6

FIG. 7A

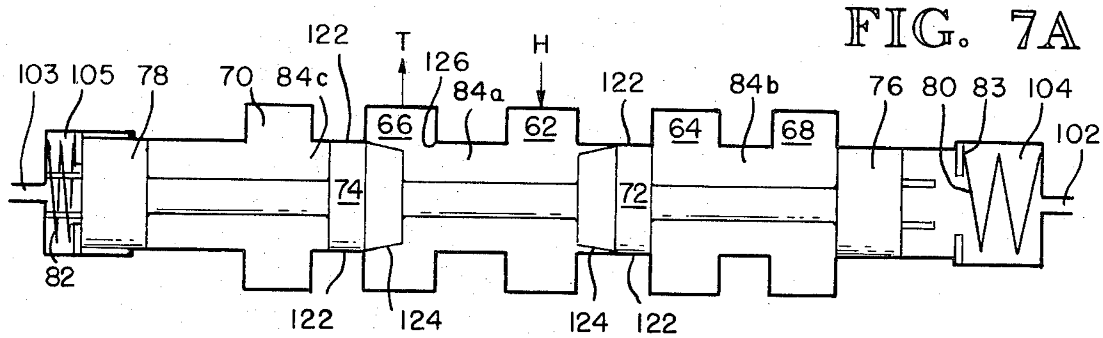


FIG. 7B

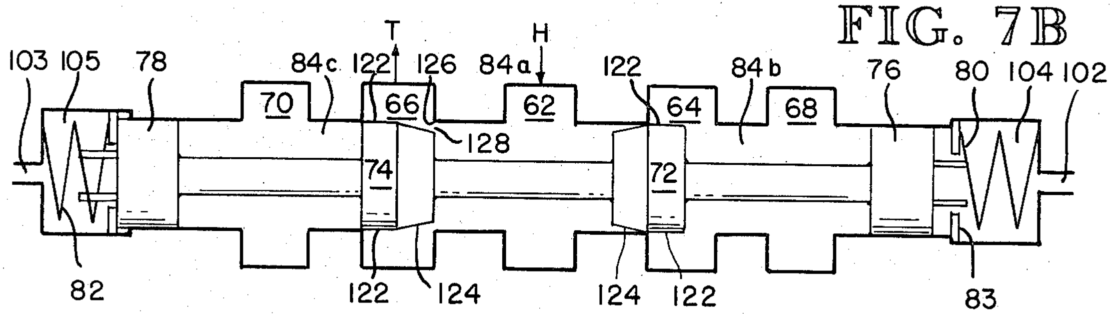


FIG. 7C

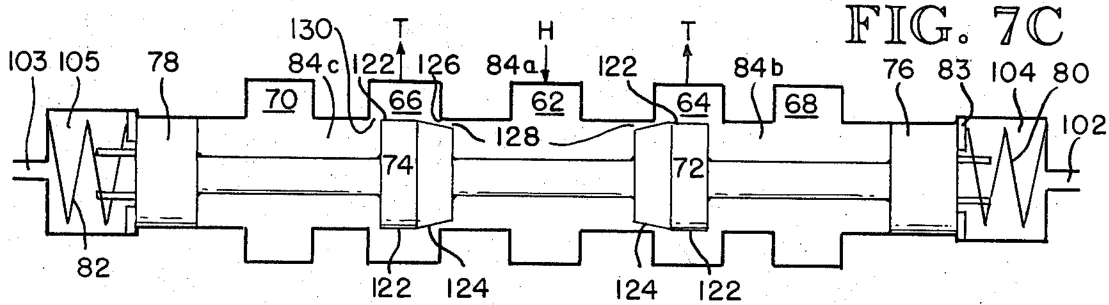


FIG. 7D

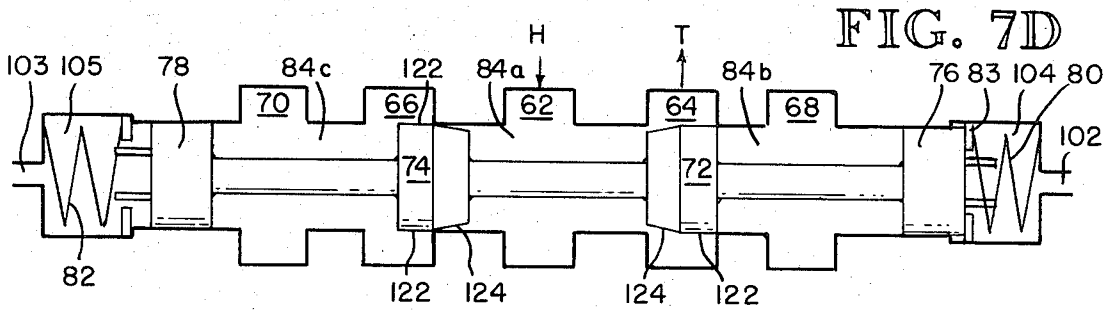


FIG. 7E

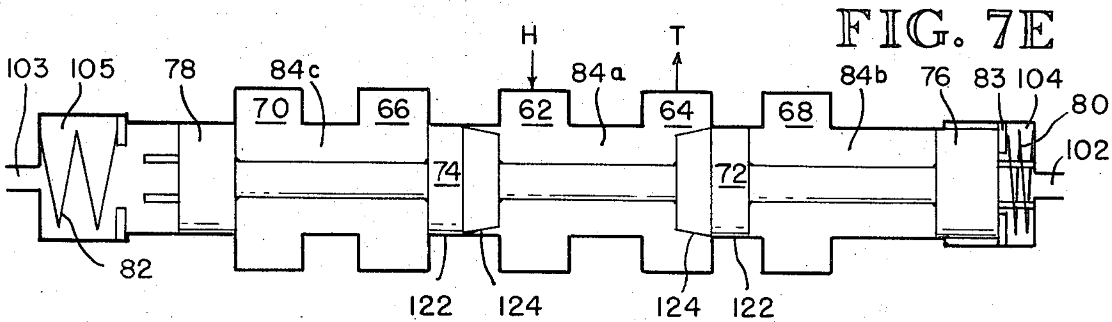


FIG. 8

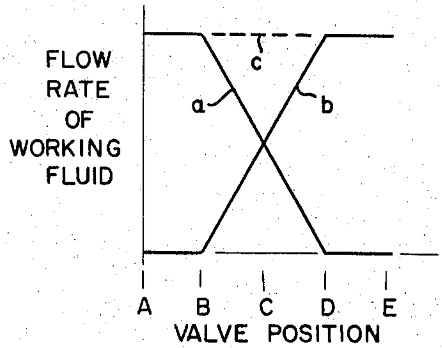


FIG. 9

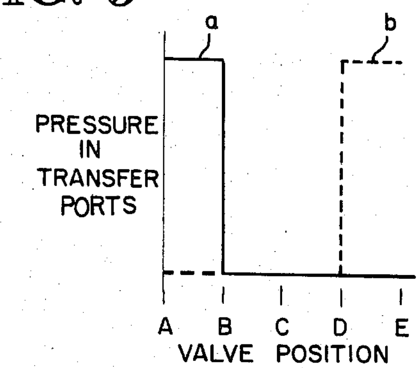
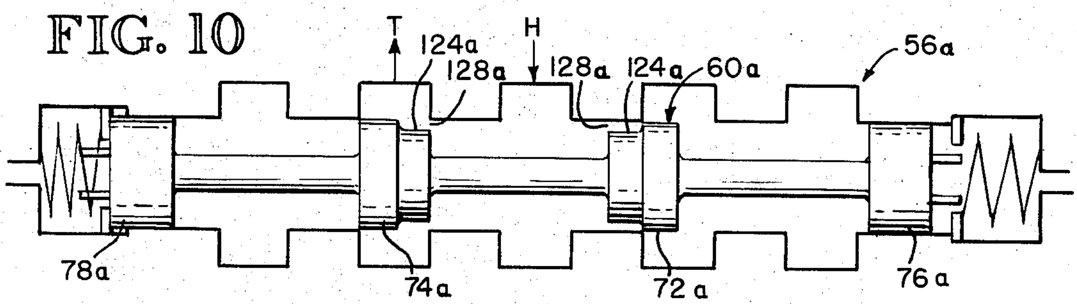


FIG. 10



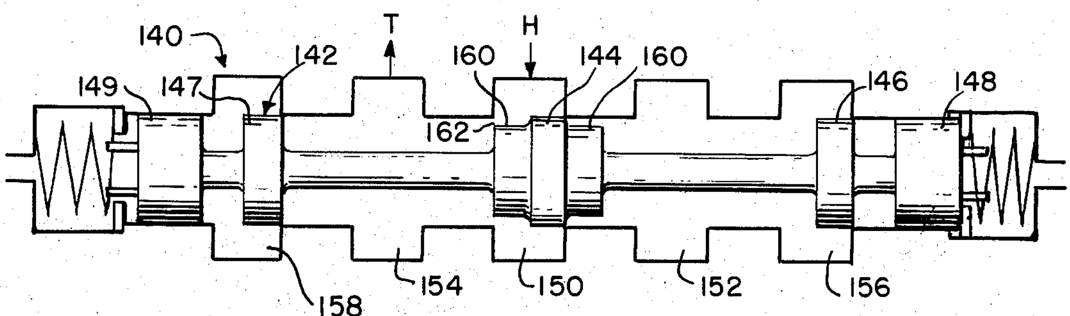


FIG. 11

HIGH PRESSURE FLUID INTENSIFIER AND METHOD

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to a high pressure fluid intensifier, such an an intensifier arranged to deliver a stream of fluid at a very high pressure, to accomplish a function such as cutting, drilling or waterblast cleaning.

2. Description of the Prior Art

There are in the prior art various pressure intensifying systems where a larger working piston is reciprocated to provide a high pressure output through smaller high pressure pistons. One of the problems associated with many such prior art devices is that of obtaining a rapid reversal of the working piston so that there is no significant interruption of the flow of high pressure output fluid. Yet another problem in the prior art is that of alleviating pressure surges in the working fluid while the working piston is being reversed. Aside from these operational problems, there are, especially when very high pressures are used, safety considerations with respect to one of the hydraulic lines or other components breaking or rupturing.

Typical of the prior art devices which show pressure intensifying systems and various valve switching mechanisms adaptable for such systems are the following U.S. Pats.: Atkinson, No. 153,296; West et al., No. 2,000,805; Rethmeier, No. 2,942,584; Murray, No. 3,045,611; Pennther, No. 3,540,349; and Bowen, No. 3,565,191.

It is an object of the present invention to provide a high pressure fluid intensifier having desirable operating features, particularly with respect to the problems and considerations mentioned above.

SUMMARY OF THE INVENTION

In the apparatus of the present invention, there is a working cylinder in which a working piston is mounted for reciprocating motion, the piston dividing the cylinder into first and second working chambers. Two high pressure pistons are connected to the working piston in a manner that reciprocation of the working piston causes a flow of high pressure fluid to be procuced alternately from the two high pressure pistons. The high pressure flow is directed through a discharge nozzle to produce a high velocity stream of water. To cause reciprocation of the working piston, there is a control valve having first and second positions to deliver pressurized working fluid to, respectively, the first and second working chambers.

According to one facet of the present invention, the control valve has a third intermediate position through which it passes in moving between its first and second positions. In the intermediate position pressurized working fluid is directed through a pressure reducing flow passage, which produces a back pressure substantially balancing the back pressure resulting from transmitting power through the working piston and the pressure intensifying pistons to produce a high pressure fluid flow through the output nozzle. Thus, during reversal of the working piston when the control valve passes through its intermediate position, any substantial surge of back pressure against the working fluid source is alleviated.

In accordance with another facet of the present invention, there are two shifting valves to cause the control valve to move between its first and second positions. Each shifting valve is made responsive not only to movement of the piston to an end limit of travel proximate the shifting valve, but is responsive also to pressurized working fluid in its respective chamber being at a predetermined level. Actuation of one or the other of the shifting valves causes a pressure imbalance at the control valve to cause rapid shifting of the control valve. A centering spring urges the control valve toward its intermediate position. If there is a pressure reduction in the working chamber that is pressurized at any particular moment, for example by a rupture in one of the high pressure output lines, the decrease in pressure causes deactivation of its related shifting valve to move the control valve to its intermediate position and stop further operation of the apparatus.

Other more specific features of the present invention will become apparent from the following detailed description.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a semi-schematic side elevational view of the overall apparatus of the present invention;

FIG. **2** is a view partly in sections of the pressure intensifying apparatus of the present invention;

FIG. **3** is a sectional view of one of the shifting valves of the apparatus of FIG. **2**;

FIGS. **4–6** are semi-schematic drawings illustrating the operating sequence of the present invention;

FIGS. 7A–7E are a series of semi-schematic drawings showing the sequence of operation of the control valve of the present invention;

FIG. **8** is a graph illustrating the flow characteristics in the control valve in the sequence of operation of FIGS. 7A–7E;

FIG. **9** is a graph illustrating the pressure characteristics of the control valve in the sequence of operation of FIGS. 7A–7E;

FIG. **10** is a semi-schematic illustration of a second embodiment of the control valve of the present invention;

FIG. **11** is a semi-schematic drawing of a third embodiment of the control valve of the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

With reference to FIG. **1**, there is shown an electric motor **10** which drives a hydraulic pump **14**, which in turn supplies working fluid to a pressure intensifier unit **16**. The intensifier **16** draws fluid (i.e., water) from a suitable source, such as a reservoir **18**, and discharges the water at a very high pressure through an output, which as shown herein is a tube **20** with a small area exit nozzle **22**. This results in a discharge of a fluid jet stream of a small diameter (e.g., 0.03 inches) and a very high velocity (e.g., 1,200 feet per second or greater).

In describing the present invention in detail, first the physical components of the pressure intensifying unit **16** will be described with reference to FIGS. **2** and **3**. Second, the overall operation of the total apparatus will be described with reference to the sequential schematic drawings of FIGS. **4** through **6**. Thereafter, the precise manner in which the control valve **38** functions to accomplish the proper operation of this apparatus will be described in more detail with reference to the sequential illustrations of FIGS. 7A through 7E, and the two graphs of FIGS. **8** and **9**, with two modified valves being shown in FIGS. **10** and **11.**

The physical components of the pressure intensifying unit **16** are illustrated in FIG. **2.** For clarity of illustration, the various fluid lines and passages built into or attached to the unit **16** are not illustrated in FIG. **2,** but rather are shown schematically in FIGS. **4** through **6.** With reference to FIG. **2,** the pressure intensifying unit **16** comprises a main housing **24,** comprising a main cylinder **26,** right and left end bell members **28** and **30,** respectively, mounted to the ends of the cylinder **26,** and right and left high pressure cylinders **32** and **34,** respectively, threaded into respective bells **28** and **30.** Connected to the housing **24** is a manifold block **36** on which is mounted a flow control valve **38.**

Mounted for reciprocating motion within the housing **24** is a unitary piston assembly **40.** This assembly comprises a larger diameter central working piston **42** mounted within the main cylinder **26,** and right and left high pressure pistons **44** and **46,** respectively, extending oppositely from the center working piston **42.** The working piston **42** divides the interior of the main cylinder **26** into right and left working chambers **48** and **50** respectively. The high pressure piston **44** reciprocates in the right high pressure chamber **52** defined by the right cylinder **32,** while the left high pressure piston **46** reciprocates in the left high pressure chamber **54** defined by the other cylinder **34.**

The aforementioned control valve **38** comprises a valve housing **56** defining a transfer chamber **58,** in which is slide mounted a valve spool **60.** In the housing **56** is a centrally located high pressure fluid inlet port **62,** right and left transfer ports **64** and **66,** respectively, on opposite sides of the inlet port **62,** and right and left low pressure outlet ports **68** and **70,** respectively, positioned outside of the two transfer ports **64** and **66.**

The valve spool **60** comprises right and left lands or pistons **72** and **74,** respectively, and right and left outermost end closure members **76** and **78,** respectively. Outside the two closure members **76** and **78** are respective right and left centering springs **80** and **82,** respectively, which urge the spool **60** to its center position in the housing **56;** each of the springs **80** and **82** has a stop collar **83** engaging a stop shoulder **83***a* to prevent either spring **80** or **82** urging the valve spool **60** beyond its center position.

The center port **62** is connected to a high pressure line from the pump **14,** while the ports **68** and **70** are connected to the low pressure return line of the pump **14.** The right transfer port **64** connects to the right working chamber **48,** and the left transfer port **66** connects to the left working chamber **50.** The groove or chamber **84***a* located between the two piston elements **72** and **74** is a high pressure fluid transfer chamber, and functions to direct high pressure fluid from the port **62** to either the right transfer port **64** or the left transfer port **66** when in, respectively, a right or left hand position. The right piston **72** and the right closure piston **76** define a groove or chamber **84***b* which is a low pressure transfer chamber that functions to connect the transfer port **64** with the low pressure outlet port **68** when the spool **60** is in its left hand position. In like manner, the left transfer piston **74** and left closure piston **78** define therebetween a groove or chamber **84***c* which functions to connect the left transfer port **66** with the low pressure outlet port **70** when the spool **60** is in its right hand position.

To move the spool **60** of the valve **38** between its right and left positions, there are two shifting valves **85** and **86,** respectively, located in, respectively, the right and left bell sections **28** and **30** of the housing **24.** For convenience of illustration only the right shifting valve **85** is shown in section in FIG. **2** (the valve **85** and **86** being substantially identical). Each of the valves **85** and **86** comprises a sleeve **88** in which is slidably mounted a plug **90.** There is an inlet port **92** and an outlet or venting port **94,** the port **92** being closed from the venting port **94** when the valve is in the closed position shown in FIG. **2.** The plug **90** is urged to its closed position by a compression spring **96.** To open the valve **85** or **86** there is provided an actuating pin **98** which butts against the plug **90** and extends through the housing to project into the end of its respective working chamber **48** or **50.** A locating stop **100** on the pin **98** properly positions the plug **90** in its closing position with the pin **98** extending into the chamber **48** or **50.** Through holes **101** in the plug **90** permit flow from the inlet port **92** to the venting port **94** when the plug **90** is pushed by pin **98** against the urging of the spring **96** to its open position shown in FIG. **3.** The inlet port **92** of the right shifting valve **85** is connected through an end opening **102** in the housing **56** of the valve **38** to a right control chamber **104** at the right end of the spool **60** of the valve **38,** while the inlet port **92** of the left shifting valve **86** is similarly connected to the left control chamber **105** through opening **103** in the left of the valve **38.**

In describing the operations of this apparatus, for convenience the overall operation will first be described and then the means for starting the pumping action. In FIG. **4,** the valve **38** is in its left hand position, so that high pressure fluid from the high pressure line **106***a* of the pump **14** is directed into the left working chamber **50,** while the right working chamber **48** is connected through a low pressure return line **106***b* to the fluid reservoir **107** of the pump **14.** This causes the working piston **42** to move to the right, as seen in FIG. **4,** which in turn causes the right high pressure piston **44** to force output fluid from the high pressure chamber **52** through a check valve **108** and out the discharge nozzle **22.** At the same time, output fluid is being drawn from the source **18** through a check valve **109** into the left high pressure chamber **54.**

The pumping pressure of the pump **14** is sufficiently large, relative to the force of the return springs **96** and the cross sectional area of the actuating pins **98** of the shifting valves **85** and **86,** that when one of the working chambers **48** or **50** is pressurized by the pump **14,** the resulting pressure on its related actuating pin **98** is sufficient to force the pin **98** against the urging of its related spring **96** to open the valve **85** or **86.** A portion of the high pressure working fluid is directed from the pump **14** to the valve control chambers **104** and **105** through respective restricted flow orifices **110** and **112,** and also to the high pressure inlet ports **92** of the shifting valves **85** and **86.** The venting port **94** of each shifting valve **85** or **86** is connected to the pump reservoir **107.** Thus, when one of the valves **85** or **86** is closed, its related valve control chamber **104** or **105,** respectively, is pressurized, but when one of the valves **85** or **86** is open, its respective valve control chamber **104** or **105** is depressurized.

With reference to FIG. 4, since the working chamber 50 is pressurized from the pump 14, the shifting valve 86 is open so that the left valve control chamber 105 is depressurized. Since the working chamber 48 is connected to the low pressure reservoir 107, the actuating pin 98 protrudes into the chamber 48, so that the shifting valve 85 is closed, with the right valve control chamber 104 being pressurized. The difference in pressure in the two valve control chambers 104 and 105 holds the spool element 60 of the valve 38 in its left hand position against the urging of the spring 82 as seen in FIG. 4.

As the working piston 42 continues to move to the right, it approaches its end limit of travel as shown in FIG. 5. Near its end limit of travel, the piston 42 engages the actuating pin 98 to push the pin 98 and the plug 90 of the shifting valve 85 to its open position to depressurize the right valve control chamber 104. With both the control chambers 104 and 105 depressurized, the left centering spring 82 pushes the spool element 60 toward its center position, as shown in FIG. 5.

When the spool element 60 reaches its center position, two things occur. First, both of the working chambers 48 and 50 become connected to the low pressure pump reservoir 107 through the low pressure return line 106*b* so as to be depressurized. Second, the high pressure supply line 106*a* from the pump 14 becomes connected through a restricted flow passage, indicated schematically at 114, to permit limited flow from the pump 14 back to the pump reservoir 107. The particular manner in which this is accomplished and how this alleviates pressure surges in the high pressure supply line will be described later herein in a more detailed description of the functioning of the valve 38.

The immediate effect of depressurizing both of the working chambers 48 and 50 is to permit the spring 96 to move the left shifting valve 86 outwardly toward the left working chamber 50 to close the left shifting valve 86 and thus cause immediate pressurization of the left valve control chamber 105. This immediately causes the spool element 60 to continue movement through its center or intermediate position to its right position, shown in FIG. 6, where high pressure working fluid is delivered to the working chamber 48, with the left working chamber 50 being connected to the low pressure line 106*b* leading to the pump reservoir 107.

With the right working chamber 48 now pressurized, the working piston 42 moves to the left so that the left high pressure piston 46 forces output fluid from the left high pressure chamber 54 through a check valve 116 and out the output nozzle 22. At the same time, additional output fluid is being drawn into the right high pressure chamber 52 through a check valve 118. With the right working chamber 48 pressurized, as the working piston 42 moves away from physical engagement with the right actuating pin 98, the pressure of the working fluid in the chamber 48 keeps the right actuating pin 98 in its retracted position to maintain the right shifting valve 85 in its open position and maintain the depressurization of the right valve control chamber 104 so that the spool element 60 remains in its right position, as shown in FIG. 6 because of the pressurization of the left valve control chamber 105 due to the left shifting valve 86 being closed.

When the working piston 42 moves toward its left end position to engage the left actuating pin 98, a similar shifting sequence occurs, as described with reference to FIG. 5, to reverse the fluid flow in the working chambers 48 and 50 and cause the working piston 42 to begin movement back to the right.

To initiate operation of the apparatus, there is provided a starting valve, this being shown schematically at 120. The valve 120 has an up position where the right valve control chamber 104 is directly connected to the reservoir 107, a down position where the left valve control chamber 105 is connected to the reservoir 107, and an intermediate position where the valve 120 provides no operative connection to the chambers 104 and 105. In the three illustrations of FIGS. 4 through 6, the starting valve 120 is shown in its center position where it has no effect on the operation of the apparatus. To describe the operation of the starting valve 120, let it be assumed that the pump 14 has been turned off and the entire system has become depressurized, with the spool element 60 of the valve 38 returning to its center position by action of the centering springs 80 and 82. Further, let it be assumed that the working piston 42 is in some intermediate position, as shown in FIG. 4.

When the pump 14 is started, with the valve spool element 60 in its center position, neither of the working chambers 48 or 50 becomes pressurized. However, both of the valve control chambers 104 and 105 are pressurized, since both of the shifting valves 85 and 86 remain closed. These circumstances contribute to the safety of the operation in that the intensifying unit 16 will not inadvertently start pumping when the hydraulic pump is started, provided that piston 42 is not at that moment holding either valve 85 or 86 open. It should be noted that when the motor 10 is turned off to stop the pumping action, there is a gradual decline of hydraulic pressure due to the inertia of the motor and pump. If the motor is turned off and the hydraulic pressure has declined below the level required to hold pin 98 in against spring 96 while the working piston 42 is in contact with one of the pins 98 of one or the other of the shifting valves 85 or 86, this will cause the control valve 38 to remain either its right or left position to pressurize the working chamber 48 or 50 at which the piston 42 is depressing the pin 98. This in turn causes the piston 42 to move out of engagement with that pin 98 allowing the valve 85 or 86 to close and cause the control valve 38 to return to its center position where neither of the working chambers 48 and 50 is pressurized. Thus, piston 42 will stop in a position remote from pins 98, and when the motor 10 is restarted, the pumping action will not start. By pushing the valve 120 to its down position, the left control chamber 105 becomes exposed to the low pressure pump reservoir 107 so that the high pressure in the right valve control chamber 104 causes the valve element 60 to move to the left (i.e., the position shown in FIG. 4.) to cause the piston 42 to move to the right. As soon as the working piston 42 reaches its full right position to engage the right actuating pin 98, the normal shifting sequence goes into effect as described above herein. By moving the starting valve 120 to its up position to cause the spool element 60 to move to the right, the working piston 42 can be caused to move to the left. So if there is some reason that the normal shifting sequence of the apparatus does not function as described above, for example by reason of excess air in the high pressure lines, then the manual starting valve 120 can be used to move the working piston 42 back and forth to clear the hydraulic lines so that the normal shifting sequence becomes operative, with the working piston **42** then reciprocating with automatic shifting of the valve **38** as described above.

In the event that there is a break in one of the high pressure output lines to the nozzle **22**, there will be an immediate drop in pressure in the output chamber **52** or **54** that is at the time pressurized, and a corresponding drop in pressure in the related working chamber **48** or **50**, which happens to be pressurized at that particular instant. When there is such a pressure drop in the pressurized working chamber **48** or **50**, its related actuating pin **98** is moved outwardly by spring **96** to its valve closing position to close its related shifting valve **85** or **86** to pressurize its related valve control chamber **104** and **105** (both chamber **104** and **105** then being pressurized) so that the valve element **60** returns to its center position by action of the centering springs **80** and **82** to vent both working chambers **48** and **50** to low pressure and halt movement of the working piston **42**. Thus, in the event of any break in the high pressure lines, the system immediately shuts itself off.

To describe in more detail the operation of the control valve **38**, reference is now made to FIGS. 7A through 7E.

In FIG. 7A, the spool element **60** is shown in its full left position (shown schematically in FIG. 4), where the right piston **72** is positioned between the high pressure port **62** and the right transfer port **64** so as to block any flow therebetween, while the left piston **74** is positioned so as to block any flow from the left transfer port **66** into the low pressure outlet port **70**. In this position, there is free flow from the high pressure port **62** to the left transfer port **66** through the center high pressure chamber **84***a* to pressurize the working chamber **50** as described above. Also, the right transfer port **64** communicates with the low pressure port **68** through the right low transfer chamber **84***b* so that the right working chamber **48** is depressurized. As described above, the spool element **60** remains in this position until working piston **42** reaches its extreme right end of travel to cause shifting of the valve spool element **60** to the right.

In FIG. 7B, the valve element **60** is shown moving from its extreme left hand position through a position where it is just about to enter its intermediate position. It will be noted that the laterally outermost portion **122** of the circumferential surfaces of each of the pistons **72** and **74** is substantially cylindrical so that it fits against the inner cylindrical surface of the housing **56**. However, the laterally inward circumferential surface portion **124** of each of the pistons **72** and **74** (i.e., those surface portions closer to the center of the spool element **60**) are tapered very moderately inwardly toward the middle of the spool element **60**. For purposes of illustration, this taper is shown to be at a somewhat larger angle than normally used, this taper ordinarily being in the order of one degree from the longitudinal axis of the spool element **60**. It can be seen that in the position shown in FIG. 7B the tapered surface **124** of the left piston **74** forms with the inner edge **126** of the left transfer port **66** a restricted circumferential flow passage **128**.

As the piston continues to move from the position of FIG. 7B toward the position of FIG. 7C, where the spool element **60** is centered, a passage opens at **130** from the left transfer port **66** through the left low pressure transfer chamber **84***c* to the low pressure outlet port **70**. Because of the very shallow taper of the surface **124**, as soon as the spool element **60** moves a very short distance from the position of FIG. 7B, the passageway **130** has a much larger cross sectional area than the passage of **128**, so that there is a large pressure drop from the inlet port **62** across the passageway **128**, and the pressure in the transfer port **66** almost immediately drops to the pump reservoir pressure that exists in the outlet port **70**. As the spool **60** continues to move to the right from position 7B to that of 7C, the left flow passage **128** becomes more restricted, while the right flow passage **128** defined by the piston **72** with the housing **56** becoming less restricted. Since the tapered surfaces **124** are both uniform, the rate of decrease of the cross section of the left restricted passage **128** is substantially the same as the rate of increase of the cross sectional area of the right restricted flow passage **128**, so that the total flow rate through both passages **128** is constant. This is illustrated in the graph of FIG. **8**, where the flow through the left restricted flow passage **128** is indicated at *a* and the right restricted flow passage **128** is indicated at *b*, with the combined flow through both passages **128** being shown by the dotted line at *c*.

In the graph of FIG. **9**, the pressure in the left transfer port **66** is indicated at *a*, while the pressure in the right transfer port **64** is indicated at *b*, in the travel of the spool element **60** from the position of FIG. 7A to that of FIG. 7E.

When the spool element **60** is traveling through its intermediate position (i.e., from the position of FIG. 7B, through the position of FIG. 7C to the position of FIG. 7D), both the right and left transfer ports **64** and **66** are at the low pump reservoir pressure since the flow passages **130** have a substantially larger cross sectional flow area than the passages **128** (in the order of perhaps one hundred times as great when the spool element **60** is in the position of FIG. 7C).

As described previously herein, as soon as the pressurized working chamber **48** or **50** becomes depressurized, there is an immediate pressure imbalance in the two valve control chambers **104** and **105**, which causes the spool element **60** to continue movement through its center postion to its other end position. When the spool element reaches the position shown in FIG. 7D, the spool element **60** is now moving from its intermediate position to its right position. At this point, the left restricted flow passage **128** is being completely closed off, while the right restricted flow passage **128** has reached its maximum effective cross sectional flow area. Simultaneously, the flow path from the right transfer port **64** into the right low pressure outlet port **68** is being abruptly closed off by the right piston **72** so that there is an abrupt rise in the transfer port **64** from pump reservoir pressure to high pressure. As the spool element **60** continues movement to the right to the extreme right position of FIG. 7E, there is substantially unrestricted flow from the high pressure port **62** to the transfer port **64** to cause pressurization of the right working chamber **48**.

It is important to note that the total cross sectional area of the two restricted flow passages **128** remains substantially constant as the spool element **60** is moving through its intermediate phase from the position of FIG. 7B to that of 7D. These two restricted flow passages **128** are, in effect, the same restricted flow passages indicated at 114 in the schematic drawings of FIGS. 4–6. The effective combined cross sectional area of the two flow passages 128 is so dimensioned that the back pressure exerted at the passages 128 is substantially the same as the pressure existing in either of the working chambers 48 and 50 when pressurized with the control valve 38 in either its right or left position. The effect of this is that when the valve element 60 is moving from its right position through its intermediate position to its left position or vice versa, the back pressure exerted on the high pressure line from the pump 14 remains substantially constant, with the only pressure surges experienced being those caused by inertial forces of reversing the working piston 42.

The pressure balance is accomplished by properly selecting the effective total cross sectional area of the passages 128 relative to the effective cross sectional flow area of the output nozzle 22 and also relative to the effective pressure areas of the working piston 42 and the high pressure pistons 44 and 46, and to the frictional forces acting on the piston assembly. To explain this relationship let us first assume these frictional forces are negligable and represent only a small correction to the relationship. It should be pointed out that the ratio of the pressure in either of the high pressure chambers 52 or 54 to the pressure in the working chamber 50 is inversely proportional to the area of the piston 44 (which is the square of the radius of the piston 44 times π) and proportional to the working area of the low pressure piston 42 (which is the square of the radius of the piston 42 times π minus the cross sectional area of the piston 44). The pressure drop through the output nozzle 22 is proportional to the square of the average velocity of fluid flow from the nozzle 22 times the density of the output fluid (i.e., water). Likewise, the pressure drop across the flow passages 128 is proportional to the square of the average flow velocity of the fluid through the passages 128 times the density of the working fluid from the pump 14. Thus, the effective cross sectional flow area of the passages 128, or of the passage shown schematically at 114, should be proportional to the effective cross sectional flow area of the nozzle 22 times the working area of the low pressure piston 42, divided by the pressure area of either of the pressure pistons 44 or 46, multiplied by the square root of the ratio of the pressure area of the working piston 42 to the pressure area of the high pressure piston 44 and 46, multiplied by the square root of the ratio of the density of the working fluid to the density of the output fluid. Expressed mathematically, this relationship can be stated as follows:

$$C_v A_v = (C_n A_n A_w/A_p) (\sqrt{A_w/A_p}) \sqrt{D_w/D_o}$$

A_w = the effective pressure area of the working piston 42

A_p = the effective pressure area of the pressure piston 44 or 46

A_n = the effective cross sectional flow area of the nozzle 22

A_v = the effective flow area of the restricted flow passages 128 of the control valve

C_n = orifice discharge coefficient of nozzle

C_v = orifice discharge coefficient of restricted flow passages 128 of the control valve

D_w = the density of the working fluid

D_o = the density of the output fluid

To give a numerical example, let it be assumed that the densities of the working fluid and output fluid are approximately equal, that the discharge coefficients are equal, and that the effective working area of the piston 42 is approximately four times the working area of either of the pressure pistons 44 or 46. The total flow of working fluid into either of the low pressure chambers 48 or 50 would be four times the flow from either of the high pressure chambers 52 and 54. According to the above formula, the total effective cross sectional flow area of the restricted valve passageway (either the passages 128 or the schematic passage 114), should be eight times the effective cross sectional flow area of the nozzle 22. In practice the passages 128 would be chosen slightly smaller to correct for the additional pressure required for overcoming friction.

In FIG. 10, there is shown a second embodiment of the control valve of the present invention, in which components corresponding to the valve 38 of the first embodiment will be given like numerical designations, with an *a* suffix distinguishing those of the second embodiment. Thus there is a housing 56*a* in which is mounted a valve spool 60*a*, comprising right and left pistons 72*a* and 74*a*, respectively, and right and left outermost end closure members 76*a* and 78*a*. However, instead of forming the inner circumferential surface portions of the pistons 72*a* and 74*a* with a tapered surface, these surface portions of the piston element 72*a* and 74*a* are stepped radially inwardly as at 124*a*. Thus, when the valve element 60*a* passes through its intermediate position (corresponding to the movement of the valve element of the first embodiment as it moves from the position of FIGS. 7B to the position of FIG. 7D), there are two restricted flow passages 128*a*, through which there are substantially equal flows. As in the first embodiment, the combined effective cross sectional flow area of the two passages of 128*a* is such as to approximate the back pressure exerted from the output nozzle back through the system to the pressurized working fluid.

A third embodiment of the control valve of the present invention is illustrated in FIG. 11. In this valve, there is a housing 140 in which there is a movable valve element 142 comprising a center piston 144 and two side pistons 146 and 147, respectively, and two end closure pistons 148 and 149, respectively. There is a center high pressure inlet port 150, right and left transfer ports 152 and 154, respectively, on opposite sides of the port 150, and right and left low pressure ports 156 and 158, respectively, located outside the ports 152 and 154.

These components are arranged so that when the valve element 142 is in the right hand position, the right transfer port 152 communicates with the low pressure port 156 to permit outflow of working fluid from its related working chamber, while high pressure fluid is being directed from the port 150 through the transfer port 154 to pressurize the other working chamber. With the valve element 142 in its left hand position, the opposite situation occurs, with high pressure fluid being directed into the port 152 and the port 154 being connected to low pressure.

The circumferential surface of the center piston 144 is stepped radially inwardly at 160 at both of its circumferential portions laterally outward from the middle of the center piston 144. These two stepped surfaces 160 form with the housing 140 restricted flow passages 162 which perform substantially the same function as the aforementioned flow passages **128** and **128a** of the first and second embodiments. For this reason, no detailed description will be provided of the operation of this valve of the third embodiment, since it is apparent from the description of the operation of the prior two embodiments of the control valve.

What is claimed is:

1. A hydraulic power apparatus for operation, for example, as a pressure intensifying system to deliver a high pressure flow of output fluid, said apparatus comprising:

a. a working cylinder,

b. a working piston mounted in said working cylinder and separating said working cylinder into first and second working chambers, said working piston and cylinder being so arranged with inlet and outlet passage means as to receive pressurized working fluid of at least a predetermined pressure level alternately into said first and second working chambers, to cause said piston to reciprocate between opposite first and second end positions in said cylinder,

c. a control valve means to direct pressurized fluid from a working fluid source alternately to said first and second working chambers to cause the reciprocation of said working piston, said control valve means having first and second positions to direct pressurized fluid alternately to respectively said first and second working chambers,

d. a first shifting valve means operatively connected to said control valve means and having an actuating position where it causes said control valve means to be positioned in its first position to deliver pressurized working fluid to said first working chamber, said first shifting valve having first actuating means to move said first shifting valve means to its actuating position, which first actuating means is responsive to the piston moving to one of its end positions of travel and also responsive to working fluid in one of said chambers being at said predetermined pressure level, and

e. a second shifting valve means operatively connected to said control valve means and having an actuating position where it causes said control valve means to be positioned in its second position to deliver pressurized working fluid to said second working chamber, said second shifting valve means having second actuating means to move said second shifting valve means to its actuating position, which second actuating means is responsive to said piston moving to the other of its end positions of travel and also responsive to working fluid in the other of said chambers being at said predetermined level.

2. The apparatus as recited in claim **1**, wherein the actuating member of each shifting valve means is positioned to be moved to its acutating position by engagement of said piston at its respective end position of travel, and also moved to its actuating position by exposure to pressure at said predtermined level in its respective working chamber.

3. The apparatus as recited in claim **2**, wherein said actuating member extends into an end of its respective working chamber and is urged by yielding means to extend into its respective working chamber.

4. The apparatus as recited in claim **1**, wherein said control valve means has first and second control chambers, which, when pressurized and depressurized, move said control valve means between its first and second positions, said first shifting valve means being operatively connected to the first control chamber so as to reduce pressure in said first control chamber to move the control valve means to its first position, and said second shifting valve means being operatively connected to the second control chamber so as to reduce pressure in said second control chamber to move the control valve means to its second position.

5. The apparatus as recited in claim **4**, wherein there are other positioning means for said valve control means, which other positioning means urges said control valve means to an intermediate position between said first and second positions, where working fluid at said predetermined pressure level is delivered to neither of the working chambers.

6. The apparatus as recited in claim **5**, wherein said other positioning means comprises spring means which resiliently urges said control valve means to its intermediate position.

7. The apparatus as recited in claim **5**, wherein said control valve means at its intermediate position functions to connect both said first and second working chambers to a low pressure area.

8. The apparatus as recited in claim **4**, further comprising other valve operating means having a first position wherein said first control chamber is exposed to low pressure, and a second position wherein said second control chamber is exposed to low pressure, whereby said control valve means can be moved between its first and second positions by operation of said other valve operating means.

9. A hydraulic power apparatus for operation, for example, as a pressure intensifying system, to deliver a high pressure stream of output fluid, said apparatus comprising:

a. a working cylinder,

b. a working piston mounted in said working cylinder and separating said working cylinder into first and second working chambers, said working piston and cylinder being so arranged as to receive pressurized working fluid of at least a predetermined pressure level alternately into said first and second working chambers, to cause said piston to reciprocate between opposite first and second end positions in said cylinder,

c. a control valve means to direct pressurized working fluid from a working fluid source alternately to said first and second working chambers to cause the reciprocation of said working piston,

1. said control valve means having a first position where it delivers pressurized working fluid to said first chamber, a second position where it delivers pressurized working fluid to said second chamber, and a third intermediate position where said first and second working chambers are connected to a lower pressure area,
2. said control valve means having first and second control chambers, which, when pressurized and depressurized, moves said control valve means between its first and second positions,
3. said control valve means having other valve positioning means arranged to urge said control valve means to its intermediate position,

d. a first and second valve shifting means operatively connected to the control chambers of the control valve means in a manner to cause movement of the control valve means between its first and second positions, the first valve shifting means having an actuating position in which it is located by said piston moving to one of its end positions of travel and also by action of said pressurized working fluid at said predetermined level in one of said chambers against said shifting valve means, said second shifting valve means also having an actuating position at which it is located by action of the piston moving to the other of its end positions of travel and also by pressurized working fluid at said predetermined level in the other of said working chambers acting against said second shifting valve means.

10. The apparatus as recited in claim **9**, wherein said first shifting valve means is operatively connected to the first control chamber so that with said first shifting valve means in its actuating position pressure is reduced in said first control chamber so as to cause movement of the control valve means to its first position, and said second shifting valve means being operatively connected to said second control chamber so that with said second shifting valve means in its actuating position pressure is reduced in said second control chamber to move said control valve means to its second position.

11. The apparatus as recited in claim **10**, wherein each of said shifting valve means comprises an actuating member which is positioned to be moved to its actuating position by engagement with said piston at its related end position of travel and also positioned to be exposed to pressure in its related working chamber to be moved to its actuating position.

12. The apparatus as recited in claim **11**, wherein said other positioning means comprises spring means which resiliently urges said control valve means to its intermediate position.

13. The apparatus as recited in claim **9**, further comprising other valve operating means having a first position wherein said first control chamber is exposed to low pressure, and a second position wherein said second control chamber is exposed to low pressure, whereby said control valve means can be moved between its first and second positions by operation of said other shifting valve means.

14. A hydraulic power apparatus for operation as a pressure intensifying system to deliver a high pressure flow of output fluid, said apparatus comprising:

a. a working cylinder,

b. a working piston mounted in said working cylinder and separating said working cylinder into first and second working chambers, said working piston and cylinder being so arranged with inlet and outlet passage means as to receive pressurized working fluid of at least a predetermined pressure level alternately into said first and second working chambers, to cause said piston to reciprocate between opposite first and second end positions in said cylinder,

c. first and second high pressure output piston means operatively connected to said working piston so as to be moved along reciprocating paths thereby to deliver a high pressure flow of output fluid,

d. a control valve means to direct pressurized working fluid from a working fluid source alternately to said first and second working chambers to cause the reciprocation of said working piston,

1. said control valve means having a first position where it delivers pressurized working fluid to said second chamber to cause said working piston to move to its first end position, a second position where it delivers pressurized fluid to said first working chamber to cause said working piston to move to its second end position, and a third intermediate position where said first and second working chambers are connected to a lower pressure area,
2. said control valve having first and second control chambers, which, when pressurized and depressurized, moves said control valve means between its first and second positions,
3. said control valve means having other valve positioning means arranged to urge said control valve means to its intermediate position,

e. a first valve shifting means having an actuating position and operatively connected to said first control chamber so as to depressurize said first control chamber when in its actuating position so as to initiate movement of said control valve means from its first position toward its second position, said first valve shifting means being responsive to move to its actuating position both by said working piston moving to its first position and by pressure in said first working chamber being at said predetermined pressure level,

f. a second valve shifting means having an actuating position and operatively connected to said second control chamber so as to depressurize said second control chamber when in its actuating position so as to initiate movement of said control valve means from its second position toward its first position, said second valve shifting means being responsive to move to its actuating position both by said working piston moving to its second position and by pressure in said second working chamber being at said predetermined pressure level,

whereby when said working piston reaches one of its end positions of travel, the working piston moves a proximate shifting valve to its actuating position to cause movement of said control valve means through its intermediate position, thereby causing a pressure reduction in the previously pressurized working chamber so that the other of said shifting valve means moves to its non-actuating position and causes continued movement of said control valve means to its opposite position, thereby effecting rapid response of said control valve means between its first and second positions.

15. The apparatus as recited in claim **14**, wherein said other valve positioning means for said control valve means comprises spring means to urge said control valve means to its intermediate position.

16. The apparatus as recited in claim **14**, wherein each of said shifting valve means comprises an actuating member extending into a related working chamber, whereby each of said actuating members is responsive to pressure in its related chamber as well as to engagement of said working piston.

* * * * *