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3,108,453

REFRIGERATING APPARATUS INCLUDING HEAT EXCHANGE STABILIZER MEANS

Filed Aug. 5, 1959

4 Sheets-Sheet 1

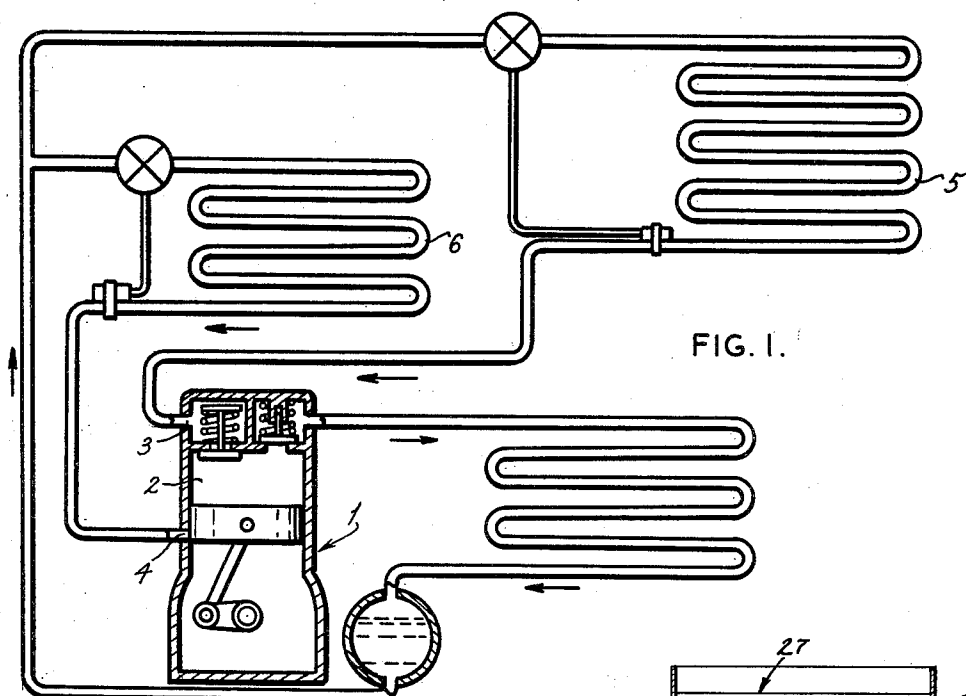


FIG. 1.

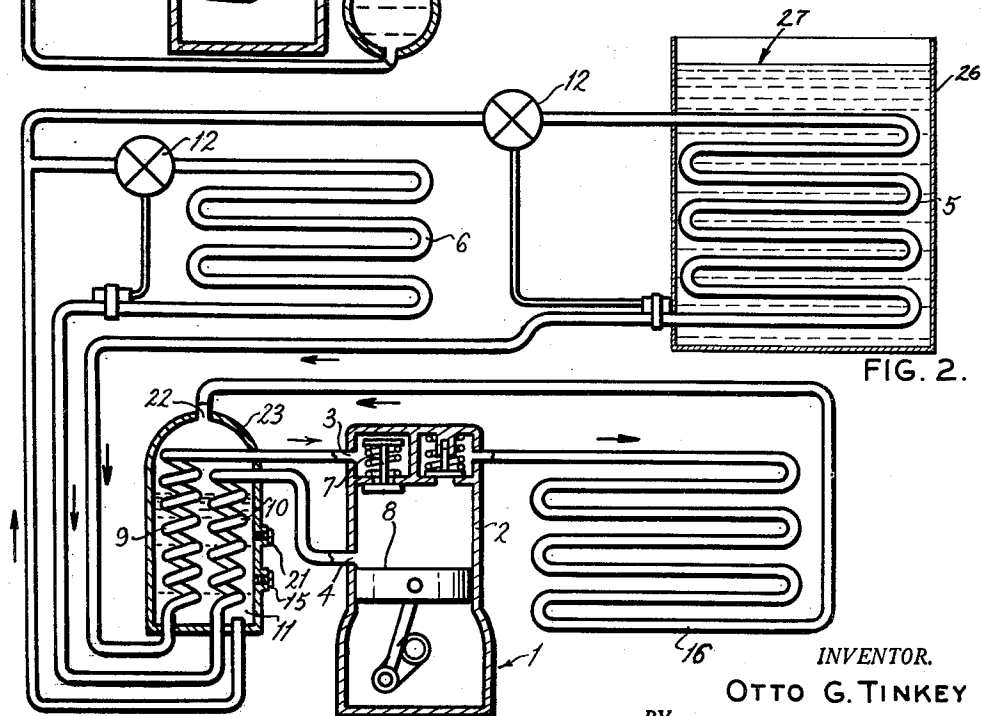


FIG. 2.

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4 Sheets-Sheet 2

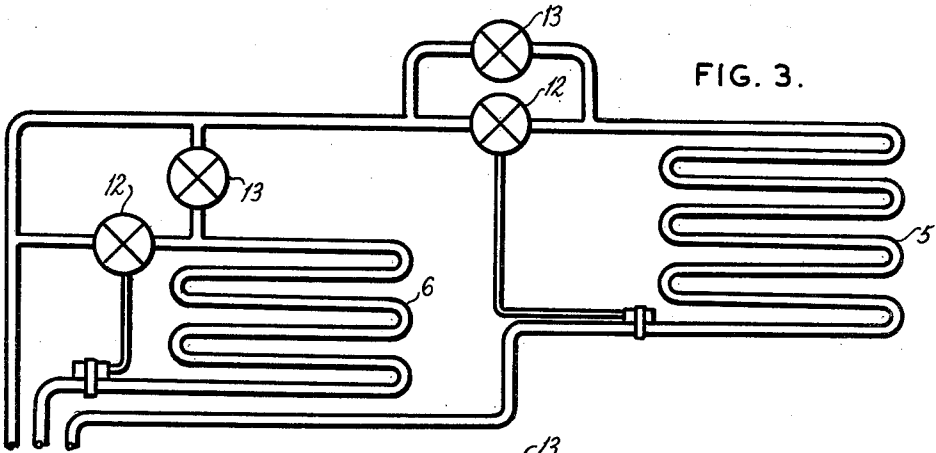


FIG. 3.

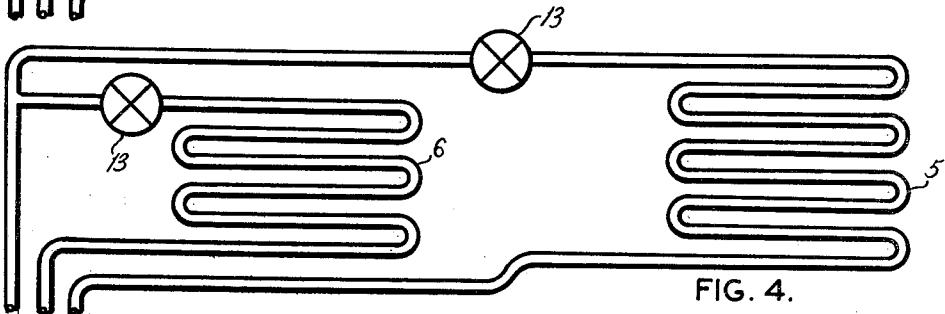


FIG. 4.

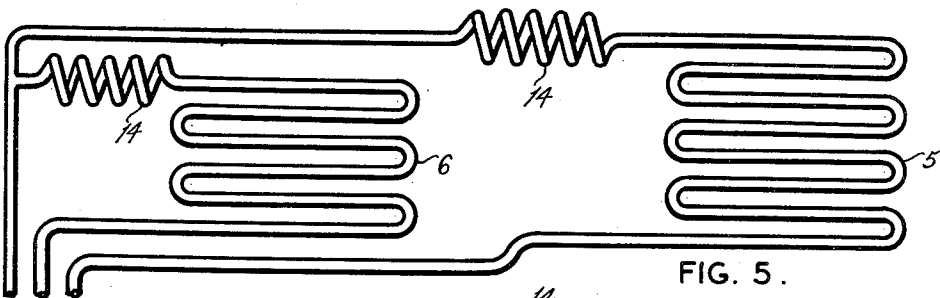


FIG. 5.

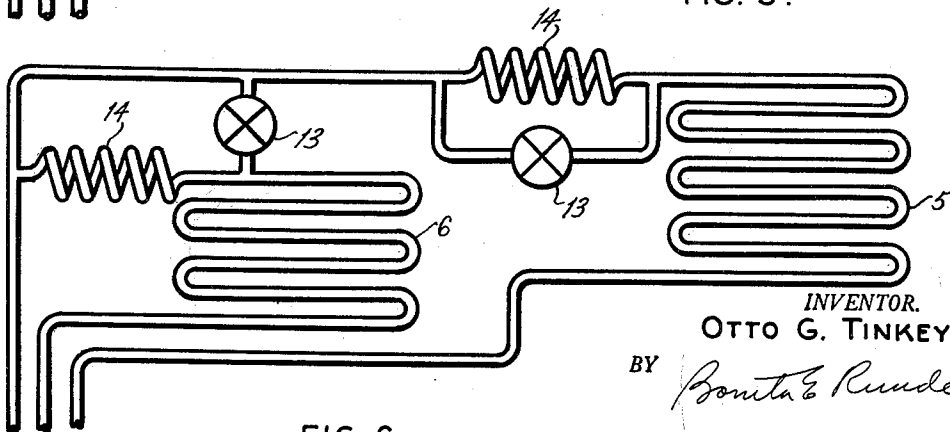


FIG. 6.

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FIG. 7.

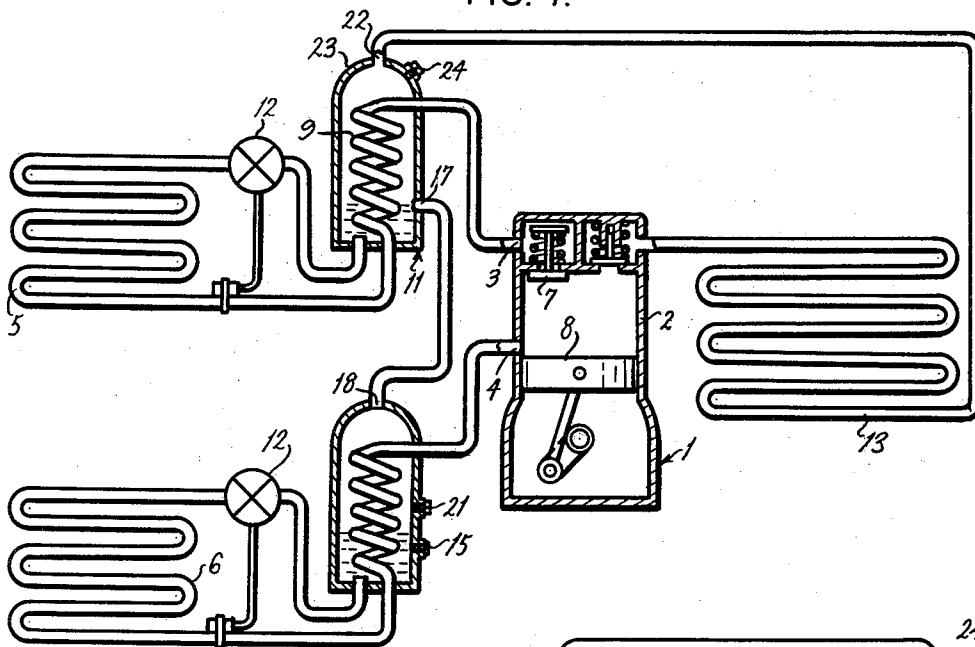
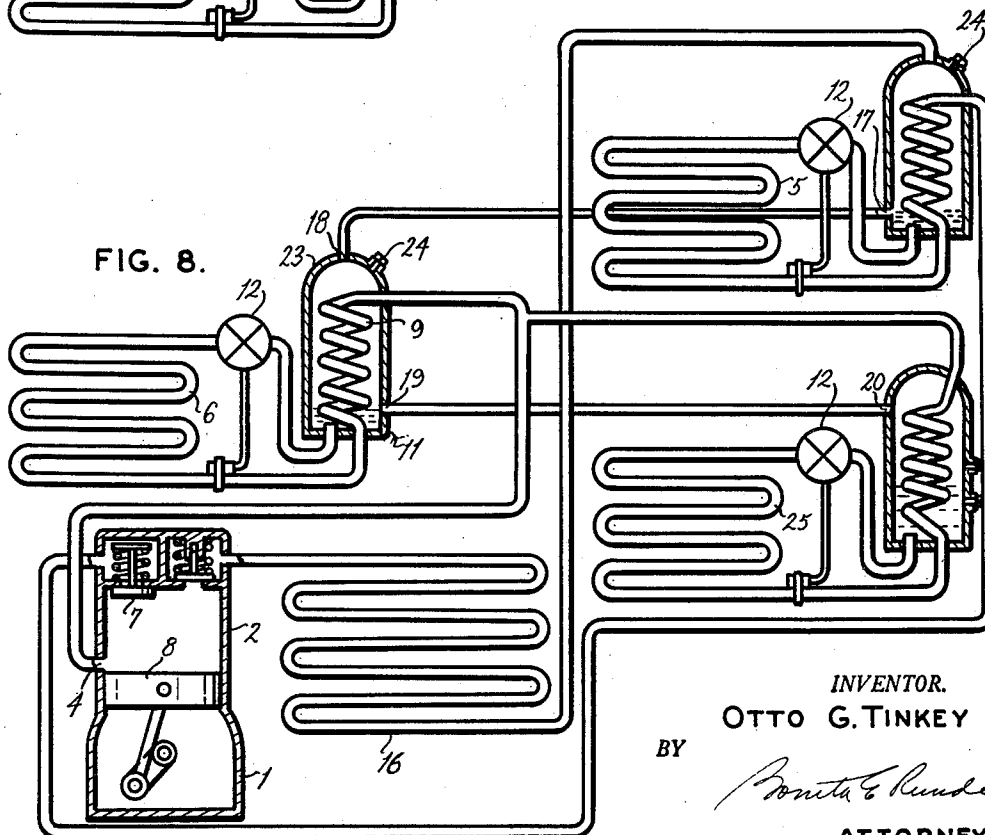


FIG. 8.



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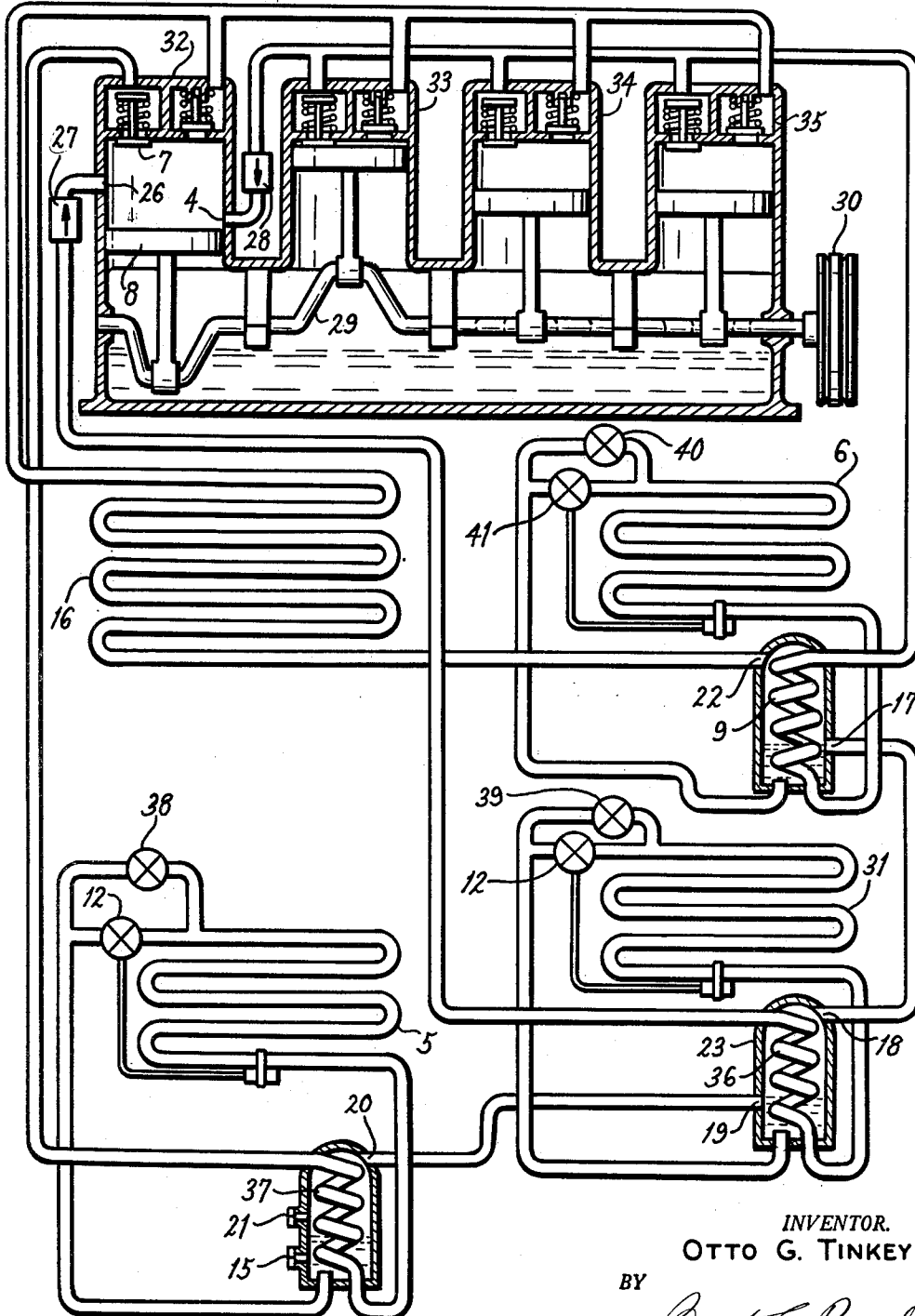


FIG. 9.

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REFRIGERATING APPARATUS INCLUDING HEAT EXCHANGE STABILIZER MEANS

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5 Claims. (Cl. 62-200)

The invention here presented is broadly in the field of refrigeration apparatus and more specifically, a self-regulating heat exchange system for hold-over solutions, mobile air conditioning and/or refrigeration systems, two temperature refrigeration systems and the like.

An object of this invention is to make simple or multiple cylinder or single and multiple-effect compressors practical and trouble free while fully automatic in operation, by eliminating the almost inevitable liquid refrigerant and oil "slugging" of the compressor.

An object is to provide means to control one or more temperatures of a multiple temperature refrigeration system independently of each other and regardless of the load on either the high or low side.

Another object is to pre-cool the liquid refrigerant going into the coils.

Another object is to warm the cold refrigerant returning from the cooling coils into the compressor.

Still another object is to hold a reserve of liquid refrigerant which is moving into the cooling coils, when using a restrictor expansion device, in order to slop over through the said cooling coils when the cooling load on the evaporator is reduced or to reduce the head pressure caused by extra high ambient temperatures.

Other objects and details of the invention will be apparent from the following description, when read in connection with the accompanying drawings wherein:

FIG. 1 is a diagrammatic drawing of a two-temperature refrigeration system;

FIG. 2 is a schematic drawing of a two-temperature system as set forth by the applicant;

FIG. 3 is a section of FIG. 2, showing the addition of automatic expansion valves;

FIG. 4 is a section of FIG. 2, but without the thermostatic expansion valve;

FIG. 5 is a section of FIG. 2, but using capillary tubes instead of automatic expansion valves;

FIG. 6 is the same as FIG. 5, with the addition of automatic expansion valves;

FIG. 7 shows two single coil stabilizers in series;

FIG. 8 shows three stabilizers connected in series;

FIG. 9 is a diagrammatic drawing of a multiple temperature refrigeration system showing the use of a multiported, multi-cylinder compressor.

There is a great consumer demand for two small refrigerators operating simultaneously, with one operating at just above freezing temperature and the other operating at below freezing temperature. Presently, some manufacturers use a separate complete condensing unit for each temperature refrigerator, thus entailing higher first cost, and higher operating costs, in addition to extra electrical, friction, and other condensing unit losses inherent to said extra equipment.

Other manufacturers use what is known in the art as secondary refrigerants, which usually requires building the freezer and refrigerator together, with the freezer on top, which is the least desirable position in the minds of most users. Still other manufacturers use various methods of holding back the above 32 degrees temperature, i.e., by means of suction pressure valves, weighted check valves, by adjustment of the refrigerant charge so it just "slops over" sufficiently to maintain an above 32 degrees temperature in the high temperature refrigera-

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tor and other methods, in an attempt to maintain a higher and lower temperature.

Practically all of these methods of using one compressor unit to maintain two different temperatures in refrigerators, greatly reduce the refrigerating capacity of the compressor when it is working on the higher temperature, by making said compressor operate at the reduced capacity of the suction pressure which goes with the low temperature, all of the time it is refrigerating the high temperature. Because the compressor must operate at times at high pressure, it cannot be proportioned for maximum efficiency at low temperature operation and accordingly, must operate at the lesser efficiency of the lower temperature.

H. J. Macintire in his "Handbook of Mechanical Refrigeration" on page 56 states that "For general refrigeration, the multiple-effect compression has not been popular because of the difficulty in adjusting the loads."

In an attempt to solve this problem, I connected a refrigeration system, as shown in FIG. 1, using a compressor 1, having a cylinder 2, which has a valve inlet port, shown at 3. I ported this cylinder, additionally, as shown at 4, thus making two inlet ports. This compressor takes a full charge of suction refrigerant into the cylinder from a low temperature evaporator 5, through the valve port 3. On uncovering the port 4, the higher back pressure suction gas from the high temperature evaporator 6, connected to the port 4, enters the latter, thereby "supercharging" the compressor. The compressor, therefore, operates at all times on the high efficiency of the higher suction gas pressure, yet does not lose any efficiency pumping the lower pressure gas from the lower temperature evaporator. However, I found such a system impractical because sooner or later, a pocket of liquid refrigerant and/or oil formed in the evaporators during the idle compressor periods. When the compressor started up on a running cycle, the refrigerant-oil "slug" would be drawn into the gas compressor and would blow a head gasket, spring a crank-shaft etc. or at least, crack a compressor intake valve. This compressor damage happened so frequently, it made the use of the unit entirely undependable.

I, then, developed the embodiment, as shown in FIG. 2, which has made the supercharged compressor entirely feasible and dependable, maintaining a temperature near 0 degrees F. in the freezer and an average of 35 degrees F. in the bottom portion to 45 degrees F. in the top portion of the above-freezing refrigerator.

In the embodiment, as shown in FIG. 2, the compressor 1 may have as many cylinders as desired, or if desired, two compressors may be used. A single cylinder 2 is shown for simplicity of description, having an inlet port 3. The compressor is fed refrigerant from the lower-temperature evaporator 5 through a compressor inlet valve 7, and, as a piston 8 reaches its bottom position, it uncovers a second port 4 in the cylinder, through which the refrigerant from the higher temperature coils 6 is admitted, thus supercharging the compressor cylinder and making it work at high efficiency at all times. A suction line from an evaporator 5 passes through its coil 9 in a stabilizer 11, and a suction line from an evaporator 6 also passes through its own coil 10 in the stabilizer. This double coil stabilizer makes possible the control of either or both temperatures of a two-temperature refrigeration system, independently of each other and regardless of the load on either the high or low side as hereinafter described.

If the temperature of either refrigerator goes too low before the unit shuts off at a pre-determined setting of a desired temperature in one refrigerator, the too-cold evaporator, or both evaporators, can be held at any minimum

temperature desired, by replacing a thermostatic admission valve 12, as shown in FIG. 2, with an automatic pressure operated expansion valve 13, shown in FIG. 4, which may be set to maintain the minimum allowable pressures in their respective evaporators. Restrictor tubes 14, shown in FIG. 5, may also be replaced with a bleeder type automatic pressure operated expansion valve. When the evaporator refrigerant pressure drops to the pressure for which its automatic valve is set, the automatic valve opens to feed its respective coil sufficient refrigerant to maintain the pressure for which the valve is set.

Better over-all refrigeration efficiency will result, if, instead of replacing thermostatic expansion valves and restrictor tubes with automatic expansion valves, an automatic expansion valve 13 is connected across a thermostatic expansion valve 12, as in FIG. 3, or, across a restrictor tube 14, as shown in FIG. 6. This allows the automatic expansion valve to supplement the thermostatic expansion valve or capillary or restrictor tube.

As soon as an entire evaporator coil drops below the desired temperature and corresponding refrigerant pressure for which its respective automatic expansion valve is set, said valve opens and any excess unevaporated liquid refrigerant, not required to cool the evaporator, "slops over" into its respective stabilizer coil, in the stabilizer, as shown at 11. The stabilizer shell 23, surrounding said coil or coils, is connected to a condenser 16 in such a manner as to act as a refrigerant receiver and in addition have space for refrigerant gas above the liquid refrigerant. The connection between the discharge end 22 of the condenser and the inlet to the stabilizer shell, must be of sufficient size to pass all of the liquid from the condenser and also pass uncondensed refrigerant gas from the condenser into the stabilizer. Therefore, any unevaporated refrigerant from the evaporator passing into said stabilizer coil 9 is immediately evaporated, thereby eliminating liquid refrigerant "slugging" of the compressor. Also, by supplementing the condenser with unevaporated refrigerant, at the point of contact between the uncondensed gas and the liquid refrigerant, the head-pressure is lowered with an air-cooled condenser or its water-tower equivalent. Or, water is saved, where cooling water is controlled by a pressure operated water valve, wasting water from a water pressure system.

The correct height of the refrigerant in the stabilizer is readily indicated by the two trycocks 15 and 21, or by gauge glass, sight glasses, float indicators or other such means. Trycocks are shown in the preferred embodiment because the topmost trycock can be used as a non-condensable gas purge for the refrigeration system. A trycock or other valving means can be used with sight glasses for a non-condensable gas purge.

Since the stabilizer shell is the liquid refrigerant receiver, all of the heat extracted from the evaporators, plus most of the heat of compression is available, because of the fact that the stabilizer is connected with the condenser 16, as shown in FIG. 2. Therefore, plenty of the heat is available around the stabilizer coils to evaporate all the liquid refrigerant returning from the evaporators. Head pressure is lowered because of the additionally cool condensing medium, thereby effecting saving in operating costs. With sufficient heat exchange surface in the stabilizer evaporating coils, it is practically impossible to slug liquid refrigerant through said stabilizer coils into the compressor, and oil returning from the evaporator has any entrained liquid refrigerant centrifuged out by the stabilizer coils or conductors, and the oil is broken into spray or mist by said centrifugal action. Therefore, no compressor slugging occurs. This is not true in the so called heat exchanger, wherein the liquid refrigerant is brought into thermo-contact with the returning suction gas from the evaporator. The amount of heat available to evaporate any unevaporated liquid returning in the suction line is limited by the heat of the

liquid capacity of the condensed high pressure liquid refrigerant.

The necessity of lowering first costs and lessening maintenance has forced manufacturers to abandon expansion valves and use restrictor tubes wherever possible. When using a restrictor tube, it is necessary to note that the size of the latter, both in cross-section and length is critical in each particular refrigeration system. The amount of refrigerant charge is critical. Likewise, the restrictor tube and the refrigerant charge must be critically balanced for a pre-determined mean pressure. Lower head pressure will cause a build-up of liquid refrigerant in the condenser coils thereby starving the evaporator by not completely filling the same. Higher head pressures cause uncondensed refrigerant from the condenser to flow through the restrictor tube into the evaporator lowering the evaporator's capacity to pick up heat. This blow-through of uncondensed refrigerant, sometimes causes liquid refrigerant to slop over into the suction line, still further reducing the evaporator capacity.

Greater self regulation will result, when using a restrictor tube as a refrigerant metering device, by the use of the stabilizer. The refrigerant charge for a stabilizer system is adjusted to just fill the evaporator at a pre-determined mean head pressure with a regulated amount of liquid refrigerant remaining in the stabilizer receiver and the normal carry-over of refrigerating effect into the stabilizer by the gas returning from the evaporator. Lower head pressure will cause a build-up of liquid refrigerant in the stabilizer, thereby starving the evaporator by not completely filling the same, thereby reducing said carry-over of refrigerating effect into the stabilizer, thus reducing the stabilizer's supplementary cooling which raises the head pressure, which in turn assists in counter balancing the lowered head pressure. High head pressure causes liquid refrigerant from the stabilizer receiver to blow through into the evaporator coil. The blow through causes unevaporated liquid to slop over into the stabilizer coil, thus supplementing the condenser, thereby lowering the head pressure. Due to this action, lowering of the head pressure, a considerably greater condenser temperature rise is necessary to blow uncondensed refrigerant into the evaporator than is the case without the stabilizer; whereas, a conventional refrigeration system with a high pressure or a motor overload cut-out must (can only) stop the compressor, thereby completely stopping all refrigeration. The stabilizer system will keep running at a much higher condensing temperature. Although refrigeration effect will be reduced in the latter, it will not be completely stopped as would be the case in present day conventional refrigeration systems.

To minimize waste of refrigeration by the cold connection between the evaporator and its respective coil in the stabilizer, the latter should be placed as close to the evaporator as it is physically possible to arrange. Where the evaporators are separated at some distance from each other, the close connection between the evaporator and its stabilizer coils may be maintained with the same over-all effect by the use of single coil stabilizers with the shells connected as shown in FIG. 7. Three or more coils may be connected with their respective single coil stabilizers, as shown in FIG. 8. When using two or more stabilizers, the connections between them should tap the stabilizer shell nearest the condenser at the top of the liquid level in the first stabilizer condenser, as shown at 17, and discharge into its next in line stabilizer shell as shown at 18.

In succeeding stabilizers, the connections between them should tap the stabilizer shell nearest the condenser at the top of the liquid level, shown at 19, and discharge into its next in line stabilizer shell, as shown at 20. When using two or more stabilizers connected in series and using heavier than air refrigerant, the uppermost stabilizers should be equipped at their tops with purge valves, 24, or cocks for purging out air and non-condensable gasses. When using lighter than air refrigerant, the purge cocks

should be located just above the liquid refrigerant in the stabilizer shells.

By using multiple stabilizers, each placed as near as possible to its respective evaporator, as above described, insulated suction lines from the evaporators to the stabilizers, with attendant refrigeration losses and possible objectionable frosting, sweating and dripping, are eliminated.

Since the compressed refrigerant from the compressor has had the superheat of compression removed by the condenser before it reaches the stabilizer, the latter coils do not superheat the refrigerant excessively as it enters the compressor, yet does superheat it to comply with manufacturers' rating requirements. In this connection, it is well to note that many manufacturers of refrigeration compressors state in their catalogue ratings that they will not guarantee the performance capacities of their compressors, unless the suction gas into said compressor is superheated to 65 degrees F.

When the suction line from the evaporator to the compressor is short, in below-freezing installations that have condensing units that are not of the hermetic type, having suction gas heated by the cooling of their motors, the suction coiled stabilizer, as shown in FIG. 2, is the most practical and almost the only way this suction gas superheating can be accomplished. This elimination of compressor slugging and the slight superheating of the suction gas from the evaporator makes possible the practical operation of the high efficiency supercharged compressor for dual or multiple temperature applications on an automatic unattended basis by porting the compressor cylinders at different points in their stroke. In multiple cylinder compressors, one or more cylinders may have their main inlet valves ports isolated and different cylinders ported at different points in the piston stroke to maintain the several temperatures in the respective evaporators. This may be illustrated by a situation such as automotive or marine air conditioning and refrigeration where one has a multiple cylinder compressor, the main purpose of which is for air conditioning as shown at 6, FIG. 9, but where one in addition wishes low temperature refrigeration evaporators as shown at 5 and 31 from the same compressor, this is attained by isolating individual cylinder inlet valves 33, 34, and 35 and cylinder ports 28 corresponding to individual evaporators 6 whereby multiple temperatures may be obtained from the same compressor.

In using an evaporator consisting of a "hold-over" or eutectic solution 27, in a container 26, as shown in FIG. 2 with refrigerant evaporating coils in contact with said solution, wherein the solution is frozen by said evaporating coils, as the solution nears complete freezing, the refrigeration load or amount of heat being given up by the solution decreases. Since the capillary tube or metering device tends to feed a constant amount of refrigerant into the cooling coils, more and more unevaporated liquid refrigerant will "flood" through said evaporating coil into the compressor. By passing the refrigerant returning from the cooling coil to the compressor through a coil inside the stabilizer member any returning liquid is evaporated, to prevent compressor slugging and at the same time the head pressure is lowered, thereby lessening the flow of refrigerant through the capillary tube, making the system self regulating to a certain extent, as well as reducing the amount of current required by the condensing unit motor.

The standby electric, friction and other losses of the compressor are, by and large, mostly taken care of on the colder cooling application. By taking advantage of the compressor supercharging application for the higher temperature refrigeration, the latter is obtained almost loss-free, thereby resulting in a large saving of power.

There are disclosed a limited number of embodiments of the structure and it is desired therefore that only such limitations be imposed on the appended claims as are stated therein, or are required by the prior art.

I claim:

1. In a two-temperature refrigeration system having refrigerant metering devices thereon and having a charge of refrigerant therethrough, a multiple-effect compressor having a cylinder therein, an upper inlet port and a lower inlet port in the latter, a condenser connected to the compressor, multiple liquid refrigerant stabilizer shell receiver means connected to the condenser, a low-temperature evaporator connected through a coil in a stabilizer shell member to the upper port, a relatively high temperature evaporator connected through a second coil in a second stabilizer shell member to the lower port, the bottom liquid receiver portion of said first stabilizer shell connected through a refrigerant metering device to the low temperature evaporator, the bottom liquid receiver portion of the second stabilizer shell connected to the high temperature evaporator through another liquid metering device, said first stabilizer shell connected to the condenser, said second stabilizer shell member tapped to the first stabilizer member at the top of the liquid level in the latter thus all of the heat of vaporization passing through said compressor is made available for evaporating slugs from any one or more of the evaporators.

2. In a two-temperature refrigeration system having a charge of refrigerant therein, a multiple-effect compressor, a condenser connected therewith, a stabilizer shell means, a cylinder in said compressor having an inlet in the upper portion thereof, a second inlet port in its bottom portion, a low temperature evaporator connected to the first mentioned port by means of a suction line connected through a coil in said stabilizer shell means, a higher temperature evaporator connected to the second inlet port through a second suction line through a second coil in the stabilizer shell means, said shell means having inlet means connected to the discharge end of the condenser, said inlet connecting means between the condenser and the stabilizer shell means, including the shell means being of sufficient size to pass all of the liquid from the condenser together with any uncondensed refrigerant gas into said stabilizer means, said shell having an outlet, and conduit means connecting the outlet with the low and high temperature evaporators.

3. The apparatus of claim 2 including adjustable automatic pressure operated expansion valves positioned in the conduit means connecting the stabilizer shell outlet with the evaporator.

4. The apparatus of claim 2 wherein the stabilizer shell means is of sufficient size to include a space for refrigerant gas above the liquid refrigerant, the stabilizer means functioning to cool the liquid therein and reduce the pressure therein below condenser pressure, and the stabilizer shell means including means to indicate the quantity of refrigerant in the shell, said last mentioned means being capable of functioning as a purge means.

5. A multi-temperature refrigeration system having a refrigerant charge therein and having refrigerant metering devices thereon, a multi-effect compressor, a condenser connected therewith, stabilizer shell means having liquid receiving portions, a plurality of cylinders in said compressor with pistons operating therein, each cylinder having an intake valve at the top of its piston stroke, the intake valve of one such cylinder isolated from its adjacent cylinders, said cylinder having a second inlet port in its bottom portion and an additional inlet port in a portion intermediate the top and bottom ports, a low temperature evaporator connected to said isolated inlet valve by means of a suction line connected through a coil in a first stabilizer shell means, a higher temperature evaporator connected to said intermediate inlet port by means of a suction line connected through a coil in a second stabilizer shell means, a high temperature evaporator connected to the bottom inlet port in said cylinder and to the top intake valves in the additional cylinder in said compressor, by means of a suction line connected through a coil in a third stabilizer shell means, the bottom liquid receiver

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portions of said shells connected by conduit means through individual metering devices to their respective low, higher and high temperature evaporators, the first shell tapped to the second shell at the liquid level of the latter, the discharge connection into said first shell being above its liquid level, the second shell tapped to the third shell at the liquid level of the latter, the discharge connection into said second shell being above its liquid level, the third shell having inlet means connected to the discharge end of the condenser.

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