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3,234,883

HYDRAULIC INTENSIFIER SYSTEM

Filed July 22, 1964

2 Sheets-Sheet 1

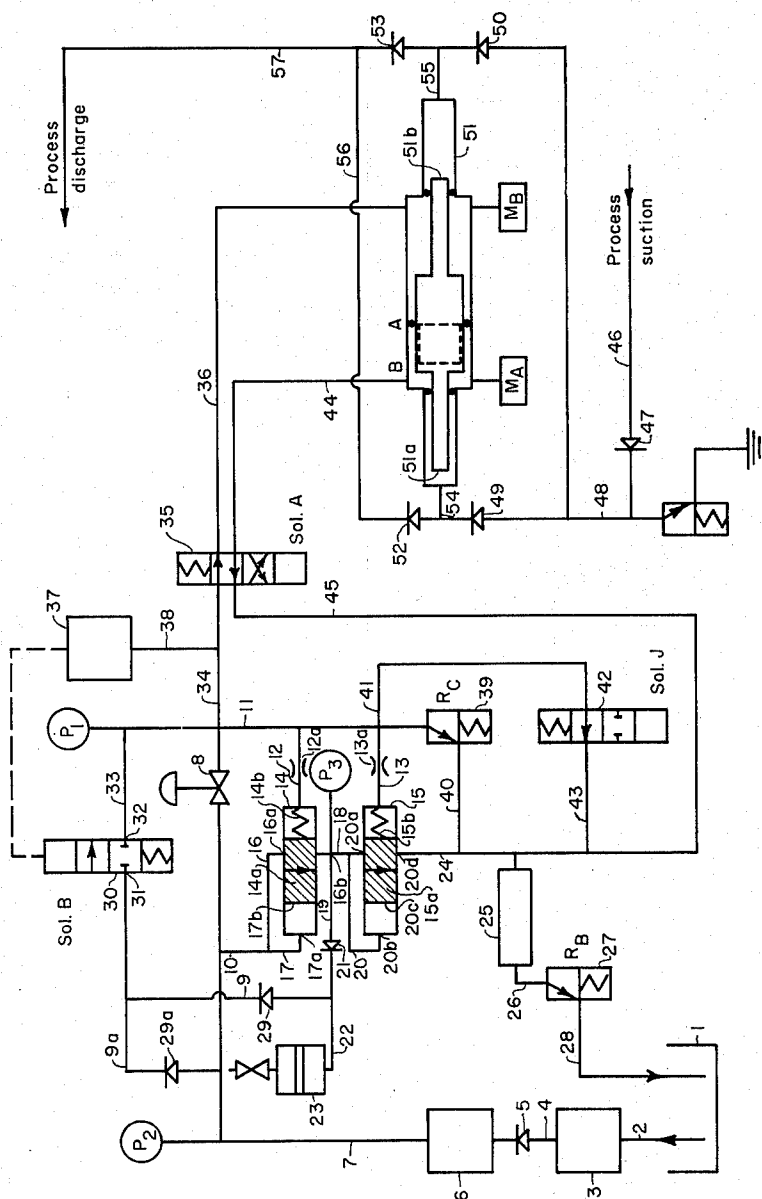


FIGURE 1

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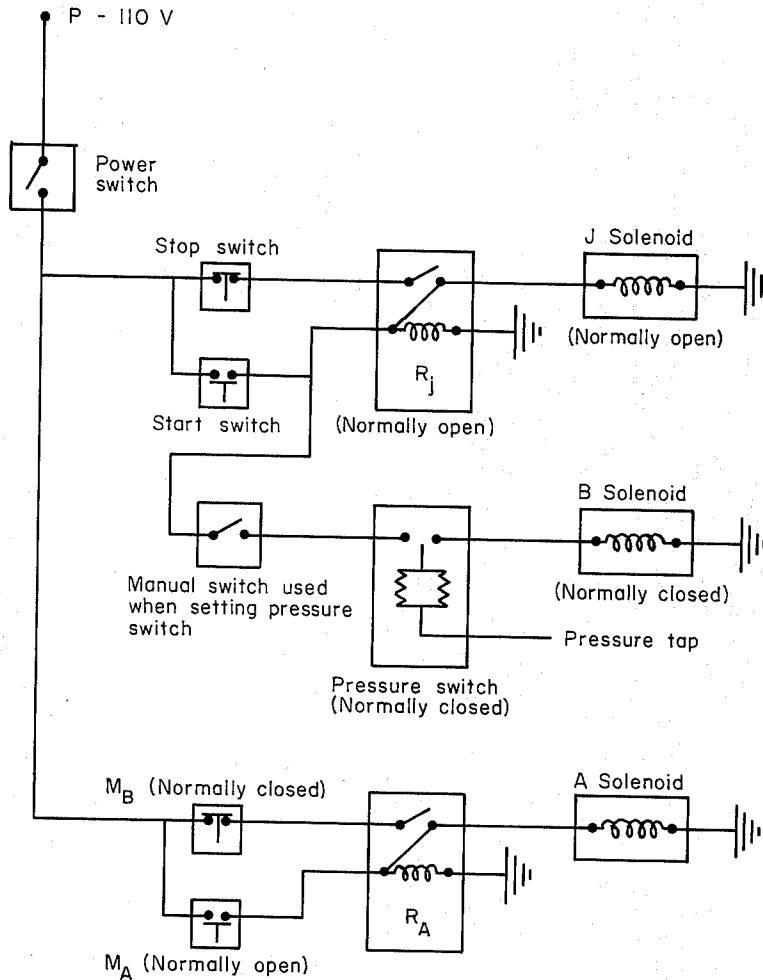


FIGURE 2

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3,234,883

HYDRAULIC INTENSIFIER SYSTEM

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This invention relates to an improvement in an intensifier assembly system and to a method which permits efficient utilization of hydraulic energy to assist in a rapid recompression and continuous delivery of a fluid to a high pressure process area. The invention utilizes hydraulic fluid energy which heretofore had not been used or had been by-passed to a reservoir or sump to assist in the recompression stroke of an intensifier assembly which can be typically a double acting intensifier assembly system.

The prior art as exemplified by U.S. Patents 3,077,838 of Feb. 19, 1963, and 2,819,835 of Jan. 14, 1958, disclose that a hydraulic fluid can be used to drive a series of low pressure piston faces which in turn can be used to force a compressible fluid via a series of smaller piston faces in a substantially continuous manner by proper sequence of discharge strokes. This is accomplished by providing a plurality of intensifiers having low and high pressure piston faces, the intensification being the ratio of the low pressure piston face area to the high pressure piston face area, said low and high pressure piston faces being operated in sequence through various valve arrangements so that a compressible fluid is continuously delivered to a high pressure process area. In particular, U.S. Patent 3,077,838 discloses in combination with at least 4 high pressure piston faces, a pump means, a throttle valve, a compensating valve and a pilot relief and shut-off valve for providing a first and second hydraulic fluid pressure levels for delivery of said hydraulic fluid to the low pressure piston faces, the arrangement being adapted to function in so-called "pipless" fashion. In the specification of this patent, as well as in known methods of operating such an intensifier assembly, however, maximum use of the energy available from the pumping system by way of hydraulic pressure is not utilized with the consequence that a good portion of the energy is by-passed to the reservoir. The instant invention is an improvement over this art and it is an object, therefore, of this invention to utilize more efficiently the energy available for aiding in the hydraulic intensifier assembly operation.

A further object of this invention is to provide a method for operating an intensifier assembly comprising at least two low and high pressure piston faces in rapid recompression and delivery of compressible fluid to a high pressure process area.

It is a further object of this invention to provide a novel method for establishing a series of hydraulic fluid pressure levels and for utilization of said pressure levels in discharge, and recompression strokes.

The foregoing objects can be better understood by reference to the attached drawings which embody a preferred arrangement of the units of the process and the electrical circuitry of this invention.

Essentially, the invention is an improvement in an intensifier assembly which comprises an intensifier unit having at least two low and high pressure piston faces, a high pressure manifold and conduit connections from each said high pressure piston face to said high pressure manifold, conduit means for supplying a compressible fluid to said high pressure piston faces, pump and conduit means for establishing a P_1 hydraulic fluid pressure level, conduit means for establishing a P_2 hydraulic fluid pressure level and conduit means to deliver one of said pressure levels to the low pressure piston faces, differential pressure reg-

ulating valve means for maintaining said P_1 and P_2 pressure levels across an orifice which supplies hydraulic fluid to the low pressure piston faces, the improvement provided herein being the provision of a second differential pressure regulating valve associated with said first valve by conduit means, a further conduit in association with said two differential pressure regulating valves for establishing a third fluid pressure level P_3 , an accumulator for momentarily storing fluid of this level and means for delivering the stored fluid to one of the low pressure piston faces to aid in the recompression stroke thereof.

The process can be briefly summarized, therefore, as an improvement in one for delivering a compressible fluid at an elevated pressure to a high pressure process area wherein said fluid is compressed and discharged sequentially and continuously from at least two enclosed high pressure displacing surface areas of small diameter and relatively non-compressible fluid is provided to drive at least two enclosed low pressure displacing surface areas of larger diameter in association with said high pressure displacing surface areas in a sequence which comprises fluid discharge, suction and recompression steps for each of said low pressure displacing surface areas and wherein a first and second fluid pressure level P_1 and P_2 are maintained for delivery of hydraulic fluid to the discharge and recompression steps of said low pressure displacing surface areas, the second pressure level being maintained above the first pressure level, the said improvement provided by this invention being in the establishment of a third hydraulic fluid pressure level, momentarily storing at least a portion of said fluid at said pressure level and subsequently delivering said fluid from storage with the fluid from one of the above pressure level to the low pressure displacing surface areas during their recompression steps.

The novel combination of units for delivering hydraulic fluid to the low pressure piston faces and the associated intensifier assembly system of this invention is useful in high pressure technology wherein it is desired to deliver a fluid, such as a gas or a compressible liquid to a high pressure area where such gas or liquid is required. In the preferred embodiment of this invention, a liquid is delivered to a high pressure area. Ordinarily, a liquid is not thought of as being a compressible fluid and under ordinary and nominal pressures of up to a few thousand p.s.i. a liquid is relatively incompressible. In high pressure technology, however, where pressures of 20,000 to 50,000 p.s.i. are used to deliver a liquid to a high pressure area, the liquid is compressed, sometimes by as much as from 10 to 20% by volume depending on the particular pressure and liquid used. Thus, at pressures of about 30,000 p.s.i., for example, specifically where a normally liquid hydrocarbon such as hexane or a mixture of benzene with hexane, etc., is used, a 10 to 12% decrease in volume is not unusual in process operations of this type (the compressibility of liquids at high pressures will be simply referred to herein as the compressibility factor or percent volume reduction). The prior art as exemplified above made little provision in the operation of intensifiers to reduce or eliminate the time factor involved in pressuring the liquid for delivery. By this it is understood that there is a time factor involved in the recompression strokes of dual low pressure piston faces wherein the piston, for a very short period of time, is functioning only to compact the compressible liquid prior to its discharge to the high pressure process area. Thus, if means are not provided for substantially reducing the time period mentioned, then the high pressure piston face in question during the recompression stroke momentarily will not deliver liquid to the high pressure process area. To reduce this delay to a negligible amount in delivery of the liquid to the high pressure area therefore, it is desired to speed up the recompression stroke of the piston so that the ensuing discharge stroke

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will not cause a substantial flow or pressure fluctuations on the process side. This invention provides a scheme where, by utilizing energy that is available from a sole or single pumping system, such energy is momentarily stored and routed in such a fashion as to substantially instantaneously provide the extra energy required to reduce the time factor involved by the compressibility factor of the liquid and thereby assure continuous or substantially continuous delivery from the high pressure piston face areas.

Reference is herein made to the attached drawings, FIGURES 1 and 2, which illustrate the preferred intensifier assembly arrangement (FIGURE 1) and the electrical circuitry in association therewith (FIGURE 2).

In FIGURE 1, vessel denotes a sump tank or reservoir. From this tank, pump 3 delivers hydraulic fluid (oil) via conduit 2 to conduit 4 through check valve 5, filter 6 and conduit 7 to the process as will be described herein. A flow control or variable orifice valve 8 is provided on conduit 7. Upstream of this valve, conduits 9 and 10 are provided for operation of the novel technique of this invention. Conduit 11 downstream of flow control valve 8 functions in association with the flow valve and conduits 12 and 13 lead to differential pressure regulating valves indicated generally at 14 and 15. Hydraulic fluid flow from conduits 7 and 10 can be diverted to conduits 16 or 17 of differential pressure regulating valve 14 and thence to inlet ports 16a or 17a and outlet port 16b to conduit 18 and thence to lines 19 or 20. The hydraulic fluid from conduit 19 can be taken via check valve 21, line 22 and to accumulator 23. Excess hydraulic fluid is by-passed via inlet port 20a to differential pressure regulating valve 15 or inlet port 20b and thence to outlet port 20c and conduit 24. From line 24 the fluid can be by-passed through heat exchanger 25, line 26, relief valve 27 and line 28 back to the sump. As will be described in more detail herein, the fluid from accumulator 23 is taken via check valve 29 to line 9 and thence to 2-way solenoid operated valve 30 via inlet port 31 and outlet port 32 to line 33. Line 33 joins line 34 from the upstream flow control valve 8 and thence to 4-way solenoid operated valve 35, line 36 through a suitable inlet port (not shown) leading directly to either low pressure piston face A or low pressure piston face B of intensifier assembly 51. The intensifier which is depicted generally in the drawing is described in more detail in U.S. Patent 3,077,838, FIGURE 2, and is typical of those intensifiers which can be used in herein. It is a double acting intensifier and for convenience here, the two integrated units will be referred to as Drive A and Drive B.

Two-way solenoid operated valve 30 is electrically connected (dotted lines) to pressure switch 37 and line 38 in association with line 34. The pressure switch is typical of the type that is adapted to sense a change in pressure in line 36 and electrically operates to open or close the solenoid of 3-way valve 30 depending on the pressure setting.

Downstream of flow control valve 8, line 11 leads to relief valve 39 which is set at a certain maximum pressure which is ordinarily not exceeded by the hydraulic system. If the pressure setting of the relief valve should be exceeded, then it would open to spill hydraulic fluid via line 40 to line 24, heat exchanger 25, etc. Line 41 in association with the first and second differential pressure regulators, as well as line 11, is used to divert flow through this line and via two-way solenoid operated valve 42 to the sump in the event that the stop switch is deactivated by an interlock system or by manual means. Valve 42 is normally open as indicated in the drawing and is activated (closed) during the operation of the pump. It is also opened or deactivated by turning off the control circuit power supply or pressing the stop switch at a control station. The hydraulic fluid during the suction stroke of either of the low pressure piston faces is discharged by lines 44 or 36 through 4-way solenoid operated valve 35, line 45 and to heat exchanger 25, etc. Process fluid

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or liquid is delivered to the high pressure piston faces 51A or 51B via line 46 through check valve 47, line 48 and check valves 49 and 50. Check valves 52 and 53 are associated with the discharge lines 54 and 55 from the discharge of the high pressure piston face areas and thence to lines 56 and 57 on to the process. As noted, the intensifier assembly is provided with microswitch MA for Drive A and MB for Drive B as indicated in the drawing.

The differential pressure regulating valves 14 and 15 are typically enclosed cylindrical chambers containing therein movable spool valves 14a and 15a which move in slidable contact in said chambers. Spring force constants 14b and 15b are provided in said chambers to oppose the spool valve faces as indicated in the drawing. Pressure differential regulating valves of this type are illustrated in U.S. Patent 3,077,838. Restrictions 12a and 13a are provided in conduits 12 and 13 leading to the chamber of differential pressure valves 14 and 15. The operation and function of these two valves will be described hereinbelow.

In the preferred embodiment of this invention, the spring force constant 14b per unit area is approximately twice the spring force constant 15b per unit area of differential pressure regulating valves 14 and 15. The inlet and outlet ports and fluid flow therethrough, as well as the various pressures acting on the faces of the spool valve dynamically balance the valve in accordance with the pressures acting thereon. In the arrangement shown in the drawing, by virtue of the establishment of hydraulic fluid pressure levels P_1 and P_2 as indicated in the drawing across flow control valve 8 in association with valve 14, the hydraulic fluid pressure P_1 and the spring force 14b normally urge the spool valve 14a to a closed position. The differential pressure regulating valves 14 and 15 are of the normally closed type, that is, their inlet ports 16a and 16b are normally closed by the pressure of springs 14b and 15b. The arrangement illustrated in the drawing as stated establishes differential pressure levels P_1 and P_2 across valve 8 with P_1 being lower than P_2 by at least the spring force per unit area 14b. When the fluid pressure from lines 10 and 17 exerted against spool valve face 17b is greater than the P_1 pressure plus the spring force, that is, 14b, the spool valve will move to the right to open inlet port 16a wider, spilling the excess flow to line 18. And, vice versa, when the P_1 fluid pressure and spring force 14b are greater than the P_2 fluid pressure level, the spool valve will move to the left to close inlet port 16a. In operation, the P_2 pressure level is normally greater than the P_1 pressure level, so excess flow is continuously spilling (throttling) to the P_2 fluid pressure level via differential pressure valve 14. The P_3 hydraulic fluid pressure level is controlled by differential pressure regulator valve 15 in substantially the same manner as the P_2 fluid pressure level is controlled by valve 14, except that in this instance, due to the nature of the spring force constant 15b, the value for this P_3 fluid pressure level is approximately between that of the P_1 and the P_2 fluid pressure levels, that is, if the P_1 fluid pressure level is 1000 and the P_2 level is 1100, the P_3 fluid pressure level is 1050. In this case, therefore, when the P_1 fluid pressure level and spring force 15b acting on spool valve face 13a is greater than the pressure exerted via line 20 through inlet 20b, acting on spool valve face 20c, the valve will move to the left closing inlet port 20a. When the fluid pressure in line 20 is greater than the spring force, that is, 15b and P_1 fluid pressure levels, the P_3 fluid pressure level will then spill via outlet port 20d to line 24. The accumulator 23, it should be noted, is directly connected to the P_3 level, so the fluid pressure stored therein is of this character. Further reference will be made to this accumulator in a brief description of the operation of the system.

In operation, assuming the position indicated in the drawing for Drives A and B, the high pressure piston face of Drive A is nearing the end of its discharge stroke. High pressure piston face B of Drive B is nearing the

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end of its suction stroke. Under these conditions the following occurs. The solenoid operated 4-way valve 35 is ported as shown in the drawing delivering hydraulic fluid to the low pressure piston face A of Drive A. A fixed differential pressure is also being maintained across the flow control valve 8 by the first differential pressure regulator 14 as explained above. The differential pressure maintained is equal to P_2 minus P_1 (pressure upstream of the flow control valve 8 less the pressure downstream thereof, that is, this value is at least equal to the value of the spring force, that is, 14b). The excess flow from the hydraulic pump is therefore diverted through the first differential pressure regulator to a third or P_3 hydraulic fluid pressure level in conduit 18. During this time the P_3 pressure level charges the accumulator 23 and after this, the excess pressure and flow is diverted through the second differential pressure regulating valve 15 (as indicated above, the P_3 pressure level is approximately half way the P_1 and P_2 values by virtue of the difference in the spring force per unit area, the second spring 15b being approximately one-half the value of the first 14b per unit area).

The flow being diverted through differential pressure regulating valve 15 is further diverted through line 24 through the heat exchanger and the relief valve 27 back to the sump. During the above operation, the solenoid valve 30 is in the normally closed position so that the accumulator maintains its charge until needed in the downstream units.

At the completion of the discharge stroke of Drive A the microswitch MA is momentarily activated due to contact of a Drive A projection (not shown) with the microswitch. In so doing, it reverses the 4-way solenoid operated valve 35 diverting the hydraulic supply P_1 fluid pressure from low pressure piston face A to low pressure piston face B. At the moment of reversal, the hydraulic supply pressure level P_1 decreases during the beginning of the recompression stroke of Drive B. At this time, the pressure switch 37 senses this pressure drop in line 34 and makes electrical contact and opens the by-pass solenoid valve 30. This diverts the maximum output of the hydraulic pump from line 7 plus the accumulator capacity (supply) through line 9 and valve 30 to the low pressure piston face B for rapid recompression. When the P_1 fluid pressure level reaches the pressure setting of the pressure switch 37, the solenoid valve 30 is deactivated and therefore closed, thus diverting the hydraulic oil flow through the control valve 8 as before.

During the period just prior to the opening of solenoid valve 30 (after the 4-way valve 35 has reversed, but prior to the actuation of pressure switch 37), both differential valves 14 and 15 have a tendency to open as P_1 pressure decreases, thus check valves 21 and 29a have been incorporated in lines 19 and 19a to prevent the unloading of the accumulator back to the sump tank through the opened differential valves. After the pressure switch 37 and solenoid valve 30 have been activated, then the differential valves 14 and 15 close until after the pressure switch 37 and solenoid 30 are deactivated by pressure P_1 .

The accumulator, which can be of any suitable configuration, is sized in order to optimize a conservation of energy and to achieve the shortest recompression time it is necessary to size it relative to the maximum pump capacity per stroke to compensate for the compressibility of the process fluid in use as hereinbefore alluded. If the compressibility factor of the process fluid is 10% by volume at a certain pressure, for example, then the accumulator is designed to store substantially this 10% by volume for delivery to the low pressure piston face during its recompression stroke. Consequently, if the accumulator is designed too small, it will not store sufficient energy to rapidly recompress the process fluid at maximum efficiency. If the accumulator capacity is made too large, then its stored P_3 fluid pressure level would not drop below the P_1 fluid pressure level and it might

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generate a non-flow control condition entering the process, especially if the response time of the pressure switch 37 and the solenoid valve 30 is slow. Therefore, the accumulator is normally precharged to a given pressure level. At any time the hydraulic system is normally operating below the precharged level, the accumulator does not respond to its function of rapid recompression. As indicated, therefore, it is necessary to maintain a precharged pressure level for the accumulator which will cover the normal minimum pressure range of the intensifier. Accumulators of the type illustrated in the drawing are old in the art and no detailed explanation of their construction need be included here. Suffice it to say, however, that as indicated generally in the drawing, a compressible inert gas such as nitrogen can be used above a piston area in an enclosed chamber so that hydraulic fluid will cause the movement of the piston to act against the pressure of the inert gas. In sizing the capacity of the accumulator for any given pressure operation on the process side, the inert gas pressure can be increased or decreased to correspond with the capacity (at the P_3 fluid pressure level) desired.

It can be appreciated from the foregoing that by virtue of this arrangement, a smaller hydraulic pump in association with an accumulator can be used for delivering a comparable quantity of fluid to the low pressure piston face for a given intensifier than without an accumulator. Conversely, a given hydraulic pump in association with an accumulator and differential pressure valve scheme can increase the potential capacity of an intensifier when rapid recompression is required.

Turning to FIGURE 2, the electrical circuits for operation of the assembly of FIGURE 1 are herein illustrated. The power switch is a normal 110 volt, 60 cycle arrangement. Under the conditions indicated above for FIGURE 1 for the hydraulic system, the electrical system functions as follows. To start the pump it was necessary to close the Power Switch and activate the Starting Switch for the J solenoid (which is normally open; NO and NC in the drawing indicating "normally open" and "normally closed" respectively) via the activation of the RJ holding coil. The Manual Switch at this point is in the closed position and the Pressure Switch (37 in FIGURE 1) which is normally closed, is in the open position because the P_1 pressure level is higher than the Pressure Switch setting for Pressure Switch 37 of FIGURE 1. Thus, in this instance the B solenoid of valve 30 of FIGURE 1 is in the normally closed position. The Pressure Tap in association with the Pressure Switch is placed on line 34 of FIGURE 1 for the purpose heretofore indicated.

On completion of Drive A, discharge stroke microswitch MA (which is in the normally open position) is momentarily closed, thus activating the holding coil RA and thus activating solenoid A of 4-way reversing valve 35. Moments after the reversal of the 4-way solenoid valve 35 by activation of the RA holding coil and solenoid A of valve 35, the P_1 pressure level decreases as indicated above and this is sensed by the Pressure Switch which closes, thus activating the 2-way solenoid valve B of valve 30 which begins the recompression cycle. Just prior to the completion of the recompression cycle, the P_1 fluid pressure level again increases thereby opening the Pressure Switch 37 of FIGURE 1 and this deactivates the solenoid B of valve 30 thereby diverting the oil flow through the flow control valve 8. On the completion of the discharge stroke of Drive B microswitch B (normally closed) is momentarily opened, thus de-energizing the holding coil relay RA and deactivating the 4-way solenoid valve 35 as before. Further sequences are a repetition of the above description.

The foregoing arrangement of intensifier assembly units is seen at once to possess unique advantages over known systems. Thus, by virtue of the establishment of the P_3 fluid pressure, its storing in an accumulator and use

during the recompression stroke of one of the low pressure piston faces, a more uniform delivery of high pressure process fluid is herein attained so that only a single double acting intensifier unit is required to substantially approach "pipless" operation. In the prior art at least two double acting units or four single acting units were required to be operated in sequence to achieve substantially "pipless" operation. By suitable arrangement of the discharge conduits of the low pressure piston faces of two single acting intensifier units so that the discharge fluid from one low pressure piston surface will force the suction stroke of the other, the system can be adapted for operation of two such single acting units in the manner encompassed by the invention herein. Also it is apparent that if desired more than two high and low pressure piston faces can be operated as above illustrated.

All of the foregoing units exemplified in FIGURES 1 and 2 are standard in the industry and the novel arrangement of certain units in establishing a P_3 fluid pressure level and momentarily storing it and thereafter delivery it for rapid recompression of Drives A or B comprise a part of this invention. Various modifications can be made to the foregoing preferred method and assembly of the units indicated herein without departing from the scope of this invention.

What is claimed is:

1. In an intensifier assembly comprising an intensifier unit having at least two low and high pressure piston faces, a high pressure manifold and conduit connections from each said high pressure piston face to said high pressure manifold, conduit means for supplying a compressible fluid to said high pressure piston faces, pump and conduit means for supplying a P_1 and P_2 hydraulic fluid pressure level, a first differential pressure regulating valve means for maintaining said P_1 and P_2 pressure levels across an orifice in association with a conduit means for supplying hydraulic fluid to said low pressure piston faces, the combination of a second differential pressure regulating valve, conduit connections from said first to said second valve for by-passing hydraulic fluid in excess of the P_2 pressure level, a conduit leading from conduit connections to a further conduit for establishing a third pressure level P_3 from said by-passed fluid from the P_2 pressure level, and conduit means for leading said P_3 pressure level to an accumulator for momentarily storing hydraulic fluid of said P_3 level, conduit connections between said accumulator and a by-pass conduit and valve for directly delivering said P_2 pressure level plus said accumulator P_3 pressure level hydraulic fluid to one of said low pressure piston faces in response to a pressure sensing means downstream of said orifice during the recompression stroke of one of said low pressure piston faces.

2. The combination of claim 1 wherein the volume of P_3 pressure level momentarily stored in said accumulator is directly proportional to the compressibility factor of the compressible fluid delivered to said high pressure piston faces.

3. The combination of claim 1 wherein the P_3 pressure level is greater than the P_1 pressure level by an amount approximately half-way between the difference in the P_1 and P_2 pressure levels.

4. The combination of claim 1 wherein said first and second differential pressure regulating means comprise enclosed cylindrical chambers, slidable spool valves therein and spring force constants of different force values.

5. The combination of claim 4 wherein the spring force constant of said first valve is approximately twice the value of said second valve spring force.

6. The combination of claim 4 wherein the intensifier is a double acting intensifier system.

7. In a process for delivering a compressible fluid at an elevated pressure to a high pressure process area wherein said fluid is compressed and discharged sequentially and continuously from two enclosed high pressure displacing surface areas of small diameter and wherein a non-compressible fluid is provided to drive two enclosed low pressure displacing surface areas of larger diameter in association with said high pressure displacing surface areas, in a sequence comprising fluid discharge, suction and recompression steps for each of said low pressure displacing surface areas, and wherein first and second fluid pressure levels are maintained for delivery to the discharge and recompression steps of said low pressure displacing surface areas, said second pressure level being maintained above said first pressure level, the improvement which comprises establishing a third fluid pressure level, momentarily storing at least a portion of the fluid at said third pressure level and subsequently delivery said fluid pressure from storage with the fluid from said second pressure level to one of said low pressure displacing surface areas during the recompression step thereof.

8. The improvement in the process of claim 7 wherein the portion of the third fluid pressure level momentarily stored is directly proportional to the compressibility factor of the compressible fluid at the delivery pressure to the high pressure process area.

9. The improvement in the process of claim 7 wherein the value of said third fluid pressure level lies approximately half-way between the values of the first and second fluid pressure levels.

10. The improvement in the process of claim 7 wherein said first, second and third pressure levels are derived from a single hydraulic fluid pressure delivery supply.

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