US 20070077155A1

(19) United States
(12) Patent Application Publication
(10) Pub. No.: US 2007/0077155 A1
(43) Pub. Date: Apr. 5, 2007
Shah et al.

(54) CENTRIFUGAL PUMP WITH HYDRODYNAMIC BEARING AND DOUBLE INVOLUTE

(75) Inventors: **Ketan R. Shah**, Olympia, WA (US); **Michael T. Crocker**, Portland, OR (US); **Daniel P. Carter**, Olympia, WA (US); **Kazimierz L. Kozyra**, Olympia, WA (US); **Gavin D. Stanley**, Puyallup, WA (US)

Correspondence Address:
SCHUBERT, OSTERRIEDER & NICKELSON, PLLC
c/o Intellevate
P.O. BOX 52050
MINNEAPOLIS, MN 55402 (US)

(73) Assignee: **Intel Corporation**

(21) Appl. No.: **11/241,624**

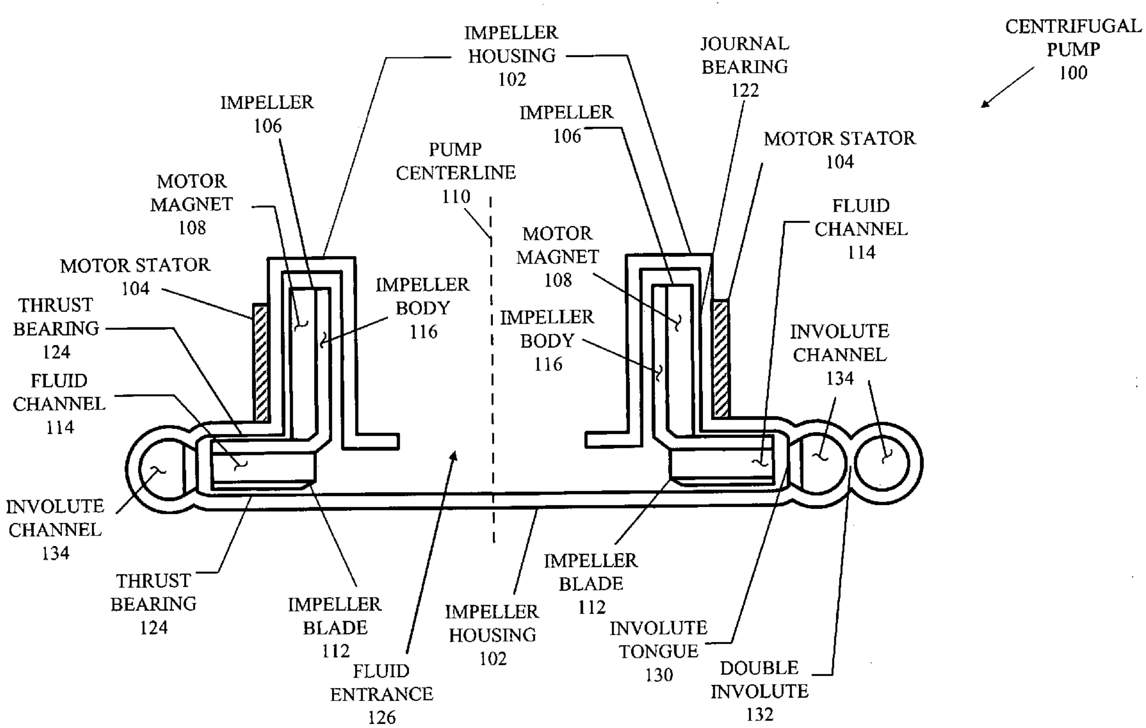
(22) Filed: **Sep. 30, 2005**

Publication Classification

(51) Int. Cl.
F04B 17/00 (2006.01)
(52) U.S. Cl. **417/353**; 417/423.12

(57) **ABSTRACT**

A centrifugal pump with a hydrodynamic bearing and a double involute is disclosed. Some embodiments may include an impeller housing comprising a fluid entrance to allow fluid to enter and an impeller located within the impeller housing, the impeller comprising a plurality of impeller blades, a plurality of fluid channels between the impeller blades, and a motor magnet. The impeller may rotate within the impeller housing about a pump centerline in response to an electromagnetic field and the fluid channels may each allow fluid to pass through when the impeller rotates. Embodiments may also include one or more hydrodynamic bearings positioned between the impeller and the impeller housing to support generated loads and a double involute coupled with the impeller housing and positioned to receive fluid exiting the plurality of fluid channels. Further embodiments may include a motor stator to generate the electromagnetic field. Other embodiments are disclosed and claimed.



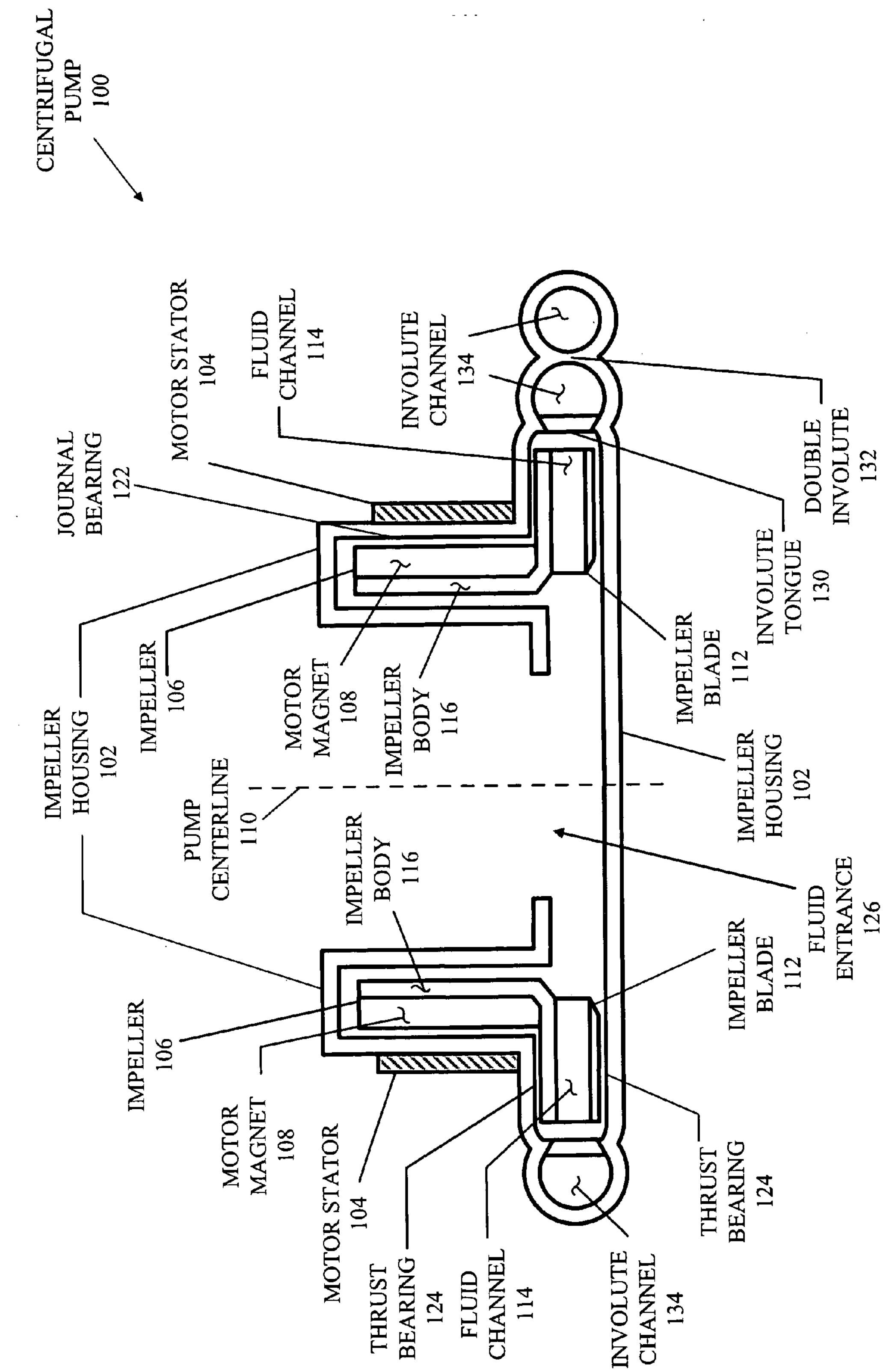

CENTRIFUGAL PUMP 100
MOTOR STATOR 104
FLUID CHANNEL 114
INVOLUTE CHANNEL 134
JOURNAL BEARING 122
DOUBLE INVOLUTE 132
INVOLUTE TONGUE 130
IMPELLER HOUSING 102
IMPELLER 106
MOTOR MAGNET 108
IMPELLER BODY 116
IMPELLER BLADE 112
PUMP CENTERLINE 110
IMPELLER HOUSING 102
IMPELLER BODY 116
FLUID ENTRANCE 126
IMPELLER BLADE 112
IMPELLER 106
MOTOR MAGNET 108
MOTOR STATOR 104
THRUST BEARING 124
FLUID CHANNEL 114
INVOLUTE CHANNEL 134
THRUST BEARING 124

FIG 1

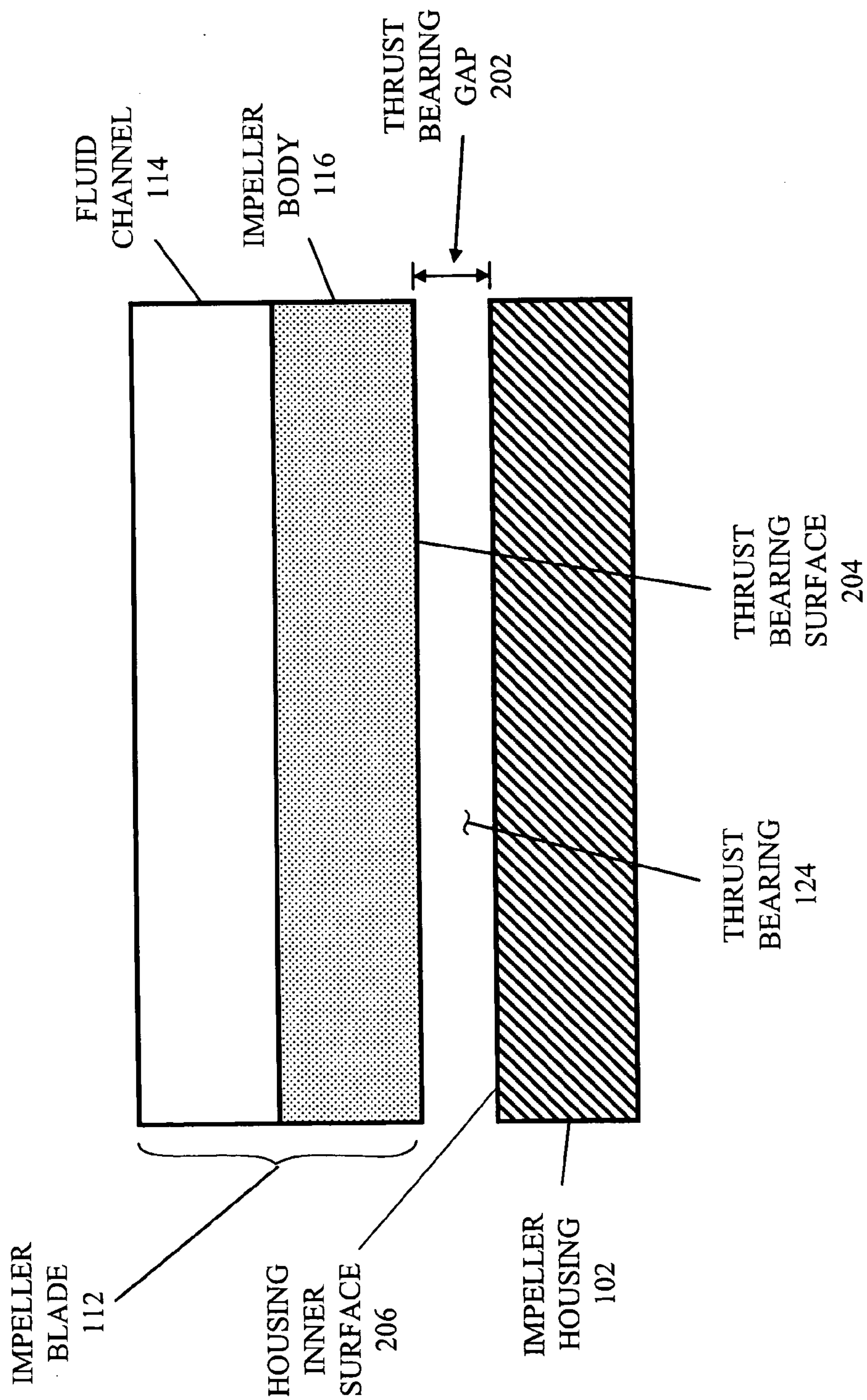

FLUID CHANNEL 114
IMPELLER BODY 116
THRUST BEARING GAP 202
THRUST BEARING SURFACE 204
THRUST BEARING 124
IMPELLER BLADE 112
HOUSING INNER SURFACE 206
IMPELLER HOUSING 102

FIG 2

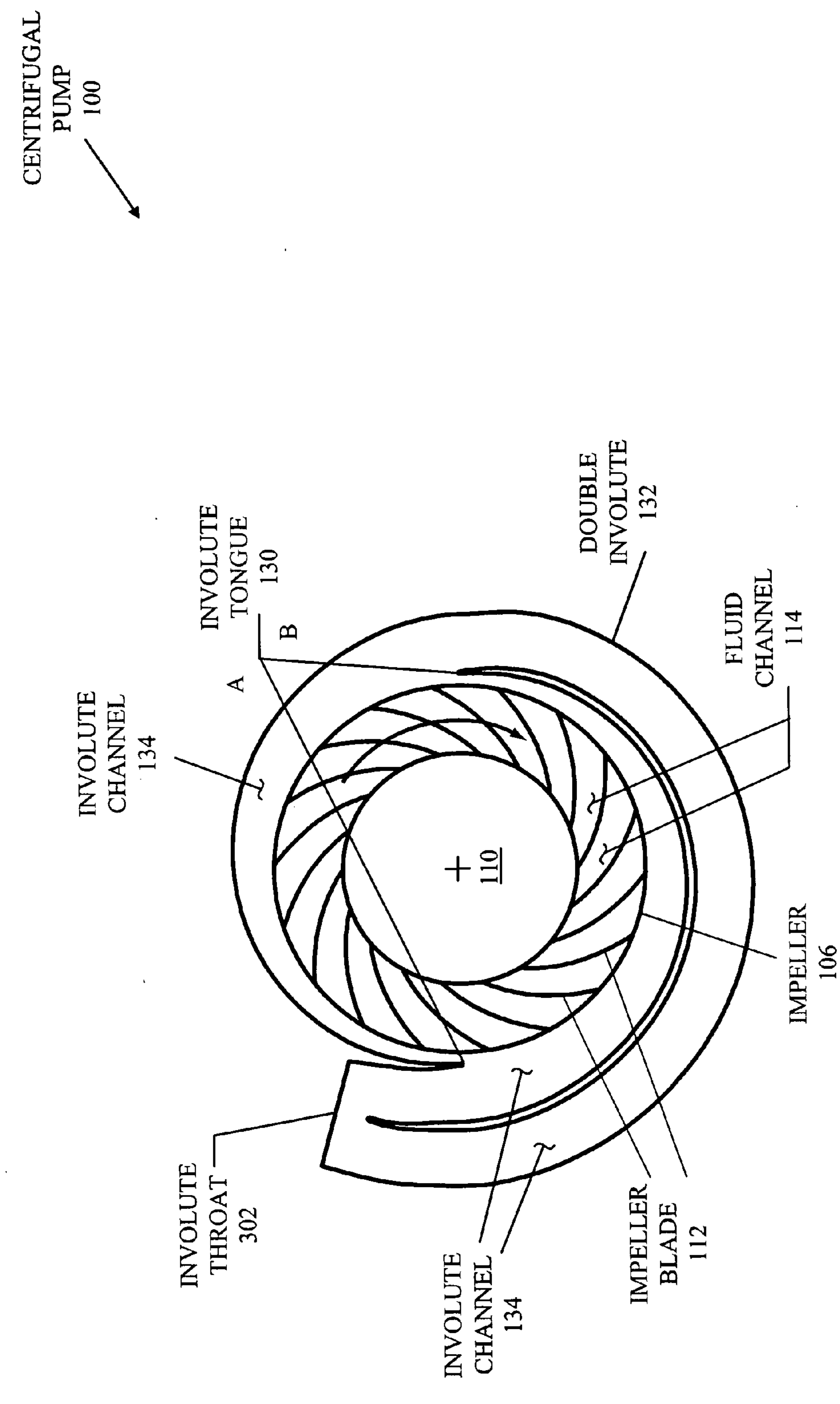

CENTRIFUGAL PUMP 100
INVOLUTE TONGUE 130
A
B
DOUBLE INVOLUTE 132
INVOLUTE CHANNEL 134
FLUID CHANNEL 114
110
IMPELLER 106
INVOLUTE THROAT 302
INVOLUTE CHANNEL 134
IMPELLER BLADE 112

FIG 3

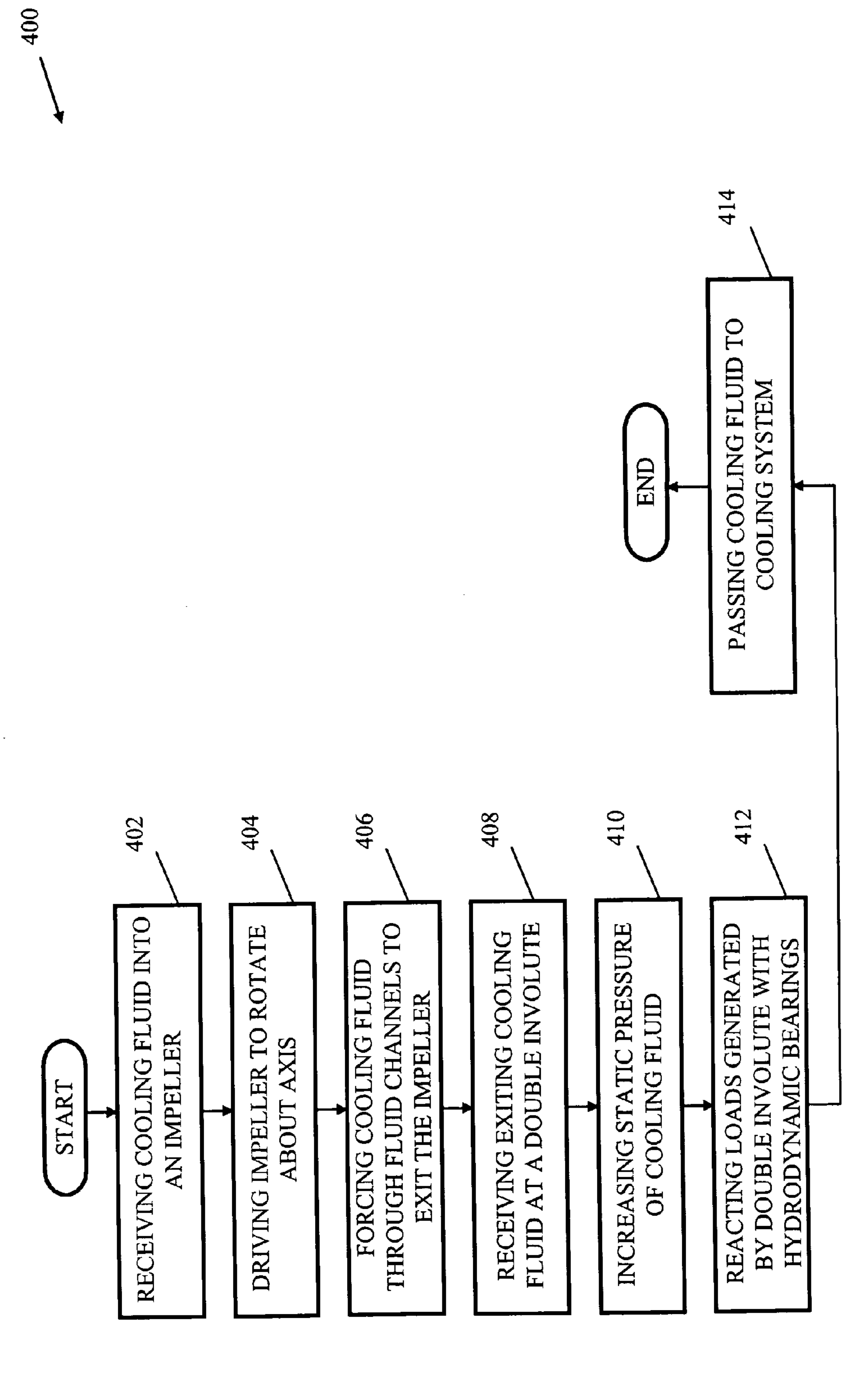

400
START
RECEIVING COOLING FLUID INTO AN IMPELLER
402
DRIVING IMPELLER TO ROTATE ABOUT AXIS
404
FORCING COOLING FLUID THROUGH FLUID CHANNELS TO EXIT THE IMPELLER
406
RECEIVING EXITING COOLING FLUID AT A DOUBLE INVOLUTE
408
INCREASING STATIC PRESSURE OF COOLING FLUID
410
REACTING LOADS GENERATED BY DOUBLE INVOLUTE WITH HYDRODYNAMIC BEARINGS
412
PASSING COOLING FLUID TO COOLING SYSTEM
414
END

FIG 4

CENTRIFUGAL PUMP WITH HYDRODYNAMIC BEARING AND DOUBLE INVOLUTE

FIELD

[0001] The present invention is in the field of centrifugal pumps. More particularly, the present invention relates to a centrifugal pump with a hydrodynamic bearing and a double involute.

BACKGROUND

[0002] Electronic devices generate heat during operation. Device designers often utilize thermal management to keep temperature-sensitive elements of an electronic device within a prescribed operating temperature. Failure to properly cool an electronic device can result in overheating, which may cause a reduction in device service life, device failure, or a reduction in operating performance. Historically, designers have cooled electronic devices using natural convection by strategically locating openings in the device packaging or case to allowed warm air to escape and cooler air to be drawn in. The advent of high performance electronic devices such as processors, however, now requires more sophisticated thermal management. Increasing electronic device performance often results in an increase in the heat generated by the device and often results in a smaller size for the electronic device, both conditions of which increase the amount of thermal energy that needs to be handled. As electronic device designs continue to increase in sophistication, these problems will be exacerbated and the need for improved thermal management will continue to increase.

[0003] One thermal management solution for high performance processors or other computer system components is the use of liquid cooling. One method of liquid cooling of components is to use a cold plate thermally coupled to the component. In this solution, a pump may pump cooling fluid through the cold plate, allowing heat to be transferred from the component to the cooling fluid through the cold plate, after which the heat is removed from the cooling fluid via a heat exchanger and then returned to the cold plate. Liquid cooling using a cold plate can be more effective than solid conduction cooling methods and can also provide additional flexibility in the size and location of the heat exchanger, as the system can pump the heated fluid to a heat exchanger located in a more desirable location. While a liquid cooling system can be effective at cooling high performance components, it can be more expensive and complicated than previous methods. Because of the cost and complexity of liquid cooling systems, liquid cooling is typically only used on higher end systems. The cost and complexity of pumps to move cooling fluid through a liquid cooling system is a significant part of the cost and complexity of the entire liquid cooling system. Reducing the cost and complexity of liquid cooling pumps can therefore make liquid cooling solutions for heat-generating components suitable for more systems.

BRIEF DESCRIPTION OF THE DRAWINGS

[0004] Advantages of the invention will become apparent upon reading the following detailed description and upon reference to the accompanying drawings in which like references may indicate similar elements:

[0005] FIG. **1** depicts a side cut-away view of a centrifugal pump with a double involute and hydrodynamic bearings according to one embodiment;

[0006] FIG. **2** depicts a side cut-away view of a hydrodynamic thrust bearing of the centrifugal pump of FIG. **1** according to one embodiment;

[0007] FIG. **3** depicts a cut-away plan view of a double involute of the centrifugal pump of FIG. **1** according to one embodiment; and

[0008] FIG. **4** depicts a flowchart of an embodiment to pump cooling fluid in a cooling system.

DETAILED DESCRIPTION OF EMBODIMENTS

[0009] The following is a detailed description of example embodiments of the invention depicted in the accompanying drawings. The example embodiments are in such detail as to clearly communicate the invention. However, the amount of detail offered is not intended to limit the anticipated variations of embodiments; on the contrary, the intention is to cover all modifications, equivalents, and alternatives falling within the spirit and scope of the present invention as defined by the appended claims. The detailed descriptions below are designed to make such embodiments obvious to a person of ordinary skill in the art.

[0010] Generally speaking, a centrifugal pump with a hydrodynamic bearing and a double involute is disclosed. Some embodiments may include an impeller housing comprising a fluid entrance to allow fluid to enter and an impeller located within the impeller housing, the impeller comprising a plurality of impeller blades, a plurality of fluid channels between the impeller blades, and a motor magnet. The impeller may rotate within the impeller housing about a pump centerline in response to an electromagnetic field and the fluid channels may each allow fluid to pass through when the impeller rotates. Embodiments may also include one or more hydrodynamic bearings positioned between the impeller and the impeller housing to support generated loads and a double involute coupled with the impeller housing and positioned to receive fluid exiting the plurality of fluid channels. Further embodiments may include a motor stator to generate the electromagnetic field.

[0011] Another embodiment comprises a method for pumping cooling fluid. Some embodiments of the method may include receiving a cooling fluid into an impeller and driving the impeller to rotate about an axis to force the cooling fluid to exit the impeller. Embodiments may also include receiving the exiting cooling fluid at two or more involutes and increasing a static pressure of the cooling fluid in the two or more involutes. Embodiments may also include reacting any loads generated by the two or more involutes and rotating impeller with one or more hydrodynamic bearings. Further embodiments may include after increasing the static pressure of the cooling fluid, passing the cooling fluid to a cooling system.

[0012] The disclosed system and methodology may advantageously provide for a centrifugal pump with two or more involutes, such as a double involute, and hydrodynamic bearings. The use of a two or more involutes may reduce radial and moment loads on the impeller and centrifugal pump and may thus allow for the use of hydrodynamic bearings instead of solid element bearing technologies, potentially reducing the cost and complexity of the centrifugal pump. The combination of a multiple involutes with hydrodynamic bearings may also allow for the use of plastic injection molded parts for the centrifugal pump, potentially further reducing the cost of the pump. Reduced cost and complexity for centrifugal pumps may allow for use of centrifugal pumps and liquid cooling in a more diverse set of circumstances and for more types of systems.

[0013] Various embodiments of the present invention provide systems and methods for pumping fluid. The following description provides specific details of certain embodiments of the invention illustrated in the drawings to provide a thorough understanding of those embodiments. It should be recognized, however, that the present invention can be reflected in additional embodiments and may be practiced without some of the details in the following description. In other instances, well-known structures and functions have not been shown or described in detail to avoid unnecessarily obscuring the description of the embodiments of the invention. While specific embodiments will be described below with reference to particular configurations and systems, those of skill in the art will realize that embodiments of the present invention may advantageously be implemented with other substantially equivalent configurations and/or systems.

[0014] Turning now to the drawings, FIG. **1** depicts a side cut-away view of a centrifugal pump with a double involute and hydrodynamic bearings according to one embodiment. The centrifugal pump **100** of FIG. **1** includes an impeller **106** having a motor magnet **108** housed within an impeller housing **102**. A motor stator **104** may be located in or outside the impeller housing **102**. When electric power is applied to the motor stator **104**, the motor stator **104** creates a magnetic field that may drive the motor magnet **108** of the impeller **106**, causing the impeller **106** to rotate within the impeller housing **102** and about a pump centerline **110**. Cooling fluid may enter the impeller housing **102** through, for example, a fluid entrance **126**. As the axial flow input enters the impeller housing **102** through the fluid entrance (and along the pump centerline **110**), it flows into the rotating impeller **106**. The rotational speed of the impeller **106** creates a centrifugal force that propels the fluid through fluid channels **114** formed by and between a plurality of impeller blades **112**. The fluid exiting the fluid channels **114** may be collected by a double involute **132** at two involute tongues **130**. The double involute **132** (as described in more detail in relation to FIG. **3**) may then convert the dynamic pressure of the fluid to a static pressure, generating the pressure difference used by the centrifugal pump **100** to drive the cooling fluid. The double involute **132** may generate equal and opposite loads at each involute which cancel out to create a relatively small net radial load. With a double involute **132**, the generated loads are equal and opposite when the impeller **106** has an even number of impeller blades **112** evenly distributed around the impeller **106** and hence an impeller blade **112** passes both involute tongues **130** at the same time. A double involute **132** may thus provide a more balanced and lower radial load on the bearings of the centrifugal pump **100** when compared to a single involute.

[0015] The centrifugal pump **100** of the disclosed embodiments may utilize hydrodynamic bearings between the impeller **106** and impeller housing **102** to support the rotating impeller **106** and its associated loads within the impeller housing **102**. Hydrodynamic bearings (which are described in more detail in relation to FIG. **2**) may rely on the gap between a stationary surface (i.e., the inside surface of the impeller housing **102**) and a moving surface (i.e., the outside surface of the rotating impeller **106**) and viscous effects of its constituent bearing film to handle the net radial force and moment associated with the development of mass flow and static pressure on the impeller blades **112**. The hydrodynamic bearings may include one or more journal bearings **122** positioned between the impeller **106** and the impeller housing **102**, such as near the motor stator **104**, and one or more thrust bearings **124** positioned between the impeller blades **112** and the impeller housing **102**. Hydrodynamic bearings may be less expensive than other types of bearings (such as roller bearings, ball bearings, needle bearings or other solid element bearings) but often have less load-bearing capacity than the more expensive bearings. The reduced loads resulting from the double involute **132** may advantageously allow the use of hydrodynamic bearings in the centrifugal pump **100** instead of more expensive and higher capacity bearings, providing for a less expensive centrifugal pump **100**.

[0016] The impeller housing **102** of the depicted embodiment is symmetrical about the pump centerline **110**, completely enclosing the impeller **106** while allowing the passage of fluid into the impeller housing **102** through the fluid entrance **126**. The fluid entrance **126** may be an opening (or multiple openings) in the impeller housing **102** or other component (such as a tube or pipe) that allows passage of cooling fluid into the impeller housing **102**. The impeller housing **102** may be made of any type of material, including metal or plastic, and may be of any shape adapted to partially or fully enclose the impeller **106**. In one embodiment, the impeller housing **102** may be manufactured using plastic injected part manufacturing methods. The impeller housing **102** may be oriented vertically (as depicted in FIG. **1**) with a gravitational force oriented downward, horizontally with the pump centerline **110** perpendicular to the gravitational force, or in any other direction. The impeller housing **102** may be part of a larger system or may be coupled to other components, such as a cold plate for a liquid cooling system.

[0017] The impeller **106** may include an impeller body **116** in addition to the motor magnet **108**. Similarly to the impeller housing **102**, the impeller body **116** may be made of any type of material, including metal or plastic, and may be injection molded plastic in one embodiment. The motor magnet **108** may be attached to the impeller body **116** in any fashion such that a rotational force applied to the motor magnet **108** also rotates the impeller body **116** (and thus the impeller **106** as a whole). This allows the impeller **106** to be magnetically coupled with the motor stator **104** through the wall of the impeller housing **102**. This eliminates the need to have a shaft or other physical coupling mechanism to connect to drive the impeller **106**. This may reduce cost and complexity, as a shaft through the impeller housing **102** to drive the impeller **106** would require, for example, rotating seals, which may be expensive and prone to leakage. In one embodiment, the motor stator **104** is outside the impeller housing **102** and shaped as a concentric circle around the outside of the impeller housing **102**. For example, the motor stator **104** may be one or more laminated steel sheets with copper wires wound on it and wrapped around the impeller housing **106** that generates a magnetic field when a direct current (DC) charge is applied to the copper wire. Other motor stator **104** designs may also be used, including solid motor stators **104** or a motor stator **104** that is partially or fully integrated within the wall of the impeller housing **102**.

[0018] The impeller body **116** and motor magnet **108** may be configured in any way. In one embodiment, for example, the motor magnet **108** may be positioned outside of the impeller body **116** so that the motor magnet **108** is closer to the motor stator **104**. This embodiment may maximize the magnetic force created by the motor magnet **108** and motor stator **104** as the motor magnet **108** will be closer to the motor stator **104** and thus in a more powerful part of the motor stator's **104** magnetic field when it is powered. In another embodiment, the motor magnet **108** may be positioned on the inside of the impeller body **116** so that it is closer to the pump centerline **110**. In other embodiments, the impeller body **116** may fully or partially enclose the motor magnet **108**. In an alternative embodiment, the motor magnet **108** may serve as the impeller body **116**, eliminating the need for a separate motor magnet **108** and impeller body **116**.

[0019] As described previously, the rotational speed of the impeller **106** creates a centrifugal force that propels cooling fluid through the fluid channels **114** between the plurality of impeller blades **112** of the impeller **106**. The impeller blades **112**, which may be part of the impeller **106**, may be any shape or size. Likewise, the fluid channels **114** formed between the impeller blades **112** to allow passage of cooling fluid through their length may be any size or shape suitable to allow passage of cooling fluid. The centrifugal force created when the impeller **106** pushes cooling fluid through the fluid channels **114** from the inside of the impeller blade **112** towards the impeller housing **102** to the outside of the impeller blade **112**.

[0020] The cooling fluid exiting the fluid channels **114** of the impeller blades **112** may be collected by an involute. An involute may be any geometry that collects fluid exiting the impeller blades **112** and efficiently increases the static pressure of the fluid before it exits the involute. The involute may accomplish the increased static pressure by converting the dynamic pressure resulting from the circumferential velocity of the fluid and converting it to static pressure of the cooling fluid. The involute may advantageously be a double involute **132** having two involute channels **134** over at least part of its length. The double involute **132** may be positioned on the exterior of the impeller housing **102** and wrapped around the circumference of the impeller housing **102**. The double involute **132** may have two involute tongues **130** (which may also be known as cutwaters). The involute tongue **130** may represent the closest point of the double involute **132** to the fluid channel exit **114** and as such is the point where the double involute **132** interacts with the cooling fluid exiting the impeller **106**.

[0021] The conversion of the velocity of the cooling fluid to an increase in static pressure of the fluid in the involute **132** may result in a reactionary force on the impeller **106**. Due to the close proximity of the involute tongue **130** to the impeller blades **112**, the involute tongue **130** may generate a reaction force on the impeller blades **112**, thus creating a radial and tangential reaction force in the impeller **106** itself. The tangential reaction force may be overcome with the torque capability of the motor (the combined motor stator **104** and motor magnet **108**). The net radial load (both its magnitude and direction) may result from the integration of the radial reaction forces generated by the double involute **132** over its entire circumference of 360 degrees. By employing two involute channels **134** and involute tongues **130** that are approximately 180 degrees apart, the double involute **132** may substantially balance the impeller loading, resulting in smaller net radial loads when compared to single involute designs. To accomplish this, the double involute **132** may generate equal and opposite loads at each involute which cancel out to create a relatively small net radial load. To help maintain a low net radial load, the impeller **106** may have an even number of impeller blades **112** evenly distributed around the impeller **106**, resulting in an impeller blade **112** passing both involute tongues **130** simultaneously and resulting in generated loads that are equal and opposite. Single involute designs may result in such high radial loads that hydrodynamic bearings are not viable and more expensive mechanical bearing approaches must be employed. A double involute **132** may thus provide a more balanced and lower net radial load on the bearings of the centrifugal pump **100** when compared to a single involute.

[0022] In an alternative embodiment, the centrifugal pump **100** may have more than two involutes instead of a double involute **132**. In this embodiment, the number of impeller blades **112** may be matched to the number of involutes to balance the net radial loads. A three involute centrifugal pump **100**, for example, may have a plurality of impeller blades **112** in a multiple of three (e.g., 3, 6, 9, 30, etc.) in order to substantially balance the loads. Similarly, a five involute centrifugal pump **100** may have impeller blades **112** in multiples of five (e.g., 5, 10, etc.) to properly balance the loads. A centrifugal pump **100** with an even number of involutes (e.g., 2, 4, 6, etc.) may have an even number of impeller blades **112** to provide load balancing. A centrifugal pump **100** with the number of involutes and the number of impeller blades **112** matched to achieve load balancing may have significantly less net radial loads than a single involute design, regardless of the number of impeller blades **112**. While a double involute centrifugal pump **100** is depicted in the Figures and described herein, one skilled in the art will recognize that other multiple involute designs may also be used.

[0023] Hydrodynamic bearings such as the journal bearing **122** and thrust bearing **124** may support the loads generated by the operation of the centrifugal pump **100**. The journal bearing **122** may support the rotating impeller **106** and motor magnet **108** and a combination of the journal bearing **122** and thrust bearing **124** may support the net radial load and force moment associated with the development of mass flow and static pressure on the impeller blades **112**. The thrust bearing **124** may also help balance component gravity loads (depending on the orientation of the centrifugal pump **100**) as well as support the moment loads generated by the offset between the journal bearing **122** surface and the impeller **106** height difference (where radial load is generated). The journal bearing **122** may be created or positioned in the gap between the motor magnet **108** and/or impeller body **116** and the impeller housing **102** in the region near the motor stator **104** (or the surface of the motor stator **104** itself). The journal bearing **122** may include a hydrodynamic fluid located in the gap between the rotating cylinder (the impeller **106**) and the stationary cylinder (the motor stator **104** or impeller housing **102**). The journal bearing **122** may be created or positioned either on the outside of the rotating cylinder, the inside of the rotating cylinder, or both. The thrust bearing **124**, on the other hand, may be created or positioned in the gap between the bottom of the impeller **106** (e.g., the disk surface of the impeller **106**) and the impeller housing **102** and/or the gap between the top of the impeller blades **112** and the impeller housing **102**.

[**0024**] The journal bearings **122** and thrust bearings **124** are hydrodynamic in nature and their performance may typically be dominated by the viscous effects between the stationary (i.e., the impeller housing **102**) and moving surfaces (i.e., the impeller **106**). The effectiveness of the hydrodynamic lubrication may be closely coupled to surface geometry and interfaces (i.e., gaps between surfaces). The effectiveness and load capability of the journal bearing **122**, for example, may be related to the eccentricity of the inner cylinder (i.e., the shape of the impeller **106**), the viscosity of the fluid in the gap, the speed of the rotating impeller **106**, the radius of the inner cylinder, and the radial clearance between the two cylinders (i.e., the gap), or other factors. For example, a reduction in the gap of the journal bearing **122** may improve the bearing performance. A gap reduction may also improve the efficiency of the motor by bringing the motor magnet **108** and motor stator **104** closer together and increasing the strength of the generated magnetic field on the motor magnet **108** and reducing the unused field, or magnetic leakage, between the two components. Improved motor efficiency may allow for less complicated control circuitry or less windings. Reduction of the gap, on the other hand, may also increase the torque losses in the motor due to the increase in shear losses in the fluid, and an optimal gap may exist based on balancing the different factors and the particular configuration. Thrust bearings **124** are impacted by similar factors as journal bearings **122**, such as by the geometry of the bearing surfaces, the speed of rotation, etc.

[**0025**] FIG. **2** depicts a side cut-away view of a hydrodynamic thrust bearing of the centrifugal pump of FIG. **1** according to one embodiment. In FIG. **2**, the thrust bearing **124** is positioned between a housing inner surface **206** of the impeller housing **102** and a thrust bearing surface **204** of the impeller body **116** part of the impeller blade **112**. The distance between the impeller body **116** and the impeller housing **102** that forms the thrust bearing **124** is the thrust bearing gap **202**. As described in relation to FIG. **1**, a fluid channel **114** may be positioned on the opposite side of the impeller body **116** from the thrust bearing **124**. While FIG. **2** is described as a thrust bearing **124** herein, the description of FIG. **2** is equally applicable to a journal bearing **122** with, for example, a motor magnet **108** replacing the impeller blade **112** or impeller body **116** in the description.

[**0026**] As described previously, the size of the thrust bearing gap **202** may impact the performance of the thrust bearing **124**. A smaller thrust bearing gap **202** may result in improved load capacity of the thrust bearing **124** while larger thrust bearing gaps **202** may result in reduced capacity. In one embodiment, the geometry and tolerances associated with the bearing surfaces (thrust bearing surface **204** and housing inner surface **206**) may be such that the thrust bearing gap **202** does not grow so large that the thrust bearing **124** cannot support the required loads. Tighter manufacturing tolerances of the housing inner surface **206** and thrust bearing surface **204**, as examples, and reduced size of the thrust bearing gap **202** may therefore provide for more consistent and predictable load bearing capability as well as increased capability. Close manufacturing tolerances may also prevent gross instabilities in the position of the impeller **106** and therefore load fluctuations on the impeller **106**, as the loading on the impeller blades **112** may be dependent on their distance in interaction with the double involute **132** geometry.

[**0027**] In one embodiment, a journal bearing **122** may be constructed with less expensive materials while still maintaining sufficiently close tolerances. The close tolerances may include both dimensional geometric aspects of the journal bearing **122** cylinders (i.e., diameters and eccentricity of the hole and cylinder). Close tolerances may be achieved using cost-effective plastic injected part manufacturing by providing support to the resulting thermoplastic walls by using components that act as a stabilizing backbone. For example, the journal bearing thermoplastic surface (equivalent to the housing inner surface **206** of the thrust bearing **124**) may be stabilized by a metal motor stator **104**. Similarly, the journal surface (equivalent to the thrust bearing surface **204** of the thrust bearing **124**) may be stabilized with the motor magnet **108** ring. The composite construction may allow for more stable dimensional control and less geometric departure than simple plastic injection molded surfaces.

[**0028**] FIG. **3** depicts a cut-away plan view of a double involute of the centrifugal pump of FIG. **1** according to one embodiment. In the embodiment of FIG. **3**, the centrifugal pump **100** includes a double involute **132** around an impeller **106** that is adapted to rotate clockwise (in FIG. **3**) about a centerline **110**. The impeller **106** may include a plurality of impeller blades **112** with fluid channels **114** to transport cooling fluid from the impeller **106** to the double involute **132** at a high velocity. The walls of the impeller blades **112** depicted in FIG. **3** are shown with no thickness for purposes of clarity. The double involute **132** may collect the high velocity fluid exiting the fluid channels **114** and may efficiently increase the static pressure of the fluid by the geometry of the double involute **132**, resulting in a higher pressure fluid for use in a cooling system.

[**0029**] The double involute **132** of the disclosed embodiments includes two involute channels **134** each beginning at an involute tongue **130** and ending at an involute throat **302**. Fluid may enter each involute channel **134** at the involute tongue **130** and travel the length of the involute channel **134** between its walls until exiting the double involute **132** at the involute throat **302**. The involute tongues **130** are the start of the involute geometry and may represent the closest point that the involute wall interacts with the fluid exiting the impeller **106**. The involute throat **302** may be of any length or configuration and may fluidly connect to a cooling system. Due to its close proximity, the involute tongue **130** generates the largest reaction force on the impeller blades **112**, thus creating a radial and tangential reaction force in the impeller **106** itself. This tangential force may be overcome with the motor's torque capability. The integration of the radial reaction forces generated by the involute walls over their entire length generates the net radial load, including both its magnitude and direction.

[**0030**] The design of the double involute **132** helps reduce net radial loads in comparison to single involute designs. In the depicted double involute **132** design, the two involute tongues **130** (labeled 'A' and 'B' in FIG. **3**) are positioned opposite each other and approximately 180 degrees apart. This may substantially balance the impeller **106** loading and results in a relatively small net radial load. The outside involute channel **134**, which begins at involute tongue **130**'A', may receive fluid exiting impeller blades **112** positioned between involute tongue **130**'A' clockwise to involute tongue **130**'B' (from approximately the 9 o'clock position to approximately the 3 o'clock position). The fluid gathered by the outer involute channel **134** then travels around the channel until it exits at the involute throat **302**, giving it approximately 180 degrees of travel within the outer involute channel **134**. The inner involute channel **134**, which begins at involute tongue **130**'B', may receive fluid exiting impeller blades **112** positioned between involute tongue **130**'B' clockwise to involute tongue **130**'A' (from approximately the 3 o'clock position to approximately the 9 o'clock position). The fluid gathered by the inner involute channel **134** travels around the channel until it exits at the involute throat **302**, giving it up to 180 degrees of travel within the inner involute channel **134**. By employing two involute channels **134** and associated geometry that are approximately 180 degrees apart the impeller **106** loading may advantageously be directly balanced and the net radial load reduced when compared to single involute designs. Positioning of the involute tongues **130** in separate halves of the double involute **132** may be considered positioning the involute tongues **130** opposite of each other. While the two involute tongues **130** are described as being approximately 180 degrees apart, they may be positioned such that they are not directly opposite. For a typical geometry, the closer the two involute tongues **130** are to directly opposite each other, however, the more the net radial load is reduced.

[0031] The geometry of the components of the double involute **132** may impact its performance. For example, changes to the length and shape of the involute channels **134**, including the area growth of the channel, may cause changes in the pressure gain in the double involute **132** as well as the loads during operation. Similarly, changes to the final area output and shape of the involute throat **302** may result in changes in the pressure gain of the double involute **132**. The double involute **132** may in some embodiments have a configuration designed to optimize pressure gain, minimize net radial loads, or a combination of these or any other factors.

[0032] FIG. **4** depicts a flowchart of an embodiment to pump cooling fluid in a cooling system. In one embodiment, one or more components of a centrifugal pump **100** may perform the elements of flowchart **400**. In the depicted embodiment, flowchart **400** begins with element **402**, receiving cooling fluid into an impeller **106**, such as through the fluid entrance **126** of the impeller housing **102**. Flowchart **400** may then continue to element **404**, where the centrifugal pump **100** drives the impeller **106** to rotate about an axis, such as the pump centerline **110**. The centrifugal pump **100** may drive the impeller **106** by, in one embodiment, providing an electric charge to the motor stator **104**, which in turn drives the motor magnet **108** of the impeller **106**. At element **406**, the centrifugal forces generated by the rotating impeller **106** may then force the cooling fluid through fluid channels **114** of the impeller blades **112** to exit the impeller **106**.

[0033] After the cooling fluid exits the impeller **106**, two or more involutes (such as a double involute **132**) may receive the cooling fluid at element **408**. In one embodiment, the double involute **132** may receive the cooling fluid at two involute tongues **130**. The double involute **132** may next, through its geometry, increase the static pressure of the cooling fluid at element **410** by converting the velocity and dynamic pressure of the cooling fluid into static pressure. The double involute **132** may accomplish this by passing the cooling fluid through the involute channels **134** as the area of each involute channel **134** increases. At element **412**, any loads, such as a net radial load, generated by the involutes at elements **408** or **410** may be reacted by hydrodynamic bearings such as one or more journal bearings **122** and one or more thrust bearings **124**. The involutes may then pass the cooling fluid back into the cooling system (at a higher static pressure), after which the method of flow chart **400** terminates. While the elements of flow chart **400** are shown sequentially, many of the elements may be performed simultaneously. Reacting the generated loads at element **412**, for example, may be performed simultaneously and in response to increasing the static pressure of the cooling fluid at element **410**.

[0034] While certain operations have been described herein relative to a direction such as "above" or "below" it will be understood that the descriptors are relative and that they may be reversed or otherwise changed if the relevant structure(s) were inverted or moved. Therefore, these terms are not intended to be limiting.

[0035] It will be apparent to those skilled in the art having the benefit of this disclosure that the present invention contemplates a centrifugal pump with a double involute and hydrodynamic bearings. It is understood that the form of the invention shown and described in the detailed description and the drawings are to be taken merely as examples. It is intended that the following claims be interpreted broadly to embrace all the variations of the example embodiments disclosed.

[0036] Although the present invention and some of its advantages have been described in detail for some embodiments, it should be understood that various changes, substitutions and alterations can be made herein without departing from the spirit and scope of the invention as defined by the appended claims. Although an embodiment of the invention may achieve multiple objectives, not every embodiment falling within the scope of the attached claims will achieve every objective. Moreover, the scope of the present application is not intended to be limited to the particular embodiments of the process, machine, manufacture, composition of matter, means, methods and steps described in the specification. As one of ordinary skill in the art will readily appreciate from the disclosure of the present invention, processes, machines, manufacture, compositions of matter, means, methods, or steps, presently existing or later to be developed that perform substantially the same function or achieve substantially the same result as the corresponding embodiments described herein may be utilized according to the present invention. Accordingly, the appended claims are intended to include within their scope such processes, machines, manufacture, compositions of matter, means, methods, or steps.

WHAT IS CLAIMED IS:

1. A centrifugal pump comprising:

an impeller housing, the impeller housing comprising a fluid entrance to allow fluid to enter the impeller housing;

an impeller located within the impeller housing comprising a plurality of impeller blades, a plurality of fluid channels between the impeller blades, and a motor magnet, wherein the impeller rotates within the impeller housing about a pump centerline in response to an electromagnetic field, and wherein further the fluid channels allow fluid to pass through when the impeller rotates;

one or more hydrodynamic bearings positioned between the impeller and the impeller housing to support generated loads; and

a double involute coupled with the impeller housing and positioned to receive fluid exiting the plurality of impeller blades.

2. The centrifugal pump of claim 1, further comprising a motor stator positioned at least partially outside the impeller housing to generate an electromagnetic field in response to an electrical charge.

3. The centrifugal pump of claim 1, further comprising a motor stator positioned at least partially within the impeller housing to generate an electromagnetic field in response to an electrical charge.

4. The centrifugal pump of claim 1, wherein the impeller housing is comprised of an injection molded plastic material.

5. The centrifugal pump of claim 1, wherein the impeller further comprises an impeller body coupled to the motor magnet.

6. The centrifugal pump of claim 5, wherein the impeller body is positioned between the pump centerline and the motor stator.

7. The centrifugal pump of claim 5, wherein the motor stator is positioned between the pump centerline and the impeller body.

8. The centrifugal pump of claim 1, wherein the impeller is at least partially comprised of an injection molded plastic material.

9. The centrifugal pump of claim 1, wherein the plurality of impeller blades are an even number of impeller blades.

10. The centrifugal pump of claim 1, wherein the plurality of impeller blades are substantially evenly distributed around the impeller.

11. The centrifugal pump of claim 1, wherein the one or more hydrodynamic bearings comprise a journal bearing positioned between the impeller housing and the impeller.

12. The centrifugal pump of claim 1, wherein the one or more hydrodynamic bearings comprise a thrust bearing positioned between the impeller housing and the impeller blades.

13. The centrifugal pump of claim 1, wherein the double involute further comprises:

two involute tongues to receive fluid from the fluid channels;

an involute throat to allow exit of fluid from the double involute; and

two involute channels to pass fluid from the involute tongues to the involute throat.

14. The centrifugal pump of claim 13, wherein the two involute tongues are positioned opposite each other.

15. A method for pumping cooling fluid, the method comprising:

receiving cooling fluid into an impeller;

driving the impeller to rotate about an axis to force the cooling fluid to exit the impeller;

receiving exiting cooling fluid at two or more involutes;

increasing a static pressure of the cooling fluid in the two or more involutes; and

reacting any loads generated by the two or more involutes and rotating impeller with one or more hydrodynamic bearings.

16. The method of claim 15, further comprising after increasing the static pressure of the cooling fluid, passing the cooling fluid to a cooling system.

17. The method of claim 15, wherein driving the impeller to rotate about an axis comprises generating an electromagnetic field to drive the impeller to rotate.

18. The method of claim 15, wherein increasing the static pressure of the cooling fluid in the two or more involutes comprises passing the cooling fluid through involute channels with increasing area.

19. The method of claim 15, wherein reacting any loads generated by the two or more involutes and rotating impeller with one or more hydrodynamic bearings comprises reacting any loads generated by the two or more involutes and rotating impeller with a journal bearing.

20. The method of claim 15, wherein reacting any loads generated by the double involute and rotating impeller with one or more hydrodynamic bearings comprises reacting any loads generated by the two or more involutes and rotating impeller with a thrust bearing.

21. A centrifugal pump comprising:

an impeller housing, the impeller housing comprising a fluid entrance to allow fluid to enter the impeller housing;

an impeller means to rotate in response to an electromagnetic field and to allow fluid to pass through during rotation;

a hydrodynamic bearing means positioned between the impeller and the impeller housing to support generated loads; and

a double involute coupled with the impeller housing and positioned to receive fluid exiting the plurality of impeller blades.

22. The centrifugal pump of claim 21, further comprising an electromagnetic field generating means to generate an electric field.

23. The centrifugal pump of claim 21, wherein the double involute further comprises:

two involute tongues to receive fluid from the fluid channels;

an involute throat to allow exit of fluid from the double involute; and

two involute channels to pass fluid from the involute tongues to the involute throat.

24. A centrifugal pump comprising:

an impeller housing, the impeller housing comprising a fluid entrance to allow fluid to enter the impeller housing;

an impeller located within the impeller housing comprising a plurality of impeller blades, a plurality of fluid channels between the impeller blades, and a motor magnet, wherein the impeller rotates within the impeller housing about a pump centerline in response to an electromagnetic field, and wherein further the fluid channels allow fluid to pass through when the impeller rotates;

one or more journal bearings positioned between the impeller and the impeller housing to support generated loads;

one or more thrust bearings positioned between the impeller and the impeller housing to support generated loads; and

two or more involutes coupled with the impeller housing and positioned to receive fluid exiting the plurality of impeller blades.

25. The centrifugal pump of claim 24, further comprising a motor stator positioned at least partially outside the impeller housing to generate an electromagnetic field in response to an electrical charge.

26. The centrifugal pump of claim 24, further comprising a motor stator positioned at least partially within the impeller housing to generate an electromagnetic field in response to an electrical charge.

27. The centrifugal pump of claim 24, wherein the impeller is at least partially comprised of an injection molded plastic material.

28. The centrifugal pump of claim 24, wherein the two or more involutes comprise a double involute, comprising:

two involute tongues to receive fluid from the fluid channels;

an involute throat to allow exit of fluid from the double involute; and

two involute channels to pass fluid from the involute tongues to the involute throat.

29. The centrifugal pump of claim 28, wherein the two involute tongues are positioned opposite each other.

* * * * *