HIGH SPEED HYDRODYNAMIC PUMP

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Filed Feb. 11, 1957, Ser. No. 639,297.
6 Claims. (Cl. 103—101)

This invention relates to rotary pumps, and particularly to high pressure pumps capable of operation at speeds of the order of 25,000 and even 45,000 r.p.m., against head pressures of the order of 3,000 p.s.i. and even 4,500 p.s.i. in special cases.

Pumps capable of such performance are desirable for use in high performance aircraft and in guided missiles. Such pumps must have a reasonable life, field of use considered, and should have an efficiency of 50% or better. They would ordinarily be driven by direct-connected turbines, and should afford a high power-to-weight ratio.

To meet these severe conditions, the invention proposes a pump of the so-called "scoop" type whose basic components are known. A dynamically balanced liquid filled cylinder is rotated on a fixed axis which carries a Pitot tube or pairs of Pitot tubes, through which discharge flow occurs under the sum of two heads, namely a pressure head developed by centrifugal force, and a pressure head developed by conversion of the velocity head of the rotating liquid.

To secure high delivery pressure and close approach to perfect dynamic balance, staged operation is adopted. The simplest form has two juxtaposed coaxial rotary chambers, with one Pitot tube in each chamber, spaced 180° apart around the fixed axis. The flow is in series through the two units. The reason for using one Pitot tube per chamber is to minimize what might be called "wake" effects, eddies and the like.

Pumps at least approximating the above basic form have been proposed, but in the crude form in which they are illustrated in such publications as are known to applicants, are not suited to the exacting duty here contemplated. Too many factors of controlling importance are ignored. Apparently their controlling importance in high speed operation was not appreciated.

One such factor is an inherent tendency for destructive accumulations of leakage liquid to occur. The present invention provides means to dissipate such accumulations at incipience and continuously.

Another such factor is the accumulation of air in the first unit. The possibility of air accumulations was known and manually controlled venting valves were suggested, but no one seems to have appreciated that the tendency to accumulate air is continuous. The invention provides means continuously effective to purge this air, and what is more difficult, to do so continuously in a closed circuit system.

Related to the air-purging concept is means to derive the energy needed for such purging from the leakage liquid, so that energy otherwise wasted is usefully applied.

Another factor not heretofore taken into account is the fact that at the speeds and pressures above suggested, the pump must operate at high temperatures, say 500° F. At such temperatures available liquids have low bulk modulus, low viscosity and increased chemical activity. This requires careful design of bearings to avoid destructive effects.

At such temperatures, the mechanical strength of materials and particularly their fatigue strength is impaired. Hence dynamic balance, freedom from vibration and free-

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stage passage 27 which leads from the Pitot unit (to be described) in the first stage chamber 26 to second stage pumping chamber 28 and discharge passage 29 which leads from the Pitot unit (hereinafter described) in chamber 28 to discharge connection 31. These three groove passages are covered and isolated from one another by an appropriately ported sleeve 32 which seats at its left-hand end against shoulder 33, is mounted with a shrink fit and is brazed to the shaft 19. The result is an approximately unitary ported shaft.

The Pitot unit 34 in first stage chamber 26 and the Pitot unit 35 in second stage chamber 28 are identical and are constructed each in one piece as shown in FIG. 5. Each has a sleeve-like hub 36, a streamlined radially ported strut 37 and a streamlined head 38 with entrance port 39. The struts have a fineness ratio of 4 and the heads have a fineness ratio of 5. The heads have a form derived from a symmetrical airfoil by imparting a curvature appropriate to the curved path of the liquid as it flows past the head.

The entrance port to each Pitot unit is shown as circular. This is preferred because it affords the maximum inlet area for a given perimeter and wetted area and affords the minimum "form drag." However, a circle is simply a special case of the ellipse having equal major and minor axes and, where space permits, it is feasible to use a true ellipse with aspect ratio radial or parallel to the axis of rotation, as circumstances may demand.

The hub sleeve 36 of the two Pitot units, an intervening spacer sleeve 41 and an end sleeve 42 make each a shrink fit with the sleeve 32. Furthermore, recourse is had to lugs 40 on each of which engages a notch in the adjacent sleeve to assure proper assembly and maintain tightness (see FIG. 4). As a precaution, a retaining ring 43 engages the end of sleeve 32.

The non-rotary components have now been described. It is convenient to state at this point, that it is practicable and advantageous under certain conditions, to replace each Pitot unit 34, 35 with a disc-like unit of the form shown in FIG. 6. In these the entrance 39a to the Pitot passage is formed in a projection 38a in the periphery of the disc 36a. The projection is streamlined as indicated by shading. The form shown in FIG. 5 is preferred, as a general rule.

The rotary components of the pump are enclosed in a cylindrical shell 45 which has an annular inward-extending head flange 46 terminating in a hub portion 47 bored to a shoulder to receive and position a bushing 48. This bushing has a hydrodynamic bearing portion 49, and beyond, a labyrinth portion 51, each of which portions encircles the bearing sleeve 42 above described as fixed on the shaft. This bearing structure, which will be discussed in further detail, sustains the inner end of the rotary assembly.

The rotary components are driven through a connector sleeve 52. This has internal splines 53 and an external flange 54 which seats against the end of hub 47. Machine screws 55 with thread-locking sleeves 56 (whose form is not here material) fix the flange 54 to the end of hub 47. The disc 57, which closes the inner end of connector 52, is brazed in place so as to be leak-proof, and flange 54 is sealed to bushing 48 and the bore of hub 47 by the O-ring 58.

Fitting into the bore of the cylindrical housing 45 and seated against the head flange 46 is a three-part assembly which encloses the two pump chambers 26 and 28. This assembly comprises a cup-shaped right-hand-end member 59, an intermediate member 61 and a left-hand-end plate 62. The member 59 encircles the bushing 48. The web of the member 61 which separates chambers 26 and 28 carries, fixed at its center, a ring internally formed as a labyrinth 63. The labyrinth surface encircles and acts with the member 41 carried by the sleeve 32, to provide a labyrinth seal between the two chambers 26 and 28. Thus the members 59, 61 and 62 define the two pumping chambers. To assure rotation of the liquid in these chambers as the shell 45 rotates, there are provided spiral ribs 64 which project from both plane boundary walls of each chamber 26, 28. These ribs are clearly shown in FIG. 2. The pitch of the spirals is such that rotation of the shell 45 tends to urge the liquid in the chambers 26 and 28 outward.

Seated against the peripheral portion of the left-hand face of the member 62 is the bearing support 65 whose form is clearly shown in FIG. 1. This is sealed to the shell 45 by an O-ring 66. A ring 67 is threaded to engage internal threads 68 formed in the left-hand end of the shell 45, and acts against the bearing support 65 to force it toward the flange 46 and the members 59, 61 and 62. The periphery of the intermediate member 61 is sealed by an O-ring 69.

In describing the sleeve 32, it was stated to be suitably ported. As will be appreciated from an examination of FIG. 1, this sleeve covers a longitudinal groove 24 which connects inlet 25 through a port in sleeve 32 with chamber 26. In sleeve 32 there are also ports connecting passage 27 at one end with the passage in the strut 37 of Pitot member 34 and at the other end with chamber 28. As indicated in dotted lines in FIG. 1, there is also communication between groove passage 29 in the shaft and the passage in the strut 37 of Pitot unit 35.

Ends 70 and 71 of the outer race 72 have a lapped planar surface 71 which contacts the lapped annular bead 72 of a running seal structure 73. This is of commercial form and hence is merely indicated in diagram. The seal 73 is designed to inhibit leakage outward from chamber 26. The bearing support 65 has an internal cone-shaped surface 74 which flares toward the left. When the rotary parts are in motion, the lapped annular bead engaging past seal 73 will be urged centrifugally along the surface 74 and out through ports 75 and 76 into the space in housing 11 between the housing and the rotary shell 45. Such leakage escapes through the drain 76. An insert 70 resists entrance into the space between the parts 11 and 45.

The seal unit 73 is sustained at its left-hand end by a flanged collar 77 which engages a shoulder 78 on the sleeve 32 (see FIG. 1) so that its position is fixed. The hub of this collar 77 serves as the support for the inner race 79 of a ball bearing whose outer race 81 is positioned by the bearing support 65.

The outer race 81 is mounted between two grease baffles 82 and 83 which are annular in form. The outer margin of the baffle 82 seats against an annular shoulder on the bearing support 65 and the left-hand baffle 83 is confined between the outer race and a threaded ring 84 which engages threads at the outer end of the bearing support 65. The baffles 82 and 83 are designed to retain in the bearing a definite amount of grease and to exclude from the bearing otherwise possible accesses of leakage oil. At the speeds contemplated, the entrance of excessive oil would be destructive to the bearing parts. The bearing, whose races appear at 79 and 81, is a combined radial and thrust ball-bearing which resists motion of the rotary components of the pump to the right and also defines the axis of rotation of these components. Motor means are incorporated in the pump to develop the axial thrust so resisting and these means comprise a port 80 which leads from the inlet passage 24 to the end of the shaft 19 so that inlet pressure, which is above atmospheric as already stated, is admitted to the interval between closure disk 57 and the end of the shaft 19. The effect is transferred to the bearing ball 60 to assure that they will never skid. The operation of the pump does not develop unbalanced axial thrust on the rotary element, so that the axial bearing is simply a function of the inlet pressure which is reasonably constant.

The member indicated at 85 is a sheet metal nut lock
designed to prevent the ring 84 from backing off. Similarly, the member 86 is a wire ring which functions as a nut lock and serves to prevent the member 67 from backing out. No novelty is here claimed for either nut lock, and various commercial units could be substituted. Hence, detail description is unnecessary.

In pumps of this type, accumulations of air in the pumping chambers, such as 26, must be prevented. Practically all oils or other hydraulic liquids carry gases either occluded or dissolved therein. Under the centrifugal forces developed in the pump, these gases separate out. In a two-stage pump such as illustrated, such separation occurs in the first stage chamber 26. One of the important features of the invention is provision of means to remove such gases continuously.

FIG. 7 shows a construction in which the gas is drawn off by an ejector whose motive fluid is liquid drawn from the interstage passage 27. This liquid is at the discharge pressure of the first stage. As shown in FIG. 7 and also in FIG. 2, a cross-drilled nozzle unit 91 is mounted in a bore 92 in the shaft member 19 and receives liquid through the port 93 leading from the interstage passage 27. The nozzle 91 delivers oil into the throat of a venturi 94 also formed in an insert. The nozzle 91 and the venturi member 94 are clamped by a threaded plug 95. The venturi throat discharges into a passage 96 which leads to connection 97 (see FIG. 3). The connection 97 would be connected to discharge into the oil reservoir of the system.

The nozzle 91 discharges into the venturi throat continuously and whenever air is present in the chamber 26, it will be drawn out through passage 90, but the ejector is virtually ineffective to withdraw oil from the chamber 26. Thus, air is removed automatically by an ejector mechanism having no moving parts and ineffective to remove any considerable quantity of oil from the pump.

The construction shown in FIG. 8 is differently proportioned from that shown in FIG. 7, but has the same functional elements. The reference numerals applied in FIG. 8 are the same as those applied in FIG. 7 but with the distinguishing letter c.

The distinguishing characteristic of the structure shown in FIG. 8 is the fact that the liquid-supplying port 93c leads, not from interstage port 27c, but from a point in the labyrinth 63c. In this way, leakage oil is tapped off and usefully applied.

The leakage oil in FIG. 8 is under the pressure developed in chamber 26, so that functionally, the two schemes are basically similar, even as to the selective removal of air as contradistinguished from oil.

One of the conspicuous features of the structure above described is the fact that the running seals (except for the seal 73) are of the labyrinth type. Static seals are of the familiar O-ring type. The labyrinth 63c prevents liquid from chamber 27c which is at high pressure, from flowing into chamber 26. The labyrinth seal 51 develops back pressure on the hydrodynamic bearing 49. As this is considered a very important detail, it is desirable to reinforce the labyrinth seal 51, and reinforcement is had by two simple labyrinth seals which have been worked in between the sleeve 52 and the driving member 52. The first of these appears at 97 where the end of the sleeve 32 enters a shallow groove turned on the inner face of the sleeve 32. The second occurs at 98 where a flange on the left-hand end of the driving member 52 enters the shallow annular groove. The clearance here, as well as at 97, is small enough to give a reasonable labyrinth seal effect.

The fact that the air ejector acts continuously and ejects air whenever air is present is a matter of extreme importance, because in any ordinary case, air will be discharged from the oil almost continuously. This circumstance is of even greater importance where the pump is used in a closed circuit. In such a case, positive ejection by the jet and venturi throat is the thing that makes the scheme practicable.

The scheme of forming all the flow passages in the shaft leads to compact and light-weight construction, and the use of the sleeve 32 gives an inexpensive construction with all the advantages of unitary construction. Incidentally, it affords some choice in the selection of material.

The tendency of any pump, operating under the conditions suggested, to run hot is a source of considerable difficulty. Advantage has been taken of the oil circulation and the jet flow used primarily to remove air, to afford effective means for dissipating at least some of the generated heat. More will be known on this head when the experimental work has progressed further. At the temperatures contemplated, it is necessary to afford some compensation in the bearings. This is had by the use of metals characterized by different coefficients of thermal expansion. Fortunately, the pump is characterized by very moderate bearing loads and serious difficulty with the bearings is not anticipated.

Streamlining of the Pitot strut and the Pitot head is a matter of considerable importance. The entrance rim at 39 (FIGS. 2 and 5) must be rounded and the trailing end which overhangs the strut is important as means for reducing wake effects and minimizing the degeneration of mechanical energy into heat.

Applicant recognizes that staged operation is important and that more than two stages can be devised. Two stages are a plurality, and will support the use of that term in the claims. It would obviously be simple to elaborate the structure by adding pumping chambers each with its own Pitot strut and head. Applicant believes that it is unwise to use more than one Pitot unit per chamber, and the present belief is that satisfactory balance can be had if these units are angularly spaced at uniform intervals.

Applicant has endeavored to describe in careful detail a satisfactory embodiment of the invention but by doing so, he does not imply that the illustrated structure is the only one which might be used within the scope of his invention. That scope is defined solely by the claims.

What is claimed is:
1. In a "scoop" pump, the combination of a fixed shaft provided with a plurality of flow ports, different ones of which serve as a supply port and a discharge port, and one serves as a loading port; a rotor enclosing a generally cylindrical pumping chamber subdivided into equal compartments, said compartments being alternately in communication with said supply port and said discharge port, and said compartments being thereby effectively filled with oil from said supply port and discharged in turn into said discharge port.

2. The combination defined in claim 1 in which the inlet port is capable of being operated at a superatmospheric pressure and serves as the loading port, whereby the pressure motor is energized by pressure fluid from said inlet port.

3. In a multistage "scoop" pump, the combination of a fixed shaft provided with a plurality of flow ports, different ones of which serve as a supply port, an interstage port, a high pressure discharge port, and a low pressure drain connection; a rotor enclosing a plurality of generally cylindrical coaxial pumping chambers which encircle said shaft and turn about their common axis as the rotor turns; liquid impelling vanes in said chambers; a plurality of scoop units, one enclosed by each of said chambers, each scoop unit comprising a ported strut and a Pitot orifice carried by said strut and communicating through the shaft and serving to develop an axial thrust resisted by said bearing means; and a connection for rotating said rotor.
with the port in its strut, said scoop units being individually mounted in successive corresponding pumping chambers and angularly spaced uniformly about the shaft axis, the supply port, the interstage port, and the discharge port in said shaft serving to complete a continuous path from supply to the interior of the first chamber, and thence by way of its enclosed scoop unit and an interstage port to the next pumping chamber, and so on, the last scoop unit delivering to the discharge port in said fixed shaft; an ejector comprising a combining tube and a jet nozzle, said combining tube opening into and connected to draw gaseous medium from the first pumping chamber at a point adjacent the surface of said shaft; conduit means in said shaft to receive liquid under pressure from a point in said continuous path between the first scoop unit and the end of the discharge port and deliver said liquid to said nozzle, said ejector delivering mixed fluid and gas to said low pressure drain connection; and a mechanical connection for rotating said rotor.

4. A pump as defined in claim 3 in which said point in the continuous path from which the nozzle receives liquid is in the interstage port between the first stage and the second stage.

5. The pump as defined in claim 1 and a hydrodynamic bearing at one end of said shaft by which said rotor is mounted on said shaft; means to admit liquid under pressure to one end of said bearing from the pumping chamber; and sealing means inhibiting the escape of liquid from the other end thereof.

6. The pump as defined in claim 5 and an anti-friction bearing between said rotor and shaft at the end opposite said hydrodynamic bearing; shields, mounted on the rotor in juxtaposition to opposite sides of said anti-friction bearing, confining an initial lubricant charge in said anti-friction bearing and serving as slingers to resist the entry of leakage liquid into said anti-friction bearing; and centrifugal impeller means on said rotor arranged to receive leakage liquid and direct it away from said anti-friction bearing.

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