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CENTRIFUGAL PUMP.

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1,080,656.

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Fig. 1.

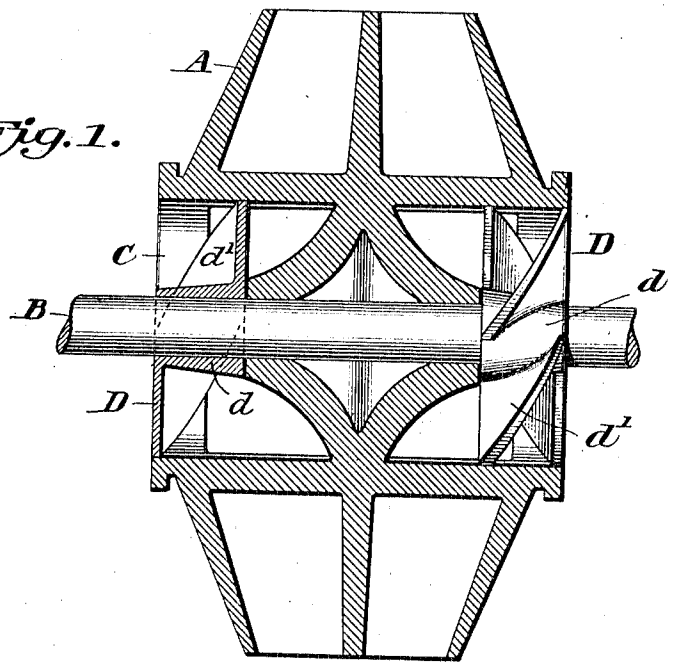
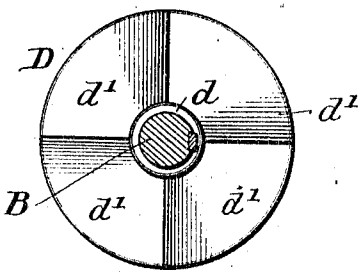


Fig. 2.



Witnesses:

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CENTRIFUGAL PUMP.

1,080,656. Specification of Letters Patent. Patented Dec. 9, 1913.

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To all whom it may concern:

Be it known that I, WILLIAM K. RICHARDSON, a citizen of the United States, residing at Leavenworth, in the county of Leavenworth, State of Kansas, have invented a new and useful Improvement in Centrifugal Pumps, of which the following is a specification.

It does not seem to have been understood by practical pump designers, that the pressure energy produced by an impeller of a centrifugal pump, is the product of the radial depth of the water subjected to centrifugal force, and its angular velocity. For this reason no provision is made in pumps now in general use for subjecting the water to centrifugal force within the area of cross section of the intake. Patents have been granted which show and describe means within the intake of a centrifugal pump for subjecting the fluid to centrifugal force, but I have found that unless the means employed be properly constructed no increased efficiency can be obtained by its employment.

The object of this invention is to utilize the area of cross section of the intake of an impeller of a centrifugal pump, whether single or multiple stage, by the employment of a screw in the eye of the runner. To accomplish this result, the diameter of the eye of the runner, its angular velocity, and the volume passing through it, must be known in order that the pitch of the screw and the number of blades constituting the screw may be determined, for it should be understood that there is a certain maximum pressure that may be generated and utilized due to any rotary velocity of the screw, delivering a predetermined quantity of fluid when properly designed. Otherwise the best results cannot be obtained, for if the pitch of the screw is too great, back pressure will be imposed upon the suction, and if the pitch be too fine, low efficiency will result.

To more particularly describe my invention, I will refer to the accompanying drawings, in which:

Figure 1 is a longitudinal section through a double suction impeller provided with a screw in the eyes of the impeller in accordance with my invention, the inclosing casing and associating parts of the pump being omitted. Fig. 2 is a plan view of one of the screws shown in Fig. 1.

As illustrated in Fig. 1, A indicates a well known form of impeller of the double suction type mounted on a shaft B to revolve therewith. Said impeller is provided on each side with an intake or eye C surrounding the shaft B. The particular construction of the impeller forms no part of my present invention, and need not be particularly described, as my invention is applicable to any form of impeller whether of the single or double suction type, which is adapted to subject water to centrifugal force.

In the eye of the impeller whether it be of the single or double suction type, is placed a screw D, said screw consisting of a central web *d* forming a sleeve of a diameter sufficient for the reception of the shaft B to which it is attached. To the sleeve are cast projecting helical blades *d'* which are inclined toward the impeller in a direction opposite to the direction of rotation. Said blades may overlap each other, but their combined surfaces must at least encompass the entire cross sectional area of the screw as shown in Fig. 2, to avoid eddying and escape of the fluid under pressure. The pitch of the screw should be so determined that twice the volume of fluid to be delivered will equal the calculated translatory displacement of the screw. The calculated translatory displacement, during one minute of a screw supposed to have blades whose ends are in radial lines, is the product of the number of R. P. M., the pitch of the screw, and the area in cross section of so much of the tubular space surrounding the shaft which is traversed by the blades in their revolution as is occupied at any instant by a figure or figures which do not overlap in the circumferential direction, containing the blade or blades, and are bounded by radial planes passing through the axis of revolution and the ends of the blade or blades. If a single blade, which at least once encircles the shaft be used, the pitch is the axial length of the blade, and the effective area in cross section is the area in cross section of the entire space traversed. If a number of equal blades be used, which have their corresponding ends in the same transverse plane, and which in the aggregate at least once encircle the shaft, the effective area in cross section is the area in cross section of the entire space traversed, and the length of the screw which is effective for one performance of the translatory func-

tion of each blade, neglecting overlapping portions, is the axial length of the blades. Neglecting slip, each blade performs its translatory function as many times during one revolution as there are non-overlapping blades, and that number multiplied by the axial length of the blades is equal to the pitch. Therefore, in any case in which the shaft is at least once encircled by the combined circumferential length of the blades, whatever their number, the calculated translatory displacement is the product of the area in cross section of the space traversed, the pitch and the number of R. P. M. A circumferential length of the blades in excess of that required to once encircle the shaft would not affect the calculated translatory displacement, but would affect the actual displacement by reducing slip in affording a greater area of surface to act upon the water.

The object of having a screw, the calculated translatory displacement of which approximately equals twice the volume delivered, is that a portion of the fluid may be longer retained within the rotary path of the screw, and thus be subjected to centrifugal force directly as the rotary velocity of the blades, and inversely as the slip. A screw having the above calculated translatory displacement, will produce a pressure due to centrifugal force and also a pressure due to impact of the fluid upon the blades, the combined pressures producing a spouting velocity toward the bit of the blades equal to the rotary velocity of the blades. Should the calculated translatory displacement materially exceed the volume stated, back pressure would be imposed upon the suction. While it may be shown that this is not the maximum centrifugal force that can be generated, it is a very close approach to the mean. For instance, the greatest centrifugal force would be when there is no delivery, and only kinetic energy due to centrifugal force of the screw is a factor. Here the centrifugal force is directly as the angular velocity of the blades, but there are other conditions to be considered. No centrifugal pump is designed to deliver any certain volume throughout the operation of the pump. This factor varies as the head and the lift of the suction, and it is also somewhat influenced by the speed. Therefore the screw should be designed to handle the maximum and a certain minimum volume throughout the general efficiency of the pump, without imposing back pressure upon the suction. In designing the screw the volume of fluid delivered by the pump while operating at its best efficiency is the rated capacity of the pump. Any axial length of a screw desired, its pitch being known, will determine the number of blades in the screw. The purpose of the screw is to produce centrifugal force of the water within the area of intake, or eye, of the runner, for forcing the fluid in the direction of flow, and for producing suctional effect.

The theory of operation is as follows: Due to rotary motion of the screw in the fluid, water is drawn into the space traversed by the screw, due to cavitation on the back side of the blades. The extent of cavitation depends upon the angular velocity of the blades, and the lift of the suction, atmospheric pressure being considered as 14.7 lbs. per square inch. The water is acted upon by the driving side of the blade, being carried forward and deflected axially, dependent upon the angular velocity of the blades, the inclination of their surface to the plane of rotation, and the resistance offered to axial flow. The screw, like the impeller, exerts a double energy, a pressure energy due to impact of the fluid upon the blades, some of which is converted into pressure, and kinetic energy in the form of centrifugal force of the fluid which is converted into useful pressure on delivery to the impeller. From the foregoing it will be seen that the combined surfaces of the blades of the screw should entirely encompass the screw and that their pitch will give a calculated translatory displacement in excess of the volume delivered. In computing the translatory displacement of a screw, the space occupied by the screw itself, should be taken into account. Proper allowance should also be made in designing a screw that is to receive fluid under either positive head or suction.

As showing one mode of computing the structure of the screw, the following is offered: Given a screw 10″ in diameter, desired length 4″, 600 revolutions per minute, delivery 3264 gallons per minute, space occupied by the screw being neglected. The superficial area of the cross section of a screw 10″ in diameter is 78.54 inches × 600 R. P. M.=47124 inches. Twice the volume 3264=6528 gallons × 231 inches (number of cubic inches in a gallon)=1507968 inches. This number divided by 47124=32 inches, which is the pitch. By dividing the pitch by the desired length we determine the number of blades which is 8. A greater number of blades may be employed in a screw of this pitch and length, but they will not increase the translatory displacement of the screw, as the overlapping portions of the blades will be a neutral zone. The lift of the suction, and high angular velocity of the blades, whether due to high speed of rotation, or large diameter of screw, will somewhat modify the pitch, for while high angular velocity will induce cavitation on the back of the blade and therefore suction, the high lift on the suction prevents rapid flow to the blades.

Having thus described my invention, what

I claim as new and desire to secure by Letters Patent, is—

1. In a centrifugal machine, the combination with a rotor adapted to subject fluid to pressure due to centrifugal force, of a plurality of blades in position to deliver fluid into the eye of said rotor and revolve therewith, said blades being inclined toward the rotor in a direction opposite to the direction of rotation, and having a calculated translatory displacement approximately twice the rated volume designed to pass the screw.

2. In a centrifugal machine, the combination with a rotor adapted to subject fluid to pressure due to centrifugal force, of a screw mounted on a shaft coaxial with said rotor and having blades in position to deflect fluid into the eye of said rotor and revolve therewith, the pitch of said screw being approximately such that twice the volume delivered past the screw according to the rated capacity of the machine is equal to the calculated translatory displacement of the screw.

3. In a centrifugal machine, the combination with a rotor adapted to subject fluid to pressure due to centrifugal force, of a screw mounted on a shaft coaxial with the said rotor and having blades in position to deflect fluid into the eye of said rotor, and revolve therewith, said blades having a total circumferential length such that they encircle the shaft, the pitch of said screw being approximately such that twice the volume delivered past the screw according to the rated capacity of the machine, is equal to the area of the cross section of the tubular space surrounding the shaft which is traversed by the blades in their revolution, multiplied by the pitch, multiplied by the number of revolutions.

4. In a centrifugal machine, the combination with a rotor adapted to subject fluid to pressure due to centrifugal force, of a screw in position to deliver fluid into the eye of said rotor and revolve therewith, the number of blades being equal to the pitch divided by the length of the screw, said blades having a total circumferential length such that they encircle the shaft, the pitch of said screw being approximately such that twice the volume delivered past the screw according to the rated capacity of the machine, is equal to the calculated translatory displacement of the screw.

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Witnesses:

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