

March 23, 1948.

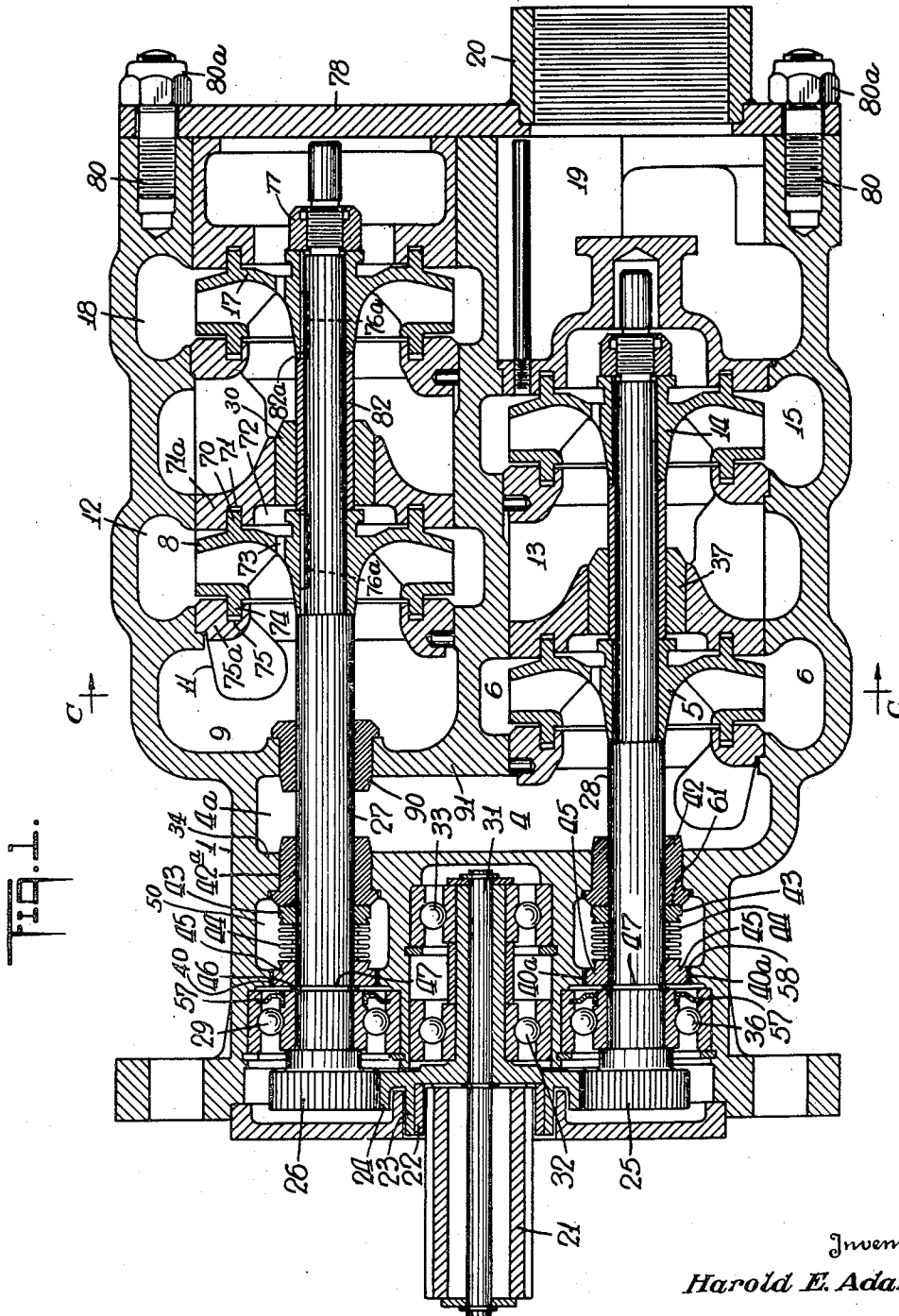
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2,438,104

PUMP

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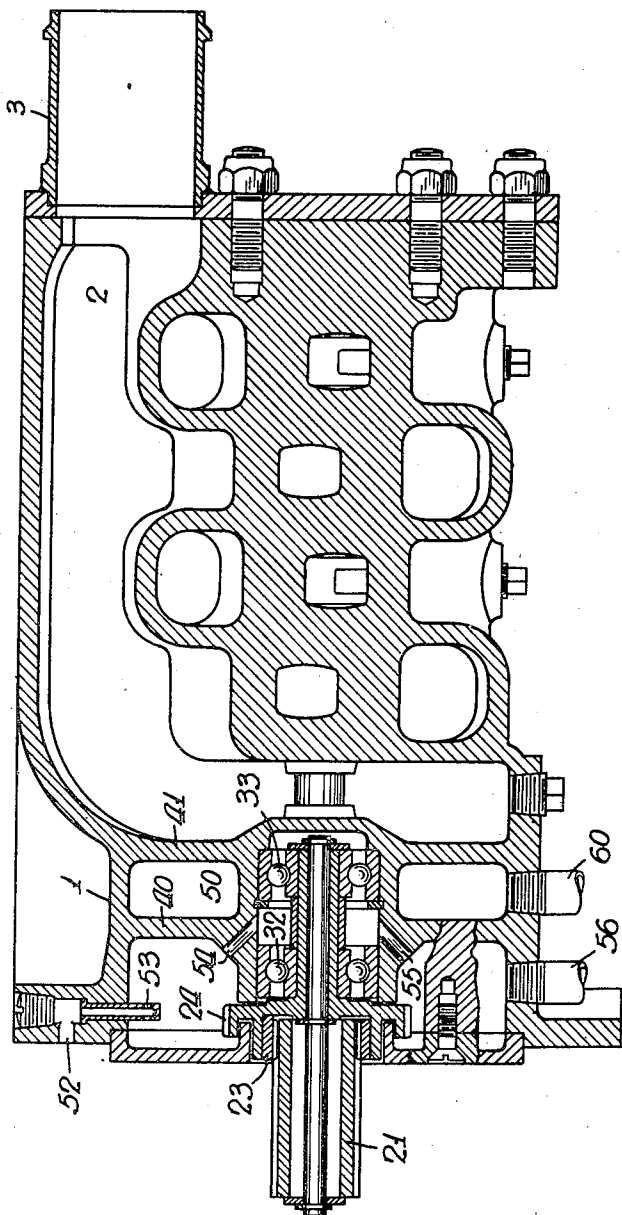
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Fig. 2.



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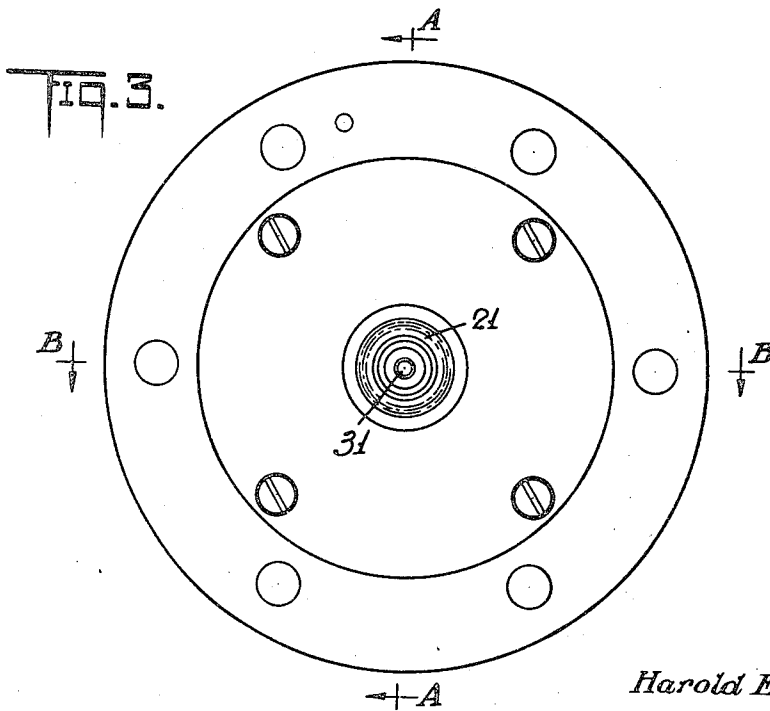
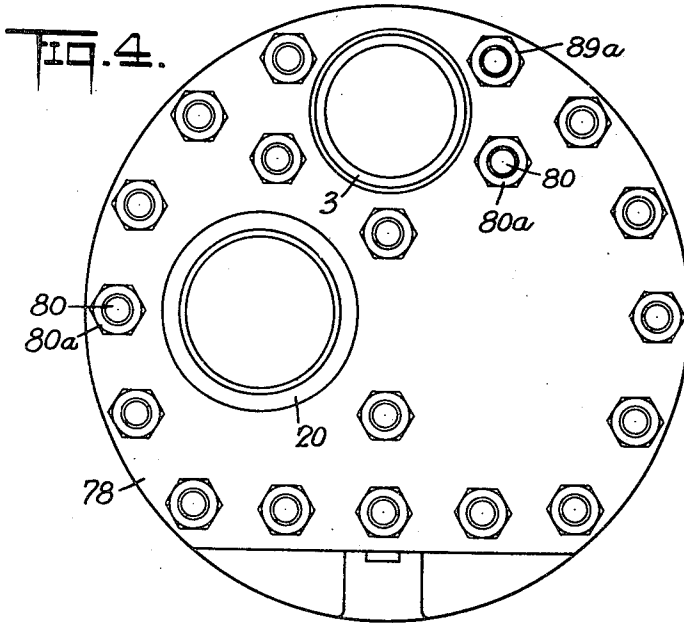
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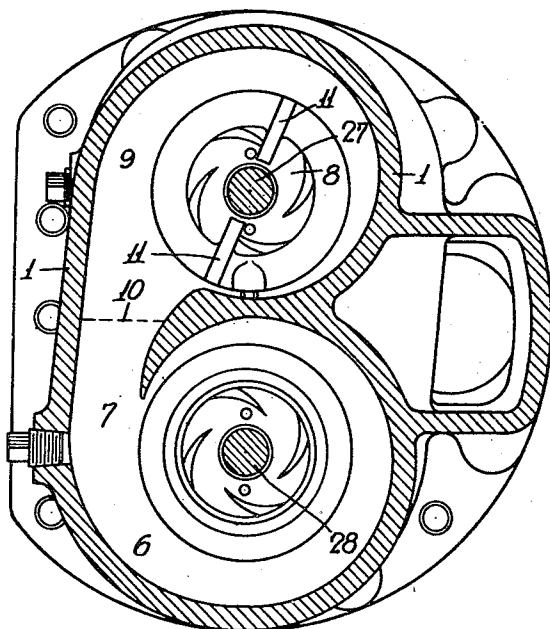
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Fig. 5.



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FIG. 6.

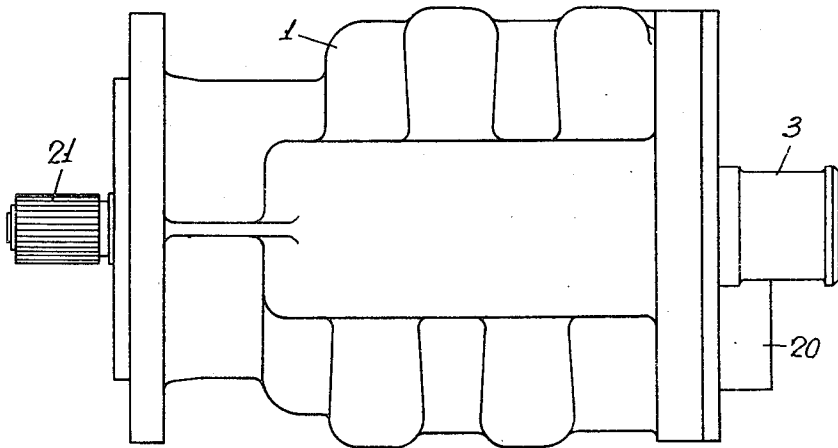
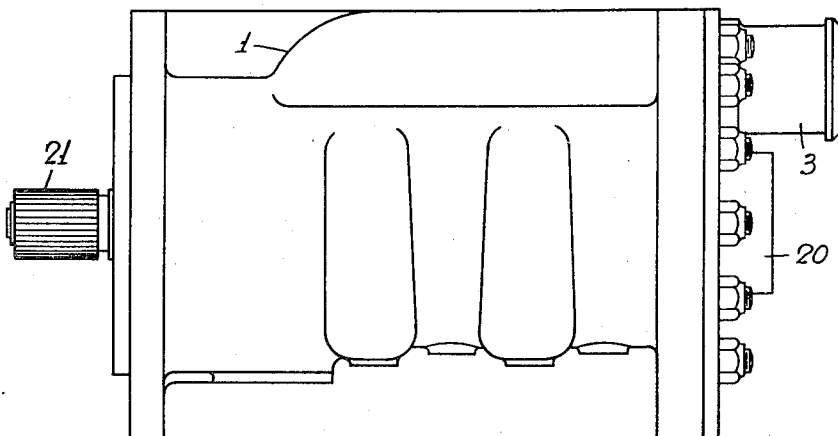


FIG. 7.



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## UNITED STATES PATENT OFFICE

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## PUMP

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The present invention relates to improvements in multi-stage centrifugal liquid pumps.

It is an object of the invention to provide a form of construction which will have the advantage over present-day conventional constructions of improved efficiency, lightness of weight, and compactness of structure.

The conventional form of multi-stage centrifugal pump comprises a single shaft on which the required number of impellers are mounted, and a suitable containing structure to convey liquid from the pump inlet through the several stages, thence to the discharge connection. The number of stages, the size of the impellers and the like obviously depends on the required pressure, capacity and rotative speed available for the pump. Such a structure involves many undesirable features, particularly so when many stages are needed. The shaft is very long and the distance between supporting bearings is often quite great. This in turn necessitates a large diameter shaft in order that its deflection will not be such as to require excessive clearance at the sealing ring, as such extra clearance causes increased hydraulic leakage with attendant lowered efficiency. Furthermore, the large diameter shaft which is required because of the length between bearings and because of the power to be transmitted results in impeller diameters that are larger than would be necessary in a single stage pump operating at the same pressure per stage and at the same speed.

The containing structure on all of these typical present-day multi-stage pumps has another serious and inherent fault, namely, that the liquid discharged from the lower pressure stage has to pass completely over the outer diameter of the volute of that stage, and thence take a sharply reversed inward path in order to arrive at the suction of the next impeller. Use of diffusing or directional vanes on the periphery or the inlet of the impellers of such pumps is well known, but does not overcome the basic disadvantage in this type of pump, that is, the reversal of flow back down toward the inlet of the next succeeding stage.

The present invention is specifically directed to eliminate the aforementioned disadvantages, and it results in a smaller, lighter and more efficient unit, particularly adapted for aircraft application. The use of two or more parallel shafts within the containing structure, as indicated, achieves a short, compact construction having small diameter shafts which, in turn, result in smaller diameter impellers for a given duty, and

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also allow for closer clearances between stages, with resultant lowered hydraulic leakage and corresponding increased efficiency. The shafts are smaller in diameter for two basic reasons, (1) because of using multiple shafts the power is divided between these shafts and, therefore, less per shaft, (2) by the use of step up gear ratios, I am not limited to the available rotational speed on the drive of the pump, as is the case with conventional pumps. I may make a wide selection of any desired speed on the multiple impeller shafts by proper choice of gear ratios. Hence, I may operate my impeller at a considerably higher speed than that available for the drive, this higher speed resulting in less torque on the shaft, and hence a smaller shaft for each of the multiple shafts.

The present application discloses a form of pump which eliminates the beforementioned stage-to-stage loss due to the tortuous passages in the conventional unit. By the structure of the present invention I am able to deliver liquid discharged from a lower pressure stage to the next pressure stage along a clean direct path with no reversal of flow, and without the use of guiding vanes to the inlet of the next higher stage. In other words, the discharge of the lower stage from the volute of that lower stage goes directly over to the adjacent impeller on an adjacent shaft directly feeding and directing this flow into the inlet of the other impeller. This construction permits me to take full advantage of gain in pressure head by proper proportioning of the passages to recover kinetic velocity as it is delivered at the inlet to the next stage impeller at the required velocity for that inlet. This is much easier and more efficiently accomplished in my construction than can be done when reversal of flow, diffusing vanes and other such devices have to be employed in order to get the liquid back into the inlet of the next stage.

Another feature of the present invention is as follows. While it is true that a gear drive can be used with a conventional multi-stage pump to step up the speed required for a given head condition, the fact remains that on such drive the thrust due to the torque on the drive gear is unbalanced. By dividing the load amongst multiple shafts, as is done on my construction, the thrust on the individual gear for each shaft is reduced because of the divided load and, even more important, the thrust on the main drive gear is balanced by the positioning of the thrusts from the multiple pump shaft taking off from the main drive gear. This further contributes to

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the possible use of higher speed and results in higher efficiency, longer life, and the use of smaller, lighter more compact construction for airborne service.

Other features and advantages of my invention will appear from the following description of a specific form of embodiment thereof having reference to the accompanying drawings, in which

Fig. 1 is a view showing a longitudinal section of said embodiment of my invention, said section being taken along the lines B—B of Figure 3, looking in the direction of the arrows;

Fig. 2 is a longitudinal section taken along the lines A—A of Figure 3, looking in the direction of the arrows;

Fig. 3 is an end elevation of the left end of Figures 1 and 2;

Fig. 4 is an end elevation of the right end of Figures 1 and 2;

Fig. 5 is a section on lines C—C of Figure 1; and

Fig. 6 is a top plan view of said embodiment; and

Fig. 7 is a side elevation thereof.

Similar reference characters refer to similar parts in each of the several views.

Casing 1, preferably of aluminum, includes an inlet passage 2, connecting with suction nipple 3. Passage 2 communicates with the inlet of the first stage impeller 5 through the lateral passageway 4. The first stage impeller 5 discharges the liquid into its volute chamber 6. From the discharge point 7 of this volute chamber the liquid is directed to the inlet of the second stage impeller 8 through the means of the passageway 9.

The division point between the outlet passage 7 of the first stage and the inlet passage 9 of the second stage is indicated approximately by the dotted line marked 10 in Fig. 5.

At the inlet of the second stage impeller 8 I place vanes 11, for the purpose of preventing rotational flow as the liquid enters the inlet of the second stage impeller. The liquid then passes through the second stage impeller 8 to its volute 12. Here the liquid is directed in a like manner into the third stage impeller 14, from which it enters discharge volute 15. From this volute it passes through passage 16 into the impeller 17 of the fourth stage, and from there it goes into the volute 18 and into the discharge chamber 19, and then out through the discharge connection 20. It will be understood, as stated above, that the use of four stages as herein described is purely illustrative and that any desired number of stages may be employed within the purview of the present invention.

The drive to the impellers is from a splined flexible coupling 21 engaging a corresponding female spline of the driving means not shown. Member 21 engages on its opposite end corresponding teeth 22 in central driving gear 23. While I have illustrated a flexible coupling drive of this conventional nature, any well known solid or flexible coupling may be employed. The central driving pinion 23 is mounted on an extension of its shaft 31. This shaft is carried in two ball bearings 32 and 33, mounted in the part 34 of the casing or housing 1. The driving pinion 23 is furnished with gear teeth 24 which engage gear teeth of corresponding nature on the impeller shafts 27 and 28, the corresponding gears on these shafts being designated as 26 and 25, respectively. Shaft 27 is mounted on bearings 29 and 30. Bearing 29 is of the ball type, so as to take up any unequal thrusts and loads on the shaft, which might develop in the operation of

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the pump. Bearing 30 is made of carbon because of its satisfactory qualities for operation in gasoline, this being the particular fluid for which the present embodiment of pump is intended. The lower shaft 28 is mounted in corresponding bearings 36 and 37, respectively.

By the above described arrangement it will be noted that the torque loads transmitted by the central drive pinion 23 are essentially equal and diametrically opposed, thus neutralizing any thrust loads on the shaft 31 and bearings 32 and 33 of the central drive pinion. This is a very desirable feature for long continued operations, inasmuch as the entire power absorbed by the structure must be transmitted through this bearing assembly.

While I have shown only two impeller shafts in the illustrated form of embodiment, it will be understood that any desired number of such multiple shafts may be employed.

It will be noted that the drive pinion is of larger diameter than the driven gears 25 and 26 mounted on the impeller shafts 28 and 27, respectively, in this case the ratio of diameters being roughly 2 to 1. By this means the rotational speed of the impeller shafts 27 and 28 may be roughly twice the rotational speed of the driver, thus resulting in the compactness of structure and lightness of weight and other advantages heretofore outlined.

While I have shown a ratio of approximately 2 to 1 from the driver to the driven pump shaft, any other ratio desired for the given result may be employed by simply changing the pitch diameters of the gear assembly.

It should be noted that the ball bearing 32 is mounted close to the pinion gear to take any torque thrust load on the shaft at that point.

The gears and ball bearings on the drive end of the pump are lubricated from a source of lubricating oil furnished to the gear casing at point 52, Fig. 2. This oil furnished under pressure is sprayed into the gear chamber 51 through a spray nozzle 53, thus forming an oil mist which is most suitable for the lubrication of gears operating at high speed. In the present application, for example, the drive gear 23 operates at approximately 7000 R. P. M. and the driven gears 25 and 26 operate at approximately 15,000 R. P. M. The oil mist lubrication also serves to lubricate the bearings 29 and 36, and provision is made for the adequate lubrication of the pinion drive bearings 32, 33 by means of the oil supply passage 54 and oil drain 55. The chamber 51 is provided with a drain 56 at a low point so as to drain off oil to keep any solid oil from reaching the level where it would submerge the gear teeth. This oil is drained off to a suitable sump and may or may not be reused in the lubricating system.

The gear and bearing assembly is separated from the rest of the pump structure by the wall or diaphragm 40. Where the rotating shaft assembly is associated with shafts 27 and 28 which pierce this wall, labyrinth seal means are provided, such as the sealing member 57 incorporated in the ball bearings 29 and 36, respectively, and the close clearance 58 between the periphery 45 and the opening 40a in the stationary wall 40.

Adjacent to the wall or diaphragm 40 is a seal drain chamber 50 formed by the outer periphery of the casing 1 and the walls 40 and 41. Wall 41 separates this chamber 50 from the suction side of the pump, and a sealing means 42 is provided in wall 41 to prevent leakage of the

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liquid being pumped (gasoline or kerosine in the illustrated case) from leaking into the intermediate chamber 50. Such leakage, however, as might occur into chamber 50 from the suction chamber 4 is drained off through drain connection 60. It will be understood, of course, that this drain also discharges any leakage of oil from the gear casing 51 through the seals 57 and 58 into this chamber 50. It will thus be seen that by the device of this intermediate chamber between the oil casing and the pumping chambers and by the provision of labyrinth sealing means I eliminate the possibility of admixture of the fluid being pumped and the oil used for lubrication.

The sealing means 42 to prevent leakage of fluid from the pump casing 4 into the intermediate chamber 50 must necessarily be more effective than the sealing means between the gear casing and this chamber because of the greater differential in pressure that is likely to exist between the pump intake and this chamber. In the present case the pressure at the inlet of the pump may be in the order of 10 to 25 pounds' gauge pressure, and the liquid is likely to be gasoline, so that very effective sealing means must be employed to keep flow along the shaft to a minimum. For this sealing means I preferably employ the following construction. The stationary member of this seal assembly is a carbon sleeve 42 which is pressed and cemented into the hole 42a around the shaft piercing the wall 41. This carbon sleeve is provided with a smooth surface 61 which engages the corresponding smooth and polished surface of the rotating member 43, preferably of hardened stainless steel. The rotating member 43 is attached in leakproof engagement to a flexible metallic bellows 44 and the bellows is likewise attached on its opposite end in leakproof engagement with the rotating seal holder member 45. This member 45 is preferably of stainless steel and is pressed on the shaft 27 so as to insure the rotation of member 45, bellows 44 and the member 43 with the shaft. The bellows assembly is positioned on the shaft by the split ring 46, which is located on the shaft by groove 47 in said shaft. The bellows 44 is of spring-like, metallic material such as brass or bronze with sufficient pressure to maintain the rotating steel member 43 in close engagement with the stationary seal face 61. The engagement of these two members at the face 61 forms a leakproof joint separating the liquid side of the pump from the chamber 50. In addition to the spring action of the bellows to maintain this seal at this point, there is an additional hydraulic effect causing increased pressure at these faces 61 with increased pressure on the pump side of the assembly. The hydraulic pressure built up in the pump is transmitted to the interior of the bellows by following along the shaft. The effective area of the interior of the bellows is slightly greater than the effective area of the seal face 61. There is, therefore, an overbalance between the static hydraulic pressure built up within the interior of the bellows and the exterior atmospheric pressure, effective to increase the pressure between the sealing members to maintain a tight joint at 61. In addition to this static hydraulic pressure obtained from the suction side of the pump, there is additional pressure built up within the interior of the bellows when the pump is in operation. This pressure is caused by the centrifugal action of the rotation of the liquid within the bellows which causes a build up of ad-

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ditional static pressure to still further maintain this seal joint tight under rotating conditions. The heat due to friction generated at surface 61 is dissipated by the continual flow of liquid being pumped past the high heat conducting member or sleeve 42.

The hydraulic liquid pumps employed in the present embodiment of my invention are essentially of standard construction common in this art but the following features thereof should be noted.

Each impeller 5, 8, 14, 17 is provided with annular projections 70 and 74 which rotate with the impeller in close clearance with corresponding annular grooves 71 and 75, respectively, located in the stationary seal ring members 71a and 75a. These seal ring members prevent leakage of the fluid being pumped from the high pressure side of the pump to the suction side. Such leakage as does come through on the closed end of the impeller escapes into the chamber 72; from this chamber the liquid is drawn back to the suction side through the hole 73 in the impeller. This releasing of the pressure back of the impeller as just indicated takes care of uneven hydraulic pressures that might be exerted on either side of the impeller which would cause a thrust on the shaft in an axial direction. The impellers are driven by their shafts by keys 76a. The impellers are furnished with collar projections 77 on their outboard end to facilitate withdrawal from the shaft. These collars provide for easy removal without danger of distorting the impeller when pulling off the shaft.

The general construction of the pump includes a continuous cast envelope 1 open only at the ends, this construction being used because of its inherent strength and pressure tightness as compared with a horizontal split casing requiring a larger diameter and increased weight because of the flange gaskets and bolts necessary for making up a tight joint at the junction of the two halves. In my construction, I only have one casing joint, which is at the end formed by the steel plate 78, which is held in leak-tight contact with the casing by means of the numerous studs 80 and nuts 80a. It should be noted that renewable hardened sleeves 82 are provided for rotation in the carbon bearing 30, each such sleeve being held against rotation relative to the shaft by a notch 82a in its end, in which notch is engaged the tip of driving key 76a.

In order to prevent the imposing of high differential pressures on the sealing means, such as 42-45, from shafts which serve impellers operating on stages above the first or inlet stage, I provide an equalizing chamber 4a which is an extension of the lateral inlet chamber 4. This equalizing chamber 4a surrounds the shaft at the entrance to the sealing means 42-45, thus imposing only the inlet pressure of the pump on this sealing means. The higher pressure of the intermediate stages is prevented from being transmitted to this chamber by the restricting bushing 90 surrounding shaft 27 with close clearance and placed in the wall or diaphragm 91 separating the inlet chamber 9 of the intermediate stage from the equalizing chamber 4a.

I have described what I believe to be the best embodiment of my invention. I do not wish, however, to be confined to the embodiments shown, but what I desire to cover by Letters Patent is set forth in the appended claims.

What I claim is:

1. A centrifugal pump comprising a plurality

of stages, each stage comprising an impeller and a volute therefor, a plurality of shafts, one of said impellers being on each of said shafts, means whereby the volute of one of said stages discharges directly into the inlet of the impeller of another stage, a casing enclosing said stages, and including an inlet chamber means, means constituting a chamber in said casing separate from the inlet chamber means through which chamber at least one of said shafts passes, a bearing for said shaft on the side of said separate chamber removed from the inlet chamber means, means for lubricating the bearing, sealing means in the separate chamber to impede leakage of fluid from said bearing into said chamber and from said inlet chamber means into said separate chamber, and means for drawing such leakage fluid as may pass said sealing means from the separate chamber.

2. A centrifugal pump comprising a plurality of stages, each stage comprising an impeller and a volute therefor, a plurality of shafts, one of said impellers being on each of said shafts, means whereby the volute of one of said stages discharges directly into the inlet of the impeller of another stage, a casing enclosing said stages and including an inlet chamber means, means constituting a separate chamber in said casing through which chamber at least one of said shafts passes, a bearing for said shaft on the side of said separate chamber, removed from the inlet chamber means, means for lubricating the bearing, sealing means to impede leakage of fluid from said bearing into said separate chamber and from said inlet chamber means into said separate chamber, said sealing means comprising a carbon sleeve surrounding said shaft and in sealing engagement with a wall of said chamber, said sleeve having a smooth bearing surface, a rotating member mounted on said shaft and having a like smooth surface for engaging said carbon sleeve, and means for mounting said rotating member on said shaft, comprising a bellows affixed to said shaft, and means for draining such leakage fluids as may pass said sealing means from the separate chamber.

3. A centrifugal pump comprising a plurality of stages, each stage comprising an impeller and a volute therefor, a plurality of shafts, a gear on each of said shafts, a plurality of said impellers being on each of said shafts, means whereby the volute of one stage discharges directly into the inlet of the impeller of another stage on another of said shafts, a casing enclosing said stages including an inlet chamber means, separate chamber means for the gears of the shafts, means constituting an intermediate chamber in said casing separate from said other chamber means through which at least one of said shafts passes, a bearing for said shaft on one side of said intermediate chamber and in said gear chamber means, means for distributing lubricating fluid over the gears and bearing in said gear housing, sealing means to impede leakage of fluid from said intermediate bearing into said chamber and from said inlet chamber means into said intermediate chamber, said sealing means comprising a carbon sleeve surrounding said shaft and in sealing engagement with a wall of said intermediate chamber, said sleeve having a smooth bearing surface,

a rotating member mounted on said shaft and having a like smooth surface for engaging said carbon sleeve, and means for mounting said rotating member on said shaft, including a bellows affixed to said shaft and means for draining from the intermediate chamber such leakage fluids as may pass said sealing means.

4. A centrifugal pump having a plurality of stages, each stage comprising an impeller and a volute therefor, a casing enclosing said stages and including an inlet chamber means, a plurality of shafts and at least one impeller on each shaft, means whereby the volute of one stage discharges directly into the inlet of the impeller of another stage on another of said shafts, gears on said shafts, separate chamber means for said gears, driving gear means entering said gear chamber means and operatively connected with said shaft gears, means constituting an intermediate chamber in said casing separate from the inlet and gear chamber means, a bearing for each of said shafts in said gear chamber means, means for distributing lubricating fluid over the gears and bearing in said gear chamber means, said shafts extending into all of said chamber means, sealing means to impede the leakage of fluid from said gear chamber means into said intermediate chamber and from said inlet chamber means into said intermediate chamber and extending within the intermediate chamber means, and means for draining from the intermediate chamber such leakage fluids as may pass said sealing means.

5. A centrifugal pump having a plurality of stages, each stage comprising an impeller and a volute therefor, a casing enclosing said stages and including an inlet chamber means, means whereby the volute of one stage discharges directly into the inlet of the impeller of another stage on another of said shafts, said casing having a part containing two separate chambers, a plurality of shafts extending in the casing and in the chambers of the casing part, a driving gear means in one of said casing part chambers, bearings in the gear chamber for supporting the driving gear means, gears on said shafts cooperating with said driving gear, a bearing on each of said shafts in said gear chamber, means for distributing lubricating fluid over the gears and the bearing within said gear chamber, sealing means to impede leakage of fluid from the gear chamber and the inlet chamber means toward the other of said chambers, and means for draining from said other chamber such leakage fluids as may pass said sealing means.

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