(54) Title: LOAD RESPONSIVE FLUID CONTROL VALVE

(57) Abstract

A direction flow control valve (95) for control of positive loads equipped with a pilot operated load responsive positive load control (13b, 63b, 82, 13), which automatically regulates valve inlet pressure to maintain a relatively constant pressure differential between inlet pressure and load pressure and which permits variation in the level of pressure differential in response to an external control signal (59) while this pressure differential is maintained constant at each controlled level.

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Description

Load Responsive Fluid Control Valve

5   This is a continuation in part of application
Serial No. 949,250, filed October 6, 1978, for "Load
Responsive Fluid Control Valve".

Background of the Invention

10   This invention relates generally to load
responsive fluid control valves and to fluid power
systems incorporating such valves, which systems are
supplied by a single fixed or variable displacement
pump. Such control valves are equipped with an

15   automatic load responsive control and can be used in a
multiple load system in which a plurality of loads is
individually controlled under positive load conditions
by separate control valves.
          In more particular aspects this invention

20   relates to direction and flow control valves capable,
in a proportional way, of controlling simultaneously a
number of loads under positive load conditions.
          In still more particular aspects this
invention relates to pilot operated load responsive

25   controls of direction control valves, which permit
variation in the level of control differential between
pump discharge pressure and the load pressure signal,
while this control differential is automatically
maintained constant at each controlled level.

30   In still more particular aspects this
invention relates to pilot operated load responsive
controls of direction control valves, which permit
variation in the controlled pressure differential
between valve inlet pressure and the load pressure, in

35   response to an external control signal.

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Closed center load responsive fluid control valves are very desirable for a number of reasons. They permit load control with reduced power loss and therefore, increase system efficiency and when

5   controlling one load at a time provide the proportional feature of flow control, irrespective of the variation in the magnitude of the load. Normally such valves include a load responsive control, which automatically maintains pump discharge pressure at a level higher, by

10   a constant pressure differential, than the pressure required to sustain the load. A variable orifice, introduced between pump and load, varies the flow supplied to the load, each orifice area corresponding to a different flow level, which is maintained

15   constant, irrespective of the variation in magnitude of the load. The application of such a system is however, limited by one basic system disadvantage.

Normally in such a system the load responsive valve control can maintain a constant pressure

20   differential and therefore constant flow characteristics when operating only one load at a time. With two or more loads, simultaneously controlled, only the highest of the loads will retain the flow control characteristics, the speed of

25   actuation of lower loads varying with the change in magnitude of the highest load. Fluid control valve for such a system is shown in U.S. Patent No. 3,488,953 issued to Haussler.

This drawback can be overcome in part by the

30   provision of a proportional valve, as disclosed in my U.S. Patent No. 3,470,694, dated October 7, 1969 and also in U.S. Patent No. 3,455,210 issued to Allen on July 15, 1969. However, while those valves are effective in proportionally controlling multiple

35   positive loads at a time, they provide a constant

pressure differential and therefore a constant
throttling loss across each valve, thus reducing system
efficiency. Also those valves use an unamplified load
pressure signal in operation of their controllers
5   requiring a control signal at a comparatively large
energy level.

Summary of the Invention

It is therefore a principal object of this
10 invention to provide an improved pilot operated load
responsive direction control valve, which permits
variation in the level of control differential between
valve inlet pressure and load pressure, while this
control differential is automatically maintained
15 constant at each controlled level.

Another object of this invention is to provide
pilot operated load responsive controls of a direction
control valve, through which control of system load can
be either accomplished by variation in area of the
20 orifice, between the valve control and a fluid motor,
while the pressure differential across this orifice is
maintained constant at a specific level, or by control
of pressure differential, acting across this orifice,
while the area of the orifice remains constant.

25 It is a further object of this invention to
provide pilot operated load responsive controls of a
direction control valve, which permit variation in the
controlled pressure differential across a metering
orifice in response to an external control signal.

30 It is a further object of this invention to
provide pilot operated load responsive controls of a
direction control valve, in which an external control
signal, at a minimum force level, can adjust and
control the pressure differential, acting across a
35 metering orifice of a load responsive direction control

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valve, while the system load is being controlled by
variation in area of the metering orifice.

It is a further object of this invention to
provide load responsive controls of direction control
5   valve, which modify control signals, supplied to the
pilot operated valve controls, to control the pressure
differential across an orifice of a load responsive
direction control valve.

It is a further object of this invention to
10 provide load responsive controls of direction control
valve, which modify control signals supplied at minimum
energy level to the amplifying stage of the valve
controls, to control pressure differential across an
orifice of a load responsive direction control valve.

15 Briefly the foregoing and other additional
objects and advantages of this invention are
accomplished by providing novel load responsive
controls of a direction control valve, to throttle
fluid supplied from the pump either in response to one
20 control input, namely variation in the area of metering
orifice, to control a contant pressure differential, at
a preselected level between valve inlet pressure and
the load pressure, or in response to another control
input, namely modification in the pressure of control
25 signal, to vary the level of the control differential
between valve inlet pressure and the load pressure,
while this control differential is automatically
maintained constant at each controlled level by the
valve controls receiving low energy control signals to
30 their amplifying stage. In this way a load can be
controlled in response to either input providing
identical control performance, or the variable pressure
differential control can be superimposed on the control
action controlling a load by variation in the area of
35 the metering orifice. Therefore this control system

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lends itself very well to an application, in which a
manual control input from an operator may be modified
by an electronic logic circuit, or a micro-processor.

Additional objects of this invention will
become apparent when referring to the preferred
embodiments of the invention as shown in the
accompanying drawings and described in the following
detailed description.

Description of the Drawings

Fig. 1 is a diagrammatic representation of a
load responsive pilot operated throttling control for
adjustment in the level of control differential from a
certain preselected level to zero level, with fluid
motor and system pump shown schematically;

Fig. 2 is a diagrammatic representation of a
load responsive pilot operated throttling control for
adjustment in the level of the control differential
from a certain minimum preselected value up to maximum
level, with fluid motor and system pump shown
schematically;

Fig. 3 is a diagrammatic representation of
another embodiment of the load responsive pilot
operated control of Fig. 1, with fluid motor and system
pump shown schematically;

Fig. 4 is a diagrammatic representation of
still another embodiment of the load responsive pilot
operated control of Fig. 1, with fluid motor and system
pump shown schematically;

Fig. 5 is a diagrammatic representation of a
four way load responsive direction control valve using
the control of Fig. 1 with system pump and reservoir
shown schematically;

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Fig. 6 is a diagrammatic representation of manual control input into the load responsive controls of Figs. 1 to 5;

Fig. 7 is a diagrammatic representation of
5   hydraulic control input into load responsive controls of Figs. 1 to 5;

Fig. 8 is a diagrammatic representation of electrohydraulic control input into load responsive controls of Figs. 1 to 5;

10   Fig. 9 is a diagrammatic representation of an electromechanical control input into load responsive controls of Figs. 1 to 5; and

Fig. 10 is a diagrammatic representation of an electromechanical control input into load responsive
15   system of Fig. 4.

Description of the Preferred Embodiments

Referring now to Fig. 1, the hydraulic system shown therein comprises a fluid pump 10, equipped with
20   an output flow control 11, connected to a reservoir 12. The output flow control 11 regulates delivery of the pump 10 into a load responsive circuit, composed of a differential throttling control, generally designated as 13, regulating the level of the pressure
25   differential developed across schematically shown variable orifice 14, interposed between the differential throttling control 13 and a fluid motor 15, operating load W. The pump 10 may be of a fixed or variable displacement type and may respond to an
30   external or internal control signal. With the pump 10 being of a fixed displacement type, responding to an external control signal, the output flow control 11, in a well known manner, regulates delivery from pump to the load responsive circuit, by bypassing part of the
35   pump flow to the system reservoir 12. The pump 10 of

fixed displacement type can also respond to maximum
pressure limiting signal, with the use of conventional
relief valve, well known in the art. With the pump 10
being of variable displacement type and capable of
5   responding to an external control signal, the output
flow control 11, in a well known manner, regulates
through a displacement changing mechanism delivery from
the pump to load responsive circuit, by changing pump
displacement. The pump 10 of variable displacement
10   type can also respond to the maximum pressure limiting
signal, such a control, well known in the art, being in
the form of a conventional pressure compensator.

Discharge line 16 of pump 10 is connected
through the differential throttling control, generally
15   designated as 13, line 17, variable metering orifice 14
and line 18 to the fluid motor 15. The fluid motor 15
is also connected by lines 18 and 19 and orifice 20
with the differential throttling control 13.

The differential throttling control, generally
20   designated as 13, composed of a throttling section,
generally designated as 13a and a signal modifying
section, generally designated as 13b, comprises a
housing 21 having inlet chamber 22, an outlet chamber
23, a first control chamber 24 and an exhaust chamber
25   25, all of those chambers being connected by bore 26,
slidably guiding a throttling spool 27. The throttling
spool 27, equipped with lands 28 and 29 and stop 30, is
provided with throttling slots 31, terminating in the
cut-off edges 32, between the inlet chamber 22 and the
30   outlet chamber 23. One end of the throttling spool 27
projects into the first control chamber 24, while the
other end projects into the exhaust chamber 25 and is
biased by a control spring 33. The first control
chamber 24 is connected by passage 34 with annular
35   space 35. Bore 36 connects annular space 35 with port

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36 and a second control chamber 37 and axially guides a
pilot valve spool 38. The pilot valve spool 38,
equipped with a metering land 39 and land 40 which
defines annular space 41, communicates with port 36 and
5   projects into the second control chamber 37, where it
engages a spring 42. The second control chamber 37 is
connected through line 19 and orifice 20 to the fluid
motor 15 and is also connected through port 43 with the
supply chamber 44, connected by bore 45 with a third
10   control chamber 46 and an exhaust chamber 47. Bore 45
slidably guides a control spool 48, equipped with land
49, provided with throttling slots 50 and positioned
between the supply chamber 44 and the third control
chamber 46, a land 51 separating the supply chamber 44
15   and the exhaust chamber 47 and a flange 52. A spring
53 is interposed in the exhaust chamber 47 between the
flange 52 of the control spool 48 and the housing 21.
The exhaust chamber 47 connected by passage 54 with the
exhaust chamber 25 and the third control chamber 46 are
20   selectively interconnected by metering orifice created
by a stem 55 guided in bore 56 and provided with
metering slots 57. The stem 55 is connected to an
actuator 58 responsive to external control signal 59.
Exhaust chambers 47 and 25 connected by passage 54 are
25   also connected by passage 60 with annular space 41 and
leakage orifice 61.

        Referring now to Fig. 2, the same components
used in Fig. 1 are designated by the same numerals.
The only difference between the load responsive
30   controls of Figs. 1 and 2 is the phasing of the
internal components of the differential control 13 and
connections of individual ports or chambers with the
fluid motor 15 and the pump 10. In both figures, in an
identical way, the pump pressure is transmitted through
35   discharge line 16, the inlet chamber 22, the outlet

chamber 23, line 17 and variable orifice 14 to the
fluid motor 15. Line 17 however, is connected in Fig.
2 through line 19 and orifice 20 to port 36 of the
pilot valve spool 38. The second control chamber 37 is
5   in direct communication through lines 62 and 19 with
load pressure Pw of the fluid motor 15. The supply
chamber 44 is connected through port 43 with port 36
and selectively interconnected with the third control
chamber 46 by the control spool 48.

10   Referring now to Fig. 3, the same components
used in Figs. 1 and 2 are designated by the same
numerals. The basic load responsive circuit of Fig. 3,
with some of the circuit components including some of
the internal components of differential throttling
15 control, generally designated as 63, are the same as
those of Figs. 1 and 2. The differential throttling
control 63 is composed of a throttling section 63a,
identical to throttling section 13a of Figs. 1 and 2, a
flow control valve section 63b and a metering valve
20 section 63c. The second control chamber 37 is
connected by port 64 with a first pressure chamber 65,
which in turn is connected through bore 66 with a
second pressure chamber 67, which guides a stem 55,
equipped with metering slots 68. The stem 55 is
25 connected to the actuator 58, responsive to the
external control signal 59. A load check 69 is
interposed between the fluid motor 15 and the outlet
chamber 23. The second control chamber 37 is also
connected through port 70 with a third pressure chamber
30 71, connected by bore 72 with an exhaust chamber 73
which is connected through passage 74 with the exhaust
chamber 25. Bore 72 axially guides a metering pin 75,
provided with metering slot 76. The metering pin 75 is
provided with a stop 77 and biased, towards position as
35 shown, by a spring 78, contained in the exhaust chamber
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Referring now to Fig. 4, the same components
used in Figs. 1, 2 and 3 are designated by the same
numerals. The basic load responsive circuit of Fig. 4
with some of the circuit components, including some of
5   the internal components of differential throttling
control, generally designated as 79, are the same as
those of Figs. 1, 2 and 3. The second exhaust chamber
37 is connected by port 80 to the chamber 81 of the
differential valve, generally designated as 82. The
10   differential valve 82 comprises a coil 83, retained in
the housing, which guides an armature 84 of a solenoid,
generally designated as 85. The armature 84 is
provided with conical surface 86, selectively
engageable with sealing edge 87 of inlet port 88 and
15   venting passage 89 terminating in bore 90, guiding a
reaction pin 91. The coil 83 is connected by sealed
connector 92 to outside of the housing 21, external
control signal being applied to the sealed connector
92. The second control chamber 37 is connected through
20   leakage orifice 93 and passage 94 to the exhaust
chamber 25 and the reservoir 12.

Referring now to Fig. 5 the same components
used in Figs. 1, 2, 3 and 4 are designated by the same
numerals. The differential throttling control 13 of
25   Fig. 1 was interconnected in Fig. 5 into a four way
valve assembly, generally designated as 95, which is
basically equivalent to variable metering orifice 14 of
Fig. 1. The four way valve assembly, generally
designated as 95, comprises a housing 96 having a
30   supply chamber 97, load chambers 98 and 99 and exhaust
chambers 100 and 101, interconnected by bore 102,
guiding a valve spool 103. The valve spool 103 is
provided with lands 104, 105 and 106, throttling slots
107, 108, 109 and 110 and signal slots 111 and 112.
35   The housing 96 is also provided with load sensing ports

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113 and 114, communicating through line 115 and orifice
20 to the second control chamber 37, of the
differential throttling control 13.

5   Referring now to Fig. 6, the stem 55 of the
actuator 58 of Figs. 1 to 5 is biased by a spring 116
towards position of zero orifice and is directly
operated by a lever 117, which provides the external
signal 59.

          Referring now to Fig. 7, the stem 55 of the
10   actuator 58 of Figs. 1 to 5, is biased by a spring 118
towards position of zero orifice and is directly
operated by a piston 119. Fluid pressure is supplied
to the piston 119 from a pressure generator 120,
operated by a lever 121.

15   Referring now to Fig. 8, the stem 55 of the
actuator 58 of Figs. 1 to 5, is biased by a spring 122
towards position of zero orifice and is directly
operated by a solenoid 123, connected by line to an
input current control 124, operated by a lever 125 and
20   supplied from an electrical supply source 126.

          Referring now to Fig. 9, the stem 55 of the
differential control, generally designated as 127, is
biased by a spring 128 towards a position, where it
isolates the third control chamber 46 from the exhaust
25   chamber 47 and is controlled by a solenoid 129. The
electrical control signal, amplified by amplifier 130,
is transmitted from a logic circuit or a micro-
processor 131, subjected to inputs 132, 133 and 134.

          Referring now to Fig. 10, a logic circuit or a
30   micro-processor 135, supplied with control signals 136,
137 and 138, transmits an external control signal to
the differential valve 82 through an amplifier 139.

          Referring now to Fig. 1, the differential
throttling control 13 is interposed between the pump 10
35   and the fluid motor 15 and controls the fluid flow and

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pressure therebetween. The differential throttling
control 13 is composed of the throttling section 13a
and signal modifying section 13b. The throttling
section 13a with its throttling spool 27 throttles with
5   throttling slots 31 fluid flow from the inlet chamber
22, connected by discharge line 16 to the pump 10, to
the outlet chamber 23, connected by line 17 and
variable orifice 14 with the fluid motor 15, to
automatically maintain a constant pressure differential
10   across variable orifice 14. This control action is
accomplished in the following way. Fluid from the
outlet chamber 23 at P_1 pressure, which is the
pressure acting upstream of variable orifice 14, is
transmitted through line 17 to port 36 where, reacting
15   on the cross-sectional area of the pilot valve spool
38, generates a force tending to move the pilot valve
spool 38 upward to connect P_1 pressure through
annular space 35 and passage 34 to the first control
chamber 24 and therefore increase the pressure level in
20   the first control chamber 24. Fluid at load pressure
Pw, which is the pressure acting downstream of variable
orifice 14, is transmitted through line 19 and orifice
20 to the second control chamber 37 where, reacting on
the cross-sectional area of the pilot valve spool 38,
25   it generates a force tending to move the pilot valve
spool downward, to connect the reservoir pressure from
annular space 41 to annular space 35, passage 34 and to
the first control chamber 24 and therefore decrease the
pressure level in the first control chamber 24. This
30   force due to pressure in the second control chamber 37
is supplemented by the biasing force of the spring 42.
Increase in pressure level in the first control chamber
24, above the level equivalent to preload of control
spring 33, reacting on cross-sectional area of the
35   throttling spool 27, will

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generate a force tending to move the throttling spool
27 from right to left, in the direction of closing of
the flow area through the throttling slots 31 and
therefore in direction of increasing the throttling
5   action of the throttling spool 27. Conversely, a
decrease in pressure level in the first control chamber
24, below level equivalent to preload of control spring
33, will result in the control spring 33 moving the
throttling spool 27 from left to right, in the
10   direction of increasing the flow area through the
throttling slots 31 and therefore in direction of
decreasing the throttling action of the throttling
spool 27. Therefore by regulating pressure level in
the first control chamber 24 the pilot valve spool 38
15   will control the throttling action of the throttling
spool 27 and consequently the pressure drop between the
inlet chamber 22 subjected to Pp pressure and the
outlet chamber 23 subjected to P_1 pressure. Assume
that the stem 55 is in the position as shown in Fig. 1,
20   isolating the third control chamber 46 from the exhaust
chamber 47 and therefore making the signal modifying
section 13b inactive. The pilot valve spool 38,
subjected to P_1 and P_2 pressures and the biasing
force of spring 42 will reach a modulating position, in
25   which by throttling action of metering land 39 will
regulate the pressure in the first control chamber 24
and therefore the throttling action of the throttling
spool 27 to throttle the pump pressure Pp to a level of
P_1 pressure which is higher, by a constant pressure
30   differential $\triangle P$, than P_2 pressure and equal to the
quotient of the biasing force of spring 42 and the
cross-sectional area of the pilot valve spool 38. In
this way the pilot valve spool 38, subjected to low
energy pressure signals, will act as an amplifying
35   stage using the energy derived from the pump 10 to

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control the position and therefore the throttling
action of the throttling spool 27. Leakage orifice 61,
connecting the first control chamber 24 through passage
60 and the exhaust chamber 24 to the reservoir 12, is
5   used, in a well known manner, to increase the stability
of the pilot valve spool 38. If P_2 pressure is equal
to Pw pressure which is the case when the stem 55 is in
the position, as shown in Fig. 1, the throttling
section 13a, by throttling fluid flow from the inlet
10   chamber 22 to the outlet chamber 23, will automatically
maintain a constant pressure differential $\triangle P$ between
the outlet chamber 23 and the second control chamber 37
and with $\triangle Py$ becoming $\triangle P$, will also maintain a
constant pressure differential across variable orifice
15   14. With constant pressure differential, acting across
an orifice, the flow through an orifice will be
proportional to the area of the orifice and independent
of pressure in the fluid motor. Therefore by varying
the area of variable orifice 14, the fluid flow to the
20   fluid motor 15 and velocity of the load W can be
controlled, each specific area of variable orifice 14
corresponding to a specific velocity of load W, which
will remain constant, irrespective of the variation in
the magnitude of the load W.

25   In the arrangement of Fig. 1 the relationship
between load pressure Pw and signal pressure P_2 is
controlled by the signal modifying section, generally
designated as 13b, and orifice 20. Assume that the
stem 55, positioned by the actuator 58 in response to
30   external control signal 59, as shown in Fig. 1, blocks
completely metering orifice through metering slots 57,
isolating the third control chamber 46 from the exhaust
chamber 47. The control spool 48 with its land 49,
protruding into the third control chamber 46, will
35   generate pressure in the third control chamber 46,

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equivalent to the preload of the spring 53.
Displacement of the stem 55 to the left will move
metering slots 57 out of bore 56, creating an orifice
area, through which fluid flow will take place from the
5   third control chamber 46 to the exhaust chamber 47.
The control spool 48, biased by the spring 53, will
move from right to left, connecting by throttling slots
50 the supply chamber 44 with the third control chamber
46. Rising pressure in the third control chamber 46,
10   reacting on cross-sectional area of control spool 48,
will move it back into a modulating position, in which
sufficient flow of pressure fluid will be throttled
from the supply chamber 44 to the third control chamber
46, to maintain the third control chamber 46 at a
15   constant pressure, equivalent to preload in the spring
53. When displacing metering slots 57, in respect to
bore 56, area of metering orifice between the third
control chamber 46 and the exhaust chamber 47 will be
varied. Since constant pressure differential is
20   automatically maintained between the exhaust chamber 47
and the third control chamber 46 and therefore across
the metering slots 57, by the control spool 48, each
specific area of metering slots 57 will correspond to a
specific constant flow level from the third control
25   chamber 46 to the exhaust chamber 47 and from the
supply chamber 44 to the third control chamber 46,
irrespective of the magnitude of the pressure in the
supply chamber 44. Therefore, each specific position
of stem 55, within the zone of metering slots 57, will
30   correspond to a specific flow level and therefore a
specific pressure drop $\triangle Px$ through the fixed orifice
20, irrespective of the magnitude of the load pressure
Pw. When referring to Fig. 1 it can be seen that
$P_1 - Pw = \triangle Py$, $P_1 - P_2 = \triangle P$, maintained
35   constant by the throttling section 13a and

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Pw - P_2 = $\triangle Px$. From the above equations, when
substituting and eliminating P_1 and P_2 a basic
relationship of $\triangle Py$ = $\triangle P$ - $\triangle Px$ is obtained.
Since $\triangle Px$ can be varied and maintained constant at

5   any level by the signal modifying section 13b, so
can $\triangle Py$, acting across variable orifice 14, be varied
and maintained constant at any level. Therefore with
any specific constant area of variable orifice 14, in
response to control signal 59, pressure differential

10   $\triangle Py$ can be varied from maximum to zero, each specific
level of $\triangle Py$ being automatically controlled constant,
irrespective of variation in the load pressure Pw.
Therefore, for each specific area of variable orifice
14 the pressure differential, acting across orifice 14

15   and the flow through orifice 14 can be controlled from
maximum to minimum by the signal modifying section 13b,
each flow level automatically being controlled constant
by the differential throttling control 13, irrespective
of the variation in the load pressure Pw. From

20   inspection of the basic equation $\triangle Py$ = $\triangle P$ - $\triangle Px$
it becomes apparent that with $\triangle Px$ = 0, $\triangle Py$ = $\triangle P$
and that the system will revert to the mode of
operation of conventional load responsive system, with
maximum constant $\triangle P$ of the differential throttling

25   control 13. When $\triangle Px$ = $\triangle P$, $\triangle Py$ becomes zero,
outlet pressure from the differential throttling
control 13 P_1 will be equal to load pressure Pw and
the flow through variable orifice 14 will become zero.
With $\triangle Px$ larger than $\triangle P$, pressure P_1 will become

30   smaller than load pressure Pw and the load check 18a
will seat.

In the load responsive system of Fig. 1 for
each specific value of $\triangle Py$, maintained constant by
the modifying section 13b through the throttling

35   section 13a of the differential control 13, the area of
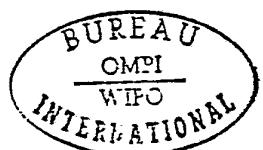
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variable orifice 14 can be varied, each area
corresponding to a specific constant flow into the
fluid motor 15, irrespective of the variation in the
magnitude in the load pressure Pw. Conversely, for
5   each specific area of the variable orifice 14 pressure
differential Δ Py, acting across orifice 14, can be
varied by the signal modifying section 13b, through the
throttling section 13a of the differential throttling
control 13, each specific pressure differential Δ Py
10 corresponding to a specific constant flow into the
fluid motor 15, irrespective of the variation in the
magnitude of the load pressure Pw. Therefore fluid
flow into fluid motor 15 can be controlled either by
variation in area of variable orifice 14, or by
15 variation in pressure differential Δ Py, each of those
control methods displaying identical control
characteristics and controlling flow, which is
independent of the magnitude of the load pressure. .
Action of one control can be superimposed on the action
20 of the other, providing a unique system, in which, for
example, a command signal from the operator, through
the use of variable orifice 14 can be corrected by
signal 59 from a computing device, acting through the
signal modifying section 13b.

25   Referring now to Fig. 2, the signal modifying
section 13b, is identical to the signal modifying
section 13b of Fig. 1 and performs in an identical way,
by modifying a control signal transmitted to the
throttling section 13a. The throttling section 13a of
30 Fig. 2 is identical to the throttling section 13a of
Fig. 1. However, the signal modifying section 13b of
Fig. 2 modifies the control signal from outlet chamber
23 and therefore from upstream of the variable orifice
14, instead of modifying the control signal of load
35 pressure Pw, as shown in the system of Fig. 1. In Fig.

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2 the load pressure signal Pw is transmitted directly
from fluid motor 15 to the second control chamber 37,
of the throttling section 13a. Then, as can be seen in
Fig. 2, $P_1 - Pw = \triangle Py$, $P_1 - P_2 = \triangle Px$ and P_2

5   $- Pw = \triangle P$, which, in a manner as previously
described, is the basic system differential and is
maintained constant by the throttling section 13a of
the differential throttling control 13. From the above
equations, when substituting and eliminating P_1 and

10 P_2, the basic relationship of $\triangle Py = \triangle P + \triangle Px$
can be obtained. Since $\triangle Px$ can be varied and
maintained constant at any level, so can $\triangle Py$, acting
across variable orifice 14 be varied and maintained
constant at any level. From inspection of the basic

15. equation $\triangle Py = \triangle P + \triangle Px$ it becomes apparent that
with $\triangle Px = 0$, $\triangle Py = \triangle P$ and that the system will
revert to the mode of operation of conventional load
responsive system, with minimum constant $\triangle P$ equal to
pressure differential of the throttling section 13a.

20 Any value of $\triangle Px$, other than zero, will increase the
pressure differential $\triangle Py$, acting across variable
orifice 14 above the level of constant pressure
differential $\triangle P$ of the throttling section 13a.
Therefore, the load responsive control arrangement of

25 Fig. 1 will control $\triangle Py$ in a range between $\triangle P$ and
zero, while the load responsive arrangement of Fig. 2
will control $\triangle Py$ in a range above the level of
constant pressure differential $\triangle P$ of the throttling
section 13a.

30   Referring now to Fig. 3, the load responsive
system is identical to the load responsive system of
Fig. 1 with the exception of the flow control valve
section 63b and the metering valve section 63c, which,
when combined together are equivalent to the signal

35 modifying section 13b of Fig. 1 and perform in a very

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similar way. The throttling section 63a of Fig. 3 is
identical to the throttling section 13a of Fig. 1. It
is apparent that the differential throttling control 63
of Fig. 3 performs in an identical way as the
5   differential throttling control 13 of Fig. 1. The flow
control valve section 63b of the differential
throttling control 63 in the housing 21, is provided
with bore 72, guiding the metering pin 75, which is
subjected to pressure in the third pressure chamber 71,
10 which is connected by port 64 with the second control
chamber 37, to the reservoir pressure in the exhaust
chamber 73 and to the biasing force of the spring 78.
Subjected to pressure in the third pressure chamber 71
the metering pin 75 will move from left to right, each
15 specific pressure level corresponding to a specific
position of metering pin 75, in respect to the housing
21 and also corresponding to the specific biasing force
of spring 78. Each specific position of metering pin
75, in respect to the housing 21, will correspond to a
20 specific flow area of metering slot 76, interconnecting
the third pressure chamber 71 with the exhaust chamber
73. The shape of metering slot 76 and the
characteristics of the biasing spring 78 are so
selected, that variation in effective orifice area of
25 metering slot 76, in respect to pressure in the third
pressure chamber 71, will provide a relatively constant
flow from the third pressure chamber 71 to the exhaust
chamber 73. To obtain special control characteristics
of the load responsive control the shape of the
30 metering slot 76 may be so selected, that any desired
relationship between the flow from the third pressure
chamber 71 and its pressure level can be obtained.
Assume that the flow control valve section 63b provides
a constant flow from the third pressure chamber 71 and
35 therefore from the second control chamber 37,

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irrespective of its pressure level. Then in a well
known manner, the flow control valve section 63b could
be substituted by a conventional flow control valve,
well known in the art. Constant flow to the third
5   pressure chamber 71 is supplied from fluid motor 15
through the metering valve section 63c, second control
chamber 37 and port 70. The metering valve section
63c, upstream of the flow control valve section 63b is
provided with a bore 66, guiding the stem 55, provided
10   with metering slots 68. Displacement of metering slots
68 past bore 66 creates an orifice, the effective area
of which can be varied by positioning of stem 55 by the
actuator 58, in response to external control signal
59. With stem 55 engaging bore 66 the flow area of the
15   metering valve section 63c becomes zero. Therefore, in
response to external control signal 59, the effective
flow area through the metering valve section 63c can be
varied from zero to a selected maximum value. Since
the flow through the metering valve section 63c is
20   maintained constant by the flow control valve 63b, each
specific area of flow through the metering valve
section 63c, in a well known manner, will correspond to
a specific constant pressure drop ΔPx, irrespective
of the variation in the load pressure Pw. Therefore
25   the load pressure signal can be modified in the second
control chamber 37 of the throttling section 63a, each
value of pressure drop ΔPx, maintained constant by
the flow control valve section 63b, corresponds to a
specific value of pressure differential ΔPy, fol-
30   lowing the basic relationship of $\Delta Py = \Delta P - \Delta Px$.
Therefore the control characteristics of the load
responsive control of Fig. 3 will be identical to those
described, when referring to Fig. 1, the pressure
differential ΔPy being varied and maintained
35   constant, at each specific level, by the control action

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of the flow control valve section 63b and the metering
valve section 63c, in response to external control
signal 59, between maximum value equal to $\triangle P$ and zero.

In a manner as previously described the shape

5   of metering slot 76 and the biasing force char-
acteristics of spring 78 can be so selected, that any
desired relationship between pressure in the third
pressure chamber 71 and the fluid flow through the
metering valve section 63c can be obtained. Assume

10   that in response to a specific external control signal
59 a specific flow area was created through the
metering valve section 63c. Then controlled increase
in flow through the metering valve section 63c, with
increase in load pressure, will proportionally increase

15   the pressure differential $\triangle Px$ and therefore pro-
portionally decrease the pressure differential $\triangle Py$,
effectively decreasing the gain of the load responsive
control with increase in the load pressure.
Conversely, a controlled decrease of flow through the

20   specific orifice area of the metering valve section
63c, with increase in the load pressure will
proportionally decrease the pressure differential $\triangle Px$
and therefore proportionally increase the pressure
differential $\triangle Py$, effectively increasing the gain of

25   the load responsive control, with increase in the load
pressure. As is well known in the art, the stability
margin of most fluid flow and pressure controllers
decreases with increase in system pressure. Therefore
the capability of adjusting the system gain, in respect

30   to system pressure, is of primary importance. With the
flow control valve section 63b the rate of change of
pressure differential $\triangle Py$, in respect to load
pressure, does not have to be constant and can be
varied in any desired way.

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Referring now to Fig. 4, the load responsive
system is similar to that of Fig. 1. The throttling
control 79, of the differential throttling control of
Fig. 4 is identical to differential throttling section
5   13a of Fig. 1. However, the differential valve 82 is
different from the signal modifying section 13b of Fig.
1, although it performs the same function and provides
identical performance. The differential valve, gener-
ally designated as 82, contains the solenoid, generally
10   designated as 85, which consists of coil 83, secured in
the housing 21 and the armature 84, slidably guided in
the coil 83. The armature 84 is provided with conical
surface 86, which in cooperation with sealing edge 87,
regulates the pressure differential Δ Px between inlet
15   port 88 and port 80. A comparatively weak unnumbered
spring can be interposed between the armature 84 and
the housing 21, to permit a back flow under deenergized
condition of the coil 83 from port 80 to inlet port
88. This feature may be of importance, when using a
20   shuttle valve logic system instead of a check valve
system, for transmittal of control signals. The sealed
connector 92, in the housing 21, well known in the art,
connects the coil 83 with external terminals, to which
the external signal 59 can be applied. A solenoid is
25   an electro-mechanical device, using the principle of
electromagnetics, to produce output forces from
electrical input signals. The force developed on the
solenoid armature 84 is a function of the input
current. As the current is applied to the coil 83,
30   each specific current level will correspond to a
specific force level, transmitted to the armature.
Therefore, the contact force between the conical
surface 86 of the armature 84 and sealing edge 87 of
housing 21 will vary and be controlled by the input
35   current. This arrangement will then be equivalent to a
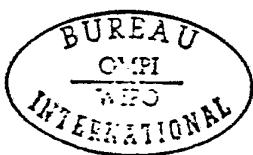
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type of differential pressure throttling valve, varying
automatically the pressure differential $\triangle Px$ between
inlet port 88 and second control chamber 37, in
proportion to the force developed in the armature 84,
5   in respect to the area enclosed by the sealing edge 87
and therefore proportional to the external signal 59,
of the input current supplied to the solenoid 85. The
pressure forces acting on the armature 84, within the
housing 21, are completely balanced with the exception
10   of the pressure force due to the pressure differ-
ential $\triangle Px$ acting on the enclosed area of sealing
edge 87. This force is partially balanced by the
reaction force, developed on the cross-sectional area
of the reaction pin 91, guided in a bore 90, which is
15   connected through venting passage 89 with inlet port
88. The cross-sectional area of the reaction pin 91
must always be smaller than the area enclosed by
sealing edge 87, so that a positive force, due to the
pressure differential $\triangle Px$, opposes the force
20   developed by the solenoid 85. The reaction pin 91
permits use of a larger inlet port 88, while also
permitting a very significant reduction in the size of
solenoid 85, also permitting the solenoid 85 to work in
the higher range of $\triangle Px$. The second control chamber
25   37 is connected by leakage orifice 93 and passage 94
with the system reservoir.
        Referring now to Fig. 5, the load responsive
system is identical to that as shown in Fig. 1 with
identical differential throttling controls being used,
30   but the variable orifice 14 of Fig. 1 was substituted
in Fig. 5 by a load responsive four way type direction
control valve, generally designated as 95. The
performance of the control embodiments of Figs. 1 and 5
is identical, the only difference being the con-
35   struction of the variable orifice. The differential

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throttling control and specifically the second load
chamber 37 are connected through orifice 20 and line
115 with the load sensing ports 113 and 114 of four way
valve 95. With the valve spool 103 in its neutral
5   position, as shown in Fig. 5, load pressure sensing
ports 113 and 114 are blocked by the land 105 and
therefore effectively isolated from load pressure,
existing in load chamber 98 or 99. Under those
conditions, in a well known manner, the differential
10   throttling control 13, automatically maintains minimum
pressure in the supply chamber 97 and equal to $\triangle$ P of
the throttling section 13a. Displacement of the valve
spool 103 from its neutral position in either
direction, first connects with signal slot 111 or 112
15   load chamber 99 or 98 with load pressure sensing port
113 or 114, while load chambers 99 and 98 are still
isolated by the valve spool 103 from the supply chamber
91 and exhaust chambers 100 and 101. Then the load
pressure signal is transmitted through load pressure
20   sensing port 113 or 114, line 115 and orifice 20 to the
second control chamber 37, permitting the differential
throttling control 13 to react, before metering orifice
is open to the load chamber 99 or 98. Further
displacement of valve spool 103, in either direction,
25   will create, in a well known manner, through metering
slot 107 or 108 a metering orifice between one of the
load chambers and the supply chamber 97, while
connecting the other load chamber, through metering
slot 109 or 110 with the exhaust chambers, in turn
30   connected to system reservoir. The metering orifice
can be varied by displacement of valve spool 103 each
position corresponding to a specific flow level into
one of the load chambers, irrespective of the magnitude
of the load controlled by four way valve assembly 95.
35   Upon this control, in a manner as previously described

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when referring to Fig. 1, can be superimposed the
control action of the signal modifying section 13b.
With valve spool 103 displaced to any specific
position, corresponding to any specific area of
metering orifice, the flow into load chambers can be
proportionally controlled by the differential
throttling control 13 with its signal modifying section
13b, each value of pressure differential Δ Py being
automatically maintained at a constant level by the
throttling section 13a and corresponding to a specific
flow level into load chambers irrespective of the
magnitude of the load controlled by the four way valve
assembly 95.

Referring now to Fig. 6, the stem 55 of the
actuator 58 of Figs. 1 to 5 is biased by spring 118
towards position of zero orifice and is directly
operated by a lever 117, which provides the external
signal 59 in the form of manual input.

Referring now to Fig. 7, the stem 55 of
actuator 58 of Figs. 1 to 5 is biased by spring 118
towards position of zero orifice and is directly
operated by a piston 119. Fluid pressure is supplied,
in a well known manner, to the piston 119 from a
pressure generator 120, operated by a lever 121.
Therefore the arrangement of Fig. 7 provides the
external signal 59 in the form of a fluid pressure
signal.

Referring now to Fig. 8, the stem 55 of the
actuator 58 of Figs. 1 to 5 is biased by a spring 122
towards position of zero orifice and is directly
operated, in a well known manner, by a solenoid 123,
connected by a line to an input current control 124,
operated by a lever 125 and supplied from an electrical
power source 126. Therefore the arrangement of Fig. 8
supplies the external signal 59 in the form of an

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electric current, proportional to displacement of lever
125.

Referring now to Fig. 9, the stem 55 of the
differential control 127 is biased by a spring 128
5   towards a position, where it isolates the third control
chamber 46 from the exhaust chamber 47. The stem 55 is
completely pressure balanced, can be made to operate
through a very small stroke and controls such low
flows, at such low pressures, that the influence of
10   flow forces is negligible. In any event, if the area
of metering slots 57 is so selected, that it provides a
linear function in respect to displacement of the stem
55 and a constant pressure is maintained in front of
the orifice, the flow forces will also be linear and
15   will add to the spring force, changing slightly the
combined rate of the spring. The stem 55 is directly
coupled to a solenoid 129. A solenoid is an
electromechanical device using the principle of
electromagnetics, to produce output forces from
20   electrical input signals. The position of solenoid
armature, when biased by a spring, is a function of the
input current. As the current is applied to the coil,
the resulting magnetic forces generated move the
armature from its deenergized position to its energized
25   position. When biased by a spring, for each specific
current level there is a corresponding particular
position, which the solenoid will attain. As the
current is varied from zero to maximum rating, the
armature will move one way from a fully retracted to a
30   fully extended position in a predictable fashion,
depending on the specific level of current at any one
instant. Since the forces, developed by solenoid 129
are very small, so is the input current, which is
controlled by a logic circuit or a micro-processor
35   131. The micro-processor 131 will then, in response to

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different types of transducers either directly control
the system load, in respect to speed, force and
position, or can superimpose its action upon the
control function of an operator, to perform required
5   work in the minimum time, with a minimum amount of
energy, within the maximum capability of the structure
of the machine and within the envelope of its
horsepower.

Referring now to Fig. 10, the control signal
10   from the logic circuit, or the micro-processor 135, in
a similar way as described in Fig. 9, is directly
transmitted through the amplifier 139 to the
differential pressure control 82, where through a
solenoid and throttling valve combination, in a manner
15   as previously described, when referring to Fig. 4,
regulates the pressure differential in response to
input current.

Although the preferred embodiments of this
invention have been shown and described in detail it is
20   recognized that the invention is not limited to the
precise form and structure shown and various
modifications and rearrangements as will occur to those
skilled in the art upon full comprehension of this
invention may be resorted to without departing from the
25   scope of the invention as defined in the claims.

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Claims

1. A load responsive valve assembly
comprising a housing (21) having an inlet chamber (22)
5   connected to a pump (10), a supply chamber (23)
connected to a fluid motor (15) and exhaust means (12),
control orifice means (14) interposed between said
·supply chamber (23) and said fluid motor (15), first
·valve means (27) having fluid throttling means (31)
10 between said inlet chamber (22) and said supply chamber
(23) controllable by a pilot valve means (38) and
operable to throttle fluid flow from said inlet chamber
(22) to said supply chamber (23) to maintain a constant
pressure differential at a preselected constant level
15 across said pilot valve means (38) and to maintain a
constant pressure differential across said control
orifice means (14), and second valve means (13b) having
means (48,55) operable through said first valve means
to vary the level of said constant pressure
20 differential across said control orifice means (14)
while said pressure differential across said pilot
valve means (38) remains constant at said constant
predetermined level.

25   2.   A load responsive valve assembly as set
forth in claim 1 wherein said control orifice means
(14) has variable area orifice means (107,108).

         3.   A load responsive valve assembly as set
30 forth in claim 1 wherein said second valve means (13b
of Fig. 2) has means (48,55) to vary the level of said
constant pressure differential across said control
orifice means (14) above the level of said pressure
differential across said pilot valve means (38)
35 maintained constant at said constant predetermined
level.

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4. A load responsive valve assembly as set
forth in claim 1 wherein said second valve means (13b)
includes constant pressure reducing means (48), orifice
means (20) upstream of said constant pressure reducing
5   means (48), and flow orifice means (57) downstream of
said constant pressure reducing means (48).

5. A load responsive valve assembly as set
forth in claim 1 wherein said second valve means (63b)
10 includes flow orifice means (55,68) and a pressure
responsive flow control means (75,76) downstream of
said flow orifice means (55,68).

6. A load responsive valve assembly as set
15 forth in claim 1 wherein said second valve means (82)
includes fluid throttling means (84) and orifice means
(93) downstream of said fluid throttling means (84)
communicable with said exhaust means (12).

20   7. A load responsive valve assembly as set
forth in claim 1 wherein said second valve means (13b
of Fig. 1) has means (48,55) to vary the level of said
constant pressure differential across said control
orifice means (14) below the level of said pressure
25 differential across said pilot valve means (38)
maintained constant at said constant predetermined
level.

8. A load responsive valve assembly as set
30 forth in claim 1 wherein said second valve means
(13b,43b,82) has means (58) responsive to an external
control signal (59).

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     9. A load responsive valve assembly as set
forth in claim 8 wherein said means (58) responsive to
an external control signal (59) includes mechanical
actuating means (117).

5

     10. A load responsive valve assembly as set
forth in claim 8 wherein said means (58) responsive to
an external control signal (59) includes fluid pressure
actuating means (120,121).

10

     11. A load responsive valve assembly as set
forth in claim 8 wherein said means (58) responsive to
an external control signal (59) includes electro-
hydraulic actuating means (127).

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     12. A load responsive valve assembly as set
forth in claim 8 wherein said means (58) responsive to
an external control signal (59) includes electro-
mechanical actuating means (124).

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13. A load responsive valve assembly
comprising a housing (21) having an inlet chamber (22)
connected to a pump (10), a supply chamber (23)
connected to a fluid motor (15), and exhaust means

5   (12), control orifice means (14) interposed between·
said supply chamber (23) and said fluid motor (15),
first and second control chambers (24,37) in said
housing, first valve means (27) having fluid throttling
means (31) between said inlet chamber (22) and said

10   supply chamber (23) provided with means (29) responsive
to pressure in said first control chamber (24), and
pilot valve means (38) operable to control pressure in
said first control chamber (24) having means (40)
responsive to pressure in said second control chamber

15   (37) and to pressure in said supply chamber (23), said
first valve means (27) operable to throttle fluid flow
from said inlet chamber (22) to said supply chamber
(23) to maintain a constant pressure differential at a
preselected constant level between said supply chamber

20   (23) and said second control chamber (37) and across
said pilot valve means (38) and to maintain a constant
pressure differential across said control orifice means
(14), pressure signal transmitting means (19) operable
to transmit control pressure signal from upstream of

25   said control orifice means (14) to said second control
chamber (37) and modifying means (13b,63b,82) of said
control pressure signal operable through said first
valve means (27) to vary the level of said constant
pressure differential controlled across said control

30   orifice means (14) while said pressure differential
between said supply chamber (23) and said second
control chamber (37) remains constant at said constant
predetermined level.

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14. A load responsive valve assembly as set
forth in claim 13 wherein said modifying means (13b of
Fig. 1) of said control pressure signal has means
(20,48,55) to vary the level of said constant pressure
5   differential across said control orifice means (14)
below the level of said pressure differential between
said supply chamber (23) and said second control
chamber (37) maintained constant at said constant
predetermined level.
10

15. A load responsive valve assembly as set
forth in claim 13 wherein said modifying means (13b) of
said control pressure signal includes constant pressure
reducing means (48), orifice means (20) upstream of
15   said constant pressure reducing means, and flow orifice
means (57,55) downstream of said constant pressure
reducing means.

16. A load responsive valve assembly as set
20   forth in claim 13 wherein said modifying means (63b) of
said control pressure signal includes flow orifice
means (55,68) and a pressure responsive flow control
means (75,76,78) downstream of said flow orifice means
(55,68).
25

17. A load responsive valve assembly as set
forth in claim 13 wherein said modifying means (82) of
said control pressure signal includes fluid throttling
means (84) and orifice means (61) downstream of said
30   fluid throttling means communicable with said exhaust
means (12).

18. A load responsive valve assembly as set
forth in claim 13 wherein said modifying means
35   (13b,63b,82) of said control pressure signal has means
(58) responsive to an external control signal (59).

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19.   A load responsive valve assembly
comprising a housing (21) having an inlet chamber (22)
connected to a pump (10), a supply chamber (23)
connected to a fluid motor (15), and exhaust means,
5   control orifice means (14) interposed between said
supply chamber (23) and said fluid motor (15), first
(24), second (37) and third (36) control chambers in
said housing, first valve means (27) having fluid
throttling means (31) between said inlet chamber (22)
10   and said supply chamber (23) provided with means (29)
responsive to pressure in said first control chamber
(24) and pilot valve means (38) operable to control
pressure in said first control chamber (24) having
means responsive to pressure in said second control
15   chamber (37) and said third control chamber (36), said
first valve means (27) operable to throttle fluid flow
from said inlet chamber (22) to said supply chamber
(23) to maintain a constant pressure differential at a
preselected constant level between said third (36) and
20   said second (37) control chambers and across said pilot
valve means (38) and to maintain a constant pressure
differential across said control orifice means (14),
first pressure signal transmitting means (19) operable
to transmit control pressure signal from upstream of
25   said control orifice means (14) to said second control
chamber (37), second pressure signal transmitting means
(17) operable to transmit control pressure signal from
said supply chamber (23) to said third control chamber
(36), and modifying means (13b,63b,82) of said second
30   control pressure signal operable through said first
valve means (27) to vary the level of said constant
pressure differential controlled across said control
orifice means (14) while said pressure differential
between said third (36) and said second (37) control
35   chambers remains constant at said constant
predetermined level.

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20. A load responsive valve assembly as set
forth in claim 19 wherein modifying means (13b) of Fig.
2) of said second control pressure signal has means
(20,48,55) to vary the level of said constant pressure
5   differential across said control orifice means (14)
above the level of said pressure differential between
said third (36) and said second (37) control chambers
maintained constant at said constant predetermined
level.
10

21. A load responsive valve assembly as set
forth in claim 19 wherein said modifying means (13b) of
said second control pressure signal includes constant
pressure reducing means (48), orifice means (20)
15   upstream of said constant pressure reducing means, and
flow orifice means (57,55) downstream of said constant
pressure reducing means.

22. A load responsive valve assembly as set
20   forth in claim 19 wherein said modifying means (63b) of
said second control pressure signal includes flow
orifice means (55,68) and a pressure responsive flow
control means (75,76,78) downstream of said flow
orifice means.
25

23. A load responsive valve assembly as set
forth in claim 19 wherein said modifying means (82) of
said second control pressure signal includes fluid
throttling means (84) and orifice means (93) downstream
30   of said fluid throttling means (84) communicable with
said exhaust means (12).

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24. A load responsive valve assembly as set
forth in claim 19 wherein said modifying means
(13b,63b,82) of said second control pressure signal has
means (58) responsive to an external control signal

5   (59).

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25. A load responsive valve assembly
comprising a housing (96) having a fluid inlet chamber
(97), a fluid supply chamber (97,23), at least one load
chamber (98 or 99), and exhaust means (12), first valve
5   means (103) for selectively interconnecting said load
chamber (98 or 99) with said supply chamber (97,23) and
aid exhaust means (12), variable orifice means
(107,108) between said supply chamber (97,23) and said
load chamber (98 or 99) operable by said first valve
10   means (103), load pressure sensing means (113,114)
selectively communicable with said load chamber (98 or
99) by said first valve means (103), and fluid
throttling means (27) interposed between said inlet
chamber (22) and said supply chamber (23), control
15   signal transmitting means (115,17) having means (17) to
transmit a first pressure signal from said supply
chamber (23) and means (115) to transmit a second
pressure signal from said load pressure sensing means
(114,113), control means (38,42) of said fluid
20   throttling means (27) having pilot valve means (38)
communicable with said first and said second pressure
signals and operable to throttle fluid flow from said
inlet chamber (22) to said supply chamber (23) to
maintain a relatively constant pressure differential at
25   a constant predetermined level across said pilot valve
means (38) and to maintain a constant pressure
differential across said variable orifice means (14),
and pressure signal modifying means (13b,63b,82) of
said control signal transmitting means (115,17)
30   operable to vary the level of said constant pressure
differential controlled across said variable orifice
means (14) while said pressure differential acting
across said pilot valve means (38) remains constant at
said constant predetermined level.

35

26. A load responsive valve assembly as set
forth in claim 25 wherein said first valve means (103)
has a neutral position in which it blocks said load
pressure sensing means (114,113), said first valve
5   means (103) when displaced from said neutral position
first connecting said load pressure sensing means
(114,113) with said control means (38,42) before
connecting said supply chamber (23) with said load
chamber (98 or 99).
10

27. A load responsive valve assembly as set
forth in claim 25 wherein said pressure signal
modifying means (13b) has means (20,48,55) operable to
vary the level of said constant pressure differential
15   controlled across said variable orifice means (107,108)
below the level of said pressure differential acting
across said valve means.

28. A load responsive valve assembly as set
20   forth in claim 25 wherein said pressure signal
modifying means (13b) includes constant pressure
reducing means (48), orifice means (20) upstream of
said constant pressure reducing means and flow orifice
means (57,55) downstream of said constant pressure
25   reducing means.

29. A load responsive valve assembly as set
forth in claim 28 wherein said orifice means (20)
upstream of said constant pressure reducing means (48)
30   has orifice area adjusting means (55,68).

30. A load responsive valve assembly as set
forth in claim 25 wherein said pressure signal
modifying means (63b) includes flow orifice means
35   (55,68) and a pressure responsive flow control means
(75,76,78) downstream of said flow orifice means.

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31. A load responsive valve assembly as set
forth in claim 25 wherein said pressure signal
modifying means (82) includes fluid throttling means
(84) and orifice means (61) downstream of said fluid
5   throttling means communicable with said exhaust means
(12).

32. A load responsive valve assembly as set
forth in claim 25 wherein said pressure signal
10   modifying means (13b of Fig. 2) has means (48,55)
operable to vary the level of said constant pressure
differential controlled across said variable orifice
means (14) above the level of said pressure
differential acting across said valve means (38).
15

33. A load responsive valve assembly as set
forth in claim 25 wherein said pressure signal
modifying means has means (58) responsive to an
external control signal (59).
20

34. A load responsive valve assembly as set
forth in claim 33 wherein said means (58) responsive to
an external control signal (59) includes mechanical
actuating means (117).
25

35. A load responsive valve assembly as set
forth in claim 33 wherein said means (58) responsive to
an external control signal (59) includes fluid pressure
actuating means (120,121).
30

36. A load responsive valve assembly as set
forth in claim 33 wherein said means (58) responsive to
an external control signal (59) includes electro-
hydraulic actuating means (127).
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37. A load responsive valve assembly as set
forth in claim 33 wherein said means (58) responsive to
an external control signal (59) includes electro-
mechanical actuating means (123).

5

38. A load responsive valve assembly
comprising a housing (96,21) having a fluid inlet
chamber (97) connected to a pump (10), a load chamber
(98 or 99) connected to a fluid motor, exhaust means
10   (12), and load pressure sensing port means (114,113),
first valve means (103) for selectively interconnecting
said load chamber (98 or 99) with said inlet chamber
(97), said exhaust means (12) and said load pressure
sensing port means (114,113), said first valve means
15   (103) having a variable orifice means (107,108) between
said inlet chamber (97) and said load chamber (98 or
99), second valve means (13) communicable with said
load pressure sensing port (114,113) means having fluid
throttling means (27) between said inlet (97) chamber
20   and said pump (10) controllable by a pilot valve means
(38) and operable to throttle fluid flow from said pump
(10) to said inlet chamber (97) to maintain a constant
pressure differential at a preselected constant level
across said pilot valve means (38) and to maintain a
25   constant pressure differential across said variable
orifice means (107,108), and third valve means (13b)
having means (20,48,55) operable through said second
valve means (13) to vary the level of said constant
pressure differential across said variable orifice
30   means (14) while said pressure differential across said
pilot valve means (38) remains constant at said
constant predetermined level.

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AMENDED CLAIMS
(received by the International Bureau on 30 December 1981 (30.12.81))

39. (New) A valve assembly comprising a housing
(21) having an inlet chamber (22) connected to a
pump (10), a supply chamber (23) connected to a fluid
motor (15) and exhaust means (12), control orifice
means (14,107,108) interposed between said supply
chamber (23) and said fluid motor (15), first valve
means (13a,63a) having fluid throttling means (31,32)
between said inlet chamber (22) and said supply
chamber (23) controllable by a pilot valve means
(38,42) and operable to throttle fluid flow from
said inlet chamber (22) to said supply chamber (23)
to maintain a constant pressure differential at a
preselected constant level across said pilot valve
means (38,42) and to maintain a constant pressure
differential across said control orifice means (14,
107,108).

40. (New) A valve assembly as set forth in claim
39 wherein said control orifice means (14,107,108)
has variable area orifice means (V).

41. (New) A valve assembly as set forth in claim
39 wherein said control orifice means (14,107,108)
has means (103) operable to selectively interconnect
said fluid motor (15) with said supply chamber (23)
and with said exhaust means (100,101,12).

42. (New) A valve assembly as set forth in claim
39 wherein second valve means (13b,63b,82) has means
(48,55,75,84,83) operable through said first valve
means (13a,63a) to vary the level of said constant
pressure differential across said control orifice

means (14,107,108) while said pressure differential
across said pilot valve means (38,42) remains con-
stant at said constant predetermined level.

43. (New) A valve assembly as set forth in claim
42 wherein said second valve means (13b) includes
constant pressure reducing means (48,53), orifice
means (20) upstream of said constant pressure reduc-
ing means (48,53) and flow orifice means (57) down-
stream of said constant pressure reducing means
(48,53).

44. (New) A valve assembly as set forth in claim
42 wherein said second valve means (63b) includes
constant flow control means (75,76,78) connected
to said exhaust means (73,74,25,12) and variable
control orifice means (55,68) upstream of said con-
stant flow control means (75,76,78).

45. (New) A valve assembly as set forth in claim
42 wherein said second valve means (82) includes
fluid throttling means (83,84) and orifice means
(93) downstream of said fluid throttling means (83,84)
communicable with exhaust means (12).

46. (New) A valve assembly as set forth in claim
42 wherein said second valve means (13b,63b,82) has
means (18,19,20,37,48,55) to vary the level of said
constant pressure differential across said control
orifice means (14) below the level of said pressure
differential across said pilot valve means (38,42)
maintained constant at said constant predetermined
level.

47. (New)   A valve assembly as set forth in claim
42 wherein said second valve means (13b) has means
(20,36,43,48,55) to vary the level of said constant
pressure differential across said control orifice
means (14) above the level of said pressure differ-
ential across said pilot valve means (38,42) main-
tained constant at said constant predetermined level.

48. (New)   A valve assembly as set forth in claim
42 wherein said second valve means (13b,63b,82) has
means (58) responsive to an external control signal
(59).

49. (New)   A valve assembly as set forth in claim
48 wherein said means (58) responsive to an external
control signal (59) includes mechanical actuating
means (117).

50. (New)   A valve assembly as set forth in claim
48 wherein said means (58) responsive to an external
control signal (59) includes fluid pressure actuating
means (120,121).

51. (New)   A valve assembly as set forth in claim
48 wherein said means (58) responsive to an external
control signal (59) includes electro-mechanical
actuating means (123,124).

52. (New)   A valve assembly as set forth in claim
48 wherein said means (127) responsive to an external
control signal (132,133,134) includes logic circuit
means (131), amplifying means (130) and electro-mech-
anical means (129).

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$$P_1 - P_W = \Delta P_y$$
$$P_1 - P_2 = \Delta P$$
$$P_W - P_2 = \Delta P_x$$
$$\therefore \Delta P_y = \Delta P - \Delta P_x$$

FIG. 1

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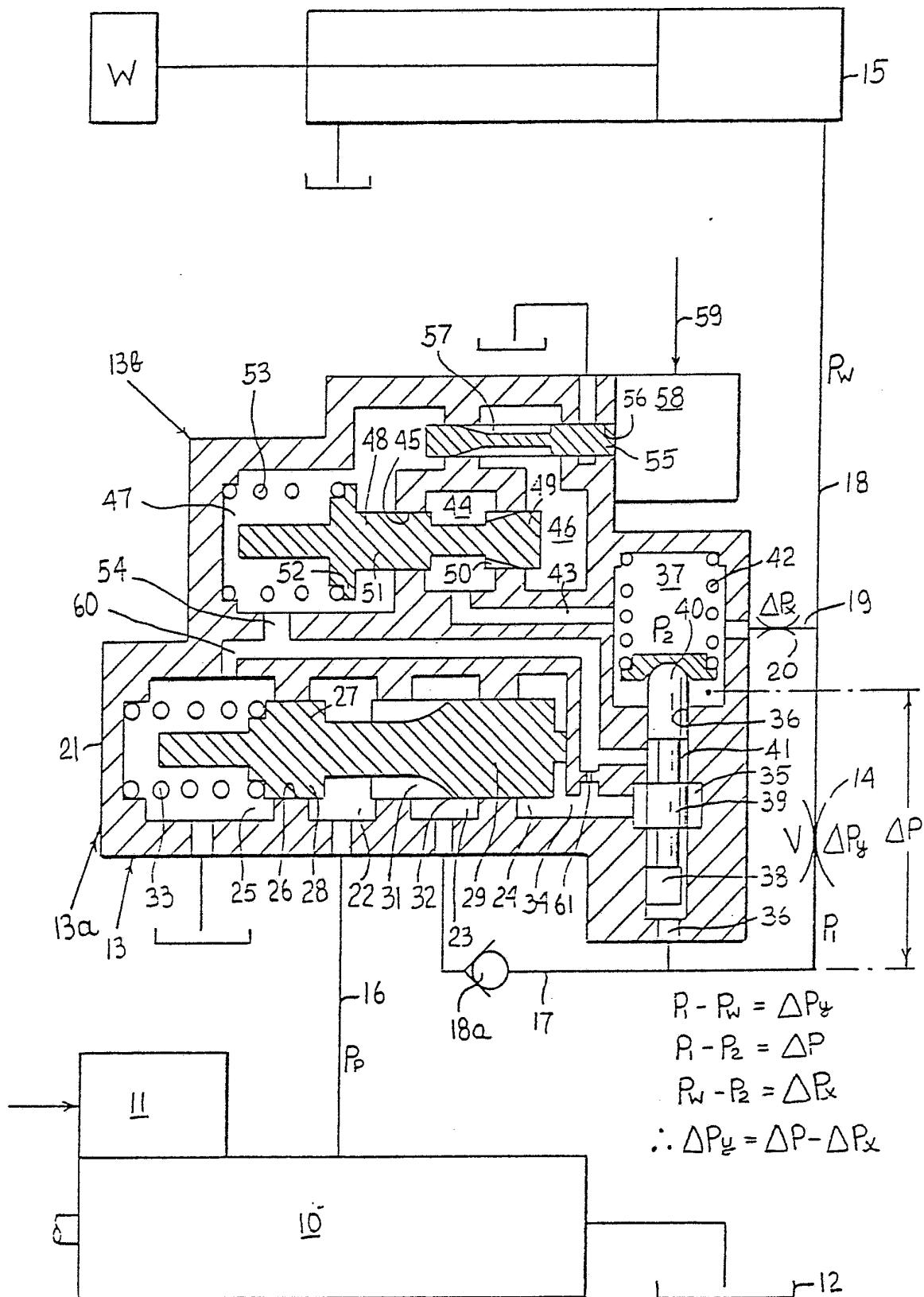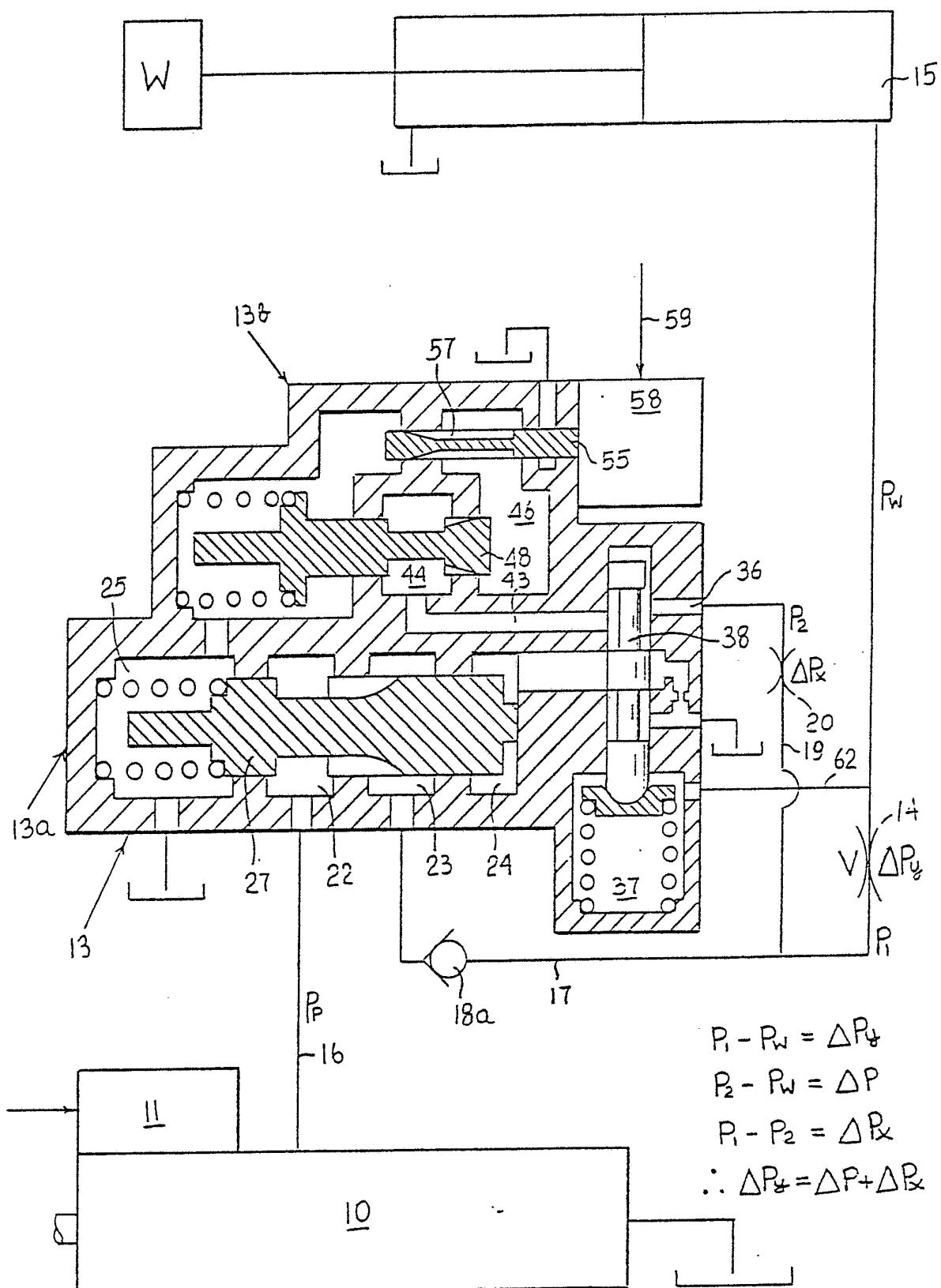
$$P_1 - P_W = \Delta P_y$$
$$P_2 - P_W = \Delta P$$
$$P_1 - P_2 = \Delta P_x$$
$$\therefore \Delta P_y = \Delta P + \Delta P_x$$

FIG. 2

SUBSTITUTE SHEET

$$P_1 - P_w = \Delta P_y$$
$$P_1 - P_2 = \Delta P$$
$$P_w - P_2 = \Delta P_x$$
$$\therefore \Delta P_y = \Delta P - \Delta P_x$$
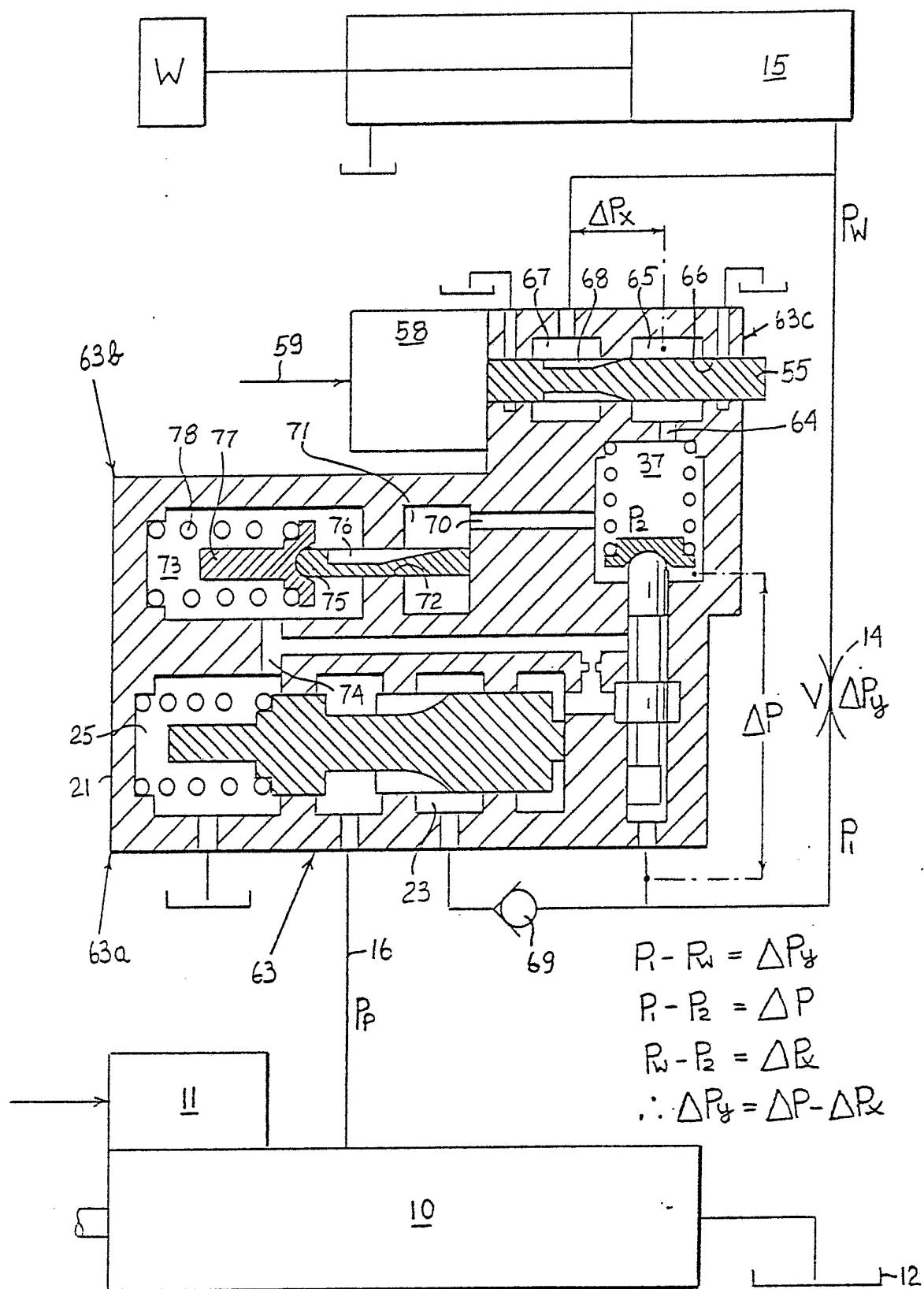
FIG. 3

SUBSTITUTE SHEET
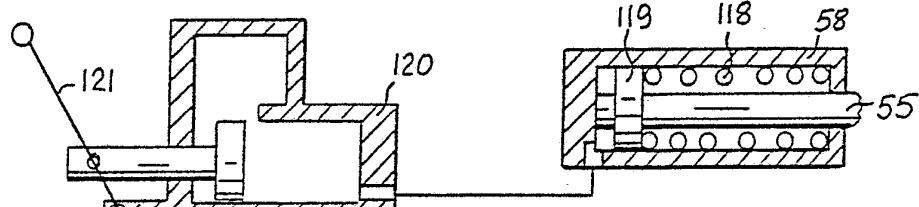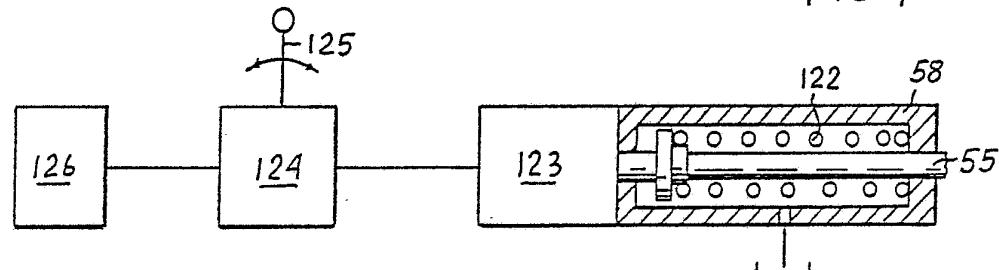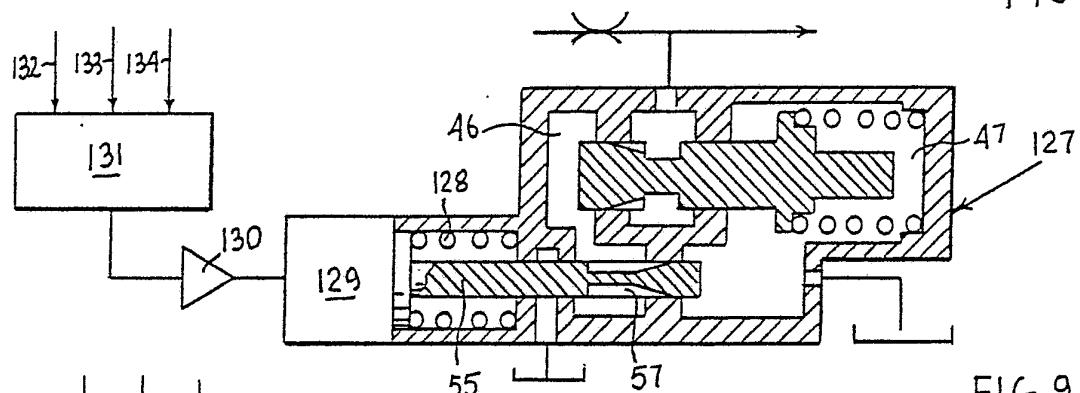
$$P_1 - P_w = \Delta P_y$$
$$P_1 - P_2 = \Delta P$$
$$P_w - P_2 = \Delta P_x$$
$$\therefore \Delta P_y = \Delta P - \Delta P_x$$

FIG. 4

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FIG. 5

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FIG 6

FIG. 7

FIG. 8

FIG. 9
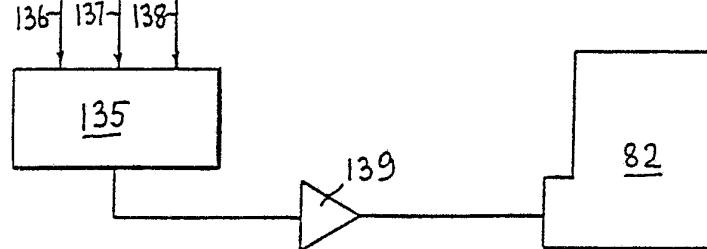
FIG. 10

BUREAU
OMPI
WIFO
INTERNATIONAL

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) ³

According to International Patent Classification (IPC) or to both National Classification and IPC

INT. CL.³ F15 B 13/04

U.S. CL. 91/446, 448; 137/596.13

II. FIELDS SEARCHED

Minimum Documentation Searched ⁴

Classification System	Classification Symbols
U.S.	91/446,448 137/596,596.13

Documentation Searched other than Minimum Documentation
to the Extent that such Documents are Included in the Fields Searched ⁵

III. DOCUMENTS CONSIDERED TO BE RELEVANT ¹⁴

Category *	Citation of Document, ¹⁶ with indication, where appropriate, of the relevant passages ¹⁷	Relevant to Claim No. ¹⁸
A	US, A, 4,153,075, Published 08 May 1979, Budzich	
A	US, A, 4,159,724, Published 03 July 1979, Budzich	
A	US, A, 4,209,039, Published 24 June 1980, Budzich.	
A	US, A, 4,282,898, Published 11 August 1981, Harmon	
A,E	US, A, 4,285,195, Published 25 August 1981, Budzich	

* Special categories of cited documents:¹⁵

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"T" later document published on or after the international filing date or priority date and not in conflict with the application, but cited to understand the principle or theory underlying the invention

"X" document of particular relevance

IV. CERTIFICATION

Date of the Actual Completion of the International Search ²	Date of Mailing of this International Search Report ³
16 October 1981	3 0 OCT 1981
International Searching Authority ¹	Signature of Authorized Officer ²⁰
ISA/US	Gerald A. Michalsky

Form PCT/ISA/210 (second sheet) (October 1977)