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Quick

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[54] **REFRIGERATION SYSTEM WITH PLURAL EVAPORATOR MEANS**

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Related U.S. Application Data

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[52] **U.S. Cl.** **62/196,** 62/278, 62/510

[51] **Int. Cl.** **F25b 41/00**

[58] **Field of Search** 62/196–200, 218, 278, 510

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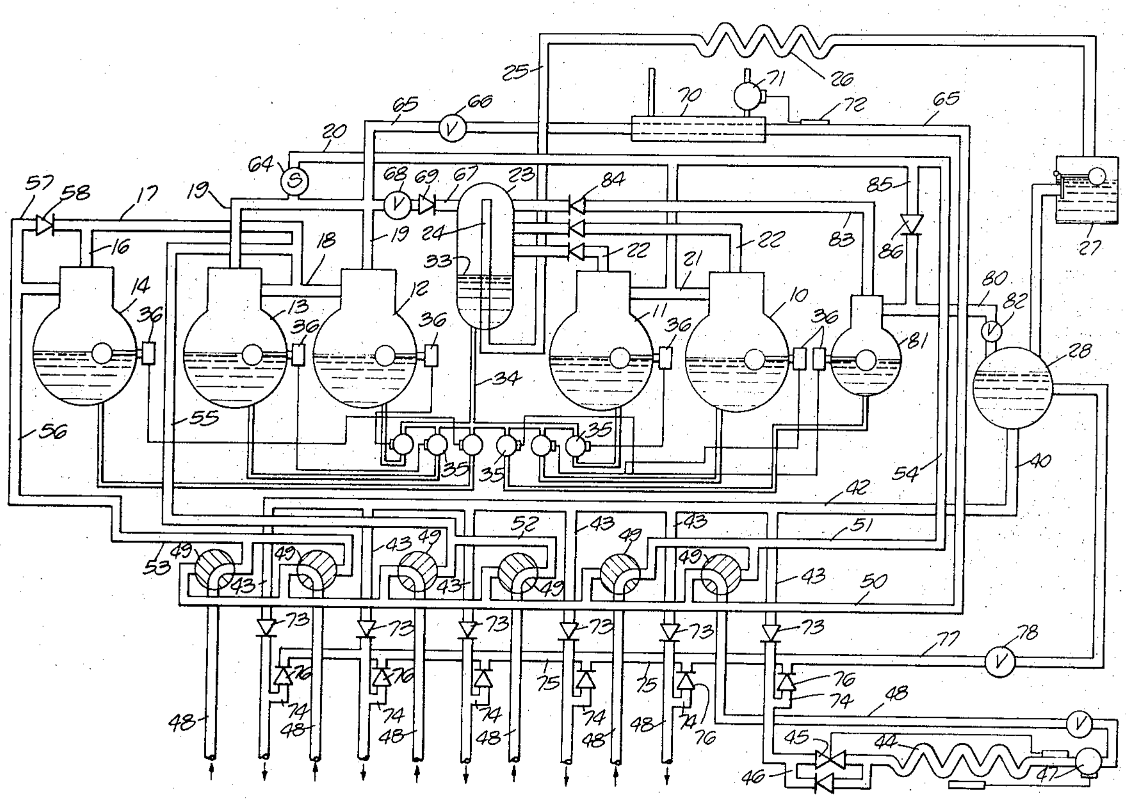
[57] **ABSTRACT**

A refrigeration system particularly adapted for supermarkets and the like employing a multiplicity of refrigerated fixtures or storage rooms operating at varying load requirements wherein centrally located compressors operate all of the evaporators which may include evaporators functioning at different temperature levels from approximately freezing to substantially below freezing. The various embodiments of the refrigeration system employ a separate compressor operating at a relatively high suction pressure which produces an intermediate refrigerant evaporation temperature far below the temperature of the condensed refrigerant in the main refrigeration system and approaching the temperatures of the evaporators of the system. In each embodiment the relatively high suction pressure and intermediate temperature developed by the separate compressor is employed for reducing the temperature of the condensed refrigerant of the main refrigeration system to that intermediate temperature before supplying same to the evaporators of the refrigerated fixtures with some of the embodiments employing direct communication between the separate compressor suction and the supply of refrigerant and others employing indirect communication by heat exchanger means. The separate compressor operates at a very efficient level and thereby increases the overall efficiency and the operational characteristics of the refrigeration system.

15 Claims, 5 Drawing Figures



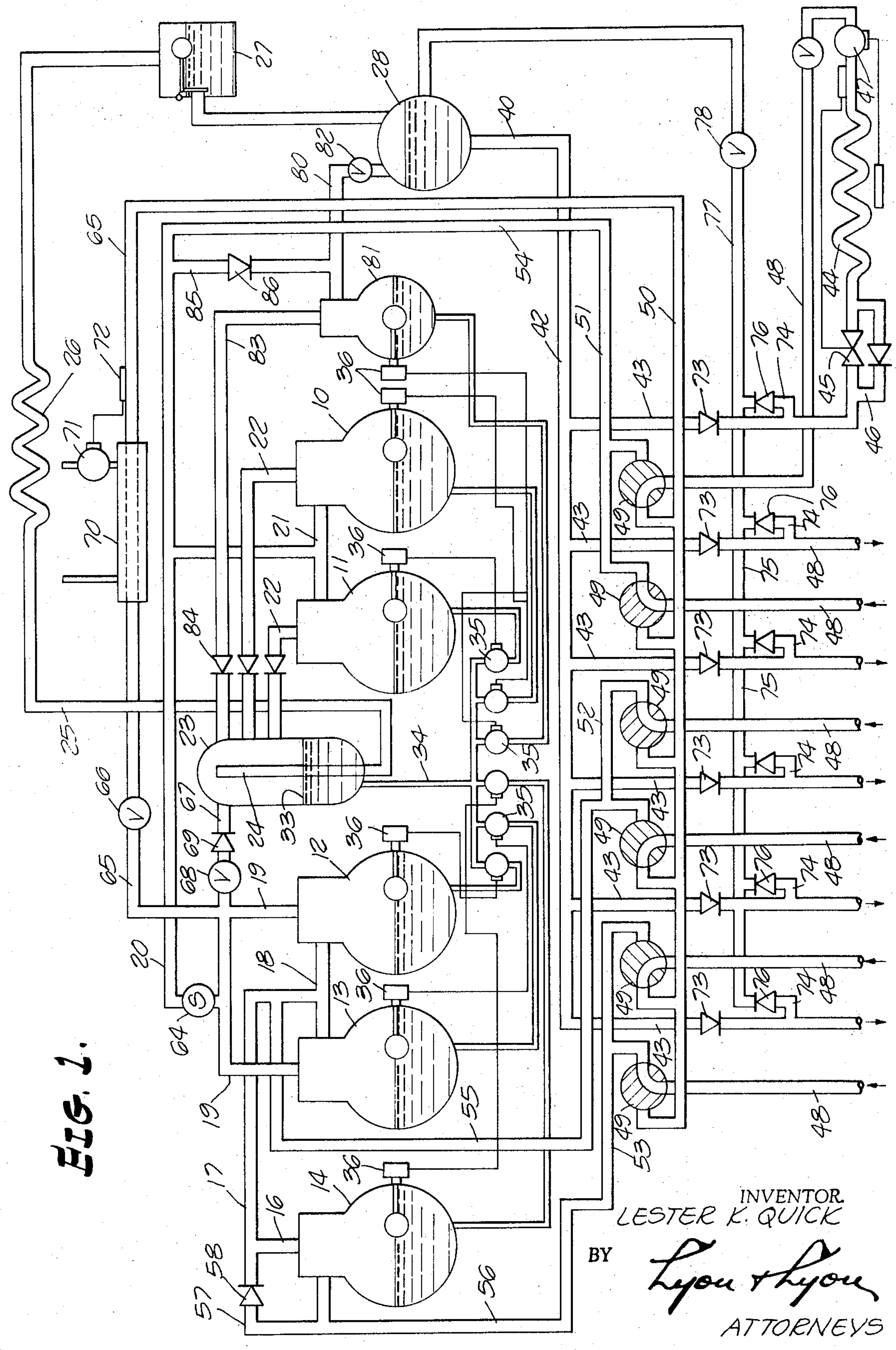


FIG. 1.

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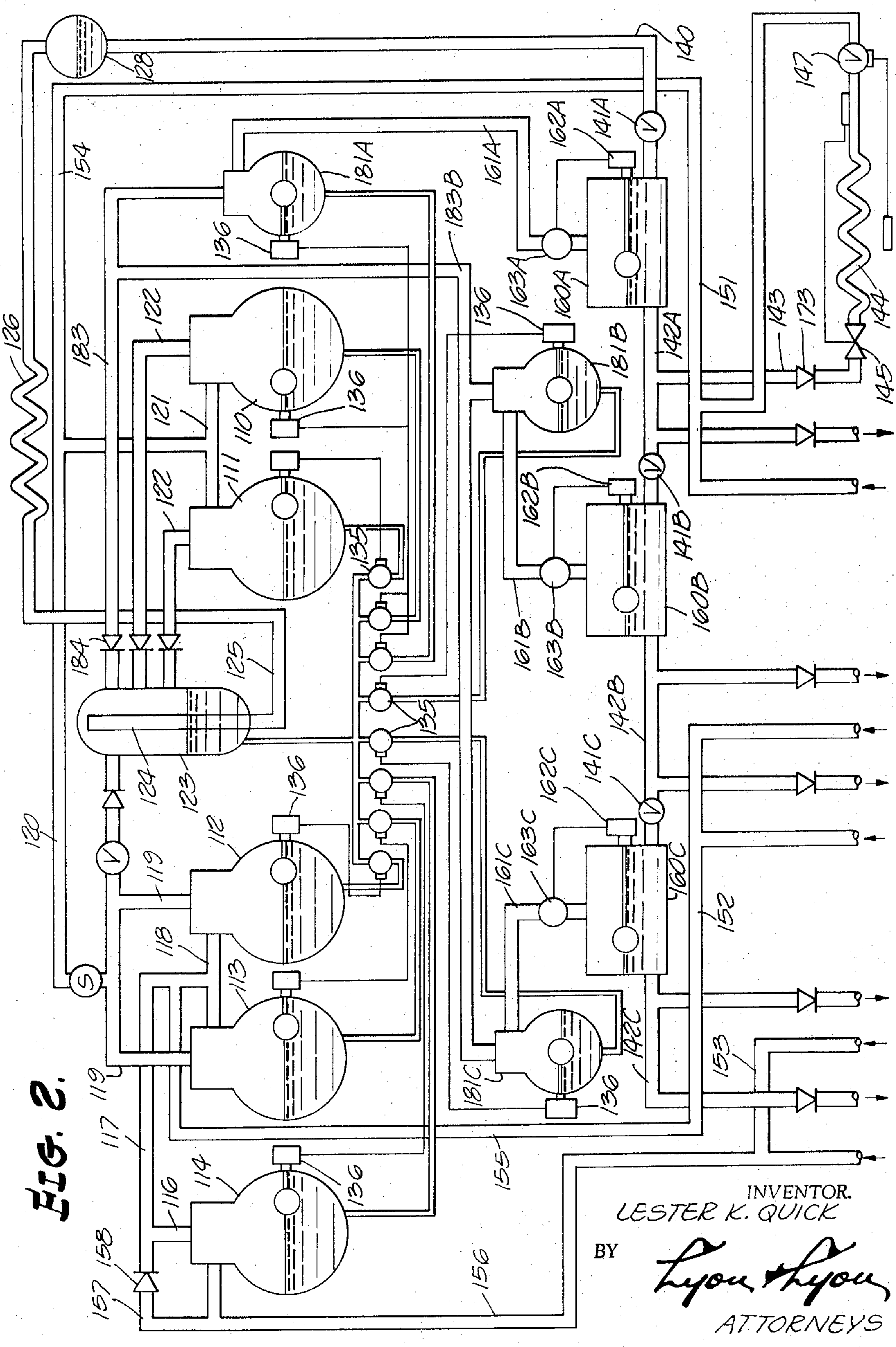

FIG. 2.

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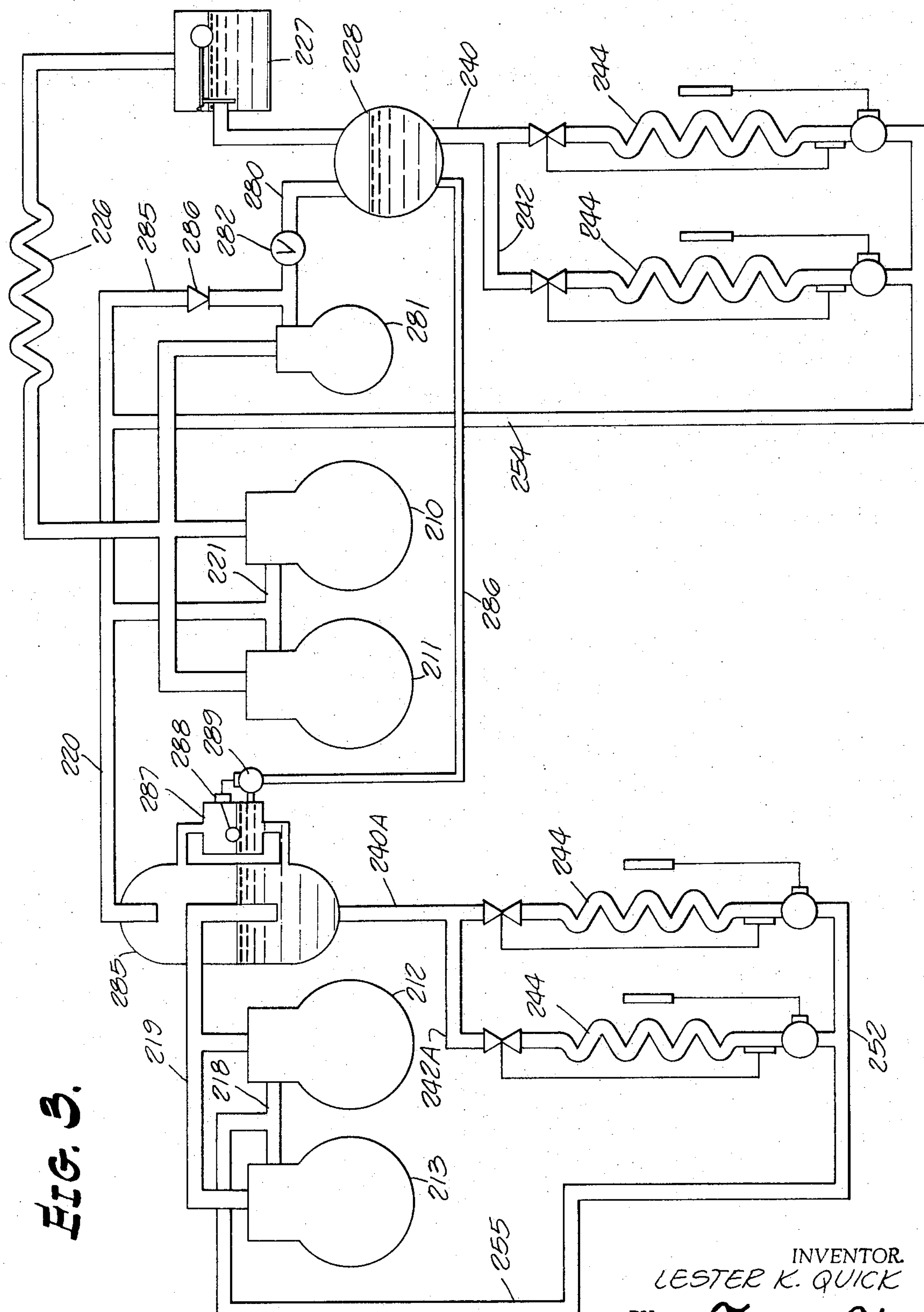

FIG. 3.

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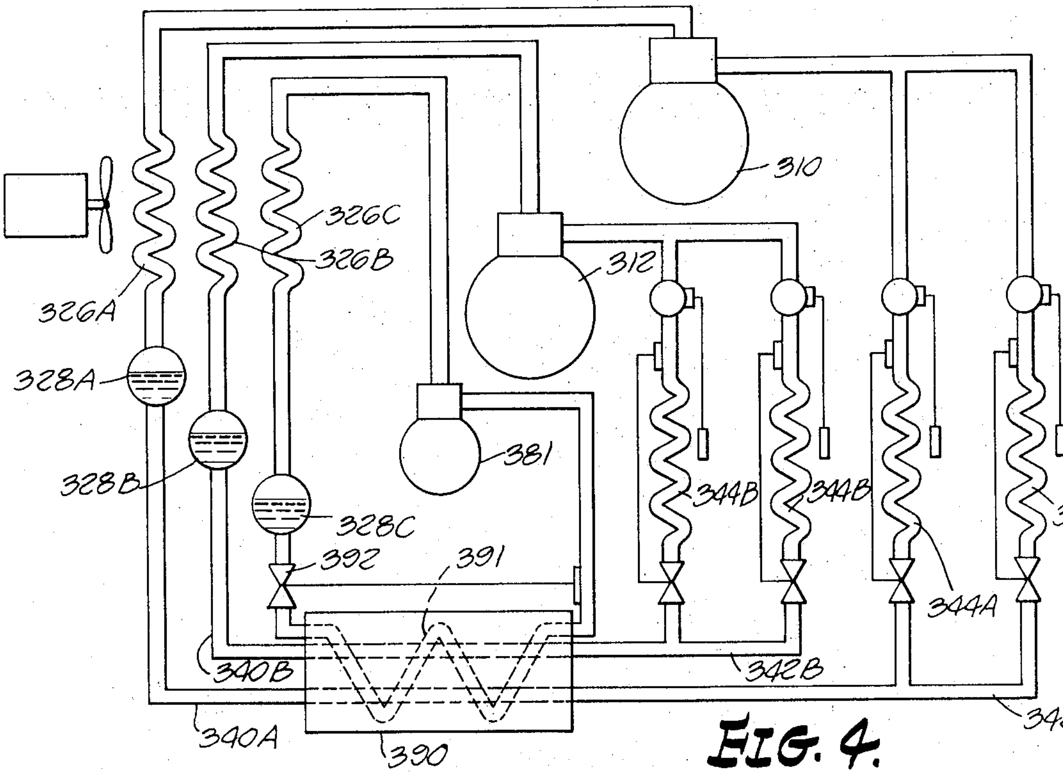

FIG. 4.

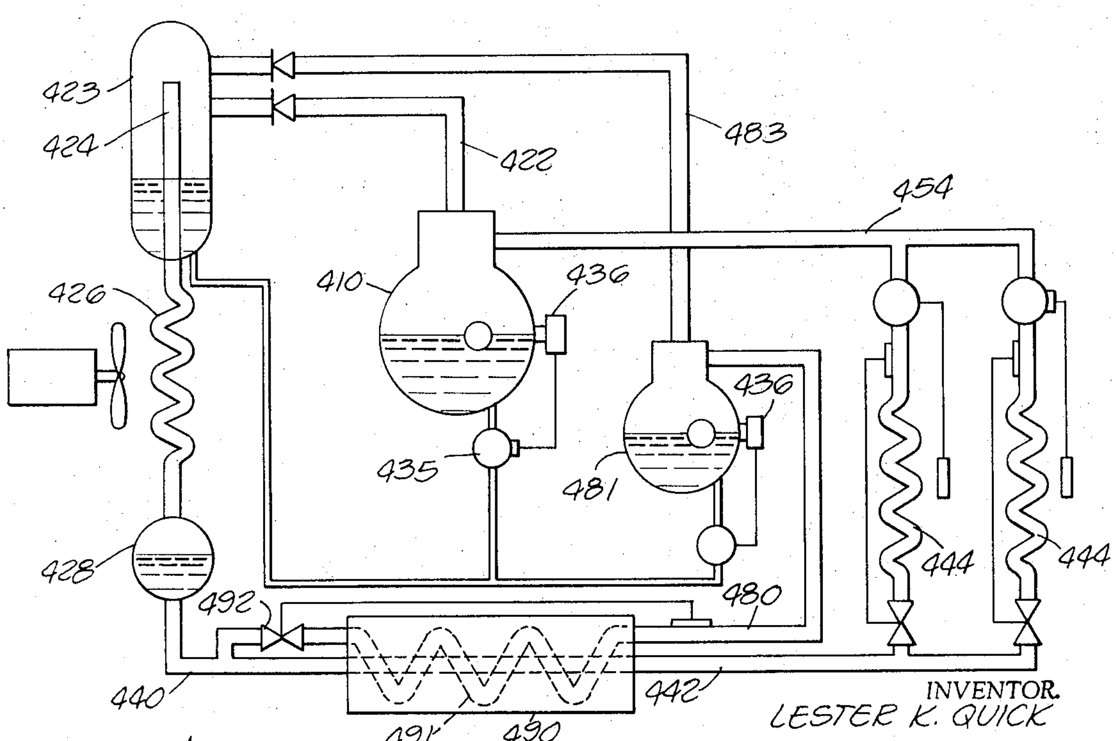

FIG. 5.

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REFRIGERATION SYSTEM WITH PLURAL EVAPORATOR MEANS

This is a continuation-in-part of my application Ser. No. 19,801, filed Mar. 16, 1970 and entitled REFRIGERATION SYSTEM WITH HOT GAS DEFROSTING, which has now issued as U. S. Pat. No. 3,645,109.

This invention relates to refrigeration systems for operating a multiplicity of separate evaporators functioning independently to cool individual refrigerated fixtures, rooms or the like such as in supermarkets or industrial installations. The invention is directed particularly to improving the operation and efficiency of these refrigeration systems by greatly reducing the temperature of the liquid refrigerant before supplying same to the individual evaporators and accomplishing such temperature reduction in a highly efficient and totally different manner.

In recent years there has been a trend toward the use of relatively large compressors centrally located in a supermarket installation for operating the refrigerated fixtures throughout the store rather than individual compressor-fitted fixtures. This trend is similar to the concept that has been employed in industrial refrigeration for certain limited applications for many years. The plurality of evaporators for refrigerating the plurality of fixtures, rooms or the like are operated by a single compressor or plural compressors combined in parallel or compounded relationship to improve the overall efficiency and dependability of the system. Such arrangements have led to the increased use of hot gas defrosting of the evaporators by the selection of only a portion of the evaporators for defrosting at any one time which could not be as readily and efficiently accomplished in simple systems matching a compressor to each evaporator.

While certain advantages and improved efficiencies, such as hot gas defrosting, have been employed in central refrigeration systems, there has been very little change between the basic refrigeration arrangement employed in a small fixture and that used in the central systems other than the mere increase in size. This is due to the highly variable requirements of the multiplicity of evaporators in a central system and other factors such as the variations in ambient conditions with the resultant variations in condensing pressures and temperatures, particularly when air cooled condensers are used. Conventionally the central refrigeration systems are designed to accommodate the most adverse refrigeration load and ambient conditions anticipated with no provisions made for affirmatively improving system efficiencies under other conditions. One such design deficiency is in allowing the temperature and pressure of the condensed liquid refrigerant to vary over a substantial range which causes substantial difficulty in the functioning of the expansion valves on the evaporators due to both the variation in pressure as well as the creation of substantial quantities of flash gas in the evaporator. Moreover since the expansion valves must function properly to accommodate the worst ambient conditions when the head pressure and refrigerant temperature are very high it is conventional to prevent the head pressure from dropping to excessively low levels in cold weather which would otherwise create an intolerable range of different pressures for the expansion valve to accommodate. Thus even the available improvement in efficiency in cold weather of allowing the head pressure to drop to low levels is sacrificed for assuring proper operation of the expansion valves and other system components.

While it is relatively common in many systems to provide sub-coolers in one form or another to cool the condensed refrigerant to thereby avoid some of the flash gas created in the evaporators and thereby improve the operation of the expansion valves, the degree of cooling has been relatively minor, perhaps 10° to 15° below condensing temperature and still far above room temperature and evaporator temperature. Thus in the conventional central refrigeration system, of all of the liquid refrigerant that is conducted to each of the evaporators where it is expanded and upon evaporation is returned through the suction line to the compressor, a substantial proportion merely turns to flash gas in passing through the expansion valve thereby performing no useful work in cooling the evaporator or the fixture but merely cools the balance of the liquid refrigerant. While the proportion of flash gas will vary for the particular system and ambient conditions, in a typical system having 120°F condensing temperature and −40°F evaporator temperature with R-502 refrigerant there will be approximately 61 percent of the refrigerant forming flash gas in passing through the expansion valve thereby performing no useful work in cooling the evaporator. The necessity of conducting this excess liquid refrigerant to the evaporator and then returning the flash gas requires the use of larger tubing than would otherwise be necessary and in central refrigeration systems the amount of tubing used is very great whereby the cost of tubing is very substantial. Moreover this flash gas must be compressed from the very low suction pressure in the evaporator to the condensing head pressure which is a substantial pressure differential and, as is well known in the art, such compression is less efficient than if the compression can be from higher suction pressure with denser gas and over a smaller pressure differential.

While there have been conventional refrigeration systems for operating a single evaporator, such as a liquid chiller for a hydronic system, wherein stepped reductions in temperature and pressure of the condensed refrigerant are accomplished, usually with centrifugal compressors, these are completely balanced systems with stable loads and the principles and equipment are not applicable to central refrigeration systems that include multiple evaporators and are subject to load fluctuations such as in supermarkets.

One of the requirements in the use of hot gas defrosting with a central refrigeration system is the disposition of the refrigerant resulting from defrosting which may be virtually any proportion of liquid and gas. Numerous methods have been developed to accomplish this objective including that which is disclosed in my copending application referred to above. Substantial inefficiencies can be introduced into the system if the method for reintroducing the defrosting refrigerant into the cycle is inefficient, and therefore it is most desirable to accomplish this necessary function with a minimum of extra equipment and power.

Another difficulty that has been encountered with respect to previously installed central refrigeration systems is the lack of capacity of the system to handle additional fixtures in the store as required by expansion. On some occasions, completely separate central refrigeration systems are installed for additional fixtures thereby losing some of the advantages of central systems. On other occasions it has been necessary to replace the large compressors with even larger units and similarly increase the capacity of the condenser all of which is extremely costly. However, again it is by reason of the fact that the original compressors must handle such a large proportion of refrigerant that is merely converted to flash gas in the evaporators that these compressors are of an insufficient capacity to accommodate additional fixtures.

Accordingly in summary the principal object of this invention is to provide an improved refrigeration system for operating a multiplicity of evaporators wherein the temperature of the liquid refrigerant supply is reduced to a temperature substantially below the room temperature by employing a separate compressor for causing such temperature reduction which compressor operates at a substantially more efficient level than the system compressors thereby improving the overall system efficiency.

Another object of this invention is to provide an improved refrigeration system with hot gas defrosting in which the resultant defrosting refrigerant is reintroduced into the liquid refrigerant supply and both sources of refrigerant are reduced to a constant temperature and pressure by the controlled evaporation through the suction of a separate compressor.

A further object of this invention is to provide a refrigeration system using piston type compressors in which the temperature of the refrigerant supply is reduced from the condensing temperature to an intermediate level between the condensing and evaporator temperatures before supplying the refrigerant to the evaporator for minimizing flash gas formed in the evaporator wherein such temperature reduction is accomplished by a separate compressor operating independently of the load requirements on the refrigeration system compressors.

Still another object of this invention is to provide an arrangement for improving the efficiency and capacity of a central refrigeration system having one or more system compressors by means of employing a separate compressor to cause a substantial reduction in the temperature of the liquid refrigerant subsequent to condensing but prior to supplying same to the evaporators.

A still further object of this invention is to provide an improved central refrigeration system operating evaporators at two or more suction and temperature levels wherein separate compressor means are employed to reduce the temperature of the liquid refrigerant being supplied to the evaporators to temperature levels more closely approximating the respective temperature levels of the evaporators whereby flash gas formed in each evaporator is minimized.

Still another object of this invention is to provide a refrigeration system operating evaporators at two or more temperature and suction levels in which the compressors are compounded with an intercooler between the compounded stages modified to provide a supply of liquid refrigerant for the lower of those two stages at a temperature more closely approximating the temperature of the evaporators of such lower stage and wherein a separate compressor is employed for reducing the temperature of the refrigerant supply for the highest temperature evaporators.

Another object of this invention is to provide a novel central refrigeration system wherein the normal liquid supply is greatly reduced in pressure and temperature at the location of the compressors with the flash gas produced by such reduction being directly and efficiently returned to such compressors. A further object is to provide such a system wherein the lengthy refrigerant supply and suction lines to the remotely located fixtures may be reduced in size by reason of such direct flash gas return. Still another object is to provide such a system wherein the pressure in the liquid refrigerant supply line to the evaporators is regulated to a substantially constant level far below the high head pressure on the compressors whereby a pressure differential incentive for the return of condensate from hot gas defrosting is always present in the liquid refrigerant supply side of the cycle. Still another object is to provide such a refrigeration system in which means are provided for regulating the pressure of the hot gaseous defrosting refrigerant to a constant level whereby the pressure differential across the evaporators is always a predetermined amount thereby permitting accurately predictable defrosting times for respective evaporators. A still further object is to provide such a system wherein the flash gas is returned to one or more compressors operating at the higher suction pressures thereby improving the efficiencies of all of the compressors operating at lower suction pressures.

Other and more detailed objects and advantages of this invention will appear from the accompanying description and drawings wherein:

FIG. 1 is a schematic illustration of the preferred form of my invention illustrating a central refrigeration system employing compressors in compounded relationship operating fixture evaporators at three different suction pressures and a separate compressor operating at a high suction pressure to reduce the temperature of the liquid refrigerant supply to substantially below room temperature as well as accommodating the gaseous refrigerant returned from the hot gas defrosting.

FIG. 2 is a schematic illustration of a modified form of the refrigeration system of FIG. 1 in which a total of three separate compressors are employed for reducing the temperature and pressure of the liquid refrigerant supply for the three separate temperature levels of evaporators.

FIG. 3 is a schematic illustration of the refrigeration system of this invention operating evaporators at two different temperature and suction levels having the compressors compounded with the intercooler therebetween providing the source of liquid refrigerant for the low temperature evaporator and a separate compressor reducing the pressure and the temperature of the liquid supply to the high temperature evaporators as well as the intercooler.

FIG. 4 is a schematic illustration of another modified form of refrigeration system of my invention wherein the liquid refrigerant for two completely separate refrigeration systems, as for example for different temperatures, is reduced to an extremely low temperature by still another completely separate refrigeration system employing a separate compressor.

FIG. 5 is a schematic illustration of another modified form of refrigeration system of my invention in which the liquid refrigerant for the system evaporators is reduced in a somewhat conventional sub-cooler with the suction of the sub-cooler connected to a separate compressor.

While the various figures illustrate varying numbers of system compressors for operating the evaporators it will be understood and readily appear to those skilled in the art that more or fewer system compressors may be used in any of the systems depending on the load requirements and other factors. However with respect to the separate compressors for reducing the temperature and sometimes the pressure of the liquid refrigerant for the overall system, it is intended and will be understood that only the actual compressors illustrated and described are employed and in each instance these are substantially smaller in their capacity and power requirement than the remaining compressors of the system. In some instances the separate compressor may be a separate portion of a larger compressor by using special heads but the operation thereof will remain the same as though it were physically separate. Furthermore while the evaporators are illustrated in varying numbers and detail in the five systems illustrated by the figures, it is to be understood and will readily appear to those skilled in the art that normally a great many evaporators will be provided and operating at each temperature level with the exact number being dependent on the size of the supermarket or industrial installation.

While only FIG. 1 includes an illustration of the appropriate components for accomplishing hot gas defrosting of the various evaporators, it will be understood and readily appear to those skilled in the art that the systems of FIGS. 2 through 5 may be provided with the appropriate components for accomplishing hot gas defrosting and that such has been omitted from these Figures merely for simplicity of illustration. Finally with respect to the versatility and adaptability ofthe compressors of the refrigeration system of this invention, the term "compressors" shall mean piston type compressors that are common in the industry rather than centrifugal type compressors that have certain capabilities such as accommodating liquid refrigerant that is not possible with piston type compressors. Thus the versatility of my disclosed systems will readily appear to those skilled in the art.

Referring now more particularly to FIG. 1, a typical refrigeration system of this invention is illustrated for accomplishing all of the refrigeration requirements in the usual supermarket and the adaptability of this system as to other types of installations will readily appear to those skilled in the art. In the usual supermarket there are at least three distinct levels of refrigerated fixture temperatures required which are normally referred to as standard temperature fixtures (i.e. diary, produce and fresh meat), low temperature fixtures (i.e. frozen foods and walk-in freezers) and ultra-low temperature fixtures, (i.e. icecream cases and special frozen food fixtures requiring very low back pressures). To provide these three temperature levels in a supermarket or any installation having similar requirements, it is necessary to provide compressors operating at three distinct suction pressures, if the best operating efficiency is to be maintained, which compressors are commonly referred to by the corresponding temperature level of the fixtures to which their suctions are connected. Thus in FIG. 1 there are two compressors 10 and 11 connected in parallel to operate the standard temperature fixtures and will be referred to as the standard temperature compressors. Two more compressors 12 and 13 are connected in parallel to operate the low temperature fixtures and compressor 14 is the ultra-low temperature compressor.

In their normal refrigeration operation compressors **10, 11, 12, 13** and **14** are compounded in this illustrated system of FIG. **1.** The discharge **16** of ultra-low temperature compressor **14** is connected through conduit **17** to the suction side **18** of the low temperature compressors **12** and **13.** The discharge sides **19** of low temperature compressors **12** and **13** are connected through conduit **20** to the suction side **21** of standard temperature compressors **10** and **11.** The discharges **22** of standard temperature compressors **10** and **11** are separately connected to a discharge accumulator and oil separator **23** which has an outlet stand pipe **24** connected to a conduit **25** leading to the system condenser **26** and thence through a high side float valve **27** to the system receiver **28.** As is well known to those skilled in the art, the high side float valve **27** functions to allow only liquid refrigerant to flow from the condenser **26** to the receiver **28** without regard for temperatures, pressures, rate of supply, etc. and the particular importance of the high side float valve in this system will be described hereinafter. As thus far described it may be seen that the entire system has a single condenser-receiver arrangement for supplying the liquid refrigerant requirement of all of the evaporators rather than providing separate condenser-receivers for each of the temperature levels. The ultra-low temperature compressor **14** is compounded with the low temperature compressors **12** and **13** which are in turn compounded with the standard temperature compressors **10** and **11.**

The above-mentioned discharge accumulator and oil separator **23** forms no part of the invention of this application although it will be understood that some form of oil separation and return must be provided in the refrigeration system of this type. The arrangement illustrated in FIG. **1,** as well as FIG. **2,** is of the type more fully disclosed in my copending application Ser. No. 815,452 entitled "Refrigeration System Oil Separator" but will be briefly described here for completeness. The vessel comprising the discharge accumulator and oil separator **23** serves to separate the oil from gaseous refrigerant discharged from the compressors and to maintain an oil supply **33** in the bottom of the vessel. An oil line **34** is connected through a header to individual conduits returning to the crank cases of each of the compressors **10** through **14.** Each such oil return conduit has a separate valve **35** the operation of which is individually controlled by a float actuated sensor **36** associated with the compressor to which the oil return line and valve are connected. Thus as the sensor **36** associated with a compressor senses a drop in the desired oil level in the compressor the associated valve is opened to cause the return of oil to that compressor by virtue of the higher pressure existing in the vessel of separator **23** whereby the desired oil levels in each of the compressors are independently maintained.

Returning to the description of the closed cycle refrigeration system of FIG. **1,** the receiver **28** is connected through a conduit **40** to a liquid refrigerant header **42,** all of which is situated at the central location of the compressors. The liquid header **42** is connected through conduits **43** to the evaporators **44** of the refrigerated fixtures. For simplicity of illustration only one evaporator **44** and six branch conduits **43** are shown although one or more separate evaporators will be associated with each conduit **43** and in most commercial installations of this type there will be a great many more than six branch conduits **43.**

As is conventional, each evaporator **44** is provided with an expansion valve **45** and a by-pass line **46** having a check valve is provided for purposes of by-passing the expansion valve **45** during hot gas defrosting in which the refrigerant flows in the reverse direction. In addition each evaporator will be provided with a temperature responsive control such as a conventional liquid line solenoid valve or, as shown in each of the drawings, a suction pressure regulator valve **47**.

Each evaporator is connected through a suction conduit **48** to a suction header through, for purposes of hot gas defrosting, a three-way valve **49**. The three-way valve **49** is operable to selectively connect the suction line **48** to either the common hot gas header **50** or one of the three respective suction headers **51, 52** or **53.** It is to be noted that as viewed in FIG. **1** the two suction conduits **48** toward the right hand side are connected through three-way valves to the standard temperature suction header **51** whereby the associated fixtures would be standard temperature fixtures, the two middle suction conduits **48** are connected through three-way valves to the low temperature suction header **52** whereby the associated fixtures would be low temperature fixtures, and the two left hand suction conduits **48** are connected through three-way valves to the ultra-low temperature suction header **53** whereby the fixtures associated with those two suction conduits would be ultra-low temperature fixtures. The standard temperature suction header **51** is connected through conduit **54** to the suction side **21** of compressors **10** and **11.** The low temperature suction header **52** is connected through conduit **55** to the suction side **18** of low temperature compressors **12** and **13.** The ultra-low temperature suction header **53** is connected through conduit **56** to the suction side of compressor **14.**

A by-pass line **57** with a check valve **58** is connected from the ultra-low temperature suction line **56** to the discharge side **16** of the ultra-low temperature compressor **14** for allowing the by-pass of refrigerant upon the failure of compressor **14** or the temporary return of an excess quantity of refrigerant to compressor **14.** In essence the suction from all of the ultra-low temperature fixtures will then be connected to the suction side **18** of the low temperature compressors **12** and **13** until the malfunction of compressor **14** is corrected or the excess return load is reduced. While the suction provided by low temperature compressors **12** and **13** will not be as low as desired for maintaining the ultra-low temperature in these fixtures the temperature maintained will be adequate for a short duration.

Thus it will be seen that during the normal refrigerating operation of a given evaporator **44** the associated three-way valve **49** is positioned to connect the suction conduit **48** to the associated suction header (**51, 52** or **53**) to return the evaporated refrigerant to the appropriate compressor whereby the appropriate suction pressure is maintained on that evaporator for providing the desired temperature and yet the liquid refrigerant is supplied to all of the evaporators from the common liquid header **42.**

In order to accomplish hot gas defrosting of the evaporators in the system of FIG. **1** as thus far described, a variety of arrangements may be used but I have illustrated a still further improvement in my invention of a refrigeration system adapted for hot gas defrosting similar to that described in my aforementioned parent application Ser. No. 19,801, now U. S. Pat. No. 3,645,109, entitled "Refrigeration System With Hot Gas Defrosting," which application is directed specifically to the improvements in hot gas defrosting. When it is desired to defrost a given evaporator **44** the associated three-way valve **49** is repositioned from its normal refrigerating connection wherein the suction conduit **48** is connected to the suction header (**51, 52** or **53**) to the position wherein the suction conduit **48** is connected to the hot gas header **50.** Simultaneously a solenoid valve **64** in conduit **20** is closed thereby eliminating the connection of the discharge sides **19** of low temperature compressors **12** and **13** to the suction side **21** of standard temperature compressors **10** and **11.** A conduit **65** connects the discharge sides **19** of low temperature compressors **12** and **13** to hot gas header **50** to supply the hot gaseous refrigerant for defrosting from compressors **12** and **13.** A pressure regulating valve **66** is provided in conduit **65** and functions to limit the pressure in conduit **65** to a predetermined constant downstream pressure of a relatively high level which is related to the temperature of gas desired to accomplishing the defrosting. In order to avoid the occurrence of excessive pressures on the discharge sides of compressors **12** and **13,** as might be produced by the throttling or pressure regulating effect of valve **66,** a by-pass conduit **67** is connected from the compressor discharge side **19** to the accumulator and oil separator vessel **23.** A pressure regulating valve **68** of the type for regulating upstream pressure is provided in by-pass conduit **67** and is set for a slightly higher pressure value than the setting of valve **66** whereby any excess refrigerant not being passed through valve **66** for hot gas defrosting purposes will be discharged through by-pass conduit **67** into the vessel **23** thereby entering the normal refrigeration cycle. A check valve **69** is provided in by-pass conduit **67** to prevent reverse flow therethrough such as would otherwise occur during normal refrigeration operation ofthe system (when there is no hot gas defrosting being performed) wherein valve **64** would be open and the pressure on the discharge sides **19** of compressors **12** and **13** would be substantially lower than the pressure in vessel **23.** By this arrangement it may be seen that defrosting gaseous refrigerant is provided at a precisely controlled and constant pressure without regard for load fluctuations or ambient conditions. If in a given installation a greater heat load capacity for defrosting is desired the valving can be rearranged to take the defrost gaseous refrigerant from the main discharge line **25** with an appropriate throttling valve for assuring the proper pressure in low ambient conditions.

In connection with the hot gas defrosting arrangement, a desuperheater **70** is provided in heat exchange relationship with conduit **65** between the pressure regulator valve **66** and the hot gas header **50** for eliminating at least most of the superheat in the gaseous refrigerant whereby the refrigerant will be more nearly its saturation temperature for that predetermined pressure as regulated by valve **66.** This avoids excessive heating of the conduits leading to the evaporators which has been found advantageous without adversely effecting the defrosting. Desuperheater **70** is shown as a water-cooled type with the water flowing therethrough as controlled by valve **71** which is in turn controlled by a temperature responsive sensor **72** associated with the conduit **65** downstream of the desuperheater. In this manner the temperature of the hot gas may be lowered to approximately the saturation temperature for the pressure setting of regulator valve **66.** It will be obvious to those skilled in the art that any form of desuperheater may be used for accomplishing this objective.

The desuperheated gaseous refrigerant passes from hot gas header **50** through the three-way valve or valves **49** associated with the evaporators to be defrosted, then passes in a reverse direction through the suction conduit **48** to the evaporator **44,** and thence through the by-pass conduit **46** into the liquid line **43.** A check valve **73** is provided in each liquid line **43** to prevent the flow of the defrosting refrigerant into the liquid header **42.** A branch conduit **74** connects each liquid line from a point between the evaporator and the check valve **73** to a condensate header **75** for conducting the defrosting refrigerant from line **43** to header **75.** Each branch conduit **74** is provided with a check valve **76** to preclude the reverse flow of defrosting refrigerant from header **75** into the liquid line **43** of any of the other evaporators operating in its normal refrigerating cycle which reverse flow would otherwise occur due to the pressure differentials as will hereinafter appear more fully. A conduit **77** having a pressure regulating valve **78** connects the condensate header **75** to the receiver **28** which, as hereinafter described, is maintained at a low pressure whereby the defrosting refrigerant will be returned directly to the refrigerating cycle regardless of its gaseous or liquid state. Valve **78** functions to maintain a constant upstream pressure (i.e. header **75**) when it is desirable to maintain a higher pressure in the suction header than is maintained in the receiver **28** and liquid header **42.** For example this higher pressure in condensate header **75** precludes the flow of liquid refrigerant into header **75** from one of the liquid lines **43** during normal refrigerating operation. However it has been found to be advantageous in many installations to provide appropriate valving between the liquid lines **43** and the header **75** and to simply allow the pressure in header **75** to drop to the pressure maintained in receiver **28** by eliminating the regulating valve **78.** By such arrangement, a very high pressure differential is created between the hot gas header **50** and header **75** thereby maximizing the rate of defrosting refrigerant flow through the evaporator to minimize the time required for accomplishing a complete defrost of the evaporator coils and related components such as fixture drip pans. Of course it is quite common for significant quantities of gaseous refrigerant to be returned with the defrosting refrigerant after passing through the evaporators to be defrosted and this is particularly true during the final stages of defrosting and with this high pressure differential. In the system of FIG. 1 this gaseous refrigerant will be returned to the receiver **28** where, as hereafter described, it will be disposed of rather than reintroducing such gaseous refrigerant into the liquid refrigerant being supplied to an evaporator where it can cause substantial malfunctioning in the normal refrigeration operation. In another operable arrangement a high side float valve is substituted for regulating valve **78** whereby the liquid defrosting refrigerant is allowed to return to receiver **28** rapidly at a low pressure but as gas starts to return the flow is restricted thereby raising pressure and temperature for the final stages of defrosting.

Referring again to the liquid refrigerant supply portion of the refrigeration system, the receiver **28** continually maintains a volume of liquid refrigerant adequate to meet the varying needs of the multiplicity of evaporators employed in this type of system as contrasted with a simple single evaporator system wherein a critical charge of refrigerant is provided in the system with no vessel corresponding to receiver **28** being required. As previously mentioned, the high side float valve **27** serves to feed only liquid refrigerant from the condenser **26** to the receiver **28** without regard for the pressure or level of refrigerant in receiver **28.** In accordance with this invention means are provided for continually evacuating receiver **28** to a predetermined reduced pressure and temperature and these means may include a suction conduit **80** connected from the top of receiver **28** to the suction of a separate compressor **81** with a pressure regulating valve **82** provided in conduit **80.** Valve **82** is of the type commonly known as an evaporator pressure regulating valve for controlling the upstream pressure, that is, the pressure in receiver **28** to a precise predetermined value. Compressor **81** operates continuously to evacuate gas from receiver **28** as required to maintain the pressure value as regulated by EPR valve **82** and, obviously, this gas evacuation causes a reduction in the temperature of the refrigerant remaining in receiver **28.** The discharge of compressor **81** is connected through conduit **83** to the accumulator oil separator **23** with a check valve **84** provided in conduit **83** to preclude reverse flow. Compressor **81** is relatively small compared to the system compressors **10** through **14** and is adapted to operate continuously. Compressor **81** is provided with the same type of oil return from separator **23** as heretofore described comprising an oil level sensor **36** operating a valve **35** to supply oil as required. Since compressor **81** is operating continuously and yet the suction load of evacuating gas from receiver **28** to the desired predetermined pressure may drop below the capacity of compressor **81,** as for example in low ambient conditions wherein the condensing pressure and temperature will be very low, I prefer to provide a suction conduit **85** with a check valve **86** connecting the suction of compressor **81** to the the suction conduit **54** leading to the suction side of the standard temperature compressors **10** and **11.** In this manner the capacity of compressor **81** may be employed continuously in a useful manner, whether partially in evacuating receiver **28** and partially assisting the standard temperature compressors or totally in evacuating receiver **28.**

The load on compressor **81** from the evacuation of receiver **28** will be comprised of the flash gas created by the liquid refrigerant passing from the high condensing pressure in condenser **26** through high side float **27** to the lower receiver pressure and the gaseous refrigerant returning from defrosting through conduit **77.** If the pressure regulating valve **78** is employed in conduit **77** to maintain a higher pressure in header **75** than in receiver **28** then the defrosting refrigerant may include some flash gas as caused by the pressure reduction through valve **78** as well as the usual quantity of gaseous refrigerant that is not condensed in passing through the evaporator, particularly in the final stages of defrosting.

The specific pressure and temperature maintained in receiver **28** may be established at different levels for different refrigeration systems depending on the requirements of the particular system but my invention contemplates that the temperature will be set at a level substantially midway between the evaporator temperature and the normal condensing temperature. However, when there are multiple evaporators operating at different temperatures from one receiver (as here) an average or compromise temperature is selected. Thus for example, without limiting this invention, a typical condensing temperature using an air cooled condenser may be **120**°F and I prefer maintaining receiver **28** at a temperature of 52°F with the warmest evaporator temperature being perhaps 10° to 20°F, and the coldest evaporator temperature being perhaps −40°F. While it is not uncommon for refrigeration systems to be provided with some form of liquid sub-cooling to reduce the temperature below the condensing temperature, this reduction is normally in the range of 10° to 15° which is all that is possible whereas in my system a temperature reduction of 70° or 80° or perhaps more is accomplished. While this desired reduction in temperature creates flash gas in receiver **28** due to the reduction in pressure, it likewise eliminates the creation of this same quantity of flash gas at the evaporator. As is well known flash gas is created at the expansion valve of the evaporator in every refrigeration system since the pressure is reduced from the liquid line pressure to the evaporator pressure and this evaporation into flash gas serves to lower the temperature of the reamining liquid from the liquid line temperature to the evaporator temperature. The quantity of flash gas created actually will depend on this temperature differential rather than the pressure differential. It is to be noted that such flash gas produces no useful work in the evaporator but rather is detrimental to the operation of the evaporator and in the cooling of the refrigerated fixture although the flash gas obviously would represent part of the load on the system compressors **10** through **14** since it must be returned and recompressed. Of course the flash gas also represents part of the volume that must be conducted by the suction conduits and, in its liquid form, part of the liquid volume that must be conducted by the liquid supply lines whereby such piping must be sized to accommodate those volumes. However any portion of the flash gas usually produced at the expansion valve that can be eliminated upstream thereof will allow a corresponding reduction in the necessary size of the liquid supply lines and suction lines and a resultant reduction in cost of installation.

However in addition to the foregoing benefits of this arrangement an even more significant improvement in the refrigeration system is in the improvement of the overall efficiency. The recompression by compressor **81** of the flash gas created in receiver **28** by reason of the pressure reduction from the condensing pressure is performed at a smaller pressure differential and with a much higher suction pressure than when the flash gas is created in the evaporators. The efficiency of the compression of gaseous refrigerant depends on the actual pressure of the refrigerant in that the higher the pressure the larger the quantity by weight of refrigerant that will be compressed in each cycle of th compressor. Thus compressor **81** is operated at a highly efficient level far above the level of a compressor in any conventional refrigeration system and far above that of compressors **10** through **14** of the system of FIG. **1**. The magnitude of the increase in efficiency of the overall system will depend on many factors and an example will be given hereinafter.

In the installation and operation of a refrigeration system as shown in FIG. **1** a variety of different pressure settings may be selected for the various regulating valves **66, 68, 78,** and **82** depending on a number of factors such as the particular refrigerant selected and the common ambient conditions encountered. For purposes of illustration and without limiting the scope of this invention, the system of FIG. **1** will now be described in connection with typical pressure settings that might be selected when using refrigerant R-502 in a supermarket refrigeration system. The pressure regulating valve **82** may be set for 100 psi with a resultant temperature of the liquid refrigerant in the receiver **28** of approximately 52°F. A desirable setting of the hot gas defrosting pressure regulating valve **66** might be 215 psi whereby the defrosting gaseous refrigerant, after desuperheating by desuperheater **70,** would be slightly over 100°F, perhaps 110°F to avoid any actual condensing by desuperheater **70.** In turn the regulating valve **68** would be set a few pounds higher than valve **66,** perhaps 218 psi. Finally if the regulating valve **78** for controlling the pressure in condensate **75** is desired, for the reasons discussed above, it might be set to maintain a pressure of perhaps 185 psi whereby a constant pressure differential of 30 psi is created between hot gas header **50** and condensate header **75** for causing the flow of defrosting refrigerant through the evaporators. Of course if valve **78** were eliminated as discussed above, the pressure in condensate header **75** would be allowed to drop to 100 psi (the pressure in receiver **28**) whereby the pressure differential would be 115 psi. In either case this predetermined pressure differential results in a predictable and consistent defrosting refrigerant flow through the evaporators whereby an appropriate defrosting time may be selected for each evaporator and a consistent defrosting will result under all ambient conditions although it will be appreciated that the time duration for defrosting a given evaporator will be reduced in relation to the magnitude of the pressure differential established between headers **50** and **75.** In a supermarket installation of this a typical suction pressures on the three levels of compressors for this arangement might be as follows: 41 psig on standard temperature compressors **10** and **11,** 12 psig on low temperature compressors **12** and **13** and 0 psig on the ultra-low temperature compressor **14.**

Thus it may be seen that in the system of FIG. **1** the liquid refrigerant will be at a constant temperature and the pressure differential across the expansion valve **45** of a given evaporator will be substantially constant under all ambient conditions, which pressure differential will be approximately 59 psi for the standard temperature evaporators, 88 psi for the low temperature evaporators and 100 psi for the ultra-low temperature evaporators. As will readily appear to those skilled in the art this will improve the ability to select and properly set the expansion valves and will greatly improve their performance. The quantity of flash gas produced in each evaporator is substantially reduced by this system over that which would occur in a conventional system. Of course the proportion of flash gas created in an evaporator will vary depending on the pressure maintained in the evaporator due to the pressure differential between the **and** psig liquid refrigerant pressure an the evaporator pressure. Moreover by providing this constant and relatively low liquid refrigerant supply pressure in receiver **28** it is possible to permit the condensing or head pressure of the system to vary over a much wider range as dictated by ambient conditions including permitting the pressure to drop to otherwise unacceptably low pressures which results in a substantial reduction in the power consumption of the compressors. Of course the improved efficiency of the overall system obviously permits the selection of system compressors of smaller sizes at lower prices when a system of this type is designed and such reductions in sizes and prices far exceeds the size and price of compressor **81.**

As an example of the improved efficiency and reduced compressor capacity required for the system of FIG. 1, is a typical installation in a supermarket using the six compressors illustrated in FIG. 1 and substantially the requirements set forth in the preceding paragraph there was an actual compressor power requirement of approximately 57 horsepower. The substantially identical refrigeration system employing only compressors **10** through **14** and omitting the receiver pump out compressor **81** required approximately **80** horsepower for the compressors. In still another competing refrigeration system employing separate refrigeration cycles for the three different temperature levels **140** horspower was required to refrigerate the same fixtures in the same store and such competing system is highly respected and accepted in the industry as being relatively efficient. Comparing the efficiencies of the systems by comparing the quantity of refrigeration produced or the power consumed, under the same high condensing temperature conditions the system of FIG. 1 in this installation produced 7.3 btu's per watt while such competing system produced only 3.9 btu's per watt. Moreover an increased efficiency in produced refrigeration per unit of power of up to 30 percent will occur under low ambient conditions whereby low condensing temperatures may be used with the system of FIG. 1, which increase is not possible with such competing system or any others in which the head pressure must be maintained at elevated levels.

Referring now to FIG. 2, a modified form of the refrigeration system of FIG. 1 is illustrated with many of the components remaining the same or substantially so and being more fully disclosed in the description of FIG. 1. To the extent that the components of FIG. 2 correspond to or are substitutes for the like components of the system of FIG. 1, a like numeral in the "100" series will be employed and the description here of that component will be minimized. It is to be noted that the system of FIG. 2 does not include the element for accomplishing hot gas defrosting of the evaporators but it will readily appear to those skilled in the art that the defrosting arrangement of FIG. 1 or any comparable arrangement may and would normally be employed therein. Compressors **110** and **111** are the standard temperature compressors having their discharges **122** connected to the accumulator and oil separator **123** with their suction sides **121** connected to both the suction conduit **154** from the evaporators and the conduit **120** connected to the discharge sides **119** of the low temperature compressors **112** and **113.** Again an ultralow temperature compressor **114** has its discharge **116** connected through conduit **117** to the suction sides **118** of the low temperature compressors **112** and **113.** The suction side of ultra-low temperature compressor **114** is connected through conduit **156** to suction header **153.** The suction side of low temperature compressors **112** and **113** is connected through conduit **155** to low temperature suction header **152.** Conduit **154** is connected from the suction sides of the standard temperature compressors to the suction header **151** for the standard temperature evaporators. The evaporators **144** and the associated controls remain the same as described with respect to FIG. **1.** The system compressors are compounded with the entire discharge passing into the vessel of separator **123** and out through the standpipe **124,** through conduit **125** to condenser **126** and then to receiver **128.**

In this embodiment of FIG. **2,** receiver **128** remains at condenser head pressure and the liquid refrigerant is supplied through conduit **140** to a pressure regulating valve **141A** which functions to maintain a reduced and constant downstream pressure. Valve **141A** is of the type commonly known in the industry as a crank case pressure regulator or CPR valve. The predetermined pressure maintained by valve **141A** is selected to be lower than the lowest condensing or head pressure permitted by the system condenser and its controls under the most extreme conditions and yet higher than the highest evaporator suction pressure such as produced by standard temperature compressors **110** and **111.** In a typical supermarket installation this valve **141A** may be set to maintain a downstream pressure of 100 psi with a resultant liquid temperature of 52°F, the same as maintained in receiver **28** of the system of FIG. **1.** The reduction in pressure by valve **141A** will produce flash gas in the manner heretofore described and therefore a separating vessel or chamber **160A** is provided immediately downstream of the valve for allowing the gaseous and liquid refrigerant to separate. The flash gas is removed through a conduit **161A** connected to the suction side of a small separate compressor **181A** similar in its function to compressor **81** of the system of FIG. **1.** Means are provided for controlling the removal of the flash gas through conduit **161A** to avoid either the excess accumulation of flash gas in chamber **160A** or the inadvertent return of liquid refrigerant to the suction of compressor **181A** and, as shown in the drawings, these means may include a float control **162A** provided in the chamber **160A** and connected to and controlling the operation of a valve **163A** in conduit **161A.** The controlled operation of valve **163A** by float control **162A** may be either an on-off switching between high and low levels of liquid in chamber **160A** or a modulating control directly related to the level of liquid in the chamber. It is to be noted that float control **162A** may also serve as an indicator or alarm for the occurrence of a below normal supply of liquid refrigerant, as for example due to the loss of refrigerant through a leak, since chamber **160A** will always be partially filled with liquid refrigerant unless there simply is no additional liquid refrigerant available from receiver **128.** Thus it may be seen that the pressure and temperature reduction of the liquid refrigerant by the regulating valve **141A** form the pressure and temperature in receiver **128** to the desired lower level is accomplished automatically and continually without employing a subcooler or the like and yet the resultant flash gas is immediately removed and returned directly to the refrigeration system by compressor **181A** which has its discharge connected through conduit **183** to the oil separator **123.** A header **142A** is connected to chamber **160A** and serves as a refrigerant supply for the standard temperature evaporators.

A second pressure regulating valve or CPR valve **141B** is provided in a conduit connecting header **142A** to a second chamber **160B** for reducing the pressure and temperature of the liquid refrigerant from the values existing in header **142A** to a temperature more closely approaching the temperatures maintained in the low temperatures evaporators as for example the temperature of perhaps 16°F. Chamber **160B** has a conduit **161B** connected to another small separate compressor **181B** with the valve **163B** and float control **162B** for evacuating the flash gas created in the pressure reduction through valve **141B** and maintaining the reduced pressure and temperature. The discharge of compressor **181B** is connected through a conduit **183B** to conduit **183** thereby returning the refrigerant to the normal cycle. The bottom of chamber **160B** is connected to a liquid header **142B** for supplying liquid refrigerant to the low temperature evaporators at this further reduced temperature. Similarly another CPR valve **141C** is provided for reducing the pressure of the liquid refrigerant from header **142B** into a chamber **160C** and thence to a liquid header **142C** for supplying the refrigerant to the ultra-low temperature evaporators. Again the magnitude of the pressure and temperature reduction may be any amount desired but ideally it will approach a temperature reduction of one-half the difference between the upstream refrigerant temperature and the evaporator temperatures thereby again maximizing the efficiency of this reduction while minimizing the flash gas formed in the evaporators. A third separate compressor **181C** may be provided with its suction conduit **161C** connected through valve **163C** to the chamber **160C** with the valve being operated by control **162C**. The discharge of compressor **181C** is connected through conduit **183C** either directly or indirectly (as illustrated) to the oil separator **123** for returning the refrigerant to the normal cycle.

The series of reductions in pressure and temperature are illustrated in FIG. **2** as an example of the manner in which this invention may be employed to maximize efficiency in systems employing multiple evaporators operating at plural temperature and suction levels but it will be appreciated by those skilled in the art that in many smaller installations the gain in efficiency at each succeeding level may be sufficiently small as to render unwarranted the provision of the additional equipment. Moreover it will also appear to those skilled in the art that one or more of the successive pressure and temperature reductions, particularly at the lower levels, may approximate the suction pressure maintained by one of the system compressors whereby the suction line **161** of the particular separation chamber **160** may be connected to the suction of a system compressor to accomplish the improved efficiency without the need for an additional compressor at that level. It should be noted that FIG. **2** also illustrates the provision of the oil return system described with respect to FIG. **1** for each of the eight compressors illustrated in FIG. **2** by means of the same individual float controls **136** and valves **135**.

Referring now to FIG. **3**, another modified form of the system of this invention is illustrated in a greatly simplified form although it will be understood and readily appear to those skilled in the art that the variety of additional components described with respect to FIGS. **1** and **2** normally would be included in the system of FIG. **3**, such as the means for oil return to the compressors and for hot gas defrosting. Again insofar as the components of this system correspond directly to the components of the systems of FIGS. **1** and **2**, numerals in the "200" series will be employed for simplicity of description and correlation. Compressors **210** and **211** are the standard temperature compressors having their suction sides **221** connected through a conduit **254** to the suction header **251** that is connected to the standard temperature evaporators **244**, only two of which are illustrated although a multiplicity thereof will normally be provided. A conduit **220** also connects the suction sides **221** of the standard temperature compressors to a vessel **285** which serves the function, among other things hereinafter described, of a flash type inner-cooler as is well known in the art. The low temperature compressors **212** and **213** have their discharges **219** connected to the vessel **285**, preferably discharging the gaseous refrigerant into the lower portion of the vessel for bubbling through the liquid refrigerant present thereby reducing the temperature of the discharge gas. The suction sides **218** of the low temperature compressors are connected through a conduit **255** to the low temperature suction header **252** which is connected to the low temperature evaporators **244** illustrated in the left hand portion of the Figure.

Thus again in the system of FIG. **3** the compressors are arranged in compounded relationship and if ultra-low temperature evaporators are included then the compressor therefor may be compounded in the manner illustrated in FIGS. **1** and **2**. Standard temperature compressors **210** and **211** discharge the compressed refrigerant to condenser **226** and the condensed refrigerant flows through the high side float valve **227** to receiver **228** in the manner described with respect to FIG. **1**. As in the system of FIG. **1**, a separate compressor **281** continually draws a suction on receiver **228** through conduit **280** with the suction pressure being controlled by the EPR valve **282** whereby a constant pressure is maintained in the receiver, as for example 100 psig and 52°F for refrigerant **502**. The discharge from separate compressor **281** is combined with the discharge from the standard compressors supplied to the condenser **226**. Again I prefer to provide a conduit **285** having a check valve **286** connected between the suction sides of the standard temperature compressors and separate compressor **281** for the latter to assist the former when the load of evacuating the receiver **282** is below the capacity of compressor **281**. The liquid refrigerant for the standard temperature evaporators (illustrated on the right hand side of FIG. **3**) is supplied directly to those evaporators from receiver **228** through conduit **240** and header **242** in the manner illustrated and described with respect to FIG. **1**. However the liquid refrigerant supply for the low temperature evaporators is supplied from the inter-cooler vessel **285** through conduit **240A** and header and header **242A** at a temperature and pressure as maintained in vessel **285** by the standard temperature compressors **210** and **211** and in the context of the heretofore described systems, this temperature may be approximately 10°F. Thus it will be seen that very little flash gas will be created in the expansion of refrigerant into the low temperature evaporators thereby minimizing the inefficiencies and problems heretofore discussed. The supply of liquid refrigerant for vessel **285** is provided through conduit **286** from receiver **228** through a float valve assembly **287** whereby an adequate supply of liquid refrigerant is maintained at all times in vessel **285** which thereby serves as a receiver for supplying the multiplicity of low temperature evaporators to accommodate their fluctuating loads. The float valve assembly may be of any convenient type such as a float sensor **288** operating a valve **289** to admit liquid refrigerant as needed to maintain the desired refrigerant level. The system of FIG. **2** provides the basic efficiencies described with respect to FIG. **1** but adds the additional efficiencies and advantages with respect to supply refrigerant of an even further reduced temperature to the low temperature evaporators similar to that which is provided by the system of FIG. **2** but by merely employing the suction of the standard temperature compressors rather than a second separate compressor.

In FIG. **4** another modified form of the system of this invention is illustrated in a greatly simplified form and again insofar as the components of this system correspond to those of the preceding Figures numerals in the "300" series will be employed. The system of FIG. **4** combines three completely separate, closed cycle, refrigeration systems which may comprise the standard temperature system, the low temperature system and the refrigeration system accomplishing the invention hereof. It will be understood and readily appreciated by those skilled in the art that this concept may be applied to more than the two fixture refrigerating systems illustrated and that such plural systems need not be operated at different temperature levels. The discharge from the standard temperature compressor is supplied to a condenser **326A**, receiver **328A** and thence through a conduit **340A** to a liquid header **342A** for supplying the evaporators **344A**. A second refrigeration system, for example a low temperature system, employs a compressor **312** discharging refrigerant to a separate condenser **326B** and receiver **328B** and thence through conduit **340B** to a header **342B** connected to the low temperature evaporators **344B**. Between the receivers and liquid headers of these conventional systems, a heat exchanger **390** of any convenient type is provided for cooling the liquid refrigerant to a predetermined level before it is conducted to the headers **342A** and **342B**. The cooling in heat exchanger **390** is accomplished by an evaporator **391** having its suction connected to a separate compressor **381** for maintaining the desired temperature in heat exchanger **390**. The discharge of compressor **381** is connected through another separate condenser **326C** to a receiver **328C** and thence through an expansion valve **392** to the evaporator **391**. In this manner again it is contemplated by his invention that the temperature reduction in the liquid refrigerant supplied to the multiplicity of fixture evaporators through the headers **342A** and **342B** will be very substantial with the resultant temperature more closely approaching the midpoint between the evaporator temperatures and the condensing temperatures and significantly below room temperature. Thus while heat exchanger **390** merely seems similar to a conventional subcooler, its function in accomplishing this extreme temperature reduction renders it substantially different. The compressor **381** operates at a highly efficient level in that evaporator **391** is maintained at this desired intermediate temperature. For example, in the context of the previously described systems, the compressor **381** and evaporator **391** together with their usual controls will serve to produce a liquid refrigerant temperature of approximately 50°F as supplied to headers **342A** and **342B**. Although the pressure of the liquid refrigerant in headers **342A** and **342B** will not be reduced as was accomplished in the systems of FIGS. **1, 2** and **3**, the temperature differential between the header and the evaporator is similar and therefore the quantity of flash gas that will be created is similarly minimized. The system of FIG. **4** is particularly advantageous when applied to either a previously installed system requiring additional capacity or in an overall system employing two or more separate refrigeration cycles (as illustrated) wherein different types of refrigerant are used in the separate systems due to the different advantageous characteristics of the refrigerants. In either situation the heat exchanger **390** and the separate refrigeration cycle including separate compressor **381** may be readily sized and installed to achieve the desired objectives. In actual practice, the capacity of an existing system of compressors can be increased by 40 to 70 percent by the addition of a compressor **381** of merely 10 to 15 percent of the total capacities of the existing compressors, whereby fixtures may be readily added to the installation.

FIG. **5** illustrates still another modified form of the system of this invention in a simplified form comprising a single refrigeration system for operating a multiplicity of evaporators at a given temperature level. Numerals in the "400" series will be employed for identifying components similar to those appearing in the preceding Figures. Compressor **410** has its discharge **422** connected to an oil accumulator and separator **423** which in turn is connected through a standpipe **424** to a condenser **426** and thence to a receiver **428**. Liquid refrigerant is supplied through a conduit **440** and a subcooler **490** to a liquid header **442** and thence to evaporators **444**. A suction conduit **454** is connected from the evaporators **444** to the suction of compressor **410** to complete the refrigeration cycle. While only two evaporators **444** are illustrated it is contemplated and this system is adapted to accommodate a multiplicity of evaporators operating at this suction pressure level. The sub-cooler **490** has an evaporator coil **491** with refrigerant supplied through expansion valve **492** from conduit **440** and the suction from the coil is connected through conduit **480** to a separate compressor **481**, the discharge of which is connected by conduit **483** to the oil separator **423**. It is to be noted that compressors **410** and **481** are provided with an oil return system of the type shown and described with respect to FIG. **1** employing valves **435** and level controls **436** for returning the oil from separator **423**. Again it is contemplated by this invention that the liquid refrigerant temperature reduction by sub-cooler **490**, as operated by compressor **481**, will be of an extreme magnitude many times that which is normally accomplished by a sub-cooler. The temperature reduction is performed at an efficient level in that compressor **481** operates with a relatively high suction pressure in the manner previously described with respect to the other systems. Most of the advantages and improvements and efficiencies are the same with this system as previously described. As an example of the improved efficiency accomplished by this invention as applied to a system of the type illustrated in FIG. **5**, the addition of an appropriately sized compressor **481** and sub-cooler **490** to a single staged system using Refrigerant-**502** operating at a −40°F suction and 120°F condensing temperature will increase the overall efficiency by 50 percent by reason of the two compressors producing approximately 69 percent more work although the separate compressor **481** has a horsepower requirement of only 16 percent of the system compressor **410.**

Having fully described my invention in connection with selected embodiments and arrangements by way of examples, it is to be understood that my invention is not limited to such specific embodiments and arrangements as disclosed herein but rather is of the full scope of the appended claims.

I claim:

1. In a closed cycle refrigeration system for a multiplicity of refrigerated fixtures, comprising a like multiplicity of evaporator means associated with and cooling the fixtures, compressor means for compressing the evaporated refrigerant, condenser means for condensing the refrigerant to a liquid at a temperature substantially above room temperature, a receiver for maintaining a supply of liquid refrigerant for the variable demand of said multiplicity of evaporator means, a separate vessel forming a chamber for receiving the liquid refrigerant from said receiver, a pressure reducing valve between said receiver and said chamber, and means employing the suction of said compressor means at a relatively high pressure level connected to said chamber to evaporate some refrigerant for reducing the temperature of all the liquid refrigerant in said chamber being supplied to said evaporator means to a temperature below both the condensing temperature and above the temperature maintained by said evaporator means.

2. In a closed cycle refrigeration system for a multiplicity of refrigerated fixtures, comprising a like multiplicity of evaporator means associated with and cooling the fixtures, compressor means for compressing the evaporated refrigerant, condenser means for condensing the refrigerant to a liquid at a temperature substantially above room temperature, a receiver means for maintaining a supply of liquid refrigerant for the variable demand of said multiplicity of evaporator means including a chamber through which all of the condensed refrigerant passes, means employing the suction of said compressor means at a relatively high pressure level connected to said chamber to evaporate some refrigerant for reducing the temperature of all the liquid refrigerant in said chamber being supplied to said evaporator means to a temperature below both the condensing temperature and room temperature and above the temperature maintained by said evapoartor means, means provided for defrosting the evaporator means by passing hot gaseous refrigerant therethrough, and means for conducting the defrosting refrigerant from the evaporator means to said chamber for separating gaseous and liquid refrigerant and returning the liquid refrigerant to the refrigerating cycle.

3. In a closed cycle refrigeration system for a multiplicity of refrigerated fixtures, comprising a like multiplicity of evaporator means associated with and cooling the fixtures, compressor means for compressing the evaporated refrigerant, condenser means for condensing the refrigerant to a liquid at a temperature substantially above room temperature, a receiver for maintaining a supply of liquid refrigerant for the variable demand of said multiplicity of evaporator means, a heat exchanger through which the liquid refrigerant passes from said receiver to said evaporator means, and means comprising a completely separate closed cycle refrigeration system having a compressor, condenser, receiver and evaporator coil with the evaporator coil forming one separate portion of the heat exchanger, said separate compressor operating at a relatively high suction pressure level for reducing the temperature of all of the liquid refrigerant passing through the heat exchanger and being supplied to said system evaporator means to a temperature below both the condensing temperature and room temperature but above the temperature maintained by said evaporator means.

4. In a closed cycle refrigeration system for a multiplicity of refrigerated fixtures, comprising a like multiplicity of evaporator means associated with and cooling the fixtures, compressor means for compressing the evaporated refrigerant, condenser means for condensing the refrigerant to a liquid at a temperature substantially above room temperature, a receiver for maintaining a supply of liquid refrigerant for the variable demand of said multiplicity of evaporator means, and means employing said compressor means both for reducing the pressure of the supply of liquid refrigerant to a constant value substantially below the head pressure produced by said compressor means and for reducing the temperature of all the liquid refrigerant being supplied to said evaporator means to a temperature below both the condensing temperature and room temperature and above the temperature maintained by said evaporator means.

5. The refrigeration system of claim **4** wherein the means for reducing the liquid refrigerant temperature includes a chamber through which all of the condensed refrigerant passes, and said suction of the compressor means is connected to said chamber for causing said evaporation of some refrigerant to cool the remaining liquid.

6. The refrigeration system of claim **4** wherein said temperature reduction of the liquid refrigerant is approximately one-half of the temperature differential between condensing temperature and evaporator temperature.

7. The refrigeration system of claim **4** wherein said compressor means includes a separate and independently operating compressor having its suction operating said means for reducing the liquid refrigerant temperature.

8. The refrigeration system of claim **7** wherein said means for reducing the temperature and pressure of the liquid refrigerant includes the receiver with said separate compressor suction connected to the receiver, means for passing only liquid refrigerant from the condenser means to the receiver, and means for controlling the pressure in said receiver to said constant value.

9. In a closed cycle refrigeration system for a multiplicity of refrigerated fixtures, comprising a like multiplicity of evaporator means associated with and cooling the fixtures, compressor means for compressing the evaporated refrigerant, condenser means for condensing the refrigerant to a liquid at a temperature substantially above room temperature, a receiver for maintaining a supply of liquid refrigerant for the variable demand of said multiplicity of evaporator means, and means employing the suction of said compressor means at a relatively high pressure level to evaporate some refrigerant for reducing the temperature of all the liquid refrigerant being supplied to said evaporator means to a temperature below both the condensing temperature and room temperature and above the temperature maintained by said evaporator means, said the multiplicity of evaporator means being operated on at least two different temperature levels and the means for reducing the liquid refrigerant temperature including at least two separate chambers through which the condensed liquid refrigerant passes consecutively, said suction of the compressor means being connected to said chambers for causing evaporation of some refrigerant to cool the remaining liquid refrigerant to two different temperatures, the higher temperature being maintained in the first chamber with the liquid refrigerant therefrom being supplied to the higher temperature evaporator means and to the second chamber, and the colder liquid refrigerant from said second chamber being supplied to the remaining evaporator means.

10. The refrigeration system of claim **9** wherein the compressor means includes a separate and independently operating compressor having its suction connected to said first chamber.

11. The refrigeration system of claim **10** wherein said compressor means includes separate compressor operating the evaporator means at different temperature levels, and compressor operating said higher temperature evaporator means has its suction also connected to said second chamber.

12. The refrigeration system of claim **10** wherein the compressor means includes a second separate and independently operating compressor having its suction connected to said second chamber.

13. The refrigeration system of claim **12** wherein a third chamber and third separate compressor operating same are provided and connected downstream of said second chamber for providing even colder liquid refrigerant to the evaporator means operating at the coldest temperature.

14. In a refrigeration system having compressor means, condenser-receiver means and evaporator means with a substantial temperature range differential between the high temperature of the liquid refrigerant resulting in said condenser-receiver means and the low refrigerant temperature maintained in the evaporator means, the improvement comprising, a separate and independent compressor means operating at a relatively high suction pressure for evaporating refrigerant at a temperature intermediate such temperature range differential, and means employing the suction of said separate compressor means in association with the high temperature liquid refrigerant for reducing the temperature and pressure of that refrigerant to substantially said intermediate temperature and to a substantially constant pressure before supplying same to the evaporator means.

15. In a refrigeration system the combination of a first compressor means, a condenser, a receiver, and a plurality of evaporator means all connected in a closed cycle with the liquid refrigerant resulting in said condenser normally having a high temperature substantially above room temperature and said evaporator emans operating at temperatures near or below freezing, valve means connected between said condenser and said receiver for permitting only liquid refrigerant to flow to said receiver, a separate continuously operating compressor means having the suction connected to said receiver for producing a relatively high suction pressure and refrigerant evaporation temperature for reducing the temperature and pressure of the liquid refrigerant in said receiver to below room temperature and to a substantially constant pressure, and said separate compressor means having its discharge connected to said condenser for returning the refrigerant to said closed cycle.

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