March 9, 1954 F. W. KRUEGER 2,671,404

TURBINE PUMP

Filed March 15, 1950 2 Sheets-Sheet 1

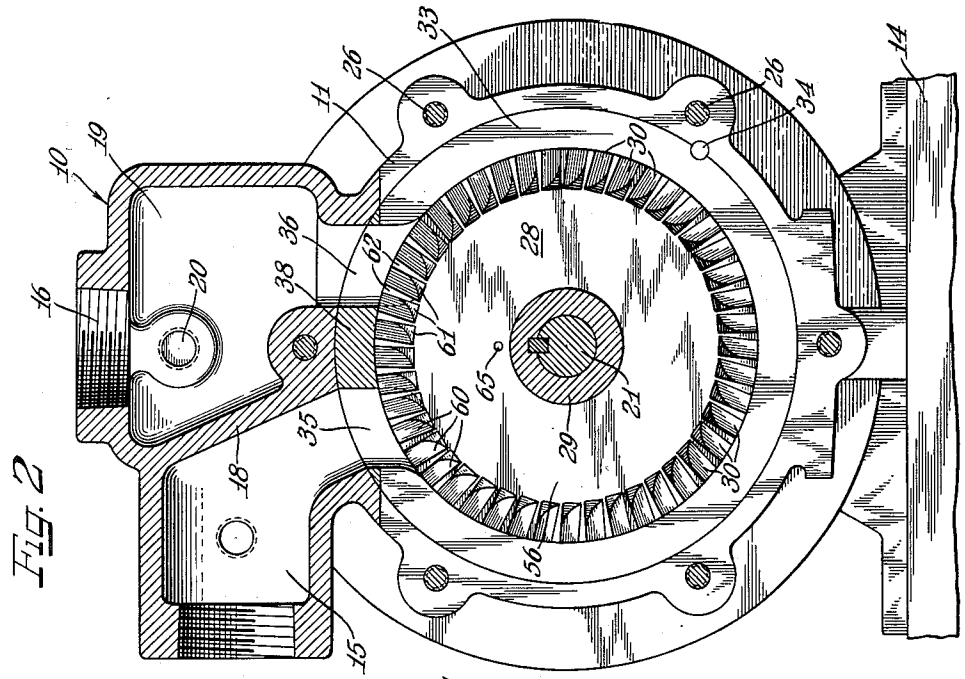


Fig. 2

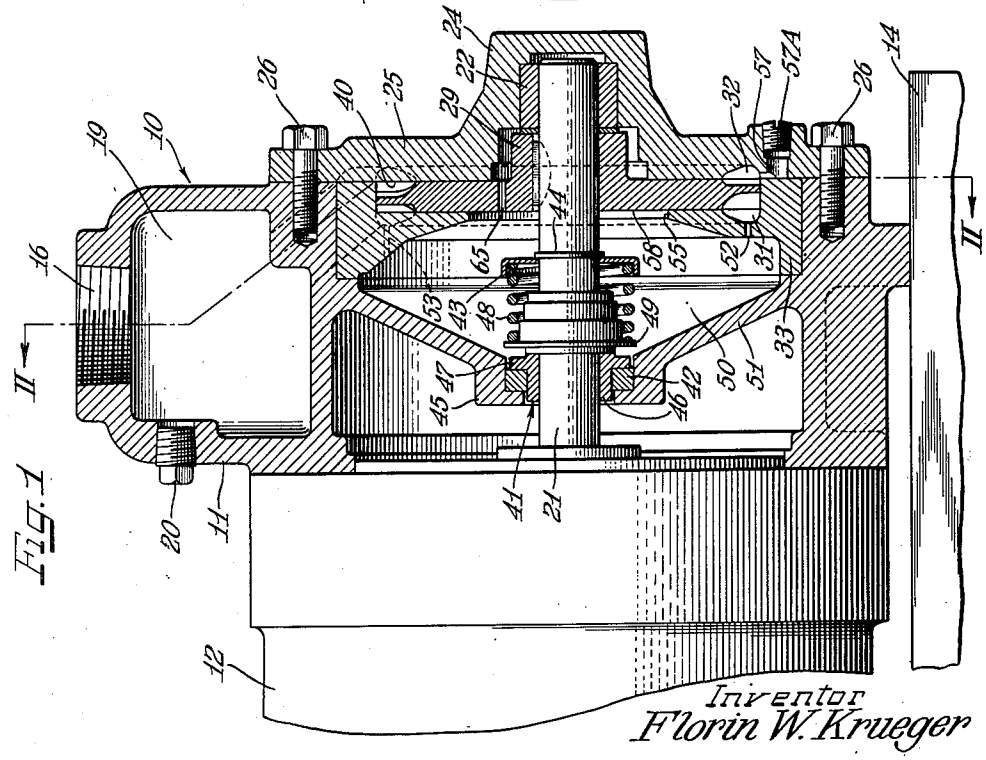


Fig. 1

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TURBINE PUMP

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Fig. 3

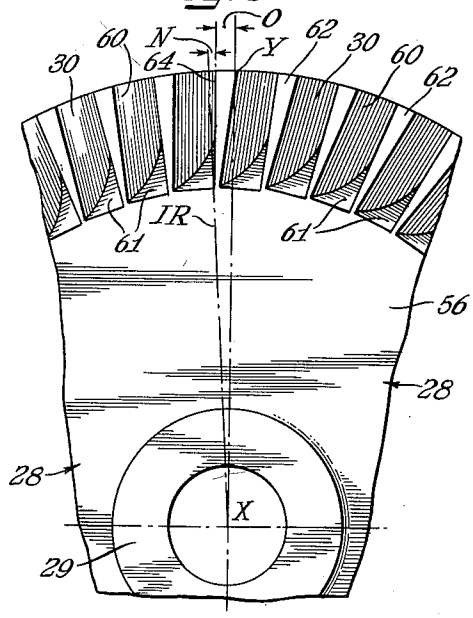



Fig. 4

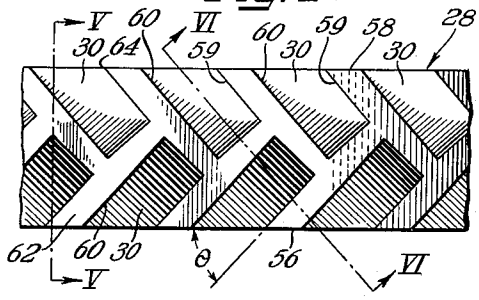



Fig. 5

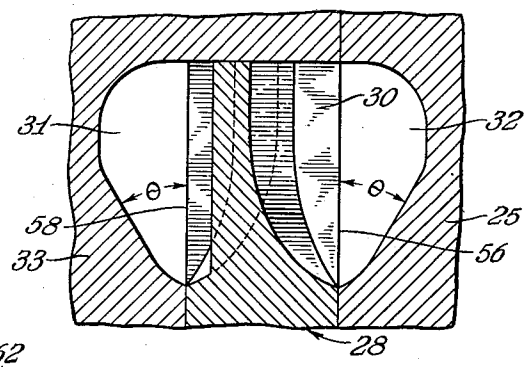



Fig. 6

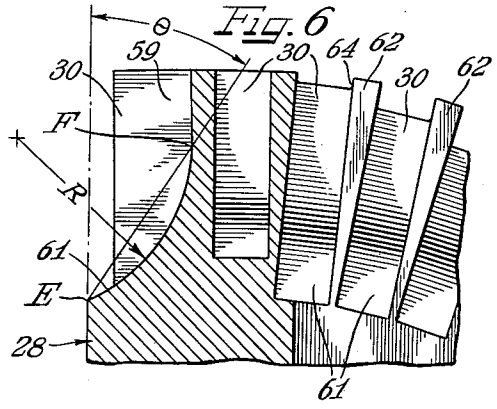



Fig. 7

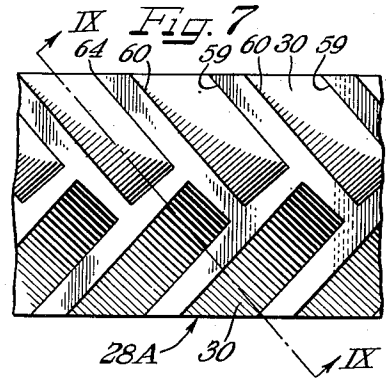



Fig. 8

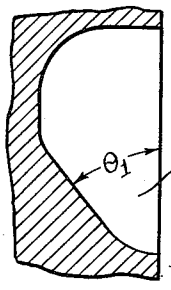
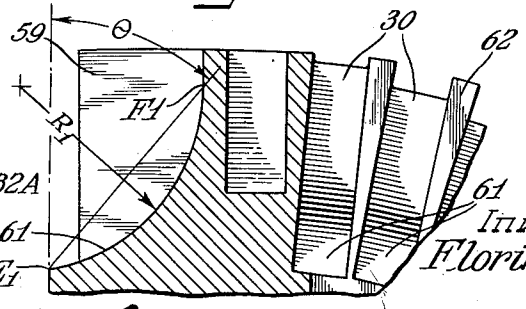



Fig. 9




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# UNITED STATES PATENT OFFICE

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## TURBINE PUMP

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Application March 15, 1950, Serial No. 149,701

8 Claims. (Cl. 103—96)

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This invention relates generally to improvements in pumps of the turbine type and has as its principal objects to provide such a pump having higher efficiency and better operating characteristics than prior turbine pumps.

A further object of my invention is to provide a new and improved form of turbine pump having a novel and efficient form of impeller and raceways, improving the flow characteristics and reducing eddy currents in the pump.

Another object of my invention is to reduce slippage and improve the efficiency of turbine pumps by confining the cross-sectional areas of the raceways to the displacement of the buckets of the pump, and the cross-sectional areas of the impelling teeth thereof.

Another object of my invention is to provide a pump of the character set forth having simply constructed and accurately fitting parts capable of being manufactured and assembled on a mass production basis.

A further object of my invention is to provide a turbine pump of the type described which can be readily disassembled for maintenance without disturbing its driving motor.

Another object of my invention is to provide a self-priming pump of the above character capable of idling without a flow of fluid therethrough for long periods of time with no damage to the parts thereof.

Still another object of my invention is to provide a silent turbine pump constructed in such a manner as to eliminate all whining or other disturbing noises.

These and other objects of my invention will become apparent as the following detailed description proceeds, and with reference to the accompanying drawings, wherein:

Figure 1 is a fragmentary vertical sectional view taken longitudinally through the axis of the drive shaft of an illustrative pump constructed in accordance with my invention;

Figure 2 is a broken transverse sectional view taken substantially along line II—II of Figure 1;

Figure 3 is an enlarged fragmentary end view of an impeller suitable for use in the pump of Figure 1;

Figure 4 is an enlarged fragmentary plan view of the pump impeller shown in Figure 3;

Figure 5 is an enlarged fragmentary transverse sectional view taken through the toothed portion of the impeller along line V—V of Figure 4, and also showing the raceways of the pump shown in Figure 1;

Figure 6 is an enlarged fragmentary sectional

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view taken through the impeller but taken in an intermediate transverse plane as indicated by the line VI—VI of Figure 4;

Figure 7 is a fragmentary plan view similar to Figure 4, but illustrating a pump impeller of greater capacity than that shown in Figures 3 and 4;

Figure 8 is an enlarged fragmentary sectional view taken transversely through one of the raceways of a pump utilizing the impeller of Figure 7; and

Figure 9 is a fragmentary transverse sectional view taken substantially along line IX—IX of Figure 7.

Referring now in particular to the drawings, the invention shown therein is shown as being embodied in a pump 10 including a housing or frame 11 mounted on a bed plate 14. Rigidly secured to the frame 11 is a driving motor 12 also mounted on the bed plate 14. The upper portion of the frame 11 is of an enlarged hollow formation, and the inner walls thereof define a suction passage 15 and a discharge passage 16, the passages being separated by a wall portion 18. The wall portion 18 is inclined toward the suction passage 15 and forms a laterally enlarged upper portion of the discharge passage 16 defining a priming pocket 19 to which access may be had as by means of a threaded plug 20. The pump is provided with a main drive shaft 21, which is herein shown as being an extension of the shaft of the driving motor 12. The shaft 21 is journalled at its outer end in a suitable bearing 22, which may be of an anti-friction type, and is shown as being housed within a hollow boss 24 of an end plate 25. The end plate 25 is detachably secured to the outer end of the frame 11, as by a series of cap screws 26.

Keyed or otherwise secured to the drive shaft 21 for rotation therewith is an impeller 28 having a hub 29. The outer peripheral portion of the impeller 28 is provided with a plurality of relatively closely spaced oppositely extending angularly disposed slots alternately opening to opposite sides of said impeller and defining fluid impelling buckets 30. Straddling the impeller and disposed in overlying relation with the buckets 30 is a pair of annular raceways 31, 32 extending around the impeller 28 from the intake passage 15 to the discharge passage 16. In the illustrative pump 10, the raceway 31 is formed as an open groove extending partially around a liner 33 mounted within the frame 11 and held from rotation by a pin 34. The raceway 32, on the other hand, is formed from an open groove ex-

tending partially around an inner face of the cover plate 25, and spaced inwardly from the periphery thereof. The cover plate 25 abuts and serves to retain the liner 33 in place within the frame 11. Both of the raceways 31 and 32 communicate with the suction passage 15 via a suction port 35 formed in the liner 33 and the end plate 25. Said raceways communicate with the discharge passage 16 via a discharge port 36 formed in said liner and end plate. The ports 35 and 36 are separated as by means of a partition or land 38 integral with the liner 33, which also separates the suction end and the discharge end of the raceway 31. The suction and discharge ends of the raceway 32 are separated by a land 40 formed on the inside of the end plate 25.

The motor end of the pump 10 is provided with a readily accessible self-contained shaft seal 41. In the present instance, such seal comprises a usual annular sealing washer 42 housed within a shouldered recess formed in a hollow boss 45 integral with the pump frame. The washer 42 extends over a flanged collar 46 on the drive shaft 21, such collar having an annular flange 47 adapted to bear against the washer 42. A yieldable axial thrust is exerted against the washer 42 as by means of an axial compression spring 48 encircling and seated against a flanged washer 49 abutting the collar 46 and slidably mounted on the drive shaft 21. The opposite end of the spring 48 is seated in a cupped seating member 43 abutting a snap ring 44 on the shaft 21.

The seal 41 is protected against damage in the event that the pump should be idled over a considerable period of time without a flow of fluid therethrough by liquid stored within the pump casing in a pocket 50. One wall of the pocket 50 is formed by a web 51 which supports the hollow boss 45 and is of a frusto conical formation. The opposite wall of the pocket 50 is formed by the adjacent side of the liner 33 and the inner side face of the impeller 28. Once fluid has entered the pump, the pocket 50 is filled with liquid through bleed passages 52, 53 and this pocket 50 tends to retain sufficient fluid to keep the seal lubricated at all times, even though the pump may be idling without pumping liquid. Also when the pump is pumping liquid and a gas or air pocket is encountered, the pump may pump the gas or air without injury to the seal or without losing its prime, and may resume pumping water with no attention or damage to the pump. A drain passage 57 closed by a drain plug 57A is provided in the end plate 25, to enable the pump and pocket 50 to be drained.

In order that the seal 41 may be replaced, inspected or repaired without disturbing the driving motor 12, the liner 33 is provided with a relatively large central aperture 55 having a diameter substantially greater than that of any of the component parts of the seal 41. The seal thus can be reached merely by removing the cover plate 25 and pulling the impeller 28, there being no occasion to disturb the liner 33. When the necessary maintenance or inspection has been completed, the seal and those parts of the pump which have been removed may be replaced as easily as they were dismantled.

The impeller 28, as shown in Figures 1 to 6, is in the form of a flat cylindrical disk having substantially parallel side faces 56 and 58, although it need not be of this form. As mentioned before, the outer peripheral portion of the impeller is angularly slotted to form a plurality of nested fluid receiving buckets 30 which open out into the opposite faces 56 and 58 of the disk as well as into the outer periphery of the disk. Each bucket 30 has lateral side walls 59 and 60 which are substantially parallel to each other and substantially perpendicular to a floor 61 of the bucket. The walls 59 and 60 also extend angularly outwardly from the transverse central portion of the impeller toward the side walls thereof in the direction of rotation of the impeller and intersect the side faces of the impeller at an acute angle, designated by the letter $\phi$ in Figure 4, which has been found to be critical insofar as the performance of the pump is concerned. For example, in a pump impeller of the foregoing type designed for operation at approximately 1750 R. P. M. and having a diameter of about three to four inches, the value of the angle $\phi$ should be somewhere between 45° and 50° and preferably about 47½°. The floor 61 of each bucket, on the other hand, is of an arcuate concave shape, the curvature of said floor changing from a generally radial direction adjacent the outer periphery of the disk to a generally axial direction as the axis of the impeller is approached. This structure gives each bucket a volume which progressively decreases towards the axis of the impeller improving the flow characteristics of the pump and reducing the eddy currents therein.

In order to take full advantage of the skewed relationship of the bucket side walls 59 and 60 with respect to the impeller side faces 56 and 58, the buckets are alternately arranged in two sets nested together in back-to-back relationship so as to create the general herringbone pattern shown in Figure 4. As the drawing indicates, ribs or teeth 62 define the advance and retreating side walls 59 and 60, respectively, of two succeeding buckets. Said teeth are of a wedge shaped cross section, diverging from the root circle of the buckets, and the advance edges thereof are inclined away from the direction of rotation of the impeller. Referring now to Figure 3, the negative rake of an advance or biting edge 64 of each tooth 62 and bucket 30, measured at the peripery of the impeller, is represented by the letter N. This rake is the distance between the peripheral end of the biting edge and a radius IR of the impeller passing through the root end or bottom of the biting edge of the bucket. The foregoing may readily be accomplished by mounting the impeller 28 in the milling machine with the cutter spaced from the vertical center line XY by an offset designated by the letter O, such offset being measured in the plane of the side face of the impeller and in the direction of rotation of the impeller. Thus, it will be appreciated that the value of the negative rake N will be proportional to the value of the offset O. While the actual values of these constants will vary with the capacity and dimensions of the pump, I have observed certain limiting factors. For example, as the rake of the biting edge of the bucket is gradually increased beyond the optimum value, the pump commences to lose its self-priming characteristics and its efficiency decreases. On the other hand, as the rake of the biting edge of the bucket is reduced from the optimum value, the power consumption increases, along with a decrease in efficiency and capacity. As the point of zero rake is passed and the edge 64 is given an increasing amount of positive rake, the power consumption tends to become excessive.

For an impeller having a diameter of three to four inches, and operating at a speed of about 1750 R. P. M., and with the buckets one-half inch deep and inclined to the side faces of the impeller at about 45° to 50°, an optimum value for the offset O lies somewhere between one-sixteenth and one-eighth of an inch with the negative rake N being approximately one-half the value of the offset O.

It will be observed from the foregoing that the impeller structure is particularly well adapted to be produced on a conventional milling machine and by the use of an ordinary milling cutter. It is only necessary to set up the impeller 28 in the machine in proper angular relation to the milling cutter and to move the cutter and the impeller relative to one another in a generally radial direction until a bucket of proper dimensions has been cut. The peripheral cutting edges of the cutter, moving on a radius R, defines the bucket floor 61 while the side cutting edges define the bucket side walls 59 and 60 (see Figures 4 to 6). After one bucket has been cut, the impeller can be shifted to the proper angle for milling out the next bucket, the machine being indexed accordingly as for the cutting of gear teeth.

Having considered the impeller per se, the general relation between the impeller and the raceways will now be considered. It should first be noted that for a given operating speed and impeller diameter, an increase in pump displacement necessitates an increase in bucket volume, which may be attained by increasing the width or thickness of the impeller together with an increase in the cross-sectional area of the raceways. Conversely, a decrease in pump displacement necessitates a decrease in impeller thickness, bucket volume, and cross-sectional area of the raceways. While the foregoing is fairly obvious when briefly analyzed, the specific relation between the contour of the impeller bucket floors 61 and the shape of the cross-sectional area of the raceways, which has been empirically developed, is far from obvious. For best efficiency over a wide range of operating pressures, I have found that a substantial portion of each raceway wall should taper outwardly from the region of the root circle of the impeller buckets at an angle $\theta$ determined by the contour of the bucket floor 61, giving each raceway an area directly proportional to the displacement of the bucket.

The volume of the raceway thus increases toward the periphery of the propeller in direct proportion to the increase in volume of the bucket. This reduces slippage and the resulting eddy currents and greatly increases the efficiency and operating characteristics of the pump.

The angle $\theta$ is determined by a chord E—F intersecting the outer edge of the bucket floor, or the root circle of the buckets and the outer extremities of the arcuate portion of the bucket floor 61 where the contour of the floor changes to a radial direction, shown as being vertical in Figures 1 and 6. The angle between the chord E—F and the adjacent side face of the impeller is indicated by the reference character $\theta$. As shown in Figure 5, the wall of each of raceways 31 and 32 starts with a slight fillet at the impeller bucket root circle or inner outer edge thereof and is then inclined outwardly at the angle $\theta$. The length of this tapered portion will be determined primarily by the volume of the raceway, which is proportional to the total displacement of its associated impeller buckets 30.

It should here be understood that a decrease in the area of the raceway increases the pressure of the fluid delivered by the pump for a given speed, and vice versa.

In order for the pump 10 to operate effectively, it is necessary to have relatively close running fits between the impeller 28, the cover plate 25 and the liner 33. In pumps of the type described herein, it has been found satisfactory to make the peripheral clearances of the impeller on the order of .005" to .008". The side clearances should be on the order of .001" to .0015". For the purpose of equalizing differential pressures which might otherwise tend to build up on opposite sides of the impeller due to such close clearances, and to reduce end thrust on the impeller due to unbalanced pressures, the impeller 28 may be provided with one or more balancing passages 65 extending therethrough in the direction of the axis of rotation thereof.

In Figures 7, 8 and 9, I show fragmentarily a portion of an impeller 28A similar to the impeller 28 and of approximately the same diameter but having a greater bucket volume attained by increasing the thickness of the impeller. In view of the similarity between the two impellers, like parts in each will be designated by the same reference numerals. Each bucket 30 of the impeller 28A is formed with substantially parallel side walls 59 and 60, and a concave floor 61 which in this instance is milled out with a cutter of radius $R_1$. Although these buckets are somewhat deeper than those of the impeller 28, they are arranged in a herringbone pattern similar to that of the impeller 28, with inclined ribs or teeth 62 separating the buckets. The biting edge of each of the ribs 62 has an appropriate amount of negative rake and the side profile of each rib is wedge shaped, as shown in the drawings.

Each of the raceways in the pump utilizing the impeller 28A is formed in the same manner as the raceways 31 and 32 described above. As in the case of the impeller 28, the angle $\theta_1$ is defined by chord E1—F1 intersecting the extremities of the arcuate portion of the bucket floor 61. Thus, in Figure 8, one of the raceways 32A, associated with the impeller 28A is shown tapering outwardly from the region of the bucket root circle at the angle $\theta_1$. The angle of the wall of the raceway 32A like the angle of the raceway 32, conforms to the displacement of the teeth 62, 62 and confines the cross-sectional area of said raceway to the shape of said teeth and the displacement thereof. Thus, since the buckets of the impeller 28A are wider than those of the impeller 28, giving them a greater displacement, the radius $R_1$ will be greater than the radius R with the result that the angle $\theta_1$ will also be greater.

In the light of the above description, the principal advantages possessed by turbine pumps constructed in accordance with my present invention may be briefly summarized. The improved efficiency, reduction in eddy currents and improved flow characteristics attained by proportioning the raceway area to the displacement of the buckets is readily apparent to those skilled in the art. The combined simplicity and precision of the parts, together with the ease with which the parts may be assembled, are also readily apparent to those skilled in the art and largely account for the low manufacturing cost and high efficiency of the pump. The symmetrical disposition of the impeller buckets and raceways and the balancing passageway between op-

posite sides of the impeller serve to balance out the axial thrust on the pump shaft and eliminate the need for thrust bearings. The staggered buckets inclined in the direction of travel of the impeller and the arcuate form of the floors thereof together with negative rake and the angular relation of the side walls of the raceways to the buckets further improve the efficiency of the pump and the flow characteristics, resulting in a pump having a much higher degree of efficiency than former pumps of the turbine type. The balanced condition of the pump and the reduction in eddy currents, coupled with the fact that the projecting end of the drive shaft is securely supported in a guide bearing further accounts in large measure for the silent operation of the pump.

Although the illustrative pumps described above have included duplicate sets of impeller buckets, each set having a separate raceway associated therewith to eliminate thrust, it should be borne in mind that certain phases of the invention are applicable to a pump having an impeller with only a single set of buckets and a single raceway associated therewith. A pump of this character would, of course, have to be provided with some sort of thrust bearing and would probably lack the capacity of a pump of comparable size built as described earlier herein. However, commercial considerations might make it desirable to produce single raceway pumps for certain specific applications.

It will be understood that modifications and variations may be effected without departing from the spirit and scope of the novel concepts of the present invention.

I claim as my invention:

1. In a turbine pump, a hollow casing having an inlet port and an outlet port, an impeller journalled within said casing and having a plurality of nested angularly disposed buckets opening in the direction of rotation of said impeller and to the periphery and opposite sides thereof, a rib forming the biting edge of each of said buckets and inclined away from the direction of rotation of the impeller, giving the biting edge thereof a negative rake, and having a cross-sectional area increasing from the root to the periphery of said bucket in accordance with the width and required displacement of the impeller, and said casing having a recessed raceway extending along each side of said impeller from said inlet to said outlet port and opening from the sides of the buckets thereof and extending outwardly from the root circles of said buckets and terminating into a peripheral annular wall portion in direct alignment with the outer periphery of said impeller.

2. In a turbine pump, a hollow casing having an inlet port and an outlet port, an impeller journalled within said casing and having a plurality of nested angularly disposed buckets opening in the direction of rotation of said impeller and to the periphery and opposite sides thereof, a rib forming the biting edge of each of said buckets and inclined away from the direction of rotation of the impeller, giving the biting edge thereof a negative rake, and having a cross-sectional area increasing from the root to the periphery of said bucket in accordance with the width and required displacement of the impeller, and said casing having a recessed raceway extending along each side of said impeller from said inlet to said outlet port and opening from the sides of the buckets thereof and extending from the root circles of said buckets and terminating in direct alignment with the outer periphery of said impeller and said raceways having side walls inclined outwardly from the root circles of said buckets and increasing in cross-sectional area in direct proportion to the displacement of said buckets.

3. A turbine pump comprising a hollow casing having an inlet port and an outlet port, an impeller journalled within said casing and having a plurality of uniformly spaced oppositely extending nested buckets angularly disposed with respect to opposite sides of said impeller and opening to the periphery of said impeller and to opposite sides thereof, each of said buckets having a closed inner wall extending generally vertically adjacent the outer periphery thereof and terminating into an arcuate floor of uniform contour terminating at a side wall of said impeller substantially tangent to a uniform root circle defining the bases of said buckets, and said casing having two facing recessed raceways extending annularly from said inlet to said outlet port, said raceways opening to the sides of said buckets only and extending from the root circles of said buckets and increasing in cross-sectional area from the root circles of said buckets in direct proportion to the displacement of said buckets and terminating in an annular wall parallel with and in direct alignment with the outer periphery of said impeller.

4. A turbine pump comprising, in combination, a hollow casing, an impeller drive shaft journalled in said casing, an impeller on said shaft having substantially parallel side faces, said impeller having a series of angularly extending parallel fluid receiving buckets opening into its outer periphery and into one of the side faces of said impeller, said impeller also having a second series of similar buckets extending in opposite angular relation with respect to said first series of buckets and opening into its outer periphery and into the other one of its side faces, each of said buckets having substantially parallel side walls and an inner wall terminating into a concave floor forming a downward continuation of said inner wall and tapering toward the axis of said impeller from a generally radial to a generally axial direction, and the side walls of said buckets having a negative rake, said casing having parallel walls in the interior thereof having running clearance with the side walls of said impeller and having annular raceways formed therein overlying the bucket openings and the side faces of said impeller, each of said raceways having an inclined wall tapering outwardly from the root circle of the buckets at an angle equal to and opposite from the angle defined by a chord intersecting the opposite extremities of the concave portion of the floors of the buckets and turning inwardly and terminating in direct alignment with the outer periphery of said impeller.

5. In a turbine type pump, an impeller of a general disk-like form and having a slotted peripheral portion having a plurality of uniformly spaced oppositely disposed angularly extending slots therein defining a plurality of fluid receiving buckets opening into the outer periphery and the opposite side faces of said impeller, each of said buckets having substantially parallel side walls and a concave floor curving upwardly from the root circles of said buckets and terminating into an inner wall perpendicular to the side walls of the associated bucket, the side walls of said buckets being inclined with respect to the axis of rotation of said impeller toward the direction of ro-

tation of said impeller and each making an acute angle with respect to said impeller side faces, and the side walls of said buckets in side elevation comprising a plurality of ribs of a wedge shaped cross section and having leading edges having a relatively slight degree of negative rake.

6. For use in a self-priming pump of the turbine type, an impeller having a peripheral portion and parallel side faces having a plurality of uniformly spaced slots therein and defining a first set of fluid receiving buckets opening into the outer periphery and the side face of said impeller, together with a second set of fluid receiving buckets opening into the outer periphery and the opposite side face of said impeller, each of said buckets having substantially parallel side walls in plan and a concave floor curving upwardly from the base of the associated bucket and terminating in a generally radially extending inner wall perpendicular to the side walls of said buckets, and said buckets being inclined at an angle with respect to the axis of rotation of said impeller to open in the direction of rotation thereof and making an acute angle with respect to the side face of said impeller into which said buckets open, and the side walls of said buckets including a plurality of wedge shaped ribs extending angularly with respect to a plane extending perpendicularly to the axis of rotation of said impeller in opposite directions from said plane and defining a herringbone pattern in plan, each of said ribs having a leading edge formed with a maximum amount of negative rake consistent with the self-priming characteristics of said pump.

7. A pump impeller of a generally disk-like form having parallel side walls and a uniform circular periphery having a plurality of oppositely extending nested buckets opening to the periphery thereof and to opposite side walls thereof, said nested buckets extending in opposite angular directions toward the side walls of said impeller and forming a herringbone pattern when looking at the periphery of said impeller, the side walls of said buckets at the outer sides of said impeller being in the form of wedge shaped teeth having a negative rake in the direction of rotation of said impeller, and each of said buckets having a closed inner wall of a generally radial contour adjacent the outer periphery of said impeller and curving downwardly into an arcuate floor of a uniform contour terminating at the base of each bucket at a side wall of said impeller substantially tangent to a uniform root circle of said buckets.



8. In a turbine pump, a hollow casing having an inlet port and an outlet port, an impeller journalled within said casing and having a plurality of angularly disposed nested buckets forming a herringbone pattern in plan and opening in opposite side walls of said impeller in the direction of rotation thereof and into the outer periphery of said impeller, each of said buckets having parallel side walls in the form of wedge-shaped biting teeth at the outer sides of said impeller and having a negative rake with respect to the direction of rotation of said impeller, said side walls having an impelling area increasing from the root to the outer periphery of said impeller, and said casing having a plurality of recessed annual races registering with said buckets and extending from said inlet to said outlet ports along the open sides of said buckets and increasing in cross-sectional area from the root circles of said buckets in direct proportion to the displacement of said biting teeth, and having outer annular wall portions extending inwardly toward said impeller in direct alignment with and parallel to the outer periphery of said impeller.

FLORIN W. KRUEGER.


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