United States Patent [19]

Kempf

[11] **3,759,628**

[45] **Sept. 18, 1973**

[54] **VORTEX PUMPS**

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[73] Assignee: **FMC Corporation,** San Jose, Calif.

[22] Filed: **June 14, 1972**

[21] Appl. No.: **262,913**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 104,315, Jan. 6, 1971.

[52] **U.S. Cl.** .... **415/213 A,** 415/206, 415/219 C

[51] **Int. Cl.** **F04d 29/44,** F04d 7/00

[58] **Field of Search** .... 415/204, 206, 213, 415/213 A, 219 C

[56] **References Cited**

UNITED STATES PATENTS

2,785,930	3/1957	Burnside	415/213 A
2,958,293	11/1960	Pray, Jr.	415/213 A
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3,171,357	3/1964	Egger	415/206
3,269,325	8/1966	Schwed et al.	415/213 A
3,316,848	5/1967	Egger	415/204
3,319,573	5/1967	Judd	415/213 A
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FOREIGN PATENTS OR APPLICATIONS

390,688	8/1965	Switzerland	415/213 A
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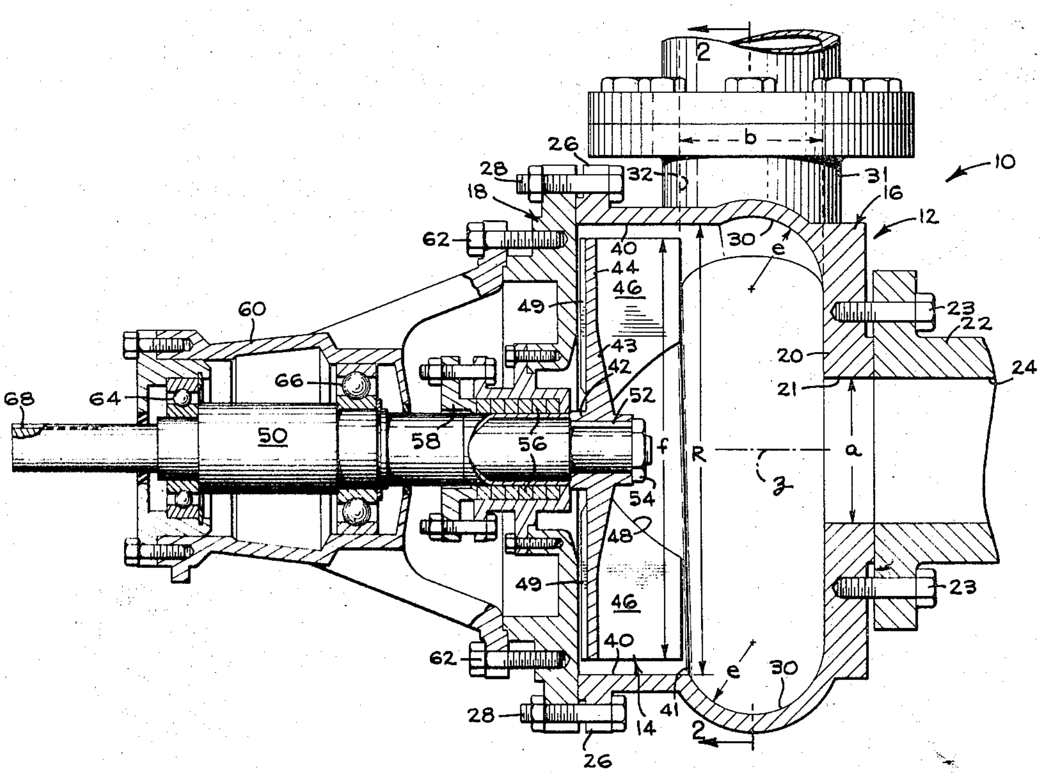
Primary Examiner—Henry F. Raduazo
Attorney—F. W. Anderson et al.

[57] **ABSTRACT**

A vortex pump casing has a generally toroidal inner wall of constant cross-sectional radius with a cylindrical axially offset impeller recess intersecting the toroidal wall. The toroidal wall is actually a volute and the impeller recess intersects the volute at an obtuse angle that increases progressively from the maximum major radius of the volute to the minimum (cutwater) major radius of the volute. Formulae for various design features, including the impeller vane depth, the radius to the volute cutwater, the impeller diameter, and the optimum number of impeller blades are presented, based on the diameter of the inlet port of the volute.

15 Claims, 18 Drawing Figures



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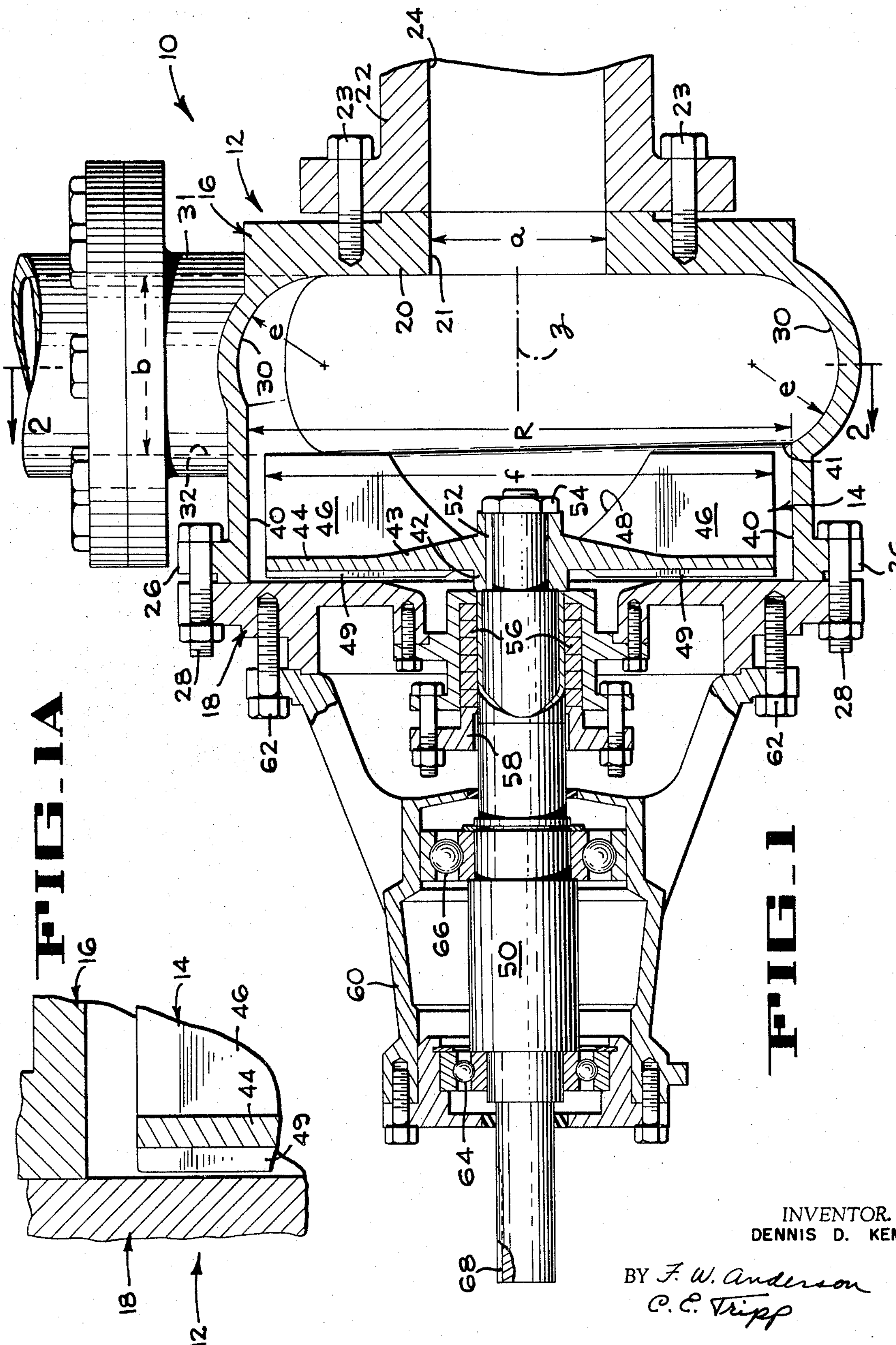

FIG_1

FIG_1A

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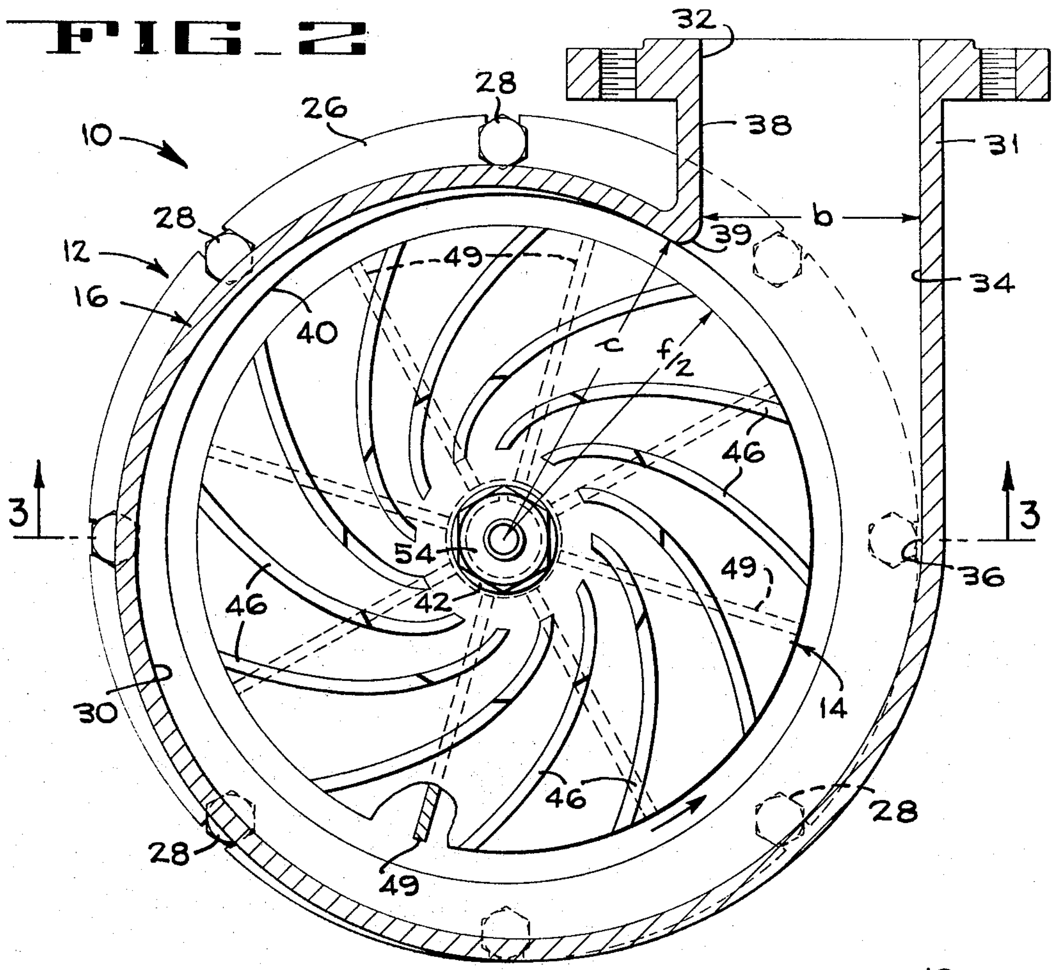
FIG_2


32
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26
10
38
31
28
12
16
b
39
34
49
40
c
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49
36
42
46
46
30
14
46
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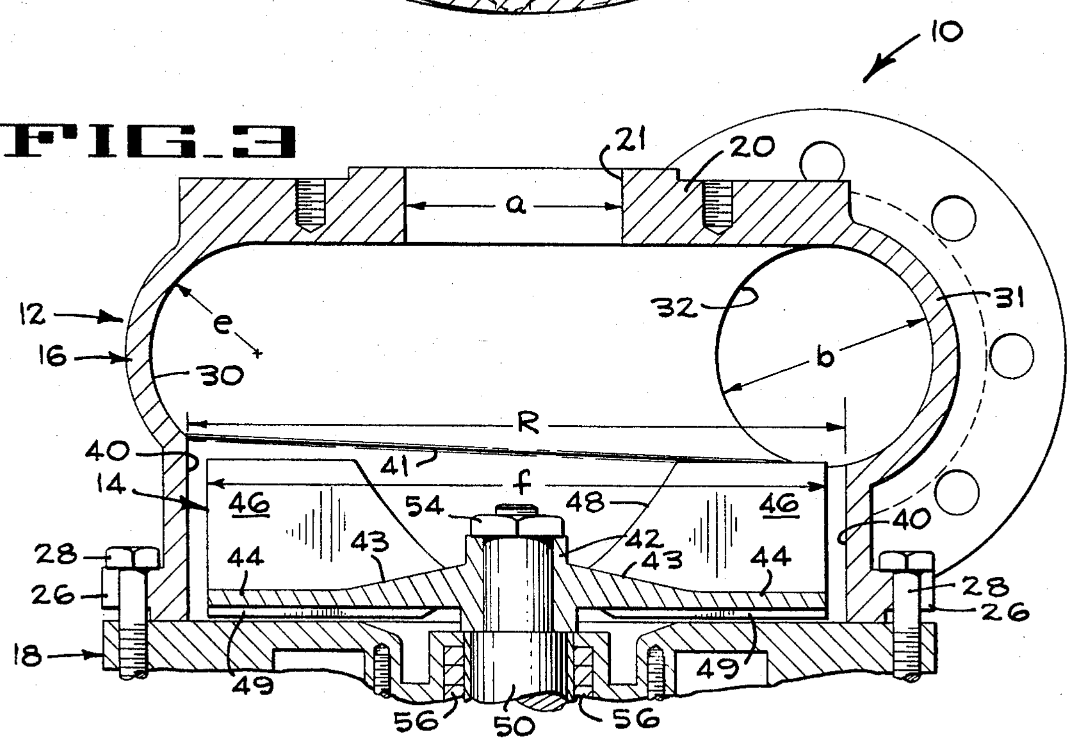
28
49

FIG_3


10
21
20
a
12
e
32
31
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30
b
R
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41
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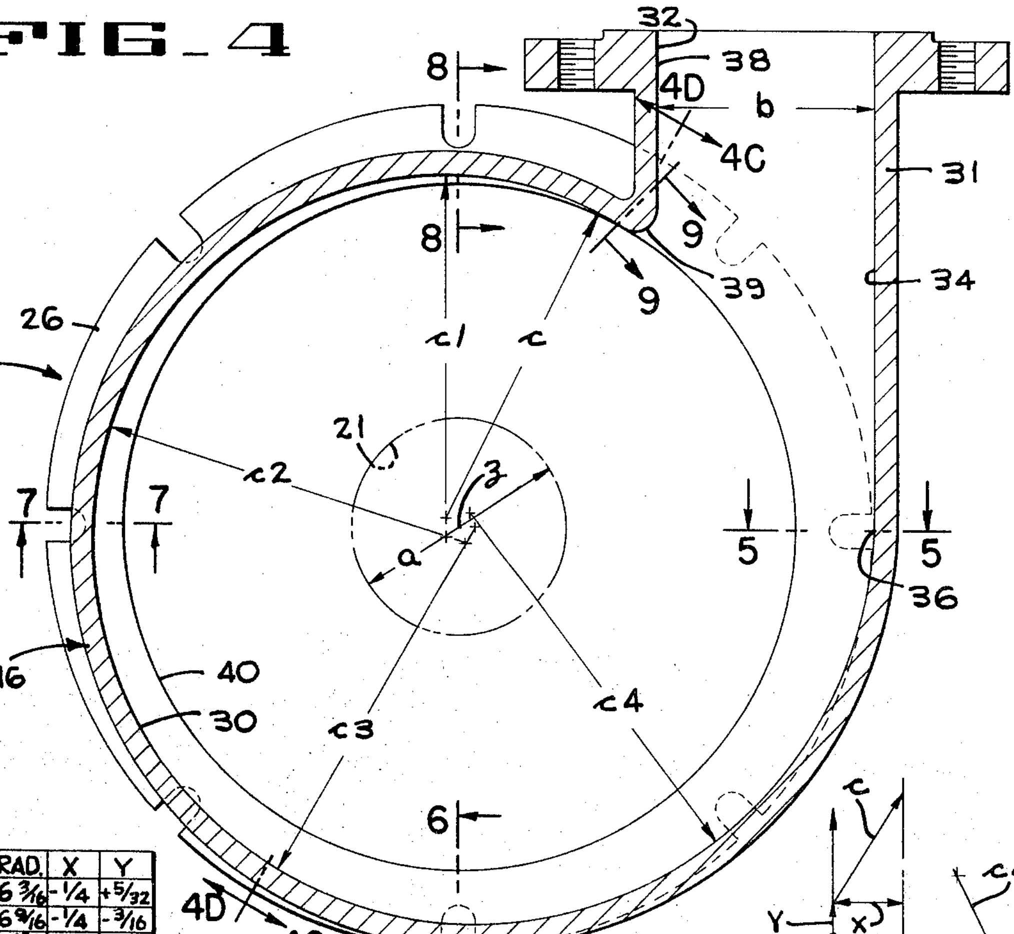
FIG_4



	RAD.	X	Y
c	6 3/16	-1/4	+5/32
c1	6 9/16	-1/4	-3/16
c2	6 15/16	+1/8	-5/16
c3	7 9/32	+5/16	0
c4	7 9/16	+3/16	+1/4

FIG_4B

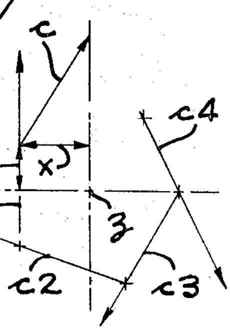


FIG_4A

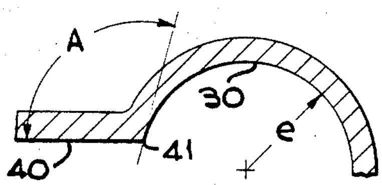


FIG_5

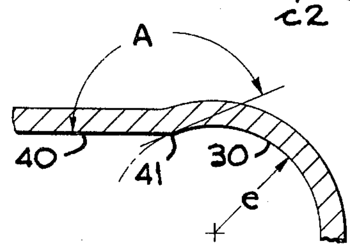


FIG_8

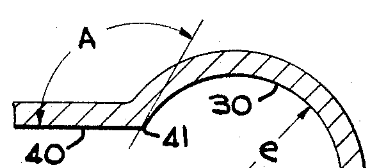


FIG_6

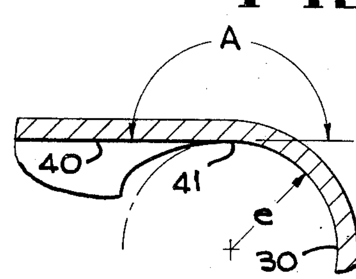


FIG_9

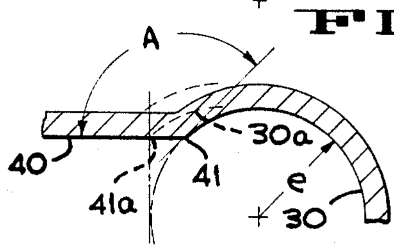


FIG_7

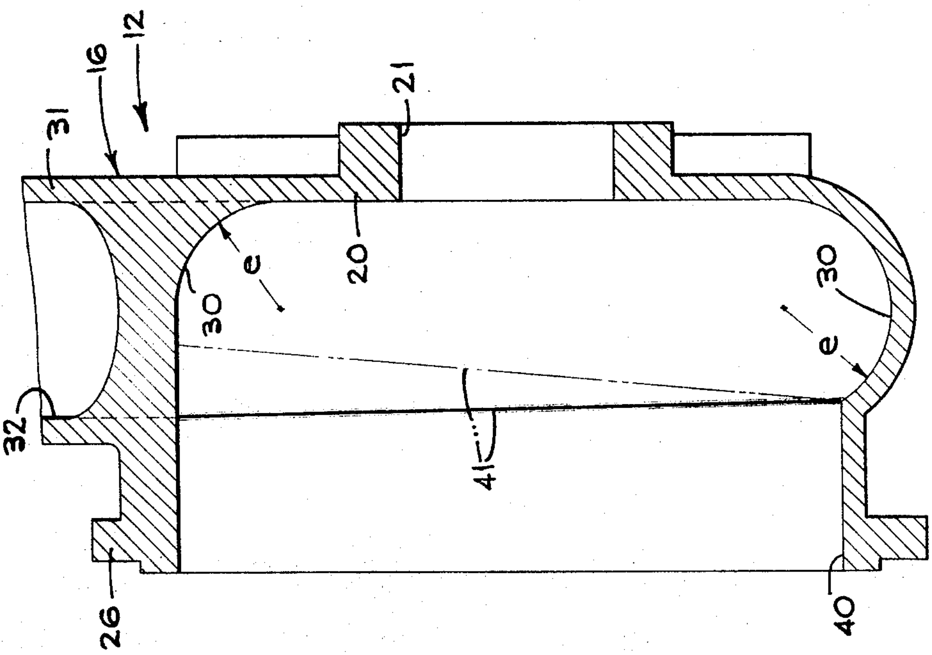


FIG_4C

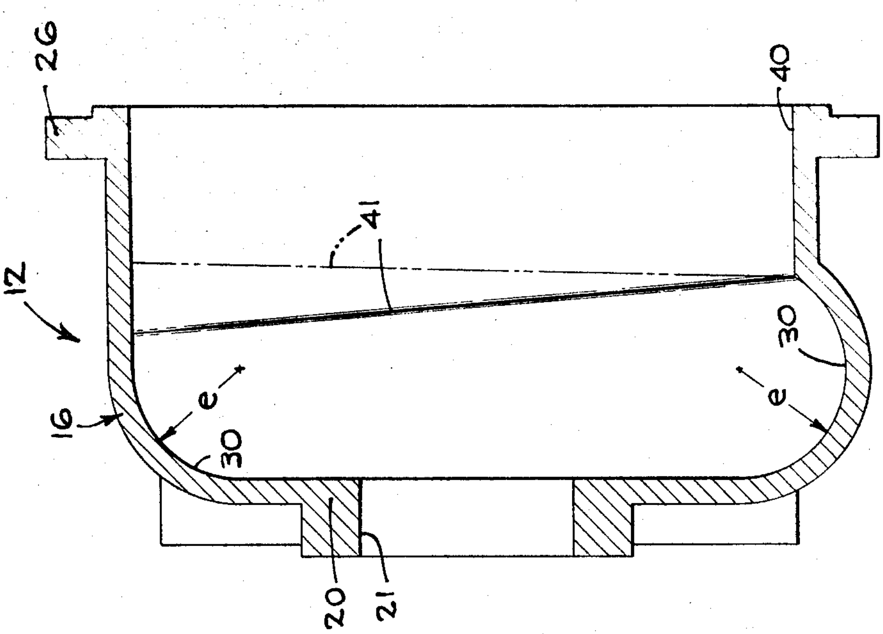


FIG_4D

FIG_10

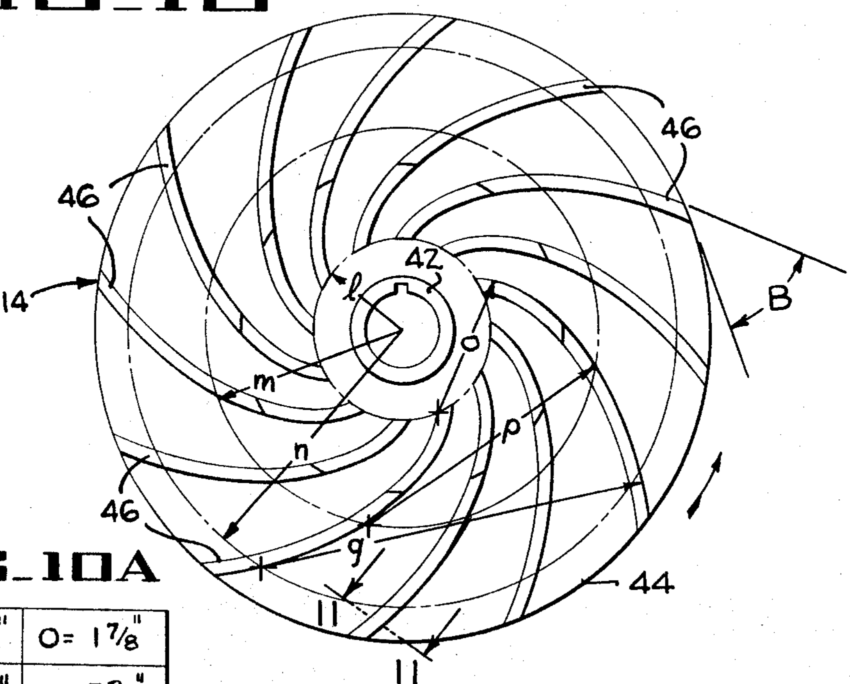


FIG_10A

ℓ = $1\frac{11}{16}$"	o = $1\frac{7}{8}$"
m = $3\frac{11}{16}$"	p = $5\frac{3}{16}$"
n = $5\frac{5}{8}$"	q = $7\frac{9}{32}$"

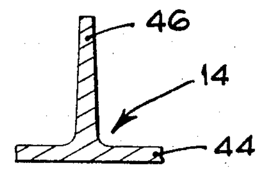


FIG_11

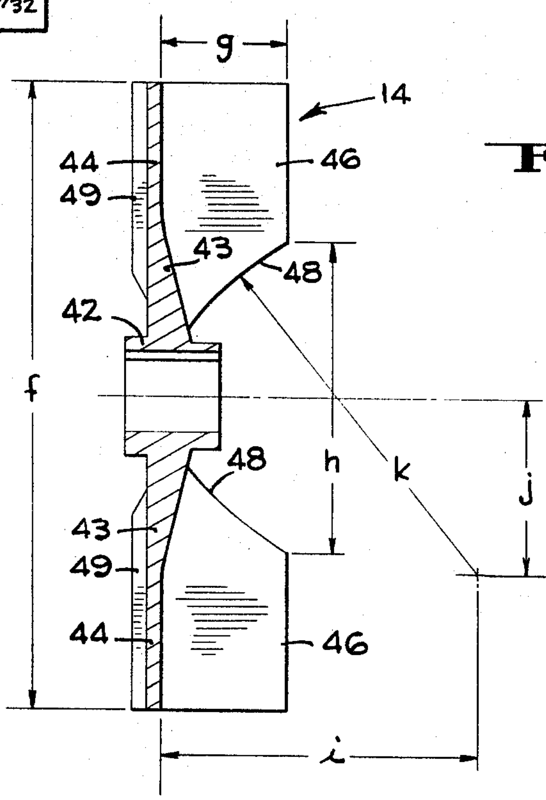


FIG_12

VORTEX PUMPS

REFERENCE TO RELATED APPLICATIONS

This application is a continuation-in-part of my co-pending application, Ser. No. 104,315, filed Jan. 6, 1971.

FIELD OF THE INVENTION

This invention relates to pumps and more particularly to pumps of the type which will be referred to as vortex pumps, which have a toroidal casing formed with an axially offset recess for receiving an open impeller. These pumps are particularly useful for handling solids-bearing liquids.

DESCRIPTION OF PRIOR ART

The U.S. Pat. to Egger No. 3,171,357, Mar. 2, 1965 discloses a vortex pump, having a recessed impeller in a circular casing with the intersection of the recess with the casing being at substantially 90°. The impeller has no recessed eye.

U.S. Pat. to Egger No. 3,316,848, May 2, 1967 (FIG. 3) shows a recessed impeller pump construction like that of the earlier Egger patent. The rear wall of the impeller has secondary vanes.

U.S. Pat. to Sence No. 3,130,679, Apr. 28, 1964 shows a volute casing in FIG. 3. The angle of intersection of the impeller recess with the volute is not shown, but the maximum diameter of the volute is spaced radially outward from the impeller recess at the cutwater. The impeller has a recessed eye and its vanes are not shrouded, but as seen in FIG. 1, the impeller is located within the volute itself.

U.S. Pat. to Droeger has found July present 18, 1961 also shows a volute casing with the impeller disposed within the volute itself and with a front cover plate on the impeller.

U.S. Pat. to Brawley No. 2,635,548, Apr. 21, 1953 shows a recessed impeller and an oddly contoured volute casing having an axial inlet (FIGS. 3 and 4). There is substantially no clearance between the impeller and the recess in the casing. The impeller recess intersects the volute in a plane perpendicular to the impeller axis and the outermost portions of the volute wall are radially offset from the impeller recess at the cutwater. The impeller does not have a recessed eye.

Burnside reissue U.S. Pat. No. 24,803, Mar. 29, 1930 shows a solids pump with a volute casing. The major diameter of the volute at the cutwater is greater than the diameter of the impeller recess (FIG. 2). The impeller vanes are shrouded but form a recessed eye.

U.S. Pat. to Glass No. 3,322,070, May 30, 1967, shows a vortex pump having a cylindrical casing containing the impeller with an axial inlet and a tangential discharge. The casing wall and the impeller recess are of the same diameter. The impeller is widely spaced from the casing wall to provide an auxiliary swirl chamber. The impeller is not shrouded and the inner ends of the impeller vanes form a very small angle with the impeller axis.

The U.S. Pat. to Schwed et al. No. 3,269,325, Aug. 30, 1966 shows a self-priming vortex-type pump wherein the casing is circumferentially concentric with the impeller. FIG. 2 shows that part of the casing opposite the discharge throat as having a curved cross section that is less than a semi-circle, but the specification does not describe the casing geometry. There is no recessed impeller eye.

German Pat. No. 179,266, Jan. 16, 1906 shows a combined inlet and impeller concentric with a circular, wherein the impeller has outwardly divergent walls that are tangent to a continuation of the casing wall.

Kay U.S. Pat. No. 1,650,873, Nov. 29, 1927, shows a rotary blower having 12 blades. The specification does not give dimensions of the parts.

The U.S. Pat. to Lykken No. 1,893,710, Jan. 10, 1933 shows a radial flange impeller with the axially outer ends of the vanes normal to the impeller axis and terminating short of the hub to form an impeller eye. The inner ends of the vanes are straight. The impeller is not in a recess to one side of the volute.

LaBouri U.S. Pat. No. 2,669,938, Feb. 23, 1954, shows a spoked impeller with slanted outer edge vanes terminating in inclined, straight inner edges.

The U.S. Pat. to Pray, Jr. No. 2,958,293, Nov. 1, 1960 shows a solids pump wherein the impeller recess is conical, the impeller vane ends are shrouded, and the impeller has no recessed eye. There is no volute and the pump casing is circular having a U-shaped cross-section, even at the cutwater.

The U.S. Pat. to Judd No. 3,319,573, May 16, 1967 shows a centrifugal pump, the casing of which has a volute that increases in depth and volume in an axial direction only and never has a radius greater than that of the impeller recess. The impeller itself has a full diameter convex nose and hence has no recessed eye.

Swiss Pat. No. 390,688 of Aug. 13, 1963 shows a vortex pump with a recessed impeller that makes a close, labyrinth fit with the impeller recess. No front elevation is provided but it appears that upstream of the discharge duct the casing wall substantially forms an extension of the impeller recess. The radial outer wall of casing does not lie in a plane, being closer to the impeller adjacent the outlet than at a point about 180° from the outlet. There is no skew or helical line of intersection between the impeller recess and the volute, assuming the latter to be present. The casing has a radial inner side wall opposite the discharge duct, with a cylindrical peripheral wall. The impeller vanes are shrouded and the impeller has a convex nose rather than a recessed eye.

A paper entitled "Some Performance Characteristics of a Centrifugal Pump with Recessed Impeller" by V.M. Lubieniecki was received at the ASME headquarters, at 345 East 47th Street, New York, on Nov. 24, 1971. The pump tested was a vortex pump with unshrouded impeller vanes that closely fits the impeller recess and a recessed eye. The casing was concentric with the impeller recess, and hence was not a volute. The conclusions were that deeper vane impellers provide better performance than shallow vane impellers for a given impeller diameter. It was also found that the gallons/minute delivery of a given pump operating at a given speed is directly proportioned to the square of the impeller diameter.

SUMMARY OF THE INVENTION

The vortex pump of the invention is particularly useful in pumping liquid transportable solids which would be damaged by excessive impeller contact and fluids containing large, stringy or abrasive solids, such as sewage. Although such pumps are generally known in prior art and have been available to the trade, the present invention represents improvements over known pumps of this type, particularly with relation to pump efficiency, the pump head at both low and high discharge volumes and the noise level.

It has been found that the above pump factors and others can be either improved or possibly rendered less desirable by various individual and combination changes in the relatively large number of pump design factors, most of which heretofore have been given only superficial consideration. At the present state of the art, improvements in pump efficiency (for example) which represent gains in only a few percentage points are considered to be of considerable economic value. Some of the design factors which have been found to have significant effects on pump performance are as follows:

- The relative radial position of the impeller recess and the internal toroidal wall of the casing volute, particularly at the cutwater.
- The angle of intersection between the impeller recess and the volute, particularly at the cutwater.
- The radial clearance between the impeller and the impeller recess in the casing.
- The c depth of the impeller blades as a function of the discharge diameter (the nominal pump size).
- The diameter of the impeller eye relative to pump size.
- The discharge angle of the impeller blades.
- The number of impeller blades relative to pump size.
- The shape of the axial inlet.

Information that will facilitate production of improved pump designs within certain discharge port size ranges is given in the detailed specification which follows. However, and by way of example, one characteristic of the present invention can be explained briefly in this summary and that is the geometry of the impeller recess with respect to the casing volute. It has been found that increased performance is obtainable if the volute itself is made shallower than usual. More specifically, the volute has the same basic cross-sectional radius throughout its working periphery. The radius of the impeller recess is equal to the smallest maximum radius of the volute, which is the maximum radius of the volute at the cutwater so that the intersection of the impeller recess with the volute wall at this angular location is radially of the cross sectional center of curvature of the volute. This means that the impeller recess intersects the volute wall at an obtuse angle substantially around the volute. This also means that the intersection of the impeller recess with the volute wall describes an approximate helix and hence, with the aforesaid obtuse angle of intersection increasing from the tangential zone of discharge (large maximum radius portion of the volute) to an angle of about 180° at the cutwater (small maximum radius portion of the volute).

As mentioned, this and a number of other design features will be best understood from a detailed description of a preferred embodiment of the invention that follows.

DESCRIPTION OF THE DRAWINGS

FIG. 1 is an axial cross section through a pump embodying the invention.

FIG. 1A is a fragmentary enlarged view through the tip of the impeller blades and the casing.

FIG. 2 is a section taken on line 2—2 of FIG. 1.

FIG. 3 is a section taken on line 3—3 of FIG. 2.

FIG. 4 is a view like FIG. 2 with the impeller removed illustrating certain geometrical relationships of the volute.

FIG. 4A is a diagram showing the location of certain nominal radaii of the volute wall.

FIG. 4B is a table showing the centers of the radii of FIG. 4A.

FIGS. 4C and 4D are sections of the pump casing taken as indicated on FIG. 4.

FIGS. 5–9 are sections of the volute wall taken as indicated on FIG. 4 (FIG. 7 shows in dotted lines a volute section having inferior operation for purposes of comparison).

FIG. 10 is a view of the impeller like that of FIG. 2 showing certain geometrical relationships relative to blade shape.

FIG. 11 is a section taken on line 11 — 11 of FIG. 10.

FIG. 12 is a section through the impeller showing other geometrical relationships of the construction.

DETAILED DESCRIPTION

GENERAL DESCRIPTION OF THE PUMP

Referring to FIGS. 1 and 2, the pump embodying the present invention is illustrated generally at 10. The pump has a main casing indicated generally at 12 with an impeller 14 in the casing. In the construction shown, the casing 12 is formed of two major parts, a main section 16 and a backplate section 18. The main section 16 includes a radial front wall 20 having an inlet opening 21 having a diameter indicated at "*a*." An inlet pipe 22 is bolted to the front wall 20 by bolts 23 and has an inlet bore 24 that forms a prolongation of the pump inlet opening 21. At the impeller side of the main casing section 16 a radial flange 26 receives bolts 28 that mount the rear casing section 18.

The pump casing is formed as a volute having a toroidal inner wall 30 which wall, in the preferred construction, has a circular cross section, with the radius of the circular cross section of the volute 30 being designated "*e*" in the drawings.

The pump casing has a tangential discharge or outlet 31 with a bore 32 having a diameter "*b*" that is preferably substantially the same as the diameter "*a*" of the pump inlet. In some designs the inlet may be larger than the outlet. The outermost portion 34 of the outlet 31 joins the wall of the volute at 36 (FIG. 2) and forms a smooth, tangential continuation of that portion of the volute wall 30. The opposite wall 38 of the discharge 31 leads from what is termed the "cutwater" portion or terminal end 39 of the toroidal volute wall 30.

A recess 40 extends rearwardly from the volute wall 30 for receiving the impeller 14. The recess has a diameter R which is about a half inch greater (for example) than the diameter "*f*" of the impeller 14 (FIG. 1). As will be described in detail presently, it is a feature of the present invention that the impeller recess 40 intersects the volute wall in an approximately helical plane, illustrated at 41 and best seen in FIGS. 4C and 4D. Also, the radius of the impeller recess 40 is substantially equal to the smallest maximum radius of the casing wall 30 at the cutwater (FIG. 4).

As seen in FIG. 1, the impeller 14 is secured to a stepped shaft 50 by a key 52 and a nut 54. The shaft extends through packings 56 in the back plate 18 of the casing and which may be taken up by a gland 58. The shaft is supported in a bearing housing 60 secured by bolts 62 to the backplate 18. The bearing housing 60 mounts radial ball bearings **64,66** which receive, in accordance with conventional engineering design, the impeller shaft **50**. The details of this bearing assembly are not critical to the present invention. The outer end of the shaft **50** is keyed at **68** to receive a driving pulley or the like (not shown) for rotating the impeller.

VOLUTE GEOMETRY

Salient features of the volute geometry are illustrated in FIGS. **4–9.** As previously described, the cross sectional radius "*e*" of the volute is constant around its peripheral extent, and hence the section is circular. selected layout radii of the volute are illustrated in FIG. **4** between the cutwater radius "*c*" to the largest layout radius "*c4*" at the tangent point **36** of the volute with the outlet wall **34** of the discharge. FIGS. **4A** and **4B** give typical design data. In the illustrated embodiment, the dimensions "*a*" and "*b*" for the inlet and outlet pipes are 4 inches. The table of FIG. **4B** taken with the diagram of FIG. **4A** shows how to lay out the centers on Cartesian coordinates for the various volute layout radii "*c***1**" to "*c4*" relative to the central axis "*z*." This procedure for laying out a volute is well known to pump designers and described in detail in the article "How to Lay Out Centrifugal Pump Volute Casings" by Edward G. Hilficker in *Machine Design*, Aug., **1951**, pages 137 to 142.

Referring to FIG. **4** it will be noticed that an important feature of the invention is that the minor radius "*c*" of the outermost radial portion of the volute wall **30** substantially coincides with the radius of the impeller recess **40** at the cutwater portion **39** of the volute. FIGS. **4C, 4D** and **9** show this also.

FIGS. **5–9** are successive transverse sections showing the intersection of the impeller recess **40** with the volute cross section **30.** This construction forms an important feature of the present invention in providing an increased efficiency of the pump with a decreased noise level. Starting with the section of FIG. **5,** at the tangent point **36** between the volute wall and the discharge wall **34** (FIG. **4**) it can be seen that the wall of the impeller recess **40** forms an obtuse angle A with a tangent to the circular arc "*e*" of the volute. The intersection **41** of the impeller recess and the volute section radius "*e*" is best seen in FIGS. **4C** and **4D**, and it is a feature of the invention that this intersection lies in a skew plane. Referring back to FIG. **5,** the angle A is obtuse because the zone of the intersection **41** of the impeller recess **40** and the inner wall **30** of the volute lies radially outside the center of curvature of the radius "*e*" of the volute. In the detailed example presented, the angle A at the section of FIG. **5** is about 102°.

Progresing around the volute from the tangent zone **36** of the discharge toward the cutwater zone **39,** it can be seen that the obtuse angle A increases steadily. For example, in FIG. **5,** the angle A is equal to approximately 102° in the present embodiment; in FIG. **6** the angle A is 118°; in FIG. **7,** the angle is 133°; in FIG. **8** the angle is 155° and in FIG. **9** (at the cutwater) the angle is 180°. Thus, at the cutwater, the impeller recess **40** is substantially tangent to the wall **30** of the volute.

FIG. **7** shows in dotted lines an inferior construction presented for comparison with that of the present invention.

In FIG. **7,** as shown in dotted lines, the volute wall, **30*a*** is of constant width around the volute. In other words, the intersection **41*a*** of the impeller recess **40** with the volute wall **30*a*** would be axially spaced from the center of curvature of volute by a distnace equal to the radius "*e.*" This construction, which widens the volute, is found to be less efficient than that of the preferred form of the invention illustrated in FIGS. **5–9.**

IMPELLER CONSTRUCTION

In addition to improving the performance of the pump over that of prior known pumps of the recessed impeller type by improving the casing design, it has been found that improvements can be made in the geometry of the impeller itself to increase the efficiency and reduce the noise of the pump. By way of example, a specific pump impeller is described and illustrated in FIGS. **10–12.** As seen in FIGS. **10** and **12** the impeller has a keyed hub **42,** tapered radial flange portion **43** and a straight radial flange portion **44.** The impeller blades **46** project axially from the flange portions **43,44** and they are curved so that their convex faces are their leading faces. The blades **46** have curved inner edges **48,** forming a recessed "eye." The edges **48** extend in continuous lines axially and radially outward from the base flange. Vanes **49** project from the rear side of the impeller flange portion **43,44** (see also FIG. **2**). The number N of impeller blades **46** is somewhat greater than that indicated by the prior art, and as will be seen presently it has been found that there is a generally optimum number of blades for maximizing impeller efficiency and minimizing the noise level.

FIG. **10** illustrates one manner for generally laying out the curvature of the impeller blades **46.** In the design shown, the blades are at three radii that merge smoothly, but each radius is struck from a different circular centerline. For example, the three circular centerlines "*l,*" "*m*" and "*n*" are illustrated, together with the corresponding radius of "*o,*" "*p*" and "*q*" struck from these centerlines respectively. In the typical pump being described wherein the inlet and outlet throats "*a*" and "*b*" have a 4 inch diameter, the dimensions for the radaii centerlines and their length is given in the table of FIG. **10A.** Other design factors of the impeller including its major diameter "*f*" previously mentioned are given in FIG. **12.**

In the typical construction shown in FIG. **12,** the impeller blades **46** have an axial width or depth "*g*" that bears a predetermined relation to the other dimensions of the impeller. The "eye" **48** formed by the curved inner edges of the vanes **46** of the impeller has a diameter "*h*" and the inner vane edges **48** are each formed on a radius "k." The coordinates for the radius "*k*" are given as an axial coordinate "*i*" and a radial coordinate "j." In the 4 inch discharge pump being described "k" equals 7 inches, "*i*" equals 5⅞ inches and "*j*" equals 3-1/16 inches. The blade depth "*g*" in the example given is 2 ⅝ inches and the major impeller diameter "*f*" is 11 ½ inches. In the example given and as shown in FIG. **10,** the number of blades N is 12 with a discharge angle B of just under 60°. The radial clearance between the periphery of the impeller **14** and the impeller recess **40** in the pump housing is about 7/16 of an inch.

Table I gives various vortex pump design data for a discharge throat "*b*" at the range of about 2 inches to 4 inches and is self-explanatory. Also included is Table II, giving the preferred number N of the impeller blades at various discharge angles B within a preferred range of those angles, which angles can be calculated from Table I data.

Table I - Vortex Pump Design Data - ($b = 2$ inches to 4 inches)

b = Volute discharge diameter - inches
N = Number of Impeller vanes
B = Impeller vane discharge angle - degrees
f = Maximum impeller diameter - inches
g = *Impeller vane depth - inches*
h = Impeller eye diameter - inches
R = Impeller recess diameter - inches
C = Volute radius at cutwater - inches
e = Volute sectional radius - inches
s = Volute spiral severity - radial inches/degree
$N = (b + 12.3 \text{ inches}) \sqrt{\tan B/2}$ (to closest whole number ±10 percent)
$g = \sqrt{b}/2 + 1\ 3/8$ inches ±10 percent
$c = (f + 0.3b - 0.25 \text{ inch}/2)$ ±10 percent
$f = 6.75$ inches $+ b$ ±10 percent
$h = 5$ inches $+ 0.25b$ ±10 percent
$c = R/2$ or $R = 2c$
s =(0.005 inch radial growth/One degree angular increment) ±30 percent

TABLE II — N, Number of Impeller Vanes (rounded off)

b	B = 20°	B = 30°	B = 45°	B = 60°	B = 75°
2"	N = 8	N = 9	N = 11	N = 12	N = 13
3"	8	10	11	13	14
4"	9	11	12	14	15
6"	10	12	14	15	17
8"	10	13	15	17	19
10"	12, 13	14	17	19	20

Typical RPM, 400 to 2,400.

Application of the design formulas of Table I will provide a high efficiency pump with the desired head-flow rate characteristics for pumps used in sewage work or the like, the formulas given for "f" (maximum impeller diameter) and for "c" (radius to volute cutwater) produced better results when applied to pumps having a discharge diameter "b" of about 2 inches to 4 inches than when applied to larger pumps wherein "b" is 6 inches to 10 inches. This does not mean that application of the formulas to Table I to the larger pumps will not serve as a positive and clear guide to designing these pumps for better characteristics than heretofore attainable (based on the specific prior art design teachings known at this time) without a lengthy, costly and carefully controlled test program. The fact is that formulas which give slightly better results for the larger pumps (6 – 10 inches) and which also apply to the 4 inches and smaller pumps have been developed and are now preferred.

These alternate and preferable (particularly for the larger pumps) formulas are:

Table III

$f = 9$ inches $+ 0.625\ b$ ±10 percent
$c = (f + 0.0625b + 0.625 \text{ inch}/2)$ ±10 percent
The remaining formulas of Table I continue to apply.

Some explanations relating to the design formulas and the various individual physical pump dimensions that must be selected for the basic desiderata at hand will now be presented. These explanations are backed up by a large number of controlled laboratory tests.

1. The volute discharge diameter "b" is determined by the limiting pipe flow velocities resulting from the flow rate desired, this being the usual practice in selecting this factor in the design of centrifugal pumps.

2. For pump applications commonly encountered in the field of pumping sewage or other solids-carrying fluids, the maximum impeller diameter "f" is ideally

determined by the formulas given in Tables I and III, preferably the latter. These formulas apply best to pump designs conventionally known as "medium" head pumps. In those cases where a considerably higher or lower head is required than can be obtained by the reasonable and usual shaft speed (e.g., 1,750 rpm) with the nominal impeller diameter, the diameter "f" can be changed in accordance with known centrifugal pump affinity laws. For example, to obtain lower head, "f" must be decreased in proportion to the ratio of the square root of the desired lower head and the square root of the head attained with a pump having the diameter "f" as given by the formula.

To obtain a higher head pump, "f" must be increased in proportion to the ratio of the square root of the desired higher head and the square root of the head attained with a pump having the diameter "f" as given by the formula.

Of course, the other dimensions of the pump that are directly or indirectly based on the diameter "f" will correspondingly change with the modified value of "f" in application of the formulas.

3. Table II shows that the impeller discharge vane angle B may be varied from about 20° to about 75°. In selecting the vane angle, it should be recognized that the lower angles give a steeper head-capacity curve (a desirable feature for the type service under consideration) but somewhat less efficiency. The higher vane angles (approaching radial) improve the efficiency of the pump but flatten the head-capacity curve. If too high a vane angle is selected, the pump becomes unfit for solids pumping service. The preferred angle of the vanes is about 30° to 45°, depending upon the steepness desired for the head-capacity curve and the efficiency required.

4. The value for the impeller eye diameter "h" given in Table I optimizes several factors. It accommodates trimming of the outside diameter "f" of the impeller for performance variations while maintaining full capacity of the impeller. This maintains maximum pump efficiency and extends the flow rate at which the efficiency begins to drop. It also decreases or eliminates leading edge cavitation of the impeller, reduces noise and maintains a reasonably good head-capacity curve slope.

5. As to selecting the number of vanes, generally speaking, the largest number of impeller vanes that can be crowded into the impeller without choking off the impeller inlet at the eye because of vane thickness is recommended. The number of vanes selected also depends on the blade entrance angle and the impeller eye diameter "h." With the impeller entrance angle and the impeller exit angle held equal, and where the impeller eye diameter "h" is determined in accordance with Table I, the equation for the number of blades N given in Table I will provide the optimum number of vanes for the pump propeller.

6. The impeller vane depth "g" is related to the desired flow rate of the pump. However, under the design principles of the present invention, it is assumed that the size of discharge diameter "b" has already been determined by the expected flow rate and hence, in accordance with Table I the vane depth "g" is made dependent on "b."

7. As to the radius "c" from the centerline to the volute cutwater (FIG. 2) it has been found, under the present invention, that this is a very important variable.

Tests have proven that a smaller radius "c" than has previously been used provides improved results. Accordingly, the formula for the cutwater radius "c" given in Tables I and III represent a valuable contribution to vortex pump design.

8. It will be noted in Table I that the impeller recess diameter R is equal to one-half the radius of "c" to the volute cutwater. A number of tests have proven that this results in optimum pump efficiency. It has further been found that holding to this relationship between R and "c" is of the most importance when maximum diameter impellers (f) are employed.

9. The cross sectional volute radius "e" is conventionally a constant radius equal to the radius of the discharge diameter "b" for the passage of large solids. Tests have shown that the wider volute resulting from using a larger value for the radius "e" results in reducing the pump efficiency. On the other hand, although use of a smaller radius "e" would enhance pump efficiency, it would preclude the passage of large solids.

It is noted that in the embodiment of the pump described, the radius "e" is that of a circle, which geometry was originally set up in the industry under the mistaken belief that a helical flow is superposed on the toroidal flow existing in the pump. However, it has been found that this is not the case and accordingly if a cross-sectional volute shape other than that of a geometric circle is desired for conveniences in casting manufacture or appearance, little performance will be lost so long as the volute width criteria outlined above are maintained. Thus, a somewhat rectangular volute cross section having ample fillets can be employed, or an oval section can be employed, so long as the cross sectional area and volute widths are maintained in accordance with the formulas herein given.

As to the volute severity, which is the radial change in radius of the periphery of the volute per degree of angular position, this quantity "s" is given in Table I. For most applications in pumping the sewage and other solids, this is the optimum value. The selection of a more severe volute flattens and lowers the head-capacity curve, particularly at shut-off and decreases both peak efficiency and efficiency of lower flow rates. On the other hand, changing to a less severe (rounder) volute has similar effects except that here there is a slight increase in low flow rate efficiency.

As previously described, it as been ound under the resent invention that a recessed impeller, vortex pump will have optimum efficiency, a head-capacity curve suitable for the pumping of solids, a low noise level, will require a minimum horsepower for a given head and will maintain a good pressure curve over the operational range of pump speeds.

Although the best mode contemplated for carrying out the present invention has been herein shown and described, it will be apparent that modification and variation may be made without departing from what is regarded to be the subject matter of the invention.

What I claim is:

1. A vortex pump of the type having a casing formed with a generally toroidal peripheral wall and a radial outer wall, an axial inlet for the casing in said radial outer wall, a cylindrical impeller recess formed in the casing opposite said inlet and joining the inner periphery of said volute, an outlet having one wall leading tangentially from said casing and an impeller with peripherally unshrouded vanes projecting axially from a generally radial flange and rotatable in said recess; the improvement wherein the peripheral casing wall is a volute, the entire internal wall of said volute having a smooth arcuate cross sectional shape with corresponding parts of the volute wall having substantially the same shape around the volute, said impeller recess intersecting said volute wall at an obtuse angle substantially around the volute, said intersection lying in a skew plane with the obtuse angle of intersection progressively increasing around the volute from the deepest portion of the volute at the tangential outlet wall to the shallowest, cutwater portion of the volute at the diametrically opposite outlet wall, said radial outer casing wall lying substantially in a plane that is equidistant from the impeller, said impeller having a substantial radial clearance with said recess, the radially inner ends of the impeller vanes being unshrouded to form a recessed eye.

2. The vortex pump of claim 1, wherein said angle of intersection at the cutwater portion of the volute is not substantially less than 180°, rendering the recess wall substantially tangent to the toroidal inner wall of said casing at the cutwater.

3. The vortex pump of claim 1, wherein the radially inner ends of the impeller vanes form substantially continuous lines that extend axially and radially outward from the impeller flange.

4. The vortex pump of claim 2, wherein R is substantially equal to $2c$, where

R = the diameter of the impeller recess in inches, and

c = the radius of the volute at the cutwater in inches.

5. The vortex pump of claim 4, wherein

b = about 2 inches to 10 inches, and

$g = (\sqrt{b}/2) + 1\ 3/8$ inches ±10 percent, where

b = diameter of the casing outlet in inches.

g = axial depth of the impeller vanes in inches.

6. The vortex pump of claim 4, wherein

b = about 2 inches to 10 inches,

$f = 9$ inches $+ 0.625b$ ±10 percent, and

$c = (f + 0.0625b + 0.625\ \text{inch}/2)$ ±10 percent, where

b = Diameter of the casing outlet in inches.

f = Impeller diameter in inches, and

c = Volute radius to the cutwater in inches,

b = Diameter of the casing outlet in inches.

7. The vortex pump of claim 6, wherein

$h = 5$ inches $+ 0.25b$ ±10 percent, where

h = the impeller eye diameter.

8. The vortex pump of claim 4, wherein

b = about 2 inches to 10 inches,

$N = (b + 12.3)\ \sqrt[3]{\tan B/2}$ to the closest whole number, ±10 percent, where

b = Diameter of the casing outlet in inches,

N = Number of impeller blades,

B = Impeller vane discharge angle of about 20° to 75°

9. The vortex pump of claim 8, wherein B, the impeller vane discharge angle, is about 30° – 45°.

10. A vortex pump of the type having a casing formed with a generally toroidal internal wall, an axial inlet for the casing, a cylindrical impeller recess formed in the casing opposite said inlet and joining the inner periphery of said volute, an outlet having one wall leading tangentially from said casing and an impeller with peripherally unshrouded vanes projecting axially from a flange and rotatable in said recess; the improvement wherein the internal casing wall is a volute, the entire internal wall of said volute having a smooth cross sec-

tional shape around the volute, said impeller recess intersecting said volute wall at an obtuse angle substantially around the volute, with the obtuse angle of intersection progressively increasing around the volute from its smallest value at the deepest portion of the volute at the tangential outlet wall to its largest value at the shallowest, cutwater portion of the volute at the diametrically opposite outlet wall, said impeller having a substantial radial clearance with said recess, the radially inner ends of the impeller vanes extending axially and radially outward from said impeller flange and being unshrouded to form a recessed eye, the value of R being substantially equal to $2c$, where

R = the diameter of the impeller recess in inches, and

c = the radius of the volute at the cutwater in inches.

11. The vortex pump of claim **10**, wherein

b = about 2 inches to 10 inches and

$g = (\sqrt{b/2}) + 1\ 5/8$ inches ±10 percent, where

b = diameter of the casing outlet in inches.

g = axial depth of the impeller vanes in inches.

12. The vortex pump of claim **10**, wherein

b = about 2 inches to 10 inches,

f = 9 inches + $0.625b$ ±10 percent, and

c = $(f + 0.0625b + 0.625\ \text{inch}/2)$ ±10 percent, where

b = Diameter of the casing outlet in inches.

f = Impeller diameter in inches, and

c = Volute radius to the cutwater in inches,

b = Diameter of the casing outlet in inches.

13. The vortex pump of claim **12,** wherein

h = 5 inches + $0.25b$ ± 10 percent, where

h = the impeller eye diameter.

14. The vortex pump of claim **10,** wherein

b = about 2 inches to 10 inches,

N = $(b + 12.3)\ \sqrt[3]{\tan B/2}$ to the closest whole number, ±10 percent, where

b = Diameter of the casing outlet in inches,

N = Number of impeller blades,

B = Impeller vane discharge angle of about 20° to 75°

15. The vortex pump of claim **14,** wherein B, the impeller vane discharge angle, is about 30° – 45°.

* * * * *

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,759,628 Dated September 18, 1973

Inventor(s) DENNIS D. KEMPF

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Abstract, last line, change "of" (second occurrence) to -- to --.

Col. 1, line 34 after "Draeger" delete "has found July present 18, 1961" and insert -- 2,992,617, July 18, 1961 -

Col. 3, line 24, change "c" to -- axial --.

Col. 5, line 52, change "progresing" to -- progressing --.

Col. 6, line 1, change "distnace" to -- distance --.

Col. 7, line 8, change "C" to (lower case) -- c --.

Col. 7, line 11, change "$\sqrt{\tan \frac{B}{2}}$ to -- $-\sqrt[3]{\tan \frac{B}{2}}$ --.

Col. 7, line 13, change "$\sqrt{b/2}$" to -- $\frac{-\sqrt{b}}{2}$ --.

Col. 10, line 34, change "$(-\sqrt{b/2})$" to -- $\frac{-\sqrt{b}}{2}$ --.

Col. 11, line 18, change "$(-\sqrt{b/2})$" to -- $\frac{-\sqrt{b}}{2}$ --.

Signed and sealed this 15th day of October 1974.

(SEAL)
Attest:

McCOY M. GIBSON JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents