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Onal

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[54] **CENTRIFUGAL PUMP APPARATUS**

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[52] **U.S. Cl.** .... **415/74,** 415/98, 415/143, 415/169 A, 415/405

[51] **Int. Cl.** **F04d 1/04,** F04d 13/04

[58] **Field of Search** .... 415/97, 98, 74, 143, 213, 415/169 A, 213; 416/186, 176; 417/407

[56] **References Cited**

UNITED STATES PATENTS

272,595	2/1883	Smith	415/98
367,564	8/1887	Wade	415/74
855,809	6/1907	Rateau	415/98
1,080,656	12/1913	Richardson	415/98
1,169,266	1/1916	Krogh	415/169 A
1,586,978	6/1926	Dorer	415/143
2,661,698	12/1953	Schellens	415/98
3,221,661	12/1965	Swearingen	415/143
3,457,869	7/1969	Janetz	415/98
3,671,138	6/1972	Ball	415/98

FOREIGN PATENTS OR APPLICATIONS

970,265	9/1958	Germany	415/143

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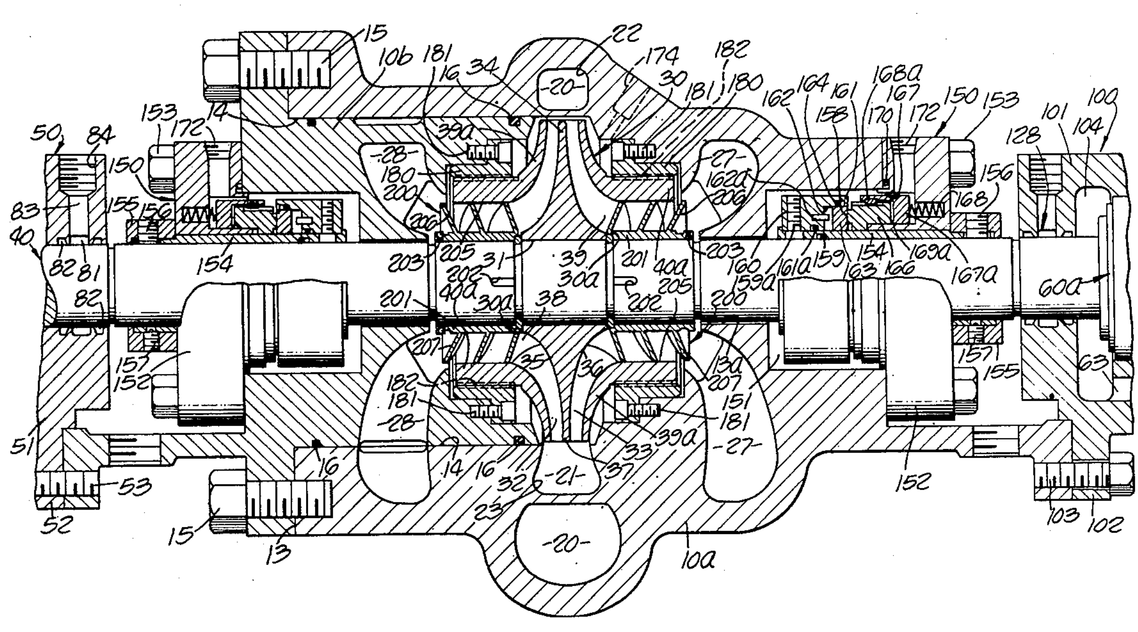
[57] **ABSTRACT**

A high speed, high capacity centrifugal pump apparatus having a radially split pump case, a pump shaft extending axially within the pump case and adapted for high speed rotation therein, and an enclosed impeller with Francis type vanes on each side thereof mounted to said pump shaft for rotation therewith within said pump case. The pump case is provided with a divided inlet flow passage for communicating liquid from a suction line to the center eye portion of the impeller on each side thereof and an outlet flow passage with two volutes extending about the periphery of the impeller for receiving the discharge of liquid therefrom and communicating this liquid to a discharge line. An inducer element operably connected to the pump shaft adjacent each side of the impeller moves the liquid from the inlet flow passage toward the vanes at each center eye portion thereby maintaining a positive pressure on the vanes during high speed operation of the pump to reduce the problems associated with cavitation.

24 Claims, 6 Drawing Figures



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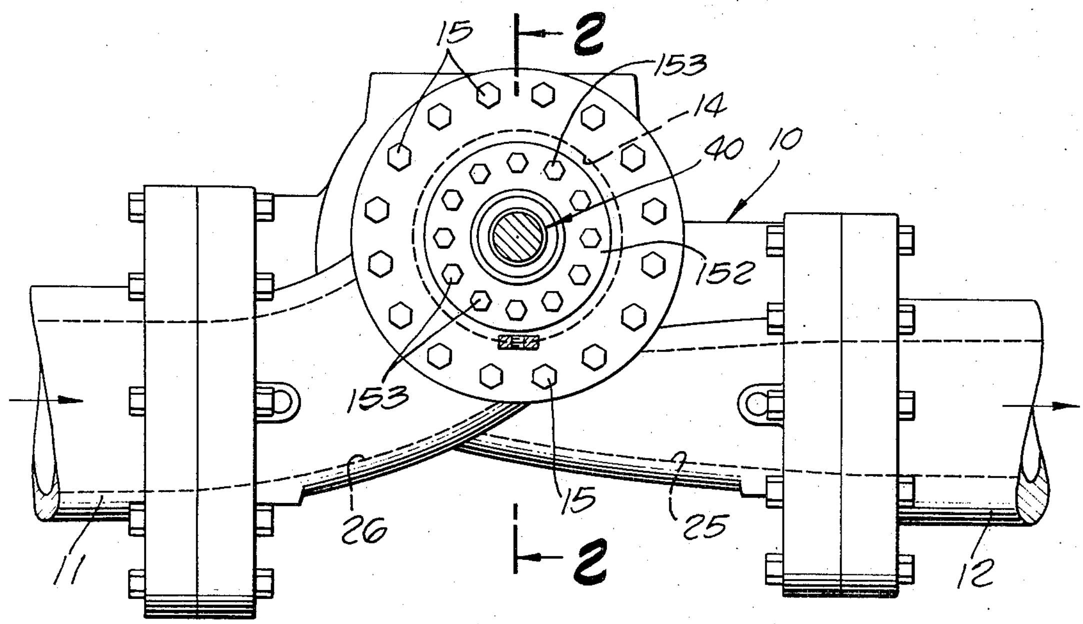

FIG. 1.

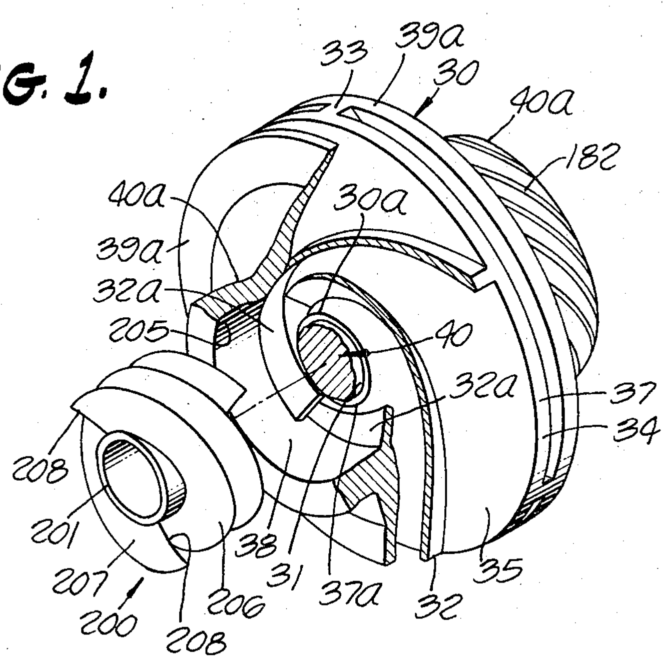

FIG. 3.

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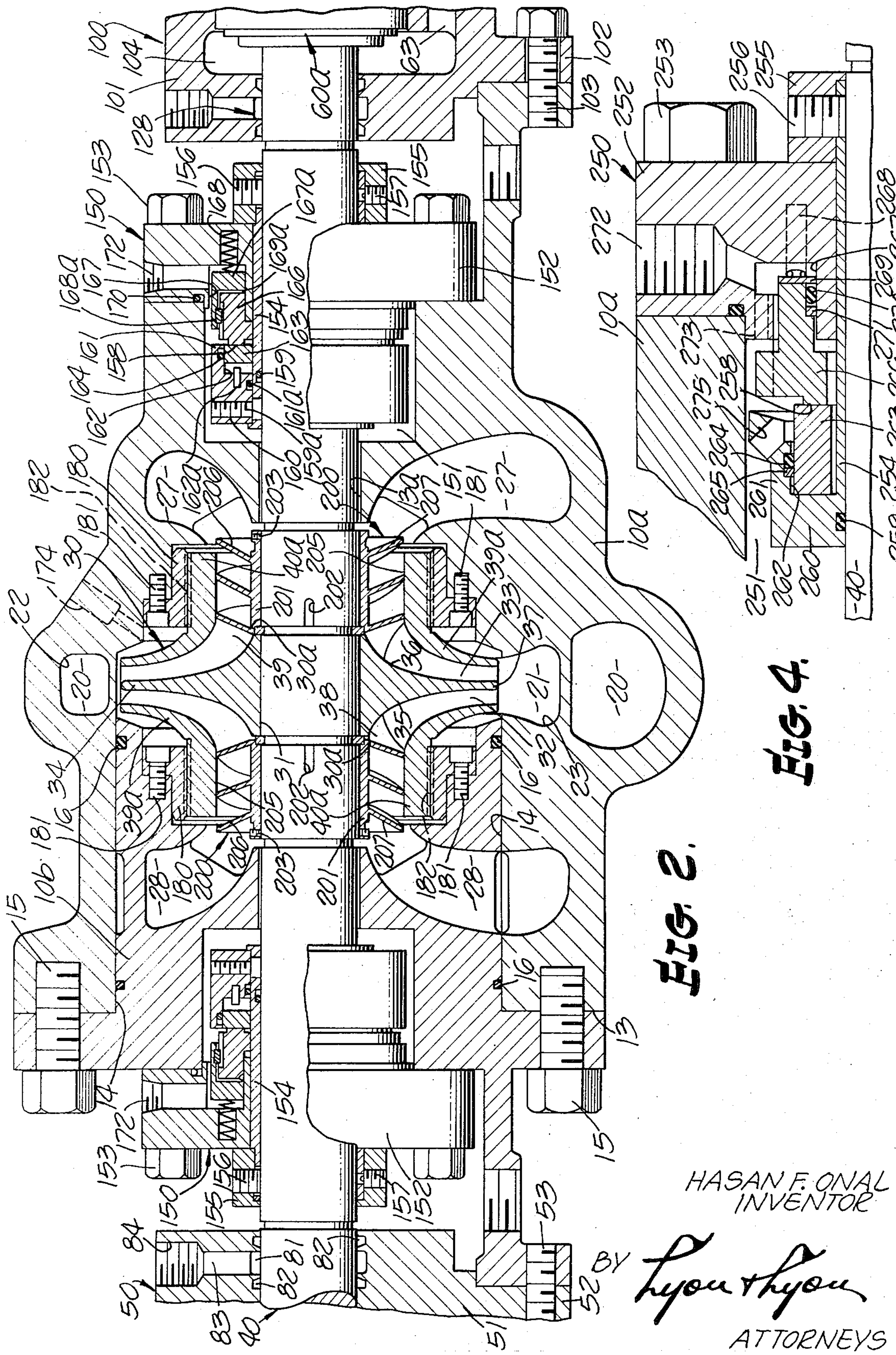

FIG. 2.

FIG. 4.

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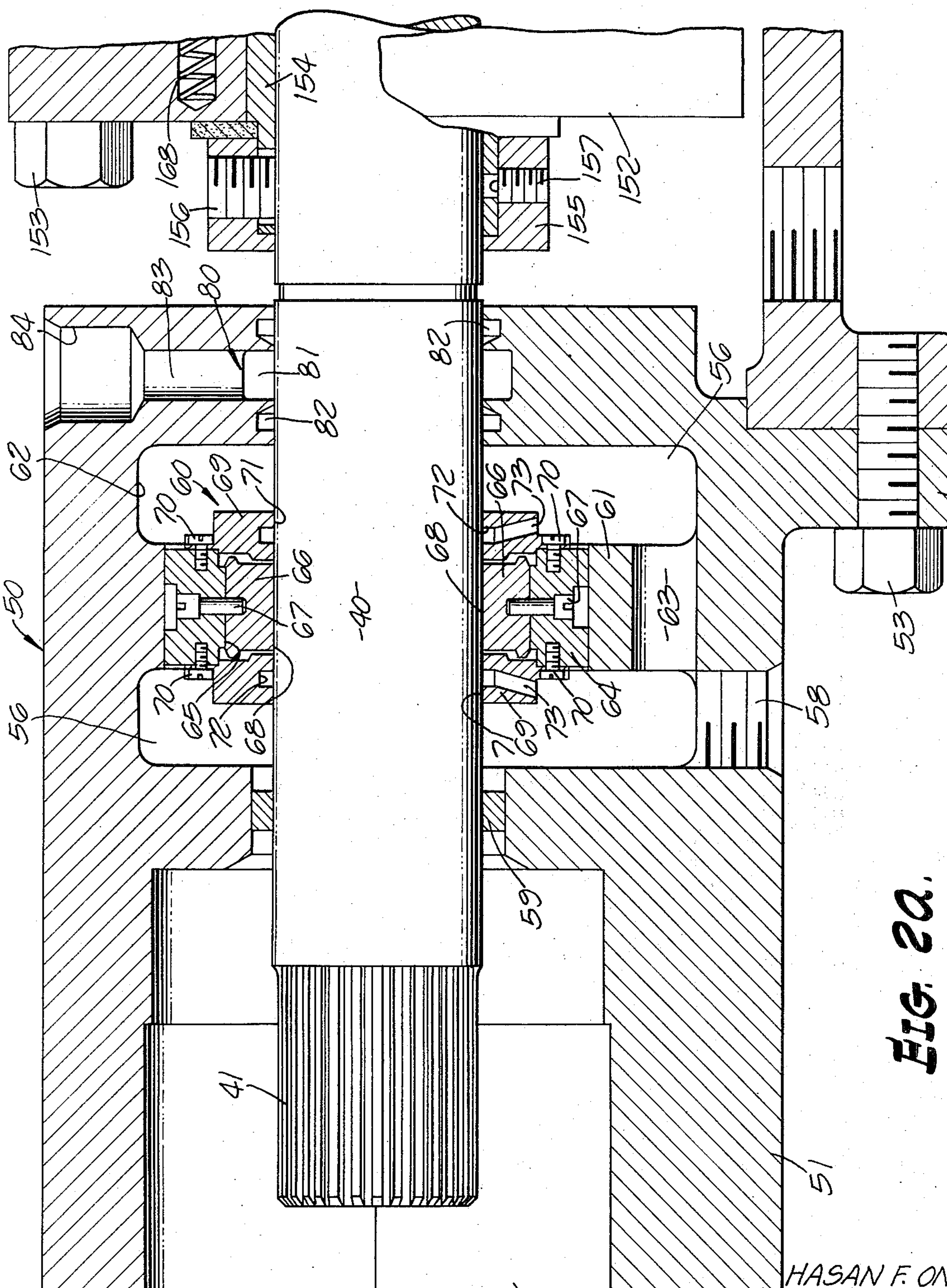

FIG. 2a.

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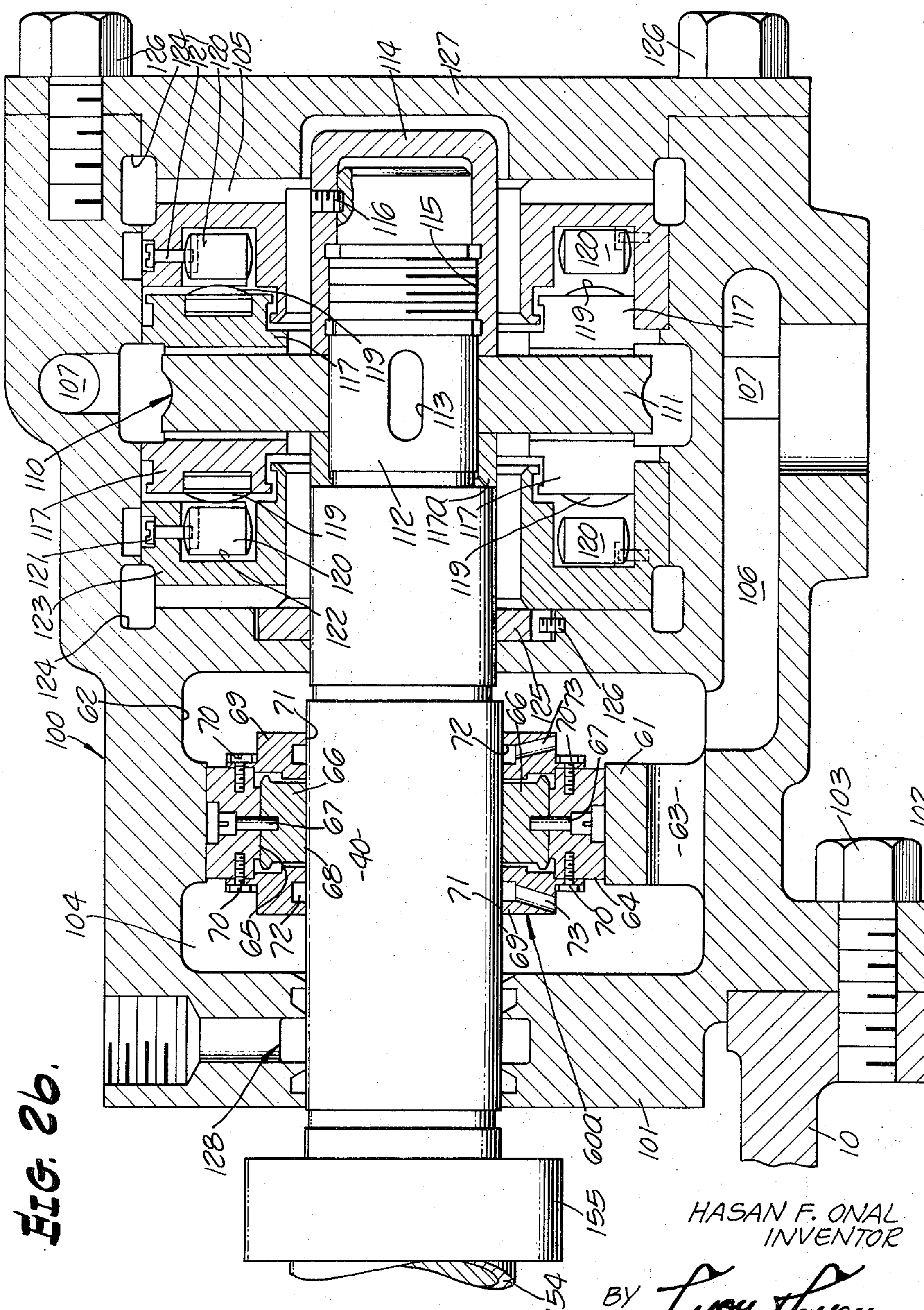

FIG. 26.

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CENTRIFUGAL PUMP APPARATUS

This invention relates to a centrifugal pump and more specifically relates to an efficient single stage centrifugal pump particularly adapted for high-flow and high-head applications.

Generally speaking, centrifrigal pumps are made up of two basic elements; one stationary, the other rotating. The function of the stationary element is to provide a suitable support and enclosure for the rotating element; it generally consists of a pump case, packing or stuffing boxes, seals, and bearings. The rotating element generally consists of a shaft on which is mounted a wheel fitted with vanes and known as the impeller. The rotating element converts the mechanical power input of the pump driver to kinetic energy and imparts this energy to be the liquid. The kinetic energy is then converted to potential energy or pressure energy in a gradually widening passage or so called volute of the pump case.

The method conventionally used in pumps of this type to produce high heads involves the employment of additional impellers in series. In such a series or multistage arrangement, the heads produced by each of the impellers acting in series are additive. However, there are several disadvantages associated with multi-stage pumps. Multistage pumps require an axially split case. At pressures above 1,000 psi it becomes difficult in such a case to prevent leakage and erosion across the face of the axial joint. This is particularly true after a pump has been in service for some time and has been dismantled for maintenance. To combat this, the practice in high pressure multi-stage pumps is to use an outer case or cylindrical case which in effect envelops the axial split case. The employment of the cylindrical case is expensive and adds substantially to the overall weight of the pump. Moreover, a multi-stage pump is very difficult to balance and the weight of the multiple impellers results in a significant deflection of the pump shaft. Shaft deflection and unbalancing create vibrations during the operation of the pump and cuase appreciable load to be applied to the pump bearings thereby increasing maintenance requirements and substantially decreasing the useful life of the pump.

The pump of the present invention in order to avoid the problems of multi-staging employs an entirely different method for producing the desired high heads. This method involves an increase in the speed of the rotating element or more specifically an increase in impeller speed to speeds which exceed 20,000 rpm. This method has been employed successfully in low flow centrifugal pumps, but prior to this invention it has been considered impractical for high flow centrifugal pumps.

A major disturbing factor at such high speeds is cavitation. The term cavitation is used loosely to describe the formation and violent collapse of vapor or of vapor and gas bubbles formed within the liquid as a consequence of extreme reductions in the absolute static pressure. In centrifugal pumps this reduction in pressure occurs at the suction side of the impeller and for satisfactory operation of any centrifugal pump operating at a particular speed and capacity there is a minimum pressure requirement at this point generally expressed in terms of net positive suction head (NPSH). Present centrifugal pumps operating at speeds up to 3,600 rpm generally operate free of cavitation, however, at higher speeds the required NPSH becomes difficult to maintain. At the speeds suggested for the present pump it was heretofore felt that the problems of cavitation would be extreme, causing destructive pitting and a substantial decrease in the efficiency of the pump.

Another factor which has dictated against the use of a high speed impeller involves the stresses acting on the impeller due to the centrifugal forces which would be generated by such speeds. Similar concern has also been expressed with regard to the stresses acting on the pump case when high heads are to be produced. It had also been anticipated that the heat generated by an impeller operating at speeds up to and exceeding 20,000 rpm would create other problems of considerable magnitude, including the problem of parts freezing or fusing together after a period of operation.

Therefore, it is a primary object of this invention to provide a single stage centrifugal pump of unique and novel design which will operate efficiently at high impeller speeds to produce high flow and high heads. In accordance with this and other objects, the centrifugal pump of the present invention briefly comprises a stationary element and a rotating element, the parts of each substantially being constructed of a material having high strength and corrosion resistant characteristics. The stationary element, which supports and houses the rotating element, includes a radially split pump case having two similar passsages or volutes with their outlets 180° apart. The wheel or impeller of the rotating element is enclosed, having a shroud or sidewall on each side of the vanes, and is designed for double suction, having inlets on each side thereof. The vanes of the impeller are Francis type vanes having a helical surface and an entrance angle which changes with radii and section. Adjacent each inlet and secured to the shaft is an inducer component having spiral blades which extend radially from the shaft and which feed the liquid being pumped to the impeller vanes.

Presently, in those instances where a high speed turbine is used to drive a centrifugal pump, it is necessary to couple any pump of the type heretofore used to the turbine through a gear box which reduces the rpm generated by the turbine. Such a gear box is expensive and require constant maintenance. Therefore, it is a further object of this invention to provide a centrifugal pump which can be directly coupled through the shaft to a high speed turbine.

Another object of this invention is to provide a high flow, high head centrifugal pump of such design to minimize the material of construction without sacrifice of either structural strength or stability.

Other and further objects and advantages of this invention will be made readily apparent from the following detailed description and the accompanying drawings wherein:

FIG. **1** is a side view of the pump of the present invention illustrating the pump case and its connection with the suction and discharge lines.

FIG. **2** is a side sectional view taken substantially along the lines **2—2** of FIG. **1** and illustrates the positioning of the pump shaft and impeller within the pump case.

FIG. **2*a*** is a side sectional view and a continuation of the left hand end of FIG. **2** on an enlarged scale illustrating the mounting of the driven end of the pump shaft.

FIG. 2*b* is a sectional view and a continuation of the right hand end of FIG. 2 on an enlarged scale illustrating the mounting of the other end of the pump shaft.

FIG. 3 is a perspective view partially broken away and exploded of the impeller and inducer component and illustrating the impeller vanes and the inducer blades and their respective relationship.

FIG. 4 is a fragmentary side view in section illustrating a modified form of a mechanical seal for use in the pump of the present invention.

Before referring in detail to the drawings, it should be noted that the particular design of any pump is influenced by hydraulic, mechanical and metallurgical considerations. The hydraulic design of the pump concerns the pump head, capacity and speed requirements and the physical characteristics of the liquid being pumped. The mechanical or structural design must satisfy the requirements for both hydraulic and material efficiency. It must meet the restrictions of form and proportions imposed by the characteristics of the materials of construction and at the same time provide for structural strength and stability. The metallurgy is concerned with the temperature and the chemical properties of the liquid as well as the physical properties and corrosion resistance of the materials of construction.

The unique and novel design of the present invention harmonizes these hydraulic, mechanical, metallurgical requirements to provide a highly efficient, high flow, high head, centrifugal pump. It is felt that this pump will be desirable for use in several different operations. However, primary use contemplated is in waterflood operations for oil fields which requires pressures from 1,100 psi to 3,500 psi and high flow and generally involves the pumping of a corrosive liquid.

Referring now to the drawings and specifically to FIG. 1, the pump case, generally designated **10**, is shown connected to an inlet or suction line **11** and to an outlet or dischrge line **12**. The pump case **10** is constructed of titanium which has a very high strength to weight ratio. Moreover, titanium is substantially corrosion free thereby permitting substantial flexibility with regard to the liquids which can be pumped. It has been found that the pump case **10** will substantially avoid any corrosion during the pumping of salt water for temperatures up to 400° F. However, it should be noted that in most instances, as for example in the case of a multi-stage pump, the cost of titanium would be prohibitive. The compactness of the pump of the present invention makes the use of titanium economical.

The pump case **10** is radially split at **13** as seen in FIG. **2**. That is, the pump case **10** is actually comprised of two castings, a body member **10*a*** and an insert member **10*b***. One side of the casting **10*a*** is provided with an axially extending bore **13*a*** having an enlarged portion **14** inorder to receive the rotating element of the pump. The other casting **10*b*** extends into the bore **14** and encloses the open end of the bore with the rotating element stationed between the insert member **10*b*** and the end of the enlarged portion of the bore. The castings **10*a*** and **10*b*** are connected along the split **13** by means **15** and are provided with appropriate sealing means at **16** to prevent leakage therebetween. A radially split pump case is generally preferred over an axially split case such as those used in multi-stage pumps wherein the pump case is split in half along a horizontal center line and the top half is removed to receive the rotating element because at high pressures it is difficult to prevent leakage at the axial joint. This is particularly true after several instances of maintenance to the pump.

The pump case **10** of the present invention is generally referred to as a double volute type pump case. In a double volute type case there are two similar, gradually widening flow channels or passages **20** and **21**, as shown in FIG. **2**, with openings **22** and **23**, respectively, to receive the discharge from the impeller positioned 180° apart. These flow channels create opposing pressures of nearly equal magnitude about the periphery of the impeller and result in an appreciable reduction in the unbalanced radial loads on the shaft and bearings. The volutes **20** and **21** discharge into a discharge passage, shown by the phantom lines **25** in FIG. **1**, which is adapted to communicate with the discharge line **12**. The pump case is also provided with an inlet passage, as shown in FIG. **1** by the phantom lines **26**, adapted to communicate with the suction line **11**. The inlet passage is split into two channels (not shown) which divide and direct the incoming flow into two inlet passages **27** and **28** on each side of the impeller, generally designated **30**. When the incoming flow is divided in this manner the pump is typically referred to as a double suction type pump. The symmetrical aspects of double suction provide for overall pump balance which avoids the necessity for a balance line normally required in other type pumps.

The impeller **30**, which is the means employed to move the mass of liquid through the pump and to generate the delivery head, includes a center eye portion **30*a*** on each side thereof and a central bore **31** through which the pump shaft, generally designated **40**, extends. The impeller **30** is designed for double suction and is thereby provided with two sets of vanes **32** and **33** separated by a radially extending central wall **34** and projecting outwardly from the exterior surfaces **35** and **36**, respectively, on each side thereof. The exterior surfaces **35** and **36** are symmetrical with respect to the radial centerline of the impeller **30** and each is a concave surface of revolution about the axis of the bore **31**, curving from an outside peripheral edge **37** first radially inward and then axially outward to the bore edge **37*a***. The impeller is also provided with two inlet areas **38** and **39**, one at each eye **30*a*** of the impeller, in order to cooperate with the two sets of vanes. The symmetry of this double suction impeller provides for hydraulic balance in the pump and thus better efficiency.

The impeller **30** is preferably an enclosed impeller having a sidewall or shroud **39*a*** outside each set of vanes with an integral cylindrical flange **40*a*** extending axially about the pump shaft and concentric therewith. Closed impellers reduce the losses from fluid leakage from the discharge side of the impeller back to the suction side and thus provide for better pump efficiency. Similar such enclosed impellers presently used would be generally unsuitable for high speed use due to their weight and the centrifugal forces which would be created. However, because the impeller **30** of the present invention is relatively light, it also being constructed of titanium, and because of the balanced nature of the pump, as will be made apparent from the following discussion, high speed use is possible.

The form of the vane surface of the impeller also has an important effect on the operational characteristics of a centrifugal pump. The vanes **32** of the present pump are Francis type vanes, which project from the exterior surface of the impeller with a varying angle of inclination and which are curved longitudinally thereby providing on each vane a helical surface **32***a*. The angle of these vanes with respect to the fluid flow changes with radii and section, thereby giving a uniform absolute velocity across the vane section and resulting in full capacity flow without undue shock. Preferably, the set of vanes on one side of the impeller are staggered with respect to the other set of vanes, as shown in FIG. 3. This increases the frequency in which the vanes pass the volute openings and thereby reduces the magnitude of the vibrations created. In the embodiment of the pump illustrated each set of vanes includes three vanes. However, it may be desirable in certain instances to use a greater number of vanes.

Referring now in detail to the specific components of the pump and beginning with FIG. 2*a*, it is noted that the driven end of the pump shaft **40** is provided with a coupling, such as a spline coupling **41**. The driver means (not shown) is preferably a high speed, high horsepower gas turbine, but the present invention is not intended to be limited to any particular type of driver means. Connection between the turbine and the pump shaft **40** through the coupling **41** is intended to be direct. That is, a gearbox such as those generally required between the driver means and a high capacity multistage centrifugal pump is not required and therefore is preferably eliminated.

The driven end of the pump shaft is supported by a radial bearing assembly, generally designated **50**. The bearing assembly includes a housing **51** which is axially split for inspection purposes when the upper half is removed and which is secured to the pump case at the flange **52** by means such as a bolt connection **53**. The housing **51** is preferably formed of ductile iron because of its corrosion resistant characteristics and because of its relatively high strength characteristics which are necessary because of the temperatures generated during high speed operation. An open end **54** of the housing **51** is sealably connected at **57** to the driver means or other means. The housing **51** defines therein an interior coupling chamber **55** and a bearing chamber **56**, each of which is filled with lubricant such as oil under pressure from an oil pressure system (not shown). Preferably, when a high speed turbine is employed as the driver, the oil pressure system of the turbine is employed for the pump bearings and coupling. Heretofore, multi-stage centrifugal pumps were unable to utilize the turbine oil pressure system because the gearbox required the excess oil from the oil pressure system. Thus, a separate oil pressure system was required for the pump coupling and bearings and such a separate system added to the expense of the pump.

Oil in the coupling chamber **55** communicates directly through the open end **54** to an oil reservoir of the oil pressure system and oil in the bearing chamber **56** communicates by means of a flexible hose or other similar conduit means through a drain opening **58**. A bronze ring **59** provides an annular seal about the pump shaft **40** and prevents communication of oil between the two chambers **55** and **56**.

The bearing arrangement on which the driven end of the pump shaft is mounted is a self-aligning tilted pad journal bearing which is adapted to withstand a radial load of 350 lbs. under full speed conditions, 18,000 – 22,000 rpm. The bearing, generally designated **60**, is secured to an annular flange **61** which extends inwardly from the sidewall **62** of the bearing chamber **56** at approximately the center thereof and is provided with a passage **63** for communication of oil to both sides of the bearing **60**. The bearing **60** includes a cylindrical outer shell **64** locked to the integral flange **61** of the housing by means (not shown) such as a bolt connection. Secured to the inner wall **65** of the shell **64** is a plurality of five circumferentially spaced pads **66** by pins **67** or other similar means, which permit a limited amount of pivotal movement of the pads circumferentially about the shaft **40**. In this manner the pads tend to be self-aligning during operation of the pump. The shell **64** and the pads **66** are preferably bronze while each pad **66** is provided on its inner face **68** with a Babbit lining which guards against deformation under high speed and high temperature conditions. During operation of the pump, the pump shaft **40** is not in contact with the Babbit lined surface **68** of the pads, but instead runs on an oil film provided in the space between the pads **66** and the shaft **40**. Preferably, the space between the pump shaft **40** and the pads **66** is about 0.0017 inches; this space being sufficiently small to avoid vibrations.

A pair of cylindrical side plates or cover plates **69** secured to each side of the shell **64** at **70** include a central bore **71** through which the pump shaft **40** extends and an annular groove **72** in the bore wall which is in communication with a drain opening **73**. The cover plates **69** protect the bearing pads **66** from foreign particulate matter by collecting such matter in the grooves **72** with the matter thereafter being discharged through the drain **73**.

To further protect the bearing **60** the bearing assembly **50** is provided with an air seal **80** which includes an annular channel **81** with a pair of annular grooves **82** which flank the larger channel **81** on each side thereof. The channel **81** and the groove **82** extend circumferentially about the pump shaft **40**. During operation of the pump air under pressure is supplied to the channel **81** through a passage **83** which is in communication with an air hose (not shown) connected to the bearing assembly at **84**. The grooves **82** form pockets for building air pressure as the air is received from the channel **81**. Thus, a seal is provided about the shaft which prevents oil from escaping the chamber **56** and prevents foreign matter from entering the chamber **56**.

A second bearing assembly, generally designated **100**, for carrying the free end of the pump shaft **40** is seen best in FIG. 2*b*. The assembly **100** includes a housing **101** which is secured to the pump casing **10** at the flange **102** by a bolt connection **103**. The case is preferably ductile iron because of its corrosive resistance characteristics and its relatively high strength at high temperatures. The housing **101** defines an interior axial bearing chamber **104** and a thrust bearing chamber **105**, both of which are filled with oil from the oil pressure system (not shown). Communication with the reservoir of the oil pressure system is provided for the chamber **104** through the drain opening **106** and for the chamber **105** through the drain opening **107**. The axial bearing, generally designated **60***a*, of the bearing assembly **100** is also a self-aligning tilted pad journal bearing and is substantially identical to the bearing **60**. Therefore, a detailed description of the components of this axial bearing is felt to be merely repetitious and instead the same reference numerals used in the description of bearing **60** are used to identify the identical components of bearing **60***a*.

The thrust bearing, generally designated **130,** is typically referred to as a self-leveling thrust bearing and is adapted to accommodate a limited amount of axial movement by the pump shaft **40** and a thrust load of about 2,500 lbs. actual axial thrust. The thrust bearing, generally designated **110,** includes a thrust disc **111** secured to the pump shaft **40** by means of a collar **112** which is provided with keyway **113.** A cap **114** threadably secured at **115** to the end of the pump shaft abuts the thrust disc **111** and maintains it in the proper axial position on the pump shaft. The cap **114** is fixed to the pump shaft by a lock screw **116** which prevents relative turning movement therebetween. An adjusting ring **117***a* on the other side of the thrust disc **111** adjusts the axial position of the pump shaft **40** to properly position the impeller with the volutes as will become evident from the following discussion.

Mounted on each side of the thrust disc **11** are a plurality of six shoes **117,** each of which includes a button press fitted thereon. The buttons provide a convex surface **119** which rides on a plurality of rectangular leveling pads **120.** The pads **120** are secured by means **121** for limited movement within an annular channel **122** of a pad support disc **123.** The pad support discs on each side of the thrust disc are affixed to the housing **101** by means (not shown) such as a bolt or screw.

Circular passages **124** provide for communication of the oil in the chamber **105** around both sides of the bearing **110.** A bronze ring **125** locked to the housing **101** by lock screw **126** provides an annular seal about the shaft **40** and prevents communication of oil between chambers **104** and **105.** An end plate **127** enclosing the rear end of the chamber **105** is removably connected to the housing **101** by bolts **126** and permits inspection and maintenance of the assembly **100.**

The bearing assembly **100** is also provided with an air seal, generally designated **127,** which is substantially identical to the seal **80** of the bearing assembly **50.** The air seal **127** prevents a loss of oil from the chamber **104** and the introduction of foreign particulate matter into the bearing chambers.

Referring now to the components of the pump contained within the pump case **10,** as best seen in FIG. **2,** it is noted that at each end of the pump case about the pump shaft **40** there is provided a mechanical seal or so called stuffing box, generally designated **150.** The stuffing boxes at each end are identical and therefore a description of only one will be set forth in detail. The stuffing box **150** is contained within an end chamber **151** of the pump case and enclosed by an end plate **152** secured to the pump case by bolts **153.** The stuffing box is provided with a sleeve **154** which is concentric about the shaft **40** and extends beyond the end plate **152** to be received by an end cap **155.** The end cap **155** is locked to the pump shaft by screws **156** and thereby locks the sleeve **154** to the shaft **40** by the lock screw **157** which secures the sleeve to the end cap. The mechanical seal prevents leakage of fluid from the suction side of the pump and dissipates the heat generated at the contacting surfaces of the rotating components and the stationary components. The sleeve **154** is provided with an O-ring or rubber gasket **150** which prevents leakage between the sleeve and the shaft **40.** Secured to the end of the sleeve **154** by bolt means **159***a* is an end wall **160** having an axially extending cylindrical flange **161** projecting therefrom. An O-ring **161***a* prevents leakage between the end wall and the exterior surface of the sleeve **154.** The cylindrical flange **161** forms an annular channel **162** between the flange **161** and the exterior sidewall of the sleeve **154.** Retained within the channel **162** by lock means **162***a* is cylindrical insert **163** of tungsten carbide. Positioned between the flange **161** and the insert **163** is a gasket **164.**

Biased against the end surface of the insert **163** at **158** is a carbon seal ring or mating ring **166.** During operation of the pump the insert **163** rorates with the pump shaft **40** whereas the seal ring **166** remains stationary with the pump case and the surface of contact therebetween at **158** forms the mechanical seal. The seal ring **166** is carried within an annular groove or channel **167** and the end wall **152.** A spacer ring **167***a* locked to the mating ring **166** by means **168***a* with a gasket member **169***a* therebetween is positioned between the mating ring **166** and spring means **168.** The spring means **168** is retained within the end wall **152** and acts on the spacer ring **167***a* and thus the mating ring **166** to maintain a proper load at the seal face **158.** An O-ring **170** prevents leakage between the end wall **152** and the pump casing.

Coolant is supplied to the stuffing box or seal through an inlet at **172** which communicates with the annular groove **167** for supply of coolant to the contacting surfaces at **158.** Preferably, the coolant for the mechanical seal is obtained from the discharge side of the pump through the tap inlet as indicated by the phantom lines **174.** Coolant is communicated from the tap **174** to the inlet **172** by a conduit (not shown), such as a flexible hose. By using the pump discharge as the supply for the coolant, the coolant is maintained at the coolant inlet **172** at a higher pressure, approximately **15** psi, than at the suction side of the pump to thereby promote any fluid flow, which might occur due to leakage, back to the suction side of the pump.

Directing attention now to the impeller and its mounting with respect to the pump case, it should be noted that the impeller is mounted on the shaft **40** by means of a spline coupling (not shown) which reduces torsional vibrations. The centering or positioning of the impeller **30** with respect to the pump case is determined by the adjusting ring **117***a* as described earlier. When properly positioned, as shown in FIG. **2,** the impeller **30** rides during operation on a pair of wearing rings **180** bolted to the pump case at **181** and interposed between the pump case and the exterior surface of the shroud of the impeller. The wearing rings **180** minimize leakage from the discharge side of the impeller back to the suction side of inlet side at **28** and **27.** Both the exterior surface of the cylindrical flange **40***a* of the impeller shroud and the surface of the wearing ring in contact therewith are provided with oppositely oriented spiral grooves **182.** That is, the wearing rings are provided with left hand spiral grooves and the exterior surfaces of the shrouds are provided with right hand spiral grooves or vice versa. These spiral grooves prevent a freezing of parts at the contacting surfaces during a shutdown of the pump.

Attention is now directed to a component which has a marked effect on the total operation of this high speed, high flow centrifugal pump. The inducer component, generally designated **200,** is provided on each side of the impeller at the eye of the impeller or the inlet just preceding the vanes. Each inducer **200** includes a cylindrical section **201** which is positioned concentrically about the pump shaft **40** and locked thereto by a keyway **202** and lock screws **203.** Projecting from the cylindrical section **201** and substantially to the inner surface of the cylindrical flange **40***a* of the shroud at **205** are a pair of axially spaced spiral blades **206** and **207** which wind about the cylindrical section **201** in the same direction as the vanes **32.** The leading edges **208** of the blades **207** and **206** are approximately 180° apart and each leading edge has a cutting angle with respect to the incoming fluid of about 7-½°. Moreover, the pitch of the blades is such that the liquid is caused to be accelerated thereby in a direction toward the vanes and in a direction correspondingly tangent with the direction of the vanes substantially at the impeller eye and in the direction of rotation of the impeller. Thus, the blades **206** and **207** are adapted to maintain a positive pressure by the fluid on the backside of the vanes **32** and **33** during high speed operation of the pump and thus substantially reduce the problem of cavitation.

FIG. **4** illustrates another form of mechanical seal or stuffing box for use in the present invention. The stuffing box shown in FIG. **4** and generally designated **250** is contained within an end chamber **251** of the pump case and enclosed by an end plate **252** secured to the pump case by bolts **253.** The stuffing box is provided with a sleeve **254** which is concentric about the shaft **40** and extends beyond the end plate **252** to be received by an end cap **255.** The end cap **254** is locked to the pump shaft by screws **256** and thereby locks the sleeve **254** to the shaft **40** by the lock screw **257** which secures the sleeve to the end cap. The sleeve **254** is provided with an O-ring or rubber gasket **259** which prevents leakage between the sleeve and the shaft **40.** Extending radially from the end of the sleeve is an end wall **260** having an axially extending cylindrical flange **261** projecting therefrom and forming an annular channel **262** between the flange **261** and the exterior sidewall of the sleeve **254.** Retained within the channel **262** is a cylindrical insert **263** of tungsten carbide. Positioned between the flange **261** and the insert **263** is a Teflon ring **264** and a gasket **265.** Locking means (not shown) maintain the insert within the channel **262.**

Biased against the end surface of the insert **263** at **258** is a carbon seal ring or mating ring **266.** During operation of the pump the insert **263** rotates with the pump shaft **40** whereas the seal ring **266** remains stationary with the pump case and the surface of contact therebetween at **258** forms the mechanical seal. The seal ring **266** is carried within an annular groove or channel **267** in the end wall **252.** Spring means **268** retained within the end wall **252** acts on one end of the seal ring **256** to maintain a proper load at the seal face **258.** A disc **269** is interposed between the seal ring **266** and the spring **268** and an O-ring **270** and Teflon ring **271** are inserted between the seal ring **266** and the inner wall of the annular groove **267.**

Coolant is supplied to the stuffing box or seal through an inlet at **272** which communicates with the annular groove **267** and a plurality of passages **273** which supply coolant to the contacting surfaces at **258.** A passage **275** extends through the flange member **261** to the area of contact between the insert **263** and the seal ring **266** to permit the removal of hot fluid caused by the heat generated due to friction at the contacting surfaces **258.**

The present invention provides a relatively compact and economical pump particularly suitable for high head, high capacity applications. In many instances it can be used where it was heretofore thought necessary to employ a less efficient, less economical multi-stage centrifugal pump.

Having fully described my invention, it is to be understood that I do not wish to be limited to the details herein set forth, but my invention is of the full scope of the appended claims.

I claim:

1. A high speed centrifugal pump apparatus comprising:

a pump case having inlet flow passage means for communication of liquid from a suction line and outlet flow passage means for communication of liquid to a discharge line;

a pump shaft mounted for rotational movement within said pump case;

an impeller mounted concentrically on said shaft for rotational movement therewith within said pump case, said impeller having a center eye portion on at least one side thereof in communication with said inlet flow passage means, said impeller extending generally radially outward from said pump shaft and having a plurality of vanes projecting laterally from at least one side thereof and extending longitudinally from said center eye portion to its peripheral edge in communication with said outlet flow passage means, each said vane having a varying angle of inclination with respect to the side of the impeller and being curved longitudinally thereby providing a continuous helical surface for moving the liquid without undue shock from said inlet flow passage means to said outlet flow passage means;

inducer means in communication with said inlet flow passage means and adapted for rotation with said pump shaft, said inducer means having at least one spiral blade winding concentrically about said shaft in the direction of rotation of said vanes and terminating adjacent the beginning of said vanes at said center eye portion of said impeller whereby said inducer means causes the liquid to accelerate in a direction toward said vanes and maintain a positive pressure against the helical surfaces thereof during high speed rotation of the impeller said inducer means separate from said impeller; and

vanes are provided on each side of said impeller and separate inducer means are provided on each side of said impeller including at least one blade terminating adjacent the beginning of a set of vanes, said impeller including a shroud associated with each side of said impeller which extends about said inducer means.

2. The apparatus of claim **1,** wherein each said inducer means includes a pair of axially spaced blades, each of which terminates adjacent the beginning of a set of vanes.

3. The apparatus of claim **1,** wherein said vanes on one side of said impeller are staggered with respect to the vanes on the other side of said impeller.

4. The apparatus of claim **1,** wherein said vanes are enclosed by a shroud, said shroud having an axially extending integral cylindrical flange concentric about said inducer means, said flange having means thereon for preventing a freezing of said impeller with respect to said pump case when the rotation of said impeller stops.

5. The apparatus of claim **1,** wherein said impeller and said pump case are titanium.

6. The apparatus of claim **1,** wherein the pitch of said inducer blade is adapted to accelerate the liquid both in a direction toward said vanes and in a direction corresponding to the direction of rotation of said vanes.

7. In a high speed centrifugal pump apparatus including a radially split pump case, a pump shaft extending axially within said pump case and adapted for high speed rotation therein, and an enclosed impeller with Francis type vanes on each side thereof mounted to said pump shaft for rotation therewith within said pump case, said pump case having a divided inlet flow passage for communicating liquid to each side of said impeller and an outlet flow passage with two volutes extending about the periphery of the impeller for receiving the discharge of liquid therefrom, said vanes extending substantially from the center eye portions at each side of the impeller to the impeller periphery and adapted to move liquid from the inlet flow passage to the volutes thereby imparting energy to the liquid, the improvement which comprises:

inducer means on each side of said impeller proximate the center eye portion thereof for maintaining a positive pressure on said vanes, each said inducer means including at least one spiral blade extending radially substantially between said pump shaft and the axially extending shroud flange in said enclosed impeller, said blade winding concentrically from a leading edge in communication with said inlet flow passage about said pump shaft and terminating adjacent said vanes at said center eye portion, and means for operably connecting said blade to said pump shaft for rotation therewith whereby rotation of said blade causes the liquid from said inlet flow passage to move toward said vanes.

8. The apparatus of claim **7,** wherein said pump case and said impeller are each cast of titanium.

9. The apparatus of claim **7,** wherein the vanes on one side of said impeller are staggered with respect to the vanes on the other side of said impeller.

10. The apparatus of claim **7,** wherein each said inducer means includes a pair of axially spaced blades, each said blade winding concentrically about said pump shaft in the direction of rotation of said impeller.

11. The apparatus of claim **7,** wherein a wearing ring is secured to said pump case concentrically about each said shroud flange and adjacent thereto, said exterior surface of said shroud flanges and said interior surfaces of said wearing rings being provided with oppositely oriented spiral grooves to prevent said wearing rings and said shrouds from freezing together.

12. A single stage high speed, high capacity centrifugal pump apparatus comprising:

a split pump case having an integrally cast body member and an integrally cast insert member, said body member having an axially extending bore enlarged at one end, said insert member extending partially into the enlarged portion of said bore and having an axially extending bore in concentric parallel alignment with the other portion of said body member bore thereby providing a continuous axially extending bore through said pump case, said body member and said insert member being sealably joined along an annular radially extending surface and forming within said pump case an impeller chamber between the inwardly extending end of said insert member and the end of the enlarged portion of said body member bore;

a pump shaft extending through said pump case bore and mounted for rotational movement therein;

an impeller member mounted concentrically on said pump shaft for rotational movement therewith within said impeller chamber, said impeller member having a central wall which tapers radially outward to its periphery from center eye portions on each side thereof adjacent said pump shaft, said generally radially extending sides of said central wall being symmetrical concave surfaces of revolution about said pump shaft, adjacent each said side of said center wall and spaced therefrom a shroud having an integral cylindrical flange extending axially about said pump shaft and concentric therewith;

inlet flow passage means formed within said pump case by said body member and said insert member for communication of liquid from a suction line, said inlet flow passage means having a pair of channels for communication of the liquid to each of said center eye portions;

outlet flow passage means formed within said pump case by said body member and said insert member for communication of liquid to a discharge line, said outlet flow passage means having a pair of volute passages extending about the periphery of said impeller, each said volute passage having an opening for receiving discharge flow from the impeller with said openings being substantially 180° apart;

a set of vanes on each side of said impeller member center wall extending laterally from said side to said shroud and extending longitudinally from a center eye portion thereof to the impeller periphery, each said vane having a varying angle of inclination with respect to said sides of said impeller member and being curved longitudinally along its length and thereby providing a continuous helical surface for moving liquid from said center eye portions to said volute openings; and

inducer means operably connected to said pump shaft for rotation therewith on each side of said impeller member substantially within the cylindrical flange of said shroud and in communication with said channel of said inlet flow passage means, each said inducer means having at least one spiral blade extending substantially between said shroud and said pump shaft said blade winding from a leading edge concentrically about said pump shaft in the direction of curvature of said vanes and terminating adjacent the beginning of said vanes at said center eye portion whereby said inducer means move the liquid from said channels toward said vanes and maintain a positive pressure on the helical surfaces thereof during high speed operation of the pump.

13. The apparatus of claim **12,** wherein the vanes on one side of said impeller are staggered with respect to the vanes on the other side of said impeller.

14. The apparatus of claim **12,** wherein said leading edge is inclined to provide a cutting angle with respect to the liquid from said channel of about 7-½°.

15. The apparatus of claim **12,** wherein each said inducer means includes a pair of axially spaced blades.

16. The apparatus of claim 15, wherein said leading edges of said part of blades are angularly spaced substantially 180° apart.

17. The apparatus of claim 15, wherein each said leading edge is inclined to provide a cutting angle with respect to the liquid from said channel of about 7-½°.

18. The apparatus of claim 12, wherein said body member and said insert member are titanium.

19. The apparatus of claim 12, wherein said impeller member is integrally cast and is titanium.

20. The apparatus of claim 12, wherein a wearing ring is secured to said pump case concentrically about each said shroud flange and adjacent thereto, said exterior surface of said shroud flanges and said interior surfaces of said wearing rings being provided with oppositely oriented spiral grooves to prevent said wearing rings and said shrouds from freezing together.

21. The apparatus of claim 12, wherein a mechanical seal is provided at each end of said pump case concentrically about said pump shaft, said mechanical seal having frictionally engaging relatively movable components adapted to provide a seal between said pump shaft and said pump case, and means for communicating liquid discharged from said impeller to said mechanical seal and thereby cooling said components.

22. The apparatus of claim 12, wherein turbine drive means are provided for driving said pump shaft, and means are provided on said pump shaft for coupling said pump shaft directly to said turbine drive means.

23. The apparatus of claim 22, wherein said pump shaft is rotatably mounted within bearing assembly means, and means are provided for connecting said bearing assembly means directly to the oil pressure system of said turbine drive means thereby providing lubricant for said bearing assembly means.

24. The apparatus of claim 22, wherein turbine drive means is adapted to drive said pump shaft at speeds in excess of 20,000 rpm.

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