

Dec. 7, 1937.

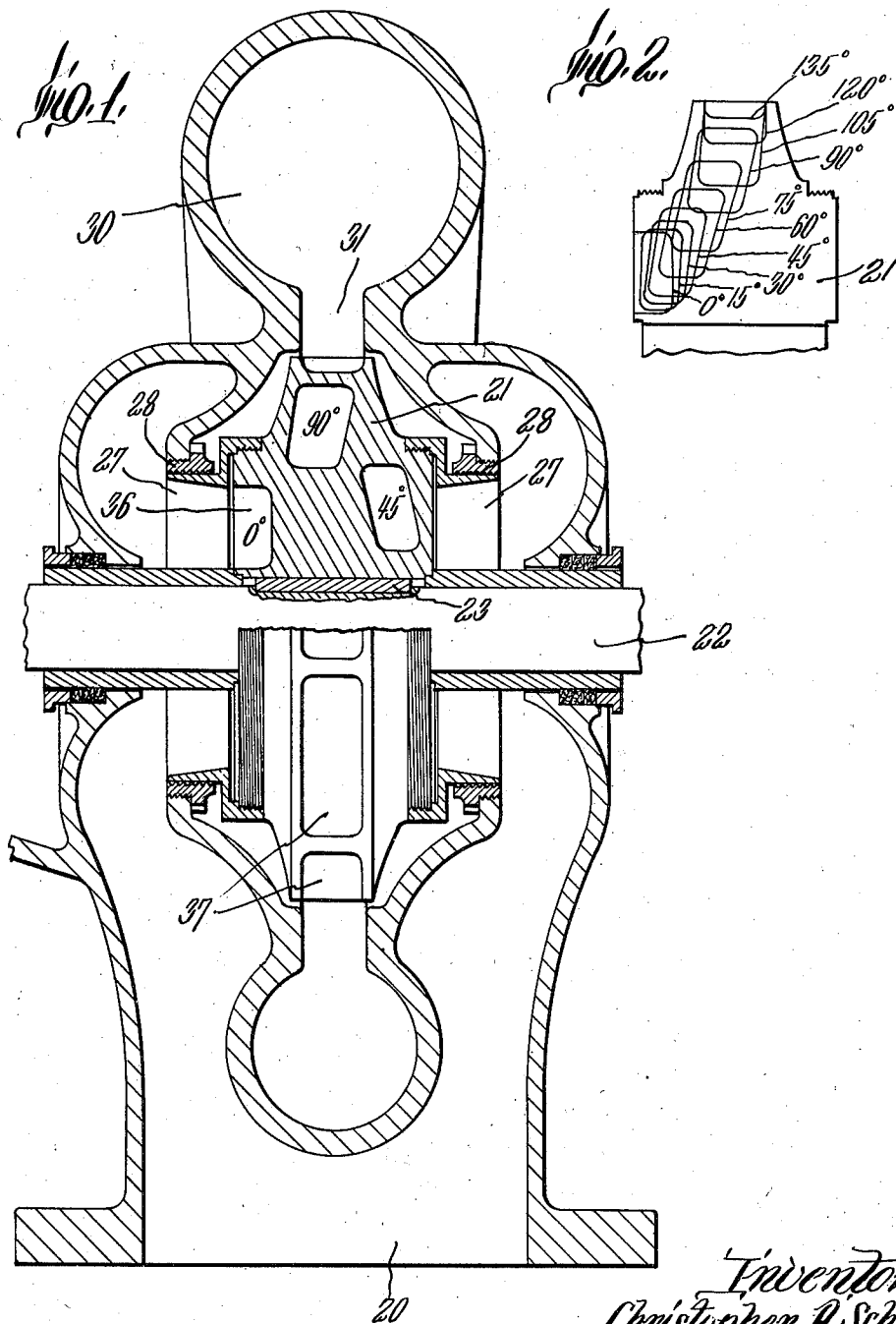
C. A. SCHELLENS

2,101,653

IMPELLER FOR CENTRIFUGAL PUMPS

Filed Sept. 1, 1934

4 Sheets-Sheet 1



Inventor
Christopher A. Schellens
by Wright, Brown, Quinby & May
Attys

Dec. 7, 1937.

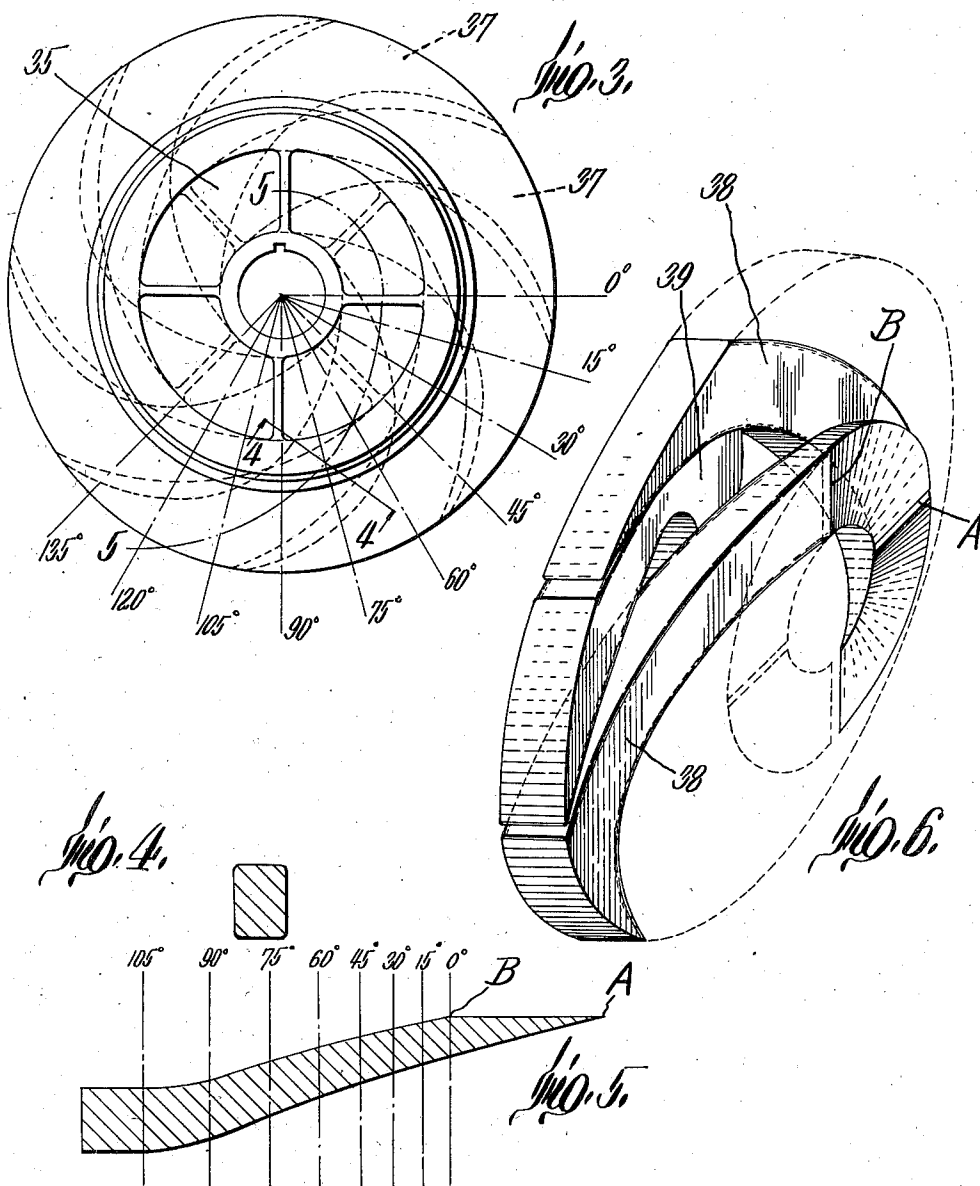
C. A. SCHELLENS

2,101,653

IMPELLER FOR CENTRIFUGAL PUMPS

Filed Sept. 1, 1934

4 Sheets-Sheet 2



Inventor
Christopher A. Schellens
by Wright, Brown, Quincy & May
Attys

Dec. 7, 1937.

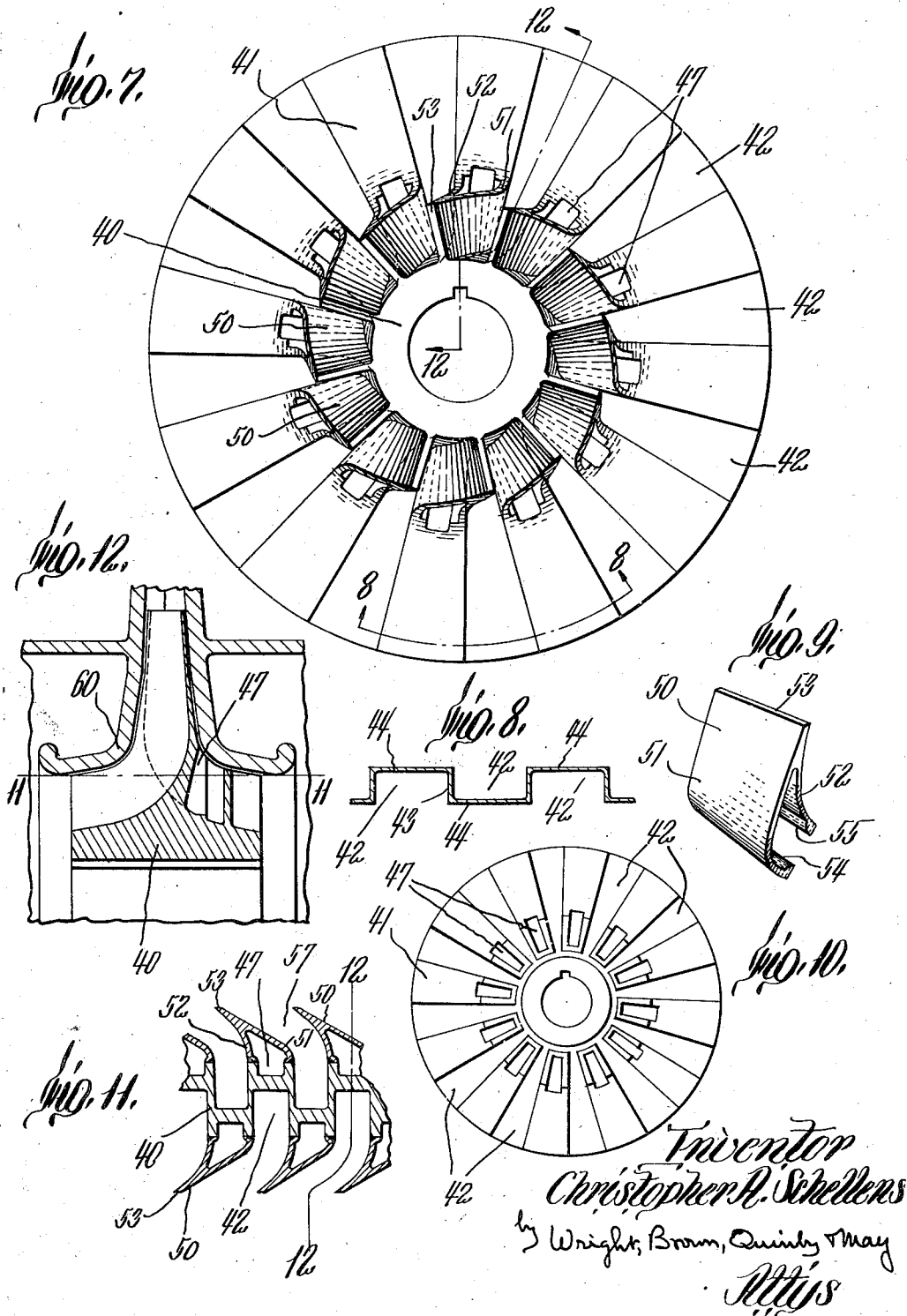
C. A. SCHELLENS

2,101,653

IMPELLER FOR CENTRIFUGAL PUMPS

Filed Sept. 1, 1934

4 Sheets-Sheet 3



Dec. 7, 1937.

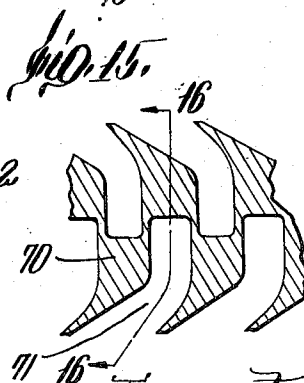
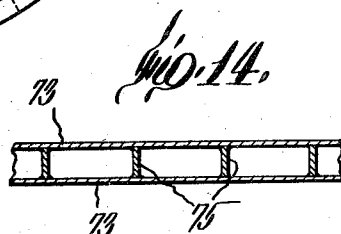
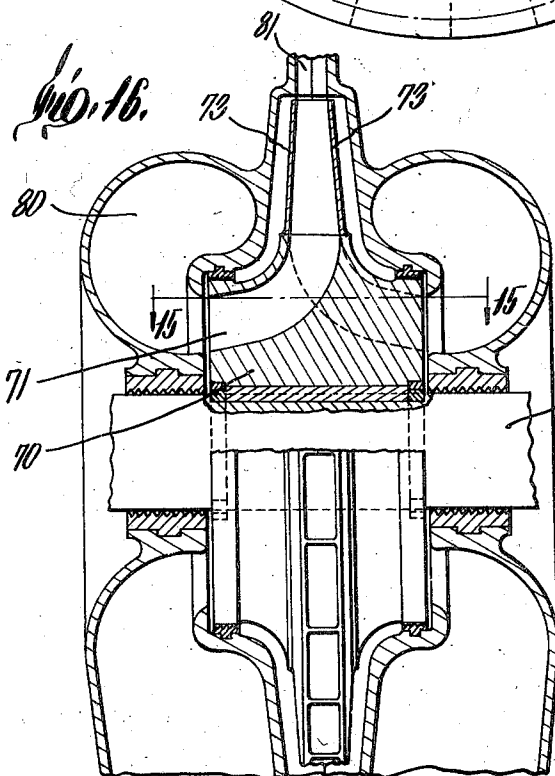
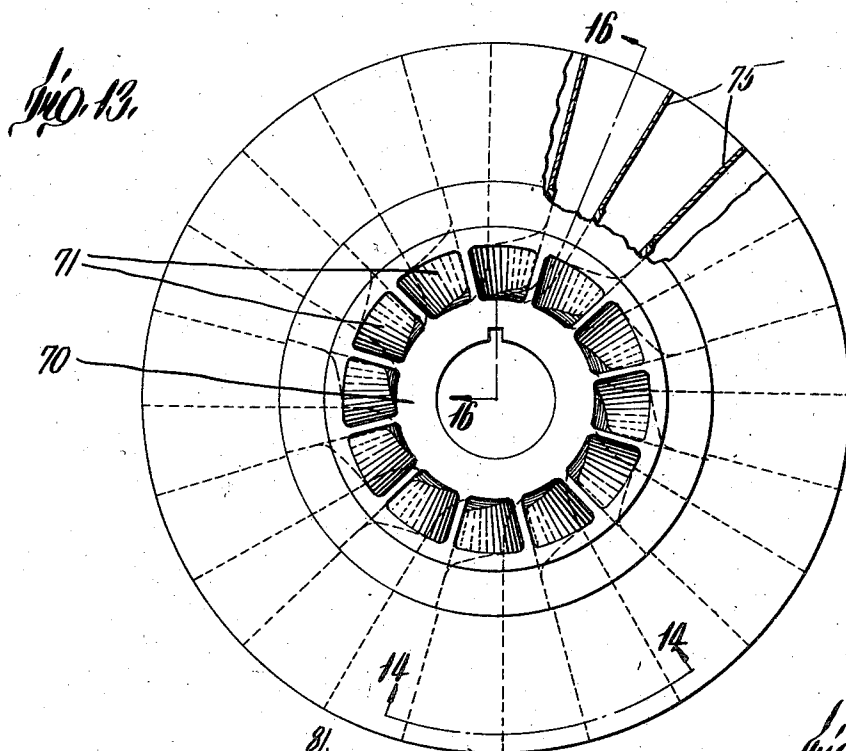
C. A. SCHELLENS

2,101,653

IMPELLER FOR CENTRIFUGAL PUMPS

Filed Sept. 1, 1934.

4 Sheets-Sheet 4



Inventor
Christopher A. Schellens
by Wright, Brown, Quincy & May
Attys

UNITED STATES PATENT OFFICE

2,101,653

IMPELLER FOR CENTRIFUGAL PUMPS

Christopher A. Schellens, Marblehead, Mass., assignor to C-S Engineering Company, Englewood, N. J., a corporation of Delaware

Application September 1, 1934, Serial No. 742,461

12 Claims. (Cl. 103—115)

This invention relates to improvements in impellers for centrifugal pumps, both for pumping liquids and for pumping gases, centrifugal pumps for the latter being customarily known as blowers or fans, and to methods for making the same.

The invention relates more particularly to centrifugal impellers of the double-eye or double-inlet type whereby the fluid which is pumped is admitted at both sides of the impeller and is discharged radially from the periphery thereof.

It is an object of the invention to provide an impeller which is relatively easy to manufacture and is of greater efficiency than impellers now known in the art, particularly for pumping fluids at delivery pressures which are relatively high in comparison with the delivery volume rate and the rate of rotation.

In centrifugal pumps for fluids, the fluid is customarily received into the impeller in an axial direction at a relatively low velocity, and, in the case of a single stage pump or the first stage of a multi-stage pump, at a relatively low pressure. By the rotary motion of the impeller, the velocity of the fluid is changed to a radial direction and increased and its pressure is built up to a greater or less extent according to the design and speed of rotation of the impeller. The changes in velocity of the fluid passing through the impeller necessarily result in shock losses in addition to which there are also unavoidable friction losses due to the frictional contact of the fluid with the walls of the passages through which it passes and also to fluid friction within the stream caused by eddy currents. For efficiency of operation, it is necessary to reduce these losses to a minimum by avoiding as far as possible abrupt changes in fluid velocity, by finishing the walls of the passages smooth, and by shaping the passages in such a way as to minimize eddy currents. If the cross-sectional area of a passage in an impeller increases materially from the inlet end to the discharge end thereof, eddy currents are liable to be formed therein, particularly if there is a marked constriction in the passage near the inlet end. Since the circumference of an impeller at the point of fluid entrance is considerably less than the circumference at the discharge orifices, there is ordinarily less space available for fluid passages near their entrance ends than there is near their discharge ends. For efficient operation, it is desirable that the driving surfaces in the portions of the fluid passages near the discharge periphery of the impeller be spaced not too far apart angularly. On the other hand, it is desirable that the partition walls separating

successive fluid passages at their entrance ports be not too close together since this results in small inlet ports and an undesirably large number of partition walls. It is an object of the invention to design the impeller so that the fluid passages therein will have an approximately constant cross-sectional area from end to end. To this end, according to the invention, I utilize both sides of the impeller for inlet ports, each inlet port having a corresponding separate passage through the impeller to the periphery thereof, the passages from the opposite sides leading to alternate orifices in a single circular series of discharge orifices in the periphery of the impeller. Thus, if the partitions between successive inlet orifices and between successive outlet orifices are relatively thin, as they should be, each inlet orifice will subtend an angle at the axis of rotation approximately twice as great as the angle subtended by one of the outlet orifices. This makes possible a relatively constant cross-sectional area in each of the fluid passages. Furthermore the provision of thin partitions between successive inlet orifices minimizes the area of obstructing surface offered to the entering stream, this resulting in marked avoidance of turbulence and abrupt velocity changes in the fluid entering the inlet orifices. This feature is vitally important where a liquid near its boiling point is being pumped. In such cases the liquid must be guided into the impeller passages as gently as possible with minimum turbulence, the pressure in the stream being maintained substantially uniform until the liquid is well inside of the several passages. It is also desirable to minimize turbulence at the discharge of the impeller. To this end the discharge orifices at the circumference are separated by thin blades. The foregoing features constitute related steps toward the production of a theoretically ideal centrifugal pump.

For pumps designed to pump liquids, the fluid passages in the impeller are also preferably designed somewhat in accordance with the characteristics of the passages in the impeller described and illustrated in my U. S. Letters Patent No. 1,664,488, granted April 3, 1928. Passages shaped in this manner receive and discharge streams of liquid with minimum turbulence and with a high degree of uniformity of acceleration of the liquid as is hereinafter pointed out in more detail.

It is also an object of the invention to provide impellers for blowers or fans adapted to be operated at high rotational speeds and to avoid undesirable constrictions in the fluid passages near the inlet end thereof.

It is a further object of the invention to provide new and improved methods of making impellers of improved design.

For a more complete understanding of the invention, reference may be made to the disclosure of certain embodiments thereof which are hereinafter described and are illustrated on the drawings, of which

Figure 1 is a sectional view of a centrifugal pump adapted for pumping liquids and including my improved impeller.

Figure 2 is a fragmentary sectional outline of a portion of the impeller or runner shown in Figure 1, successive sectional outlines of a fluid passage being superimposed thereon.

Figure 3 is a side elevation of an impeller embodying the invention.

Figure 4 is a transverse section of a fluid passage taken on the line 4—4 of Figure 3.

Figure 5 is a longitudinal median section of a fluid passage taken on the line 5—5 of Figure 3.

Figure 6 is a perspective view of three successive cores for fluid passages, these cores being shown in the position they would occupy when used for making a mold for casting an impeller such as is illustrated in Figures 1 and 3.

Figure 7 is an end view of an open-type impeller adapted for use in a blower or fan.

Figure 8 is a section on the line 8—8 of Figure 7.

Figure 9 is a perspective view of one of the entrance vanes used in the construction of the impeller shown in Figure 7.

Figure 10 is an end elevation, on a smaller scale, of a machined blank to which entrance vanes are to be attached to form an impeller such as is shown in Figure 7.

Figure 11 is a developed fragmentary section taken on the line 11—11 of Figure 12.

Figure 12 is a fragmentary section of the impeller shown in Figure 7, taken on the line 12—12 of Figure 11, together with adjacent portions of the pump casing.

Figure 13 is an end elevation of an impeller of the closed type suitable for a blower or fan.

Figure 14 is a section on the line 14—14 of Figure 13.

Figure 15 is a fragmentary developed section on the line 15—15 of Figure 16.

Figure 16 is a fragmentary sectional view of the impeller shown in Figure 13, taken on the line 16—16 of Figures 13 and 15, and including portions of the pump casing.

The pump illustrated in Figure 1 is intended for use particularly with liquids, and is more especially designed to deliver liquids at a relatively low volume rate but at a relatively high pressure. The pump may include a casing of conventional shape having an inlet opening 20 leading to both sides of the impeller or runner 21. This impeller is keyed to a driving shaft 22 as by a suitable key 23, customary provision being also made to lock the impeller against axial movement relative to the shaft 22. At each side of the impeller 21 an inlet eye 27 may be secured by a screw thread, as shown, or in any desired manner, these inlet eyes being of any preferred design. A gland member 28 may abut the outer surface of each inlet eye whereby fluid discharged from impeller 21 is restrained from short circulating excessively back to the inlet thereof. Surrounding the periphery of the impeller is a discharge chamber 30 having a narrow connecting passage 31 leading from the periphery of the impeller. At each side of the impeller is a circular

series of inlet orifices 35, four such orifices being provided at each end in the particular embodiment of the invention illustrated in Figures 1 and 3. The orifices 35 at the opposite sides of the impeller open into separate individual fluid passages 36 each of which is of a spiral-helicoidal shape, these fluid passages terminating in a single series of peripheral outlets 37. Since there are twice as many discharge orifices 37 at the periphery of the impeller as there are inlet orifices 36 at each side of the impeller, the inlet openings each subtend an angle at the axis of rotation approximately twice as great as the angle subtended by the corresponding discharge orifice. Thus the greater circumference at the periphery of the impeller compared with the circumference of the eyes at the ends of the impeller is to a large extent offset so that the fluid passages have an approximately constant cross-sectional area from one end to the other as is evident from Figure 2 which shows a series of cross sections of a fluid passage taken on radial planes spaced by angles of 15° as indicated in Figure 3. Thus rapid increase in cross section of the passages is avoided, particularly in the portion adjacent to the inlet end of each passage, so that not only are eddy currents minimized, but cavitation or flashing is avoided in liquids which are at temperatures near the boiling point adjacent to the impeller inlet.

Another advantageous feature of structure which helps to minimize turbulence in the fluid passing through the pump is the relatively thin dividing blades between successive inlet orifices and between successive outlet orifices. As a result, the stream which enters each inlet eye 27 is smoothly divided into four separate streams which flow through respective passages, as described, and are discharged with minimum turbulence through orifices which alternate with orifices through which are discharged the streams entering through the opposite eye 27.

The character of the fluid passages in the impeller can be seen at a glance from Figure 6 which shows in perspective three of the eight alternating cores 38, 39, used in making a mold for casting the impeller. Each core illustrates graphically the shape and position of a fluid passage in the impeller. Figure 6 thus shows how the successive passages are arranged so that alternate discharge orifices, represented by the outer end faces of the cores, are connected with inlets at opposite sides of the impeller. Each fluid passage is characterized by the fact that its two walls, which at the inlet constitute partitions separating it from the adjacent passages, merge adjacent to the outlet into parallel planes perpendicular to the axis of rotation. Furthermore, the walls of each passage, which at the inlet are cylindrical, become thin partitions at the outlet separating the passage from adjacent passages. It will be noted that, in the specific embodiment illustrated on the drawings, the axial inlet opening of each passage subtends an arc of approximately 90°, that is, from the point A to the point B (Figures 5 and 6). Between these two points, the side walls of the passage are cylindrical so that no radial velocity is imparted to the liquid until it passes the point B. From the point B to the discharge orifice the passage is enclosed by four walls. Thus the fluid enters the enclosed portion of the passage with no radial velocity whatever, the radial velocity of the stream being acquired between the point B and the discharge orifice. The curvature of the pas-

sage determines the radial acceleration of the liquid at any point, such acceleration being preferably as nearly uniform as possible. The series of cross sections shown in Figure 2 and the longitudinal section shown in Figure 5 indicate the gradual increase in cross-sectional area of the fluid passage. This avoids undesirable eddies in the stream since the increase in cross-sectional area from the reference point B through an angle of 45° or more is very slight. By the time the stream reaches the 45° mark, considerable pressure has been built up therein due to the rotation of the impeller, so that cavitation associated with eddy currents further along the passage is thus prevented. In designing impellers for ratings other than that shown, it may be necessary to modify the cross-sectional shape and/or curvature of the passages at certain points to preserve an adequate wall thickness between adjacent passages at all points.

An impeller particularly intended for pumping gaseous fluids is illustrated in Figures 7 to 12 inclusive. This impeller preferably consists of a member having a hub portion 40 and an outer portion 41 integral therewith. This member may be conveniently cut from a solid blank. According to the invention, radial channels 42 are gouged out alternately in opposite faces of the blank so that the peripheral edge of the impeller has a crenelated shape as indicated in Figure 8. This portion of the structure may be said to comprise a series of thin radial blades 43, each blade having its width parallel to the axis of rotation. The side edges of each blade 43 are each connected to the opposing side edge of one adjacent blade by a web 44. Thus the outer portion of the impeller has open radial channels in both faces, the channels in one face alternating with the channels in the other face. This radial form of structure for the blades 43 and the connecting webs 44 is necessary in the case of blowers or fans which are intended for operation at high rotational speeds on account of the great radial stresses set up by centrifugal forces. The crenelated form of disk portion is also advantageous in that it readily permits an increase of circumference with minimum stress. This is important since, in blowers which are operated at high speeds, the air passing through the channels may undergo a large increase of pressure during its travel from the inner ends of the channels 42 to the outer ends thereof. This increase of pressure is accompanied by the usual rise in temperature, so that, during the operation of the blower, the peripheral portion may be maintained at a considerably higher temperature than the hub portion. Unless provision is made for thermal expansion of the peripheral portion, large mechanical stresses may be set up in the structure of the impeller in addition to the stresses produced by centrifugal forces. The thermal expansion of the peripheral portion of the impeller illustrated in Figure 7 is readily taken care of by the crenelated form of this portion of the impeller.

In addition to gouging out the channels 42, I preferably hollow out the outer portion adjacent to the hub portion as at 47 so as to reduce the weight of and consequent centrifugal forces acting on the impeller.

The outer portion 41 of the impeller can also be made in other ways such as by stamping or forging sheet metal blanks to form alternating radial channels 42 in the opposite faces thereof.

In order to provide suitable inlet channels with

contiguous entrances leading to the radial channels, I preferably secure to each side of the impeller member 41, as by welding, a series of entrance vanes 50. Each of these vanes may be in the form illustrated in Figure 9. As therein shown, each vane 50 consists of two sheets 51 and 52 diverging from a thin blade at the common edge 53 and having separate edges 54 and 55 respectively which are independently welded or otherwise secured to the impeller member 41 as indicated in Figures 7 and 11. The entrance vanes 50 thus have a cross section resembling a V shape with a central hollow which saves weight. As is evident from Figure 11, the legs 51 and 52 of the V serve to stiffen the vane as a whole and to prevent vibration such as is liable to occur in the case of vanes consisting of single blades. Pairs of successive vanes 50 form entrance channels 57 which communicate with respective channels 42. Thus half of the radial channels connect with entrance channels at one side of the impeller, while the other half of the radial channels connect with passages at the other side of the impeller, and the angle subtended by each entrance channel at the axis of rotation is approximately twice as great as the entrance channel subtended by a radial passage. The divergence of the legs or side elements 51 and 52 of the vanes 50 not only stiffens and strengthens the vanes but also minimizes eddy currents in the fluid passages in the region of sharpest curvature thereof, that is, near the inlets, by maintaining an approximately constant width of passage in such regions as is evident from Figure 11.

The impeller is designed to rotate in a casing 60 having portions fitting closely adjacent to the side faces of the impeller, as shown in Figure 12. Thus the casing itself cooperates with the channels 42 and 57 to form enclosed passages from the entrance orifices to the peripheral discharge orifices.

As herein employed, the word "passage" is intended to have a broad meaning including an open or closed way to guide the flow of a fluid medium. The word "channel" is intended to refer to an open passage such as a groove.

An impeller of the closed type for a blower or fan is illustrated in Figures 13 to 16 inclusive. The fluid passages in this impeller are enclosed from end to end and are made radial for the greater part of their length on account of the large centrifugal stresses imposed on their walls by high speeds of rotation. The shape of the individual fluid passages may be substantially similar to the shape of the passages in the impeller illustrated in Figure 7. Instead of open channels, however, the impeller shown in Figure 13 is provided with enclosed passages from the inlet port to the discharge orifice. This form of impeller may be made by casting a central member 70 containing a series of entrance passages 71 entering the member 70 at each side thereof and curving so as to open out radially at the periphery thereof. As shown in Figure 15 the member 70 is shaped so that the portions separating successive inlets are in the form of thin blades at the orifices of the inlets. As indicated in Figures 13 and 15, these entrance passages enter alternately at the opposite sides of the hub member. The outer portion of the impeller may consist of a pair of slightly conical angular plates 73 which are secured to the periphery of the central member 70 as by welding. Between these plates are a series of radial blades 75, each blade having its width parallel to the axis of rotation and its side

edges secured as by welding or otherwise to the respective plates 73, as indicated in Figure 14. Each blade 75 is also welded or otherwise permanently secured to the periphery of the hub member 70 so as to register with a partition separating a pair of successive discharge orifices of the entrance passages 71. In assembling an impeller of this kind, one of the plates 73 may be conveniently welded to the periphery of the hub member 70. The blades 75 may then be welded along one side edge to the plate 73 and along the inner end edge to the periphery of the hub member 70. Finally, the second plate 73 may be welded to the periphery of the hub member 70 and to the free side edges of the blades 75. As indicated in Figure 16, this impeller may be mounted on a suitable shaft 22 for rotation in a casing having a suction chamber 80 and a discharge passage 81.

It is to be noted that in the form of my invention illustrated in Figures 7 and 13, in contradistinction to the form illustrated in Figure 3, the walls of the fluid passages which form driving surfaces at the passage entrance extend into walls which form corresponding driving surfaces at the discharge from the passages, while the non-driving walls are substantially surfaces of revolution throughout their length. In this form of construction, the curvature of the passages near their inlet ends is of necessity greater than in the form illustrated by Figure 3, as can be seen by an inspection of Figures 11 and 15 contrasted with Figure 6. It will be seen, however, that the walls of channels 57 and 71 of Figures 11 and 15 respectively are free from the tendency to diverge rapidly in the direction of fluid flow in the region of the sharpened curvature of the passage and the formation of eddies is therefore minimized. The non-divergence of the above passage is made possible by the inactive spaces between walls 51 and 52.

While I have described the form of my invention as shown in Figure 3 as being particularly suited to the pumping of liquids, and the forms shown in Figures 7 and 13 as being more suitable for the pumping of gases, either form may in a particular case be best adapted to the pumping of either fluid.

It is understood that the embodiments of the invention herein illustrated and particularly described are set forth not by way of limitation but by way of illustration and that the invention is not to be limited thereto in its scope except as defined in the following claims.

I claim:—

1. In a centrifugal pump, an impeller having fluid passages extending from both sides thereof to the circumference thereof, each said passage having two walls which adjacent to the inlet end form partitions between said passage and other similar passages and which adjacent to the outlet side are disposed in parallel surfaces of revolution substantially perpendicular to the axis of rotation of the impeller, each said passage having two other walls which at the inlet are disposed in substantially cylindrical surfaces of revolution and which adjacent to the outlet end form partitions to separate said passage from fluid passages leading from the opposite side of the impeller.

2. A centrifugal impeller comprising a central portion having a series of inlet channels entering each side axially with contiguous entrances and curving to radial directions, and an outer portion having a single series of contiguous radial channels communicating with respective inlet

channels, successive radial channels being alternately open to opposite sides of said outer portion.

3. A centrifugal impeller comprising a series of radial blades, the width of each blade being parallel to the axis of the impeller, said blades constituting walls of a series of radial passages terminating in a series of contiguous discharge ports, and a hub portion projecting axially beyond said blades in both directions, said hub portion having inlet passages extending from series of contiguous inlet ports in the sides thereof to said radial passages, the radial passages being connected in alternation with the inlet passages of the opposite sides.

4. A centrifugal impeller comprising a series of radial blades, the width of each blade being substantially parallel to the axis of the impeller, said blades constituting walls of a series of radial passages terminating in a series of contiguous discharge ports, and a hub portion having inlet passages extending from series of contiguous ports in the sides thereof to said radial passages, each inlet passage subtending at its entrance an angle at the axis approximately twice as great as the angle subtended by each radial passage.

5. A centrifugal impeller comprising a series of radial blades, the width of each blade being parallel to the axis of the impeller and tapering outwardly, substantially radial walls each extending from an edge of a blade to the corresponding edge of a blade next adjacent thereto whereby a single series of radial passages are formed terminating in a series of contiguous discharge ports, and a hub portion having inlet passages extending from series of contiguous inlet ports in the sides thereof to respective radial passages there being half as many inlet passages from each side as the total number of radial passages.

6. A double-eye centrifugal impeller comprising a one-piece hub member having inlet passages therein starting inward axially from the ends thereof and curving to radial directions, a pair of spaced annular disks permanently secured to the periphery of said member, and a series of radial partition elements permanently secured along their side edges to the opposing faces of said disks and along their inner end edge to said member, said disks and partition elements forming radial passages which successively communicate with respective inlet passages from alternate ends of the impeller.

7. A double-eye centrifugal impeller comprising a one-piece member having a series of inlet passages entering each end thereof axially and curving within said member to open radially at the periphery thereof in a single circular series of orifices communicating alternately with the opposite ends of said member, a pair of annular spaced disks permanently secured to the periphery of said member on either side of said series of orifices, and radial partition elements each permanently secured along its side edges to said disks and along its inner end edge to the periphery of said member between two successive orifices.

8. A centrifugal impeller having a circular series of four substantially contiguous inlet ports on each side thereof and a series of eight substantially contiguous outlet ports at the periphery thereof, said impeller having eight helicoidal passages connecting said inlet ports with respective outlet ports, the inlet ports on each side being connected with alternate outlet ports at the periphery.

9. In a centrifugal pump, an impeller having a circular series of substantially contiguous inlet ports at each side thereof and a single series of outlet ports at the circumference thereof, the inlet ports of each series being in a plane perpendicular to the axis of rotation of the impeller, said impeller having passages therein connecting said outlet ports with inlet ports at alternate sides, said passages starting helically from the inlet orifices and curving toward the outlets with not more than a slight change in cross-sectional area.

10. In a centrifugal pump, an impeller having a circular series of substantially contiguous inlet ports separated by thin blades at each side adjacent to the axis thereof and a single series of substantially contiguous discharge ports separated by thin blades along the circumference thereof, successive discharge ports communicating alternately with inlet ports at opposite sides of said impeller.

11. In a centrifugal pump, an impeller having a circular series of inlet ports separated by thin

blades at each side adjacent to the axis thereof, the ports of each said series being disposed in a plane perpendicular to said axis, and a separate fluid passage extending from each said inlet port to the periphery of said impeller, said passages terminating in a single series of discharge outlet ports separated by thin blades.

12. A centrifugal impeller having a circular series of substantially contiguous quadrilateral entrance orifices at each side thereof, the orifices of each series being disposed in a plane perpendicular to the axis of rotation of the impeller, each said orifice being defined by a pair of radial lines and a pair of arcs concentric with said axis and separated from each other by thin blades, said impeller having passages leading from said orifices to the periphery thereof and terminating in a single series of substantially contiguous discharge orifices separated by thin blades in said periphery and corresponding in alternation to the inlet orifices on the opposite sides of the impeller.

CHRISTOPHER A. SCHELLENS.