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(54) AN IMPELLER FOR A CENTRIFUGAL PUMP, A CENTRIFUGAL PUMP AND A USE THEREOF

(57) The present invention relates to an impeller (40) for a centrifugal pump, a centrifugal pump and a use thereof. The present invention relates especially to a novel closed impeller (40) structure for a centrifugal pump. The impeller (40) is characterized by pump-out vanes (48) on the rear shroud and pump-out vanes (50) on the front shroud (44). The front pump-out vanes (50) are dimensioned according to a specific formula. The centrifugal pump utilizing the impeller (40) of the present invention is suitable for pumping both clean liquids and solids-containing liquids like for instance fibrous suspensions of pulp and paper or board industry.

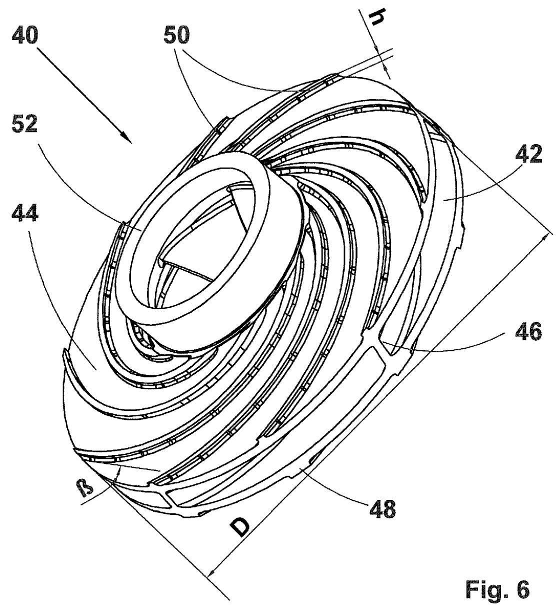

Fig. 6

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Description

Technical field

[0001] The present invention relates to an impeller for a centrifugal pump, a centrifugal pump and a use thereof. The present invention relates especially to a novel closed impeller structure for a centrifugal pump. The centrifugal pump utilizing the impeller of the present invention is suitable for pumping both clean liquids and solids-conta8ining liquids like for instance fibrous suspensions of pulp and paper or board industry.

Background art

[0002] Energy saving, in other words efficiency, is becoming nowadays a more and more important factor in the development and design of all kinds of machines and machine elements including centrifugal pumps and their impellers. It has always been a known fact that the work the impeller of a centrifugal pump subjects to the fluid it pumps is not totally converted to kinetic and/or potential energy but a part of it is wasted in phenomena taking place between the fluid and both the rotary impeller and the static pump volute or volute casing. Such phenomena include, among others, surface friction between the fluid and the surfaces of both the impeller and the pump volute, and various leakage flows between the impeller and the volute casing.

[0003] The energy aspects of pumping have also been taken into account by the European Union a few years ago when they established a framework for the setting of ecodesign requirements for energy-related products. In 2012 the European Commission has introduced implementing measures for products used in electric motor systems, such as water pumps. In accordance with the EU water pumps forming parts of electric motor systems are essential in various pumping processes, and there is a total cost-effective potential for improving the energy efficiency of these pumping systems by approximately 20 % to 30 %. Even though the main savings can be achieved by motors, one of the factors contributing to such improvements is the use of energy-efficient pumps. Consequently, water pumps are a priority product for which ecodesign requirements should be established

[0004] Therefore the EU has set a goal to pump manufacturers to manufacture pumps having a certain efficiency as a function of specific speed of the pump. Figure 1 illustrates schematically two efficiency curves in relation to the specific speed, and Figure 2 the specific speed and its relation to basic pump construction. What Figure 2 in practice teaches is that the specific speed is the higher the larger is the capacity of the pump. In other words, small sized pumps have a low specific speed.

[0005] Specific speed (n_s) means a dimensional value characterizing the shape of the pump impeller by head (H), flow (Q) and speed (n). Specific speed is calculated by using the following equation:

$$n_s = n * Q_{BEP}^{1/2} / H_{BEP}^{3/4} \text{ [min}-1\text{]},$$

where

- head (H) means the increase in the hydraulic energy of water in meters [m], produced by the pump at the specified point of operation,
- rotational speed (n) means the number of revolutions per minute [rpm] of the shaft,
- flow (Q) means the volume flow rate [m^3/s] of fluid through the pump, and
- best efficiency point (BEP) means the operating point of the pump at which it is at the maximum hydraulic pump efficiency measured with clean cold water.

[0006] There is one more variable that needs to be specified, i.e. hydraulic pump efficiency or mere efficiency (η), which is the ratio between the mechanical power transferred to the liquid during its passage through the pump and the mechanical input power transmitted to the pump at its shaft

[0007] Now coming back to Figure 1 the solid curve A shows the efficiency required by the EU, and the dashed curve B the efficiency of a series of today's pumps having a semi-open impeller. By a series of pumps is meant pumps having the same basic construction but a differing capacity/flow designed to cover, more or less, all the pumping needs (in view of flow) of the customers. What is noteworthy is that for the most part of the operating range (specific speed) of the pump series the semi-open impellers have an efficiency well above that required by the EU. However, at the lower end of the specific speed range the efficiency curve B drops below the EU- curve A.

[0008] Thus it appears that in order to fulfill the requirements of the EU, the efficiency of pumps having a low specific speed has to be improved. Since it was already above explained that both the surface friction and the leakage flows are clearly the causes of the reduction of the pumping efficiency, they have to be considered in more detail.

[0009] It has also been customary practice to use, for pumping pure water, centrifugal pumps having closed impellers, shrouds with smooth faces opposite to the working vanes and wear rings. However, since the specific speed of a centrifugal pump correlates to efficiency, it has been understood now when studying the pumps having a low specific speed that they have low efficiency due to two impeller-related factors having a relatively high impact to efficiency. The first factor being high leakage flow, in relation to the total flow, via the wear rings. And the second factor is the energy wasted on the smooth faces of the shrouds in relation to the total power used by the pump.

[0010] The leakage flows appear in the case of open impellers at the opposite side edges of the impeller vanes, as there has to be a certain running clearance between the side edges of the vanes and the walls of the volute casing, whereby a part of the fluid to be pumped is able to pass via such a clearance from a preceding vane cavity to a succeeding vane cavity.

[0011] In the case of semi-open impellers the above mentioned leakage flow appears only on one side of the impeller as at the other side, usually the rear side of the impeller, the working vanes are attached to a rear shroud, also called as a hub, of the impeller. However, another type of leakage flow may be found in semi-open impellers, as the pumped fluid has such a high pressure at the radially outer edge of the rear shroud of the impeller that it is capable of forcing the fluid round the impeller circumference to the rear side of the impeller between the rear shroud and the rear wall of the volute casing.

[0012] In the case of closed impellers, i.e. impellers having both rear and front shrouds fastened to both the rear and front side edges of the working vanes, the leakage flow round the side edges of the working vanes is naturally prevented, but the leakage flows round the radially outer edges or circumferences of the shrouds are a fact.

[0013] The further consideration based, on the one hand to the EU- requirements, and on the other hand, to the properties and construction of pumps having a low specific speed has now taught that the efficiency of a small-sized semi-open impeller is very hard, if not impossible, to improve to such an extent that the efficiency would be above the EU- curve A in Figure 1. Therefore, the consideration led to taking the closed impeller in use at the lower end of the specific speed range.

[0014] The closing of the side edges of the working vanes in closed impellers not only creates a leakage flow round the radially outer circumferential edge/s of the shroud/s but also subjects the face/s of the shroud/s opposite to the working vanes to the pressure of the pumped fluid. The pressure distribution at the rear side of the shroud is parabolic, i.e. at its highest at the outer circumference of the impeller from where it reduces gradually when moving towards the shaft of the impeller. The pressure results, both with semi-open and closed impellers, in an axial thrust pushing the impeller towards the pump inlet, as the full area of the rear shroud is subjected to the fluid pressure. The axial thrust is clearly greater in semi-open impellers than in closed impellers, as, in semi-open impellers there is no front shroud to the front side of which the pressure could act like in closed impellers. Yet, in both impeller types the impeller needs to be balanced such that the bearings of the shaft of the pump are not subjected to a too high axial load. Also, without any measures the pressure affects the shaft sealing, and has to be limited for preventing the sealing from deteriorating. The axial force is balanced by arranging to the rear face of the shroud pump-out vanes the purpose of which is to increase the speed of the fluid entering the rear side of the shroud such that its pressure is reduced. Thus, the rear pump-out vanes act somewhat like the impeller working vanes. However, because they are normally much smaller, the pressure they develop cannot overcome that developed by the working vanes. Instead, the back pump-out vanes simply act to break down that discharge pressure to a value between suction pressure and discharge pressure. Another measure to affect the pressure at the rear side of the rear shroud is to provide the shroud close to the shaft with holes extending through the shroud via which holes the pressure is able to be balanced.

[0015] At the front side of the closed impeller the situation is different. There is no need to fight the pressure, which is one of the major tasks of the rear pump-out vanes, as there is no reason to try to lower the pressure due to the fact that the area of the front shroud face opposite to the working vanes is much smaller than the area of the rear shroud face opposite the working vanes. The front face of the shroud has to be provided with means to minimize the leakage flow round the impeller circumference to the front side of the front shroud. At its worst there is a significant recirculating leakage flow from the pressure side of the impeller back to the suction side of the impeller through the gap between the front shroud of the impeller and the volute casing. Such a leakage flow takes a substantial amount of energy used for pumping, whereby the efficiency of the impeller is decreased remarkably. There are two ways that the leakage flow may be controlled, i.e. either by arranging a sealing, most often called as a wear ring, between the impeller and the volute casing, or by arranging front pump-out vanes on the front face of the front shroud, i.e. on the face opposite to the working vanes.

[0016] Wear rings, which function basically as a slide ring sealing, restrict efficiently the amount of discharge fluid that tries to circulate back to the suction side of the impeller. Wear rings provide an adequate solution for applications that handle clear water or occasionally handle light solids. However, as the wear ring has a certain operating clearance, the wear ring must be replaced, when the clearance becomes excessive. The flow restriction created by the tight clearance between the stationary and rotating wear ring faces causes very high local velocities and hence a high wear rate. If the fluid to be pumped contains abrasive particles, wear rings, because they are subject to a very high flow velocity, will have an unacceptably short life span, even when made of hard materials or when their surfaces have been specifically treated in view of wear. Thus the use of a wear ring is not desirable when pumping liquids containing solids.

[0017] Pump-out vanes offer a better alternative for handling abrasive solids. The use of such pump-out vanes is known from slurry pumps like, for instance, those discussed in US-A1-20090226317. Pump-out vanes control the leakage through a pumping action creating a head to prevent or at least counter any leakage or recirculation from an outer high pressure peripheral outlet of the impeller radially inwardly in-between the impeller and the volute casing. The pump-out vanes are typically almost radial, or arranged at an angle of 10 - 30 degrees from the radial direction.

[0018] The disadvantage of known pump-out vanes is that they consume considerable amount of power while controlling leakage. When new, a pump impeller equipped with pump-out vanes will likely have a lower efficiency than its wear ring counterpart. However, it will come close to maintaining its "as installed" efficiency throughout its operational life. An impeller with wear rings loses efficiency rapidly as the rings wear. It is not uncommon to have several outages to replace wear rings over the life of a single impeller when wear rings are used in an aggressive solids application. Thereby, the use of pump-out vanes on the front face of the front shroud has been accepted especially in connection with pumps designed to pump slurries or other abrasive liquids in spite of their power consumption, as the energy efficiency is not the main issue in slurry pumps.

[0019] A further known disadvantage of closed impellers is that the smooth front and back shrouds (not having pump-out vanes), rotating in close proximity to the casing walls, generate disc friction that lowers the efficiency of the pump relative to that found in open impeller designs.

[0020] Yet another disadvantage is that the closed impeller is more easily plugged. Large solids that might otherwise be broken up by the grinding action generated by a rotating open impeller and the stationary casing wall, can easily become lodged in the eye of a closed impeller. This may create a mechanical or hydraulic imbalance that has the potential to damage the pump, or at the least causes a pre-mature outage to remove the blockage. In other words, there are two separate methods of restricting internal recirculation that can lower the efficiency of the pump and generate a lot of unwanted heat to the fluid to be pumped.

Brief summary of the Invention

[0021] Thus, an object of the present invention is to find a way to improve the construction of the centrifugal pumps at least at the lower end of the specific speed range of a series of pumps such that the efficiency for the entire range of pumps is above the EU efficiency curve.

[0022] Another object of the present invention is to change the construction of the impeller such that the efficiency of an impeller may be raised.

[0023] Yet another object of the present invention is to design the impeller such that its pump-out vanes both prevent the leakage flow and function in an energy efficient manner, i.e. the pump-out vanes are to be designed such that they prevent the leakage flow in an optimal way in view of the total efficiency of the impeller.

[0024] A still further object of the present invention is to design a novel impeller that is able to prevent the recirculating leakage flow of liquids containing solids without the use of wear ring/s.

[0025] At least one of the above objects of the present invention, among others, are fulfilled by an impeller for a centrifugal pump, the impeller having a front shroud, a rear shroud, and one or more working vanes therebetween, the front shroud having a front face opposite to the face having the working vanes, the rear shroud having a rear face opposite to the face having the working vanes, the front shroud having an outer circumference and a plurality of front pump-out vanes attached to the front face of the shroud, the rear shroud having a plurality of rear pump-out vanes attached to the rear face of the shroud, wherein the front pump-out vanes are dimensioned in accordance with an equation:

$$\sum_{i=1}^{Z}(l_i) \;/\; D > 8,$$

where

Z is the number of front pump-out vanes,
I is the vane length measured along the leading surface of each front pump-out vane, and
D is the outer diameter of the front shroud.

[0026] Other characterizing features of the impeller of the present invention become evident in the accompanying dependent claims.

[0027] The centrifugal pump impeller of the present invention brings about several advantages in comparison to prior art centrifugal pumps. At least the following advantages may be found

- to prevent the leakage typical for a closed impeller,
- to make it possible to use a closed impeller or vane passages for pumping suspensions having solids, and
- to reduce the power needed to win the friction between the shroud and the volute casing. This is performed by optimizing the liquid flow in the volume between the shroud and the volute casing to have a circumferential velocity component that results in minimum power loss.

Brief Description of Drawing

[0028] The impeller of the present invention is described more in detail below, with reference to the accompanying drawings, in which

Fig. 1 illustrates schematically a comparison between the efficiency curves based on EU- regulations and on a present series of centrifugal pumps,

Fig. 2 explains schematically the correlation between the impeller type and the specific speed,

Fig. 3 illustrates schematically a partial axial cross sectional view of a prior art centrifugal pump,

Fig. 4 illustrates schematically a partial axial cross sectional view of another prior art centrifugal pump,

Fig. 5 illustrates schematically the basic functional differences between the pump-out vanes of the impeller of the present invention by comparing such in the total head vs.

flow rate coordinates with both the working vanes and the pump-out vanes of the front shroud of a prior art impeller,

Fig. 6 illustrates the impeller in accordance with a preferred embodiment of the present invention,

Fig. 7 illustrates schematically a comparison between the efficiency curves based on EU- regulations and on centrifugal pumps utilizing the impeller of the present invention,

Detailed Description of Drawings

[0029] Figure 3 is a schematical cross sectional illustration of a prior art centrifugal pump having a closed impeller. The pump of Figure 3 comprises a volute casing 2, a rear wall 4, a shaft 6 and an impeller 8 attached to the end of the shaft 6. The volute casing 2 comprises an inlet or suction duct 10, and an outlet or discharge duct 12. The rear wall 4, which is fastened to the volute casing 2 comprises some kind of sealing means 14 for axially sealing the shaft 6. Here a stuffing box type sealing is shown. The impeller 8 is, as mentioned already above, a closed one, which means that the working vanes 16 of the impeller 8 are at their both sides covered by a shroud, a rear shroud 18 and a front shroud 20. To the sides of the shrouds 18, 20 opposite to the working vanes 16 so called pump-out vanes 22, 24, respectively, have been arranged. The vanes 22, 24 are usually radial though also somewhat (of the order of 10 - 30 degrees from radial direction) inclined pump-out vanes have been used. The impeller may also be provided with a series of balance holes (not shown) arranged to run through the rear shroud 18 close to the shaft 6. The impeller 8 is arranged to run in the volute casing 2 at a small clearance, i.e. such that the clearance between the rear pump-out vanes 22 and the rear wall 4 is as small as practically possible, i.e. of the order of 0,4 - 1,0 mm. The front side of the impeller 8 is sealed by means of a so called wear ring 26 in relation to the volute casing 2. Usually the wear ring 26 is a cylindrical sleeve arranged at the end of the inlet duct 12 facing the impeller 8. The impeller 8 is provided with a cylindrical extension 28 fitting within the wear ring 26 with a small clearance. The cylindrical extension 28 may also be provided with a specifically treated surface or a specific ring facing the wear ring 26 of the volute casing.

[0030] Figure 4 is a schematical cross sectional illustration of a prior art centrifugal pump having a closed impeller. The centrifugal pump of Figure 4 is identical to the pump of Figure 3 except for the front end of the impeller. Now that the impeller of Figure 3 included the lengthy cylindrical extension 28 cooperating with the wear ring arranged to the casing surface, the casing surface of the pump of Figure 4 is not provided with any wear ring, but the shorter cylindrical extension of the impeller is arranged at a distance 30 from the counter surface of the volute casing such that liquid to be pumped may flow relatively freely to or from the volume between the front shroud and the volute casing.

[0031] To be able to improve the efficiency of the impeller, or that of the pump, the treatment of the leakage flow has to be thought over once again. And, since a centrifugal pump cannot be designed merely for pumping pure water, liquid or suspensions containing more or less solids has to be taken into account, too. Thus, the use of the wear ring remains a secondary means for fighting the leakage flow, as the wear ring is susceptible to considerable wear and difficult maintenance operation if the liquid to be pumped contains solids. Therefor the main concern is the design of pump-out vanes in a novel way. In other words, the aim of the invention is to design pump-out vanes such that they prevent the leakage flow in an optimal way in view of the total efficiency of the impeller. Since the main task of the front pump-out vanes is to prevent the leakage flow, it has to be accepted that they consume power, but their power consumption has to be minimized. In view of their efficiency, it is also important to adjust the pressure difference of the pump-out vanes to be correct at the optimal flow of the pump at or close to the best efficiency point (BEP). The pressure difference is considered to be correct when it produces the smallest total loss of the rotor.

[0032] In view of the above, the front pump-out vanes in the volume between the front shroud and the volute casing are designed to improve the efficiency by means of following three mechanisms:

1. The velocity field thereof is dimensioned such that the friction subjected to the shroud surface is as low as possible, preferably lower than when using a smooth-faced shroud.
2. The pressure the pump-out vanes create is dimensioned such that the pump does not leak at its BEP (best efficiency point) from its outer circumference to the suction duct.
3. The hydraulic energy transferred via the front volume is kept at such a low level that only a minimal flow is allowed via the front volume. Thereby, even if the efficiency of the front pump-out vanes themselves is weak, its effect on the total efficiency of the impeller is negligible. Thus, substantially all of the hydraulic energy is produced by the working vanes operating in high-efficiency closed liquid passages.

[0033] The above represents fresh thinking as this far the front pump-out vanes have been understood and accepted as a necessity that is allowed to decrease the impeller efficiency significantly. Now the front pump-out vanes have been designed in view of minimal friction loss between the shroud and the volute casing. After extensive testing it has been learned that the friction losses are at their minimum when the circumferential velocity component of the liquid in the volume between the front shroud and the volute casing is one half of that of the front shroud.

[0034] When the impeller is constructed in accordance with the above guidelines, the impeller has a front and a rear shroud and liquid passages formed between the shrouds and each successive pairs of working vanes. Both the front and the rear shrouds are provided with front and rear pump-out vanes, respectively. The pump-out vanes create a field of pressure. When pumping liquid with the pump a small or negligible flow compared to the flow via the liquid passages is allowed to be guided to the effective area of the front pump-out vanes. Thereby the losses based on the movement of the impeller in relation to the volute casing are subjected to the front pump-out vanes, which maintain potential energy, while a major part or almost all of the energy of the pump is transferred by the high-efficiency closed liquid passages between the shrouds.

[0035] By connecting a wear ring arranged between the impeller and the volute casing in series with the front pump-out vanes maintaining the potential energy the energy transferred via the front pump-out vanes may be minimized with all volume flows of the pump.

[0036] However, the impeller should be designed to work without the wear ring in case the liquid to be pumped contains solids.

[0037] Therefore the present invention introduces a manner by which the total efficiency of the impeller may be raised in impellers having a low specific speed.

[0038] In traditional pumps like the one cited earlier (US-A1-20090226317) the purpose of the front pump-out vanes is to create a mass flow between the front shroud and the volute casing. However, the pumping of a mass flow takes place with a very low or poor efficiency, as the front pump-out vanes form liquid passages having a very low specific speed (narrow vanes in relation to their length, see Fig. 2), which is not able to get even close to its maximum efficiency. The reason for this is the energy spent by the shroud in high friction in comparison to hydraulic energy recovered from this kind of liquid passages. In pumps like the one cited above the circumferential velocity component of the liquid in the volume between the shroud and the volute casing is almost identical to the velocity of the shroud, whereby the energy lost in friction is nearly at its highest.

[0039] Based on the novel design of the pump-out vanes of the impeller in accordance with the present invention the power needed for running the front pump-out vanes is negligible compared to traditional pump-out vanes. However, the pump-out vanes of the present invention are still able to maintain rotation in the liquid between the front shroud and the volute casing and prevent the leakage flow with minimal power consumption.

[0040] The thinking behind the novel impeller design is that the power consumption of the front pump-out vanes has to be kept low. Figure 5 is a schematic representation of the behavior of the front pump-out vanes of the invention (curve C) compared to the working vanes (curve D) and the pump-out vanes of conventional slurry pumps (curve E) in total head vs. flow rate coordinates. Figure 5 illustrates clearly that the pump-out vanes of the present invention lose their ability to create head when the flow rate increases.

[0041] The mass flow or flow rate is kept small so that the mixing of liquids having different energies (meaning different speed and different direction of speed) is minimized. Additionally, the aim is that when the mass flows of the working vanes and the pump-out vanes meet they would have as closely matching dynamic and static energies as possible so that there is no need to convert static energy to dynamic or vice versa in the energy interface area. If there is a difference the equalizing of the energies means loss. When it is a question of an impeller having no wear ring the circumferential velocity component of the mass flow is kept in about a half of that of the impeller, as has already been discussed earlier in this specification. And when it is a question of an impeller provided with a wear ring, the liquid has to be accelerated to a circumferential velocity higher than a half of the impeller circumferential velocity.

[0042] An exemplary impeller 40 of the present invention is shown in Figure 6. The impeller has a rear shroud 42, a front shroud 44 and working vanes 46 therebetween. The rear shroud 42 has pump-out vanes 48, and the front shroud 44 has pump-out vanes 50, too. The front pump-out vanes 50 have a height h of at most 2%, preferably between 0.5 - 1.5 % of the diameter D of the front shroud of the impeller. The front pump-out vanes may be of equal length, but they may also be of variable length. An option is to have a certain number of full-length vanes and an equal number of shorter vanes, or shorter vanes twice the number of full-length vanes. The number of front pump-out vanes 50 may be higher, the same or lower than that of the working vanes 46. Here, in Figure 6, the number of pump-out vanes 50 is twice that of the working vanes 46. In practice, the front pump-out vanes 50 of the present invention are designed in accordance with the following guidelines:

The number of pump-out vanes 50 may be defined by using the equation

$$\sum_{i=1}^{Z}(l_i) \;/\; D > 8,$$

where

- Z is the number of pump-out vanes 50,
- I is the vane length measured along the leading surface of each front pump-out vane 50, whereby the term $\sum_{i=1}^{Z}(l_i)$ represents the sum of the lengths of the front pump-out vanes, and
- D is the outer diameter of the front shroud 44.

Additionally, the angle of inclination of each pump-out vane 50 at the outer circumference of the vanes β < 25 degrees, the vanes being backwardly curved. Typically, the number of vanes Z = 10, the vane length I = +0.9 ... 1.1 * D (when the vanes are of equal length), preferably I = D and β = 22°. Fig. 6 also shows the cylindrical extension 52 of the front face of the front shroud 44 of the impeller 40, the extension 52 being suitable for cooperating, when in use, with a wear ring arranged to the volute casing of a centrifugal pump.

[0043] When testing the front pump-out vanes 50 it has been learned that such a vane may not extend radially outside the outer circumference of the front shroud 44, as, if it does, the vanes 50 start acting like those of a side channel pump, which is known to have a very low efficiency. However, in view of the working of the present invention the front pump-out vanes 50 should, preferably but not necessarily, irrespective of their length, extend radially to the outer circumference of the front shroud 44, i.e. to the same outer diameter as the working vanes.

[0044] Figure 7 shows the efficiency curve F of the impellers in accordance with the present invention. In other words, the impellers of the pumps having a low specific speed in the series of pumps have been manufactured in the manner described above, and the result is that the entire series of pumps has an efficiency higher than what the EU ecodesign requires.

[0045] As can be seen from the above description a novel centrifugal pump impeller construction has been developed. While the invention has been herein described by way of examples in connection with what are at present considered to be the preferred embodiments, it is to be understood that the invention is not limited to the disclosed embodiments, but is intended to cover various combinations and/or modifications of its features and other applications within the scope of the invention as defined in the appended claims.

Claims

1. An impeller for a centrifugal pump, the impeller (40) having a front shroud (44), a rear shroud (42), and one or more working vanes (46) therebetween, the front shroud (44) having a front face opposite to the face having the working vanes (46), the rear shroud (42) having a rear face opposite to the face having the working vanes (46), the front shroud (44) having an outer circumference and a plurality of front pump-out vanes (50) attached to the front face of the shroud (44), the rear shroud (42) having a plurality of rear pump-out vanes (48) attached to the rear face of the shroud (42), **characterized in that** the front pump-out vanes (50) are dimensioned in accordance with an equation:

$$\sum_{i=1}^{Z}(l_i) \, / \, D > 8,$$

where

Z is the number of front pump-out vanes (50),
I is the vane length measured along the leading surface of each front pump-out vane (50),
D is the outer diameter of the front shroud (44).

2. The impeller as recited in claim 1, **characterized in that** each front pump-out vane (50) has a backward angle of inclination β at the outer circumference of the front shroud equalling to less than 25°.

3. The impeller as recited in claims 1 or 2, **characterized in that** the front pump-out vanes (50) have a height h of less than 2% of the diameter D of the front shroud (44) of the impeller (40).

4. The impeller as recited in accordance with any one of the preceding claims, **characterized in that** the front pump-out vanes (50) have a height h of 0.5 - 1.5 % of the diameter D of the front shroud (44) of the impeller (40).

5. The impeller as recited in accordance with any one of the preceding claims, **characterized in that** the front pump-out vanes (50) are of equal length, whereby the vane length I = 0.9 ... 1.1* D.

6. The impeller as recited in accordance with any one of the preceding claims, **characterized in that** the number of front pump-out vanes is 10.

7. The impeller as recited in any one of claims 2 to 6, **characterized in that** the backward angle of inclination β of each front pump-out vane (50) at the outer circumference of the front shroud is 22°.

8. The impeller as recited in accordance with any one of the preceding claims, **characterized in** a cylindrical extension (52) of the front face of the front shroud (44) of the impeller (40), the extension (52) being suitable for cooperating, when in use, with a wear

ring arranged to a volute casing of a centrifugal pump.

9. A centrifugal pump using the impeller of any one of claims 1 - 8.

10. Use of the centrifugal pump of claim 9 for pumping liquids and solids-containing liquids.

11. Use of the centrifugal pump of claim 9 for pumping fibrous suspension of pulp and paper or board industry.

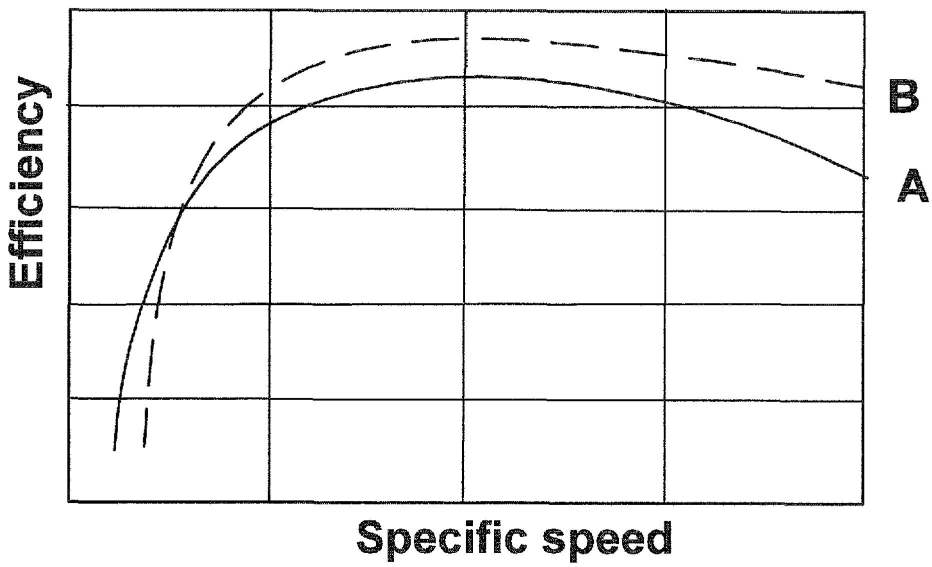

Fig. 1

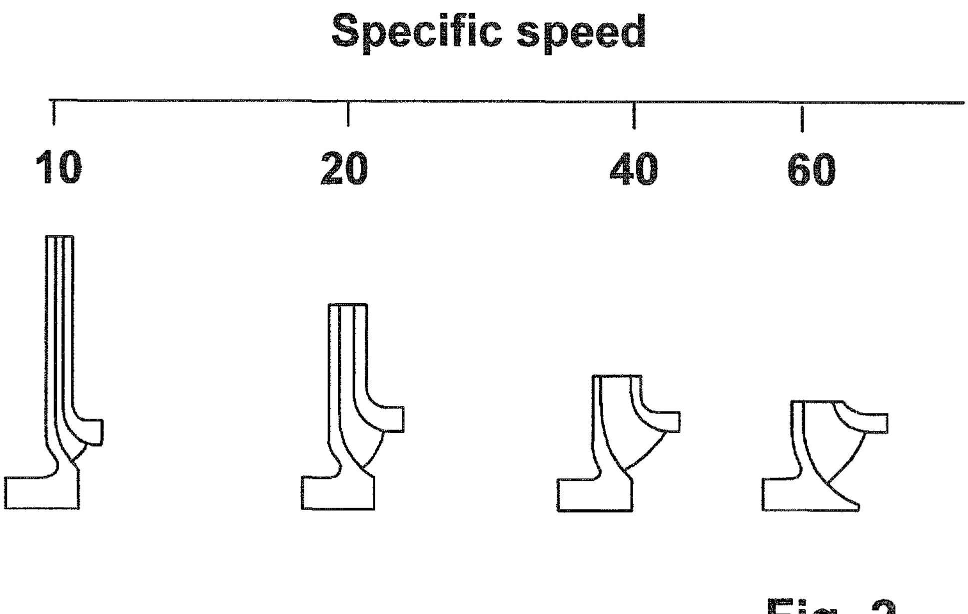

Fig. 2

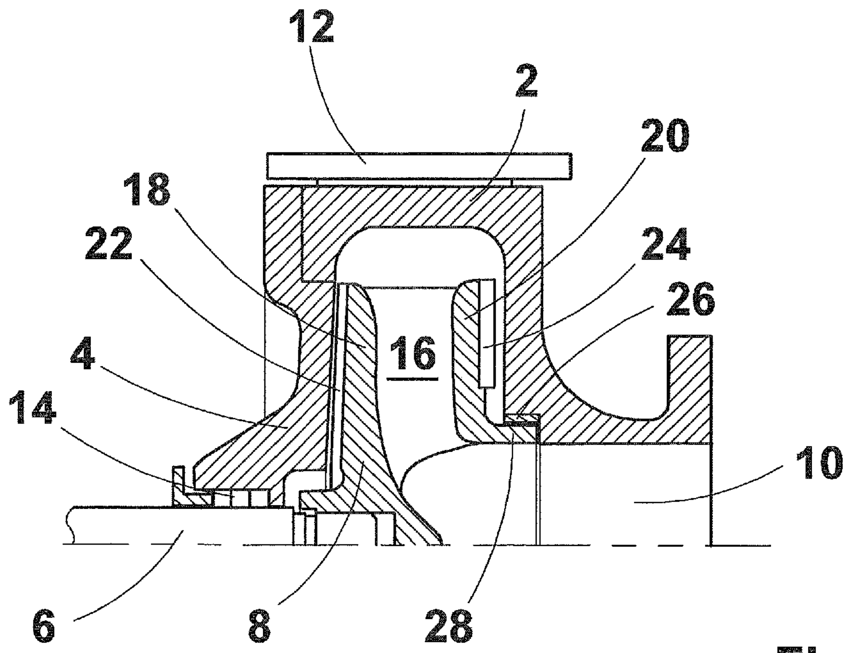

Fig. 3

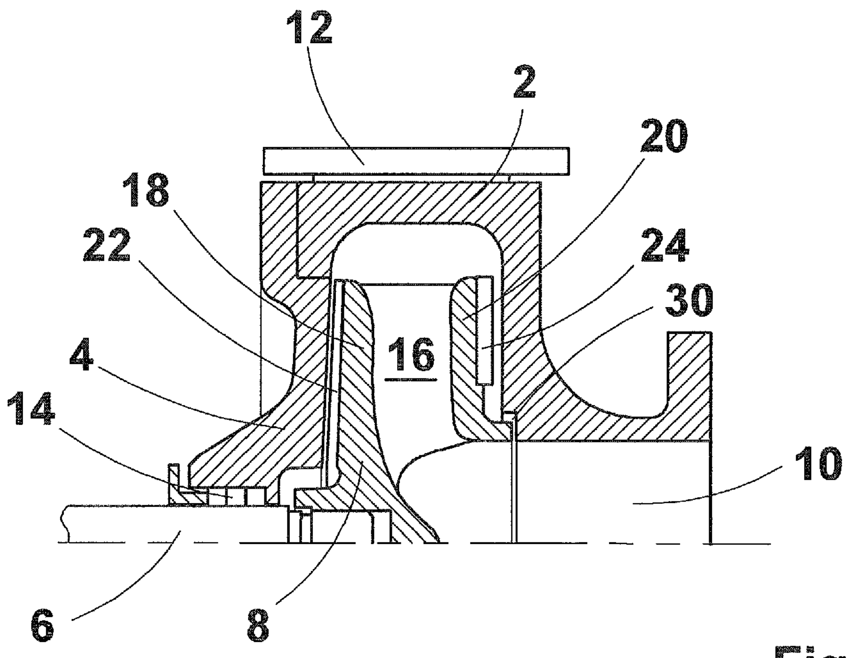

Fig. 4

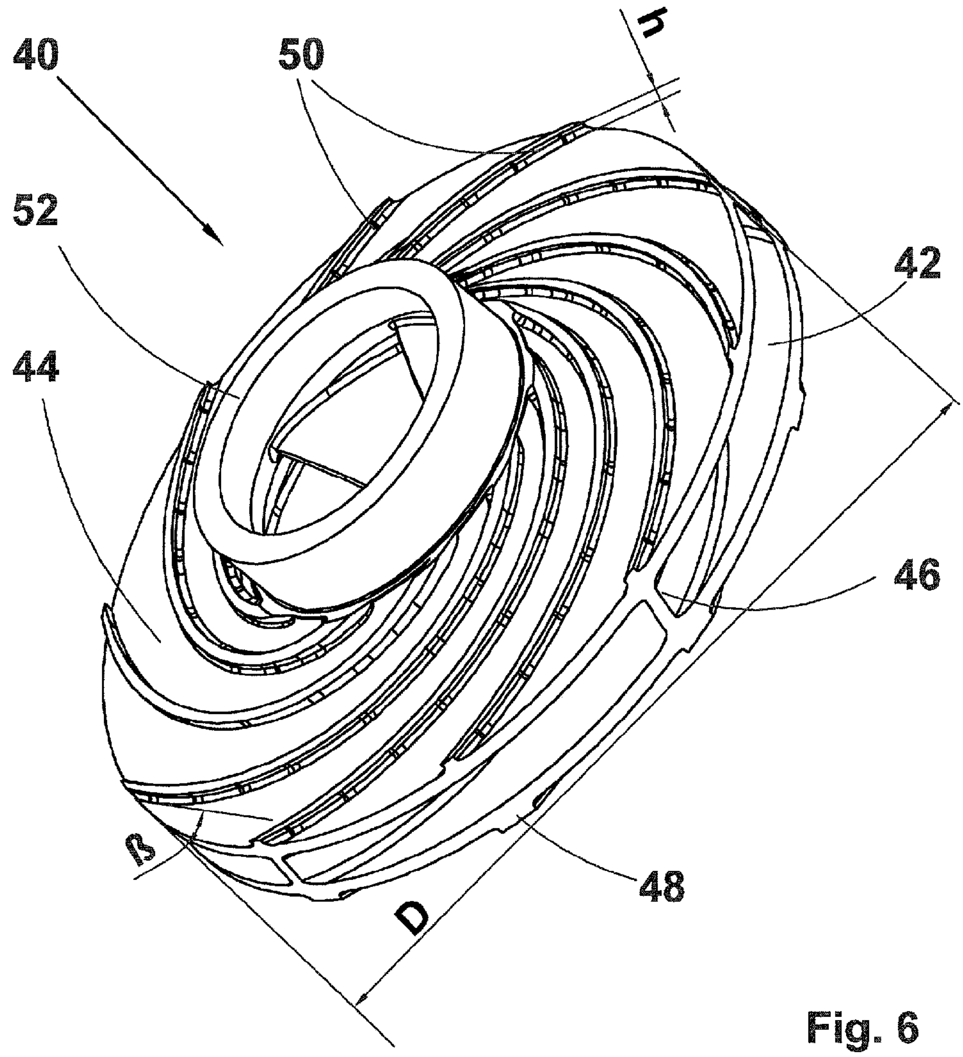

Fig. 6

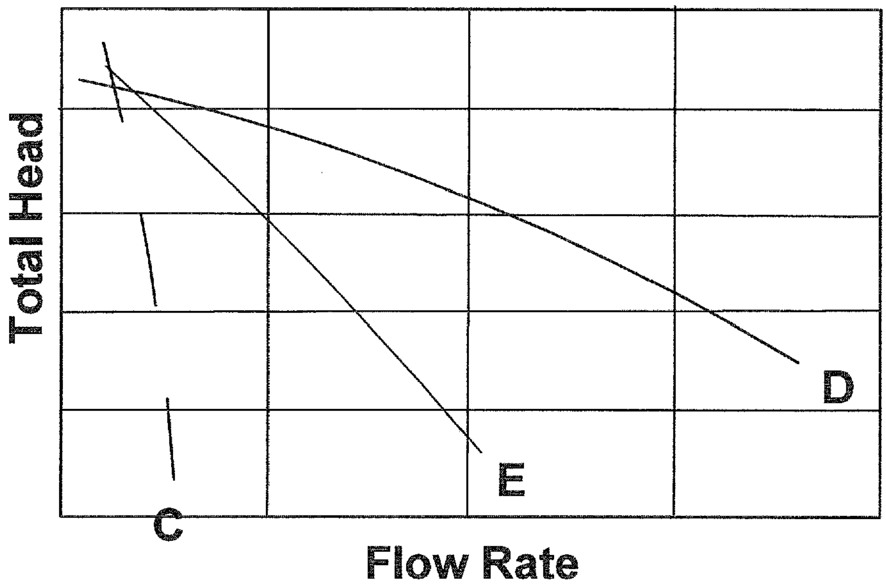

Fig. 5

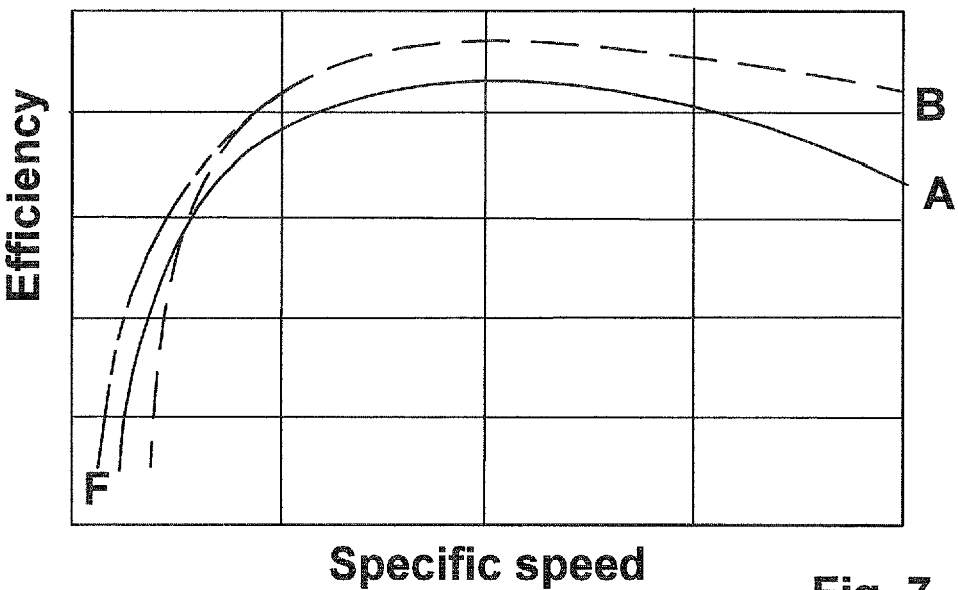

Fig. 7

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Application Number
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Place of search	Date of completion of the search	Examiner
Munich	3 September 2015	Nicolai, Sébastien

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