(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 3 530 947 A1

(12) # EUROPEAN PATENT APPLICATION

(43) Date of publication:
28.08.2019 Bulletin 2019/35

(21) Application number: **18157850.1**

(22) Date of filing: **21.02.2018**

(51) Int Cl.:
F04D 1/06 *(2006.01)*
F04D 29/041 *(2006.01)*
F04D 15/00 *(2006.01)*

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA ME
Designated Validation States:
MA MD TN

(71) Applicant: **Sulzer Management AG**
8401 Winterthur (CH)

(72) Inventors:
- **Gåsman, Teemu**
  48600 Kotka (FI)
- **Tiitinen, Kalle**
  33950 Pirkkala (FI)

(74) Representative: **Intellectual Property Services GmbH**
Langfeldstrasse 88
8500 Frauenfeld (CH)

(54) **A CENTRIFUGAL PUMP AND A METHOD OF BALANCING AXIAL FORCES OF THE CENTRIFUGAL PUMP**

(57) The present invention relates to a single- or multi-stage centrifugal pump having a novel balancing arrangement for balancing the axial forces of the centrifugal pump. The balancing arrangement comprises a dynamically adjustable constriction (52) provided in parallel with a slide bearing (28) of the shaft of the centrifugal pump.

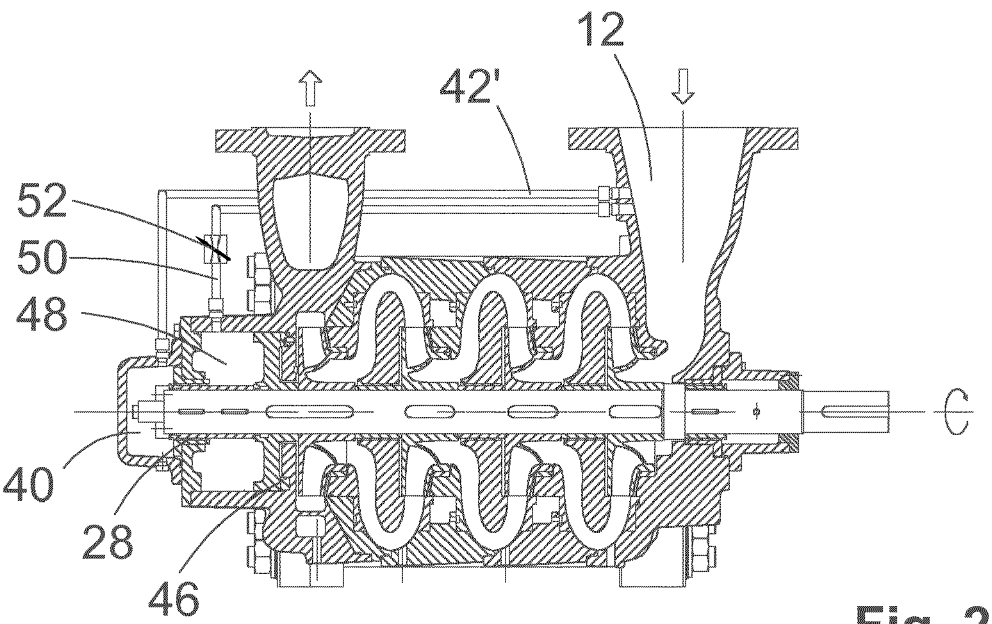

Fig. 2

EP 3 530 947 A1

Description

Technical field

[0001] The present invention relates to a centrifugal pump in accordance with the preamble of claim 1 and a method of balancing axial forces of the centrifugal pump in accordance with the preamble of claim 15. More specifically the present invention relates to single- or multi-stage centrifugal pumps having novel disc- type means for balancing the axial forces of the centrifugal pump.

Background art

[0002] The means for balancing axial forces or thrust of centrifugal pumps are normally in use in multistage pumps, which have a high pressure head, and are provided with several subsequent centrifugal impellers on the same shaft. An axial force is generated while an impeller, or a plurality of impellers, draws liquid axially in the pump and discharges the liquid radially from the pump. The axial force tends to draw the impeller/s towards the pump inlet, whereby the bearings of the pump shaft are subjected to a considerable axial force when keeping the pack of impellers in place. In order to reduce the axial force subjected to the bearings, and, thus, to make it possible to use smaller or lighter bearings or different types of bearings various means for balancing the axial force have been developed.

[0003] Prior art knows two basic types of means for balancing the axial force. One is a so-called drum-type balancing means, and the other a disc-type balancing means. Also hybrid balancing means are known, i.e. one comprising both a balancing drum and a balancing disc. In most cases the balancing means are positioned on the pump shaft behind the last impeller when viewed from the pump inlet towards the pump outlet. However, it is possible, if desired, to construct a centrifugal pump such that the balancing means are between the stages of a multi-stage centrifugal pump or in front of the impeller/s thereof. The disc-type balancing means may be considered as the preferred choice of the two basic balancing means as it adjusts its operation automatically, i.e. slight wear does not affect the operation of the balancing means at all, whereas even the slightest wear of the drum-type balancing means results in a change in the balancing capability of the balancing means. Furthermore, the disc-type balancing means occupies also less space in the axial direction than drum-type balancing means.

[0004] The disc-type balancing means the present invention discusses later on in more detail is formed of a balancing disc fastened on the shaft of the pump and a stationary counter member. In most cases the stationary counter member is arranged to extend from the pump volute or casing radially inwardly between the last impeller, which is the most used construction, or between one of the impellers, and the balancing disc. Most usually the stationary counter member is the rear wall of the last pumping stage of the multi-stage centrifugal pump. The balancing disc and the stationary counter member leave a radially extending annular cavity, so called balancing space, therebetween. Either the balancing disc or the counter member or both have an annular axial extension, sometimes a separate circular ring, at the outer periphery of the balancing disc, i.e. radially outside the annular balancing space, for forming a thin gap between the balancing disc and the counter member. The purpose of the thin gap is to limit the leakage flow of the pressurized balance liquid from the pump. However, it should be understood that the balancing means, i.e. the balancing disc, its stationary counter member and the balancing space, may be located between a pair of impellers and also in front of the impeller/s when viewed from the direction of the inlet opening of the pump. In the latter case it is required that the pressurized liquid is taken to the balancing space along a separate flow passage.

[0005] The disc-type balancing means functions in its most used constructional alternative such that a part of the liquid pressurized by the impeller or the pack of impellers enters, as is well known in centrifugal pumps, to the cavity behind the impeller of the last pumping stage, and finds its way via the gap between the shaft of the pump or the shaft sleeve of the balancing disc and the stationary counter member to a radially extending balancing space between the balancing disc and the stationary counter member. Thus the above described part of the liquid may be called 'balance liquid'. Now that the pressure of the balance liquid is, in practice, not reduced the full pressure of the pumped liquid affects on the rotary balancing disc pushing the balancing disc, and the pump shaft attached thereto, away from the inlet of the pump, i.e. contrary to the axial force or thrust created by the impellers. Thereby, the true axial thrust loading the bearings of the pump (in case the bearings are capable of absorbing any axial force) is the difference of the two axial forces having opposite directions. By properly dimensioning the balancing means the two opposite forces may be equalized resulting in zero thrust, whereby the shaft bearings may be changed into slide bearings that, by nature, are not capable of carrying any axial load.

[0006] However, while the pressurized balance liquid flows radially outwardly in the balancing space between the balancing disc and its stationary counter member, the balance liquid reaches the thin annular gap between the balancing disc and its stationary counter member close to the outer perimeters thereof. Now that the annular gap is very thin, i.e. its axial dimension is very small, and the pressure difference radially over the gap is relatively high (depending mostly on the head of the pump), the flow velocity of the balance liquid in the thin gap is high. Due to the high velocity of the balance liquid the pressure in the gap between the rotary balancing disc and the stationary counter member is low resulting in that in the area of high flow velocity, i.e. at the gap area, the disc is not able to create any significant axial force, whereby the gap, i.e. the axial dimension thereof, tends to decrease in appropriate conditions unless specific measures are taken. The reduction in the size or axial dimension of the gap may lead to increase in friction in the gap, to generation of heat in the liquid in the gap, to evaporation of the balance liquid, and finally, at its worst, to mechanical contact, either impacts or continuous contact between the surfaces of the rotary balancing disc and its stationary counter member resulting in wear and, in the long run, in the need to replace the worn parts. Additionally, as soon as the friction in the balancing arrangement starts to increase it has, naturally, a negative effect on the power consumption, and overall efficiency of the centrifugal pump.

[0007] The above mentioned specific measures relate to preventing the excess decrease in the size or axial dimension of the gap by dimensioning the volume flow via the balancing means sufficient for all operating conditions. It is a well-known fact that the size or axial dimension of the gap is at its smallest when the capacity or volume flow of the pump is the smallest. Therefore, the balancing arrangement is, in all prior art arrangements, dimensioned to provide sufficient balance liquid film in the narrow or thin gap of the balancing means in such low capacity conditions. This kind of dimensioning, naturally, prevents premature wear of the balancing means and provides the pump with a relatively long lifetime. However, as soon as the capacity of the pump starts to increase, the size of the gap widens and the volume flow via the balancing arrangement increases, though there is no need for such. Such an increase in the volume flow of the balance liquid taken from the main flow decreases the net capacity of the pump and reduces the efficiency of the pump significantly, i.e. a significant amount of the power used for pumping is wasted in pumping the excess balance liquid via the balancing arrangement.

[0008] Thus an object of the present invention is to offer such a novel balancing arrangement for a centrifugal pump that does not reduce excessively the efficiency of the centrifugal pump.

[0009] Another object of the present invention is to offer such a novel balancing arrangement for a centrifugal pump that dynamically controls the volume flow through the balancing means.

[0010] A further of the present invention is to offer such a novel balancing arrangement for a centrifugal pump that keeps the volume flow through the balancing arrangement at its optimum, whereby the power used in pumping the balance liquid via the balancing arrangement is minimized.

[0011] A still further object of the present invention is to offer such a novel balancing arrangement for a centrifugal pump that adjusts automatically its operating clearance.

Disclosure of the Invention

[0012] At least one of the problems is solved and at least one of the objects of the present invention are met with a centrifugal pump having an arrangement for balancing axial forces of a centrifugal pump, the centrifugal pump comprising a pump casing with an inlet, an outlet and at least one centrifugal pumping stage therebetween, a shaft mounted rotatably to the pump casing by means of bearings, an impeller for each pumping stage fastened on the shaft for rotation therewith and a means for balancing axial forces of the centrifugal pump; the balancing means comprising a balancing disc fastened on the shaft for rotation therewith, a stationary counter member extending from the casing radially inwardly between the at least one pumping stage and the balancing disc, an annular balancing space located between the stationary counter member and the balancing disc, a thin gap forming a variable constriction positioned radially outside the annular balancing space between the stationary counter member and the balancing disk, a recirculation flow path leading away from the variable constriction, the recirculation flow path comprising at least one fixed constriction arranged in series with the variable constriction and downstream thereof, and a recirculation pipeline, wherein at least one dynamically adjustable constriction is provided either in parallel with the at least one fixed constriction or in connection therewith, and means for following a variable used for controlling the dynamically adjustable constriction.

[0013] Other characteristic features of the present invention become apparent in the appended dependent claims.

[0014] The present invention brings about the following advantages over the prior art balancing means

- the overall reliability of the centrifugal pump is increased as the mechanical contact between the components of the balancing arrangement is prevented,
- lower power consumption than traditional disc-type balancing means,
- continuous adjustment of the gap between the balancing disc and its counter surface, and
- the balancing means adjusts itself dynamically for different pump properties, especially to varying head.

Brief Description of Drawings

[0015] The centrifugal pump and the method of balancing axial forces thereof of the present invention are discussed more in detail below with reference to the accompanying drawings, of which

Fig. 1 illustrates schematically, and in an axial cross section, a prior art multi-stage centrifugal pump including a disc-type balancing means;

Fig. 2 illustrates schematically an axial cross section of a multi-stage centrifugal pump in accordance with a first preferred embodiment of the present invention;

Fig. 3 illustrates schematically an axial cross section of a multi-stage centrifugal pump in accordance with a second preferred embodiment of the present invention;

Fig. 4 illustrates schematically an axial cross section of a multi-stage centrifugal pump in accordance with a third preferred embodiment of the present invention;

Fig. 5 illustrates schematically an axial cross section of a multi-stage centrifugal pump in accordance with a fourth preferred embodiment of the present invention;

Fig. 6 is a schematical representation of a prior art arrangement of balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 1;

Fig. 7 is a schematical representation of a novel arrangement of balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 2 in accordance with the first preferred embodiment of the present invention;

Fig. 8 is a schematical representation of a novel arrangement of balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 3 in accordance with the second preferred embodiment of the present invention;

Fig. 9 is a schematical representation of a novel arrangement of balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 4 in accordance with the third preferred embodiment of the present invention;

Fig. 10 is a schematical representation of a novel arrangement of balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 5 in accordance with the fourth preferred embodiment of the present invention;

Fig. 11 is a schematical representation of a novel arrangement of balancing the axial forces of a centrifugal pump in accordance with a fifth preferred embodiment of the present invention; and

Fig. 12 is a schematical representation of a novel arrangement of balancing the axial forces of a centrifugal pump in accordance with the sixth preferred embodiment of the present invention.

Detailed Description of Drawings

[0016] Figure 1 illustrates an axial cross section of a prior art multi-stage centrifugal pump having a casing 10 with an inlet 12 and an outlet 14, the casing 10 housing a plurality of, here four, impellers 16 attached on a shaft 18 for rotation therewith and a disc-type balancing means. The disc-type balancing means is formed of a disc 20 fastened on the shaft for rotation therewith and a stationary counter member, preferably, but not necessarily, an annular ring 22 fastened on an intermediate wall 24 extending as a part of the casing 10 radially inwardly between the impeller 16 of the last, i.e. in the drawing the leftmost impeller, pumping stage and the balancing disc 20. The shaft 18 is supported on a first and a second slide bearing 26 and 28 at opposite ends of the casing 10, i.e. at the drive end 30 and at the free end 36 of the shaft 18. At the drive end 30 of the shaft 18 the pump casing 10 is provided with a shaft seal 32 (shown schematically) arranged at such a distance from the first slide bearing 26 that a first cavity 34 is left between the shaft seal 32 and the first slide bearing 26. The opposite, free end 36 of the shaft 18 is isolated from the atmosphere by means of an end cover 38, which leaves a second cavity 40 at a side of the second slide bearing 28 opposite the balancing disc 20, i.e. between the end cover 38 and the second slide bearing 28. The first cavity 34 and the second cavity 40 are connected by means of a pipeline 42.

[0017] The balancing of the axial forces of the centrifugal pump is performed such that a part of the liquid to be pumped is allowed to flow, in the shown embodiment, as a balance liquid, from the leftmost impeller 16 in the outlet pressure of the pump to a balancing space 44 between the balancing disc 20 and the intermediate wall 24 and further to a gap 46 radially outside the balancing space 44 between the balancing disc 20 and its stationary counter member 22. As the gap 46 throttles the balance liquid flow the pressure acting on the balancing disc 20 pushes the disc 20 and the shaft 18 fastened thereto to the left, i.e. in a direction opposite to the direction of the thrust created by the impellers 16. The gap 46 may thus be considered a variable constriction, as the pressure acting on the disc 20 tends to change the axial dimension of the gap. From the gap or variable constriction 46 the liquid flows to a balance chamber 48 between the balancing disc 20, or, in view of the function of the balance chamber 48, between the variable constriction 46 and the second slide bearing 28 and further through the second slide bearing 28 to the second cavity 40 (at the side of the second slide bearing 28 facing away from the balancing disc 20) from where the balance liquid advances via the pipeline 42 to the first cavity 34 and therefrom through the first slide bearing 26 to the inlet 12 of the centrifugal pump. In other words, the same liquid that is used for balancing the axial forces is utilized in lubricating the slide bearings at both ends of the shaft. Thus, the flow path of the liquid used for balancing the axial forces, i.e. the thrust of the centrifugal pump, and for lubricating the slide bearings from the impeller 16 of the last pumping stage (in this embodiment, though the balance liquid could be taken from any one of the pumping stages) to the inlet 12 of the centrifugal pump has two fixed constrictions, i.e. the first and the second slide bearings, 26 and 28, and one variable constriction, i.e. the gap 46.

[0018] The above has to be understood as a general example of a prior art construction of a multi-stage centrifugal pump having a disc-type balancing means. There are also multi-stage pumps that are not circulating the balance liquid through the bearings, but use a separate fixed constriction instead. In other words, in more general terms, the balancing arrangement of prior art most often comprises a variable constriction in connection with the balancing disc and at least one fixed constriction arranged in series with the variable one.

[0019] The prior art construction has, however, a relatively serious drawback, as the balancing arrangement may be dimensioned to function optimally in only one operating condition. Such being the case it is self-evident that the operating condition chosen as the design condition is the one where the mechanical durability of the disc- type balancing means is the dictating factor. The risk of mechanical contact between the balancing disc and its counter member is at its highest when the capacity of the pump is at its lowest. Now that the size or axial dimension of the gap, i.e. the variable constriction, is, in theory, without any specific measures, directly proportional to the volume flow through the gap/constriction the low capacity also means that the volume flow through the variable constriction, i.e. the gap 46, is relatively small. Thereby the surfaces facing each other at the gap are at risk of getting in mechanical contact with one another. To prevent this from taking place the fixed constriction/s in series of the balancing means, i.e. the constriction 46, is/are dimensioned to allow such a volume flow through the series of constrictions that is sufficient for keeping the surfaces facing one another out of mechanical contact. However, now that such low capacities are rarely needed the higher capacities in the normal operating range result in a wider gap and in volume flows higher than actually needed. That is, in itself, not a problem, but when taking the efficiency of the pump in consideration, the excess volume flow is a clear problem. In practice, for example, the required minimum volume flow for making the balancing arrangement of a centrifugal pump work, i.e. preventing mechanical contact between the stationary and rotary parts of the balancing arrangement, is of the order of 0.1 - 1 % of the total capacity of the pump, depending largely on the pump characteristics. The volume flow through the balancing arrangement in normal operating conditions of the centrifugal pump may be, again depending on the pump characteristics, of the order of 0.5 - 5 % of the total capacity. In other words, the excess flow through the balancing arrangement may, in this exemplary case, vary between 0.4 and 4% of the total flow of the pump. Thus, in such a case that the flow of the balance liquid through the balancing arrangement may be kept at a level of 0.1 - 1.0 % of the total flow of the pump over the entire operating range of the centrifugal pump, the lifetime savings are considerable. In a pump having a nominal power consumption 3 MW already a reduction of 0.5 % in the balance liquid flow means that the entire sales price of the pump is saved in about 20 years, i.e. during the lifetime of the pump. Naturally, if the reduction is 2.5 % the sales price of the pump may be saved in about 4 years.

[0020] Fig. 2 illustrates schematically an axial cross section of a multi-stage centrifugal pump in accordance with a first preferred embodiment of the present invention. The multi-stage centrifugal pump of this embodiment is for the most parts thereof identical with that of Figure 1, only the parts differing therefrom are discussed in the following. Here, the fixed constriction is formed of a single slide bearing 28, whereby the recirculation pipeline 42' from the second cavity 40 at a side of the second slide bearing 28 facing away from the balancing disc leads directly to the inlet 12 of the pump. In accordance with the first preferred embodiment of the present invention the balance chamber 48 is arranged in communication with the inlet 12 of the pump by means of a recirculation pipeline 50 to which a dynamically adjustable constriction 52 is arranged. The dynamically adjustable construction 52 is, thus, coupled in parallel with the fixed constriction, i.e. the slide bearing 28. Now, the balancing arrangement may be dimensioned such that the volume flow through the variable constriction, i.e. the gap 46, in connection with the balancing disc, is always sufficient but not excessive. For instance, if the capacity of the pump is, for some reason, decreasing, whereby both the pressure acting on the surfaces of the balancing space and the volume flow (as well as the size or axial dimension of the gap 46) in the balancing arrangement also tend to decrease, by opening the dynamically adjustable constriction 52 the volume flow may be maintained, whereby the size of the gap 46 or variable constriction remains sufficient for preventing mechanical contact between the facing surfaces.

[0021] Fig. 3 illustrates schematically an axial cross section of a multi-stage centrifugal pump in accordance with a second preferred embodiment of the present invention. The multi-stage centrifugal pump of this embodiment is for the most parts thereof identical with that of Figure 1, only the parts differing therefrom are discussed in the following. Thus, the fixed constriction is formed of the two slide bearing 26 and 28, whereby the recirculation pipeline 42 from the second cavity 40 at a side of the second slide bearing 28 facing away from the balancing disc leads to the first cavity 34 between the first slide bearing 26 and the shaft seal 32. In accordance with the second preferred embodiment of the present invention the balance chamber 48 is arranged in communication with the inlet 12 of the pump by means of a recirculation pipeline 50 to which an dynamically adjustable constriction 52 is arranged. The dynamically adjustable constriction 52 is, thus, coupled in parallel with the fixed constriction, i.e. the slide bearings 26 and 28. Now, the balancing arrangement may be dimensioned such that the volume flow through the variable constriction, i.e. the gap 46, in connection with the balancing disc, is always sufficient but not excessive. For instance, if the capacity of the pump is, for some reason, decreasing, whereby both the pressure acting on the surfaces of the balancing space and the volume flow (as well as the size or axial dimension of the gap 46) in the balancing arrangement also tend to decrease, by opening the dynamically adjustable constriction 52 the volume flow may be maintained, whereby the size of the gap 46 or variable constriction remains sufficient for preventing mechanical contact between the facing surfaces.

[0022] Fig. 4 illustrates schematically an axial cross section of a multi-stage centrifugal pump in accordance with a third preferred embodiment of the present invention. The multi-stage centrifugal pump of this embodiment is for the most parts thereof identical with that of Figure 1, only the parts differing therefrom are discussed in the following. Thus, the fixed constriction is formed of the two slide bearing 26 and 28, whereby the recirculation pipeline 42 from the second cavity 40 at a side of the second slide bearing 28 facing away from the balancing disc leads to the first cavity 34 between the first slide bearing 26 and the shaft seal 32. In accordance with the third preferred embodiment of the present invention the balance chamber 48 is now arranged in communication with the recirculation pipeline 42 by means of a branch pipeline 60 to which a dynamically adjustable constriction 62 is arranged. The dynamically adjustable constriction 62 is, thus, coupled in parallel with the fixed constriction, i.e. the second slide bearing 28. Now, the balancing arrangement may be dimensioned such that the volume flow through the variable constriction, i.e. the gap 46, in connection with the balancing disc, is always sufficient but not excessive. For instance, if the capacity of the pump is, for some reason, decreasing, whereby both the pressure acting on the surfaces of the balancing space and the volume flow (as well as the size or axial dimension of the gap 46) in the balancing arrangement also tend to decrease, by opening the dynamically adjustable constriction 62 the volume flow may be maintained, whereby the size of the gap 46 or variable constriction remains sufficient for preventing mechanical contact between the facing surfaces.

[0023] Fig. 5 illustrates schematically an axial cross section of a multi-stage centrifugal pump in accordance with a fourth preferred embodiment of the present invention. The multi-stage centrifugal pump of this embodiment is for the most parts thereof identical with that of Figure 1, only the parts differing therefrom are discussed in the following. Thus, the fixed constriction is formed of the two slide bearing 26 and 28, whereby the recirculation pipeline 42 from the second cavity 40 at a side of the second slide bearing 28 facing away from the balancing disc leads to the first cavity 34 between the first slide bearing 26 and the shaft seal 32. In accordance with the fourth preferred embodiment of the present invention the balance chamber 48 is arranged in communication with the recirculation pipeline 42 by means of a first branch pipeline 60 to which a first dynamically adjustable constriction 62 is arranged. The dynamically adjustable constriction 62 is, thus, coupled in parallel with the fixed constriction, i.e. the second slide bearing 28. In addition to the first branch pipeline 60 and the first dynamically adjustable constriction 62 the recirculation pipeline 42 has a second branch pipeline 70 to which a second dynamically adjustable constriction 72 is arranged. The second dynamically adjustable constriction 72 is thus coupled in parallel with the first slide bearing 26. Now, the balancing arrangement may be dimensioned such that the volume flow through the variable constriction, i.e. the gap 46, in connection with the balancing disc, is always sufficient but not excessive. For instance, if the outlet pressure of the pump is, for some reason, decreasing, whereby both the pressure acting on the surfaces of the balancing space and the volume flow (as well as the size or axial dimension of the gap 46) in the balancing arrangement also tend to decrease, by opening at least one of the dynamically adjustable constrictions 62 and 72 the volume flow may be maintained, whereby the size of the gap 46 or variable constriction remains sufficient for preventing mechanical contact between the facing surfaces.

[0024] Fig. 6 is a schematical representation of a prior art arrangement for balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 1. The reference numerals used already above are used here, too. In other word, it is shown that liquid for the balancing arrangement is taken from the outlet 14 of the pump so that the liquid flows through the variable constriction 46 and the second slide bearing or fixed constriction 28 to the recirculation pipeline 42 which takes the liquid via the first slide bearing or fixed constriction 26 to the inlet 12 of the pump. Thus, as there are no adjustable constrictions in the flow path of the liquid used for both balancing the thrust and lubricating the slide bearings, the volume flow through the variable constriction 46 cannot be controlled dynamically, i.e. such that it would match the varying operating conditions of the pump, such as changes in outlet pressure and/or capacity. Such a balancing arrangement has the above discussed drawbacks and problems.

[0025] Fig. 7 is a schematical representation of a novel arrangement for balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 2 in accordance with the first preferred embodiment of the present invention. The reference numerals used already above are used here, too. In other word, it is shown that liquid for the balancing arrangement is taken from the outlet 14 of the pump so that the balance liquid flows through the variable constriction 46 and the second slide bearing or fixed constriction 28 to the recirculation pipeline 42', which takes the liquid directly to the inlet 12 of the pump. To solve the problem relating to the control of the volume flow through the variable constriction 46 a dynamically adjustable constriction 52 is arranged in a pipeline 50 in parallel with the fixed constriction 28. The dynamically adjustable constriction 52 receives its control instructions from the pressure difference between the balance chamber or the pipeline 50 upstream of the dynamically adjustable constriction 52 and the inlet 12 of the pump or the pipeline 50 downstream of the dynamically adjustable constriction 52. i.e. the pressure difference over the dynamically adjustable constriction 52. Thus, as there is a dynamically adjustable constriction in the flow path of the liquid used for both balancing the thrust and lubricating the slide bearings, the volume flow through the variable constriction 46 may be controlled dynamically.

[0026] Fig. 8 is a schematical representation of a novel arrangement for balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 3 in accordance with the second preferred embodiment of the present invention. The reference numerals used already above are used here, too. In other word, it is shown that liquid for the balancing arrangement is taken from the outlet 14 of the pump so that the liquid flows through the variable constriction 46 and the second slide bearing or fixed constriction 28 to the recirculation pipeline 42 which takes the balance liquid via the first slide bearing or fixed constriction 26 to the inlet 12 of the pump. To solve the problem relating to the control of the volume flow through the variable constriction 46 a dynamically adjustable constriction 52 is arranged in a pipeline 50 in parallel with the fixed constrictions 26 and 28. The dynamically adjustable constriction 52 receives its control instructions from the pressure difference between the balance chamber or the pipeline 50 upstream of the dynamically adjustable constriction 52 and the inlet 12 of the pump or the pipeline 50 downstream of the dynamically adjustable constriction 52. i.e. the pressure difference over the dynamically adjustable constriction 52. Thus, as there is a dynamically adjustable constriction in the flow path of the liquid used for both balancing the thrust and lubricating the slide bearings, the volume flow through the variable constriction 46 may be controlled.

[0027] Fig. 9 is a schematical representation of a novel arrangement for balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 4 in accordance with the third preferred embodiment of the present invention. The reference numerals used already above are used here, too. In other word, it is shown that liquid for the balancing arrangement is taken from the outlet 14 of the pump so that the balance liquid flows through the variable constriction 46 and the second slide bearing or fixed constriction 28 to the recirculation pipeline 42, which takes the liquid via the first slide bearing or fixed constriction 26 to the inlet 12 of the pump. To solve the problem relating to the control of the volume flow through the variable constriction 46 a dynamically adjustable constriction 62 is arranged in a by-pass pipeline 60 in parallel with the fixed constriction 28. The by-pass pipeline 60 connects the balance chamber to the recirculation pipeline 42 downstream of the dynamically adjustable constriction 62. The dynamically adjustable constriction 62 receives its control instructions from the pressure difference between the balance chamber or the by-pass pipeline 60 upstream of the dynamically adjustable constriction 62 and the inlet 12 of the pump or the by-pass pipeline 60 downstream of the dynamically adjustable constriction 62, i.e. the pressure difference over the dynamically adjustable constriction 62. Thus, as there is a dynamically adjustable constriction in the flow path of the liquid used for both balancing the thrust and lubricating the slide bearings, the volume flow through the variable constriction 46 may be controlled dynamically.

[0028] Fig. 10 is a schematical representation of a novel arrangement for balancing the axial forces of a centrifugal pump, i.e. those of the centrifugal pump of Figure 5 in accordance with the fourth preferred embodiment of the present invention. The reference numerals used already above are used here, too. In other word, it is shown that liquid for the balancing arrangement is taken from the outlet 14 of the pump so that the balance liquid flows through the variable constriction 46 and the second slide bearing or fixed constriction 28 to the recirculation pipeline 42, which takes the liquid via the first slide bearing or fixed constriction 26 to the inlet of the pump. To solve the problem relating to the control of the volume flow through the variable constriction 46 a dynamically adjustable constriction 72 is arranged in a by-pass pipeline 70 in parallel with the fixed constriction 26 and a dynamically adjustable constriction 62 in by-pass pipeline 60 in parallel with fixed constriction 28. The dynamically adjustable constrictions 62 and 72 receive their control instructions from the pressure difference between the balance chamber or the by-pass pipeline 60 upstream of the dynamically adjustable constriction 62 and the inlet 12 of the pump or the by-pass pipeline 60 downstream of the dynamically adjustable constriction 62. i.e. the pressure difference over the dynamically adjustable constriction 62. Thus, as there is a dynamically adjustable constriction in the flow path of the liquid used for both balancing the thrust and lubricating the slide bearings, the volume flow through the variable constriction 46 may be controlled.

[0029] In the above the dynamically adjustable constriction receives its control information from the pressure difference over the constriction itself. However, data concerning other control variables measurable when the pump is running, like the temperature in the balance chamber 48, the temperature difference between the pump inlet 12 and the balance chamber 48 (see figure 3, for instance), the axial shift of the shaft or the total flows in the recirculation pipelines 42 and 50, may be used to control the operation of the dynamically adjustable constriction 52 (see figure 3, again). Preferably, such data is taken to a control unit, which either compares the data with preprogrammed values or manipulates the data such that appropriate control signals may be sent to control the operation of the dynamically adjustable constriction/s. The temperature difference between the inlet and the balance chamber may be utilized either such that when the difference exceeds a predetermined value the dynamically adjustable constriction/s is/are opened, or such that when the temperature value in chamber 48 exceeds a predetermined value the dynamically adjustable constriction/s is/are opened. In a corresponding manner the axial shift of the shaft is utilized such that when the value of the axial shift reaches a predetermined value (corresponding to a minimum allowable gap in the balancing arrangement between the rotary balancing disc and its stationary counter member) the dynamically adjustable constriction/s is/are opened. The axial shift of the shaft may be determined by means of specific device arranged in connection with the shaft or, for instance, by means of measuring the axial position of the balancing disc in relation to the casing of the pump. The changes in the axial shift mean, naturally, equal changes in the axial dimension of the balancing gap. The total flows are preferably measured by means of arranging flow meters in the corresponding pipelines and passing the flow data to the control unit. Again, in the control unit the flow data is compared to the information stored in the memory of the control unit, and when the overall flow in the recirculation pipelines decreases to a predetermined value the dynamically adjustable constriction is opened. The basis for this kind of an action is the fact that the axial dimension of the balancing gap is directly proportional to the flow through the gap.

[0030] In practice, there are three alternatives for the dynamically adjustable constrictions 52, 62 or 72 discussed above. In its simplest mode the constriction may be an on-off constriction valve, which is arranged to open, when the pressure difference over the constriction falls below a predetermined value. Another alternative is a springloaded constriction valve, which starts opening when the pressure difference over the constriction falls below a predetermined ideal value. If the pressure difference does not return to its ideal value or continues to decrease, the valve continues opening until a balance, i.e. ideal value, is reached. And a third alternative is to input the value of the variable to be followed (pressure difference, temperature, temperature difference, axial shift, total recirculation flow) to a control unit, which, when needed, provides an electrically controlled dynamically adjustable constriction/s with operating instructions.

[0031] In addition to what is disclosed above, the present invention includes a number of alternatives. For instance, when, in the above, the balancing disc and its counter member are discussed, the detailed construction thereof has a number of alternatives.

[0032] Thus, either the balancing disc or the intermediate wall having the counter surface is provided with a raised annular part such that the balancing space is formed radially inside the raised annular part, though such a raised annular part may also be found on both facing components. The raised annular part or parts may be either integral elements of the balancing disc and/or the intermediate wall or separate elements fastened thereon. The surface of the raised annular part and the surface facing thereto are, at least for the most part thereof, machined smooth surfaces so that the friction between the facing surfaces is as small as possible in case the surfaces get into contact with one another. The two surfaces facing one another are parallel and their direction may be radial, though also an inclined direction is possible, as long as the surfaces are not cylindrical.

[0033] Also, in the above, the slide bearing/s has/have been presented as the only options for the fixed constriction arranged in series of the variable constriction. However, if, for instance, desired or needed, the shaft of the centrifugal pump may be sealed between the balancing disc and the slide bearing, whereby it is not easy to arrange the bearing to function as the fixed constriction. Also, when the balancing disc is arranged between the pumping stages of a multi-stage centrifugal pump, the bearing is hardly at a side of the pumping stage but at a distance thereof, whereby it is not easy to arrange the bearing to function as the fixed constriction. Thus, a separate fixed constriction has to be used as disclosed in Figure 11. The separate fixed constriction 80 is preferably, but not necessarily, a throttling provided in a conduit or pipeline taking the balance liquid to the inlet of the pump or to some other position having a lower pressure than that downstream of the variable constriction or the balancing gap 46. Figure 11 also shows, as a fifth preferred embodiment of the present invention, a dynamically adjustable constriction 82 arranged in parallel with the fixed constriction 80 and means for controlling the dynamically adjustable constriction by means of following the pressure difference over the constriction.

[0034] And further, in the above, the recirculation pipeline bringing back the fluid used in balancing the axial thrust of the pump is taught to terminate either to the pump inlet 12 or the first cavity 34 between the shaft sealing and the slide bearing (see Fig. 3, for instance). However, the pipeline may, in practice, lead to any such place that is in a lower pressure than that in the balance chamber. Thus, the pipeline may lead to a tank or another vessel upstream of the pump inlet 12, or to one of the pumping stages of the actual pump, just to name a few alternatives. In the claims the various above discussed positions for receiving the balance liquid is generally referred to as 'balance liquid discharge cavity'.

[0035] Further, now that the above discussed embodiments and various alternatives always require that there is, in series with the variable constriction, i.e. the balancing gap, a fixed constriction either in the form of at least one slide bearing (Figs. 2 - 10) or a separate fixed constriction (Fig. 11), and then a dynamically adjustable constriction coupled in parallel with the fixed constriction, the present invention also proposes, as a sixth preferred embodiment of the present invention, in Figure 12, to combine the functions of the fixed constriction and the dynamically adjustable constriction to one device 92. The device 92 is constructed such that it has an always open cross section corresponding to the fixed constriction and a further openable cross section corresponding to the dynamically adjustable constriction, whereby it is perfectly justified to call the device a dynamically adjustable constriction. In its simples form the new dynamically adjustable constriction is a valve where the valve opening is not capable of closing entirely but leaving an open cross section, which may be then be opened further by means of the three different manners discussed already above. Figure 12 shows the novel dynamically adjustable constriction 92 in series with the variable constriction 46 as well as means for controlling the dynamically adjustable constriction by means of following the pressure difference over the constriction.

[0036] In other words, the phrase "dynamically adjustable constriction" refers here in this application to such a constriction that is controlled by continuously following at least one of the pressure difference over the dynamically adjustable constriction, the temperature difference between the pump inlet and the balance chamber 48, the temperature in the balance chamber, the axial shift of the shaft and the total flows in the recirculation pipelines.

[0037] While the invention has been described herein by way of examples in connection with what are, at present, considered to be the most preferred embodiments, it is to be understood that the invention is not limited to the disclosed embodiments, but is intended to cover various combinations or modifications of its features, and several other applications included within the scope of the invention, as defined in the appended claims. The details mentioned in connection with any embodiment above may be used in connection with another embodiment when such combination is technically feasible.

Claims

1. A centrifugal pump having an arrangement for balancing axial forces of a centrifugal pump,

a) the centrifugal pump comprising a pump casing (10) with an inlet (12), an outlet (14) and at least one centrifugal pumping stage therebetween, a shaft (18) mounted rotatably to the pump casing (10) by means of bearings (26, 28), an impeller (16) for each pumping stage fastened on the shaft (18) for rotation therewith and a means for balancing axial forces of the centrifugal pump;

b) the balancing means comprising a balancing disc (20) fastened on the shaft (18) for rotation therewith, a stationary counter member (24) extending from the casing (10) radially inwardly between the at least one pumping stage and the balancing disc (20),, an annular balancing space (44) located between the stationary counter member (24) and the balancing disc (20), a thin gap (46) forming a variable constriction positioned radially outside the annular balancing space (44) between the stationary counter member (24) and the balancing disk (20), a recirculation flow path leading away from the variable constriction (46) to a balance liquid discharge cavity (12, 34), the recirculation flow path comprising at least one fixed constriction arranged in series with the variable constriction (46) and downstream thereof, and a recirculation pipeline (42, 42') arranged between the fixed constriction and the balance liquid discharge cavity (12, 34),

characterized in at least one dynamically adjustable constriction (52, 62, 72, 82, 92) provided either in parallel with the at least one fixed constriction (26, 28) or in connection therewith and means for following a variable used for controlling the dynamically adjustable constriction (52, 62, 72, 82, 92).

2. The centrifugal pump as recited in claim 1, **characterized in** the recirculation flow path further comprising a balance chamber (48) between the at least one fixed constriction (28) and the variable constriction (46).

3. The centrifugal pump as recited in claim 2, **characterized in** the recirculation flow path further comprising a second recirculation pipeline (50) providing fluid communication from the balance chamber (48) to the balance liquid discharge cavity (12, 34), the dynamically adjustable constriction (52) being provided in the second recirculation pipeline (50).

4. The centrifugal pump as recited in claim 2, **characterized in** the recirculation flow path further comprising a first by-pass pipeline (60) providing a flow path from the balance chamber (48) to the recirculation pipeline (42), the dynamically adjustable constriction (62) being provided in the first by-pass pipeline (60) in parallel with the fixed constriction (28).

5. The centrifugal pump as recited in claim 2 or 4, **characterized in** the recirculation flow path further comprising a second by-pass pipeline (70) providing a flow path from the recirculation pipeline (42) to the balance liquid discharge cavity (12, 34), the dynamically adjustable constriction (72) being provided in the second by-pass pipeline (70) in parallel with the fixed constriction (26).

6. The centrifugal pump as recited in any one of the preceding claims, **characterized in** the fixed constrictions (26, 28) being the first and second slide bearings supporting the shaft (18) of the centrifugal pump.

7. The centrifugal pump as recited in claim 6, **characterized in** the shaft (18) having a free end (36) and a drive end (30), the shaft (18) having a shaft seal (32) at its drive end (30), the shaft seal (32) leaving a first cavity (34) between itself and the first slide bearing (26), the casing (10) being provided at the free end (36) of the shaft (18) with a cover (38) leaving a second cavity (40) between itself and the second slide bearing (28).

8. The centrifugal pump as recited in claim 1, **characterized in** the dynamically adjustable constriction (52, 62, 72, 82) being a valve opening at a certain value of a control variable.

9. The centrifugal pump as recited in claim 1, **characterized in** the dynamically adjustable constriction (52, 62, 72, 82) being a valve starting to open at a certain value of a control variable.

10. The centrifugal pump as recited in claim 1, **characterized in** the dynamically adjustable constriction (92) being a valve having a continuously open cross section and capable of being opened further at a certain value of a control variable.

11. The centrifugal pump as recited in claim 8, 9 or 10, **characterized in that** the control variable is one of a pressure difference over the dynamically adjustable constriction (52, 62, 72, 82, 92), a temperature difference between the pump inlet (12) and the balance chamber (48), the temperature in the balance chamber (48), an axial shift of the shaft (18) and the total flows in the recirculation pipelines (42, 42’, 50, 60, 70).

12. The centrifugal pump as recited in claim 11, **characterized in** a control unit provided to receive data concerning at least one control variable and to manipulate such for controlling the operation of the dynamically adjustable constriction (52, 62, 72, 82, 92).

13. The centrifugal pump as recited in claim 1, **characterized in** the balance liquid discharge (12, 34) cavity having a lower pressure than that in the balance cavity (48).

14. The centrifugal pump as recited in claim 1 or 13, **characterized in** the balance liquid discharge cavity being one of the pump inlet (12), the first cavity (34) between the shaft sealing (32) and the first slide bearing (26), a vessel upstream of the pump inlet (12) and any one of the pumping stages of the centrifugal pump.

15. A method of balancing an axial thrust of the centrifugal pump of claim 1, **characterized by** the steps of

a) determining, for the balance liquid, a volume flow sufficient for preventing mechanical contact between the rotary balancing disc (20) and the stationary counter member (24)

b) determining such a borderline value for a control variable that ensures the sufficient volume flow for the balance liquid,

c) dimensioning the at least one fixed constriction to allow the sufficient volume flow in all operating conditions of the centrifugal pump,

d) providing the balancing arrangement with means for monitoring the control variable,

and when the pump is running,

e) monitoring values of the control variable,

f) comparing the values of the control variable with the borderline value,

g) opening at least one dynamically adjustable constriction, when the value of the control value reaches the borderline value.

16. The balancing method of claim 15, **characterized by**, in step e) monitoring the values of the control variable by collecting such to a control unit, manipulating the values of the control variable to be able to perform step f), and performing step g) by instructing the at least one dynamically adjustable constriction to open.

17. The balancing method of claim 15, **characterized by** using pressure difference over the at least one dynamically adjustable constriction as the control variable and performing step d) by directly utilizing the pressure difference to open the at least one dynamically adjustable constriction.

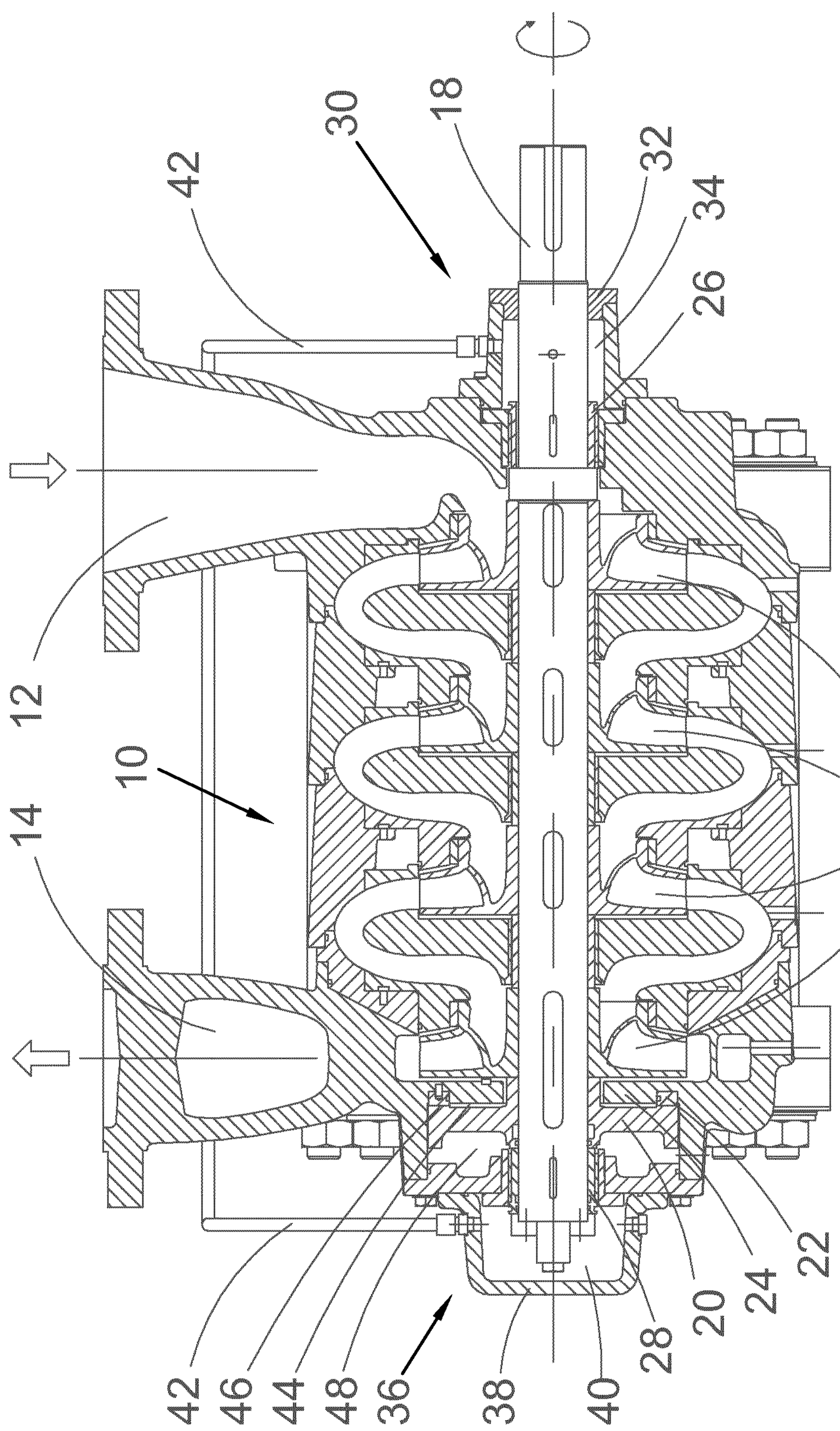

Fig. 1 (Prior Art)

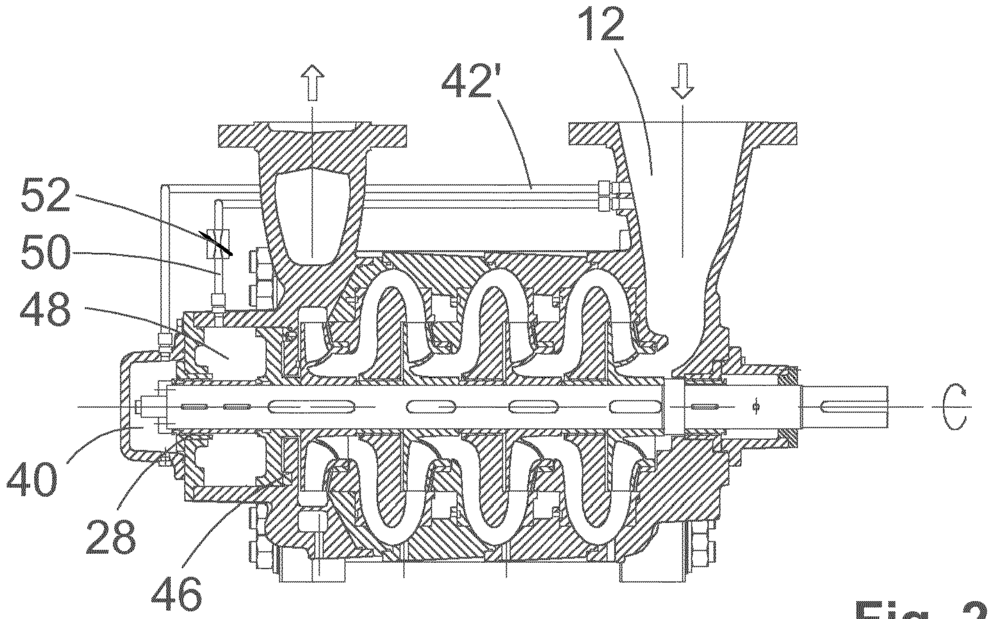

Fig. 2

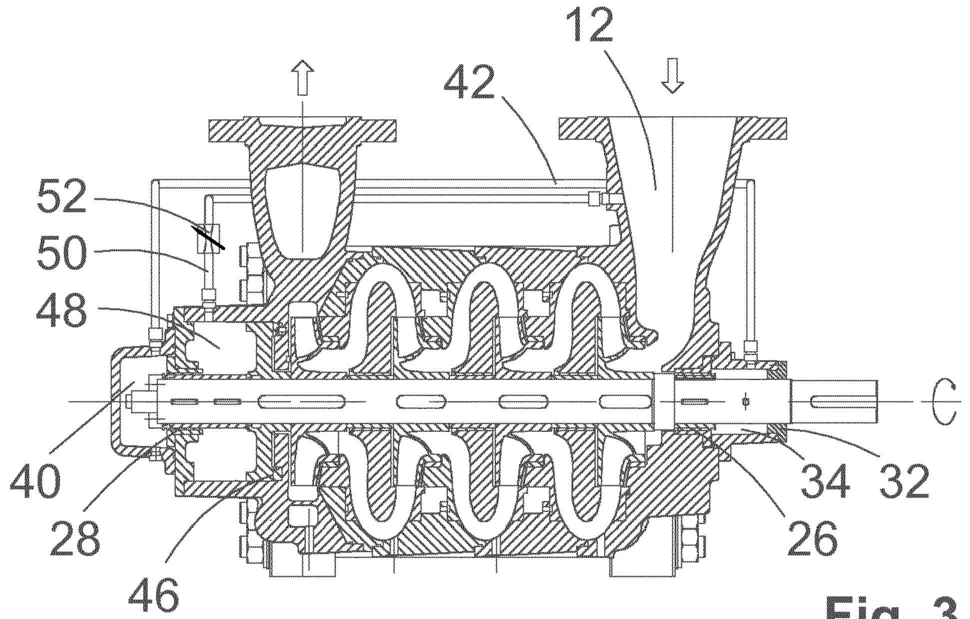

Fig. 3

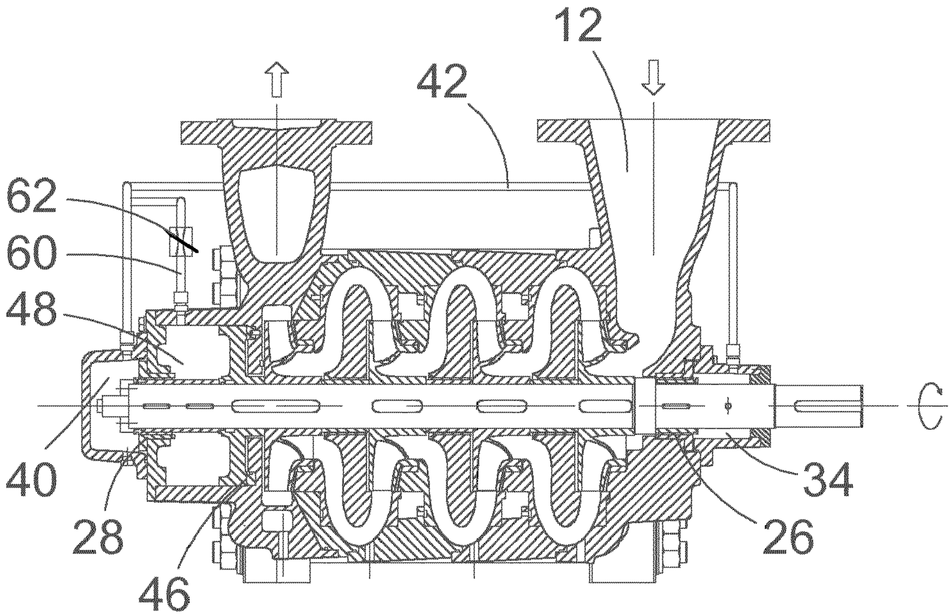

Fig. 4

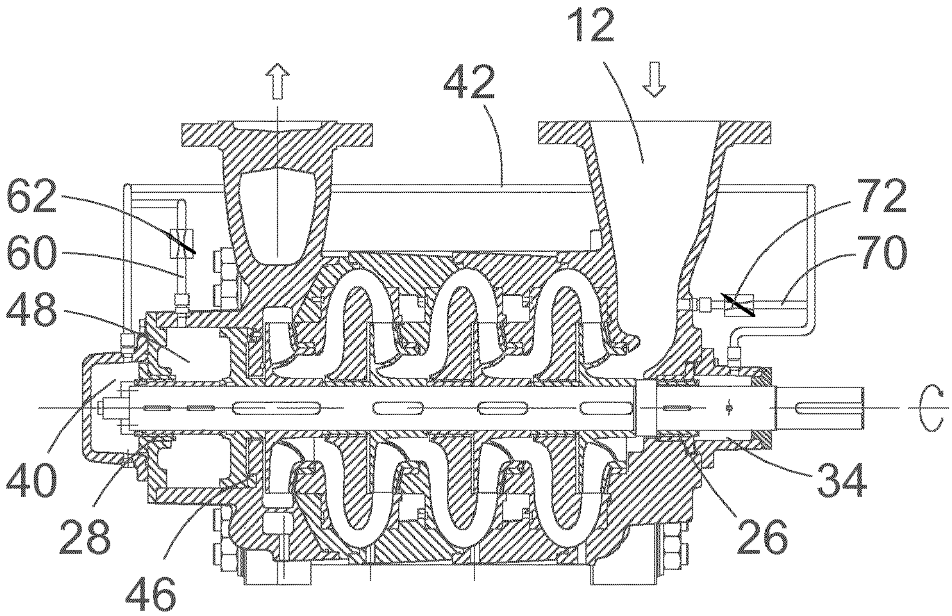

Fig. 5

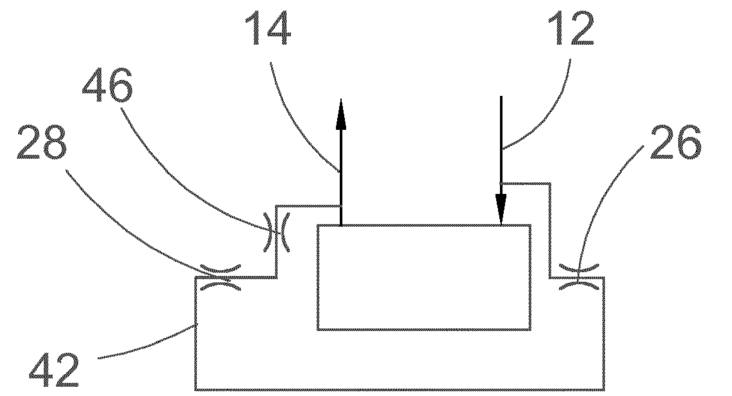

Fig. 6 (Prior Art)

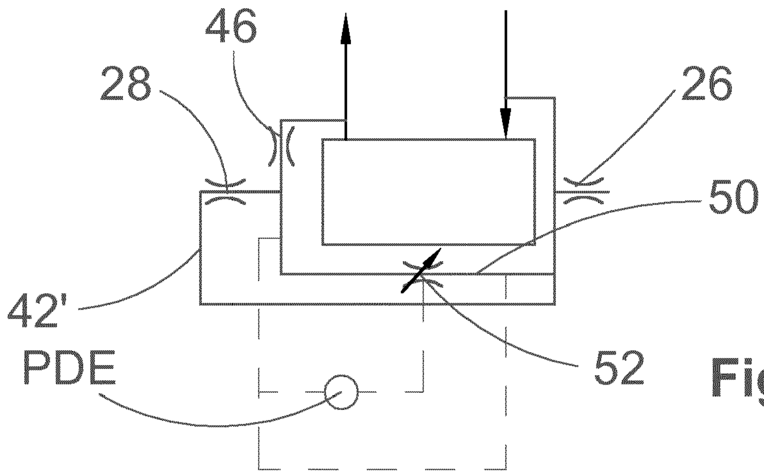

Fig. 7

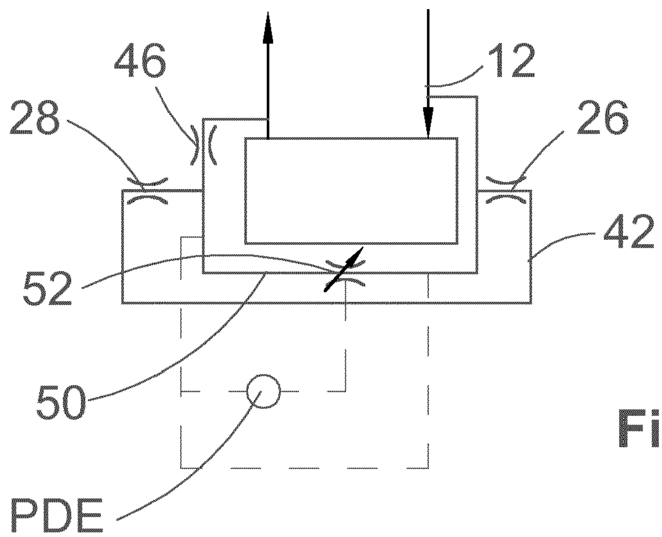

Fig. 8

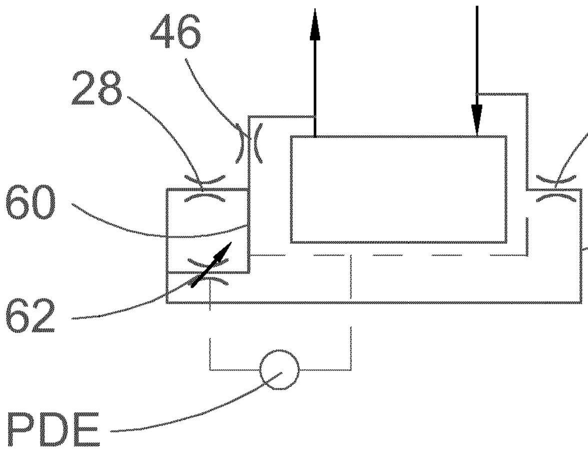

Fig. 9

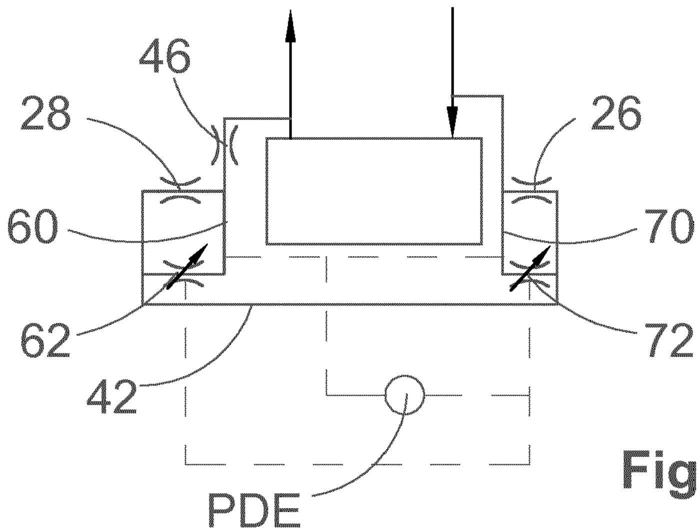

Fig. 10

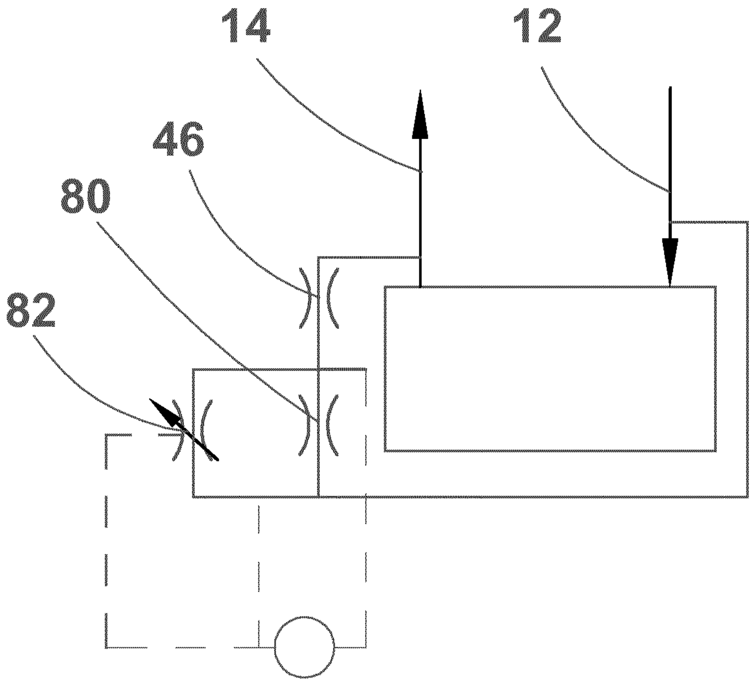

Fig. 11

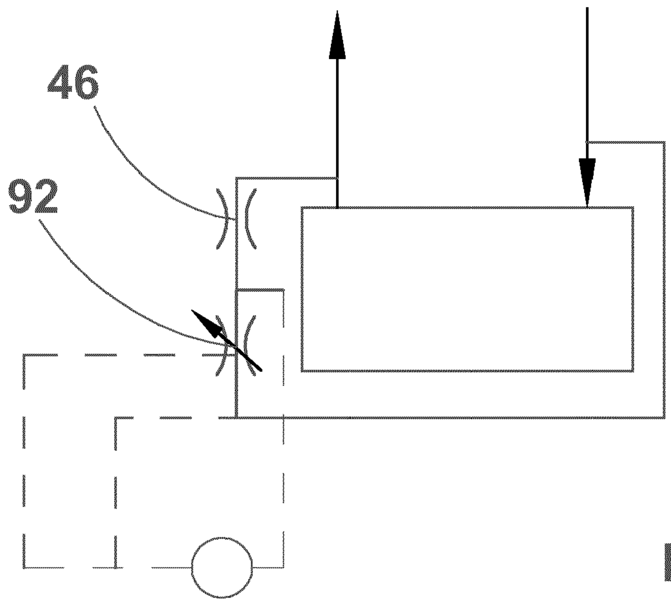

Fig. 12

Europäisches Patentamt
European Patent Office
Office européen des brevets

EUROPEAN SEARCH REPORT

Application Number
EP 18 15 7850

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	DE 11 84 642 B (KLEIN SCHANZLIN & BECKER AG) 31 December 1964 (1964-12-31) * column 1, line 1 - column 4, line 4 * * figures 1-2 * -----	1-17	INV. F04D1/06 F04D29/041 F04D15/00
X	GB 17268 A A.D. 1909 (LEGROS HENRI [BE]) 20 January 1910 (1910-01-20) * page 2, line 21 - page 3, line 4 * * figure 1 * -----	1-17	
X	DE 933 849 C (KLEIN) 6 October 1955 (1955-10-06) * page 1, line 1 - page 3, line 22 * * figure 1 * -----	1-17	

TECHNICAL FIELDS SEARCHED (IPC)

F04D
F01D

The present search report has been drawn up for all claims

2

Place of search	Date of completion of the search	Examiner
The Hague	21 August 2018	Lovergine, A

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C01)

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 18 15 7850

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report. The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

21-08-2018

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
DE 1184642 B	31-12-1964	NONE	
GB 190917268 A	20-01-1910	NONE	
DE 933849 C	06-10-1955	NONE	

EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82