

J. L. COKER, JR.
CENTRIFUGAL PUMP.
APPLICATION FILED DEC. 9, 1910.

998,751.

Patented July 25, 1911.

2 SHEETS—SHEET 1.

Fig. 1.

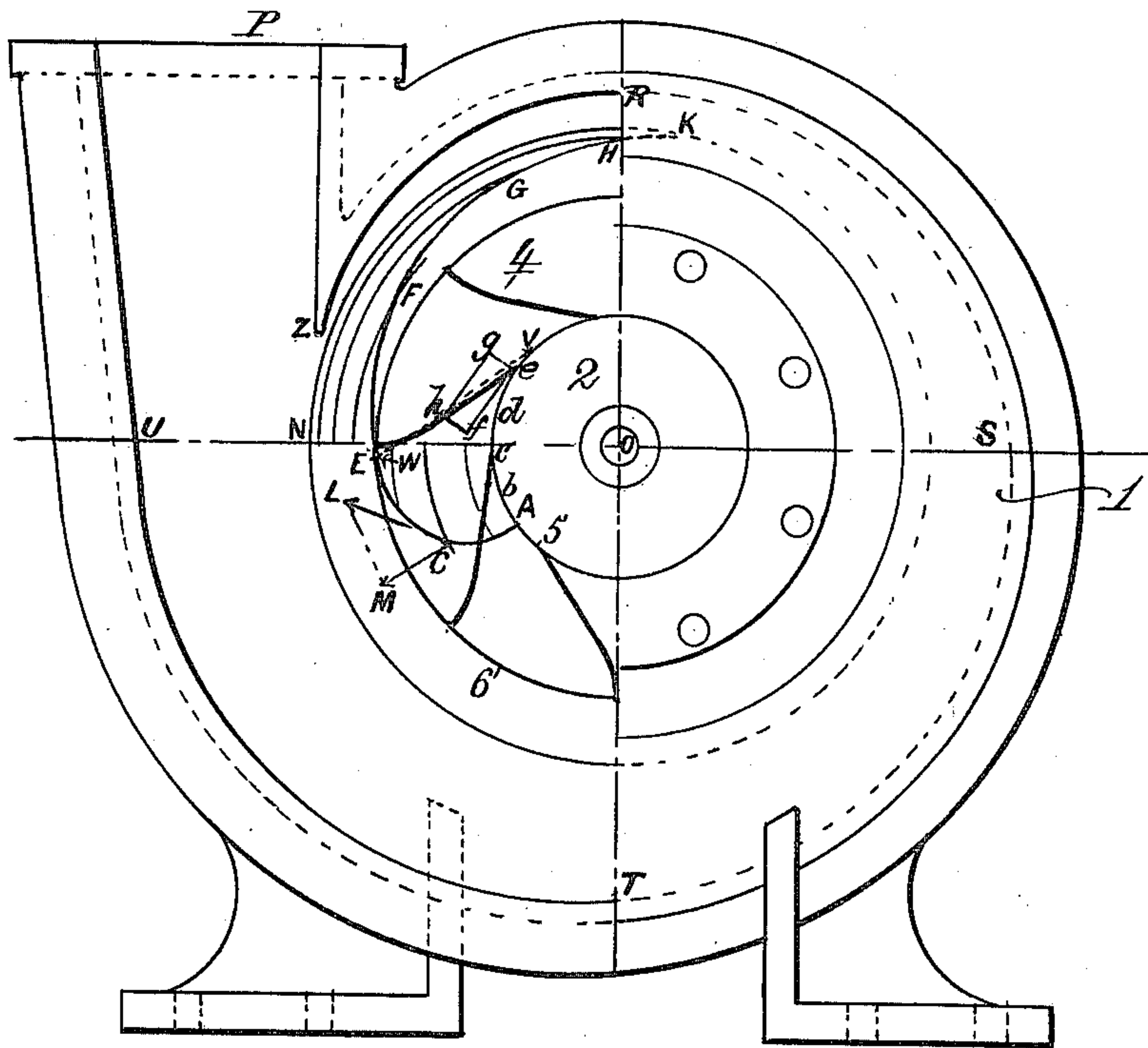
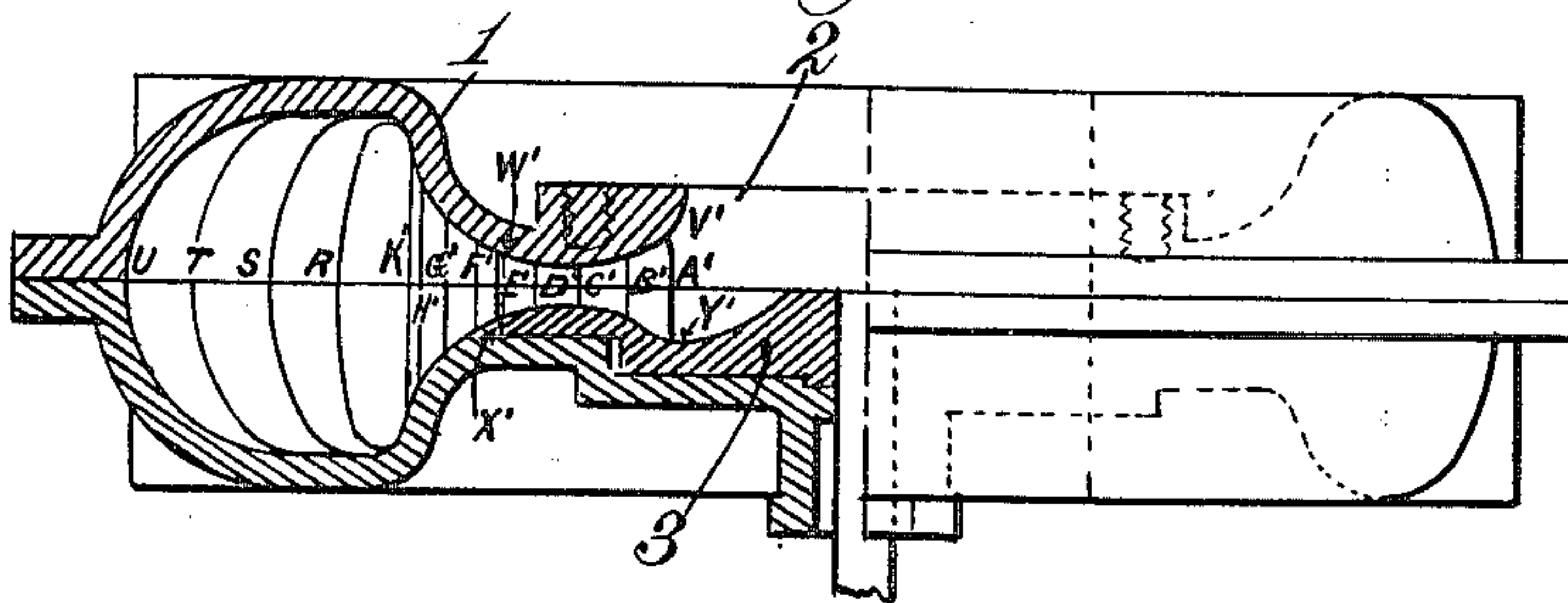


Fig. 2.



Witnesses:
Robert Goult.
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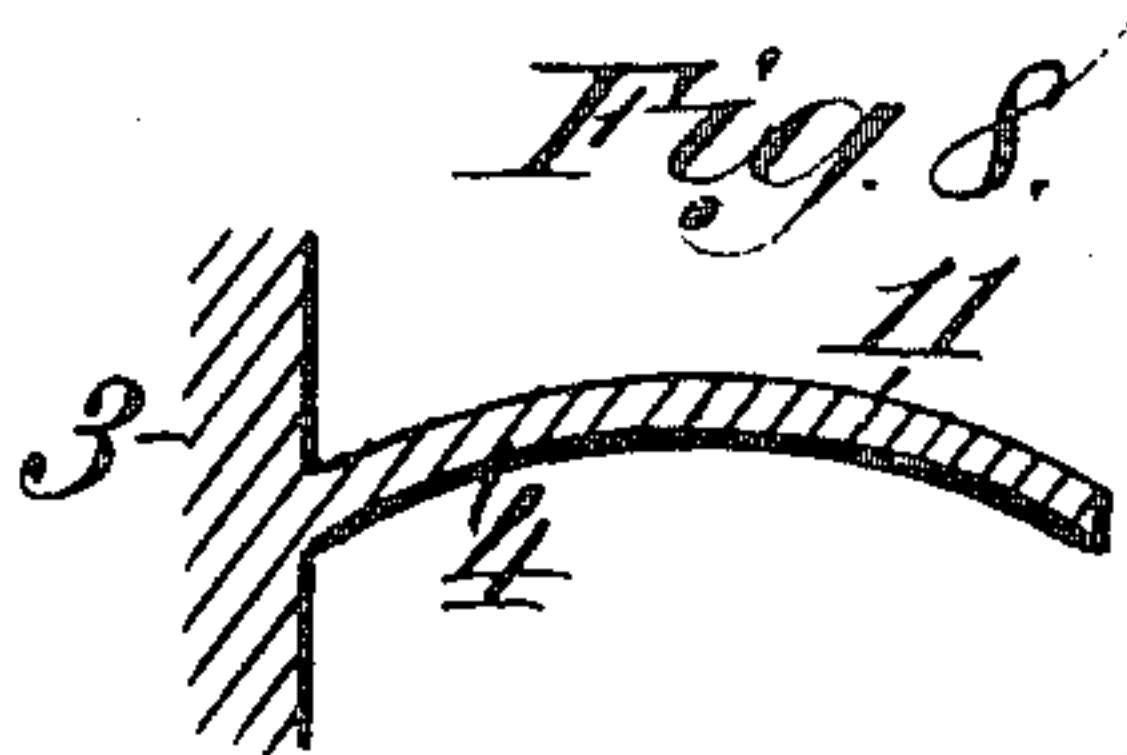
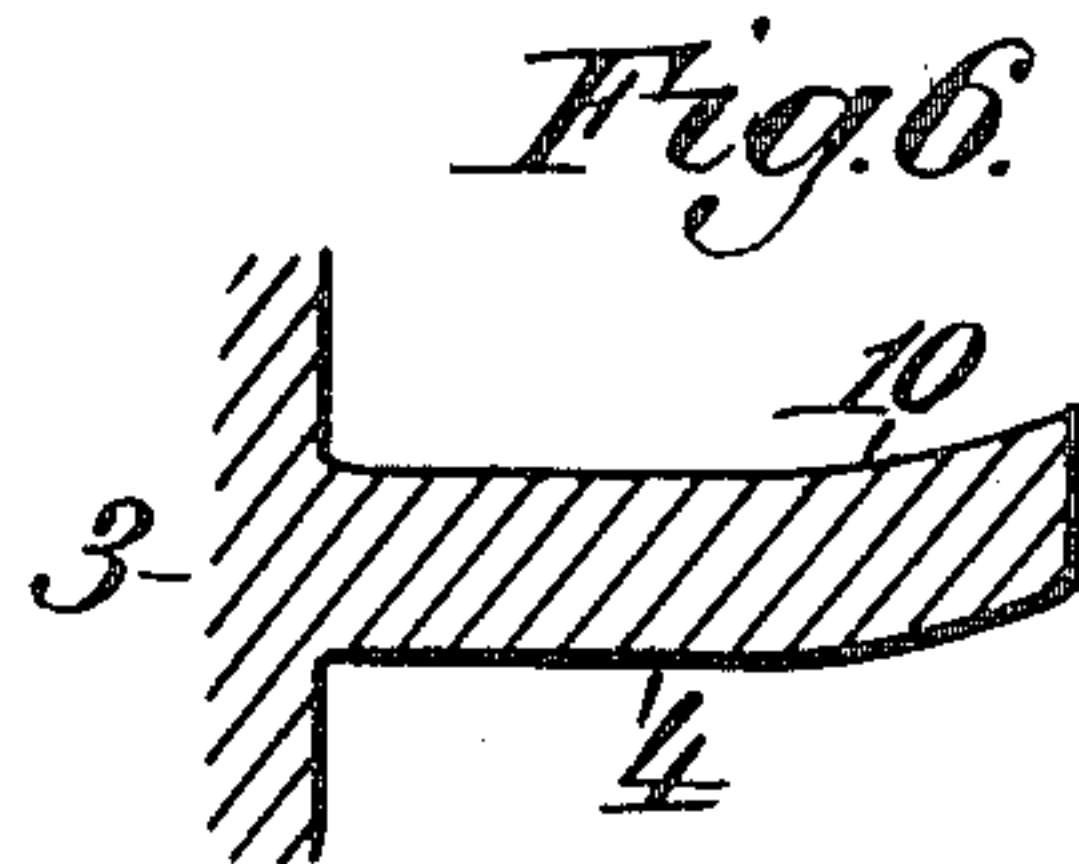
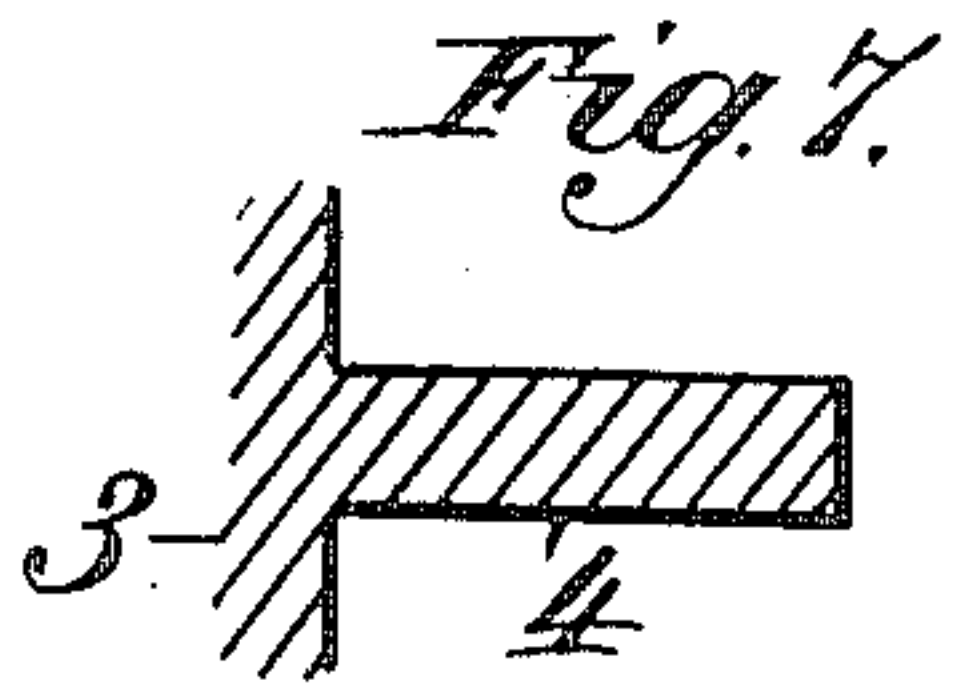
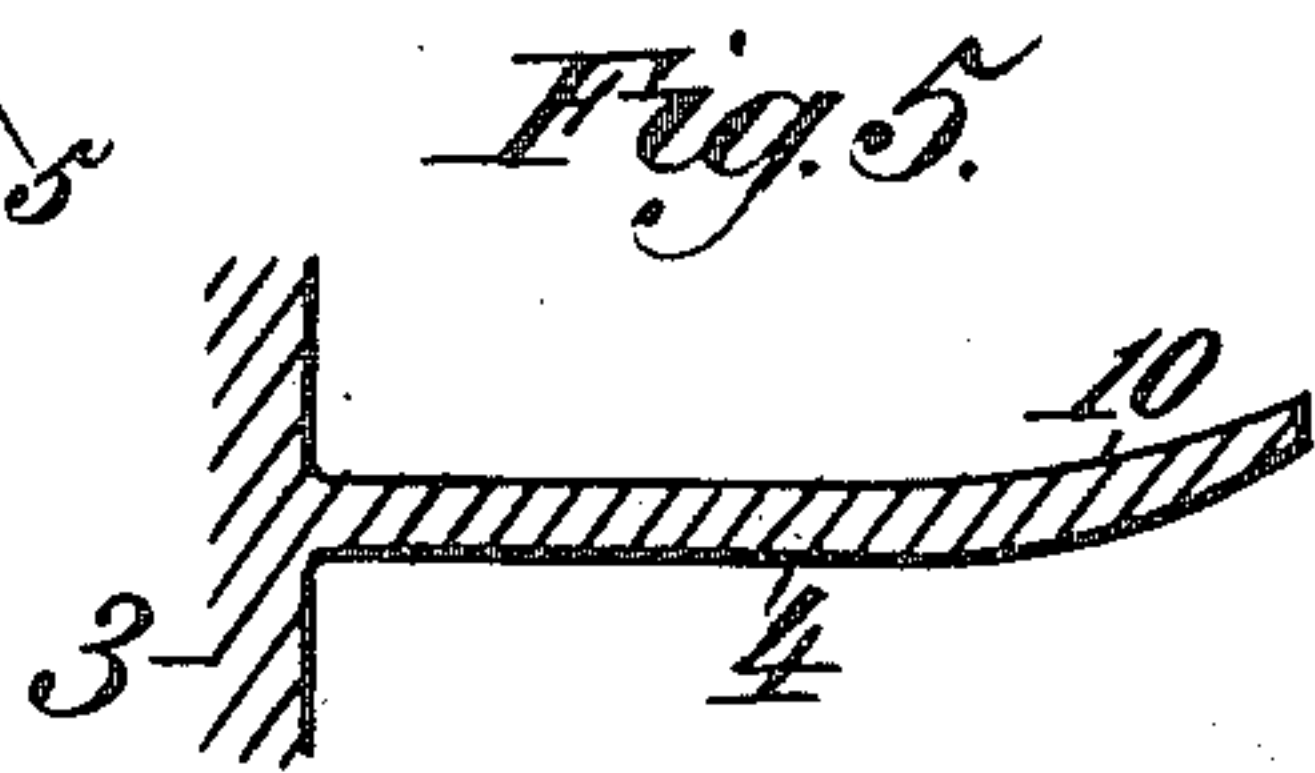
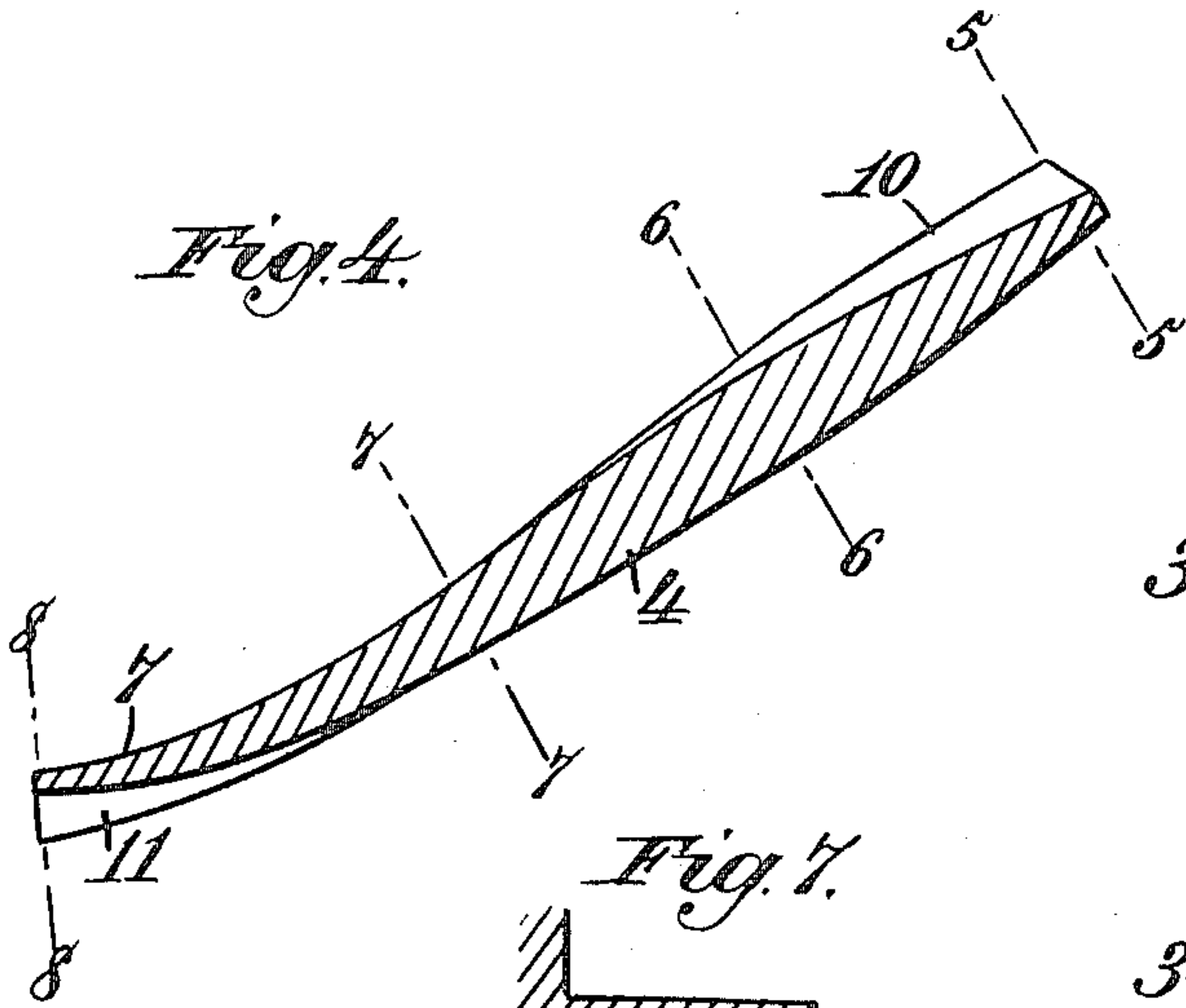
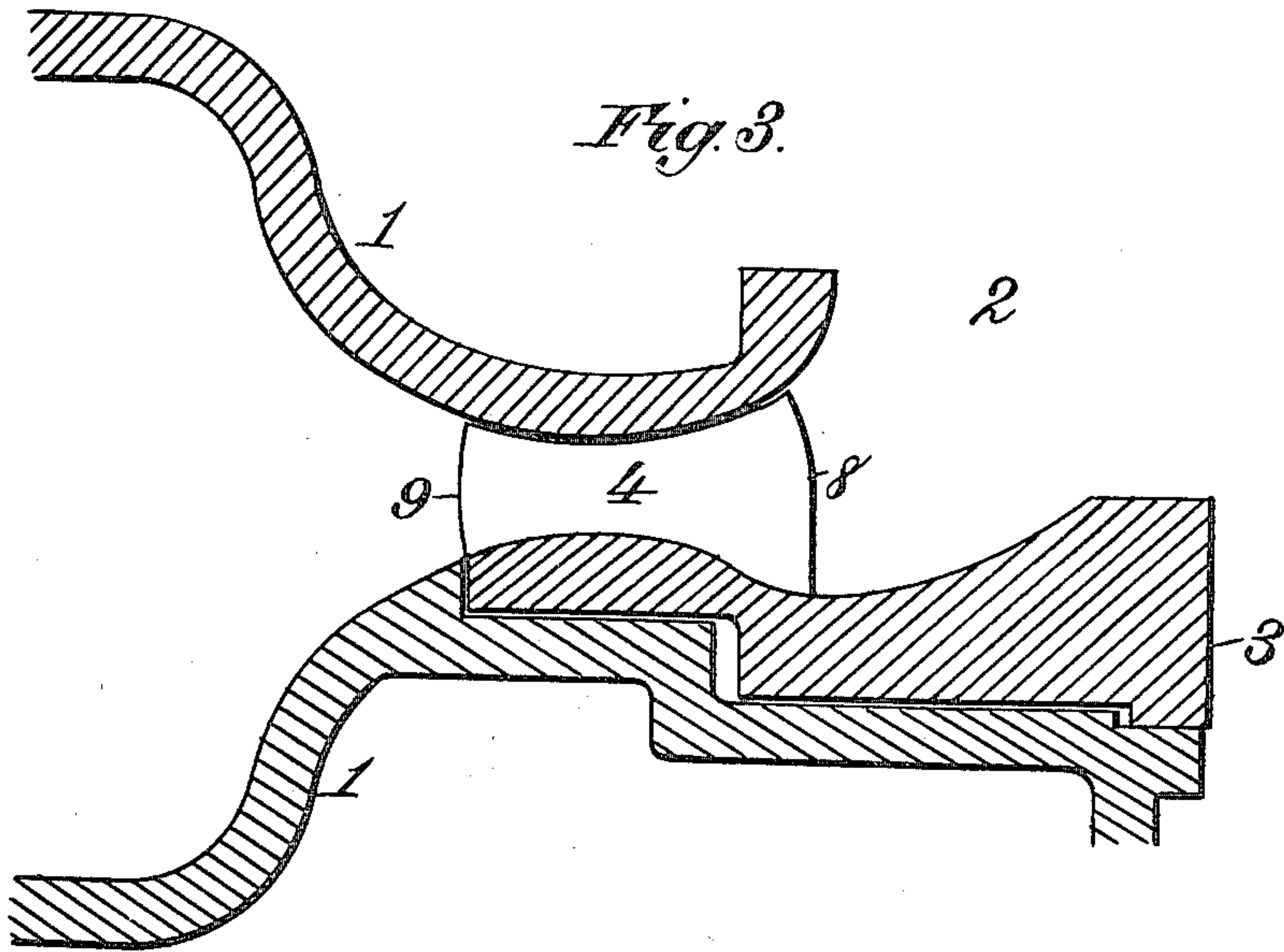
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J. L. COKER, JR.
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2 SHEETS—SHEET 2.



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UNITED STATES PATENT OFFICE.

JAMES L. COKER, JR., OF HARTSVILLE, SOUTH CAROLINA.

CENTRIFUGAL PUMP.

998,751.

Specification of Letters Patent.

Patented July 25, 1911.

Application filed December 9, 1910 Serial No. 596,515.

To all whom it may concern:

Be it known that I, JAMES L. COKER, JR., a citizen of the United States, and a resident of Hartsville, in the county of Darlington and State of South Carolina, have invented certain new and useful Improvements in Centrifugal Pumps, of which the following is a specification.

This invention is designed to produce a centrifugal pump of high efficiency, with which end in view its primary object is to obtain uniform acceleration of velocity of the intake water along the shortest and smoothest practical path between the inner and outer circles which bound the annular runway wherein the impeller or runner blades are included.

My invention resides in the means hereinafter set forth whereby this object is accomplished, and refers particularly to the shape and configuration of the runner blades and the annular runway in which said blades move.

I shall first describe in connection with the accompanying drawings the best way now known to me of carrying my improvements into practical effect, and will then point out more particularly in the claims those features which I believe to be new and of my own invention.

In said drawings—Figure 1 is a side elevation of a centrifugal pump embodying my invention, with a portion of the case removed to enable me to better illustrate the manner of laying out the runner blades. Fig. 2 is a top plan, half in axial section. These two figures are largely diagrammatic, and are intended more particularly to indicate graphically the method of designing those portions of the pump to which my invention relates. Fig. 3 is an axial section on a larger scale of so much of a pump as will show a cross section of the annular runway, together with a view of one of the runner blades therein. Fig. 4 is a longitudinal section of one of the blades, this section being on the median line of the blade. Figs. 5, 6, 7, 8, are cross sections of the blade on lines 5—5, 6—6, 7—7, and 8—8, respectively of Fig. 4. In each of these figures is also shown a portion of the runner to which the blade at one edge is attached.

The problem is to increase the velocity of the intake water by uniform acceleration along the shortest and smoothest practical path from the inner to the outer circle of

the impeller blades. This path may be normal to, or in some instances—as for example in high lift or high revolution pumps—make an angle with, the inner circle of the blades, but with the outer circle of the blades it should make the least possible angle. The acquired velocity of the water in this path, when it reaches the outer circle of the blades, should equal that due to the total head against which the pump is to work. After passing the outer circle of the blades the water should still follow a smooth, continuous path with uniform retardation to a velocity corresponding to that of the discharge pipe.

The single-side intake pump, because of its lower cost of construction, is to be preferred where the degree of pull or suction necessary to bring the water to the pump is not large, and also where the head against which the pump operates is small. Such a pump is illustrated in the drawings; wherein, so far as the general construction is concerned—referring more particularly to Figs. 1 and 2—it is sufficient to say that, 1 is the scroll pump casing having intake 2 and discharge pipe P; 3 is the runner disk mounted as usual upon a rotary power driven shaft passing through a stuffing box in one head of the pump casing; 4 are the impeller blades; and 5 and 6 are the inner and outer circles respectively of said blades.

The flow of water along the paths and with the velocities above referred to is secured by maintaining suitable relations between the form of pump casing and blade and speed of runner, the method pursued being as follows:

I. Knowing the quantity of water to be handled and the head against which it is to be lifted, the first step in the designing will be to determine the sizes of the intake and discharge pipes. Where economy of operation is the main consideration the velocity of flow of water in the intake pipe should not exceed substantially 6 feet per second for ordinary suction lengths, and the velocity in the discharge pipe should range between 7 feet and 12 feet per second. The diameters for these pipes will be determined by the equation

$$D = 2r = 2\sqrt{\frac{Q \times 231}{60 \times 12 \times V \times \pi}} \quad (\text{Eq. 1})$$

where Q is quantity in gallons per minute and V is given a value close to the limits

above suggested, and which will give a value to D corresponding to commercial size of the pipe.

In the particular pump illustrated in the diagram, it is planned to discharge 1,000 gallons per minute against a 9 foot head, the velocity of flow of the water at the intake being approximately 6.4 feet per second. Therefore substituting these values for Q and V in Eq. 1, and solving—

$$D = 2r = 2\sqrt{\frac{1,000 \times 231}{60 \times 12 \times 6.4 \times 3.14}} = 8 +$$

we obtain a value of 8 inches for diameter of intake pipe. So, also, we get a value of 7 inches for diameter of discharge pipe by giving V , at that point, a value of approximately 8.3 feet per second in the same equation.

II. In determining the size of the runner, having reference more particularly to the diameters of the intake and outer circles 5 and 6, between which the impeller blades are included, the source of motive power operating the pump must be considered; for the outside diameter will vary inversely with the speed of rotation of the runner, and directly with the quantity of water to be handled. Where practical, the tangential velocity of the outer edge of the blade should conform to that due to the head against which the pump is to operate. From one or more tentative layouts, depending upon the foregoing considerations, upon a path of acceleration (shown at A—E, Fig. 1) assumed on the principle hereinbefore referred to, and upon the diameter of the intake pipe, definite values may be determined for the inner and outer circles. For example, in the pump illustrated in the diagram the head against which it is to operate is 9 feet.

From any manual of engineering we find $V^2 = 2gh$, where V is the velocity acquired by a body falling from a height h , g is velocity due to gravity acting for one second, and h is height, or, in hydraulics, head. Substituting 9 for h , and solving, we have

$$V = \sqrt{64 \times 9} = 24 \text{ (feet per second)}$$

approximately, which is the proper velocity for the outer tip of blade.

Assuming the pump is to be belt driven, a convenient size of runner will be one having an outer blade circle, 6, of 15.4 inches diameter or about 48 inches in circumference, thus requiring a speed of rotation for the runner of 360 revolutions per minute to accomplish the required tangential speed at 24 feet per second of the outer edges or tips of the blades. The diameter of the intake circle 5, under these conditions, may be efficiently taken as 8.5 inches, this figure being controlled by the diameter of the intake pipe in connection with the curve of entrance at V' , Fig. 2, which should be tangent to the

side of the intake pipe, and also to the curve of the annular space of the case wherein the impeller blades operate. The circles 5 and 6 bound the throat or runway in which the impeller blades move.

III. The circumferences of these circles being known, together with the radial velocity and quantity passing through the pump, the widths of said throat or runway on the circles 5 and 6 are obtained by the equation—

$$x = \frac{Q \times 231}{60 \times 12 \times V_R \times C} \quad (\text{Eq. 2})$$

in which x = width of throat in inches; Q = quantity of water in gallons per minute; V_R = radial component of velocity in feet per second; C = circumference in inches of that particular blade circle under consideration. The width of the throat or runway at these circles is indicated in Fig. 2 by the ordinates $V^1—Y^1$ and $W^1—X^1$, respectively. Applying this equation (Eq. 2) to the pump illustrated in Fig. 2, and assuming frictional losses in the intake pipe will result in the water arriving at the intake circle 5 at an actual velocity of 6 feet per second along a path normal to that circle and in planes perpendicular to the axis of rotation of the runner, then, substituting known values in Eq. 2, and solving, we get—

$$x = \frac{1,000 \times 231}{60 \times 12 \times 6 \times 26.7} = 2'' \text{ approximately,}$$

as the width $V^1—Y^1$, of the throat and impeller blade at the intake circle; and similarly—

$$x = \frac{1,000 \times 231}{60 \times 12 \times 5.6 \times 48} = 1.18'',$$

as the width $W^1—X^1$, of throat and impeller blade at the outer or discharge circle 6. The length of the ordinates at other sections of the throat, as at B^1 , C^1 , E^1 , Fig. 2, depends upon the acceleration curve A—E, (Fig. 1); and of the ordinates beyond the outer circle 6, as at F^1 , G^1 , H^1 , K^1 , upon the retardation curve E—K (Fig. 1)—the latter being a continuation of the former. These two curves are determined in the tentative lay out, and may be arcs of mathematical spirals, or a series of arcs of circles smoothly joined, the curves themselves being governed at their extremities by the consideration that they must be smooth and continuous at E where they join; also that the acceleration curve should be normal to the intake circle, but should make the smallest possible angle with the outer circle; and that the retardation curve must discharge at such an angle that the radial component of velocity of the water will give practical and suitable dimensions for the interior of the pump casing on the circular section at N K, Fig. 1. It may be here remarked, that the

circle in Fig. 1 on which N and K are located is diagrammatic and representative of the outer boundary of the annular space between E and N, Fig. 1, corresponding to the 5 retarding section between E¹ and K¹, Fig. 2.

In the design illustrated in Figs. 1 and 2, the acceleration curve makes with the outer circle 6 an angle whose natural tangent is

$$\frac{5.6}{24},$$

which is the smallest practical angle for this particular design. Having settled upon the acceleration and retardation curves, and ascertained their developed lengths by usual methods of measurement, the accelerating or retarding velocity of the water may be found at any or all points on the curves. Knowing the tangential velocity of water along the path, as C L, at C (Fig. 1), the radial component of velocity C M may be found, and the width of throat corresponding to the ordinate at C¹, Fig. 2, determined for the section through C by the use of Eq. 2, as above. In a similar manner, as many points as desired may be found from A to K. To obtain the various points on the curves for computing these sections, it is convenient to divide the curves on a time basis, it requiring the same length of time for a particle of water to flow from A to B as from B to C, and so on. To obtain these division points, use is made of the formulas—

$$V_E^2 = 2aS + V_A^2 \quad (\text{Eq. 3})$$

$$V_E = at + V_A \quad (\text{Eq. 4})$$

in which— V_E =velocity of water along path at E; V_A =velocity of water along path at A; a =acceleration in feet per second, per second; S =length of path A E in feet; t =time in seconds required for water to pass from A to E.

In Equation 3, all quantities are known except a . Its value is found and substituted in Equation 4, thus giving the actual time for a particle of water to pass from A to E, while traveling in the mid-plane A¹—K¹, Fig. 2. There would naturally be slight variations for particles moving along the curved walls V¹—W¹ and Y¹—X¹, of the throat or runway on either side of the mid plane, which variations may be compensated for as indicated later.

The total time required for water to pass from A to E may be divided, as hereinbefore indicated, into a convenient number of equal parts—in this instance four, represented by the spaces A—B, B—C, C—D, D—E. The velocity of water along path at A and E has been already assumed. Like values at B, C, D may be obtained at these times by the use of Equation 4, giving t in that equation its appropriate value. The values V_B , etc., having been thus ascertained,

are then substituted for V_E in Equation 3, and S found and laid off from A to B, A to C, etc. As before stated, similar points F, G, H, on the retardation curve, between E and K, and corresponding ordinates F¹, G¹, H¹, (Fig. 2), can be computed on the same principle as for B, C, D, and B¹, C¹, D¹.

IV. Knowing the revolutions per minute of the runner (360) and the time required for a particle of water to move from A to E, it is readily found how far the inner tip of the impeller blade will move in the same time. This distance is A e and is equal to—

$$t \text{ (Eq. 4)} \times \frac{r.p.m. \times 2\pi r_A}{60} \quad 80$$

Since the increasing distances A—B, B—C, C—D, etc., are assumed to be traversed in equal times by a particle of water, the equally spaced points b, c, d , etc., (Fig. 1) will represent the position of the inner tip of air impeller blade when the water particle is at B, C, D, etc., both having passed the point A at the same time.

From these data, the form of impeller blade can be conveniently obtained as follows: On a piece of tracing cloth placed on the drawing, Fig. 1, and pivoted to rotate about O as a center, mark a point A₂ (not shown) directly over A, Fig. 1; then rotate the cloth until its point A₂ is directly over b , and directly over B in the drawing, mark a point B₂ on the tracing cloth; do also the same at c, d and e , and there will be obtained on the tracing cloth a series of points A₂—E₂, which, when connected, will give a form of blade—in longitudinal edge view—similar to the form of blade E e, Fig. 1. The form thus obtained should check with the velocity construction shown at e, f, g, h , in which $e—f$ represents velocity of tip of impeller blade and $e—g$ velocity of particle of water at e , both in feet per second, thus giving the resultant velocity $e—h$ relative to the runner. A tangent to the blade at e should coincide with this resultant $e—h$, to insure the absence of shock at the point e . Should there be lack of coincidence here, some changes in the assumptions must be made, as for example in the water path A—E, until the two constructions agree.

It will be noted that the impeller blade, as shown in the diagrammatic Figs. 1 and 2, inclines rearwardly relatively to the direction of revolution of the runner (indicated by the arrow in Fig. 1) and is narrower at a point between its ends, and thence gradually increases in width toward each end to conform to the contour of the runway in which it moves; also that the outer end or portion of said rearwardly inclined blade, is curved forwardly or in the direction of rotation. Both of these features I believe to be new with me. Allowance for thick-

ness of blades may be made by considering an increase in the diameter of the intake circle 5, and also by an increase in the ordinates, as at A^1 , B^1 , C^1 , etc.; modifications in the values of the affected terms in the equations hereinbefore given should be made accordingly.

Since the foregoing computations are theoretically true only for particles of water flowing in the median plane A^1-K^1 , Fig. 2, modification of the form of the impeller blade may be made to vary its longitudinal contour slightly from that represented by the line e E , Fig. 1, to that represented by dotted lines V W in the same figure, this modification being obtained by erecting curved ordinates normal to the direction of flow of particles not in the median plane, instead of the straight ordinates at A^1 , B^1 etc., Fig. 2. Also, according to the action of the water in flow, the form of the blade at the inner and outer extremities may be varied from straight lines as shown in Figs. 1 and 2, to curved lines. These modifications are desirable as tending to insure uniform acceleration of all the particles of water, but in many cases are practically negligible, their importance for a given size of runner increasing with the increased quantity of water to be handled. In Figs. 3 and 4, etc., I have shown an impeller blade 4 which embodies all these features. The throat or runway in Fig. 3 varies somewhat in form and dimensions from that shown in Fig. 1; but this is not material as concerns the present purpose. It will be noted that the blade has the longitudinal forward curvature 7 of its outer end which characterizes the blade graphically indicated at E e , Fig. 1. The edges of its two ends have a slight lengthwise curvature as indicated at 8, 9, Fig. 3; its inner end has a transverse concave formation 10, which gradually diminishes in depth and finally vanishes as it approaches the longitudinal center of the blade, as indicated in Figs. 4, 5 and 6; its outer end, which has the forward curvature 7, hereinbefore referred to, is also curved transversely, having a convex formation, 11, which is practically confined to the forwardly curved part 7 and decreases as it recedes from the outer end of the blade until it finally vanishes. These curves 8, 9, 10, 11, as before said, while desirable are not indispensable.

The lengthwise transverse dimensions of the blade are clearly indicated in Figs. 5-8. It is narrowest at a point between its ends as indicated in Fig. 7, and thence, as indicated in Figs. 5, 6, 8, gradually increases in width toward each end, its edges conforming to the curvature of that portion of the opposed walls of the runway between which it is included.

The curve through R , S , T , etc., Fig. 1, is

the usual spiral water way, which should be proportioned so that sections K^1 , R^1 , etc., Fig. 2, will pass their proportionate amount of water on its way to the discharge pipe P .

The fin Z , Fig. 1, is formed on such a construction line as will give the best division of the two streams, one making for the discharge pipe, and the other for the spiral circuit.

In obtaining the form of blades and the channels by the process herein set forth, suitable allowance, of course, must be made for friction values.

The principles of construction here involved apply to double intake, as well as to the single intake which is here illustrated.

In this particular pump the intake water is given a radial direction of flow in the mid-plane of the blade and at the inner tip of the blade, but this is not an essential feature, as the water may be taken to have a flow at any initial angle, and the design worked out accordingly along the lines herein specified. This initial angle may be necessary for best work in high-lift and in double-intake pumps, and may be established by a suitable shape of intake opening or by directing vanes.

As before pointed out, the blades while inclined rearwardly from the point of intake, have at the same time a forward curvature at their outer ends. I remark that this curvature is not arbitrary, but may, and will, be of greater or less extent, according to the varying factors of the problem.

Where the water to be handled contains solid matter in suspension, such as strings, chips, gravel, etc., the impeller blades should be stronger than for pure water, and should extend past the intake circle toward the center of the runner, to prevent clogging.

Having described my improvements and the best way now known to me of carrying the same into practical effect, I state in conclusion that I do not limit myself strictly to the structural details herein illustrated, since manifestly the same can be varied in a number of particulars without departure from my invention: but

What I claim as new and of my own invention is as follows:

1. The combination with the pump casing and runner, of impeller blades narrowest at a point between their ends and thence gradually increasing in width toward each end, secured to the runner and inclined rearwardly from the point of intake, and an annular throat or runway in the casing for said blades, having opposed faces convex in cross section and of a contour to conform to the longitudinal concave edges of the blades in the position which they occupy in the runway, substantially as and for the purpose hereinbefore set forth.

2. The combination with the pump casing and runner, of impeller blades carried by the runner, narrowest at a point between their ends and thence gradually increasing
5 in width toward each end, inclined rearwardly from the point of intake, and having a forward curvature at their outer ends, and an annular throat or runway in the casing for said blades, having opposed faces convex
10 in cross section and of a contour to conform

to the longitudinal concave edges of the blades in the position which they occupy in the runway, substantially as and for the purposes hereinbefore set forth.

In testimony whereof I affix my signature in presence of two witnesses.

JAMES L. COKER, JR.

Witnesses:

JOHN L. FLETCHER,

W. LEE HELMS.