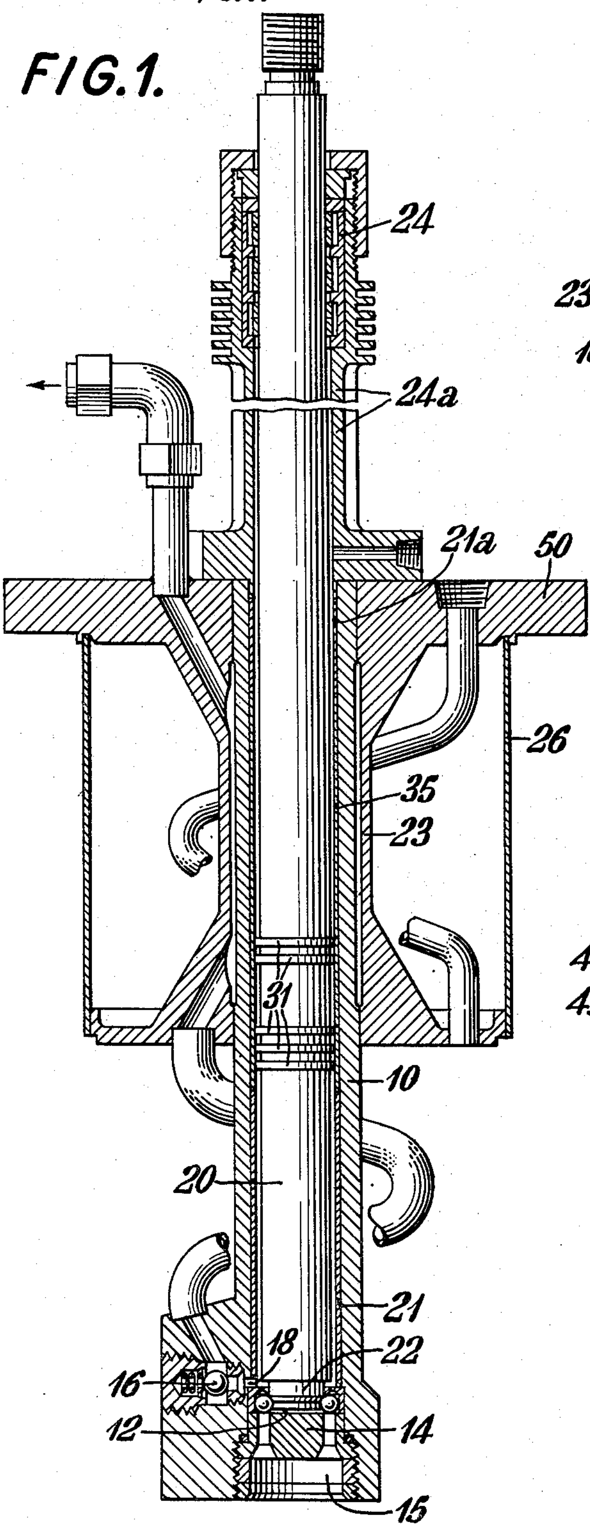
CRYOGENIC PUMP SEALING RINGS

Filed Dec. 13, 1960 5 Sheets-Sheet 1

FIG. 1.



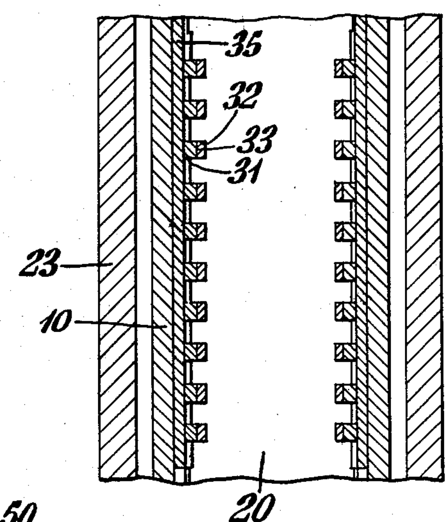


FIG. 2.

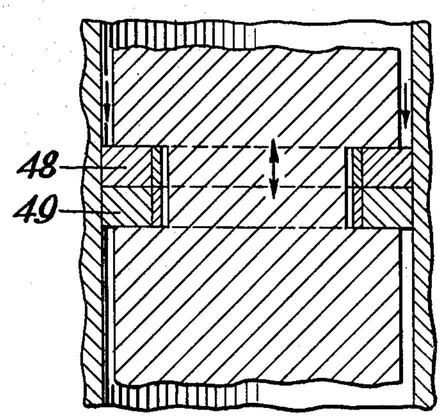


FIG. 11.

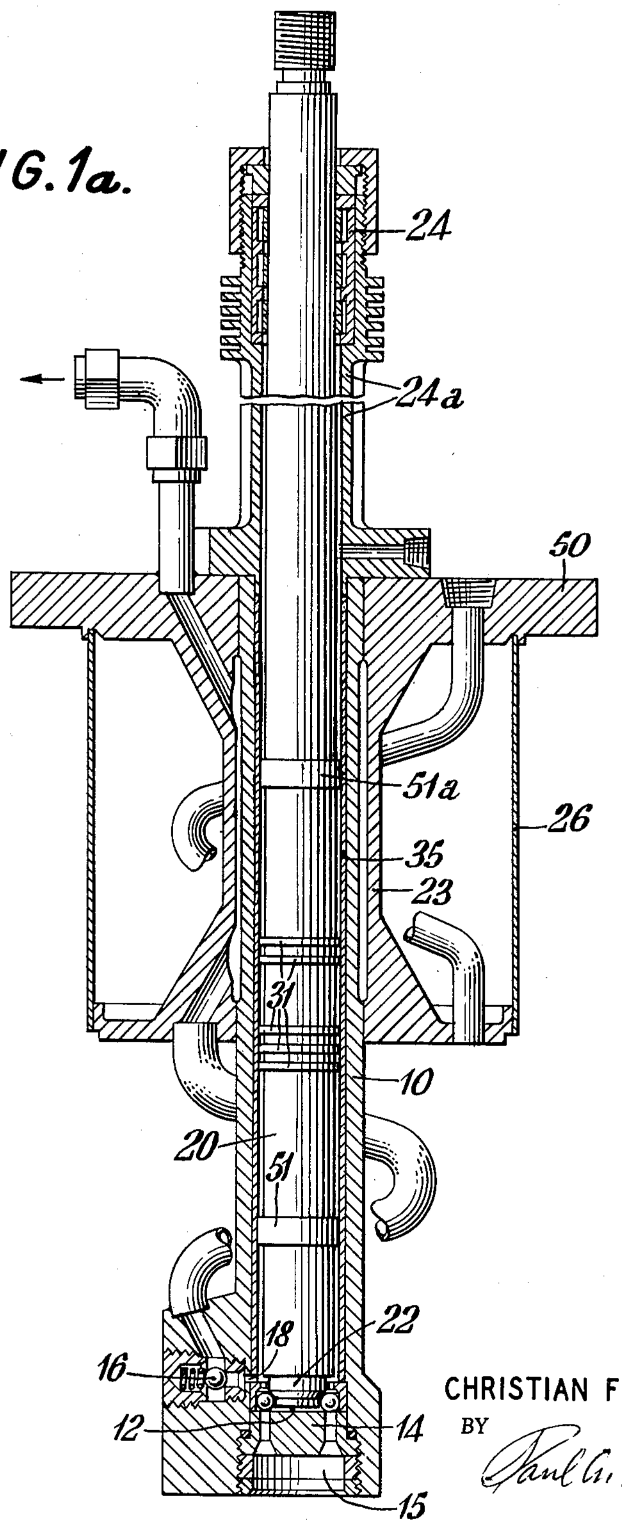
INVENTOR.
CHRISTIAN F. GOTTZMANN
BY
ATTORNEY

Aug. 25, 1964 C. F. GOTTZMANN 3,145,629

CRYOGENIC PUMP SEALING RINGS

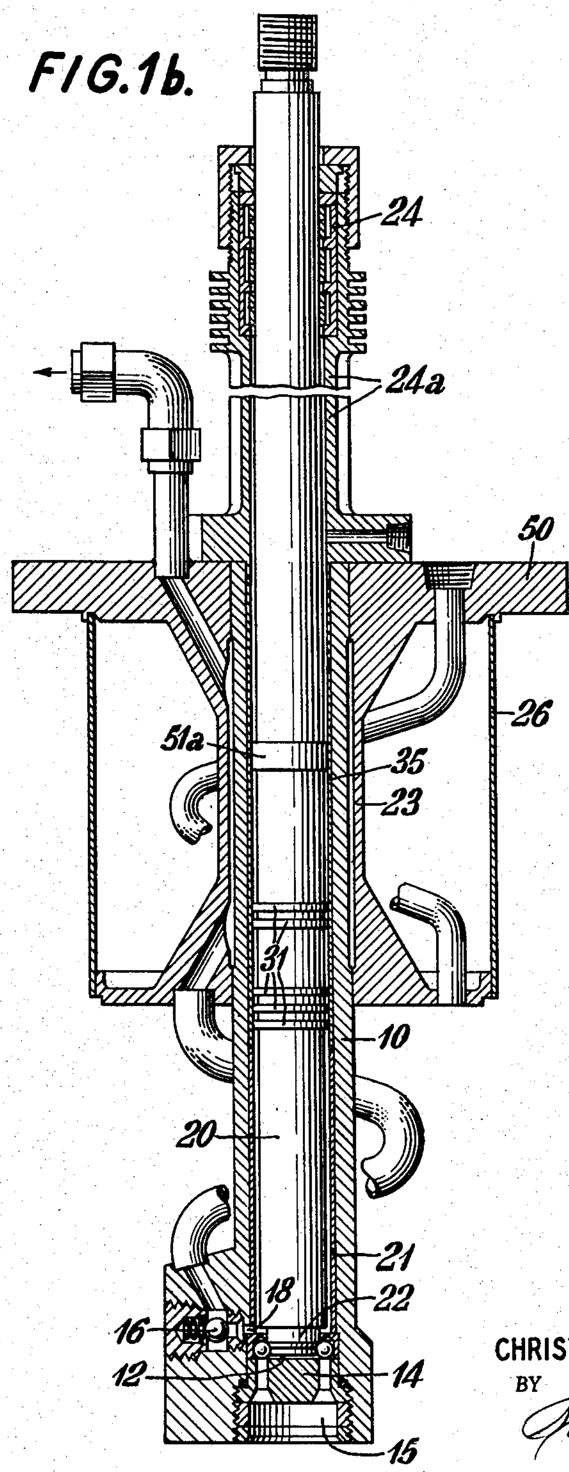
Filed Dec. 13, 1960 5 Sheets-Sheet 2

FIG. 1a.



INVENTOR.
CHRISTIAN F. GOTTZMANN
BY
Paul G. Rose
ATTORNEY

FIG. 1b.



INVENTOR.
CHRISTIAN F. GOTTZMANN
BY
Paul A. Rose
ATTORNEY

Aug. 25, 1964 C. F. GOTTZMANN 3,145,629

CRYOGENIC PUMP SEALING RINGS

Filed Dec. 13, 1960 5 Sheets-Sheet 4

FIG. 3

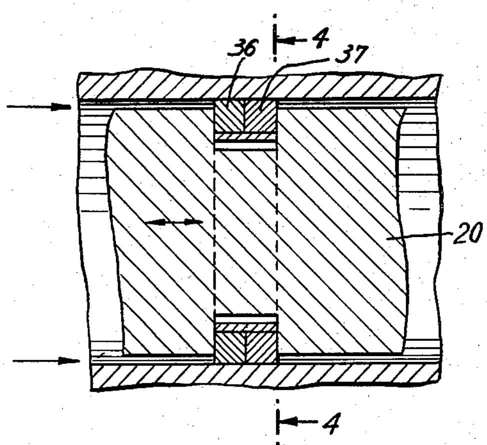


FIG. 4

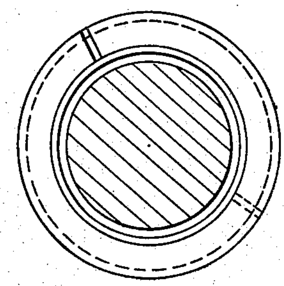


FIG. 5

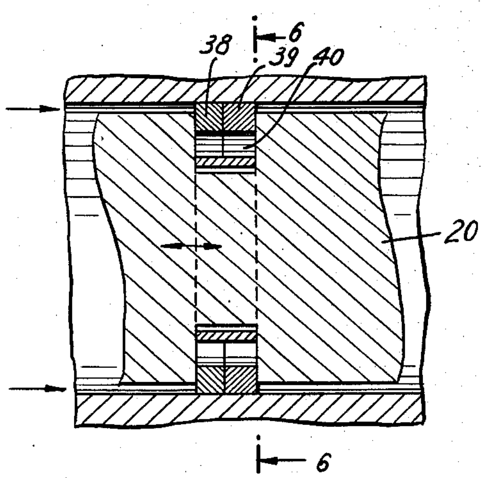
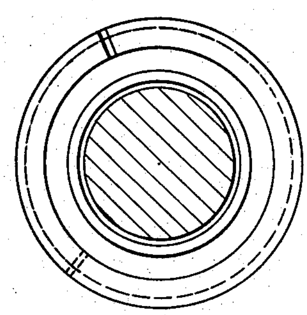


FIG. 6



INVENTOR.
CHRISTIAN F. GOTTZMANN

BY Paul A. Rose

ATTORNEY

CRYOGENIC PUMP SEALING RINGS

Filed Dec. 13, 1960 5 Sheets-Sheet 5

FIG. 7

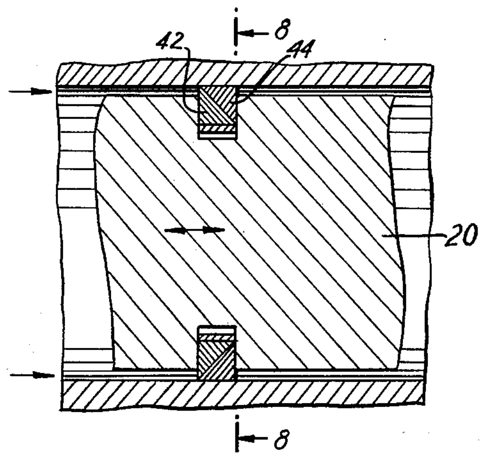


FIG. 8

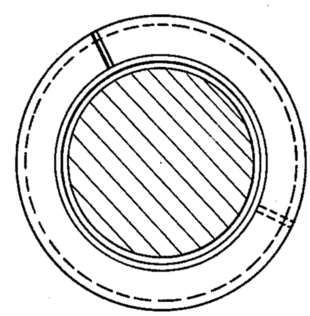


FIG. 9

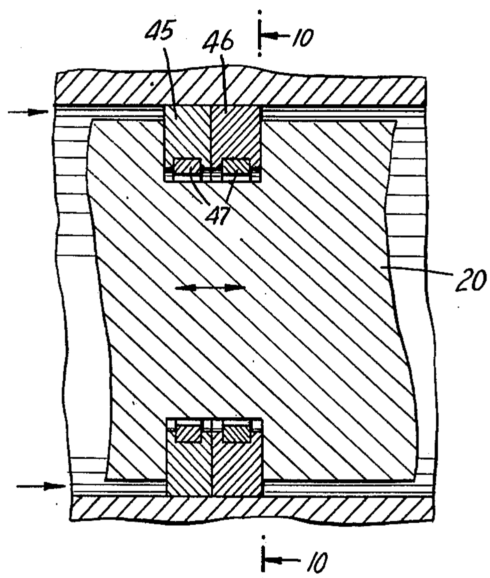
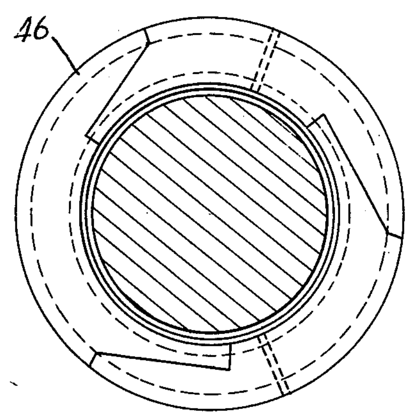


FIG. 10



INVENTOR.
CHRISTIAN F. GOTTZMANN
BY Paul A. Rose
ATTORNEY

United States Patent Office

3,145,629

Patented Aug. 25, 1964

1

3,145,629

CRYOGENIC PUMP SEALING RINGS

Christian F. Gottzmann, Buffalo, N.Y., assignor to Union Carbide Corporation, a corporation of New York

Filed Dec. 13, 1960, Ser. No. 75,598
18 Claims. (Cl. 92—155)

This invention relates to cryogenic pump sealing rings, and more particularly, to such sealing rings used for pumping liquefied gases having a boiling point below 273° K. and usually at discharge pressures above 200 p.s.i.g. and piston speeds above 100 feet per minute.

This application is in part a continuation of my co-pending application Serial No. 12,130 filed March 1, 1960, for "High-Pressure Pump for Cryogenic Fluids," the entire disclosure of which by this reference is incorporated into this application.

In high-pressure cryogenic liquid pumps handling very low-boiling cryogenic liquids such as hydrogen or helium, which have low densities and viscosities, sealing the reciprocating plunger against high pressures is extremely difficult. In prior art high-pressure pumps for higher-boiling liquid nitrogen and oxygen, sealing was satisfactorily accomplished by providing a high-pressure resistance path back to suction through the annular clearance space between the plunger and liner. By making the radial clearance between the plunger and liner sufficiently small and the length of the leakage path sufficiently long, blow-by leakage can be kept at relatively low values when pumping liquid oxygen and nitrogen. However, for a liquid hydrogen or helium pump of this construction, blow-by leakage is considerable and improved plunger sealing means are required for efficient operation of the pump.

As liquid hydrogen and helium possess insignificant lubricating qualities and only provide a sink for the undesirable friction heat, lubrication between the rubbing elements has to rely on the self-lubricating properties of the materials used. Prior expedients for materials of this type have resulted in excessive wear rates.

It is therefore the main object of the present invention to provide improved self-lubricating sealing rings which have better wear resistance and lower friction than rings previously used in cryogenic pumps. Another object is to locate the self-lubricating sealing rings of cryogenic pumps near the warm end of the pump body. At the warmer location, the blow-by gas being sealed will have increased in both temperature and specific volume, and is much easier to seal than the colder, denser gas. The sealing material functions better, and the coefficient of friction thereof is reduced at the warmer temperature.

According to the present invention, a filled plastic sealing ring is mounted in an annular groove in the pump plunger, and engages a mating surface of the pump barrel without externally supplied lubrication. The sealing ring is preferably Teflon filled with carbon or glass. The groove and ring are preferably located near the warm end of the plunger. The pump plunger is suitably guided at its ends by bushings or rider rings, which are also made of a self-lubricating material. While these sealing rings and bushings are regarded as a necessity for liquid hydrogen and helium pumps, they are also beneficial in reciprocating plunger pumps handling other higher-boiling cryogenic liquids such as oxygen, nitrogen, and argon.

2

In the drawings:

FIGURE 1 is a vertical section through a cryogenic pump provided with sealing means according to the preferred embodiment of the present invention;

FIGURE 1*a* is a vertical section through a cryogenic pump similar to FIGURE 1 provided with sealing means according to the present invention;

FIGURE 1*b* is a vertical section through a cryogenic pump similar to FIGURE 1 provided with guiding means according to the present invention;

FIGURE 2 is an enlarged section of a portion of the plunger and barrel shown in FIG. 1;

FIGURES 3, 5, 7, 9, and 11 are modifications of FIG. 2; and

FIGURES 4, 6, 8, and 10 are sections along the respective lines indicated.

As shown in FIG. 1, the pump comprises an elongated body **10** preferably in the form of a barrel having a pumping chamber **12** in the cold end thereof provided with an inlet valve assembly **14** which controls an inlet port **15** in the bottom of the body **10**. A discharge valve **16** controls an outlet passage **18** from the pumping chamber **12**.

A reciprocating plunger **20** extends through the opening in the top of the barrel **10** and has an inner pumping end portion **22** interfitting with the inlet valve assembly **14** and substantially filling the clearence space thereof. The opposite or warm end portion of the plunger **20** is provided with warm end packing means **24** spaced a substantial distance from the pump mounting flange **50** in an extension **24*a*** of the pump body **10**. The extension **24*a*** absorbs sufficient heat to avoid freezing water vapor or condensible gases in the packing. The plunger **20** is guided at its cold end by guide bushing **21** and at the warm end by guide bushing **21*a***. Because the guide bushings are made from a self-lubricating material, small radial clearances can be satisfactorily provided between plunger and guide bushing at the cold end which comprises the pumping chamber **12**, and pump clearance volume can be thus maintained quite small without causing operating difficulties. The upper end of the pump body is surrounded by a heat exchanger **23** and vacuum jacket **26**.

As shown in FIGS. 1 and 2, several piston rings **31**, preferably constructed of filled plastic material, are located in annular grooves **32** machined into the plunger **20**. These rings are preferably backed by metal rings **33** which exert a substantially uniform outward force to keep piston rings **31** bearing against the mating pump inner wall which may be a removable metal sleeve or liner **35**, preferably of surface hardened or plated seamless steel.

The piston rings **31** are preferably constructed of carbon-filled polytetrafluoroethylene (sometimes abbreviated as TFE and well known as "Teflon"), although other similar filled Teflon materials such as glass-filled Teflon (GFT) may be used successfully. Back-up rings **33** are preferably made of beryllium copper, although stainless steel or other metals not embrittled by low temperatures may also be used.

Filled Teflon is a very satisfactory piston ring material for cryogenic pumps. This material is ideally suited for this sealing requirement, because compressive yield strength increases from about 2,000 p.s.i. at ambient temperature to 14,000–18,000 p.s.i. at liquid nitrogen temperature. Bench tests have indicated the coefficient of friction of these filled Teflon materials at liquid nitrogen temperature is below about 0.1 and as low as 0.04, as compared to 0.13 to 0.15 for the Teflon impregnated porous metals under similar conditions. Such filled Teflon materials are also desirable as guide bushing materials.

It is pointed out that the sealing rings of this invention do not provide an absolute sealing means for the pump plunger, but instead permits some blow-by which is controlled to a small amount with very little frictional heat being generated. The low friction produced by the sealing rings and guide bushings permits increased pump plunger operating speeds, which is desirable since pump size and investment costs may thereby be reduced and overall performance also improved. Using this construction, plunger speeds above 100 feet per minute and even up to about 400 f.p.m. may be used.

It will be noted that piston rings **31** are preferably located near the warm or upper end of the pump body. At the warmer location, the blow-by gas being sealed will have increased significantly in both temperature and specific volume, thus making the sealing inherently more effective than attempting to seal the colder, denser gas. Filled Teflon plastic material has better sealing characteristics at a warmer temperature level than if located nearer the very cold lower end of the pump. The coefficient of friction of the ring material is reduced somewhat at a warmer temperature level unless unit load and speed became excessive. Also, the frictional heat of the piston rings may be more easily controlled near the warm end of the pump barrel, and thus prevented from being conducted to the cold suction end of the pump and adversely affecting pump performance.

Therefore, the piston rings are preferably located as near the warm end of the pump body as possible and in that portion of the extended pump barrel where a heat exchanger may be most effectively employed to recover by the high-pressure discharge fluid the heat leak due to conduction down the plunger. The location of this heat exchanger is usually in the upper half of the pump body. Also, the sealing rings are so located as to still remain within the limits of the heat exchanger during the plunger upstroke. This permits the major part of the frictional heat generated by the sealing rings to be absorbed by the compressed discharge fluid passing through the heat exchanger. It also locates the rings within a sufficiently low-temperature zone to make use of the improved low-temperature strength of filled Teflon materials, while still being sufficiently removed from the very low-temperature parts of the pump so that use can be made of the improved sealing and frictional characteristics of the rings at this intermediate temperature level.

Thus, the first piston ring should be removed from the cold end of the plunger rod by a distance at least equal to the pump working stroke plus 2 inches, and preferably by a distance equal to at least twice the pump stroke length. The 2-inch distance will provide the minimum required guidance of the plunger at the cold or bottom end, and the required number of sealing rings for effective sealing and heat transfer considerations will then determine the body length of the pump.

In general, the number and design of the piston rings used depends upon the pump discharge pressure required, the gas blow-by quantity which can be tolerated in a particular pump application, and the ring "life" desired. For example, for normal discharge pressures of 1,000–2,000 p.s.i.g., about 4–6 rings would usually be used, and a step-type ring gap would usually be used to obtain the most effective sealing per ring. However, for higher pump discharge pressures such as 2,000–10,000 p.s.i., about 8–12 rings would usually be used. The first rings sealing against these higher pressure levels would preferably be designed with a plain gap, i.e., butt ends, to provide less pressure differential across these rings and thereby serve as "pressure breaker" rings to prevent excessive ring wear rates. Rings operating at the lower pressure levels would preferably have step-type gaps for more effective sealing per ring. By sealing less pressure difference across each of several rings, the unit loading of each ring will be less and the ring life will thereby be increased. Thus, the period of time between pump overhaul and replacement of rings for a particular application is directly related to the number and type of piston rings used. A minimum life of 500 hours between overhaul is expected.

The guide bushings **21** and **21***a* preferably comprise plastic impregnated metal bushings which are usually rolled into a cylinder from a flat piece and thus have a single gap or split, which facilitates installation without undesirable distortion. The total desired length for each bushing may be provided by a single cylindrical piece, or may be made up of several separate bushings, depending upon the individual bushing lengths available. Wall thickness is whatever is required to give dimensional stability, and should be at least .045 inch. The guide bushings may be composed entirely of filled Teflon material, or such material may be bonded onto a metal backing.

In another desirable modification shown in FIG. 1*a*, plastic rider rings **51** and **51***a* may be substituted at the warm and cold ends of the plunger instead of guide bushings, also a plastic rider ring **51***a* may be substituted for guide bushing **21***a* at the warm end of the plunger. This arrangement has the advantage of permitting extending the metal liner **35** to the top flange **50** of the pump, which makes for easier installation of the plunger and piston ring assembly into the pump bore. Another advantage is that the piston rings can be located somewhat nearer the warm end of the pump and thus provide more effective sealing and better control of the heat of ring friction.

If desired, two piston rings of rectangular cross-section may be used in each groove, as shown in FIGS. 3 and 4, to provide improved sealing over that obtained with a single ring. Both rings are preferably butt gap type, with the gap location staggered radially to prevent leakthrough. If desired, ring **36** could have a butt-type gap and ring **37** have a step-type gap. In the latter arrangement, the ring with the butt gap is preferably placed on the side exposed to the higher pressure, and the gaps circumferentially staggered to prevent leakthrough.

Also, if desired, three piston rings are used in one groove, as shown in FIGS. 5 and 6. In this arrangement, two narrow rectangular butt gap type rings **38** and **39** are employed side by side, ring **40** having a width equal to the first two is employed underneath, and a metal back-up ring is used beneath the three plastic rings. The main advantage of this arrangement is that fluid leakage by all possible paths is completely sealed.

In another useful arrangement shown by FIG. 7, two plastic piston rings are used in a single groove with a metal back-up ring, whereby the cross-section of the two plastic rings **42** and **44** have a mating diagonal surface. In this arrangement, the higher pressure preferentially forces piston ring **44** outward against the liner surface to help compensate for wear.

Segmented type sealing rings could be used in the ring grooves of the plunger rod, with internal springs, as shown in FIGS. 9 and 10, segments **45**, **46**, and springs **47**. The springs **47** may have either rectangular or circular cross-section. Due to the larger unsupported areas usually involved, these segmented rings are more generally useful for lower discharge pressures of up to, say, 1,000 p.s.i.g. One disadvantage of segmented type rings is the greater cross-section usually required for adequate strength, and particularly the greater groove depth required to accommodate this type packing ring, so that it is principally applicable for relatively larger diameter plunger rods, say, above about 2 inches diameter.

Also, a combination of one piston ring **48** and one segmental ring **49** with tangential cuts could be used, as illustrated by FIG. 11, with either a common or separate metal back-up rings. Here also, the piston ring is preferably placed towards the higher pressure side for most effective sealing.

Other desirable modifications of plunger sealing rings for cryogenic pumps may be used, of which the above serve as examples. Also, while this plunger pump is preferably used in a vertical upright position to substantially remove thrust forces due to gravity, it may be successfully used in any position. Also, while the sealing rings have been described for a reciprocating single-action plunger type cryogenic pump, they are also useful for many other types of reciprocating cryogenic pumps such as double-acting pumps, multi-plunger pumps, and pumps having port-type suction valves.

What is claimed is:

1. In a reciprocating plunger pump for cryogenic liquids boiling below 273° K. pumped to pressures above 200 p.s.i., and having cold and warm end portions, the improvement comprising a metal pump barrel having an inner mating surface; a pump plunger slideably fitting in the barrel and having an external annular groove; a self-lubricating annular sealing ring within the annular groove having an outer surface which engages the barrel mating surface; and first and second self-lubricating plunger guiding means disposed in the cold and warm pump end-portions respectively such that the sealing ring and grooves are positioned therebetween.

2. In a reciprocating plunger pump for cryogenic liquids boiling below 273° K. pumped to pressures above 200 p.s.i., and having cold and warm end portions, the improvement comprising a metal pump barrel having a hard-surfaced metal liner fitted therein, the inner surface of which comprises a barrel mating surface; a pump plunger slideably fitting in the barrel and having an external annular groove; a self-lubricating annular sealing ring within the annular groove having an outer surface which engages the barrel mating surface; and first and second plunger guiding means comprising self-lubricating guide bushings fitted within said barrel at the respective cold and warm pump end-portions of said barrel respectively such that the sealing ring and groove are positioned therebetween.

3. In the improvement in a reciprocating plunger pump as claimed in claim 1, the annular groove and sealing ring being positioned in the warm pump end-portion of said plunger.

4. In the improvement in a reciprocating plunger pump as claimed in claim 1, the self-lubricating guide bushings at the cold and warm pump end-portions consisting of Teflon-impregnated metal; and the self-lubricating sealing ring consisting of filled Teflon.

5. In the improvement in a reciprocating plunger pump as claimed in claim 1, said self-lubricating sealing ring being composed of Teflon filled with a filler of the group carbon and glass.

6. The improvement in a reciprocating plunger pump as claimed in claim 1, in which a pair of said self-lubricating sealing rings are mounted in the same groove, one of said pair being segmental.

7. In the improvement in a reciprocating plunger pump as claimed in claim 1, sealing ring back-up means comprising a metal ring providing a substantially uniform outward radial force at all temperature levels on the sealing ring.

8. In the improvement in a reciprocating plunger pump as claimed in claim 2, the annular groove and sealing ring being positioned in the warm pump end portion of said plunger, and spaced away from the cold end thereof by at least the distance of the pump stroke plus two inches.

9. In the improvement in a reciprocating plunger pump as claimed in claim 2, said pump having piston speeds above 100 feet per minute, and said guide bushing being filled Teflon.

10. In the improvement in a reciprocating plunger pump as claimed in claim 1, in which a pair of said sealing rings having parallel sides and diagonal mating surfaces are mounted in the same groove.

11. In the improvement in a reciprocating plunger pump as claimed in claim 1, said barrel having a hard-surfaced metal liner fitted therein, the inner surface of which comprises said barrel mating surface; and the first and second plunger guiding means comprising self-lubricating rider rings positioned in the plunger at the respective cold and warm pump end-portions thereof and having outer surfaces which engage said barrel mating surface.

12. In a reciprocating plunger pump for cryogenic liquids below 273° K. pumped to pressures above 200 p.s.i., and having cold and warm end portions, the improvement comprising a metal pump barrel having a hard-surfaced metal liner therein forming an inner barrel mating surface; a pump plunger slideably fitting in the barrel and having an external annular groove; a self-lubricating annular sealing ring consisting of filled Taflon within the annular groove having an outer surface which engages the barrel mating surface; and first and second plunger guiding means disposed in the cold and warm pump end-portions respectively such that the sealing ring and groove are positioned therebetween, such guiding means comprising self-lubricating rider rings consisting of filled Teflon positioned at the respective cold and warm pump end-portions thereof and having outer surfaces which engage said barrel mating surface.

13. In the improvement in a reciprocating plunger pump as claimed in claim 12, said self-lubricating sealing ring being composed of Teflon filled with a filler of the group carbon and glass.

14. In the improvement in a reciprocating plunger pump as claimed in claim 12, in which a pair of said self-lubricating sealing rings are mounted in the same groove, one of said pair being segmental.

15. In the improvement in a reciprocating plunger pump as claimed in claim 12, sealing ring back-up means comprising a metal ring providing a substantially uniform outward radial force at all temperature levels on the sealing ring.

16. In the improvement in a reciprocating plunger pump as claimed in claim 12, in which a pair of said sealing rings having parallel sides and diagonal mating surfaces are mounted in the same groove.

17. In a reciprocating plunger pump for cryogenic liquids below 273° K. pumped to pressures above 200 p.s.i., and having cold and warm end portions, the improvement comprising a metal pump barrel having a hard-surfaced metal liner therein forming an inner barrel mating surface; a pump plunger slideably fitting in the barrel and having an external annular groove; a self-lubricating annular sealing ring consisting of filled Teflon within the annular groove having an outer surface which engages the barrel mating surface; and first and second plunger guiding means disposed in the cold and warm pump end-portions respectively such that the sealing ring and groove are positioned therebetween, such guiding means comprising self-lubricating rider rings consisting of Teflon-impregnated metal positioned at the respective cold and warm pump end-portions thereof and having outer surfaces which engage said barrel mating surface.

18. In a reciprocating plunger pump for cryogenic liquids below 273° K. pumped to pressures above 200 p.s.i., and having cold and warm end portions, the improvement comprising a metal pump barrel having a hard-surfaced metal liner therein forming an inner barrel mating surface; a pump plunger slideably fitting in the barrel and having an external annular groove; a self-lubricating annular sealing ring consisting of filled Teflon within the annular groove having an outer surface which engages the barrel mating surface; and first and second plunger guiding means disposed in the cold and warm pump end-portions respectively such that the sealing ring and groove are positioned therebetween, the cold pump-end portion guiding means comprising a self-lubricating guide bushing

fitted within said barrel at the cold pump end-portion of the metal liner and the warm pump-end portion guiding means comprising a self-lubricating rider ring positioned in the plunger at the warm pump end-portion thereof and having an outer surface which engages said barrel mating surface.

References Cited in the file of this patent

UNITED STATES PATENTS

2,135,076	Hoferer	Nov. 1, 1938
2,198,301	Campbell	Apr. 23, 1940
2,607,644	Smith et al.	Aug. 19, 1952
2,784,013	Groen	Mar. 5, 1957
2,831,325	White	Apr. 22, 1958
2,846,280	Berg	Aug. 5, 1958
2,855,859	Petzold	Oct. 14, 1958
2,857,218	Pachernegg	Oct. 21, 1958
2,931,673	Gondek	Apr. 5, 1960
2,973,978	Oppenheim	Mar. 7, 1961
2,991,003	Petersen	July 4, 1961

FOREIGN PATENTS

1,102,582	France	May 11, 1955