April 26, 1949. R. M. THAYER 2,468,246

ROTARY PUMP FOR LIQUEFIED GASES

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Fig. 1.

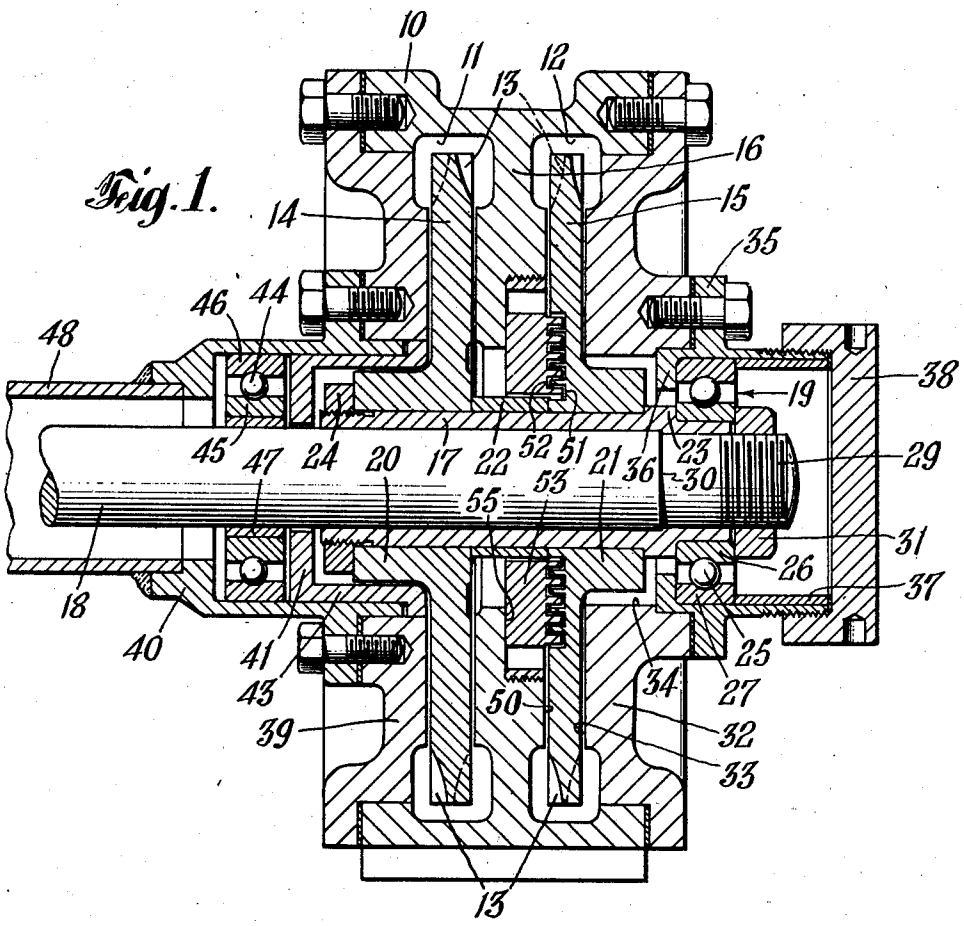
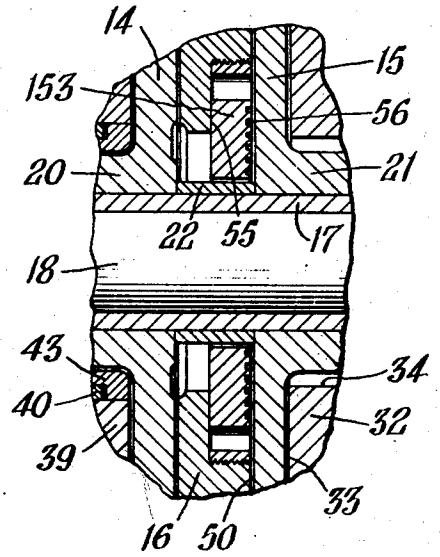


Fig. 2.



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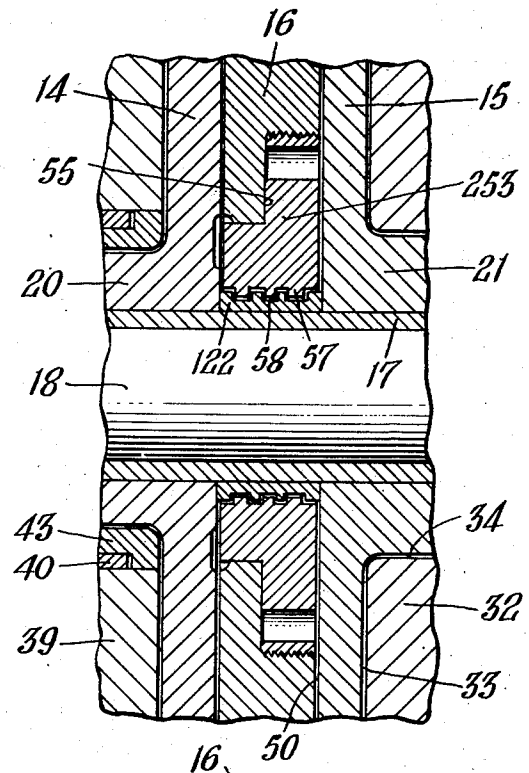


Fig. 3.

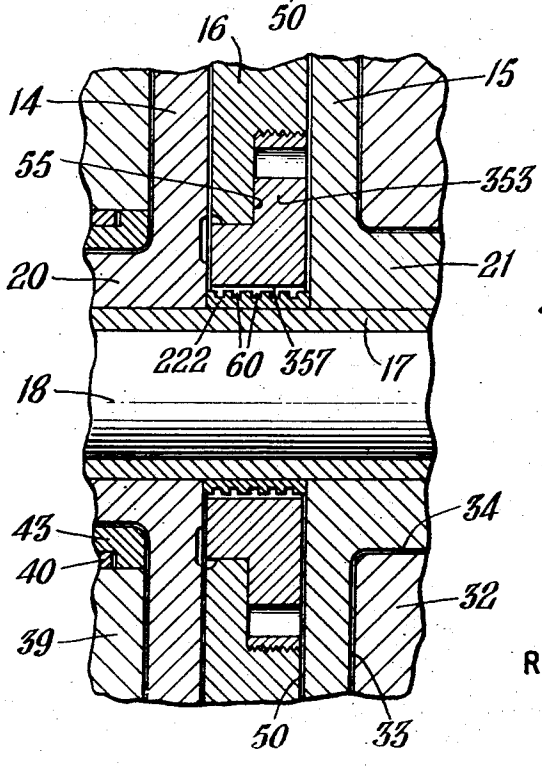


Fig. 4.

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ROTARY PUMP FOR LIQUEFIED GASES

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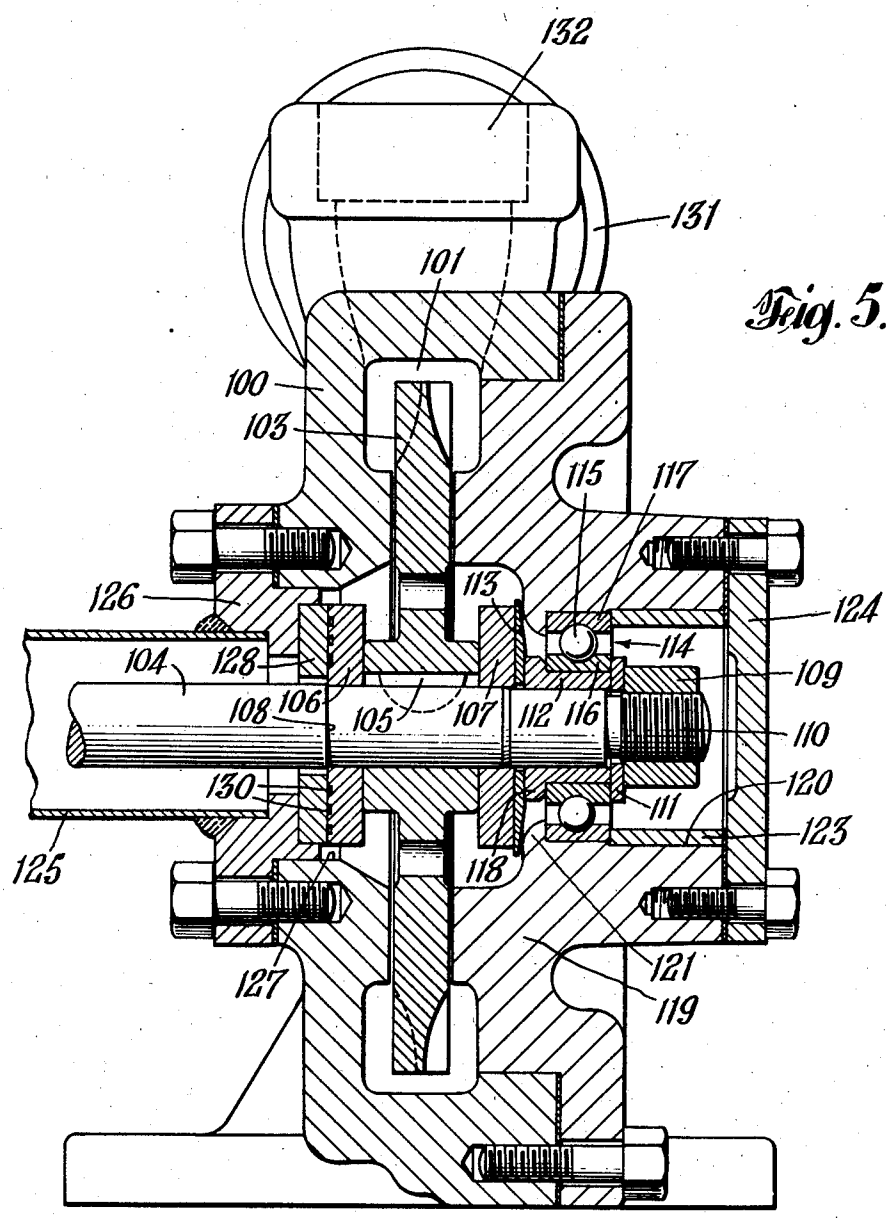


Fig. 5.

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UNITED STATES PATENT OFFICE

2,468,246

ROTARY PUMP FOR LIQUEFIED GASES

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12 Claims. (Cl. 103—96)

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This invention relates to rotary pumps for liquefied gases, particularly liquefied gases having normal boiling points substantially below 233° K., such as liquid nitrogen.

The pumping of liquefied gases having low boiling points involves unusual problems due to the great volatility of the liquid and its extremely low temperature. The construction must allow for severe contraction of the pump parts due to the low temperature without incurring increased friction or affecting the pumping efficiency. The volatility of the liquefied gas makes it essential to avoid undue generation of heat during operation of the pump as such heat generation may cause gas binding and failure of operation. The low temperature also makes it impossible to use customary fluid or semi-solid lubricants, so that bearings of the customary type have not been employed.

To obtain higher delivery pressures when pumping a liquefied gas by rotary pumps it is found necessary to increase the pressure in at least two stages and when several stages are provided in a single casing with a single drive shaft the difficulties of accommodating the contraction of the parts have caused complicated arrangements to be employed. The problem of interstage sealing is then introduced because the increased energy of the liquid in the higher pressure stage causes vaporization if it leaks to a lower pressure stage. Such interstage sealing should preferably be accomplished without incurring excessive friction and also the sealing arrangement must be unaffected by expansion and contraction and by the character of the liquefied gas.

It has been proposed to use bearings made of bonded carbon or graphite in pumps for low temperature liquefied gas, particularly liquid oxygen, since such material is self-lubricating and unaffected by the low temperature. When attempting to pump liquid nitrogen, however, it has been discovered that such material is not satisfactory because the bearing wears excessively or disintegrates. When bonded graphite bearings are employed in liquid oxygen pumps it has been found necessary to reduce the bearing loads by various expedients for counterbalancing the resultant force of the radial increase of pressure around the impeller and an axial force due to the difference of pressure on opposite sides of the impeller, particularly the last stage impeller of a multi-stage pump. Such expedients add greatly to the complexity of the pump structure, thus solutions that may suggest themselves for many

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of the difficulties are conflicting in that they may enhance others.

According to the present invention, instead of the previously employed carbon sleeve type of inboard bearing there is employed a hard metal bearing of the type having rolling elements between inner and outer annular races, specifically ball and roller bearings, it having been discovered that when in contact with clean low temperature liquefied gas, such as liquid nitrogen in particular, such bearings operate successfully without requiring oily liquid or semi-solid lubricants.

The principal objects of the present invention are therefore to provide an improved pump for liquefied gases embodying a novel combination of features and association of parts that successfully overcome the difficulties set forth above.

Other objects of the invention are to provide a pump for low temperature liquefied gas that avoids the use of bonded carbon inboard bearings and the complications attendant on such use; to provide a multi-stage pump for liquefied gas having an improved type of interstage sealing means that is frictionless to avoid the production of frictional heat; to provide in a rotary pump for liquefied gas an inboard bearing arrangement that maintains the axial impeller position so accurately that special thrust bearings for the one or more impellers and provisions for giving axial adjustability to the one or more impellers during operation are not needed.

Other objects and features of novelty will be apparent from the following description and the accompanying drawings in which:

Fig. 1 is a view of a longitudinal cross-section through a pump of the two-stage type embodying the principles of the present invention;

Figs. 2, 3, and 4 are fragmentary sectional views of several embodiments of the improved interstage fluid seal employed between the impellers of the pump shown in Fig. 1; and

Fig. 5 is a longitudinal sectional view of a single stage rotary pump embodying features of the invention.

Referring now to the drawings and particularly to Fig. 1, the pump, which is of the turbine type, is provided with a casing 10 having two annular pumping channels 11 and 12 therein. The channels 11 and 12 are provided with the customary inlet and outlet passages, the outlet passage of channel 11 being connected to the inlet of channel 12, so that the channel 11 serves for the low pressure stage and the channel 12 for the high pressure stage. The inlets and outlets are not

shown in the drawing because their particular arrangement forms no part of the present invention. They may, however, be arranged as illustrated in Fig. 4 of United States Patent No. 2,340,787. The bladed perimeters 13 of impellers 14 and 15 operate in the channels 11 and 12. These impellers are positioned on opposite sides of a partition 16 formed in the casing 10. The impellers 14 and 15 are supported in axial alignment and rigidly spaced apart upon a sleeve 17 that is fixed to a drive shaft 18 which is concentric with the pumping channels 11 and 12 and is disposed so that one end extends out of the casing, the other or inboard end being supported by a bearing indicated generally at 19 which is of the type having rolling elements between annular races. The sleeve 17 fits tightly in the bore of the hub portions 20 and 21 of the impellers 14 and 15, the hub 20 extending leftward and the hub 21 extending rightward. A spacer ring 22 is disposed around the sleeve 17 between the hubs 20 and 21 and the hubs and the spacer 22 are clamped together between a collar 23 formed on the sleeve 17 adjacent the hub 21 and a ring nut 24 which is threaded onto the left end of the sleeve 17 to bear against the end of hub 20. The sleeve 17 is preferably secured to the shaft 18 by suitable means, for example it is axially fixed by the nut 31 hereinafter described, and a key may also be used for securing it rotatively.

As herein illustrated, the inboard bearing 19 is a single-row ball bearing having a row of balls 25 mounted between inner and outer races 26 and 27, the inner race 26 being mounted upon a portion of the sleeve 17 extending to the right of the collar 23. A suitable type of ball separator or retainer may be used to space the balls around the races. The right end 29 of the shaft 18 is of reduced diameter providing a shoulder 30 which engages a corresponding internal shoulder of the sleeve 17. The nut 31 is threaded on the end 29 of the shaft to engage the side of the inner race 26 and hold it tightly against the collar 23 and also to tightly fix the sleeve 17 axially to the shaft 18. The outer race 27 is supported in fixed relation to the casing 10 which has an end cover 32 having an inner surface 33 closely adjacent the impeller 15 and a central opening 34 about the hub 21. The opening 34 is closed by a bearing support member 35 that supports the outer race 27, the race 27 being clamped between a flange 36 at the inner end of the bearing support 35 and a ring 37 engaged between the race 27 and a cap 38 that is threaded onto the bearing support 35 for closing its outer end.

The left side of the casing 10 is closed by a similar cover 39 having a central opening which is closed by a bearing housing 40. Within the housing 40 is provided a fluid barrier 41 positioned to the left of the hub 20 and having a hole through which the shaft 18 passes with a free running clearance. The barrier may be supported in any suitable manner, for example as shown, by being integral with a sleeve 43 fitted into the housing 40. Also within the housing 40 another bearing may be optionally provided. Such bearing may also be of the annular single row ball type including a row of balls 44 between internal and external races 45 and 46, the outer race 46 being mounted within the housing 40 with a clearance space on each side of it so as to permit axial adjustment of the bearing with respect to the housing 40. The inner race 45 may be secured in any suitable manner to the shaft 18, for example it is fitted onto a collar or ring 47 which may

be a portion of or be tightly secured on the shaft 18. Since the impellers 14 and 15 and the bearing race 26 are mounted in fixed relation to each other, it will be seen that the sleeve 17 could be an integral portion of the shaft 18, or that these parts could be mounted directly on the shaft.

Usually it will be desirable to position a shaft seal for the shaft 18 at a distance from the housing 10 and to this end a shaft housing extension 48 is secured to the housing 40. The outer end of housing 48 with the seal is not illustrated because it may be of any suitable type, for example such as that indicated in the aforementioned Patent No. 2,340,787. The outer end of the shaft 18 must be free to adjust itself axially because the position of the inner end is fixed by the bearing 19.

Associated with the partition 16 there is preferably provided an interstage sealing means which depends on close clearance between adjacent surfaces to impede the flow of liquid from the high pressure channel 12 toward the low pressure channel 11. Sufficient interstage sealing may be obtained by holding the clearance space 50 between the side of the impeller 15 and the side of the partition 16 very thin. The clearance between the inner end or rim of the partition 16 and the spacer sleeve 22 may also be made very narrow. It will be advantageous to provide additional barrier means to the flow of liquid through the clearance space by grooves of various forms as illustrated in the several figures. Thus in Fig. 1 a form of labyrinth seal is provided by forming a series of annular radially spaced grooves 51 in the left side of the impeller 15. These grooves and the lands between them intermesh with corresponding grooves 52 and lands of a seal member 53 which is secured in the partition 16. The seal member is seated fluid-tightly against the bottom 55 of the counterbore.

For satisfactory sealing it is not essential that both surfaces be grooved because it is found that the sealing effect may be sufficient by merely providing grooves in one of the surfaces. The grooves may be in either the stationary or the rotating surface. As shown in Fig. 2, the impeller 15 is not grooved but the seal member 153 that corresponds to seal member 53 of Fig. 1 is provided with a radially arranged series of grooves 56. These grooves may be annular but in some instances it may be preferable to arrange them in spiral form so that liquid trapped therein will be given an outward flow characteristic in order that advantage may be taken of the centrifugal force for impeding flow of liquid from the pumping channel 12 toward the shaft.

As mentioned heretofore, the sealing surfaces may alternatively be formed between the inner rim of the partition and the spacer sleeve 22. In Fig. 3 a sealing member 253 is arranged to form the inner rim of the partition 16, having a cylindrical surface 57 that is close to a spacer ring 122 that is of the same width as sleeve 22 but is provided with external grooves 58 which intermesh with corresponding lands and grooves in the surface 57.

As previously mentioned, either surface may be grooved and the other surface may be ungrooved, thus as illustrated in Fig. 4 the surface 357 of the seal member 353 is ungrooved and the sleeve 222 is provided with grooves 60. The grooves 60 may be annular or they may be helical in form with the helix angle arranged with respect to the direction of rotation to drive fluid toward the right.

The principles of the invention are also of

great advantage when applied to other rotary pumps for liquefied gas such as a centrifugal type or a single stage pump as illustrated in Fig. 5. This pump is provided with the customary casing 100 having a pumping channel 101 in which the periphery of impeller 103 operates. The impeller 103 is mounted tightly on the shaft 104 in any suitable manner such as by a key 105, and by clamping between spacer rings 106 and 107. The ring 106 abuts against a shoulder 108 on the shaft 104 while the spacer ring 107 is pressed toward the impeller 103 by a nut 109 that is threaded on the inboard end 110 of the shaft 104. The nut 109 bears on a washer 111 that secures a mounting sleeve 112 and a slinger disk 113 toward the spacer ring 107. The sleeve 112 provides a mounting means for an inboard bearing of the rolling element type indicated generally at 114. The inboard bearing shown has a single annular row of hard metal balls 115 acting between inner and outer races 116 and 117, the inner race 116 being supported on the sleeve 112 and clamped between a collar 118 at the left end of the sleeve, and the washer 111. The pump casing 100 has a side cover 119 provided with a central bore 120 in which the outer race 117 of the bearing is mounted. The bore 120 is provided with a bottom shoulder 121 against which the race 117 is secured by a ring 123 that is secured in place by a cover 124 that seals the outer end of the bore 120. An extended shaft housing 125 for the outboard portion of the shaft 104 is secured to a ring member 126 that covers a shaft opening 127 in the left side of the casing 100.

For the single stage pump it is preferable to provide a fluid barrier to prevent fluid from passing from the pumping channel into the shaft housing 125 at an excessive rate. To this end there is provided a barrier ring or partition 128 secured in the ring member 126 and having a surface adjacent the surface of the ring 106 to provide a narrow clearance therebetween. If desired, the sealing action may be enhanced by providing one or both of the surfaces of the barrier 128 and ring 106 with grooves 130 as previously described in connection with the interstage seal of the two-stage pump. The slinger disk 113 preferably has a tapered peripheral portion so as to be effective for centrifugally throwing outward any material of high density that may come near it. The slinger disk is particularly effective to prevent any particles of solid matter from reaching the bearing 114. The pumping channel 101 is provided with the customary inlet and outlet which are indicated in Fig. 5 at 131 and 132 respectively.

As previously indicated, the invention relies in part upon the discovery that hard metal bearings of the rolling element type may be successfully operated without the customary lubricants when they are subjected to low temperature liquefied gases provided such gases do not attack the metal of which the bearing is made. It has been universal practice to run ball bearings in an oily or grease type lubricant even though there is very little if any sliding friction. The lubricant is employed to preserve the fine-finished surfaces of the balls and races and to remove the localized heat produced at the rolling contact points due to the metal resilience. In the pump according to the invention no oily lubricant can be used, and although the liquefied gas such as liquid nitrogen does not appear in itself to have any lubricating qualities for sliding friction, it is inert with respect to the metal of the bearing and it effectively removes rolling contact heat.

The metal of the bearing should be chosen according to the character of the liquefied gas being pumped. With non-oxidizing gases such as liquid nitrogen the bearing balls and races may be made of the customary carbon or low alloy tool steels provided that care is taken to prevent rusting before the pump is put into service. In a pump for liquid oxygen rusting during idle periods may be difficult to prevent and it will be preferable then to use bearings made of the hard stainless metals such as stainless steel. Also certain hard-surfaced corrosion-resistant surfacing treatments for the balls and races may be used.

The use of the inboard bearing or bearings permits a great simplification of construction in rotary pumps for low temperature liquefied gases of both the single and multiple stage types. Previously it was necessary to provide axially adjustable connections between the drive shaft and the impellers; to provide thrust bearing means for centering each impeller in its pumping channel; to provide means such as balance cylinders for counteracting the resultant radial force of the liquid pressure acting on the high pressure stage impeller; and to provide means for reducing the axial pressures acting upon the impeller hub. According to the invention, such complications are avoided by utilizing a rolling element type of bearing positioned close to the impeller, the bearing being constructed and mounted to withstand substantial radial loads and also to resist substantial axial forces. A two-point contact ball bearing of the type illustrated is found to give good results in resisting the combined radial and axial forces because the rotating speeds of the pumps are in excess of 1,500 R. P. M. For heavier duty, a double-row ball bearing may be advantageously substituted for the single-row type. The inboard bearing which is to axially position the shaft and impeller or impellers should be mounted close to the impeller so that relative axial movements between the impeller and casing due to expansion and contraction will be small even through the coefficients of expansion of the shaft and casing material may differ slightly.

For efficient operation of a multi-stage pump, excessive leakage from the high pressure stage toward the low pressure stage should be prevented. Previously this was accomplished by providing an interstage seal employing rubbing surfaces, one of which was made of bonded carbon. It is found that a small amount of leakage can be tolerated without measurably affecting the pumping and, according to the invention, the interstage sealing is obtained by maintaining close clearance spaces between adjacent rotating and stationary surfaces. Thus sufficient sealing is maintained by making the run clearance 50 between the side of the partition 16 and the side of the second stage impeller quite small or the clearance between the sleeve 22 and the bore through the partition may be maintained very small. The sealing efficiency is increased, according to the invention, by providing a series of grooves in one or both of the adjacent surfaces. In any of the seal arrangements there is no frictional contact between the rotating and stationary surfaces, and thus generation of frictional heat is avoided. The use of the inboard ball bearing makes it possible to maintain the running clearance of

the sealing surfaces at the desired constant value.

Although several modifications of the invention have been described in detail, it is contemplated that changes other than those disclosed herein may be made and that certain features may be used without others without departing from the spirit and scope of the invention.

What is claimed is:

1. In rotary pumps and the like for pumping liquefied gas having a boiling point at atmospheric pressure below 233° K., a casing having at least one pumping channel, a rotary impeller in said casing having a peripheral portion cooperating with said channel, said casing having portions disposed at very close clearance spacing from the sides of said impeller, an axially disposed drive shaft for said impeller, and at least one bearing rotatively supporting said shaft in the casing, said bearing being of the type having rolling elements between annular races, said rolling elements being free of lubricants and exposed to the liquefied gas being pumped, the inner and outer races of said bearing being fixed respectively to the drive shaft and to the casing whereby said bearing axially positions the drive shaft and impeller with respect to the casing to maintain the said close clearance spacing irrespective of severe temperature changes.

2. A rotary pump according to claim 1 which includes a slinger disk secured to said shaft between said bearing and the impeller.

3. A multi-stage rotary pump having a casing with at least two pumping channels, rotary impellers having peripheral portions cooperating with said pumping channels for providing lower and higher pressure pumping stages, an axial drive shaft carrying said impellers, an inboard bearing for both radially and axially supporting said shaft in the casing, said casing having an interstage partition between the pumping channels and the said impellers, and interstage fluid sealing means associated with said partition and comprising rotating and stationary surfaces maintained closely spaced with respect to each other.

4. A muti-stage rotary pump according to claim 3 in which said sealing means comprises a series of radially spaced grooves in a portion of said partition spaced closely adjacent to an annular surface on one of said impellers.

5. A multi-stage rotary pump according to claim 3 in which said sealing means comprises a series of radially arranged annular grooves and lands intermeshing at close spacing with corresponding lands and grooves on one of said impellers.

6. A multi-stage rotary pump according to claim 3 in which said sealing means comprises closely adjacent annular surfaces, one of which is on said partition and the other of which is on the impeller of the higher pressure stage, at least one of said adjacent surfaces having a series of grooves.

7. A multi-stage rotary pump according to claim 3 in which said sealing means comprises means on said shaft between the impellers and having an external surface of revolution and a corresponding closely adjacent surface of revolution on the inner rim of said partition, at least one of said surfaces of revolution being grooved.

8. A multi-stage rotary pump according to claim 3 in which said sealing means comprises a sleeve secured to said shaft between the impellers and having an external surface of revolution and a corresponding closely adjacent surface of revolution on the inner rim of said partition, both of said surfaces of revolution being provided with annular grooves separated by lands positioned so that the lands of each surface intermesh with the grooves of the opposite surface.

9. A multi-stage rotary pump according to claim 3 in which said sealing means comprises means on said shaft between the impellers and having an external surface of revolution and a corresponding closely adjacent surface of revolution on the inner rim of said partition, the surface of revolution on said shaft being provided with a series of grooves.

10. In rotary pumps and the like for pumping liquefied gas having a boiling point at atmospheric pressure below 233° K., a casing having at least one pumping channel, a rotary impeller in said casing having a peripheral portion cooperating with said channel, said casing having portions disposed at very close clearance spacing from the sides of said impeller, an axially disposed drive shaft for said impeller having an inboard end within the casing and an opposite end extending to a point without the casing, a bearing supporting said shaft in the casing, said bearing being of the type having rolling elements between annular races, said rolling elements being free of lubricants and exposed to the liquefied gas being pumped, the inner race of said bearing being secured to the inboard end of the shaft and the outer race being secured to the casing at one side of the impeller, the impeller being secured to the shaft at a fixed distance from the bearing, and sealing means on the side of said impeller opposite said bearing comprising surfaces maintained closely spaced with respect to each other, one of said surfaces being stationarily fixed to said casing and the other being secured to rotate with said shaft.

11. A multi-stage rotary pump having a casing with at least two pumping channels, rotary impellers having peripheral portions cooperating with said pumping channels for providing lower and higher pressure pumping stages, an axial drive shaft carrying said impellers in fixed relation to each other on the shaft, said shaft having an inboard end adjacent the impeller of the higher pressure stage and an opposite end extending to a point without the casing, a bearing for the inboard end of the shaft of the type having rolling elements between annular races, the inner race being secured to the shaft and the outer race being secured to the casing, said casing having a partition between said impellers, interstage fluid sealing means associated with said partition and comprising rotating and stationary surfaces maintained closely adjacent each other, a second bearing of the type having rolling elements between inner and outer races positioned on said shaft on the side of said impellers opposite to said first-mentioned bearing and having its inner race secured on the shaft and its outer race supported radially by the casing in a manner allowing axial adjustment, and means associated with the casing between said second bearing and the lower pressure stage impeller providing a fluid barrier therebetween.

12. In rotary pumps and the like for pumping liquefied gas having a boiling point at atmospheric pressure below 233° K., a casing having a pumping channel, a rotary pumping element in said casing cooperating with said pumping chan-

nel, an axially disposed drive shaft for said pumping element, and an inboard bearing rotatively supporting said shaft in the casing, such bearing being of the type having hard metal balls between inner and outer annular races, the metal of said balls and races being inert to the liquefied gas being pumped and free of lubricant, the inner race being fixed to said shaft and the outer race being secured to said casing whereby said bearing radially and axially positions the drive shaft and pumping element with respect to the casing, said bearing balls being exposed to and cooled by the liquefied gas being pumped.

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