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(54) **CENTRIFUGAL PUMP**

(57) The present invention relates to a centrifugal pump (1) with semi-open impeller (2) having a back plate (3), a number of blades (4) provided onto the back plate (3), and a front shroud (5, 7) arranged upstream of the impeller (2) at a defined distance above the blades (4), wherein onto the face of the front shroud (5, 6) directed towards the blades (4) a number of macroscopic grooves (5, 7) is provided for improved mixing and transport of gas-liquid two-phase flows.

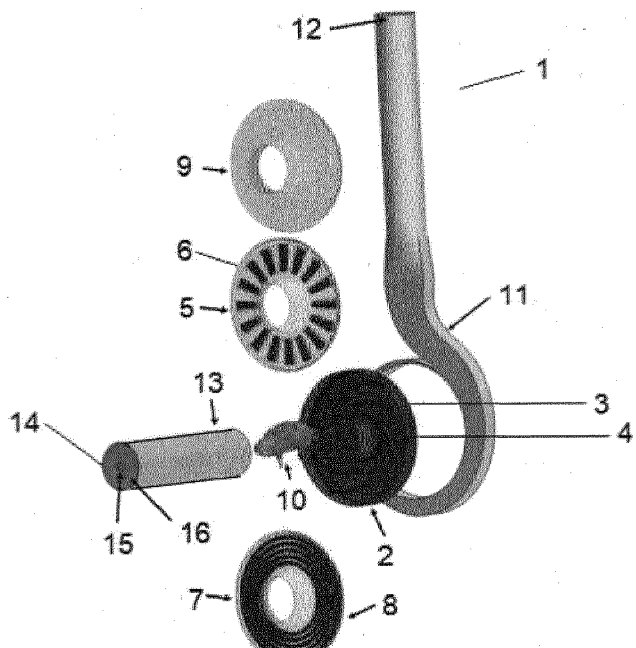

Fig. 1

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Description

[0001] The present invention relates to centrifugal pumps with semi-open impeller for transporting gas-liquid two-phase flows. In particular, the present invention relates to a design for centrifugal pumps with increased transport delivery capacity for gas-liquid two-phase flows.

[0002] Such centrifugal pumps comprise a casing, typically having a volute form surrounding the impeller said impeller comprising a back plate onto which is mounted a number of blades, with a free space between adjacent blades, said free space being referred to "channel".

[0003] Further, a front shroud is arranged above the blades at a defined distance, the tip clearance gap or, simply, gap. Depending on the size of the tip clearance gap between impeller and front shroud such impeller arrangement is known as semi-open impeller with standard gap and semi-open impeller with increased gap, respectively.

[0004] Centrifugal pumps are well known devices for energy conversion where rotational energy of an impeller is transferred to the fluid that is to be transported.

[0005] They are used in many industrial applications such as petroleum industry, chemical industry, power plants food production industry and so on. For single-phase operating conditions such as liquids only, the performance is reliable and consistent.

[0006] There are many applications which require the transport of two-phase flows (gas-liquid) by centrifugal pumps like medical treatment, paper industry, geothermal and nuclear plants.

[0007] However, for two-phase flow conditions a steep decline in performance is observed already as such a low gas content as 1 volume %. The main issue is the strong tendency of the gas content to accumulate and stay in the pump, blocking the impeller channels. At a higher gas content such as about 5 to 10 volume %, pump "breakdown" occurs, that is, the pump is no longer able to transport the fluid. Apart from this, there is a continuous accumulation and discharging of huge gas pockets leading to severe instabilities of the work performance of the pump, known as pump "surging".

[0008] Up to now various techniques were developed in an attempt to improve the performance of pumps working with gas-liquid two-phase flows:

- Employing semi-open impeller rather than closed ones, to utilize the leakage flow to provide higher resistance to gas accumulation and better gas handling capability;
- increasing the tip clearance gap of semi-open impellers to increase the secondary flow and the gas accumulation resistance;
- employing an inducer (axial impeller) together with the main pump impeller, which positively helps to increase the two-phase mixing at the pump inlet, providing better pump performance; or
- increasing the rotational speed and/or turbulence levels to improve the two-phase mixing.

[0009] However, the gain in two phase-mixing and pumping performance of these measures is limited or valid only for a specific flow-rate range.

[0010] Further, it was known to modify the impeller to delay the gas accumulation and increase the pump performance.

[0011] For example, DE 35 44 566 A1 describes impellers for a semiradial or "mixed flow" type pump comprising openings in the back plate within the channels formed by adjacent blades for allowing the recirculation of the gaseous and liquid phases in order to ensure their mixing and avoiding accumulation of the gas.

[0012] Nevertheless, there is an on-going need for improvement of the overall performance of centrifugal pumps for transporting gas-liquid two-phase flows.

[0013] Thus, it was the object of the present invention to provide centrifugal pumps with semi-open impeller with improved transport capability for gas-liquid two-phase flows, in particular even at high gas content.

[0014] It was a further object of the present invention to provide an alternative to known centrifugal pumps with increased tip clearance gap and inducer, respectively, which does not require an additional component such as an inducer and which can be readily integrated into conventional pumps with impeller with standard gap.

[0015] According to the present invention this object is solved by a centrifugal pump with semi-open impeller, wherein macroscopic grooves are provided to the surface of the front shroud directed towards the impeller blades.

[0016] The macroscopic grooves of the present invention are characterized by having dimensions such as depth and width in the order of millimeter.

[0017] It was observed that providing such macroscopic grooves on the front shroud increases the strength of secondary flow and improves two-phase mixing.

[0018] The mixing effect of the secondary flow in the tip clearance gap across the impeller blades is maximized by creating intensified vortices, thereby delaying and reducing gas accumulation. In particular, good results can be obtained even at high gas content.

[0019] The increased mixing obtainable by the macroscopic grooves on the front shroud make handling of higher amounts of gas contents much easier and, thus, more economically.

[0020] The alignment of the grooves on the surface of the front shroud can be circumferential, radial, combinations of both of them, such as curved with radial and circumferential components, or any other configuration. The cross-section profile can be selected from rectangular, semi-circular, rectangular with curved corners, etc.

[0021] The parameters defining the design of the macroscopic grooves of present invention include number of grooves, alignment, dimensions of grooves, cross-section profile, the inter-groove distance, and in case of radial grooves the length. It is understood that these parameters can vary depending on the particular configuration of the pump, working condition etc.

[0022] As set out above at least one of the width, depth and radius, respectively (in case of grooves with semi-circular profile), is in the order of millimeter (mm), preferably width, depth and radius, respectively, are all in the order of millimeter (mm).

[0023] For example, the depth can be selected from 2 mm to 12 mm, preferably 3 mm to 10 mm, the width from 6 mm to 12 mm, preferably from 8 to 10 mm and the radius from 1 to 5 mm, respectively. Also the number of grooves is not particularly restricted, but can be selected according to need depending on the circumstances, such as design of pump and size of front shroud.

[0024] A typical number of grooves ranges from 3 to 20, preferably 5 to 16.

[0025] Also the length of the radial grooves is not particularly restricted. The radial grooves extend radially across the surface of the shroud.

[0026] The length of the radial grooves can be less than the width of the front shroud, i.e. the distance between the inner and outer diameter of the front shroud.

[0027] For example, the radial grooves can start at a distance from the inner circumference and end at a distance from the outer circumference of the front shroud.

[0028] The grooves in the front shroud increase the turbulence of the flow and mixing of the phases with development of strong secondary flow and many vortices, thereby avoiding or at least delaying gas accumulation and phase segregation.

[0029] Thus, the transport of gas-liquid two-phase flows can be improved.

[0030] The outer diameter and total area, respectively, of the front shroud can vary according to need, for example depending on the design of the impeller and blades. Typically the outer diameter can range from 100 mm to 500 mm.

[0031] Further, the number of grooves can be adapted to the respective outer diameter. A larger area of the front shroud can require a larger number of grooves.

[0032] It is shown in

Figure 1	an exploded view of a centrifugal pump with semi-open impeller according to the prior art and of the present invention, respectively,
Figures 2a and 2b	a cross-sectional view of an impeller without inducer (Figure 2a) and with inducer (Figure 2b),
Figures 3a and 3b	a front and side view of a front shroud of the present invention with an arrangement of circumferential grooves (Figure 3a) and an arrangement of radial grooves (Figure 3b),
Figures 4a, 4b, 4c, 4d	embodiments of grooves with various cross-sectional profile (a: rectangular, b: semi-circular, c: rectangular with curved corners, d: radial),
Figure 5	a diagram with a comparison of the mixing coefficient at low load of centrifugal pumps with semi-open impeller and front shroud according to prior art, and centrifugal pumps of present invention with front shrouds having macroscopic grooves,
Figure 6	a diagram with a comparison of the mixing coefficient at nominal load of the pumps referred to in Figure 5,
Figure 7	a diagram with a comparison of the mixing coefficient at overload of the pumps referred to in Figure 5, and
Figure 8	a diagram with a comparison of the efficiency of a centrifugal pump according to the present invention with radial macroscopic grooves and pumps with semi-open impeller and front shroud according to the prior art.

[0033] In Figure 1 the structure of a typical centrifugal pump 1 with semi-open impeller 2 is shown with a representation of individual components in an exploded view.

[0034] The impeller 2 has a back plate 3 onto which a number of diverging curved blades 4 is arranged. Above the blades 4 a front shroud 5, 7, 9 is provided at a defined distance forming a gap. In Figure 1 three different types of front shrouds are shown: a front shroud 9 of prior art, a front shroud 5 with radial grooves 6, and one 7 with circumferential grooves 8, respectively, according to the present invention with the grooves 6, 8 being provided on the surface of the front shroud 5, 7 facing the blades 4.

[0035] Additionally an inducer 10 is provided for purpose of illustration upstream of the impeller 2 for increasing the static pressure of the fluid before it enters the impeller 2 and improving the phase mixing. Impeller 2 and inducer 10 are mounted on the same shaft (not shown).

[0036] The pump 1 has a volute casing 11 for housing the impeller 2 with pressure outlet 12 and inlet pipe 13 with mass flow inlet 14.

[0037] The mass flow inlet 14 is divided into circular air inlet 15 located in the center and an annular water inlet 16 surrounding the circular air inlet 15.

[0038] An overview of geometrical parameters essential for the design of an impeller 2 is given in table 1 and explained with reference to Figure 2a and Figure 2b showing cross-sectional views of impeller 2 without inducer 10 and with inducer, respectively.

[0039] The impeller shown in Figures 2a and 2b is equipped with 6 non-twisted blades, while the inducer has 3 helical blades to provide low inlet solidity as described in Gülich, J.F., 2008, Centrifugal Pumps, Springer, with a detailed description of design procedures for pumps.

[0040] The front shroud 5, 7, 9 is always fixed whereas the impeller 2 and inducer 10 are mounted rotating on a common shaft (not shown).

Table 1: Impeller Geometrical Dimensions

Parameter	Symbol
Impeller blade inlet diameter	D_1
Impeller blade outlet diameter	D_2
Impeller blade inlet width	b_1
Impeller blade outlet width	b_2
Suction pipe diameter	Ds
Inducer hub to tip diameter ratio	D_h/D_t
Inducer blade axial length	L_b/D_s
Tip clearance gap (gap)	S
Radial tip clearance gap of inducer	S_i
width of tip clearance gap	S/b_2
width of inducer tip clearance gap	S_i/b_2

[0041] In Figure 3a a sample design of a front shroud 7 with circumferential grooves 8 is shown as top view and as section through line A - A of the top view. Here, five circumferential grooves 8 are provided in coaxial manner on the front shroud 5, the circumferential grooves 8 having a rectangular cross-section profile as shown in the section view with W indicating the width and D the depth of each groove 8.

[0042] A sample design of a front shroud 5 with radial grooves 6 is shown in Figure 3b as top view and as section through line A-A of the top view. Here a number of sixteen radial grooves 6 is regularly distributed over the surface of the front shroud 5 and extends radially outward. In this embodiment the width of the grooves 6 increases with increasing diameter of the front shroud 5 where the referenced width being the width at mid-length of the radial grooves 6 resulting in a symmetrical trapezoid form. That is, here reference to width of the radial grooves relates to width at mid-length of the respective groove.

[0043] In Figure 3b the length of the radial grooves 6 is kept constant with the grooves starting and ending at a defined distance from the inner and outer diameter of the front shroud, respectively.

[0044] The cross-section profile of the radial and circumferential grooves 6, 8 is rectangular.

[0045] Some examples for different designs of the grooves for the front shroud according to the present invention are shown in Figures 4a **to** 4d as a section view through impeller 2 with back plate 3, blade 4, and front shroud with grooves.

[0046] Shown in Figure 4a are circumferential grooves with rectangular cross-section profile with W being width and D being depth, Figure 4b circumferential grooves with semi-circular cross-section profile with r being the radius, Figure

4c circumferential grooves with rectangular cross-section profile and curved corners, and Figure 4d a front shroud with radial grooves with L being the length of the grooves and D being the depth.

Examples

[0047] In the examples the mixing performance for gas-liquid two-phase flows of pumps according to the present invention with grooved front shrouds was compared with prior art pumps with semi-open impeller with a standard gap, increased gap, and additional inducer, respectively, and using the best convention settings known for the prior art impellers. The settings for the geometrical parameters are summerized in Table 2 below.

[0048] The pumps according to the present invention had a semi-open impeller with standard gap with the front shroud being provided with either radial or circumferential grooves. The setting for the impeller were the same as used for the prior art pump with standard gap as set out in Table 2 with reference to Figures 2a and 2b.

[0049] The outer diameter of the front shroud was each 347 mm.

Table 2: Impeller Geometrical Dimensions

Parameter	Symbol	Value
Impeller blade angle [2] (defining curvature of blades)	β_1	24°
Impeller blade outlet angle [2] (defining curvature of blades)	β_2	24°
Impeller blade inlet to outlet diameter ratio	D_1/D_2	47.55 %
Inducer hub to tip diameter ratio	D_h/D_t	41.85 %
Inducer hub solidity (Blade chord to pitch at hub)	$\sigma_h = C_h/P_h$	2.836
Inducer tip solidity (Blade chord to pitch at tip)	$\sigma_t = C_t/P_t$	1.807
Inducer area solidity	σ_a	23.33 %
Inducer hub blade angle	θ_h	34°
Inducer tip blade angle	0_t	58°
Inducer blade axial length	L_b/D_s	1.0
Impeller with standard gap	S/b_2	2.5 %
Impeller with increased gap	S/b_2	5 %
Inducer tip clearance gap	S_i/b_2	12.5 %

[0050] The parameters changed in case of circumferential grooves were the depth (D), width (W), number of grooves (n_g) and the cross-section profile, i. e. rectangular (with sharp corners), rectangular with curved corners and semi-circular as shown in Figure 4a to 4c.

[0051] The parameters changed in case of radial grooves were depth (D), width (W), and number of grooves (n_g) with the radial grooves stretching radially along the shroud. The length of the radial grooves was always constant and shorter than the blade radial length. The diameter at which the radial grooves starts and ends was kept at $D_1 + 0.5b_2$ and $D_2 - 0.5b_2$ respectively, with D_1 = impeller blade inlet diameter and D_2 = impeller blade outlet diameter.

[0052] In Table 3 the details of all grooves combination according to this invention used in the examples are listed.

Table 3: Parameter of grooves

<table>
<tr><th>Groove type</th><th>Groove name</th><th>Groove crosssection</th><th>No. of grooves</th><th>Width (mm) W</th><th>Depth (mm) D</th><th>Radius (mm) r</th><th>Length (mm) L</th></tr>
<tr><td rowspan="5">Circumferential grooves</td><td>C1</td><td rowspan="4">Rectangular</td><td rowspan="3">5</td><td rowspan="2">10</td><td>5</td><td rowspan="4">0</td><td rowspan="5">-</td></tr>
<tr><td>C2</td><td>3</td></tr>
<tr><td>C3</td><td>8</td><td>8</td></tr>
<tr><td>C4</td><td>3</td><td>10</td><td>5</td></tr>
<tr><td>C5</td><td>Semicircular</td><td>3</td><td>0</td><td>0</td><td>5</td></tr>
</table>

(continued)

<table>
<tr><th>Groove type</th><th>Groove name</th><th>Groove crosssection</th><th>No. of grooves</th><th>Width (mm) W</th><th>Depth (mm) D</th><th>Radius (mm) r</th><th>Length (mm) L</th></tr>
<tr><td rowspan="5">Radial grooves</td><td>R1</td><td rowspan="5">Rectangular</td><td rowspan="4">16</td><td rowspan="3">10</td><td>10</td><td rowspan="5">0</td><td rowspan="5">71</td></tr>
<tr><td>R2</td><td>8</td></tr>
<tr><td>R3</td><td>5</td></tr>
<tr><td>R4</td><td>8</td><td>10</td></tr>
<tr><td>R5</td><td>12</td><td>10</td><td>5</td></tr>
</table>

[0053] All runs were carried out for three different load conditions, i. e. part load (50 % of the nominal flow of the pump), nominal load (100 % nominal flow), and overload (150 % of the nominal flow).

[0054] Pumps were used having a principal configuration as shown in Figure 1 and impeller configuration as shown in Figures 2a and 2b which can be roughly divided into three zones with inlet pipe, rotating zone including the impeller (or impeller with inducer), and the volute with exit pipe. The length-to-diameter ratios of the inlet and outlet pipes were approximately 4 and 5, respectively.

[0055] The size of the air mass flow inlet 15 of the mass inlet flow 14 was calculated so that the relative velocity between the two phases (water and air) was always equal to zero, i. e., $V_a = V_w$, for any given flow rate.

[0056] For this the radius of the air inlet surface was set to $D_s\sqrt{\varepsilon}$ with D_s being the suction pipe diameter and ε the inlet gas volume fraction with ε being defined as the ratio of air volume flow rate (Q_a) to the total volume flow rate of the two phases $Qt = Q_a + Q_w$, with Q_w being the water volume flow rate.

[0057] The rotation speed of the pumps was 650 rpm with the specific speed of pump being approximately $n_q = 21\ \mathrm{min}^{-1}$.

[0058] The gas volume fraction ε was set to 3 %. The results obtained for this setting of the gas volume fraction can be representative for higher gas volume fractions as has been shown in previous studies [1,2]. These studies show that a pump keeps its performance for higher gas volume fraction as found by a similar analysis concerning turbulence and mixing.

[0059] The mixing coefficient of all pumps used in these examples was calculated as described below.

[0060] For quantifying the two-phase mixing the surface uniformity S_u (also called uniformity index) was used which measures the homogeneity of the distribution of a particular phase on a given surface.

[0061] The S_u of the gas phase (air) is given in equation 1:

$$Su = 1 - \frac{\sum_f |c-\bar{c}| A_f}{2|\bar{c}| A_f} \qquad (1)$$

with

A_f being the area of an elementary cell-face,
c = local gas volume fraction,
$\bar{c}$ = average gas volume fraction over the surface area A.

[0062] The mixing coefficient M_c is obtained normalizing the surface uniformity values according to equation 2:

$$M_c = \frac{Su - Su{,}min}{Su - Su{,}min} \qquad (2)$$

[0063] The value S_u always ranges from $S_{u,\,min}$ and 1, with the minimum value being found at the inlet surface where the two phases are completely separated at injection.

[0064] The non-dimensional mixing coefficient M_c ranges between 0 and 1, where 0 indicates no mixing at all (gas and liquid are perfectly separated like in the injection plane of the inlet surface), while 1 indicates that the two phases are perfectly mixed.

[0065] The average mixing coefficient for the different pump configurations used in these examples are given in Figures 5, 6 and 7 for three load conditions: Figure 5 - part load, Figure 6 - nominal load (100%), and Figure 7: overload (150% of nominal load), respectively.

[0066] It was found that the pumps with grooved front shrouds of the present invention results in smaller and more dispersed gas bubbles compared to those of the prior art pumps. In particular, the radial grooves show an even higher number of smaller structures compared to the circumferential grooves particularly for nominal and overload conditions. The presence of the grooves results in formation of strong secondary flows with many vortices for both the circumferential and the radial grooves. As the impeller blades move past the grooves, the vortices were stretched along the blades and the recirculation zones were fed continuously by the impeller rotation, which amplifies the mixing of the gas and the liquid. Higher turbulence and mixing levels are obtained which significantly support two-phase pumping, particularly at high gas volume fractions.

[0067] Comparing Figures 5 to 7 the values of the mixing coefficient are highest at part-load conditions and lowest at overload, which is due to the maximum and minimum residence time available for mixing, respectively, which is longest for part load and lowest for overload.

[0068] At part-load conditions shown in Figure 5 the pump with inducer shows the best results of the prior art pumps for improving two-phase mixing.

[0069] Likewise the pumps with circumferential groves according to the present invention have improved flow mixing at part-load compared to the prior art pumps without inducer, yet less than the pump with inducer. On the other hand, all the examples of pumps with radial grooves according to the present invention are clearly better than all other configurations inclusive the prior art pump with inducer, leading to a strong improvement in two-phase mixing.

[0070] At nominal load as shown in Figure 6, all examples of the present invention show improved mixing compared to the prior art configurations inclusive the pump with inducer, with the examples with radial grooves showing superior results not only compared to the prior art pumps but also with respect to the examples with circumferential grooves according to the present invention.

[0071] At overload conditions as shown in Figure 7, the examples with radial grooves according to the present invention show at least as good results than the best prior art pump, i.e. the pump with inducer, with example R1 being much better than the prior art pump with inducer.

[0072] The mixing coefficient of the examples with circumferential grooves according to the present invention are comparable with the prior art pump with standard gap but, as the prior art pump with standard gap, are not as good as the result of the prior art pumps with inducer and with increased gap, respectively.

[0073] Summarizing, the examples with radial grooves according to the present invention show the best results for all three conditions not only compared to the prior art pumps but also compared to the examples with circumferential grooves according to the present invention.

[0074] The examples with circumferential grooves according to the present invention show better results compared to all three prior art pumps at nominal load, similar results compared to the prior art pumps with standard gap and increased gap at low load.

[0075] At overload as shown in Figure 7, the performance of the examples with circumferential grooves according to the present invention is comparable to that of the prior art pump with standard gap, but not as good as the prior art pumps with inducer and with increased gap.

[0076] Again, the performance of the examples with radial grooves according to the present invention is significantly better than both that of the examples with circumferential grooves according to the present invention and the prior art pump with standard gap exceed the performance of the prior art pump with inducer, with R1 exceeding all the other pumps studied.

[0077] For pump R1 with radial grooves also a comparison of the efficiency with the three prior art pumps is shown in Figure 8,

[0078] Pump R1 of present invention with radial grooves provides the best efficiency for the three different flow conditions considered.

[0079] It is to be noted, that pump R1 has the highest number of grooves, 16, and largest cross-section area. Consequently, it is believed that there is a positive correlation between the total area of the radial grooves and the resulting efficiency. The total area of the radial grooves A_G is calculated by multiplying the number of grooves by the width and the depth.

[0080] These results show that the novel pumps according to the present invention with grooved front shroud have good or excellent gas-liquid two-phase mixing performance.

Reference list

[0081]

1 centrifugal pump
2 impeller
3 back plate

4 blades
5 front shroud, radial
6 radial grooves
7 front shroud, circumferential
8 circumferential groove
9 front shroud, prior art
10 inducer
11 volute casing
12 pressure outlet
13 inlet pipe

14 mass flow inlet
15 air inlet
16 water inlet
17 inlet pipe surface
18 pump body
19 gap

References cited:

[0082]

[1] Gülich, J.F., 2008. Centrifugal pumps. Springer.
[2] Mansour, M., Parikh, T., Engel, S., Wunderlich, B., Thevenin, D., 2019. Investigation on the influence of an inducer on the transport of single and two-phase air-flows by centrifugal pumps, in: 48thTurbomachinery & 35th Pump Symposia.
[3] Mansour, M., Wunderlich, B., Thevenin, D., 2018b. Effect of tip clearance gap and inducer on the transport of two-phase air-water flows by centrifugal pumps. Exp. Therm Fluid Sci. 99, 487-509. [4] Parikh, T., Mansour, M., Thevenin, D., 2020.
[4] Parikh, T., Mansour, M., Thevenin, D., 2020. Investigations on the effect of tip clearance gap and inducer on the transport of air-water two-phase flow by centrifugal pumps. Chem. Eng. Sci. 218,115554.

Claims

1. Centrifugal pump (1) with semi-open impeller (2),

   wherein the semi-open impeller (2) has a back plate (3), a number of blades (4) provided onto the back plate (3), and a front shroud (5, 7) arranged at a defined distance upstream to the blades (4),
   wherein on the face of the front shroud (5, 7) directed toward the blades (4) a number of macroscopic grooves (6, 8) with dimensions in the millimeter range is provided.

2. Centrifugal pump (1) according to claim 1,
   wherein at least one of the depth and width of the macroscopic grooves (6, 8) has a magnitude in the millimeter range.

3. Centrifugal pump (1) according to claims 1 or 2,
   wherein the macroscopic grooves (6) are aligned radially on the front shroud (5).

4. Centrifugal pump (1) according to claim 3,
   wherein the radial macroscopic grooves (6) extend over the front shroud (5) with a distance from the inner and outer diameter of the front shroud (5).

5. Centrifugal pump (1) according to claims 3 or 4,
   wherein the number of the radial grooves (6) is from 8 to 18.

6. Centrifugal pump (1) according to claims 3 or 5,
   wherein the number of the radial macroscopic grooves (6) is from 8 to 18, the width from 8 to 12 mm and the depth from 4 to 12 mm.

7. Centrifugal pump (1) according to any of the claims 3 to 6, **wherein** the radial macroscopic grooves (6) have a number of 12 to 16, a width of 8 to 10 mm and a depth of 5 to 10 mm.

8. Centrifugal pump (1) according to claims 1 or 2, **wherein** the macroscopic grooves (8) are aligned circumferentially on the front shroud (7).

9. Centrifugal pump (1) according to claim 8, **wherein** the number of circumferentially macroscopic grooves (8) is from 3 to 7.

10. Centrifugal pumps (1) according to claim 8 or 9, **wherein** the width of the circumferentially macroscopic grooves (8) is from 4 mm to 12 mm and the depth from 3 mm to 12 mm.

11. Centrifugal pump (1) according to any of the preceding claims, **wherein** the cross-section profile of the macroscopic grooves (6, 8) is selected from rectangular, rectangular with curved corners, and semi-circular.

12. Centrifugal pump (1) according to any of the preceding claims, **wherein** the outer diameter of the front shroud (5,7) is from 100 mm to 500 mm.

13. Use of a centrifugal pump (1) according to any of the claims 1 to 12, for transporting two-phase flows.

14. Use of a centrifugal pump (1) according to claim 13, for transporting gas-liquid flows.

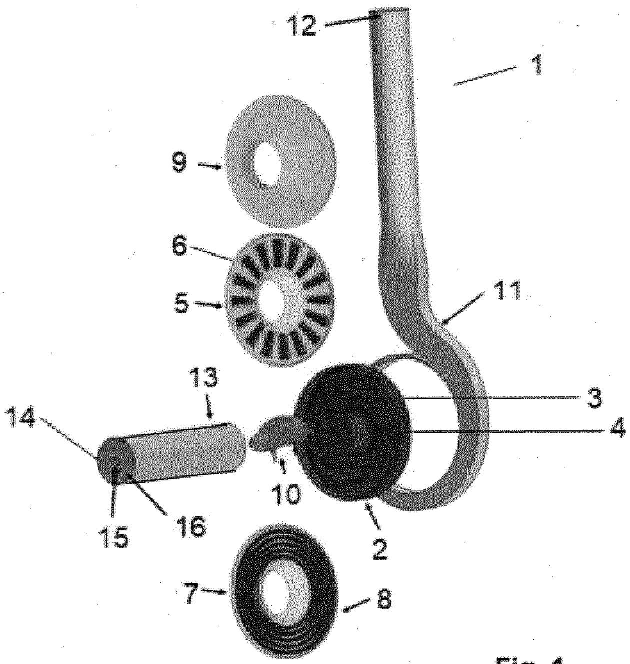

Fig. 1

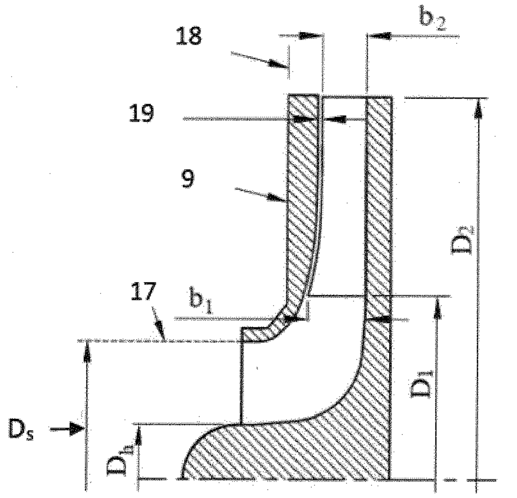

Fig. 2a

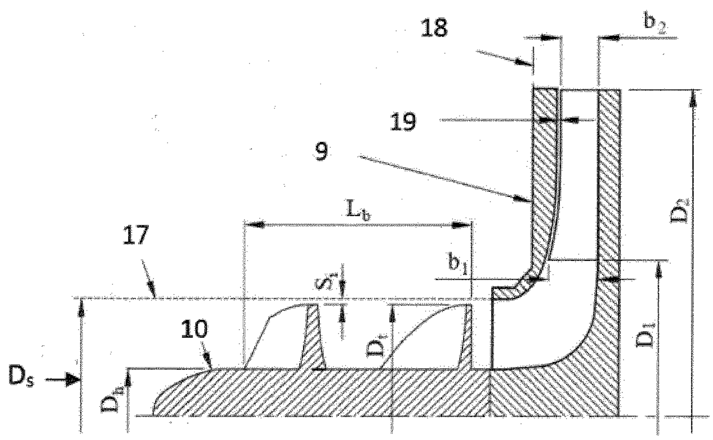

Fig. 2b

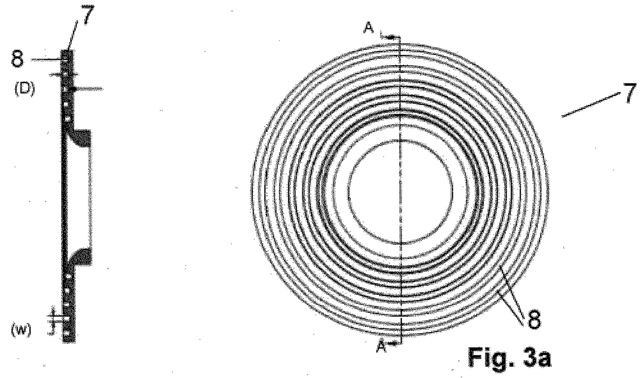

Fig. 3a

D = depth
W = width

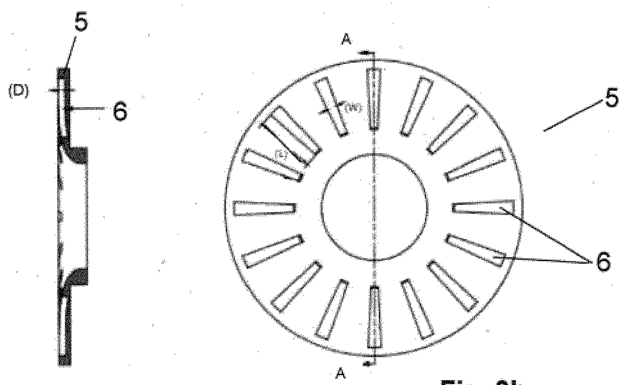

Fig. 3b

L = length
W = width

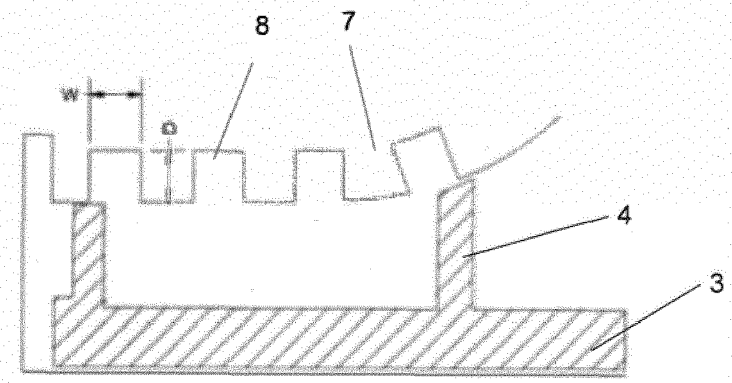

Fig. 4a

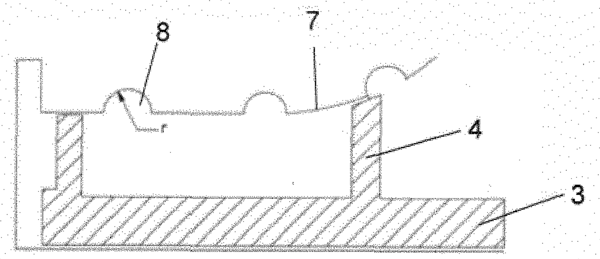

Fig. 4b

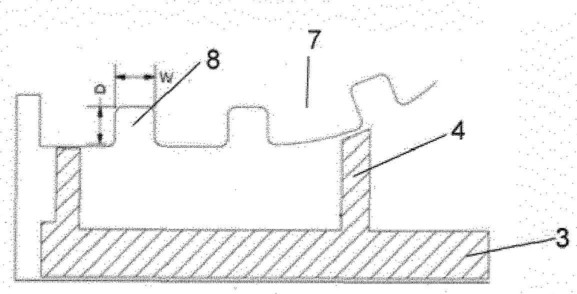

Fig. 4c

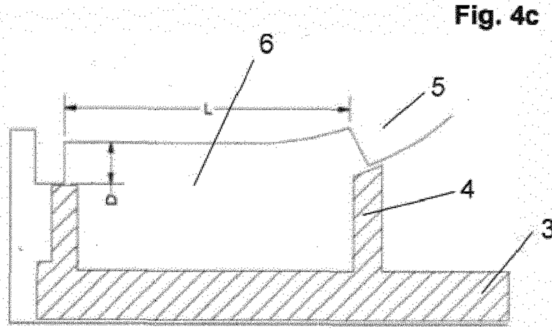

Fig. 4d

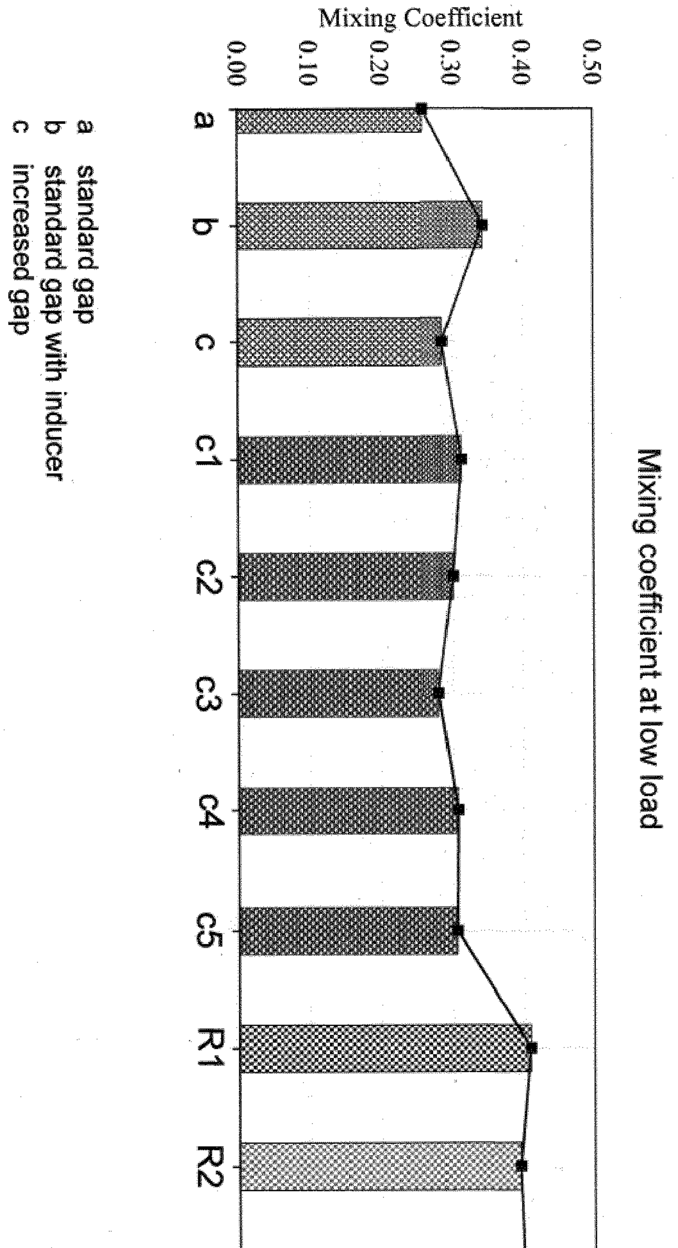
Mixing coefficient at low load
Mixing Coefficient
0.00
0.10
0.20
0.30
0.40
0.50
a
b
c
c1
c2
c3
c4
c5
R1
R2
R3
a standard gap
b standard gap with inducer
c increased gap

Fig. 5

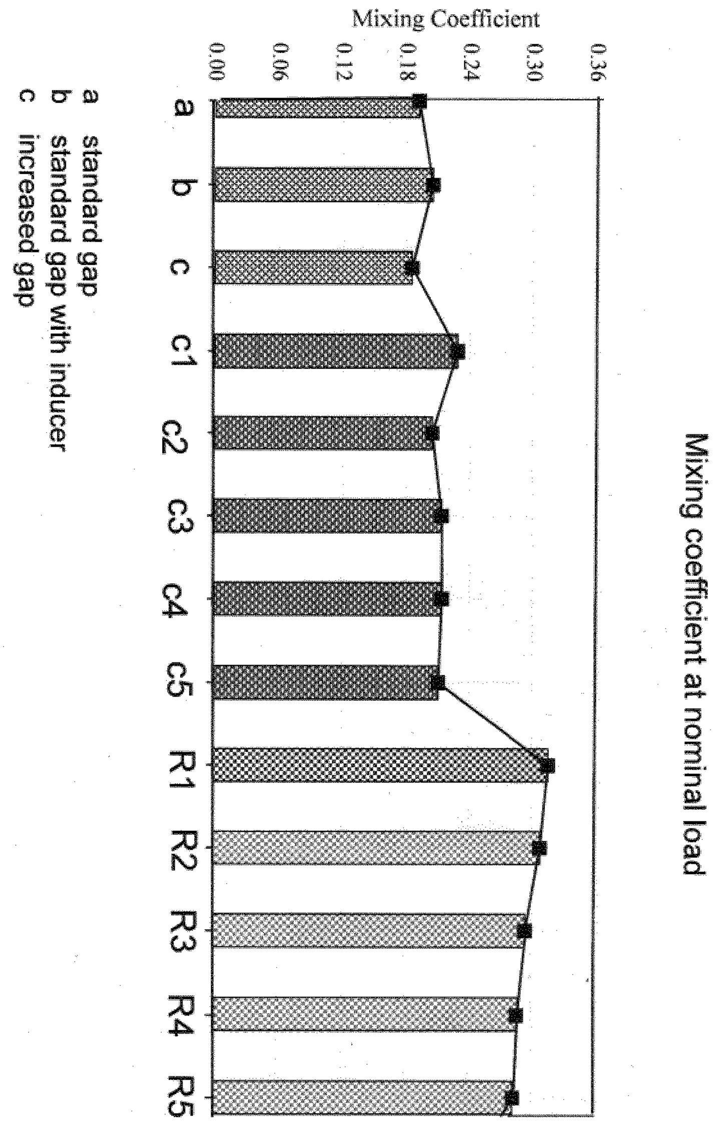
Mixing coefficient at nominal load
Mixing Coefficient
0.00
0.06
0.12
0.18
0.24
0.30
0.36
a
b
c
c1
c2
c3
c4
c5
R1
R2
R3
R4
R5
a standard gap
b standard gap with inducer
c increased gap

Fig. 6

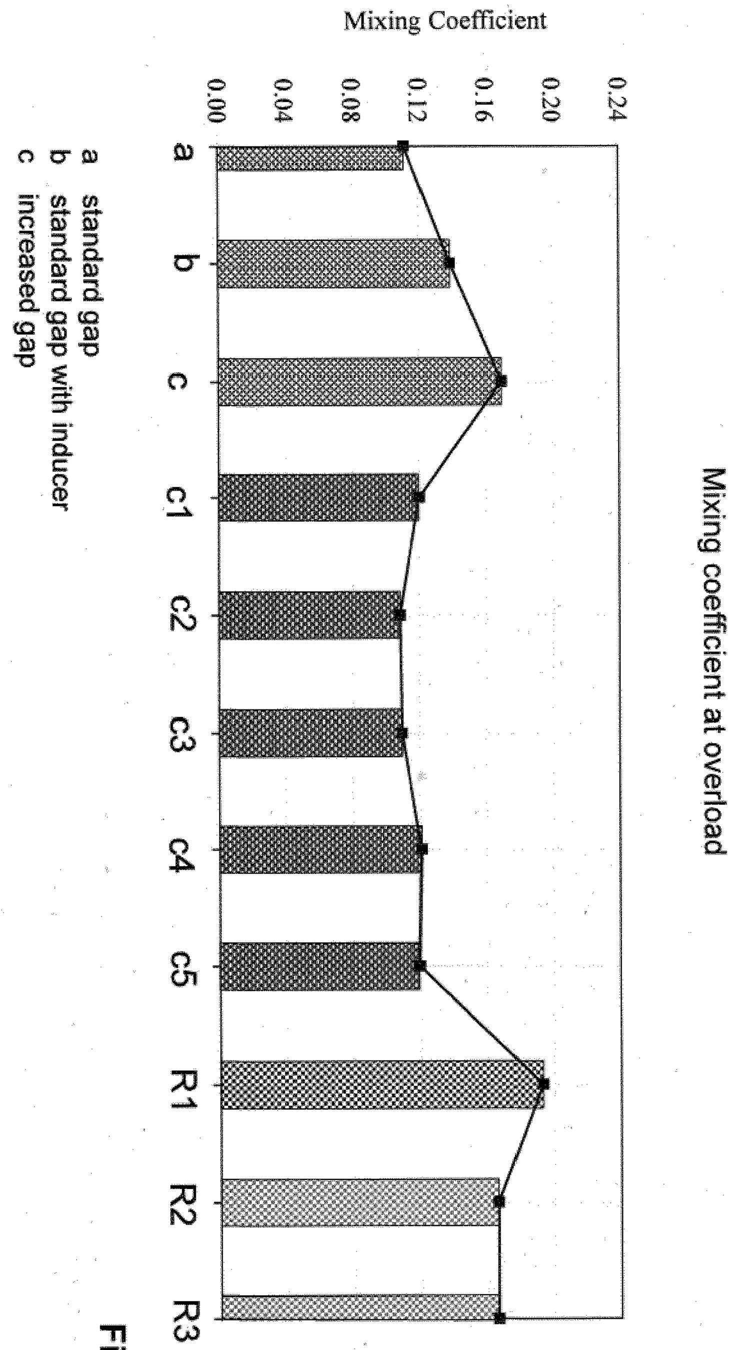
Mixing coefficient at overload
Mixing Coefficient
0.00
0.04
0.08
0.12
0.16
0.20
0.24
a
b
c
c1
c2
c3
c4
c5
R1
R2
R3
a standard gap
b standard gap with inducer
c increased gap

Fig. 7

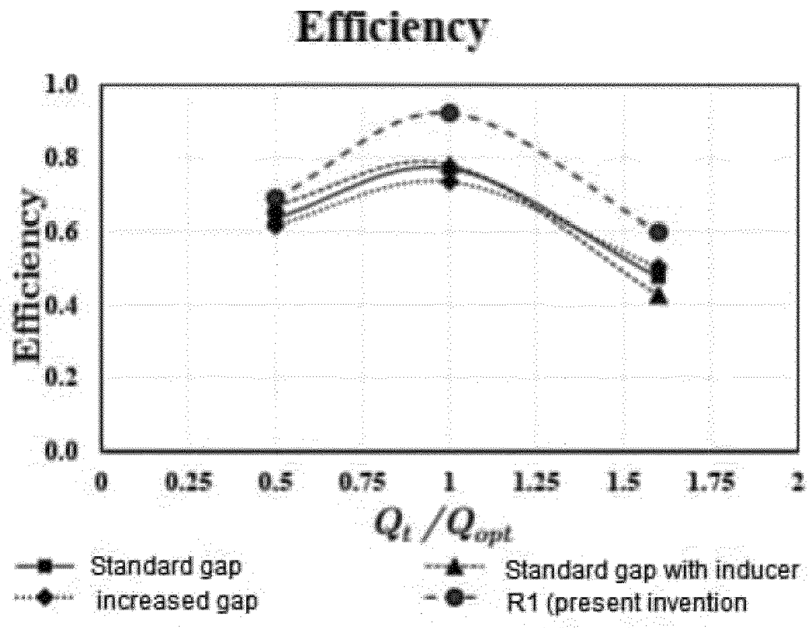
Efficiency
1.0
0.8
0.6
0.4
0.2
0.0
Efficiency
0
0.25
0.5
0.75
1
1.25
1.5
1.75
2
Q_t / Q_{opt}
Standard gap
increased gap
Standard gap with inducer
R1 (present invention

Fig. 8

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EUROPEAN SEARCH REPORT

Application Number
EP 22 17 5498

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
X	JP H09 4585 A (TORISHIMA PUMP MFG CO LTD) 7 January 1997 (1997-01-07) * abstract * * figures 1-3 *	1-13	INV. F04D31/00 F04D1/00 F04D7/04 F04D29/22 F04D29/42 F04D29/16 F04D29/44
X	US 2017/016457 A1 (JAPIKSE DAVID [US]) 19 January 2017 (2017-01-19) * paragraph [0074] - paragraph [0096] * * figures 5-31 *	1-14	
A	US 2012/051897 A1 (DAVIS MARK ALAN [US] ET AL) 1 March 2012 (2012-03-01) * paragraph [0028] - paragraph [0045] * * figures 1-3b *	1-12	
			TECHNICAL FIELDS SEARCHED (IPC)
			F04D

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
The Hague	31 October 2022	Lovergine, A

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31-10-2022

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
JP H094585 A	07-01-1997	NONE	
US 2017016457 A1	19-01-2017	CN 106574636 A	19-04-2017
		CN 113685377 A	23-11-2021
		EP 3161324 A1	03-05-2017
		JP 6866019 B2	28-04-2021
		JP 2017523338 A	17-08-2017
		JP 2021102967 A	15-07-2021
		JP 2022091926 A	21-06-2022
		KR 20170028367 A	13-03-2017
		US 2015369073 A1	24-12-2015
		US 2017016457 A1	19-01-2017
		US 2018258950 A1	13-09-2018
		WO 2015200533 A1	30-12-2015
US 2012051897 A1	01-03-2012	AU 2011281111 A1	07-02-2013
		BR 112013001314 A2	17-05-2016
		CA 2806043 A1	26-01-2012
		CN 103154522 A	12-06-2013
		CN 109185163 A	11-01-2019
		US 2012051897 A1	01-03-2012
		WO 2012012622 A2	26-01-2012
		ZA 201300487 B	25-09-2013

EPO FORM P0459

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Patent documents cited in the description

- DE 3544566 A1 **[0011]**

Non-patent literature cited in the description

- **GÜLICH, J.F.** Centrifugal Pumps. Springer, 2008 **[0039]**
- **GÜLICH, J.F.** Centrifugal pumps. Springer, 2008 **[0082]**
- **MANSOUR, M. ; PARIKH, T. ; ENGEL, S. ; WUNDERLICH, B. ; THEVENIN, D.** Investigation on the influence of an inducer on the transport of single and two-phase air-flows by centrifugal pumps. *48thTurbomachinery & 35th Pump Symposia,* 2019 **[0082]**
- **MANSOUR, M. ; WUNDERLICH, B. ; THEVENIN, D.** Effect of tip clearance gap and inducer on the transport of two-phase air-water flows by centrifugal pumps. *Exp. Therm Fluid Sci.,* 2018, vol. 99, 487-509 **[0082]**
- **PARIKH, T. ; MANSOUR, M. ; THEVENIN, D.** Investigations on the effect of tip clearance gap and inducer on the transport of air-water two-phase flow by centrifugal pumps. *Chem. Eng. Sci.,* 2020, vol. 218, 115554 **[0082]**