United States Patent [19]
Lieberman
[11] 3,733,845
[45] May 22, 1973

[54] CASCADED MULTICIRCUIT, MULTIREFRIGERANT REFRIGERATION SYSTEM

[76] Inventor: Daniel Lieberman, 2931 Piedmont Avenue, Berkeley, Calif. 94705

[22] Filed: Jan. 19, 1972
[21] Appl. No.: 218,870

[52] U.S. Cl. 62/335, 62/175, 62/114, 62/95, 62/120, 62/502, 62/510, 62/512
[51] Int. Cl. F25b 7/00
[58] Field of Search..........................62/175, 335, 79, 62/114, 95, 120, 332, 502, 510

[56] References Cited

UNITED STATES PATENTS

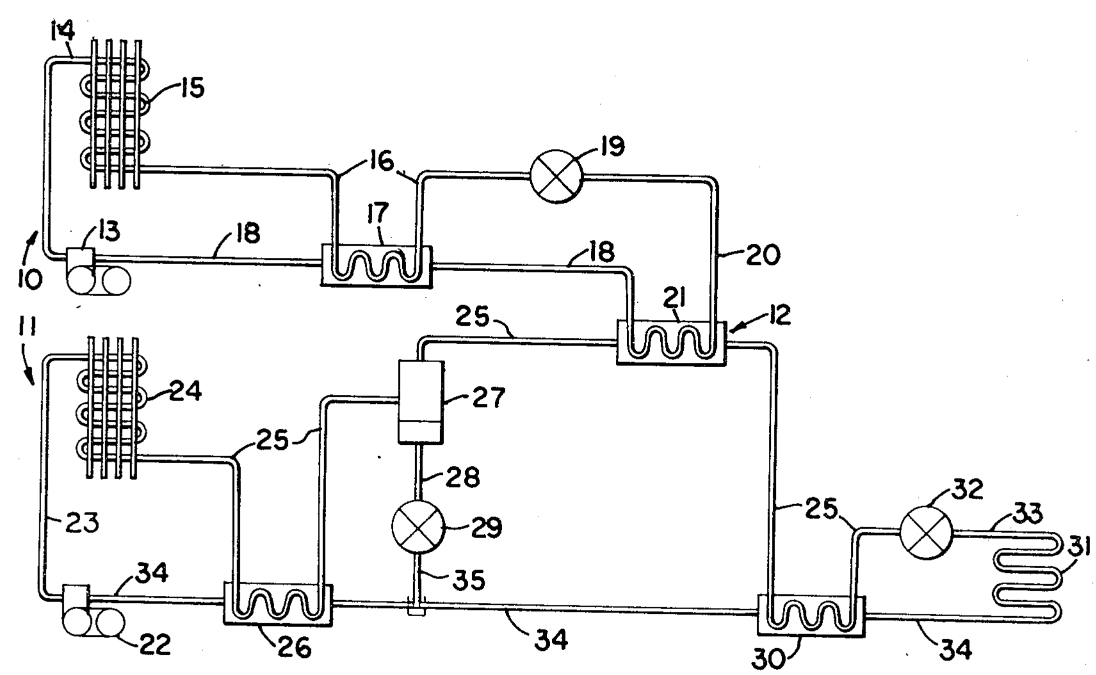
2,680,956	6/1954	Haas	62/175
2,717,765	9/1955	Lawler	62/175
3,019,614	2/1962	Schubert	62/502
3,203,194	8/1965	Fuderer	62/114
3,392,541	7/1968	Nussbaum	62/175

Primary Examiner—William J. Wye
Attorney—Carlisle M. Moore

[57] ABSTRACT

A refrigeration system having two or more closed refrigerant circuits in which each circuit has a compressor, a condenser connected to the compressor outlet, a high-pressure line connecting the condenser to an evaporator and a low-pressure line returning from the evaporator to the compressor inlet and in which each circuit after the first has at least one vapor-liquid separator in the high-pressure line and is charged with a mixture of refrigerants. The evaporator of each circuit is in heat exchange with the high-pressure line of the next successive circuit at a point downstream of the first separator therein. The system enables very low temperatures to be reached in the evaporator of the final circuit with conventional, relatively low-pressure, oil-lubricated compressors being used throughout.

10 Claims, 5 Drawing Figures



Patented May 22, 1973 3,733,845

3 Sheets-Sheet 1

FIG_1

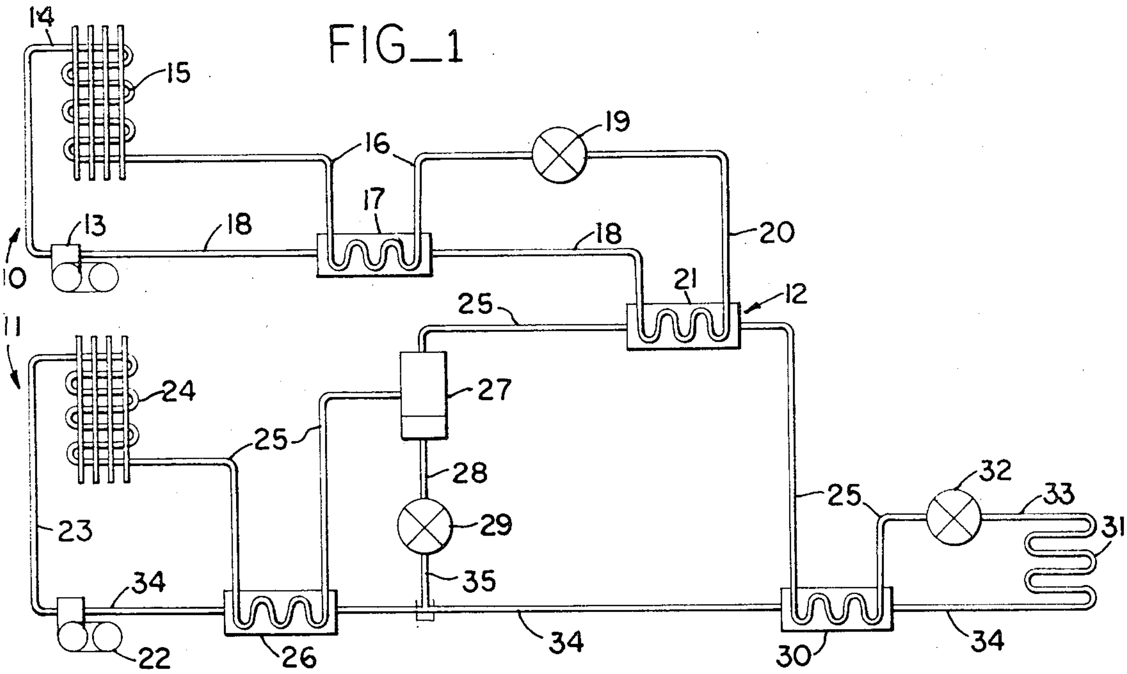
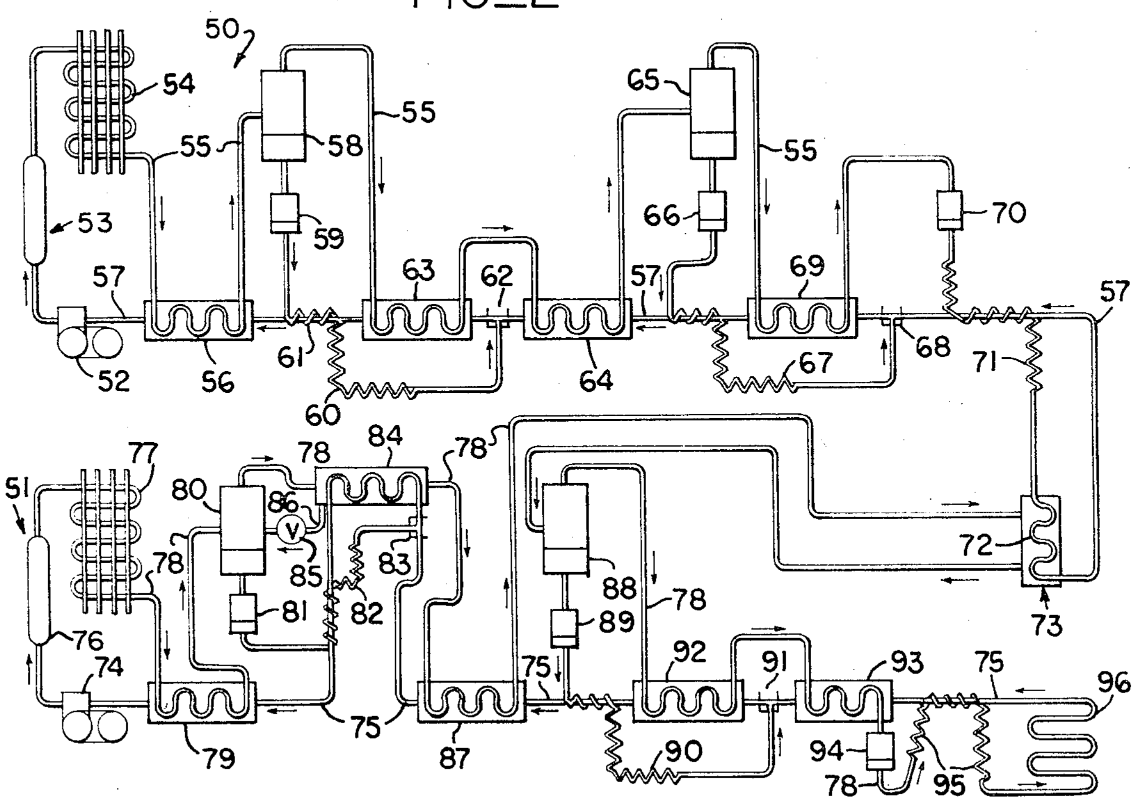


FIG_2



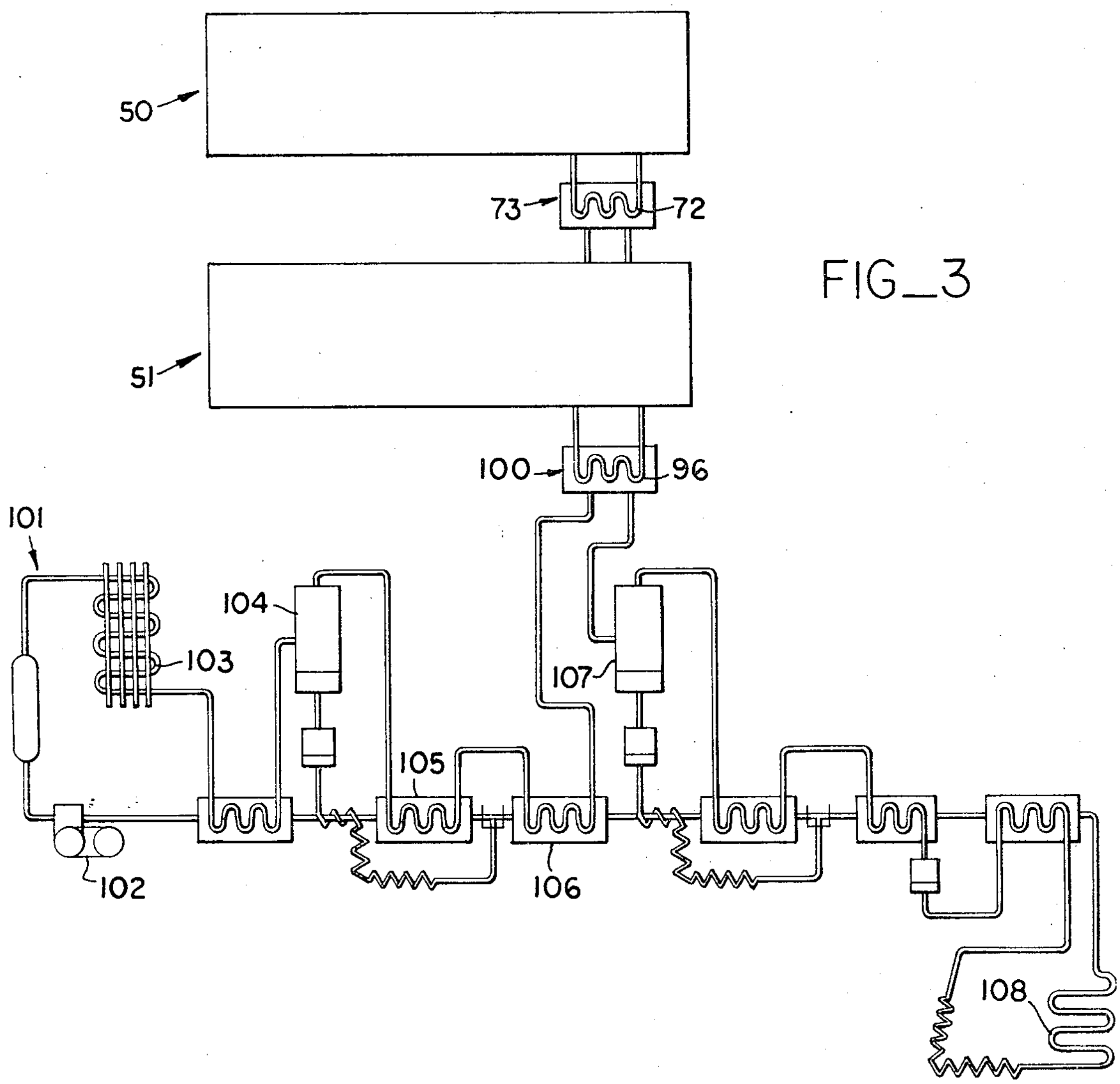

50
73
72
51
100
96
101
103
104
107
105
106
102
108

FIG_3

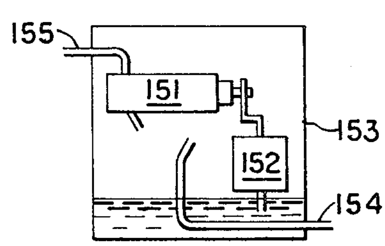

155
151
152
153
154

FIG_5

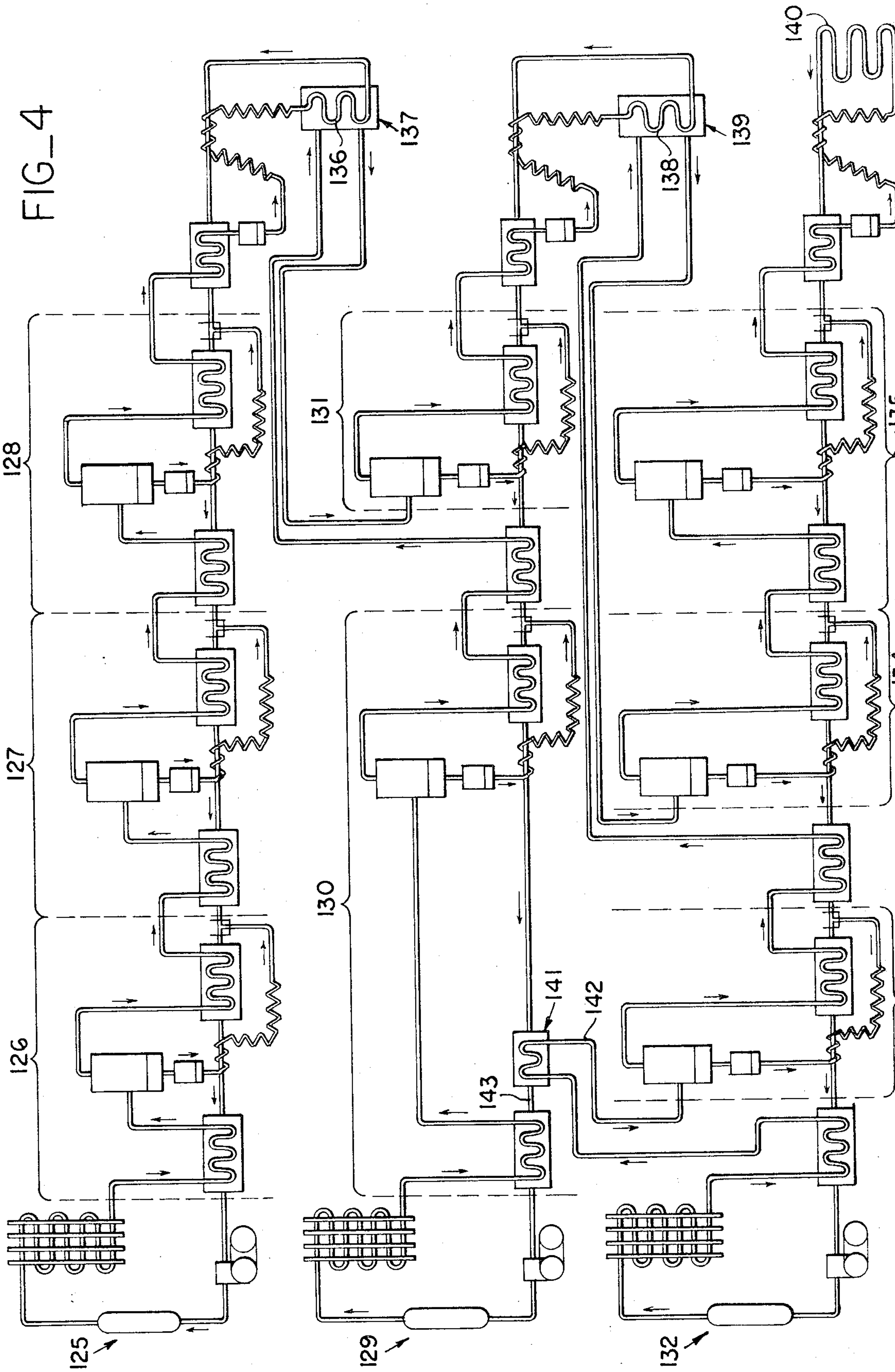

125
126
127
128
129
130
131
132
133
134
135
136
137
138
139
140
141
142
143

FIG_4

CASCADED MULTICIRCUIT, MULTIREFRIGERANT REFRIGERATION SYSTEM

BACKGROUND OF THE INVENTION

This invention relates to compression refrigeration systems. More particularly, the present invention is concerned with a novel system for achieving a wide range of extremely low refrigeration temperatures.

In a typical single refrigerant system of compression refrigeration, the refrigerant vapors are compressed, the vapors are condensed by heat exchange with ambient air or water, the condensate is cooled by expansion, the liquid refrigerant is evaporated at low pressure to produce refrigerating temperatures, with the refrigerant vapors being returned to the compressor so that the cycle may be repeated. When commercially available single-stage hermetic and semi-hermetic oil-lubricated compressors are employed in refrigeration systems of the above type, such systems have been limited to achieving temperatures on the order of −40° F. Where lower temperatures have been required, particularly temperatures approaching the cryogenic range (i.e., −250° F. and below), the simple refrigeration cycle has required substantial modification, including the use of high-pressure gas systems, expendable refrigerants or specially designed multi-stage compressors, turbo-compressors or high-pressure oilless compressors. Such systems are expensive to manufacture, operate and maintain and often require skilled personnel in constant attendance.

Relatively low refrigeration temperatures have been achieved by employing two or more closed refrigeration systems in conventional cascade connection wherein the final evaporator of one stage forms a heat exchanger with the initial condenser of the next lower stage. Such systems have not been effective in producing practical systems operating at or near cryogenic temperatures, particularly due to freezing problems with compressor lubricating oils in the low-temperature stage, unless the lowest stage included the use of high-pressure hazardous hydrocarbon refrigerants and special low-temperature compressors.

A further proposal for achieving low refrigeration temperatures involves the use of a mixture of refrigerants in a single refrigeration cycle. In such a system, a compressed mixture of refrigerant gases undergoes a partial condensation in a first condensation stage so that a liquid fraction which is rich in the higher boiling refrigerant is formed. The liquid fraction is separated from the remaining vapors and the vapors are transferred to a second condensation stage where they are condensed by the evaporation of the expanded and cooled liquid component from the first condensation stage. The final refrigeration temperature is achieved by expanding and evaporating the condensate from the second stage of condensation. The refrigeration cycle is closed by mixing the vapor leaving the final evaporator with the vapor formed by evaporation of the higher boiling refrigerant in the second condensation stage and returning the vapor mixture to the compressor.

A further proposal is that disclosed in my copending United States application Ser. No. 87,423, filed Nov. 6, 1970, now abandoned and entitled "Low-Temperature Refrigeration System," wherein a single, closed-cycle compression refrigeration system is shown which employs a mixture of non-flammable, non-explosive and non-toxic refrigerants having different boiling points, and which includes at least two vapor-liquid separation stages and at least two heat interchanges in which an expanded liquid refrigerant condensed in an earlier stage of the cycle is evaporated to cause further condensation of remaining refrigerant vapors.

It was discovered that by employing successive stages of vapor-liquid separation and utilizing the expanded liquid phase to provide cooling for the next successive stage of condensation, a refrigeration system which is capable of achieving a wide range of low temperatures, including temperatures in the cryogenic range, at unexpectedly low discharge pressures and compression ratios was achieved. This, unlike prior art refrigeration processes, made it possible to achieve extremely low refrigeration temperatures utilizing conventional single-stage hermetic or semi-hermetic compressors.

The refrigeration process described in my copending application is applicable to a wide combination of refrigerants, partial condensation stages and separation stages. Thus, for example, the process therein could be employed to achieve low temperatures at low discharge pressures and compression ratios with a binary mixture of refrigerants utilizing two or more vapor-liquid separation stages and an equal number of heat interchangers. Alternatively, refrigeration systems comprising a mixture of three or four refrigerants and two, three or more stages of vapor-liquid separation in a single, closed refrigeration cycle operated by a single compressor are possible. In general, the process contemplates compression refrigeration systems employing a mixture of any number of refrigerants and at least two distinct vapor-liquid separation stages. In the preferred embodiments the system comprises a mixture of N refrigerants and N-1 vapor-liquid separators and heat interchangers where N equals 3 or more refrigerants. However, the invention is equally applicable where the number of separators is equal to or exceeds the number of refrigerants in the mixture.

The ultimate low temperatures achieved by the refrigeration system of my previous system is governed by the boiling points of the respective refrigerants employed and is limited by the evaporation pressure of the lowest boiling refrigerant in the mixture. It was discovered that temperatures of less than −290° F. might be achieved by employing nitrogen as one of the refrigerants and the principles of the invention would be equally applicable to lower boiling refrigerants which would achieve even lower temperatures.

However, the multirefrigerant system described in my copending application has inherent pressure and temperature limitations which make it difficult to achieve very low, cryogenic temperatures by the use of conventional compressors. More particularly, in each stage, a refrigerant which has been condensed is allowed to evaporate by reducing the pressure thereon from the high pressure in the separator to the low pressure in the return line to the compressor. As such refrigerant evaporates, it absorbs heat from the gaseous refrigerant mixture leaving the separator, and reduces the temperature thereof to a level below the boiling point (at high pressure) of the next refrigerant. Thus, the pressure in the high-pressure line must be sufficiently high so that the boiling point of the next refrigerant is sufficiently low so that it will be liquefied by the evaporation of the first refrigerant. The greater the spread in boiling points (at low pressure) of the refrigerants, the higher must be the head pressure of the system. As a result, for a given number of refrigerants in the refrigerant mixture, higher and higher head pressures must be utilized to achieve lower and lower final temperatures. To achieve a desired low temperature, the number of refrigerants may be increased, in order to reduce the spread in boiling points between adjacent refrigerants. However, an increase in the number of refrigerants will increase the head pressure necessary to condense the added refrigerants.

Thus, in order to achieve cryogenic temperatures with a closed, single-loop multirefrigerant circuit, it is necessary to use head pressures that are not obtainable with conventional commercial oil-lubricated single-stage compressors, such compressors having a maximum working capacity of about 350 psi.

SUMMARY OF THE INVENTION

It is the principal object of the present invention to provide a multirefrigerant refrigeration system capable of achieving very low temperatures with no external cooling except conventional ambient-temperature air or water condensers, with relatively low head pressures, and with provision whereby any oil introduced into the refrigerant mixture will be prevented from freezing. Although the invention is not so restricted, it has the significant advantage that it will enable the system to operate at room temperatures using conventional commercially available hermetic or semi-hermetic oil-lubricated single-stage compressors.

In general, the objects of the invention are achieved by using a plurality of closed-circuit refrigeration circuits each having a compressor and an initial condenser cooled by ambient temperature. The second circuit is of the type described in my copending application Ser. No. 87,423, filed Nov. 6, 1970, now abandoned and entitled "Low-Temperature Refrigeration System" wherein one or more separator stages and two or more refrigerants are used. The final evaporator of the first circuit is in heat exchange relation with the high-pressure line of the next circuit downstream of the first separation therein. In this way, the oil separation function of the first separator in each circuit is unaffected by the other circuits and there is no danger of oil passing through the circuits to a point wherein the oil will freeze. With the cooling of one circuit by the preceding circuit, the spread between boiling points of the refrigerants in a circuit may be increased without a corresponding increase in head pressure. Increasing the spread between boiling points of the refrigerants will thus enable a lower final temperature to be achieved without increasing the number of refrigerants in a circuit.

The multicircuit system has a distinct advantage over a single circuit system designed to reach the same low temperature in that the circuits of the multicircuit system can operate at lower head pressures. Accordingly, the various heat exchangers throughout may be designed for lower-pressure operation, with thinner walls for more efficient heat exchange. Correspondingly, the other system components may be of more inexpensive construction.

Other objects and advantages will become apparent in the course of the following detailed description.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings accompanying this application,

FIG. 1 is a schematic representation of a two-circuit system wherein the high-temperature circuit is charged with a single refrigerant and the low-temperature circuit is charged with a mixture of two refrigerants and has one separator stage therein, there being one thermal interchange between the two circuits.

FIG. 2 is a schematic representation of a two-circuit system, each circuit having two separator stages and being charged with a mixture of three refrigerants, there being one thermal interchange between the circuits, the system being designed to operate at approximately −300° F.

FIG. 3 is a schematic illustration of a three-circuit system, each circuit having two separator stages and a mixture of three refrigerants, there being a single thermal interchange between successive circuits, the system being designed to operate at approximately −345° F.

FIG. 4 is a schematic illustration of a three-circuit system, the high-temperature circuit having three separator stages and four refrigerants, the intermediate-temperature circuit having two separate stages and three refrigerants and the low-temperature circuit having three separator stages and four refrigerants, there being one thermal interchange between the high- and intermediate-temperature circuits and two thermal interchanges between the intermediate- and low-temperature circuits, the system being designed to operate at approximately −440° F.

FIG. 5 is a schematic illustration of a typical single-stage oil-lubricated hermetic compressor.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. 1 illustrates a simplified embodiment of the present invention having a high-temperature closed circuit **10**, a low-temperature closed circuit **11**, and a single cascade heat exchanger **12** therebetween. This system will use a single refrigerant in the high-temperature circuit and a binary mixture in the low-temperature circuit. In operation, the refrigerant in the high-temperature circuit **10** is aspirated by compressor **13** and compressed and discharged through line **14** to condenser **15**. Condenser **15** may be cooled by air or water, as desired. The refrigerant condenses and flows through the high-pressure line **16** to heat exchanger **17**, wherein the fluid refrigerant is subcooled by returning cold refrigerant gas in the low-pressure suction line **18**. The subcooled liquid continues through high-pressure line **16** to expansion valve **19**. Such valve is illustrative of expansion devices such as thermal expansion valves, capillary tubes, etc. The pressure is reduced through the valve and the fluid temperature drops. This fluid flows through line **20** to the evaporating coil **21** of cascade heat exchanger **12** to condense the low-temperature fraction of the low-temperature circuit **11**. In so doing, the high-temperature circuit refrigerant absorbs thermal energy and in general will become completely evaporated to gas in the evaporating coil **21**. The gas proceeds through low-pressure line **18** to heat exchanger **17** where it subcools the liquid refrigerant and in turn becomes superheated. This gas then continues through line **18** back to the inlet of compressor **13** to complete the high-temperature-stage portion of the cycle.

In the low-temperature circuit, a mixture of two refrigerants is used, the refrigerants having different boiling points. The higher-boiling-point refrigerant is one in which lubricating oil is highly miscible. The binary refrigerant mixture in the low-temperature circuit 11 is aspirated by compressor 22, is compressed thereby and discharged through line 23 to condenser 24. Condenser 24 may be air- or water-cooled as required and condenses a portion of the refrigerant mixture to liquid. The liquid consists primarily of the high-boiling-point refrigerant of the binary mixture, entrained oil which is scrubbed out of the vapor by the condensing refrigerant, and a fraction of the low-boiling-point refrigerant of the binary mixture. The composition of the vapor-liquid mixture leaving the condenser 24 will vary with choice of refrigerants, pressure and condenser temperatures. The mixture goes through high-pressure line 25 to heat exchanger 26 wherein supercooling and some additional condensation may take place. The vapor-liquid mixture continues through high-pressure line 25 to separator 27. Here, by means of gravity and velocity changes, the vapor and liquid portions are separated. The liquid portion containing the oil flows from the liquid outlet of separator 27 through line 28 to expansion valve 29. The liquid mixture is reduced in pressure going through the valve and experiences a drop in temperature. The high-boiling-point refrigerant is chosen so that its temperature will not drop below the freezing point of the oil used in lubricating the compressor. Typically such oils will not freeze unless their temperature is reduced to about −175° F.

The vapor portion of the low-boiling-point refrigerant leaves the vapor outlet of separator 27 and continues through the high-pressure line 25 to the cascade heat exchanger 12. Here it is condensed into a liquid by thermal exchange with the high-temperature circuit refrigerant. The cooled liquid refrigerant in the low-temperature circuit leaves cascade heat exchanger 12, continues through high-pressure line 25 and enters heat exchanger 30 where it is subcooled by returning gases from the final evaporator coil 31. The subcooled liquid flows through high-pressure line 25 to expansion valve 32 (again representative of any applicable expansion device). In going through expansion valve 32 it is reduced in pressure and its temperature drops. The low-pressure fluid goes through line 33 to evaporator 31. Here it is evaporated and draws thermal energy from its surroundings, producing the desired refrigeration effect. The vapor flows back through low-pressure line 34 to heat exchanger 30 where it subcools the liquid and is itself superheated. This vapor then continues through low-pressure line 34 and is mixed with the fluid from line 35 entering the low-pressure return line 30. The mixture is used to subcool liquid in heat exchanger 26 and returns in vapor form to compressor 22 through the low-pressure line 34 to complete the cycle.

FIG. 2 shows an embodiment of the invention having two multirefrigerant circuits with their initial condensers cooled at ambient temperature and designed to produce a temperature in the final evaporator of approximately −300° F. The high-temperature circuit 50 is charged with a mixture of three refrigerants: R21 (dichloromonofluoromethane, also known as "Freon" 21, boiling point = −22° F.), R13B1 (bromotrifluoromethane, boiling point = −72° F.) and R14 (carbontetrafluoride, boiling point = −195° F.). The low-temperature circuit 51 is charged with a mixture of the following three refrigerants: R21, R14 and R728 (nitrogen, boiling point = −320° F.).

The system of FIG. 2 operates as follows. The refrigerant mixture in the high-temperature circuit 50 will be compressed, in a vapor state, by compressor 52, which may be a single-stage, hermetic or semi-hermetic, oil-lubricated compressor, with the compressed gas mixture being delivered to the receiver tank 53. This tank is utilized to provide storage of gas to prevent excessive refrigerant pressure during such time as the system is turned off and is not in operation, although other conventional techniques may be used to obtain this result. From the receiver tank 53 the compressed gas mixture flows to the inlet of condenser 54, which may be cooled by ambient air or water as desired. As the refrigerant mixture is cooled in the condenser 54, the highest-boiling-point refrigerant R21 condenses to liquid, with any oil in the mixture which may have been introduced into the mixture as it passed through the compressor 52 being scrubbed out of the mixture by the condensation of refrigerant R21. Most of the refrigerants R13B1 and R14 will remain in their vapor phase since they will be at a temperature considerably above their boiling points.

The mixture of liquid and vapor leaves condenser 54 and flows through the high-pressure line 55 to heat exchanger 56 wherein the mixture is further cooled by the refrigerant mixture returning through the low-pressure line 57. The cooled vapor-liquid mixture continues through high-pressure line 55 to the inlet of the vapor-liquid separator 58 wherein separation occurs between the liquid and vapor fractions of the mixture. The liquid fraction, containing the entrained oil which had been scrubbed during the condensing process, flows from the liquid outlet of the separator, through drier-strainer 59 to capillary tube 60. Here the pressure on the liquid fraction is reduced, thereby lowering the temperature thereof. If desired, a portion 61 of capillary tube 60 may be in heat-exchange relation with the low-pressure line 57 to provide additional cooling of the liquid fraction in the capillary tube by the returning refrigerant mixture. The cooled liquid fraction with lubricating oil mixed therein enters the low-pressure line 57 at junction 62 to return to the compressor. As the cooled liquid fraction passes through heat exchange 63 it will evaporate and absorb heat.

The vapor fraction in separator 58 will leave the vapor outlet of the separator and will flow through the high-pressure line 55 to heat exchanger 63. Here a fraction of the vapor is condensed to liquid by thermal interchange with the low-temperature vapor-liquid mixture coming into the heat exchanger 63 from the junction 62.

The vapor-liquid mixture in the high-pressure line then flows to and through auxiliary heat exchanger 64 to be further cooled therein by the returning refrigerant vapors in low-pressure line 57. From heat exchanger 64, the vapor-liquid mixture in the high-pressure line flows to separator 65 wherein separation of the vapor and liquid fractions again occurs. The liquid fraction is predominantly the middle-boiling-point refrigerant R13B1 but will contain some of the higher- and lower-boiling-point refrigerants R21 and R14.

The liquid fraction flows through drier-strainer 66 and capillary tube 67 to join the low-pressure line 57 at junction 68. As before, the pressure on the liquid fraction will be reduced and the temperature thereof lowered, and the liquid fraction will then evaporate in heat exchanger 69.

The vapor fraction in separator **65** is predominantly the lowest-boiling-point refrigerant R14 with some refrigerant R13B1 and a very small amount of refrigerant R21 mixed therewith. The vapor fraction leaves the vapor outlet of separator **65** and flows through high-pressure line **55** to heat exchanger **69** where the vapor fraction is cooled and condensed to liquid. The liquid refrigerant then flows through drier-strainer **70** and capillary tube **71** wherein the pressure is reduced, thus lowering the temperature of the liquefied refrigerant. The refrigerant then flows through the evaporating coil **72** of cascade heat exchanger **73** wherein it evaporates and absorbs heat from the low-temperature circuit **51.** The resulting vapor then flows through the low-pressure line **57** back to the compressor **52,** mixing with other fractions from junctions **68** and **62** and rising in temperature as it passes through heat exchangers **69, 64, 63** and **56.**

In the low-temperature circuit **51,** the compressor **74** compresses the refrigerant vapors coming from the low-pressure return line **75** and delivers the compressed vapors through storage receiver **76** to condenser **77** wherein the highest-boiling-point refrigerant R21 condenses to liquid, scrubbing out any oil from the compressor **74** which may have been entrained with the refrigerant vapors. Again, the compressor is preferably of the type as compressor **52** used in the high-temperature circuit **50,** and the condenser **77** may be cooled by ambient air or water.

The vapor-liquid refrigerant flows through the high-pressure line **78** and heat exchanger **79** to separator **80.** As before, the liquid fraction, containing the scrubbed oil, exits the liquid outlet of the separator, passes through the drier-strainer **81** and the capillary tube **82** wherein the pressure and temperature of the liquid fraction is reduced, the liquid fraction joining the low-pressure return line **75** at junction **83** and then passing through heat exchanger **84** wherein the liquid fraction evaporates and absorbs heat. The vapor fraction leaving separator **80** is predominantly the R14 and R728 refrigerants with a small amount of R21 mixed therein, and this vapor fraction is cooled as it flows through heat exchanger **84.** It is not necessary to operate the low-temperature circuit at a head pressure wherein the boiling point of the refrigerant mixture in the high-pressure line is raised to a point wherein condensation of the mixture will occur in heat exchanger **84,** as was the case in heat exchanger **63** of the high-temperature circuit. Thus refrigerants may be used in the low-temperature circuit which have a considerably greater difference in boiling points without the attendant need for extremely high-pressure operation.

The heat exchanger **84** will condense any of the R21 vapors still in the mixture, and, if desired, the heat exchanger **84** may be operated as a reflux condenser by opening valve **85** to allow condensed liquid to flow back through line **86** to separator **80** and mix with the liquid fraction therein. Such operation would further ensure that all oil is removed from the refrigerant mixture before such mixture is reduced in temperature to a point wherein the oil could freeze. In this regard, it is important that the high-boiling-point refrigerant be chosen so that it will not produce low temperatures such that the oil entrained therewith can freeze.

The cooled vapor mixture leaving heat exchanger **84** flows through heat exchanger **87** wherein it is further cooled and then through the high-pressure line **78** to

cascade heat exchanger **73.** In this heat exchanger the evaporation in coil **72,** at low pressure, of the refrigerant R14 in the high-temperature circuit will condense a fraction of the higher-pressure refrigerant R14 in the low-temperature circuit, the condensed fraction being predominantly the refrigerant R14 in the mixture.

The vapor-liquid refrigerant mixture will then flow from cascade heat exchanger **73** to separator **88** wherein separation of the liquid and vapor fractions again takes place. The liquid fractions, predominantly R14, passes through drier-strainer **89** and capillary tube **90** to junction **91** in the low-pressure return line **75** and then through heat exchanger **92** wherein the liquid fraction evaporates. The vapor fraction, predominantly R728, passes from separator **80,** and flows through the high-pressure line **78** to heat exchanger **92** wherein the vapor condenses to liquid. The head pressure, of course, in the high-pressure line must be sufficiently high so that the boiling point of the vapor fraction is sufficiently high that the vapor fraction will be condensed in the heat exchanger **92.**

The liquid refrigerant then flows through heat exchanger **93** for further cooling by the returning refrigerant and then flows through drier-strainer **94** and capillary tube **95** wherein the pressure and temperature are reduced. The low-pressure liquid is then evaporated in the final evaporator coil **96** to absorb thermal energy from the surroundings and perform useful refrigeration. The stabilized temperature at this point will be approximately −300° F.

As before, the refrigerant vapor from evaporator coil **96** will flow back through the low-pressure return line **75,** mixing with the other fractions and rising in temperature as the refrigerant passes through the various heat exchangers until the entire mixture reaches the compressor inlet to complete the cycle.

FIG. **3** shows yet another embodiment of the invention, wherein all initial condensers are cooled by ambient water or air and where conventional oil-lubricated compressors may be used to achieve a final temperature of approximately −345° F.

The FIG. **3** system utilizes three closed multirefrigerant circuits, cascaded together in accordance with the invention. If desired, the two circuits of FIG. **2,** i.e., **50** and **51,** may be used as the low- and intermediate-temperature circuits, and with the evaporator coil **96** of circuit **51** being employed in the cascade heat exchanger **100** between the intermediate-temperature circuit and the low-temperature circuit **101.** Preferably, the low-temperature circuit **101** is charged with a refrigerant mixture of R21 (Freon), R728 (nitrogen) and R720 (neon). The entire system would thus be charged with refrigerant mixtures as follows (the boiling points being at atmospheric pressure):

High Temperature Circuit 50		Intermediate Temperature Circuit 51		Low Temperature Circuit 101	
Refrigerants	Boiling Point	Refrigerants	Boiling Point	Refrigerants	Boiling Point
R21	−22° F.	R21	−22° F.	R21	−22° F.
R13B1	−72° F.	R14	−195° F.	R728	−320° F.
R14	−195° F.	R728	−320° F.	R720	−406° F.

The low- and intermediate-temperature circuits will operate as previously described, resulting in a temperature in the evaporating coil **96** of approximately −300° F., produced by the evaporation of a refrigerant fraction which is predominantly R728 (nitrogen).

The low-temperature circuit **101** operates in the same manner as previously described in connection with circuit **51.** Any oil in the refrigerant mixture, introduced from compressor **102,** will be scrubbed from the mixture as the highest-boiling-point refrigerant condenses in condenser **103** and separated out with the liquid fraction in separator **104.** The nitrogen and neon vapor fraction from separator **104** will be cooled but not condensed in heat exchangers **105** and **106.** As the vapor fraction passes through the cascade heat exchanger **100,** a fraction (predominantly nitrogen) will be condensed by the evaporating nitrogen in evaporator coil **96.** The vapor-liquid mixture will be separated in separator **107,** with the liquid fraction being used to condense the vapor fraction (predominantly neon). This fraction then has its pressure and temperature lowered and it then evaporates in the final evaporator coil **108** to perform useful refrigeration at approximately −345° F.

FIG. **4** illustrates yet another embodiment of the invention, utilizing three closed multirefrigerant circuits and designed to produce a final temperature of approximately −440° F. resulting from the evaporation of a refrigerant fraction which is predominantly R704 (helium).

The high-temperature circuit **125** has three separator stages **126, 127** and **128,** the intermediate-temperature circuit **129** has two separate stages **130** and **131,** and the low-temperature circuit **132** has three separator stages **133, 134** and **135.** As is apparent from the preceding description, a separator stage includes a vapor-liquid separator having its inlet and vapor outlet connected in the high-pressure line and its liquid outlet connected to the low-pressure line, the stage operating so that the liquid fraction from the separator is reduced in pressure and temperature and evaporated to cool the vapor fraction from the separator.

The system of FIG. **4** is charged with the following refrigerant mixtures:

High Temperature Circuit 125 Refrigerants	Boiling Point	Intermediate Temperature Circuit 129 Refrigerants	Boiling Point	Low Pressure Circuit 132 Refrigerants	Boiling Point
R21	−22° F.	R 21	−22° F.	R21	−22° F.
R13B1	−72° F.	R728	−320° F.	R728	−320° F.
R14	−195° F.	R720	−406° F.	R720	−406° F.
R728	−320° F.			R704	−453° F.

In the high-temperature circuit **125,** the various refrigerants will be progressively condensed, resulting in a refrigerant fraction which is predominantly liquid R728 (nitrogen) at the end of separator stage **128.** This refrigerant is then evaporated at low pressure in evaporator coil **136** in cascade heat exchanger **137.**

In the intermediate-temperature circuit **129,** refrigerant R21 with any entrained oil therein will be separated out and returned to the compressor in separator stage **130.** The vapor fraction — nitrogen and neon — will pass through cascade heat exchanger **137** and a fraction thereof, predominantly nitrogen, will be condensed. The condensed liquid fraction is separated out and used to condense the remaining refrigerant which is predominantly neon. This refrigerant is then evaporated at low pressure in evaporator coil **138** of cascade heat exchanger **139.**

In the low-temperature circuit **132,** refrigerant **R21** is again liquefied to scrub out any entrained compressor oil and is separated and returned to the compressor in separator stage **133.** The vapor fraction from this state, a mixture of nitrogen, neon and helium vapors is passed through cascade heat exchanger **139** to liquefy the nitrogen and some of the neon and to cool the liquid-vapor mixture to a very low temperature. The liquid, predominantly nitrogen, separates out in separation stage **134** and is used to condense essentially all of the neon, which is then separated out in separator stage **135** and used to condense the remaining refrigerant, which is neon and helium. This refrigerant is then reduced in pressure and temperature and evaporated in the final evaporating coil **140.** If it is desirable to operate above the helium liquefaction point the final high pressure circuit may use a Joule Thomson expansion device rather than a liquid expansion valve.

FIG. **4** also shows the use of an intercircuit heat exchanger **141** connected to exchange heat between the high-pressure line of one circuit, in this case high-pressure line **142** of low-temperature circuit **132,** and the low-pressure return line of another circuit, in this case low-pressure line **143** of intermediate circuit **129.** Such heat interchange may be employed to balance compressor operations and increase the efficiency of the system. For example, if, without such heat exchanger, the compressor in the low-pressure circuit **132** were operating at a higher pressure than the compressor in circuit **132** were operating at a higher pressure than the compressor in circuit **129,** then the heat exchanger **141** could be employed so that the refrigerant mixture in the low-pressure line **143** of circuit **129** would absorb heat from, and thereby cool, the refrigerant mixture in the high-pressure line **142** of circuit **132.** To compensate for the additional heat being absorbed, the compressor in circuit **129** would have to be operated at a higher pressure, but, with the additional cooling provided, the compressor in circuit **132** could then be run at a lower pressure. Correspondingly, if the head pressure in circuit **129** were higher than that in circuit **132,** an intercircuit heat exchanger could be employed whereby the refrigerant mixture in the low-pressure line of circuit **132** cools the refrigerant mixture in the high-pressure line of circuit **129.** Similarly, such heat interchange can be provided, if required, between circuits **132** and **125** or between **129** and **125.** The exact location in the circuits of such heat interchange is not critical except that is must be located at such point wherein the loss or gain of heat by the refrigerants passing therethrough will not be sufficient to interrupt normal operation of the circuits. Also, such heat interchange must not be located such that if a refrigerant mixture having entrained oil passes therethrough, such mixture will not be lowered in temperature to a point whereby the oil will freeze.

It is to be understood that the foregoing recitals of specific refrigerants in connection with the circuits of FIGS. **2, 3** and **4** is for the purpose of illustration. Other refrigerants may of course be used and a wide variety of combinations will readily occur to those persons skilled in the art by inspection of standard refrigerant tables and charts.

The relative amounts of each refrigerant in a system is not critical. However, changing relative quantities will affect final temperatures. Sufficient amounts of each refrigerant, of course, must be present to ensure an adequate flow of liquid from each stage of the process when the system is in full operation. The optimum weight ratio of refrigerants in any particular circuit will depend upon their respective molecular weight, which influences their individual partial pressures, their liquid densities and the amount of liquid required at each stage of separation. In general, the amount of lowest-boiling-point refrigerant in a circuit should be maintained at the minimum necessary to achieve the required refrigeration effect of the circuit, since greater amounts of the lower-boiling-point refrigerant will tend to increase the necessary head pressure of the circuit.

As set forth in the foregoing specification, although the present invention is not so limited, it has a significant advantage in that it enables standard single-stage oil-lubricated hermetic compressors to be used in achieving very low temperatures. FIG. **5** illustrates schematically a typical compressor of this type. A piston compressor unit **151** and the actuating motor **152** are both disposed within a hermetically sealed housing **153.** The incoming refrigerant vapors from the return line of the refrigeration circuit enter the housing through inlet **154** and flow across the exterior of the motor and compressor unit to cool them before the vapors enter the compressor unit. The vapors are then compressed within the unit and leave through the housing outlet **155** to the high-pressure line of the refrigeration circuit. Oil from the sump formed by the lower part of the housing is drawn up into the motor to lubricate the motor, the oil also being used to lubricate the piston in the compression unit. Such compressors provide trouble-free operation for years since the oil is not exposed to air and cannot leak from the housing. Similarly, there is no problem of refrigerant leakage from the unit since there are no seals required in the compression unit to seal against escape of refrigerant to atmosphere as would be necessary in a non-hermetic unit. The contact of refrigerant vapor and oil in the compressor will cause some entrainment of oil in the refrigerant vapor. A semi-hermetic compressor operates in the same manner and differs only in that portions of the housing are removable in the event repair of the internal parts is required. Compressors of the type described may be used for the compressors **13** and **22** of FIG. **1,** compressors **52** and **74** of FIG. **2,** compressor **102** of FIG. **3,** and the compressors of FIG. **4.**

Having thus described my invention, I claim:

1. A multicircuit, multirefrigerant refrigeration system comprising:

a. first and second closed refrigerant circuits, each circuit having
  i. a compressor,
  ii. a condensor connected to receive refrigerant vapors under pressure from said compressor,
  iii. an evaporator,
  iv. a high-pressure line connecting from the condenser outlet to the evaporator inlet,
  v. a low-pressure line connecting from the evaporator outlet to the compressor inlet,
b. a vapor-liquid separator in said second circuit having its inlet and vapor outlet connected in the high-pressure line and its liquid outlet connected to the low-pressure line,
c. said second circuit being charged with a mixture of at least two refrigerants having different boiling points,
d. means forming a heat exchanger between the evaporator of said first circuit and the high-pressure line of said second circuit at a point therein downstream of said separator.

2. A cascaded, multicircuit, multirefrigerant refrigeration system comprising:

a. a plurality of closed refrigerant circuits, each circuit having
  i. a compressor,
  ii. a condenser connected to receive refrigerant vapors under pressure from said compressor,
  iii. an evaporator,
  iv. a high-pressure line connecting from the condenser outlet to the evaporator inlet,
  v. a low-pressure line connecting from the evaporator outlet to the compressor inlet,
  vi. a plurality of separator stages successively connected between said condenser and evaporator, each stage having a vapor-liquid separator having its inlet and vapor outlet connected in said high-pressure line and means connecting between the fluid outlet of said separator and said low-pressure line to expand liquid refrigerant and to exchange heat between the high-pressure line downstream of said separator and the expanded refrigerant,
b. means forming a heat exchanger between the evaporator of a circuit and the high-pressure line of the next successive circuit at a point therein downstream of the first vapor-liquid separator therein,
c. each circuit having a mixture of refrigerants therein, said refrigerants in circuit being one more in number than the number of separator stages therein and having different boiling points at the same pressure,
d. i. the highest-boiling-point refrigerant in each circuit having a boiling point sufficiently high to enable said refrigerant to condense prior to delivery to the first vapor-liquid separator in said circuit,
  ii. the difference between the highest and lowest boiling point of the refrigerants in each circuit being progressively greater for each circuit,
  iii. the boiling point of the lowest-boiling-point refrigerant being progressively lower for each circuit.

3. A refrigeration system as set forth in claim **2** and further including means forming a heat exchange between the low-pressure line of one circuit and the high-pressure line of another circuit.

4. A refrigeration system as set forth in claim **2** wherein the lowest-boiling-point refrigerant of the last circuit is selected from the group consisting of neon, hydrogen and helium.

5. A refrigeration system as set forth in claim **2** wherein all of the compressors are of the hermetic oil-lubricated type.

6. A refrigeration system as set forth in claim **5** wherein the highest-boiling-point refrigerant in each circuit is a halocarbon.

7. A refrigeration system as set forth in claim **5** wherein the lowest-boiling-point refrigerant of the last circuit is selected from the group consisting of neon, hydrogen and helium.

8. A refrigeration system as set forth in claim **7** wherein the highest-boiling-point refrigerant in each circuit is a halocarbon.

9. A refrigeration system as set forth in claim **5** and further including means for exchanging heat between the low-pressure line of one circuit and the high-pressure line of another circuit.

10. A refrigeration system as set forth in claim 9 wherein said means for exchanging heat is located in the high-pressure line of the next successive circuit, upstream of said first separator therein and is located in said low-pressure line at a point therein where the temperature in said low-pressure line is above the freezing point of the oil used to lubricate the compressor in said next successive circuit.

* * * * *