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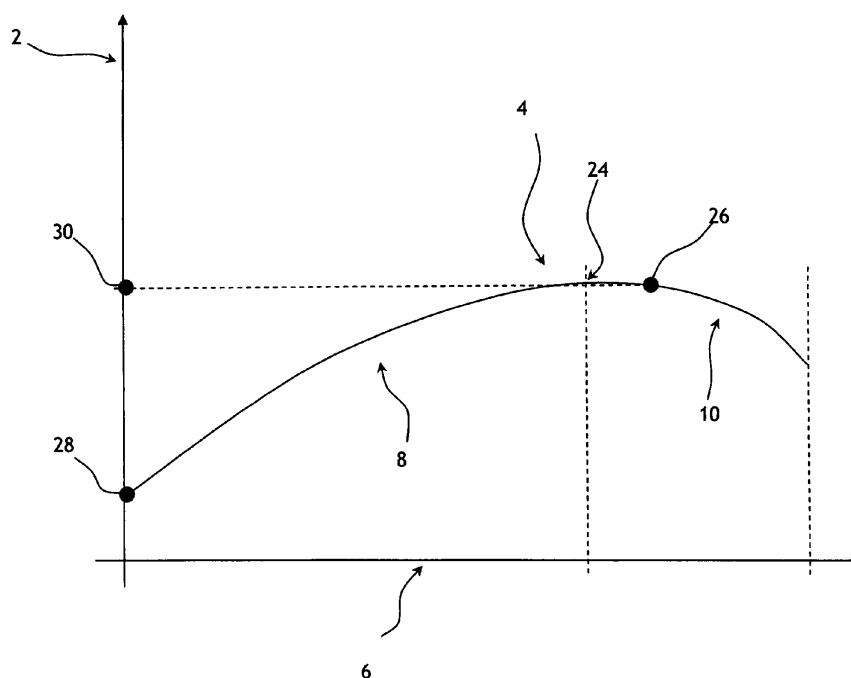
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(54) **Centrifugal pump**

(57) A centrifugal pump having at least one impeller (32), a pump housing and an electrical motor. The pump has a Q-H pump curve (4) having a head H_0 (28) at zero flow and a head H_{ref} (30) corresponding to the highest hydraulic power and that H_{ref} (30) is greater than H_0 (28).

The pump has low energy consumption rate, especially at low flow corresponding to the conditions in which the pump is operated most of the time. Thus, the pump according to the invention is less energy consuming than the prior art centrifugal pumps.

Fig. 1



Description

[0001] The present invention generally relates to a single speed circulator pump. The invention more particularly relates to a circulator pump having a low energy consumption rate in the typical mode of operation.

[0002] It is known to reduce the energy consumption rate of a pump by regulating the speed of the pump. This may by way of example be done by using a frequency converter in a pump. This solution is however relative technical demanding and expensive. Therefore, it is desirable to have a cheaper alternative to this solution.

[0003] When the speed of the pump can be changed during operation it is possible to fit the pump speed to the actual pressure and flow demand, however when a single speed pump is operated a lot of energy is used to build up a higher pressure than required. Therefore, unregulated pumps normally are energy consuming. Normally a pump is only required to perform to its maximum about 5-10% of the time and hence a lot of energy can be saved by adjusting the pump to actual demand.

[0004] It is possible to regulate a pump by using various regulation methods such as proportional pressure and constant pressure regulation. Speed regulation of pumps, however, requires regulation means such as a frequency converter that is an expensive additional feature to the pump.

[0005] Accordingly, it is an object of the present invention to provide less expensive pump that has a reduced energy consumption rate.

[0006] The object of the present invention is achieved by a pump having the features defined in claim 1.

[0007] Preferred embodiments, benefits and further scope of applicability of the present invention will become apparent from the claims and the description given hereinafter.

[0008] The centrifugal pump according to the invention comprises at least one impeller, a pump housing and an electrical motor. The pump has a Q-H pump curve with a head H_o at zero flow and a head H_{ref} corresponding to the highest hydraulic power and H_{ref} is greater than H_o . Therefore, the pump has low energy consumption rate, especially at low flow corresponding to the conditions in which the pump is operated most of the time. Thus, the pump according to the invention is less energy consuming than the prior art centrifugal pumps.

[0009] In one embodiment of the invention the at least one impeller comprises impeller blades that are shaped in a manner so that that H_{ref} is greater than H_o . If the impeller blades are forward swept, by way of example, H_{ref} would be greater than H_o in the Q-H pump curve (where the head at zero flow is denoted H_o and where the head corresponding to the highest hydraulic power is denoted H_{ref}).

[0010] The forward swept blades are swept or curved from the radial inner side to the radial outer side in rotational direction.

[0011] In one embodiment of the invention the pump

the first part of the Q-H curve is an increasing function of the flow. Hereby it can be achieved to have a Q-H curve where H_{ref} is greater than H_o and more specific a pump having a low energy consumption rate at low flow.

[0012] It is also possible to have pump where the entire Q-H curve is an increasing function of the flow.

[0013] In another embodiment of the invention the last part of the Q-H curve is a decreasing function of the flow. Hereby it can be achieved that a pump has a decreasing power consumption rate that so that overload of the motor can be avoided. There may be several ways of achieving that the last part of the Q-H curve is a decreasing. This may by way of example be achieved by choosing a pump housing geometry that restricts the flow rate at high head. For example the pump housing may be designed in such way that the cross sectional area of the volute is reduced or can be reduced as a function of the head. This will cause a restricted flow at high head. Further, it may for example be also possible to use the special design of the impeller to achieve a restricted flow at high head. For example the impeller may be configured so that the distance between the front plate and the back plate can be altered as a function of the head.

[0014] In one embodiment of the invention the pump housing and/or the impeller is configured to introduce flow restriction causing that the end part of the Q-H curve as function of the flow is decreasing. By the term flow restriction is meant means that restrict the flow. Flow restriction means may by way of example be an impeller or a pump housing having a specific geometry.

[0015] In another embodiment of the invention the impeller has forward swept blades. Forward swept impeller blades may contribute to an increasing Q-H curve. Moreover, the size of the impeller may be minimised because an impeller with forward swept impeller blades is capable of creating a higher flow than an impeller with backward swept impeller blades given the same conditions. The impeller may be constructed in a various ways even though the impeller has forward swept impeller blades.

[0016] In another embodiment of the invention the pump has a synchronous motor. This may be an advantage due to the relative high efficiency of synchronous motors especially at low flow.

[0017] The synchronous motor operates synchronously with line frequency. The rotational speed is determined by the number of pairs of poles and the line frequency. A synchronous motor is highly efficient and thus by using a synchronous motor it is possible to achieve a pump with a low energy consumption rate.

[0018] In one embodiment according to the invention the motor is working with constant speed during operation. This can be achieved by using a synchronous motor.

[0019] In one embodiment according to the invention the pump is a circulator pump. The circulator pump may be a glandless (wet-runner) pump. This pump may be used heating, domestic hot water and air-conditioning applications by way of example.

[0020] In another embodiment according to the inven-

tion the motor is a line start permanent magnet motor. A line star permanent magnet motor is basically a combination of an asynchronous motor and a synchronous motor with fixed magnetisation. In a line start permanent magnet motor there is no field winding, instead permanent magnets are used in order to provide the necessary excitation flux.

[0021] A synchronous motor without a rotor winding has no net torque at the speeds different from the synchronous. In order to start the motor from a constant frequency supply (such as the mains) some kind of start winding in the rotor has to be used. During the start, currents are induced in the rotor winding. These currents interact with the stator flux field to produce an asynchronous torque that accelerates the rotor. When the rotor speed is sufficiently close to synchronous speed, and on condition that load torque and inertia are not too high, the rotor will be pulled into synchronism. After the rotor has been synchronised the asynchronous torque vanishes and the motor acts as a synchronous motor except that the rotor magnetisation is supplied by permanent magnets and not by a DC-current in a field winding.

[0022] In one embodiment according to the invention the impeller blades are arced and distributed symmetrically along the periphery of the impeller plate. By this impeller construction it is possible to generate a great flow and achieve the desired Q-H pump curve where

- the first part of the Q-H curve is an increasing function of the flow;
- that last part of the Q-H curve is a decreasing function of the flow and
- H_{ref} is greater than H_0 (where H_{ref} is the head corresponding to the highest hydraulic power and H_0 is the head at zero flow).

[0023] In another embodiment according to the invention the impeller comprises a first set of impeller blades and a second set of blades, wherein the first set of impeller blades the impeller blades are longer than the second set of impeller blades and where the first set of impeller blades and the second set of impeller blades are distributed alternately along the periphery of the impeller plate. Hereby a Q-H curve having the desired properties can be achieved.

[0024] In one embodiment according to the invention $(2/3) \cdot H_{ref} \geq H_0$. A pump having a Q-H curve with these properties will be significantly less energy consuming than the prior art centrifugal pumps.

[0025] It would also be possible to have a pump according to the invention where $(3/5) H_{ref} \geq H_0$. A pump with such Q-H curve would also be significantly less energy consuming than the prior art centrifugal pumps.

[0026] The present invention will become more fully understood from the detailed description given herein below and the accompanying drawings which are given by way of illustration only, and thus, are not limitative of the present invention, and wherein:

Fig. 1 - shows a Q-H curve of one embodiment according to the invention;

Fig. 2 - is a prior art Q-H curve;

Fig. 3a - illustrates the prior art Q-H curve shown in Fig. 2;

Fig. 3b - shows the power-flow curve for a pump having the Q-H curve illustrated in Fig. 3a;

Fig. 4a - illustrates the prior art Q-H curve shown in Fig. 1;

Fig. 4b - shows the power-flow curve for a pump having the Q-H curve illustrated in Fig. 4a;

Fig. 5 - is a comparison of the power-flow curves illustrated in Fig. 3a and Fig. 4a

Fig. 6a - shows the Q-H curve according to another embodiment of the invention;

Fig. 6b - shows the Q-H curve according to a third embodiment of the invention;

Fig. 7a - illustrates a schematically view of Q-H curves for different impeller blade angels;

Fig. 7b - illustrates a schematically view of three different impeller types and

Fig. 8 - illustrates an impeller according to one embodiment of the invention.

[0027] Other objects and further scope of applicability of the present invention will become apparent from the detailed description given hereinafter. It should be understood, however, that the detailed description and specific examples, while indicating preferred embodiments of the invention, are given by way of illustration only, since various changes and modifications within the spirit and scope of the invention defined by the claims will become apparent to those skilled in the art from this detailed description.

[0028] The pumping performance of a centrifugal pump is frequently expressed in the form of a Q-H curve, depicting the head H (normally measured in m) as function of the flow Q (for instance measured in m^3/h) of the pump. The slope of the Q-H curve is determined by the pump construction and particularly by the design of the impeller.

[0029] The majority of circulator pumps are provided with impellers having backwards swept impeller blades. These types of impellers generate a Q-H curve where the head is decreasing with increasing flow (see Fig. 2).

[0030] The hydraulic power P_h is given by the following equation (1)

$$(1) \quad P_h = H \cdot g \cdot \rho \cdot Q$$

Where H is the head, g is gravity, ρ is the density of the fluid and Q is the flow.

[0031] In order to calculate the hydraulic efficiency η_h of the hydraulic part of the pump (the pump housing and the impeller) one needs to know the power P_n supplied to the hydraulic part of the pump as well as the power P_h that the pump transfers to the fluid. This is given by the following equation (2)

$$(2) \quad \eta_h = \frac{P_h}{P_n}$$

[0032] In order to calculate the total efficiency η_t of the pump one have to know the total power P_t supplied to the control (if any) and the motor as well as the power P_h that the pump transfers to the fluid. This is given by the following equation (3)

$$(3) \quad \eta_t = \frac{P_h}{P_t}$$

[0033] The total efficiency η_t of the pump is given by the following equation (4)

$$(4) \quad \eta_t = \eta_{control} \cdot \eta_{motor} \cdot \eta_h$$

where $\eta_{control}$ is the efficiency of the control and η_{motor} is the efficiency of the motor.

[0034] The flow where the pump has the highest efficiency is referred to as the best point.

[0035] When working with the Q-H curve one often focus on the head H_0 at zero flow and the head H_{ref} corresponding to the highest hydraulic power $P_{h, max}$. These points are characteristics of the pump. In the Q-H curves of the prior art centrifugal pumps H_0 is greater than H_{ref} and H curve is normally a decreasing function of Q. If we look at the power-flow curve for the prior art centrifugal pumps the power consumption is relative high especially at low flow. The pumps are operated in the low flow area in the majority of the time. Therefore, it would be advantageous to have a pump that is less energy consuming especially in the low flow area.

[0036] Speed regulated pumps are used to adjust the generated pressure according to the actual demand. Speed regulation requires a regulation of the motor. In

many pumps a frequency converter is used to regulate the speed of the motor, however; such solution is expensive and technical demanding. On the other hand, many unregulated motors have a low efficiency. A high efficiency, especially at low loads, can be achieved by using a line start permanent magnet motor. A line start permanent magnet motor has typically a significant position dependent difference in the inductance (difference in the D- and Q-axis inductance). This difference gives a reluctance torque, so that the total torque production from the motor is given by the combination of the alignment torque and the reluctance torque. By tailoring the geometry to the hydraulic load and the specific application the reluctance torque can be used to increase the efficiency of the motor at lower load (at a slightly reduced efficiency at maximum load). Hereby the energy consumption can be lowered.

[0037] Combining a line start motor and a pump having a Q-H pump curve where H_{ref} is greater than H_0 may eliminate the use of a frequency converter. A pump with a high efficiency may be achieved by combining a line start motor and a pump having a Q-H pump curve where H_{ref} is greater than H_0 . Therefore, the present invention may make it possible to make a high efficiency that is cheaper than the prior art high efficiency pump.

[0038] Traditionally, unregulated pumps are equipped with manual speed change-over means e.g. a rotary knob that may be set in three different speeds. Most pump manufacturers have focused on producing pumps having different regulation curves. Line start motors are generally used for applications in which an exact and constant speed is required. One example of such application is a conveyor belt.

[0039] If a pump is provided with a line start motor there is no speed regulation option. Therefore, pump manufacturers use other types of motors for their pumps. The present invention, however, the pump is equipped with a line start motor. Hereby it is achieved that the efficiency is increased compared with traditional asynchronous motors especially at the lower loads. Therefore, the line start motor makes it possible to save energy.

[0040] Referring now in detail to the drawings for the purpose of illustrating preferred embodiments of the present invention, a Q-H curve 4 according to a preferred embodiment of the present invention is illustrated in Fig. 1. The Q-H curve 4 illustrates the hydraulic head (H) 2 (hereinafter referred to as "head") as function of the flow (Q) 6. It can be seen from Fig. 1 that the first part 8 (approximately the first two thirds) of the Q-H curve 4 has a positive slope. This means that the first part 8 of the Q-H curve 4 is increasing. Moreover, it can be seen that the last part 10 (approximately the last third) of the Q-H curve 4 has a negative slope. Accordingly, the last part 10 of the Q-H curve 4 is decreasing. It can be seen that H_{ref} 30 is greater than H_0 28 (where H_{ref} is the head corresponding to the highest hydraulic power and H_0 is the head at zero flow). Moreover, the global maximum 24 of the Q-H curve 4 is indicated and it can be seen that

$(QH)_{ref}$ is offset slightly to the right side of the global maximum 24 of the Q-H curve 4.

[0041] Fig. 2 shows a prior art Q-H curve 4 (the head 2 as function of the flow 6). It can be seen that the head (H) 2 is a decreasing function of the flow (Q) 6. This Q-H curve 4 corresponds to the Q-H curve of a typical centrifugal type circulator pump. It can be seen that H_0 28 is greater than H_{ref} 30 (where H_{ref} is the head corresponding to the highest hydraulic power and H_0 is the head at zero flow).

[0042] Fig. 3a illustrates the Q-H curve 4 shown in Fig. 2 and Fig. 3b illustrates the corresponding power-flow curve 12 (the power (P) 14 as function of the flow (Q) 6) for a prior art pump having the Q-H curve 4 illustrated in Fig. 3a. In Fig. 3b it can be seen that the maximal flow $Q_{100\%}$ 22, the flow $Q_{25\%}$ 16 corresponding to 25% of the maximal flow $Q_{100\%}$ 22, the flow $Q_{50\%}$ 18 corresponding to 50% of the maximal flow $Q_{100\%}$ 22 and the flow $Q_{75\%}$ 20 corresponding to 75% of the maximal flow $Q_{100\%}$ 22 are lying relative high (the indicated power values 16, 18, 20 and 22) on the power-flow curve 12.

[0043] Fig. 4a illustrates the Q-H curve 4 shown in Fig. 1 and Fig. 4b illustrates the power-flow curve 12 (the power (P) 14 as function of the flow (Q) 6) for a pump having the Q-H curve 4 illustrated in Fig. 4a. In Fig. 4b it can be seen that the flow $Q_{25\%}$ 16 corresponding to 25% of the maximal flow $Q_{100\%}$ 22, the flow $Q_{50\%}$ 18 corresponding to 50% of the maximal flow $Q_{100\%}$ 22 and the flow $Q_{75\%}$ 20 corresponding to 75% of the maximal flow $Q_{100\%}$ 22 are associated with lower power values 16, 18, 20 than in the prior art pump curve 4 illustrated in Fig. 3b.

[0044] Fig. 5 shows a comparison of the power-flow curves illustrated in Fig. 3a and Fig. 4a. It can be seen from Fig. 5 that the maximal flow $Q_{100\%}$ 22', 22" of both the prior art power-flow curve 38 and for the power-flow curve 40 corresponding to a pump having the Q-H curve 4 according to the invention (illustrated in Fig. 4a) are almost coinciding. If we look at the power-flow curve 40 corresponding to the invention it can be seen that the power value at the flow $Q_{25\%}$ 16" corresponding to 25% of the maximal flow $Q_{100\%}$ is significantly lower than the prior art power value at the flow $Q_{25\%}$ 16'. It can also be seen that the power value at the flow $Q_{50\%}$ 18" corresponding to 50% of the maximal flow $Q_{100\%}$ is significantly lower than the prior art power value at the flow $Q_{50\%}$ 18'. Besides, the power value at the flow $Q_{75\%}$ 20" corresponding to 75% of the maximal flow $Q_{100\%}$ is significantly lower than the prior art power value at the flow $Q_{75\%}$ 20'. Accordingly, the pump according to the present invention will have a low energy consumption rate.

[0045] Fig. 6a illustrates the Q-H curve 4 according to an embodiment of the invention. In the Q-H curve 4 the head (H) 2 is plotted against the flow (Q) 6. Approximately the first two thirds 8 of the Q-H curve 4 has a positive slope and thus the first part 8 of the Q-H curve 4 is increasing. Approximately the last third 10 of the Q-H curve 4 has a negative slope and therefore, the last part 10 of

the Q-H curve 4 is decreasing. H_{ref} 30 is greater than H_0 28 (where H_{ref} is the head corresponding to the highest hydraulic power and H_0 is the head at zero flow). Besides, the global maximum 24 of the Q-H curve 4 is and $(QH)_{ref}$ 26 are almost coinciding.

[0046] Fig. 6b illustrates a Q-H curve 4 according to another embodiment of the invention. This Q-H curve 4 is almost similar to the Q-H curve 4 shown in Fig. 6a, however; the global maximum 24 of the Q-H curve 4 is and $(QH)_{ref}$ 26 are displaced relative to one another. $(QH)_{ref}$ 26 is located to the right for the global maximum 24 of the Q-H curve 4.

[0047] Fig. 7a illustrates a schematically view of theoretical Q-H curves 42, 44, 46 for different impeller blade angles. In these curves 42, 44, 46 the height 2 is plotted against the flow 6. The blade angle β is indicated in

[0048] Fig. 7b and represents the angle between the outer periphery of the impeller and the outer side of the impeller blade. Fig. 7a shows that backward swept impellers have a decreasing theoretical Q-H curve 46.

[0049] Fig. 7a also shows that forward swept impellers have an increasing theoretical Q-H curve 46. It can also be seen that the theoretical Q-H curve 44 of a neutral impeller construction where the blade angle β between the outer periphery of the impeller and the outer side of the impeller blade is 90 degrees is flat (horizontal).

[0050] By the term forward swept blades is meant that the angle β is greater than 90°, where β is defined as the angle between the outer periphery of the impeller 32 and the outer side of the impeller blade 34. By the term backwards swept blades is meant that the angle β is less than 90°. By the term neutral swept blades 34 is meant that the angle β is equal to 90°.

[0051] Fig. 7b illustrates a schematically view of three different impeller 32 types where the blade angle β is under 90 degrees, equal to 90 degrees and more than 90 degrees respectively. The blades 34 as well as the rotational direction of the impeller 36 are indicated in the figure.

[0052] Fig. 8 shows an impeller 32 according to one embodiment of the invention. The impeller 32 comprises first set of impeller blades 34 and a second set of blades 35, where the first set of impeller blades 34 are longer than the second set of impeller blades 35 and where the first set of impeller blades 34 and the second set of impeller blades 35 are distributed alternately along the periphery of the impeller plate 48. The where the first set of impeller blades 34 comprises ten blades and where the second set of impeller blades 35 comprise also ten blades. When one take a look at the rotational direction 36 of the impeller 32 it can be seen that both the first set of impeller blades 34 and the second set of blades are forward swept because the angle between the outer periphery of the impeller 32 and the outer side of the impeller blades 34, 35 is greater than 90°.

[0053] Fig. 9 illustrates a typical impeller 32 having a number of back wards swept impeller blades 34, i.e. the impeller blades are swept or curved against the rotational

direction 36. It is indicated that the rotational direction 36 is counter clockwise - the rotational speed is denoted ω and the radius r of the impeller 32 is also indicated. The absolute velocity C of the fluid is given by the sum of the tangential velocity U of the impeller 32 and the relative velocity W relative to the impeller 32. These velocities C , U and W are indicated with arrows. The magnitude of the tangential velocity U of the impeller 32 is given by the product of the radius r and the rotational speed ω :

$$(5) \quad |U| = r \cdot \omega$$

[0054] The blade angle β is less than 90 degrees.

[0055] Fig. 10 illustrates an impeller 32 with a forward swept impeller blade 34, i.e. the blades are swept in rotational direction 36. It is indicated that the rotational direction 36 is counter clockwise like in Fig. 9. The projection C_u of C in the tangential plane is indicated and it can be seen that this forward swept impeller has the following characteristic:

$$(6) \quad C_u > U$$

List of reference numerals

[0056]

2	- Hydraulic head
4	- Q-H curve
6	- Flow
8	- First part of the Q-H curve
10	- Last part of the Q-H curve
12	- Power-flow curve
14	- Power
16, 16', 16'''	- $Q_{25\%}$
18, 18', 18''	- $Q_{50\%}$
20, 20', 20''	- $Q_{75\%}$
22, 22', 22''	- $Q_{100\%}$
24	- Global maximum of the Q-H curve

26	- $(QH)_{ref}$ corresponding to $P_{h, max}$
28	- H_o
30	- H_{ref}
32	- Impeller
34	- Impeller blade
35	- Impeller blade
36	- Rotational direction
38	- Power curve
40	- Power curve
42	- Theoretical Q-H curve
44	- Theoretical Q-H curve
46	- Theoretical Q-H curve
48	- The angle between the outer periphery of the impeller and the outer side of the impeller blade.
48	- Impeller plate

Claims

1. A centrifugal type circulator pump having at least one impeller (32), a pump housing and an electrical motor, where the pump has a Q-H pump curve (4) having a head H_o (28) at zero flow and a head, H_{ref} (30), corresponding to the highest hydraulic power **characterised in that** H_{ref} (30) is greater than H_o (28).
2. A centrifugal pump according to claim 1 **characterised in that** the at least one impeller (32) comprises impeller blades (34) that are shaped in a manner so that that H_{ref} (30) is greater than H_o (28).
3. A centrifugal pump according to claim 1 or claim 2 **characterised in that** the first part (8) of the Q-H curve (4) is an increasing function of the flow (6).
4. A centrifugal pump according to one of the preceding claims **characterised in that** last part (10) of the Q-H curve (4) is a decreasing function of the flow (6).
5. A centrifugal pump according to one of the preceding claims **characterised in that** the pump housing and/or the impeller (32) is configured to introduce flow restriction causing that the end part (10) of the Q-H curve (4) as function of the flow (6) is decreas-

ing.

6. A centrifugal pump according to one of the preceding claims **characterised in that** the impeller (32) has forward swept blades (34). 5
7. A centrifugal pump according to one of the preceding claims **characterised in that** the motor is a synchronous motor. 10
8. A centrifugal pump according to one of the preceding claims **characterised in that** the centrifugal pump is a wet-runner type pump.
9. A centrifugal pump according to one of the preceding claims **characterised in that** the motor is a line start permanent magnet motor. 15
10. A centrifugal pump according to one of the preceding claims **characterised in that** the impeller blades (34) are arced and distributed symmetrically along the periphery of the impeller plate (48). 20
11. A centrifugal pump according to one of the preceding claims **characterised in that** the impeller (32) comprises a first set of impeller blades (34) and a second set of blades (35), where the first set of impeller blades (34) are longer than the second set of impeller blades (35) and where the first set of impeller blades (34) and the second set of impeller blades (35) are distributed alternately along the periphery of the impeller plate (48). 25 30
12. A centrifugal pump according to one of the preceding claims **characterised in that** $(2/3) \cdot H_{ref} \geq H_o$ preferable $(3/5)H_{ref} \geq H_o$ 35

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Fig. 1

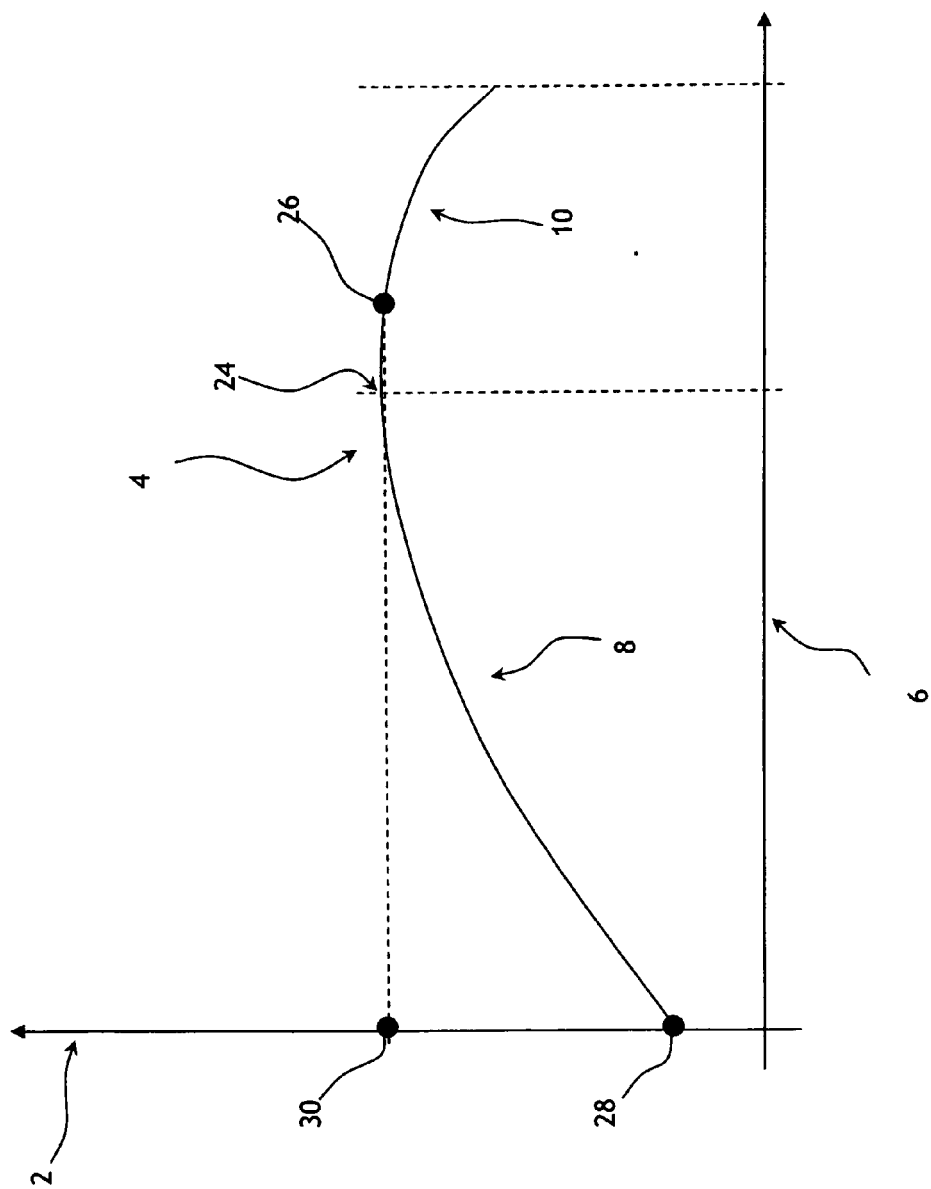


Fig. 2

Prior art

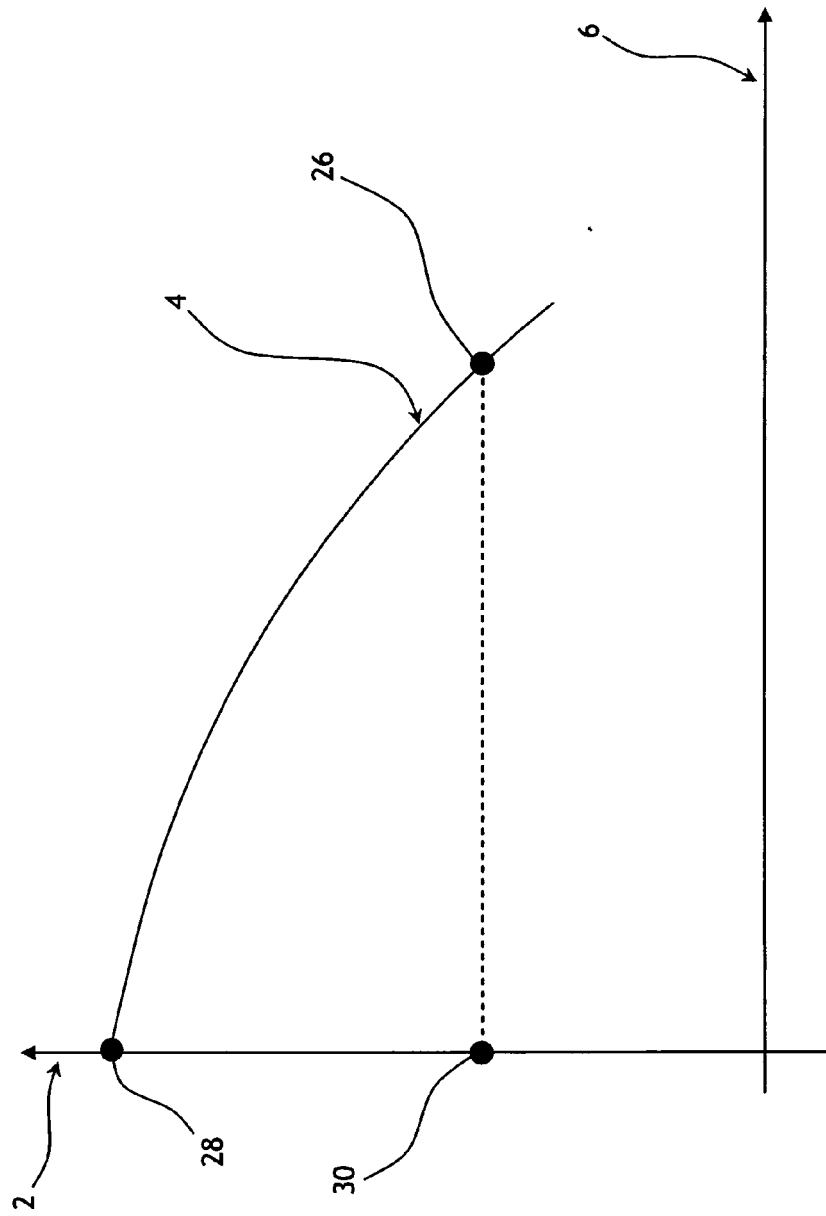
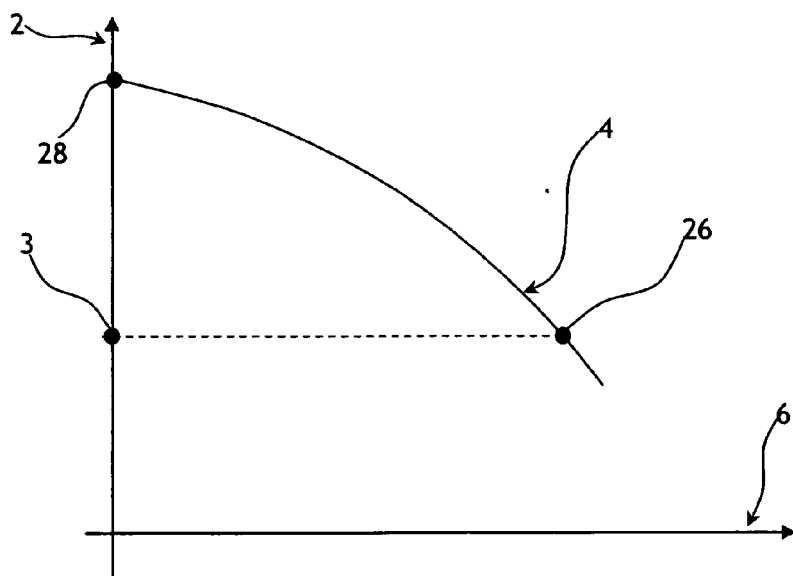


Fig. 3

Prior art

a)



Prior art

b)

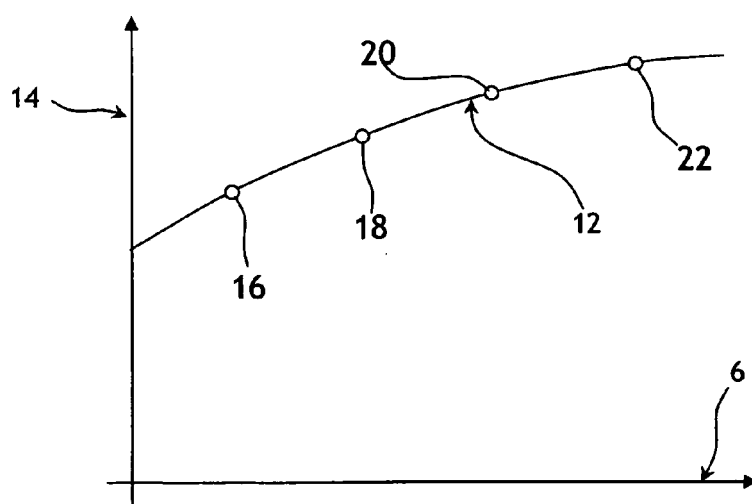
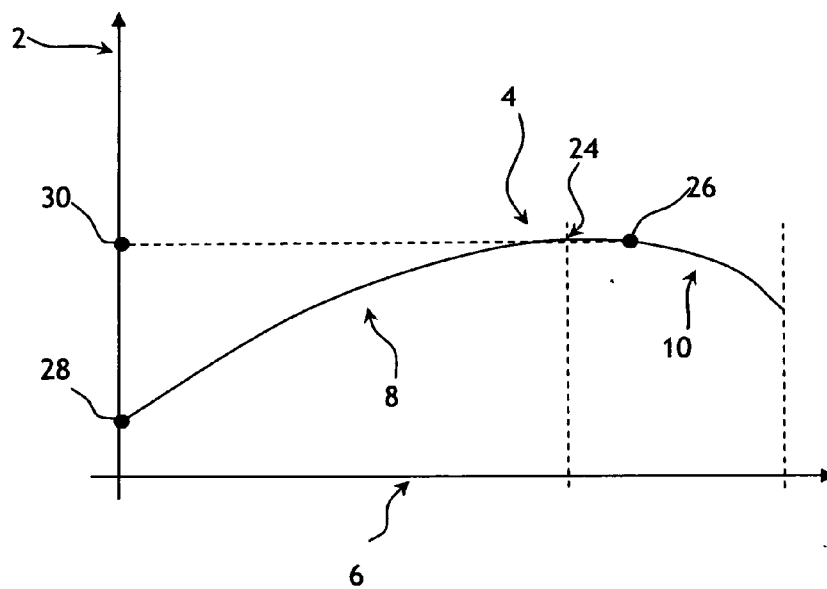


Fig. 4

a)



b)

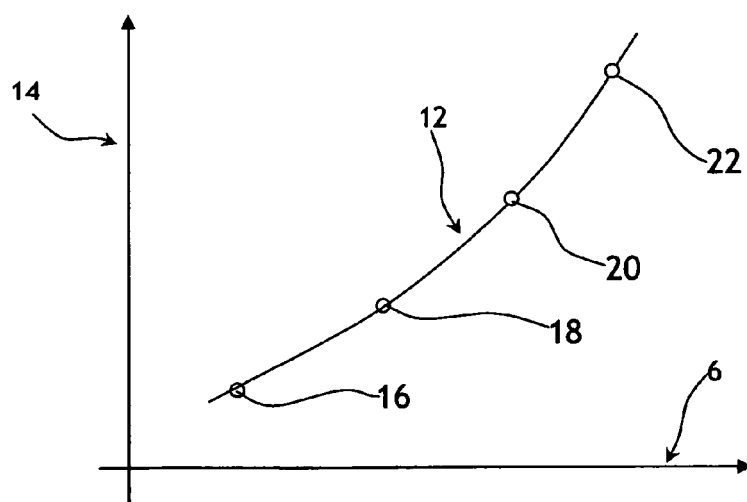


Fig. 5

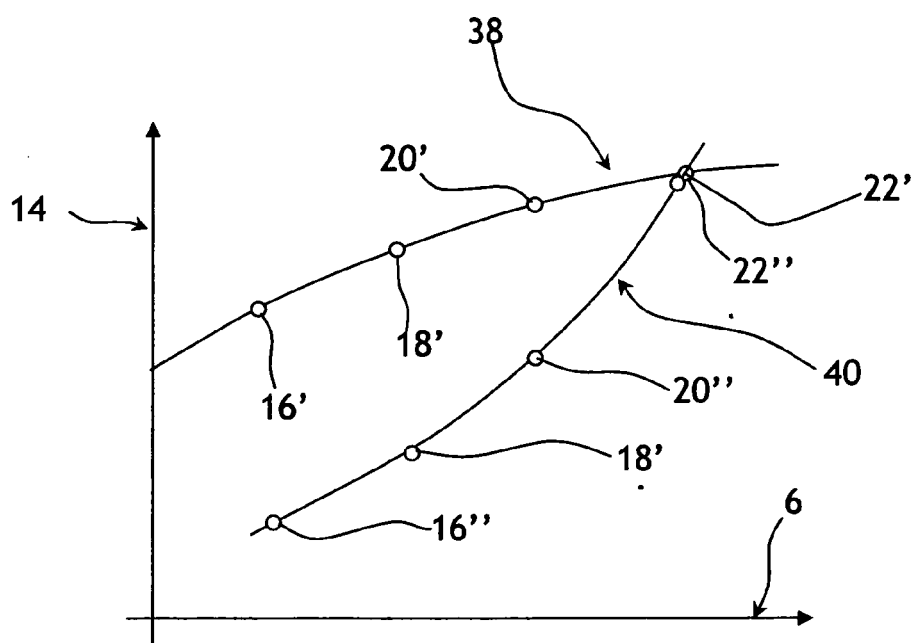
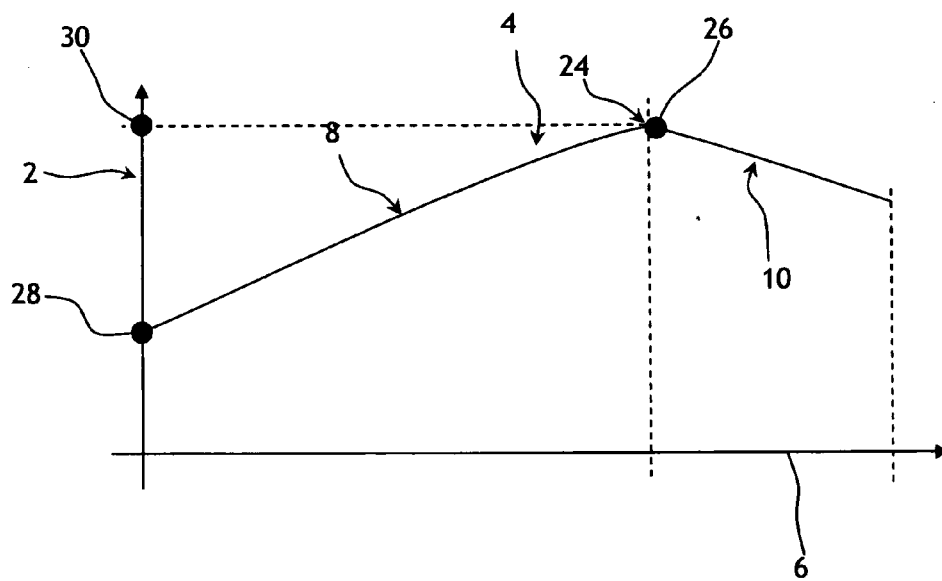


Fig. 6

a)



b)

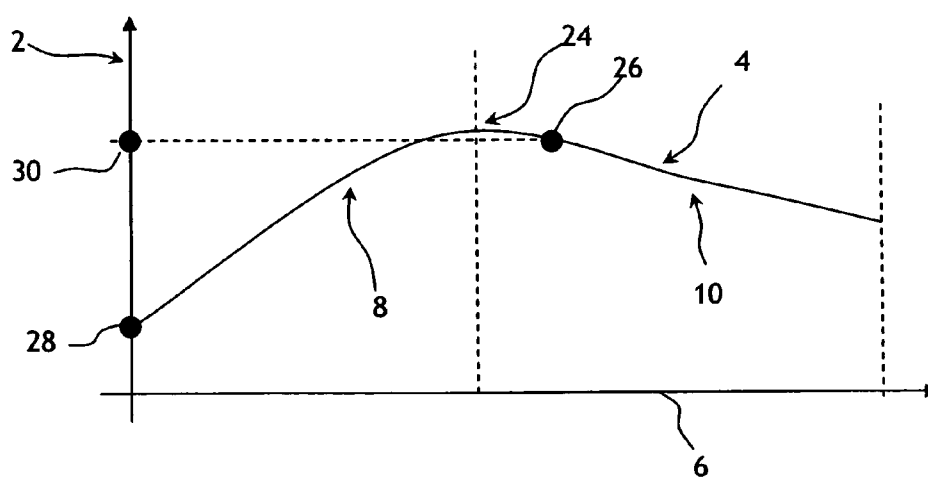


Fig. 7

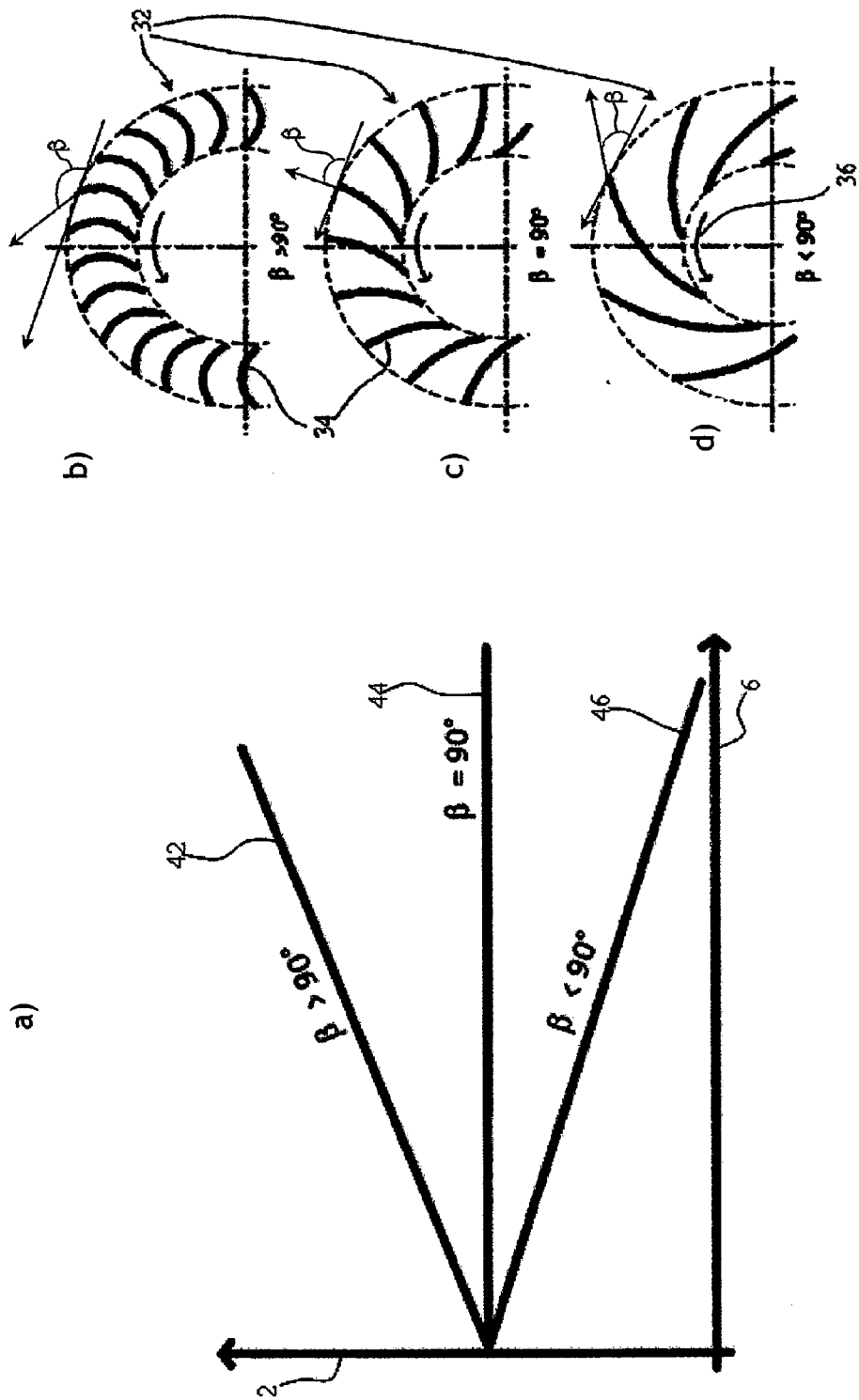
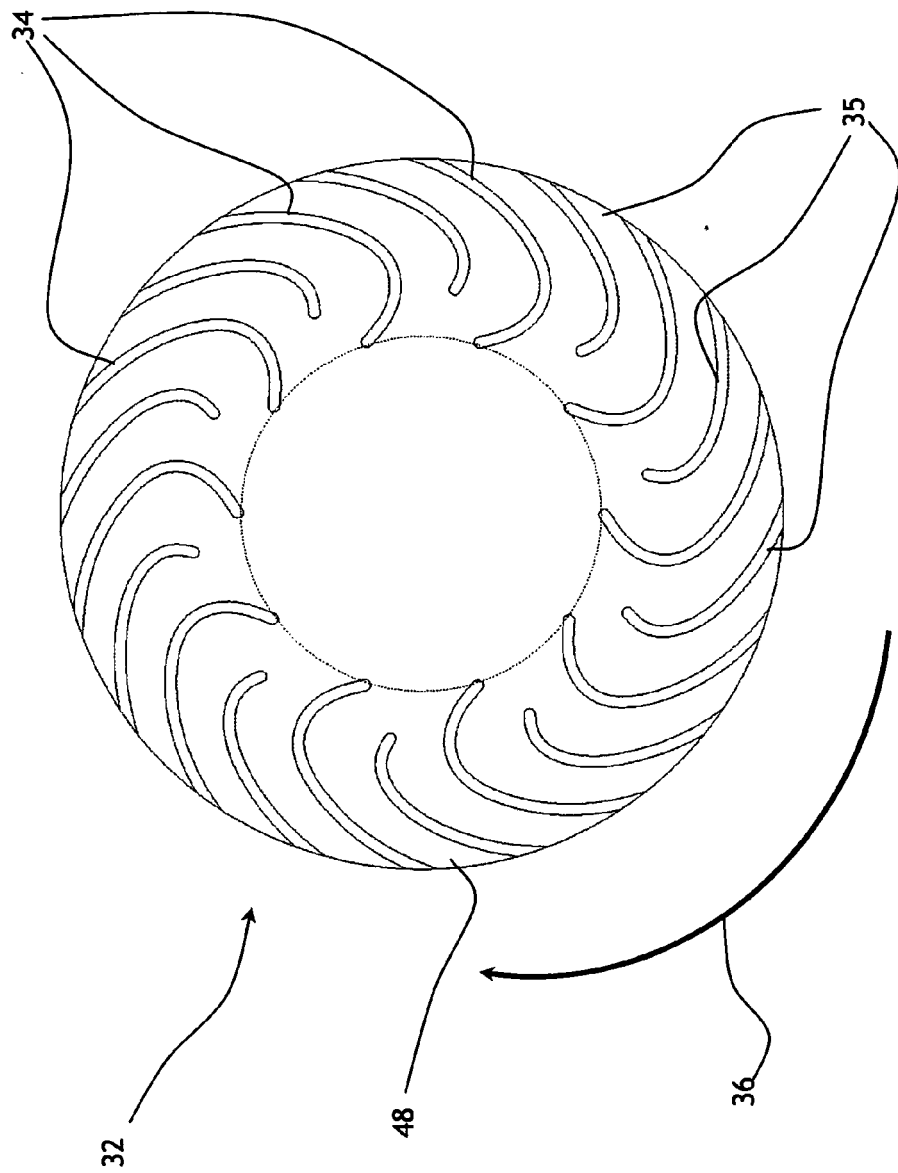


Fig. 8





EUROPEAN SEARCH REPORT

Application Number
EP 10 00 8738

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
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Y	* page 1, line 84 - line 108; claim 1; figure 2 *	7-9,11	F04D29/22 F04D29/24
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