

[54] PLUNGER AND CYLINDER ARRANGEMENT

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92/154, 156, 157, 162, 86.5, 127, DIG. 1,
DIG. 2

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[57] **ABSTRACT**

A fluid-bearing plunger and cylinder arrangement comprises a cylinder, a plunger member closely fitted within the cylinder for reciprocation therein, said cylinder defining a pressure chamber therein for actuating the plunger, a plurality of pressure recesses formed on one of the plunger member and the cylinder at a location removed from the chamber, and a pair of exhaust grooves formed on one of the cylinder and the plunger member and axially spaced one on each side of the pressure recesses. The recesses circumferentially surround the member and in addition are substantially equally spaced from one another. The grooves circumferentially and continuously surround the plunger member, and one of the grooves is disposed between the pressure recesses and the chamber. Means are provided for supplying a pressurized fluid to each of the recesses and for conveying from the exhaust grooves any fluid collected therein.

In other applications of the invention flow restricting means are provided for accurately metering individual flows into said pressure recesses respectively.

12 Claims, 5 Drawing Figures

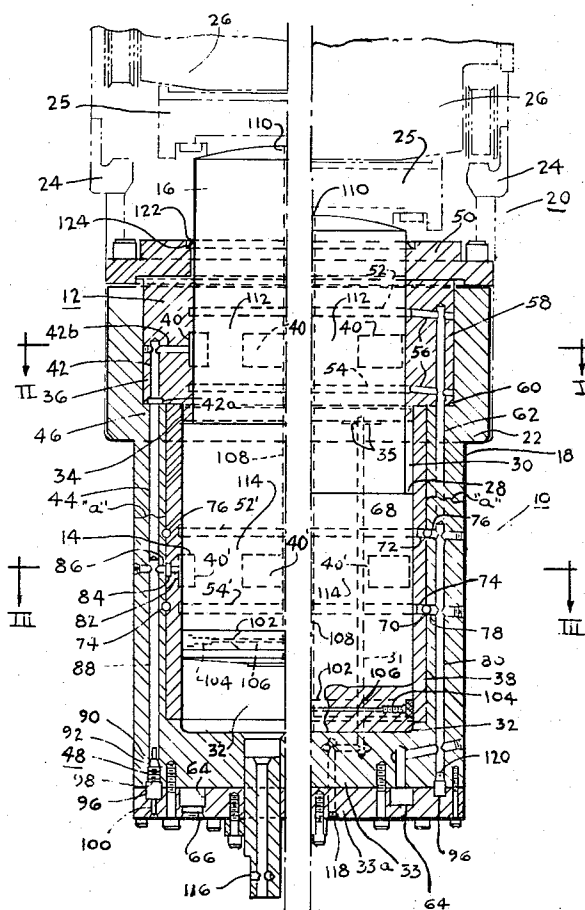


FIG. 2

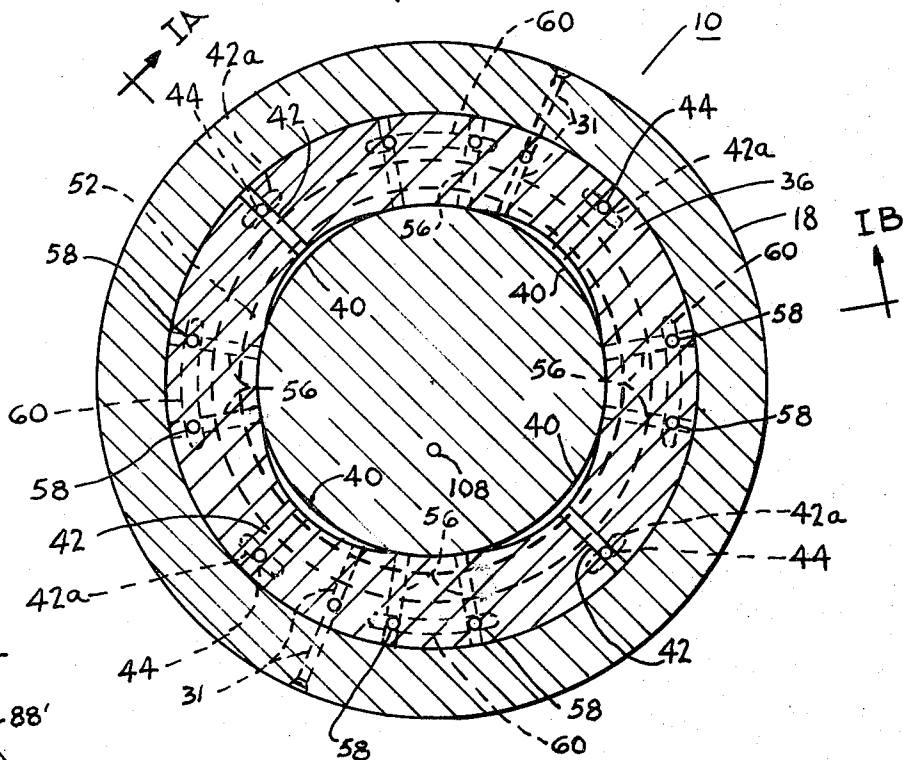


FIG. 4

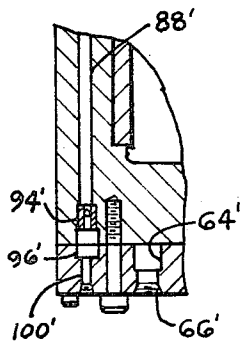
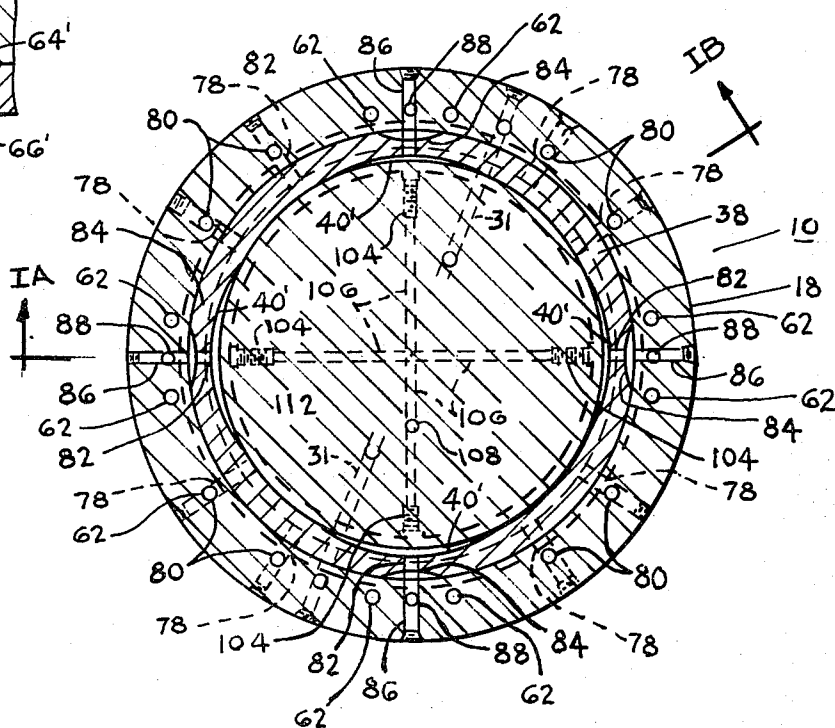


FIG. 3



PLUNGER AND CYLINDER ARRANGEMENT

The present invention relates to an improved plunger and cylinder arrangement, and more particularly to an arrangement of the character described wherein the plunger or piston is rendered virtually frictionless and is centered within the cylinder by means of one or more fluid bearing arrangements. The invention, therefore, also relates to an improved fluid bearing arrangement, and more particularly to such arrangement which is advantageously functional under essentially static conditions capable of providing both bearing and lubrication.

In the manufacture and use of varied types of piston or plunger and cylinder arrangements, there has been a perennial need for a reliable fluid bearing. Many types of fluid bearings have been proposed for highly specialized applications and then subsequently abandoned. Fluid bearing plunger and cylinder arrangements in the past have been based largely on hydrodynamic principles, with the result that in many cases or under unforeseen conditions, such fluid bearings or devices utilizing the same, have been difficult to start up, owing to the lack of adequate lubrication or bearing forces under essentially static conditions. When in operation, difficulty has been experienced in properly segregating the fluid bearing system and the main cylinder chamber, while providing adequate bearing and lubrication. In certain applications, where devices utilizing fluid bearings are constantly in motion, or otherwise involved relatively few starts or stoppages, hydrodynamic fluid bearings have performed satisfactorily. For example, in rapidly rotating shafts or similar machine components, an adequate hydrodynamic bearing wedge can be maintained for long periods of time. In applications where the moving or rotating member is operated intermittently, especially in the case of a plunger or piston and cylinder arrangement, hydrodynamic wedges are frequently interrupted and such fluid bearing arrangements are of little avail. Although hydrostatic bearings have been developed for rapidly rotating shafts, no one, insofar as I am aware, has solved the problems of combining such bearing in a plunger or piston and cylinder having a pressure chamber.

For the applications contemplated by this invention, conventional fluid bearing arrangements are totally unreliable, and the member supported thereby is apt to "hang up" from excessive friction when subject to intermittent motions or to closely controlled reciprocations in the plunger and cylinder arrangement. In this respect such bearing arrangements suffer from the same disadvantages as in the case of conventional packing such as chevron seals, O-rings, and the like. In most cases such conventional packing is loaded by various mechanical devices to increase the frictional engagement between the plunger or other moving or movable member and its housing. With either the loaded packing arrangement or with known fluid bearings, the plunger or piston member is apt to hang up or freeze within the cylinder or other housing if the member has not been moved for a time. In many applications, for example in earth moving or excavating equipment the piston and cylinder combinations thus employed are comparatively small and their tendency hang up can be obviated by application of excessive fluid pressures. If this fails, the bucket or blade attached to the cylinder rods can be gently snubbed

against the ground or other solid object to shake the pistons loose.

In completely stationary applications or in applications involving the larger sizes of pistons or plungers and cylinders, these field-expedients are not available. In many of these applications the use of conventional packing or of conventional fluid bearings or both is largely obviated owing to the aforementioned unreliability and a tendency of the piston or plunger to freeze within the cylinder or similar housing. As an example, for many years attempts have been made to substitute more or less conventional plunger and cylinder arrangements for the mechanical screw-ups and screw-downs employed in individual mill stands of a high speed rolling mill. Such substitution would be highly advantageous as the strip sheet gage could be controlled at the mill stand "screw-up" cylinders owing to the more rapid and facile action of the one or more plungers substituted for the screw-ups and/or screw-downs. However, a rolling mill of high quality product has not as of this writing been constructed owing to the aforementioned unreliability of the necessarily enlarged piston or plunger and cylinder arrangements. In a typical rolling mill, the thickness of the strip may be sensed about twenty times per second or more (for highest quality strip), with the result that the plunger and cylinder arrangement, if they could be used, would be subject to a relatively high frequency, intermittent movement. With conventional bearing and/or packing systems, and the high rate of operation of conventional rolling mills, which may range around 5,000 feet per minute, an extensive footage of sheet or strip would be out of spec., if the plunger should happen to freeze or hang up only momentarily.

When the piston or plunger thus hangs up within the cylinder, application of excessive pressures to dislodge the piston or plunger is necessitated, which can result in considerable personnel hazard. In addition, in the application just mentioned the introduction of excessive pressures causes the piston to overshoot its mark when dislodged. In the case of rolling mills a resultant single overshoot of the workroll gap can result in a loss of 20 to 30 feet or more of steel strip or other rolled material.

When employed in conjunction with a rolling mill, and as a hydraulic cylinder arrangement therefor a reliable and invariable, low friction is provided. This results from the elimination of conventional packing which notoriously exhibits widely varying frictional coefficients, which may range in the aforesaid example from 20 or 30 percent of the output force to more than 100 percent when the piston or plunger hangs up and jams within the cylinder. In previous rolling mills where hydraulic cylinders have been tried with varying degrees of unsuccess, it has been necessary to shut down the rolling mill periodically in order to change the cylinder packings. As much as 1 week per mill stand is required for such operations, which amount to considerable loss in production.

Owing to the constant low frictional coefficient of the novel piston or plunger and cylinder arrangement of my invention, rolling loads can be determined simply by measuring the cylinder pressure. In this way a number of expensive and complicated load cells can be eliminated. Moreover, as intimated previously, a plunger and cylinder arrangement of my invention can be used directly for an automatic gage control of the

strip or other material being rolled, owing to the reliably low coefficient friction. Incorporation of the "screw up" cylinder into the automatic gage control system is also made possible by the quick response of my novel plunger and cylinder arrangement. Thus, my piston and cylinder arrangement is not susceptible to hunting and can provide a constant force at the roll bite of the rolling mills. With the extremely low coefficient of friction attainable in my arrangement, considerable power savings can be realized.

A rolling mill application of my improved piston or plunger and cylinder arrangement is merely exemplary and non-limitative, as my plunger and cylinder arrangement is novel per se and can be advantageously employed in many applications where compactness, low friction, or an essentially total reliability is a prerequisite. Similarly, my novel plunger and cylinder arrangement is advantageously used where space is at a premium, as the bulky packing arrangement of a conventional piston and cylinder arrangement is eliminated. Finally, my novel arrangement is useful both for lubricating and supporting any reciprocating plunger or piston and for centering the plunger or piston within a cylinder or other housing.

Owing to the desirability, as aforesaid, of providing a fluid bearing plunger and cylinder arrangement, a number of attempts have been made in the past to modify known fluid bearing arrangements for this purpose. Such attempts are typified by the U.S. Pat. Nos. to Davis 2,367,009; Strimel 3,015,315; Foster 2,495,516; Bayer 2,833,602; Cormier 2,570,647; Swarthout 3,104,619; McCrory et al. 3,229,900; Guillot 3,527,074 and Guillot 3,538,727.

The Guillot patents are not fluid bearing arrangements at all, but are of interest in their relationship to one application for which my improved plunger and cylinder arrangement is admirably suited. The Guillot arrangements involve the use of an extremely viscous hydraulic fluid in conjunction with more or less conventional packing arrangements. The Guillot devices would not appear to obviate the problems, then, entailed in the use of various forms of mechanical packing. In this letter respect, the McCrory et al and Swarthout arrangements likewise employ conventional piston packings. Moreover, the continuous fluid sealant grooves of the latter-named references are employed in conjunction with conventional packings and are inappropriate of the problems solved by the present invention.

The Strimel, Davis Jr. and Cormier devices are of lesser interest in that fluid bearings are provided by a introduction of pressurized fluids into a continuous circumferential groove or grooves on the plunger or cylinder. If used for the applications contemplated by my invention, the bearing forces would become unbalanced, particularly under static conditions as fluid pressure would apt to be lost owing to unequal clearances between the cylinder and the plunger or other movable member. Thus, bearing fluid would be diverted from the best-sealed portions where it is needed most to the worse-sealed portions of the fluid bearing groove. Accordingly these devices fail to provide adequate fluid bearing forces under essentially static conditions.

It appears that the Bayer piston would be subject to the same limitations. The pressure flats provided on the Bayer piston are interconnected such that the available

bearing forces would tend to be reduced at the better-sealed bearing flats where the bearing forces are most needed. The Bayer arrangement likewise would appear inappropriate for reciprocatory or other intermittent motion, particularly in applications where the piston or plunger or other movable member is heavily loaded.

The Foster arrangement likewise is inappropos of the problems confronted by the present invention. Although contemplative of circumferentially end-to-end pressure recesses, such recesses are not uniformly spaced about the piston or plunger or other movable member, nor are they communicated uniformly with a common, continuous exhaust groove or recess. Thus, the Foster fluid bearing arrangement would not appear to be suitable for hydrostatic centering and lubricating of an intermittently movable member. The Foster plunger, when employed in applications contemplated by the present invention, would tend to hang up and jam on the ribbing separating the various pressure and relief recesses. There is no employment of a constantly metered flow to a plurality of pressure areas as in certain of the modifications of the present invention. Instead, Foster employs an intricate cross-valving system of intermittent operation. In the Foster device, bearing surfaces are used to support the plunger or piston when the pressure recesses are depressurized. Thus the Foster device apparently is non-hydrostatic.

As noted previously, hydrostatic fluid bearings for rapidly rotating spindles or shafts are known, an example of which is set forth in the Mar. 1965 issue of Lubrication Engineering pages 89-96, Some Advantages of Hydrostatic Bearings in Machine Tools, Robert S. Hahn. There are no teachings in any of the references cited herein, of a workable and reliable incorporation of hydrostatic bearing principles in a plunger and cylinder arrangement, particularly one which can be heavily loaded. The problems of properly segregating the cylinder chamber and fluid bearing, prevention of leakage therefrom, adequate plunger lubrication, and exhaust of spent bearing fluid without possibility of back-pressures have not been solved. Because the plunger and cylinder may be heavily loaded a quick and reliable self-centering and, in some cases, anti-cocking capabilities are required.

The advantages accruing from elimination of conventional or mechanical packing are self-evident. The packing itself is complicated to manufacture and install in the first instance. Considerable downtime on the equipment in which the piston and cylinder arrangement or other machine elements requiring packing are utilized is necessitated by the inevitable maintenance and periodic packing changes. Maintenance, of course, includes frequent adjustment of various types of loading devices, where the packing is mechanically loaded. Replacement of such packing with a fluid bearing arrangement permits the use of non-critical cylinder and plunger tolerances and other design parameters, and considerably reduces cylinder friction. In experimental installations, I have found that friction can be reduced from the 10 to 33 percent or more produced by conventional mechanical packings, to a figure of the order of 1 percent or less. In addition to elimination of any tendency of the plunger to hang up or jam, this small amount of fluid friction can be made essentially invariable, such that the power output of the plunger and cylinder arrangement is essentially linear and therefore totally predictable. Constant and repeti-

tive forces are readily delivered by my novel piston and cylinder arrangement.

For heavily loaded applications, no acceptable fluid bearing arrangement has been proposed to date, to the best of my knowledge. Such applications include for example the larger sizes of excavating, earthmoving and/or strip mining equipment and the like, or usage in hypothetical hydraulic rolling mills as noted below. Those fluid bearings which have been proposed previously have encountered excessive leakage of hydraulic fluid. In an experimental operation, with a plunger of about 3 inches in diameter, I have attained a very low leakage of about one drop of hydraulic fluid every 2 seconds. In this activity a relatively light hydraulic fluid of about 300 S.S.U. was employed, in contrast to the high viscosity of the Lubrication Engineering article. The advantage and surprising facility of operation of my novel fluid bearing system is immediately apparent from the absence of any requirement for closely controlled tolerances. This results at least in part from the surprising efficacy of the self-centering feature of my novel fluid bearing system. A potentially great advantage of this system is its proper operation regardless of position or orientation of the plunger and cylinder arrangement and of the fluid bearing system incorporated therewith. This aspect of the invention is particularly useful when the improved plunger and cylinder arrangement of the invention is employed in excavating or earth-moving operations or in analogous operations involving frequent changes in position or orientation of other machine elements using my novel fluid bearing system.

I accomplish the unexpected results outlined above and overcome the deficiencies of the prior art in these respects by providing a fluid bearing plunger and cylinder arrangement comprising a cylinder, a plunger member closely fitted within said housing for reciprocation therein, said cylinder defining a pressure chamber therein for actuating said plunger, a plurality of pressure recesses formed on one of said plunger member and said cylinder, said recesses circumferentially surrounding said plunger member and in addition being substantially equally spaced from one another, a pair of exhaust grooves formed in one of said plunger member and said cylinder and axially spaced one on each side of said recesses, said grooves circumferentially and continuously surrounding said plunger member and one of said grooves being disposed between said recesses and said chamber, and conduit means for supplying a pressurized fluid to each of said recesses and for conveying from said exhaust grooves any fluid collected therein.

I also desirably provide a similar plunger and cylinder arrangement wherein said exhaust grooves are equally spaced from said recesses.

I also desirably provide a similar plunger and cylinder arrangement wherein each of said pressure recesses is circumferentially spaced from adjacent recesses and axially spaced from said exhaust grooves such that the fluid resistance of the surrounding interfacial areas of said member and said housing at least aid in establishing an essentially hydrostatic pressure condition in at least some of said recesses, said one groove interrupting any leakage from said pressure chambers to said interfacial areas.

I also desirably provide a similar plunger and cylinder arrangement wherein said conduit means include a sep-

arate inlet duct coupled to each of said recesses, and discrete flow restricting means are coupled to said ducts respectively, said flow restricting means and said interfacial areas and said grooves being interrelated such that leakage from said pressure recesses is substantially less than the exhaust capability of said grooves.

As in the preceeding provision, the last-mentioned relationship can be readily calculated from viscosity of the bearing fluid, extent of the interfacial areas, systemic pressures, flow parameters of the particular flow restricting means used, and sizes of the exhaust grooves and drain ducts coupled thereto, as evident to the artisan.

I also desirably provide a similar plunger and cylinder arrangement wherein a plurality of outlet or exhaust ducts are coupled at spaced locations to each of said grooves in avoidance of pressure build-up and differential pressures within said grooves.

I also desirably provide a similar plunger and cylinder arrangement wherein flow restricting means for each of said pressure recesses are mounted respectively in ducts for said recesses formed in said cylinder.

During the foregoing discussion, various objects, features and advantages of the invention have been set forth. These and other objects, features and advantages of the invention together with structural details thereof will be elaborated upon during the forthcoming description of certain presently preferred embodiments of the invention and presently preferred methods of practicing the same.

In the accompanying drawings I have shown certain presently preferred embodiments of the invention and have illustrated certain presently preferred methods of practicing the same, wherein:

FIG. 1A is a longitudinally sectioned, partial view of one form of my novel plunger and cylinder assembly as provided with my unique fluid bearing arrangement and showing the plunger thereof in its extended position. FIG. 1A is taken generally along reference lines IA of FIGS. 2 and 3;

FIG. 1B is a similar view of the apparatus illustrated in FIG. 1A but showing the plunger thereof in its retracted position. FIG. 1 is taken generally along reference line IB of FIGS. 2 and 3;

FIG. 2 is a cross-sectional view of the apparatus as shown in FIGS. 1A and 1B and taken substantially along reference line II—II thereof;

FIG. 3 is a similar view but taken along reference line III—III of FIGS. 1A, 1B; and

FIG. 4 is a longitudinally sectioned partial view similar to FIG. 1A but illustrating a modification of the invention for use with a different form of metering device.

With initial reference to FIGS. 1A and 1B of the drawings, the exemplary form of the plunger and cylinder arrangement 10 shown therein is provided with a pair of my novel fluid bearing systems 12, 14. The fluid bearing arrangements 12, 14 are axially spaced along the length of plunger 16 or other movable member, which is reciprocable or otherwise movable within cylinder or housing 18. A spaced pair of the fluid bearing arrangement 12, 14 can be utilized in the illustrated manner where it is anticipated that cocking or canting forces may be applied to the piston or plunger 16. In other applications, it is contemplated that a different number of the fluid bearing arrangements of the inven-

tion can be employed. For example in a relatively short plunger or piston only one such fluid bearing arrangement need be employed. For pistons or plungers which are relatively longer than that shown three or more such fluid bearing arrangements can be used.

In the illustrated application the cylinder 18 is mounted in the bottom portion of a rolling mill stand 20 where it is supported by means of shoulders 22. The cylinder 18 thus is supported directly below the mill stand housing rails 24 for engagement with a roll change plate 25 and the lower back-up roll bearing chock 26, which is thereby raised and lowered for the usual purposes.

The piston or plunger 16 is considerably elongated in this example to accommodate a spaced pair of my fluid bearing systems 12, 14 as aforesaid, in order to prevent cocking or canting. The piston 16 is stepped at 28 to afford a fluid space 30 (FIG. 1B) into which introduction of a high pressure hydraulic fluid serves as a safety hold-down and also ensures downward movement of the piston 16 when the pressure in the cylinder blank end space 32 is removed. A duct 31 is bored through the cylinder wall 18 and through its blank end 33 and end cap 33a so that a hydraulic fluid can be introduced therethrough and into the hold down space 30 in this example through a pair of apertures 35, as evident from FIG. 1B. A safety stop 34 is formed on an upper one 36 of the cylinder liners 36, 38 and avoids any possibility of the piston 16 being blown out of the cylinder 18. Although, FIGS. 1A and 1B illustrate my novel plunger and cylinder arrangement in an upright disposition, such disposition is merely exemplary, and it will be readily apparent from the following description that my plunger and cylinder arrangement 10 can be used in a variety of positions or successive positions.

The cylinder 18 proper is considerably oversized to accommodate the liner members 36, 38 between the reduced or upper portion of the piston 16 and its lower portion and the respectively juxtaposed areas of the cylinder 18.

In the illustrated embodiment of my invention (FIGS. 1A, 1B) one of my novel fluid bearing arrangements, for example the arrangement 12, is formed on the upper cylinder liner 36 while the other fluid bearing arrangement 14 is formed on the lower liner 38. As better shown in FIGS. 1A, 1B and 2 the upper fluid bearing arrangement comprises a plurality of recesses or "pads" 40 formed in this example on the inner surface of the upper cylinder liner 36. As shown in FIG. 2 four such pads 40 are employed, each of which is elongated in the circumferential direction. The pads 40, moreover, are disposed in end-to-end juxtaposition and in a circumferential equally spaced array about the upper liner 36 and the juxtaposed portion of the piston or plunger 16. Each of the pads 40 is coupled through ducts 42 and 44 (FIGS. 1A and 2) which extend respectively through the upper liner 36 and the wall of the cylinder 18 as illustrated. Desirably each of the ducts 42 terminate at their lower ends in a circumferentially extending recess 42a to maintain communication between the ducts 42, 44 despite slight misalignment of the upper liner 36 relative to shouldered portion 46 of the cylinder 18.

For purposes of illustration the upper portions of FIGS. 1A, 1B are sectioned at angles of about 45° to the lower portions thereof (cf. FIGS. 2 and 3), as denoted by break lines "A". Thus, the axial ducts 44,

which are bored through the cylinder wall 18, extend to the blank end 33 of the cylinder 18 and through the blank end cap 33a thereof, where they are coupled to flow restriction means, such as the flow restriction means 48 described below in connection with the lower fluid bearing system 14. With this arrangement individual portions of pressurized fluid can be introduced through each of the ducts 42, 44 and individually metered into the pressure pads 40 through the aforementioned flow restriction means, as likewise described in greater detail below. The pressure of such fluid portions and the particular distribution of the pressure pads 40 provide a very accurate centering action for the juxtaposed portion of the plunger or piston 16. Moreover, as a small but constant flow of fluid is emitted from each of the pads 40 the passage of the plunger 16 through the upper liner 36 is continuously lubricated. Of course, the amount of fluid emitted in this matter is controlled by the relatively close fit, which can involve normal manufacturing tolerances or even larger, of the plunger 16 within the cylinder liner 36 and by the axial distances along the junction therebetween.

To prevent the fluid thus discharged from the pressure pads from leaking outwardly through the rod-end cap 50 of the cylinder 18, a circumferential collection or exhaust groove 52 is provided as shown in FIGS. 1A, 1B and 2. To militate against the possibility of differential pressure buildup within the groove 52, in the illustrated example, four pairs of exhaust ducts are provided for the groove 52. These exhaust ducts, as presently described, are similarly connected to a second circumferential exhaust groove 54 which is spacedly disposed on the upper liner 36 and between the pressure pads 40 and the holddown space 30 described above. The second exhaust groove 54 thus collects downward leakage from the pressure pads 40 and upward leakage from the holddown space 30, as neither the plunger 16 nor the cylinder liners 36, 38 are provided with any form of packing in these areas.

The circumferential exhaust grooves 52, 54 are connected in this example through eight vertical pairs of transverse ducts 56 formed in the upper liner 36. Each pair of transverse ducts 56 are connected to a single axial duct 58 likewise formed in the cylinder liner 36. Each adjacent pair of axial ducts 58 as better shown in FIG. 2 adjoin a circumferentially extending recess 60 formed on the underside (as viewed in the FIG. 1B; see also FIG. 3) of the upper liner 36. Thence, the axial ducts 58 adjoin through the circumferential recess 60 a like number of axial ducts 62 formed in the cylinder wall 18. The cylinder ducts 62 are extended through the lower portion of the cylinder 18 and its blank end 33 to a collecting manifold 64 and outlet 66 formed in the blank-end cap 33a. The circumferential recesses 60 permit slight misalignment of the upper liner 36 with the cylinder 18 while preserving communication between the axial liner ducts 58 and the axial cylinder ducts 62.

Owing to the free flowing characteristics of the effluent collected in the upper exhaust groove 52 (as in all of the exhaust grooves) and the multiple ducting arrangement 56-62 therefor, no leakage of hydraulic fluid will be evident at the rod-end cap 50 of the cylinder 18. A circumferential gasket 122 seated in groove 124 merely prevents the entry of foreign matter into the plunger-cylinder arrangement 10.

In the illustrated application a relatively large number of exhaust ducts 56-62 are utilized to avoid any possibility of differential pressure build up within the exhaust grooves 52, 54 or in any portion of these grooves. In other applications of the invention it is to be understood that a greater or lesser number of exhaust ducts, such as the ducts 56-62 can be employed. Similarly, a greater or lesser number of pressure pads 40 and individual ducts 42-44 therefor can be employed, as long as two or more such pressure pads are circumferentially and equidistantly spaced about the plunger or other movable member to be centered thereby.

The lower fluid bearing arrangement 14 is similarly constructed and thus includes a similar spatial arrangement of exhaust grooves 52', 54' and pressure pads 40'. As evident from FIG. 3 the exhaust grooves 52', 54' are likewise provided with multiple outlet connections, in this example eight such connections, although a different number can be employed as denoted previously. In the illustrated embodiment the exhaust grooves 52', 54' and the pressure pads 40' are each recessed into the inner surface of the lower liner 38. Thus, there are projections extending from the inner surface of the liner 38 on which the juxtaposed portions of the plunger 16 can hang up. The exhaust grooves 52, 54 and pressure pads 40 of the previously described fluid bearing arrangement 12 are similarly recessed upon the inner surface of the upper liner 36.

In the case of the lower fluid bearing arrangement 14, however, an additional drainage or exhaust groove 68 or 70 is juxtaposed respectively to each exhaust groove 52' or 54'. The drainage grooves 68, 70 are formed in this example partially on the outer surface of the lower liner 38 and on the juxtaposed inner surface of the cylinder 18. A number of transverse ducts 72 and 74 extend radially through the lower liner 38 and couple the exhaust grooves 52' and 54' with their respectively juxtaposed drainage grooves 68 and 70. The drainage grooves 68, 70 collect any fluid which may enter the junction between the outer surface of the lower liner 38 and the cylinder 18 but more importantly compensate any rotational misalignment of the lower liner 38 with the cylinder 18. Such misalignment may occur of course either upon assembly of the plunger and cylinder arrangement 10 or upon subjection to vibratory or shock forces during its subsequent use.

The drainage grooves 68, 70 in this example are coupled through a like number of radial passages 76, 78 in the cylinder 18 to a number of axial ducts 80 likewise extending through the wall of the cylinder 18. As evident from FIG. 1A each axial cylinder duct 80 is coupled to an axial pair of the radial ducts 76, 78. From FIG. 3, a total of eight axial ducts 80 are employed, although a different number can be used depending upon the application of the invention.

In the blank end portion 33 of the cylinder 18 each of the ducts 80 communicates through an appropriate passage with the aforementioned manifolding groove 64 and outlet 66 in the blank end cap 33a. The lower end portions of the axial cylinder ducts 64 for the exhaust grooves 52, 54 of the upper bearing 12 can be similarly adjoined at their lower ends (not shown) to the manifold groove 64.

As in the case of the upper bearing 12, each of the pressure pads 40' of the lower bearing 14 are provided with an independently metered source of pressurized

fluid. Thus, the pads 40' each include an interconnecting radial passage 82 (FIGS. 1A and 3) extending through a juxtaposed portion of the lower liner 38. The radial passages 82 in turn are connected respectively to expanded recesses 84 formed on the outer surface of the lower liner 38 and thence to radial ducts 86 in the cylinder wall 18. The recesses 84, which are expanded in the circumferential direction (FIG. 3) compensate for any rotational misalignment of the lower liner 38 relative to the cylinder 18, which may occur during assembly or use of the apparatus. The radial ducts 86 are coupled individually and respectively to a like number of axial cylinder ducts 88, each of which is provided with the flow restricting or metering means 48 mentioned previously. One of such means 48 is shown in FIG. 1A, and each such restricting means 48 is mounted in the lower end portion of the associated axial cylinder passage 88. As noted previously, similar flow restricting or metering means 48 are similarly located in the lower end portion (not shown) of each of the axial cylinder passages 44 for the upper bearing arrangement 12.

In the plunger and cylinder arrangement 10 (FIG. 1A) the restriction means 48 can take the form of a restrictor, in this case a bar 90 of steel or other suitable structural material provided with an accurately bored passage (not shown) extending longitudinally there-through. The length and diameter of the bored passage are selected to provide desired, very accurate flow rate through the restrictor, in accord with known fluid parameters. A portion of the upper surface of the bar 90 can be threaded at 92 for engagement with a tapped end portion of each of the axial ducts 88 of the lower bearing 14 or of the axial ducts 44 of the upper bearing 12. Alternatively as shown in FIG. 4 the flow restricting means 48 of the aforementioned axial passages can take the form of a jet structure, such as the structure 94, installed in the axial duct 88'. The jet 94 can be fabricated after the manner of jet structures available from Lee Company, Westbrook, Connecticut. It is contemplated also that the restrictor 90 or the jet structure 94 can be replaced with an orifice plate, a length of capillary tubing, or a needle valve and the like, depending upon a particular application of the invention.

It will be understood, of course, that one or both of the exhaust grooves and the pressure pads of the fluid bearing 12 or 14 can be similarly formed on the outer surfaces of the plunger 16. In that case the several supply and exhaust ducts for the pressure pads and exhaust grooves respectively, can be extended (not shown) through the plunger 16. It is also contemplated that, in certain applications of the invention where the centering forces may be less critical or where tighter tolerances or larger interfacial areas 112 or 114 are available, that separate metering of the pressurized fluid into the pressure pads 40 or 40' can be eliminated while maintaining acceptable centering forces at the respective pressure pads and a proper segregation between the fluid bearing 14 and the main pressure chamber 32. On the other hand, it is apparent that the discrete flow restricting means 48 or 94 can be replaced by a restriction formed integrally by the wall surfaces of the ducts 82-88 for the lower pressure pads 40' or of the ducts 42-44 for the upper pressure pads 40. For example, the radial ducts 86 or 42b respectively can be provided with sufficiently small diameters to accomplish this modification of the invention.

Referring again to FIGS. 1A and 1B of the drawings pressurized fluid is supplied to each of the axial cylinder ducts 88 of the lower bearing arrangement 14 through a circumferentially extending manifold 96 formed in this example partially on the outer surface of the cylinder blank end 33 and on the juxtaposed inner surface of the blank end cap 33a. It will be understood, of course, that the manifold 96 can be formed entirely on one or the other of these items. The inlet ends 98 of each of the restrictors 92 communicate with the manifold 96, as evident from FIG. 1A. The restrictors (not shown) for the upper bearing arrangement 12 are similarly disposed with respect to the manifold 96. A source (not shown) of pressurized fluid is coupled to the manifold 96 through its inlet 100 in the cylinder blank-end cap 33a.

The upper exhaust groove 52' of the lower bearing arrangement 14 is thereby disposed between the hold down cylinder space 30 and the pressure pads or recesses 40'. The upper exhaust groove 52', then, quickly removes any leakage and avoids any pressure build up as a result of fluid flowing upwardly (as viewed in FIGS. 1A, 1B) from the pressure pads 40'. Similarly the upper groove 52' removes leakage and prevents pressure buildup of fluid flowing downwardly from the hold-down space 30. The lower exhaust groove 54' of the lower fluid bearing 14 similarly removes and prevents pressure build up in any fluid moving downwardly and axially from the pressure pads 40' and any fluids moving upwardly from the main pressure chamber 32 of the cylinder 18.

In many applications, the lower end portion of the plunger 16 can be elongated such that the normal interfacial tolerances at the adjacent portion of the cylinder or liner 38 are sufficient to minimize flow of the hydraulic fluid from the main cylinder chamber 32 axially to the adjacent exhaust groove 54'. In those cases where the tolerances are less restricted, leakage from the main cylinder chamber 32 to the exhaust groove 54' can be minimized by further lengthening the lower end portion of the cylinder 16. Thus, even in the fully extended position of the plunger 16 (FIG. 1A) there is sufficient axial distance between the lower end of the plunger and the lower exhaust groove 54' to provide a form of labyrinthine seal, with sufficiently high fluid friction to minimize leakage.

However, it is an unexpected feature of the invention that the adjacent exhaust groove 54' and associated ductwork uniformly removes all such leakage before it reaches the pressure pads 40 of the fluid bearing 14. Similarly the same groove uniformly removes any leakage from the pads 40' before it reaches the main cylinder chamber 32. Most importantly, in view of the possibility of considerable leakage counterflows in this area, either from excessive wear or from an initial employment of loose tolerances, the groove 54' and associated multiple exhaust ducts prevent the development of differential pressures in any portion of the exhaust groove. Thus, as in the case of the other exhaust grooves used in the plunger-cylinder 10, all potential leakage is exhausted to areas of essentially zero pressure.

There may be a few applications, where labyrinthine sealing is not sufficiently availing, and leakage from the main cylinder chamber 32 to the adjacent exhaust groove 54' can be minimized further by use of one or more piston rings such as the ring 102. It will be under-

stood, of course, that conventional piston rings can be employed for this purpose. In the illustrated embodiment, however, the piston ring 102 is of the split variety and can be fluid-actuated to improve its sealing capabilities.

As better shown in FIGS. 1A and 1B, the piston ring 102 is provided with a number of flow restricting pressure means 104, with four such means being utilized in this example as shown in FIG. 3. It will be appreciated that a greater or lesser number of flow restricting means 104 can be employed. I prefer to use the flow restricting means 104 against the event that excessive leakage may occur in one area of the piston ring 102. In that case the use of the flow restricting means for that area does not result in loss of fluid pressure for the entire piston ring 102. In consequence the remainder of the piston ring 102 maintains its sealing function to minimize leakage of hydraulic fluid. Although the flow restricting means 104 are illustrated as jets similar to the jets structures described above (FIG. 4), it will be appreciated that other flow-restricting means such as discussed previously can be substituted.

Each flow restricting means 104 is connected through radial ducts 106, and thence to a longitudinal or axial duct 108 to the "rod" end of the plunger 16. At this point a suitable source (not shown) of pressurized fluid is coupled to the tapped end portion 110 of the axial passage 108.

In most applications the piston ring 102 can be eliminated, as noted above and if necessary the plunger 16 can be further elongated in the region of its blank end to enhance the labyrinthine effect of the thus elongated interfacial surfaces of the piston 16 and the liner 38, for purposes of segregating the main cylinder chamber 32 and the adjacent fluid bearing 14, while providing adequate lubrication.

In operation, it will be recalled that the pressure pads 40 or 40' of the fluid bearing systems 12, 14 are equally spaced about the circumference of the plunger 16. Such equal spacing affords a centering action upon the plunger 16. When coupled with an accurate and individual metering into the pressure recesses 40 or 40', a very accurate centering action results. Desirably but not essentially the centering action is applied at two spaced locations along the length of the plunger 16 by the use of two or more fluid bearing systems 12, 14. This arrangement ensures that the plunger 16 will not be canted or cocked within the cylinder 18 or other housing regardless of the operating position of the plunger and cylinder arrangement 10.

To render the fluid bearing arrangement 12 or 14 effective for these purposes it is essential that each of the bearing pads 40 or 40' be capable of maintaining a relatively high pressure and/or a low leakage relationship with the adjacent surfaces of the plunger 16. This is accomplished by individually and accurately metering a flow of pressurized fluid into each of the pressure pads such that the bearing characteristics of the pocket of fluid entrapped therein are essentially hydrostatic. The hydrostatic relationship results partially from the individual metering of the relatively small fluid flow into each of the pressure pads 40 or 40' and also to the relatively large expanse of plunger—liner or —cylinder interfacial areas 112 or 114, which surround each of the pressure pads 40 or 40' respectively. As noted above in connection with the main cylinder chamber 32, these interfacial areas or clearances, under normal condi-

tions, considerably maximize fluid friction and minimize leakage from each of the pressure pads, such that a controlled very small flow into each pad coupled with a similarly controlled effluent therefrom promotes an essentially hydrostatic relationship.

Of equal or perhaps greater importance, the individual and accurate metering of pressurized fluid into each of the pressure pads maintains the fluid bearing and centering capabilities of the remainder of the pads, should one or more of the pads be subjected to a deterioration in the sealing characteristic of its interfacial sealing areas, as when the plunger 16 is driven off center with respect to the cylinder 18 or one or both of the liners 36, 38. In such cases excessive leakage from the less-sealed pad or pads cannot occur, and the pressure available for recentering the plunger 16 at the remainder of the pads cannot be diminished, owing to the individual control of the fluid flows to each of the pressure pads by the flow restricting means 48. Thus, the full system pressure of the fluid bearing 12 or 14 is available for recentering the plunger 16 until the diminished sealing capability of the affected pressure recess is restored.

It will now be apparent that the same course (not shown) of hydraulic fluid can be coupled to the various points of utilization in the plunger—cylinder arrangement 10, i.e. to the upper and lower pressure pads 40, 40' through their common inlet 100, the main cylinder chamber 32 through its inlet 116, and to the holddown space 30 through its inlet 118. It will be understood of course that the axial ducts 44 for the upper pressure pads 40 can be coupled to a separate manifold (not shown), similar to the manifold 96, provided with its own inlet (not shown). At this point it may be noted that while the axial exhaust ducts 80 are bored for convenience through the inlet manifold 96 as evident from FIG. 1B, there is no communication therebetween owing to plugs 120. It will also be appreciated that certain applications of the invention can result in elimination of one of the fluid bearing arrangements 12 or 14 or of the cylinder hold down space 30. It is also evident that separate sources (not shown) of pressurized fluid can be coupled respectively to the hold down space 30 and the pressure pads 40, 40'. Owing to the complete segregation of the various pressure utilizing items of the plunger—cylinder 10, by means of the exhaust grooves 52, 54 and 52', 54', it is possible to use pressure sources of a relatively higher or lower pressure with respect to the main cylinder chamber 32.

From the foregoing it will be apparent that novel and efficient forms of Improved Plunger and Cylinder Arrangement have been described herein. While I have shown and described certain presently preferred embodiments of the invention and have illustrated presently preferred methods of practicing the same it is to be distinctly understood that the invention is not limited thereto but may be otherwise variously embodied and practiced within the spirit and scope of the invention.

I claim:

1. A fluid bearing plunger and housing arrangement comprising a housing, a plunger member closely fitted within said housing for reciprocation therein, said housing defining a pressure chamber therein for actuating said plunger member, a plurality of discrete pressure recesses formed in an inner surface of said housing and juxtaposed to said plunger member, said recesses

circumferentially surrounding said plunger member and in addition being substantially equally spaced from one another, a pair of exhaust grooves formed in said inner housing surface and juxtaposed to said plunger member, said exhaust grooves being axially spaced one of each side of said recesses, said grooves circumferentially and continuously surrounding said plunger member, one of said grooves being disposed between said recesses and said chamber, and conduit means for supplying a pressurized fluid to each of said recesses and for conveying from said exhaust grooves any fluid collected therein, each of said pressure recesses being circumferentially spaced from adjacent recesses and axially spaced from said exhaust grooves to an extent that the fluid resistance of surrounding first interfacial areas of said member and said housing establish an essentially hydrostatic pressure condition in said recesses, said first interfacial areas extending uninterruptedly between adjacent one of said pressure recesses and uninterruptedly between said recesses and said exhaust grooves, said plunger member extending beyond said one groove and toward said chamber so that second interfacial areas of said member and said housing between said chamber and said one groove effectively isolate said one groove from said chamber, said second interfacial areas and said one groove interrupting any leakage from said pressure chamber to said first interfacial areas and said pressure recesses.

2. The combination according to claim 1 wherein said recesses are elongated and are disposed in circumferentially spaced end-to-end relation.

3. The combination according to claim 1 wherein said exhaust grooves are equally spaced from said recesses.

4. The combination according to claim 1 wherein flow restricting means for each of said pressure recesses are mounted respectively in ducts for said recesses formed in said housing.

5. The combination according to claim 1 wherein an expandable piston ring is mounted on said plunger member between said pressure chamber and said one exhaust groove, and fluid means are provided for expanding said piston ring, said fluid means including a branched duct extending within said plunger member to several points about said ring and flow restricting means in each of the duct branches.

6. The combination according to claim 1 wherein a plurality of outlet or exhaust ducts are coupled at spaced locations to each of said grooves in avoidance of pressure build-up and differential pressures within said grooves.

7. A fluid bearing plunger and housing arrangement comprising a housing, a plunger member closely fitted within said housing for reciprocation therein, said housing defining a pressure chamber therein for actuating said plunger member, a plurality of pressure recesses formed in an inner surface of said housing and juxtaposed to said plunger member, said recesses circumferentially surrounding said plunger member and in addition being substantially equally spaced from one another, a pair of exhaust grooves formed in said inner housing surface and juxtaposed to said plunger member, said exhaust grooves being axially spaced one on each side of said recesses, said grooves circumferentially and continuously surrounding said plunger member, one of said grooves being disposed between said recesses and said chamber, conduit means for supply-

ing a pressurized fluid to each of said recesses and for conveying from said exhaust grooves any fluid collected therein, and ducts for said recesses coupled to an intake manifold formed in a blank end portion of said housing, the major portion of said ducts extending from said manifold through a wall of said housing.

8. A fluid bearing plunger and housing arrangement comprising a housing, a plunger member closely fitted within said housing for reciprocation therein, said housing defining a pressure chamber therein for actuating said plunger member, a plurality of pressure recesses formed on one of said plunger member, and said housing, said recesses circumferentially surrounding said plunger member and in addition being substantially equally spaced from one another, a pair of exhaust grooves formed in one of said plunger member and said housing and axially spaced one on each side of said recesses, said grooves circumferentially and continuously surrounding said plunger member, one of said grooves being disposed between said recesses and said chamber, conduit means for supplying a pressurized fluid to each of said recesses and for conveying from said exhaust grooves any fluid collected therein, and exhaust ducts for said grooves coupled to an exhaust manifold formed in a blank end portion of said housing, said exhaust ducts extending to said manifold through one of said plunger and a wall of said housing.

9. The combination according to claim 1 including at least one additional array of said pressure recesses being similarly formed and spaced along the length of said plunger member from the first-mentioned array of said recesses, and an additional exhaust groove similarly disposed on the outer side of said additional array, the additional fluid bearing system thus formed minimizing or preventing altogether canting or cocking of said plunger member relative to said housing.

10. A fluid bearing plunger and housing arrangement comprising a housing, a plunger member closely fitted within said housing for reciprocation therein, said housing defining a pressure chamber therein for actuating said plunger member, a plurality of discrete pressure recesses formed on one of said plunger member and said housing, said recesses circumferentially surrounding said plunger member and in addition being substantially equally spaced from one another, a pair of exhaust grooves formed in one of said plunger member and said housing and axially spaced one on each side of said recesses, said grooves circumferentially and continuously surrounding said plunger member, one of

said grooves being disposed between said recesses and said chamber, conduit means for supplying a pressurized fluid to each of said recesses and for conveying from said exhaust grooves any fluid collected therein, said housing being enlarged and spaced from said plunger member, and a liner member closely fitted within the space between said plunger member and said housing, said conduit means being extended through said housing and said liner member.

11. The combination according to claim 10 wherein a circumferential extension of said duct means is formed on one of said liner member and said housing in compensation for circumferential displacement of said liner member relative to said housing.

12. A fluid bearing plunger and housing arrangement comprising a housing, a plunger member closely fitted within said housing for reciprocation therein, said housing defining a pressure chamber therein for actuating said plunger, a plurality of discrete pressure recesses formed in an inner surface of said housing and juxtaposed to said plunger member, said recesses circumferentially surrounding said plunger member and in addition being substantially equally spaced from one another, a pair of exhaust grooves formed in said housing inner surface and juxtaposed to said plunger member, said exhaust grooves being axially spaced one on each side of said recesses, said grooves circumferentially and continuously surrounding said plunger member, one of said grooves being disposed between said recesses and said pressure chamber, conduit means for supplying a pressurized fluid to each of said recesses and for conveying from said exhaust grooves any fluid collected therein, each of said pressure recesses being circumferentially spaced from adjacent recesses and axially spaced from said exhaust grooves such that the fluid resistance of the surrounding interfacial areas of said plunger member and of said housing at least aid in establishing an essentially hydrostatic pressure condition in said recesses, said one groove interrupting any leakage from said pressure chamber to said interfacial areas, said conduit means including a separate inlet duct coupled to each of said recesses, and discrete flow-restricting means coupled to said ducts respectively, the respective configurations of said flow-restricting means and of said interfacial areas and of said grooves being structurally interrelated such that leakage from said pressure recesses is substantially less than the exhaust capability of said grooves.

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