Dec. 7, 1965 J. S. SWEARINGEN 3,221,661

LOW-SUCTION HEAD PUMPS

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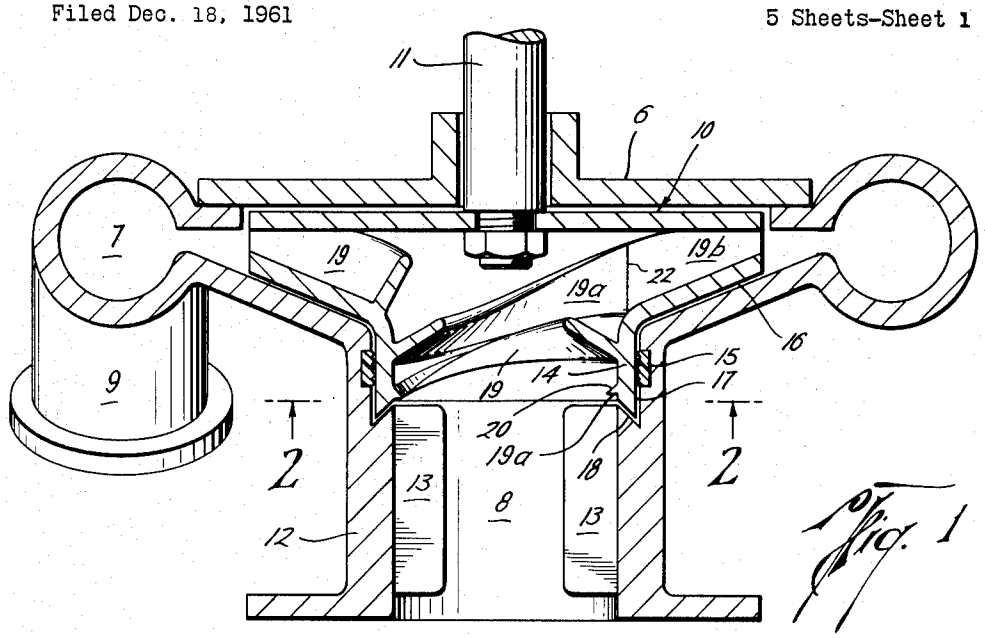
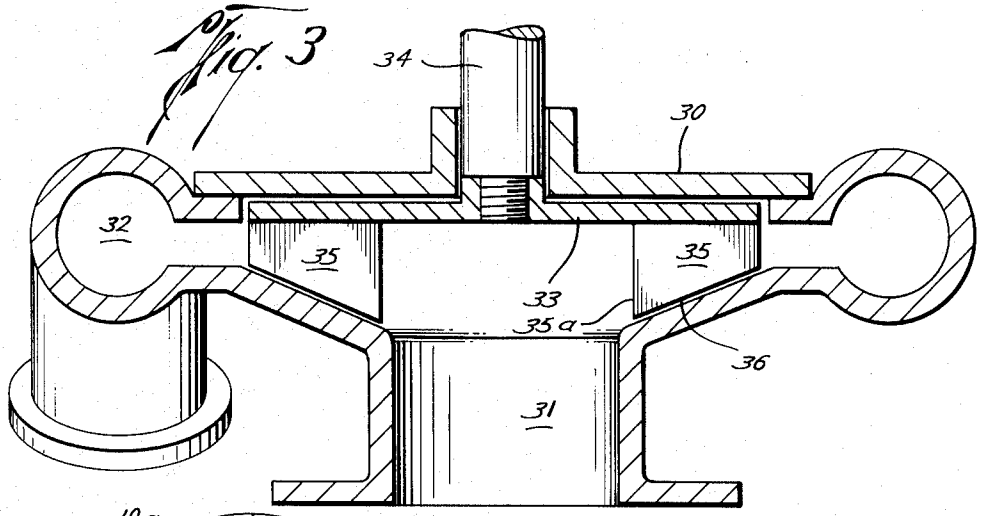


Fig. 1

Fig. 3



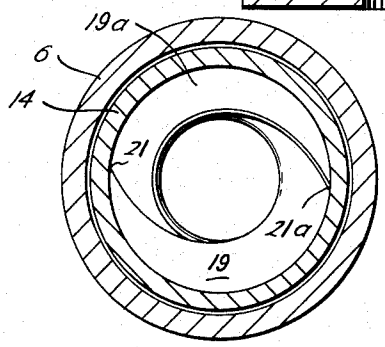


Fig. 2

Judson S. Swearingen
INVENTOR.

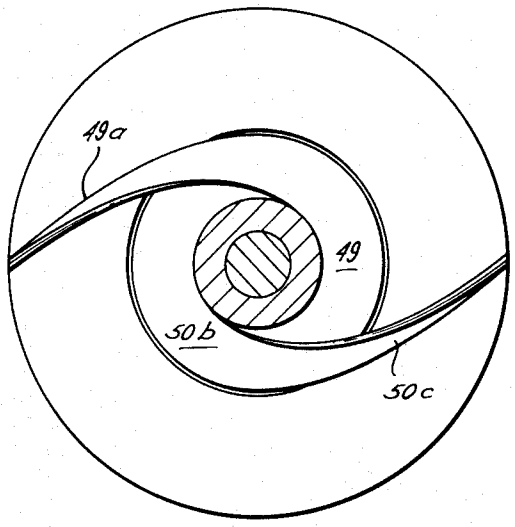
BY Browning, Simmons, Hyer & Eickenroth
ATTORNEYS

Dec. 7, 1965 J. S. SWEARINGEN 3,221,661

LOW-SUCTION HEAD PUMPS

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Fig. 5



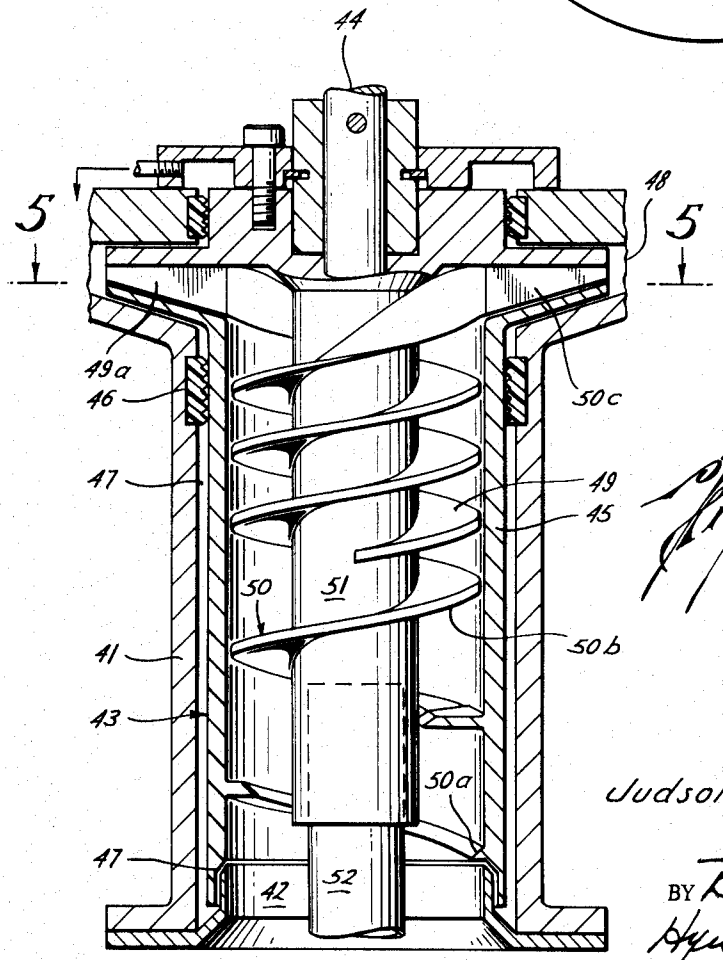


Fig. 4

Judson S. Swearingen
INVENTOR.

BY Browning, Simmons, Hyer & Brickenstein
ATTORNEYS

Dec. 7, 1965 J. S. SWEARINGEN 3,221,661

LOW-SUCTION HEAD PUMPS

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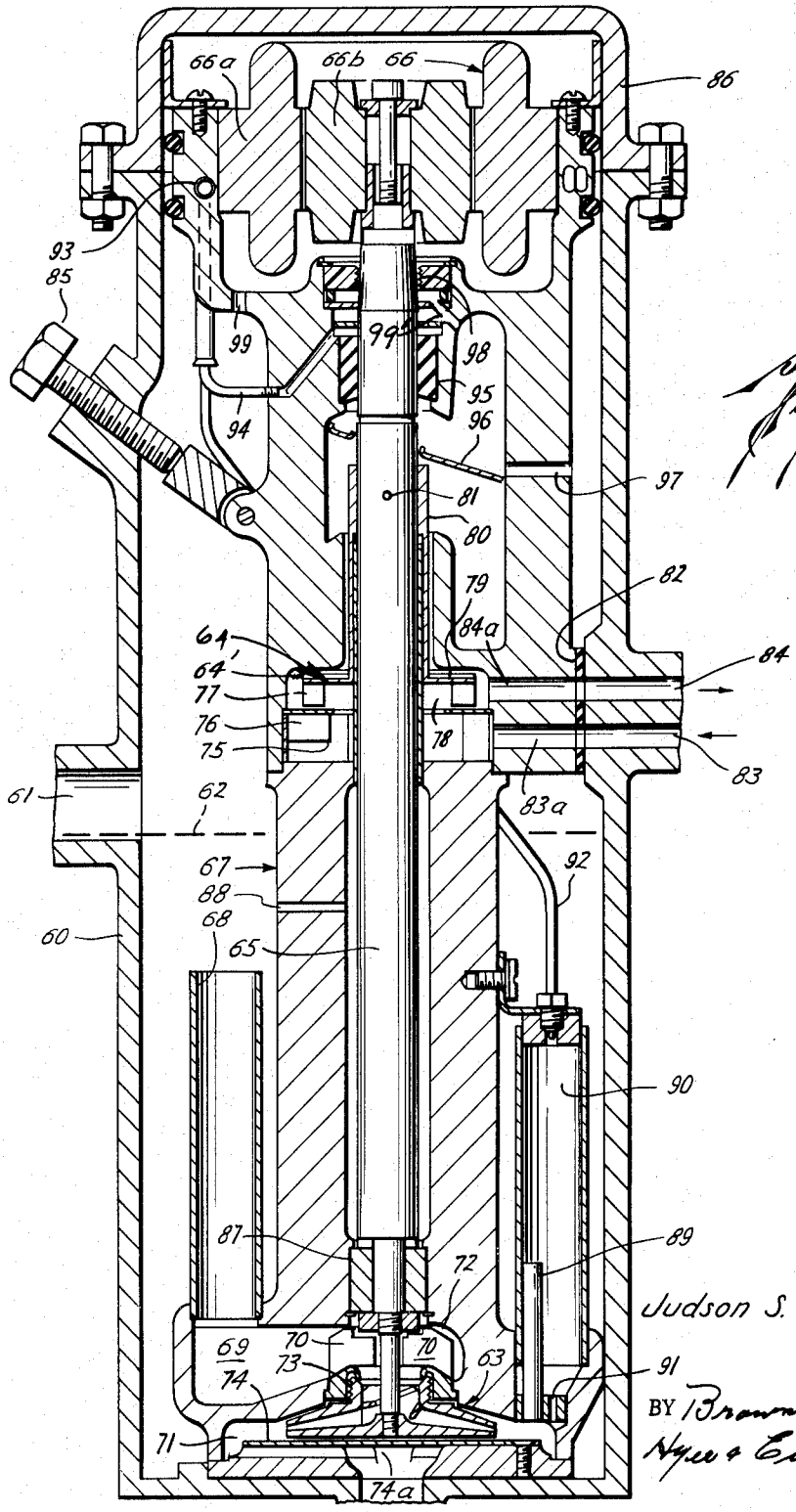


Fig. 6

Judson S. Swearingen
INVENTOR.

BY Browning, Simmons, Nyce & Eisenrauch
ATTORNEYS

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LOW-SUCTION HEAD PUMPS

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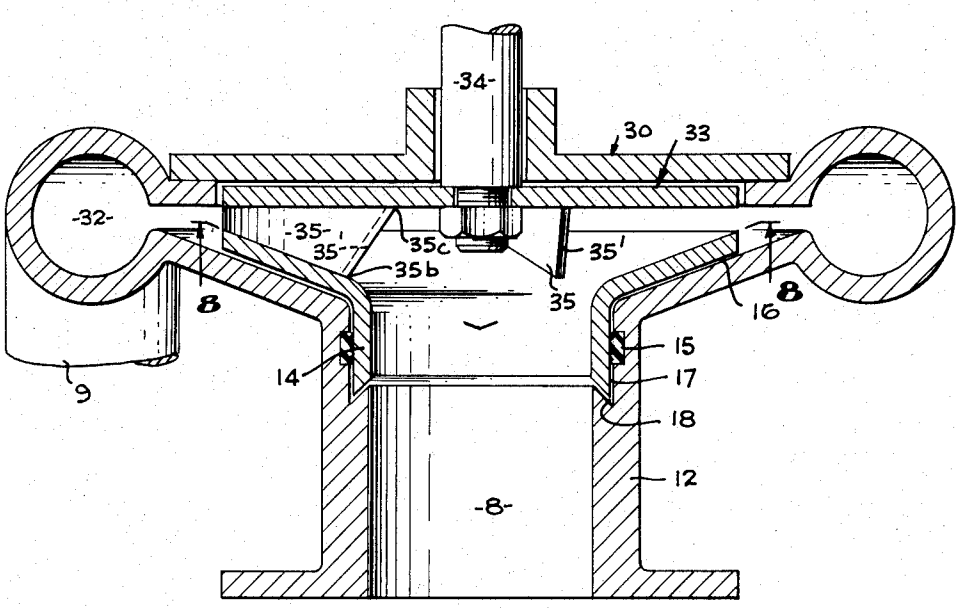


Fig. 7

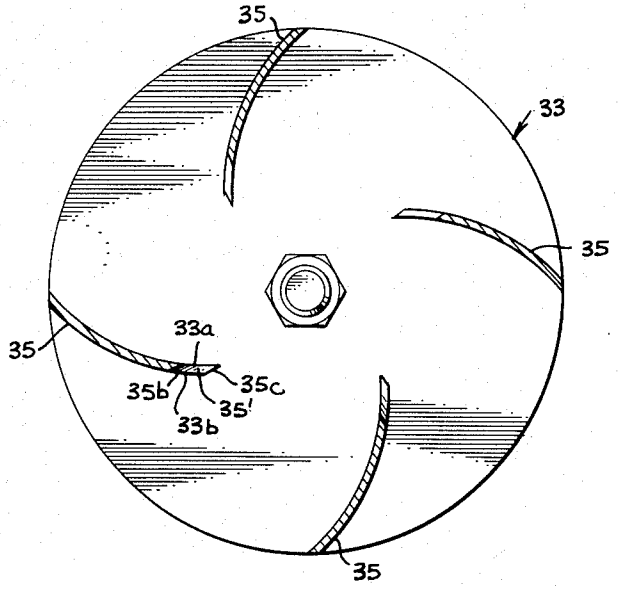


Fig. 8

INVENTOR.
JUDSON S. SWEARINGEN
BY Browning, Semina, Hyer & Eickenroht
ATTORNEYS

Dec. 7, 1965 J. S. SWEARINGEN 3,221,661

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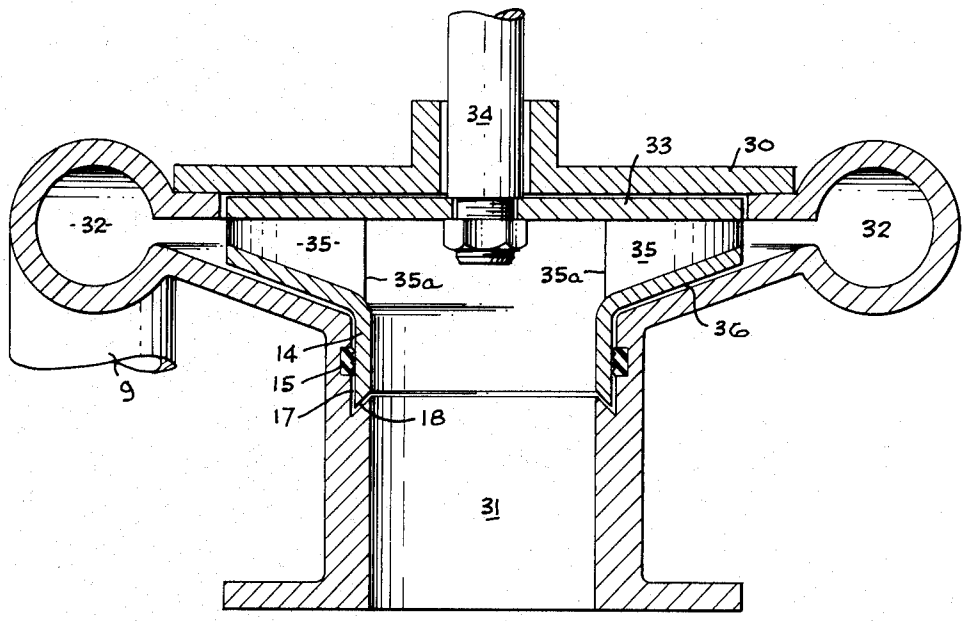


Fig. 9

INVENTOR.
JUDSON S. SWEARINGEN
BY Browning, Simms
Hyer & Eickemeyer
ATTORNEYS

United States Patent Office

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LOW-SUCTION HEAD PUMPS

Judson S. Swearingen, Los Angeles, Calif., assignor, by mesne assignments, to Electronic Specialty Co., Los Angeles, Calif., a corporation of California

Filed Dec. 18, 1961, Ser. No. 159,901
23 Claims. (Cl. 103—87)

This invention relates to improved centrifugal pumps. In one aspect, it relates to improvements in centrifugal pumps which are particularly useful in handling liquids under low head at the suction of the pump, such as liquids at temperatures substantially at their boiling points at pressures existing at the inlet of the pump; and in another aspect, it relates to improved pumping systems particularly useful in handling liquid refrigerants and liquid absorbents in absorption-refrigeration systems.

Conventional centrifugal pumps require a minimum total absolute pressure on liquid entering the inlet of the pump to prevent cavitation. At the critical point for initiation of cavitation, pressure required usually is about 10 to 15 percent of the total head developed by the pump. At only slightly lower inlet pressures, cavitation occurs, and becomes serious; the rate of flow falls markedly, and extensive damage to the pump may follow.

There are several conditions present at the inlet of a centrifugal pump which cause cavitation unless there be sufficient pressure at the inlet. One of these factors is turbulence created by the rapidly rotating pump impeller. As an impeller blade meets liquid entering the pump, the blade must plow into the incoming liquid. It plows into it at a rather flat angle, and the liquid is thrown away from the blade violently because the blade moves through the liquid at several times the absolute velocity of the liquid so that high turbulence is created. This throwing of liquid back in outward direction must be overcome by pressure available to force the liquid into the inlet in the face of this impact, else cavitation results.

Another factor which requires pressure to overcome it is the whirling action of an incoming stream of liquid caused by friction with the rotating impeller. Rotating impeller parts, such as a disc carrying the blades, the nut holding the disc onto a drive shaft, or other parts of the impeller in contact with the incoming liquid, give it an initial whirling action in the pump inlet. The whirling action of liquid in this location is communicated to liquid further out from the pump. Many inlets to centrifugal pumps have converging walls, and the combination of converging walls and whirling of liquid flowing therethrough creates a vortex which will develop a pressure which opposes the input pressure at the inlet to the pump.

A third factor tending to produce cavitation which frequently exists at the inlet to a centrifugal pump and requires pressure to overcome it is the effect of a jet of liquid returning from an outlet of the pump to the inlet through a seal, usually of labyrinth type, between the impeller body or shroud and the housing of the pump. Leakage through a seal of this type usually discharges in a direction countercurrent to the incoming stream, or at least discharges radially inward at right angles to the direction of flow of the stream. Usually leakage of this type makes up a sizable fraction of the total flow induced by the pump and may amount to several percent of the flow. The leakage stream is usually under high pressure and thus causes turbulence which must be overcome by pressure at the inlet of the pump.

In many applications of centrifugal pumps, the pressure available at the inlet of the pump is a limiting factor

in pump design, especially in pumps for the handling of boiling liquids or in the removal of liquids from zones of low absolute pressure. One method which has been used for securing sufficient pressure upon liquid entering the inlet of a centrifugal pump is to use a small supercharger pump developing low head which discharges into the inlet of a larger pump. Another known method is to use a jet pump inducing liquid into the inlet of a centrifugal pump. In still another known method, pressure is increased sufficiently at the inlet of a centrifugal pump merely by submerging the pump to such depth that the required minimum pressure is available as liquid head at the inlet of the pump.

All these methods for securing sufficient pressure in the inlet of a centrifugal pump have disadvantages which are very marked under many circumstances. For example, it may well be impossible to submerge a pump in liquid to a depth sufficient to get the required pressure from liquid head; and the use of auxiliary pumps frequently may require space which is not available. In any case, the provision of auxiliary pumps is expensive, and their use involves additional repair and operating expense.

It is an object of the present invention to provide a centrifugal pump which will operate satisfactorily at very low inlet pressures as compared to the total head developed by the pump.

Another object is to provide a centrifugal pump which is especially useful in pumping a liquid which is substantially at its boiling point under very low pressure at the inlet of the pump.

Another object is to provide a pump of the above type which will operate at low absolute pressures.

Another object is to provide a pump of the above type which will not cavitate and lose capacity and efficiency at low inlet pressure.

Another object is to provide a pump of this type which has long passages through the impeller between the blades.

Still another object is to provide an improved pumping system for absorption-refrigeration systems capable of pumping refrigerant and absorbent at very low absolute pressures.

Another object is to provide a pump of the above type in which the inlet pressure need be only one percent or less of the total head developed by the pump.

Another object is to provide an improved pumping system for an absorption-refrigeration system in which the pumps will not cut off sharply at a minimum total net pressure at the inlet of the pump.

Another object is to provide an improved pumping system for absorption-refrigeration systems in which pumping of refrigerant and absorbent may be carried out at very low absolute pressure with the refrigerant substantially at its boiling point.

Still another object is to provide an improved pumping system for absorption-refrigeration systems in which multiple pumps of the above type provide circulation of boiling refrigerant and of absorbent. Still a further object is to provide for such pumps so arranged to be operated by a single drive motor, and bearings are cooled and lubricated by a portion of the pumped absorbent stream.

In the present invention, a pump is provided which has means for minimizing back pressure at the inlet section, including impeller blades arranged so that there is an impact on the liquid which generates a minimum force on liquid entering the pump, which force is in opposition to the inlet pressure. The blades are so located and shaped that impact on the liquid is minimized as stated

above, and the forces imposed on the liquid are substantially rotational forces and forces directed to the outlet and to minimize the forces directed to the inlet. The blades are so arranged as to take advantage of this outward impact to cause it to create a head on the liquid which is in the direction of the discharge of the pump instead of in the reverse direction.

As a result of the rotation of the liquid induced by the rotating impeller, a parasitic rotation of the liquid ahead of the eye is produced known as pre-rotation. This pre-rotation may have the effect of increasing the back pressure which opposes the inlet pressure. As an additional feature which aids in reducing the required minimum pressure at the inlet to the pump, the pump of the present invention has means for minimizing the pre-rotation of the liquid in the inlet. Radial vanes attached at their outer edges to the interior of the inlet section may be placed in this section to reduce pre-rotation.

As an additional feature in order to reduce the back pressure on the inlet ensuing from pre-rotation, I so form the geometry of the inlet as to avoid decreasing the radii of curvature of the rotational component of flow of the elements of the liquid flowing through the inlet to the eye of the impeller. Thus, where the inlet is conical decreasing in diameter toward the eye, or any other geometry with decreasing radius, the radius of curvature decreases and the rotational kinetic energy of the rotating fluid increases at the expense of its pressure. This requires a higher inlet pressure to compensate for this loss of pressure.

I, however, may employ an inlet of such geometry as to either maintain or increase the radius of curvature of the rotational component of the elements in their flow in the inlet toward the eye of the pump impeller. This I may accomplish by employing an inlet section of increasing cross-sectional area in the direction of flow or one of constant, for example, a cylindrical inlet.

In pumps employing imperfect seals between the impeller discharge and the inlet, objectional jetting action of liquid leaking through the seal may produce an undesirable back pressure which opposes the inlet pressure. It is an additional feature of the pump of my invention to reduce this effect by locating the labyrinth seal as far from the point of junction of the leakage stream and inlet stream in the pump inlet as is practical and providing an enlarged passageway from the seal to the inlet of the pump preferably having its terminal portion directed into the pump toward the impeller, so that liquid flowing along this passage will have lost much of its velocity due to the enlarged passage and does not cut into the entering stream at a right angle or at an angle to oppose entrance of the stream but is directed inward at an oblique angle so that its effect is to further increase pressure on the liquid in the inlet of the pump.

Each of the above features has the effect of reducing the input head required for the operation of the pump, and they may be employed serially or in any combination, or they may all be employed to minimize the required magnitude of the input head.

The preferred arrangements of pump parts can be best understood by reference to the folowing detailed description and the attached drawings wherein:

FIG. 1 is a cross-section through one preferred type of pump constructed according to principles of the present invention;

FIG. 2 is a plan of impeller blades of FIG. 1 as seen from the inlet of the pump at the line 2—2 of FIG. 1;

FIG. 3 is a section through a simple pump including principles of the present invention;

FIG. 4 is a section through a modification of the pump of FIG. 1 having a drive shaft through the inlet and increased length of impeller blades;

FIG. 5 is a cross-section through the impeller of the pump of FIG. 4 taken in the line 5—5; and

FIG. 6 is a section through an outside sump of an absorption-refrigeration system showing two pumps of the present invention in a preferred arrangement;

FIG. 7 is a vertical section through a modified form of my invention;

FIG. 8 is a section taken in line 8—8 of FIG. 7;

FIG. 9 is a vertical section through another form of my invention.

In one preferred type of pump of this invention, illustrated in FIG. 1, the pump comprises a case or housing **6** having a peripheral volute section **7** connected to an outlet **9**, and having a central inlet **8**. An impeller, designated generally as **10**, is disposed coaxially with inlet **8** in the housing and is driven by rotation of shaft **11**. The housing **6** has an inlet section having nonconverging walls, illustrated as cylindrical walls **12**, of sufficient length to prevent rotation of liquid due to frictional drag of rotating impeller parts extending to incoming liquid outside the nonconverging walls **12**. Flat radial vanes **13**, attached to walls **12**, may be used in the inlet if desired, but usually are not necessary to prevent pre-rotation of incoming liquid if the nonconverging inlet section is of sufficient length to extend beyond the zone of pre-rotation.

The impeller **10** has a cylindrical shroud portion **14** sealed to the pump casing by a labyrinth seal **15**. This labyrinth seal is exposed on one side to high pressure in volute section **7** through clearance **16** between the walls of impeller **10** and the casing wall and the other side of the seal communicates with the inlet **8** through an enlarged passageway **17** preferably having sufficient length and cross-section so as to reduce the momentum of the liquid entering from the passage to a degree such that on entering into the mainstream in the inlet section **8**, no substantial back pressure is created. An additional feature is the provision of a terminal section **18** arranged to direct a leakage stream flowing through labyrinth seal **15** into the pump at an angle in a direction toward the volute section **7**.

A plurality of spiral impeller blades, illustrated as two blades, **19** and **19***a*, or one of them alone, are attached to the inner wall of shroud portion **14**. The shape of these spiral impeller blades at the inlet may be best appreciated from consideration of FIG. 2. The spiral blades **19** and **19***a* begin at points **21** and **21***a* at almost zero width and increase in breadth as the blades spiral inward into the body of impeller **10**. The leading edges of these blades are not square, but are sharp as shown at **20** (FIG. 1); and as the spiral extends deeper into the pump, the spiral blades twist in a smooth curve through a sufficient fractional or multiple revolution so as to produce the directional forces described above. As illustrated in FIGS. 1 and 2, sections **19***b* of the blades are substantially perpendicular to the plane of the blades at **21** and **21***a*.

Sections **19***b* have the generally involute shape of blades **49***a* and **50***c*, best shown in FIGS. 4 and 5. The spiral blades thus are in effect a continuation of the involute impeller blades into the pump inlet. The pitch of the spiral blades is not constant but increases as the spiral extends into the impeller body.

The spiral blades **19** and **19***a* may be integral throughout their length or may be constructed in sections. The blade **19** is illustrated as an integral blade of this type, while blade **19***a* is illustrated as constructed in sections abutting each other at the line **22**. However, when the sectional construction is used, it is preferred that the blade sections abut against each other so as to form a smooth, uninterrupted spiral surface, forcing the liquid outward into the volute section **7**. The sectional construction usually will be preferred because the entire impeller body and sections **19***b* of the impeller blades can be cast in one foundry operation, thus providing savings in cost.

The operation of a pump of this type is obvious. The pressure on liquid entering inlet **8** must be great enough

to induce flow through the inlet **8** at a rate equivalent to the capacity of the pump, as otherwise the liquid could not enter the pump at all. The head necessary to cause liquid to flow into the pump at this rate is very small, and may, for example, be considerably less than one percent of the total head developed by the pump.

The three principal causes of back pressure inducing cavitation in a centrifugal pump have been eliminated in a pump of this design. The length of the inlet section having nonconverging walls, and the radial vanes in this section if the latter are used, prevents the conversion of rotational energy of the rotating fluid in the inlet section **31** into a back pressure. It will be seen that liquid entering the inlet of this pump encounters the rotating spiral blades and that these blades "screw onto" the stream in a gradual way such that turbulence at the inlet of the pump is minimized. The sharp edges of the impeller blades cut into the liquid without imposing on the liquid a substantial component of force directed toward the inlet in opposition to the inlet pressure, and the force applied by the blades upon the liquid is in a tangential direction causing the liquid to rotate and develop centrifugal force tending to move the liquid outward. The inner face of the next oncoming blade then encounters the liquid, and the more centrifugal force is developed, the greater the outward pressure will be. This outward pressure aids in urging the liquid further into the impeller and in a direction generally toward the volute section and outlet **9**.

The leakage stream from liquid under high pressure in volute section **7** flowing through labyrinth seal **15** loses velocity in the enlarged passageway **17**, and the resulting stream at low velocity is discharged into the inlet in such direction that its momentum carries it into the pump toward the volute section and substantially no back pressure is developed.

The simpler modification shown in FIG. 3 also applies the impact and frictional forces between the liquid flowing into the inlet of the pump and the impeller blades in a manner to aid in moving the liquid outward into a volute section of the pump. This pump comprises a housing **30** having a nonconverging inlet section **31** and a volute section **32**. Within the housing, an impeller, including a plate **33** having a smooth surface facing the inlet, driven by a shaft **34**, and carrying a plurality of blades **35**, is disposed concentrically with the inlet. The blades **35** are offset from the inlet; that is, the inner edges **35***a* of impeller blades **35** are arranged at the inlet to the eye of the impeller on a circle of greater diameter than the diameter of inlet **31**. The blades **35** are curved in any conventional shape as shown by the outer terminal portion of blades **49***a* and **50***c* shown in FIG. 5. The offset position of the inner edges **35***a* of blades **35** results in substantially all impact and frictional forces between the blades and liquid entering the inlet **31** being in a tangental direction, thus tending to cause rotation of the incoming liquid in body section **36**, which is tapering outward to the volute section. The smooth inner surface of plate **33** offers little friction to the liquid, and such friction as develops aids in creating centrifugal force which forces the liquid outward in body section **36** where it will come in contact with rotating blades **35**. It will be noted that this impeller has no shroud section, and consequently no labyrinth seal in required. The problems due to turbulence of a leakage stream entering the inlet thus do not exist in a pump of this type.

As shown in FIG. 3, the edges **35***a* of the blade **35** are each parallel to the axis of the impeller eye and the inlet **31** to the housing. The blade edges terminate on an imaginary cylinder of diameter greater than the diameter of the inlet. The ends of the edges of the blades, which edges are positioned closest to the plate **33**, i.e., the end remote from the inlet, may, however, be disposed so that the ends of the edges of the blades remote from the inlet are placed nearer the axis than are the ends nearest the outlet. The part of the edges more remote from the inlet may be positioned on an imaginary circle whose diameter is less than the diameter of the inlet to the eye of impeller. The ends of the edges nearest the inlet are preferably placed on an imaginary circle whose diameter is greater than the inlet to the eye of the impeller.

FIGS. 7 and FIG. 8 illustrate one such configuration. The pump of FIG. 7 and FIG. 8 is the same as FIG. 3, except for the modification of the blades and the provision of the seal such as is described in connection with FIG. 1. All like parts in FIG. 7 and FIG. 8 have the same numbers as in FIG. 3, and the seal parts bear numbers of like parts in FIG. 1. It will be observed that the edges of the blades **35** are cut at an angle to the axis of the eye of the impeller, to give an angularly positioned surface.

The blade ends **35***b* nearest the inlet, i.e., farthrest from the plate **33**, are positioned on a circle whose diameter is greater than the diameter of the inlet. The ends **35***c* are positioned on a circle whose diameter is less than the diameter of the circle on which the ends **35***b* are positioned. Thus, this diameter may be as shown in FIG. 7, less than the diameter of the inlet **31**. The surface of **35'** may be faired from the edge **33***a* at the concave side into the convex surface of the blade.

The blade thus has a leading edge which meets the liquid flow at an angle to the direction of flow of the liquid, slicing into the liquid, and directs the flow toward the outlet with a minimum thrust backward toward the inlet.

FIG. 9 shows the structure of FIG. 3 to which the seal employed in FIG. 1 has been added. Like parts on FIG. 9 bear the same number as like parts on FIGS. 1 and 2.

The modification shown in FIG. 4 is particularly useful when pumping liquid in such locations that only a very low liquid head can be applied to liquid flowing into the inlet of the pump.

In FIG. 4, the pump illustrated has a central drive shaft extending through the inlet. This pump comprises a housing **41** having a cylindrical inlet section **42**. An impeller, designated generally as **43**, is disposed within housing **41** to be driven by shaft **44**. The impeller has an elongated shroud section **45**, concentric with inlet **42** and drive shaft **44**, sealed to housing **41** by a labyrinth seal **46**. A large passageway **47** betwen the interior of housing **41** and the exterior of shroud **45** leads from one side of the labyrinth seal **46** to the inlet section of the pump and has a terminal section **47** directed into the inlet of the pump. The opposite side of the labyrinth seal is of course exposed to outlet pressure from a peripheral volute, shown fragmentarily at **48**.

The impeller includes a pair of spiral blades **49** and **50** of which the latter has the greater number of turns. In the prefererd construction illustrated, blade **50** is made up of three sections **50***a*, **50***b* and **50***c*. The section **50***a* begins nearest the inlet of the pump at a substantially zero width; and after a full turn or so, increases to a width substantially equal to clearance between the interior of shroud **45** and the exterior of a sleeve **51** to which section **50***b* is attached. The inner edges of section **50***a* are sharp and this section preferably is attached, as by integrally casting with or welding to the shroud **45**. The wider end of section **50***a* preferably abuts a corresponding end of section **50***b* to form a smooth spiral surrounding sleeve **51**. In the last turn of this spiral, the blade preferably is twisted so that the perpendicular face of the end section of blade **50***c* assumes the position of the customary involute blades of an impeller, similar to the sections **19***b* shown in FIG. 6. The end sections **50***c* and **49***a* of the two blades preferably are constructed separate from the middle section of the blades in order to simplify foundry practice.

It is preferred to mount blades **49** and **50** on a sleeve attached to and rotating with shaft **44** rather than on the shaft itself when pressure on liquid flowing into inlet **42** amounts to only two or three inches of head of that

liquid. This construction permits the use of an inner sleeve 52 which is stationary and eliminates the necessity for seals between the shaft and a rotating part of the impeller. Sleeve 52 extends up within sleeve 51 as indicated by the dotted lines to a distance which is normally above the level that would be attained by liquid flowing into the inlet due to pressure thereon. The junction of blades 50*a* and 50*b* occurs below this level, and the blade section 50*b* is wide enough at this point to substantially fill the annular space between sleeve 51 and the inner wall of the shroud 45.

Thus, it will be seen that rotation of the spiral blade of the pump creates substantially no back pressure at the inlet because of its geometry, as described above. Impact and friction of blade 50*a* upon the liquid causes the liquid to swirl around inner sleeve 52, and the centrifugal force tends to move the liquid outward to where it is given a stronger inward impulse by the next revolution of the blade. At the point where the increasing width of tapering blade section 50*b* fills the annular space between sleeve 51 and the shroud 45, the liquid is trapped and forced by rotation of the spiral blade into the space before the involute continuation of the spirals 49*a* and 50*c*. As blade 50 rises upon sleeve 51, the pitch increases to give room for the beginning of blade 49 which also ends in an involute section 49*a*.

In the configurations shown above, I have described pumps which embody a number of features which minimize the inlet pressure which is required for the operation of the pump.

Of these features, the most important is the means provided whereby the liquid is given a rotation, but the component of the liquid rotation which generates a force which is directed in opposition to the input pressure is minimized. The preferred means is to employ an impeller whose blades are designed of such geometry so that the blade slices the rotating liquid to generate a rotation of the liquid in which substantially the entire energy imparted to the liquid by the impeller blades is centifugal in nature and is directed toward the outlet, and the component of this rotational energy directed to the inlet is but a minor fraction of thet total energy or is entirely absent.

The difficulties solved by the above features of my invention constitute an avoidance of a major proportion of the parasitic pressures generated at the inlet. Since this results in the avoidance of the major proportion of the pressure drop experienced at the inlet by the incoming liquid, in its travel to the eye of the impeller, the use of the impeller design described above may by itself satisfactorily solve the problem created by parasitic pressure drop.

An additional reduction in the parasitic pressure drop is the design of the seal leakage path described above. Acting by itself, it will reduce the pressure drop which otherwise is inherent where leaky seals are employed which jet into the liquid at the inlet. Thus, the design of the leakage path of the seal of my invention acts to eliminate one of the serious sources of back pressure at the inlet.

Where pre-rotational parasitic pressures resulting from the component of the rotational energy of pre-rotation directed in opposition to the pressure of the incoming liquid is substantial, I may add the feature of preventing the decrease in the radius of rotation of the incoming fluid by means of the geometry of the inlet passageway to hold the cross-section of the inlet either constant or of increasing magnitude in direction of travel. This expedient has value, irrespective of the design of the impeller blades, to reduce the pressure drop in the inlet. However, when used with the design of the blades described above, it will act conjointly with the impeller design to reduce the pressure necessary at the inlet for proper operation of the pump.

These features become of great significance where the available inlet pressures are very small. In such systems the pressure losses due to the parasitic pressure drop described above may be so large that the inlet pressure becomes insufficient for the operation of the pump. A particularly significant service where minimizing of parasitic pressure drop is important is where the liquids pumped are at their boiling point, when pure components, or at their bubble point, when multiple component liquids. A very small pressure drop thus results in a partial vaporization of the liquid. Since the vapor volume may be many times the liquid volume, such vaporization may result in vapor bound pump or a pump of very low efficiency and capacity.

The pump 63, shown in FIG. 6, is particularly adapted to service when the liquid pump is at boiling or bubble point. Such service is in an absorption refrigeration system described in my copending application Serial No. 157,170, filed December 5, 1961, now Patent No. 3,146,602 which is herewith incorporated by this reference. In such case the pumps transfer the liquids employed in the system, as described below.

Pumps of this type have been found to be highly efficient for pumping water at its boiling point. At an inlet pressure of only two inches of water, a pump of this type has developed pressure equivalent to a head of 40 feet of water without cavitation.

In one preferred arrangement of pumps of the present invention for recirculating liquid refrigerant and liquid absorbent in absorbent-refrigerant systems, two pumps are mounted on a shaft driven by a motor in a hermetically sealed housing as shown in FIG. 6. This assembly comprises a hermetically sealed housing 60 which is connected to an absorber section (not shown) of an absorption-refrigeration system by a line 61 in such manner that the housing 60 serves as an outside sump for the absorber section. Spent absorbent draining from the absorber section through line 61 accumulates in a lower part of the housing 60 so that the liquid level therein is approximately at the level indicated by the line 62. A plurality of pumps constructed substantially according to FIG. 1, 3 or 4, illustrated as two pumps designated generally as 63 and 64 similar in construction to FIGS. 1 and 3, respectively, are centrally mounted in the housing to be driven by a single shaft 65. Shaft 65 is driven in turn by a motor designated generally as 66. The motor, shaft and pumps are supported in a suitable supporting member, designated generally as 67.

Spent or partially spent absorbent entering the lower part of the housing by line 61 flows downward through downcomer 68 into a chamber 69 in the supporting member 67 which communicates with the inlet of pump 63. A series of flat radial vanes 70 is arranged at the inlet of pump 63 to prevent liquid from swirling as it enters the pump.

Pump 63 has spiral blades according to the construction shown in FIG. 1, has a central drive shaft extending through the pump inlet, and has a modified volute section 71. A section 72 of chamber 69 containing the vanes 70 to prevent swirling of liquid entering the pump serves as a side inlet to the pump. A labyrinth seal 73 between the shroud section of the impeller and the pump housing discharges a leakage stream obliquely downward and centrally into the pump.

The outlet from volute section 71 comprises a space beneath a baffle plate 74, subdivided by a plurality of vanes or ribs 74*a* to prevent swirling of liquid in the space below the baffle.

Pump 64 is essentially the pump of FIG. 3, modified to allow for the presence of a central drive shaft. This pump comprises a cylindrical inlet section 75, having vanes 76 therein to prevent swirling of liquid in the inlet. Blades 77 are of involute shape and are set back from the path of flow of fluid from inlet 75 into impeller chamber

78. Blades 77 are mounted on a plate 79 carried by a sleeve 80 attached to shaft 65 by pin 81.

Pump 64 is arranged to receive refrigerant at a temperature substantially at its boiling point through an inlet line 83 and to discharge the refrigerant under higher pressure through line 84. The inlet and outlet openings 83 and 84 pass through a gasket 82 carried by the supporting member 67 and communicate respectively with passageways 83*a* and 84*a* in the supporting member. The connection through this gasket is hermetically sealed around the inlet and outlet passageways by pressure of gasket 82 against the inner wall of housing 60. This pressure is applied by a screw 85 extending through a wall of housing 60.

This construction is preferred so that, when a head 86 is removed, the entire pimp and motor assembly can be lifted out of housing 60 by loosening a single screw 85 at any time repair or replacement of a part is required.

Both upper and lower bearings of shaft 81 are cooled and lubricated with spent absorbent. This is accomplished by locating the lower bearing 87 where its upper side is submerged in spent absorbent admitted through opening 88 in support 67 and the lower side is exposed to absorbent in chamber 69 and by taking a side stream off the outlet section from pump 63 for lubricating and cooling the upper bearing.

This side stream is taken off by an upright line 89 communicating with the volute section 71 of pump 63 and with a settling chamber 90. Spent absorbent passes up through line 89 into settling chamber 90, and any sediment contained therein is returned to the volute section 71 by a small sediment-return opening 91. Clean, spent absorbent then passes under pressure through line 92 to a cooling coil 93 arranged to cool the stator of motor 66 and is returned through line 94 to a location within the supporting member 67 above the upper bearing 95. Spent lubrication thus moves down through bearing 95 around shaft 65 to a suitable drip pan 96 arranged to lead liquid coming down through the bearing to an opening 97 through the support member and so returns it to the pool of liquid in the bottom of housing 60. Any liquid not flowing downward through the bearing flows upward and overflows through drain 99′ into the pool of liquid in the bottom of 60. If, through a misoperation, liquid should enter the motor cavity, it would drain out through emergency drain 99.

The pressure within the housing 60, when used to pump liquids in the vacuum portion of a salt-water absorption refrigeration system, may, for example, be in the range from about .1 atmosphere to about .01 atmosphere, depending upon the particular refrigerant-absorbent pair used. At such pressures, the gas surrounding the motor will be water vapor, but water vapor has at this concentration approximately twice the electrical breakdown resistance of air, and is a favorable environment for an electric motor.

From the foregoing it will be seen that this invention is one well adapted to attain all of the ends and objects hereinabove set forth, together with other advantages which are obvious and which are inherent to the apparatus.

It will be understood that certain features and subcombinations are of utility and may be employed without reference to other features and subcombinations. This is contemplated by and is within the scope of the claims.

As many possible embodiments may be made of the invention without departing from the scope thereof, it is to be understood that all matter herein set forth or shown in the accompanying drawings is to be interpreted as illustrative and not in a limiting sense.

The invention having been described, what is claimed is:

1. An improved pumping system for an absorption refrigeration system comprising in combination, a hermetically sealed housing disposed to receive spent absorbent; a centrifugal pump disposed in said hermetically sealed housing, said pump including a central inlet and a peripheral outlet section; a pump impeller shaft, bearing for said shaft a rotary impeller substantially coaxial with said inlet mounted on said shaft; means for preventing rotation of liquid in the inlet, including an inlet section having nonconverging walls immediately adjacent to the impeller; and means for preventing turbulence in the inlet section, including impeller blades arranged to impart impact and frictional forces to the liquid in the pump substantially entirely in directions to move the liquid in radially outward and tangential directions; an electric drive motor disposed in the hermetically sealed housing, said electric motor including a stator and a rotor mounted on said pump impeller shaft; and means in the hermetically sealed housing for cooling and lubricating the bearing of the shaft by a portion of a pumped stream including a settling chamber communicating with the peripheral outlet section of the pump, a cooling coil in the stator of the drive motor a line arranged to withdraw a part of the pumped stream and to introduce the same into said cooling coil and to said bearing to be lubricated and cooled.

2. The system of claim 1 in which a plurality of centrifugal pumps each have an impeller mounted on the shaft; one of said pumps being disposed at a central location on the shaft and connected to pump a stream of refrigerant, and at least one pump disposed adjacent to a lower end of the shaft is connected to pump partially spent absorbent.

3. The system of claim 1, wherein the pumps are mounted in a supporting member in the hermetically sealed housing; inlet and outlet passageways to the centrally located pump are disposed in the supporting member in position to cooperate with inlet and outlet passageways through a wall of the housing; a sealing member is disposed in the housing around said inlet and outlet passageways; and a means for applying pressure to the supporting member, forcing said supporting member against said seal, is disposed in the side of the housing across said passageways.

4. A centrifugal pump comprising in combination a housing having a central inlet section and a peripheral outlet section, rotary impeller rotatably mounted in said housing the rotary impeller having a shroud section cylindrical adjacent to the inlet and divergent adjacent to the peripheral outlet section and tapered spiral blades having sharp leading edges and a terminal involute section are attached to the interior of said shroud section, said terminal involute section being in the divergent portion of the shroud section.

5. A centrifugal pump comprising in combination a housing having a central inlet in a peripheral outlet section, a rotary impeller having an impeller eye to said impeller and impeller blades, said impeller blades having a terminal involute section and an initial spiral section said rotary impeller comprising a shroud with a depending opening positioned in said central opening and a plurality of impeller blades positioned about said shroud opening, a rotary seal between said depending section of said shroud and said central opening, a passageway of substantial length positioned between said depending opening and said central opening, a passageway connecting said first-mentioned passageway and the interior of said central opening, said last-named passageway directing the fluid outlet from said last-named passageway in a direction toward said impeller blades.

6. In the pump of claim 5 said inlet section being axially aligned with said eye and having an end adjacent the said eye and a portion more remote from said eye, the cross-sectional area of the portion adjacent said eye being at least as great as the cross-sectional area of the portion of said axially aligned inlet section, more remote from said eye.

7. In the pump of claim 6 where said inlet section is cylindrical and of the diameter substantially equal to the diameter of said eye at said inlet.

8. In a centrifugal pump comprising, in combination, a housing having a central opening, a rotary impeller comprising a shroud with a depending opening positioned in said central opening, and a plurality of impeller blades positioned about said shroud opening at the eye of said impeller, a rotary seal between said depending section of said shroud and said central opening, a passageway of substantial length positioned between said depending opening and said central opening, a passageway connecting said first-mentioned passageway and the interior of said central opening, said last-named passageway directing the fluid outlet from said last-named passageway in a direction toward said impeller blades.

9. In the pump of claim 8, said inlet section being axially aligned with said eye and having an end adjacent the said eye and a portion more remote from said eye, the cross-sectional area of the portion adjacent said eye being at least as great as the inlet section more remote from said eye.

10. A centrifugal pump comprising in combination a housing having a central inlet and a peripheral outlet section, rotary impeller blades each having an inner edge positioned about an eye to said impeller, said eye having a cross-sectional area adjacent said inlet greater than the cross-sectional area of said inlet adjacent said eye, said rotary impeller comprising a shroud with a depending opening positioned in said central opening and a plurality of impeller blades positioned about said shroud opening, a rotary seal between said depending section of said shroud and said central opening, a passageway of substantial length positioned between said depending opening and said central opening, a passageway connecting said first-mentioned passageway and the interior of said central opening, said last-named passageway directing the fluid outlet from said last-named passageway in a direction toward said impeller blades.

11. In the pump of claim 10, where said inlet section is cylindircal and of the diameter substantially equal to the diameter of said eye at said inlet.

12. A centrifugal pump comprising in combination a housing having a central inlet and a peripheral outlet section, rotary impeller blades each having an inner edge positioned about an eye to said impeller, said eye having a cross-sectional area adjacent said inlet greater than the cross-sectional area of said inlet adjacent said eye, in which said edge adjacent said inlet is angled inwardly toward the axis of the eye in a direction away from the portion adjacent said inlet to said eye, said rotary impeller comprising a shroud with a depending opening positioned in said central opening and a plurality of impeller blades positioned about shroud opening, a rotary seal between said depending section of said shroud and said central opening, a passageway of substantial length positioned between said depending opening and said central opening, a passageway connecting said first-mentioned passageway and the interior of said central opening, said last-named passageway directing the fluid outlet from said last-named passageway in a direction toward said impeller blades.

13. In the pump of claim 12, where said inlet section is cylindrical and of the diameter substantially equal to the diameter of said eye at said inlet.

14. A centrifugal pump comprising in combination, a housing having a central inlet and a peripheral outlet section, rotary impeller blades each having an inner edge positioned about an eye to said impeller, said eye having a cross-sectional area adjacent said inlet greater than the cross-sectional area of said inlet adjacent said eye, and in which said edge is parallel to the axis of said eye, said rotary impeller comprising a shroud with a depending opening positioned in said central opening and a plurality of impeller blades positioned about said shroud opening, a rotary seal between said depending section of said shroud and said central opening, a passageway of substantial length positioned between said depending opening and said central opening, a passageway connecting said first-mentioned passageway and the interior of said central opening, said last-named passageway directing the fluid outlet from said last-named passageway in a direction toward said impeller blades.

15. In the pump of claim 14, where said inlet section is cylindrical and of the diameter substantially equal to the diameter of said eye at said inlet.

16. A centrifugal pump comprising in combination a housing having an elongate central inlet section and a peripheral outlet section; a rotary impeller including impeller blades and a shroud having a depending section which is positioned in and spaced from the wall of said inlet mounted in said housing forming a passageway, one end of said passageway communicating with said outlet section, said impeller having an inlet eye to the impeller said inlet section having non-converging walls and means minimizing back pressure directed toward the inlet section, said means including said impeller blades arranged to impart a rotation to the liquid in the pump to generate a force on said liquid which is substantially entirely centrifugal when directed toward said outlet, the component of rotation which is directed to said inlet being but a minor fraction of the force imparted to said liquid by said impeller blades, said passageway forming a seal between said wall of the inlet and said depending section of the shroud, and an outlet section for said passageway into the interior of said depending section at said inlet section, said outlet section from said passageway directed toward said impeller blades.

17. A centrifugal pump comprising a housing having an inlet section and a peripheral outlet section, an impeller rotatably mounted in said housing, said impeller having a shroud section, said shroud section being cylindrical adjacent to the inlet and divergent adjacent to the peripheral outlet section and a terminal involute section attached to the interior of said shroud section, said terminal involute section being in the divergent portion of said shroud section.

18. A centrifugal pump comprising in combination a housing having a peripheral outlet, a rotary impeller mounted for rotation in said housing, an impeller shroud, means for mounting said impeller shroud for rotation with said impeller, said shroud having a dependent section positioned in fluid communication with said opening and a passageway between said shroud and said housing and communicating with said outlet and a second passageway connecting said first-mentioned passageway and the interior of said central opening, means including said last-named passageway for directing fluid into said opening from said last-named passageway in a direction toward said impeller blades.

19. In combination with the pump of claim 18, a restriction in said first-mentioned passageway between said housing and said shroud.

20. The centrifugal pump of claim 18, a labyrinth seal in said first-named passageway.

21. A centrifugal pump comprising a housing having an inlet section and a peripheral outlet section a rotary impeller rotatably mounted at said inlet section and a shroud mounted for rotation with said impeller, said shroud being spaced from said housing, a passageway from said peripheral outlet section between said housing and a second passageway connecting said first-named passageway and said inlet section, said last-named passageway directing fluid into said inlet section in the direction toward said impeller blades.

22. In the centrifugal pump of claim 21, a restriction in said first-named passageway.

23. The centrifugal pump of claim 21, a labyrinth seal in said first-named passageway between said last named passageway and said outlet section.

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LAURENCE V. EFNER, *Primary Examiner.*

ROBERT M. WALKER, *Examiner.*