March 4, 1952 J. A. ROBERTS ET AL 2,588,164

PNEUMATICALLY POWERED HYDRAULIC PUMP

Filed April 7, 1948 6 Sheets-Sheet 1

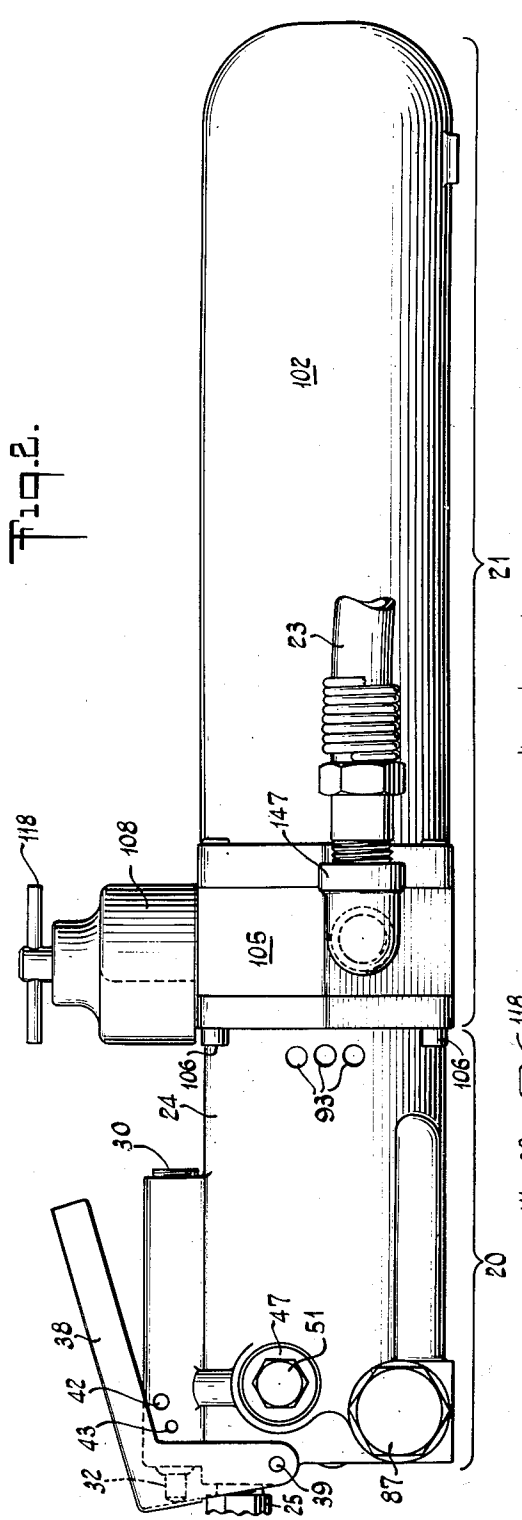


Fig. 2.

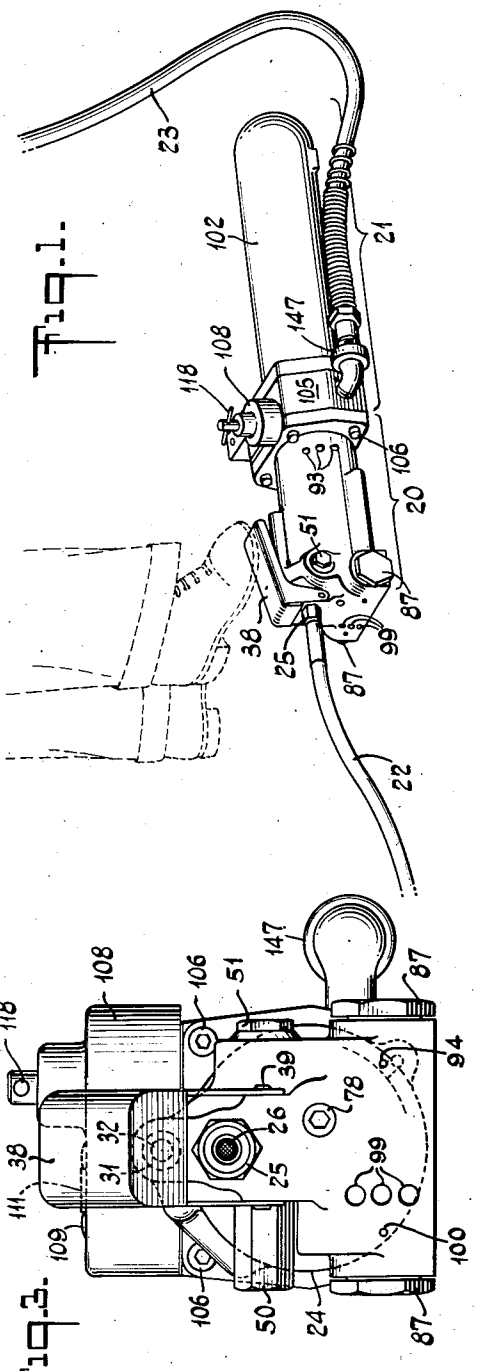


Fig. 1.

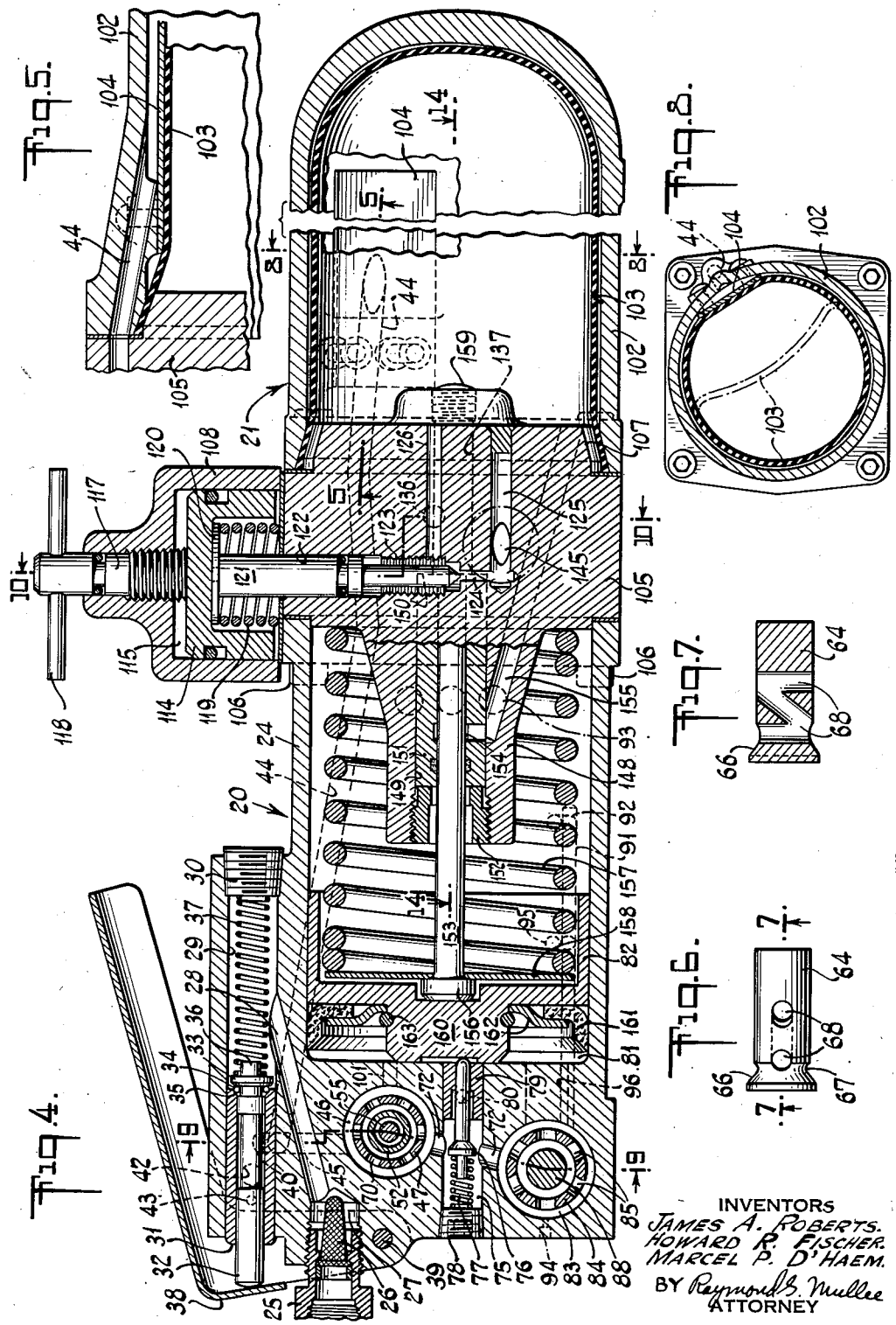
Fig. 3.

INVENTORS
JAMES A. ROBERTS
HOWARD R. FISCHER
MARCEL P. D'HAEM.
BY Raymond G. Mullee
ATTORNEY

March 4, 1952 J. A. ROBERTS ET AL 2,588,164

PNEUMATICALLY POWERED HYDRAULIC PUMP

Filed April 7, 1948 6 Sheets-Sheet 2



INVENTORS
JAMES A. ROBERTS.
HOWARD R. FISCHER.
MARCEL P. D'HAEM.
BY Raymond B. Mullee
ATTORNEY

March 4, 1952 J. A. ROBERTS ET AL. 2,588,164

PNEUMATICALLY POWERED HYDRAULIC PUMP

Filed April 7, 1948 6 Sheets-Sheet 3

Fig. 9.

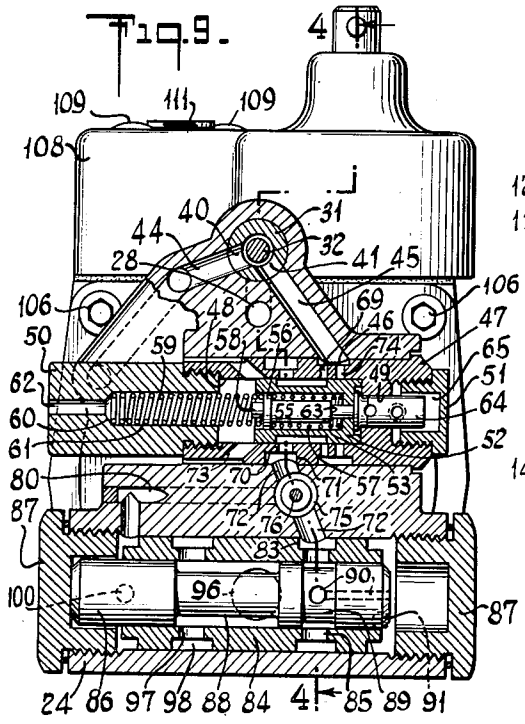


Fig. 10.

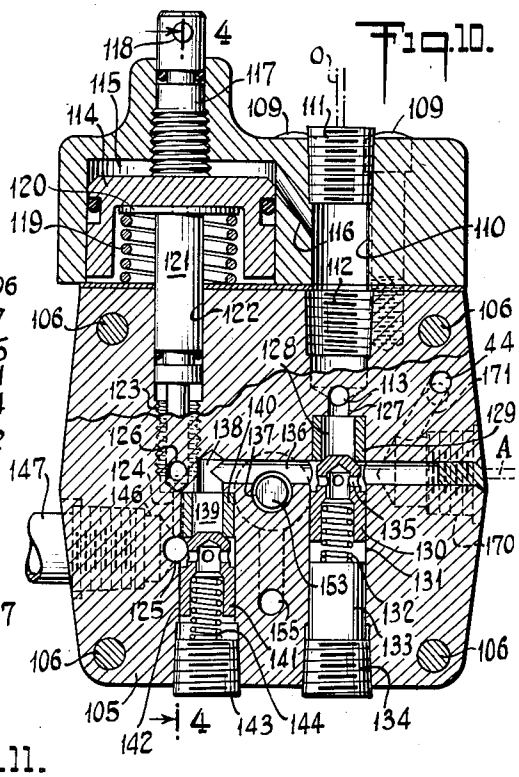
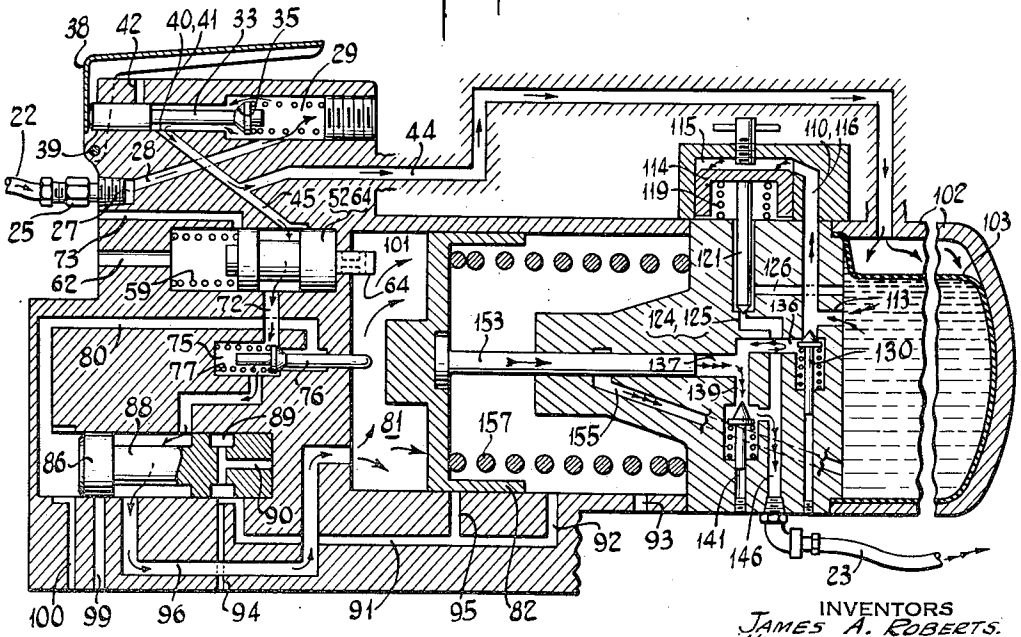


Fig. 11.



INVENTORS
JAMES A. ROBERTS.
HOWARD R. FISCHER.
MARCEL P. D'HAEM.
BY Raymond G. Mullee
ATTORNEY

March 4, 1952 J. A. ROBERTS ET AL 2,588,164

PNEUMATICALLY POWERED HYDRAULIC PUMP

Filed April 7, 1948 6 Sheets-Sheet 4

Fig. 12.

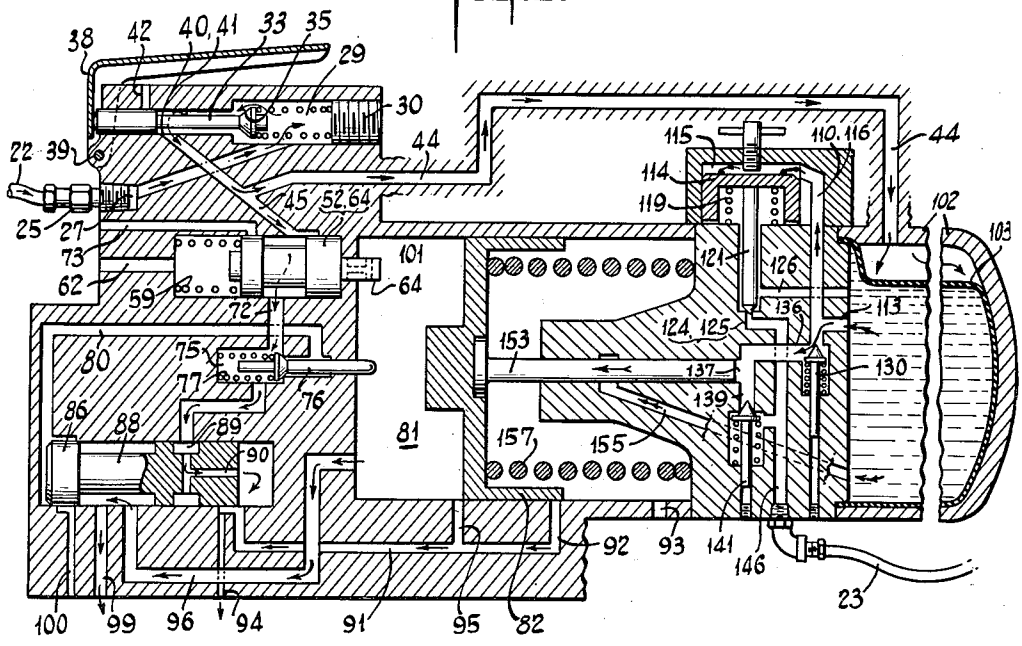
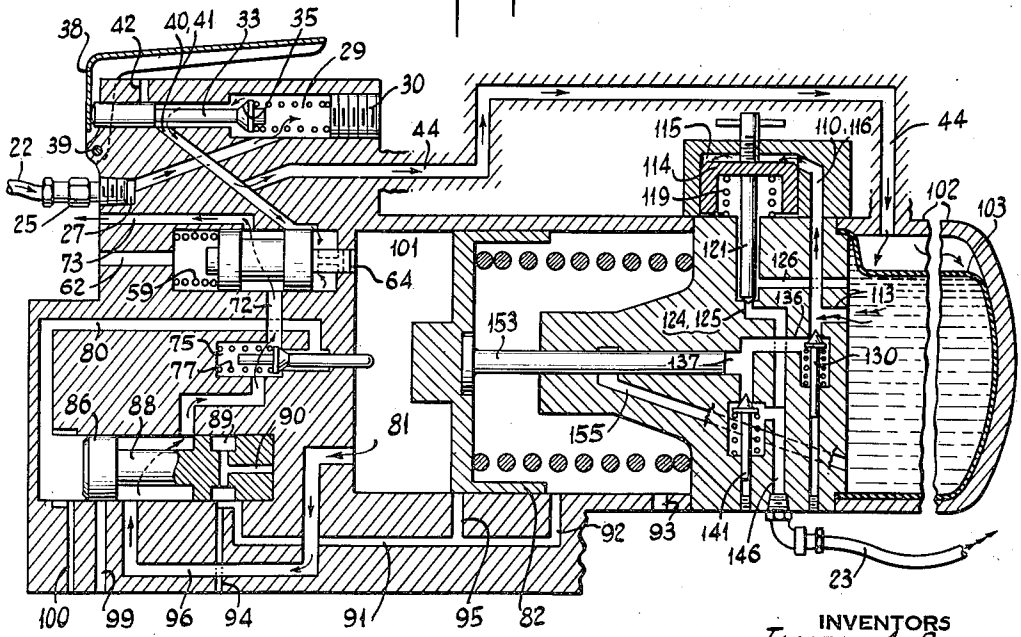


Fig. 13.



INVENTORS
JAMES A. ROBERTS.
HOWARD R. FISCHER.
MARCEL P. D'HAEM.
BY Raymond G. Mullee
ATTORNEY

March 4, 1952 J. A. ROBERTS ET AL 2,588,164

PNEUMATICALLY POWERED HYDRAULIC PUMP

Filed April 7, 1948 6 Sheets-Sheet 5

Fig. 14.

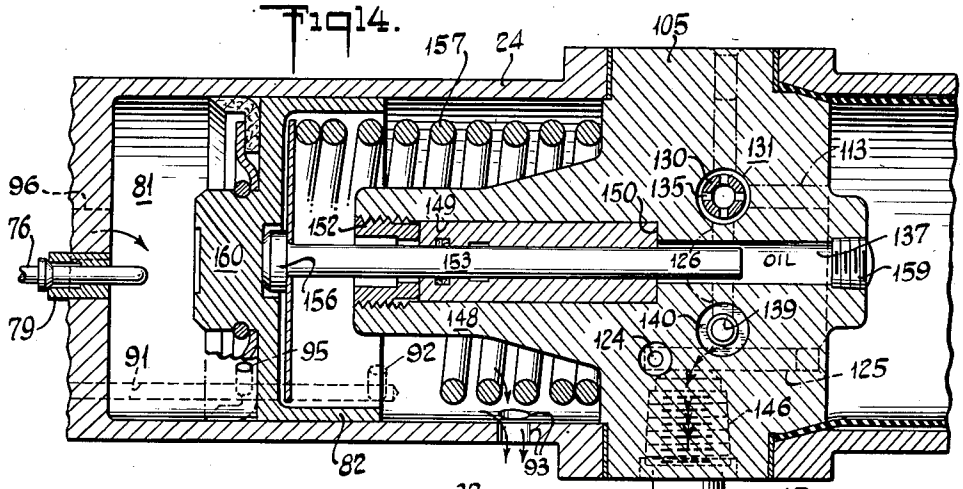


Fig. 15.

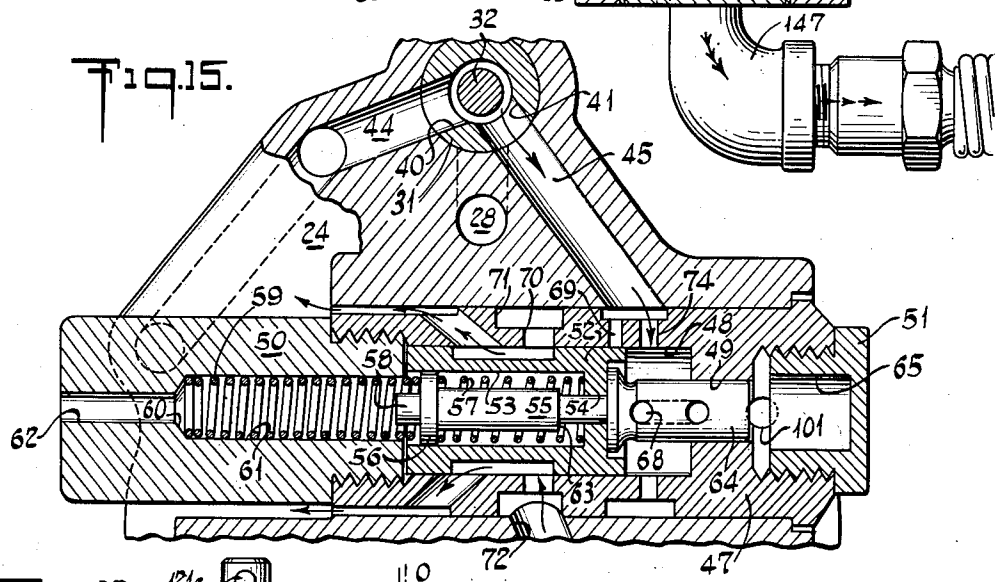


Fig. 16.

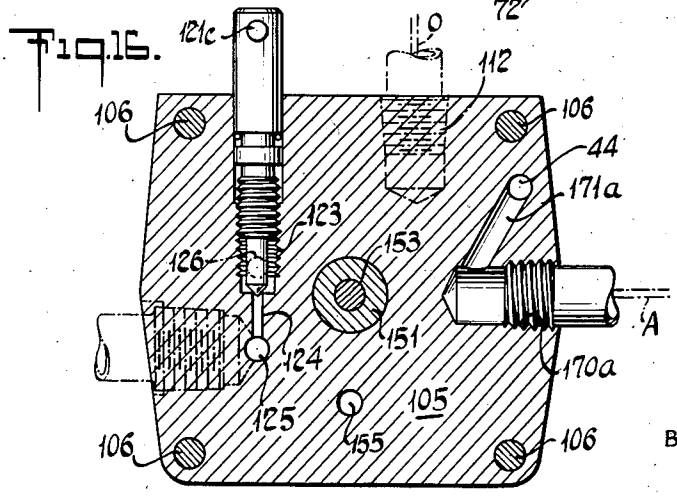
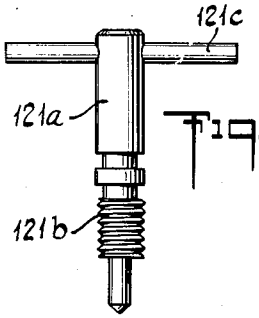


Fig. 17.



INVENTORS
JAMES A. ROBERTS.
HOWARD R. FISCHER.
MARCEL P. D'HAEM.
BY Raymond G. Mullee
ATTORNEY

March 4, 1952 J. A. ROBERTS ET AL 2,588,164

PNEUMATICALLY POWERED HYDRAULIC PUMP

Filed April 7, 1948 6 Sheets-Sheet 6

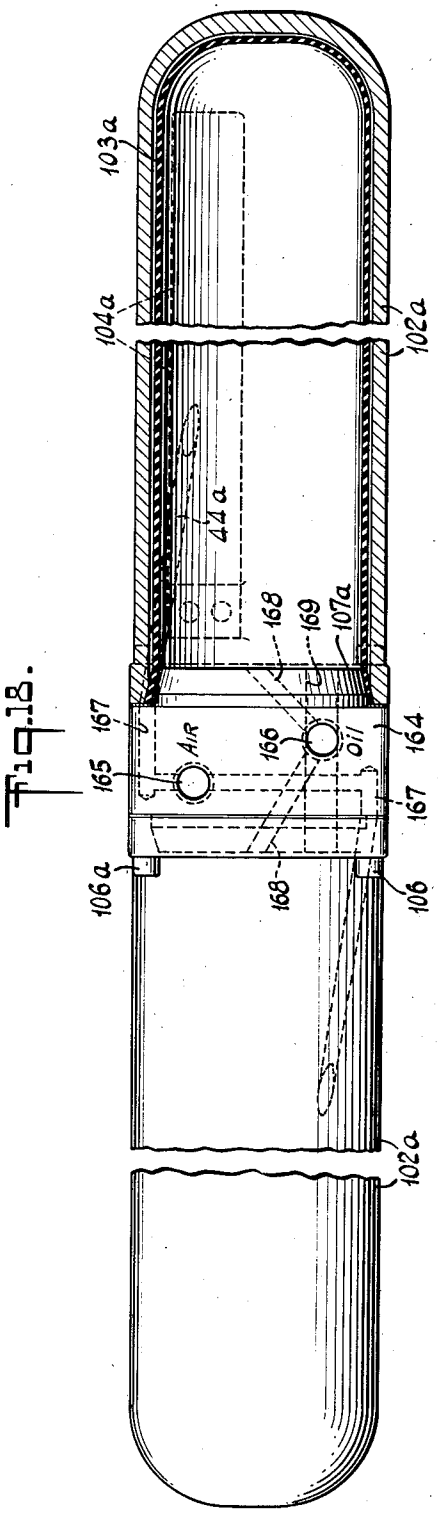


Fig. 18.

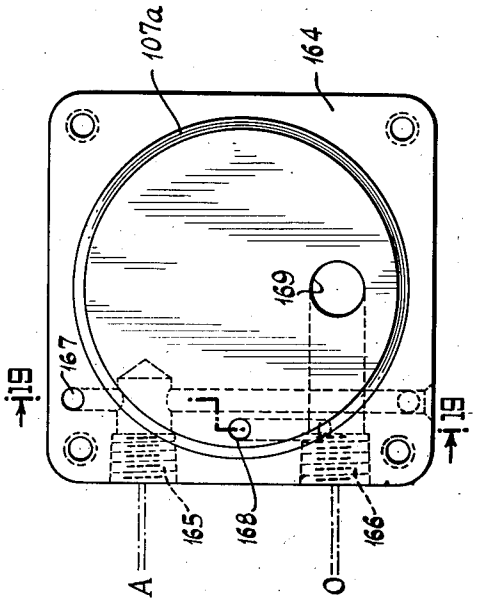


Fig. 20.

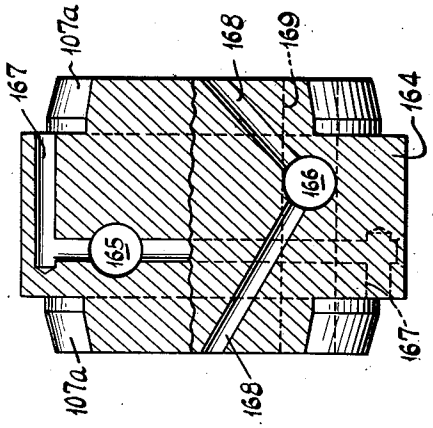


Fig. 19.

INVENTORS
JAMES A. ROBERTS
HOWARD R. FISCHER.
MARCEL P. D'HAEM.
BY Raymond G. Mullee
ATTORNEY

Patented Mar. 4, 1952

2,588,164

UNITED STATES PATENT OFFICE

2,588,164

PNEUMATICALLY POWERED HYDRAULIC PUMP

James A. Roberts, Howard R. Fischer, and Marcel P. D'Haem, Detroit, Mich., assignors to Chicago Pneumatic Tool Company, New York, N. Y., a corporation of New Jersey

Application April 7, 1948, Serial No. 19,478

16 Claims. (Cl. 103—48)

1

This invention relates to pumps and more particularly to pumps which comprise a pneumatic and hydraulic combination.

In the smaller type industrial shops and especially in service garages for motor vehicles, it is often required to have a high-pressure hydraulic source for such tools as power presses, rams, body and frame jacks, cylinder liner removers, as well as other hydraulic service tools. It is usual for such shops and garages to maintain a live air line for various purposes, and the present pump has been adapted to operate with the usual pressure carried by such an air line. With an air line pressure of 90 pounds per square inch gauge, the pump will deliver a hydraulic pressure of 8,000 pounds per square inch gauge. It is a feature of the invention that the control means are extremely versatile in that the hydraulic pressure delivered can be automatically maintained or released after pump operation has stopped.

The prime object of this invention is to provide a new and novel pump which will furnish a high pressure hydraulic delivery when connected to a low pressure pneumatic source.

Another object of this invention is to provide a pump which is simple to install and operate, has a variety of applications, has a versatile control arrangement, and is reliable under all service demands.

A further object of this invention is to provide a pump which has low initial and maintenance costs, is structurally compact, and of relative small size and low weight.

These and further objects and features of the invention will become more apparent when viewed in the light of the following specification and the accompanying drawings wherein:

Fig. 1 is a perspective view of a pump embodying the invention;

Fig. 2 is an enlarged side elevation view of the pump;

Fig. 3 is an end view of the pump as seen from the throttle or back end in Fig. 2;

Fig. 4 is a further enlarged longitudinal section of said pump, as indicated by the lines 4—4 in Figs. 9 and 10, with the parts in non-operative position;

Fig. 5 is a fragmentary longitudinal section, as indicated by the line 5—5 in Fig. 4, showing the oil reservoir bag maintained in spaced relation from its housing;

Fig. 6 is an elevation view of a pressure regulator valve used in the pump, the figure being double size as compared with Fig. 4;

2

Fig. 7 is a longitudinal section taken along the line 7—7 in Fig. 6;

Fig. 8 is a cross-section as indicated by line 8—8 of Fig. 4, but on a smaller scale, showing the oil reservoir bag inflated in full lines and partly collapsed in broken lines;

Fig. 9 is a cross-sectional view of the pump as indicated by the irregular line 9—9 in Fig. 4, showing particularly the valves controlling the supply of live air to the cylinder;

Fig. 10 is a cross-sectional view of the pump as indicated along the irregular line 10—10 in Fig. 4, showing the inlet check valve, discharge check valve and pressure release valve in the hydraulic system;

Figs. 11, 12 and 13 are diagrammatic views of the pump showing respectively the relative position of the various parts during an early stage of the power stroke, return stroke, and just after maximum pressure has been reached;

Fig. 14 is a longitudinal section, looking downward as indicated by the arrows 14 in Fig. 4, showing the parts just prior to the attainment of maximum pressure;

Fig. 15 is a fragmentary cross-sectional view in the same plane as Fig. 9 but on double scale and showing the pressure regulator valve and distributor valve shifted, in response to the attainment of maximum pressure, to automatically cut off the supply of live air to the cylinder;

Fig. 16 is a cross-section through the oil valve block, similar to Fig. 10 but with a modified pressure relief valve of the manual or screw type;

Fig. 17 is an elevational view of the screw type valve used in Fig. 16;

Fig. 18 is a two-third size elevation view of an auxiliary reservoir system which is adapted to be connected by an air hose and an oil hose to the valve block of Fig. 10 or Fig. 16;

Fig. 19 is a full-size sectional view of a part of same taken along line 19—19 in Fig. 20; and

Fig. 20 is an end view of the part shown in Fig. 19.

The pump is shown in Fig. 1 in operating position, it being understood, as will later appear evident, that the pump can be readily controlled by foot or hand, and will operate satisfactorily in any position with regard to the relation of the pump axis to the horizontal plane, any particular operating set-up being in accordance with the greatest convenience thereby afforded. The pneumatic and hydraulic portions are indicated generally therein by the respective numerals 20 and 21 (Figs. 1, 2 and 4). An air supply line or hose 22 is connected to the pneumatic

portion 20, while a high-pressure hydraulic delivery line or hose 23 is connected at one end to the hydraulic portion 21, the other end of the hose 23 being connected to a hydraulically powered tool, such as a ram, not shown. The supply and delivery hoses 22 and 23 respectively are preferably of the flexible type to facilitate portage of the pump, as will be appreciated. The pump is designed to deliver a maximum hydraulic pressure of approximately 8,000 pounds per square inch gauge, when using a 90 pound per square inch gauge air supply. However, intermediate hydraulic pressures can be obtained, if desired, by proper throttle control, as will later be seen.

In constructing the pump advantage has been taken of all known mechanical expedients to insure a maximum of efficiency in its operation. In the specification, no specific mention is made ordinarily of elements such as piston rings or seals, gaskets, and the like, but it should be understood that these elements are provided in the pump wherever necessary.

Pneumatic system—Live air supply

Considering first the structure of the pneumatic portion 20 and referring to Figs. 2, 3, 4, 6, 7 and 9, the numeral 24 indicates a cylinder, being most conveniently in the form of a casting, which is arranged to house and support the various elements required for the operation of the pneumatic cycle. The air hose 22 is connected to a standard ¼ inch pipe connection 25 which is screwed to the rear end wall of the cylinder 24, the connection 25 being provided with an air strainer 26 (Fig. 4), which is arranged to partially extend into a cavity 27, formed in the cylinder end wall. An air passageway 28 in the cylinder 24, leads upwards from the cavity 27 and into a throttle chamber 29 which is located at the top of the cylinder 24.

The throttle chamber 29 is cylindrical in form and has an axis which is generally parallel to the axis of the pump proper. One end of the throttle chamber 29 is sealed by means of a standard ¼ inch pipe plug 30, while at the opposite end, a bushing 31 is press-fitted into the chamber for a distance of a little less than one-half the overall length of the chamber.

A throttle valve 32 is slidably fitted in part within the bushing 31, the forward portion of the throttle valve having a collar 33, which operatively serves to force a sealing ring 34 against a seat 35 formed on the inner or front end of the bushing 31. Forwardly of the collar 33, the valve has a pilot portion 36 which serves to position a compression spring 37 located within the throttle chamber 29, and interposed between collar 33 and a pipe plug 30 which closes the front end of throttle chamber 29. The rear portion of the throttle valve 32 projects in part beyond the bushing 31, the rear extremity of the valve being in contact at all times with a throttle lever 38, which is generally L-shaped, one leg of which is bifurcated and pivotally connected to the cylinder 24 by means of a pin 39, the other leg of the lever 38 extending in a direction over the throttle chamber 29 as shown. The throttle valve 32 has a reduced diameter mid-portion, which is of such length as to uncover ports 40 and 41 (Figs. 4, 9 and 15), within the bushing 31, during all positions of the throttle valve. An exhaust port 42 extends through the bushing 31 and cylinder 24, and is so positioned that it is uncovered by the reduced diameter mid-portion of the throttle valve 32, when said valve is in non-operative position, and is covered by the rear portion of the throttle valve, when said valve is in full operative position. An oil hole 43, extending through the cylinder 24 and bushing 31, is provided for the convenient lubrication of the throttle valve 32.

It will now appear obvious that the throttling arrangement is such that when throttle lever 38 is fully depressed, the throttle valve 32 will be forced to a forward or full operative position, thereby covering exhaust port 42, unseating the ring 34, further compressing the spring 37, and thus allowing air in the throttle chamber 29 to pass around the reduced diameter mid-portion of the throttle valve, and enter simultaneously ports 40 and 41. When the throttle lever 38 is released, spring 37 (assisted by the pressure of the live air) will force the throttle valve 32 to a backward or non-operative position, thereby seating the ring 34 to restrict further air passage from the throttle chamber 29, and will simultaneously uncover exhaust port 42 to allow exit to the atmosphere of air in passageways leading from ports 40 and 41.

The port 40 opens into an air passageway 44 (Figs. 9 and 15) in the cylinder 24, said passageway extending forward to another section of the pump, the detailed description and function of which will be discussed later in the specification. The port 41 opens into an air passageway 45 (Figs. 4, 9 and 15) which passes downwardly in the cylinder and into a circumferential recess 46 formed on the outer surface of a valve bushing 47.

The valve bushing 47, which is best illustrated in Fig. 15, is cylindrical in form and is press-fitted within a transverse bore in the body of the cylinder 24, in the manner shown. The valve bushing 47 is open from end to end and has two smooth finished bores, a larger bore 48 and a smaller bore 49 which is concentric with and adjacent to bore 48. Both ends of the valve bushing 47 are internally threaded, the threads concentric with and adjacent to the larger bore 48 being adapted to receive a distributor valve stop 50, while the threads at the opposite end of the valve bushing are concentric with the smaller bore 49 and are adapted to receive a pressure regulator valve cap 51.

Distributor valve

Reciprocally positioned within the larger bore 48 is a distributor valve 52, which has a reduced external diameter mid-portion and is internally provided with a counterbore 53 and bore 54, both of which are smooth finished. Within the larger bore 53 of the distributor valve 52, is positioned a pressure regulator valve push pin 55, which has a narrow integral collar 56, slidably fitted to the bore 53. Also positioned in bore 53 is a distributor valve spring 57 which is maintained under compression, and which surrounds a major portion of the push pin 55, and is interposed between collar 56 and the bottom of the counterbore in the distributor valve 52. Beyond collar 56, the pin 55 has a pilot 58 for the positioning of a pressure regulator valve spring 59, maintained under compression and interposed between the collar 56 and a shoulder 60 in a bore 61 of the distributor valve stop 50. A hole 62 within the valve stop 50 is provided for open communication of the bore 61 with the atmosphere. The push pin 55 has a reduced diameter, at the end opposite the collar 56, which is arranged to slidably reciprocate within the bore 54 of the distributor valve 52. A shoulder 63 formed on the push

pin 55 limits movement of the push pin in one direction.

Pressure regulator valve

A pressure regulator valve 64 has a stem portion of cylindrical form reciprocably arranged in part within the bore 49 of the valve bushing 47 and projecting into a cavity 65 formed in the valve cap 51, the diameter of said cavity being somewhat greater than the diameter of the adjacent portion of the valve 64. The opposite end of valve 64 abuts against distributor valve 52. The regulator valve 64, best seen in Figs. 6 and 7, has a shoulder 66 (engageable with bushing 47 to limit travel of the valve in one direction), and a circumferential groove 67 which connects at two points with a zig-zag hole arrangement 68 within the valve 64, the function of said groove 67 and hole arrangement 68 being discussed in the operation of the pump as found later in the specification.

The distributor valve 52 and regulator valve 64 are shown in Fig. 9 in a non-operative position. In such position a plurality of ports 69, radially positioned in the distributor valve bushing 47 allow for the passage of live air from the circumferential recess 46 to the bore 48 of the valve bushing 47, around the reduced diameter mid-portion of the distributor valve 52, through a plurality of radially arranged ports 70 in the valve bushing 47, into a circumferential recess 71 formed on the outer surface of the valve bushing 47, and finally into an air passageway 72 which extends downwardly in the cylinder 24. However, when the distributor valve 52 and regulator valve 64 are shifted in the valve bushing 47 to the extreme left or operative position, i. e. when the distributor valve 52 is abutting the valve stop 50, as shown in Fig. 15, ports 69 are covered by the distributor valve 52, and a plurality of ports 73 in the valve bushing 47 are placed in communication with the reduced diameter mid-portion of the distributor valve 52 as shown. Thus in the operative (Fig. 15) position, any air under pressure in passageway 72 will pass through recess 71 and ports 70, around the reduced diameter mid-portion of the distributor valve 52, through ports 73, around a reduced diameter end portion of the valve bushing 47, and out to the atmosphere. It is to be noted that a plurality of ports 74 radially disposed within the valve bushing 47, are uncovered by the distributor valve 52 when in operative position, the reason for which will be discussed in the operation of the pump, as found later in the specification. Summarizing, the distributor valve 52 normally connects the passageway 72 to live air, as shown in Fig. 9, until displaced by the pressure regulator valve 64, after which it connects passageway 72 to exhaust, as shown in Fig. 15.

Shift valve

The air passageway 72 exits into a shift valve chamber 75 (Figs. 4 and 9) which contains a shift valve 76 and a shift valve spring 77 maintained under compression. One end of the spring 77 abuts against a pipe plug 78, which closes the outer end of chamber 75, while the opposite end abuts against a shoulder on the shift valve 76. The shift valve 76 has an elongated projection which is snugly fitted for reciprocation in a shift valve bushing 79, which is press-fitted within the cylinder 24 at the inner end of the shift valve chamber 75. Forwardly of the shoulder of shift valve 76 is a conical surface which at times seats against the end of the shift valve bushing 79. The shift valve bushing 79 has a rearwardly open counterbore which is somewhat larger than the diameter of the shift valve elongated porjection. An air passageway 80 opens into the counterbore, the function of said passageway being discussed shortly. The elongated projection of the shift valve 76 projects beyond the end of the shift valve bushing 79 into a main air chamber 81 in the cylinder 24, and is in contact with a piston 82 during the pump non-operative position, and also during a part of the movement of the piston 82. It can be seen that when the pump is in non-operative position, and while the shift valve 76 remains unseated by contact with the piston 82, egress of air from the shift valve chamber 75 takes place along two paths, the first of which is a continuation of the passageway 72 and which passes downwardly in the cylinder 24, the other path being around the elongated porjection of the shift valve 76, within the counterbore of the shift valve bushing 79, and then through the passageway 80. When the pump is in operative position, movement of the piston 82 permits seating of the shift valve 76, due to the action of the spring 77, cutting off flow of air through the passageway 80, but allowing for the flow of air through passageway 72.

Automatic valve

Passageway 72 leads into a circumferential recess 83 (Fig. 9) formed on the exterior surface of an automatic valve bushing 84, which is press-fitted to a cylindrical transverse bore in the cylinder 24, as shown. A plurality of radially disposed ports 85, connect the recess 83 to the bore of the valve bushing 84, while reciprocally arranged and snugly fitted within the valve bushing is an automatic valve 86. Each end of the transverse bore for the valve bushing 84 has a threaded portion which is adapted to receive a valve bushing cap 87, each valve bushing cap having a cavity which is concentric with the automatic valve 86, and which has a diameter somewhat larger than the maximum diameter of the automatic valve 86. The automatic valve 86 has two reduced diameter portions 88 and 89, portion 88 being of greater length than portion 89, whereas portion 89 has one end of a hole arrangement 90 which provides an air passageway between portion 89 and the right extremity of the automatic valve 86, as seen in Fig. 9.

When the automatic valve 86 is in the extreme left or non-operative position, two discrete circuits for air flow are established. In the first of the two circuits, the air in passageway 72 finds principal exit through circumferential recess 83 and ports 85, around reduced diameter portion 89, and through the holes 90 of the automatic valve 86, then through a passageway 91 (Figs. 4, 9 and 14) in the cylinder 24, and forward to cylinder port 92 in front of piston 82, then out to the atmosphere through ports 93, which are best seen in Figs. 2 and 14. Some of the air in the above described circuit also finds egress through a small vent 94 which is formed in the back wall of the cylinder 24, said vent being approximately in line with the passageway 91. It is to be noted that the passageway 91 has a second port 95 which opens into the main air chamber 81, said port 95 being covered by the peripheral wall of the piston 82 during a major portion of the stroke of the piston. The function of port 95 will be further discussed

in the description of the pump operation which follows in the specification.

In the second of the two circuits, any air in the main air chamber 81 finds exit through a passageway 96 (Figs. 9 and 14) which connects the main air chamber to the bore of the valve bushing 84, around the reduced mid-portion 88 of the automatic valve 86, through a plurality of radially disposed ports 97 in the valve bushing 84, to a circumferential recess 98 on the outer surface of the valve bushing 84, then on to the atmosphere through a series of ports 99 formed in the rear wall of the cylinder 24, the ports 99 best seen in Fig. 3. A small vent 100 (Figs. 3 and 9) connects the cavity in the left bushing cap 87 with the atmosphere, the purpose of which will be discussed in the operation of the pump, as found later in the specification.

A final detail in the structure of the pneumatic portion 20 of the pump concerns a short air passageway 101 (Figs. 4 and 15) which connects the main air chamber 81 and the cavity 65 of the regulator valve cap 51, the function of the passageway 101 being discussed later in the specification.

Hydraulic system

Considering now the structure of the hydraulic portion 21 and referring to Figs. 2, 4, 5, 8, and 10, the numeral 102 indicates a reservoir for housing an ail reservoir bag 103, which is made of a flexible material preferably a synthetic rubber, such as neoprene. The oil reservoir bag 103 contains oil and completely fills the reservoir 102 when the pump is in non-operative position, with the exception of a longitudinal region on the inside of the reservoir 102, wherein a flat flexible strip 104 (Figs. 4 and 5) keeps the bag 103 away from the inner wall of the reservoir for the entire length of the strip 104. The strip 104 is riveted at one end to the wall of the reservoir 102 and is so arranged that the air passageway 44 exits into the space formed between the strip and the inner wall of the reservoir. In this manner the compressed air emerging from passageway 44 during pump operation, will act along the entire length of the oil reservoir bag 103 and thus avoid uneven collapse of the bag under pressure. In Fig. 8 the dashed lines represent a showing of a partially collapsed position of the oil bag 103.

The reservoir 102 is securely attached to an oil valve block 105 (Fig. 4) by means of four screws 106 (Figs. 2 and 11), which pass through the block 105 and are arranged to also securely attach the cylinder 24 by flange means thereon, to the opposite side of the block 105. The air passageway 44 is arranged to pass through the oil valve block 105 so that an unrestricted path is provided for the flow of air between those portions of the passageway 44 in the cylinder 24 and in the reservoir 102 respectively. The open end of the oil reservoir bag 103 is wedged between a tapered pilot 107 on the oil valve block 105 and a matched tapered surface on the inner end of the reservoir 102, so that a liquid tight seal is thereby effected.

Hydraulic pressure release

In the preferred embodiment the pump is so designed that the hydraulic pressure developed can be released simultaneously with release of the throttle lever 38, by an automatic pressure release valve; or if desired, by a manipulative member after the throttle valve has been released, in case the operator decides to lock the automatic pressure release valve in closed position.

The numeral 108 indicates an automatic pressure release valve cylinder block which is securely attached to the top of the oil valve block 105 by means of two screws 109 (Fig. 10). An oil filler hole 110, passes downwardly through the valve cylinder block 108 and continues into the oil valve block 105. A pipe plug 111 is screwed into the entrance to the oil filler hole 110 in the valve cylinder block 108. A threaded portion 112 is arranged at the mouth of the oil filler hole 110 in the oil valve block 105, to receive the plug 111 in the event that the automatic pressure release valve means is to be substituted by a screw type pressure release means, hereinafter described in connection with the modification of Fig. 16. An oil passageway 113 in the oil valve block 105, connects the oil filler hole 110 with the open end of the oil reservoir bag 103, as shown in Figs. 10 and 14.

An automatic pressure release valve piston 114 is positioned for reciprocal movement within the valve cylinder block 108. On the upper side of the piston 114 is formed a chamber 115 (Figs. 4 and 10), which is connected to the oil filler hole 110 by means of an oil passageway 116 sloping downwardly in the valve cylinder block from the upper region of the chamber 115. It will be apparent that with such an oil passageway arrangement, the pneumatic pressure exerted on the oil in the oil reservoir bag 103 forces oil up into the chamber 115, and the unit hydraulic pressure attained by the oil in the bag 103, is transmitted to the upper surface of the piston 114. Projectable within the chamber 115 is a pressure release valve block screw 117, which can be rotated by means of a handle 118, for engagement with the piston 114, in the event that the operator desires to prevent the automatic opening of the release valve.

On the under side of the piston 114 is a pressure release valve spring 119 (Figs. 4 and 10) maintained under compression at all times, and which is in abutting relationship with the under side of the piston 114 through the intermediary of a washer 120, the opposite end of the spring being in abutment with the top surface of the oil valve block 105. Concentrically positioned beneath the piston 114 is a pressure release valve 121, having the form of a cylindrical plunger, and which is adapted for reciprocal movement within a smooth finished bore 122 extending downwardly in the oil block 105. The lower portion of the release valve 121 has a reduced diameter which terminates in a conical point. At the lower end of the bore 122, and surrounding a major portion of the reduced diameter of the release valve 121, is a threaded portion 123 which is adapted to function in the event that the screw type pressure release valve is substituted as shown in Fig. 16. At the bottom of the bore 122, and concentrically arranged with the axis of the bore 122 and release valve 121 respectively, is a small oil passageway 124 (Figs. 4, 10 and 14) which extends downwardly within the oil valve block 105 and connects with a larger diameter longitudinal oil passageway 125. Another longitudinal oil passageway 126 connects the threaded portion 123 of the bore 122 to the open end of the oil reservoir bag 103, as illustrated in Figs. 4 and 10.

When the conical point of the pressure release valve 121 is seated in the opening of the passageway 124, the flow of oil from passageway 125

through passageway 124, about the reduced diameter portion of the valve 121, through the passageway 126, to the oil reservoir bag 103, is prevented; and conversely, when the pressure release valve is unseated, the flow of oil through said passageways 124 and 126 to the oil reservoir bag 103, is permitted, thereby providing a short circuit between the high and low pressure ports of the hydraulic system for a purpose hereinafter set forth.

Hydraulic pumping circuit

A short vertical oil passageway 127 (Fig. 10) conects the longitudinal passageway 113 to a chamber 128 formed in a bushing 129, which is positioned above a first check valve 130 (Figs. 10 and 14). The check valve 130 is adapted for reciprocable movement within a smooth finished vertical bore 131 formed in the oil valve block 105, the lower end of said bore opening on the bottom surface of the block 105. The check valve 130 is tubular in form, the top end being closed and having a beveled edge engageable with a valve seat on the lower end of bushing 129 which functions to close the bottom of chamber 128. Within the check valve 130 and extending therefrom, is a compression spring 132 which abuts against an internal shoulder in the check valve 130 at the upper end, and a check valve spring spacer 133 at the lower end. The spring spacer 133 is smoothly fitted to the bore 131 and is positionally maintained therein by means of a pipe plug 134, which is threadably affixed at the lower end of said bore 131. At the upper end of the check valve 130 is a plurality of ports 135 which are radially disposed in the wall of the check valve, the outside diameter of the check valve 130 at the region of the ports 135 being somewhat less than the outside diameter of the lower portion of the check valve. Entering into the check valve bore 131 and adjacent the bottom edge of the bushing 129, is a transverse oil passageway 136.

Check valve 130 functions as an inlet valve for the pump. It will be apparent that the operation of check valve 130 is mainly dependent upon the relative pressure of the oil in chamber 128 and passageway 136. When the pressure of the oil in chamber 128 acting against the top of the closed end of the check valve 130, is greater than the combined pressure of the oil in passageway 136 and the compressive pressure of the spring 132 acting on the bottom of the closed end of check valve 130, the check valve will be unseated, and oil will flow from the chamber 128, around the reduced diameter portion of check valve 130, to the passageway 136. Conversely, when the pressure of the oil in chamber 128 is less than the combined pressure of the oil in passageway 136 and the compressive pressure of the spring 132, the check valve will be seated and there will be no flow of oil between the chamber 128 and the passageway 136.

The passageway 136 leads through an oil pressurizing chamber 137 (Figs. 10 and 14), which is cylindrical in form and which is concentrically arranged about the longitudinal axis of the pump, thence to a short passageway 138 which exits into a chamber 139 formed in a bushing 140, which is positioned above a second check valve or discharge valve 141. Check valve 141 is structurally like the check valve 130, and is reciprocably fitted to a smooth finished vertical bore 142 formed in the oil valve block 105. The lower end of the bore 142 opens into the bottom surface of the block 105 and is adapted to receive a pipe plug 143, by

means of a threaded portion at the end of the bore. A compression spring 144, arranged within and extending from the check valve 141, abuts against a shoulder therein at one end, and the pipe plug 143 at the other end, thereby serving to seat the check valve 141 at the lower inner edge of the bushing 140. An opening 145 (Fig. 4) in block 105, connects the check valve vertical bore 142 at the region adjacent the bushing 140 with the longitudinal oil passageway 125. The passageway 125 is also connected at the region of the opening 145, to a cavity 146 (Figs. 4, 10 and 14) formed in the side of the oil valve block 105, said cavity having a threaded portion adapted to receive an eli 147 (Figs. 2, 10 and 14), to which is threadably connected the hydraulic delivery hose 23.

It will be apparent that when the pressure of the oil acting on the closed end of check valve 141 within chamber 139, is greater than the combined pressure of the spring 144 and the pressure of the oil in passageway 125 acting on the under side of said closed end, the check valve 141 will be unseated and oil will flow from chamber 139, around the reduced diameter of the check valve 141, through opening 145, into passageway 125, to cavity 146 and on to the hydraulically powered tool (not shown) connected to the remote extremity of the hose 23. Conversely, when the oil pressure acting on the closed end of the check valve 141 within the chamber 139 is less than the combined pressure of the spring 144 and the pressure of the oil in passageway 125, the check valve 141 will be seated and there will be no flow of oil between the chamber 139 and the hose 23. Thus, the periodic lowering and raising of pressure in the oil pressurizing chamber 137 alternately draws oil from the reservoir bag 103, past the inlet check valve 130 into the pressurizing chamber; and then forces the oil, out of said chamber past the discharge check valve 141 to the delivery hose 23.

Pump apparatus

On the side of the oil valve block 105 opposite the oil reservoir 102, is an elongated projection 148 (Figs. 4 and 14) which is concentrically arranged with respect the axis of the cylinder 24. The projection 148 has a counterbore 149 which extends the entire length of said projection and continues into the oil valve block 105 for a short distance, where it meets a bore which forms the pressure chamber 137. Since the diameter of the counterbore 149 is larger than the diameter of the bore forming the pressurizing chamber 137, a shoulder 150 is formed at the junction of the two bores, as seen in Figs. 4 and 14. A piston ram bushing 151 is snugly fitted to the counterbore 149, one end of the bushing 151 abutting against the shoulder 150, the other end of said bushing being in abutment with a bushing nut 152, which is screwed into a threaded portion at the rear end of the counterbore 149, as shown. The bushing 151 has a bore which is smooth finished for reciprocable movement therein of a piston ram 153. A circumferential recess 154, formed in the bore of the bushing 151, surrounds the piston ram 153 and is arranged to connect with a passageway 155, which passes through the bushing 151 and the oil valve block 105, and exits into the open end of the oil reservoir bag 103. In this manner any oil which leaks by the piston ram 153 during pump operation, finds ready passage back to the oil reservoir proper.

The piston ram 153 has a head portion 156, which is maintained in abutting relationship with

the piston 82 by means of a compression spring 157, through the intermediary of a piston ram washer 158, the opposite end of the spring 157 abutting against the wall of the oil valve block 105 around the base of the projection 148. The strength of spring 157 is sufficient to fully retract the piston 82 when the air in the main air chamber 81 is exhausted to the atmosphere.

The length of the piston ram 153 is such that when the piston 82 is in non-operative position, the end of the ram within the oil valve block 105 is approximately in line with the shoulder 150, as shown in Fig. 4. The diameter of the piston ram 153 is approximately two-thirds the diameter of the pressurizing chamber 137. The forward end of the pressurizing chamber 137 is closed by a pipe plug 159 which provides a liquid tight seal for that end of the pressurizing chamber 137.

On the side of the piston 82 within the main air chamber 81 is a cylindrical projection 160 arranged concentrically with the pump axis, said projection 160 serving to limit the extreme backward movement of piston 82, as is clearly evident in Fig. 4. A piston ram seal 161, made of a flexible material such as leather for example, surrounds the projection 160 and slidingly engages the walls of the cylinder 24, serving to provide a sealing means for the air in the main air chamber 81. The seal 161 is positionally maintained by means of a washer 162 and a retaining ring 163, which is fitted to the projection 160, as shown.

Operation

Before entering into a description of the operation of the pump, it is desired to point out certain features or expedients which are embodied in the structure of the pump, some of which may not have appeared obvious.

First, the length of the pressurizing chamber 137 is such that the end of the piston ram 153 is free from contact with the closed end of said pressurizing chamber, i. e. the plug 159, when the piston 82 has moved to the extreme limit of its working stroke.

Second, the quantity of oil in the pump hydraulic system at the completion of an operating cycle i. e., after the oil has returned to the reservoir, is the same as the quantity of oil in the pump hydraulic system before the start of an operating cycle, excepting of course, that which may be lost due to any leakage. This is arranged by the use of a check valve hose coupling, not shown, of the type having a spring seated valve which is unseated or opened only when the coupling is connected. The male portion of the coupling is attached to the end of the hydraulic hose 23, while the female portion is attached to the hydraulically operated tool. Thus when the coupling is connected, the pump forces oil from the hose 23 past the open check valve hose coupling and into the hydraulically operated tool connected thereto, whereas the return stroke movement of the hydraulically operated tool forces the oil past the check valve hose coupling into the hose 23 and back to the pump reservoir proper. The use of a check valve hose coupling of such type, provides for ready make-and-break connection of the pump to whatever type of hydraulically operated tool it is desired to use at the time, and still prevents the oil from draining from the hose 23 when the hose is not attached to a tool.

Third, regarding the capacity or output of the pump, it is obvious that such is merely a matter of proportion of the various pump parts. However, for a commercial embodiment of this invention, an oil delivery flow of 38 cubic inches per minute at 8,000 pounds per square inch is attained from an oil reservoir capacity of 45 cubic inches. To augment the oil capacity of a pump of any given size, the use of an auxiliary reservoir as later described and illustrated in the drawings, may be provided.

Fourth, it is to be noted that the pump is supported for normal upright position by means of flat portions on the bottom of the cylinder 24, oil valve block 105 and oil reservoir 102, as best seen in Fig. 2. However, as has been mentioned heretofore, the pump will operate satisfactorily while maintained in any desired position.

Finally, in accordance with the usual construction of machines in this general class, an air line oiler of any suitable type may be provided in order that a small amount of oil may be introduced into the live air as a lubricant for the valves and the air operated piston.

Turning now to a consideration of the operating cycle of the pump, Figs. 4, 9 and 10 illustrate the relative position of the various movable parts of the pump when the pump is in non-operative position. Since the foregoing description of the pump's structure has been made with reference particularly to the pump in non-operative position, it is not deemed necessary to again discuss the relative position of the movable parts of the pump when in non-operative position. If the operator desires that the hydraulic pressure developed by the pump and delivered to the hydraulically operated tool, is to be automatically released after the throttle lever 38 is released, he merely observes that the handle 118 (Fig. 4) has been rotated so that the release valve screw 117 is free from contact with the release valve piston 114; whereas, if he wishes the developed hydraulic pressure to be maintained after the throttle lever is released, he checks to see that the handle 118 is rotated so that the release valve screw 117 is in contact with the piston 114.

Assuming then that the pump is connected to a source of live air by means of the air hose 22, and the delivery hose 23 is connected to a hydraulically operated tool, and that the automatic release valve is set for automatic release (as in Fig. 4), the operator initiates the pump operating cycle by fully depressing the throttle lever 38. The throttle valve 32 is thereby unseated and live air flows from the throttle valve chamber 29, past the valve seat 35, about the reduced diameter of the throttle valve 32 and into ports 40 and 41. The air entering port 40 passes through the air passageway 44, enters the air reservoir 102 and acts upon the exterior of the oil bag 103, thereby pressurizing the oil contained therein and forcing oil into the oil passageway 113, from which it passes into oil passageways 116 and 127 (Fig. 10). The oil in passageway 116 flows upward into the chamber 115, and acts upon the piston 114, which results in the seating of the pressure release valve 121 in the opening of the passageway 124. The oil in passageway 127 enters the chamber 128, where the pressure of said oil unseats the inlet check valve 130, the oil thereby advancing through passageway 136 into the pressurizing chamber 137, wherein it is eventually pressurized by the advance of the piston ram 153.

The air going through port 41 (Fig. 9) passes through passageway 45 and ports 69 into the bore of the valve bushing 47, about the reduced diameter mid-portion of the distributor valve 52,

and through the radial ports 70 and annular recess 71 to the air passageway 72. The air from passageway 72 enters the shift valve chamber 75 (Fig. 4), some of the air passing onward through said chamber and into the continuation of passageway 72, while some flows past the unseated shift valve 76 into the passageway 80. The air in passageway 80 enters the cavity in the left cap 87 (Fig. 9), thereby shifting the automatic valve 86 to the right, where the end of the valve 86 abuts against the bottom of the cavity formed in the other cap 87. The automatic valve 86 in shifting to the right, uncovers the vent 100 so that air in the cavity of the left cap 87 is exhausted to the atmosphere.

The air which flows downward from the shift valve chamber 75 through the continuation of passageway 72 (Fig. 9), is allowed to exhaust to the atmosphere before the automatic valve 86 is shifted to the right, such exhausting taking place by way of annular groove 83, bushing ports 85, valve recess 89, passageways 90 in the automatic valve 86 and through the passageway 91 (Figs. 4 and 11), port 92, and then through cylinder exhaust ports 93. After the automatic valve 86 is shifted to the right, the air from passageway 72 flows about the reduced diameter portion 88 of the automatic valve 86, into passageway 96 (Figs. 9, 11 and 14), from where it enters the main air chamber 81. As the air builds up pressure in the main air chamber 81, piston 82 begins to move, which results in the compression of the spring 157 and the simultaneous pressurizing of the oil in chamber 137 by means of the advance therein of the piston ram 153. As the piston continues its movement which constitutes the power stroke, contact with the shift valve 76 is released and ended, as the shift valve 76 is allowed to seat by action of the spring 77, thereby cutting off the flow of air from chamber 75 to the passageway 80.

Power stroke

Fig. 11 illustrates the first stage in the power stroke of the pump, as just discusssed. The single-headed arrows indicate air at no greater than air line pressure, the double-headed arrows indicate oil under air line pressure, and the triple-headed arrows indicate oil under higher pressure as a result of the pressurizing which takes place in the pressurizing chamber 137. It is to be noted that the inlet check valve 139 has been seated, while the discharge check valve 141 has been unseated, each as a result of the pressurization of the oil in the pressurizing chamber 137, and that the oil is flowing under pressure into the oil delivery hose 23. As has been indicated in the preceding discussion of the pump's structure, the compression springs 132 and 144 of the check valves 139 and 141 respectively, provided for the automatic actuation of the said valves in accordance with the relative pressure of the oil in the pressurizing chamber 137 and the oil in the oil bag 103.

It may be pointed out that the maximum pump delivery pressure of 8,000 pounds per square inch is not ordinarily realized after the completion of the first power stroke, but rather is attained only after a series of successive power strokes, in accordance with the amount of oil that will have to be pumped to the hydraulic tool which is connected to the pump. In other words, some tools may require full oil capacity of the pump before the maximum oil pressure can be built up, in which case the piston 82 will undergo a certain number of reciprocations to pump the required amount of oil, and hence the maximum oil pressure of 8,000 pounds per square inch will be achieved only on the final power stroke. Ordinarily, the operator keeps the throttle lever 38 fully depressed continuously during a succession of piston reciprocations until the desired oil pressure is attained, after which he releases the throttle lever 38. By opening the throttle valve only part way and by closing it when desired, the operator may control the movement of the ram (not shown) on the driven tool. Thus the ram may be run out at full speed or it can be inched forward, stopped, held in position, or retracted, as desired.

Return stroke

Fig. 12 illustrates the relative position of the various parts of the pump just after completion of a power stroke and at the beginning of the return stroke. It will be seen that when the piston 82 is advanced during the power stroke to a certain position, which can be termed maximum forward position, cylinder port 95 is uncovered and port 92 is simultaneously covered by the piston head, thereby allowing air in the main air chamber 81 to flow through port 95 into passageway 91, to the reduced diameter mid-portion 89 of the automatic valve 86, and into hole arrangement 90 within said automatic valve. The air which exists from the hole arrangement 90, builds up pressure behind the right end of automatic valve 86 within the cap 87, and forces the automatic valve to the left, i. e. the end being forced into abutment with the bottom of the cavity in the other cap 87. Upon movement to the left the valve 86 displaces the air in left cap 87 which escapes to atmosphere around the periphery of valve 86 and through the small port 100 (Fig. 9). The shift of valve 86 to the left places the passageway 96 (Figs. 9 and 12) in communication with the ports 93 by way of the reduced diameter portion 88 of the automatic valve 86, thereby exhausting the air in the main air chamber 81 to the atmosphere. As soon as the pressure in the main air chamber is released, the spring 157 acts to force the piston 82 rearward to its initial position within the cylinder 24. It is to be noted that the port 95 is uncovered only for that instant during which the piston 82 is in maximum forward position, and that some of the air in passageway 72 is bled to the atmosphere during the return stroke of the piston via the hole arrangement 90, passageway 91, ports 92 and 93, and also restricted vent 94.

During the return stroke of the piston, the pressurizing chamber 137 expands in size to reduce the pressure on the oil therein, while the air in passageway 44 is still acting upon the oil bag 103, thereby forcing a slug of oil past the inlet check valve 139 and into the pressurizing chamber 137, in preparation for the next pressurizing stroke of piston ram 153.

As the piston 82 repeatedly reciprocates, the oil pressure gradually rises in the discharge hose 23, causing a corresponding rise in the resistance of discharge valve 141 to opening movement and in the resistance offered by the oil in pressurizing chamber 137 to forward movement of the piston ram 153. Against this increased opposition the forward motion of the piston 82 becomes less rapid as compared with preceding strokes. Under ordinary operating conditions, however, the opposition is still small enough, and the forward travel of the piston still fast enough, to prevent the air in cylinder chamber 81 from approaching line pressure (say 90 p. s. i.) as the chamber ex-

pands on the power stroke. Accordingly the pressure regulator valve 64 and distributor valve 52 remain in the non-operated position of Fig. 9 under normal operation.

Oil pressure release

It has been pointed out that the piston 82 starts to reciprocate when the throttle lever 38 is depressed and continues in a succession of pumping strokes until the throttle lever is released. When this occurs, the oil pressure in hose 23 may not be automatically released, depending upon the setting of the manipulative screw 117. If the screw (Fig. 4) is forced down on pressure release valve piston 114, the pressure in hose 23 will be maintained because the oil in that hose has no means of escape back to the oil reservoir bag 103.

If, on the other hand, the operator desires to have the oil released from hose 23 automatically upon closing of the throttle, he first conditions the machine for automatic pressure release by withdrawing the screw 117 out of the path of the pressure release piston 114. When operating with such a setting the valve piston 114 is held down by the pressure of oil admitted through passage 116, which is maintained only so long as live air is delivered through the throttle valve and passage 44. When such pressure is released the pressure release valve 121 rises under the influence of spring 119 to open a path from the oil hose 23 direct to the reservoir bag 103, independently of the check valves, as above described.

Manual pressure release

In the modification of Fig. 16 the automatic pressure release valve piston 114, and its associated housing, spring, etc., are omitted to simplify the structure and reduce the manufacturing cost. Instead of the automatic valve 121, the modification comprises a manual valve 121*a* (Figs. 16, 17) having a threaded portion 121*b* engageable with the screw threads 123 in the oil valve block 105. Seating and unseating of the modified pressure release valve 121*a* is effected by turning the handle 121*c*. In other respects the valve 121*a* is comparable to the automatic valve 121. In the modification of Fig. 16, the threaded opening 112 is directly engaged by a pipe plug for closing its upper end.

Pressure regulator

In the foregoing discussion it was assumed that the operator would release the throttle lever and stop the pumping operation before the pressure in oil hose 23 attained a dangerous value. If he fails to release in time, however, the pressure regulator automatically comes into operation and stops the reciprocations of the piston, without however, releasing the pressure of the oil already pumped into the hose.

Figs. 13 and 14 illustrate the relative position of the various parts of the pump just at the time that maximum pump pressure is reached. Assuming that the maximum oil pressure of the pump under discussion is 8,000 pounds per square inch, then the effective air pressure in the main air chamber 81 will not realize a maximum value until the maximum oil pressure is realized in the pressurizing chamber 137. When this occurs, the piston 82 is short stroked and the air in the main air chamber 81 has time to build up and approach line pressure, and by way of passageway 101, to force the regulator valve 64 toward the left in the valve bushing 47, which results in the simultaneous shifting to the left of the distributor valve 52, so that the end of said valve 52 abuts the distributor valve stop 50, as shown in Fig. 15. In such a position, the distributor valve 52 allows exit to the atmosphere of the air in the main air chamber 81, by way of passageway 96, about the portion 88 of the automatic valve 86, through passageway 72 and onward about the reduced diameter midportion of the distributor valve 52, and finally through ports 73 in the bushing 47.

When the distributor valve 52 is in its shifted position (Fig. 15), the ports 69 are covered by the distributor valve, while the ports 74 are uncovered. This allows the live air from passageway 45 to act upon the end of the distributor valve 52 which is in abutment with the regulator valve 64, and to hold the distributor valve in shifted position independently of the pressure exerted upon the distributor valve by the regulator valve 64. Under these conditions the distributor valve 52 may be released from shifted position, only upon release of the throttle lever 38, which results in the cutting off of air flow to passageway 45, and allows escape of air from passageway 45 to the atmosphere by means of port 42, thus permitting the regulator valve spring 59 to shift the distributor valve 52 to its first or normal position. The strength of the regulator valve spring 59 is a factor which determines the point of cut-off for the hydraulic delivery pressure, it being obvious that a spring with a lower compression factor will allow the distributor valve 52 to be shifted at a lower value of hydraulic delivery pressure, which is directly reflected in the pressure of air in the main air chamber 81. It will be seen that the zig-zag hole arrangement 68 of the regulator valve 64, prevents the trapping of air within the chamber formed in the valve bushing 47 at the end of the distributor valve 52, and thus allows the unrestricted seating of said distributor valve.

Auxiliary reservoir

While it is apparent that the oil capacity of any given pump of the present type is constant once the dimensions of the parts are established, extra oil capacity can be obtained by means of an auxiliary oil reservoir, which is readily connected to the pump, for use of the pump with hydraulic tools requiring more oil capacity for operation than that normally afforded by the pump. Figs. 18, 19 and 20 illustrate an auxiliary reservoir system, which when connected to the pump, increases the oil capacity by approximately three times. In identifying parts of the auxiliary reservoir system, numerals which have been previously used in the detailed description of the pump will be reused, with the letter "*a*" following, to identify identical or corresponding parts.

The auxiliary reservoir system is simple in structure and basically consists of three main parts, a connection block 164 and two reservoir portions 102*a*, each containing an oil reservoir bag 103*a*. The reservoir portions 102*a* are connected to the block 164 on opposite sides thereof by means of four bolts 106. A tapered pilot 107*a* is located on opposite sides of the block 164, the purpose of said pilots being for centering the reservoirs 102*a* with respect to the block 164, and also for fixing the reservoir bags 103*a* in liquid tight manner. Extending into the block 164 from one face are two bores 165 and 166, each having a threaded portion at the entrance thereof. Passageways 167, leading from the bore 165 at two points, exit on each side of said block 164 in a region near the base of each pilot 107*a*, each of said passageways 167 being located in line with

the opening of passageway 44a in the reservoir portion 102a. The bore 166 has passageways 168 leading therefrom at two points, and opening on the surface of each pilot 107a in the region of the open end of the oil bag 103a. A transverse passageway 169 having a diameter approximately equal to the diameter of the bore 166, extends through the block 164 and connects the bore 166 to the surface of each pilot 107a, the ends of said passageway 169 also being in the region of the open end of the oil bag 103a.

The auxiliary reservoir system is connected to the pump by means of two pipes, not shown, but represented diagrammatically by the broken lines O and A respectively. The first end of pipe O is threadably connected by means of standard pipe connections to the bore 166 in block 164 (Fig. 20), the second end of said pipe O being threadably connected to the oil filler hole 110 in the oil valve block 105 (Fig. 10); or alternatively to threaded opening 112 in Fig. 16. In similar manner the first end of the other pipe A is connected to the bore 165 in the block 164 (Fig. 20), the second end of said pipe A being connected to a threaded bore 170 (Fig. 10), which is formed in the oil valve block 105; or alternatively to a threaded bore 170a (Fig. 16). The bore 170 (or 170a) is connected to the oil passageway 44 by means of a passageway 171 (or 171a). When the pump is not connected to an auxiliary reservoir system, as set forth above, the oil filler hole 110 and the bore 170 are plugged by means of standard pipe plugs 111.

The operation of the auxiliary reservoir system and the manner in which it functions to supplement the oil capacity of the pump is as follows: Some of the air in passageway 44 enters passageway 171 and bore 170 in the oil valve block 105, then passes through the interconnecting pipe A to the bore 165 and through passageways 167 in the block 164, and exits into each oil reservoir 102a to exert pressure upon the oil in the oil bag 103a, contained therein. As a result of the pressure on the oil bag 103a, oil is forced out of each oil bag 103a into passageways 168, and bore 166 in the block 164, through the interconnecting pipe O and into the hole filler hole 110 in the oil valve block 105, the oil thus becoming part of the oil system of the pump.

A divisional application, Serial No. 252,762, filed October 16, 1951, contains claims relating to an auxiliary reservoir system, and to an air motor per se, all as shown and described herein.

What is claimed is:

1. A pump comprising in combination a pneumatic portion, a hydraulic portion coupled to the pneumatic portion, and a reciprocable piston contained in part within each of said portions, said pneumatic portion being arranged to receive a pneumatic medium and having a passageway arrangement which directs an amount of said medium for motivation of said piston and another amount to the hydraulic portion, said hydraulic portion having an oil reservoir the volume of which can be decreased by action of said pneumatic medium, a check valve system including a liquid pressurization chamber, an oil delivery outlet located at the exterior of the pump, and connecting said outlet with said reservoir means forming a liquid return flow passageway, said check valve system being interposed between the reservoir and the outlet and further positioned so that a portion of the piston extends into the pressurization chamber, all arranged so that liquid is forced from said reservoir by action of the pneumatic medium, passes through the check valve system while being pressurized by action of the piston within the pressurization chamber, is delivered to the outlet under increased pressure, and is returned to the reservoir by the return flow passageway after pressurization ceases.

2. A pump comprising in combination a pneumatic portion, a hydraulic portion coupled to the pneumatic portion, and a reciprocable piston contained in part within each of said portions and being movable in one direction by pneumatic pressure and in the opposite direction by mechanical means, said pneumatic portion being arranged to receive a pneumatic medium and having a pressure responsive valve system which automatically directs a given quantity of said medium for motivation of said piston, said hydraulic portion having an oil reservoir which is collapsible under action of pneumatic medium received from the pneumatic portion, a check valve system including a liquid pressurization chamber, an oil delivery outlet located at the exterior of the pump, and connecting said outlet with said reservoir means forming an oil return flow passageway, said check valve system being interposed between the reservoir and the outlet and further positioned so that a portion of the piston extends into the pressurization chamber, said hydraulic portion being arranged so that oil is forced from the reservoir by pneumatic pressure through a first part of said check valve system, thereafter through said pressurization chamber where the oil is pressurized by reciprocable action of said piston, and through a second part of said check valve system to the oil delivery outlet, and is returned to the reservoir by the return flow passageway after the oil pressurization is stopped.

3. A pump according to claim 2, wherein said piston movement mechanical means comprises a pre-compressed spring.

4. A pump according to claim 3 wherein said pneumatic portion has a throttle means operable to initiate the operating cycle of said pump.

5. A pump according to claim 4 wherein an oil delivery pressure release means is included in the oil return flow passageway.

6. A pump according to claim 5 wherein said pneumatic portion has a valve which automatically releases to the atmosphere the pressure medium acting upon the piston when the oil in the pressurization chamber has been pressurized a predetermined amount.

7. A pump according to claim 6 having means adapted for the reception of an auxiliary reservoir system which automatically operates to supplement the oil delivery capacity of the pump, said auxiliary reservoir system including one or more oil reservoirs which are collapsible under action of pneumatic medium delivered by hose means from the pump to thereby force oil from the reservoirs through hose means to the hydraulic portion of the pump.

8. In a pump of the character described, a hydraulic portion comprising a reservoir having a bag containing a liquid medium, said reservoir being arranged for the collapse of said bag by pneumatic pressure; a valve block connected to said reservoir and containing in part a reciprocable piston means, a first chamber having a first check valve, a first passageway connecting said first chamber to an open end of said bag, a pressurizing chamber being arranged for the

pressurization of liquid therein by action of said piston means, a second passageway connecting said pressurizing chamber to said first chamber, a second chamber having a second check valve, a third passageway connecting said second chamber to said pressurizing chamber, a liquid outlet means, a fourth passageway at one point connecting said liquid outlet means to said second chamber, a third chamber having a pressure release valve reciprocably arranged therein, and a fifth passageway connecting said third chamber to an open end of said bag, said third chamber being also connected to another point of said fourth passageway, said first check valve being resiliently maintainable in a first position to prevent passage of liquid from said first passageway into said second passageway and being hydraulically maintainable in a second position to allow passage of liquid from said first passageway into said second passageway, said second check valve being resiliently maintainable in a first position to prevent passage of liquid from said third passageway into said fourth passageway and being hydraulically maintainable in a second position to allow passage of liquid from said third passageway into said fourth passageway, said pressure release valve being maintainable in a first position to prevent passage of liquid from said fourth passageway into said fifth passageway and being maintainable in a second position to allow passage of liquid from said fourth passageway into said fifth passageway.

9. In a pump hydraulic portion according to claim 8, automatic means for positioning said pressure release valve in said first and second positions comprising a piston having a top surface forming a wall of a chamber which is connected to the first passageway in said valve block and a bottom surface which abuts one end of said pressure release valve, and a resilient means in abutment with the bottom surface of said piston, all so arranged that a liquid under pressure flowing from said first passageway into said chamber urges said piston in a direction to force said pressure release valve into said first position, while when said liquid is caused to flow from said chamber into said first passageway said resilient means urges said piston in a direction to allow movement of said pressure release valve into said second position.

10. In a pump hydraulic portion automatic means according to claim 9, a piston control means comprising a member arranged to project into said chamber and to contact the top surface of said piston, said member being manually adjustable to restrict movement of said piston to an extent that said pressure release valve is maintained in said first position.

11. In a fluid pressure operated hydraulic pump including a motor end and a pump end, a cylinder in the motor end providing a piston chamber, a piston reciprocating therein, means in the motor end including a throttle valve for supplying live air to the piston chamber to effect such reciprocations, a valve block attached to the cylinder for closing an open end of the piston chamber, an oil reservoir attached to the opposite side of the valve block, a high pressure outlet on said block, a piston extension of reduced diameter projecting forwardly into the valve block, said block having a pressurizing chamber receiving the piston extension, said valve block having two separate passageways between the reservoir and the outlet, one passageway leading from the reservoir through an inlet check valve to the pressurizing chamber thence through a discharge check valve to the outlet, the other passageway extending from the outlet through a pressure relief valve back to the reservoir.

12. In a fluid pressure operated pump according to claim 11 an arrangement whereby the pressure relief valve is adapted to be closed when the throttle valve is open and to be opened when the throttle valve is closed, whereby the operator may discontinue the pumping operation and release the oil pressure at the same time.

13. In a fluid pressure operated pump according to claim 11, an arrangement whereby the throttle valve supplies live air to close the pressure relief valve only as long as the throttle valve is open and also supplies air to the reservoir for forcing the oil into the pressurizing chamber during the rearward stroke of the piston.

14. In a fluid pressure operated hydraulic pump according to claim 11 an arrangement whereby a pressure regulator valve is adapted to cut off the flow of live air from the throttle valve to the piston chamber upon development of a predetermined opposition to the forward stroke of the piston.

15. A pump pneumatically operable to pressurize a hydraulic medium within the pump for delivery to a device served by the pump, the pump being arranged for the return of said medium for storage when in non-operative condition comprising, a pneumatic portion including valve means and a piston chamber, a hydraulic portion connected to the pneumatic portion, and a piston contained in part within each of said portions and pneumatically movable to pressurize hydraulic medium entrapped in a pressurizing chamber formed in the hydraulic portion and movable in the opposite direction by resilient means, said pneumatic portion having a pneumatic inlet means and passageways leading therefrom to conduct pneumatic medium to the exterior of a collapsible reservoir located within the hydraulic portion and containing hydraulic medium and to the valve means, said valve means arranged to admit pneumatic medium to the piston chamber containing part of the piston whereby the piston is moved to pressurize the hydraulic medium and further arranged to release pneumatic medium from the piston chamber so that the piston can be moved by the resilient means, said valve means including a pressure regulating valve which cuts off delivery of pneumatic medium to the piston chamber when a predetermined hydraulic pressurization is realized, said hydraulic portion including a hydraulic medium delivery means and valves to maintain pressurized hydraulic medium within the delivery means and further to allow return of the hydraulic medium to the reservoir when the pump is in non-operative condition.

16. A pump comprising a pneumatic portion including a main chamber and pneumatic valve means, a hydraulic portion coupled to the pneumatic portion and including an oil reservoir, an oil pressurizing chamber, an oil outlet, and check valve means, piston means slidably arranged in the main chamber and including a ram which extends into the pressurizing chamber, a passageway arrangement for conducting pneumatic medium to the pneumatic valve means and to the oil reservoir, said hydraulic portion having a passageway connecting the reservoir to the pressurizing chamber via a check valve and a passageway connecting the pressurizing chamber to the oil outlet via a check valve, said check valves arranged for the one way flow of oil from the reser-

voir, through the pressurizing chamber and to the oil outlet, oil hose delivery means connected to the oil outlet for attachment to a device served by the pump, said pneumatic valve means adapted to automatically admit pneumatic medium to the main chamber to move the piston means whereby oil is forced from the pressurizing chamber under increased pressure, and further adapted to cut off flow of pneumatic medium to the main chamber when the pressurization of oil attains a predetermined value, and oil pressure valve means in the hydraulic portion arranged to maintain oil pressure in the oil hose delivery means after cessation of piston movement and further arranged to release the oil pressure and allow return of oil to the reservoir when the pump is in non-operative condition.

JAMES A. ROBERTS.
HOWARD R. FISCHER.
MARCEL P. D'HAEM.

REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
1,485,138	Johnson	Feb. 26, 1924
1,743,968	Hatfield	Jan. 14, 1930
2,001,190	Ginter	May 14, 1935
2,042,744	Tear et al.	June 2, 1936
2,080,695	Cargile	May 18, 1937
2,134,734	Reinhold	Nov. 1, 1938
2,324,701	Herman	July 20, 1943
2,347,379	Teeter	Apr. 25, 1944
2,406,747	Davis	Sept. 3, 1946
2,433,759	Hess	Dec. 30, 1947