Aug. 23, 1955 J. M. TOWLER ET AL 2,715,875

AXIAL TYPE RECIPROCATING ENGINE

Filed Oct. 21, 1949 3 Sheets-Sheet 1

Fig. 1

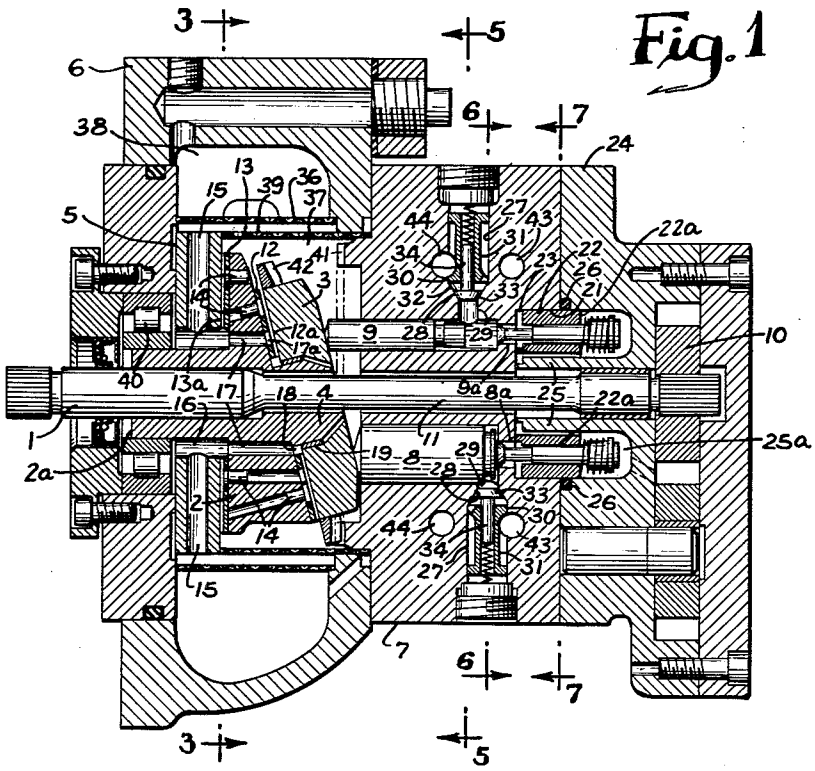
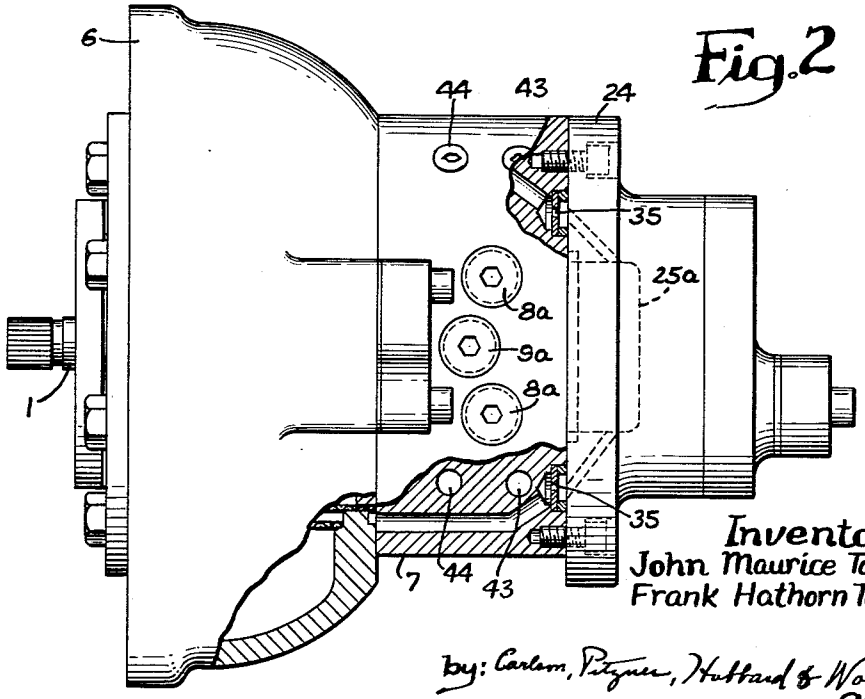


Fig. 2



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Fig. 3

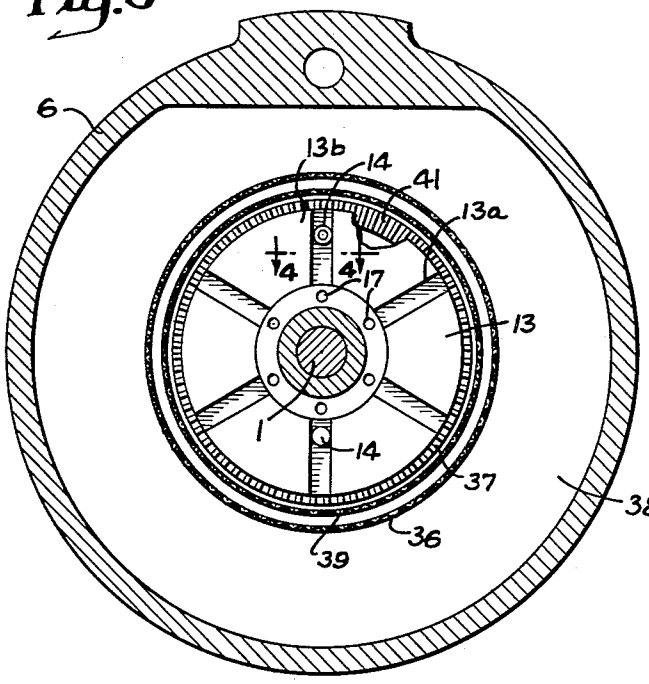


Fig. 4

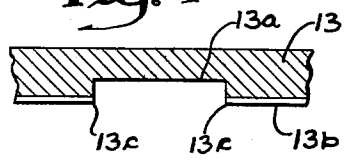
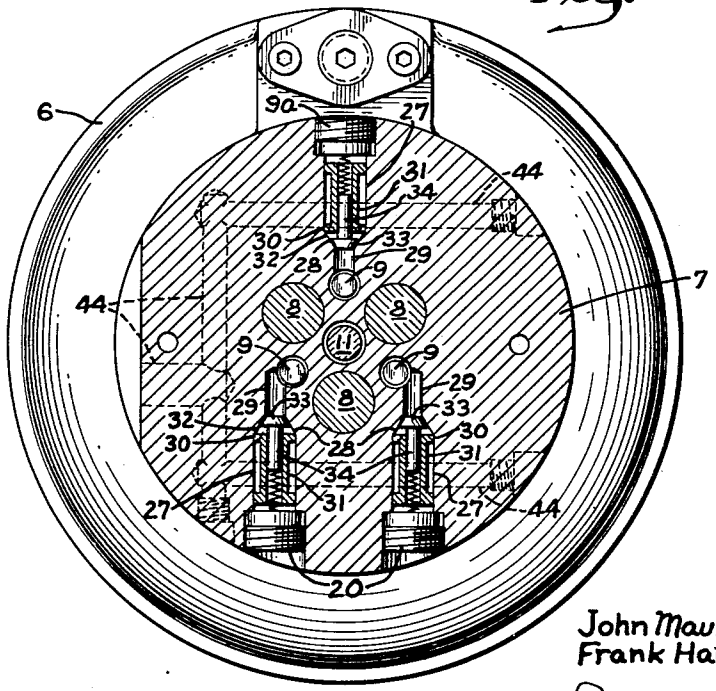


Fig. 5



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Fig. 6

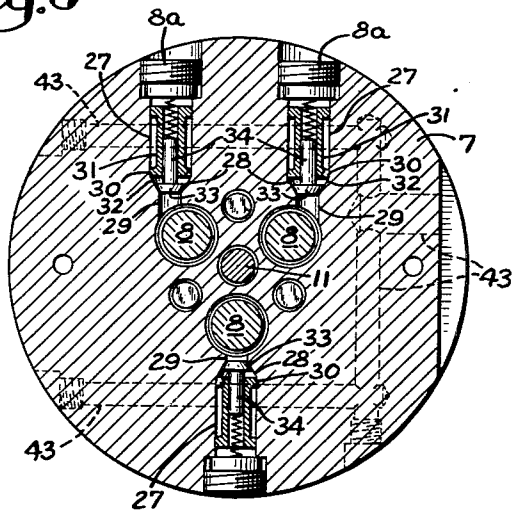
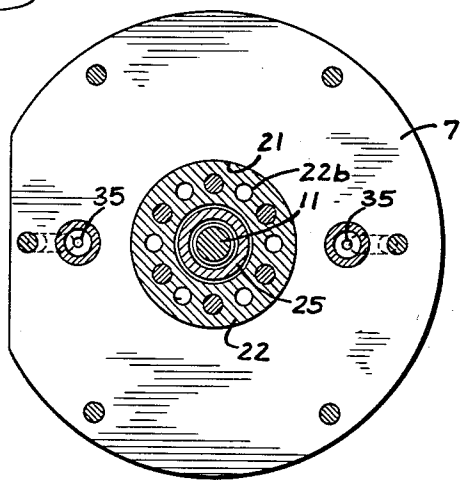


Fig. 7



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AXIAL TYPE RECIPROCATING ENGINE

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Application October 21, 1949, Serial No. 122,698

Claims priority, application Great Britain October 26, 1948

12 Claims. (Cl. 103—173)

This invention relates to hydraulic multi-ram pumps of the kind having a cylinder block in which the pump rams are disposed around a common axis the rams being reciprocated either directly or through push rods without the inter-position of connecting rods by means of a ram actuating device comprising a wobbler which is given angular oscillation in two planes in relation to the axes of the rams by means of a swash member. The above description is to be read as a definition of the term "of the type specified" used in the claims at the end of this specification. It is to be understood that if a pump or compressor of the above type is supplied with pressure fluid hereinafter referred to as liquid and provided with suitable valve gear it will also operate as a rotary engine or motor. Accordingly the term "pump" hereinafter used in this specification is to be read as including a motor as well as a compressor.

One of the objects of the present invention is to provide a pump of the above type which has improved means for taking the thrust loads on the wobbler and the swash member which will not crush under the heaviest loads the pump is likely to be subjected to. A further object of the invention is to improve generally the efficiency of such pumps.

In an axial ram pump according to the present invention the axial load on the wobbler and on the swash member is taken by bearings consisting of opposed plane and relatively rotatable surfaces which are held out of metal to metal contact by an interposed film of oil. Each said bearing may comprise a washer one of which is disposed between the wobbler and the inclined face of the swash member and the other between the rear face of the swash member and a supporting member or reaction plate, with said washers secured against rotation relative to the swash member, one of the co-acting surfaces of each bearing being composed of a relatively soft bearing metal and one of said surfaces in each bearing being formed with radial grooves which are open at their inner ends to a space in which is available, during use, a continuous supply of oil.

Where washers are used the soft surfaces are on the outside faces of the assembled washers so as in one case to contact the rear face of the wobbler and in the other case the opposing face of the reaction plate or supporting surface for the swash member. Where no washers are used the soft metal surfaces may be applied to the wobbler and the supporting member for the swash member respectively, with the faces of the swash member of hard metal. Alternatively the two faces of the swash member may be of the soft metal when the co-acting faces of the wobbler and the supporting member will be of hard metal. In the preferred construction, both the wobbler and the swash member are given freedom to tilt in relation to their supporting surfaces and one manner of obtaining this effect will be explained in the following description of the example illustrated in the accompanying drawings in which:

Fig. 1 is a longitudinal section through an axial ram pump embodying the features of the present invention.

Fig. 2 is a view of the same pump in elevation with a small part broken away to expose a feature which is hereinafter referred to.

Fig. 3 is a transverse sectional view taken in a plane substantially on the line 3—3 of Fig. 1.

Fig. 4 is a fragmentary sectional view taken in a plane substantially on the line 4—4 of Fig. 3.

Figs. 5, 6 and 7 are transverse sectional views respectively taken in planes substantially on the lines 5—5, 6—6 and 7—7 of Fig. 1.

The axial ram pump illustrated in the drawings comprises a driving shaft **1**, a swash member **2** that is a press fit on the driving shaft, a wobbler **3** mounted on a nose or extension **4** of the swash member, a reaction plate **5** for supporting the swash member against the axial load, a crank case **6** enclosing the swash mechanism, a pump body **7** containing axially disposed rams **8** and **9** with their valves, and a simple gear pump **10** functioning as a booster and driven by a flexible extension **11** of the driving shaft **1**.

In the pump illustrated the thrust on the wobbler and the thrust on the swash member are taken by bearings comprising a washer **12** disposed between the wobbler and the inclined face of the swash member and a further washer **13** disposed between the rear face of the swash member and the opposing face of the reaction plate **5**. These washers are held against rotation in relation to the swash member by dowel pins **14** which fit in holes in the swash member. These washers, on the sides which oppose the wobbler and the reaction plate respectively, are faced with a covering of a relatively soft bearing metal **13***b* and such soft facings are traversed by radial grooves **13***a* for the distribution of oil as shown in Fig. 4. We have found that it is an advantage that these radial grooves shall have sharp edges **13***c* to avoid "taper lead" at the leading edge of each bearing sector.

Oil is caused to circulate to the bearing surfaces by being first drawn inwards through radial holes **15** in the reaction plate **5** into an annular space **16** between the reaction plate and rear boss **2***a* on the swash member. From this annular space **16** some of the oil passes up the radial grooves in the face of the washer **13** and the remaining oil flows through a series of axial holes **17** into the grooves in the opposing face of the other washer **12**. Owing to the angularity of the forward face of the swash member some of these holes **17** will open into the grooves part way along their length whilst others, of these holes will open into an annular space **18** at the centre of the washer **12**. The oil in the space **18** is free to find its way between the wobbler **3** and a bush **19** on the nose **4** of the swash member.

In the pump illustrated the pump bores are divided into two sets with the bores of one set spaced alternately with the bores of the other set, and the area of one set being less than, preferably half, the area of the other set. This arrangement has as one of its features the disposing of the delivery valve bores in planes normal to the axis of the pump shaft with the delivery bores associated with one set of rams disposed in a plane which is nearer to the swash member than is the plane in which are disposed the delivery bores associated with the other set of pump bores. The separate delivery passages for each set of pump bores are also disposed in different planes, those, for example, associated with the bores of smaller area being disposed nearer to the swash member than those associated with the bores of greater area or vice-versa.

According to a preferred arangement six ram bores and rams are provided in two sets of three, for rams **8** and **9** respectively, with all six of the delivery valve

bores **20** being parallel to each other and grouped on the opposite sides of the pump body. The delivery bores in each group are staggered, for example two **8***a* being associated with one set of pump bores and one **9***a* with the other set at one side of the body as shown in Fig. 2, the delivery passages (not shown) being arranged so as to unite two delivery bores on one side of the pump body with the one of the same set on the other side of the pump body and to connect all three with the delivery main.

In the improved pump of the present invention, a novel construction and arrangement of inlet valve guides and suction passages is used, this consisting in forming in the end of the pump body opposite to that housing the rams, a single cylindrical cavity **21** concentric with the pump driving shaft and into this cavity is pressed an annular bush **22** to as to leave a gap or space **23** between the end of the bush and the bottom of the cavity with part of the bush projecting out of the pump body to form an annular spigot. Concentric with the pump bores and suction valve bores a corresponding number of parallel bores are formed in the bush and these constitute the suction valve guides. The suction passages may conveniently be formed by providing holes in the bush between the guide bores leading to the space **23** between the end of the bush and the corresponding bore **21** in the pump body.

In the above construction the end of the pump body is closed by means of a cover plate **24** which locates on the projecting portion of the suction valve guide and is itself provided with an annular portion **25** which projects through a hole in the centre of the aforesaid annular suction valve guide bush **22** thus forming the inlet manifold **25***a* and makes a joint at **26** with the pump body **7**. The cover plate may conveniently form a part of the gear type boost pump **10** and the extension **11** of the main driving shaft for driving this pump can pass through the above mentioned annular projecting portion **25** of the cover.

The bores **27** which incorporate the delivery valve seats are cylindrical and terminate at their inner ends in frusto-conical portions **28**, which themselves terminate in cylindrical passages **29**, the junctions between the frusto-conical portions and the cylindrical passages forming the delivery valve seats.

Each delivery valve guide **30** has a short cylindrical portion at its outer end which registers in the above mentioned cylindrical bore **27**, a long reduced portion **31** and a short frusto-conical portion **32** which registers with the outer end of the frusto-conical portion of the delivery valve bore. This portion has slots formed in it to provide passages for the oil.

Each delivery valve consists of a short frusto-conical portion **33** and a long cylindrical portion **34** of a smaller diameter, the said cylindrical portion being guided in a corresponding cylindrical hole in the delivery valve guide **30**. A compression spring is also housed in the delivery valve guide and presses on the end of the cylindrical portion of the delivery valve. The included angle of the flanks of the frusto-conical portion of the delivery valve is preferably slightly less than the included angle of the flanks of the frusto-conical portion of the delivery valve bore so that the delivery valve seats on the junction between the inner end of the frusto-conical portion **28** of the delivery valve bore and the cylindrical passage.

Surplus oil from the boost pump passes through a constricted orifice **35**, see Fig. 2, into the crank case to cool the lubricating oil and back again to the reservoir. This orifice **35** is such that, provided that there is an adequate supply of surplus oil to cool the crank case, there will be sufficient pressure to force the oil through the suction valves into the pump chambers and force the pump rams against the face of the wobbler. On the other hand if the gear pump is starved of oil, or if for any reason the gear pump fails to provide an adequate supply of oil, the flow of oil through the orifice into the crank case will be so reduced as to reduce the back pressure to such an extent that the oil will no longer be forced into the pump chambers and therefore the pump will cease to pump. This is a safeguard because it ensures that the bearing members in the crank case will not be loaded, (and they cannot be loaded if the pump is not pumping), unless there is an adequate surplus of oil to cool the crank case.

In order to prevent contamination of the lubricating oil with that carried in from the boost pump a thin annular metal ring **36** is interposed between the lubricating oil and the cooling oil so that the crank case **6** is divided into a central chamber **37**, containing the parts of the ram actuating device and the lubricating oil, and a surrounding annular chamber **38** containing the cooling oil. This thin annular dividing ring or screen consists for preference of perforated sheet metal supporting extremely fine gauze.

A further annular screen **39** of large mesh is provided inside the fine screen **36** to prevent the oil within the chamber **37** from swirling, which would interfere with the inward flow of oil to the radial holes **15** in the reaction plate **5**.

The oil within the central chamber **37** is replenished by that which leaks past the pump plungers, and although this leakage is minute, it is sufficient to renew the oil within the screen with reasonable frequency during the operation of the pump thereby maintaining its lubricating quality.

As previously mentioned in this specification the wobbler **3** is free to tilt in relation to the swash member **2**. Similarly the swash member is also permitted a small amount of tilt in relation to the reaction plate **5** and this is brought about by forming the main driving shaft so that the extension **11** is free to flex and supporting the main shaft **1** against the radial load by means of a single journal bearing **40**. The bearings in both cases are thus similar in effect to each other as the wobbler is also supported against the radial load by a comparatively short journal bearing constituted by the bush **19**, a loose fit betweeen the two giving to the wobbler the desired freedom to tilt.

The ram ends may be either spherical or conical but a quasi-conical form is preferred with the flanks of the quasi-conical surface curve to a large radius in such a manner that under full load the area of contact on the ram end spreads nearly to the two ends of the quasi-cone. In a typical case with a ram diameter of one half inch, a working pressure of 7,000 p. s. i. an angle of tilt of 12° of the swash member and the radius of the curved flank of the quasi-cone **6''** the departure of the quasi-cone from the true was of the order of 0.001".

The driving shaft **1** and swash member **2** rotates, whereas the pump body **7** is stationary, but the wobbler **3** is made to rotate very slowly in a reverse direction to the driving shaft by means of bevel gearing, one gear wheel **41** being attached to the pump body and the other wheel **42** being attached to the wobbler. The ratio of the bevel gearing is as the secant of the tilt angle of the swash-member, with the result that the relative reverse rotation of the wobbler ensures substantially pure rolling contact between the plunger ends and the surface of the wobbler, the plungers being free to rotate. This geared wobbler in combination with film-lubricated thrust-washers enables the pump to be operated continuously at very high pressures without appreciable wear but the invention is not limited to a pump having this gear.

The two delivery manifolds **43** and **44** are in communication with a combined unloading-relief valve not shown which is so arranged that, when a predetermined pressure is reached, the larger plungers are unloaded or, in other words, the delivery manifold **44** is by-passed to

inlet, and pressure is maintained in delivery manifold 43 by a non-return valve. Finally, when the maximum pressure is reached, the delivery of the small plungers along manifold 43 blows off at the relief valve.

The combined unloading-relief valve can be arranged to operate as a sustained pressure valve, in which case there is only a difference of 200 to 300 p. s. i. between the predetermined pressure at which the large plungers are unloaded or by-passed and the maximum pressure at which pressure is sustained with the delivery of the small plungers blowing off at the relief valve. Alternatively, the valve may be arranged to operate the pump as a two-stage high pressure pump, in which case there is a difference of 2000 or 3000 p. s. i. between the predetermined pressure at which the large plungers are unloaded or by-passed and the maximum pressure at which the small plungers blow off at the relief valve. Furthermore, the booster pump 10 may be increased in capacity, thereby providing low pressure for the rapid approach stroke of a press ram, with the addition of sustained pressure or two-stage high pressure for the pressing operation, as described above.

The unloading-relief valve may be adjustable so that the maximum blow-off pressure may be varied at will to suit the pressing operation.

We claim:

1. An axial ram pump of the type described comprising, in combination, a body having a plurality of parallel pump cylinders annularly arranged at equal radii about a central axis, a plurality of pump plungers slidably reciprocable in said cylinders of the annular series, said cylinders being alternately relatively large and small in diameter, power operated means for reciprocating said plungers in progressive overlapping timed sequence each with an intake stroke in one direction and a pressure stroke in the reverse direction in each cycle, a fluid supply manifold, inlet valve means for admitting fluid from said manifold to each cylinder in the course of the intake stroke of the associated plunger, a pair of fluid delivery manifolds in said pump body extending generally transversely of each of said cylinders and spaced apart axially of the cylinders, and discharge valve means individual to each cylinder and operative in the course of the pressure stroke of the associated piston to deliver pressure fluid to a particular one of said delivery manifolds.

2. A pump according to claim 1 in which said power operated means comprises a device having a working face relatively inclined to a plane normal to said axis and located in opposed separable operative contact with the outer ends of said plungers, power drive means for effecting a relative gyratory rocking motion between said face and said body, and means including a driving member on the pump body cooperating with a driven member carried by said device for effecting a relative rotational displacement between said face and said body.

3. A pump according to claim 1 in which the discharge valve means for the small cylinders comprises individual valves located in one common plane transverse to the axis, and for the large cylinders comprises individual valves located in another common plane transverse to said axis and located at a greater distance along said axis than said first mentioned common plane from said power operated means.

4. A pump according to claim 3 in which the small cylinders are the same in number as the large cylinders, and the respective individual discharge valves have valve bores extending transversely of the axis, the valve bores for one-half of the small and large cylinders extending in closely-spaced parallel laterally-staggered relation to one side of the pump body, and the valve bores for the other one-half of said cylinders extending similarly to the opposite side of the pump body.

5. An axial ram pump of the type described comprising, in combination, a body having a plurality of parallel pump cylinders annularly arranged about a central axis, a plurality of pump plungers slidably reciprocable in said cylinders, power operated means for reciprocating said plungers in progressive overlapping timed sequence each with an intake stroke in one direction and a pressure stroke in the reverse direction in each cycle, a fluid supply manifold, inlet valve means to admit fluid from said manifold to each cylinder in the course of the intake stroke of the associated plunger, and discharge valve means to permit delivery of fluid from each cylinder in the course of the pressure stroke of the associated piston, said power operated means comprising a power drive shaft aligned with said axis, an annular stationary abutment member encircling said shaft and presenting a first bearing face, a swash plate rotatable with said shaft and having on the side remote from said member an active face inclined to a plane normal to said axis, and a flat annular washer interposed in operative bearing assembly between said member and swash plate and secured for rotation with said swash plate and having a second bearing face in bearing engagement with said first bearing face, a wobbler plate coacting with said inclined face and in bearing engagement with the outer ends of said plungers to effect reciprocation of said plungers upon rotation of said shaft.

6. An axial ram pump of the type described comprising, in combination, a body having a plurality of parallel pump cylinders annularly arranged about a central axis, a plurality of pump plungers slidably reciprocable in said cylinders, power operated means for reciprocating said plungers in progressive overlapping timed sequence each with an intake stroke in one direction and a pressure stroke in the reverse direction in each cycle, a fluid supply manifold, inlet valve means to admit fluid from said manifold to each cylinder in the course of the intake stroke of the associated plunger, and discharge valve means to permit delivery of fluid from each cylinder in the course of the pressure stroke of the associated piston, said power operated means comprising a power drive shaft aligned with said axis, an annular stationary abutment member encircling said shaft and presenting a first bearing face, a swash plate rotatable with said shaft and having on the side remote from said member an active face inclined to a plane normal to said axis, and a flat annular washer interposed in operative bearing assembly between said member and swash plate and secured for rotation with said swash plate and having a second bearing face in bearing engagement with said first bearing face, an annular wobbler plate encircling said axis and having a third bearing face on one side in opposed relation to said inclined face, and having an active face on the other side in bearing engagement with the outer ends of said plungers to effect reciprocation of said plungers, a flat annular washer interposed in operative bearing assembly between said swash plate and wobbler and secured for rotation with said swash plate and having a fourth bearing face in bearing engagement with said third bearing face, and means for positively constraining said wobbler plate to effect relative rotation between said swash plate and wobbler plate upon rotation of the former by said shaft.

7. A pump according to claim 6 in which one of said first and second bearing faces is formed with a plurality of narrow generally radial grooves extending from the inner periphery to the outer periphery of said first mentioned washer, and having means for supplying lubricant to said grooves.

8. A pump according to claim 7 in which the sides of said grooves have sharp marginal edges in the plane of bearing face in which they are formed, thereby avoiding taper lead.

9. A pump according to claim 7 in which the bearing face with the grooves is formed of relatively soft metal and the other bearing face is formed of relatively hard metal.

10. A pump according to claim 6 in which one of said

first and second bearing faces and one of said third and fourth bearing faces are each formed with a plurality of generally radial grooves extending from the inner periphery to the outer periphery of the associated washer, and having means for supplying lubricant to said grooves.

11. An axial ram pump of the type described comprising, in combination, a body having a plurality of parallel pump cylinders annularly arranged about a central axis, a plurality of pump plungers slidably reciprocable in said cylinders, power operated means for reciprocating said plungers in progressive overlapping timed sequence each with an intake stroke in one direction and a pressure stroke in the reverse direction in each cycle, a fluid supply manifold in said pump body, passages in the pump body for supplying fluid to said manifold, inlet valve means to admit fluid from said manifold to each cylinder in the course of the intake stroke of the associated plunger, discharge valve means to deliver fluid from each cylinder in the course of the pressure stroke of the associated piston, said plungers being separable from said power operated means and being held operatively against said power operated means when the pump is operating by the fluid pressure of the fluid received in the cylinder from said manifold, said power operated means having cooperative relatively movable bearing surfaces and lubricant-conducting passages opening thereto, a lubricating oil containing chamber enclosing said power operated means, and means including passages in the pump body for directing fluid from said manifold around said chamber for cooling lubricating oil within said chamber.

12. A pump according to claim 11 in which the last mentioned means comprises a flow passage from said manifold to said chamber, and a restricted orifice in said flow passage effective to build up a back pressure in said manifold upon a predetermined rate of fluid flow therethrough, whereby an inadequate flow for cooling purposes through said orifice will result in a reduced pressure in said manifold inadequate to maintain said plungers in operative association with said power operated means.

References Cited in the file of this patent

UNITED STATES PATENTS

1,971,601	Dilg	Aug. 28, 1934
2,131,857	Lauret	Oct. 4, 1938
2,238,252	Davis	Apr. 15, 1941
2,270,263	Butler	July 20, 1942
2,385,990	Huber	Oct. 2, 1945
2,433,222	Huber	Dec. 23, 1947
2,454,563	Mercier	Nov. 23, 1948
2,461,279	Huber	Feb. 8, 1949
2,495,685	Beaman et al.	Jan. 31, 1950
2,520,632	Greenhut	Aug. 29, 1950

FOREIGN PATENTS

165,539	Switzerland	1934
588,614	Great Britain	1947
912,421	France	1946