Dec. 12, 1967 R. E. RAYMOND 3,357,363

HYDRAULIC MACHINE

Filed Nov. 15, 1966 6 Sheets-Sheet 1

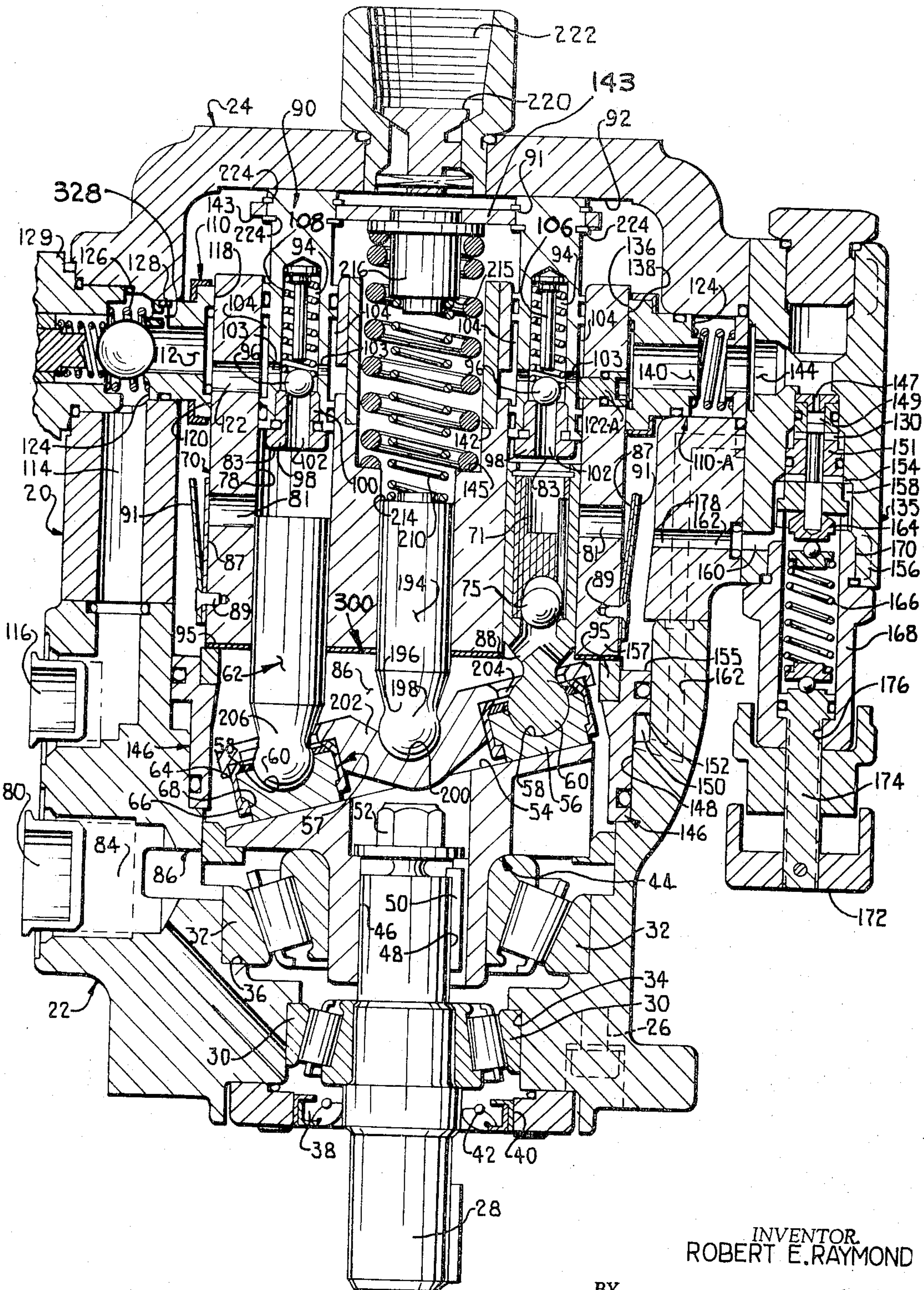

FIG.1

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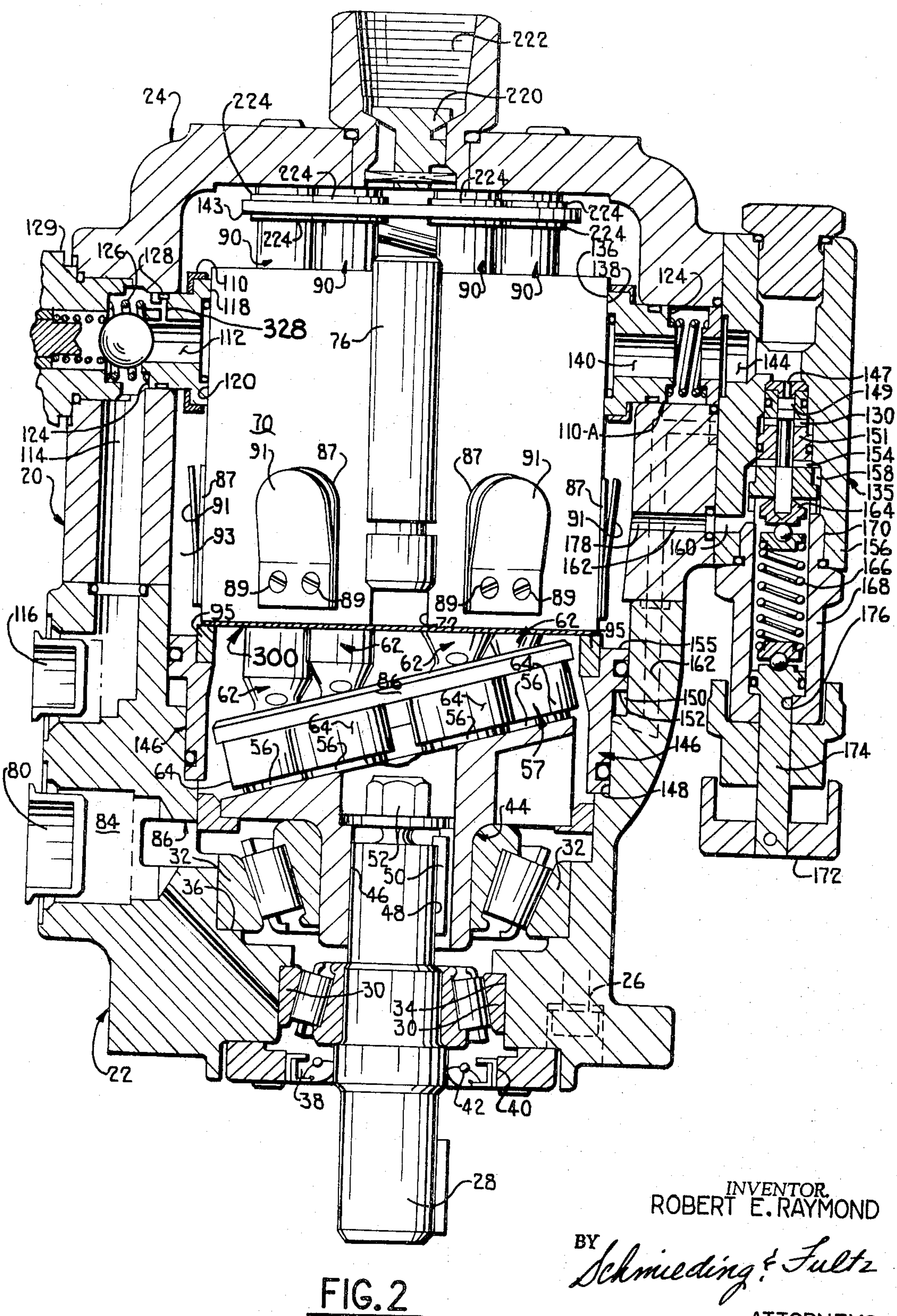

FIG. 2

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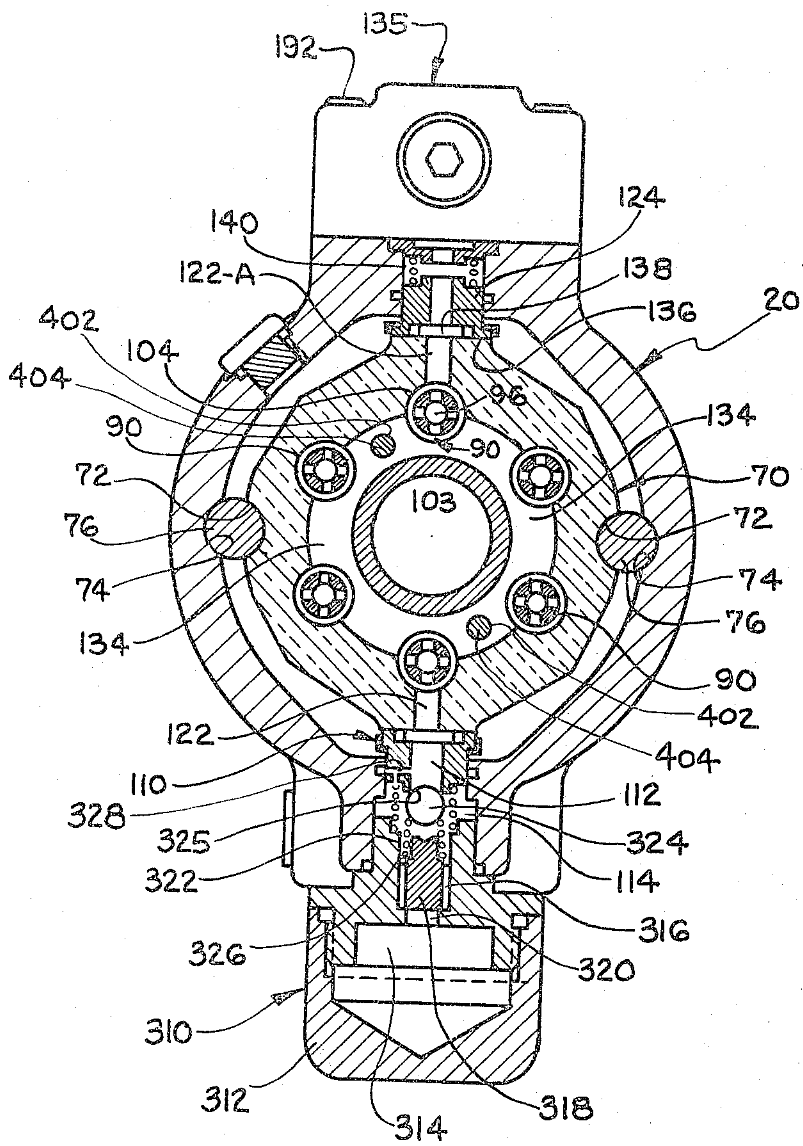

FIG. 3.

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FIG. 4.

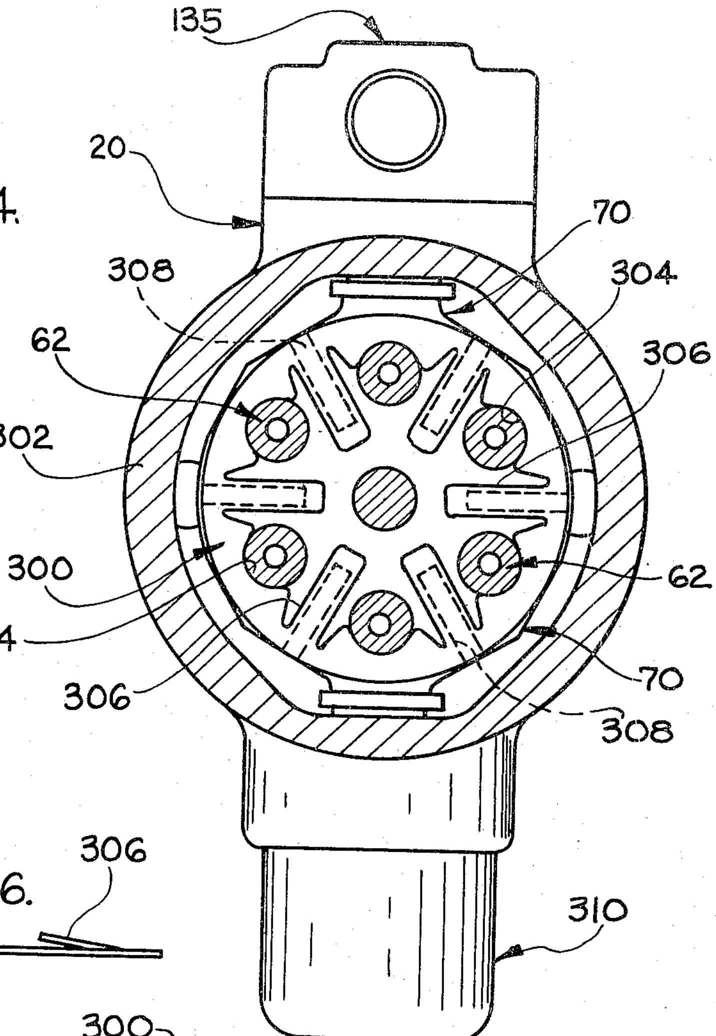
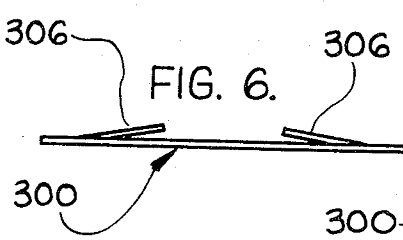


FIG. 6.



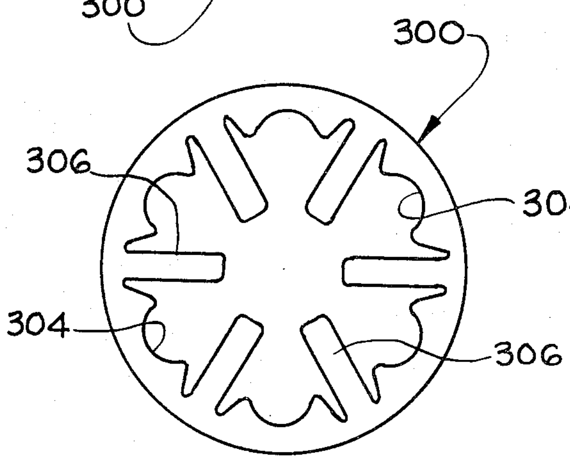


FIG. 5.

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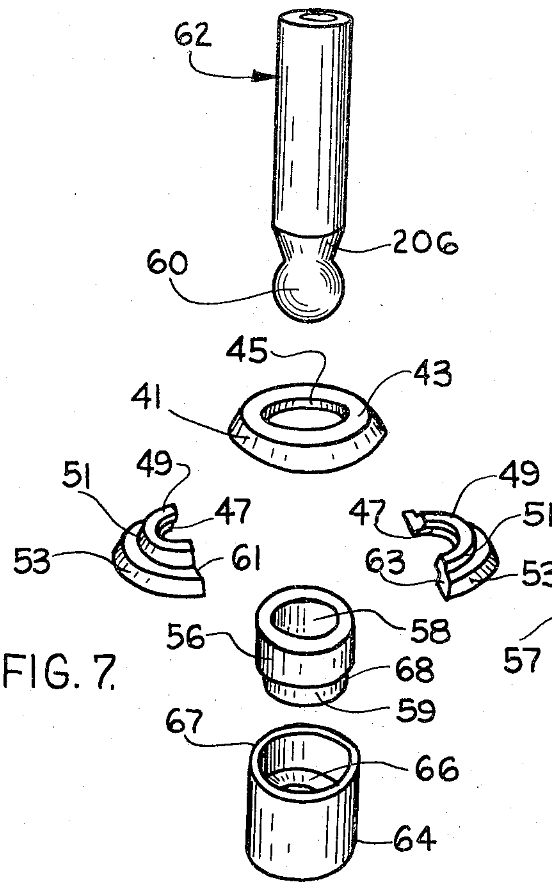

FIG. 7.

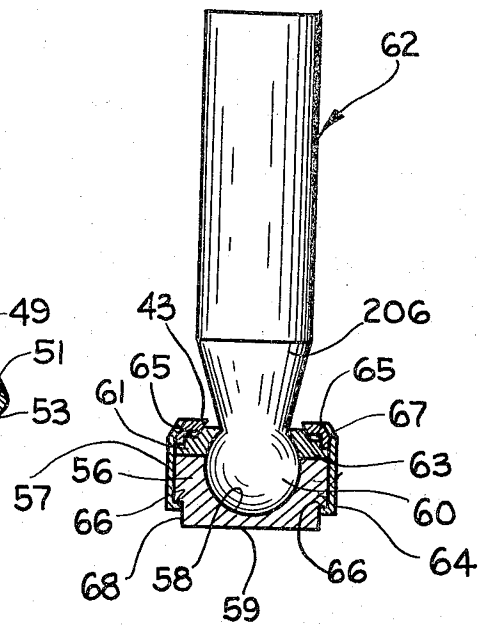

FIG. 8.

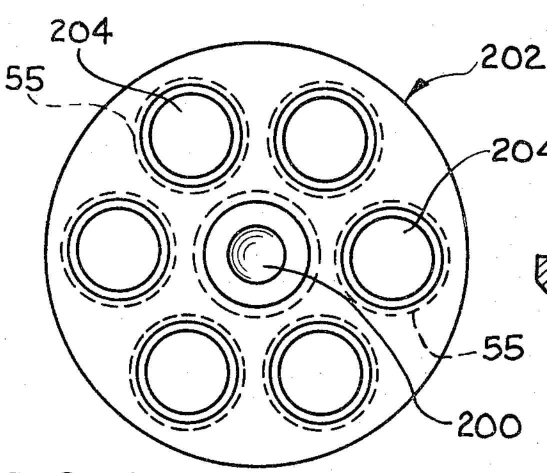

FIG. 9.

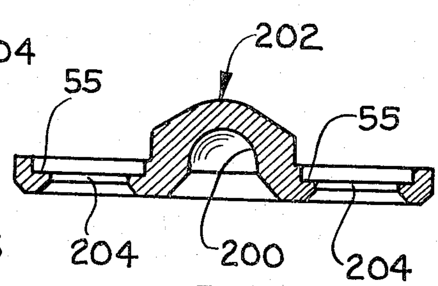

FIG. 10.

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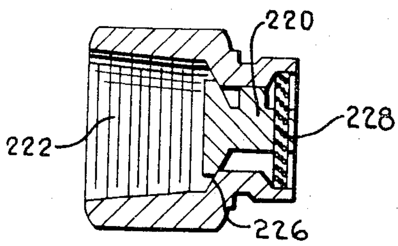

FIG. 11.

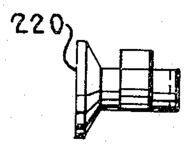

FIG. 12.

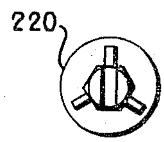

FIG. 13.

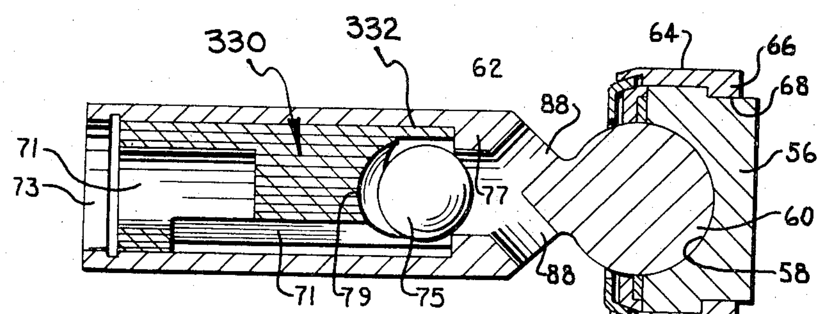

FIG. 14.

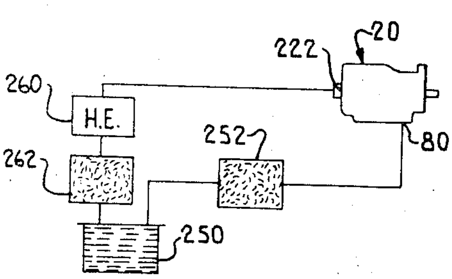

FIG. 15.

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United States Patent Office 3,357,363
Patented Dec. 12, 1967

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3,357,363
HYDRAULIC MACHINE

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Filed Nov. 15, 1966, Ser. No. 594,350
23 Claims. (Cl. 103—173)

ABSTRACT OF THE DISCLOSURE

A variable displacement hydraulic pump comprising an axially shiftable cylinder barrel the movement of which with respect to the pumping pistons varies the displacement of the machine. The cylinder barrel separates the interior of the pump housing into a cooling chamber and an intake chamber and pressure relief valve means is provided on the cylinder barrel for releasing pressurized fluid from said cooling chamber to said intake chamber responsive to shifting of said cylinder barrel.

The present invention relates generally to hydraulic machines and in particular to a novel piston type pumping apparatus.

This application is a continuation-in-part of my copending application Ser. No. 511,291 filed Dec. 1, 1965.

In general, the hydraulic machine of the present invention comprises an improved axial piston type variable displacement pump wherein the volumetric displacement of the pumping cylinders is automatically varied by shifting a cylinder barrel axially with respect to the pumping pistons of the machine.

Certain of the improved features of the present invention are also applicable to fixed displacement pumps and fluid motors as will be understood from the following description wherein the novel features are incorporated in the above mentioned variable displacement machine.

As one aspect of the present invention the pump is provided with a novel piston shoe construction whereby the pumping pistons are actuated by a cam means. Such piston shoe construction comprises a metal encased piston shoe means formed of low friction material and mounted on a ball-shaped piston foot. The piston shoe means includes a novel multiple segment ring bearing and retaining cap means disposed in overlying relationship with the upper portion of the piston foot and each include tapered surfaces which engage one another such that vertical force in a downward direction urges the multiple segment bearing in towards the piston foot.

In accordance with the present invention the novel construction of the piston shoe means includes a multiple segment ring bearing which is constantly urged into engagement with the piston foot to take up any clearance or backlash and to compensate for wear between component parts over long periods of operation.

As another aspect of the present invention, the multiple segment ring bearing is covered by a continuous metal retaining ring which locks it in engagement with the piston foot.

As another aspect of the present invention, a novel piston and piston shoe construction is provided wherein each piston incorporates a spherical foot portion provided with a composite shoe means including socket forming components formed of low friction material, and a metal casing that not only reinforces the socket forming components against laterally directed tensile stresses, but also serves to retain the components in assembled relationship on the spherical piston foot.

As another aspect of the present invention, the novel piston and piston shoe construction is used in combination with a novel yoke formed of low frictional material.

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As another aspect of the present invention the pump is provided with a novel by-pass flow chamber relief apparatus wherein the interior of the pump housing is divided into a lower pressure intake chamber and a higher pressure by-pass flow or cooling chamber. In addition a novel pressure relief means is located between these two chambers and functions to rapidly release fluid from the by-pass flow chamber to the intake chamber upon axial shifting of the above mentioned cylinder barrel to vary the displacement. This results in high response displacement control of the pumping apparatus.

As another aspect of the present invention the pump further includes a novel floating barrel driving ring that functions to form a seal between the above mentioned by-pass flow and intake chambers. In addition the barrel driving ring forms a self-aligning driving means for the previously mentioned cylinder barrel and also serves to mount the above mentioned pressure relief means for the by-pass flow chamber.

As another aspect of the present invention the pump is provided with a novel pulse filtering apparatus located at the high pressure discharge manifold of the pump which provides capacitive and resistive control of the high pressure outlet flow from the pump. This provides means for filtering out pulse effects of the pumping pistons on the high pressure outlet flow from the pump.

As another aspect of the present invention the above mentioned pulse filtering apparatus includes an integral outlet check valve and orifice mechanism that functions in a novel manner to isolate a pressure control mechanism on the pump from stored energy that for any reason may be present in the system downstream of the pump mechanism.

As still another aspect of the present invention the pump includes hollow pumping pistons that are provided with novel inlet check valves that include resilient inserts that serve multiple functions by:

(1) Forming shock absorbing stops for inlet ball check valves that provide low noise level valve operation.

(2) Providing guides for the paths of movement of the ball check valves.

(3) Providing space fillers for the cavities in the hollow pistons thereby making possible high compression ratios.

(4) Forming maximum opening stops for the ball check valves in the piston.

It is therefore an object of the present invention to provide an improved hydraulic machine that includes a novel piston shoe construction which includes a multiple segment bearing ring constantly urged into engagement with the piston foot and which takes up any clearance or backlash between the component parts. Further, the bearing surfaces are constantly urged together to compensate for wear between the engaging parts over long periods of use.

It is another object of the present invention to provide an improved hydraulic machine that includes a novel piston shoe construction wherein the multiple segment ring bearing is locked in place and covered by a continuous metal retaining ring.

It is another object of the present invention to provide an improved hydraulic machine that includes a novel piston shoe construction that includes components formed of low friction material which are retained in assembled relationship by a metal casing that also serves to prevent deformation and fracturing of the components under laterally directed tensile stresses.

It is another object of the present invention to provide an improved hydraulic machine that includes a novel piston shoe construction that eliminates the need for high pressure lubrication in the hydraulic machine and which further will not gall when abrasive substances are encountered between the shoe and the cam means that drives it.

It is another object of the present invention to provide an improved hydraulic machine that includes a novel piston shoe construction that eliminates the need for holding close tolerances in the fitting of the drive components of a hydraulic machine.

It is another object of the present invention to provide an improved variable displacement pump that comprises a novel bypass flow chamber relief apparatus that functions to provide high response variable displacement control.

It is another object of the present invention to provide an improved variable displacement pump that comprises a novel floating barrel driving ring that functions to form both an inter-chamber seal and a self-aligning driving means for the axially shiftable cylinder barrel of the pump.

It is another object of the present invention to provide an improved pump that comprises a pulse filtering apparatus that functions to filter out pulse effects of the pumping pistons on the high pressure outlet flow from the pump.

It is still another object of the present invention to provide an improved pump that comprises novel inlet check valve apparatus which provide efficient quiet intake valving and high compression ratio capacities.

Further objects and advantages of the present invention will be apparent from the following description, reference being had to the accompanying drawings wherein a preferred form of embodiment of the invention is clearly shown.

In the drawings:

FIG. 1 is a side sectional view of a pumping apparatus constructed in accordance with the present invention, the section being taken along a vertical plane through the centerline of the apparatus;

FIG. 2 is a side view of a pumping cartridge assembly constructed in accordance with the present invention, the remainder of the apparatus being shown in section as seen in FIG. 1;

FIG. 3 is an end sectional view of the pump of the preceding figures, the section being taken along the line 3—3 of FIG. 1;

FIG. 4 is another end sectional view of the pump of the preceding figures, the section being taken along the line 4—4 of FIG. 1;

FIG. 5 is a plan view of a by-pass flow relief apparatus comprising a portion of the pump of the preceding figures;

FIG. 6 is a side elevational view of a valve plate comprising a portion of the pressure relief valve apparatus of FIG. 5;

FIG. 7 is an exploded perspective view of the piston and piston shoe construction incorporated in the pump shown in FIG. 1;

FIG. 8 is a side elevational view, partially in section, of the piston and piston shoe construction shown in FIG. 7;

FIG. 9 is a plan view of a piston return yoke used in combination with the piston and piston shoe construction of FIGS. 7 and 8;

FIG. 10 is a side sectional view of the piston return yoke of FIG. 9, the section being taken along the line 10—10 of FIG. 9;

FIG. 11 is a side sectional view of a portion of the apparatus shown in FIG. 1 illustrating a check valve for the cooling outlet port in the housing, the section being taken along a vertical plane through the centerline of the apparatus;

FIG. 12 is a side view of a portion of the valve assembly shown in FIG. 11;

FIG. 13 is an end view of that portion of the valve assembly shown in FIG. 12;

FIG. 14 is a side sectional view of a portion of the apparatus of the present invention illustrating the novel piston and piston inlet check valve construction, the section being taken along a vertical plane through the centerline of the piston; and

FIG. 15 is a diagrammatic view of a typical closed circuit cooling system wherein the fluid may be forced through an oil filter to clean the fluid before returning the pump inlet port.

Referring in detail to the drawings, a variable displacement pump constructed in accordance with the present invention is illustrated in FIGS. 1 and 2 and comprises a housing indicated generally at **20** that includes a front housing portion indicated generally at **22** and a rear housing portion indicated generally at **24**. The housing portions are joined together at the central portion of the pump and held by a plurality of studs **26**.

A drive shaft **28** is mounted in the forward end of the housing by tapered roller bearing assemblies **30** and **32** which are pressed into recesses **34** and **36**.

An oil seal **38** is pressed into a recess **40** in housing **20** and includes an annular resilient element **42** that wipes the periphery of drive shaft **28**.

As seen in FIG. 1, the inner end of drive shaft **28** carries a cam means indicated generally at **44** which includes a central bore **46** provided with a keyway **48** that receives a key **50** for preventing rotation of cam means **44** relative to shaft **28**. The cam means is retained on shaft **28** by a nut **52** which is tightened into locked relationship on a threaded inner end of shaft **28**.

With continued reference to FIG. 1, cam means **44** includes an inclined surface **54** which engages a plurality of shoes **56** preferably formed of low friction material, the latter including sockets **58** which form pivotal ball joints with ball-shaped ends **60** formed on a plurality of pumping pistons **62**.

Each of the shoes **56** is surrounded by a metal casing **64** that is crimped around the ball-shaped end **60** of its respective piston **62**. Each metal casing **64** also includes an inwardly extended annular protrusion **66** that snaps into an annular recess **68** formed in the base of the shoe.

Reference is next made to FIGS. 7 and 8 which illustrate in detail a piston and piston shoe constructed in accordance with the present invention.

The shoe means **57** is of composite construction and includes a lower bearing portion **56** that forms an upper spherical surface **58** for receiving spherical piston foot **60** and a lower bearing surface **59** that engages the upper surface **54** of cam means **44**.

With continued reference to FIGS. 7 and 8, a multiple segment ring formed by segments **61** and **63** comprises an upper bearing portion of shoe means **57**.

Segments **61** and **63** include spherical surfaces **47** that conform with the upper portions of ball-shaped piston foot **60** and upper and lower tapered surfaces **51** and **53** each of the tapered surfaces **51** being formed on a rib or shoulder portion **49** and the lower tapered surfaces being formed on the base portion of each segment.

A continuous retaining ring **43**, preferably formed of metal, covers segments **61** and **63** and includes tapered surfaces **45** and **41** which engage tapered surfaces **51** and **53** respectively such that a gap or clearance **65** exists between segments **63** and **61** and ring **43**, as best seen in FIG. 8.

A metal casing, indicated generally at **64** surrounds the lower bearing portion **56**, ring **43**, and ring segments **61** and **63** and includes an upper edge **67** that is crimped over retaining ring **43** such that ring **43** locks segments **61** and **63** in place and an inner shoulder **66** that surrounds a recess **68** of reduced diameter provided on the lower end of lower bearing portion **56**.

Lower bearing portions **56**, multiple segments **61** and **63** are preferably formed of lower friction material, such as nylon, aluminum or the like, whereby galling is eliminated between the pivotal bearing portions in metal ball-shaped piston foot **60** and also between lower bearing portion **56** and metal cam means **44**.

It will be understood that the above mentioned nylon to metal bearing surfaces, although under relatively heavy axial loads, will not gall or cause failure when metal chips or other foreign particles are encountered.

When shaft **28** of the pump is driven by a prime mover, not illustrated, rotation of cam means **44** exerts axial forces on pistons **62** and causes them to reciprocate in their respective cylinders.

On the return stroke of each piston **60** a vertical force is exerted on retaining ring **43** by the surface of shoe mounting recess **55** of yoke **202** and ring **43** pushes against ring bearing segments **61** and **63**.

Engagement between the respective tapered surfaces **51**, **53**, **45**, and **41** forces multiple ring bearing segments **61** and **63** in toward piston foot **60** and thereby maintains engagement between the spherical surfaces **47** of bearing segments **61** and **63** and the outer bearing surface of piston foot **60**.

The original dimension of the tapered surfaces determine the amount of clearance **65** that exists. The clearance or gap **65** that exists between the bottom of the downwardly facing surface on ring **43** and the top of the upwardly facing surfaces of segments **61** and **63** represents the amount of clearance or backlash that may be taken up between the spherical surfaces **47** and spherical piston foot **60**.

It will be understood that wear between surfaces **47** and foot **60** over long periods of operation is compensated for by the novel construction of shoe means **57** as spherical bearing surfaces **47** are being urged inwardly by the action of the double tapered surfaces on ring **43** and segments **61** and **63** respectively.

When the high axial stresses are imposed on the shoe means **57**, and if lower bearing members **56** are made of low friction material such as nylon, for example, they have the advantageous characteristic of possessing high strength under compression. However, these nylon bearing portions then have the undesirable characteristic of being relatively weak when subjected to tensile stresses. Since the spherical piston foot **60** is axially forced into the hemispherical socket **58** of the lower bearing portion, the lower bearing portion is subjected to radially directed tensile forces and since the low friction resinous materials such as nylon are relatively weak under tension, the metal casing **64** confines bearing portion **56** against radial deformation and rupture.

With continued reference to FIGS. 1 through 3, a cylinder barrel cartridge assembly indicated generally at **70** is axially slideably mounted within housing **20** and guided by a plurality of guide grooves **72** which receive longitudinally extending bearing members **76**.

Members **76** may be best described as side rail bearings and function not only to absorb piston side thrust reaction imposed on barrel **70** but, in addition, bearing members **76** function as keys against cylinder barrel reaction and thereby serve to absorb torque.

Pistons **62** are disposed in respective barrel cylinders **78** which receive low pressure oil or hydraulic fluid in a novel manner via housing inlet port **80**, passage **84** in front housing portion **22**, an inlet chamber indicated generally at **86** within housing **20** and inlet ports **88** formed in each piston **62**.

Now referring to FIGS. 1 and 14, inlet ports **88** in pistons **62** communicate with a cavity **71** formed in each piston **62** which in turn communicates with a respective cylinder **78** through an orifice **73** in the rearward end of piston **62**.

A ball-check valve **75** is freely carried in the opening to cavity **71** between a valve seat portion **77** and a stop portion **79** formed on a resilient insert indicated generally at **330**. These inserts serve multiple functions by:

(1) Forming shock absorbing stops for inlet ball check valves that provide low noise level valve operation.

(2) Providing longitudinally extending legs that form guides for the paths of movement of the ball check valves.

(3) Providing space fillers for the cavities in the hollow pistons thereby making possible high compression ratios.

(4) Forming maximum opening stops for the ball check valves in the pistons.

On the suction stroke of the pistons **62**, fluid is drawn into a respective cylinder **78** via the passage and inlet means described above as the low pressure fluid forces ball-check valve **75** away from seat portion **77** to allow fluid to flow into cavity **71** and then through orifice **73** into cylinders **78**.

Now referring to FIGS. 1 and 2 each cylinder **78** includes a unique outlet port construction whereby a first cylinder outlet port **81** is disposed in a side wall of each cylinder **78** and a second cylinder outlet port **83** is axially spaced from outlet port **81** and formed in the rearward end of cylinders **78**.

It is important to point out that the axially spaced cylinder outlets **81** and **83** may both be disposed in a side wall of cylinder **78** without departing from the spirit of the invention although the structure shown is preferred. Further, the inlet port may also be formed in the cylinder instead of in the pistons as shown in the preferred embodiment without departing from the spirit of the present invention.

As seen in FIGS. 1 and 2 each of the cylinders **78** includes a respective reaction plug, indicated generally at **90** in free self-aligning engagement with the inner end surface **92** of rear housing portion **24**.

Each reaction plug **90** is provided with a central bore **94** that carries an outlet check valve **96** which is freely retained in bore **94** by a threaded plug **98**.

Each threaded plug **98** includes a seat portion **100**, a longitudinal passage **102**, and a radial passage **103**, the latter communicating with an annular passage **104** formed in the outer wall of reaction plug **90**.

With continued reference to FIG. 1, bore **94** in each reaction plug **90** includes a valve stop **106** and a compression spring **108** which serve to limit the stroke of the ball and bias it toward a closed position.

A plurality of valve means **87**, attached to the outer surface of barrel **70** by screws **89**, covers each of the first cylinder outlets **81** and is in the form of a substantially flat, flexible reed-type valve. Valve means **87** is novmally biased in a closed position and the degrees of valve opening is limited by a stiff back-up plate **91** attached by screws **89** to valve means **87**.

It is important to point out that the reed-type valves **87** are preferred because they provide efficient valve means and yet conveniently take up little space. Further, valves **87** are easily incorporated into the cylinder barrel cartridge assembly **70** to provide an integral package to offer the attractive commercial advantages previously mentioned.

Pressurized fluid from cylinders **78** is first discharged on the compression stroke of the pistons **62** through first cylinder outlet **81**, the fluid forcing valve means **87** open, and flows into an outlet chamber or cooling chamber **93** formed in rear housing portion **24**. Outlet chamber **93** is isolated from the inlet fluid in inlet chamber **86** by an annular floating barrel driving ring and seal **95** engaging barrel **70** and an annular cylinder barrel driving piston, indicated generally at **146** which will be described in detail later herein.

Outlet chamber **93** completely surrounds cylinder barrel assembly **70** and forms a passage means for the outlet flow of fluid through a check valve **220** disposed in a housing outlet port **222** formed in the rear housing portion **24**.

As best seen in FIG. 1, as a piston **62** advances during the compression stroke, outlet port **81** is closed by the outer wall of cylinder **78** and flexible reed valve **87** will close.

It is important to point out that the flow through outlet port **81** depends upon the position of barrel **70**, hence port **81**, relative to the stroke of piston **62**. Further, since the fluid from outlet port **81** flows completely around barrel **70**, a very effective cooling flow is developed.

When piston **62** advances to the position where outlet port **81** is closed, the pressurized fluid is then discharged through the second cylinder outlet port **83** through longitudinal passages **102** in reaction plugs **90**, a plurality of small radially extending passages **103**, annular recesses **104**, annular manifold **134**, barrel outlet port **122**, radial passage **112**, an outlet member **110**, passage **114**, and a second housing outlet port **116** to the load.

Hollow outlet member **110** includes the central passage **112** that communicates with high pressure discharge passage **114** that in turn leads to housing outlet port **116**.

As seen in FIG. 1, outlet member **110** also includes a foot portion provided with a surface **118** that is in slideable sealed engagement with a longitudinally extending surface **120** formed in the outer wall of cylinder barrel **70**.

It will be noted from FIG. 1 that when cylinder barrel **70** is axially shifted relative to the housing means **20** an outlet port **122** formed in the cylinder barrel always remains in communication with central passage **112** in outlet member **110** notwithstanding axial movement of cylinder barrel **70**.

With continued reference to FIGS. 1 and 2, pressure biased outlet member **110** includes a piston surface **124** that causes the pressurized hydraulic fluid in passage **112** to bias the surface **118** on outlet member **110** downwardly into sealed engagement with longitudinally extending surface **120** on barrel **70**.

A spring **126** augments the biasing force of the high pressure oil on piston surface **124** and also serves to retain surface **118** in sealed engagement with surface **120** at low pressures and at the outset of operation.

The outer peripheral surface of outlet member **110** is provided with an annular seal **128** and a threaded plug **129** is screwed into the hole forming passage **112** and includes an inner protrusion that forms a retainer for the end of spring **126**.

Pressurized oil is also released to a variable displacement hydraulic control unit indicated generally at **135** by a second pressure biased outlet member **110–A**, FIGS. 1 and 2, which is substantially identical to outlet member **110** previously described. It will be noted that outlet member **110–A** includes a base surface **136** that is hydraulically biased into sealed engagement with a longitudinally extending surface **138** formed in the outer wall of cylinder barrel **70**. Outlet member **110–A** is biased downwardly against longitudinally extending surface **138** by a force exerted by a piston surface **124**, pressurized oil in a passage **140**.

Referring to FIG. 1, cylinder barrel **70** is constantly biased towards the front of the housing means by a control spring **142** which is interposed between a spider **143** and an annular shoulder **145** formed on cylinder barrel **70**.

With reference to FIG. 3, the cylinder barrel **70** is provided with a plurality of bores **402** which communicate with annular exhaust manifold **134**. Each bore **402** slideably carries a reaction pin **404** that includes an outer end in force transmitting engagement with spider **143** which in turn abuts the rear wall **92** of housing **20**. The pressurized oil in annular manifold **134** exerts a force on the inner ends of reaction pin **404** which force is equal to the product of the exhaust pressure and the cross-sectional area of reaction pins **404**. This hydraulic force augments the mechanical force exerted on cylinder barrel **70** by control spring **142** and hence can be utilized to reduce the mechanical spring force required to counteract the barrel shifting force exerted by control piston **152**.

It should be pointed out that control spring **142** is required, however, in addition to the hydraulic force exerted by reaction pins **404**, since the hydraulic force is not present at the starting of the pump.

Cylinder barrel **70** is hydraulically shifted axially against the biasing force of control spring **142** by an annular cylinder barrel driving piston indicated generally at **146** in FIG. 1. Piston **146** is mounted in a cylindrical surface **148** and forms therewith control cylinder **150** for receiving pressurized oil in a manner later to be described. A small annular piston surface **152** of large diameter provides sufficient axial force with low control pressures to shift cylinder barrel **70** against the force of control spring **142**.

With continued reference to FIG. 1, piston **146** includes a rear end **155** in force transmitting engagement with floating barrel driving ring and seal **95** which in turn is in force transmitting engagement with an annular base portion **157** on cylinder barrel cartridge **70**.

Referring particularly to FIGS. 1 and 3 pressurized oil is delivered through control apparatus **135** to control cylinder **150** via outlet port **122–A**, passage **140**, passage **144**, orifice **147**, radial passages **130** in spool housing **151**, lateral passage **154** in control block **156**, longitudinal passage **158** in control block **156**, vertical passage **160** in control block **156** and passage **162** in housing **20** which connects to control cylinder **150**.

As is best seen in FIGS. 4–6, when control cylinder **150** is pressurized, and cylinder barrel **70** is shifted upwardly, as viewed in FIG. 2, the cylinder barrel is adapted to shift rapidly by the inclusion of a by-pass flow chamber relief apparatus. Such apparatus comprises a relief valve plate indicated generally at **300** which is formed from a thin sheet of spring steel or the like. As seen in FIG. 1 and FIG. 4, relief valve plate **300** engages an end surface **302** in cylinder barrel **70** and is retained in place by annular floating barrel driving ring and seal **95**.

It should be further pointed out that relief valve plate **300** is accurately locted and retained in place against radial shifting by the side walls of pumping pistons **62** which confront inwardly facing arcuate surfaces **304** formed on relief valve plate **300**.

With continued reference to FIGS. 4 through 6, relief valve plate **300** includes a plurality of radially inwardly extending projections **306** each of which covers a respective barrel slot **308** formed in end surface **302** of the cylinder barrel.

When the barrel is shifted rapidly, by the previously described control flow to control chamber **150**, the projections **306** resiliently yield and open barrel slots **308** in the barrel thereby permitting fluid to rapidly be displaced from a by-pass flow or outlet chamber **93**, FIG. 2, and inlet chamber **86**. As soon as the barrel has shifted to a new equilibrium position resilient protrusions **306** of relief valve plate **300** will close barrel slots **308** and by-pass flow chamber **93** will be maintained at a higher pressure than the inlet chamber **86**.

Referring again to FIGS. 1 and 2 spool housing **151** carries a longitudinally shiftable spool member **164** that is normally biased towards a closed position by a spring **166**, the latter being contained in a spring housing **168** that is threaded into control block **156** at a threaded hole **170**. Compression spring **166** is selectively compressed or compressed by manipulating a control knob **172** that includes a shank **174** in threaded engagement with spring housing **168** at a threaded hole **176**. As seen in FIG. 1, a radial passage **178** communicating with passage **162** in housing **20** provides a direct drain back to tank for any fluid that many flow back from control cylinder **150** during shock conditions.

As best seen in FIG. 3, control block **156** is mounted on housing **20** by a plurality of studs **192**. It will be understood that other types of control apparatus responsive to various load conditions can be readily adapted to replace control apparatus **135**.

The pumping pistons **62** are returned and biased against cam means **44** by a single centrally disposed return rod

194, FIG. 1, which includes an arcuate socket 196. A ball 198 fits into socket 196 of rod 194 and also into a socket 200 formed in a piston return yoke 202. Yoke 202 includes a plurality of circumferentially spaced holes 204 that fit around neck portions 206 and which are large enough to permit free oscillation of neck portions 206 of the piston means 62.

The periphery of yoke 202 further includes a plurality of piston shoe mounting recesses 55 that are shaped to form snugly fitting sockets for the tops of the piston shoe means.

Piston return yoke 202 applies force to the rear sides of the ball-shaped piston ends 60 and in turn receives force from piston return rod 194 via the pivot joint formed by ball 198 and sockets 196 and 200. A compression spring 210 is disposed between a shoulder 214 on the rear end of piston return rod 194, a shoulder 214 on a spring retainer plug 216 which in turn bears against the front side of spider 143. Spider 143 is restrained from rearward movement by the shoulder 224 formed on the ends of reaction plugs 90. It should be pointed out that reaction plugs 90 are fitted loosely into respective holes 91 in spider 143 and are in free engagement with the inner surface 92 of the housing whereby the plugs are self-aligning with respect to pump cylinders 78.

As is best seen in FIG. 3 the pump further includes a pulse filtering apparatus indicated generally at 310 which comprises a housing 312 that forms a capacitance chamber 314 for receiving pressurized oil from discharge passage 112 and hollow outlet member 110.

It should be pointed out that capacitance chamber 314 is filled with oil from the outlet flow leaving the pump via passages 112 and 114 and the oil in chamber 314 is constantly in communication with the outlet flow via an annular passage 316 formed around spring retainer 318 which includes radially extending legs 320 that are positioned in an annular bore 322 in housing 312.

With continued reference to FIG. 3 the pulse filtering apparatus is provided with a ball check valve 324 which is biased against a seat 325 by a valve return spring 326.

It should be pointed out that the purpose of outlet check valve 324 is to isolate the previously described pressure controller 135 from any energy which may be stored in the system downstream from the pump outlet passage 114.

With continued reference to FIG. 3 the pulse filtering apparatus further includes a bleed back orifice 328 which, when check valve 324 is closed, connects the interior of the pump with outlet passage 114 via annular passage 316, bleed back orifice 328, and passage 112 in hollow outlet member 110.

It will now be understood that outlet check valve 324 and bleed back orifice 328 provide for large volumetric outlet flows but limit any bleed back flows to the relatively small flow capacity of bleed back orifice 328.

In operation, when the pumping apparatus is driven by a prime mover, cam 44 reciprocates pistons 62 which, on the suction stroke draw fluid into cylinders 78 via inlet port 80, inlet passage 84, inlet chamber 86, inlet ports 88, cavity or passage 71, and orifice 73.

It is important to point out that inlet port means may be disposed in the cylinders 78 themselves instead of in pistons 62 without departing from the spirit of the invention.

The intake fluid forces ball-check valve 75 away from seat portion 77 as it enters ports 88 and flows through cavity 71 and out of orifice 73 into cylinders 78.

On the suction stroke, valves 87 are normally biased in a closed position.

On the compression stroke, fluid is delivered first through first cylinder outlet ports 81 as the flexible valves 87 are forced open with the degree of opening being limited by back-up plate 91. The amount of flow through ports 81 depends upon the position of ports 81 relative to the stroke of pistons 62 which in the variable displacement pump illustrated, varies according to the position of barrel 70. Fluid then flows into and through outlet chamber 93 to housing outlet port 222 and out one way check valve assembly 220, best seen in FIGS. 11, 12, and 13.

Referring to FIGS. 11, 12, and 13, check valve 220 is biased against a valve seat portion 226 by a spring 228 and prevents fluid from flowing back into outlet chamber 93. Pressurized fluid from cylinder outlets 81 flows through chamber 93 and forces valve 220 open to permit fluid to flow out of housing outlet port 222.

Many types of conventional valves or surge control orifices may be used in place of outlet check valve 220 to prevent flow back through port 22 on the suction stroke of pistons 62.

The fluid from port 222 may then be delivered to a load or recirculated for cooling purposes. A typical closed circuit cooling system is illustrated diagrammatically in FIG. 15.

The fluid from port 222 leads to a heat exchanger 260 and then is forced through a very fine filter 262 before returning to reservoir 250. The fluid from reservoir 250 is pulled through 2 conventional wire screen filter 252 to inlet port 80. Therefore, the oil circulated through the pump housing 20 is cooled and cleansed before returning to the interior of the housing. It is important to point out that a substantial flow through cylinder outlets 81 occurs which is delivered out of outlet port 222 to provide very efficient cooling of the hydraulic system.

After the wall of piston 62 passes outlet port 81, the remainder of the pressurized fluid flows through second cylinder outlet port 83 to the other housing outlet port 116 via passages 102 in plugs 90, check valves 96, radial passages 103, annular grooves 104, manifold 134, passage 112 in hollow outlet member 110 and outlet passage 114 which leads to outlet port 116.

The fluid in inlet chamber 86 is isolated from the fluid in outlet chamber 93 by annular seal 95.

Some of the pressurized oil in manifold 134 is delivered to variable control apparatus 135 for shifting annular piston 146 and cylinder barrel 70 as previously mentioned.

A constant pressure at the load is obtained by arranging spool 164 to open only when a predetermined selected load pressure is exceeded. Control knob 172 is adjusted to compress spool control spring 166 so as to bias spool 164 with the proper force to permit its opening when the predetermined selected operating pressure is exceeded. When the pressure at the load rises above the operating pressure oil from the manifold passes through passage 144 and orifice 147 to spool cylinder 149. The increased pressure in the spool chamber overcomes the preset control force exerted by spool control spring 166 whereby the spool is shifted to the left, as viewed in FIG. 1, and oil is released through radial passages 130 in spool housing 151 and thence through the previously described passages to the control cylinder 150. This shifts annular piston 146 and cylinder barrel 70 to the left as viewed in FIG. 1, whereby cylinder outlet ports 81 are moved to the right relative to the pistons 62. This movement increases the flow of fluid through ports 81 because said ports are closed later in the compression stroke whereby more oil is free to flow out of said ports. Accordingly, this decreases the flow of oil per piston compression stroke out of second cylinder outlets 83 to automatically cut back the pressure in that circuit.

When the pressure drops to the preselected control pressure the biasing force exerted by spool control spring 166 shifts spool 164 to the right, as viewed in FIG. 1, whereby the spool closes radial ports 130 and the flow of oil to control cylinder 150 is terminated.

It is important to point out that the present invention has been described only by way of illustration, with respect to variable displacement operation and a cooling circuit but is not limited to such an application.

The pumping apparatus may be readily adapted to perform as two fixed displacement pumps with both housing outlet ports **116** and **222** connected to a load. The cylinder barrel **70** may be fixedly positioned to divide the flow between ports **81** and **83** in any manner desired or a variable flow control may be used to control one circuit with the excess delivered to the second circuit.

It is also important to point out that other valve means than the type shown may be employed such as, for example, an orifice which permits substantial flow in only one direction may be used to replace valves **87** and outlet port **81**, particularly for recirculating cooling fluid, without departing from the spirit of the present invention.

While the form of embodiment of the present invention as herein disclosed constitutes a preferred form, it is to be understood that other forms might be adopted, all coming within the scope of the claims which follow.

I claim:

1. A hydraulic machine comprising, in combination, housing means; cam means rotatably mounted in said housing means; cylinder barrel means axially shiftably mounted in said housing means and including cylinder means; piston means mounted for reciprocation in said cylinder means and including a piston foot; piston shoe means mounted on a ball shaped portion of said piston foot and including a lower bearing portion for engagement with said cam means, an upper multiple segment ring bearing means overlying the upper portion of said piston foot and retaining means mounted in force transmitting relationship over said multiple segment ring bearing; shoe casing means surrounding said piston shoe means; sealing means separating the interior of said housing means into a cooling chamber and an intake chamber; and pressure relief valve means on said cylinder barrel means for releasing pressurized fluid from said cooling chamber to said intake chamber responsive to shifting of said cylinder barrel means.

2. The hydraulic machine defined in claim **1** wherein certain of said ring bearing means and retaining means includes a tapered surface whereby downward force on said retaining means forces said segments of said ring bearing means inwardly against said ball shaped portion of said piston foot.

3. The hydraulic machine defined in claim **1** wherein said upper ring bearing means includes a first tapered surface and said retaining means includes a second tapered surface in force transmitting relationship with said first tapered surface.

4. The hydraulic machine defined in claim **1** wherein said shoe casing means includes a lower portion provided with means for retaining said lower bearing portion in said shoe casing means.

5. The hydraulic machine defined in claim **1** wherein said lower bearing portion includes a recess and said shoe casing means includes an inturned retaining shoulder disposed in said recess.

6. The hydraulic machine defined in claim **1** wherein said shoe casing means includes a lower portion provided with means for retaining said lower bearing portion in said shoe casing means, and wherein said lower bearing portion includes a recess and said shoe casing means includes an inturned retaining shoulder disposed in said recess.

7. The hydraulic machine defined in claim **1** wherein said ring bearing means includes a base portion provided with a first tapered surface and a shoulder portion provided with a second tapered surface, and wherein said retaining means includes a first tapered surface engaging said first tapered surface on said base and a second tapered surface engaging said second tapered surface on said shoulder portion.

8. A hydraulic machine comprising, in combination, housing means; cylinder barrel means axially shiftably mounted in said housing means and including cylinder

means; piston means mounted for reciprocation in said cylinder means and engaging said cam means; sealing means separating the interior of said housing means into a cooling chamber and an intake chamber; and pressure relief valve means on said cylinder barrel means for releasing pressurized fluid from said cooling chamber to said intake chamber responsible to shifting of said cylinder barrel means.

9. The hydraulic machine defined in claim **8** wherein said pressure relief valve means comprises a sheet of resilient material that includes radially extending fingers.

10. The hydraulic machine defined in claim **8** wherein said cylinder barrel means includes an end provided with a valve opening and wherein said pressure relief valve comprises a sheet of resilient material that includes a resilient valve flap normally closing said valve opening.

11. The hydraulic machine defined in claim **8** wherein said pressure relief valve means comprises a sheet of resilient material including a valve flap portion and arcuate guide portions thta engage the sides of said piston means.

12. The hydraulic machine defined in claim **8** that includes an annular barrel driving ring and wherein said pressure relief valve means comprises a sheet of resilient material retained against an end of said cylinder barrel means by said barrel driving ring.

13. A hydraulic machine comprising, in combination, housing means; cam means rotatably mounted in said housing means; cylinder barrel means axially shiftably mounted in said housing means and including cylinder means; piston means mounted for reciprocation in said cylinder means and engaging said cam means; an annular barrel driving piston mounted in said housing means; an annular barrel driving ring and seal between said barrel driving piston and said cylinder barrel means; sealing means separating the interior of said housing means into a cooling chamber and an intake chamber; and pressure relief valve means on said cylinder barrel means for releasing pressurized fluid from said cooling chamber to said intake chamber responsive to shifting of said cylinder barrel means.

14. A hydraulic machine comprising, in combination, housing means; cam means rotatably mounted in said housing means; cylinder barrel means axially shiftably mounted in said housing means and including cylinder means piston means mounted for reciprocation in said cylinder means and including a piston foot engaging said cam means, said piston means including a longitudinally extending bore having an inner end forming a piston valve seat; piston passage means through a side wall of said piston means and communicating with said bore; a ball check valve in said bore and engageable with said seat; valve stop means formed of resilient material mounted in said bore for resiliently arresting the opening strokes of said ball check valves; sealing means separating the interior of said housing means into a cooling chamber and an intake chamber; and pressure relief valve means on said cylinder barrel means for releasing pressurized fluid from said cooling chamber to said intake chamber responsive to shifting of said cylinder barrel means.

15. The hydraulic machine defined in claim **14** wherein said valve stop means includes axially extending legs for guiding the path of movement of said ball check valve in said bore.

16. A hydraulic machine comprising, in combination, housing means; cam means rotatably mounted in said housing means; cylinder barrel means axially shiftably mounted in said housing means and including cylinder means, a high pressure manifold and an outlet passage; piston means mounted for reciprocation in said cylinder means and engaging said cam means; a pulse filtering apparatus including a capacitance chamber and a filter passage connecting said capacitance chamber with high pressure manifold sealing means separating the interior of said housing means into a cooling chamber and an intake chamber; and pressure relief valve means on said cylinder barrel means for releasing pressurized fluid from said cooling chamber to said intake chamber responsive to shifting of said cylinder barrel means.

17. The hydraulic machine defined in claim **16** that includes a ball check valve between said manifold and said outlet passage.

18. The hydraulic machine defined in claim **16** that includes a ball check valve between said manifold and said outlet passage; and a bleed back orifice between said outlet passage and said manifold.

19. The hydraulic machine defined in claim **1** wherein said cylinder barrel means includes an annular manifold and a plurality of spaced cylinders in said manifold having ports communicating with said annular manifold.

20. The hydraulic machine defined in claim **1** wherein said cylinder barrel means includes an annular manifold and a plurality of spaced cylinders in said manifold; and a plurality of reaction plugs having inner ends disposed in said cylinders and outer ends freely engaging said housing means.

21. The hydraulic machine defined in claim **1** wherein said cylinder barrel means includes a plurality of cylinders, a manifold communicating with said cylinders, and an axially extending outer surface provided with a manifold outlet port; and a hollow outlet member shiftably mounted in said housing means and including an inner surface engaging said outer surface of said cylinder barrel means and an outlet passage communicating with said manifold.

22. The hydraulic machine defined in claim **1** wherein said cylinder barrel means includes a plurality of cylinders provided with intake ports; a control cylinder in said housing means; and an annular piston disposed in said control cylinder and in force transmitting relationship with said cylinder barrel means for varying the location of said intake ports relative to said piston means.

23. The hydraulic machine defined in claim **1** wherein said cylinder barrel means includes a plurality of circumferentially spaced cylinders and a central axially extending bore; a piston return yoke engaging said piston shoe means; and yoke biasing means slideably disposed in said bore and in force transmitting relationship between said yoke and said housing means.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,357,363 Dated December 12, 1967

Inventor(s) Robert E. Raymond

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

In Fig. 1, reference numeral 130 has been added with a lead line to the dotted passage communicating passages 130 and 162. A lead line has been added between reference numeral 120 and the barrel periphery. In Figs. 1, 2, refernce numeral 302 has been added with a lead line to the bearing surface of barrel 70 adjacent valve 300. In Fig. 4, the lead line for reference numeral 302 has been extended to the central portion of the bearing surface of the barrel. In Fig. 3, reference numeral 124 has been deleted and reference numeral 110-A has been added for the threaded plug located between cap 312 and the pump housing. In Fig. 7, the lead line for reference numeral 41 has been replaced by a dotted line leading to the inside tapered surface of ring 43. In Fig. 8, reference numerals 41 and 45 have been added for the tapered surfaces of retainer rings 63 and 61 respectively. Reference numeral 47 has been added for the spherical surface of ring 63. Column 8, cancel lines 56 - 68; same column 8, cancel lines 20 - 25 and substitute the following: -- cylinder 150 via cylinder block outlet passage 122-A, passage 140 in outlet member 110-A, passage 144, orifice 147 and a radial passage 130 that connects with a housing passage 162 which in turn connects with control cylinder 150 as seen in Fig. 1.

With continued reference to Fig. 1, a control spool 149 is axially slideably mounted in a spool housing 151 and is biased toward a left position by compression spring 166 in which position spool 149 normally isolates radial passage 130 from control orifice 147.

When the fluid pressure through control orifice 147 exceeds a predetermined pressure value, established by the selected setting of a control knob 172 that includes a shank 174 in threaded engagement with a spring housing 168 at a threaded hole 176, then spool 149 shifts to the right thereby overcoming spring 166 via self-aligning spool end 164. Radial

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Patent No. 3,357,363 Dated December 12, 1967

Inventor(s) Robert E. Raymond PAGE - 2

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

passage 130 is thereby opened and pressurized fluid is released to control cylinder 150 as explained above. This serves to shift cylinder block 70 to the left, Fig. 1, to a new position of equilibrium wherein the force exerted on the cylinder block by compression spring 142 equals the force exerted on said block by annular control piston 146.

Referring again to Fig. 1, a radial passage 178 in the pump housing provides a drain from the control apparatus 135 back to the relatively low pressure outlet chamber 93. When pressure of oil through orifice 147 acting on the end of spool 149 drops below the preset value established by control spring 166, then spool 149 shifts to the left closing passage 130 to pressurized flow and subsequently opening passage 130 to drain passage 178 in the pump housing via passages 154, 158 and 160. This permits annular control piston 146 to move to the right discharging fluid from annular chamber 150 back to low pressure outlet chamber 93. This permits the cylinder block to shift to the right to a new position of equilibrium. --.

Signed and sealed this 30th day of May 1972.

(SEAL)
Attest:

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