Aug. 25, 1964 V. BUSH 3,145,660

FREE PISTON HYDRAULIC PUMP

Filed Feb. 13, 1962 4 Sheets-Sheet 1

FIG. 1.

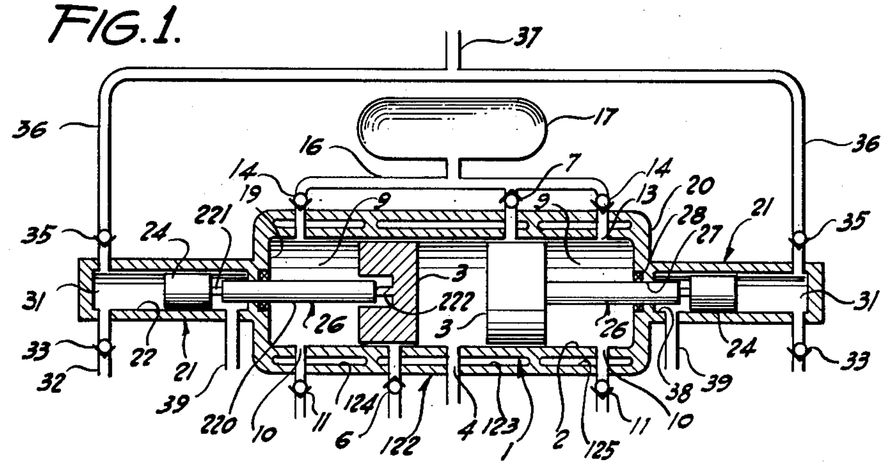


FIG. 2.

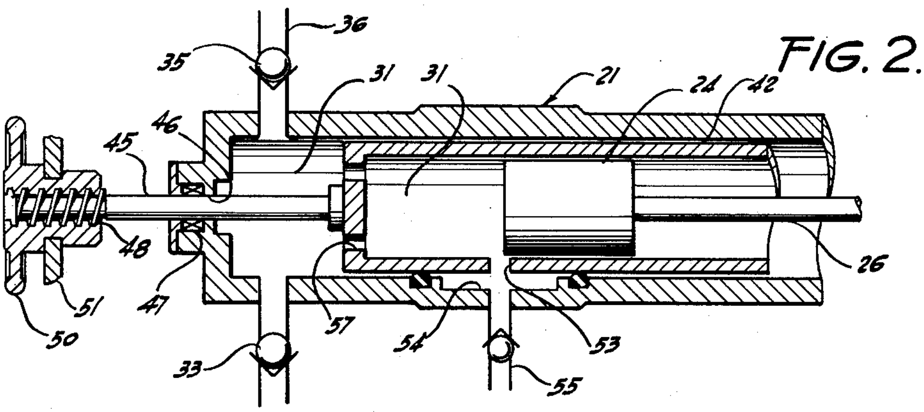
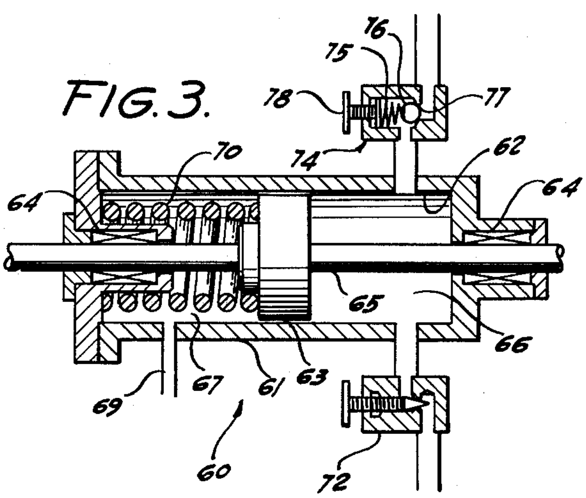


FIG. 3.



INVENTOR

Vannevar Bush

By A. L. Downs

Attorney

Aug. 25, 1964 V. BUSH 3,145,660

FREE PISTON HYDRAULIC PUMP

Filed Feb. 13, 1962 4 Sheets-Sheet 2

FIG. 4.

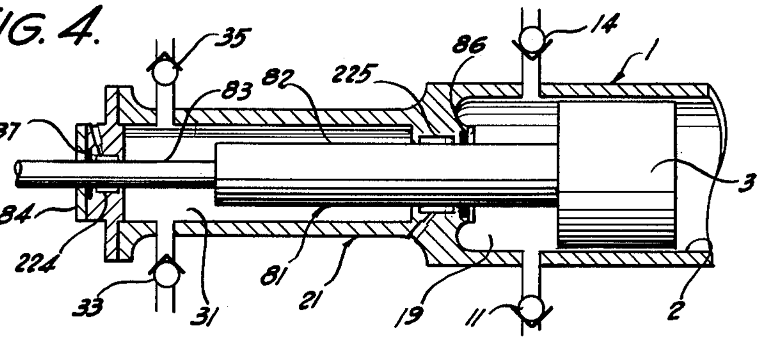


FIG. 5

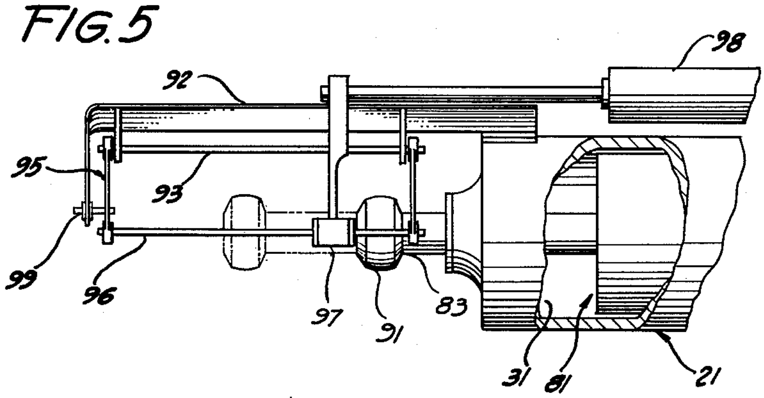
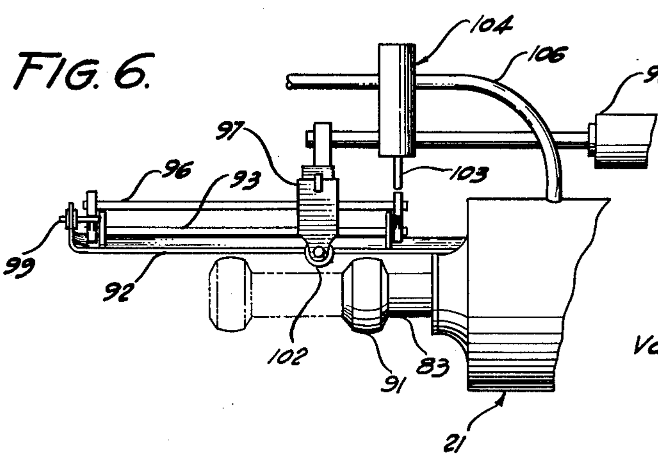


FIG. 6.



INVENTOR

Vannevar Bush

By. A. G. Duval

Attorney

FREE PISTON HYDRAULIC PUMP

Filed Feb. 13, 1962

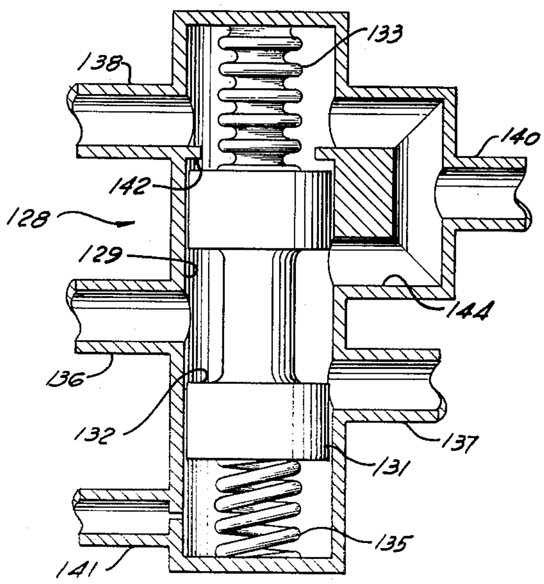


FIG. 8.

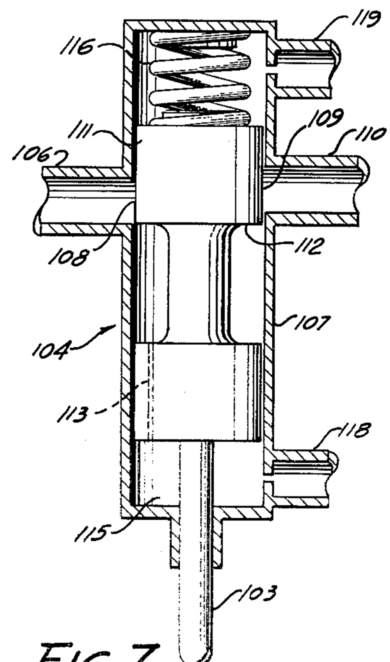


FIG. 7.

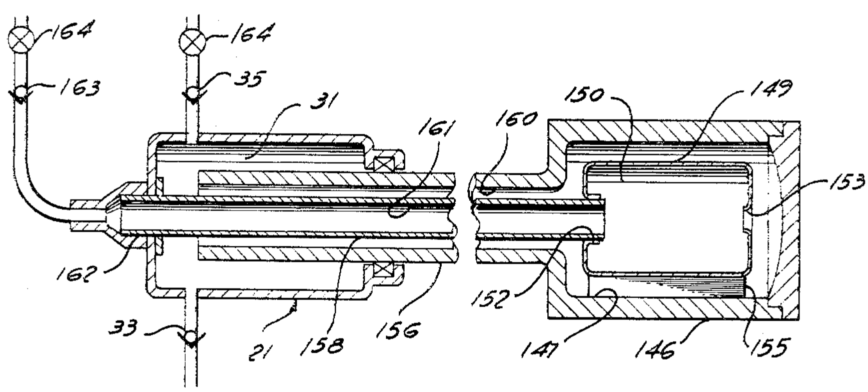


FIG. 9.

INVENTOR
Vannevar Bush

By A. G. Douvas
Attorney

FREE PISTON HYDRAULIC PUMP

Filed Feb. 13, 1962 4 Sheets-Sheet 4

FIG. 10.

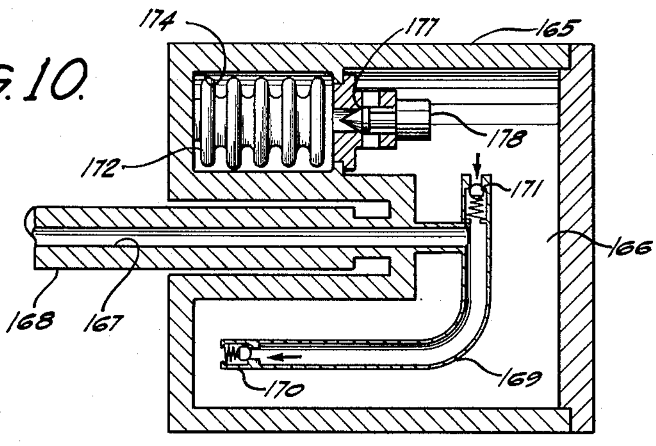


FIG. 11.

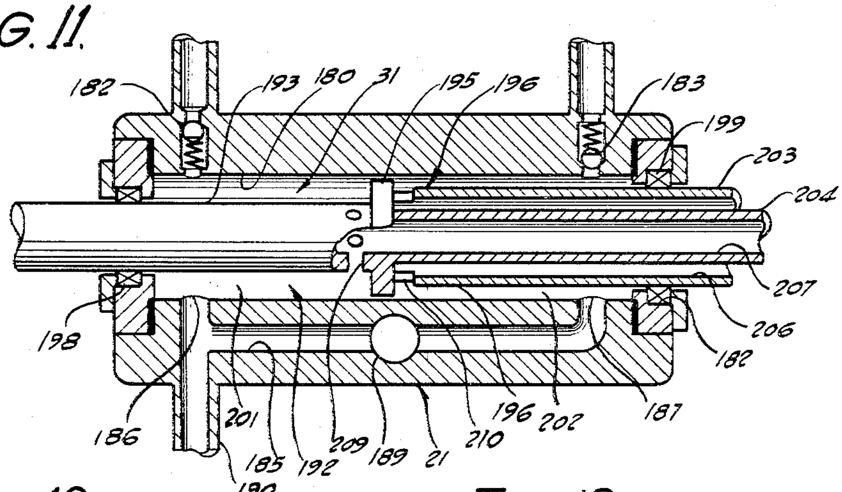


FIG. 12.

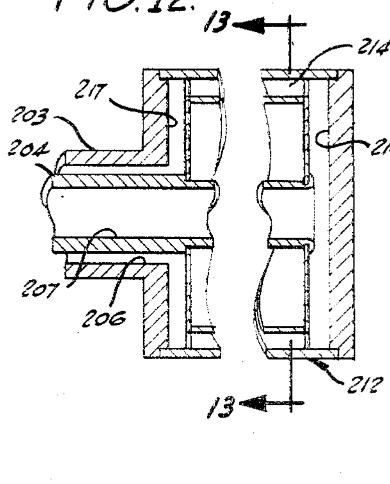
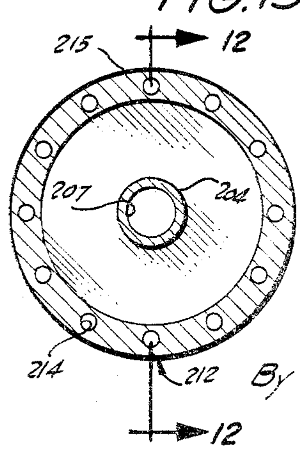


FIG. 13.



INVENTOR
Vannevar Bush

By A. G. Downs
Attorney

United States Patent Office

3,145,660
Patented Aug. 25, 1964

3,145,660

FREE PISTON HYDRAULIC PUMP

Vannevar Bush, 304 Marsh St., Belmont 78, Mass.

Filed Feb. 13, 1962, Ser. No. 173,008
16 Claims. (Cl. 103—54)

This invention relates to a free piston hydraulic pump having a free piston engine and a hydraulic pump combined as a single interacting unit, the output power of the free piston engine directly providing the input power for the hydraulic pump.

A free piston engine is disclosed and claimed in my Patent No. 2,983,098 granted May 9, 1961, and entitled "Gas Lubricated Free Piston Engines With Supercharging Arrangements." The free piston engine therein disclosed consists of two simple cylindrical pistons reciprocable in a closed end cylinder of uniform diameter. A diesel cycle occurs in the volume between these pistons and drives them apart. Air is compressed between each piston and the cylinder end, and the energy thus represented is used, after a transformation, to supercharge the diesel cycle. Bounce chambers on the outward ends of the cylinder utilize the remaining kinetic energy of the pistons to return the pistons to center for the next diesel cycle. The exhaust gases from the diesel cycle are used to drive a turbine to perform useful work.

The outwardly imposed motion of the pistons caused by the diesel cycle is not utilized directly to perform useful work. Instead the kinetic energy of the reciprocating pistons merely permits repeated diesel cycles from which hot exhaust gases are generated. The exhaust gas of the diesel cycle in excess of that necessary to maintain piston oscillation is taken from the free piston engine to drive an auxiliary independent device such as a power turbine from which the ultimate output work is derived.

It is desirable in many applications that output work be in the form of hydraulic fluid under high pressures and flows. This is because of the ease of transforming fluid pressure energy to useful mechanical work, the ease of conveying and storing fluid pressure energy, the economy of installation and operation, and the large output powers available within a confined space. Heretofore the hydraulic power unit has not itself been combined with a prime mover for related dependent operation therewith. Instead the prime mover, in such forms as an electric motor, turbine or combustion engine, drives the hydraulic unit, with its operations being independent of that of the hydraulic unit and generally only brought to common or related speeds, etc., by means of a mechanical transmission device. This necessitates at least two units somewhat more costly than a single combined unit since the cost of each unit is required, and of somewhat less overall efficiency since the losses of the two units must be overcome.

Accordingly, an object of this invention is to provide in a single unit for unitary combined action, a free piston engine and a reciprocating hydraulic pump, the output power of the free piston engine directly providing the input power for the hydraulic pump.

Another object of this invention is to provide means for adjusting the output per cycle of the free piston hydraulic pump unit in the form of constant hydraulic flow at variable pressures or of constant fluid pressure at variable hydraulic flows.

A further object of this invention is to provide in a free piston hydraulic pump unit means to maintain uniform clearances between the pistons and cylinder bores throughout all operating conditions of the units to minimize leakage or flow-by gas losses of the unit.

These and other objects will be more fully appreciated and understood after a disclosure of the subject invention

given in the following specification and drawings forming a part thereof, wherein:

FIG. 1 is a longitudinal cross-section of the subject free piston hydraulic pump, many of the essential details being eliminated for clarity of disclosure;

FIG. 2 is a longitudinal cross-section of a form of hydraulic pump operable with the free piston hydraulic pump of FIG. 1, and particularly showing an output adjustment arrangement for the unit;

FIG. 3 is a longitudinal cross-section of a control valve operable in the free piston hydraulic pump of FIG. 1;

FIG. 4 is a longitudinal cross-section of a second form of hydraulic pump operable with the free piston hydraulic pump of FIG. 1;

FIG. 5 is a plan view, partly broken away and in section, of the end portion of the free piston hydraulic pump of FIG. 4, showing another output adjustment arrangement for the subject free piston hydraulic pump;

FIG. 6 is an elevational view of the unit of FIG. 5, with certain additional details being shown;

FIG. 7 is a longitudinal cross-section of a relief valve operable in the free piston hydraulic pump unit of FIG. 1;

FIG. 8 is a longitudinal cross-section of a thermostat valve operable in the free piston hydraulic pump unit of FIG. 1;

FIG. 9 is a longitudinal cross-section of another form of hydraulic pump and showing a piston construction operable in the free piston hydraulic pump unit of FIG. 1;

FIG. 10 is a longitudinal cross-section view of another piston construction operable in the subject free piston hydraulic pump unit of FIG. 1;

FIG. 11 is a longitudinal cross-section of another form of hydraulic pump operable in the subject free piston hydraulic pump unit of FIG. 1;

FIG. 12 is a longitudinal cross-section view, as seen generally from line **12—12** of FIG. 13, of another piston construction operable in the subject free piston hydraulic pump of FIG. 1; and

FIG. 13 is a view as seen generally from line **13—13** of FIG. 12.

FIG. 1 shows diagrammatically a cross-section of the subject free piston hydraulic pump unit. This is much simplified for clarity of disclosure, and important details omitted will be described later. There is an oscillator cylinder **1** of uniform bore **2** which receives for oscillation therein a pair of pistons **3**. When the pistons **3** are near their point of closest approach, fuel is injected from nozzle **4** to the combustion chamber defined between the pistons. When the pistons **3** are near their farthest separation, exhaust occurs through exhaust valve **6** and air from intake valve **7** scavenges the cylinder. Valves **6** and **7**, of course, are connected to circumferentially disposed ports (not shown).

The end volumes **9** of the cylinder **1** acts as blowers (blowers rather than compressors) since they produce very little pressure and absorb very little power. Inlet ports **10** having check valves **11** allow influx of atmospheric air to the end volumes **9**, and outlet ports **13** with outward check valves **14** cause this air to be blown through conduit **16** to intake valve **7** and thence into the combustion chamber, there being a reservoir **17** to smooth the flow. The volumes **19** between the cylinder ends **20** and the ports **10** and **13** act as bounce chambers to keep the pistons **3** in oscillation.

Thus far the unit of FIG. 1 is similar to that described in the above-mentioned Patent No. 2,903,098, except that there is no utilization of power developed in the diesel cycle.

Thus the pistons **3** are maintained in synchronous oscillation with respect to the cylinder **1** and to each other by controlled gaseous interchange between the bounce chambers **19** and by positive centering means actuated by differential pressures in the cylinder; as taught in Patent No. 2,903,098. Means dependent upon gaseous pressures in the cylinder further cause fuel ignition of the engine, open and close the intake valves **7** and exhaust valves **6** of the engine, and start initial oscillation of the pistons for starting the engine.

The free piston hydraulic pump as herein disclosed alters the arrangement of the above-mentioned Patent No. 2,983,098. On the ends of the main cylinder **1** are attached added cylinders **21** each of small diameter bore **22** having therein piston **24**. The pump pistons **24** are directly connected to the oscillator pistons **3** by rods **26** extending through openings **27** in the cylinder ends **29**. Appropriate packing **28** separates the cylinder bores **2** and **22** from each other. Each piston **24** and receiving cylinder **21** form on the outer end of the cylinder an expansible oil chamber **31**. Conduits **32** from an oil sump (not shown) connect through inlet check valves **33** to the chambers **31**. Outlet check valves **35** from chambers **31** discharge through conduits **36** and **37** to the driven hydraulic unit (not shown). The oil chambers **31** are interconnected to equalize the load on the oscillator pistons **3**. Chambers **38** on the inward side of the pistons **24** are vented through conduit **39** to the oil sump (not shown).

Oscillation of the pistons **3** and **24** directly causes a pressure rise of oil in the chamber **31** and the resultant discharge therefrom through outlet check valves **35**. This pumping of oil at high pressures and flows utilizes the power output developed by the diesel cycle. The energy released in the diesel cycle is thus used to drive the pistons apart and to maintain them in oscillation, which in turn, transfers energy to the oil pump pistons and finally to the hydraulic oil.

An equalizing pressure dome (not shown) can be in the output lines **36** and **37** to smooth the flow of pressure, or to take up initial pressure shock upon operation of the pump. Similarly to prevent oil pressures in excess of a maximum, an overpressure relief valve (not shown) can be included on the discharge side of the pump.

For rapid pump operation, it is desirable to pressurize the oil in the inlet side to avoid cavitation. This can be accomplished by maintaining at moderate pressure an equalizing dome (not shown) on the inlet side of the oil system.

The output of the hydraulic system can be controlled for either constant volume or constant pressure operation. Thus, assuming constant amplitude and cycling frequency of the oscillator pistons **3** and pump pistons **24**, the oil flow at any setting of the control is constant and independent of output pressure, or varies with the output pressure in some predetermined manner. In each system the output pressure or the volume of oil pumped adjusts automatically according to fuel input and to the required output power. The abovementioned Patent No. 2,983,098 discloses means to maintain constant amplitude and frequencies of piston oscillation with changing loads on the free piston engine.

A constant volume system is shown in FIG. 2. The hydraulic cylinder **21** has an inner sleeve **42** in which the oil piston **24** operates. This sleeve can be moved axially within the cylinder by appropriate structure such as a rod **45** attached to the sleeve **42** and extending through an opening **46** in the end of cylinder **21**. Packing **47** seals the opening **46** to prevent leakage from the defined oil chamber **31**. A threaded portion **48** on the end thus can be moved by screw member **50** supported by slide bearing **51**. The sleeve **42** has a port **53** communicating through annular recess **54** with a conduit **55** leading to the oil sump (not shown). It will be apparent that port **53** is duplicated by a diametrically opposite port omitted from the drawing for the sake of clarity of disclosure.

The piston **24**, sleeve **42** and cylinder **21** define the expansible fluid chamber **31** as above noted. Openings **57** in the end of the sleeve **42** permit free flow of oil throughout the chamber **31**. Flow check valves **33** and **35** operate in the manner explained above. Whenever port **53** is covered by piston **24**, the pump operates in the usual manner through check valves **33** and **35**. Whenever port **53** is uncovered back flow in conduit **55** causes negligible pressure to be developed in the oil chamber **31**. Thus by adjusting the axial position of sleeve **42**, and thus of port **53**, the effective stroke of the pump, and hence its rate of delivery is altered.

When port **53** is in its full left-hand position (FIG. 2), the pump is idling and delivering little or no fluid. As the port is moved to the right, the fluid delivery is increased. In the idling position, there is merely a surge of oil from inlet **33**, through chamber **31** and port **53** to the oil sump. This might cause heating of the oil in the pump, but this can readily be remedied by an oil cooler (not shown) disposed in the line. If inlet check **33** and port **53** are connected to opposite sides of this cooler, oil will circulate therethrough on idling, since check **33** passes oil one way only.

The hydraulic pump piston **24** is of small effective cross-section and operates at relatively long strokes and high speeds. In order to reduce fluid friction losses, the sleeve **42** can be cut away almost completely, leaving material only to guide it and the piston, and to cover the port **53**. The piston itself is made somewhat longer than the stroke to ensure that during the effective pumping stroke the port **53** remains covered regardless of the adjustment setting. The oil leakage past the piston **24** within the hydraulic cylinder is piped back to the sump by conduit **39** as above noted.

A constant pressure system is provided with a slight change of FIG. 2. The screw member **50** and threaded portion **48** of rod **45** of that arragement are replaced by a control **60** as shown in FIG. 3. The control **60** includes a cylinder **61** of uniform bore **62** receiving a control piston **63**. The control piston **63** is connected to rod **65** which is connected by appropriate structure (not shown) to rod **45** for actuating the sleeve **42**. Packings **64** between rod **65** and the cylinder **61** seal the bore **62** of leakage as required. The control piston **63** defines on one side a pressure tight chamber **66** whereas the opposite side **67**, is vented through conduit **69** to the sump (not shown). Spring **70** engages the piston **63** and urges it to the right in FIG. 3.

A needle valve **72** connects the control chamber **66** to the sump. A relief valve **74** connects the high pressure side of the pumping system to the control chamber **66**. Valve **74** has a spring **75** which holds moving valve member **76** on the valve seat to control the valve opening **77**. The force of spring is adjustable by a control screw **78**. The pressure at which this valve opens depends on the setting of the control screw **78**. Even after relief valve **74** opens the flow through the valve increases rather slowly with increase of pressure; that is, the valve opening **77** is small.

The control **60** is appropriately connected to the sleeve **42** so that spring **70** in the control urges the sleeve to maximum pumping position, or to the right in FIG. 2. The forces acting on the control piston **63** are: that of spring **70** urging it to the right, fluid pressure in chamber **66** urging it to the left, that imposed by rod **65**, and a small value of friction. Fluid pressure in control chamber **66** when moving the control piston **63** moves the sleeve **42** to the left (FIG. 2) toward the idle or no pumping adjustment of the sleeve.

Assume that 100 p.s.i. is the minimum desired pressure to be maintained in the hydraulic system. Thus 100 p.s.i. in the control chamber **66** acting on the control piston **63** will fully compress the spring **70** and cause the sleeve **42** to move from full pumping to idle position. To establish pressure on the control piston **63**, the control screw **78** releases spring pressure on relief valve **74** which is hence easily opened. The leak through needle valve **72** being small, full system pressure soon appears in chamber **66** to act on the control piston **63**. At starting conditions, the sleeve **42** is in full pumping position and there is no pressure in the control chamber **66**. Fluid volume is delivered until the pump produces 100 p.s.i., which is conveyed to chamber **66** to move sleeve **42** to idle position; thereafter the oil flow stops but with the outlet oil at the set pressure.

Consider adjusting control screw **78** to increase the force of spring **75** to that capable of confining fluid pressures up to 1000 p.s.i. Until the output of the pump arrives at this pressure, full pump volume occurs, for, since relief valve **74** is closed, there is no pressure in the control chamber **66**. At 1000 p.s.i. output pressure relief valve **74** begins to open. The pressure build-up in the control cylinder **66** depends upon the rate of opening of relief valve **74** and the set leakage of needle valve **72**. Somewhat about 1000 p.s.i. output pressure the pressure in chamber **66** rises to 100 p.s.i. to move sleeve **42** to idle pump output. If the leak through needle valve **72** is small compared to the delivery from relief valve **74**, the pump will go from full to idle discharge with a small change in delivered pressure; and thus be very sensitive. If the leak through needle valve **72** is made larger, there is a greater pressure variation before the control piston **63** makes a full excursion. The rate of control can be further regulated to prevent hunting, for instance, with a needle valve (not shown) between relief valve **74** and the control **60**.

FIG. 4 shows another form of hydraulic pump that, at certain times, appears desirable over the arrangement previously described. The oscillator piston **3** is connected to a two-diameter hydraulic piston **81** having a large pump portion **82** and a smaller rod portion **83** extending through the end **84** of the pump cylinder **21**. The large pump portion **82** is sealed with appropriate packing **86** at the inward end of the pump cylinder, while the rod portion **83** is sealed by packing **87** at the outward end of the pump. The pumping of the oil from chamber **31** for each stroke, or the effective area of the pump piston **81**, is determined by the differential area between the piston portions **82** and **83**.

Reciprocation of the oscillator piston **3** causes the connected pump piston **81** to change the volume of the oil chamber **31**. Outward movement of the piston **81** (to the left as seen in FIG. 4) reduces the volume of chamber **31**. The inlet **33** and the outlet **35** check valves communicate with the oil chamber **31** and operate as above noted. Since the effective pumping volume per stroke in the oil cylinder is represented by the differential in cross-section of the two portions **82** and **83**, the desired cross-section as compared to the cross-section of the oscillator piston **3** can readily be provided.

Another means of adjusting the effective pumping stroke is shown in FIGS. 5 and 6. The smaller diameter rod portion **83** secured to the pump piston **81** and projecting from the cylinder end **84** is provided with a rounded ring **91** on its outward end. Bracket **92** mounted on the cylinder **21** supports a bearing shaft **93** extending parallel to rod **83**. Carriage **95** is mounted on bearing shaft **93** and includes a bar **96** extending parallel to the stroke of the pump piston. A slider **97** mounted on the bar **96** moves along the length thereof and is actuated by a power cylinder **98**, similar to the control **60** described above, to any position along the bar. Carriage **95** is free to rotate about shaft **93** in one direction, resting on stop **99** to oppose movement in the opposite direction.

As tahe ring **91** oscillates it contacts a roller **102** on the slider **97** and lifts the slider and carriage bar **96** momentarily. Carriage bar **96** thus strikes plunger **103** of valve **104** to actuate the valve. Valve **104** is a relief valve connected by conduit **106** to the pump chamber **31**, as will be described below.

Relief valve **104** is shown in FIG. 7. In this figure, as in all figures of the specification, only one port of the pair of ports is shown for simplicity. It will be understood, however, that, as in all balanced valves, each port is duplicated by one equal and diametrically opposite thereto.

The valve **104** consists of a cylinder **107** having an inlet port **108** from conduit **106** and an outlet port **109** to conduit **110**. Conduit **106** connects to the oil chamber **31**; while conduit **110** connects to the sump of the hydraulic system. A piston **111** having central annular recess **112** oscillates within the cylinder **107**. A small opening **113** connects the recess to one end chamber **115**. The ports **108** and **109** are generally aligned intermediate the ends of the cylinder **107** with port **109** being slightly further from end chamber **115**. Compression spring **116** on the other side of piston **111** urges the piston toward the end chamber **115** with a generally small force. Plunger **103** on the piston **111** extends to outside of the valve **104** and is adapted to be engaged by the carriage **95**, as above mentioned. Chamber **115** is connected through conduit **118** and a needle valve (not shown but similar to **72** above), to the sump. The spring end of the cylinder is vented by conduit **119** to the sump.

When ring **91** strikes the roller **102** to actuate the projecting plunger **103**, the piston **111** is moved vertically in the FIG. 7 to open the inlet port **108** from conduit **106** communicating with the pump chamber **31** to the recess **112**. This communicates pressure from recess **112**, through opening **113** to chamber **115** which gives a positive impulse on the piston acting against the force of light spring **116**. Port **109** is thus opened to permit bypass flow from conduit **106** across recess **112** to conduit **110**. The pressure drop through the outlet port **109** plus the controlled setting of the unshown needle valve hold the valve open, even though the force on plunger **103** is removed.

The action of relief valve **104** with the hydraulic cylinder **21** is as follows: The pump starts its power stroke (to the left in FIGS. 5 and 6) with relief valve **104** closed and oil delivery to output. At the outward point of the stroke determined by control **60** (as shown with solid lines in FIGS. 5 and 6) the valve **104** is opened and pumping through outlet check **35** stops as the fluid in chamber **31** is by-passed across the valve **104** to the sump. As was above mentioned, valve **104** remains open for the balance of the pump stroke (shown with phantom lines in FIGS. 5 and 6). On the return stroke of the pump piston **81**, spring **116** closes the valve ready for the next by-pass action.

In the arrangement shown, the ring **91** would also strike the roller **102** on the return stroke to actuate valve **104**. However, since there is little output pressure in the conduit **106**, there is no harm in this action. Of course, this extra action can be easily eliminated by simple additional actuating linkage (not shown) that responds only to outward piston movement.

It is noted that output adjustment by the mechanism shown in FIG. 2 causes the first part of the outward pumping stroke to by-pass fluid, while the remaining part of the stroke is the actual pumping stroke. FIGS. 5 and 6 disclose an arrangement by which the adjustment is made by causing the first portion of the outward stroke to be the pumping stroke, while the remaining portion is the by-pass stroke. Depending on the purposes desired, at times it is preferable that the active part of the pump stroke be at the beginning of the power stroke, instead of at the end. This avoids a sudden onset of pressure when the piston is at high velocity since the piston starts pumping from central piston position where the piston velocity is zero.

It is also noted that the adjustment arrangements of FIG. 2 and FIGS. 5 and 6, respectively, occur from structure within the pump cylinder and structure located outside of the pump cylinder. The arrangement shown in FIGS. 5 and 6 is of the constant pressure system operable to maintain a constant output oil pressure at variable flows. However, it is apparent that slider **97** can be actuated axially along the carriage bar 96 independently of pressure, if desired, by structure similar to the screw member 50 disclosed in FIG. 2.

While it is possible to construct an oscillator device as herein described using conventional piston construction involving piston rings and oil lubrication, it is preferable to utilize air lubrication of the piston as taught in previously mentioned Patent No. 2,983,098. The operating temperatures can be higher with improved efficiency since temperatures are not limited by the presence of lubricating oil.

The clearance between oscillator pistons and cylinder should be small, to minimize losses due to gas leakage past the pistons. The criterion for allowable clearance is the maximum tolerable leakage during full cycling operation of which the duration of the high pressure causing flow past is very brief. If, during this brief interval, the weight of gas lost from the high pressure volume is small compared to the total weight of gas in the volume, that is, say, one or two percent, the clearance is admissable.

It is desirable to construct the pistons and cylinders with small initial clearances and to maintain these during operation. If the thermal expansions of the cylinder and pistons are closely regulated throughout the starting, operating and stopping of the unit, the close minimum clearances can be maintained at all times. This is possible by maintaining certain temperature ratios in the cylinder and pistons consistent with uniform expansion thereof, or by eliminating or greatly reducing themal expansion of the cylinder and pistons.

A possible method for maintaining uniform clearance is to make the cylinder and pistons of certain materials and to hold the temperature changes thereof inversely proportional to the coefficients of expansion of the materials.

The oscillator cylinder can be maintained at a uniform and predetermined temperature by supplying it with a coolant jacket 122 (FIG. 1) through which a coolant flows. The coolant jacket can be divided into several parts, and the temperature of each controlled independently. For example, there may be one jacket 123 about the central portion of the cylinder, and separate interconnected jackets 124 and 125 about the ends. The fraction of the coolant flow through each is controlled independently by thermostats.

It is possible to cool the cylinder by circulating the working hydraulic fluid as the coolant fluid. If a silicone oil or some other high temperature oil is used as the working fluid, operating temperature can be as high as 600° F. to ensure high efficiencies. A heat exchanger can dissipate excess heat generated in the working fluid as required. An extremely sensitive positive action thermostatic valve suitable for holding constant temperatures will now be described, although other types similarly can be used.

FIG. 8 shows a thermostatic valve 128 having a uniform bore 129. A piston 131, with annular center groove 132, is operated in the bore 129 by bellows 133 and spring 135. Tube 136 is the inlet for the working fluid. Tube 137 connects to the coolant jacket. Tube 138 is the return from the coolant jacket. Tube 140 is the outlet for the working fluid which connects to another jacket or to the hydraulic system. Tube 141 drains leakage back to the sump. Bellows 133 is filled with a high pressure gas, or with a liquid which has a high and rapidly variable vapor pressure at the range of temperatures where control is to be expected.

The temperature of the returning oil from the jacket is a good measure of the cylinder temperature. The bellows 133 will be at nearly the temperature of this returning oil. When the jacket temperature is low the piston 131 is in its furthest raised position (FIG. 8), and flow to the jacket is small. There is a stop 142 for piston 131 so that there is always some oil moving from inlet tube 136 across groove 132 to tube 137 to the jacket. The main portion of the oil flows through the by-pass passage 144 to outlet tube 140. As the jacket returns temperature rises and approaches the set temperature, the piston 131 moves downwardly in FIG. 8 to increase jacket flow (to tube 137) and correspondingly throttle by-pass flow (to passage 144). Provision can be made for adjusting spring 135 to adjust the operating temperature.

The oscillator pistons tend to become hotter than the cylinder, for they receive heat directly from the very hot exhaust gases and the paths by which they lose this heat are not of high thermal conductance. In their motion they lose heat to the cylinder through a very thin film of air. Generally the temperature distribution in the pistons is not uniform since heat is received on one face and has to be conducted elsewhere. Uniformity of piston temperatures can be approached by several means to control the expansion of the piston.

The oscillator pistons can be cooled by a coolant, and if desired, by the working fluid with an arrangement as shown in FIG. 9. Oscillator piston 146 is hollow and defines an internal cavity 147. A smaller shell 149 defines a center volume 150 communicating to the outside through the openings 152 and 153 in the opposite ends of the shell. The shell 149 is disposed within cavity 147 of piston 146 and held in place therein by circumferentially spaced ribs 155 (only one of which is shown). A large tube 156 is connected to the piston 146. A smaller diameter tube 158 is positioned within tube 156 and secured to outer end of the pump cylinder 21. Tube 156 moves relative to tube 158 upon the stroke of the piston 146, the tube 158 sliding freely within opening 152. The tubes 156 and 158 define an annular passage 160 between the tubes and a passage 161 within the tube 158 communicating, respectively, with cavity 147 and the center volume 150.

The difference between the exterior cross-sections of tubes 156 and 158 determines the change in volume per stroke in the expansible fluid chamber 31. The external cross-section of inner tube 158 determines the change of volume per stroke within the volume 150. The total pump output is the sum of the two volumes and is communicated across check 35, as previously mentioned, and through conduit 162 having outward check 163. Balanced type thermostatic valves 164 are in the outlet lines operable to regulate flow from conduit as determined by the required coolant flow.

Upon inward movement of the piston, oil is communicated from chamber 31 through the annular passage 160 and cavity 147 to volume 150. Depending upon the setting of the thermostatic valves 164, the effective flow through the conduit 162 and thus the piston can be varied as desired. However, regardless of the through-flow in piston 146, the fluid therein is circulated at least partially for every pump stroke.

FIG. 10 shows a section of an oscillator piston thermally controlled by a coolant circulated within the piston by structure disposed therein. The hollow oscillator piston 165 defines chamber 166 communicating through passage 167 in hollow connecting rod 168 with the oil chamber (not shown) of the hydraulic cylinder at the outward end of the rod. A tube 169 communicating with the passage 167 extends to a remote portion of the chamber 166. Light check valves are shown at 170 and 171 to permit flow as indicated by the arrows. A bellows 172 is located within a separate volume 174 sealed from the piston chamber 166 by a valve 177. The bellows is filled with compressed gas at approximately 50 to 60 p.s.i. (being greater than the inlet oil pressure but less than the output pressure). Thus on applying greater external pressure to the bellows, say of 100 p.s.i., the bellows collapses against a stop (not shown) and decreases to a minimum volume, so that further external pressure increases do not collapse the bellows further. Thermostat 178 opens valve 177 when the temperature of oil in piston chamber 166 rises to the predetermined operating temperature. The change in volume of bellows **172** is a fraction of the volume displacement per stroke of the hydraulic pump, approximately 10%.

When thermostat **178** is closed there is no difference in action from that of a solid piston, except for a small effect due to oil compressability. When thermostat **178** opens, bellows **172** alters in volume each stroke to circulate oil within the piston chamber **166** through the check valves **170** and **171**. This limited oil circulation acts to maintain uniform piston temperature and thus uniform piston expansion.

FIG. 11 shows another pump construction particularly adaptable for operation with a completely circulating coolant through an oscillator piston of construction similar to that of FIG. 12. The pump cylinder **21** includes a bore **180** having inlet check valve **182** and outlet check valve **183** communicating therewith at its opposite ends. A through-passage **185** communicates with the bore **180** at ports **186** and **187** in general alignment with the check valves **182** and **183**. A thermostat valve, which can be similar to that shown in FIG. 6, is shown schematically at **189**. Conduit **190** is connected to a relief valve similar to valve **104** shown in FIG. 7.

The piston pump **192** includes a multiple diameter arrangement having an outward smallest portion **193**, and intermediate largest portion **195** in close proximity with the periphery of the bore **180**, and an inward intermediate sized portion **196**. The end portions **193** and **196** of the piston pump **192** extend through the ends of the pump cylinder **21** as above noted and are sealed by appropriate means such as packings **198** and **199**. The pump chamber **31** thus is defined generally within the entire bore **180** in the volume **201** on the outward side of the intermediate portion **195** and that volume **202** defined on the inward side of intermediate portion **195**.

The inward portion **196** is formed from a tube **203** receiving a smaller tube **204**. Thus the tubes **203** and **204** define an annular passage **206** and a passage **207**. Radial openings **209** in the outward portion **193** of the piston communicate the defined volume **201** with the passage **207** in the smaller tube **204**. Radial openings **210** in the inward portion **196** of the piston (in tube **203**) communicate the defined volume **202** with the annular passage **206**.

The intermediate portion **195** of the piston **192** need not form a sealing fit with the periphery of the bore **180** but is sufficient if it presents a high resistance leakage path for the fluid. Reciprocation of the pump thus causes pumping of fluid from pump chamber **31** out check valve **183** as the differential area of the piston portions **193** and **196**. The fluid from volume **201** is caused to flow: (1) through the by-pass passage **185**; (2) past the intermediate portion **195** and the periphery of the bore; and, (3) throught openings **209** to passage **207**, through piston structure to be described, and returned through the annular passage **206** and radial openings **210**, to the defined volume **202** and out the outlet check **183**. When thermostat control **189** is wide open there is evidently very little circulation through the piston, on account of relative resistance of paths. When thermostat control **189** is closed nearly the full volume pumped is forced through the piston path. The operation here is that, while there is flow through the piston in both directions, the flow in one direction is much greater than in the other, the ratio being determined by relative areas of piston portions **193** and **196**. Thus, on the outward stroke, for example, 120% of pumped volume would pass through the piston, and on the return stroke 20% would pass back.

An engine piston structure particularly adaptable for a completely circulating coolant fluid is shown in FIGS. 12 and 13. The outer tube **203** and inner tube **204** define separate flow passages **206** and **207**. Instead of having oil flow merely within the piston **212**, the oil flows through longitudinal holes **214** in the wall **215**. These

holes **214** are connected to the gratings or manifolds **216** and **217** in ends of the piston, each manifold being connected to only one of the passages **206** and **207**. The inner tube passage **207** is connected directly to the inward manifold **216**, which communicates with the longitudinal holes **214**, which communicates with the outward manifold **217** and thence to annular passage **206**.

It is desirable that the flow through the piston be controlled by the difference in temperature between piston and cylinder. The coolant flow will thus be regulated to hold these two temperatures nearly equal, even during transient periods such as starting up, when temperatures have not reached their controlled maximums. A thermostat for this purpose could be similar to FIGURE 8, except that the spring **135** is replaced by a second bellows (not shown). One bellows is bathed in oil returning from the cylinder and the other by oil returning from the piston. When the piston temperature is higher than that of the cylinder, the valve moves to increase piston flow, and vice versa. The bellows in this case are preferably filled with gas of rather high pressure, to give nearly constant sensitivity over the range of operation.

When the hydraulic working fluid is also the cooling fluid, there is a practical limit to the operating temperature of cylinders and pistons, even when using a fluid such as silicone oil. When efficiency is very important, it will be desirable to increase the operating temperature higher than this. The coolant and the working fluids can then be separated.

Another way for maintaining uniform clearances is to make the pistons of material with very low coefficient of expansion, and to cool the cylinder only. Two constructions of such a piston are possible. The first is to use a solid piston of Lithrafax (tradename of The Carborundum Company), quartz, pyrex, porcelain, or the like; while the second is to use a low expansion material that is protected from the heat by insulating slabs on the exposed piston ends.

Consider first a solid piston made of one of the materials listed above. Generally, each of the materials is brittle, but has high compressive strength. The pistons are subjected to compressive stresses only; and they receive no sudden blows. All the materials can readily be made to very high precision at moderate cost of fabrication.

The coefficient of expansion of fused quartz, in the range from 0 to 800° C. is about $0.55\times10^{-6}/°$ C. The coefficient of expansion of Lithrafax is about $0.69\times10^{-6}/°$ C. Thus a piston 4" in diameter heated from 70° to 570° F. expands on the radius about only .0003". It is thus possible to make a piston the expansion of which in operation is very small. The only temperature control then necessary is to sufficiently cool the cylinder to maintain the clearance generally constant in operation to hold down flow-past losses.

The temperature distribution of the piston can be computed accurately if the hotter face is assumed at a uniform temperature, and the resistance across the air film is neglected in comparison with the resistance of the body of the material, which is quite justifiable. This permits computation of the actual shape of the piston after it has come to temperature equilibrium. The ends of the piston can be tapered slightly, a few thousandths of an inch to an inch, to allow for this expansion, and for the small portion of the piston ends that gets really hot.

The improvement of thermal efficiency by letting the face of the piston become hot should be substantial. At the time of fuel injection, the active volume in the cylinder is bounded principally by faces of pistons, and by only a narrow ring of cylinder wall. Even with reduced cylinder wall temperatures, the heat of the piston faces largely is transferred to the gas during the compression, and somewhat during the latter part of the power stroke. Furthermore, to ensure completely atomized fuel and properly mixing with the air, fuel can be intentionally injected against the hot piston faces to vaporize quickly. With rapid vaporization better fuel burning should result, and the time of injection before dead-center reduced.

The second method available for controlling the expansion of the piston without actually cooling the piston is to use a low expansion metal piston faced with slabs of refractory material. The slabs would be of low conductivity, preferably but not necessarily low expansion coefficient, with full stability at maximum operating temperature and of adequate strength. Nickel steel and porcelain have the required characteristic and thus could be materials of choice. In mounting the slabs on the metal piston, to counteract the differential expansion of the porcelain and the steel over a given temperature change, a metal sheet having nearly the same expansion coefficient as the rear face of the slab can be used. The porcelain can be bonded to the metal sheet with the metal sheet in turn being brazed to the end of the piston.

To reduce the effect between the differential expansion of the metal sheet and the piston end, it is desirable that the following construction be used. The slab of porcelain has fuzed to it a thin sheet of metal. The end of piston has relatively narrow concentric rings terminating on a planar face and defining deep concentric grooves. The slab is brazed onto the face of the rings. The outer ring is not included, so that the expansion of the slab will not distort it. The outer surface of this last ring is tapered slightly. The slab diameter is less than cylinder diameter by an amount amply sufficient to take care of slab expansion. During expansion of the slab the rings are bent, but well within their elastic limits, as is the stress on the slab and metal sheet due to expansion.

For the body of the piston, when used with end slabs, low expansion nickel steel, with about 42% nickel, is acceptable. Its conductivity is relatively low so that its temperature will rise slightly in conducting heat from the face slab to the air film, but because of its low coefficient of expansion, thermal expansion will be small. It appears that a solid piston of this material, faced with porcelain slabs, will have almost negligible expansion in use.

It is necessary to use care in regard to the method of interconnecting the large oscillator piston and the small hydraulic piston. In the construction shown in FIG. 1 (also shown in FIG. 10), the connecting rod 26 therein shown has a large diameter portion 220 over most of its length, and small diameter portions 221 and 222 adjacent each piston. Both the small and large portions are designed to be stable as a column under the loads to be encountered. However, the short small diameter portions 221 and 222 are relatively flexible. This allows the pistons 3 and 24 to adjust in alignment, through very small angles without bringing great resisting couples to bear. The rod 26 is connected at portion 222 to piston 3 near its center of gravity. The thrust, even if slightly offset from center, then has very small tendency to throw the pistons 3 and 24 off of axial alignment. The assembly is constructed with the cylindrical outer surfaces of the pistons in line with the axis of the rod 26.

It is also possible and at time desirable to interconnect the pistons in a manner to maintain each piston centered in its cylinder. FIG. 4 shows one end of an oscillator cylinder 1, and an oil cylinder 21 mounted thereon. The oscillator piston 3 is secured to the oil pump piston 81 made in two diameters 82 and 83. The moving piston assembly 3–81 is made sufficiently stiff that deflection because of gravity or applied forces is virtually eliminated, the longitudinal center axis and circumferential surfaces always remaining concentric and parallel. Special journals 224 and 225 of the type to be described hold the piston assembly 3–81.

The basis of the construction lies in the positive centering feature of a hydrostatic bearing. See page 52 of Oil Hydraulic Power and Its Industrial Applications, Second Edition, by Walter Ernst. The shaft (formed by portions 82 or 83) passes through a cylindrical journal (225 or 224, respectively) having a comparatively shallow annular passage intermediate adjacent land areas. From this passage a number (3 or more) of circumferentially spaced holes extend radially to a connection (not shown) to a source of high pressure oil. In each of these radial holes is a fine restriction, made by pressing in a tube of small bore generally only a few thousandths of an inch in diameter. While the hydraulic working oil is not considered as a good lubricating oil, it can be used in this type bearing since metal-to-metal contact is prevented.

Such a bearing has a very strong centralizing effect on the shaft. The shaft can move axially and/or rotate at relatively high speed. The leakage of oil can be held to a few cubic inches per minute, and this can be caught and returned to the source. To prevent most oil leakage from flowing out the ends of the bearing, internal grooves near the end of the bearing can pick it up, and these can be drained to return the oil.

The radial clearance in the bearings 224 and 225 (between the shaft and land areas) should be small compared to the clearance of the oscillator piston 3, such as .0002″ compared to .001″. If the parts are accurately made and concentric and not subjected to forces from expansions or the like to throw them out of line, the distortion of the moving piston assembly 3–81 of FIG. 4 is negligible. Then, evidently, piston 3 can never come in contact with periphery of bore 2 of the cylinder. In fact, its clearance can never depart far from .001″ at any point because of the strong centralizing effect of bearings 224 and 225. Hence the oscillator can be operated with as small piston clearance as desired, limited only by the practical mechanical precision, symmetry of construction and the uniformity of materials, etc.

With a piston supported by gas flotation, it is necessary to have a rather large ratio of length to diameter to obtain best performance. With the present positive support means there is no such limitation. It thus is entirely possible to increase the piston diameter and correspondingly shorten its length and stroke to new limits regulated by the consideration of leakage only. Such a "square" design makes for stiffness of the moving part, low gravity deflection, and in some ways better oil pump construction. It also gives a more compact oscillator larger in diameter and shorter in overall length. Since the effective oil pumping area is the difference between the areas of the two rods, the proper pumping value for the oscillation size can readily be provided.

The part of the total piston stroke used for action of the buffer is of some importance in determining the design of the unit. If the buffer volume is too small, the maximum buffer pressure increases to increase leakage past the piston during buffer action. The same sort of flow-past loss occurs here, as in the diesel volume, and buffer flow-past loss occurs due to flow past only a fraction of the piston length, namely from the face of the piston to the compressor ports, rather than past the full piston length. Both losses depend intimately on piston clearance, which in turn depends on the practical precision attainable in clearance control. Conversely, if the buffer volume is made too large, the fraction of stroke used for pumping of scavenging air is reduced, and scavenging will probably be made incomplete.

Consider a unit with an oscillator having a piston stroke of about S in each piston. The power stroke of .7S drives the piston outwardly from the cylinder center. The blower ports are open this portion of the stroke to put into the reservoir a volume of air for scavenging at a pressure of a few p.s.i. The blower ports then close, while the exhaust valve opens. The buffer action begins and lasts approximately .3S, making this long to keep down flow-past losses from the buffer volume. Shortly after the exhaust valve opens, the inlet check opens to permit air flows from the reservoir through the cylinder. Since the piston keeps on moving outwardly the .3S after the inlet valve opens, there is plenty of time for complete scavenging.

The total length of the oscillator cylinder will be from 3 to 4 times the stroke, varying with various piston lengths, port arrangements etc. Each oil pump cylinder needs to be S long, plus room for piston length, sleeves, bearings, etc. In some designs, rods project out of these a distance slightly greater than S. The overall lengths of the subject free piston hydraulic pump will be from 6S to 12S depending on design.

The unit can be started generally in a manner disclosed in the above-mentioned Patent 2,983,098, with the following modifications. It is possible to bring the pistons to center by clamping shut the outlet valves of the hydraulic cylinders and subjecting the oil chambers 31 to oil under moderate pressure from a small reservoir kept full by a check valve during normal operation. This positions the oscillator pistons near their point of closest approach. The outlet valves are then unclamped and the source of moderate pressure disconnected from the chamber 31. A short pulse of high pressure air admitted between the main pistons starts the oscillation. An air tank for this purpose is also kept pumped up automatically. It is readily possible to provide that all of these steps will be taken in proper sequence upon pressing a single starting button.

An injection control system can be used as disclosed in the above-mentioned Patent No. 2,983,098. It is sometimes desirable however that injection be positively timed. This can be effected in the following manner by the use of a valve actuated directly by the reciprocating piston assembly.

A projection on the piston assembly can be used to actuate the valve. This projection is so placed that, near the end of the piston stroke toward center, it impacts the stem of a small poppet valve connected in a line with the injector control and a high pressure air reservoir automatically recharged. The possible motion of this valve stem should be long enough so that it will not bottom even if the stroke departs considerably from normal. When the poppet valve closes it should open a vent to release pressure from the injectors to actuate the injection. The whole valve should be movable axially, and held in position by a screw, so that it may be set for the optimum timing of injection.

While specific embodiments of the subject free piston hydraulic pump have been shown, it is apparent that many modifications can be made therein without departing from the concept of the invention. Accordingly, it is desired that the subject invention be limited only by the scope of the following claims.

What is claimed is:

1. A free piston hydraulic pump unit, comprising in combination, a free piston engine and a hydraulic pump, the output of the engine providing the input for the pump, said engine and pump each including a cylinder, an engine piston reciprocable in the engine cylinder and having a uniform radial clearance, a pump piston reciprocable in the pump cylinder and defining therewith an expansible oil chamber, means interconnecting the engine and pump pistons for related dependent movements, means adjustable relative to the stroke of the pump piston for adjusting the hydraulic output of the unit when operating at a constant piston stroke and cyclic frequency, and means including a thermostat, coolant, and coolant passage means within the engine piston, cylinder and interconnecting means operable to maintain related temperatures thereof for maintaining said radial clearance uniform throughout the operating conditions of the unit.

2. A free piston hydraulic pump unit, comprising in combination, a free piston engine and a hydraulic pump in operational interconnection, the output of the engine providing the input for the pump, said engine including a cylinder of uniform bore, a piston reciprocable in the engine bore and having a uniform diameter less than that of the bore to produce uniform radial clearance, said pump including a pump cylinder having a bore in general alignment with the longitudinal center axis of the engine bore, a pump piston reciprocable in the pump cylinder and defining therewith an expansible oil chamber, inlet and outlet means for the oil chamber, means mechanically interconnecting the engine and pump pistons for related dependent movements, means adjustable axially of the stroke of the pump piston for adjusting the effective output stroke of the unit when operating at a constant piston stroke and cyclic frequency, and means including a thermostat, coolant, and coolant passage means within the engine piston, cylinder and interconnecting means operable to maintain related temperatures thereof for maintaining said radial clearance uniform throughout the operating conditions of the unit.

3. A free piston hydraulic pump unit, comprising in combination, a free piston engine and a hydraulic pump, the output of the engine providing the input for the pump, said engine including a cylinder of uniform bore, a pair of pistons reciprocable in the engine bore each having a uniform diameter less than that of the bore establishing uniform radial clearances, said pump including a pump cylinder disposed outward of each end of the engine cylinder and having a bore in general alignment with the longitudinal center axis of the engine bore, a pump piston reciprocable in each pump cylinder and defining therewith an expansible oil chamber, inlet and outlet means for each oil chamber, means mechanically interconnecting the adjacent engine and pump pistons for related dependent movements, means including structure adjustable axially of the stroke of the pump piston for adjusting the output of the unit when operating at a constant piston stroke and cyclic frequency, and means including a thermostat, a coolant for, and coolant passage means in the engine piston and cylinder to maintain the temperatures thereof related for maintaining said radial clearances uniform throughout the operating conditions of the unit.

4. A free piston hydraulic pump unit, comprising in combination, a free piston engine and a hydraulic pump, the output of the engine providing the input for the pump, said engine including a cylinder of uniform bore, a piston reciprocable in the engine bore and having uniform radial clearance, said pump including a pump cylinder having a bore in general alignment with the longitudinal center axis of the engine bore, a pump piston reciprocable in the pump cylinder and defining therewith an expansible oil chamber, inlet and outlet means for the oil chamber, means mechanically interconnecting the engine and pump pistons for related dependent movements, means for adjusting the output of the unit when operating at a constant piston stroke and cyclic frequency, said adjusting means including a member movable in line with the piston stroke to adjusted positions along the stroke, passage means from the chamber by-passing the outlet means, said member being operatively associated with said passage means for determining the portion of the stroke effected by said passage means, and means to move the member, and means including a thermostat, coolant, and coolant passage means within the engine piston, cylinder and interconnecting means operable to maintain related temperatures thereof for maintaining said radial clearance uniform throughout the operating conditions of the unit.

5. A free piston hydraulic pump unit, comprising in combination, a free piston engine and a hydraulic pump each having at least one reciprocable piston in direct mechanical interconnection, the output of the engine providing the input for the pump, said engine and pump including axially aligned cylinders each of uniform bore, the engine and pump pistons being freely received in the respective bore and each having a generally uniform radial clearance, said pump cylinder and piston defining an expansible oil chamber, inlet and outlet means for each oil chamber, means mechanically interconnecting

the engine and pump pistons, means for adjusting the hydraulic energy output of the unit when operating at a constant piston stroke and cyclic frequency, and means for maintaining said radial clearances uniform throughout the operating conditions of the unit, said last-mentioned means including at least jacket means disposed about the cylinder operable to carry a coolant therethrough, means including a thermostat operable to vary the flow of coolant responsive to variation from a desired temperature thereof, and means for controlling the thermal expansion of the engine piston.

6. A free piston hydraulic pump unit, comprising in combination, a free piston engine and a hydraulic pump, the output of the engine providing the input for the pump, said engine including a cylinder of uniform bore, a pair of pistons reciprocable in the engine bore each having a uniform diameter less than that of the bore establishing uniform radial clearances, said pump including a pump cylinder disposed outward of each end of the engine cylinder and having a pump bore in general alignment with the engine bore, a pump piston reciprocable in each pump bore and defining therewith an expansible oil chamber, inlet and outlet means for the oil chambers, means mechanically interconnecting the engine and pump pistons for related dependent movements, means for adjusting the hydraulic output of the unit when operating at a constant piston stroke and cyclic frequency, said adjusting means including a member movable parallel to the direction of the stroke to a plurality of positions along the stroke, means generally independent of the outlet means for removing hydraulic output from the oil chambers, said member being operatively associated with said independent means for related operation therewith to determine the effective stroke of the pistons, and means to move the member, and means for maintaining said radial clearances uniform throughout the operating conditions of the unit, said last-mentioned means including at least coolant means on the cylinder operable to maintain a desired temperature thereof, and means for controlling the thermal expansion of the engine piston.

7. A free piston hydraulic pump, comprising in combination, a free piston engine and a hydraulic pump in mechanically defined interaction, the engine having a cylinder of uniform bore, a pair of engine pistons of uniform diameter mating in the bore, means to drive the engine pistons apart from a position of generally closest approach, means to reverse the movement of the engine pistons after a piston stroke outward from the closest approach, the hydraulic pump directly utilizing the outward movement of the engine pistons for transformation to output in the form of hydraulic energy, said hydraulic pump including a pump cylinder at each outward end of the engine cylinder and having a bore in genreal alignment with the engine bore, a pump piston reciprocable in each pump bore and defining therewith an oil chamber of maximum volume with the piston being at the closest approach, inlet and outlet means for each oil chamber, means for interconnecting mechanically each pump piston and its adjacent engine, means for adjusting the output of the unit at constant piston stroke and cyclic frequency, said adjusting means including structure movable along a path generally in line with the piston stroke to a plurality of positions, means associated with the structure and communicable with the oil chamber operable to vary the effective piston stroke with respect to the actual piston stroke, and means to adjust the position of the member, and means for maintaining radial clearances of the engine piston and cylinder substantially uniform throughout the operating conditions of the unit, said last-mentioned means including at least coolant passage means on the cylinder operable to maintain a desired temperature thereof, and means for controlling the thermal expansion of the engine piston.

8. A free piston hydraulic pump unit, comprising in combination, a free piston engine and a hydraulic pump, the engine having a cylinder of a uniform bore, a pair of pistons received in the bore in general sealing relationship with the periphery thereof, said engine pistons being driven apart by a combustion cycle between the pistons initiated when the pistons are at generally closest approach, said hydraulic pump directly utilizing the outward movement of said engine pistons for transformation to output in the form of hydraulic energy, said pump including a pump cylinder disposed at each end of the engine cylinder and defining a pump bore in general alignment with the engine bore, a pump piston reciprocable in each pump bore and defining therewith an expansible oil chamber of maximum volume with the piston being at closest approach, inlet and outlet means for each oil chamber, means interconnecting each pump piston and its adjacent engine piston and forming a unitary assembly having common movements, said interconnecting means having a generally uniform exterior cross-section at least along the portion thereof intermediate its connection with the pistons, means surrounding the intermediate portion operable to separable the bores of the engine and pump from each other, means operable to adjust the effective stroke of the unit when operating at a constant piston stroke and cyclic frequency, said adjusting means including structure movable generally in line with the piston stroke and having at least one portion thereof adapted to be traversed by at least one portion of the assembly, and means to move said structure, and means for maintaining the radial clearance of the engine piston and cylinder uniform throughout the operating conditions of the unit, said last-mentioned means including at least jacket means disposed about the cylinder operable to carry a coolant therethrough, means including a thermostat operable to vary the flow of coolant responsive to variations from a desired temperature thereof, and means for controlling the thermal expansion of the piston.

9. A free piston hydraulic pump according to claim 8, wherein said interconnecting means is formed by a rod having said intermediate portion thereof of high degree stiffness throughout its length, and wherein the portions of the rod proximate the connections with the pistons being of generally lesser degrees of stiffness with at least the connection of the rod to the larger of the pistons being generally at the longitudinal and lateral center of said piston.

10. A free piston hydraulic pump unit according to claim 8, wherein said pump piston and interconnecting means define a composite rigid structure having at least a second generally uniform exterior cross-section outward of, in line with, and less than, said first-mentioned cross-section, the engine piston being supported rigidly at one end of the rigid structure and disposed with its outer peripheral surfaces in line with those of the composite sections, and spaced bearing means supported by the pump cylinder and closely surrounding, respectively, cross-sections operable to maintain the unitary assembly uniformly disposed within the cylinder bores.

11. A free piston hydraulic pump unit, comprising in combination, an engine cylinder having a uniform bore, a pair of engine pistons of uniform diameter mating in the cylinder each having a uniform radial clearance, means to drive the engine pistons apart from a position generally at closest approach, means to reverse the direction of the engine pistons at the end of the outward stroke, means for utilizing the outward stroke of the engine pistons for transformation to hydraulic energy, said last-mentioned means including a hydraulic pump having a pump cylinder disposed outward of the engine cylinder and having a bore in general alignment with the bore thereof, a pump piston in each pump cylinder defining an expansible oil chamber of maximum volume at said closest approach, inlet and outlet means for each oil chamber, means interconnecting mechanically each pump piston with the adjacent engine piston, means adjustable

axially of the stroke of the pump piston for adjusting the effective output stroke of the unit when operating at a constant piston stroke and cyclic frequency, and means for maintaining said radial clearances uniform throughout the operation of the unit, said last-mentioned means controlling the thermal expansion of the cylinder and pistons and including a coolant, coolant passage means and temperature responsive means for the engine operable to maintain related temperatures of the cylinder and pistons.

12. A free piston hydraulic pump unit according to claim 12, wherein said coolant passage means include at least one coolant jacket disposed about the cylinder and operable to carry coolant fluid therethrough, and wherein said temperature responsive means include at least one thermostat operable to vary the flow of coolant fluid responsive to variations from the desired temperature thereof.

13. A free piston hydraulic pump unit according to claim 12, wherein said coolant passage means further include the engine pistons defining internal coolant chambers operable to receive therein coolant fluid, the coolant fluid being conveyed to said chambers from outside of the pistons by the coolant passage means further including at least one passage in such interconnecting means, and wherein said temperature responsive means further include at least one thermostat for varying the coolant circulation within the piston chambers.

14. A free piston hydraulic pump unit, comprising in combination, a free piston engine having a cylinder of uniform bore, a piston of uniform diameter received in the bore and having generally uniform radial clearance, said piston defining with the bore a fluid power cylinder of given effective area, and means to reciprocate the piston in the cylinder responsive to a combustion cycle therein, and a hydraulic pump directly utilizing the energy of said engine piston responsive to the combustion cycle for transformation to output in the form of hydraulic energy, said pump including a pump cylinder defining a pump bore, a pump piston reciprocable in the pump bore and defining therewith an expansible oil chamber of maximum volume before the initiation of the combustion cycle, the effective pumping area of the pump piston being substantially less than said effective engine area, inlet and outlet means for the oil chamber, means for interconnecting the pump piston and the engine piston for common related movements, means adjustable axially of the stroke of the pump piston to regulate the effective output of the unit when operating at a constant piston stroke and cyclic frequency, and means including a thermostat and a coolant and coolant passage means in the engine piston and cylinder and in the interconnecting means for maintaining the temperature thereof related to maintain the radial clearance substantially uniform throughout the operation of the unit.

15. A free piston hydraulic pump unit according to claim 14, wherein said adjustable means include a member disposed outside of the pump cylinder and movable generally in line with the stroke of the pump piston to a plurality of adjustable positions, said member having at least one portion thereof traversed by at least a portion of the pump piston during the stroke, means including a by-pass passage operatively associated with the member at said one portion thereof and communicating with the oil chamber effective to by-pass output therefrom independently of said outlet means, and means to adjust the member between the adjusted positions.

16. A free piston hydraulic pump according to claim 14, wherein said last-mentioned means include passage means formed in part in the interconnecting means from the oil chamber to a coolant chamber defined within the engine piston, and wherein thermostat means in the passage means regulate coolant circulation of the working fluid within the coolant chamber responsive to variations from the desired temperature thereof.

References Cited in the file of this patent

UNITED STATES PATENTS

2,978,986	Carder et al.	Apr. 11, 1961
2,981,198	Nettel	Apr. 25, 1961
2,983,098	Bush	May 9, 1961