

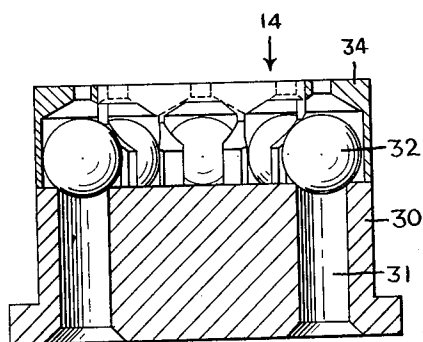
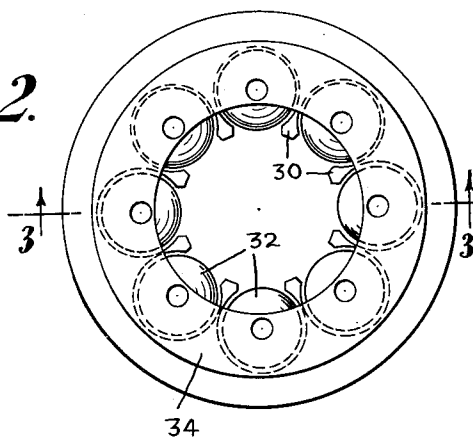
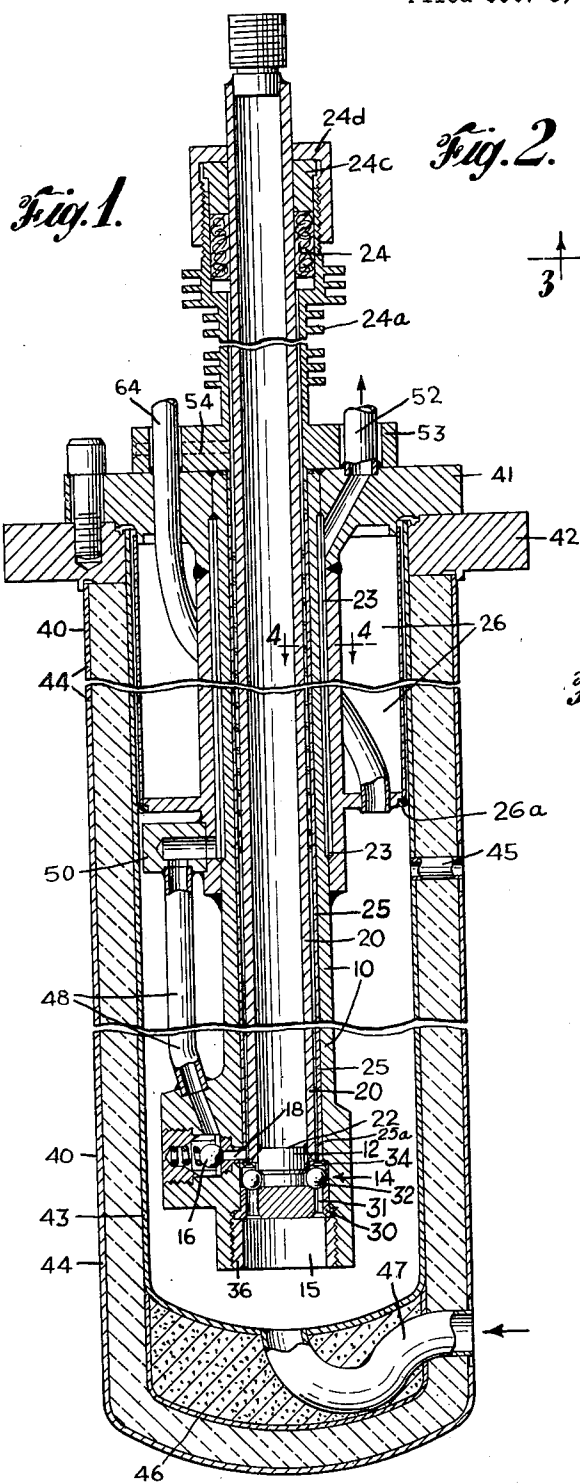
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HIGH-PRESSURE PUMP FOR CRYOGENIC FLUIDS

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HIGH-PRESSURE PUMP FOR CRYOGENIC FLUIDS
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This invention relates to apparatus for storing and pumping a volatile liquid having a boiling point temperature at atmospheric pressure materially below 273° K., and more particularly to a reciprocating pump for pumping 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. The present invention is a continuation-in-part of my copending applications Serial No. 692,311, filed October 25, 1957, now U.S. Patent 3,016,717, and Serial No. 12,130, filed March 1, 1960, now abandoned.

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 considerations, arise which tend to decrease the efficiency of the heretofore known pumps in this ultra-high pressure discharge range, and these difficulties could eventually make the pumps inoperative. These unique problems 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 unfilled 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 surrounding liquefied gas, to open the suction valve. This characteristic becomes critical in ultra-high pressure operation, since the increased vapor flashoff results in a higher pressure in the pumping chamber at the end of the discharge 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 increase the operating costs of the system.

An additional problem not sufficiently overcome by the previously proposed systems for pumping volatile liquids to high pressures is the substantial transmission of heat from the atmosphere through the pump mounting assembly 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 result-

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ant 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 container. 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 immersion 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 requires a larger diameter pump cylinder or body to achieve a given pumping rate, and this in turn necessitates a massive pump body resulting in a high rate of heat transmission into the liquid bath 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 assembly.

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 surrounding 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 minimized.

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 efficiency and operation dependability.

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

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

FIGURE 2 is a plan view of the improved valve means; FIGURE 3 is a section taken along the line 3—3 of FIGURE 2; and

FIGURE 4 is a fragmentary view of a partial section through the pump barrel and plunger, taken along line 4—4 of FIGURE 1.

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 pumping 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 provided to control a discharge outlet passage which communicates 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 interfitting with the inlet valve. Thus, the clearance space in the pump of the present invention is appreciably reduced by having the plunger fill up at the end of the pump stroke a major part of the clearance space into which the suction valve projects into the pumping chamber. In the preferred

embodiment of this invention, multiple suction valves are used since they permit a smaller combined volume than a single valve and thus require less clearance space while also providing a desired total opening area. Ball type inlet and discharge valves are preferred in the immersion pump of the present invention because they provide the most satisfactory and trouble-free pressure seal at ultra-high pressures. 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 8 percent. 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.

Another advantage of using multiple ball-type inlet or suction valves is that a lower pressure drop is required to lift small multiple balls from their seats during the suction stroke than is required to lift a large single ball. Pressure drop across the suction valve is critical in low temperature liquefied gas pumps because the liquid has a strong tendency to flash, thereby reducing the capacity of the pump and perhaps causing loss of prime. An inlet valve port with a large cross-sectional area is advantageous in minimizing this pressure drop. With a single ported valve, a large port size requires a large heavy-weight ball valve; whereas, with multiple ports, the same inlet port area may be obtained by using several valves, all of which are of uniform minimum weight. Furthermore, light-weight multiple balls are subjected to less impact during operation, and thus result in a longer seat life. The required pressure differential may be further reduced in the pump of the present invention by using hollow ball-type inlet valves.

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 walls of the opening in the sump chamber through which the pump is inserted, so as to minimize heat leak and liquid evaporation.

Referring now to the drawings, and specifically 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 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 clearance space therein. The opposite or upper end portion of the plunger 20 is provided with warm end packing means 24 spaced a substantial distance from the pump mounting flange 41 in an extension 24a of the pump body 10. The extension 24a carrying packing gland absorbs sufficient heat to avoid freezing water vapor or condensable gases in the packing. A flow restricting and plunger guide sleeve 25 is mounted in the pump body 10 closely fitting the plunger with a sliding clearance. The upper end portion of the pump body 10 is surrounded by a vacuum jacket 26.

Plunger 20 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. The flow restricting liner or guide sleeve 25 is preferably disposed between the body portion 10 and the plunger 20, the sleeve 25 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 20 and the guide sleeve 25 of 0.0020

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 25 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 20 and the sleeve 25 of less than about 0.35. A low thermal conductivity plastic spacer 25a is positioned in the sleeve 25 to restrict the flow of heat generated by friction due to plunger movement. The effect of the plastic spacer 25a 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 25a is preferably located just above the inner end of the plunger 20 when it has reached the end of the suction stroke. Suitable materials for the plastic spacer 25a 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 body portion 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 20 from the pumping chamber 12, a passage means is provided preferably in the form of one or more vent passages or holes 54 in the upper part of the body portion 10. These holes 54 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 or weight of leakage loss is kept very small. The vent holes 54 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 54.

The warm end of the sleeve 25 is retained by a mounting flange 53 at the lower end of the extension 24a. A packing box portion of the extension 24a is filled with the packing means 24 which is retained by a gland follower 24c and a packing nut 24d. The warm end of the plunger 20 is secured to a reciprocating mechanism by means not shown. A suitable type of such warm end packing may be composed of asbestos and graphite composition rings which provide a sliding gas seal. When the liquefied gas to be pumped is liquid oxygen, a suitable type of packing may be similar to that described in United States Patent No. 2,292,543 to J. F. Patterson.

The upper end portion of the pump body 10 is surrounded by a vacuum jacket 26 which reduces heat leakage into the cold region of sump chamber 40 into which the pump body is inserted. Vacuum jacket 26 preferably surrounds the pump body for about the length of annular heat exchanger 23.

An annular heat exchanger 23 surrounds the upper portion of the pump barrel 10 about 16 inches below the mounting flange 41, and comprises an annular space surrounding the upper end of the pump body. The re-

duced wall thickness of the cylinder and the fins 27 shown in FIG. 4 increase the heat transfer. Alternatively to fins 27, the space may be provided with helically wound strips which may be corrugated, wrinkled or folded, to act as turbulence promoters.

The inlet valve assembly 14 is preferably multiple and as shown in FIGS. 2 and 3 comprises an annular valve seat plate or disk 30 having a plurality of inlet ports 31 drilled therein, preferably eight as shown in an annular row. Each ball valve 32 is assembled over a corresponding port in the valve seat, and a valve retainer cage 34 is provided on the valve seat 30 about the balls and secured in place on assembly. The cage fills the space between the balls and the pump barrel wall and provides a central opening for the plunger end. The valve assembly is slipped into the lower end of the pump cylinder and secured by an externally threaded annular or hollow nut 36 threaded into the end of the pump body to bear against the flanged lower end of the valve seat. This valve arrangement readily accommodates the requisite number of $\frac{3}{8}$ – $\frac{1}{2}$ inch diameter balls for optimum simplicity and pressure drop.

The plunger 20 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 substantially as shown. The ball valves 32 being arranged around the periphery of the valve body, the end 22 of the plunger 20 is stepped down so that it fills the space inside the circle of valves and annular ball cage 34. The clearance volume obtained by this construction and cooperation between the plunger end and valve assembly is reduced to about 3½%.

For some plunger materials, for example Type 18–8 stainless steel, the plunger 20 is plated with about .005 inch thick chromium plate to improve the surface hardness and wearing characteristics of the stainless steel and to reduce friction.

The diameter of the bottom 5½ inches is reduced about .003 inch to avoid severe rubbing between this section of the plunger and the liner due to any unavoidable curvature or bow in the plunger. Relieving the end of the plunger is highly beneficial in reducing the generation of friction in those parts of the pump in contact with suction liquid.

The pump is installed in a fore-chamber or sump 40 by bolting mounting flange 41 against sump flange 42. The sump consisted of inner container 43 and outer container 44 with the space therebetween vacuum insulated with a high quality insulation such as alternate layer aluminum foil and glass fiber material as described in copending patent application Serial No. 597,947 to L. C. Matsch filed July 16, 1956, now U.S. Patent No. 3,007,596, and Serial No. 824,690 to L. C. Matsch filed July 2, 1959, now U.S. Patent No. 3,009,601, and provided with liquid inlet and gas phase connections. Gas phase connection 45 is positioned just below the heat exchanger 23 and vacuum jacket 26. A lower extension of the sump with perforated head 46 forms a space for an adsorbent such as Calcium "A" Zeolite in accordance with U.S. Patent 2,900,800. Sump liquid inlet connection 47 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 disturbing internal connections.

Liquid is drawn from the sump 40 through the hollow nut 36, through suction ports 31 and around ball valves 32 into the pumping chamber 12. On the discharge stroke, liquid is expelled through discharge port 18 around discharge valve 16 into the discharge tube 48. The fluid then passes through connection 50 into the annular heat exchanger 23.

As the cold fluid flows through heat exchanger 23 upward and around this section of the pump, it intercepts and removes heat which otherwise would reach the pump-

ing chamber. Thus, the exchanger chills the upper region of the pump below the mounting flange 41 and thereby reduces substantially the temperature increment between the warm mounting flange 41 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 20 and the liner 25.

From the heat exchanger 23 the pump discharge fluid passes into discharge line 52 and thence to usage. The line 64 shown opposite the discharge line 52 extends through the packing housing flange 53, pump flange 41 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.

The vent conduit 54 is located in the extension flange 53 and serves to vent the blowby or fluid leakage as previously described. The vented material may be blown to atmosphere or conducted to receiving means as desired.

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 extension 24a. 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 26 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 cold sump fluids from the warm upper 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 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 as shown at 26a.

The space between the vacuum jacket 24 and the sump wall 43 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 about .015 in.

Adequate benefits of the vacuum space are achieved simply by evacuating the space 26; 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 the pump assembly and with all joints sufficiently leak tight to obtain dependable, high-vacuum service. An adsorbent or getter may be sealed in the lower end of the jacket to insure 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 assembly 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.

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 positioned to control flow through said port having an inlet valve protruding into said pumping chamber such that a space is provided at the lower end of said pumping chamber not occupied by the valve assembly; a reciprocating plunger extending through said opening in the pump body having an inner pumping end portion operable in said pumping chamber and constructed and arranged to interfit with said inlet valve to substantially fill said space 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 such that the liquid expelled through said discharge valve-controlled port passes through said heat exchanger.
4. A reciprocating pump for liquefied gases 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 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 positioned to control flow through said port having an inlet valve protruding into said pumping chamber such that a space is provided at the lower end of said pumping chamber not occupied by the valve assembly; a reciprocating plunger extending through said opening in the pump body having an inner pumping end portion operable in said pumping chamber and constructed and arranged to interfit with said inlet valve to substantially fill said space 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 controlled outlet passage; and a sump chamber comprising inner and outer walls and having an open end for receiving said pump body and said vacuum jacket such that the insertion of said pump body fluid tightly seals said open end and such that said vacuum jacket is removable from said sump without breaking the vacuum therein, the interior of said sump chamber communicating with a fluid inlet conduit and the pump inlet valve-controlled port.

13. 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; multiple inlet valves in said pumping chamber and surrounding a central space therein, and an inlet valve-controlled port near the end of said pumping chamber opposite said opening; a discharge valve and a discharge valve-controlled outlet passage from the pumping chamber; a reciprocating plunger extending through said opening in the pump body having an inner pumping end portion operable in said pumping chamber and having a reduced end of a size and shape for substantially filling said central space at the end of the discharge stroke; 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 according to claim 13 wherein insulating means surrounds a portion of said pump body adjacent the warm end thereof.

15. A reciprocating pump according to claim 13 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.

16. A reciprocating pump for liquefied gases according to claim 13 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.

17. A reciprocating pump for liquefied gases according to claim 13 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.

18. A reciprocating pump according to claim 13 wherein said plunger is hollow, constructed of low-conductive material, and sealed at both ends.

19. A reciprocating pump according to claim 18 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.

20. A reciprocating pump according to claim 14 wherein said insulating means comprises a vacuum jacket constructed to maintain a low positive pressure below atmospheric about 10 microns of mercury absolute.

21. A reciprocating pump according to claim 20 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.

22. A reciprocating pump according to claim 15 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.

23. A reciprocating pump according to claim 15 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.

24. 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 positioned to control flow through said port having multiple inlet valves protruding into said pumping chamber such that a space is provided at the lower end of said pumping chamber not occupied by the valve assembly; a reciprocating plunger extending through said opening in the pump body having an inner pumping end portion operable in said pumping chamber and constructed and arranged to interfit with said multiple inlet valves to substantially fill said space 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.

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