June 15, 1954 A. HILKEN 2,681,010

DEVICE COMPENSATING THE AXIAL THRUSTS EXERTED ON THE SHAFT CARRYING THE TURBINE'S WHEEL OF CIRCULATING PUMPS

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Fig.1

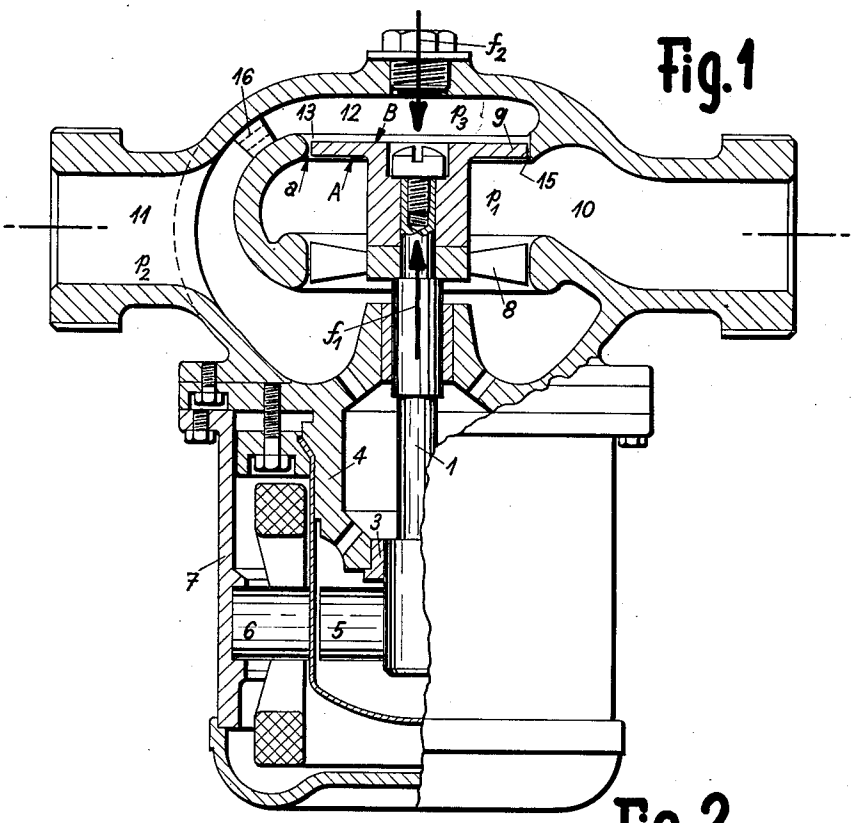
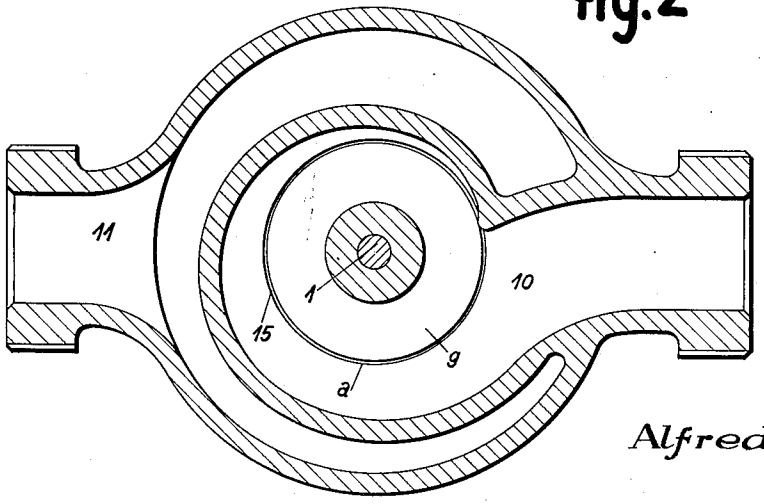


Fig.2



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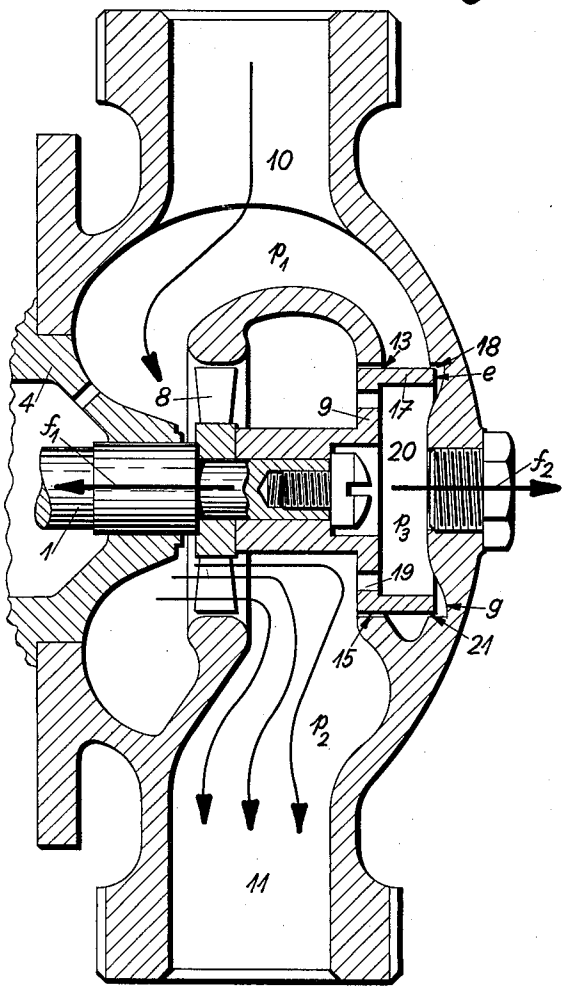
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Fig.3



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DEVICE COMPENSATING THE AXIAL THRUSTS EXERTED ON THE SHAFT CARRYING THE TURBINE'S WHEEL OF CIRCULATING PUMPS

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6 Claims. (Cl. 103—89)

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It is known that, in a circulating pump, the driving shaft of the turbine's wheel is subjected to a non-negligible axial thrust generated by the pushes exerted by the liquid on the impeller wheel. In consequence, it is necessary to provide a large size and well lubricated axial thrust bearing, capable of absorbing this axial thrust.

The present invention relates to a device compensating the axial thusts exerted on the shaft of the impeller wheel of a circulating pump fed by an intake duct and forcing the liquid back in a discharge duct. This device is characterized by a disc carried by the shaft on the impeller wheel and arranged in a passage connecting a space connected to the intake duct with a space connected to the forcing-back duct.

The attached drawing shows, schematically, and by way of example, two embodiments of a circulating pump fitted with the compensating device in accordance with the invention.

Fig. 1 is a part cross-section view of same.

Fig. 2 is a cross-section taken along line II—II of Fig. 1.

Fig. 3 is a cross-section view of another embodiment.

The circulating pump represented on the drawing is of the type having bearings lubricated by the liquid to be set in circulation, the rotor of the impeller driving motor bathing in this liquid, but it is obvious that the invention can be applied to other types of circulating pumps, the pump of which is constituted by an impeller.

In accordance with Figs. 1 and 2 of the attached drawing, the circulating pump comprises, like other known circulating pumps, a horizontal shaft 1 revolving in bearings 2, 3, provided in an integral cast central support 4. On one of the ends of this shaft is fixed the rotor 5 of an electric motor, the stator 6 of which is carried by a casing 7 fixed to the central support. The other end of this shaft 1 carries, on the one hand, the wheel 8 of the impeller and, on the other hand, a disc 9. The impeller wheel is fed by a scroll-shaped intake duct 10 and forces the liquid back in a discharge duct 11. The latter is linked with a space 12 connected to the intake scroll 10 by a passage 13 closed by a disc 9. The face A of this disc is therefore subjected to the pressure p_1 prevailing in the intake scroll 10, whilst the face B of this disc is subjected to the pressure p_3 prevailing in the chamber 12. Now, the latter being connected to the discharge duct, the pressure prevailing in this chamber 12 will be equal to the discharge pressure p_2. Consequently, the disc is subjected to an axial thrust f_2, whilst the axial component

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f_1 of the resultant of the thrusts acting on the turbine's wheel is of opposite direction. Thus, the thrust f_2 compensates, at least partly, the thrust f_1. Now, if the passage 13 is designed with a diameter slightly larger than that of the disc 9, a leakage-annular-slot 15 is obtained all around the disc. This leak's discharge tends to reduce the difference between pressures p_1 and p_3 and thus to diminish the value of the thrust f_2. An automatic equalization of the two thrusts f_2 and f_1 is therefore obtained, on condition that the leak discharge be a function of the axial position of the disc 9 in the passage 13. Now, this can be easily achieved. Indeed, as represented on Fig. 1, as soon as the plane of the face A of the disc 9 protrudes on the plane of the circular edge a, the length of the leakage slot 15 decreasing, the leak discharge increases for a same difference in pressure. The full compensation of the axial thrusts sustained by the driving shaft of the turbine's wheel is thus automatically and very easily obtained. Practical tests have shown that the turbine's shaft assumes immediately a well defined position and that no swinging effect is to be feared.

In order to cut down as far as possible the leak discharge necessary to this compensation, one can connect the chamber 12 to the discharge duct 11 by means of a duct comprising a gauged throttling 16. In this way, a slight increase of the leak discharge, can already provoke an appreciable reduction in the pressure p_3, as the chamber 12 can only be fed with a discharge limited by the gauge 16. It is thus possible to provide a very slight leak discharge for attaining the automatic compensation, so that the total efficiency of the circulating pump fitted with the compensating device will not be appreciably affected.

Fig. 3 of the attached drawing shows another embodiment of the compensating device in accordance with the invention.

The turbine part of the circulating pump has only been illustrated in this figure, the motor part being similar to that represented on Figs. 1 and 2. In addition, the already described parts and elements are designated by the same reference ciphers.

The axle 1, driven by the motor (which is not represented) carries, on the one hand, the impeller wheel 8 and, on the other hand, a disc 9. The wheel 8 is fed by the intake duct 10 and forces the liquid back in the discharge duct 11. The disc 9 revolves in a passage 13 connecting the intake duct to the discharge duct, as de-

scribed in detail with reference to Figs. 1 and 2. However, in this second embodiment, the disc 9 carries a sleeve 17, the end of which revolves inside a boring 18. In addition, the disc 9 comprises openings 19 connecting the discharge duct 11 to a chamber 20 which is limited by the disc 9 and the sleeve 17, and which is connected to the intake duct 10 by means of an annular slot 21.

According to the axial position of the shaft 1, the sleeve 17 is engaged more or less deeply in the boring 18, so that the discharge traversing the annular slot 21 is, for a same difference in pressure, variable according to the axial position of the shaft 1.

The working of this second embodiment is as follows:

The difference in pressure p_1, p_2, generated by the turbine 8 exerts an axial thrust f_1 proportional to the difference between the pressures p_1 and p_2, and to the effective surface of the blading of the wheel 8 increased by the pressure loss in said blading. The disc 9 is subjected, not only to the thrust resulting from the static difference between pressures p_2 and p_3, but also to a dynamic thrust generated by the direction change of the liquid streams leaving the turbine. This dynamic thrust is a function of the kinetic energy of the liquid forced back by the turbine. The pressure p_3 in chamber 20 varies in accordance with the dimensions of the annular slot 21, and therefore in accordance with the axial position of the shaft 1. Now, it is clear that p_3 can vary between the two extreme limits:

$$p_3=p_1 \text{ and } p_3=p_2$$

In order to obtain a full compensation of the axial thrust f_1, it is therefore necessary to ensure that the sum f_2 of the static and dynamic axial thrusts acting on the disc 9 be equal and of opposite direction to the thrust f_1. Now, it is easy to ensure, by design, that for one of the extreme axial positions of the shaft 1, this sum f_2 of the dynamic and static thrusts acting on the disc 9 should have a greater value than the axial thrust f_1, whilst for the other extreme axial position of this shaft 1, this sum f_2 of the dynamic and static thrusts acting on the disc 9 should have a value smaller than the axial thrust f_1.

Thus, when starting the circulating pump, the shaft 1 automatically assumes an axial position for which $f_1=f_2$ and for which $p_1<p_3<p_2$.

Since it is possible to provide, as stated above, a very great variation of the value of the thrust f_2, it is possible by means of the described device, not only to compensate the axial component of the resultant of the thrusts exerted on the turbine's wheel, but also at least a part of the weight of the revolving part of the circulating pump (when the shaft 1 is not horizontal) and also at least a part of the axial magnetic pull exerted on the rotor of the electric motor driving the shaft 1.

It is possible to ensure that for one of the extreme axial positions of the shaft 1, the sleeve 17 should be situated entirely outside the boring 18. For this position of the shaft 1, the slot 21 has a very large passage section, and the pressure p_3 will be approximately equal to p_1. Moreover, it is also possible to ensure that, for the other extreme axial position of the shaft 1, the frontal face e of the sleeve should be situated very close to the bottom g of the boring 18. For this position of the shaft, the length of the slot 21 being very large and its passage section very small, the pressure p_3 will be approximately equal to the pressure p_2.

In designing, by choosing the dimensions of the various elements and organs of the compensating device, it is therefore possible to ensure that a comparatively small axial displacement of the shaft 1, for instance a few millimetres, causes a very great variation of the pressure p_3. This feature of the described device enables to make a circulating pump, the driving shaft of which places itself automatically in a steady axial equilibrium position. Moreover, thanks to the fact that the thrust f_2 can reach a high value, equal to, or greater than the sum of the hydraulic axial thrusts acting on the turbine's wheel plus the magnetic pull acting on the motor's rotor, it is possible to compensate fully the axial thrusts sustained by the shaft of a circulating pump. In a variation of execution of the described device, one could revert the turbine's revolution direction, the intake taking place through the centre as in the embodiment represented by the Figs. 1 and 2.

However, in this case, the disc 9 is no longer subjected to the dynamic thrust caused by the direction change of the liquid streams leaving the turbine's wheel. It will then be necessary to provide a disc 9 of larger diameter, in order to obtain a same compensating thrust f_2.

The leakage discharge through the slot 15 or 21 will be reduced as far as possible, in order to render practically nil this leakage discharge, so that the general efficiency of the circulating pump should not be influenced.

Two embodiments of the object of the invention have been described here by way of example and with reference to the attached schematic drawing, but it is obvious that numerous variations in design and execution can be provided without departing from the scope of the present invention.

One could, for instance, form in front of the face A of the disc a chamber connected to the intake scroll by a gauged passage, so that the pressure in this chamber increases in function of the leakage discharge. In this latter case, one obtains an effect similar to that obtained thanks to the gauge 16, so that the latter could possibly be suppressed.

In order to reckon with the differences which can take place in the dimensions of cast parts and to avoid a strict checking of these dimensions, and possible expensive machining, one could very well replace the gauge 16 by a duct, the passage section of which can be modified, chosen and fixed by means of a needle-screw, for instance, or by any other known and used adjustment device now in current use.

I claim:

1. In a construction for compensating automatically the thrust exerted on the impeller wheel shaft of a pump, the combination comprising a casing having an inlet duct and an outlet duct, a partition within the casing and defining a chamber communicating with the inlet duct and having two passages, one of the partition passages communicating with the outlet duct, an impeller wheel in the partition passage communicating with the outlet duct and having a shaft journaled in the casing with axial play, an extension of the impeller shaft disposed within the partition chamber, means carried by the shaft extension and slidable with the shaft in the other partition passage in spaced relation thereto to provide an annular variable leakage path con-

centric with the shaft axis from the inlet duct to the outlet duct, the means and the casing wall adjacent thereto defining a chamber connecting with the outlet duct, the leakage path being controlled by the position of the means within the other partition passage and thereby controlling the value of the thrust exerted by the liquid on the means and the shaft.

2. The combination according to claim 1 and wherein said means is a disc peripherally spaced from the other partition passage to provide an annular leakage path.

3. The combination according to claim 1 and also comprising variable gauge means in the connection between the chamber and outlet duct.

4. In a construction for compensating automatically the thrust exerted on the impeller shaft of a pump the combination comprising a casing having an inlet duct and an outlet duct, a partition defining a chamber within the casing and communicating with the inlet duct and having two passages in opposite relation, one of the passages communicating with the outlet, an impeller wheel in the partition passage communicating with the outlet duct and having a shaft journaled in the casing with axial play, an extension of the impeller shaft, a disc secured to the extension and slidable in the other partition passage and carrying a sleeve and defining with the other partition passage an annular variable leakage path concentric with the shaft axis from the inlet duct to the outlet duct, the disc and casing wall adjacent thereto defining a chamber connecting with the outlet duct, the disc having at least an opening connecting the partition chamber to the second chamber.

5. The combination according to claim 4 and wherein the casing wall confronting the disc has on its inner side a depression receiving the sleeve in one of its extreme positions.

6. The combination according to claim 4 and wherein the casing wall confronting the disc has on its inner side a depression receiving the sleeve in one of its extreme positions and having a bottom engaged thereby.

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