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Missimer

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[54] **SELF-BALANCING LOW TEMPERATURE REFRIGERATION SYSTEM**

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[52] **U.S. Cl.** **62/84,** 62/114, 62/470, 62/473, 62/502

[51] **Int. Cl.** **F25b 1/00**

[58] **Field of Search** 62/84, 114, 468, 62/470, 502

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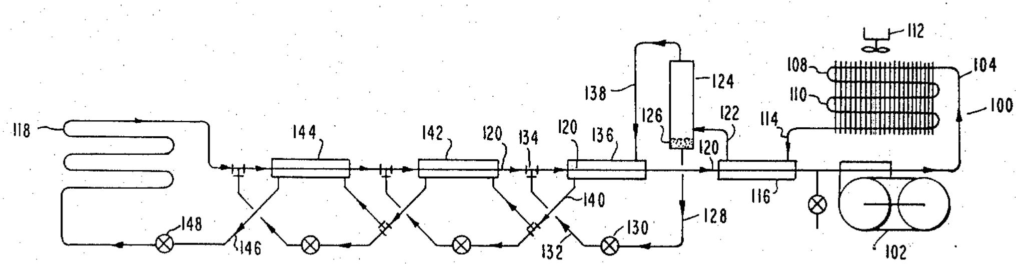
[57] **ABSTRACT**

Extremely low temperatures, in the range of −40°F. to −300°F., are achieved in a single circuit compression refrigeration system which is capable of being operated by a conventional compressor without the aid of special start-up or stand-by equipment. The system relies upon a series of intermediate cooling stages in which each stage includes the steps of withdrawing a portion of the liquid condensate from a compressed vapor-liquid refrigerant mixture which enters that stage, throttling the withdrawn condensate to a lower pressure, mixing the throttled condensate with refrigerant being recycled to the compressor from the final evaporator and evaporating the throttled condensate to absorb heat from and at least partially condense the compressed unconcensed vapor in the compressed mixture.

14 Claims, 3 Drawing Figures



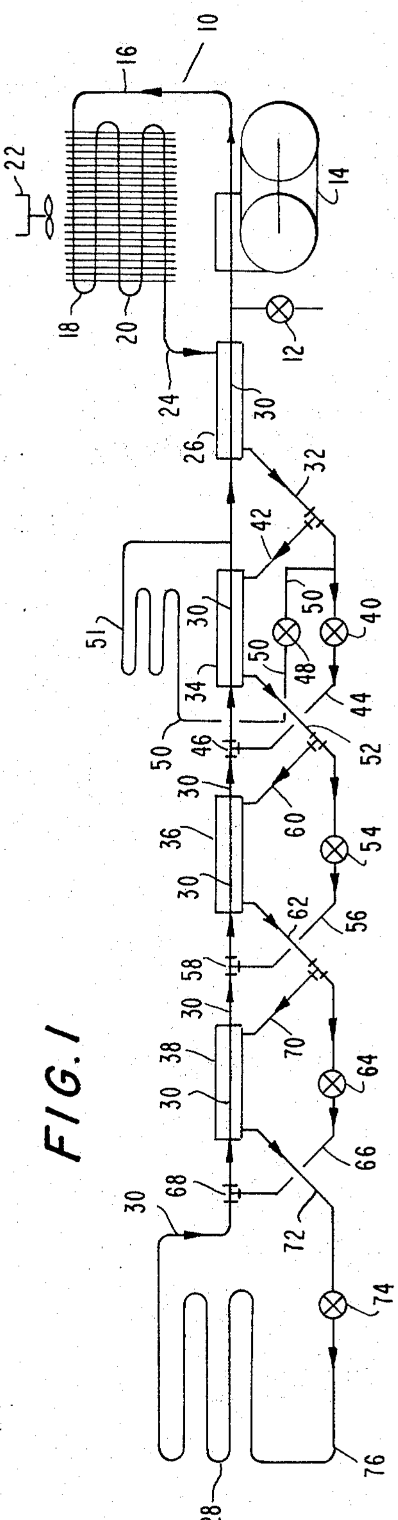


FIG. 1

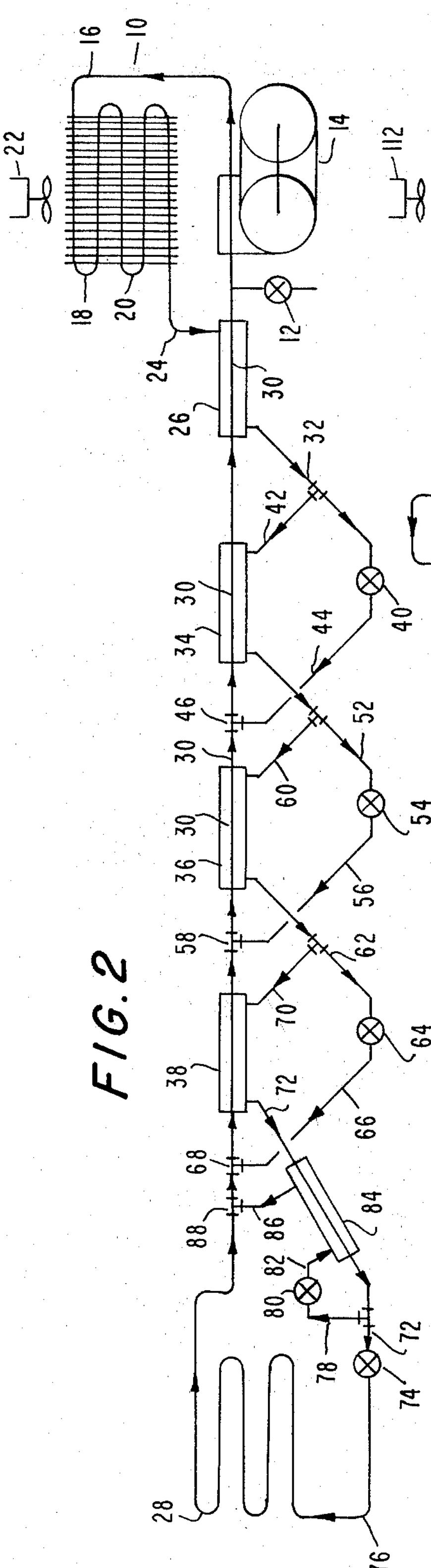


FIG. 2

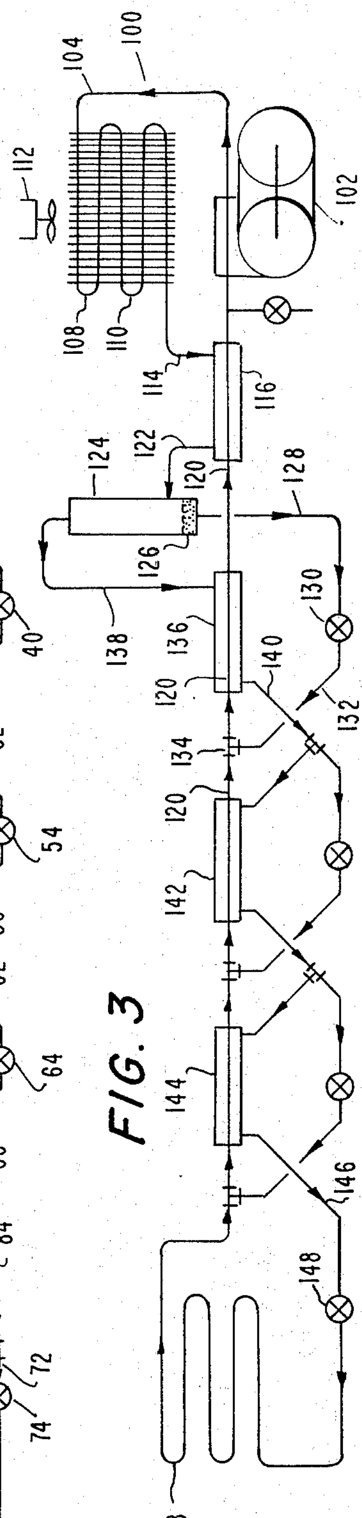


FIG. 3

SELF-BALANCING LOW TEMPERATURE REFRIGERATION SYSTEM

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 employing a mixture of refrigerants.

In the known systems of compression refrigeration, refrigerant vapors are compressed, the vapors are condensed by heat exchange with ambient air or water, the condensate is throttled to a low pressure and evaporated to produce the refrigerating effect, and the refrigerant vapors are recycled to the compressor. When commercially available single stage air conditioning or refrigeration 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, the simple refrigeration system has required substantial modification including the use of high pressure gas systems, expendable cryogenic refrigerants, specially designed multi-stage compressors, or high pressure oilless compressors. Such systems are expensive to manufacture and operate and frequently require skilled personnel in constant attendance.

Relatively low refrigeration temperatures have been achieved by employing two or more complete refrigeration circuits in cascade connection. In such systems the evaporator of one stage forms a heat exchanger with the condenser of the next lower stage. Such systems have the disadvantage of both size and expense in that each stage of the cascade includes all of the components of a complete refrigeration system. Moreover, such systems have not been effective in producing practical systems due to freezing problems caused by circulation of compressor lubricating oils within the system unless the lower stage included the use of high pressure hazardous hydrocarbon refrigerants, special lubricants, a highly efficient oil separation system or other components of a conventional cryogenic system.

A further method of achieving low refrigeration temperatures involves the use of a mixture of refrigerants in a single refrigeration circuit. publications and plublications describing such refrigeration techniques, in both opened and closed circuit systems, include U.S. Pat. No. 3,203,194, issued Aug. 31, 1965 to A. Fuederer; U.S. Pat. No. 3,218,816, issued Nov., 1965 to Grenier; U.S. Pat. No. 2,041,725 issued May, 1936 to Podbielniak; U.S. Pat. No. 3,487,653, issued Jan., 1970 to Myre; U.S. Pat. No. 2,952,139, issued Sept. 13, 1960 to Kennedy, et al. and A.P. Kleemenko, "One Flow Cascade System," *Progress in Ref. Science & Technology*, Permagon Press (1960). In such systems, low temperatures are achieved by a series of intermediate cooling stages in which the refrigerant is partially condensed, the condensate is separated into vapor and liquid phases, the separated liquid phase is throttled, and the throttled liquid is evaporated in heat exchange relationship with the separated compressed vapors in order to achieve further cooling and further partial condensation. While such systems are capable of achieving very low final refrigeration temperatures at relatively low pressures and compression ratios, closed circuit systems employing this concept are not commercially practical unless they are equipped with a rather complicated series of expansion tanks, control valves or other

devices which will enable such systems to operate properly under all conditions. Specifically, it has been found that when these systems operate at higher temperatures, for example during start-up from room temperature or after power off and stand-by conditions, the refrigerants employed exert substantially higher vapor pressures and there is an increased likelihood of reaching compressor discharge pressures and compression ratios which would damage the conventional compression equipment which such systems are designed to utilize. This difficulty is not readily correctable by varying the amount of each refrigerant charged to the system since the amounts of refrigerant which would be desirable to avoid excessive start-up pressures are not the optimum amounts once the system begins to cool to its designed operating temperature level, i.e. the refrigerant charge is quite critical. In addition, because such systems employ throttling devices having a constant or limited maximum mass flow for a given pressure and refrigerant quality or vapor content as well as vapor-liquid separators at each intermediate stage of condensation, the amount of condensate formed at each stage tends to be critical and may adversely affect the proper operation of the system either due to liquid hold-up or insufficient liquid flow. Indeed, such systems have exhibited a tendency to undergo "self-refrigeration," i.e. the lowest system temperature is reached at one of the intermediate cascade condensers rather than at the final evaporator, because liquid condensate formed up stream at a partial condensation step when throttled and evaporated reduces the discharge pressure to a level so low that an inadequate amount of liquid condensate is formed in the final step and thus, an insufficient amount of liquid condensate is available for feeding the throttling device which in turn feeds the final evaporator. Although this situation can be corrected by the addition of more of the lowest boiling point refrigerant the addition of such additional refrigerant will upset the balance required both for start-up and for final operating design temperatures.

It is an object of the present invention to provide a novel improved refrigeration system for achieving a broad range of low temperatures.

It is another object of the present invention to provide a refrigeration system capable of achieving temperatures approaching the cryogenic range in a single refrigeration circuit.

It is another object of the present invention to provide a novel, self-balancing refrigeration system employing a mixture of refrigerants which does not require the use of vapor-liquid separators and which eliminates any criticality as to the amount of each refrigerant charged to the system or the amount of liquid condensate formed at any intermediate point in the system.

A further object of the invention is to provide a novel refrigeration system capable of achieving extremely low temperatures at relatively low discharge pressures and compression ratios such that conventional mass-produced air conditioning type compressors may be employed.

A further object of the invention is to provide a sealed self-balancing compression refrigeration system employing multiple refrigerants and intermediate cooling stages containing a full refrigerant charge which is capable of rapidly achieving low temperatures without the use of control systems, expansion tanks or other devices designed to alleviate problems associated with high start-up pressures and compression ratios.

The above and other objects of the invention are accomplished in a novel, single cycle compression refrigeration system which employs a mixture of non-flammable, non-explosive and relatively non-toxic refrigerants having different boiling points and which includes at least one, but preferably more than one, intermediate cooling stage in which a compressed mixture of the refrigerants is at least partially condensed to form a mixture consisting of compressed vapors and compressed liquid condensate, a portion of the liquid condensate is withdrawn to feed a throttling device and the remainder of the liquid condensate is permitted to flow downstream with the uncondensed vapor. Further cooling and at least partial further condensation is accomplished in each intermediate cooling stage by evaporating the throttled liquid condensate to absorb heat from and at least partially condensed the remaining vapor in the mixture of compressed refrigerant which includes both vapor and the liquid condensate which was not fed through the throttling device at that stage. This novel system may also include sub-cooling and oil separation apparatus as will be more fully described herein.

It has now been discovered that by eliminating the successive stages of vapor-liquid separation which have characterized prior compression refrigeration processes utilizing mixtures of refrigerants to achieve low temperatures, the criticality of the amount of each refrigerant charged to a sealed system is eliminated and the system becomes inherently self-balancing, i.e. step wise partial condensation of the refrigerant mixture is achieved by a proper flow of liquid condensate to each throttling device, the remainder passing downstream without the liquid hold-up which has characterized systems which employ separators. Moreover, the system of the invention is capable of achieving rapid cooling from start-up or stand-by conditions without excessive discharge pressures and compression ratios and without the use of bypass control devices or expansion tanks to limit the amount of refrigerant in the system so as to avoid such pressures. While not wishing to be limited to any particular theory of operation, it is presently believed that the elimination of vapor-liquid separators allows the novel refrigeration system of the invention to be inherently self-balancing, i.e. to provide for the appropriate amount and flow of liquid condensate to each throttling device in the system and to vary the amount of that flow as dictated by changes in the operating conditions of the system, albeit that a fixed amount of each refrigerant is initially charged into the system. This approach also appears to improve the thermodynamic efficiency of the system due to the fact that a more nearly equilibrium condition between the vapor and liquid phases is achieved.

Irrespective of the theory of operation, it has been found that by withdrawing 10 to 95 percent of the liquid condensate formed in each intermediate cooling stage rather than making a complete separation of the vapor and liquid phases the outstanding results heretofore described are achieved. The exact amount of condensate withdrawn through each throttling device is not critical and will, of course, vary with the design and operating conditions of the system including the nature and amount of the refrigerants being employed, the type of throttling device, the operating temperature and similar parameters, the control of which are within the skill of the art.

The novel refrigeration system of the invention is not limited to use with particular refrigerant combinations and a wide variety of refrigerant combinations may be employed to achieve operation over a wide temperature range, e.g. −40°F. to −300°F. The refrigerants in any particular system will range from high boiling point refrigerants of the type normally employed in conventional centrifugal type air conditioning systems to extremely low boiling point refrigerants such as nitrogen, argon, neon, helium and the like. Preferably, the higher boiling refrigerants will be selected from the group consisting of well-known halogenated hydrocarbons and their azeotropic mixtures. Hydrocarbon refrigerants may also be used provided adequate safety precautions are employed.

The boiling points of the individual refrigerants in the mixture must be sufficiently far apart to permit step wise condensation at each intermediate cooling stage of the system. Typically, each refrigerant in the mixture will differ in boiling point from the next closest boiling refrigerant by 50°F. to 180°F. The differences in boiling point between adjacent refrigerants may vary widely within these ranges but, in general, may be smaller in those instances where a large number of intermediate cooling stages are employed.

The relative amount and type of each refrigerant in the system is not critical and ordinarily sufficient amounts of each refrigerant will be present to insure an adequate flow of liquid at each stage of the process when the system is in full operation. Among the considerations as to the type and amount of refrigerant charged to the system, as will be apparent to those persons skilled in the art, are the design-operating temperature and pressures of the system; the nature of the condensing media; the size of various heat exchangers and throttling devices in the system; the compressor displacement and the nature of the refrigerants being employed. The optimum weight ratio of refrigerants in any particular system will also depend upon their respective molecular weights which influence their individual partial pressures; their liquid densities; and the amount of liquid required at each intermediate cooling stage. In general, the amount of lowest boiling refrigerant will be maintained at the minimum necessary to achieve the required refrigeration effect of the system since higher amounts of the lower boiling refrigerants tend to increase the discharge pressures of the system. In one typical system designed to operate at refrigerating temperatures as low as −230°F., 22.5 mol percent trichlorofluoromethane (R–11), 29.8 mol percent dichlorodifluoromethane (R–12), 15.0 mol percent chlorotrifluoromethane (R–13), 16.3 mol percent carbontetrafluoride (R–14) and 16.4 mol percent argon (R–740) were employed. The method by which the individual refrigerants are charged to the system is not critical and charging can be based upon volume, weight or increase in partial pressure. Ordinarily, each refrigerant component is added to the system separately in order of decreasing boiling point.

The system of the invention will be further understood by reference to the accompanying drawings wherein:

FIG. 1 is a schematic representation of a refrigeration system having three intermediate cooling stages;

FIG. 2 is a schematic representation of a refrigeration system similar to that shown in FIG. 1 modified to include apparatus for additional sub-cooling prior to the final evaporator; and

FIG. 3 is a schematic representation of a refrigeration system similar to that shown in FIG. 1 modified to include apparatus and a technique for removing compressor lubricating oils from the system.

Referring specifically to FIG. 1, a mixture of two or more refrigerants having different boiling points is charged into a single closed refrigeration circuit generally identified as **10** through a service valve **12** or other conventional charging means such as a tube, pipe or the like, which will be sealed after the charging step. The amount of each refrigerant charged to the system may be predetermined by volume or weight or, in the case of lower boiling point refrigerants, by allowing each refrigerant gas to circulate through the system until a predetermined partial pressure and a predetermined total pressure for the system are reached.

Subsequent to the charging step, the vapors are aspirated by a compressor **14** and passed through conduit **16** to condenser **18** where partial condensation occurs. Condensation occurs by heat exchange with ambient air forced over condenser pipes **20** by a fan **22** or, alternatively, condensation may be carried out using a readily available source of water.

The partially condensed refrigerant mixture flows through conduit **24** to an auxiliary condenser **26** where, after the system is in operation, further condensation may occur by heat exchange with the cooler vapors returning to compressor **14** from the final evaporator **28** through conduit **30**. Utilization of an auxiliary condenser is not critical to the refrigeration system but such additional heat exchange at this point serves to improve the thermodynamic efficiency of the system.

The partially condensed refrigerant mixture leaves auxiliary condenser **26** through conduit **32** and passes to the first of a series of successive intermediate cooling stages. The compressed mixture at this point comprises a liquid which is rich in the higher boiling refrigerant or refrigerants and a vapor which is rich in the lower boiling refrigerant or refrigerants of the mixture. However, each fraction will contain at least minor amounts of each of the refrigerants in the mixture. FIG. 1 illustrates a refrigeration system including three intermediate cooling stages each consisting of a cascade condenser **34, 36** or **38,** a throttling device **40, 54** or **64** and associated conduits. The number of intermediate cooling stages in any system is not critical, provided at least one such stage is present, and the selection of the ultimate number of stages, for example two to six stages, may be readily determined by those persons of ordinary skill in the art depending upon the operating load and other conditions for which the system is designed.

A portion of the compressed liquid condensate in conduit **32** is throttled in throttling device **40**. Such throttling devices are well known in the art and may consist of a capillary tube, a thermal expansion valve, a float valve or a similar device which permits the pressure on the liquid flowing therethrough to be dropped from the discharge pressure of the system to the suction pressure of the system. Since the mass flow of liquid through a throttling device is, inter alia, a function of the inlet pressure to the throttling device, it will be apparent to those skilled in the art that throttling device **40** as well as the other throttling devices in the refrigeration system **10** will not be capable of handling the full flow of liquid condensate under all of the variety of operating conditions which may be encountered during operation of the refrigeration system. Accordingly, throttling device **40** is not designed to permit the flow of all of the liquid condensate in conduit **32** therethrough. A portion of the compressed condensate, as well as the compressed vapors formed in auxiliary condenser **26** flows through conduit **42** to cascade condenser **34**. The throttled low pressure liquid leaving throttling device **40** passes through conduit **44** and is intermixed at point **46** with the cold vapors in conduit **30** which are returning to compressor **14** from final evaporator **28**. Thereafter, this low pressure mixture flows through the portion of conduit **30** which is disposed in cascade condenser **34** where the throttled liquid is at least partially evaporated and absorbs heat from the compressed mixture of liquid condensate and vapor which entered cascade condenser **34** through conduit **42** thereby at least partially further condensing the same.

It is a further and optional feature of the invention that a portion of the liquid condensate flowing in conduit **32** (or from successive similar conduits between intermediate cooling stages) may be split off from conduit **32** through a suitable throttling device **48** and a conduit **50** and evaporated in evaporator **51** to obtain an independent refrigeration effect. In such event that portion of the throttled liquid which is evaporated would by-pass cascade condenser **34** and be returned to conduit **30** at a point located between cascade condenser **34** and auxiliary condenser **26.**

A tri-axial or three stream type heat exchanger may be used in place of cascade condenser **34** and evaporator **51**. In this case, valve **48** and conduit **50** would be eliminated and throttling device **40** would be selected so that the evaporating refrigerant flowing in the low pressure side of the new heat exchanger would be at a rate sufficiently great to absorb heat from both the partially condensing high pressure stream and from the external load.

The at least further partially condensed compressed mixture obtained from heat exchange in cascade condenser **34** passes to the second intermediate cooling stage through conduit **52**. Thereafter, the cycle described in connection with the first intermediate cooling stage is repeated, i.e. a portion of the compressed liquid condensate is withdrawn and throttled through throttling device **54** and passes through conduit **56** to point **58** of conduit **30** where the throttled liquid is mixed with recycling vapors. A portion of the compressed liquid condensate as well as the compressed uncondensed vapors flowing in conduit **52** are withdrawn through conduit **60** and enter into cascade condenser **36** where further at least partial condensation occurs by heat exchange with the throttled liquid passing through the portion conduit **30** disposed within cascade condenser **36**.

The compressed further condensed vapor-liquid mixture from cascade condenser **36** passes to the next successive intermediate cooling stage through conduit **62** and a portion of the liquid condensate is throttled in throttling device **64**, passed through conduit **66** to point **68** where it is mixed with cold vapors being recycled from the final evaporator **28** through conduit **30.** As before, the compressed uncondensed vapors and a portion of the compressed liquid condensate in conduit

62 is withdrawn through concuit 70 and passed to cascade condenser 38 where further condensation occurs as a result of the at least partial evaporation of the liquid throttled in throttling device 64. The further condensed compressed mixture in cascade condenser 38 is withdrawn through conduit 72 and passes through final throttling device 74 to the evaporator inlet 76 which is at the coldest system temperature and essentially at the suction pressure. The liquid condensate is partially or completely evaporated in evaporator 28 to achieve the final refrigeration temperature of the system. The refrigeration circuit is closed by returning the vapors and any residual liquid from evaporator 28 through conduit 30 back to compressor 14, the vapor being mixed with additional throttled liquid portions prior to its passage through each of the cascade condensers associated with each of the intermediate cooling stages as previously described.

FIG. 2 illustrates a modification of the refrigeration system described in FIG. 1 wherein the condensate emanating from the final intermediate cooling stage through conduit 72 is subcooled prior to the final evaporation stage. The operation of the compressor 14, condenser 22, auxiliary condenser 26 and the intermediate cooling stages is identical to that previously described in connection with FIG. 1. Sub-cooling is accomplished by dividing the compressed condensate flowing in conduit 72 into two streams and utilizing the first stream to sub-cool the second stream. More particularly, a conduit 78 is provided for drawing off a said first stream and the first stream is thereafter passed to throttling device 80 where the condensate is throttled to the suction pressure of the system. The throttled liquid which flows into conduit 82 is colder than the compressed condensate flowing into conduit 72. Conduit 82 discharges into said sub-cooler 84 and the throttled liquid is employed to further cool the compressed condensate flowing in line 72. The throttled and at least partially evaporated liquid leaves sub-cooler 84 through conduit 86 and is mixed at point 88 with cold vapor from evaporator 28 being recycled to the compressor through line 30.

The sub-cooled compressor condensate in line 72 (except for that portion withdrawn through line 78) is passed to final throttling device 74 and then to the final evaporator 28, all as previously described with respect to FIG. 1.

FIG. 3 is illustrative of a refrigeration system similar to that described in FIGS. 1 and 2 in which the system has been modified to provide for the use of a compressor lubricating oil admixed with the refrigerant and for the removal of that lubricating oil at a point in the system which is well in advance of the lowest temperatures which the system is capable of producing. While the circulation of lubricating oil is acceptable in compression refrigeration systems operating at relatively high final evaporator temperatures, it cannot be tolerated in systems operating at low temperature since good lubricants have relatively high pour points and will not flow thereby coating heat transfer surfaces, clogging conduits throughout the circuit and possibly resulting in inadequate compressor lubrication. In order to overcome these problems a technique is provided for removing the lubricating oil from the refrigerant mixture which technique includes the steps of adding a relatively high boiling separation fluid to the refrigerant mixture. The separation fluid has a boiling point which is about 35°F. to 115°F., preferably 40° to 70°F. higher than the highest boiling refrigerant in the refrigerant mixture and has a high degree of miscibility or solubility with the lubricating oil being employed. In view of the high boiling point of the separation fluid, the first condensation step in the refrigeration system depicted in FIG. 1 will result in condensation of substantially all of the separation fluid which, due to its high miscibility and solubility will entrain substantially all of the lubricating oil. Thereafter, the compressed liquid condensate including the separation fluid and the lubricating oil may be separated from the compressed vapors, throttled to essentially the suction pressure and recycled to the compressor along with the vapors being recycled to the compressor from the final evaporator.

A wide variety of relatively high boiling, oil miscible separation fluids may be employed in the oil separation method and the choice of a particular fluid will depend on a number of factors including the boiling point of the fluid; the boiling point of the highest boiling refrigerant and the nature of the lubricating oil. The preferred separation fluids are halocarbons since these materials are relatively non-toxic, non-flammable and non-explosive. Typical halocarbons are those which are normally employed as a refrigerant in high temperature refrigeration systems and include such materials as trichlorotrifluoroethane, methylene chloride, trichlorofluoromethane, dichlorofluoromethane, dichlorotetrafluoroethane, or combinations of these materials, Ordinarily, the lubricating oil employed for lubrication of the compressor will be a hydrocarbon.

Ordinarily, enough separation fluid will be present after the condensation step to insure an adequate fluid flow in the separation system and to insure that substantially all of the lubricating oil will enter into solution.

Referring specifically to FIG. 3, which illustrates the refrigeration system of the present invention including an oil separation step, a mixture of refrigerants, separation fluid and compressor lubricating oil as above described is charged into the closed refrigeration circuit generally identified as 100 through a service valve or other conventional charging means as previously described. Subsequent to the charging step the vapors are aspirated by compressor 102 and passed through conduit 104 to condenser 108 where partial condensation occurs by heat exchange with ambient air forced over condenser pipes 110 by a fan 112 or alternate condenser means as described in connection with FIG. 1. As a further option, the partially condensed refrigerant mixture may flow through conduit 114 to auxiliary condenser 116 where, when the system is in operation, further condensation may occur by heat exchange with the cooler vapors returning to compressor 102 from the final evaporator 118 through conduit 120. The partially condensed refrigerant mixture leaves auxiliary condenser 116 through conduit 122 and passes to a vapor-liquid separator 124. The liquid at this point is rich in the separation fluid and compressor lubricating oil as well as the higher boiling refrigerants while the vapor is rich in the lower boiling refrigerant or refrigerants of the mixture.

The liquid separated in separator 124 passes through an optional dryer-strainer 126 where particulate matter is filtered from the stream and residual moisture is removed and then through conduit 128 to throttling device 130 which throttles the liquid, i.e. the pressure of the liquid drops from the discharge pressure to the suction pressure of the system. The throttled liquid next passes through conduit **132** to point **134** of conduit **120** where the throttled liquid is mixed with vapors being recycled from the final vaporator to the compressor. Thereafter this mixture flows in conduit **120** through heat exchanger **136** where it is evaporated and used to absorb heat from and further partially condense the separated vapors from separator **124** which enter heat exchange **136** through conduit **138.** Following the heat exchange the vapor in conduit **120,** which includes the separation fluid and the lubricant, are recycled to the compressor through auxiliary condenser **116.**

The compressed mixture of condensate and uncondensed vapor formed in heat exchanger **136** leaves exchanger **136** through conduit **140** and enters the first of a series of successive intermediate cooling stages. FIG. **3** depicts two intermediate cooling stages **142** and **144.** In each of these intermediate cooling stages a portion of the compressed condensate is throttled and used to cool and further condense a mixture consisting of the remainder of the compressed condensate and the compressed vapors all as previously described in connection with the system illustrated in FIG. **1.** The compressed condensate leaving the final intermediate cooling stage **144** flows through conduit **146** to final throttling device **148,** with or without sub-cooling as described in FIG. **2,** and the throttled liquid from the final throttling device is partially or fully evaporated in final evaporator **118** to achieve the final refrigeration temperature of the system.

In lieu of the system described above and depicted in FIG. **3,** the condenser may be divided into two sections with a heat exchanger and the vapor-liquid separator installed between the sections. In that arrangement, the vapors are aspirated by the compressor, passed through a conduit to the first condenser section where they are desuperheated and partially condensed by heat exchange with ambient air (or alternately by water). The vapor and partially condensed mixture is then passed through the heat exchanger for further cooling and partial condensation and then to the liquid-vapor separator. The condensed liquid at this point is very rich in the separation fluid and contains almost all of the lubricating oils pumped by the compressor along with the aspirated vapors.

The liquid separated in the separator passes through a conduit to a throttling device where the pressure is reduced to essentially the suction pressure. This low pressure mixture of separation fluid and lubricating oil is then passed back through the heat exchanger and the separation fluid is evaporated by counter-current heat exchange with the high pressure stream feeding the vapor-liquid separator. The lubricating oil and evaporated fluid is then recycled to the compressor suction connection.

The vapor mixture exiting from the separator is passed through a conduit to the second condenser section where additional heat is removed by further partial condensation and the condensed mixture is then fed to the intermediate cooling stages as previously described.

The sizing of throttling devices, heat exchangers and other apparatus employed in the system is not critical and will, of course, depend upon the operating conditions for which a particular system is designed. For example, in determining the appropriate size of throttling devices such as capillary tubes in which flow capacity is dependent upon the pressure and quality of entering condensate, the total weight of refrigeratant to be circulated in the system is calculated and this amount is divided between the various throttling devices in the system. Ordinarily, the throttling device feeding the final evaporator will be designed to handle about 30 to 50 percent of the total mass flow, the remainder being divided equally among the throttling devices feeding associated with each intermediate cooling stage. The optimum size of each throttling device is, of course, an empirical determination.

The optimum design for system heat exchangers is also an empirical determination based on well known principles of heat and mass transfer. It has been found however that ordinarily the intermediate heat exchangers should be designed to handle about twice the evaporator load for systems having a final operating temperature of about −80°F. and four or more times the load of systems designed for a final evaporator temperature of −180°F. or lower. Although no critical, the heat exchangers in the system are generally designed to handle approximately equal temperature drops rather than equal thermal loads which results in exchangers of varying sizes due to differences in mass flow in different parts of the system.

The invention will be further understood by reference to the following illustrative examples:

EXAMPLE:

A mixture containing approximately 21.5 wt. percent (16.0 mol percent) trichlorofluoromethane (R–11), 21.5 wt. percent (18.2 mol percent) dichlorodifluoromethane (R–12), (23.8 wt percent) (23.1 mol percent) chlorotrifluoromethane (R–13), 30.2 wt. percent (35.0 mol percent) carbontetrafluoride (R–14), and 3.0 wt. percent (7.7. mol percent) argon (R–740) was charged into the refrigeration system of FIG. **3** at room temperature. The system employed a conventional air conditioning type hermetic compressor and the final evaporator was connected to a chest style ultalow temperature freezer. After charging the system, the refrigeration system was completely sealed off so that the refrigerant mixture would freely circulate without loss. The initial charge pressure was 100 p.s.i.g.

The above-described system was started and permitted to run. System pressure and temperature were periodically measured to determine the ultimate operating characteristics of the system as well as the start-up conditions. The result of these measurements is shown in Table I.

TABLE I

Time (from start)	Air Temp.	Suct. Press. p.s.i.g.	Disch. Press. p.s.i.g.
0.0 hrs.(start-up)	+ 73° F.	24	365
0.25 hrs.	+ 54° F.	22	314
0.50 hrs.	+ 8° F.	19.5	275
0.75 hrs.	− 43° F.	19	260
1.00 hrs.	−100° F.	18	240
1.25 hrs.	−143° F.	16	205
1.50 hrs.	−161° F.		
1.75 hrs.	−169° F.		
2.00 hrs.	−175° F.	9	150
2.25 hrs.	−178° F.	8	137

As can be seen from the Table, the start-up pressures were 24 p.s.i.g. suction pressure and 365 p.s.i.g. discharge pressure (compression ratio 9.75/1). Cooling commenced promptly and continued at a good rate until −178° F, the ultimate low temperature of the system, was reached in just over two hours. Subsequently, the system was shut down and then re-started. No difficulty was encountered in again achieving the operating characteristics set forth in Table I.

For the purpose of comparison, a refrigerant mixture identical to that described above, was charged into a refrigeration system which was identical to that shown in FIG. 3 with the exception that vapor-liquid separators were employed between each cascade condenser. The balanced at rest initial charge pressure was 100 p.s.i.g. The results of the run are set forth in Table II.

TABLE II

Time (from start)	Air Temp.	Suct. Press. p.s.i.g.	Disch. Press. p.s.i.g.
0.0 hrs.(start-up)	+ 72° F.	21	370
0.25 hrs.	+ 52° F.	16.5	260
0.50 hrs.	+ 34° F.	12.5	205
0.75 hrs.	+ 20° F.	10	180
1.00 hrs.	+ 8° F.	8	155
1.25 hrs.	− 2° F.	7	140
1.50 hrs.	− 11° F.	6.5	135
1.75 hrs.	− 19° F.	6	133
2.00 hrs.	− 28° F.	7.5	145*
2.25 hrs.	− 47° F.	10	170
2.50 hrs.	− 68° F.		
2.75 hrs.	− 91° F.		
3.00 hrs.	−114° F.		
3.25 hrs.	−136° F.		
3.50 hrs.	−154° F.		
3.75 hrs.	−171° F.	12	170
4.00 hrs.	−180° F.	11	160
4.25 hrs.	−183° F.	10	155
4.50 hrs.	−185° F.	9.5	148

*4 wt. % (5 mol %) of carbontetrafluoride (compared to the total charge of the mixture) was added to the system at this point.

Table II reveals that the start-up pressure (370 p.s.i.g.) and compression ratio (370/21 = 10.5/1) were higher for the system employing separators. In addition, a comparison of Tables I and II indicates that the rate of cooling was much more rapid without the separators. For example, less than 0.75 hours was required to reach −25°F. without the use of separators (Table I) while the same temperature was not achieved for almost 2.00 hours using separators. Indeed, after 2.00 hours, the rate of cooling in the system employing separators was so low as indicated by the extremely low suction pressure that it became obvious that a larger amount of a lower boiling refrigerant was required for proper operation of the system. Although the addition of carbon-tetrafluoride increased the suction and discharge pressures and the rate of cooling, that rate was still only one-half as fast as the system of FIG. 3 and the ultimate low temperature of −185°F. was not achieved for 4.5 hours. At this temperature, the system pressure was about 6.6 percent higher than the system of FIG. 3.

Of greater significance, as compared to the system of the invention, is the fact that the system utilizing separators could not be restarted after it had been shut down due to excessive start-up pressures.

In addition to the data listed in Table I & II above, a number of temperature measurements were made throughout the systems, with both systems operating at the same discharge and suction pressures and with identical refrigerant mixtures charged into them. Detailed analyses of these data show that approximately 60 percent of the liquid condensate formed by partial condensation in a cooling stage was withdrawn, throttled and mixed with the returning stream in the self-balancing system of the invention. The result, in addition to the self-balancing effect of the system at various operating conditions, was an improvement of 17.4 percent in heat transfer rate in each cascade condenser as determined by the reduction of the log mean temperature difference across each condenser.

A third system employing vapor-liquid separators between each cascade condenser and additionally employing an auxiliary discharge vapor tank similar to that described at column 5, lines 3–19 of Fuderer U.S. Pat. No. 3,203,194 was employed. The discharge tank was connected to the system across the compressor so that excess high pressure vapors could be stored during start-up (with the low pressure connection closed and the high pressure connection open). The system employed the same refrigerant mixture at the same balanced at rest pressure as employed with the system of FIG. 3. The results of this run are set forth in Table III.

TABLE III

Time (from start)	Air Temp.	Suct. Press. p.s.i.g.	Disch. Press. p.s.i.g.
0.0 hrs.(start-up)	+ 65° F.	22	370
0.25 hrs.	+ 47° F.	17	240
0.50 hrs.	+ 27° F.	15	230
0.75 hrs.	+ 8° F.	13	200
1.00 hrs.	− 10° F.		
1.25 hrs.	− 28° F.		
1.50 hrs.	− 44° F.	10	168*
1.75 hrs.	− 63° F.	11.5	178
2.00 hrs.	− 99° F.		
2.25 hrs.	−138° F.		
2.50 hrs.	−167° F.	14	185
2.75 hrs.	−176° F.		
3.00 hrs.	−178° F.		
3.25 hrs.	−181° F.	7.5	134

*Vapors in auxilliary tank permitted to flow into system.

As compared to the system of FIG. 3, the results indicate that the rate of cooling in the system employing the auxilliary discharge tank was about one-half the cooling rate for the self-balancing system of the invention until such time as operating conditions were reached which made it appropriate to permit the excess high pressure vapors to flow into the system at the suction pressure (by closing the high pressure connection and opening the low pressure connection between the tank and the system). Once the full refrigerant charge was permitted to flow in the system, operating results were comparable to the system of the invention. However, use of the auxiliary tank was essential to restarting the system after shut-down.

These comparative examples illustrate that the system of the invention is simplier to contruct and operate and achieves far superior operating results as comapred to mixed refrigerant systems which require complete separation of vapor and liquid phases at each intermediate cooling stage.

Having thus described the general nature as well as specific embodiments of the invention, the true scope will not be pointed out in the appended claims.

What is claimed is:

1. A compression refrigeration process employing a mixture of refrigerants having different boiling points, comprising the steps of:

a. compressing a vaporous mixture of said refrigerants;

b. partially condensing the compressed refrigerant vapor to form a mixture consisting of compressed condensate and compressed uncondensed vapor;
c. subjecting the compressed mixture from step (b) to at least one intermediate, cooling stage, each said intermediate cooling stage including the steps of throttling a portion of said compressed condensate to a lower pressure, mixing said throttled condenate with the mixture of refrigerants being recycled to the compressing step from the final evaporator, evaporating said throttled condensate to absorb heat from and at least partially condense the compressed vapor in the remaining mixture of the compressed condensate and compressed vapor, returning the mixture of evaporated, throttled condensate and recycled refrigerant mixture to step (a), and passing said at least partially condensed compressed mixture to the next successive intermediate cooling stage;
d. throttling said compressed mixture obtained from the last intermediate cooling stage to a lower pressure;
e. at least partially evaporating the throttled condensate produced in step (d) to produce the final refrigerating temperature and recycling the at least partially evaporated mixture of refrigerants to said compressing step.

2. The process of claim **1** wherein the portion of compressed condensate throttled in each intermediate cooling stage is determined by the capacity of the throttling device.

3. The process of claim **1** wherein 10 to 95 percent of the compressed condensate entering each successive intermediate cooling stage is throttled.

4. The process of claim **2** wherein said throttling device is a capillary tube.

5. The process of claim **2** wherein said throttling device is a thermal expansion valve.

6. The process of claim **1** including 2 to 6 successive intermediate cooling stages.

7. The process of claim **1** wherein said refrigerant mixture comprises 2 to 7 individual refrigerants.

8. The process of claim **7** wherein the difference in boiling point between each refrigerant and the closest boiling other refrigerant in said mixture is 50°F. to 180°F.

9. the process of claim **7** wherein said refrigerant mixture includes at least two halocarbon refrigerants.

10. The process of claim **7** wherein said refrigerant mixture includes at least two halocarbon refrigerants and at least one inert gas selected from the group consisting of argon, nitrogen and neon.

11. The process of claim **1** wherein only a portion of the cooling effect produced by evaporating said throttled condensate in an intermediate cooling stage is employed to absorb heat from and at least partially condense the compressed uncondensed vapor and the remainder is employed to produce an external refrigeration effect.

12. The process of claim **1** wherein the compressed mixture obtained from the last intermediate cooling stage is divided into first and second streams, throttling said first stream to a low pressure, evaporating said throttled first stream to absorb heat from and sub-cool said second stream, mixing said throttled, evaporated first stream with refrigerant being recycled to the compressor from the final evaporator, throttling said second stream to a lower pressure, and passing said throttled second stream to step (e).

13. The process of claim **1** wherein said refrigerant mixture further includes a compressor lubricating oil and a separation fluid, said separation fluid having a boiling point which is 35° to 115°F. higher than the highest boiling refrigerant in said refrigerant mixture and being soluble in said lubricating oil, separating the compressed mixture obtained in step (b) into liquid and vapor phases whereby said lubricating oil and a substantial portion of the separation fluid remains in said liquid phase, throttling said separated oil-laden separation fluid to a lower pressure, mixing said throttled oil-laden separation fluid with refrigerant being recycled to the compressor from the final evaporator and passing said compressed vapor phase to the first intermediate cooling stage.

14. The process of claim **13** wherein the separated compressed vapor phase is at least partially condensed by heat exchange with said mixture of throttled oil-laden separation fluid and recycled refrigerant prior to said first intermediate cooling stage.

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