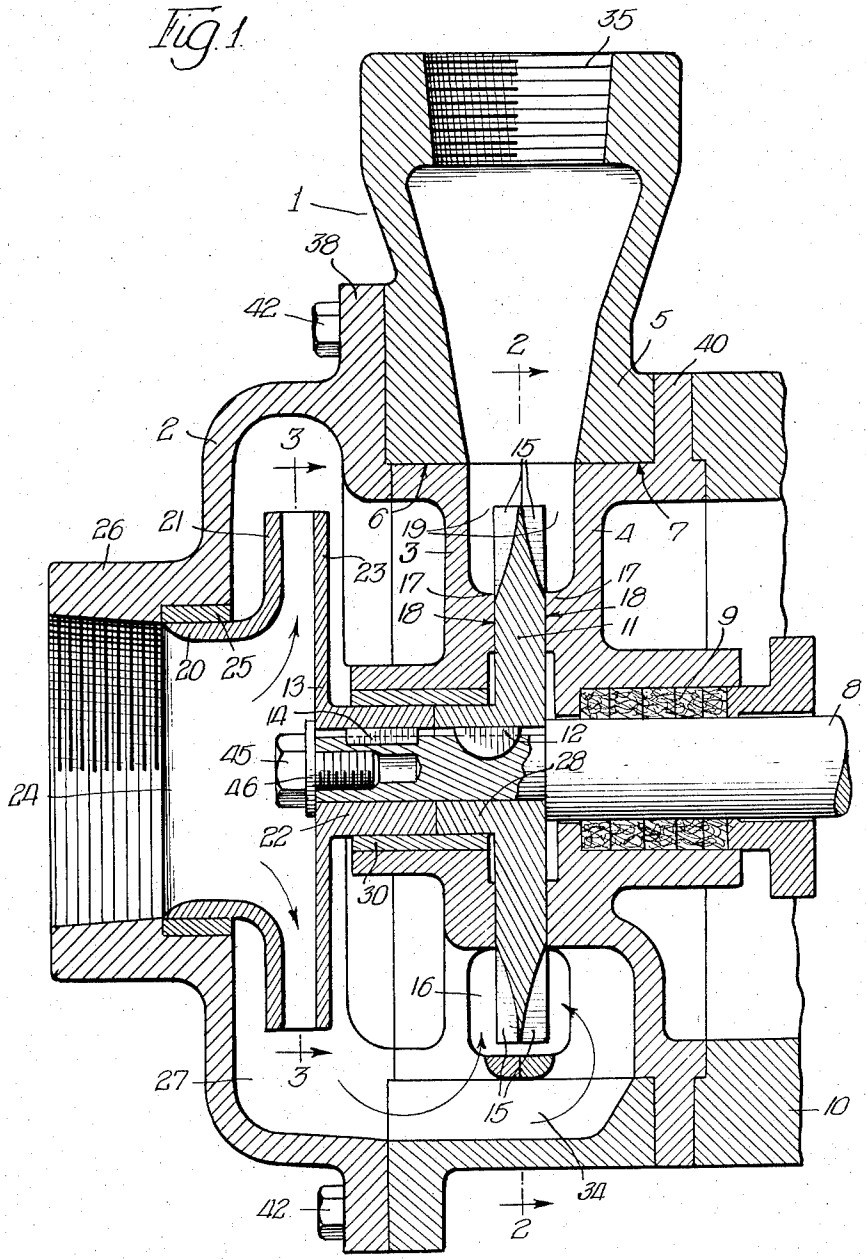
March 3, 1959 L. C. ROTH 2,875,698

COMBINATION CENTRIFUGAL-TURBINE PUMP

Filed March 16, 1955 2 Sheets-Sheet 1

Fig. 1



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Fig. 2

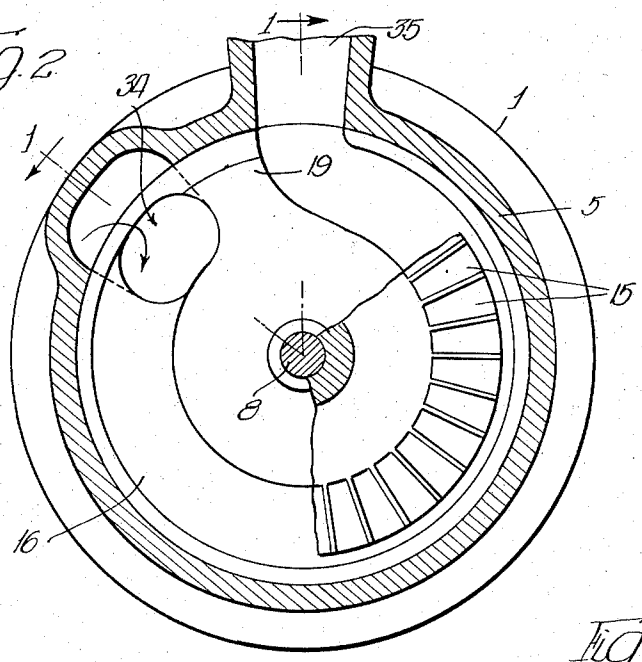


Fig. 3

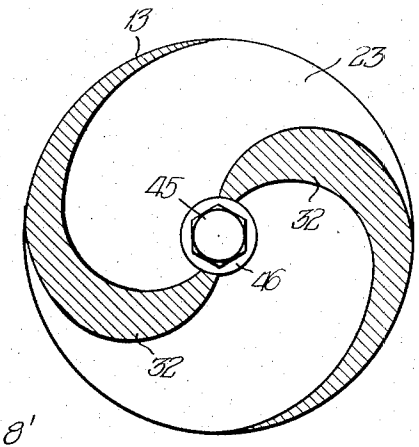
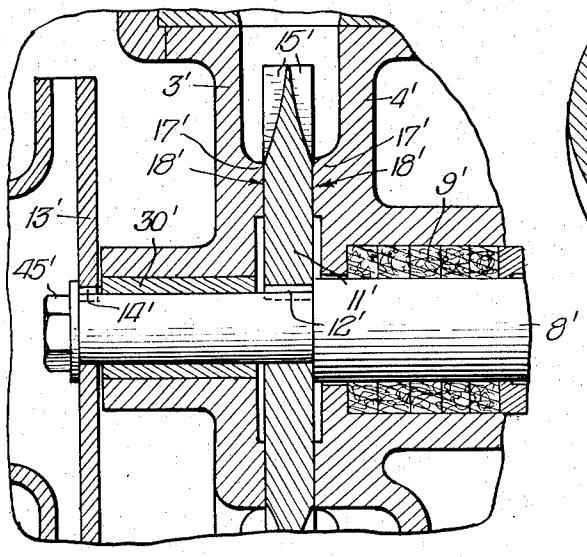


Fig. 4



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2,875,698

COMBINATION CENTRIFUGAL-TURBINE PUMP

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Application March 16, 1955, Serial No. 494,632
8 Claims. (Cl. 103—108)

This invention relates to an improved form of combination centrifugal-turbine pump, and is particularly concerned with improvements which impart to the pump the ability to pump liquids at or near the boiling point.

Propane is one of the many liquids with low boiling points. It has a boiling point of —40° F. and, at ordinary room or ambient temperatures, produces vapor pressures varying from 25 lbs. per sq. in. to 180 lbs. per sq. in. Anhydrous ammonia also boils at temperatures below normal ambient temperatures and is commonly employed in refrigeration systems where it is deliberately boiled for refrigeration.

This invention is concerned particularly with pumping the above mentioned liquids into storage and transport tanks and with the filling of cylinders and the like.

The present invention is also concerned with the pumping of various other liquids handled at or near boiling point in refineries, gasoline plants, chemical and petrochemical plants. Among these are butadiene, styrene, propane, butane, ethane, ammonia, water and various water solutions. Such liquids may be at boiling point by virtue of temperature or vacuum conditions.

The invention is also concerned with pumping liquid refrigerants such as Freon, ammonia, methyl chloride, sulphur dioxide, and carbon dioxide in the refrigeration process.

The invention is also concerned with pumping low temperature liquefied gases such as liquid oxygen, nitrogen, and argon into storage and transport tanks and cylinders.

A liquid at the boiling point is in a state where slight increases in temperature or slight decreases in pressure create an active boiling which produces vapor. It is obvious that it is impossible to "lift" such a liquid from a lower level to the pump. However, it is not so obvious that, even with a static suction head, it is difficult to pump liquids in this state unless the suction heads are so high as to prevent boiling of the liquid within the pump. Under many conditions these suction heads must be so great as to be impractical.

The turbine pump consisting of a vaned impeller and annular passage with a peripheral suction entrance and vertical discharge outlet has been found widely satisfactory for such applications with liquid suction heads of 2 to 6 feet when operating at 1750 R. P. M., and suction heads of 4 to 12 feet when operating at 3500 R. P. M. The range given at each speed is due to the individual characteristics of specific models.

Basic turbine pump design is such that the impeller can be located to run free of contact with the side plates and water block. Also due to the repeated impulses provided by the vanes of the impeller there is a gradual pressure build-up which counteracts any increased vapor pressures caused by efficiency losses. This has resulted in the wide use of this design for high pressure handling of non-lubricating liquids.

However, below the minimum suction heads indicated above, there is a phenomena encountered peculiar to turbine pumps. Partly due to suction entrance pressure losses and partly due to internal by-pass of high pressure liquid a gradual increase in vapor building up occurs with the progressive lowering of the suction head until at heads of 1 to 3 feet at 1750 R. P. M., and 2 to 6 feet at 3500 R. P. M. a complete vapor lock occurs within the pump.

Many instances are found in industry where it is essential to pump liquids at boiling point with suction heads of 1 foot or less. At present, standard designs, turbine or otherwise, cannot fill this need. Previous designs involving use of rotary gear or rotary vane pumps in conjunction with centrifugal stage are not suitable for non-lubricating liquids and have a limited life in all liquids because of the metallic contact of the gears or vanes.

It is also desirable to provide pumping equipment to handle boiling liquids with 1 foot suction head that can operate at conventional speeds for electric motors—namely 1750 and 3500 R. P. M. permitting direct electric motor drives.

Accordingly, the present invention provides a special mechanical device introducing preliminary to the turbine pump and running at the same speed which will provide a mechanical equivalent to the minimum liquid suction heads previously indicated.

In a turbine pump there are two principal factors which cause the active boiling of the liquid and produce sufficient vapor to make the pump "vapor bind" or stop pumping. One is a condition in which the pumped liquid, such as a refrigerant, is at a very low temperature and is still at the boiling point. In this case, the heat transferred through the pump casing from the ambient temperature outside causes the liquid to boil and produces vapor, even when the pump is at rest. Another factor is that the regenerative turbine action of the turbine pump adds heat to the liquid, and some of this superheated liquid is carried over into the pump suction, where it flashes into a vapor, "vapor binding" the pump.

The cause of "vapor binding" is therefore the presence or production of vapor in the liquid in the pump, but this does not completely explain what takes place in a turbine pump when it "vapor binds." Because its pumping elements consist of a series of buckets and vanes in the periphery or in the face or faces of the impeller, which elements rotate in an annular channel or raceway interrupted by a "liquid block" or stop, and because these rotating elements are travelling at relatively high peripheral speeds, the lower inertia of the vapor allows it to fill the buckets between the vanes, displacing the liquid, and effectively preventing liquid from entering the buckets unless there is a positive force available at the suction of the pump (such as adequate suction head) to compress the vapor sufficiently to initiate the pumping action. Once the pumping action is initiated, the compression of the vapor proceeds rapidly, as the pump builds up pressure in the annular channel, until the vapor is condensed or delivered to the pump discharge.

Attempts have been made to improve turbine pumps in this respect by constructing the suction entrance of the pump in a way to allow the full weight of static suction head to force the liquid into the suction of the pump under sufficient pressure to prevent the pump from "vapor binding." In practice, however, improvements of this sort have not been adequate.

The present invention resides essentially in the combination of a centrifugal pump first stage and a turbine pump second stage, and utilizes the following characteristics of both types of pumps:

A. The centrifugal stage is used as the first stage, because: (1) Its pumping action is not regenerative, and the exterior of the impeller is smooth; therefore it does not add appreciable heat to the pumped liquid. (2) The centrifugal pump does not carry any liquid over from the high pressure discharge to the low pressure suction in the impeller elements themselves (as does a turbine impeller). (3) The centrifugal pump does not produce

high discharge pressures in a single stage, so that whatever leakage takes place between the discharge and suction areas is small in proportion to the capacity of the pump. (4) The centrifugal pump will handle large quantities of vapor providing there is a static suction head available. (5) The centrifugal pump is at least as efficient (horsepower consumption vs. horsepower output) as a turbine pump under all operating conditions.

B. Because of the above characteristics of the centrifugal pump and because it operates effectively at the same rotative speeds with similar impeller diameters as the turbine pump, the centrifugal pump stage delivers to the turbine pump stage liquid at sufficiently high discharge pressures to effectively prevent, under all reasonable conditions, "vapor binding" of the turbine stage.

C. The turbine pump is used as the second stage because: (1) Its regenerative pumping action produces high pressures in a single stage—10 to 20 times the pressure of a centrifugal impeller of the same diameter rotating at the same speed. (2) There being no metal-to-metal contact in the rotating parts of a turbine pump, a turbine pump can handle lubricating or non-lubricating liquids without wear. (3) The turbine pump operates best at the same rotative speeds as the centrifugal pump.

At least one form of pump with a centrifugal first stage and a turbine second stage, as previously proposed for a different purpose, would not be suitable for the purpose for which the present invention is provided, because it has a leakage patch between a relatively high pressure area of the turbine stage and the low pressure area in the suction entrance of the centrifugal stage of the pump. Even with a sealing surface along this leakage path, the normal wear occasioned by the presence of abrasive particles in the pumped liquid would soon open up such a sealing surface to the point where the superheated liquid from the turbine stage, which leaks across this sealing surface, would produce so much vapor that the centrifugal stage would "vapor bind" completely and stop the pumping action.

In the aforementioned prior form of pump the construction is also significantly different in that the centrifugal impeller and turbine impeller are arranged in a manner that would not suffice for the purpose of the present invention. The size of the suction entrance of the centrifugal impeller of such pump is limited by the inside diameter of a sealing surface inside the vanes and annular channel. This is a serious defect, because the suction entrance of a pump for the purpose for which the pump of the present invention is provided, should be unusually large to handle satisfactory volumes of liquid and vapor.

One of the main objects of the present invention is to provide an improved combination centrifugal-turbine pump in which there is coactive utilization of the characteristics of both types of pumps, and in which the arrangement is such that there can be no direct leakage between a relatively high pressure area of the turbine stage and the low pressure area in the suction entrance of the centrifugal stage of the pump.

Another object is to provide an arrangement of the centrifugal and turbine stages of the pump in which sealing surface means is provided for restraining leakage of the high pressure liquid from the turbine stage and wherein any liquid that does leak from the turbine stage, passes to an area of the centrifugal stage which is under the pressure of the centrifugal force generated there, which effectively prevents the liquid from flashing into vapor.

Another object is to provide an arrangement in which the only leakage that can occur into the suction entrance of the centrifugal first stage is from the discharge of this stage. Because the differential pressures generated in this stage are small relative to the capacity, and since the centrifugal impeller itself adds little heat to the liquid, the possibility of "vapor binding" this stage is minimal, and takes place only under extreme conditions not usually encountered in actual practice, such as attempting to operate the pump at terminal or shutoff pressure while handling a boiling liquid.

Another object is to provide an arrangement of the centrifugal and turbine stages of the pump which will not limit the size of the suction entrance of the centrifugal stage, but will permit the suction entrance of the centrifugal stage to be unusually large so as to handle satisfactory volumes of liquid and vapor.

Further objects and advantages of the invention will appear from the following detailed description, taken in connection with the accompanying drawings which show the construction and operation of an illustrative embodiment of the invention.

In the drawings:

Figure 1 is a longitudinal sectional view taken substantially along the lines 1—1 of Figure 2;

Figure 2 is a transverse sectional view taken along the line 2—2 of Figure 1;

Figure 3 is a transverse sectional view through the centrifugal impeller taken along the line 3—3 of Figure 1; and

Figure 4 is a fragmentary detailed section showing another sealing arrangement for sealing off relatively high pressure liquid at the center of the turbine stage from escaping into the lower pressure centrifugal stage.

Referring to the drawings for a more detailed description, the pump selected for illustration is a combination centrifugal-turbine pump having a pump casing designated in its entirety by the numeral 1. The casing 1 comprises a casing part 2, a pair of casing parts 3 and 4 and a sealing ring or annular casing 5 closely surrounding the cylindrical surfaces 6 and 7 of the casing parts 3 and 4.

The pump shaft 8 projects through the pump casing part 4, which contains a stuffing box 9 or other suitable sealing means; also through the casing part 3 and into the casing part 2. The pump shaft 8 is driven by any suitable or preferred prime mover (not shown), and the casing extension 10 may be provided with suitable bearings (not shown) for supporting the shaft 8, and, if desired, with a supporting base. A turbine impeller 11 is fixed, for example, by a key or spline 12 or the like to rotate with the shaft 8. A centrifugal impeller 13 coaxial with and spaced axially from the turbine impeller 11 is also fixed, for example, by a key or spline 14, or the like, to rotate with the shaft 8.

The turbine impeller 11 has at its outer periphery two sets of vanes or blades and buckets 15 cut into its opposite sides at the periphery thereof. One set of these vanes 15 opens from one side of the outer marginal portion of the impeller 11, and the other set of vanes 15 opens from the other side of the outer marginal portion of the impeller. Both sets of vanes open radially from the outer periphery of the impeller 11, and both sets of vanes operate in an annular liquid channel 16 during the rotation of the impeller 11.

The casing parts 3 and 4 have opposed generally annular ribs or flanges 17 between which the impeller 11 is accurately and closely spaced, or which abut or cooperate with the opposite sides of the impeller 11 to form sealing surfaces 18 which prevent most of the pressurized liquid from escaping from the discharge of the annular channel 16 radially inwardly between the casing parts 3 and 4 and the opposite sides of the impeller 11. The liquid channel 16 is interrupted by stop or block members 19 which may be formed integral with the casing parts 3 and 4 and prevent the pressurized liquid from escaping to the suction of the turbine stage of the pump from the discharge of the annular channel 16.

The centrifugal impeller 13 which, as already set forth, rotates with the shaft 8 and the turbine impeller 11 is preferably of the enclosed type as shown more

in detail in Figures 1 and 3. It has an annular hub **20** integral with its wall **21** and an annular hub **22** integral with its wall **23**. The hub **20** defines the suction inlet **24** of the centrifugal stage of the pump and with the arrangement described this suction inlet opening **24** is not limited in size but may be as large as desired. This is an important practical advantage, because the suction entrance of this type of pump should preferably be unusually large to handle satisfactory volumes of liquid and vapor.

The hub **20** rotates within a bushing **25** which lines the inner end of the opening in the hub **26** of the casing part **2**. The bushing **25** provides a sealing surface around the suction opening **24** to prevent the escape of liquid from the discharge of the volute channel **27** to the suction inlet **24** of the centrifugal stage of the pump. The hub **22** of the centrifugal impeller **13**, along with the hub **28** of the turbine impeller **11**, rotates within a bushing **30** which seals off relatively high pressure liquid at the center of the turbine stage from escaping into the lower pressure discharge of the centrifugal stage.

The construction for sealing the relatively high pressure liquid at the center of the turbine stage from escaping into the lower pressure discharge of the centrifugal stage may be altered in various ways within the scope of the present invention. For example, as shown in Figure 4, the shaft **8'** itself, which corresponds with the impeller shaft previously described, may project through a hole or bushing **30'** between the two stages to prevent the high pressure liquid from escaping to a lower pressure portion of the pump. Other parts of Figure 4, which are similar to parts shown in Figures 1–3, are designated by primed reference characters corresponding to those used in Figures 1–3.

The suction inlet **24** of the centrifugal stage opens into the spaces between the walls **21** and **23** and the enclosed blades **32** of the impeller **13** and these spaces open radially outwardly to the volute channel **27**. The volute channel **27** opens into the suction entrance **34** of the turbine stage of the pump. The discharge of the annular channel **16** of the turbine stage delivers the pressurized liquid to the pump outlet **35** which thus opens from the discharge of the annular channel **16**.

The casing part **5** which, as already set forth, surrounds or encircles the casing parts **3** and **4** fits between a flange **38** on the casing part **2** and a flange **40** on the casing part **4** and the casing parts are secured together, for example, by screws or bolts **42**. The impeller **13** is secured in place on the adjacent reduced diameter end of the shaft **8** by a nut **45** between which and the adjacent wall of the impeller **13** a washer **46** is interposed.

In the operation of the pump, the liquid enters the suction entrance **24** of the centrifugal impeller **13** and is discharged from the periphery of this impeller into the volute channel **27** at increased pressure and higher velocity. Under the pressure of the centrifugal force of the first stage the liquid passes through the suction entrance **34** into the annular liquid channel **16** of the turbine stage. The regenerative pumping action of the turbine impeller **11** builds up sufficient pressure to deliver the liquid to the outlet **35** of the turbine stage, by constant recirculation of the liquid through the impeller vanes and annular channel **16**.

Since the liquid in the suction entrance of the turbine second stage is always under the pressure of the centrifugal force from the centrifugal first stage, vaporization cannot occur, and this stage will not "vapor bind." Further, it has been found that the centrifugal type pump, when properly designed for handling liquids at or near the boiling point, and operating with a static suction head, will itself not "vapor bind" except at terminal or shutoff pressure, because, since there is no recirculation of the liquid and because of the smooth exterior of the impeller, very little heat is added to the pumped liquid. Since the second stage will not "vapor bind" if the first stage does not "vapor bind," and since under all pumping conditions the first stage will not, it is evident the complete device as described will not "vapor bind."

It will be apparent from the foregoing description that the combination centrifugal-turbine pump of the present invention coactively utilizes the characteristics of both types of pumps and that the arrangement is such that there can be no direct leakage between areas of sufficient differential pressure to "vapor bind" or stop the pump. The high pressure liquid from the turbine second stage is first restrained along radial paths by the sealing surfaces **18** and the opposite sides of the turbine impeller **11** inwardly of the vanes **15**. Whatever liquid may leak by the sealing surfaces **18** is restrained from leaking to the centrifugal stage by the sealing surface around the hubs **22** and **28** of the centrifugal and turbine impellers **13** and **11**, and between these hubs and the bushing **30** in Figure 1, or between the shaft **8'** and the bushing **30'** in Figure 4. Even then such liquid as may leak to the centrifugal stage is still under the discharge pressure of the centrifugal stage or the centrifugal force generated there, which effectively prevents the liquid from flashing into vapor.

The only leakage into the suction entrance **24** of the centrifugal first stage which can occur is from the discharge of the centrifugal stage across the sealing surface around the hub **20** of the centrifugal impeller **13** as provided by the bushing **25**. Because the differential pressures generated in the centrifugal stage are small relative to the capacity, and since the centrifugal impeller **13** itself adds little heat to the liquid, the possibility of "vapor binding" this stage is minimal, and, as previously set forth, takes place only under extreme conditions not usually encountered in actual practice, such as attempting to operate the pump at terminal or shutoff pressure while handling a boiling liquid.

The structural arrangement described does not limit the size of the suction entrance **24** of the centrifugal stage of the pump, but enables making this suction entrance **24** unusually large or, at least, as large as desired to handle satisfactory volumes of liquid and vapor for the purposes contemplated.

The use of the pump of the present invention is not limited to handling a particular liquid or particular class of liquids, but the pump may be employed for pumping any liquids at or near the boiling point.

The centrifugal impeller provided in the pump of the present invention is of special type of low volumetric efficiency and very low pressure creating characteristic. Since maximum discharge head of 10 feet at 1750 R. P. M., and 20 feet at 3500 R. P. M. is required all other considerations have been sacrificed to the need of pumping boiling liquid with 1 foot or less suction head.

To this end the special features of the centrifugal impeller are as follows:

(1) Disproportionately large suction entrance about 50% of the full impeller diameter.

(2) Shrouded impeller eye machined inside and outside to a very thin wall.

(3) Deep blades at the impeller eye parabolic with reference to the radius and helical with reference to the axis.

(4) A rapid but even volumetric closure of the impeller cavity at the periphery.

(5) Streamlining of all surfaces in contact with the liquid stream.

By the foregoing means I provide a centrifugal impeller of special design for developing heads up to 10 feet at 1750 R. P. M. and up to 20 feet at 3500 R. P. M. in capacities up to 200 G. P. M specifically for use in conjunction with the turbine impeller at the above speeds to take boiling liquids, volatile liquids or liquified gases with suction heads of 1 foot or less and produce discharge heads up to 800 feet per turbine stage.

The claims of the present application are not limited

to a combination centrifugal-turbine pump with a centrifugal stage and a single turbine stage but are of a scope to cover a centrifugal-turbine combination with two or more turbine stages arranged in series (or tandem).

The embodiments of the invention shown in the drawings are for illustrative purposes only, and it is to be expressly understood that the drawings and the accompanying specification are not to be construed as a definition of the limits or scope of the invention, reference being had to the appended claims for that purpose.

I claim:

1. A centrifugal-turbine pump capable of pumping liquids and liquified gases at or near the boiling point thereof with suction heads not greater than about 1 foot, comprising a pump casing having a first pumping channel and a second pumping channel disposed laterally of said first pumping channel, a driving shaft disposed axially of said pumping channels, a high pressure regenerative turbine impeller fixed for rotation at all times and in the same direction with said driving shaft and having peripheral vanes forming liquid-propelling buckets in opposite sides of the periphery of said turbine impeller and opening peripherally therefrom, said second pumping channel extending generally annularly about and spaced outwardly from the periphery of said turbine impeller so that the liquid will enter the buckets at both sides of said turbine impeller and be circulated peripherally into and out of said buckets for almost a full revolution of said turbine impeller, said second pumping channel having a discharge outlet opening generally radially from one end thereof, said casing having block means separating the discharge outlet end of said second pumping channel annularly from said first pumping channel, said casing having a suction entrance for the turbine stage opening laterally from said first pumping channel to said other end of said second pumping channel, and a centrifugal impeller of low volumetric efficiency and low pressure creating characteristics fixed for rotation within said first pumping channel and at all times and in the same direction with said driving shaft and said high pressure regenerative turbine impeller, said centrifugal impeller having a large diameter axial suction inlet eye of substantially the same diameter as a suction inlet for the centrifugal stage defined by said casing and closed blades spaced inwardly from the periphery of said first pumping channel so that liquid entering said suction inlet eye will be thrown out by centrifugal force into said first pumping channel and pass through the suction entrance for the turbine stage into said other end of said second pumping channel to prevent vapor-formation within the pump and provide a mechanical equivalent to the minimum suction head at which the regenerative turbine impeller is operable without vapor binding.

2. In a rotary pump for pumping liquids and liquified gases at or near the boiling point thereof, a high pressure regenerative turbine impeller having peripheral vanes forming liquid-propelling buckets in opposite sides of the periphery thereof and opening peripherally from said turturbine impeller to produce a regenerative pumping action and high pressures, said turbine impeller having an annular wall separating said liquid-propelling buckets on one side of said impeller from said liquid-propelling buckets on the opposite side of said impeller, a centrifugal impeller of low volumetric efficiency and low pressure creating characteristics positioned ahead of said turbine impeller with respect to the direction of movement of the liquid through the pump and fixed to turn at all times in the same direction and at the same rotative speeds as said turbine impeller, a casing having a first pumping channel for said centrifugal impeller and provided with a large diameter axial suction inlet for the centrifugal impeller and a single peripheral outlet, said casing having a second pumping channel extending generally annularly about and spaced outwardly from the periphery of said turbine impeller so that the liquid will enter the buckets at both sides of said turbine impeller and be circulated peripherally into and out of said buckets for almost a full revolution of said turbine impeller, said second pumping channel having a peripheral suction entrance in communication with the single peripheral outlet of said first pumping channel and a single vertical discharge outlet, and block means separating said discharge outlet of said second pumping channel annularly from said peripheral suction entrance of said second pumping channel.

3. A rotary pump according to claim 2 wherein the block means separating the discharge outlet of said second pumping channel from the suction entrance of said second pumping channel is integral with the interior of said casing and extends generally radially inwardly therefrom between the discharge outlet and the peripheral suction entrance of said second pumping channel.

4. A rotary pump according to claim 2 wherein the centrifugal impeller has closed blades constructed parabolically with reference to the radius and spirally with reference to the axis thereof.

5. A rotary pump according to claim 2 wherein the turbine impeller and the centrifugal impeller have oppositely extending tubular hubs which abut one another and through which a driving shaft connected to turn said turbine and centrifugal impellers therewith extends with one end of said shaft terminating at about the inner surface of a radial wall forming a part of a closure for the blades of the centrifugal impeller, and a nut coacting with said radial wall of said centrifugal impeller and screwed axially into the adjacent end of said shaft for securing the centrifugal impeller in place on said shaft.

6. A rotary pump according to claim 2 wherein the axial suction inlet for the first pumping channel is about half the size of the rotor diameter of the centrifugal impeller.

7. A rotary pump according to claim 2 wherein there are annular ribs integral with the interior of the casing and cooperating with opposite sides of the turbine impeller radially inwardly of the liquid-propelling buckets in the periphery thereof for preventing liquid from moving radially inwardly of the turbine impeller.

8. A rotary pump according to claim 2 wherein the turbine impeller and the centrifugal impeller have oppositely extending tubular hubs which abut one another and through which a driving shaft connected to turn said turbine and centrifugal impellers therewith extends with one end of said shaft terminating at about the inner surface of a radial wall forming part of a closure for the blades of the centrifugal impeller, a nut coacting with said radial wall of said centrifugal impeller and screwed axially into the adjacent end of said shaft for securing the centrifugal impeller in place on said shaft, and an annular bushing surrounding the oppositely extending tubular hubs of the turbine and centrifugal impellers and cooperating with said casing to seal the first and second pumping channels from one another.

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