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3,220,202

APPARATUS FOR STORING AND PUMPING A VOLATILE LIQUID

Original Filed Dec. 29, 1961

2 Sheets-Sheet 1

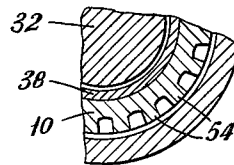
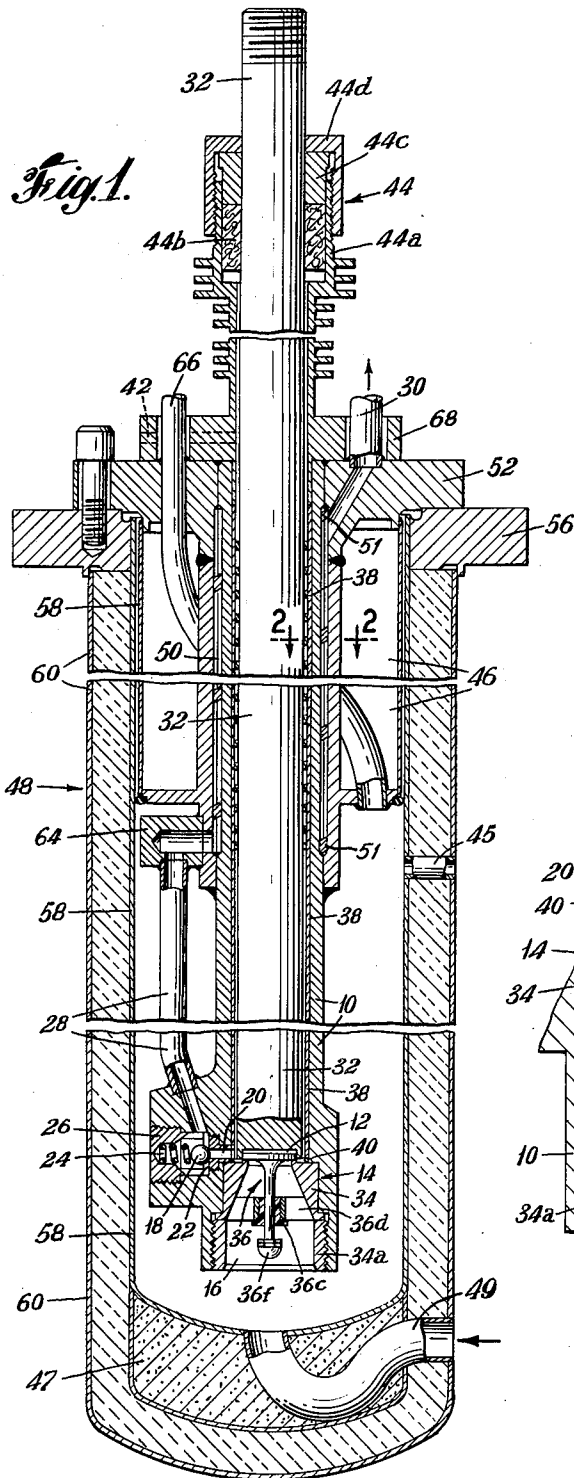


Fig. 2.

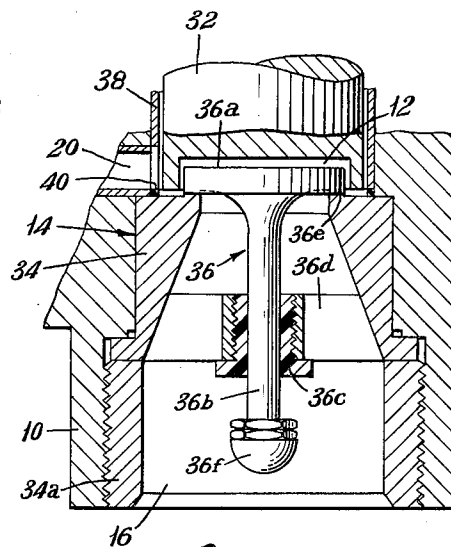


Fig. 1a.

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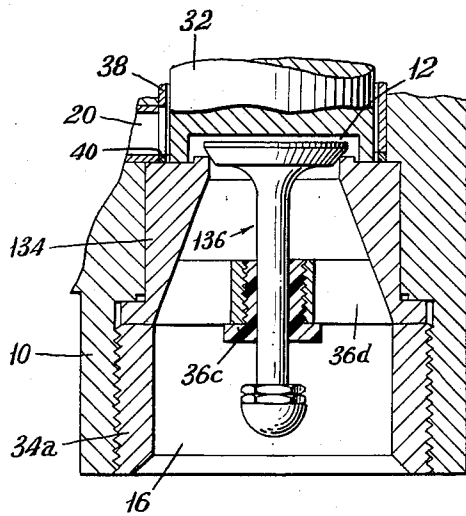


Fig. 3.

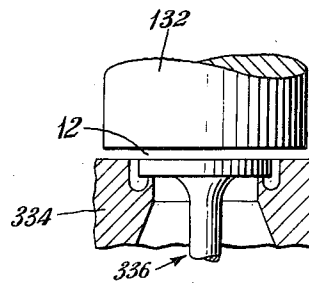


Fig. 5.

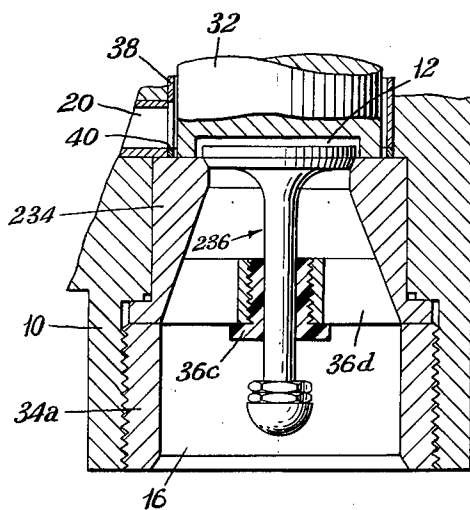


Fig. 4.

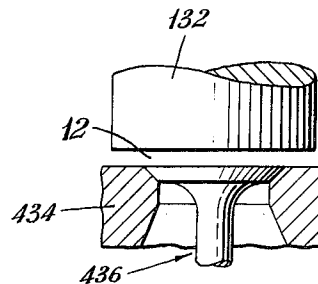


Fig. 6.

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3,220,202 APPARATUS FOR STORING AND PUMPING A VOLATILE LIQUID

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Continuation of application Ser. No. 163,297, Dec. 29,
1961. This application May 15, 1964, Ser. No. 369,352
16 Claims. (Cl. 62—55)

This application is a continuation of copending appli-
cation Serial No. 163,297, filed December 29, 1961, now
abandoned.

This invention relates to apparatus for storing and
pumping a volatile liquid having a boiling point tempera-
ture at atmospheric pressure materially below 273° K.,
and more particularly to a reciprocating pump for pump-
ing a liquefied gas having a normal boiling point below
233° K., such as liquid oxygen, nitrogen, and the like,
to an ultra-high pressure, for example, 10,000 p.s.i.

Pumps heretofore proposed for pumping liquefied gases
to high pressures have involved difficulties due to the
physical properties of the highly volatile liquids. For
example, liquefied gases have greater compressibility
than water, and thus present greater heat of compression
problems. However, these difficulties have been largely
overcome when the desired discharge pressure was below
about 3,000 p.s.i. On the other hand, the previously
known pumps are either inefficient or unworkable in the
ultra-high pressure range above about 5,000 p.s.i. This
is because special problems, in addition to strength con-
siderations, arise which tend to decrease the efficiency
of the heretofore known pumps in this ultra-high pres-
sure discharge range, and these difficulties could eventu-
ally make the pumps inoperative. These unique prob-
lems are due to the increased pressure drop across the
leakage path between the pump plunger and surrounding
cylinder, greater plunger forces and therefore higher
plunger friction, and the larger amount of heat added
to the liquefied gas during compression. The latter fact
leads to vapor flashoff from the liquid trapped in the
clearance space; that is, the portion of the pumping
chamber not taken up or filled by the plunger at the end
of the discharge stroke, after the pressure in the pumping
chamber has been released. Vapor flashoff in this un-
filled portion of the pumping chamber limits the amount
of liquid that can enter the pumping chamber during
the suction stroke, and may cause the pump to become
vapor bound and lose prime. The heretofore proposed
pumps have relatively large clearance spaces, which were
tolerable with discharge pressure below about 3,000 p.s.i.,
but cannot be tolerated when a pump must operate in
the ultra-high pressure range.

Another limitation of the heretofore known immersion
pumps is the requirement of a relatively high pressure
differential between the pumping chamber and the sur-
rounding liquefied gas, to open the suction valve. This
characteristic becomes critical in ultra-high pressure op-
eration since the increased vapor flashoff results in a higher
pressure in the pumping chamber at the end of the dis-
charge stroke. Thus, to maintain the same pressure
differential across the suction valve, it is necessary to
supply the liquefied gas in the surrounding container at
a relatively higher pressure. This requirement may in-
crease the operating costs of the system.

An additional problem not sufficiently overcome by the
previously proposed system for pumping volatile liquids
to high pressures is the substantial transmission of heat
from the atmosphere through the pump mounting as-
sembly into the container. This heat leak is partly due
to conduction and also results from splashing of the

liquefied gas against the pump mounting assembly, with
resultant wetting of the warm parts of this assembly,
thereby increasing the liquid evaporation rate. Because
of this wetting, the heat leak problem is particularly acute
when the immersion pump-liquid container assembly is
subject to considerable vibration and movement, or when
the pump is mounted substantially horizontally in the con-
tainer. In the latter case, the pump mounting assembly
is either directly immersed in the liquefied gas, or in
relatively close proximity thereto.

Still another disadvantage of the presently used im-
mersion pumps is the slow plunger speed necessary to
prevent excessive generation of frictional heat and to
avoid the previously discussed excessive pressure drop
across the suction valve. A relatively low plunger speed
required a larger diameter pump cylinder or body to
achieve a given pumping rate, and this in turn necessi-
tates a massive pump body resulting in a high rate of
heat transmission into the liquid both inside and outside
the pump and requiring a large opening in the container
wall for pump installation. The percentage of the liquid
which leaks between the plunger and cylinder wall also
increases at slow plunger speeds. Finally, the forces
imposed on the plunger and driving mechanism for a
given pumping rate are higher at slower speeds which
adds to the expense and massiveness of the entire as-
sembly.

One object of the present invention is to provide
a highly efficient immersion pump for pressurizing liquefied
gas to an ultra-high pressure;

Another object of the present invention is to provide
a reciprocating-type pump having a minimum clearance
space;

A still further object is to provide a reciprocating-type
immersion pump which requires a relatively low pressure
differential between the pumping chamber and the sur-
rounding liquefied gas, to open the suction valve;

An additional object is to provide an immersion pump-
liquid container assembly whereby heat leak through
the pump mounting assembly into the container is mini-
mized; and

Another object is to provide a compact immersion
pump which operates at a relatively high plunger speed
but is characterized by low frictional heat and low suction
valve pressure drop thereby achieving low cost, high effi-
ciency and operation dependability.

These and other objects and advantages of this as-
sembly will be apparent from the following description
and the accompanying drawings in which:

FIG. 1 is a view of a vertical section through a pump
according to the preferred embodiment of the present
invention.

FIGURE 1a is a view of a vertical section through the
inner end of the pump plunger and the plate valve.

FIG. 2 is a fragmentary view of a partial section
through the pump barrel and plunger, taken along line
2—2 of FIG. 1.

FIGS. 3—6 are cross-sectional views illustrating other
inlet valve assembly constructions.

The reciprocating pump of one embodiment of this
invention comprises an elongated pump body having a
pumping chamber at one end and an opening at the other
end. An inlet or suction valve protrudes into the pump-
ing chamber and controls the inlet port, which is situated
near the end of the pumping chamber opposite the open
end of the pump body. A discharge valve is also pro-
vided to control a discharge outlet passage communicat-
ing with the pumping chamber for discharge of ultra-
high pressure liquid therethrough. A reciprocating
plunger extends through the pump body opening, the
plunger having an inner end portion with a recessed

section or cavity which covers at least most of the inlet valve at the end of each discharge stroke. Alternately, the suction valve could be positioned within the pump inlet head so as not to protrude into the pumping chamber. In such case, the plunger inner end portion would not have the aforementioned recessed section or cavity. Thus, the clearance space in the pump of the present invention is appreciably reduced by having the plunger fill up a major part of the clearance space between the suction valve and the pumping chamber inner wall at the end of the pump stroke, after the plunger has pumped the liquefied gas to an ultra-high pressure and forced it through the discharge valve-controlled outlet passage. In the preferred embodiment of this invention, a plate-type suction valve is used since it permits a smaller clearance volume than prior art valves, requiring less clearance space while also providing a desired total opening area. Furthermore, a plate type suction valve employed in the manner to be described will provide a tighter seal than prior art ball-type suction valves and yet also reduce suction valve pressure drop. For example, it is estimated that the pressure drop inherent in the present invention will be about $\frac{2}{3}$ of the suction valve pressure drop of the pumps disclosed in the aforementioned applications using multiple ball type inlet valves. Thus, it can be seen that the present invention provides a reciprocating pump having a substantially smaller clearance volume than the heretofore proposed pumps; that is, less than about 4 percent and preferably about 2.5 percent of the pumping chamber working volume. This may be accomplished by separately and in combination providing a plunger with a recessed inner end portion which covers at least most of the plate-type inlet valve assembly at the end of the discharge stroke, thus filling up most of a relatively smaller clearance space. One advantage of minimizing this space may be illustrated by the fact that for an immersion pump operating at 10,000 p.s.i. discharge pressure and 15° F. subcooling, it is estimated that there will be a 3 percent loss in volumetric efficiency for each 1 percent increase in clearance volume.

A still further novel feature of the present invention provides an improved means for sealing the annular space between the pump body and the concentrically positioned entrance tube forming the walls of the opening through which the pump is inserted, so as to minimize heat leak and liquid evaporation.

Referring to the drawings, and especially to FIG. 1, the pump comprises an elongated body 10 preferably in the form of a barrel having a pumping chamber 12 in one end thereof, preferably the bottom, and an opening in the top or other end thereof. Mounted in the pumping chamber 12 is an inlet valve assembly 14 which controls an inlet port 16 in the bottom of pump body 10. A discharge valve 18 controls an outlet passage 20 from the pumping chamber 12. Discharge valve 18 is preferably a ball valve which is urged to its seat 22 by a spring 24 which is retained by a cap 26. From the side of the discharge valve 18, a discharge conduit 28 passes the pumped liquefied gas to conduit 30 in a manner to be described subsequently. Ball valve 18 is preferably metallic because of hardness and impact requirements. Suitable metals include stainless steel, nickel steel and hard surface aluminum.

The pump is provided with a reciprocating pumping element, the inner working or pumping portion of which is a plunger 32 which enters the end of the working or pumping chamber 12 which is opposite the head 34. The inner end of the plunger 32 has a recessed part or cavity machined therein, which covers a plate valve 36 at the end of the discharge stroke. The inner end surface of plunger 32 at this point of the pumping cycle is in close proximity with the inner surface of the head 34 and thereby minimizes the clearance volume within the pumping chamber. It is to be noted that in this embodiment

of the present invention, the clearance space or volume consists of essentially only that part of the space within the cavity 40 which is not filled by the plate valve 36 and is usually expressed as a percentage of the total pumping chamber working volume.

Plate valve 36 preferably comprises a substantially flat top portion 36a (which is preferably circular but may be rectangular; and the either hollow or solid) attached to a stem 36b and centrally guided by a bushing 36c which may be constructed of a low-friction self-lubricating material such as a glass-filled Teflon. Bushing 36c is supported by spider section 36d, the openings of which provide the necessary flow area for the suction liquid drawn into the pumping chamber 12. Plate valve 36 is lifted on the suction stroke of plunger 32 by the pressure differential across the valve, and closes on the discharge stroke. The valve seat surface 36e mates tightly with head 34. Both the valve 36 and head 34 are constructed of hardenable material having good low-temperature ductility and impact properties, such as nitrided AISI Type 304 stainless steel. Head 34 is preferably retained within pump body 10 by externally-threaded ring nut 34a. While the valve 36 may be retained from above by a cage similar to those employed by the aforementioned applications, it is preferable that stem 36b have an enlarged end 36f to act as a valve-stop, the valve lift being determined by the length of stem 36b extending downward past bushing 36c when the valve 36 is seated on head 34.

While the valve seating surface 36e is preferably flat as shown in FIG. 1, it may be beveled and seat against a sharp edge of head 34; or valve seating surface 36e may be beveled and seat against a like-beveled surface on head 34. These embodiments are depicted in FIGS. 3 and 4 respectively. In all these arrangements, the valve 36 is at least partially enclosed within the lower end of plunger 32 in order to reduce the pump clearance to an absolute minimum. Alternately, in those embodiments wherein the plunger would not have a recessed cavity, the plate valve would be partially enclosed within head 34. FIGS. 5 and 6 are examples of such constructions.

Plunger 32 preferably fits the bore of the working chamber fairly closely to provide an easy sliding fit. There should, however, be a small clearance at working temperatures to permit a slight liquefied gas flow along the plunger. A flow restricting liner or guide sleeve 38 is preferably disposed between the upper portion of body 10 and the plunger 32, the sleeve 38 having an internal cylindrical surface of a diameter similar to that of the pumping chamber 12. It was found that a diametrical clearance between the plunger 32 and the guide sleeve 38 of 0.002 to 0.0035 inch at the low operating temperature permits free movement of the plunger, and at the same time gives reasonably low leakage rates. Although self-lubricating material, for example, a bonded graphite or carbon is suitable for the sleeves 38 of pumps operating with discharge pressures in the range of about 3,000 p.s.i., such material has been found to be unsuitable for service in ultra-high pressure pumps because of the increased strength and ductility requirements. As a consequence, lubricant-containing metals are preferred as the sleeve material for the pump of the present invention. Suitable materials include high graphite content sintered bronze, porous bronze impregnated with plastics such as polymerized tetrafluoroethylene, or chlorotrifluoroethylene, and porous bronze impregnated with a low pour point oil of a character compatible with the liquid pumped. These materials provide a low friction factor between the plunger 32 and the sleeve 38 of less than about 0.35. A low thermal conductivity plastic spacer 40 is positioned in the sleeve 38 to restrict the flow of heat generated by friction due to plunger movement. The effect of the plastic spacer 40 is to break up the otherwise smooth temperature gradient between the warm end of the pump and the pumping chamber 12, thereby reducing the temperature of the walls in the pumping chamber. The

spacer 40 is preferably located just above the inner end of the plunger 32 when it has reached the end of the suction stroke. Suitable materials for the plastic spacer 40 include polytetrafluoroethylene, glass cloth reinforced melamine resin, and cloth or fiber reinforced phenol formaldehyde resin for processing inert gases such as nitrogen. Polytetrafluoroethylene is also suitable for oxygen service. Also, in order to minimize heat conduction from the cold or operating end of the pump, the upper portion of body 10 is made as long and thin as is consistent with adequate strength. Preferably, the material employed also has a low thermal conductivity to minimize longitudinal heat leak.

For venting the liquefied gas flowing along the plunger 32 from the pumping chamber 12, a passage means is provided preferably in the form of one or more vent passages or holes 42 in the upper portion of body 10. These holes 42 are preferably located as far towards the warm end of the pump as practical in order to provide the longest possible leakage path. This is because a long leakage path minimizes the amount of liquid leakage; and since liquid changes to gas with great increase in volume, the actual quantity of leakage loss is kept very small. The vent holes 42 are closed by the plunger, but since there must be some small clearance for easy operation, a small amount of the gasified material pumped will escape from the pumping chamber 12 through the vents 42 back to the container in which the pump body is immersed. Thus, the pump "blow-by" is not lost to the atmosphere but recycled for repumping.

The warm end of the sleeve 38 is retained by a stuffing-box 44 comprising a gland 44a which forms a bottom for the stuffing box section of the upper portion of body 10. This section may be filled with packing rings 44b which are retained by a gland follower 44c and a packing nut 44d. The warm end of the plunger 32 is secured to a reciprocating mechanism by means not shown. A suitable type of warm end packing may be composed of asbestos and graphite composition rings which provide a gas seal. When the liquefied gas to be pumped is liquid oxygen, a suitable type of packing at 44b may be similar to that described in United States Patent No. 2,292,543 to J. F. Patterson.

The upper portion of pump body 10 is preferably surrounded by a vacuum jacket 46 which reduces heat leakage into the cold region of sump chamber 48 into which the pump body is inserted. Vacuum jacket 46 preferably surrounds the pump body 10 for about the length of annular heat exchanger 50.

Annular heat exchanger 50 surrounds the upper portion of pump body 10 for about 16 inches of the pump barrel, below pump mounting flange 52, and comprises an annular space surrounding the upper portion of the pump body. The reduced wall thickness of the cylinder and the fins 54 shown in FIG. 2 increase the heat transfer. Alternately to fins 54, the space may be provided with helically wound strips 51 as shown in FIGURE 1 which may be corrugated, wrinkled or folded, to act as turbulence promoters.

The plunger 32 is preferably constructed of a precipitation-hardenable low conductive stainless steel. If desired, conduction may be further reduced by making the plunger of thick-walled tubing with plugs welded into both ends. For some plunger materials, for example Type 18-8 austenitic stainless steel, the plunger 32 is plated with 0.005 inch thick chromium plate to improve the surface hardness and wearing characteristics of the stainless steel and to reduce friction.

The pump is installed in a sump 48 by bolting mounting flange 52 against sump flange 56. The sump consists of inner container 58 and outer container 60 with the space therebetween vacuum insulated with a high quality insulation such as alternate layer, aluminum foil and glass fiber material as described in Patent No. 3,007,596 to L. C. Matsch and copending patent application Serial No.

824,690 to L. C. Matsch filed July 2, 1959 and provided with liquid inlet and gas phase connections. Gas phase connection 45 is positioned just below the heat exchanger 50 and vacuum jacket 60. A lower extension of the sump perforated head 47 forms a space for an adsorbent such as Calcium "A" Zeolite in accordance with U.S. Patent 2,900,800. Sump liquid inlet connection 49 is located to pass through this space. The pump body can be withdrawn from the sump without breaking the vacuum in either vacuum space or distributing internal connections.

Liquid is drawn from the sump 48 through the hollow nut 34a, through suction port 16 and around valve 36 into the pumping chamber 12. On the discharge stroke, liquid is expelled through discharge valve 18 into the discharge tube 28. The fluid then passes through connection 64 into the annular heat exchanger 50.

As the cold fluid flows through heat exchanger 50 upward and around this section of the pump, it intercepts and removes heat which otherwise would reach the pumping chamber. Thus, the exchanger chills the upper region of the pump below the mounting flange 52 and thereby reduces substantially the temperature increment between the warm mounting flange 52 and the pump chamber 12. In addition to removing atmospheric heat passing down the pump by conduction, the discharge fluid helps remove friction heat generated in the upper section of the pump between the plunger 32 and the liner 38.

From the heat exchanger 50 the pump discharge fluid passes into discharge line 30 and thence to usage. The line 66 shown opposite the discharge line 30 extends through the packing housing flange 68, pump flange 52 and through the vacuum jacket gas tightly and into the gas phase space of the sump, and provides a connection for a safety valve, bursting disk, and a pressure gauge.

A vent conduit 42 is located in the packing extension flange 68 and serves to vent the blowby or fluid leakage, as previously described, between the plunger 32 and the liner 38. This vent and its location are beneficial by relieving pressure from the plunger packing 44, and by heat exchanging the cold vent gas with the entire length of the pump body to reduce heat leakage and help remove friction heat.

The vacuum jacket around the upper (warm) region of the pump prevents convection between cold and warm levels along the outside of the pump body, and it also prevents sump (or tank) liquid from coming in contact with the warm mounting flange. An effective insulator is very important when pumping hydrogen or helium because it is imperative to minimize heat transmission into the sump liquid.

Although the upward flowing vent gas and discharge fluid greatly reduce the temperature of the pump inside the sump, it is unavoidable that the upper portions of the pump near the mounting flange will be relatively warm due to solid conduction from the flanges and upper portions of the pump. Sump vapor coming in contact with such warm parts will circulate by convection and transfer heat to the colder areas in the sump. The addition of a heavy-walled heat exchanger jacket below the mounting flange tends to raise the temperature of the upper portion of the pump due to increased cross section area for longitudinal heat conductance and to radial heat conductance from the relatively warm compressed discharge fluid. Thus, the heat exchanger structure would normally aggravate the convection problem. However, the vacuum jacket 46 insulates the heat exchanger from the sump vapor and eliminates this problem. The outer wall of the jacket is relatively thin and does not conduct an appreciable quantity of heat. Similarly, the vacuum jacket prevents sump liquid from contacting the warm parts of the pump in the event that the liquid splashes or otherwise rises above normal level.

In order to exclude effectively the cold sump fluids from the warm regions surrounding the pump the vacuum

jacket must make a moderately good seal against the sump wall at the lower (cold) end of the jacket. This must also be a sliding seal so that the pump can be easily installed in and removed from the sump. One method of effecting this seal is shown to provide the edge of the jacket below the bottom closure for the vacuum space with a flared portion so that it makes a forced, sliding fit into the sump. Uniform close clearance is thus maintained between the jacket and the sump wall at this point. A preferred method of sealing is to provide a plastic O-ring (e.g., Teflon or equivalent) around the lower end of the jacket.

The space between the vacuum jacket 46 and the sump wall 58 must be very narrow throughout the length of the jacket in order to minimize or eliminate convection in this space. The diametral clearance between the two walls is preferably .015 in.

The benefits of the vacuum space are achieved simply by evacuating the space; no filler, reflective shields or reflective surfaces are used. A "warm" vacuum of about 1 to 10 microns is adequate. In practice, it is important that the vacuum jacket be removable from the sump without breaking the vacuum in order to facilitate pump maintenance. This permits construction of the jacket as a factory-sealed unit with all joints sufficiently leaktight to obtain dependable, high-vacuum service. An adsorbent or getter may be sealed in the jacket to assist in maintaining a good vacuum.

In combination, the above-described features provide a more efficient pump for cryogenic service than heretofore possible. The unavoidable generation of friction heat by the moving plunger is minimized in that portion of the pump which contacts suction liquid, and is centered primarily in an intermediate section of the pump which is in thermal contact with the discharge fluid. The discharge liquid, together with the fluid unavoidably leaking by the plunger, absorbs and removes the major portion of the friction heat and also reduces heat inleak by chilling the intermediate section. A vacuum jacket removable with the pump body prevents suction fluids from contacting the warm end parts and the intermediate section of the pump where friction heat is being removed. Spacing the packing well beyond the chilled section on a housing extension and venting the plunger leakage afford warm temperature and low pressure at the packing, thereby reducing friction.

It is contemplated that various modifications of the pumping apparatus may be without departing from the spirit and scope of the invention herein described.

What is claimed is:

1. A reciprocating pump for liquefied gases having a boiling point below 273° K., said pump comprising an elongated pump body having a pumping chamber therein adjacent one end and an opening at the other end; an inlet valve controlled port near the end of said pumping chamber opposite said opening; an inlet valve assembly comprising a plate valve positioned to control flow through said port and having a portion extending into said pumping chamber; a reciprocating plunger extending through said opening in the pump body having an inner pumping end cavity operable in said pumping chamber and constructed and arranged for enclosing the portion of said plate valve extending into said pumping chamber and substantially minimizing the clearance volume within said pumping chamber when at the end of a discharge stroke; a discharge valve and discharge valve-controlled port near the end of said pumping chamber; warm end packing means for the portion of the plunger extending through said opening, said packing means being spaced at a substantial distance from said pumping chamber; and a flow restrictive sleeve in said pump body closely fitting said plunger.

2. A reciprocating pump according to claim 1 wherein insulating means surrounds a portion of said pump body adjacent the warm end thereof.

3. A reciprocating pump according to claim 1 wherein a heat exchanger passage surrounds a portion of said pump body adjacent the warm end thereof and is interposed in said discharge valve controlled outlet passage.

4. A reciprocating pump for liquefied gas according to claim 1 in which said flow restricting sleeve comprises portions made of a lubricant-containing metal and at least one low thermal conductivity plastic spacer interposed between said portions so as to minimize the flow of frictional heat along the sleeve.

5. A reciprocating pump for liquefied gases according to claim 1 in which said flow restricting sleeve comprises portions made of a lubricant-containing metal which affords a friction factor between said plunger and the sleeve of less than about 0.35; said plunger and sleeve being sized so as to provide a diametrical clearance of about 0.0020 to 0.0035 inch at the low operating temperature.

6. A reciprocating pump according to claim 1 wherein said plunger is hollow, constructed of low-conductive material, and sealed at both ends.

7. A reciprocating pump according to claim 6 wherein said pump body has a bore receiving said hollow low-conductive plunger with a clearance forming an extended blowby passage in heat exchange with said plunger and a plunger covered vent is provided in the pump body wall adjacent the warm end thereof.

8. A reciprocating pump according to claim 2 wherein said insulating means comprises a vacuum jacket constructed to maintain a low positive pressure below atmospheric of about 10 microns of mercury absolute.

9. A reciprocating pump according to claim 8 wherein said pump body is installed in a sump and the diametrical clearance between the vacuum jacket and the inner wall of said sump is sufficiently close in order to provide a sliding seal between said vacuum jacket and said sump, and including sealing means to effect such seal.

10. A reciprocating pump according to claim 3 wherein said heat exchanger comprises an outer wall, an inner wall having radially outward projecting heat transfer fins, and a fluid space therebetween connecting to said discharge valve controlled outlet passage.

11. A reciprocating pump according to claim 3 wherein said heat exchanger comprises an outer wall, an inner wall having outward projecting helically wound heat transfer strips, and a fluid space therebetween connecting to said discharge valve controlled outlet passage.

12. A reciprocating pump for liquefied gases having a boiling point below 273° K., said pump comprising an elongated pump body insertable in a sump chamber adapted to receive said pump body, said pump body having a pumping chamber therein adjacent one end and an opening at the other end; an inlet valve controlled port near the end of said pumping chamber opposite said opening; an inlet valve assembly comprising a circular plate valve positioned to control flow through said port and having a portion extending into said pumping chamber; a reciprocating plunger extending through said opening in the pump body having an inner pumping end cavity operable in said pumping chamber and constructed and arranged for enclosing the portion of said plate valve extending into said pumping chamber and substantially minimizing the clearance volume within said pumping chamber when at the end of a discharge stroke; a discharge valve and discharge valve-controlled port near the end of said pumping chamber; warm end packing means for the portion of the plunger extending through said opening, said packing means being spaced at a substantial distance from said pumping chamber; and a flow restrictive sleeve in said pump body closely fitting said plunger; a heat exchanger surrounding a portion of said pump body adjacent the warm end thereof and a vacuum jacket surrounding said heat exchanger, said heat exchanger comprising inner and outer walls and a fluid space therebetween connecting to said discharge valve con-

trolled outlet passage; and a sump chamber comprising inner and outer walls and having an open end for receiving said pump body, the insertion of which fluid-tightly seals said open end, the interior of said sump chamber communicating with a fluid inlet conduit and the pump inlet valve-controlled port.

13. A reciprocating pump for liquefying gases having a boiling point below 270° K., said pump comprising an elongated pump body having a pumping chamber therein adjacent one end and an opening at the other end; an inner valve control port near the end of said pumping chamber opposite said opening; an inlet valve assembly comprising a plate valve positioned to control flow through said port, said plate valve comprising a substantially flat top portion extending into said pumping chamber, a spider positioned in said inlet valve controlled port, a bushing supported by said spider, and a stem attached to the top portion and centrally guided by said bushing and having an enlarged end so constructed and arranged to abut against said bushing at the end of a plunger suction stroke so as to act as a valve-stop; a reciprocating plunger extending through said opening in the pump body having an inner pumping end cavity operable in said pumping chamber and constructed and arranged for enclosing the portion of said plate valve extending into said pumping chamber and substantially minimizing the clearance volume within said pumping chamber when at the end of a discharge stroke; a discharge valve and discharge valve-controlled port near the end of said pumping chamber; warm end packing means for the portion of the plunger extending through said opening, said packing means being spaced at a substantial distance from said pumping chamber; and a flow restrictive sleeve in said pump body closely fitting said plunger.

14. A reciprocating pump for liquefied gases having a boiling point below 270° K., said pump comprising an elongated pump body having a pumping chamber therein adjacent one end and an opening at the other end; an inlet head having an inlet valve-controlled port near the end of said pumping chamber opposite said opening; an inlet valve assembly comprising a plate valve positioned to

control flow through said port, said plate valve comprising a top portion at least partially enclosed within said inlet head, and a stem attached to the top portion extending into said inlet head and having an enlarged end so constructed and arranged to abut against a section of said inlet head at the end of a plunger suction stroke so as to act as a valve-stop; a reciprocating plunger extending through said opening in the pump body having an inner pumping end operable in said pumping chamber and constructed and arranged for minimizing the clearance volume within said pumping chamber when at the end of a discharge stroke; a discharge valve and discharge valve-controlled port near the end of said pumping chamber; warm end packing means for the portion of the plunger extending through said opening, said packing means being spaced at a substantial distance from said pumping chamber; and a flow restrictive sleeve in said pump body closely fitting said plunger.

15. A reciprocating pump for liquefied gases according to claim 14 wherein a spider is positioned within said inlet valve-controlled port with said stem extending through said spider and abutting against said spider at the end of a plunger suction stroke.

16. A reciprocating pump for liquefied gases according to claim 14 wherein a spider is positioned within said inlet valve-controlled port and a bushing is supported by said spider with said stem extending through said spider and abutting against said bushing at the end of a plunger suction stroke.

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3,011,450	12/1961	Tyree	62—55
3,016,717	1/1962	Gottzmann	62—55

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