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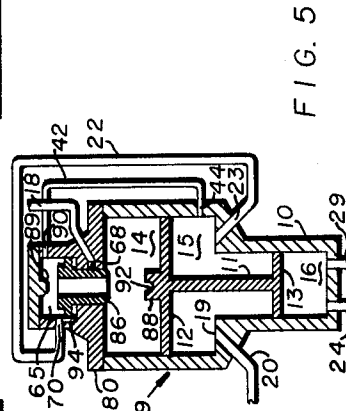
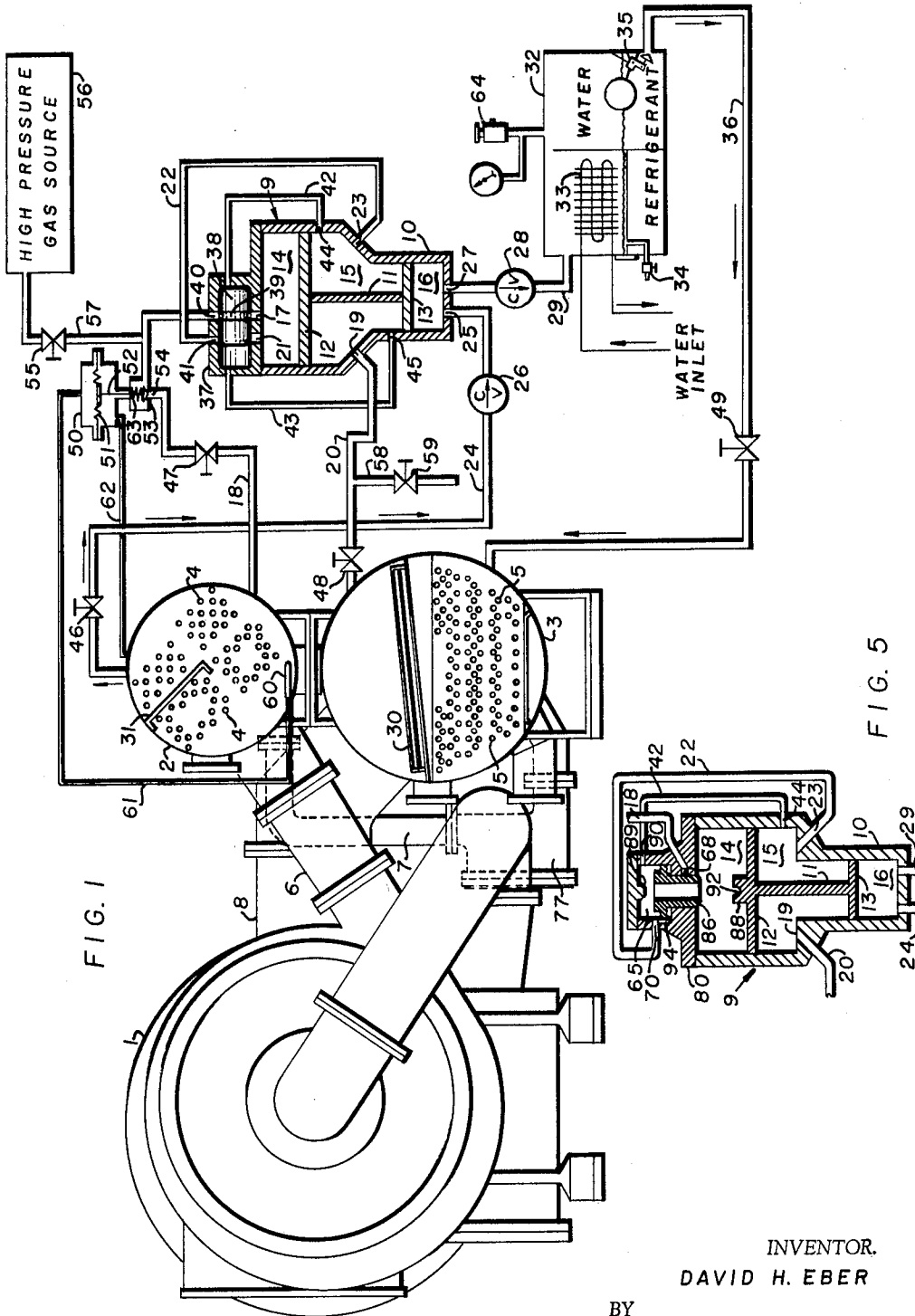
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PURGING APPARATUS FOR REFRIGERATION SYSTEM

Filed Sept. 29, 1964

2 Sheets-Sheet 1



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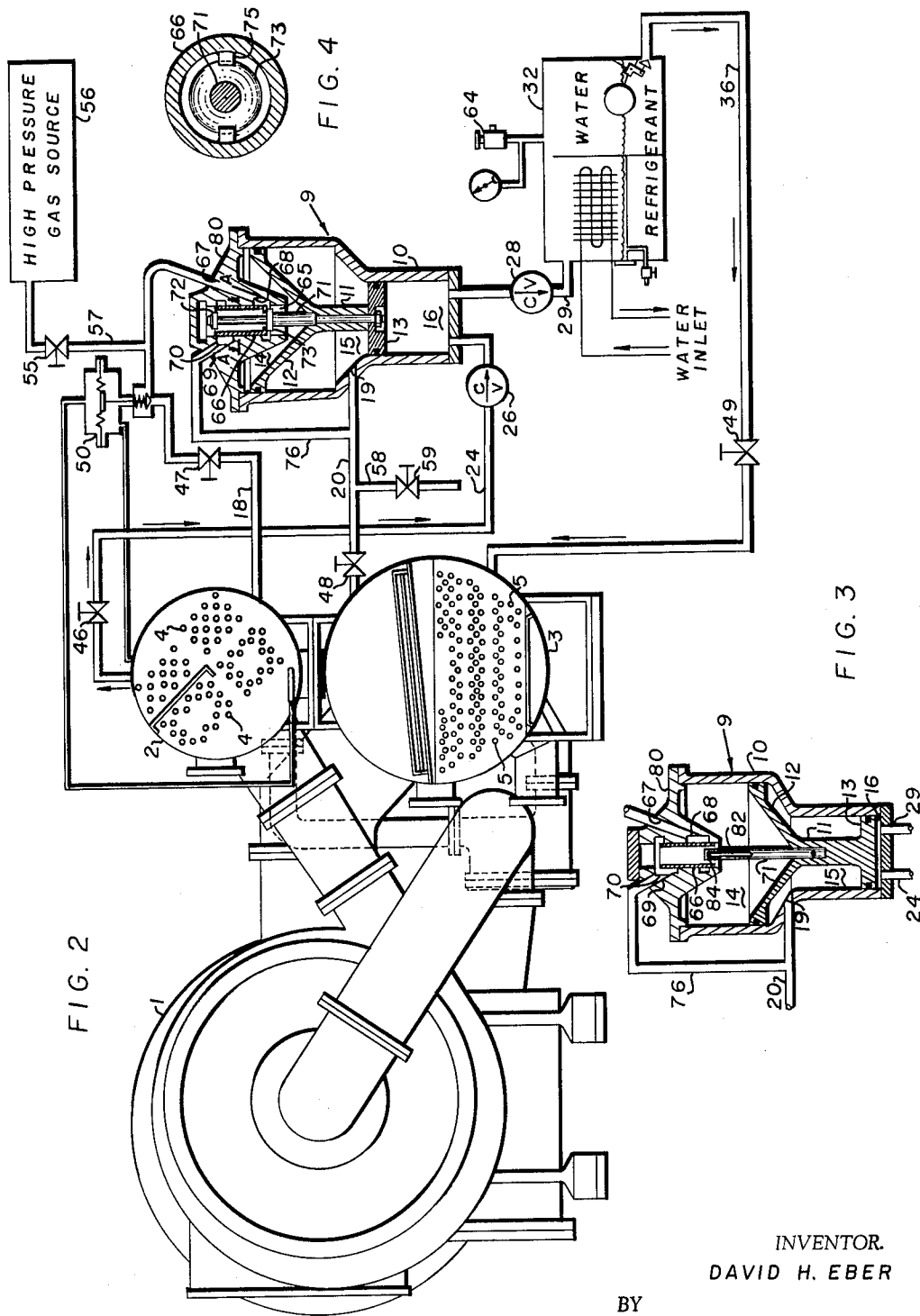
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2 Sheets-Sheet 2



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PURGING APPARATUS FOR REFRIGERATION SYSTEM

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This invention relates generally to refrigeration systems, and more particularly to a centrifugal refrigeration system incorporating new and improved means for purging noncondensable gases therefrom.

Noncondensable gases, such as air, which accumulate in the upper part of the system condenser, are conventionally removed by a motor-driven purge pump of the reciprocating type. This type of purge apparatus has several disadvantages: an external source of electric power and attendant controls are required to energize the pump motor; the use of an oil-lubricated pump results in oil contamination of the refrigerant—noncondensable gas mixture being pumped and thus requires the installation of an oil separator in the purge system; and the shaft seals on the reciprocating pump are a source of air in-leakage.

It is, therefore, a primary object of this invention to provide a refrigeration system with a new and improved purging apparatus which eliminates these difficulties.

A further object of the invention is to provide purging apparatus having a purge pump which is actuated by the pressure differential in the refrigeration system of which it is a part, and which thus requires no external power source.

A further object of this invention is to provide a purge pump of the free-piston type which requires no oil lubrication and which may be hermetically incorporated into the refrigeration system.

Another object of the invention is to provide a refrigeration system purge pump of the aforesaid type wherein the flow of pressurizing gas to and from the driving portion of the pump is controlled by a valve actuated by the pressure differential within the system.

A fifth object of the invention is to provide a refrigeration system purge pump of the aforesaid type which is automatically actuated in response to an increase in condenser pressure caused by accumulations of noncondensable gases in the condenser.

A further object of the invention is to provide a refrigeration system purge apparatus of the aforesaid type arranged for operation by a source of gas under pressure which is external of the refrigeration system.

These and other objects and advantages of the invention will be clearly apparent from the following description made with reference to the accompanying drawings, in which:

FIGURE 1 is a schematic illustration of a typical centrifugal refrigeration system incorporating the purging apparatus of the invention;

FIGURE 2 is a schematic view similar to FIGURE 1 showing a modified form of the purge pump and pump control valve;

FIGURE 3 is a schematic view of a modification of the purge pump and pump control valve arrangement of the invention;

FIGURE 4 is a cross-sectional view of the control valve piston taken along line A—A of FIGURE 2; and

FIGURE 5 is a schematic view of a further modification of the purge pump and control valve combination.

With reference to FIGURE 1 of the drawings, the refrigeration system includes a centrifugal compressor 1, condenser 2, and evaporator 3. Conduit 6 connects the compressor discharge to condenser 2 and conduit 7 interconnects the condenser outlet and the inlet of evaporator 3.

Pressure reducing valve 77 located in conduit 7 divides the system into a high pressure side upstream thereof and a low pressure side downstream thereof. Evaporator 3 is connected to the compressor inlet by suction line 8. The condenser and evaporator are of the shell and tube type. The condenser contains a tubing bundle, generally designated by reference numeral 4, through which cooling water circulates. The fluid to be cooled circulates through tubes 5 in the evaporator. An eliminator 30 is located in the evaporator and serves to separate entrained droplets of refrigerant liquid from the vapor rising towards the inlet of suction tube 8. Baffle plate 31 in the condenser 2 serves to shield the upper portion of the condenser from the incoming compressed gas so as to form a zone of low turbulence in which noncondensable gases collect. In operation, the compressor 1 discharges refrigerant gas under pressure into the condenser shell through conduit 6. Condensed refrigerant flows from the condenser through conduit 7 and pressure reducing valve 77 to the evaporator 3 where the refrigerant boils to produce the desired cooling effect. Refrigerant gas generated in the evaporator is drawn into the compressor through suction line 8.

The novel purge means of the invention consists of a free piston pump 9 comprising a housing 10 containing piston 11. The piston 11 has a driving end portion 12 of relatively large diameter and a pumping end portion 13 of smaller diameter. The end portions 12 and 13 of the piston 11 form three chambers 14, 15 and 16 within the housing 10. The pump 9 is motivated by applying the pressure differential which exists between the condensing or high side pressure and the evaporating or low side pressure of the refrigeration system across driving end portion 12 of piston 11. High pressure refrigerant gas is admitted to chamber 14 through inlet port 17 which is connected to the condenser 2 by means of conduit 18. Chamber 15, on the underside of driving piston portion 12, is continuously vented to the low pressure side of the system through port 19. Port 19 may be connected to any point in the low pressure side of the system, e.g., suction line 8 or evaporator 3. As a matter of piping convenience in the system illustrated, port 19 is connected to evaporator 3 through conduit 20. The connection between conduit 20 and the evaporator is made at a point above eliminator 30 in the vapor space at the upper portion of the evaporator.

Chamber 14 also contains an exhaust port 21 through which high pressure gas is vented to evaporator pressure. The vented gases are directed through tubular passageway 22 which preferably interconnects with conduit 20 by way of port 23 in low pressure chamber 15 for reasons set forth below. It is also important to note that high pressure gas conduit 18 is connected to the condenser in the lower portion thereof where there will be a minimum of noncondensable gases present. This is done in order to avoid introducing noncondensable gases into chamber 14 and ultimately venting them back into the evaporator through passageway 22, low pressure chamber 15 and conduit 20.

The intake, compression and pumping of noncondensable vapors to the purge condenser at a predetermined pressure is accomplished by the pumping end portion 13 of the piston 11 in chamber 16. Noncondensable gases and refrigerant vapor are conducted from the low turbulence zone in the top of condenser 2 through conduit 24 into chamber 16 through intake port 25. Flow through port 25 is controlled by check valve 26 which may preferably take the form of a flapper valve located directly in port 25. Compressed gases are discharged from chamber 16 through outlet port 27 into conduit 29 communicating with the inlet of purge drum 32. Flow through

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outlet port 27 into conduit 29 is controlled by check valve 28.

The compressed gas entering the purge drum will normally be a mixture of noncondensable gases, refrigerant vapor and water vapor. This gas mixture is directed over water cooled condensing coil 33, as a result of which the refrigerant and water vapor are condensed. The refrigerant, being heavier, settles to the bottom while the water floats on top of the refrigerant and is drained off through manual valve 34. As the depth of refrigerant in the drum increases, float valve 35 opens and liquid refrigerant is forced back to the lower part of the evaporator through conduit 36 by the higher pressure in the purge drum. The noncondensable gas accumulates at the top of the drum and is purged to the atmosphere through pressure relief valve 64 as the pressure in the drum increases to a predetermined value.

The intake and exhaust of high pressure gas to and from chamber 14 through ports 17 and 21 respectively is preferably controlled by sliding piston valve 37. Located within the cylindrical valve housing is a piston 38 provided with an annular groove 39 at its central portion extending completely around its periphery. Ports 40 and 41 are provided in the valve housing in alignment with inlet and exhaust ports 17 and 21 respectively. With valve piston 38 in the extreme right position as shown, inlet port 17 is in communication with high pressure gas conduit 18 through groove 39 in the piston and port 40 in the valve housing. Piston 38 is reciprocated between this position and a position to the extreme left, where exhaust port 21 is placed in communication with exhaust passageway 22 through groove 39 and port 41 in the valve body. The reciprocation of piston 38 is accomplished by alternately introducing high pressure gas on the opposed end faces of the piston through passageways 42 and 43. Tubing passageways 42 and 43 are connected to pump body 10 through openings 44 and 45 in the manner shown so as to be intermittently in communication with high pressure gas in chambers 14 and 16 respectively.

The operation of purge pump 9 and control valve 37 is as follows: A typical cycle of operation may begin with the control valve piston 38 in the position shown. High pressure gas at condensing pressure is introduced into chamber 14 on the upper side of driving piston 12 through conduit 18, port 40, groove 39 and inlet port 17. Chamber 15 on the underside of driving piston 12 is at the lower evaporator pressure. Chamber 16 on the underside of pumping piston 13 is at condensing pressure due to its connection with the top of the condenser through conduit 24. This pressure differential and the area ratio of driving piston 12 and pumping piston 13 are such as to produce a resultant force in the downward direction. Thus piston 11 moves downwardly and pumping piston portion 13 forces compressed gas through outlet port 27 and check valve 28 into conduit 29 and the purge drum 32. Piston 11 will continue moving downwardly until the upper face of driving piston portion 12 clears opening 44. At this time high pressure gas from chamber 14 will be directed against the right end of valve piston 38 through passageway 42. Since the left end of valve piston 38 is vented to evaporator pressure through passageway 43, the piston will shift to the left, exhausting the high pressure gas from chamber 14 through port 21, groove 39, port 41 and tubular passageway 22 to the lower evaporator pressure in chamber 15. Chamber 15 is vented to the evaporator through port 19 and conduit 20. Since both chambers 14 and 15 are now at evaporator pressure and chamber 16 is at the higher condensing pressure, piston 11 will move upwardly. Exhausting chamber 14 through conduit 22 to chamber 15 has the particular advantage that the high pressure gas venting through chamber 15 to the evaporator will create a temporary pressure increase in chamber 15. This assists in moving piston 11 upwardly on the intake stroke of pumping pis-

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ton portion 13. Piston 11 will continue moving upwardly until the bottom face of pumping piston 13 clears opening 45 in the pump housing. At this time, high pressure gas will pass through tubular passageway 43 to the left end of valve piston 38. Since the right end of piston 38 is now vented to evaporator pressure through passageway 42, the valve piston will shift back to the right and the operating cycle will be repeated.

FIGURE 2 illustrates a variation of the purge pump shown in FIGURE 1 wherein the pump control valve is operated mechanically rather than by refrigerant gas pressure. Like reference numerals in FIGURE 2 designate like elements of FIGURE 1. A generally cylindrical chamber 65 is formed within the control valve body 80, and hollow piston 66 is slidably positioned therein. Passageway 67 through the control valve body communicates at its outer end with high pressure gas conduit 18 and at its inner end with inlet port 68 to chamber 14. A second passageway 69 through the control valve body communicates at its outer end with exhaust conduit 76 and at its inner end with exhaust port 70 from chamber 14. Conduit 76 communicates with conduit 20 leading to the evaporator 3.

Rod 71 is connected at one end to pump piston 11 and extends upwardly through hollow valve piston 66 for reciprocation therein. Spaced apart collars 72 and 73 are mounted on rod 71 so as to sequentially abut against opposite faces of lugs 75 extending inwardly from one end of hollow piston 66, as is more clearly shown in the cross-sectional view of valve piston 66 illustrated in FIGURE 4.

Purge pump 9 is operated by refrigerant pressure in the same manner as described above with respect to FIGURE 1. Chamber 15 is continuously vented to evaporator pressure through port 19 and conduit 20. With valve piston 66 in the position shown in FIGURE 2, inlet port 68 is open and gas at the relatively high condensing pressure flows through conduit 18, passageway 67 and inlet port 68 to chamber 14. The force thus exerted by this high pressure gas on driving piston portion 12 causes piston 11 to move downwards on the pumping stroke of pumping piston portion 13. Rod 71 will be carried downwardly by piston 11 and eventually collar 72 will come into contact with the upper face of lugs 75 of piston 66. Valve piston 66 will then be carried downwardly until its side wall closes inlet port 68 and its top end clears exhaust port 70. Chamber 14 will then be vented to evaporator pressure through chamber 65, the clearance space between collar 72 and the inner wall of piston 66, exhaust port 70, passageway 69 and conduits 76 and 20. With chambers 14 and 15 now at evaporator pressure and chamber 16 still at condensing pressure, the resulting upward force will move piston 11 and rod 71 upwardly. Eventually collar 73 will strike the lower surface of lugs 75 and move piston 66 upwardly until it closes exhaust port 70 and opens inlet port 68. The aforesaid cycle of operation is then repeated.

FIGURE 3 illustrates a variation of the mechanically actuated control valve wherein the valve piston 66 is actuated by a pin and slot arrangement rather than by collars on rod 71. Like reference numerals indicate like elements of FIGURES 1 and 2. Pin 84 extends horizontally through valve piston 66 and is received within slot 82 formed in rod 71. The operation of the control valve is the same as that described with respect to FIGURE 2 except for the manner of moving valve piston 66. As is indicated in FIGURE 3, the upper end of slot 82 will come into contact with pin 84 and carry valve piston 66 downwardly as rod 71 is reciprocated downwardly. Piston 66 will eventually move into the position shown in FIGURE 3 wherein it serves to open exhaust port 70 and to close high pressure gas inlet port 68, thus creating a pressure differential across pump piston 11 causing it to move upwardly. The bottom end of slot 82 will then come into contact with pin 84 and carry valve piston

66 upward into a position where it closes exhaust port 70 and opens inlet port 68.

In FIGURE 5 there is shown a variation of the purge pump and control valve combination which provides for the movement of the control valve piston by a combination of mechanical and pneumatic means. Like reference numerals refer to like elements of FIGURES 1-3. Purge pump 9 is generally the same in design and operation as the purge pump shown in FIGURE 1. The control valve comprises a hollow piston 86 slidably positioned within chamber 65 formed in valve housing 80 in an arrangement similar to that shown in FIGURES 2 and 3. Fluid motor inlet port 68 communicating with chamber 14 is located in one wall of valve housing 80 and is connected to high pressure gas conduit 18. Exhaust outlet port 70 is located in the opposite wall of housing 80 and is vented to the low pressure side of the refrigeration system through tubular passage 22, port 23, chamber 15, port 19, and conduit 20 in the same manner provided for the venting of chamber 14 in FIGURE 1. Vertical groove 94 in the wall of valve housing 80 communicates with port 70 and serves to continuously vent the space beneath the upper, larger diameter portion of piston 86 to evaporator pressure. Chamber 15 is continuously vented to evaporator pressure through port 19 and conduit 20. The top end of valve chamber 65 is connected through port 90 and tube 42 with port 44 in the wall of housing 10 of purge pump 9. Port 44 is so positioned as to be exposed to high pressure gas in chamber 14 when driving piston 12 is at the end of its driving stroke. Projection 89 extends downwardly from the top wall of chamber 65 so as to seat within the bore of valve piston 86 when the piston reciprocates upwardly. An upwardly extending boss 88, notched at 92, is provided at the center of the top surface of driving piston portion 12.

Valve piston 86 operates in cooperation with purge pump 9 to control the flow of high pressure gas to and from chamber 14 in the following manner: With valve piston 86 in the position shown, high pressure inlet port 68 is blocked and chamber 14 is vented to low side pressure through the center of hollow piston 86 and outlet port 70. With both chambers 14 and 15 now at low side pressure, and chamber 16 at high side pressure, there will be a resulting force on piston 11 in an upward direction. Free piston 11 will then move upwardly until boss 88 strikes the bottom of valve piston 86 and moves it upwardly to a position where port 68 is uncovered and projection 89 blocks the vent passage through the center of piston 86. Notch 92 in boss 88 permits the continued venting of high pressure gas from chamber 14 through the bore of piston 86 and port 70, after boss 88 strikes the bottom of piston 86 and before the vent passage through the piston bore is closed by projection 89. The introduction of high pressure gas into chamber 14 through port 68 creates a resulting force on piston 11 in the downward direction. Free piston 11 then moves downwardly on the pumping stroke of pumping piston 13 until driving piston 12 clears port 44. At this time high pressure gas will pass from chamber 14 through port 44, tubular passageway 42 and port 90 into valve chamber 65 above piston 86. Gas at high side pressure will now be present in chamber 65 acting on the entire larger diameter top face of valve piston 86. Since the same surface area on the bottom faces of piston 86 is subjected to a combination of low side pressure acting through port 70 and groove 94 and high side pressure from chamber 14 acting on the extreme bottom face, there will be a resulting force on valve piston 86 in a downward direction. Piston 86 will then slide downwardly to a position where the vent passage through the bore of piston 86 and port 70 is opened, and port 68 is blocked. Valve piston 86 is now back in the position shown in FIGURE 5, free piston 11 is ready to move upwardly again, and a complete cycle of operation has been completed.

The arrangement shown in FIGURE 1 has the disadvantage that the high pressure gas directed from chamber 16 through port 45 and tube 43 to one end of the control valve housing is eventually vented back through port 45, chamber 15 and conduit 20 to either the low side of the system or to the atmosphere through conduit 58. Since the gas being pumped in chamber 16 is a mixture of refrigerant vapor and noncondensable gases, a small amount of refrigerant vapor is periodically lost by being vented to the atmosphere through conduit 58 when high pressure gas source 56 is being used to drive the fluid motor; and a small amount of noncondensable gas is returned to the low side of the system when operating the purge pump by refrigerant pressure. The design shown in FIGURE 5 eliminates these undesirable results by doing away with tube 43 and the port 45 required in the FIGURE 1 design. The shifting of the control valve piston into a position where it opens the high pressure inlet port to chamber 14 is accomplished on the suction stroke of pumping piston 13 by the upward force of boss 88 on valve piston 86 rather than by conducting high pressure gas from chamber 16 to one end of the valve housing.

Although the pump has been shown mounted vertically, the pump 9 and the control valve 37 may actually be mounted in any position. The pressure exerted by high pressure gas through port 21 on valve piston 38 will create friction between the piston and the inner wall of the valve housing, and thus prevent undesired sliding movement of the piston from the position shown in FIGURE 1. When piston 38 is reciprocated in the opposite direction, this same friction effect is achieved by the high pressure gas acting through port 40 on the piston wall. This friction feature which operates to hold piston 38 in the desired position is particularly important when control valve 37 is mounted vertically.

Manually operable valves 46, 47, 48 and 49 are located in conduits 24, 18, 20 and 36 respectively in order that the purge pump may be disconnected for servicing or replacement without affecting the operation of the refrigeration system.

By adjusting the flow through high pressure gas conduit 18 and evaporator pressure conduit 20, the pressure differential across driving piston 12 and therefore the speed of the pump may be regulated.

In order to permit purge pump 9 to operate when the centrifugal compressor 1 is not running, a high pressure gas source 56 may be provided. Conduit 57 having valve 55 therein interconnects gas pressure source 56 with high pressure conduit 18. Conduit 58 is connected to low pressure conduit 20 and may be opened to the atmosphere through valve 59. Thus, in order to operate purge pump 9 on gas pressure when the compressor is not running, valves 47 and 48 would be closed and valves 55 and 59 would be opened. By opening valve 59, chamber 15 will be vented to the atmosphere rather than to the low pressure side of the system. Thus the pump will be operated by the pressure differential between the high pressure gas source and atmospheric pressure applied across driving piston 12. The cycle of pump operation will be identical with that described above.

Provision has also been made for operating purge pump 9 automatically in response to a predetermined pressure rise within condenser 2. For this purpose, pressure-responsive valve 50 of the snap-action type is positioned in high pressure gas conduit 18. Valve 50 is spring loaded in a normally closed position with valve element 53 seated in port 54. Obviously, with valve 50 in this normally closed position, high pressure gas cannot flow through conduit 18 and control valve 37 to chamber 14, and purge pump 9 will be rendered inoperative. Valve 50 is designed to open upon a predetermined pressure rise in the condenser 2 due to the accumulation of noncondensable gases therein. To this end, sensing bulb 60 charged with the same liquid refrigerant as that in the system is located in the bottom of the condenser shell where it will

always be exposed to liquid refrigerant. Capillary tube 61 connects bulb 60 with the space in the top of the valve housing above horizontally extending diaphragm 51. The underside of diaphragm 51 is subjected to the total pressure in the condenser by means of tubing 62 which runs from the top of the condenser to the space in the bottom of the valve housing. Ordinarily, the pressure in the top of the condenser transmitted through tube 62 will be the saturation pressure of the condensing refrigerant. However, as noncondensable gases accumulate in the top of the condenser, the total pressure therein will be the sum of the refrigerant, saturation pressure and the partial pressure of air and other noncondensables. The latter pressure will of course vary depending upon the quantity of noncondensable gases present. The liquid refrigerant flowing out of the bottom of the condenser will be at a temperature corresponding to its saturation pressure, regardless of the total pressure in the condenser. Bulb 60 senses this temperature of the refrigerant liquid, and thus the refrigerant in bulb 60, which is the same as that in the system, will always be at a pressure corresponding to the refrigerant saturation pressure. Therefore the pressure on top of diaphragm 51 as transmitted by capillary tube 61 will always be refrigerant saturation pressure. As long as there are no noncondensable gases present in the condenser, the total pressure transmitted through tube 62 will also be refrigerant saturation pressure and spring 63 will hold valve element 53 seated on port 54. However, as noncondensable gases accumulate in the top of the condenser, the total pressure on the underside of diaphragm 51 will increase to a value equal to the sum of the refrigerant saturation pressure and the partial pressure of the noncondensable gases present. Since the top side of diaphragm 51 will still be subjected to refrigerant saturation pressure, a pressure differential equal to the partial pressure of the noncondensable gases in the condenser will be created across diaphragm 51 in an upward direction. When this pressure differential exceeds the pressure exerted by spring 63, diaphragm 51 will move upwardly, carrying valve stem 52 with it and thus causing valve element 53 to be lifted off of seat 54. The valve will remain open, permitting high pressure gas to flow through conduit 18 and actuate purge pump 9, until the operation of the purge pump reduces the partial pressure of the noncondensable gases in the condenser to a value less than the pressure of spring 63.

A solenoid valve could be employed in place of pressure differential valve 50. In that case, the solenoid valve would be energized by a relay responsive to the differential between the total pressure in the top of the condenser (refrigerant saturation pressure plus noncondensable gas pressure) and the refrigerant saturation pressure.

The purging apparatus described above will operate satisfactorily on any of the refrigerants, such as Refrigerant 11 or Refrigerant 113, commonly employed in centrifugal refrigeration systems. The purge pump 9 and control valve 37 are constructed of materials compatible with the refrigerant employed in the system. Piston 38 in control valve 37 may be constructed of nylon, "Teflon," or other suitable plastic.

It will be apparent from the foregoing description that the purge pump of the invention achieves the objectives set forth above. Since purge pump 9 is operated by the pressure differential existing in the refrigeration system to which it is connected, no external power source is required. The purge pump is of the free-piston type requiring no oil lubrication and having no shaft seals. Thus the pump may be hermetically incorporated into refrigeration systems and the oil separator conventionally employed downstream of motor-driven purge compressors is eliminated. By doing away with the electric motor normally utilized to drive the purge compressor as well as the oil separator, considerable cost savings are achieved. The purge pump of this invention is relatively compact

and takes up considerably less space than that required by existing purge pump-motor combinations.

I do not desire to limit my invention to the particular embodiment shown and described, which is illustrative only. It is contemplated that changes may be made without departing from the spirit and scope of the invention as defined by the following claims.

I claim:

1. In combination with a closed circuit refrigeration system having a compressor, condenser, pressure reducing valve and evaporator interconnected in refrigerant flow relationship, said valve being located between the condenser and evaporator so as to form within said system a high pressure side including the compressor discharge line, said condenser and the inlet of said valve and a low pressure side including the outlet of said valve, said evaporator and the compressor suction line; a purge pump for removing noncondensable gases from said system, said pump having an inlet connected by a first conduit to the condenser so as to receive a mixture of noncondensable gases and refrigerant vapor therefrom; a fluid motor drivingly connected to said pump, and conduit means connecting said fluid motor to said refrigeration system whereby the pressure differential within said system imparts the necessary motive force to said fluid motor.

2. Apparatus as recited in claim 1 wherein said fluid motor is provided with an inlet and an exhaust outlet and said means connecting said fluid motor to said refrigeration system includes a second conduit connecting said inlet to said high pressure side of said system and third conduit means connecting said exhaust outlet to said low pressure side of said system.

3. Apparatus as recited in claim 2 including normally closed valve means in said second conduit, and motor means operative to open said valve means in response to a predetermined pressure rise in said condenser.

4. Apparatus as recited in claim 3 wherein said motor means consists of a diaphragm, a valve element connected to said diaphragm for actuation thereby; a first tube connecting one side of said diaphragm to the top of said condenser and a second tube connecting the other side of said diaphragm to a sensing bulb located in the refrigerant liquid containing portion of said condenser, said bulb containing an expansible fluid.

5. Apparatus as recited in claim 2 wherein said second conduit is connected to the lower portion of said condenser at a point where noncondensable gases would normally not be present.

6. Apparatus as recited in claim 2 further including a source of high pressure gas, a conduit with a valve therein connecting said gas source to said fluid motor inlet, and another conduit having a valve therein and connecting said fluid motor exhaust outlet to the atmosphere.

7. Apparatus as defined in claim 2 wherein said purge pump comprises a housing, a free-piston mounted within said housing for reciprocating movement therein, said piston including a small diameter pumping piston portion connected to a relatively large diameter driving piston portion for actuation thereby, said driving piston portion serving as said fluid motor.

8. Apparatus as defined in claim 7 wherein a chamber is formed within said housing between said driving piston portion and said pumping piston portion, and including conduit means connecting said chamber to said low pressure side of said system regardless of the position of said free-piston.

9. Apparatus as defined in claim 7 and further including valve means comprising a body defining an inner chamber containing a valve element movable therein, said body having first passage means therethrough connected to said second conduit and said fluid motor inlet and second passage means therethrough connected to said third conduit means and said fluid motor exhaust outlet, said valve element being movable between a first position in which it permits flow through said first passage means

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and a second position in which it permits flow through said second passage means.

10. Apparatus as defined in claim 9 including means alternately connecting opposite sides of said valve element to said high pressure side of said system whereby said valve element is moved between said first and second positions.

11. Apparatus as recited in claim 10 wherein said means alternately connecting opposite sides of said valve element to said high pressure side of said system comprises a first valve operating tube connected to one end of said valve body and to a port in said pump housing, said port being so positioned that it is exposed to gas at the pressure of said high pressure side when said pumping piston portion is at the end of its suction stroke, and a second valve operating tube connected to the opposite end of said valve body and to another port in said pump housing so positioned that it is exposed to gas at the pressure of said high pressure side when said driving piston portion is at the end of its driving stroke.

12. Apparatus as recited in claim 9 including a mechanical linkage interconnecting said free-piston and said valve element whereby said valve element is moved between said first and second positions by the reciprocating action of said piston.

13. Apparatus as defined in claim 12 wherein said valve element is in the form of a hollow piston and said mechanical linkage comprises a rod connected at one end to said free-piston and slidably positioned within said hollow valve piston, said rod having two spaced apart col-

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lars attached thereto so constructed and arranged as to alternately abut against different end surfaces of said hollow piston as said rod reciprocates with said free-piston.

14. Apparatus as defined in claim 12 wherein said valve element is in the form of a hollow piston having a pin extending therethrough and said mechanical linkage comprises a rod connected at one end to said free-piston and slidably positioned within said hollow valve piston, said rod having a slot therein through which said pin extends, whereby one end of said slot comes into contact with said pin and moves said hollow piston to said first position as said rod slides in one direction and the other end of said slot comes into contact with said pin and moves said hollow piston to said second position as said rod slides in the opposite direction.

15. Apparatus as recited in claim 9 including means connected to said free piston so constructed and arranged as to contact one end of said valve element and move said valve element into said first position as said free piston reciprocates in one direction, and means connecting the opposite end of said valve element to said high pressure side when said free piston reaches the end of its stroke in the opposite direction.

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30 ROBERT A. O'LEARY, *Primary Examiner*.