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DESCRIPTION

CROSS REFERENCE TO RELATED APPLICATIONS

FIELD OF THE INVENTION

[0001] The present invention generally relates to centrifugal pumps, such as, for example, centrifugal pumps having impellers of radial, Francis vane, mixed flow, and axial flow design. More specifically, the present invention relates to an impeller and casing for centrifugal pumps that may produce a high head output and high efficiency, while also being capable of pumping shear sensitive liquids or liquids having suspended solids without applying damaging forces to the liquid or the solids.

BACKGROUND OF THE INVENTION

[0002] Conventional centrifugal pumps include an impeller that rotates within a cavity in the body of the pump. Fluid entering from an inlet in the cavity typically flows toward the impeller and near to the impeller's center of its rotation. Further, the rotation of the impeller typically forces fluid to flow radially outward toward an outlet of the cavity that is often at a location that is radially adjacent to the impeller.

[0003] Producing high head output by centrifugal pumps often requires that the impeller be rotated at accelerated speeds. However, such accelerated speeds are typically associated with the generation of a relatively significant shearing force that is applied to the fluid that is flowing through the pump. Yet such shearing forces may be unacceptable for at least certain types of fluids and/or solids that are passing through the pump. For example, food processing systems, pharmaceutical processing systems, and clay slurries, are examples of applications in which a high shearing force may be unacceptable due to the potential damage that such shearing forces may cause to the structure of the fluid and/or the solids within the fluid. Thus, in applications in which the fluid or solids flowing through the pump should not be subjected to such shearing forces, typically the impeller may be operated at a low pump speed and have a low head output. Moreover, to avoid and/or minimize the generation of such shearing forces, the total head generation capability of the centrifugal pumps may be limited or centrifugal pumps may not be used in such applications.

[0004] Additionally, low shear centrifugal pump designs, particularly food grade pumps, have relatively lower efficiencies than standard industrial centrifugal pumps. Thus, low shear centrifugal pump designs often result in pumps that have more internal recirculation of fluids and/or solids within the pump and have higher power requirements.

[0005] A centrifugal pump comprising the features of the preamble of claim 1 is known from WO 2005/100796 A1, i.e. an open or half- open channel impeller for a centrifugal pump for waste-water. The channel impeller has a substantially circular periphery and comprises a central hub with a hub axis and a hub plane to which the hub axis extends perpendicularly. The channel impeller further comprises at least one blade which extends from the hub to the periphery.

[0006] Further, US 2003/007871 A1 teaches an impeller for a centrifugal pump. The impeller is particularly suited for use in pumps in which a high head is required and in which only low shear forces must be applied to the fluid moving through the pump. The impeller comprises vanes which sweep an arc around an impeller axis to provide a smooth path past the impeller and through the pump. The vanes of the impeller are formed to cause the fluid moving over the vanes to apply a hydrodynamic force to the vane that opposes the force applied to the vane by fluid as the vane urges fluid through the pump. An impeller according to this invention does not require the supporting structures that are required by known impellers.

BRIEF SUMMARY OF THE INVENTION

[0007] The disadvantages and limitations of known impeller centrifugal pumps can be overcome by providing an impeller that subjects the fluid moving through the pump to lower shear forces than known centrifugal pump impellers.

[0008] In particular, the centrifugal pump of the invention comprises the features of claim 1 to overcome the disadvantages and limitations of known impeller centrifugal pumps.

[0009] Moreover, the method for pumping shear sensitive liquids or liquids having suspended solids according to claim 7 comprises: providing such a centrifugal pump, providing a liquid to be pumped at the inlet, and rotating the impeller to pump the liquid from the inlet to the outlet.

[0010] Test results show that, when pumps employing the claimed impeller and casing are used in certain dairy processing applications, the acid degree value of the milk does not increase as a result of pumping. An increase in acid degree value typically serves as an indicator that the fat globules in the milk have been damaged due to mechanical shearing. Accordingly, the claimed impeller and casing cause less damage to the milk. This advantageous result would also benefit other applications beside dairy processing systems, such as food processing systems, pharmaceutical processing systems, and clay slurries.

[0011] These and other objects and advantages of the impeller and/or casing described in this disclosure will be understood from the following description and drawings of exemplary embodiments of an impeller and casing.

BRIEF DESCRIPTION OF SEVERAL VIEWS OF THE DRAWINGS

[0012]

Figure 1 illustrates an isometric view of an embodiment of the inlet side of an impeller.

Figure 2 illustrates an inlet side view of the impeller shown in Figure 1.

Figures 3A and 3B illustrate side elevation views of the impeller shown in Figure 1.

Figure 4 illustrates an isometric view of the impeller shown in Figure 1.

Figure 5 illustrates a rear view of an impeller according to an illustrated embodiment.

Figure 6 illustrates a side cross sectional view of a casing according to an illustrated embodiment.

Figure 7 illustrates a partial cross sectional view of an impeller assembly having an impeller, casing, and a motor according to an illustrated embodiment.

[0013] The following reference characters are used in the specification and figures:

10	Impeller
11	Front side
12	Shroud
13	Backside
14a,b	Vane(s)
16	Hub
17	Orifice
18	Impeller axis
19	Hub protrusion
20	High pressure surface
22	Low pressure surface
24	Upper vane surface
26	Leading edge
28	Trailing edge
30	Lower leading edge
31	Lower trailing edge
32	Lower vane body
33	Central axis
34	Lower leading surface
35	Vane edge

36	Lower trailing surface
37	Casing
38	Inlet orifice
40	Sidewall
42	Front wall
43	Inlet port
44	Cavity
45	External thread
46	Discharge port
48	Outlet orifice

DETAILED DESCRIPTION OF THE INVENTION

[0014] Figures 1-5 illustrate an embodiment of an impeller 10 according to the present disclosure. In the illustrated embodiment, the impeller 10 is a radial impeller that includes a shroud 12, at least two vanes 14a, 14b, and a generally central hub 16. According to certain embodiments, vanes 14a, 14b and the shroud 12 may be part of a single, integral construction. The hub 16 may extend from a front side 11 of the shroud 12 and be positioned along an impeller axis 18. Further, the hub 16 may have a variety of different configurations, including, for example, being generally cylindrical. Additionally, according to certain embodiments, the shroud 12 and/or hub 16 may be configured to be operably connected to a drive shaft, such as, for example, to an impeller shaft that is used to rotate the impeller 10 about the impeller axis 18. For example, the impeller shaft may be used to rotate the impeller 10 in a circumferential rotation direction R_o , as indicated in Figure 2.

[0015] Referencing at least Figures 4 and 5, the impeller 10 may include an orifice 17 that is configured for connecting the impeller 10 to the impeller shaft. For example, according to certain embodiments, the orifice 17 may include an internal thread that is configured for a threaded connection with an external thread of the impeller shaft or a coupling used to connect the impeller 10 to the impeller shaft. Alternatively, the orifice 17 may be sized to receive a portion of the impeller shaft and may include one or more slots that are configured for a keyed connection between the impeller 10 and the impeller shaft.

[0016] Further, according to certain embodiments, the orifice 17 may pass through a hub protrusion 19 that extends outwardly from a backside 13 of shroud 12, the backside 13 being on a side of the shroud 12 that is opposite of the front side 11 (i.e., the side containing vanes 14a, 14b). According to certain embodiments, the hub protrusion 19 may be sized to space at least a portion of the shroud 12 from an adjacent wall of a casing. Further, according to certain embodiments, the hub protrusion 19 may be sized to receive a set screw that is used to at

least assist in securing the impeller 10 to the impeller shaft. For example, the hub protrusion can be about 0.25 mm (0.01") to about 2.54 mm (0.1"), such as about 0.76 mm (0.03").

[0017] In the illustrated embodiment, the impeller 10 has two vanes 14a, 14b that extend radially outwardly from the hub 16. Moreover, the two vanes 14a, 14b extend from two locations that are spaced equidistantly around the circumference of the hub 16. While other embodiments of the impeller 10 may utilize more than two vanes 14a, 14b, a two vane 14a, 14b configuration may enhance the overall hydraulic balance of the impeller 10.

[0018] Each vane 14a, 14b defines a high pressure surface 20 and a low pressure surface 22. As best shown by Figures 2, 3B, and 7, when positioned within a casing 37, the low pressure surface 22 faces partially outwardly along the impeller axis 18 toward an inlet orifice 38 of the casing 37. Conversely, the high pressure surface 20 faces partially along the impeller axis 18 away from the inlet orifice 38. Further, each vane 14a, 14b has an upper vane surface 24 that lies in a plane that is generally perpendicular to the impeller axis 18. The upper vane surface 24 meets the high pressure surface 20 along a leading edge 26. Additionally, the upper vane surface 24 meets the low pressure surface 22 along a trailing edge 28.

[0019] According to the invention, each vane 14a, 14b extends along the hub 16 to a lower vane body 32. According to the illustrated embodiment, the lower vane body 32 may extend along the front side 11 of the shroud 12. Further, the lower vane body 32 extends along the front side 11 of the shroud 12 about a central axis 33 that lies in a plane that is perpendicular to the impeller axis 18. The lower vane body 32 also includes a lower leading surface 34 and a lower trailing surface 36. The lower leading surface 34 meets the high pressure surface 20 at a lower leading edge 30. The lower trailing surface 36 meets the low pressure surface 22 at a lower trailing edge 31.

[0020] Each vane 14a, 14b extends along the hub 16 from the upper vane surface 24 to the lower vane body 32 and sweeps an arc around the hub 16 in a circumferential direction from the leading edge 26 toward the trailing edge 28 that is opposite the circumferential rotation direction R_0 . The vane 14a, 14b may sweep an arc around the impeller axis 18 so that the cord length for the leading edge 26 of the upper vane surface 24 to the lower trailing edge 31 achieves a solidity ratio to the vane spacing or pitch of at least 0.46:1.

[0021] Figure 6 illustrates a cross sectional side view of a casing 37 according to an illustrated embodiment of the present disclosure. The casing 37 includes a sidewall 40 and a front wall 42 that generally define a cavity 44 of the casing 37. The sidewall 40 and front wall 42 may include a variety of recesses, protrusions, and/or shoulders. For example, as shown in Figures 6 and 7, the front wall 42 may include an inlet port 43 having an inlet orifice 38 that is in fluid communication with the cavity 44. Similarly, the sidewall 40 may include a discharge port 46 having an outlet orifice 48 that is in fluid communication with the cavity 44. The inlet port 43 may be configured for an operable connection to a supply line that is used in the delivery of fluid and/or solids to the inlet orifice 38. Similarly, the discharge port 46 may be configured for an operable connection with a discharge line that receives fluids and/or solid that is exiting the

casing 37. For example, according to certain embodiments, the inlet and discharge ports 43, 46 may be configured for mechanical connection with the supply or discharge lines, respectively, such as a clamped, threaded, or compression engagement, among other connections. In the illustrated embodiment, the inlet and discharge ports 43, 46 each include an external thread 45 that is configured for an operable connection with the associated supply or discharge line or associated couplings or connector(s). However, the inlet and discharge ports 43, 46 may be configured for a variety of other connections with the associated supply or discharge lines, including, for example, welded or soldered connections, among others.

[0022] Referencing Figure 3B, according to certain embodiments, the height ("H") of the impeller 10 between the upper vane surface 24 and the front side 11 of the shroud 12 is generally equal to the diameter of the outlet orifice 48 of the discharge port 46. The arc swept by the vane 14a, 14b (from upper vane surface 24 along the impeller axis to the lower vane body 32) extends the high pressure surface 20 extends the acceleration distance and thereby decreases the shear forces applied to fluid moved by the impeller 10 to diminish damage that such forces may cause. The sweep of the vane 14a, 14b and ratio of the swept arc to impeller height provides relatively gentle re-direction of the liquid and/or solids in the cavity 44 of the casing 37, thereby reducing abrupt changes in direction for the liquid and/or solids being moved within the cavity 44 and increases overall pump efficiency.

[0023] As shown by at least the leading edge 26 and trailing edge 28 as illustrated in Figure 2, each vane 14a, 14b may be formed so that the distance between the high pressure surface 20 and the low pressure surface 22 increases as the distance away from the hub 16 increases to a distance R. By increasing the distance between the leading and trailing edges 26, 28 as the distance away from the hub 16 increases, the length of a slip path along the high pressure surface 20 in a direction from the hub 16 toward the vane edge 35 may also be increased. The longer slip path may decrease the amount of fluid and/or solids that can travel over the high pressure surface 20 to and around the vane edge 35 to the low pressure surface 22, thereby reducing recirculation of fluid and/or solids around the impeller 10 and increasing pumping efficiency.

[0024] Reducing recirculation around the vane edge 35 reduces the chances of damaging any fluid and solids entrained in the fluid. The wide slip path on vane surfaces 22 and 24 makes the transit of the liquid from the high pressure side of the impeller to the low pressure side difficult. A tight mechanical tolerance between the pump casing and the upper vane surface 42 makes this design highly efficient as it reduces the liquids ability to recirculate inside the pump. In addition to the wide area of the slip path, the integral rear shroud limits recirculation from the high pressure to low pressure thus eliminating the liquids ability to recirculate at the back of the impeller, further improving the efficiency of the pump.

[0025] As shown in at least Figure 7, when positioned in the casing 37, the shroud 12 is positioned axially behind the vanes 14a, 14b. Further, the shroud 12 has generally the same or similar outer diameter as the impeller 10. More specifically, the shroud 12 has a radius from the impeller axis 18 that is similar to the distance from the impeller axis 18 to the vane edge

35. The thickness of the integral shroud, as a ratio of the impeller height, is determined to be about 0.337. The shroud serves to offset the impeller axially away from the back of the casing and, more particularly, forward from the casing discharge port.

[0026] The front of the casing consists of two concentric radii from the central axis. The major diameter D_1 is axially rearward and of sufficient size beyond the impeller diameter to facilitate efficient transfer from kinetic to potential energy, as understood in the art. The height of the major diameter is equal to the diameter of the outlet port. The minor diameter D_2 is axially forward and is the same diameter as the impeller plus that which is necessary for mechanical clearance (e.g., the minimum clearance between a vane edge of the impeller and the casing at the minor diameter is about 0,5 mm (0.02")). The height of the minor diameter is equivalent to that of the impeller shroud. The transition from minor to major casing diameter is stepped such that there is a 90° angle from the major diameter to a transition step that is perpendicular to the axis and a 90° angle from the transition step to the minor diameter. This stepped casing provides a narrowing fluid channel from the axial front to the axial rear as the fluid translates from the impeller hub to the impeller periphery. This channel provides a smooth and efficient path while limiting recirculation and therefore improving pump efficiency, both of which result in lower fluid and solids damage.

[0027] The impeller described in this disclosure provides a centrifugal impeller and casing which can pump shear sensitive and high solids liquids with high efficiencies and low product damage. The helical vane sweep induces laminar flow. The impeller vanes, shroud, and casing reduce recirculation and assist inducement of laminar flow, therefore requiring less power.

[0028] One metric used in the dairy industry to measure the quality of milk is the acid degree value ("ADV"). The ADV measures the presence of long chain fatty acids in the milk. There is a correlation between the ADV and the flavor of milk because rancidity results from the release of free fatty acids in the milk. When used in dairy processing applications, conventional pumps typically produce an undesirable increase in the ADV of the milk as a result of fat globule damage due to mechanical shearing. This increase in the ADV can negatively affect the taste of the milk. In contrast, when a pump employing the claimed impeller and casing is used to pump milk, there is either no significant change in the ADV level as a result of pumping or even a decrease in the ADV level. This advantageous result reflects that pumps employing the claimed impeller and casing cause less product damage due to mechanical shearing than conventional systems.

[0029] This beneficial result was confirmed by two independent tests, the results of which are summarized in the working examples and Tables 1 and 2 below.

Example 1: Tests performed by Silliker, Inc.

[0030] The ADV levels of various milk samples were measured before pumping and after

pumping using a pump employing the claimed impeller and casing—namely, the Bowpeller model B3258 8" centrifugal pump—in Trials A and B and a competitor's conventional 8" centrifugal pump in Trials C and D. The results, summarized in Table 1, show that in Trials C and D, the ADV of the milk consistently increased as a result of pumping using the competitor's conventional pump, thereby indicating undesirable mechanical agitation and foaming of the milk due to pumping. However, Trials A and B show that the ADV of the milk consistently decreased (or at least did not change) as a result of pumping using the claimed impeller and casing—a highly desirable outcome.

Example 2: Tests performed by Eurofins DQCI LLC.

[0031] The ADV levels of various milk samples were measured before pumping and after pumping using a pump employing the claimed impeller and casing—namely, the Bowpeller model B15154 4" centrifugal pump—in Trials E and F and a competitor's conventional 4" centrifugal pump in Trials G and H. The results, summarized in Table 2, show that in Trials G and H, the ADV of the milk consistently increased as a result of pumping using the competitor's conventional pump, thereby indicating undesirable mechanical agitation and foaming of the milk due to pumping. However, Trials E and F show that the ADV of the milk consistently decreased (or at least did not change) as a result of pumping using the claimed impeller and casing—a highly desirable outcome.

[0032] This data confirms that a pump employing the claimed impeller and casing is capable of pumping shear sensitive liquids (such as milk) without applying damaging forces to the liquid. This result would also have beneficial application in food processing systems, pharmaceutical processing systems, and clay slurries.

TABLE 1

Tests Performed by Silliker, Inc.			
Pump	Acid Degree Value (Dairy Tank; Before Pumping)	Acid Degree Value (Truck Tanker; After Pumping)	Change in Acid Degree Value Due to Pumping
Applicant Trial A	0.98	0.97	-0.01
Applicant Trial B	0.99	0.95	-0.04
Competitor Trial C	0.90	0.94	+0.04
Competitor Trial D	0.86	0.94	+0.08

TABLE 2

Tests Performed by Eurofins DQCI LLC			
Pump	Acid Degree Value (Raw Milk; Before Pumping)	Acid Degree Value (Raw Milk; After Pumping)	Change in Acid Degree Value Due to Pumping
Applicant Trial E	0.82	0.73	-0.09
Applicant Trial F	0.82	0.69	-0.13
Competitor Trial G	0.63	0.68	+0.05
Competitor Trial H	0.63	0.77	+0.14

REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- [WO2005100796A1 \[0005\]](#)
- [US2003007871A1 \[0006\]](#)

P A T E N T K R A V

1. Centrifugalpumpe omfattende:

et pumpehjul (10) og et pumpehus (37);

hvor pumpehjulet (10) omfatter:

- 5 (a) et nav (16), som udstrækker sig langs en pumpehjulakse (18);
 (b) mindst to skovle (14, 14b), som har en spiralformet skovlkrumning, og som udstrækker sig fra navet (16) i en radial retning væk fra pumpehjulaksen (18) til en skovlkan-
 10 t ved den fjerneste udstrækning af hver skovl (14a, 14b) fra navet (16), hvor hver skovl (14a, 14b) udstrækker sig langs navet (16) i retning af pumpehjulaksen (18) fra et første sted til et andet sted, hvor retningen langs pumpehjulaksen (18) fra det første sted til det andet sted definerer en aksial indløbsretning,
 15 hvor hver skovl (14a, 14b) udstrækker sig om navet (16) i en første rundtgående retning for at overstryge en bue fra et første sted langs pumpehjulaksen (18) til et andet sted langs pumpehjulaksen (18), hvor hver skovl (14a, 14b) definerer:
 20 en højtryksoverflade (20), der vender mindst delvist langs den aksiale indløbsretning, hvor højtryksoverfladen (20) udstrækker sig fra en højtryksoverfladeforkant ved det første sted i den aksiale indløbsretning i den første rundtgående retning til en højtryksoverfladebagkant ved det andet sted, hvor højtryksoverfladeforkanten og højtryksoverfladebagkanten udstrækker sig udad fra navet (16) væk fra pumpehjulaksen (18),
 25 en lavtryksoverflade (22), der vender mindst delvist langs pumpehjulaksen (18) i en anden aksial retning, som er modsat den aksiale indløbsretning, hvor lavtryksoverfladen (22) er adskilt fra højtryksoverfladen (20) i den første rundtgående retning, hvor adskillelsen mellem højtryksoverfladen (20) og lavtryksoverfladen (22) i den første rundtgående retning øges med afstanden fra pumpehjulaksen (18) til et første sted tæt tilgrænsende den radiale skovlkan-
 30 t;
 og
 35 (c) en fuld cirkulær beklædning (12) af en diameter lig med den af pumpehjulet (19), som er orienteret og i ét stykke med den aksiale bagside af pumpehjulet (10), hvor beklædningen (12) desuden omfatter en frontside (11);

k e n d e t e g n e t ved, at forholdet mellem højden af beklædningen (12)

og pumpehjulskovlens aksiale højde er 0,337, hvor den aksiale højde af pumpehjulskovlen er afstanden mellem et første plan defineret af det første sted og et andet plan defineret af det andet sted, hvor det første og andet plan hver er vinkelret på pumpehjulaksen (18);

5 hvor hver skovl (14a, 14b) udstrækker sig langs navet (16) til et nedre skovllegeme (32), hvor det nedre skovllegeme (32) udstrækker sig langs frontsidens (11) af beklædningen (12) om en central akse (33), der ligger i et plan, som er vinkelret på pumpehjulaksen (18);

10 hvor det nedre skovllegeme (32) af hver skovl (14a, 14b) omfatter en nedre forkant (34) og en nedre bagkant (36), hvor den nedre forkant møder højtryksoverfladen (20) ved en nedre forkant (30), og den nedre bagkant (36) møder lavtryksoverfladen (22) ved en nedre bagkant (31).

2. Centrifugalpumpe ifølge krav 1, hvor de mindst to skovle (14a, 14b) definerer en øvre skovloverflade, som indeholder det første sted, og som udstrækker sig fra forkanten af højtryksoverfladen (20) til at møde lavtryksoverfladen (22) for at danne en øvre skovloverfladebagkant.

3. Centrifugalpumpe ifølge krav 2, hvor højtryksoverfladeforkanten (26) og den nedre skovloverfladebagkant (31) af hver skovl (14a, 14b) definerer en generelt ret linje, som udstrækker sig radialt væk fra pumpehjulaksen (18), og hver skovl (14a, 14b) overstryger en bue om pumpehjulaksen (18) således, at en linjelængde fra højtryksoverfladeforkanten (26) ved det første sted i den aksiale indløbsretning i den første rundtgående retning til den nedre skovloverfladebagkant (31) ved det andet sted af det aksiale udløb opnår en tæthedsgrad til skovlafstanden på mindst 0,46.

4. Centrifugalpumpe ifølge krav 1, hvor pumpehuset (37) omfatter:

25 en husudløbsåbning (46),

en husindløbsåbning (43), og

et hulrum (44), som tilvejebringer fluid forbindelse mellem husudløbsåbningen (46) og husindløbsåbningen (43), hvor hulrummet (44) er defineret af en største diameter (D_1) og en mindste diameter (D_2), hvor den største diameter (D_1) er centralt koncentrisk med den mindste diameter (D_2), og

30 begge diametre (D_1 , D_2) er centralt koncentriske med pumpehjulaksen (18), hvor den største diameter (D_1) er anbragt direkte bag den mindste diameter (D_2), og er adskilt med et trin, der er vinkelret på pumpehjulaksen (18),

hvor dybden af den største diameter (D_1) er ækvivalent med udløbsdiameteren,

35 hvor den mindste diameter (D_2) er diameteren af pumpehjulet (10) plus frigangen, som er nødvendig for at hindre mekanisk indgriben, og hvor dybden af den mindste diameter (D_2) er som nødvendigt for at acceptere balancen af pumpehjulets højde (H).

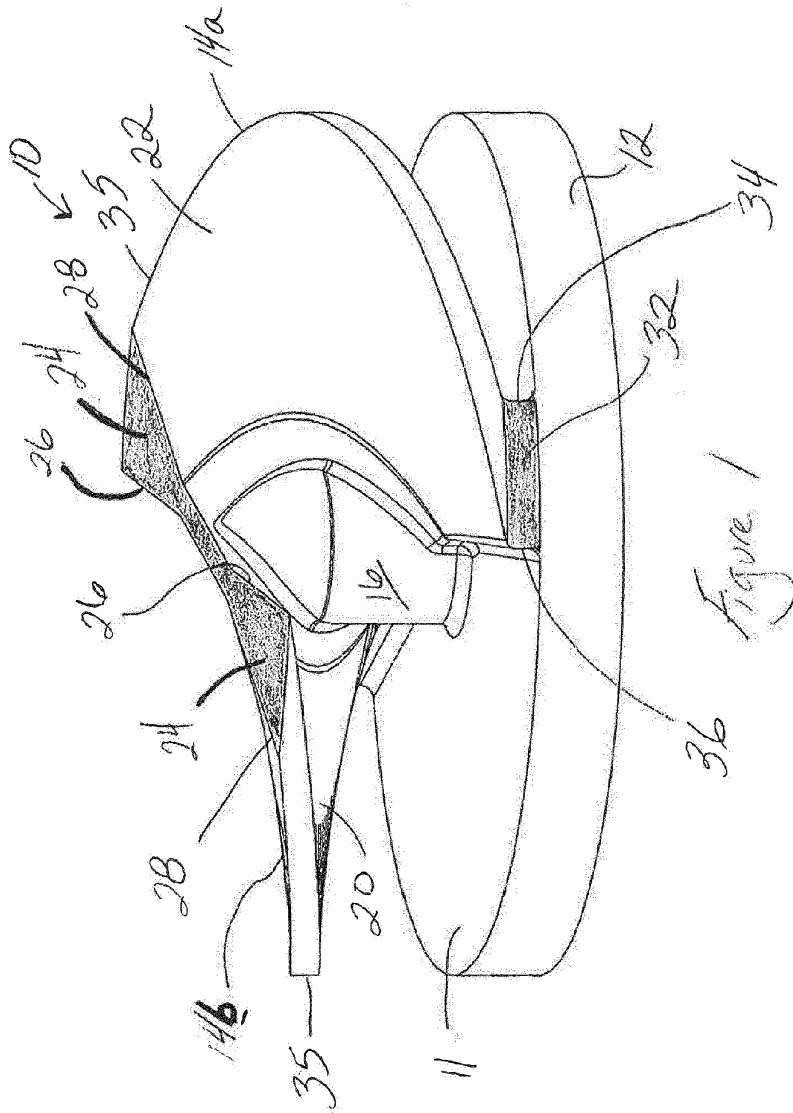
5. Centrifugalpumpe ifølge krav 4, hvor den aksiale bagside af pumpehjulbeklædningen (12) er positioneret inden for huset (37) i opretning med den aksiale bagside af den indvendige diameter af husudløbsåbningen (46).

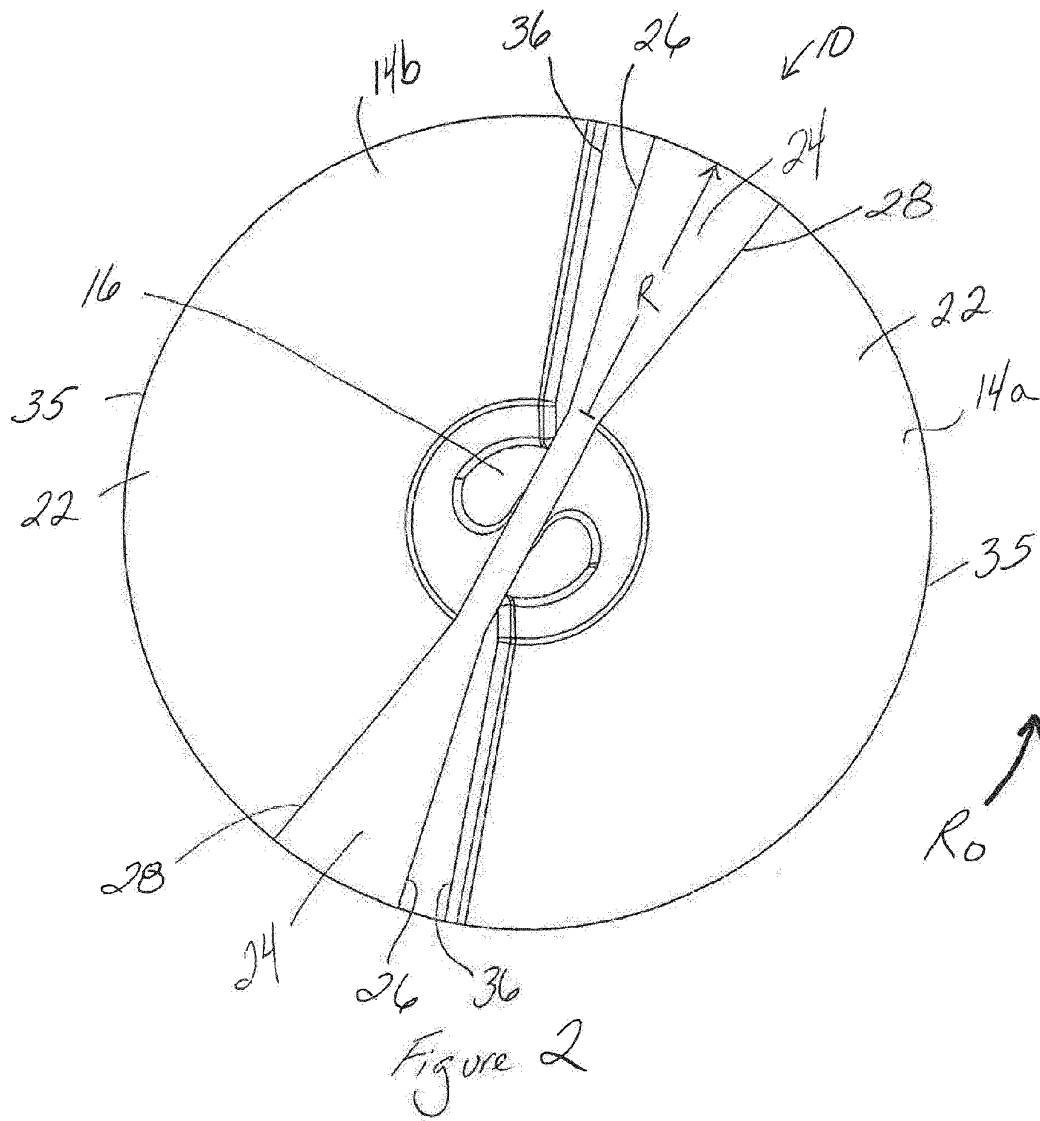
5 6. Centrifugalpumpe ifølge krav 1, hvor pumpehjulet (10) omfatter to skovle (14a, 14b).

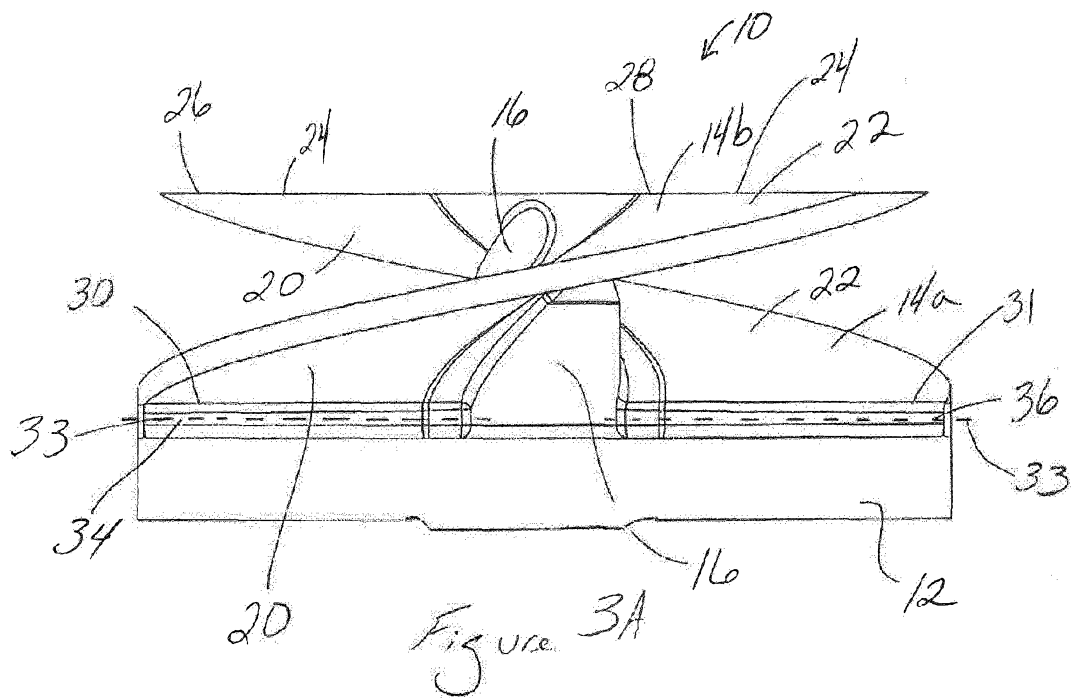
7. Fremgangsmåde til at pumpe forskydningsfølsomme væsker eller væsker, der har suspenderede faststoffer, omfattende:

- a. at tilvejebringe centrifugalpumpen ifølge ethvert af kravene 1 til 6,
- b. at tilvejebringe en væske, som skal pumpes, ved indløbet, og
- 10 c. at rotere pumpehjulet (10) for at pumpe væsken fra indløbet til udløbet.

DRAWINGS







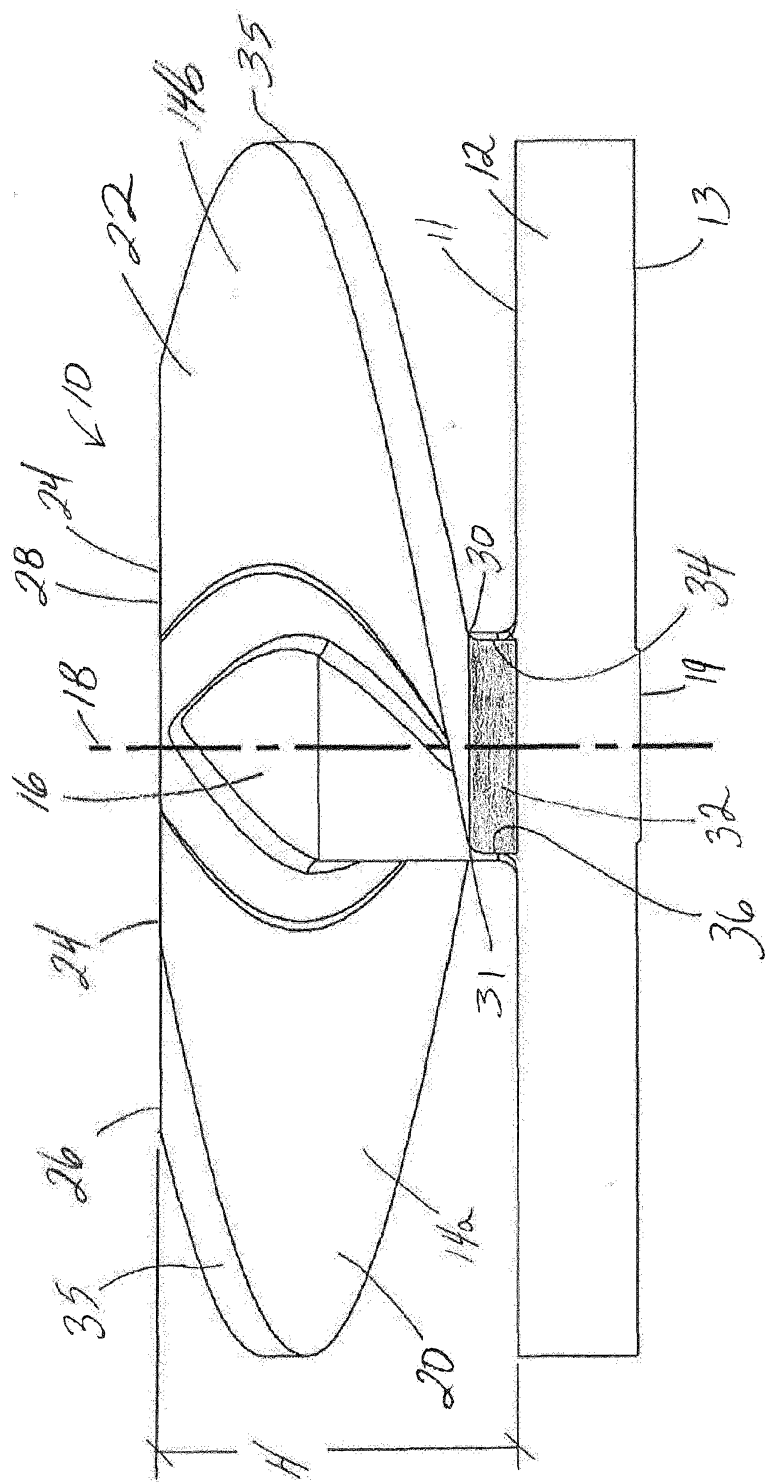
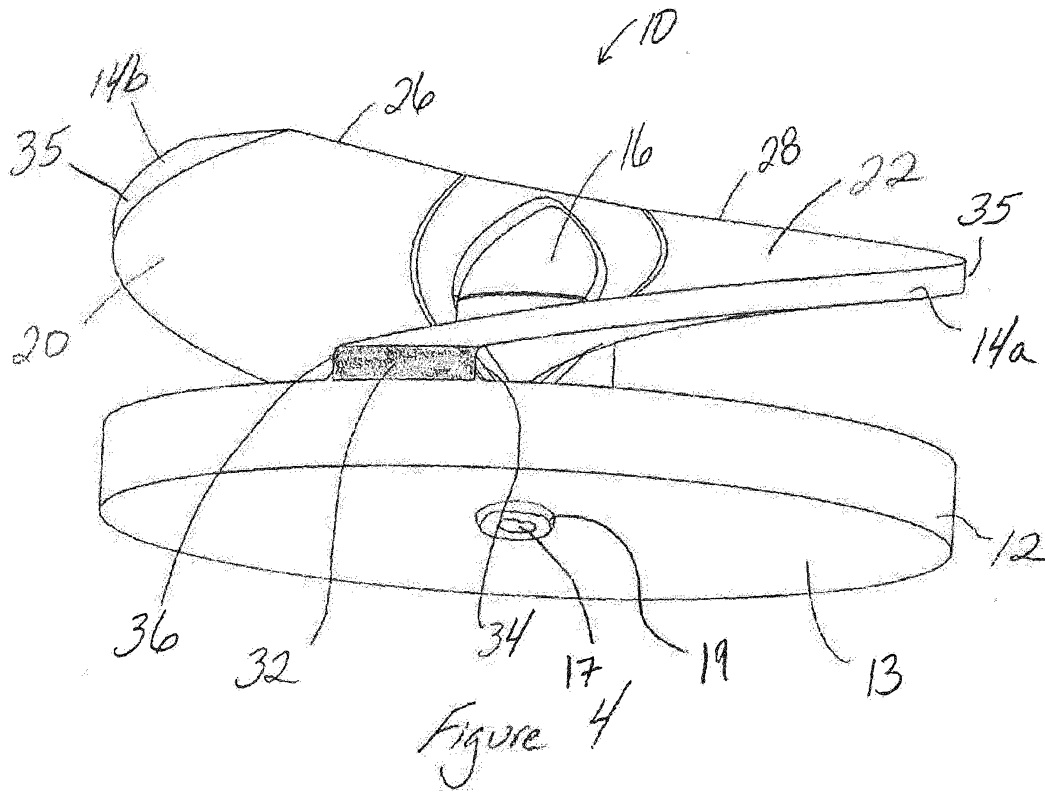


Figure 3B



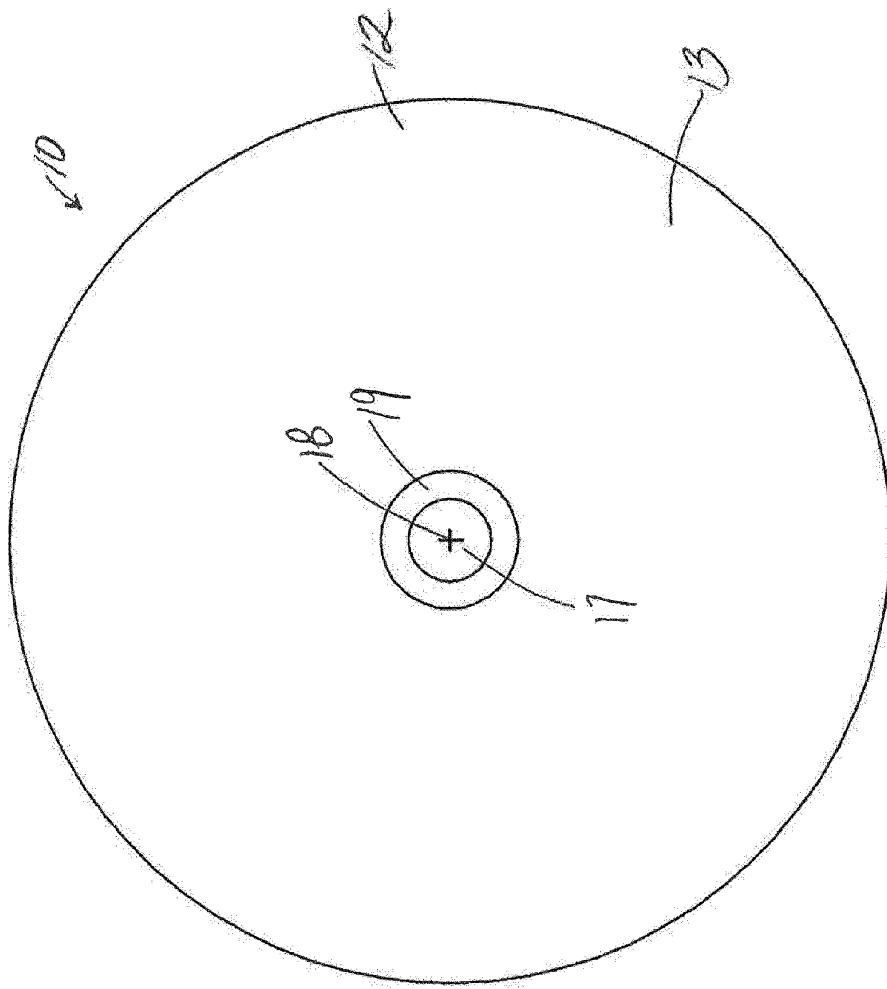


Figure 5

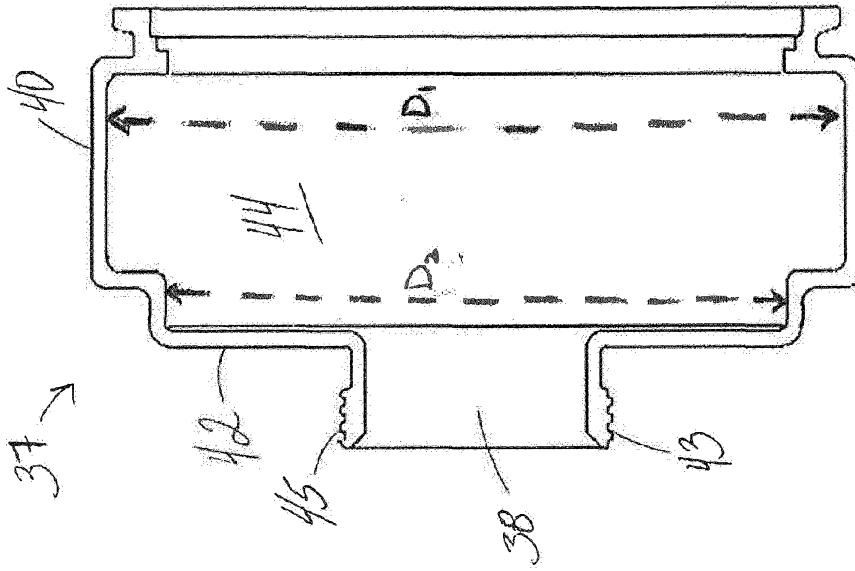


Figure 6

