Feb. 17, 1953 E. W. ZEARFOSS, JR 2,628,478

METHOD OF AND APPARATUS FOR REFRIGERATION

Filed Dec. 13, 1949 3 Sheets-Sheet 1

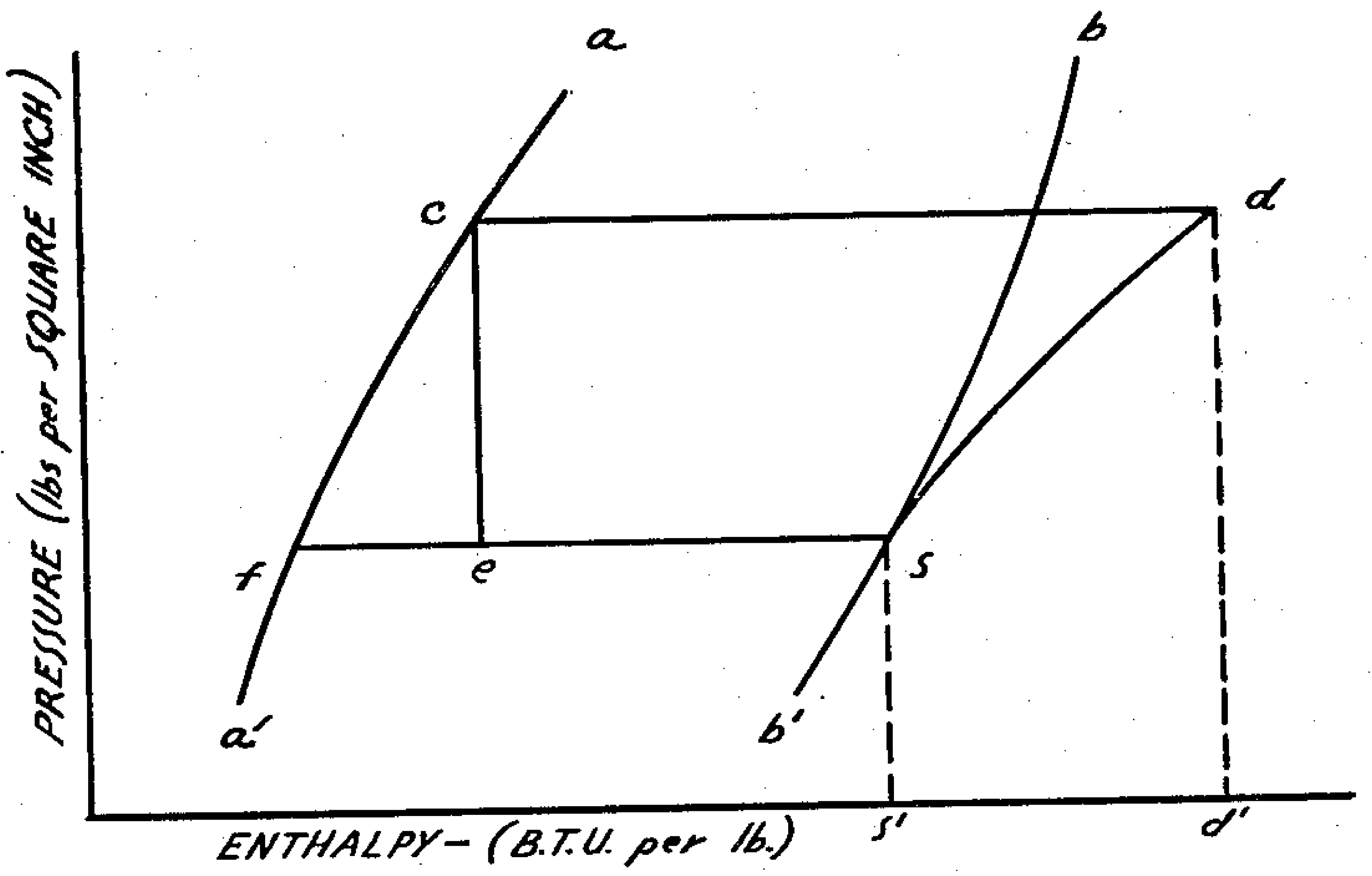


FIG. 1.

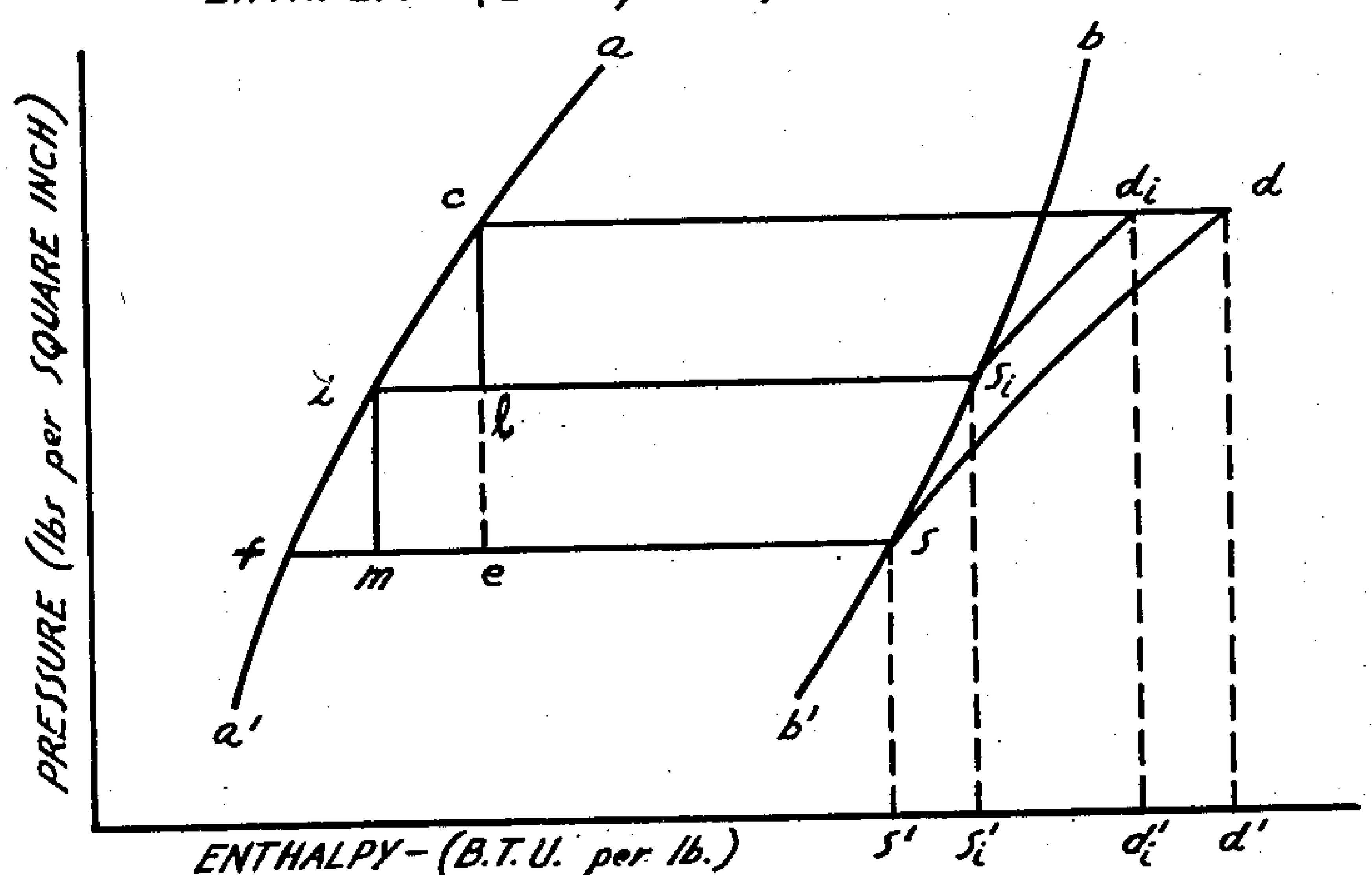


FIG. 2.

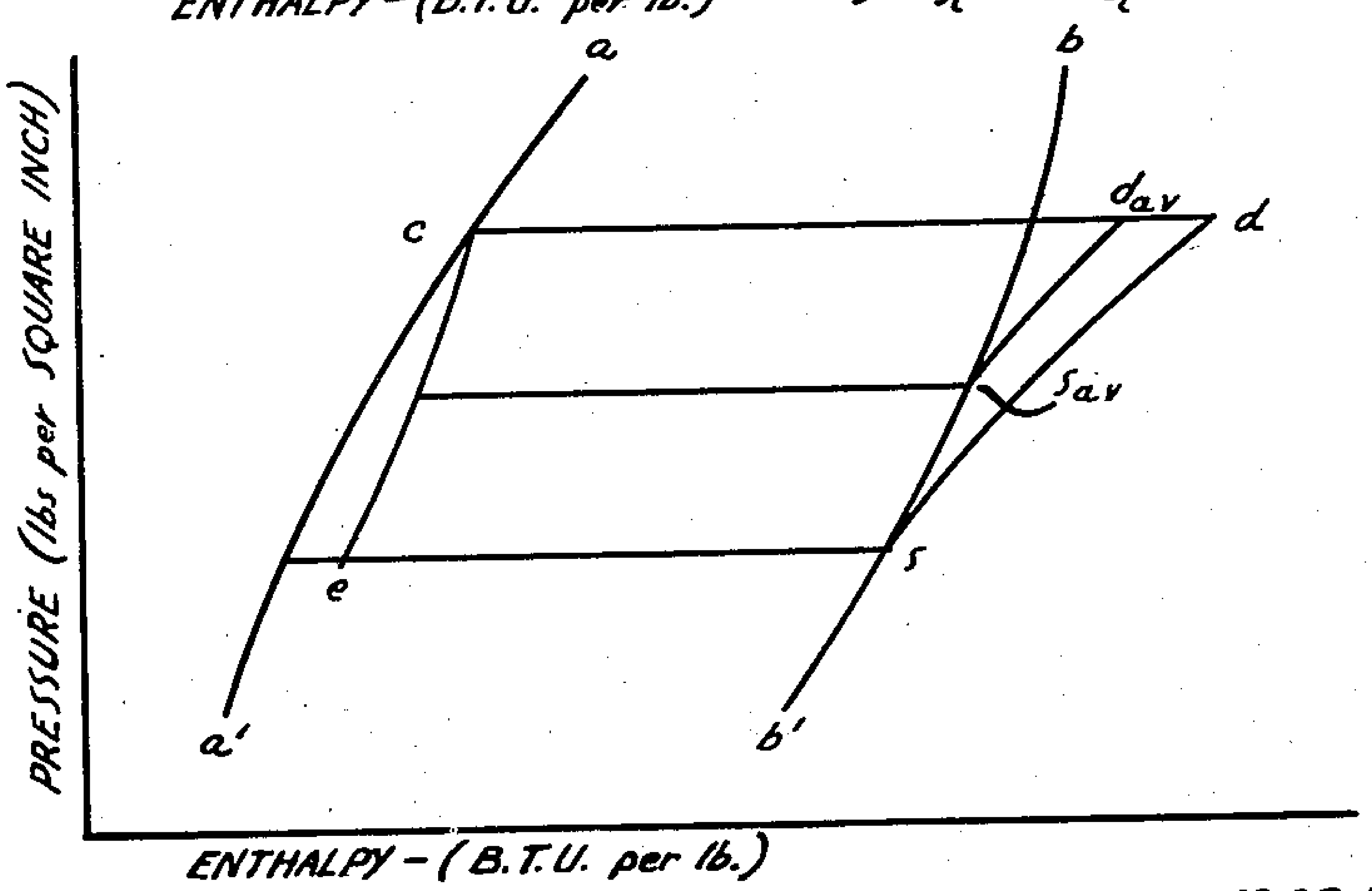


FIG. 3.

INVENTOR.
ELMER W. ZEARFOSS, JR.
BY
Brown, Denk & Synnestvedt
AGENTS

Feb. 17, 1953 E. W. ZEARFOSS, JR 2,628,478

METHOD OF AND APPARATUS FOR REFRIGERATION

Filed Dec. 13, 1949 3 Sheets-Sheet 2

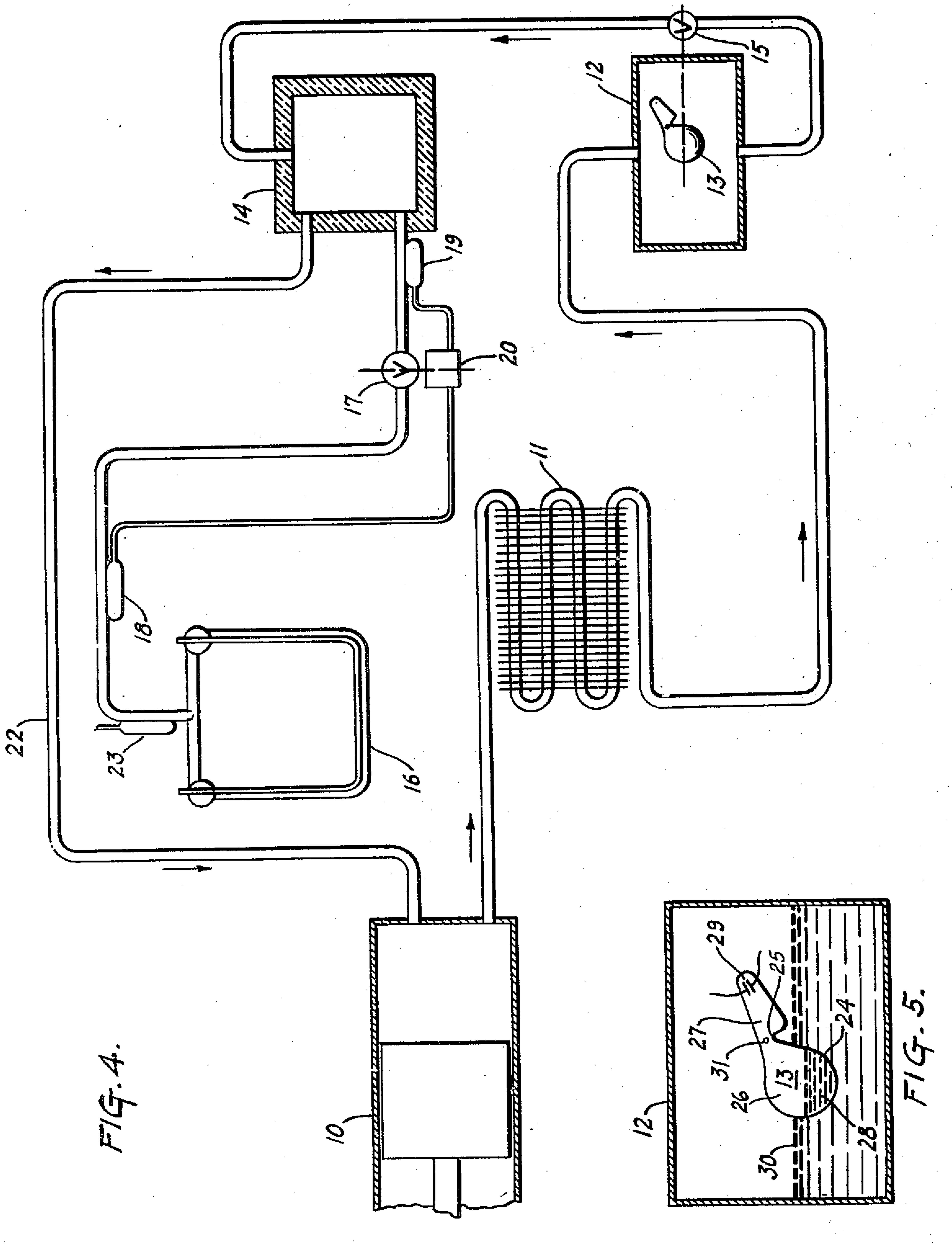


FIG. 4.

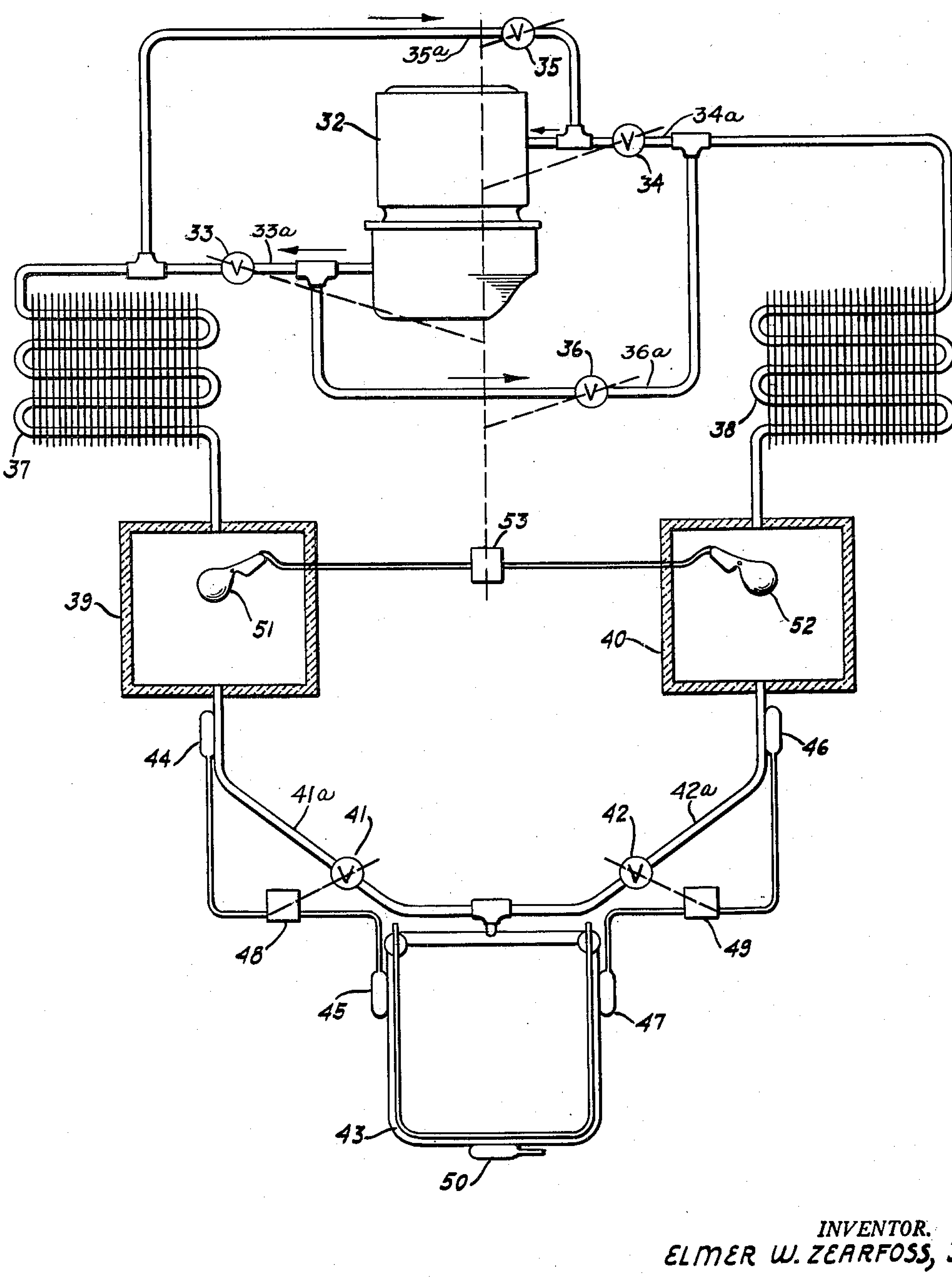
FIG. 5.

INVENTOR.
ELMER W. ZEARFOSS, JR.
BY
Brown, Denk & Synnestvedt
AGENTS

METHOD OF AND APPARATUS FOR REFRIGERATION

Filed Dec. 13, 1949 3 Sheets-Sheet 3

FIG. 6.



INVENTOR.
ELMER W. ZEARFOSS, JR.
BY
Brown, Denk & Synnestvedt
AGENTS

Patented Feb. 17, 1953 2,628,478

UNITED STATES PATENT OFFICE

2,628,478

METHOD OF AND APPARATUS FOR REFRIGERATION

Elmer W. Zearfoss, Jr., Philadelphia, Pa., assignor to Philco Corporation, Philadelphia, Pa., a corporation of Pennsylvania

Application December 13, 1949, Serial No. 132,766

14 Claims. (Cl. 62—3)

1

The present invention relates to refrigeration, the instant disclosure being a continuation-in-part of my copending application, Serial Number 49,387, filed September 15, 1948, now abandoned. More particularly, the invention relates to a refrigeration method and system in which volatile refrigerant is evaporated and then condensed to obtain the desired refrigerating effect.

Refrigerating systems of the kind here contemplated are generally provided with an evaporator in which low pressure liquid refrigerant vaporizes and absorbs heat in the process; with a compressor in which low pressure vaporized and heat-laden refrigerant is compressed to increase its pressure to a value corresponding to a saturation temperature well above normal atmospheric temperature; with a condenser in which high pressure vaporized refrigerant rejects to the ambient medium and reverts to a liquid state; and with a pressure reducing device in which the high pressure liquid refrigerant is subjected to an expansion process effective to bring the refrigerant down to the low evaporating pressure and temperature.

In the expansion process, the pressure of the liquid is lowered and part of the liquid evaporates into gas, known as flash gas, the heat of vaporization required to produce this flash gas removing heat energy from the refrigerant and leaving the remaining liquid refrigerant in a condition in which it can absorb heat energy in the evaporator. The flash gas produced has no further value as refrigerant, since it has already absorbed its heat of vaporization, and it is advantageous that such flash gas be removed from the refrigeration system at the point of expansion and before it can enter the evaporator. It is thence returned to the compressor where it can be recombined with the remaining refrigerant in the refrigerating system.

For the simple refrigeration cycle outlined above, the work of compression, which is the only work input required, increases as the difference between the evaporation and condensation pressures of the refrigerant increases. This pressure range is fixed, for the useful portion of the refrigerant, by the prevailing operating conditions, but I have discovered that a possibility of bettering the performance of the system lies in reducing the pressure range of that portion of the refrigerant which becomes flash gas during the expansion process. Formation of flash gas begins as soon as the expansion of the refrigerant starts. When the pressure has dropped a finite increment, a finite fraction of refrigerant has become

2

saturated vapor and is therefore no longer capable of providing a refrigerating effect. If it were possible to extract this vapor and recompress it immediately, the required work of compression would be much less than is the case when the vapor is first expanded and lowered in temperature to the final evaporator pressure and temperature, and then is recompressed through the entire pressure range of the system. If the vapor formed after the first finite interval were extracted, then the remaining liquid could be further expanded and the resulting flash gas again removed, and this process, by adjusting the pressure range to be covered in each partial expansion, could be repeated as often as desired before reaching the final evaporator pressure.

As will be seen, from the foregoing discussion, the average pressure range over which the flash gas must be recompressed will be smaller, the greater the number of expansion increments, or stages; and, in the limiting case, said average pressure range will be smallest with an infinite number of stages, which corresponds to a continuous removal of flash gas. Such a continuous removal of flash gas would therefore require the least work of compression, thus yielding the most efficient operation of the refrigeration system.

It has been found impractical to increase the number of expansion increments, or stages, to more than two or three, due to the prohibitive increase in equipment necessitated by such an increase in the number of stages. The removal of flash gas, by an infinite number of discrete stages, as hereinbefore outlined, is, of course, impractical (see Pages 47 and 48, "Refrigeration and Air Conditioning," B. F. Raber and F. W. Hutchinson, Wiley, 1945).

It is a feature of the present invention that there is effected continuous removal of saturated vapor flash gas formed in the transition from high pressure to low pressure refrigerant.

It is a primary object of the present invention to provide a method and apparatus for removing, in the most efficient manner possible, flash gas formed during a refrigeration cycle.

It is a further object of the present invention to increase the coefficient of performance of a refrigeration system to a value considered impossible of attainment heretofore.

It is still another object to increase the coefficient of performance of a refrigeration system, while, simultaneously, reducing the volumetric capacity of the compressor required to give a predetermined amount of refrigeration.

These and other objects of the invention, and the manner in which they are attained, will be more fully understood from the following description and with reference to the accompanying drawings in which:

Figure 1 shows a conventional pressure-enthalpy diagram, well known in thermodynamics and illustrating the thermodynamic performance of an ordinary, simple refrigeration system;

Figure 2 shows a conventional pressure-enthalpy diagram, illustrating the thermodynamic performance of an ordinary, multi-stage refrigeration system;

Figure 3 shows a conventional pressure-enthalpy diagram, illustrating the thermodynamic performance achieved by the apparatus of the present invention;

Figure 4 represents, somewhat diagrammatically, a refrigeration system embodying the present invention;

Figure 5 illustrates, on an enlarged scale, a portion of a control mechanism used in the system shown in Figure 4; and

Figure 6 is a further, somewhat diagrammatic representation illustrative of a refrigeration system comprising an alternative embodiment of the present invention.

Now making more detailed reference to Figure 1, lines $a—a'$ and $b—b'$, respectively, represent the saturated liquid and saturated vapor lines of a volatile refrigerant such as dichlorodifluoromethane (Freon 12). Point c indicates the state of a pound of the refrigerant upon issuance from the condenser of a simple refrigeration system of the type previously described, and point e represents the state of said refrigerant after constant enthalpy expansion along line $c—e$, as is usual. In the course of this expansion, the formation of flash gas causes a reduction of the available heat capacity per pound (represented graphically by line $f—s$), of the refrigerant, by the amount represented graphically by the line $f—e$. This is not due only to the formation of the volume of flash gas which is unavoidably associated with an expansion such as the above, but also due to the fact, that since no flash gas is removed until the completion of the expansion, those portions of the flash gas which are formed in the early part of the expansion must further be cooled during subsequent expansion, and thus cause a still further reduction in the heat capacity originally available per pound of refrigerant. The liquid refrigerant, which is sometimes separated from the flash gas, is passed through the evaporator, changing its condition along line $e—s$. Both flash gas and refrigerant are then recompressed along line $s—d$, necessitating an expenditure of work per pound which, in terms of heat units, is equal to the projection $s'—d'$ of $s—d$ on the enthalpy axis. The compressed refrigerant and flash gas are then returned through the condenser to state c and the cycle is ready to start again.

With reference to Figure 2, it will now be seen that if the flash gas formed during that portion of expansion $c—e$ denoted by segment $c—l$, is removed in the state described by point l, this flash gas can be recompressed to the initial pressure condition along line $s_1—d_1$, which corresponds to a smaller expenditure of compressor work per pound of flash gas than would have been required had this aforementioned portion of flash gas been permitted to expand to the final evaporator pressure represented by point e. The refrigerant remaining, after removal of the flash gas at point

l, is now again a saturated liquid in a state described by point i and can now proceed to expand again at constant enthalpy, which will bring it, this time, to the state described by point m. Then, only the flash gas formed during expansion $i—m$ need be removed at this time and recompressed over the total pressure range along line $s—d$, together with the evaporated refrigerant which provides the useful refrigeration.

Having shown that the work per pound needed to recompress flash gas along line $s_1—d_1$ (represented by the projection $s_1'—d_1'$ of $s_1—d_1$ on the enthalpy axis) is smaller than the work per pound needed to recompress flash gas along line $s—d$ (represented by the projection $s'—d'$ of $s—d$ on the enthalpy axis), it now follows that the average work per pound required to recompress all the flash gas formed during the cycle of Figure 2 will be less than that required to recompress all the flash gas formed during the cycle of Figure 1. An additional saving in the cycle of Figure 2 will result from the fact that no flash gas need be formed, during expansion $i—m$, to effect further cooling of flash gas previously formed during expansion $c—l$. Thus there exists a greater heat capacity available per pound of refrigerant at the final pressure, the increase being represented by segment $e—m$ of Figure 2, and which increase raises the heat capacity available from an amount represented by $s—e$, to an amount denoted by the line $s—m$.

Figure 3 illustrates what occurs, in this thermodynamic refrigeration cycle, when the number of stages of expansion and flash gas removal is actually, or effectively, increased to infinity. As shown in Figure 3, the expansion line $c—e$ then follows an isentropic, or reversible adiabatic. This can be explained as follows: It is physically impossible to increase the number of discrete expansion stages to infinity, therefore alternative means, hereinafter described, are used to accomplish the equivalent effect, namely the continuous removal of flash gas immedaitely upon its formation and at the pressure at which it is formed. If, then, each infinitesimal amount of flash gas is removed as soon as it is formed, cooling thereof will not be required, as would have been the case had it been permitted to remain in the system during subsequent expansion, and the expansion becomes a reversible adiabatic process as hereinbefore stated.

Comparative consideration of Figures 2 and 3 might, at first sight, lead to the erroneous conclusion that, if the expansion of Figure 3 is the same as the expansion of Figure 2—with the number of stages carried to infinity as a limit and, conversely, the size of each stage reduced to an infinitesimal decrement—then the line in Figure 3, showing that expansion, should coincide with the saturated liquid line $a—a'$. The error of this conclusion can readily be seen if it is recalled that it is the practice in thermodynamics to draw diagrams, such as Figure 2, using the refrigerant in the initial saturated liquid state as a base of reference. This means that at point i of Figure 2, a new reference has been introduced, namely the quantity of liquid refrigerant remaining in the system at that pressure and after having subtracted the flash gas previously formed. This remaining liquid will form more flash gas during expansion $i—m$, losing a portion of its available heat capacity in the process. Thus, in Figure 2, the proportion of liquid and flash gas denoted by point m refers to the original quantity of refrigerant present at point i, while the proportion denoted by point l refers to the larger original quantity of refrigerant denoted by point c. This same reasoning must be applied to Figure 3, where each proportion indicated by any point on line c—e, must be referred to the original total quantity of refrigerant. In short, it must be borne in mind that Figures 2 and 3 are conventional graphical representations of the thermodynamic changes in the respective processes, and the showings of said figures are quite independent of any physical apparatus employed to effect these changes.

The average pressure range, shown by line s_{av}—d_{av} of Figure 3, over which such continuously removed flash gas must be recompressed, will now be smaller than that which could be obtained with any finite number of stages and, in addition, the heat capacity of the low pressure liquid refrigerant will be highest, since no cooling of flash gas takes place.

The former of these two effects, namely the decrease in average pressure range, makes possible the decrease in compression work, and the latter, namely the increase in heat capacity availability, makes possible the reduction in volume of refrigerant needed to produce a certain amount of refrigeration and, thereby, reduces the volumetric capacity and cost of the whole system.

The advantages of a method of removing flash gas continuously and as soon as it is formed are hence twofold. First, only as much flash gas is formed as is necessary to lower the refrigerant pressure and temperature to the values required by operating conditions and, secondly, that flash gas is removed in the manner requiring the least possible work, namely at the pressure at which it is formed.

The terms used in the foregoing analysis are well-known in the field of thermodynamics and accordingly, no further explanation is deemed necessary here. If a further clarification is desired, reference may be had to "Refrigeration and Air Conditioning Engineering," B. F. Rabor and F. W. Hutchinson, Wiley, 1945.

Referring now to Figure 4, there is diagrammatically illustrated a refrigeration system which comprises a preferred embodiment of the present invention.

The system includes a conventional compressor 10, a conventional condenser 11, and a receiving tank 12, containing a horizontally hinged mercury pool switch 13. The compression side of compressor 10, the condenser 11 and the receiving tank 12 are connected by refrigerant flow conduits. The system further comprises refrigerant flow conduits of substantially non-restrictive dimensions connecting receiving tank 12 to an insulated expansion tank or chamber 14 through a valve 15, and refrigerant flow conduits leading from insulated expansion tank 14 to the top header of evaporator 16 through a valve 17. The tank 14 is preferably of such a nature that it is of low thermal mass, in order that the heat loss therein will be negligible.

Mercury switch 13 is connected to control means such as a solenoid (not shown) designed to operate valve 15 in a manner described below. Further means are provided, such as temperature sensitive bulbs 18 and 19, for detecting differences in refrigerant temperature between the input of evaporator 16 and the output of expansion tank 14, said means being connected to a control 20 arranged to actuate valve 17 in the manner described in detail below.

The suction port of compressor 10 is connected to an opening near the top of expansion chamber 14. This compressor connection is effected by way of a refrigerant conduit 22 of substantially non-restrictive dimensions. The reason for this arrangement will become apparent from the detailed description of its operation.

A temperature sensitive cycle switch control bulb 23 is placed at the output of evaporator 16, to initiate and control the operation of compressor 10 in response to the temperature of the evaporator. This latter cycling control is conventional and, for the purpose of the present invention, no further description or illustration is necessary.

To obtain an understanding of the operation of the system, let it first be assumed that the system is at rest, the compressor, and other components not having recently been in operation. The ambient temperature and the temperature of the system itself are, of course, considerably in excess of the temperature necessary to cause control bulb 23 to start the compressor 10 through conventional relays (not shown). When the compressor begins operation, valves 15 and 17 are both closed, as will become evident from the subsequent discussion.

At this time the refrigerant is dissolved in the compressor oil, except for such refrigerant as still exists in the gaseous state throughout the remainder of the system, the pressure of the dissolved refrigerant being equal to the pressure of the gaseous refrigerant.

In operation, the compressor distills refrigerant from the oil in which it is dissolved and forces it into condenser 11, where it is condensed into liquid refrigerant at a high pressure and temperature as represented thermodynamically by that portion of the process shown graphically by line d–c of Figure 3. This distillation reduces the pressure of that portion of the refrigerant remaining dissolved in the oil, and operation of the compressor simultaneously reduces the pressure of such a small quantity of refrigerant as exists as vapor in tank 14.

The liquid thus formed begins to fill receiving tank 12. The horizontally pivoted mercury switch 13, located inside this tank, is so connected that rising of the liquid above a certain level causes an electrical contact to be made which in turn causes actuation of solenoid operated valve 15.

Since substantially no liquid refrigerant exists in the system, until this initial filling of tank 12, switch 13 will evidently have heretofore maintained valve 15 closed, as hereinbefore stated. The switch is so arranged that it will permit outflow of refrigerant through valve 15 until a level is reached in tank 12 which is very much lower than that at which the switch makes contact. Switch 13 then breaks contact and solenoid operated valve 15 closes, permitting renewed filling of receiving tank 12. The delayed opening of mercury switch 13 may be accomplished in any desired manner. For example, switch 13 may be constructed as shown in Figure 5, and comprise a horizontally pivoted float 24 with a ridge 25 separating the high-buoyancy section 26 from the contact section 27 and containing mercury pool 28 and normally open contacts 29 connected by leads to an external source (not shown) of electric potential. When the liquid level denoted by reference numeral 30 rises to a predetermined height, section 26 also rises about pivot 31 and some mercury spills over ridge 25 into contact section 27. Since the cohesion of mercury is greater than its adhesion, substantially all the mercury is drawn into contact section 27 and an electrical path is established between contacts 29, with the results hereinbefore described. When the liquid level 30 falls, on the other hand, the ridge 25 prevents the mercury from returning to section 26, and thereby opening contacts 29, until the liquid level has fallen to a predetermined low level, thus permitting valve 15 to remain open until tank 12 has been substantially emptied as hereinbefore described.

Receiving tank 12 will, in most commercially practicable systems, be located in a surrounding medium which is at the same ambient temperature as the medium surrounding condenser 11. Since the refrigerant temperature drop due to heat exchange through the conduits connecting condenser 11 and tank 12 will, in most cases, be practically negligible, for the foregoing implies that the temperature of the refrigerant in tank 12 will, ordinarily, be higher than that of the surrounding medium. In that case, receiving tank 12 should be uninsulated, as shown in Figure 4, in order to take advantage of the favorable heat exchange between the refrigerant in tank 12 and the lower temperature surrounding medium, and thus obtain further partial cooling of the hot refrigerant. If receiving tank 12 should be located in a surrounding medium which is at a higher temperature than the temperature of the refrigerant as it arrives in the tank, then it would, of course, be advantageous to insulate tank 12, to prevent the refrigerant from being unnecessarily heated by the surrounding medium.

The opening of valve 15 permits the high pressure refrigerant to flow out of receiving tank 12 and into expansion chamber 14, still at substantially the same high pressure and temperature, but the refrigerant is prevented from reaching evaporator 16 by closed valve 17. The operation of valve 17 is controlled by temperature sensitive bulbs 18 and 19 which are connected to opposed bellows which, in turn, actuate a conventional system of levers generally designated by reference numeral 20 and arranged to open and close valve 17, as follows. Bulb 18 is gas charged at 20° F. and bulb 19 is gas charged at 55° F., and the bellows and mechanical linkages 20 are so arranged that valve 17 is closed when the temperature at bulb 19 exceeds the temperature at bulb 18 by 30° F., or more, and open when the temperature at bulb 19 exceeds that at bulb 18 by 5° F. or less. Such arrangements are well-known in the art and require no further discussion here. The temperature differentials listed above are not critical, but merely indicative of a preferred set of conditions. Since the normal room temperature is assumed to be higher than 55° F., the effective 35° temperature differential between bulbs 18 and 19—created by gas charging thereof—is sufficient to keep valve 17 initially closed, as hereinbefore stated. The high-pressure liquid in tank 14 is now confined by closed valves 15 and 17, its only remaining communication with the other elements of the system being by way of conduit 22, which leads directly to the suction port of compressor 10. Each successive suction stroke of compressor 10 will then result in the creation of a pressure differential between tank 14 and the compressor cylinder. This, in turn, will cause the formation and removal at the pressure of formation, of a given incremental volume of gaseous refrigerant, or flash gas, from tank 14. Formation of each such volume of flash gas results in cooling of the remaining portion of liquid refrigerant in tank 14.

This process of decreasing the pressure of the refrigerant, forming flash gas and thereby cooling the liquid refrigerant, but (and importantly) removing each increment of flash gas to the compressor as soon as such flash gas is formed, constitutes the fundamental process of the present invention.

From the foregoing, it is clear that evacuation of gaseous refrigerant from tank 14 will proceed at a rate determined solely by the gradual reduction of pressure in tank 14 due to the operation of the compressor 10 and entirely independently of the pressure conditions anywhere else in the system.

When the refrigerant in tank 14 has been cooled to within 5° F. of the temperature at which bulb 18 is gas charged, this difference in temperature between bulbs 18 and 19 will be sensed and control 20 will operate to open valve 17. Since the refrigerant in tank 14 is now still at a higher pressure than any refrigerant in evaporator 16—by virtue of the aforesaid 5° temperature differential—the refrigerant will flow into evaporator 16.

By the simple expedient of introducing this refrigerant into the top header of the evaporator, as hereinbefore briefly stated, any back-flow of liquid refrigerant into the expansion tank is prevented, even though the evaporator may be located above the expansion tank.

With the passage of the liquid reference from tank 14 into evaporator 16, unrestricted communication between the top header of this evaporator and the suction port of compressor 10 is established. This communication is by way of valve 17 (which remains open for the time being, since there exists, as yet, no temperature differential sensible to feeler bulbs 18 and 19 which could cause this valve to close), through tank 14, which is now devoid of liquid refrigerant, and through conduit 22.

The liquid refrigerant in the evaporator, now absorbs heat from the medium to be refrigerated, in the usual manner, becoming vaporized in the process. Continued operation of compressor 10 will then result in the withdrawal of this vaporized refrigerant from the evaporator and in delivery thereof to the compressor where it is recompressed for renewed delivery to the condenser and recirculation through the system.

To preclude the possibility of delivery of a new load of hot high pressure refrigerant to expansion tank 14, which would cause valve 17 to close prematurely, thereby preventing successful evacuation of the vaporized refrigerant from evaporator 16, the system is so designed that, for a given volumetric capacity of the physical components, there is not enough refrigerant available to enable tank 12 to fill up to its relief level before substantial evacuation of evaporator 16 has taken place. While many arrangements may be employed which fulfill the preceding requirement, for the sake of simplicity it may be assumed that the components are so proportioned that substantially all the refrigerant in the system is needed to fill tank 12 to a level high enough to cause valve 15 to open (as hereinbefore described). This arrangement, as hereinbefore pointed out, prevents the premature admission of hot refrigerant to tank 14, thereby preventing the premature closure of valve 17. In addition, there is, thus, evidently no possibility of tank 12 filling to a level which would cause valve 15 to open before expansion tank 14 has emptied into evaporator 16, thereby precluding any possibility of hot refrigerant contacting, or mixing with cold refrigerant.

When in steady state operation the closing of valve 17 is effected by the first sensing by bulb 19, of the arrival of hot refrigerant from tank 12, which bulb then acts to cut tank 14 off from evaporator 16 until said refrigerant is sufficiently cooled.

Figure 6 represents an alternative embodiment of the refrigeration system of the present invention. This system is differentiated from the one shown in Figure 4 in that it has a compressor 32 which can, by means of conventional solenoid-controlled reversing valves, 33, 34, 35, and 36, acting upon conduits *33a*, *34a*, *35a* and *36a*, respectively associated therewith, be made to interchange its suction and its compression sides with respect to an externally connected system. It further has two condensers 37 and 38. It has two insulated tanks 39 and 40, which are used alternately as receiving and expansion tanks. It further comprises valves 41 and 42, acting upon conduits *41a* and *42a*, respectively, which connect a single evaporator 43 to tank 39 and tank 40. Evaporator 43 is located below tanks 39 and 40, in order to enable refrigerant to flow from these tanks to evaporator 43 under the influence of the force of gravity. One pair of temperature sensitive feeler bulbs 44 and 45, similar to bulbs 18 and 19 of Figure 4, are arranged to sense the temperature at the output of tank 39 and in the evaporator 43, respectively, while another pair of feeler bulbs 46 and 47, also similar to bulbs 18 and 19 of Figure 4, are placed to sense the temperature at the output of tank 40 and in the evaporator 43, respectively. Bulbs 44 and 45 and bulbs 46 and 47 are each connected to a pair of opposing bellows and linkages respectively designated by reference numerals 48 and 49 and similar to control 20 of Figure 4. Control 48 is arranged to control the opening and closing of valve 41 while control 49 controls the operation of valve 42. Bulbs 44 and 46 are gas charged at 55° F. and bulbs 45 and 47 are gas charged at 20° F. Controls 48 and 49 are so arranged as to close valves 41 and 42, respectively, when the temperature at the output of each respective tank exceeds the temperature in the evaporator by 30° F., or more, and to open valves 41 and 42, respectively, when the temperature at the output of each respective tank exceeds the temperature in the evaporator by 5° F. or less. Thus each pair of temperature sensitive feeler bulbs (such as bulbs 44 and 45) cooperates with the control connected thereto (such as control 48) to operate the valve associated with the control (such as valve 41) in response to the difference in temperature existing between the respective tank and evaporator 43.

A cycling bulb 50, similar to bulb 23 of Figure 4, is arranged to sense the temperature of evaporator 43 and to control the operation of compressor 32 by means of conventional relays (not shown).

Tanks 39 and 40 are each equipped with horizontally pivoted mercury pool switches respectively numbered 51 and 52 and similar in construction to the switch illustrated in Figure 5. The set of contacts in each switch is connected to a source of electric potential (not shown) and each set of contacts is further connected to a conventional double-throw locking relay—represented schematically by box 53—in such a manner that closing of one switch throws the relay in one direction and locks it there, until closing of the other switch releases the relay from that position and throws it in the opposite direction. Operation of the relay 53, due to the action of switch 51, makes suitable electrical contacts (not shown), thus energizing the solenoids associated with valves 33 to 36, opening valves 35 and 36 and closing valves 33 and 34, while operation of switch 52 causes relay 53 to reverse, making suitable electrical connections to energize solenoids to close valves 35 and 36 and open valves 33 and 34. Thus, effectively, closing switch 51 connects tank 39 to the suction side of compressor 32 while closing switch 52 connects tank 40 to the suction side of compressor 32. Since relay 53 is of the locking type, in each position, the first mercury switch to operate the relay loses control of relay 53 once it has caused it to operate, and only the other switch is capable of reversing the relay position.

To assist full comprehension of the operation of this system, the following detailed description is presented. The system is assumed to be at rest—the compressor 32 and the other components of the system not having been recently operated—and valves 33 and 34 having remained open from the time when the system had last been operated. The ambient temperature and the temperature of the system itself, are, of course, considerably in excess of the temperature necessary to cause cycling bulb 50 to initiate operation of the compressor, with the result that the compressor commences operation and the refrigerant in the compressor is forced into condenser 37 where it is condensed into hot liquid. This hot liquid refrigerant being at least at the ambient temperature and therefore at a higher temperature than is required to close valve 41, this valve will close and the hot liquid refrigerant will accumulate in tank 39. When it has reached a level sufficiently high to tip and close switch 51—thereby reversing relay 53 and connecting tank 39 to the suction side of compressor 32 as hereinbefore described—flash gas will start to form in tank 39 and will be removed by compressor 32, in a manner similar to the formation and removal of flash gas from tank 14 of the embodiment shown in Figure 4.

Again, removal of flash gas immediately upon, and at the pressure of formation, constitutes the basic characteristic of this embodiment of the present invention.

The flash gas removed from tank 39 is compressed in compressor 32, condensed in condenser 38 and, having closed valve 42 by the effect of its high temperature on bulb 46, begins to accumulate in tank 40. When the liquid refrigerant in tank 39 has, for example, been cooled to 25° F., valve 41 opens, as hereinbefore described and the cold refrigerant is gravity-fed into evaporator 43, where it absorbs heat from the surrounding medium, vaporizes and is exhausted by the suction of compressor 32, to which evaporator 43 is now connected. The gaseous refrigerant thus removed will be compressed in compressor 32, condensed in condenser 38 and also stored in tank 40. When sufficient refrigerant has accumulated in tank 40 to tip and close mercury switch 52, the valves associated with compressor 32 reverse, as hereinbefore described, and tank 40 is now connected to the suction side of compressor 32, with the result that flash gas is now removed from tank 40 with the attendant cooling of the liquid refrigerant remaining in tank 40. This flash gas is again compressed in compressor 32, condensed in condenser 37 and delivered to tank 39 where it closes valve 41 immediately upon arrival, thus permitting accumulation of hot liquid refrigerant in tank 39. When the temperature of the refrigerant in tank 40 has been lowered to 25° F., valve 42 opens as hereinbefore described and the refrigerant from tank 40 is gravity-fed into evaporator 43. The vapor formed in evaporator 43 is again exhausted and compressed by compressor 32, condensed in condenser 37 and stored in tank 39, the mercury switch 51 in the latter again reversing the compressor connections when the proper liquid level is reached, thus initiating a new cycle. Here again, the temperature differentials referred to are not critical but are intended only to be indicative of a preferred set of conditions. Furthermore, it will be understood that mercury switches 51 and 52 are so adjusted in tanks 39 and 40 as to permit maximum utilization of each tank and are, in addition, so located that each will tilt and make contact when substantially the same quantity of liquid refrigerant has accumulated in one tank as would be required to make the switch tilt in the other tank.

It is clear that the means for controlling valve 17 of Figure 4 and valves 41 and 42 of Figure 6 are not, per se, an essential feature of the invention, and that any conventional means suitable for accomplishing the desired results outlined above, may be employed. A practical system which is suitable for this purpose, as indicated above, consists of two opposed bellows, each of which is subjected to the pressure existing in one of the pair of temperature sensitive bulbs associated with each of the above valves, together with a system of linkages so arranged as to be responsive to the difference between the movements of the bellows. A suitable system of bellows and linkages such as outlined above is disclosed in Bauman United States Patent No. 2,440,628, issued April 27, 1948, and assigned to the assignee of the present invention. The linkages may operate the valves directly, as shown by the broken line connecting each valve to its respective control, or, if desired, may serve to actuate an electrical system (not shown) used to open and close the respective valves.

As will be evident from the foregoing description, both of the described embodiments accomplish and bring out the principle of the present invention, namely the substantially continuous removal of flash gas from the expanding refrigerant as soon as such flash gas is formed, and at the pressure and temperature at which it is formed. As shown in the preceding discussion this results in the lowest possible requirement of work input into the system, for a given amount of refrigeration, and therefore in the highest possible coefficient of performance.

While two embodiments of the invention have been described with particularity, it will be understood that the invention is susceptible of changes and modifications, without departing from the essential spirit thereof. For example means, other than the compressor arrangements shown, might be employed to effect the transition of the refrigerant from the gaseous to the liquid state. Similarly, as set forth above, control of the flow of the refrigerant toward the evaporator may be accomplished in a variety of ways. However, it will be evident that such changes and

modifications are contemplated, as may come within the scope of the appended claims.

I claim:

1. In a refrigeration system including a condenser and an evaporator connected in circuit, and an expansion chamber disposed in the circuit intermediate said condenser and evaporator, the method of operation which includes the following steps: delivering to said chamber liquid refrigerant derived from said condenser; isolating the refrigerant in said chamber from said evaporator; gradually decreasing the pressure of the liquid refrigerant in said chamber to provide for the incremental formation of flash gas and consequent gradual cooling of the liquid refrigerant; effecting separation of increments of flash gas upon formation thereof and under substantially the temperature and pressure conditions at which said increments are formed; and delivering to said evaporator the resultant cool, low-pressure liquid refrigerant in said chamber.

2. In a refrigeration system including a compressor, a condenser and an evaporator connected in circuit, and an expansion chamber disposed in the circuit intermediate said condenser and evaporator, the method of operation which includes the following steps: compressing volatile refrigerant fluid in the gaseous state; condensing said fluid; delivering to said chamber a predetermined quantity of liquid refrigerant derived from said condenser; isolating the refrigerant in said chamber from said evaporator; gradually decreasing the pressure of the liquid refrigerant in said chamber to provide for the incremental formation of flash gas and consequent cooling of the liquid refrigerant; effecting separation of increments of flash gas immediately upon formation and under substantially the temperature and pressure conditions at which said increments are formed, returning separated flash gas to the compressor under the influence of the suction of the compressor and recompressing said flash gas for subsequent re-use in the system; and delivering to said evaporator the cool, low-pressure liquid remaining in said chamber.

3. In a refrigeration system including a compressor, a condenser and an evaporator connected in circuit, and an expansion chamber disposed in the circuit intermediate said condenser and evaporator, the method of operation which includes the following steps: compressing volatile refrigerant fluid in the gaseous state; condensing the compressed fluid; delivering a predetermined quantity of said condensed fluid to said expansion chamber; isolating the refrigerant in said chamber from said evaporator, gradually reducing the pressure in said expansion chamber under the influence of the suction of said compressor to provide for the incremental formation of flash gas in said chamber and consequent cooling of the liquid refrigerant; effecting separation of each increment of flash gas upon formation thereof and under substantially the temperature and pressure conditions at which each increment is formed; recompressing the said separated flash gas for subsequent re-use in the system; and vaporizing the remaining cool, low-pressure liquid in heat exchange relation with the evaporator.

4. In a refrigeration system including a condenser and an evaporator connected in circuit, means for delivering liquid refrigerant from said condenser to said evaporator at predetermined conditions of pressure and temperature, said means comprising: apparatus for gradually de-

creasing the pressure of the liquid refrigerant prior to its delivery to said evaporator to provide for incremental formation of flash gas and consequent cooling of the liquid refrigerant; means for effecting separation of increments of flash gas immediately upon formation and under substantially the temperature and pressure conditions at which said gas is formed; and valve means effective to close off said chamber from said evaporator during cooling of the liquid refrigerant and to initiate substantially unrestricted delivery of the resultant cool, low-pressure liquid to said evaporator.

5. In a refrigeration system including a compressor, a condenser and an evaporator connected in circuit, means for delivering liquid refrigerant from said condenser to said evaporator at predetermined conditions of pressure and temperature, said means comprising: apparatus for gradually decreasing the pressure of the liquid refrigerant under the influence of the suction of said compressor and prior to the delivery of the refrigerant to said evaporator, t o provide for formation of flash gas and consequent cooling of the liquid refrigerant, the compressor operating to effect separation of increments of flash gas upon formation and under substantially the temperature and pressure conditions at which each said increment is formed; and means effective to prevent flow of refrigerant from said chamber to said evaporator during cooling and to initiate substantially unrestricted delivery of the resultant cool, low-pressure liquid refrigerant to said evaporator.

6. In a refrigeration system including a compressor, a condenser and an evaporator connected in circuit, means for delivering liquid refrigerant from said condenser to said evaporator at predetermined conditions of pressure and temperature, said means comprising: an expansion chamber disposed in the circuit intermediate said condenser and said evaporator and within which gradual cooling of the liquid refrigerant and gradual reduction of its pressure takes place; means for removing increments of flash gas from said chamber and for returning each said increment to said compressor under substantially the pressure and temperature conditions at which it is formed; means for isolating the contents of said chamber from said evaporator during formation of flash gas; and means for delivering the resultant cool, low-pressure liquid refrigerant in said chamber to said evaporator.

7. In a refrigeration system including a compressor, a condenser and an evaporator connected in circuit, means for delivering liquid refrigerant from said condenser to said evaporator at predetermined conditions of pressure and temperature, said means comprising: an expansion chamber disposed in the circuit intermediate said condenser and said evaporator and within which cooling of the refrigerant and reduction of its pressure takes place; valve means for delivering to said chamber a predetermined quantity of liquid refrigerant derived from said condenser; conduit means interconnecting said chamber and said compressor to provide for separation of the flash gas formed in said chamber and the return of the same to said compressor under the influence of the suction of the compressor, such separation and return of each increment of flash gas being effected upon formation thereof and under substantially the temperature and pressure conditions at which each such increment is formed; means for isolating the contents of said chamber from said evaporator during formation of flash gas; and means providing for delivery to said evaporator of the remaining cool, low-pressure liquid refrigerant in said chamber.

8. A system in accordance with claim 7, and further characterized in that said conduit means is of substantially non-restrictive dimensions.

9. A system in accordance with claim 7, and further characterized in that said last means includes valve means responsive to a predetermined temperature differential between said evaporator and said chamber.

10. In a refrigeration system, comprising: a compressor; a condenser; a storage tank; an expansion chamber; and an evaporator, said compressor, condenser, tank, chamber and evaporator being disposed in flow circuit; valve means disposed between said storage tank and said expansion chamber; and float means disposed within said storage tank and operatively connected to said valve means to maintain said valve means closed, whereby the flow of refrigerant from said storage tank to said expansion chamber is interrupted until a predetermined quantity of refrigerant has accumulated in said storage tank, said float means being arranged to sense the accumulation of said predetermined quantity of refrigerant and to open and maintain fully open said valve means in response to said accumulation, whereby the substantially unimpeded flow of refrigerant from said storage tank to said expansion chamber is re-established and substantial emptying of said storage tank is effected and said float means being further arranged to sense the substantial emptying of said predetermined quantity of refrigerant from said storage tank and to close and maintain closed said valve means in response to said emptying of said storage tank, whereby renewed accumulation of said predetermined quantity of refrigerant is initiated.

11. In a refrigeration system, comprising a compressor; a condenser; a storage tank; an expansion chamber, said expansion chamber being insulated to prevent substantial heat exchange between the contents of said chamber and the medium surrounding it; and an evaporator, said compressor, condenser, tank, chamber and evaporator being disposed in flow circuit, first valve means disposed between said storage tank and said expansion chamber and arranged so as to open in response to accumulation of a predetermined quantity of liquid refrigerant derived from said condenser and to admit into said expansion chamber said quantity of liquid refrigerant; conduit means connecting said expansion chamber to the suction side of said compressor; second valve means disposed between said expansion chamber and said evaporator; a pair of temperature sensitive elements disposed to sense, respectively, the temperature of the contents of said expansion chamber and of said evaporator; control means responsive to said temperature sensitive elements and effective to close and maintain closed said second valve means when the temperature of the contents of said expansion chamber exceeds the temperature of the contents of said evaporator by at least a predetermined differential, and to open, and maintain open, said second valve means when the temperature of the contents of said expansion chamber exceeds the temperature of the contents of said evaporator by less than a predetermined differential.

12. A refrigeration system, comprising: a compressor; a condenser; a receiver; an expansion chamber; and an evaporator, the aforesaid com-

ponents being connected in circuit and adapted for the undirectional flow of fluid refrigerant from said compressor, through said condenser, receiver and expansion chamber to said evaporator, and from thence returning to said compressor; valve means interposed between said receiver and said expansion chamber, said valve means being effective to interrupt flow of refrigerant from said receiver to said chamber until a predetermined quantity of refrigerant has accumulated in said receiver, said valve means being responsive to accumulation of said predetermined quantity to re-establish said flow of refrigerant from said receiver to said chamber, and said valve means additionally being responsive to substantial emptying of said receiver to again interrupt the flow of refrigerant from said receiver to said chamber, whereby renewed accumulation of said predetermined quantity of refrigerant is initiated; conduit means by-passing said evaporator and connecting said compressor with said chamber to provide for reduction of the pressure in said chamber and consequent production of flash gas and return of the same to the compressor under the influence of the suction of the latter; valve means interposed between said expansion chamber and said evaporator; and control apparatus providing for delivery to said evaporator of the liquid refrigerant in said chamber, said last means being effective to provide such delivery in accordance with the temperature differential between said evaporator and said chamber.

13. In a refrigerant flow system including a compressor and an evaporator: a pair of parallel refrigerant flow paths interconnecting said compressor and evaporator, each of said paths comprising a condenser and an expansion chamber in series flow circuit, valve and conduit means so disposed as interchangeably to connect one of said condensers to the suction side of said compressor and the other one of said condensers to the compression side of said compressor, whereby refrigerant is alternately removed from one of said paths, compressed, and introduced into the other path, and control means operatively connected to said valve means to effect interchange of said compressor connections in response to a predetermined accumulation of liquid refrigerant in that path connected to the compression side of said compressor prior to said interchange.

14. In a refrigerant flow system including a compressor, and an evaporator: a first refrigerant flow path interconnecting said compressor and evaporator, said first path including a first condenser and a first expansion chamber in series flow circuit; a second refrigerant flow path paralleling said first path and interconnecting said compressor and evaporator, said second path including a second condenser and a second expansion chamber; means arranged to connect one of said paths to the suction side of said compressor and the other path to the compression side of said compressor, said compressor connections being interchangeable; first valve means disposed intermediate said first expansion chamber and said evaporator; second valve means disposed intermediate said second expansion chamber and said evaporator; and first and second control means operatively associated, respectively, with said first and second valve means, and responsive to predetermined temperature differentials between each corresponding expansion chamber and said evaporator to open and close said associated valve means.

ELMER W. ZEARFOSS, Jr.

REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
1,106,287	Doelling	Aug. 4, 1914
1,718,312	Shipley	June 25, 1929
1,994,037	Gay	Mar. 12, 1935
2,094,565	Wolfert	Sept. 28, 1937
2,299,811	Feicht	Oct. 27, 1942
2,403,220	Hintze	July 2, 1946
2,454,537	Atchison	Nov. 23, 1948
2,500,688	Kellie	Mar. 14, 1950