United States Patent [19]

Mayer

[11] **3,893,790**

[45] **July 8, 1975**

[54] **DUAL SINGLE ACTION RAM INTENSIFIER**

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[22] Filed: **May 29, 1973**

[21] Appl. No.: **364,595**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 139,098, April 30, 1971, abandoned.

[52] **U.S. Cl.** **417/346**; 60/371

[51] **Int. Cl.**[2] **F04B 17/00**; F16D 31/02

[58] **Field of Search** ..... 417/225, 226, 344–347, 417/342; 60/97 E, 371, 378

[56] **References Cited**

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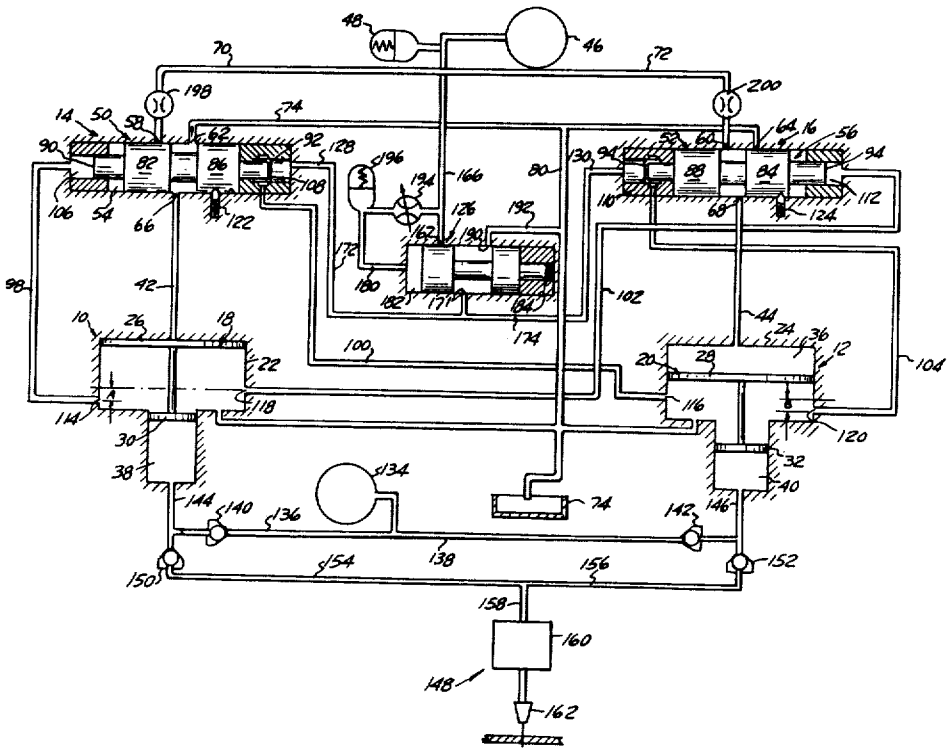
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[57] **ABSTRACT**

A ram type intensifier capable of producing a continuous flow of liquid at pressures on the order of 70,000 psi is disclosed, comprised of a pair of alternately and cyclically stroked ram cylinders, in which the flow of pressurized operating fluid to the ram chambers is from a constant pressure source controlled such that simultaneous pressurization of both ram chambers is provided at the end of each cycle so that precompression of the cylinder beginning its pressure stroke is carried out before depressurization of the cylinder ending its pressure stroke. This arrangement prevents pressure variations occurring as a result of the substantial compression of the liquid occurring at these pressure levels.

1 Claim, 9 Drawing Figures



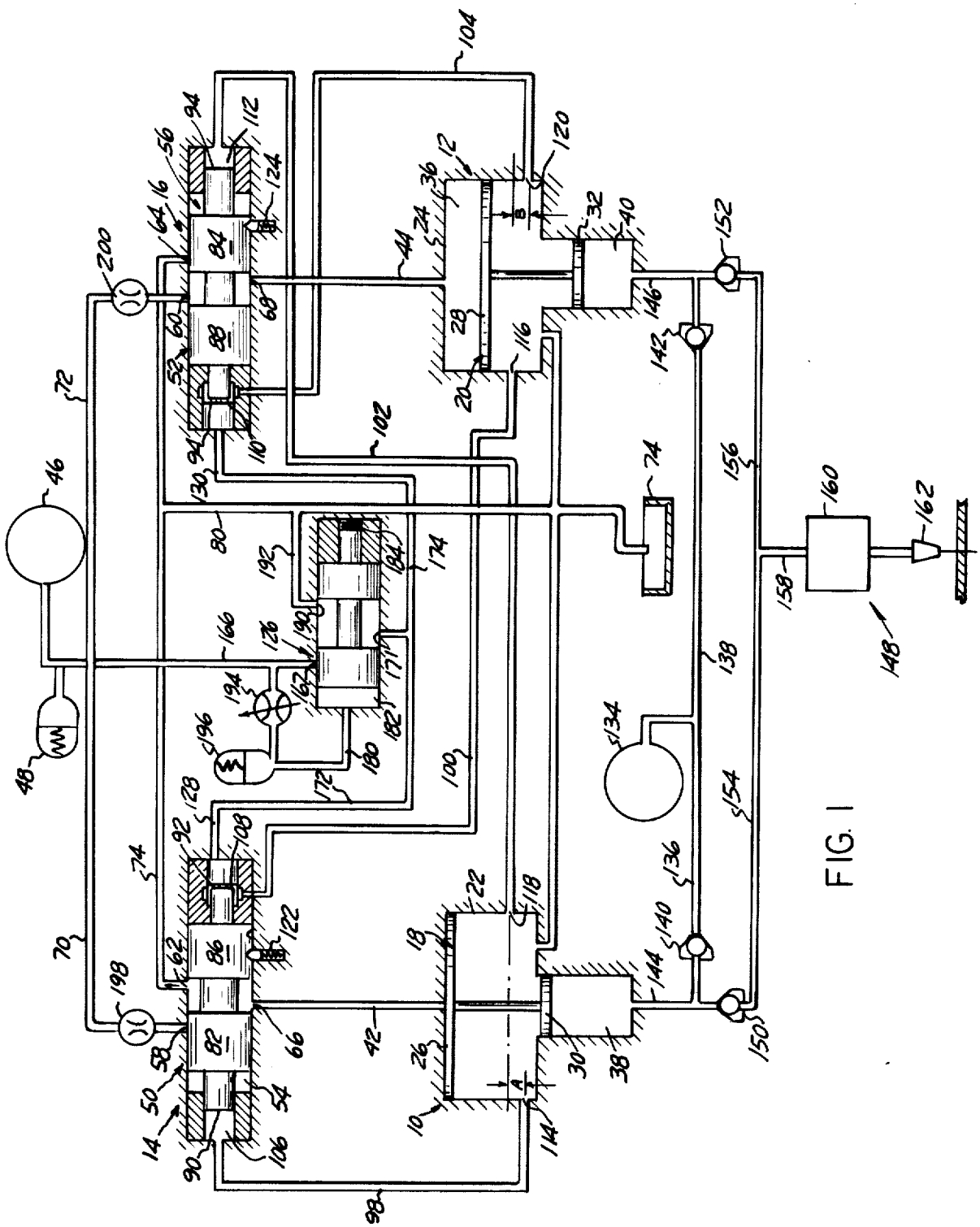


FIG. 1

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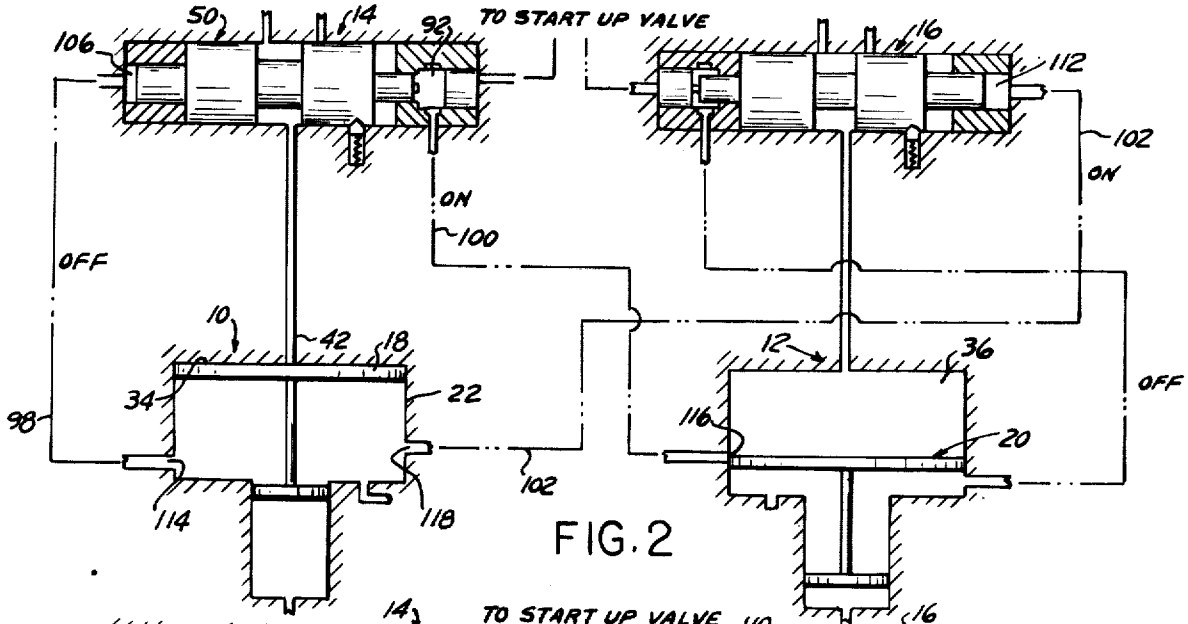


FIG. 2

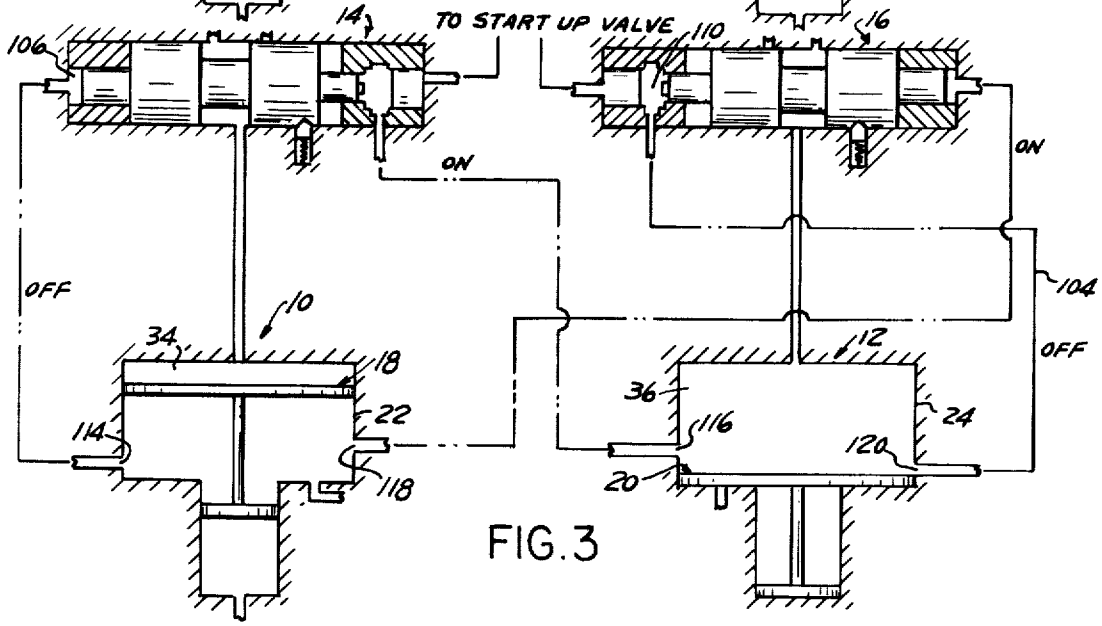


FIG. 3

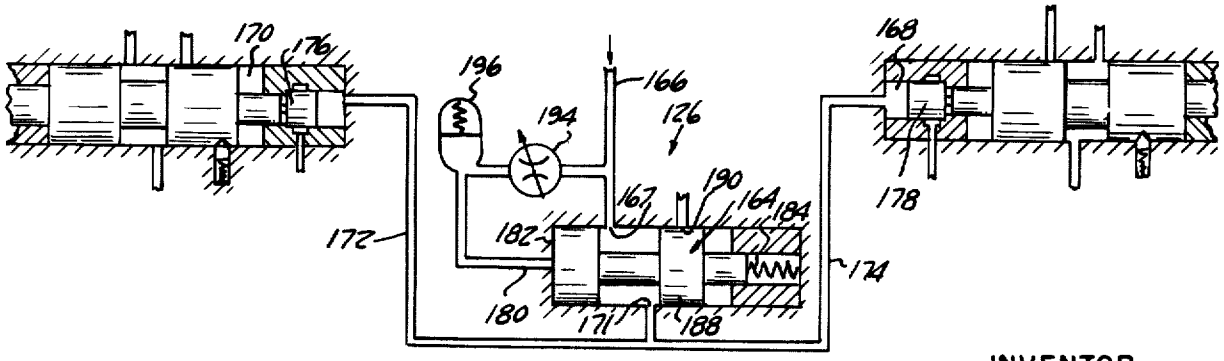


FIG. 4

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FIG. 5

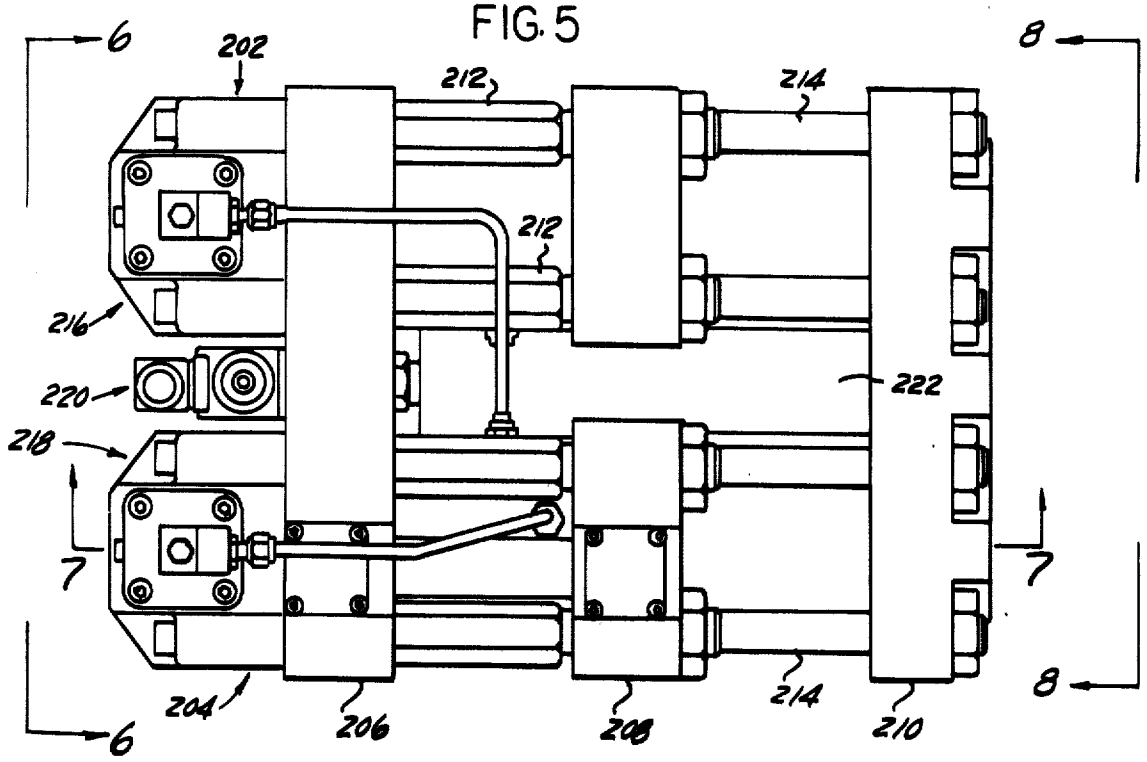
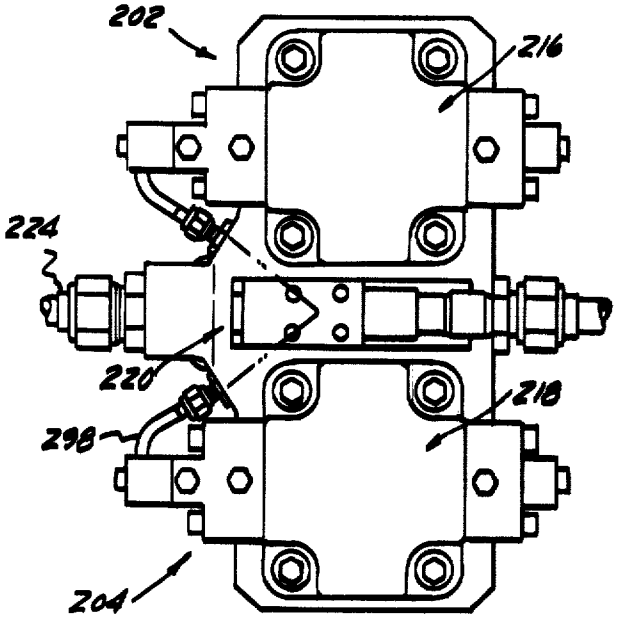


FIG. 6



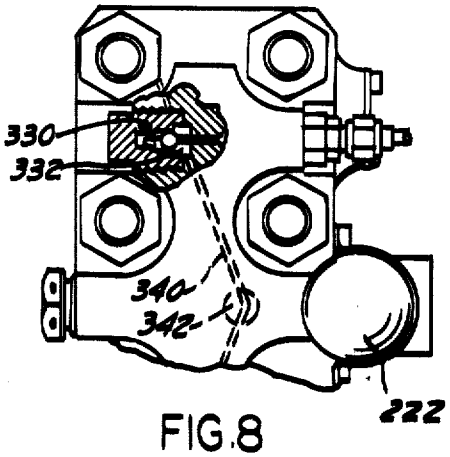


FIG. 8

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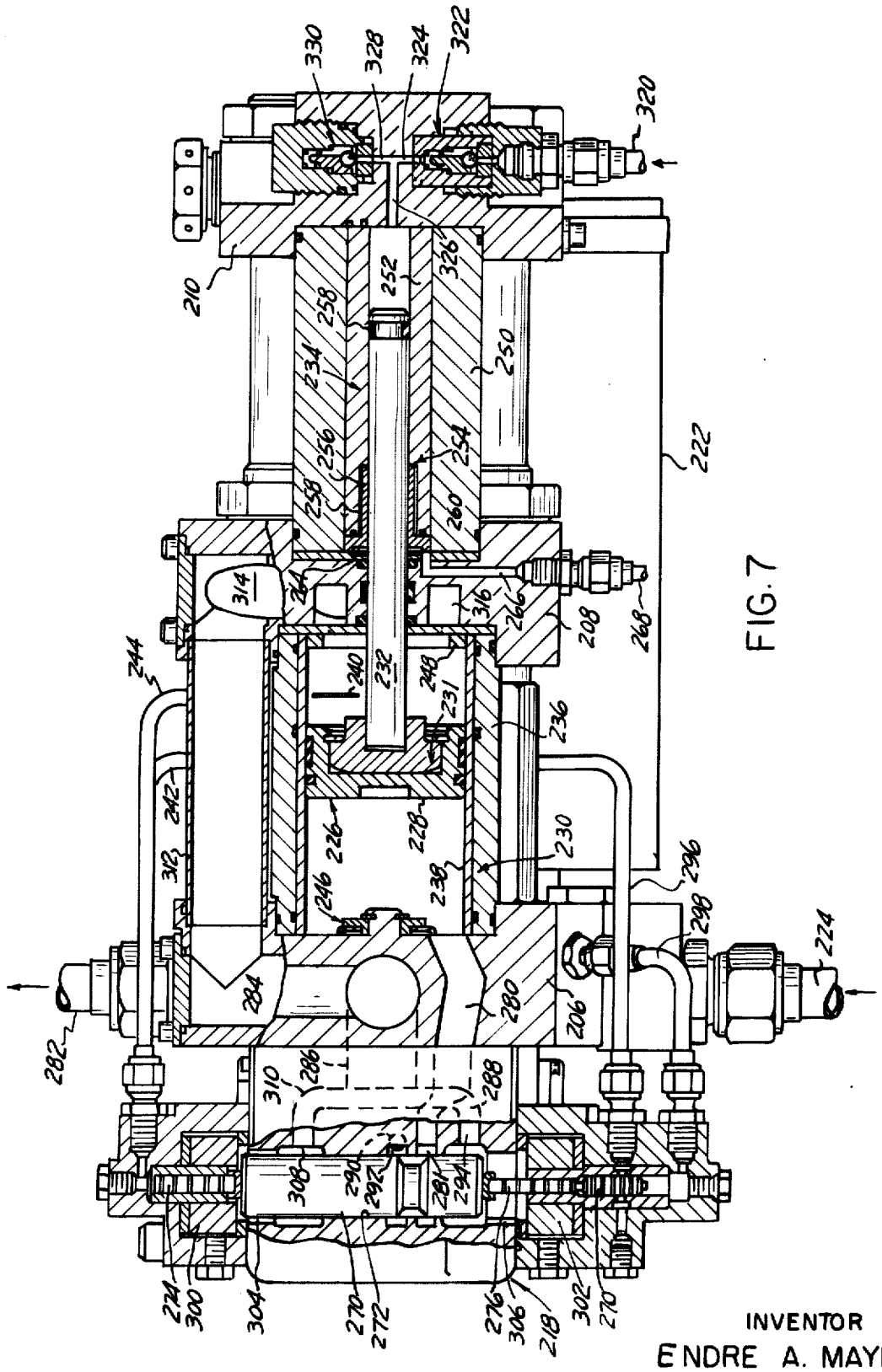
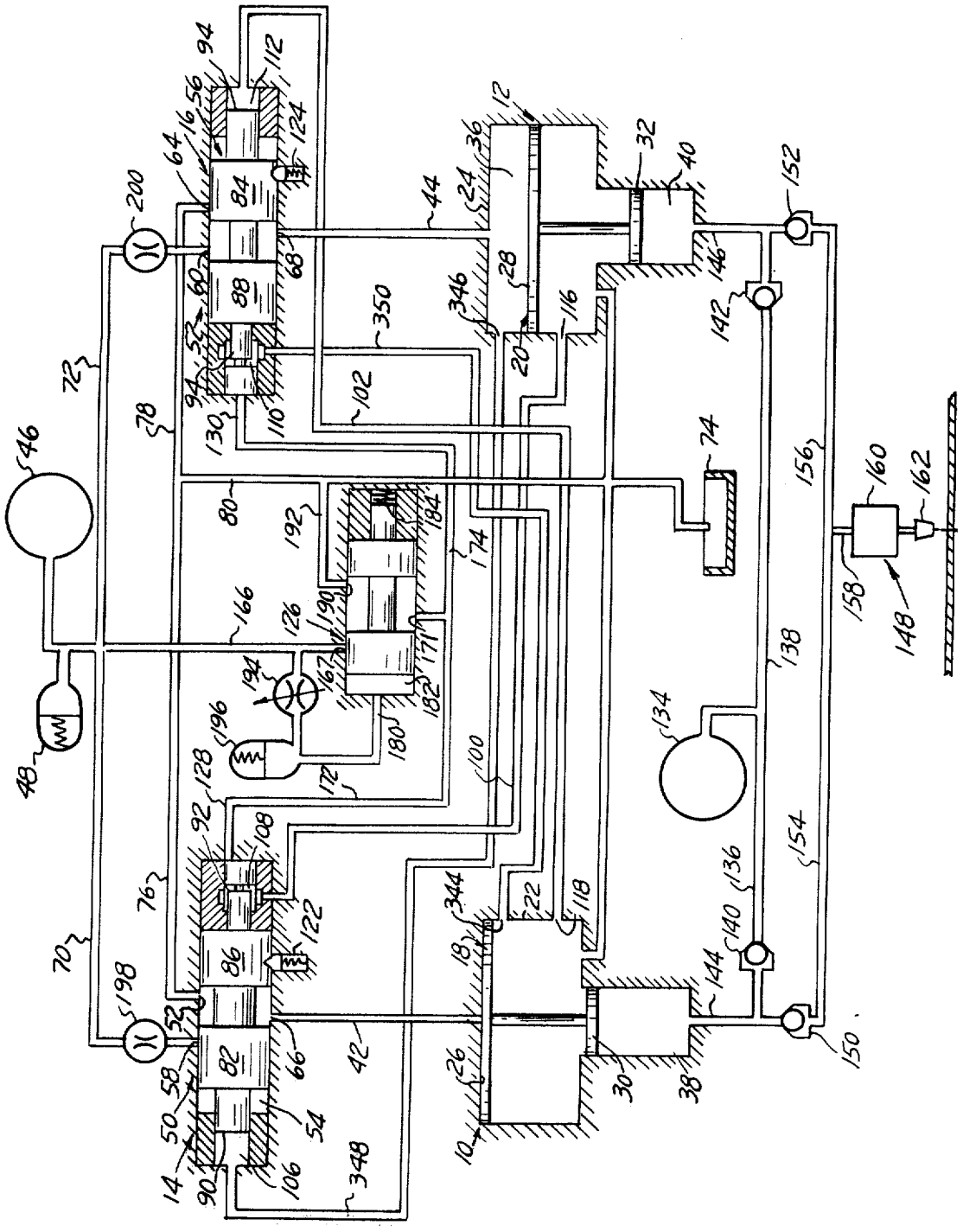


FIG. 7

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FIG. 9


200
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94
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124
84
88
68
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44
24
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32
40
152
146
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92
108
122
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198
86
82
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38
10

DUAL SINGLE ACTION RAM INTENSIFIER

CROSS REFERENCE TO RELATED APPLICATIONS

This application is a continuation-in-part of copending application Ser. No. 139,098 filed Apr. 30, 1971, now abandoned.

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention concerns intensifiers and particularly ram type intensifiers capable of producing pressures of 70,000–100,000 psi.

2. Description of the Prior Art

Fluid Jet cutting in which a high pressure liquid jet is used to cut various materials, has long been proposed and demonstrated experimentally. However, practical industrial application of this concept has heretofore been precluded because of several technical difficulties, one of which being the problem of providing a suitable source of high pressure liquid.

This is a difficult problem since pressures in excess of 30,000 psi are generally required and most such applications would call for pressures on the order of 50–70,000 psi or even higher, and liquid under these pressures must be supplied continuously in order to yield clean cuts of the workpiece.

Furthermore, in many situations the jet assembly must be moved over the workpiece, such as in automated pattern cutting, and since flexible hydraulic lines and couplings are not available which are capable of handling such pressures, a source which is reasonable in size and weight is highly desirable.

Heretofore, the only commercially available systems for continuously producing a flow of liquid under pressures on this order have been double acting differential piston type ram intensifiers, as for example that disclosed in U.S. Pat. No. 2,631,542.

This approach has not been very successful since the deflections tending to occur at these very high pressure levels are particularly troublesome in this device as "three-point" support is involved and the parts must be designed to be very large and stiff to prevent binding of the parts during stroking. Furthermore, the ram tends to be excessively long and bulky since the chambers are arranged end-to-end.

These deflections also tend to aggravate the piston sealing problems and the heavy construction requires considerable seal servicing effort and consequent excessive downtime.

A prior art intensifier as described in U.S. Pat. No. 2,579,670 to Hjarpe discloses a dual free piston intensifier which arrangement would alleviate some of the difficulties described above, but the valving arrangement disclosed in that patent does not provide simultaneous pressurization of the separate cylinders sufficient to precompress the liquid in the cylinder beginning its pressure stroke.

Another prior art intensifier described in U.S. Pat. No. 3,331,329 to Bauer describes such a single acting twin cylinder intensifier which does involve "overlap" or simultaneous pressurization of both cylinders at the end of both cylinders at the end of each cycle to alleviate pressure surges occurring because of the transition conditions existing during the changeover of the load from one to the other cylinder. However, as stated therein, this system is designed for pressures in which

compression of the liquid is negligible. Specifically, the fluid which operates the rams is supplied from a constant volume source. This type of source would not function in the environment of the extremely high pressures causing substantial compression of the working fluid since the compression of the liquid would create a substantial drop in pressure in the ram chambers, in turn causing a substantial drop in the output pressure to thereby defeat the purpose of the overlap.

It has been discovered by the present inventor that a great deal of compression of working liquids which are considered incompressible is encountered which would severely hinder the production of a continuous flow of liquid at these higher pressures with the arrangement described in the aforementioned patents since compression of the liquid would take place for a substantial portion of the pressurization stroke of each cycle and hence the maximum pressure thereof would not be developed during this portion of the stroke.

Therefore, it is an object of the present invention to provide an intensifier which is relatively lightweight and compace and yet capable of providing a continuous flow of liquid at pressures on the order of 70,000 psi.

SUMMARY OF THE INVENTION

This object and others which will become apparent upon a reading of the following specification and claims is accomplished by providing a dual single action ram type intensifier which is controlled so that during a portion of each compression stroke of each piston the other piston is fully pressurized to create a substantial overlap in the operation. The return stroke of each piston is powered by the working liquid which is supplied thereto under sufficient pressure to minimize the effects of air bubbles on the volumetric efficiency of the intensifier and on the continuity of flow, as well as to eliminate the need for an arrangement to create return strokes thereof by the operating fluid.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic representation of the dual single action ram intensifier arrangement together with its associated valving and hydraulic circuitry.

FIG. **2** is a schematic representation of the dual pistons and their associated valving at an intermediate stage of the intensifier cycle.

FIG. **3** is a schematic representation of the dual pistons and their associated valving at an advanced state of the intensifier cycle.

FIG. **4** is a schematic representation of the start-up valve and the control valves during initiation of the intensifier operation.

FIG. **5** is a front elevational view of a specific embodiment of the dual piston intensifier assembly.

FIG. **6** is a view of the intensifier assembly in the direction of the arrow **6—6** in FIG. **5.**

FIG. **7** is a view of the section taken along the line **7—7** in FIG. **5.**

FIG. **8** is a view of the intensifier in the direction of the arrow **8—8** in FIG. **5.**

FIG. **9** is a schematic representation of a second embodiment of the dual single action ram intensifier according to the present invention.

DETAILED DESCRIPTION

In the following detailed description, certain specific

terminology will be utilized for the sake of clarity and a specific embodiment described in order to provide a complete understanding of the invention, but the invention is not so limited and may be practiced in a variety of forms and embodiments.

Referring to the Figures and particularly FIG. **1**, the intensifier of the present invention includes first and second single action ram intensifier means **10** and **12** controlled by a pair of associated pilot pressure operated control valves **14** and **16.**

Each ram intensifier cylinder **10** and **12** includes a respective piston member **18** and **20** disposed in a respective cylinder means **22, 24.** The piston members consist of connected large diameter heads **26** and **28** and smaller diameter heads **30** and **32** which together with the cylinder means define in upper chambers **34,** 36 and lower chambers **38** and **40** in the cylinder means **22** and **24.**

This arrangement constitutes the basic intensifying apparatus, with a lower pressure fluid introduced in each upper chamber **34** and **36** which, acting on the pistons **18** and **20,** serves to raise the pressure of a working liquid introduced to the lower chambers **38** and **40** by virtue of the difference in area of the connected piston heads.

For example, in one design a large diameter head of 2.50 inches diameter and a small diameter head of 0.625 inches diameter was used to create an intensification ratio of 18 to 1. Thus, using a 5000 psi supply in the upper chambers, a 70,000 psi pressurization of the working liquid can be obtained.

The supply operating fluid under pressure to each upper chamber is provided by means of conduits **42, 44** disposed entering the top of each cylinder **22** and **24** so as to pressurize the volume above each large diameter head **26** and **28.**

A source of operating fluid **46** is provided such as a **5000** psi oil pump which may include an accumulator **48,** and its communication with the conduits **42, 44** is controlled by the pilot pressure operated control valves **14** and **16,** respectively.

This source **46** and the associated accumulator **48** must provide a means for supplying fluid under pressure to the upper chambers **34** and **36** at a substantially constant pressure over the range of flow requirements encountered during the operating cycle. This is particularly important during the overlap portion of the cycle to be described in detail herein, since the pressurization of the cylinder upper chamber at the beginning of its pressure stroke involves a rather rapid increase in the volume of the upper chamber due to compression of the working fluid. This resulting large momentary flow must be provided by the source **46** without causing a substantial pressure drop in the chamber of the cylinder approaching the end of its pressure stroke lest a corresponding drop in output pressure occur.

Each control valve **14** and **16** includes a valve spool **50, 52** slidably disposed in a valve bore **54, 56,** respectively.

The valve spools **50** and **52** by means of their position in each bore **54, 56** control communication between ports **58, 60,** and **62, 64** with ports **66, 68,** respectively, hence providing a three-way function of the valves **14** and **16.**

Ports **58** and **60** are connected via conduits **70, 72** with the source **46,** while ports **62, 64** are connected to the sump **74** via conduits **76, 78,** and **80.** Ports **66** and **68** are connected to the conduits **42** and **44,** respectively.

Thus, in the position of the valve spool **50,** shown in FIG. **1,** the region of upper chamber **34** of cylinder **22** above the large diameter head is connected to the sump **74** as port **58** is blocked by the land **82,** while in the position of the valve spool **52** shown therein, the upper chamber **36** of cylinder **24** above the large diameter head **28** is connected to the source **46** as the land **84** blocks the port **64.** In the opposite positions of the valve spools, the opposite would be true, as the land **86** would block the port **62,** while the land **88** would block the port **60.**

The position of each of the valve spools **50** and **52** in their respective bores **54** and **56** is controlled by pilot pressures applied to opposite ends **90, 92** and **94, 96** thereof, with these pilot pressures being generated by movement of the pistons **18** and **20** in the cylinders **22** and **24.** This is accomplished by means of sensing lines **98, 100** and **102, 104** communicating with chambers **106, 108, 110,** and **112,** respectively, each being behind a respective valve spool end **90, 92, 94,** and **96.**

Sensing line **98** is connected to a sensing port **114** positioned in cylinder **22** so that in the full down position of piston **18,** it will be uncovered and direct source **46** pressure to chamber **106.** Sensing line **100** is connected to port **116** positioned in cylinder **24** so that as the piston **20** moves downwardly at an intermediate point in its pressurization travel, port **116** will be uncovered and direct source pressure to chamber **108.** Sensing line **102** is connected to sensing port **118** positioned in cylinder **22** so that in a similar manner, at an intermediate point in the downward travel of the piston **18,** the port **118** will be uncovered and direct source pressure to chamber **108.** Sensing line **104** is connected to sensing port **120** positioned in cylinder **20** so that at the full down point in travel of the piston **28** it likewise is uncovered to direct source pressure to chamber **110.**

In order to maintain the spools **50, 52** in the positions created by the application of the pilot pressures, mechanical detents **122** and **124** may be provided. Alternate approaches such as hydraulic lock of the chambers **106, 108, 110,** and **112** or magnetic detents described infra could be utilized.

In order to insure proper synchronization of the pistons **18** and **20** during start up, a start-up valve **126** is provided which serves to momentarily pressurize conduits **128** and **130** and cause the valve spools **50, 52** to be positioned in a manner to be described more complete infra, while normally these conduits are connected to the sump **74** via conduit **132** as shown in FIG. **1** and do not affect the operation of the device.

The source **134** of working liquid provides a means for supply of the working liquid to each of the lower chambers **38** and **40** via conduits **136** and **138,** check valves **140** and **142,** and conduits **144** and **146.** For many cutting applications water is preferable as a cutting fluid and the system will be described as such. Since oil is the preferred liquid for hydraulic systems as the intensifier arrangement of the present invention, a two-liquid system results. The water is supplied under relatively high pressure, i.e., 3000 psi for the system described as contrasted with prior art intensifiers utilizing differing operating and working fluids for the above-described reason which have supplied the working fluid under substantially no pressure, and have relied on the operating fluid to provide the return stroke.

This approach minimizes the effects of bubbles in the working liquid source **134** on the volumetric efficiency of the system as their volume is much reduced initially. In addition, this eliminates the need for a separate return cycle arrangement with the operating liquid, as the high pressure working liquid provides a means for returning the pistons to their return position after a pressurization cycle since it forces each of the pistons upwardly whenever the operating fluid is not applied to their respective upper chambers **34** and **36.**

The working liquid, during the compression stroke of each piston **18, 20** is directed to a common outlet means comprising the utilization device, such as the fluid jet cutting system **148** described via check valves **150, 152** and conduits **154, 156,** and **158.** Thus alternating and overlapping pressurization of the cylinders will cause the liquid under high pressure to be directed to the fluid jet cutting system. From this description it can be appreciated that check valves **150** and **152** prevent communication of the lower chambers **38** and **40** with the common outlet means except when these chambers are pressurized.

The fluid jet cutting system may include a surge vessel **160** and a nozzle **162** of a construction as described in copending application Ser. No. 119,758, filed Mar. 1, 1971, "Nozzle for Producing Fluid Cutting Jet", M. C. Kurko et al.

By reference to FIGS. **2** and **3,** the operation of this device after start up may be readily understood. Assuming that the intensifier is pressurized in the manner depicted in FIG. **1,** i.e., the upper chamber **36** of cylinder **24** is pressurized and the upper chamber **34** of cylinder **22** connected to the sump, the associated piston **20** will move downwardly, pressurizing the liquid in the lower chamber **40.**

When the piston **20** has reached the point intermediate in its pressurization travel depicted in FIG. **2,** the sensing port **116** will be uncovered and pressurized by the operating fluid under pressure. This pressure acting via sensor line **100** in chamber **92** will force valve spool **50** to the left as viewed in FIG. **2** to the position depicted therein, which will serve as means to cause pressurization of the upper chamber **34** associated with returned piston **18** via conduit **42.**

This pressurization will cause the piston **18** to move to compress the working liquid in the lower chamber **38,** while at the same time the pressurization of the upper chamber **36** is maintained substantially unabated.

Thus, preceding the return movement of piston **20,** both pistons **18** and **20** will then be pressurized and move downwardly together (although more slowly than piston **20** alone) until the piston **20** reaches the point depicted in FIG. **3,** at which point the sensing port **120** is uncovered. The resulting pressurization of sensor line **104** and connected chamber **110** causes the valve spool **52** to shift to the right to the position depicted in FIG. **3,** which in turn causes chamber **36** to be connected to the sump **74** and piston **20** to the return position.

In the meantime, piston **18** has descended to the position depicted in FIG. **3** to provide substantially full pressurization and continues to descend, thus maintaining the output pressure.

The amount of overlap is designed so that all compression of the liquid, conduits, etc., has taken place at the beginning of each pressurization cycle and the pressure of the working liquid compressed by intensifier cylinder **10** has risen to the operating pressure of the device before the depressurization of intensifier cylinder **12,** so that no fluctuations of the output flow or pressure is encountered.

It has been determined by the present inventor that the degree of overlap necessary at pressures on the order indicated, i.e., 70,000 psi, is considerable since substantial travel of the piston at the beginning of its pressure strike must take place to "precompress" the working liquid in the lower chamber before reducing the pressure in the cylinder ending the pressure stroke.

For example, in the system described and with a three-inch stroke, the sensor ports **116** and **120** of intensifier cylinder **12** and **114** and **118** of intensifier cylinder **10** would be positioned 0.5 inches apart and the resulting "compression" travel would be on the order of 1.0 inches in order to minimize such pressure variations.

Piston **18** then continues downwardly (although more rapidly) until port **118** is uncovered causing pressurization of sensor line **104** and chamber **112,** causing intensifier cylinder **12** to be turned on once again in a similar manner as with the cycle of piston **20.** When piston **18** reaches bottom and port **114** is uncovered, sensor line **98** and chamber **106** are pressurized causing intensifier **10** to be turned off, and piston **8** to be returned to the return position.

Thus, the cycle repeats itself over and over as long as an operating fluid is supplied thereto. For the specific system described a cycle rate of 3.65 Hz has been calculated to produce an output flow rate of 1 gpm at 70,000 psi.

By referring to FIG. **4,** the operation of the start-up valve **126** may be understood. In the position shown, the start-up valve spool **164** directs via conduit **166** and port **167** fluid under pressure from source **46** to chambers **168** and **170** of control valves **14** and **16** via port **171,** conduits **172** and **174.** This causes free pistons **176** and **178** to force valve spools **50** and **52** into the position shown in FIG. **4** to properly synchronize their operation. Valve spool **164** stays in this position only momentarily until pressure builds up in conduit **180** and chamber **182** to cause the valve spool **164** to move to the right as viewed therein against the bias of a spring **184** to the position shown in FIG. **1.** Land **186** then blocks port **167** while land **188** moves to uncover port **190** and place conduits **172** and **174** in communication with the sump **74** via branch conduit **192.**

The response of the valve spool **164** is controlled by an orifice **194** and accumulator **196** so that its movement is delayed until the valve spools **50** and **52** have shifted. Alternatively, an electronic operator and delay or other arrangements could be used to provide this function.

This arrangement insures that the valve spools are in the proper position at the beginning of operation to initiate the cycling activity.

Also, in connection with start-up, flow into the supply chambers must be limited sufficiently so that too rapid descent of the pistons does not occur since they encounter relatively little resistance until the output load builds up, which in turn can cause a loss of of synchronization if the return stroke of the piston **18** is not completed before piston **20** reaches port **116** and thus initiates a premature power stroke of piston **18.**

To accomplish this end, flow limiting orifices **198** and **200** can be incorporated in conduits **70, 72** or the asso-

ciated ports could merely be properly sized to limit the maximum flow.

Referring to FIGS. 5–8 a specific hardware embodiment of the present invention is depicted.

This includes a pair of intensifier cylinders **202, 204** in side by side relationship held together by means of a top manifold plate **206,** intermediate manifold plate **208,** and bottom manifold plate **210,** which in turn are held together by sets of studs **212** and **214.**

Each intensifier cylinder **202** and **204** includes a control valve assembly **216** and **218,** while a start-up valve **220** is mounted therebetween.

An accumulator **222** corresponding to the accumulator **48** depicted schematically in FIG. **1** may be mounted between and below the intensifier cylinders **202** and **204,** and in communication with conduit **224** and passages (not shown) in the top manifold plate **206.**

FIG. **7** shows intensifier cylinder **204** and its associated control valve **218** in section, which is typical of both intensifier cylinders **202** and **204.** This includes a piston **226** having a large diameter upper head **228** slidably disposed in an upper cylinder **230,** connected by a self-aligning coupling **231** to a smaller diameter lower head **232** passing through intermediate manifold plate **208** and disposed in a lower cylinder **234.**

The upper cylinder **230** includes an outer load bearing sleeve **236** and an inner wear sleeve **238,** which has the sensor ports formed therein with the lower shut-off sensor port **240** shown both which are preferably shaped as a narrow slit. This form of sensor port reduces seal scuffing and abrasion as the piston **226** traverses the inner sleeve **238.**

Each sensor port communicates with openings (not shown) in the outer sleeve **238** in turn connected to sensor lines **242** and **244.**

The piston **228** is cushioned at the top of cylinder **230** by a spring washer stop **246** and at the bottom by a resilient ring **248.**

The bottom cylinder **234** includes an outer sleeve **250** stretched over an inner sleeve **252** so as to preload the inner sleeve **252** against the pressure loading. This has been found to be an efficient cylinder design for withstanding very high pressures.

The dynamic sealing of the small diameter head **232** during the compression stroke is accomplished in this embodiment by means of a "controlled leakage" bushing **254.** This arrangement, which has been described in the literature, creates a seal under high pressure conditions by creating a pressure differential across the sleeve **256.** Liquid which gets past the packing **258** flows into both clearance **260** and the space between the small diameter head **232** and the sleeve. Since clearance **260** ends in a static packing **262,** the liquid is trapped, while flow past the small diameter head **232** is allowed to escape and be collected by annular slot **264,** passage **266** and line **268.** Thus, during the compression stroke a pressure differential across the sleeve **256** is created which causes the sleeve **256** to grip the small diameter head tightly, but during the return stroke to allow free movement therethrough.

The control valve **218** includes a valve spool **270** slidably disposed in a bore **272** and positioned by means of pilot pistons **274, 276,** and **278.** Valve spool **270** controls the communication of passage **280** and port **281** connected with the return line **282** and connected passages **284, 286,** and **288** and the supply line **224** and connected passage **290,** by either blocking port **292** or **294** depending on its position in bore **272.**

Pilot piston **274** is operated by pressure received from sensor port **240** and sensor line **244,** pilot piston **276** is operated by pressure received via sensor line **296** connected to the on sensor port of the opposite intensifier cylinder **202,** while piston **278** is operated by pressure received via line **298** from the start-up valve **220** (FIG. **6**).

A pair of permanent magnets **300, 302** are utilized to provide the detent function, holding the valve spool **270** in one position or the other by magnetic attraction therebetween until overcome by the pilot pistons.

Leakage flow past the pilot pistons **274, 276,** and **278** is collected via relieved areas **304, 306** at each end of the bore **272,** and ports **294** and **308,** and passages **310** and **288,** and passed into the return passages **286, 284.**

In order to collect any leakage flow past the large diameter head **228** and from the sensor ports, as well as to allow return flow to fill the space behind the large diameter head **228** during the return strike, a line **312** communicating with passage **314** in turn communicating with annular passage **316** and opening **318** is provided.

Thus, as the operating fluid is displaced from above the head **228** during the return, it may readily flow via port **294,** passages **288, 286, 284** into line **312** and passages **314, 316,** and **318** into the area behind the head, thus minimizing the resistance during the return stroke.

The working liquid is supplied via line **320** inlet check valve **322,** and enters the lower cylinder **234** via passages **324, 326.** During the compression stroke, flow passes into passages **326, 328** past outlet check valve **330** into slot **332,** passage **340** (FIG. **8**) to the output passage **342.** The fluid jet cutting nozzle assembly could be threaded directly into the passage **342** to receive the output flow.

This specific embodiment operates as described in reference to the schematic drawings of FIGS. **1–5.**

In the embodiment described in FIGS. **1–5,** sequencing difficulties may occur due to unequal pressures generated in the lower chambers from the same applied pressures in the upper chambers **34** and **36** if significant manufacturing variations result in differing piston areas, differences in friction, etc. This difference in lower chamber pressure could cause arresting of the motion of the piston approaching the end of its pressure stroke before it has uncovered the final ports **114** or **120.**

The alternate embodiment shown in FIG. **9** obviates this potential difficulty by modifying the means for generating the shut off signal. In this embodiment the shut-off signal is generated from ports **344** and **346** in cylinders **10** and **12** respectively, these ports being located at a level such that the pistons **18** or **20** will have carried out their precompression at the point at which these ports are uncovered. These ports **344** and **346** are connected via lines **348** and **350** to chambers **106** and **112** of valves **14** and **16** so that the piston which is approaching the end of its cycle is turned off when the other piston has moved sufficiently to uncover its sensing port **344** or **346.** Thus, the aforementioned lock up cannot occur, since even if the piston would be arrested in its motion, as the piston starting up reaches full pressure sequencing will proceed since it does not depend on further travel of the piston at the end of its stroke.

While specific embodiments have been described, the invention is not to be so limited and many varia-

tions are of course possible within the scope of the present invention.

I claim:

1. A fluid actuated intensifier comprising:

first ram intensifier means including a piston member disposed in a cylinder means to form an upper chamber defined by said cylinder means and one head of said piston and a lower chamber defined by the other head of said piston and said cylinder means;

second ram intensifier means including a piston member disposed in a cylinder means to form an upper chamber defined by said cylinder means and one head of said piston and a lower chamber defined by the other head of said piston and said cylinder means;

means for introducing fluid into said lower chambers;

means for pressurizing said upper chambers including a constant pressure source capable of supplying said fluid to said upper chambers without a substantial drop in pressure due to substantial flow into said upper chambers resulting from compression of said fluid in said lower chambers;

said means for pressurizing said upper chambers also including control means cyclically pressurizing said upper chambers by communication with said constant pressure source so as to cause said piston members to alternately move to pressurize said fluid introduced into said lower chambers and including means causing each of said pistons to move to a return position after said pressurization cycle, said control means further including means causing simultaneous pressurization of both of said upper chambers and means causing both piston members to simultaneously move to pressurize said lower chambers preceding the return movement of either of said piston members by said simultaneous pressurization, said control means including a pair of control valve means operatively associated with each of said ram intensifiers operable independently of each other to produce said simultaneous pressurization of said ram intensifier means, and further including start-up valve means causing said control valves to momentarily pressurize one of said upper chambers and depressurize the other of said upper chambers during start-up of said intensifier, whereby said piston members initially assume oppositive positions in said cylinder means;

common outlet means receiving fluid pressurized in said lower chambers, including means providing communications of said lower chamber with said common outlet means during said pressurization and discontinuing communication during said return movement.

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