

May 12, 1953

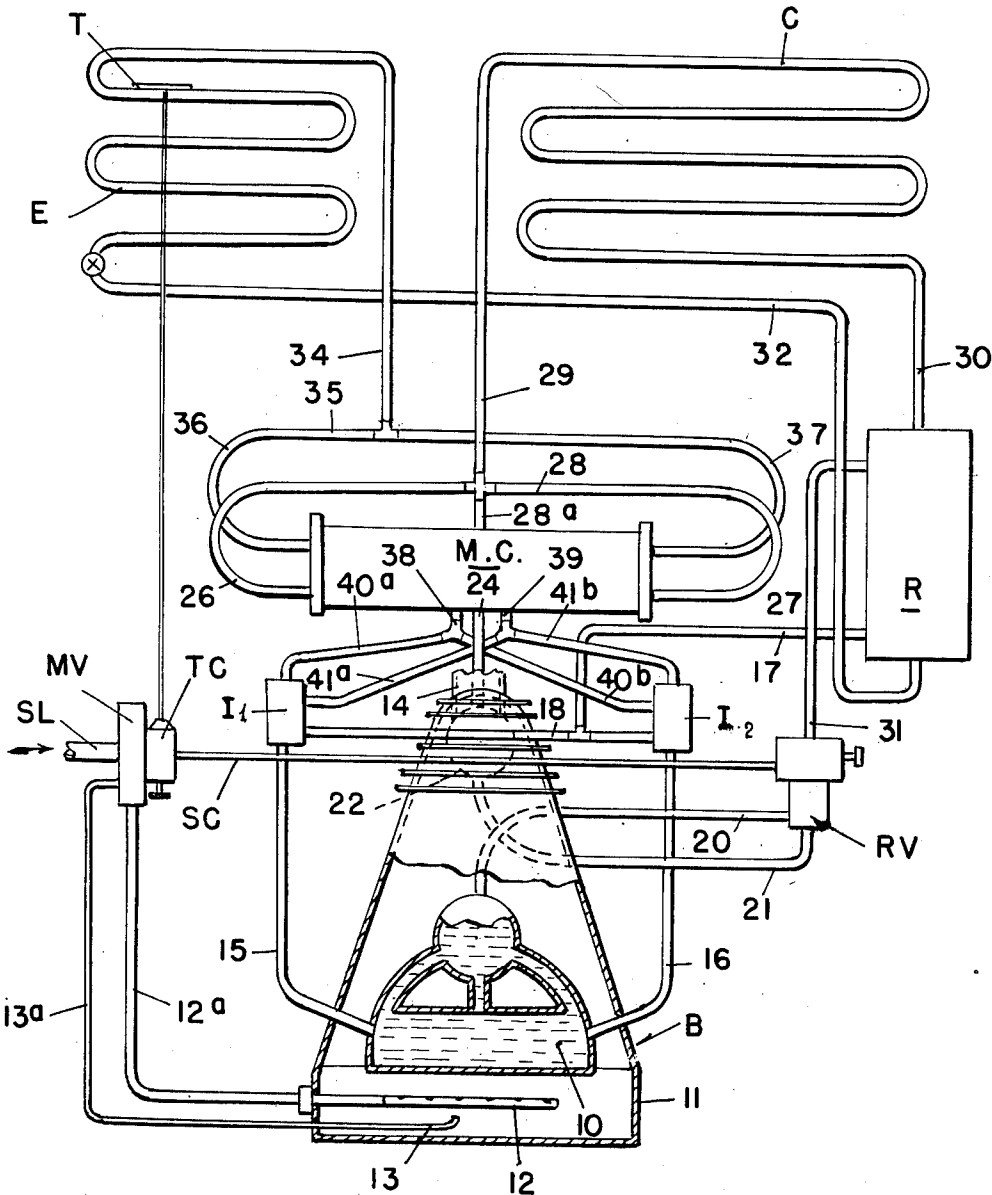
C. D. RUSSELL
REFRIGERATION MACHINE

2,637,981

Filed Dec. 30, 1948

4 Sheets-Sheet 1

FIG. 1



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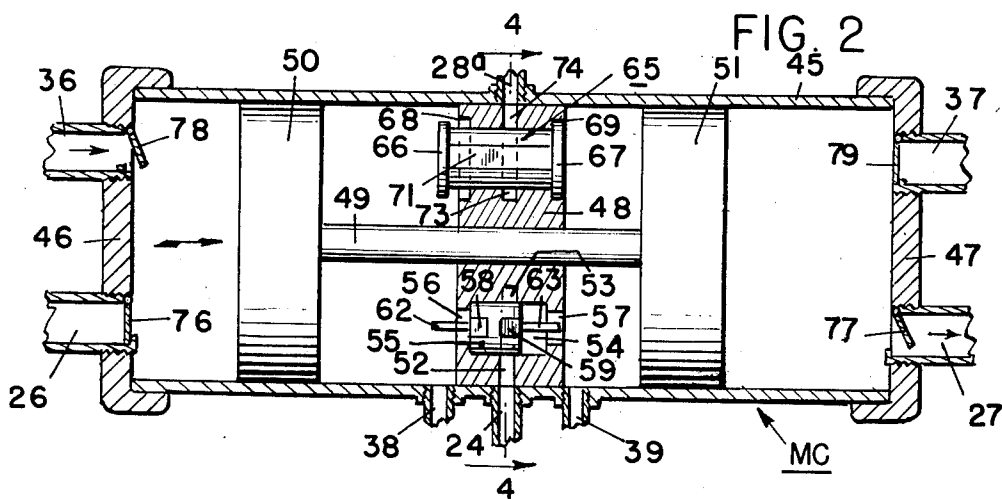
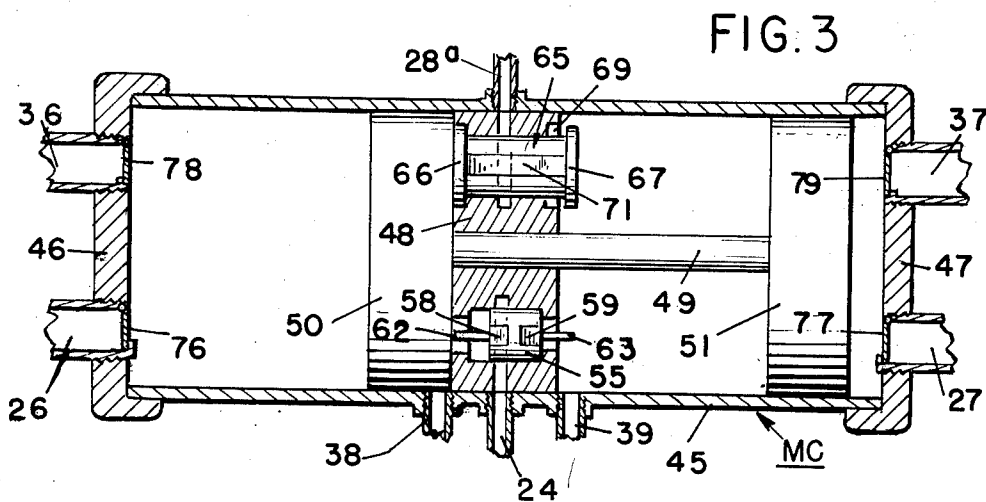
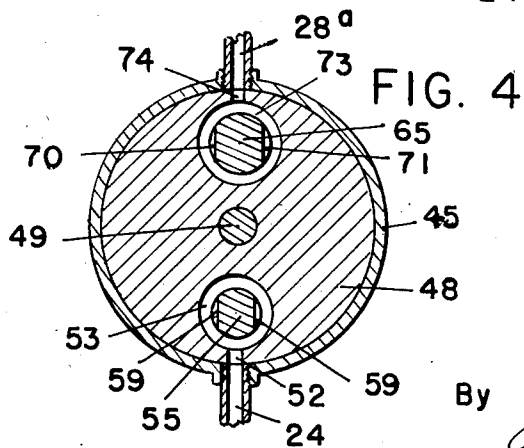
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4 Sheets-Sheet 2



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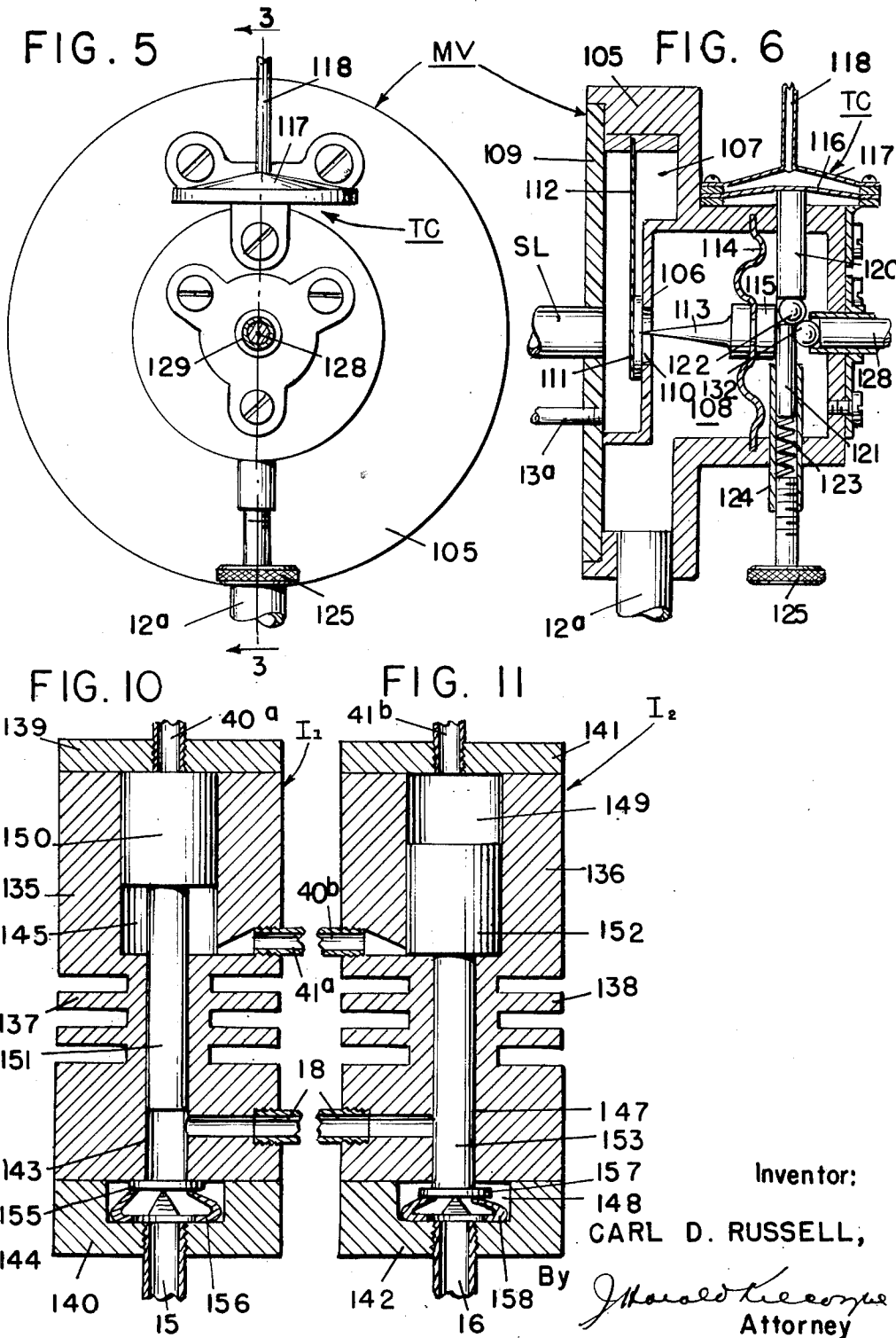
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REFRIGERATION MACHINE

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FIG. 7

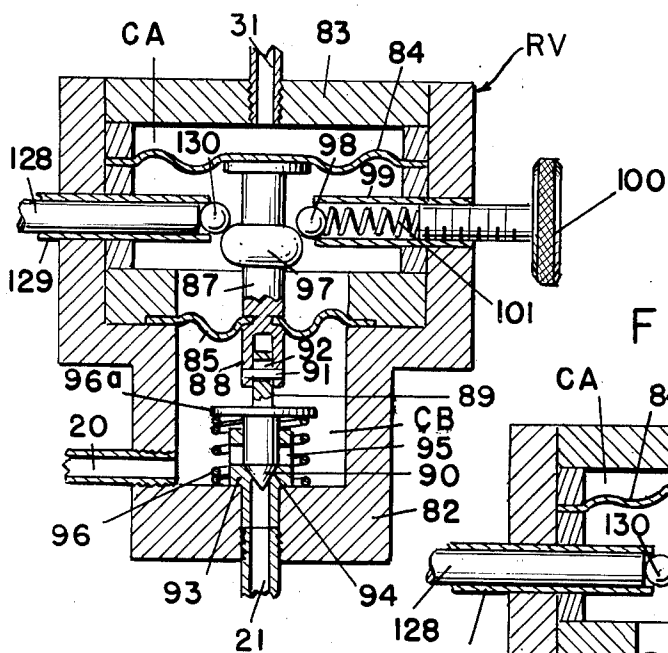


FIG. 8

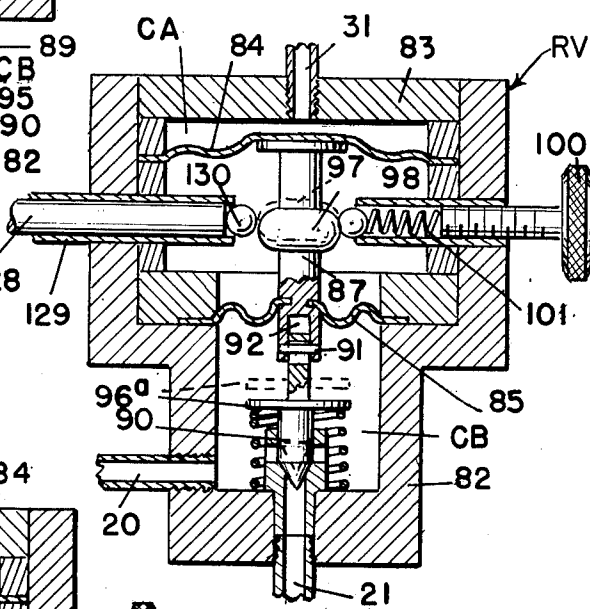
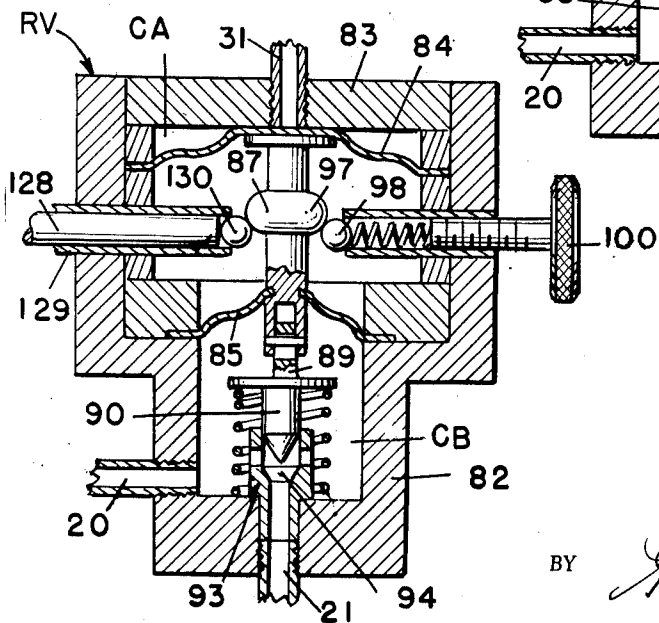


FIG. 9



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REFRIGERATION MACHINE

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Application December 30, 1943, Serial No. 68,141

19 Claims. (Cl. 62-4)

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This invention relates to improvements in refrigeration apparatus, and more particularly to improved refrigeration apparatus employing a single supply of refrigerant serving as a motive fluid for the compressor as well as providing refrigeration effect.

Broadly stated, it is a principal object of the invention to provide an efficient, practical and thoroughly dependable heat-operated mechanical refrigeration apparatus of the type employing a low boiling point refrigerant which when heated provides a high pressure vapor serving as the driving or motive fluid for the compressor.

The invention also aims to provide an improved motor-compressor unit for use in refrigeration apparatus of the stated character, which is so designed and constructed as to efficiently utilize the energy of the high pressure refrigerant vapor in driving the compressor component of the unit.

The invention also contemplates a novel, integrated system of controls for a heat-operated refrigeration apparatus as described, which provides in the first instance for the supply of a metered volume of fuel to the heat source in accordance with the cooling effect desired, and the admission of refrigerant vapor to the motor-compressor unit at the requisite pressure to motivate the same against the condenser pressure.

Yet another object of the invention is to provide an improved and efficient means powered by the high pressure refrigerant vapor for supplying make-up liquefied refrigerant to the generator or boiler in accordance with the demands of the apparatus.

The above and other objects and features of advantage of the invention will be seen from the following detailed description, taken with the accompanying drawings, wherein

Fig. 1 is a diagrammatic view illustrating an improved refrigeration apparatus according to this invention, together with the controls therefor;

Figs. 2 and 3 are longitudinal sections taken through the improved motor-compressor unit according to the invention, Fig. 3 illustrating the valve reversing function of the pistons;

Fig. 4 is a section taken along line 4-4 of Fig. 2;

Figs. 5 and 6 are, respectively, elevation and vertical sectional views of the fuel metering valve and controls associated therewith;

Figs. 7, 8 and 9 are vertical sectional views illustrating the anti-percolating or pressure regulating valve functioning to insure the requisite high pressure of the refrigerant vapor supplied to the motor-compressor unit; and

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Figs. 10 and 11 are vertical sectional views taken through the twin fuel injectors, illustrating their alternate operation by pressure supplied from the motor-compressor unit.

Referring to the drawings, wherein like reference characters designate like parts throughout the several views, a generalized description of the improved refrigeration apparatus as a whole will first be given, to be followed by a more detailed description of the component parts making up the same.

In Fig. 1, illustrating diagrammatically the apparatus as a whole, reference character B designates a boiler or generator of high pressure vapor, shown to comprise a heating vessel 10 enclosed within the shell of a small heater or furnace 11, the heat source therefor being shown as a main burner 12 and a pilot burner 13. The heated products of combustion are conducted from the heater by a flue generally designated 14. Liquefied make-up refrigerant is supplied to the boiler B alternately through lines 15, 16 from a receiver R through a liquid line 17 connected into a common supply line 18 extending between and delivering to twin injectors I₁, I₂ (Figs. 10 and 11), to the discharge sides of which the make-up lines 15, 16 are connected.

The liquid refrigerant is vaporized in the boiler B, and the high pressure vapor is conducted from the boiler through line 20 to an anti-percolating or pressure regulating valve RV (Figs. 7, 8 and 9) which functions to regulate the pressure at which the refrigerant vapor generated in the boiler is supplied to a motor-compressor unit MC (Figs. 2 and 3). Assuming the pressure of the vapor to have reached a predetermined high value, the regulating valve RV opens to pass the high pressure vapor through line 21 which re-enters the boiler and connects with a secondary heater 22 contained in its flue section 14, wherein superheat is added and any entrained liquid is vaporized. From the secondary heater, the superheated high pressure refrigerant vapor is conducted to the motor-compressor unit MC through supply line 24.

As will be described, the motor-compressor unit MC is of the connected piston type, the two pistons thereof being alternately driven by the high pressure refrigerant vapor to compress the warm or low pressure refrigerant vapor drawn from the evaporator E, the compressor discharging through lines 26, 27 connected to the opposite ends of the motor-compressor cylinder into a common line 28 into which also connects an exhaust line 28a extending into the midportion of the cylinder of the motor-compressor unit. The com-

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pressed refrigerant vapor passes from common line 28 through line 29 to a condenser C, from which the liquefied refrigerant may drain to receiver R through line 30. Liquefied refrigerant is conducted from receiver R to pressure regulating valve RV through line 31, and to the evaporator E through line 32 wherein, in providing cooling effect, it vaporizes under relatively low pressure, the low pressure vapor being drawn from the evaporator through line 34 into a common line 35 delivering to compressor inlet lines 36, 37, connected to the ends of the motor-compressor cylinder.

The burners 12, 13 are supplied with gas through individual supply lines 12a and 13a under the control of a fuel metering valve MV, to which gas supply line SL delivers. As will be later described, fuel (heating gas) is supplied to the pilot burner 13 at a constant low rate through the metering valve MV, and fuel is supplied to the main burner 12 through line 12a under the control of the metering valve, whose functioning is controlled conjointly by a thermostatic control TC responding to the rate of evaporation in the evaporator E and a supervisory control generally designated SC responding to the intended functioning of the regulating valve RV. The twin injectors I₁, I₂ are alternately operated in response to the differences in pressure existing in the power or inner-end spaces of the motor-compressor unit MC, and, to this end, take-off lines 38, 39 are connected to said cylinder, of which line 38 connects into branch lines 40a and 40b, and line 39 connects into branch lines 41a, 41b, said branch lines leading to said injectors, as will be more fully described hereinafter.

Now considering the component parts of the aforesaid described refrigeration apparatus, reference is had to Figs. 2, 3 and 4 illustrating the details of the improved motor-compressor unit which is motivated by the high pressure refrigerant vapor generated in the boiler B. As illustratively shown, the motor-compressor unit MC comprises an elongated cylinder 45 closed at its ends by the heads 46, 47, into which the previously described compressor discharge lines 26, 27 and the suction (inlet) lines 36, 37 are connected. The interior cylinder space is divided by a central partition 48 in which a connecting rod 49 has sliding bearing, the connecting rod mounting at its ends the pistons 50, 51 which accordingly reciprocate in unison, that is to say, as one of the pistons 50, 51 is motivated in outward direction by high pressure refrigerant vapor supplied to its under face, it imparts corresponding travel to the other piston.

High pressure refrigerant vapor enters one or the other power spaces of the cylinder from line 24 through a radial port 52 provided in the partition 48 and which opens into an annular passage 53 formed in the partition to encircle and open into a cylindrical cavity or bore 54 for a reversible admission valve 55. Said valve is preferably of the cylindrical plug type and has shorter length than the valve cavity 54 in which it operates, and in its end positions, it controls opening and closing of the smaller diameter end passages 56, 57 through which the valve cavity opens into the power spaces of the cylinder, as the term "power spaces" is employed to designate the cylinder inner-end spaces between the partition and under faces of the pistons 50, 51.

As best seen in Fig. 4, the admission valve 55 is provided at each end with a pair of

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diametrically opposed and axially aligned "flats," the aligned flats 58, 59, of the pairs having non-communicating relation due to the full-circular central portion of the valve body. Accordingly, when the valve 55 is disposed in its leftwise position, as in Fig. 1, high pressure refrigerant vapor is distributed to the inner-end space to the right of the partition 48 through port 52, annular passage 53, valve flats 59, right end of valve cavity 54, and the small diameter end passage 57 to cylinder. Conversely, when the valve 55 has moved to the right, as in Fig. 3, high pressure refrigerant vapor is distributed to the cylinder inner-end space to the other side of the partition 48 through the left end valve flats 58 and cavity end passage 56. Due to the full circular central portion of the valve body 55, supply of high pressure fluid to either of the cylinder spaces is interrupted as the valve is in its intermediate or dead-center position, whereby the possibility of a pressure lock occurring within the cylinder consequent to high pressure fluid being admitted to both power spaces at the same time is precluded.

To effect reversing action of the valve 55 requisite to its distributing function, it is provided with actuating pins 62, 63 projecting axially from the ends thereof. The action of said pins in effecting valve travel will be seen from a comparison of Figs. 2 and 3. In Fig. 2, wherein the valve is shown in its leftwise end position to which it has been previously actuated by engagement of piston 51 with pin 63, it will be seen that the left-hand pin 62 projects into the power space to the left of partition 48. As piston 50 moving to the right approaches the end of its stroke (this piston position being illustrated in Fig. 3), its under face will engage the pin 62 and shift the valve 55 to the right. Righthand pin 63 will now project into the power space to the right of the cylinder partition and is thus engageable by the piston 51 as the latter approaches the limit of its leftwise travel, such engagement effecting valve reversal and return movement of valve to its Fig. 2 position.

The partition 48 also mounts a spool-type reversible exhaust valve 65 which, depending upon its axial end position, controls exhaust from the power or inner-end cylinder space not being supplied with high pressure vapor. Said valve comprises a cylindrical body having enlarged diameter ends 66, 67, the body sliding in a cylindrical through passage or bore formed in the partition and which is counterbored at both ends to provide seats 68, 69 for the large-diameter valve ends 66, 67. The total length of the valve 65 is greater than the axial thickness of the partition 48, and its body portion is provided with opposed "flats" 70, 71 (Fig. 4) extending between the enlarged ends 66, 67 of the valve. Said flats provide communication between one or the other of the enlarged counter-bores 68, 69 and an annular passage 73 formed in the partition to encircle the valve passage and which opens to a radial port 74 into which the exhaust line 28a (Fig. 1) is connected. The action of the exhaust valve 65 will also be seen from a comparison of Figs. 2 and 3. In Fig. 2, the valve is shown in its leftwise position to which it has been previously actuated by piston 51 in its leftwise travel. In such position, the power space to the right of the partition 48 is being supplied with high pressure refrigerant vapor from the boiler, and such pressure is effective on the face of the large-diameter

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right end 67 of the valve, whereby said valve is not only seated but its end 67 has sealing engagement with the internal shoulder formed by the counter-bore 68. However, the left end of the valve is spaced from its counter-bore 68, with the result that the power or inner-end space to the left of the partition, in which expansion has taken place and the expanded vapor is being compressed by the under face of piston 50 which is now acting as a compression face, may exhaust through the left end of the valve to line 28a. Upon piston 50 approaching the end of its travel to the right, it engages against the enlarged end 66 of the valve and shifts same to the right, closing the left end of the valve, and permitting the power space to the right of the partition 43 to exhaust through the right end of the valve into exhaust line 28a. As diagrammatically shown in Figs. 2 and 3, outwardly opening check valves 76, 77 are provided between the outer-end compression spaces of the cylinder and the discharge lines 26, 27, respectively, and similar but inwardly opening check valves 78, 79 are provided between said compression spaces and suction lines 36, 37, respectively.

With the construction so far described, and assuming admission valve 55 to be in its leftwise position shown in Fig. 2, high pressure refrigerant vapor from the boiler is being supplied to the power space between partition 43 and under face of piston 51 acting as a power piston, and accordingly piston 51 is actuated to the right, the outer face of the piston compressing the warm vapor previously drawn into the right outer-end compression space of the cylinder, the compressed vapor discharging through line 27 to condenser C, suction line 37 being meanwhile closed by check valve 79. The exhaust valve 65 is also in its leftwise position, and the high pressure vapor admitted to the under face of piston 51 is also effective against the face of the enlarged valve end 67. Due to the positive connection between pistons 50, 51 provided by connecting rod 49, piston 50 is also motivated to the right. Check valve 78 is open as shown, so that the outer face of the piston 50 functions as a pump face, drawing warm gas from the evaporator E into the left outer-end space of the cylinder through line 36, check valve 76 meantime closing off the discharge line 26. The under face of piston 50 is now also functioning as a compression face, and the expanded vapor compressed in the left inner-end cylinder space is exhausting through the open left end of valve 65 into exhaust line 28a, this exhaust combining with the compressed vapor discharging through line 27 and being conducted to condenser through line 29.

Upon piston 50 reaching the end of its rightwise travel, it effects reversal of both admission valve 55 and exhaust valve 65 and movement thereof to the right, as seen in Fig. 3. Accordingly, the high pressure vapor boiler is now effective against the under face of the piston 50, causing it to move to the left and impart corresponding leftwise movement to piston 51. Leftwise travel of piston 50 compresses the warm gas previously drawn into the compression space at the left outer end of the cylinder through line 36, discharging the compressed vapor through line 26 to condenser. The under face of piston 51 is now acting as a compression face, the compressed vapor exhausting through valve 65 to lines 28a, 29 and thence to condenser. The outer face of piston 51 acts as a pump face, drawing the warm vapor from the evap-

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orator into the right outer-end of the cylinder. This cycle is repeated at a rate which varies with the demands of the system, such corresponding to the rate of evaporation in the evaporator.

Considering that the pistons 50, 51, when functioning as compressor pistons, are working against condenser pressure which is effective on a total piston area equivalent to substantially twice the area of a single piston face, it becomes obvious that the motivating fluid (high pressure refrigerant vapor from the boiler) must have a pressure value at least equal to twice the condenser pressure. This disregards evaporator pressure effective on the outer face of the piston then acting as a pump face, but for the purpose of discussing the principles of the present invention, evaporator pressure, though recognized, can be discounted. To insure the requisite high pressure of the refrigerant vapor supplied to the power spaces of the motor-compressor unit for the reason assigned in the foregoing, the above referred to pressure regulating valve RV is provided.

Referring to Figs. 7, 8 and 9, illustrating the details thereof, the pressure regulating valve RV preferably comprises a stepped diameter casing 82 having a closed bottom, and being closed at its top as by a plate 83 into which line 31 from receiver R is connected. Mounted in the large-diameter upper end of the casing is a flexible diaphragm 84 defining with the under face of the closure plate 83 a sealed chamber CA which is at receiver and hence condenser pressure. Mounted in the small diameter end of the casing is a flexible diaphragm 85 defining a lower sealed chamber CB into which connects the vapor line 20 from the boiler and the vapor line 21 extending to the secondary heating chamber 22. Accordingly, pressure in chamber CB corresponds to the pressure in the vapor generated in the boiler B, or more simply, to boiler pressure.

In a system wherein the pressure of the motive fluid must be at least twice the condenser pressure, the diaphragm 84 has an effective face area which is twice that of the diaphragm 85, and said diaphragms are connected to move in unison by an axial stem 87, having a depending tubular end 88 extending into the chamber CB, which receives a pin 89 projecting upwardly from a needle valve 90. A lost-motion connection is provided between the lower end 88 of the stem 87 and the valve pin 89, such illustratively taking the form of a pin and slot connection 91, 92, which permits limited upward movement of the diaphragm without corresponding upward movement of needle valve 90.

Needle valve 90 operates in the bore of a tubular end fitting 93 with which line 21 is in communication, the fitting being shaped to provide a conical valve seat 94 for the needle valve 90. An opening 95 in the circular side wall of the fitting 93 provides communication between chamber CB and line 21, said opening being closed when the needle valve is full-seated, as illustrated in Figs. 7 and 8 (full lines). The needle valve 90 is normally seated so as to seal off line 21 from line 20 by the preponderance of pressure in chamber CA, against the bias of a relatively light spring 96 whose ends react between the bottom wall of the chamber CB and a radial flange 96a formed on the valve body intermediate its pointed valve end and its pin end.

Intermediate its ends, the diaphragm connecting stem 87 is formed with a knob-like enlargement 97 functioning as a double wedge cam, and positioned in the path of travel of said enlargement is a ball 98 which partially projects from, although being retained in, the end of a tubular retainer sleeve 99, the ball being backed by a spring 100 whose tension is adjustable by a thumb screw 101 threaded into said sleeve. The ball 98 bears against the stem enlargement 97 with spring pressure and it serves, in conjunction with said enlargement, to exercise a holding action on the stem when the latter moves to one or the other of its extreme positions as illustrated in Figs. 7 and 9, consequent to a substantial decrease of pressure in one or the other of the chambers CA or CB.

The functioning of the regulating valve will now be briefly analyzed. If both the pilot and the main burners 13, 12, respectively, are unlighted, pressure in chamber CB is of course low and needle valve 90 is closed on its seat 93 by the relatively high condenser pressure effective on diaphragm 84, so that the apparatus is completely inoperative even though the thermostat T associated with the evaporator may call for cooling. Consequent to the low pressure in chamber CB, the diaphragm stem 87 will be actuated to its low position illustratively shown in Fig. 7, the ball 98 thereupon exercising holding action on said stem. Such arrangement provides a valuable safety feature in that the system is rendered inoperative if the pilot light should be extinguished accidentally.

Normally, however, the pilot burner 13 is lighted and its adjustment is such that it provides sufficient heat for the boiler to vaporize the liquefied refrigerant to a predetermined normal operating pressure which is such as to result in a predetermined pressure difference obtaining in the chambers CA and CB. Under this condition of predetermined pressure difference, diaphragms 84 and 85 assume their normal operating i. e. balanced position illustrated in full lines, Fig. 8, but needle-valve 90 remains closed as permitted by the pin and slot connection 91, 92 between tubular end 88 of the diaphragm connecting stem 87 and the upwardly projecting valve pin 89. In said position, ball 98 bears on the upper middle zone of the stem enlargement 97, having been plied to the right by the cam action of said enlargement, and the system is accordingly conditioned for operation upon the thermostat T calling for cooling.

If, now, the thermostat T calls for cooling, fuel metering valve MV becomes effective, as will be described, immediately to supply fuel to the main burner 12 through line 12a. Additional heat is accordingly supplied to boiler B, thereby increasing the rate of vaporization and the pressure within the boiler B. This increased pressure becomes almost immediately effective in chamber CB and causes slight upward movement of the diaphragms 84, 85 and stem 87, which slightly raises the stem enlargement 97 to its dotted line position (Fig. 8) and also effects limited upward or unseating movement of valve 90, as the permissible relative movement between stem 87 and valve 90 as provided by the lost motion connection 91, 92 has now been exceeded. Upon the needle valve 90 being raised from its seat, spring 96 becomes immediately effective, and snaps said valve to full open position. Accordingly, full volume of high pressure vapor from chamber CB

may now pass through the valve into line 21 and thence to motor-compressor unit MC, placing the same in operation. As valve opening movement under the urge of spring 96 as aforesaid is independent of diaphragm and stem movement, said diaphragms maintain a working position in which the full middle zone of the stem enlargement 97 is engaged by the ball 98. Thus, assuming the demand for cooling to continue, there results a regulated supply of high pressure vapor being passed through the regulating valve RV to the motor-compressor unit, and the latter thereby operates at a rate determined by the cooling effect called for by the thermostat T.

However, should pressure within the boiler tend to build up to unsafe limits, the diaphragms 84, 85 and connecting stem 87 also tend to move upwardly to the position shown in Fig. 9. Once this move has been initiated, the ball 98 exercises camming effect on the lower rounded surface of the stem enlargement 97 and thereby completes raising movement of stem and diaphragms. This full raising movement of the diaphragms and stem maintains full communication between lines 20, 21 irrespective of cooling requirements, and, as will be described, also cuts off the supply of gas to the main burner 12 until pressure in chamber CB lowers sufficiently to reestablish the normal difference in pressure in the chambers CA and CB, whereupon the system operates in the manner above described in response to a demand for cooling.

From the above general statement of functioning, it will be observed that while the pressure regulating valve RV regulates the pressure of the refrigerant vapor supplied to the motor-compressor unit, the primary control for normal operating conditions is exercised through the rate at which fuel is supplied to the main burner 12, through fuel metering valve MV, responsively to the rate of evaporation in the evaporator E, as through the temperature control TC. However, it is a feature of the invention that the regulating valve RV exercises supervisory control on the fuel supply to burner 12, and to permit a clearer understanding of the integration between the regulating valve, the metering valve, and the system controls, the metering valve will now be described in detail.

Referring to Figs. 5 and 6, the aforesaid fuel metering valve MV preferably comprises a cylindrical, step-diameter housing 105 having an interior partition 106 dividing the housing into chambers 107, 108. The large-diameter end of the casing is closed by a face plate 109 into which connects the fuel supply line SL. Communication between chambers 107 and 108 is provided by an opening 110 in the partition 106, which is normally closed by a disc-valve 111 illustratively shown to be mounted at one end of a leaf spring 112 contained in housing 107, the other end thereof being fixed generally as shown. Gas line 13a to pilot burner 13 is also connected into the face plate 109 so as to communicate with chamber 107 thereof, whereby gas from the supply line SL passes at a constant low rate through gas line 13a to the pilot burner 13. Disc valve 111 is adapted to be opened against the bias of spring 112 upon leftwise movement of a pointed stem 113 mounted on the axis of the housing chamber 108 and carried by and to one side of a flexible sealing diaphragm 114 extending across it, the stem having an enlarged boss-like end 115 extending to the other side of the diaphragm 114.

The small diameter end of the valve housing

also mounts externally thereof a thermostat diaphragm 116 sealed within an enclosure 117, the diaphragm being responsive to expansion and contraction of a heat-sensitive medium contained in the space above the diaphragm and in a capillary 117 extending to the thermostat T associated with evaporator E, said parts being collectively referred to as the thermostat control TC. The diaphragm 116 bears on a pin 120 journaled for limited sliding movement through the side wall of the small diameter end of the housing 103 and whose upper end projects into the diaphragm enclosure 117 to engage with said diaphragm. The pin 120 is biased in upward direction by means of a coaxially disposed spring-loaded pin 121 acting on the lower end of the pin 120 through the intermediary of a ball 122. Said pin 121 is spring-loaded by a spring 123 operating in a sleeve 124 into which an adjusting thumb screw 125 is threaded, said adjusting screw providing a simple means for adjusting the pressure at which deflection of diaphragm 116 is effective to cause downward movement of pin 120 against the bias of spring 123.

According to the invention, the valve 111 controlling supply of fuel to the main burner 12 is opened upon the evaporator thermostat T calling for cooling, and provided the normal pressure difference obtains in chambers CA and CB of regulating valve RV. To satisfy such requirement, a control rod 128 of the referred to supervisory control SC extends between regulating and metering valves, the rod operating in a sleeve 129. The right end of the rod and sleeve projects into the regulating valve casing 32, said sleeve end mounting a ball 130 which closely abuts the end of the control rod and engages against stem enlargement 97, being aligned with the previous described ball 98, having similar engagement.

The other or left end of the control rod 128 and its sleeve 129 projects into the small diameter chamber 108 of the metering valve housing 105, said sleeve end mounting a ball 132 closely abutting the related end of the control rod. When the control rod 128 is plied to the left by stem enlargement 97 engaging against the control rod end-ball 130 (Fig. 8), which occurs when the condition of normal pressure difference obtains in chambers CA and CB of the regulating valve RV, said control rod end-ball 132 is disposed in the path of movement of the ball 122, as seen in Fig. 6.

Thus, upon the thermostat T calling for cooling, diaphragm 116 is distended downwardly, and pin 120 is also forced downwardly, such forcing ball 122 downwardly into alignment with ball 132 at the metering valve end of the control rod 128, the position of the latter being considered fixed under the condition of the normal pressure difference obtaining in the chambers of the regulating valve. Such downward movement of ball 122 causes it to be cammed to the left by ball 132, whereupon it engages against the enlarged end 115 of the valve actuating stem 113, causing leftwise movement thereof which results in opening of the valve 111. Accordingly, fuel may pass through the metering valve to the main burner 12 in accordance with the cooling effect called for.

In the event the pilot burner 13 has become accidentally extinguished, or if the pressure in chamber CB of the regulating valve has exceeded safe limits, control rod 128 may move freely to the right the limited distance permitting the thermostat actuated ball 122 to lower

without forcing the stem 113 against valve 111. Thus, even if the thermostat calls for cooling under the above conditions, valve 111 remains closed. Hence, it will be seen that the regulating valve RV exercises a supervisory control over the functioning of the fuel metering valve by insuring the supply of fuel to the main burner only under the normal condition of predetermined pressure difference obtaining in the regulating valve, or under the operating condition of said pressure difference not substantially exceeding the predetermined difference.

As previously forecast, the injectors I₁, I₂ are operated alternately to supply make-up liquid to the boiler B by the difference in pressure of the refrigerant vapor in the motor-compressor unit MC. Referring to Figs. 10 and 11, illustrating the construction and operation of the injectors, each comprises a thick-walled cylinder, of which cylinder of injector I₁ is designated 135 and the cylinder of injector I₂ is designated 136. Both said cylinders may be provided with heat dissipating fins 137, 138 formed by circumferential grooves in the mid portions of the cylinders. The cylinder 135 is closed at its ends by an upper end member 139 and a lower end member 140, the cylinder 136 being closed by similar end members 141, 142. The cylinder 135 is provided with a vertically disposed plunger bore 143 opening at its lower end into a valve cavity 144 provided in the bottom end member 140 and in its upper end into a piston bore 145 which is closed at its upper end by the end member 139. The cylinder 136 is provided with a similar plunger bore 147 opening at one end to valve cavity 148 in end member 142 and at its other end to piston bore 149 closed at its upper end by end member 141.

Mounted to reciprocate in the piston and plunger bores of injector cylinder 135 is an interconnected piston 150 and plunger 151, and a similar interconnected piston and plunger 152, 153 is mounted in the piston and plunger bores of the injector cylinder 136.

Disposed in the cavity 144 is check valve 155, preferably of disc form, which is biased to its raised (closed) position by an annular spring 156 provided with flexible valve supporting fingers. Disposed in cavity 148 is a similar disc valve 157 biased to its closed position by an annular spring 158 similar to spring 156.

The previously described branch line 40a of the take-off line 38 which is connected to the inner-end space of motor-compressor cylinder to the left of the partition 48 thereof is connected into the upper end member 139 of the injector cylinder 135 to supply high pressure refrigerant vapor to the upper face of piston 150, and branch line 40b is connected to the cylinder 136 at a point thereof as simultaneously to supply high pressure vapor to the under face of piston 152. Branch line 41a from take-off line 39 conducts the high pressure refrigerant vapor from the cylinder inner-end space to the right of partition 48 to cylinder 135 at a point thereof so as to supply high pressure vapor to the under face of piston 150, the other branch line 41b being connected into upper end member 141 of cylinder 136 so as to supply vapor to the upper face of piston 149.

The liquid line 18 through which liquefied refrigerant from receiver R is conducted to the injectors is connected at its ends into the plunger bores 143 and 147 of the respective cylinders, connection being made at a point thereof immediately below the end face of the plungers 151, 153

when the latter are in their uppermost position. The liquid make-up lines 15, 16 connect into the cavities 144, 143, respectively, of the cylinder end members 140, 142.

Assuming high pressure refrigerant vapor being admitted to the cylinder space to the right of partition 48 of the motor-compressor cylinder, such high pressure vapor is conducted through take-off line 39 and branch lines 41a, 41b simultaneously to the under face of piston 150 and the upper face of piston 152. Although refrigerant vapor is being supplied also to the upper face of piston 150 and the under face of piston 152 through take-off line 38 connected to the cylinder space to the left of said partition 48, the pressure therein is lower than in the cylinder space to which the high pressure vapor from the generator is being supplied, due to the fact that expansion has occurred in the former cylinder space. Accordingly, piston 150 is motivated upwardly and piston 152 is motivated downwardly, with corresponding movement being imparted to the piston plungers 151, 153. As piston plunger 151 moves upwardly, it draws a charge of liquefied refrigerant from line 18 into its bore and, as plunger 153 moves downwardly, it forces the charge of liquefied refrigerant previously drawn into its bore past valve 157 and to the boiler B through make-up line 16. Upon reversal of the valve 55 of the motor-compressor unit being effected, the reverse action takes place, namely, the injector piston 150 is driven downwardly to force the charge of liquefied refrigerant contained in bore 143 past the valve 155 to make-up line 15, and the piston 152 is driven upwardly to draw a charge of liquefied refrigerant into the lower end of its bore 147. Since the rate at which the injectors I₁ and I₂ operate depends on the rate of operation of the motor-compressor unit, and as the latter depends on the rate of cooling called for by the thermostat T, it will be seen that make-up refrigerant in liquefied form is supplied to the boiler B in accordance with the rate of evaporation in the evaporator E.

Refrigeration apparatus as described in the foregoing, in which a single supply of refrigerant, such as "Freon" serves as the working fluid for the motor-compressor unit, provides many practical advantages. The apparatus may be made substantially smaller and more compact than conventional refrigeration apparatus while at the same time giving at least equal refrigeration service. It is self-lubricating by simply admixing or dissolving oil in the refrigerant. The motor-compressor may be built as a completely sealed unit with no leakage. It permits of reduction in size of the condenser consequent to high efficiencies attained, and of the elimination of electric motors for both compressor and condenser cooling fan. Another advantage of the improved refrigeration system resides in its substantially noiseless operation.

As many changes could be made in carrying out the above constructions without departing from the scope of the invention, 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.

I claim:

1. In refrigerating apparatus employing a compressor-condenser-expander circuit and utilizing high-pressure vaporized refrigerant as the motive fluid, a fluid motor-compressor-pump unit comprising a cylinder having end closures

and a transverse partition subdividing its interior space, pistons mounted for reciprocatory travel in the cylinder end-spaces, a piston rod having sliding bearing in the partition and interconnecting said pistons for movement in unison, means for distributing high-pressure vaporized refrigerant through the partition alternately to the cylinder inner-end spaces thereby to motivate said pistons alternately throughout their compression strokes, means for exhausting the cylinder inner-end space not then being supplied with the vaporized refrigerant through the partition, whereby the piston operating in said space is actuated throughout its combined pump and exhaust stroke by the piston moving throughout its compression stroke, valved conduits connected to the cylinder end-closures for admitting low pressure vaporized refrigerant into the cylinder outer-end spaces under the suction effect of the pistons as the latter move throughout their exhaust stroke, and valved conduits connected to the cylinder end-closures for delivering the refrigerant compressed by the pistons as the latter are motivated throughout their compression strokes to the condenser.

2. Refrigerating apparatus as set forth in claim 1, wherein the means for exhausting the cylinder inner-end spaces and the compressed refrigerant conduits are connected into a common line delivering to the condenser.

3. Refrigerating apparatus as set forth in claim 1, wherein the distributing and exhaust means are actuable by the pistons upon the latter approaching the end of their exhaust strokes.

4. Refrigeration apparatus comprising a fluid motor-compressor unit, a condenser and an evaporator, said motor-compressor unit comprising a cylinder having a central partition, pistons interconnected to reciprocate in unison within the opposite ends of the cylinder upon motive fluid being supplied alternately to the adjacent under faces of said pistons, the cylinder being closed at its ends and suitable valved connections between the cylinder outer-end spaces and each of said condenser and evaporator, whereby the outer faces of the pistons function alternately as compression and pump faces, the cylinder inner-end spaces being connected to condenser whereby the underface of the piston not being supplied with motive fluid also functions as a compressor face, means for vaporizing under relatively high pressure a controlled portion of the refrigerant liquefied in the condenser, and means for supplying the high pressure refrigerant vapor alternately to the working faces of said pistons.

5. Refrigeration apparatus comprising a fluid motor-compressor unit, a condenser and an evaporator, said motor-compressor unit comprising a cylinder having a central partition, pistons interconnected to reciprocate in unison within the opposite ends of the cylinder upon motive fluid being supplied alternately to the adjacent under faces of said pistons, the cylinder being closed at its ends and suitable valved connections between the cylinder outer-end spaces and each of said condenser and evaporator, whereby the outer faces of the pistons function alternately as compression and pump faces, the cylinder inner-end spaces being connected to condenser whereby the under face of the piston not being supplied with motive fluid also functions as a compressor face, means for vaporizing a portion of the refrigerant liquefied in the condenser under a sufficiently

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high pressure that it will motivate the one piston on which it is effective against the condenser pressure effective on the two faces of the pistons then functioning as compression faces, and means for supplying the refrigerant vapor at the afore-

6. Refrigeration apparatus as set forth in claim 4 combined with means actuated by the high pressure refrigerant vapor for supplying make-up liquefied refrigerant to the vaporizing means for vaporization therein.

7. Refrigeration apparatus as set forth in claim 4 combined with means for supplying liquefied refrigerant from the condenser to the vaporizing means, said last means being actuable by the difference in pressure effective on the underfaces of said pistons.

8. Refrigeration apparatus as set forth in claim 4, wherein said vapor supplying means includes a reciprocating valve actuable by said pistons in their reciprocation.

9. Refrigeration apparatus as set forth in claim 4, wherein said vapor supplying means includes a reversing valve mounted in the cylinder partition and being actuable to reverse positions by said pistons.

10. Refrigeration apparatus as set forth in claim 4, wherein said last means includes a reversing valve mounted in the cylinder partition so as to be actuable to reverse positions by said pistons in which it admits the high pressure vapor to one or the other cylinder inner-end spaces, said valve being operative in its intermediate position to positively interrupt admission to both said spaces.

11. Refrigeration apparatus as set forth in claim 4, wherein first and second reversible valve means are mounted in the cylinder partitions so as to be actuable to reverse positions by said pistons, the first valve means being operative to admit the high pressure vapor to one cylinder inner-end space for expansion therein and the second valve means being operative simultaneously to exhaust the expanded vapor being compressed in the other cylinder inner-end space to condenser.

12. In heat-operated refrigeration apparatus, a fluid motor-compressor unit, a condenser and an evaporator, a boiler supplied with liquefied refrigerant from the condenser and operating to vaporize a controlled portion of the liquefied refrigerant, heating means for the boiler, pressure responsive means effective to supply the vaporized refrigerant at a predetermined working pressure to the motor-compressor unit, and means for controlling the actuation of the heating means, said last means being jointly responsive to the rate of evaporation in the evaporator and to any deviation in the pressure of said vaporized refrigerant from the predetermined working pressure.

13. In heat-operated refrigeration apparatus, a fluid motor-compressor unit, a condenser and an evaporator, a boiler supplied with liquefied refrigerant from the condenser and operating to vaporize a controlled portion of the liquefied refrigerant, heating means for the boiler comprising pilot and main burners and fuel supply lines thereto and a fuel metering valve normally closing the supply line to main burner and operative to supply a constant low volume of fuel to the pilot burner which is such as to effect vaporization of the liquid refrigerant to a predetermined normal pressure, pressure regulating means for regulating the pressure of the vaporized refrig-

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erant and being effective to supply refrigerant vapor at a working pressure in excess of the normal pressure to the motor-compressor unit, and means responsive to the rate of evaporation in the evaporator and the maintenance of the predetermined normal pressure for actuating said metering valve to open the fuel supply line to the main burner.

14. In heat-operated refrigeration apparatus, a fluid motor-compressor unit, a condenser and an evaporator, a boiler supplied with liquefied refrigerant from the condenser and operating to vaporize a controlled portion of the liquefied refrigerant, heating means for the boiler, a pressure regulating valve means connected between boiler and motor-compressor unit for supplying the vaporized refrigerant to the latter, said valve means responding to both condenser and boiler pressures and being operative to establish a predetermined high pressure of the vaporized refrigerant supplied to the motor-compressor unit relative to condenser pressure, means for controlling the heating means responsively to the rate of evaporation in the evaporator, and supervisory control means actuable by the pressure regulating valve means to render said first control means ineffective upon the heating means becoming inoperative or upon the boiler pressure substantially exceeding said predetermined pressure.

15. In heat-operated refrigeration apparatus, a fluid motor-compressor unit, a condenser and an evaporator, a boiler supplied with liquefied refrigerant from the condenser and operating to vaporize a controlled portion of the liquefied refrigerant, heating means for the boiler, a pressure-regulating valve means connected between boiler and motor-compressor unit for supplying the vaporized refrigerant to the latter, said valve means including a first large-area diaphragm responsive to condenser pressure, a second small-area diaphragm responsive to boiler pressure, a stem connecting said diaphragm whereby they move in unison, and a valve element for controlling passage of the vaporized refrigerant to the motor-compressor unit, the valve element having a lost-motion connection with said stem providing for movement of diaphragms and stem to a normal position without corresponding opening movement of the valve element, in which normal position the boiler pressure exceeds the condenser pressure by a predetermined amount determined by the difference in the areas of said diaphragms, said heating means being adjusted normally to supply heat to the boiler sufficient to maintain said normal position of said diaphragms, means responding to the rate of evaporation at the evaporator for supplying additional heat to the boiler resulting in increased boiler pressure and thereby corresponding further movement of said diaphragms and limited opening of said valve element, and spring means associated with said valve element for actuating said valve element to full open position upon its opening movement being initiated, as aforesaid, thereby to supply a full volume of vapor at the increased boiler pressure to the motor-compressor unit.

16. Heat-operated refrigeration apparatus as set forth in claim 13, wherein said pressure regulating means includes a first large-area diaphragm responding to condenser pressure, a second smaller-area diaphragm responding to boiler pressure, said diaphragms being connected to move in unison by a stem having a cam-like en-

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largement thereon, said pilot burner being adjusted so as normally to supply sufficient heat to the boiler to maintain the predetermined normal pressure of the vaporized refrigerant, and wherein the means for actuating the metering valve to open position includes a valve actuating element, a member movable responsively to the rate of evaporation in the evaporator from an inactive to an active position in which it is conditioned to actuate said valve actuating element, and a supervisory control member actuable by said stem enlargement to a position in which it is effective to translate movement of said movable member to active position into motion of said valve actuating element and opening of said metering valve.

17. In heat-operated refrigeration apparatus, a fluid motor-compressor unit, a condenser and an evaporator, a boiler supplied with liquefied refrigerant from the condenser and operating to vaporize a controlled portion of the liquefied refrigerant, heating means for the boiler, pressure responsive means effective to supply the vaporized refrigerant at a predetermined working pressure to the motor compressor unit, means for supplying fuel to the heating means in accordance with the demand for evaporation in the evaporator, and means for inter-relating the actuation of the fuel supplying means with that of the pressure responsive means.

18. In heat-operated refrigeration apparatus, a fluid motor-compressor unit, a condenser and an evaporator, a boiler supplied with liquefied refrigerant from the condenser and operating to evaporate a controlled portion of the liquefied refrigerant, heating means for the boiler comprising pilot and main burners and fuel supply lines thereto and a fuel metering valve normally closing the supply line to main burner and operative to supply a constant low volume of fuel to the pilot burner which is such as to effect vaporization of the liquid refrigerant to a predetermined normal pressure, pressure regulating means for regulating the pressure of the vaporized refrigerant and being effective to supply refrigerant vapor at a working pressure in excess of the normal pressure to the motor-compressor unit, means for actuating the fuel metering valve to open and supply line to main burner in response to a demand for evaporation in the evaporator, and means inter-relating the actuation of said last means with that of the pressure regulating means in such manner that the fuel

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metering valve is operative to open said supply line to main burner only upon the predetermined normal pressure of the refrigerant having first been established.

19. Refrigeration apparatus comprising a fluid motor-compressor unit, a condenser, and an evaporator, said motor-compressor unit comprising a cylinder having a central partition, pistons interconnected to reciprocate in unison in the opposite ends of the cylinder, both the outer and the inner end spaces of each cylinder end being connected to the condenser, the arrangement being such that the pistons are motivated by working fluid supplied alternately to the underfaces of the pistons at a pressure sufficiently that the under-face of the piston not being supplied with working fluid acts as a compression face for delivering the expanded fluid previously supplied thereto as the working fluid to the condenser, a boiler for vaporizing a portion of the liquefied refrigerant under a pressure which is sufficiently high as to motivate the piston against the condenser pressure effective on the two piston faces then acting as compression faces, means for supplying the high pressure refrigerant vapor alternately to the inner-end spaces of said cylinder, and means for supplying liquefied refrigerant from the condenser to the boiler comprising a pair of injectors each including a cylinder, a power piston and a plunger actuated by said piston to draw a charge of liquefied refrigerant into the cylinder bore and thereupon to forward said charge to boiler, a branched take-off line connecting each inner-end cylinder space of the motor-compressor unit with both injector cylinders, the branches of each line delivering to the opposite faces of the two power pistons, whereby said power pistons are actuated in opposite directions by the difference in the pressure of the refrigerant vapor existing in said inner-end cylinder spaces.

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