United States Patent

Rumsey

[15] **3,694,101**

[45] **Sept. 26, 1972**

[54] **REENTRY CENTRIFUGAL PUMP/MIXERS**

[72] Inventor: **Rollin Douglas Rumsey,** 148 Summer St., Buffalo, N.Y. 14222

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[21] Appl. No.: **112,865**

[52] **U.S. Cl.** **415/52**
[51] **Int. Cl.** **F01d 1/12**
[58] **Field of Search** 415/52–56

[56] **References Cited**

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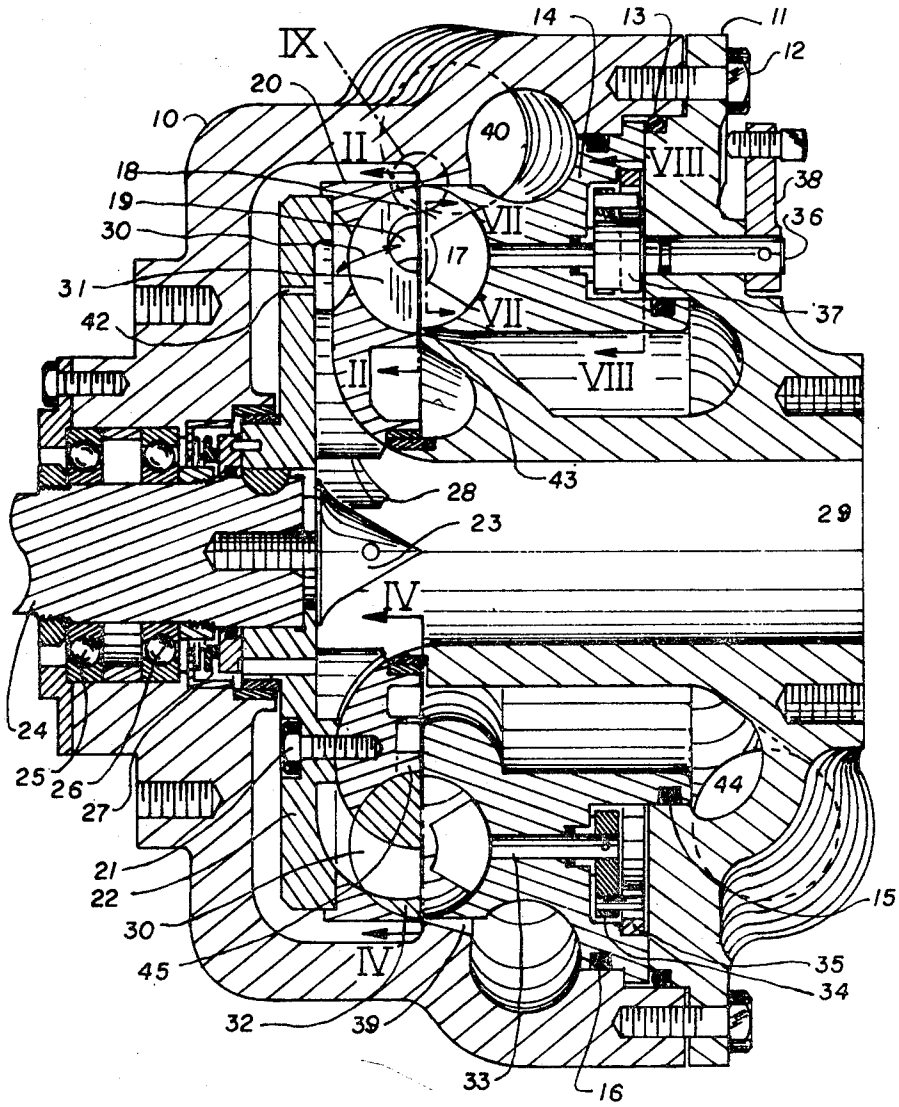
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Primary Examiner—C. J. Husar
Attorney—R. Douglas Rumsey

[57] **ABSTRACT**

A reentry centrifugal pump has within its housing a toroidal working chamber one-half of which is stationary and the other half of which is blade filled and contained in a rotor coupled to a shaft. Fluid inlet to the working chamber is along the inside periphery on one version and from the blades at the centerline of the toroid (the fluid vortex center) on another or both. When more than one inlet is employed, extremely efficient mixing of the fluids results. Provision is made to prerotate the inlet fluid using a conventional centrifugal pump runner in order to avoid inlet shock. Fluid outlet from the working chamber is provided along the outside periphery to a diffusor on one embodiment and through the stationary casing adjacent to an abutment dam blocking the stationary housing portion of the toroidal working chamber on another.

12 Claims, 18 Drawing Figures



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SHEET 1 OF 2

FIG. 1

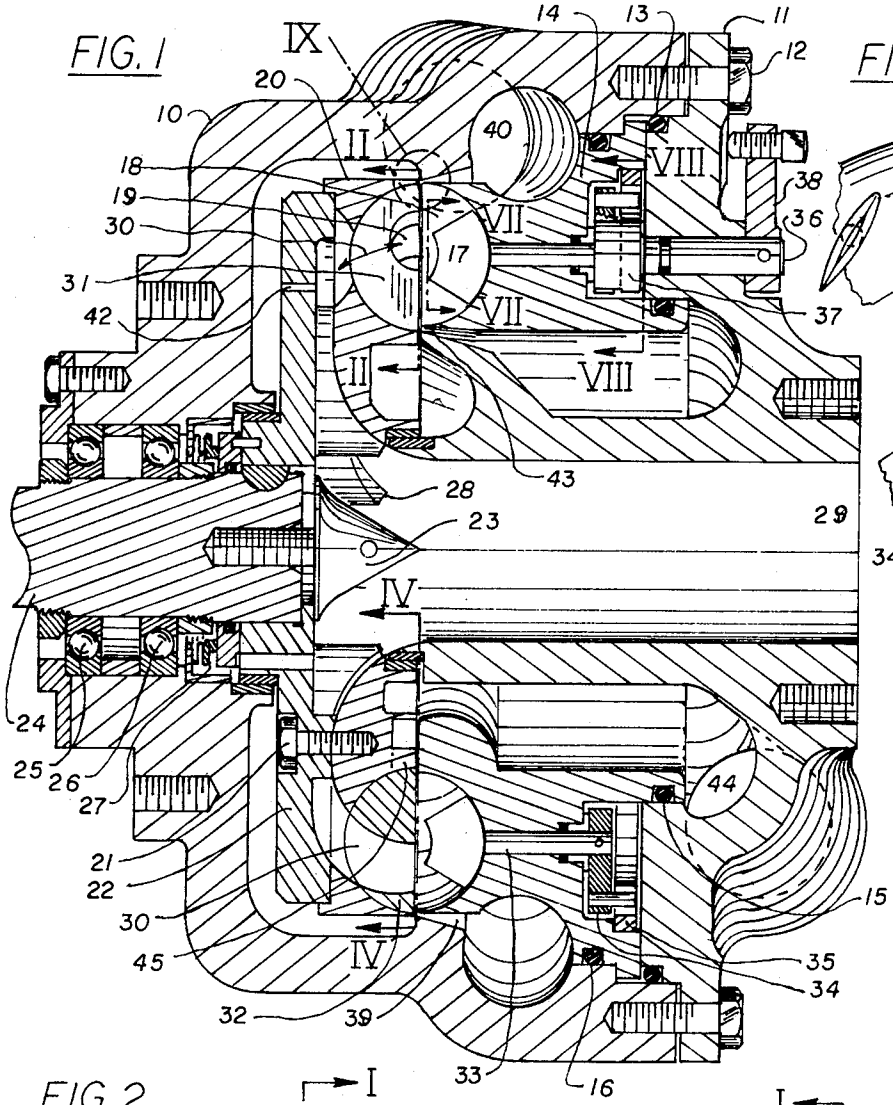


FIG. 7

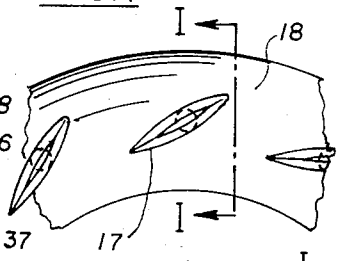
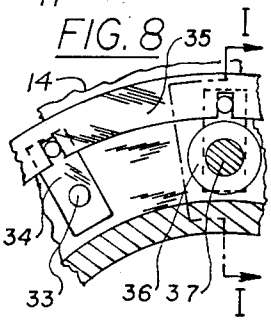


FIG. 8



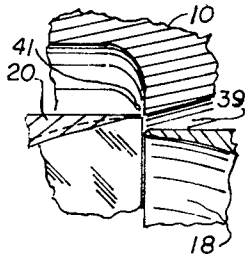
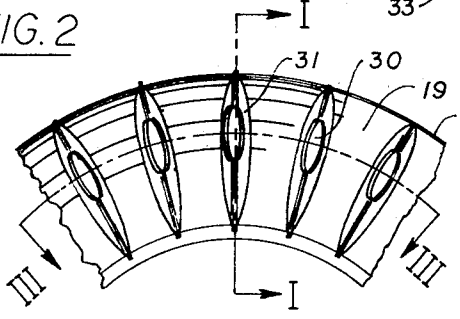


FIG. 9

FIG. 2



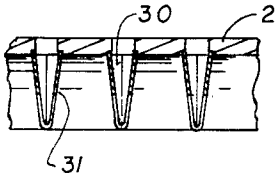


FIG. 3

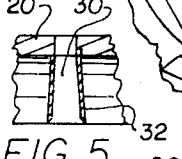


FIG. 5

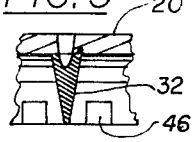


FIG. 6

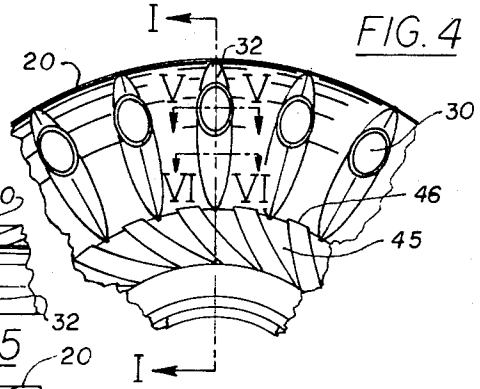


FIG. 4

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FIG. 10

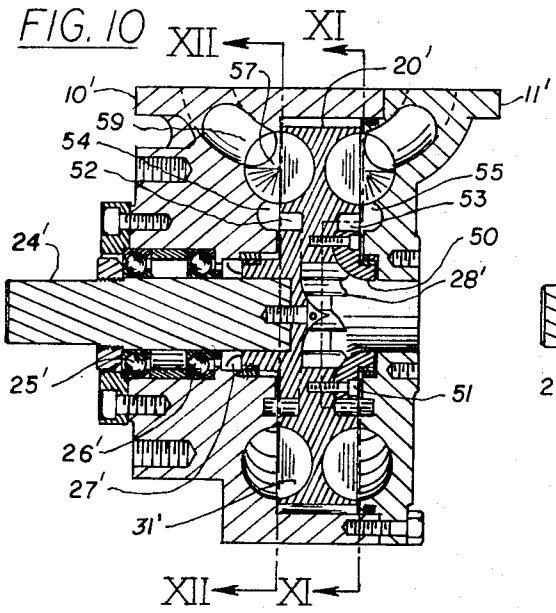


FIG. 17

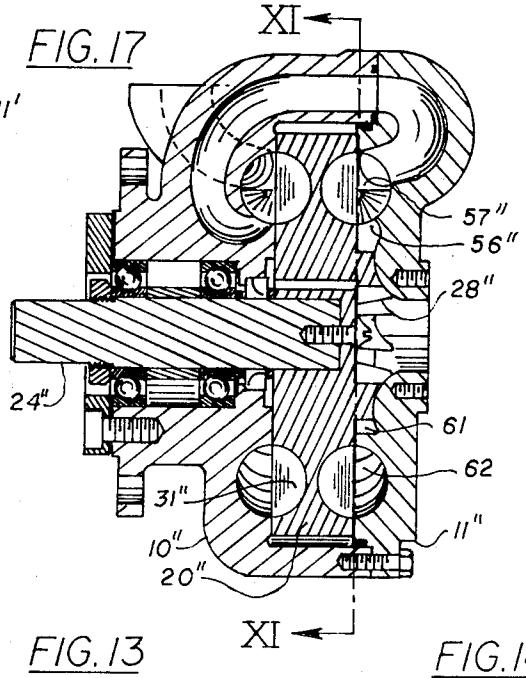


FIG. 11

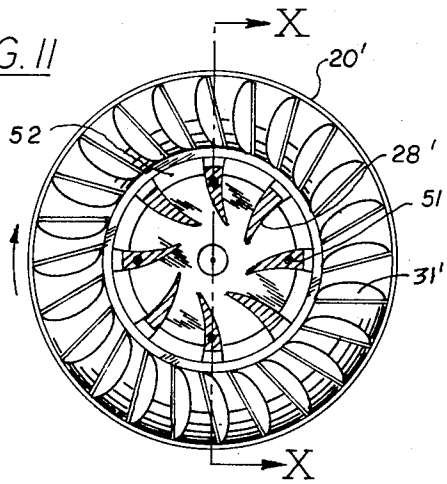


FIG. 13

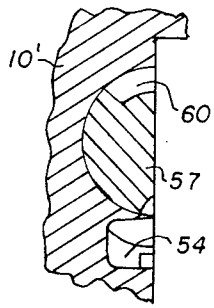


FIG. 14

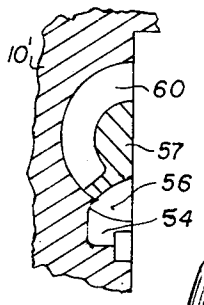


FIG. 16

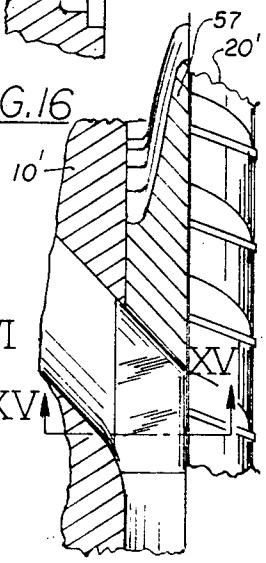


FIG. 15

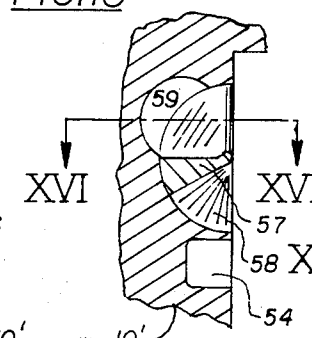
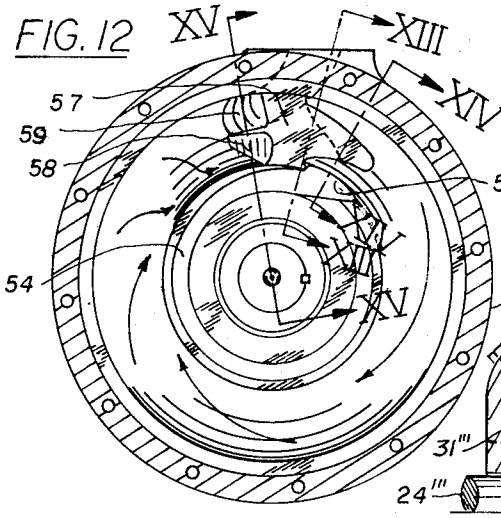


FIG. 12



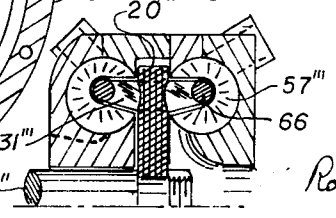


FIG. 18

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REENTRY CENTRIFUGAL PUMP/MIXERS

This invention relates to fluid pumps generally known as reentry turbine or regenerative vortex pumps. Such pumps have the unique property of developing extremely high pressures compared to standard centrifugal pumps, often approaching the characteristics of positive displacement pumps.

The prior known reentry and side channel pumps of the general type under consideration contain a bladed rotor operating within a generally toroidal working chamber fitted at one point with an abutment or dam on either side of which are located the inlet and outlet ports. During operation the rotor establishes a whirl motion in the working fluid beginning at the inlet and increasing in intensity toward the outlet. When this highly energized whirl flow reaches the abutment it is abruptly arrested and forced out the discharge at reduced velocity and increased pressure. Such pumps suffer severe shock losses at the outlet, turbulence losses in the blading channels while passing the dam and severe blading shock losses at the inlet. All of which contribute to the generally low efficiency of such pumps — on the order of 20 to 30 percent overall efficiency.

It is accordingly an object of the present invention to depict a number of improved embodiments that overcome the above mentioned deficiencies and provide means for reducing shock and turbulence losses; thereby increasing either or both delivery head and efficiency.

The technology of centrifugal pumps is well founded and results in overall efficiencies as high as 90 percent in large sizes and up to 60 to 80 percent in small units. Careful attention to proper blading angles versus flow have produced these efficiencies, where stall and turbulence are minimized by the use of proper channel sizes, shapes, and blading. It is a part of this invention to apply some of these principles to the reentry pump to improve its performance.

It is a common need in many industries to blend and thoroughly mix two or more fluids. One of the objects of the present invention to demonstrate a method whereby two or more inlets for different fluids may be provided for the pump whereby highly efficient mixing action may be accomplished within the pump; thereby eliminating the need for separate mixing and pumping mechanisms.

A further object of the present invention is to provide means whereby a single rotor reentry pump with vanes on either side of the rotor may be adapted to operate with both sides in parallel for Maximum flow volume or, by replacement of some parts, in series for maximum pressure generation.

Other objects, features and advantages of the present invention will be readily apparent from the following detailed description of certain preferred embodiments thereof taken in conjunction with the accompanying drawings in which:

FIG. 1 is a longitudinal developed sectional detail view through a pump-mixer embodying features of the invention, and taken along substantially the centerline according to line I—I of FIGS. 2, 4, 7, and 8.

FIG. 2 is a fragmentary elevational view of the rotor blading taken along a line II—II of FIG. 1.

FIG. 3 is a detail section of the rotor viewed along line III—III of FIG. 2.

FIG. 4 is a fragmentary elevational view of an alternate rotor blading including prerotation inlet vanes as taken substantially along line IV—IV of FIG. 1.

FIG. 5 is a fragmentary detail sectional view of a rotor blade taken along line V—V of FIG. 4.

FIG. 6 is a fragmentary detail sectional view of a rotor blade taken along line VI—VI of FIG. 4.

FIG. 7 is a fragmentary elevational view of moveable stator vanes taken along line VII—VII of FIG. 1. FIG. 8 is a fragmentary elevation and sectional view of the stator vane actuation linkage taken along line VIII—VIII of FIG. 1.

FIG. 9 is an enlarged fragmentary detail section view of the diffusor inlet, covering substantially the area IX of FIG. 1.

FIG. 10 is a longitudinal developed sectional detail view through a double sided rotor parallel operating pump according to the invention taken substantially along a line X—X of FIG. 11.

FIG. 11 is a sectional and elevational view of and through the impeller of the pump of FIG. 10 or FIG. 17 taken substantially along a line XI—XI of FIG. 10 and FIG. 17.

FIG. 12 is a sectional and elevational view of the stator housing and abutment dam of the pump of FIG. 10 taken along a line XII—XII.

FIG. 13 is an enlarged fragmentary detail sectional view of the abutment and inlet taken along line XIII—XIII of FIG. 12.

FIG. 14 is an enlarged fragmentary detail section along line XIV–XIX of FIG. 12.

FIG. 15 is an enlarged fragmentary detail sectional view of the outlet taken on line XV—XV of FIG. 12.

FIG. 16 is an enlarged fragmentary developed sectional detail view of the abutment taken substantially along line XVI—XVI of FIG. 15.

FIG. 17 is a longitudinal developed sectional detail view along substantially the centerlines of a double sided rotor two stage (series operating) pump according to the invention taken substantially along a line corresponding to X—X of FIG. 11.

FIG. 18 is a fragmentary longitudinal sectional view through a double sided rotor pump having a reduced blade size and vortex core rod tieing the blades together.

The accompanying drawings are essentially to scale of which FIG. 1 depicts a reentry mixing pump having dual or triple inlets in which cup shaped housing 10 in combination with flanged member 11 held together by bolts 12 form a closure sealed by O-ring 13, said parts clamping between them stator housing 14 which is independently sealed by O-rings 15 and 16 and contains moveable stator vanes 17 located in semi toroidal stator channel 18 in close abutting running clearance with a similar semitoroidal rigidly mounted vane filled channel 19 incorporated in rotor 20 which is attached by suitable means such as bolts 21 to flanged disc member 22 which in turn is mounted by screw 23 on drive shaft 24 which protrudes from aforesaid housing 10 and is supported by bearings 25 and 26 and sealed by sealing mechanism 27.

It is well known in the centrifugal pump art that in order to avoid stalling of the inlet vanes and prevent cavitation as well as excessive turbulence losses, that the vane angle should conform to the fluid flow lines within 0° to 10° and that normal design practice would

require a positive angle of attack on the order of 3° to 6° at the design flow and speed condition. On reentry pumps past practice has been to favor the blading configuration for discharge and regenerative considerations with the result that the inlet blade angle to fluid flow line is on the order of a 90° blade angle of attack. Therefore, stalling, early cavitation and excessive turbulence have been normal limitations of such pumps, which in part explains their low overall efficiency.

It will be noted that in FIG. 1, an essentially standard centrifugal pump impeller blading 28 is arranged on disc 22 which serves to prerotate the incoming fluid from inlet 29 and imparts considerable pressure as well as velocity to the fluid by the time it enters the regenerative or reentry section of the pump into which it may be effectively introduced through passages 30 in the vanes in such a manner that it reaches the toroidal section at the center and lowest pressure portion of the whirl vortex generated therein.

Two forms of rotor vanes including inlet passages therein are depicted in FIG. 1, vanes 31 in the upper portion constructed for example of sheet metal are seen further in views of FIG. 2 and 3. In the lower portion of FIG. 1 cast vane elements 32 are illustrated and further shown in FIGS. 4, 5, and 6. Vanes such as 32 could similarly be integrally molded in plastic or cast with the impeller using an investment casting technique.

When the fluid reaches the toroidal chamber it is flung radially outward by the rotor vanes under the action of centrifugal force and as it crosses into the stationary half of the tore it slows down slightly and returns radially inwardly until it reengages the rotor vanes. This action builds up until the input energy rate and friction balance — that is under no flow conditions. One of the problems on existing reentry pumps is the rate at which the whirl builds up under conditions of high flow. Unless there is high whirl velocity the delivery pressure will not be high and as a result many such units have a straight line head-capacity characteristic curve. In order to achieve rapid whirl build up and at the same time control it at a fixed level, adjustable guide vanes as seen at 17 in FIG. 1 and 7 are desirable. Such vanes may be mounted on a shaft 33 on the outboard end of which is mounted a pin carrying lever 34 engaging slots in a synchronizing ring 35. One of said levers may be straddled by a fork 36 formed in shaft 37 and actuated by adjusting lever 38.

Fluid leaving the toroidal working chamber is collected in annular diffusor 39 and conducted by a volute to discharge port 40, an enlarged view of the diffusor entrance being seen in FIG. 9. In order to achieve maximum conversion of velocity head to pressure in the diffusor; it is desirable to bleed off the boundary layer fluid, emerging from the rotor, into clearance 41 and return it to the inlet through controlled size port 42 in flanged disc 22.

It has previously been pointed out that a high whirl velocity is present in the toroidal chamber, the velocity of which in the embodiment of FIG. 1 can be controlled by vanes 17. The stationary half of the toroidal working chamber may be made slightly smaller than the rotor section such that the whirl velocity can induce fluid into the inside diameter clearance between them as at 43 thus providing a secondary inlet for blending a second fluid with the primary working fluid, which secondary fluid may be introduced into the machine through inlet port 44.

Inasmuch as the above described secondary fluid inlet suffers from the lack of prerotation, hence would exhibit high shock losses unless its flow rate was very low, a preferred secondary inlet provision is shown in the lower portion of FIG. 1 and in FIG. 4 where prerotation impeller vanes 45 are provided on the forward face of the rotor, and exhaust into the toroid between its vanes as at 46 in FIGS. 4 and 6.

It may be appreciated that the induced secondary inlet 43 could be employed in addition to the inlets 46 if it were desireable to simultaneously mix three separate fluids.

A alternate design embodiment of the invention is shown in FIG. 10 wherein housing 10′ and closure 11′ each contain stationary partial toroids with a double sided rotor 20′ located between them, mounted on shaft 24′, journalled in bearings 25′ and 26′ and sealed by member 27′. It will be noted that the rotor again is equipped with centrifugal pump type inlet vanes 28′ which are partially shrouded by cover plate 50 attached to the impeller rotor by screws 51. In this embodiment the flow splits between the vanes to enter twin diffusor channels 52 and 53 and thence volutes 54 and 55, which in the case of 54, leads into the torroidal working chamber 18′ through inlet 56 located adjacent to abutment dam 57 shown in FIG. 12. For stable operation without pulsation and for maximum pressure conversion within the pump, it is desirable to radially slope the vanes rearward and axially forward as shown in FIG. 11. Such an arrangement also reduces inlet shock and facilitates machining, although such rotors end themselves particularly well to plastic molding.

In reentry pumps using an abutment, although a whirl flow exists in the fluid working toroidal chamber, its intensity increases around the periphery from inlet to outlet, as shown by the flow arrows in FIG. 12; it is therefore highly desireable that the entering and leaving fluid flows conform as closely as practical to this whirl pattern; hence, the face of the abutment facing the departing fluid should employ a helical face 58 to aid the fluid in its path to the outlet port 59 refer FIGS. 12 and 15, and the inlet side of the abutment should also employ a helical face or higher pitch and / or a smaller and gradually increasing area. A configuration employing gradually increasing channel area to match the circumferential flow input rate is illustrated in FIGS. 12 through 16 where channel 60 FIG. 13 is quite small at a point where inlet 56 is just beginning to open into the working chamber area and in FIG. 14 said channel has reached approximately half the total area at a point where half of the inlet 56 area is exposed, refer FIG. 12. It will be noted that by positioning the dam in the center of the center of the tore channel at the inlet, the maximum differential centrifugal head with minimum turbulence is available to initiate the whirl flow.

A second alternate embodiment of the invention is shown in FIG. 17 arranged for series or two stage operation wherein housing member 10″ mates with closure 11″ to enclose double sided impeller 20″ mounted on shaft 24″. In this instance, inlet impeller vanes 28″ are mounted on the rotor outside face and

direct the fluid radially outward to volute **61**; thence into the first working chamber **62** through inlet **56''** adjacent to abutment **57''**, thence around the unit and out through outlet **63** into second inlet **64** and second working chamber **65** where the second stage generated pressure is added to the first and exhausted at **40''**. Since this embodiment is hydrostatically unbalanced in an axial direction, extremely high capacity thrust bearing provision must be made.

Although the rotor blades are illustrated as occupying one-half (180°) of the working chamber, since all the fluid trapped between the blades is returned to the inlet side of the pump, in some instances it is desirable—particularly at low flow rates and on compressible fluids—that the vanes be considerably shorter, for example preferably as shown in FIG. **18** where a hoop member **66** attached to the blade tips forms the vortex core and if hollow could form the inlet; alternatively the inlet could be similar to FIG. **10** or **17**. In this embodiment, abutment **57''** would preferably utilize helical faces on either side, of low angle or pitch on the exit side and high angle on the inlet. The rotor could conveniently be manufactured using laminations as shown.

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

I claim:

1. In a reentry centrifugal pump having a housing defining a generally toroidal fluid working chamber therein and a shaft concentric with said chamber and extending outward from said housing:

a rotor within said housing having at least one cavity formed therein, said cavity forming a portion of said toroidal working chamber;

a plurality of circumferentially spaced blades extending from said rotor into said fluid working chamber;

a stationary abutment dam at one location in said working chamber, blocking at least half the area of said chamber including essentially all the area not swept by said blades;

a dual thread helical face on said abutment facing the approaching blade direction wherein said helical surfaces are phased one from the other such that when the first of said helical surfaces is in alignment with a blade surface in the radial outer portion of the working chamber, the second helical surface will be in alignment with the same blade surface in the inner portion of the working chamber and where the pitch of said helical surfaces approximates the helix angle of the fluid whirl near the discharge from the pump at the pump's design flow rate;

a divergent discharge passage extending from said working chamber to the exterior of aforesaid housing, said passage being essentially tangential to said chamber at its outer portion and in line with and an extension of said first helical surface on said abutment;

a circumferential surface on said abutment in close running sealing engagement with the exposed portion of said rotor blades, said surface being essentially one blade spacing in circumferential length;

a partial circular bent cone or horn portion essentially centered within the working chamber extending in the direction of rotation from said abutment, a relatively high angle helical face surface surrounding said horn and beginning at the end of the outer portion of said abutment circumferential sealing surface;

inlet passage means from the exterior of said housing opening into the radial inner portion of said working chamber, commencing at the end of the inner portion of said abutment circumferential sealing surface and extending approximately three blade spacings away from said abutment in the direction of rotation.

2. A reentry pump according to claim **1**, wherein said inlet passage means extend from the exterior of said housing into engagement with said rotor near its center, radial outward extending impeller vanes on said rotor forming fluid channels therebetween connecting to said inlet means, an annular volute joining the radial outward portion of said fluid channels and leading to the aforesaid inner portion of said working chamber adjacent to said abutment dam and extending from said abutment in the direction of rotation.

3. A reentry pump according to claim **1**, fluid prerotation vane means on said impeller between said exterior inlet and said working chamber inlet opening.

4 A reentry pump according to claim **1**, conduit means joining said discharge passage means from one toroidal working chamber on one side of said rotor to a second radially inward inlet to a second toroidal working chamber on the other side of said rotor and a second discharge passage to said housing exterior for two stage operation.

5. In a reentry centrifugal pump mixer structure having a housing defining a generally toroidal fluid working chamber therein and a shaft concentric with said toroid extending therefrom:

a rotor within said housing having a cavity formed therein, said cavity forming a portion of said toroidal working chamber;

a plurality of circumferentially spaced vanes extending from said rotor into said working chamber:

a plurality of inlet port means in said housing and fluid communication means connecting said port means to the generally inner portion of said toroidal working chamber;

a discharge port in said housing with a flow channel connecting said port to the outer portion of said toroidal working chamber;

fluid seal means and bearing support means between said shaft and said housing.

6. A pump mixer according claim **5**, sealing block in said working chamber essentially one vane spacing in circumferential length and blocking all of said working chamber area not swept by said vanes.

7. A fluid pump mixer according to claim **5** wherein one of said inlet port fluid communication means extend within the rotor from near its center, radially outward into the interior of aforesaid vanes and exiting from said vanes into said toroidal working chamber.

8. A fluid pump mixer according to claim **7** wherein said inlet fluid communication means exit from said vanes near the fluid vortex center in said toroidal working chamber.

9.

In a reentry centrifugal pump mixer structure having a housing defining a generally toroidal fluid working chamber therein and a shaft concentric with said toroid extending therefrom:

a rotor within said housing having a cavity formed therein, said cavity forming portion of said toroidal working chamber;

a plurality of circumferentially spaced vanes extending from said rotor into said working chamber;

inlet port means in said housing and fluid communication means connecting said port means to passages within said vanes thence to said toroidal working chamber;

a discharge port in said housing with a flow channel connecting said port to said toroidal working chamber;

fluid seal means and bearing support means between said shaft and said housing.

10. A fluid pump according to claim **9**, a second inlet port in said housing and fluid connection means from said second inlet port to the radially inward portion of said toroidal working chamber.

11. A fluid pump according to claim **9**, wherein said inlet fluid communication means exit from said vanes near the fluid vortex center in said toroidal working chamber.

12. A fluid pump according to claim **9**, wherein said vanes are generally semi-circular in shape and said inlet fluid communication means enter said working chamber near the center of said semi-circles.

* * * * *