ROTARY PUMP

Filed Aug. 6, 1968 Sheet 1 of 3

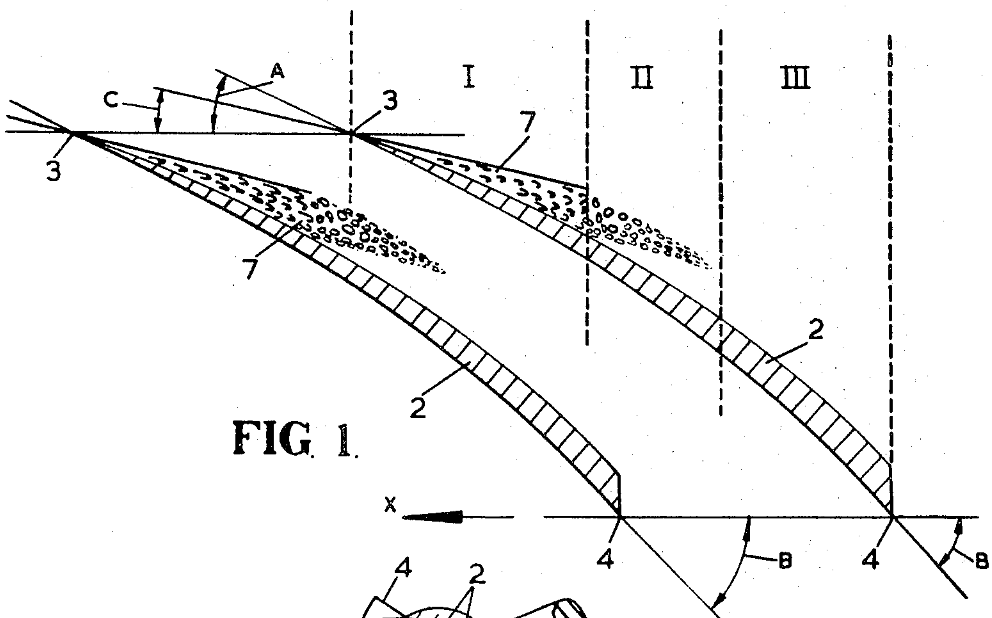

FIG. 1.

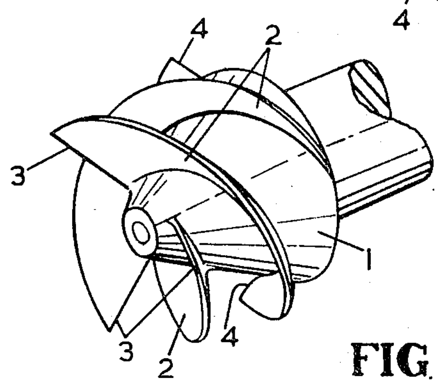

FIG. 2.

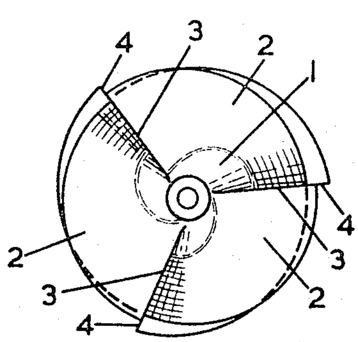

FIG. 3.

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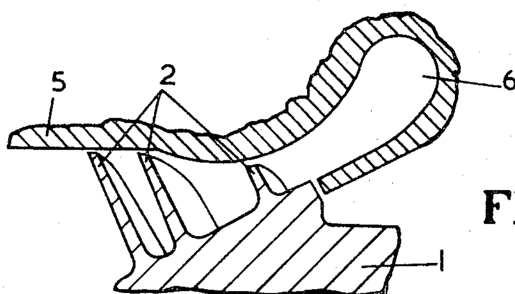

FIG. 4.

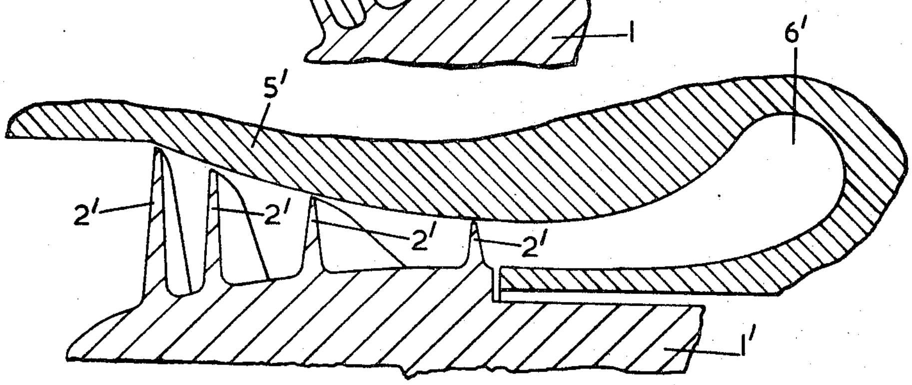

FIG. 5.

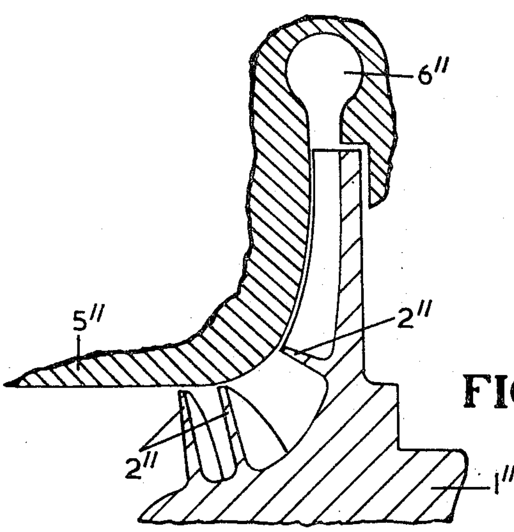

FIG. 6.

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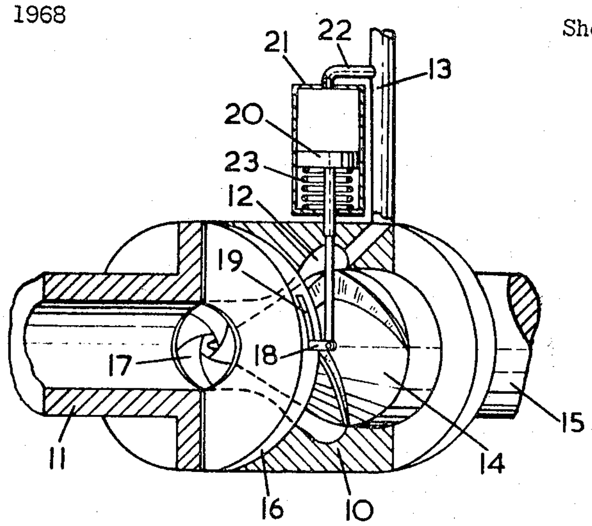

FIG. 7

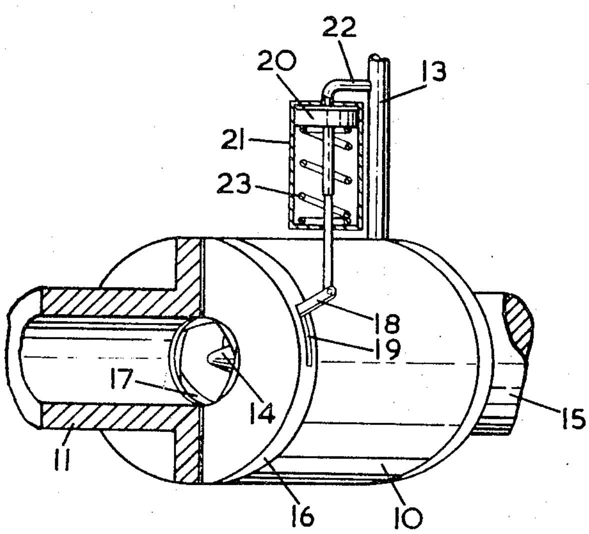

FIG. 8

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ROTARY PUMP

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U.S. Cl. 103—88 6 Claims

ABSTRACT OF THE DISCLOSURE

A rotary pump for liquids or liquids containing gas or vapour and capable of operating at a high rotational speed while pumping from a low suction head. The pump has an impeller having at least one cambered blade of screw-like configuration and having a thin leading edge of finely-tapered, wedge-like form extending throughout the whole effective span of the blade from the root to the tip thereof and a blade inlet angle of small magnitude, thereby to intentionally produce cavitation on the suction side of the blade over the whole of the effective span. The or each blade has a pitch which increases from the leading to the trailing edge thereof and a blade angle which increases progressively from the small inlet angle to a larger outlet angle, thereby to terminate the cavitation in a region between the inlet and outlet ends of the impeller.

The invention is a continuation-in-part of Ser. No. 557,217, filed June 13, 1966, now abandoned, and relates to a rotary pump for liquids or liquids containing gas hereinafter referred to as "liquids."

For each constant speed characteristic of known rotary pumps, there is a critical suction head below which the delivery head and the efficiency will fall very rapidly. The critical suction head increases as the rotational speed of the pump impeller increases and so hitherto it has not been possible to operate a pump at a high rotational speed where it has been necessary to pump from a low suction head. In such cases, where the pump impeller is to be driven by a turbine, it has been necessary to interpose a reduction gear between the turbine and the impeller or to use a large low speed turbine, thereby increasing the bulk and weight of the pump-turbine assembly. An object of the invention is to provide a rotary pump which can pump efficiently from a low suction head and be driven at high speed, e.g., by a turbine, without the need for a reduction gear.

Furthermore a high impeller speed in combination with a low suction head is a condition which results in the formation of cavities of gas or vapour at the inlet to the impeller. If these cavities should be allowed to form in a random manner, they would lead to obstruction of the flow of liquid through the impeller. A further object of the invention is to form said cavities in a controlled manner, thereby to produce as little disturbance to the flow of liquid as possible.

According to the invention, a rotary pump for liquids includes an impeller having at least one cambered blade thereon of screw-like configuration, said blade having a leading edge extending throughout the whole effective span of the blade from the root to the tip thereof, said leading edge being of finely-tapered, wedge-like form having an included angle of substantially 3° and thickness of between 0.5% and 2% of the blade height at inlet and said blade having a blade angle, defined between the direction of rotation at a point on said blade and the tangent to the pressure face of said blade at said point, said blade

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angle increasing from the leading to the trailing edge of said blade from a value of between 8° and 12° at inlet and a value at least 7° larger than the value at inlet at a position substantially mid-way in the axial length of the impeller, a fluid inlet angle, defined between the direction of rotation at a point on said leading edge and the direction of fluid flow relative to said blade of between 1° and 5°, a hub to tip ratio at the inlet to said impeller of between 0.2 and 0.3 and a mean blade diameter at outlet at least equal to that at the inlet to the impeller and the impeller having a meridional flow area at inlet of between 30% and 50% larger than that at the outlet thereof.

It has been found that a pump including an impeller having a blade geometry as set out in the immediately preceding paragraph, when operated under conditions involving a cavitation number K of between 0 and 0.04, where

$$K=\frac{P_{stat}-Pv}{\rho\frac{V_r^2}{2g}}$$

and where

P_{stat}=Static pressure in the pump inlet passage
P_v=Vapour pressure of the liquid being pumped
ρ=Density of liquid, and
V_r=Velocity of liquid relative to the blade at the tip thereof at the impeller inlet

will operate under conditions of controlled cavitation.

During such operation, the following regions will be defined within the flow passage through the impeller:

(I) an inlet region in which cavitation is developed on the suction side of the blading of the impeller over the whole effective span of said blading in a direction from root to tip thereof,

(II) an intermediate region in which the cavitation is terminated, and

(III) a delivery region in which the flow passage is filled by the liquid being pumped and work is effected thereon.

By producing cavitation in region I as aforesaid, the full area of flow through the impeller at the inlet region thereof is reduced to a smaller effective value sufficient to enable an adequate flow of liquid into the impeller. In other words, the inlet area of the impeller is sufficient to accommodate the required liquid flow plus the volume occupied by the cavity produced by the or each blade. This enables pumping to take place under such conditions of high rotational speed and low suction head that would not have been possible with pumps designed to operate with little or no cavitation.

In region II the cavity produced by the or each blade is terminated by causing it to collapse in the flow passage through the impeller away from the blade or other solid surfaces, as will be explained hereinafter, thereby reducing erosion damage.

In region III, after the cavitation has been terminated, the impeller will run full and work will be effected on the liquid being pumped.

The pump in accordance with this invention tends to be self-adjusting thereby to accommodate changes in suction or delivery head, within the design range of the pump, by movement of the said intermediate region II towards one or other end of the impeller.

The blade may be formed on hub portions of various shapes. For example, the hub portion may be conical having its apex at the inlet end of the impeller and may carry a single blade or several blades intertwined around it in the manner of a multi-start screw thread.

Impellers having screw-like cambered blades in accordance with the invention and pumps including said impellers are shown by way of example in the accompanying drawings, in which:

FIGURE 1 shows a diagrammatic development of a cascade formed by two adjacent blades of a multi-bladed impeller or by opposite flanks of the same blade in two positions spaced apart by one revolution, in an impeller having only one blade;

FIGURE 2 is a perspective view looking from the inlet end of a first impeller;

FIGURE 3 is a view of the inlet end of the first impeller;

FIGURE 4 is a half axial section of the first impeller shown in a pump housing;

FIGURE 5 is a half axial section similar to FIGURE 4 of a second impeller shown in a pump housing;

FIGURE 6 is a half axial section similar to FIGURES 4 and 5 of a third impeller shown in a pump housing;

FIGURE 7 is a diagrammatic part-sectional perspective view of a pump including an impeller of the kind shown in FIGURES 2, 3 and 4 and having an iris upstream of the pump inlet for effecting partial admission thereto, the iris being shown in one extreme position in which the effective inlet to the impeller has a minimum area, and

FIGURE 8 is a view similar to FIGURE 7 in which the iris is shown in the other extreme position in which the effective inlet to the impeller has a maximum area.

Referring to FIGURES 2, 3 and 4, the first impeller comprises a conical hub **1** having screw-like blades **2** extending therefrom. On the impeller illustrated, there are three blades arranged in the manner of the threads of a multi-start screw. Each blade **2** has a thin finely-tapered, wedge-like leading edge **3** having an included angle of substantially 3° and a thickness within the range of 0.5% to 2% of the blade height and/or span at inlet. The edge may be a sharp knife-edge or it may be rounded by a convex curve of small radius. These forms of leading edge are to promote clean separation of the liquid from the flank of the blade at the suction side thereof to produce the region of cavitation, as will be described hereinafter with particular reference to FIGURE 1, and to present minimum cross-sectional area in the direction of flow, thereby to avoid undue interference by the cavitation with the flow of liquid into the impeller. The leading edge plan-form of each blade may be straight and extend radially from the axis of the impeller, or be inclined rearwardly or forwardly, or be curved. The leading edge also has a small blade inlet angle A defined between the direction of rotation of any point on the leading edge and the tangent to the pressure face of the blade, of between 8° and 12°, also for the purpose of intentionally producing a region of cavitation, as will be explained hereinafter with particular reference to FIGURE 1. Each blade **2** has an outlet edge **4**.

In FIGURE 1, the blade inlet angle A, as defined hereinbefore, is the angle at the inlet edge **3** between the tangent to the pressure surface, i.e., the lower surface of a blade in FIGURE 1, and the direction of motion X of the blades, i.e. the direction of rotation of the impeller. The blade outlet angle B is the angle at the outlet edge **4** between the tangent to the working surface and the direction of motion X of the blades. The angle B is greater than the angle A by an amount such that the blade angle at a position substantially mid-way in the axial length of the impeller is at least 7° larger than the blade inlet angle A. Also the fluid angle at inlet, the angle C, defined between the direction of motion X, i.e., the direction of rotation, and the direction of fluid flow relative to said blade is between 1° and 5°.

The flow passage between two adjacent blades or blade positions, of the impeller defines three regions during operation. The regions are indicated in FIGURE 1 with respect to one of the blades shown therein and are as follows:

(I) the inlet region in which cavitation is developed on the suction side (i.e. the upper surface in FIGURE 1) of the blades over the whole span thereof, i.e., from root to tip;

(II) the intermediate region in which the cavitation is terminated, and

(III) the delivery region in which the flow passage is filled by the liquid being pumped, as in a normal pump.

Although upright broken lines indicate the end of one region and the beginning of the next region, it should be understood that the relative lengths of the regions may vary considerably from those shown, particularly as it is a feature that the position of the region of termination of the cavitation shall be variable by self-adjustment in the length of the impeller, thereby to accommodate changes in suction or delivery head.

For full span cavitation in the inlet region I the blade leading edge must be of finely-tapered, wedge-like form as aforesaid, the values of angles A and C must be within the aforesaid limits and the value of the cavitation number K as defined hereinbefore, must be between 0 and 0.04 and the hub to tip ratio at inlet must be between 0.2 and 0.3.

The following features are necessary for the collapse, i.e., termination, of the cavity produced by the or each blade, within the region II:

Area ratio

The ratio of the areas at the inlet and outlet of the impeller taken perpendicularly to the meridional flow line through the impeller (that is the meridional flow area) must be such that the inlet area is between 30% and 50% larger than the outlet area. The latter area is as in conventional pumps, since the outlet region III of the impeller runs full of liquid.

Mean blade diameter

In addition to the required area ratio, the mean blade diameter at outlet must be at least equal to that at inlet and is preferably of the order of twice the mean diameter at inlet.

Blade angle

In order to achieve the correct conditions for the collapse of the cavity, the blade angle of the or each blade should become greater towards the outlet such that its value at substantially mid-way in the axial length of the impeller is at least 7° larger than that at inlet, the value at outlet therefore being at least 7° larger than that at inlet.

The cavities produced in region I are indicated by bubbles 7 in FIGURE 1. The cavities are believed to collapse in the region II in the inter-blade spaces away from the blade or other solid surfaces due to the combined effect of centrifugal force and free surface flow and thus erosion damage normally produced by cavitation is avoided or reduced.

As will be seen in FIGURE 4, the impeller is enclosed by a housing **5** having a volute chamber **6** in which the kinetic energy at the outlet is diffused as the liquid is delivered to a delivery pipe, not shown.

A pump having the impeller shown in FIGURES 2, 3 and 4 is suitable, for example, as a high speed feed pump for a propellent in a rocket engine, as a fuel pump in a gas turbine engine or as a boiler feed pump.

The pump shown in FIGURE 5 has an impeller of which blades **2'** are formed on a hub **1'** of which the diameter increases slightly from inlet to outlet or is substantially cylindrical. The blades **2'** are similar to the blades **2** of the first example. The impeller is enclosed by a housing **5'** having a delivery volute chamber **6'**. The pump is intended to give a lower head rise than the pump shown in FIGURE 4 and is suitable, for example, for use as a boost or transfer pump in many applications.

The pump shown in FIGURE 6 has an impeller of which the blades are formed on a hub **1''** in the manner of the blades of a centrifugal pump. The blades 2'' are similar to the blades 2 of the first example. The impeller is enclosed by a housing 5'' having a delivery volute chamber 6''. The pump of FIGURE 6 is intended to give a higher head rise than the pump of FIGURE 4 and is suitable, for example, for the pumping of liquefied gases, particularly hydrogen for rocket engines.

Since in the cavitation region I, the flow passage between adjacent blades is not completely filled with liquid, any of the foregoing pumps can accept substantial quantities, e.g., of the order of 30% by volume, of vapour or gas without seriously affecting its hydraulic operation. Such vapour or gas may be separated from solution in the liquid by turbulence upstream of the pump inlet and therefore a pump in accordance with this invention is suitable for operation under such conditions. In liquefied gases, there may be bubbles of gas present and therefore the pump provided by this invention is suitable for pumping liquefied gases.

Where liquid is admitted over the whole span of the blading at inlet, a pump provided by this invention will be passing its maximum flow, this being determined by the reduction of the effective inlet area by the cavitation. Additionally, the effective span of the blading at inlet may be controlled by means for producing partial admission of fluid to the impeller. By producing partial admission and keeping the net positive suction pressure constant, the flow rate will be varied. Alternatively, by producing partial admission and varying the net positive suction pressure appropriately, the flow may be maintained constant.

As a result of producing partial admission and of the self-adjusting ability of the impeller, as described hereinbefore with regard to the relative axial lengths of the regions I, II and III, to accommodate changes in suction or delivery head, the power required to drive the impeller under conditions of controlled reduced flow, is reduced, thus maintaining the pump efficiency. This aspect is of particular benefit where it is of importance to minimise the temperature rise of liquid passing through the pump.

A suitable way in which partial admission can be effected is to provide an iris immediately upstream of the inlet of the impeller, thereby to reduce the effective tip diameter or to increase the effective root diameter of the blading at inlet. FIGURES 7 and 8 illustrate diagrammatically a pump having an impeller in accordance with this invention and an iris controlled in accordance with the delivery pressure of the pump. In FIGURES 7 and 8, the pump comprises a cylindrical housing **10**, having an inlet pipe **11** and a delivery volute chamber **12** leading to a delivery pipe **13**. The housing **10** contains an impeller **14** similar to that shown in FIGURES 2 to 4 and mounted on a shaft **15**. Immediately upstream of the impeller inlet there is an iris device **16** comprising overlapping obturating shutters **17** movable between the two closed and open limiting positions shown in FIGURES 7 and 8 respectively by an operating lever **18** movable in a peripheral slot **19** in the device **16**. The lever **18** is moved by a piston **20** movable in a cylinder **21** by delivery pressure applied through a branch pipe **22** leading from the delivery pipe **13**. The delivery pressure is opposed by a return spring **23**. Instead of the piston **20**, cylinder **21** and spring **23** another fluid-responsive device, such as a bellows, may be employed

to operate the iris device in response to variation of pump delivery pressure.

What we claim as our invention and desire to secure by Letters Patent of the United States is:

1. A rotary pump for liquids or liquids containing gas including an impeller having at least one cambered blade thereon of screw-like configuration, wherein the improvement comprises, said blade has a leading edge extending throughout the whole effective span of the blade from the root to the tip thereof, said leading edge being of finely-tapered, wedge-like form having an included angle of substantially 3° and thickness of between 0.5% and 2% of the blade height at inlet and said blade has a blade angle, defined between the direction of rotation at a point on said blade and the tangent to the pressure face of said blade at said point, said blade angle increasing from the leading to the trailing edge of said blade from a value of between 8° and 12° at inlet and a value at least 7° larger than the value at inlet at a position substantially mid-way in the axial length of the impeller, a fluid inlet angle, defined between the direction of rotation at a point on said leading edge and the direction of fluid flow relative to said blade of between 1° and 5°, a hub to tip ratio at the inlet to said impeller of between 0.2 and 0.3 and a mean blade diameter at outlet at least equal to that at the inlet to the impeller and the impeller has a meriodional flow area at inlet of between 30% and 50% larger than that at the outlet thereof.

2. A pump as claimed in claim **1** in which the impeller has a conical hub having its apex at the inlet end of the impeller and having at least two screw-like cambered blades intertwined around the hub in the manner of a multi-start screw thread.

3. A pump as claimed in claim **1** in which said blade extends substantially perpendicularly to the surface of the impeller hub.

4. A pump as claimed in claim **1** having variable admission means positioned immediately upstream of the impeller.

5. A pump as claimed in claim **4** in which the variable admission means is an iris device operable to vary the effective area of admission immediately upstream of the impeller.

6. A pump as claimed in claim **5** in which said iris device is controllable by means responsive to the pressure of fluid delivered by the pump.

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U.S. Cl. X.R.

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