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REFRIGERATION SYSTEM EMPLOYING BELLOWS SOLUTION PUMP

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2 Sheets-Sheet 1

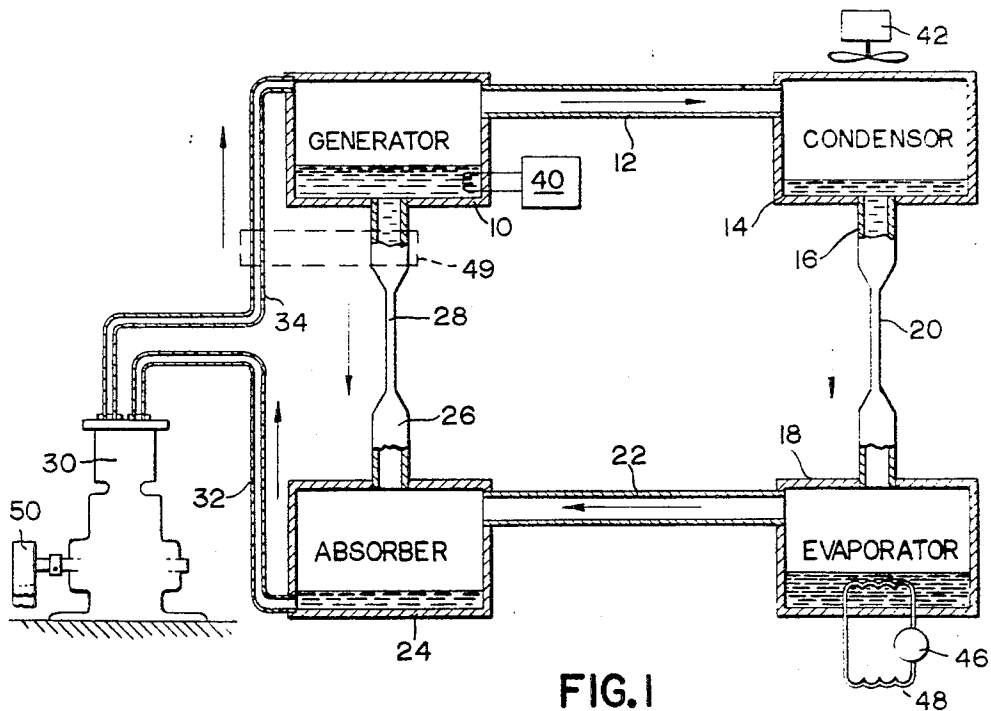


FIG. 1

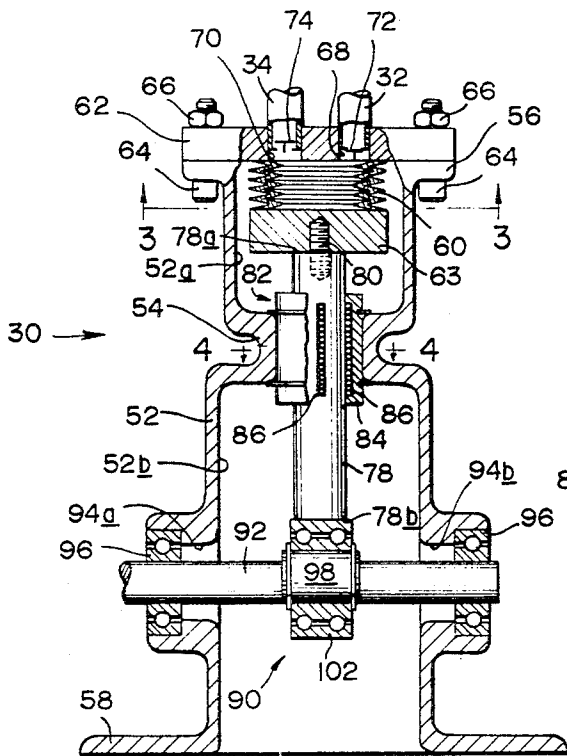


FIG. 2

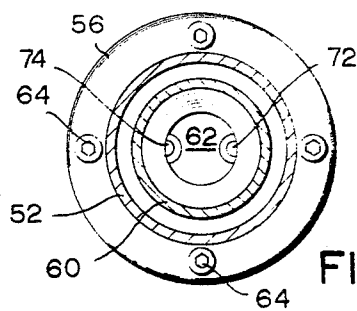


FIG. 3

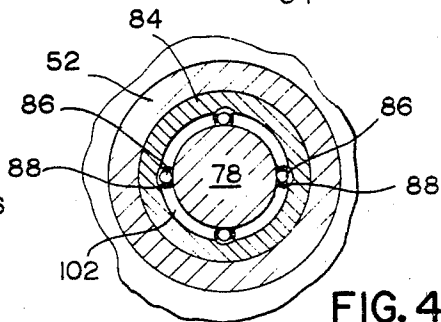


FIG. 4

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3,473,347 REFRIGERATION SYSTEM EMPLOYING BELLOWS SOLUTION PUMP

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7 Claims

ABSTRACT OF THE DISCLOSURE

The fluid circulator in a closed loop absorption refrigeration system is a positive displacement welded bellows pump connected to pump the working fluid from the absorber directly to the generator. The pump itself comprises a bellows having one end sealed to a header and communicating with intake and outlet valves in the header. The other end of the bellows is closed and is moved toward and away from the header by a shaft driven by an eccentric. The drive mechanism is bearing mounted and designed to move the bellows only parallel to the longitudinal axis of the bellows. These factors minimize the stresses on the pump elements and enable the pump to run much faster and for longer periods than prior pumps used for this purpose. Also, the pump can operate with a very low suction head which enables the overall system to function effectively at relatively low ambient temperatures.

BACKGROUND OF THE INVENTION

This invention relates to a hermetically sealed absorption refrigeration system and to means for circulating working fluids through such a system. As such, the invention has application also in air-conditioning and chilling systems.

Refrigeration apparatus of the type with which we are concerned here have four basic sections, to wit, a generator, a condenser, and evaporator and an absorber, connected together in a closed loop. Working fluids are circulated between the different sections to obtain the refrigeration effect.

For best results, the refrigeration system should meet the following three requirements. First, the system should be completely leak proof. This is because its working fluids are usually corrosive or toxic. Secondly, the system should have a long useful life and require a minimum amount of maintenance under a variety of operating conditions. Specifically, the system should be able to function effectively even at temperatures below 30° F. Finally, the cost, size and weight of the refrigerator should be kept to a minimum.

There are several available absorption refrigeration systems which are superior with respect to one or another of the above criteria. However, none of them exhibit superior performance with respect to all of these factors. This is due in large part to the means used heretofore to circulate the working fluids in the system. More particularly, conventional systems rely on a piston pump or a diaphragm pump to circulate the fluids around the refrigeration loop. And the pump is generally not placed directly between the absorber and the generator, which is the most desirable location for it. The piston pump is unsatisfactory because its seals tend to leak after a relatively short time due to exposure to the corrosive working fluids. The diaphragm pump, on the other hand, which is most commonly used in these systems, must be operated at relatively low speeds, i.e. on the order of 100–300 r.p.m. Consequently, it tends to deliver the fluid in pulses, with the result that it must be followed by an accumulator

to average out these fluid pulses. Further, because it is a low speed device, the diaphragm pump requires either a separate drive system or elaborate gear reduction from its usual power takeoff at the main high speed circulating pump used in these systems. Of course, all of these extra components increase the cost, size and weight of the overall refrigerator.

The diaphragm pump has another serious drawback in that it requires a certain minimum positive pressure head, on the order of 10 p.s.i.g., at the input side of the pump. This minimum pressure head is needed to return the pump diaphragm after each pressure stroke to ready it for the next pressure stroke. The inability of the diaphragm pump to operate with a low suction head, in turn, limits the refrigeration effect which can be obtained from the system as a whole. This is because the refrigeration effect increases as the pressure at the low pressure side of the refrigeration loop (i.e. at the suction side of the pump) decreases. Since present day diaphragm pumps limit this suction head to 10 p.s.i.g., they also limit the refrigeration effect which can be obtained from a given amount of refrigerant. More specifically, they prevent the refrigerator from working effectively when the ambient temperature falls below 30° F. because, at this temperature, the pressure at the pump inlet becomes less than 10 p.s.i.g.

SUMMARY OF THE INVENTION

Accordingly, it is an object of the present invention to provide an absorption refrigeration system which can function efficiently even when the ambient temperature is relatively low, i.e. on the order of 30° F. and below.

Another object of the invention is to provide an absorption refrigeration system which can operate for long periods without maintenance.

A further object of the invention is to provide an absorption refrigeration system having a minimum size and weight for a given capacity.

Another object of the invention is to provide an improved solution pump for an absorption refrigeration system which minimizes the overall cost of the system.

A still further object of the invention is to provide a fluid-tight, positive displacement pump for pumping corrosive and toxic fluids.

Another object of the invention is to provide a solution pump for an absorption refrigeration system which can operate at relatively high speeds.

A still further object of the invention is to provide a solution pump which can operate effectively with a relatively low suction head.

Still another object of the invention is to provide a positive displacement pump whose components suffer a minimum amount of stress and wear in use.

Other objects of the invention will in part be obvious and will in part appear hereinafter.

The invention accordingly comprises the features of construction, combination of elements, and arrangement of parts which will be exemplified in the constructions hereinafter set forth, and the scope of the invention will be indicated in the claims.

Briefly, the refrigeration system comprises a generator, condenser, evaporator and absorber arranged in a closed loop. The generator and condenser are at the high pressure side of the loop, while the evaporator and absorber are at the low pressure side thereof. The system employs two working fluids, to wit, a refrigerant fluid and an absorbent fluid. The refrigerant circulates all around the loop while the absorbent circulates between the generator and the absorber only. A positive displacement, welded bellows solution pump is connected directly between the generator and the absorber to pump the re-

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frigerant and absorbent fluids from the absorber at a low pressure, generally atmospheric pressure, to the generator at the high pressure side of the loop.

The solution pump comprises a housing having upper and lower compartments. A welded bellows is situated in the upper compartment. One end of the bellows is sealed to a header forming the top wall of the upper compartment and inlet and outlet check valves in the header communicate with the interior of the bellows. The other end of the bellows is sealed to a rigid plate and is free to move.

A vertical shaft having one end secured to the plate extends down through a bearing-lined bushing into the lower compartment. The lower end of the shaft rides on a bearing encircling an eccentric carried by a transverse drive shaft journaled in the sidewalls of the lower compartment. The drive shaft is driven, in turn, by a source of rotary power. Thus, when the drive shaft turns, the vertical shaft moves up and down.

On the upstroke of the vertical shaft, the bellows is compressed so that any fluid inside the bellows is pumped out through the outlet valve in the header and thence to the generator. The expansion of the bellows, on the other hand, is not due directly to the downward movement of the vertical shaft. Rather, the bellows itself functions like a compression spring. That is, the bellows is still under some compression when the vertical shaft is in its full down position. Consequently, the free end of the bellows follows the shaft on its downstroke, rather than being forceably pulled down by that shaft. When this occurs, fluid is drawn from the absorber into the bellows through the inlet valve in the header.

This process repeats itself during each revolution of the drive shaft so that the working fluids are forceably and continuously pumped from the absorber to the generator.

The employment of my bellows solution pump between the absorber and the generator enables the overall refrigeration system to function effectively even when the ambient temperature is below 30° F. This is because the pump does not require a minimum positive pressure head at its suction side. In fact, the present pump will work satisfactorily even with a small negative suction head.

Also, the elements of the pump are all designed to minimize stresses on the bellows and also to minimize frictional losses within the pump. Accordingly, the pump can operate at greater speeds for a longer period than can conventional bellows pumps which are presently being used for other purposes. The fact that the present pump can operate at high speed produces two distinct advantages. First, the pump can be driven via a direct belt drive from the high speed water circulating pump used in these systems. This eliminates costly speed-reducing equipment which contributes to the weight and bulk of the overall system. Also, fluid pulses in the output from the present pump are much less pronounced than is the case with prior comparable solution pumps. Consequently, the present system does not require an accumulator to average out pulses in fluid being pumped to the generator. This also results in a concomitant decrease in the cost and size of the overall refrigeration system.

BRIEF DESCRIPTION OF THE DRAWINGS

For a fuller understanding of the nature and objects of the invention, reference should be had to the following detailed description taken in connection with the accompanying drawings, in which:

FIG. 1 is a schematic diagram of an absorption refrigeration system embodying the principles of my invention;

FIG. 2 is a vertical section with parts in elevation and other parts cut away of the solution pump employed in the FIG. 1 system;

FIG. 3 is a sectional view along line 3—3 of FIG. 2;

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FIG. 4 is a sectional view along line 4—4 of FIG. 2; and

FIG. 5 is a vertical section with parts in elevation of another pump embodiment employing the principles of the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to FIG. 1 of the drawings, the present system comprises a generator 10 connected by way of a pipe 12 to a condenser 14. A pipe 16 communicates between the bottom of condenser 14 and the top of evaporator 18. Pipe 16 has a restriction at 20 provided by an expansion valve. A pipe 22 connects the top of evaporator 18 to the top of an absorber 24. Further, a pipe 26 connects the top of absorber 24 to the bottom of generator 10. Pipe 26 also has a restriction 28 similar to restriction 20 in pipe 16. Thus, the generator, condenser, evaporator and absorber are arranged in a closed fluid-tight loop.

A fluid circulating pump 30, to be described in detail later, is connected between absorber 24 and generator 10. Specifically, a pipe 32 communicates between the bottom of absorber 24 and the suction side of pump 30 and a pipe 34 leads from the pressure side of pump 30 to the top of generator 10. Thus, restrictions 20 and 28 (and pump 30) divide the loop into a high pressure side comprising generator 10 and condenser 14 and a low pressure side comprising evaporator 18 and absorber 24.

The system uses two working fluids. A refrigerant fluid, such as ammonia, is circulated clockwise in various phases all around the loop, while an absorbent liquid such as water circulates only between generator 10 and absorber 24. The purpose of the absorbent liquid is to enable pump 30 to return refrigerant vapor from absorber 24 to generator 10 after each evaporation cycle.

During operation of the system, a rich aqua-ammonia solution in generator 10 is heated by any suitable means indicated by an electric heater 40. Ammonia is boiled off and passes through pipe 12 to condenser 14 as a vapor. In condenser 14, the ammonia vapor is cooled by suitable external cooling means indicated by a fan 42. The vapor condenses and the condensate passes through restriction 20 to the low pressure side of the loop. Specifically, the liquid ammonia flashes into evaporator 18 and cools water-carrying chiller coils 44 located therein. A pump 46 circulates the cold water from coils 44 to external cooling coils 48 which then cool the space around them. The coils 44 and 48 and pump 46 form a closed loop and pump 46 is normally driven at a high speed on the order of 3600 r.p.m.

From evaporator 18, the ammonia vapor passes through pipe 22 into absorber 24. At the same time, water from generator 10, at a high pressure, is forced through restriction 28 in pipe 26 into absorber 24, maintained at a lower pressure. The ammonia vapor is dissolved in the water and the rich aqua-ammonia mixture settles to the bottom of the absorber. Pump 30 then pumps the rich aqua-ammonia solution from the absorber back to generator 10 under high pressure, and the cycle commences again.

Thus, it is the function of absorber 24 to maintain the desired pressure at the low side of the loop and to dissolve the ammonia refrigerant vapor in the water absorbent so that the refrigerant can be returned to the high pressure side of the loop for recycling. Preferably, the pressure in the absorber is kept low, e.g. —½ p.s.i.g. This is because, as mentioned previously, when the pressure in evaporator 18 is decreased, the amount of vapor leaving the evaporator is increased. Consequently, the refrigeration effect is also increased.

To further maximize the efficiency of the system, pipes 26 and 34 may be placed in heat exchange relationship as indicated by a heat exchange shell 49 shown in dotted lines. In this way, the heated water flowing to absorber

24 is used to preheat the cool aqua-ammonia solution being pumped to generator 10.

Pump 30, to be described shortly, can operate at high speeds, on the order of 900-1200 r.p.m. Therefore, pump 30 can be driven through a conventional speed reduction belt drive 50 directly off high speed pump 46. Consequently, the FIG. 1 device does not require a separate drive means or speed reducer which would add to its cost and size. Moreover, because it operates at high speeds, the pump 30 delivers fluid to generator 10 relatively uniformly so that there is no need for an accumulator to smooth its output as is the case with solution pumps in other comparable systems.

The FIG. 1 system is able to obtain a maximum refrigeration effect because pump 30 is connected directly between absorber 24 and generator 10 and is able to operate with suction pressures well below 10 p.s.i.g. which is the limit for the diaphragm pumps normally used in these systems. Consequently, the pressure through the evaporator 18 at the low pressure side of the loop is less than is possible with comparable conventional systems. This means that the amount of vapor leaving the evaporator is increased with a concomitant increase in the refrigeration effect produced by the system. In actual practice, the system still operates efficiently even though ambient temperature drops below 30° F.

Referring now to FIG. 2, pump 30 comprises an up-standing, generally cylindrical housing 52 which is formed with a restriction 54 dividing the housing into upper and lower open-ended compartments 52a and 52b, respectively. The upper end of housing 52 is outwardly flanged at 56 and the lower end of the housing is flanged at 58. Flange 58 serves as a base by which pump 30 can be secured to a suitable support.

An open-ended welded bellows 60 is situated in upper compartment 52a. One end of bellows 60 is sealed to a discoid header 62 which closes off the top of compartment 52a. The other end of the bellows is sealed to a circular plate 63. Header 62 is coextensive with upper flange 56 and is secured thereto by a series of threaded bolts 64 extending through the flange and header. Correspondingly threaded nuts 66 are turned down on the bolts to maintain a fluid-tight joint between flange 56 and header 62. Suitable gasketing may be provided there, if needed.

As best seen in FIGS. 2 and 3, a pair of vertical openings 68 and 70 are formed in header 62 which communicate with the interior of bellows 60. Opening 68 contains an inlet valve in the form of a conventional check valve 72 which allows fluid to flow into, but not out of, bellows 60; opening 70 contains a similar check valve 74 which is arranged to allow fluid to flow only out of bellows 60. The ends of pipes 32 and 34 (FIG. 1) are screwed into openings 68 and 70, respectively, and thus communicate through the valves with the interior of bellows 60.

A vertical cylindrical shaft 78 is secured at one end 78a to plate 63 by means of a double-ended, threaded stud 80 prior to securing header 62 to housing 52.

As best seen in FIGS. 2 and 4, shaft 78 extends down into lower compartment 52b through a bushing 82. Bushing 82 comprises an open-ended sleeve 84 which is secured to housing 52 within restriction 54 so that its longitudinal axis is aligned with the longitudinal axis of bellows 60. Vertical arrays of ball bearings 86 in cages 88 are distributed around the inside of sleeve 84 to provide low friction bearing surfaces for shaft 78.

Referring again to FIG. 2, shaft 78 is moved up and down along the axis of bellows 60 by means of a drive unit indicated generally at 90 in lower compartment 52b of housing 52. Drive unit 90 comprises a horizontal drive shaft 92 extending through flanged openings 94a and 94b on opposite sides of compartment 52b. Specifically, drive shaft 92 is journaled in conventional double race ball bearing units 96 mounted in openings 94a and 94b. Also, one end of drive shaft 92 is coupled to belt drive 50 (FIG. 1).

As shown in FIG. 2, drive shaft 92 carries an eccentric 98 which is positioned directly below bushing 82. Also, a cam-follower 102 in the form of a double race ball bearing unit encircles eccentric 98 with the inner race being keyed to the eccentric. Shaft 78 extends down into compartment 52b so that its lower end 78b normally rides on cam-follower 102.

When shaft 92 is rotated, shaft 78 is moved up and down by eccentric 98 through cam-follower 102, thereby actuating bellows 60. Thus, as eccentric 98 moves toward its up position, shaft 78 compresses bellows 60, thereby forcing any fluid in the bellows out through valve 74 and into generator 10 (FIG. 1). During this time, the inlet valve 72 is closed. Then, as eccentric 98 moves toward its down position, bellows 60, which is under compression, pushes against shaft 78 so that the shaft follows the eccentric. During this part of the pump cycle, inlet valve 72 opens, drawing in more fluid from absorber 24, while outlet valve 74 remains closed. Thus, fluid is drawn into and pumped out of bellows 60 during each revolution of drive shaft 92.

It is important to note that all of the potential wear surfaces of the moving elements 78 and 92 are supported in ball bearing elements, i.e. bushing 82 and bearing units 96 and 102. Therefore, these elements suffer a minimum amount of wear. For the same reason, pump 30 is able to operate at relatively high speeds as mentioned previously.

It should be emphasized also that the pump components are arranged to keep the stresses on its different elements to a minimum. Thus, bellows 60, shaft 78 and eccentric 98 are all disposed along the very same vertical axis. Also, shaft 78 is loose and rides on cam-follower 102. Consequently, drive unit 90 does not transmit any appreciable sidewise forces to shaft 78 as would tend to cock it. Moreover, shaft 78 itself is constrained by bushing 82 so that it can only move along the axis of bellows 60. Consequently, no appreciable sidewise forces are imparted to the bellows as would tend to unduly stress or distort it with the result that bellows 60 is able to operate at high speeds for relatively long periods without breaking.

Bellows 60 is still under some compression in its fully open condition. Therefore, it is able to operate properly even though the pressure on the suction side of the pump, i.e. in pipe 32, drops to a relatively low value. In fact, the present pump works effectively with suction heads as low as $-\frac{1}{2}$ p.s.i.g. Resultantly, the FIG. 1 system cools efficiently even when the ambient temperature drops below 30° F.

FIG. 5 shows a duplex pump made in accordance with the present invention. The FIG. 5 pump is utilized where higher flow rates are desired in the refrigeration loop or in applications requiring low flow rates, but more uniform fluid velocities.

The FIG. 5 pump comprises two identical pump sections indicated generally at 112 and 114. The components of the two pump sections are all arranged in a straight line and both sections are driven in unison by a single drive shaft 116 between them. As with the FIG. 2 pump, the in-line construction minimizes stress and wear on the pump components.

The duplex embodiment comprises a generally cylindrical, open-ended housing 118. The opposite ends of housing 118 are flanged at 118a and 118b. Also, housing 118 has reduced diameter portions 120 and 122 midway along the two pump sections in which are mounted similar bushing 124 and 126.

A welded bellows 128 is situated in pump section 112 outwardly of bushing 124. One end of bellows 128 is sealed to a generally circular header 130 which is bolted to flange 118a, thereby closing off the open end of pump section 112. The other end of bellows 128 is sealed to a circular plate 132. A pair of passages 134 and 136 extend through header 130 and communicate with the interior of bellows 128. Passage 134 contains a check valve 138 which allows fluid to flow out of, but not into, bellows 128;

passage 136 contains a similar check valve 140 which only permits fluid to flow into the bellows. Pump section 112 draws fluid into bellows 128 from an inlet pipe 142 connected to passage 136 and pumps fluid out of bellows 128 through an outlet pipe 144 connected to passage 134.

Plate 132 is secured by a threaded stud 146 to one end of a power transmitting shaft 148 received in bushing 124. The opposite end of shaft 148 is bifurcated, forming a pair of spaced, parallel arms 150 and 152. Shaft 148 is arranged so that arms 150 and 152 extend perpendicular to drive shaft 116.

A rubbing member 154 is positioned between arms 150 and 152. A pin 156 extends loosely through member 154 and is secured to arms 150 and 152. Thus, member 154 is positioned between the arms, but can cock or pivot slightly above the pin.

Drive shaft 116 extends through flanged openings 158 and 160 in housing 118. Specifically, shaft 116 is journaled in double race ball bearing units 162 and 164 positioned within openings 158 and 160, respectively. Also, shaft 116 carries an eccentric 166 which is positioned directly opposite member 154. Member 154 rides on a ball bearing cam follower 168 carried by the eccentric. Thus, when shaft 116 is rotated toward a position 180° from its position indicated in FIG. 5, shaft 148 is urged to the left, thereby compressing bellows 128. This causes any fluid in the bellows to be pushed out through valve 138 and then to the outlet pipe 144.

On the other hand, when the shaft is rotated toward the illustrated position, bellows 128 expands. The expansion of bellows 128 draws additional fluid from pipe 142 into the bellows through valve 140. As mentioned previously, bellows 128 is still under some compression when shaft 148 has moved to the right as far as it can go. Therefore, the bellows does not require positive pressure in the inlet passage 136 to expand it.

The provision of the cocking member 154 minimizes the sidewise forces that might be transmitted from eccentric 166 to shaft 148. Moreover, the shaft is constrained by bushing 124 to move only along the axis of bellows 128. Consequently, the bellows is not subjected to sidewise stresses which might tend to distort or break it, as fully described above in connection with FIG. 2.

The elements of pump section 114 are identical to those of section 112. Therefore, we will not describe them again in detail. The section 114 components corresponding to those in section 112 carry the same identifying number followed by the letter *a*. It should be noted, however, that the pumping cycles of sections 112 and 114 are 180° out-of-phase. That is, when fluid is being pumped out of bellows 128 in section 112, fluid is being drawn into bellows 128*a* in section 114.

Thus, the two pump sections 112 and 114 can be connected together so that the overall device pumps fluid at twice the rate of the FIG. 2 pump. More particularly, outlet pipes 144 and 144*a* can both be connected through a suitable T-coupling (not shown) to outlet pipe 34 (FIG. 1). Similarly, inlet pipes 142 and 142*a* in the FIG. 5 pump can be connected through a T-fitting (not shown) to inlet pipe 32 (FIG. 1). Thus, when the pump is operated in this fashion, it delivers fluid at a very uniform velocity and with very little pulsing. Needless to say, also, the duplex pump has all the advantages described above in connection with the single action pump.

It will be appreciated that the same principles may be followed in constructing a pump having three or more such sections, all of which are driven from a single drive shaft. In this event, the eccentrics on the drive shaft would be arranged much like those on an automobile cam shaft to actuate the bellows in the various pump sections at the appropriate times. Also, the inputs and outputs to and from the various sections would be connected together as described above in connection with FIG. 5. Thus, the overall pump would be capable of delivering fluid at extremely uniform velocities and with practically no pulsing

at all. This feature is useful in applications where extremely uniform and precise flow rates are desired.

It will thus be seen that the objects set forth above, among those made apparent from the preceding description, are efficiently attained. Also, certain changes may be made in the above constructions without departing from the scope of the invention. For example, the pump may employ drive means between the drive shaft 92 or 116 and the bellows plates 63 or 132 which affords a positive linkage between those elements. Therefore, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

It is also to be understood that the following claims are intended to cover all of the generic and specific features of the invention herein described, and all statements of the scope of the invention which, as a matter of language, might be said to fall therebetween.

We claim:

1. An absorption refrigeration system comprising
 - (A) a generator,
 - (B) a condenser,
 - (C) an evaporator,
 - (D) an absorber, and
 - (E) a pump connected to circulate working fluid within the system, said pump comprising
 - (1) a housing,
 - (2) a bellows having one end secured to a wall of said housing,
 - (3) a pair of check valves mounted in said housing wall and communicating with the interior of said bellows,
 - (4) a rigid plate closing off the other end of said bellows;
 - (5) a rotary drive shaft
 - (a) extending into said housing, and
 - (b) oriented perpendicular to the axis of said bellows, and
 - (6) drive means extending to said drive shaft directly below said plate and engaging said plate for alternatively compressing and expanding said bellows upon rotation of said drive shaft.
2. An absorption refrigeration system comprising
 - (A) a generator,
 - (B) a condenser
 - (C) an evaporator,
 - (D) an absorber, and
 - (E) a fluid circulating pump having its inlet connected to said absorber and its outlet connected to said generator, said pump comprising
 - (1) a housing,
 - (2) a bellows situated in said housing, one end of said bellows being sealed to a wall of said housing, the other end of said bellows being closed and movable within said housing,
 - (3) means defining inlet and outlet passages in said housing wall with both of said passages communicating with the interior of said bellows,
 - (4) a check valve in said inlet passage arranged to permit fluid to flow only into said bellows,
 - (5) a check valve in said outlet passage arranged to permit fluid to flow only out of said bellows,
 - (6) a drive shaft mounted for rotation in said housing, said shaft being disposed generally perpendicular to the axis of said bellows,
 - (7) a cam carried by said shaft, said cam being aligned with said bellows axis, and
 - (8) force transmitting means aligned between said cam and said other end of said bellows, said cam moving said transmitting means along said bellows axis upon rotation of said drive shaft so as to compress said bellows without

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applying any appreciable sideways forces there-
to.

3. A system as defined in claim 2 and further includ-
ing bearing means mounted in said housing for rotative-
ly supporting said drive shaft.

4. A system as defined in claim 2 wherein said cam
comprises a circular eccentric mounted on said drive
shaft.

5. A system as defined in claim 4 and further includ-
ing a double-race ball bearing unit

(A) encircling said eccentric, and

(B) engaging said transmitting means.

6. A system as defined in claim 2 wherein said bellows
is maintained under compression so that it tends to ex-
pand, causing said transmitting means to follow said cam
when the latter is moving away from said housing wall
and thereby draw fluid from said absorber into said bel-
lows, even when there is no positive pressure head in said
absorber.

7. A system as defined in claim 2 and further including
a tubular bushing

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(A) having its axis aligned with said bellows axis, and
(B) slidably receiving said transmitting means so as
to constrain the latter to move only along said bel-
lows axis.

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LLOYD L. KING, Primary Examiner

U.S. Cl. X.R.

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