

(12) **EUROPEAN PATENT APPLICATION**

(21) Application number: 86305218.9

(51) Int. Cl.⁴: **F 25 B 25/00**

(22) Date of filing: **07.07.86**

(30) Priority: **10.07.85 GB 8517445**

(43) Date of publication of application:
14.01.87 Bulletin 87/3

(84) Designated Contracting States:
BE DE FR IT NL SE

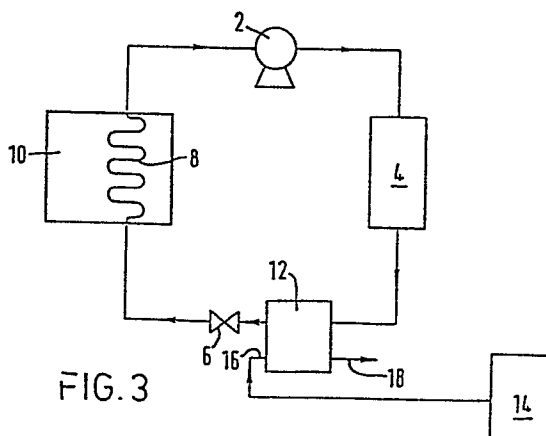
(71) Applicant: **The BOC Group plc**
Chertsey Road
Windlesham Surrey GU20 6HJ(GB)

(72) Inventor: **Miller, Jeremy Paul**
Jasmine Cottage, Littleworth
Near Farringdon, Oxfordshire(GB)

(74) Representative: **Wickham, Michael et al,**
c/o Patent and Trademark Department The BOC Group plc
Chertsey Road
Windlesham Surrey GU20 6HJ(GB)

(54) **Refrigeration method and apparatus.**

(57) In a mechanical refrigeration apparatus of the vapour compression kind, comprising a compressor 2, condenser 4, expansion valve 6, and evaporator 8, a sub-cooler 12 is employed to reduce the temperature of the condensed refrigerant to a value at which at least 90% by volume and preferably all of it remains in the liquid phase on passage through the expansion valve 6. The sub-cooler 12 is cooled by means of a supply of liquid nitrogen from vessel 14. Alternatively, liquid nitrogen is evaporated in a circulating heat exchange atmosphere in the sub-cooler 12. Increased heat load in chamber 10 may thus be met without there necessarily being any increase in the evaporating temperature and pressure.



DESCRIPTION OF THE INVENTION

REFRIGERATION METHOD AND APPARATUS

This invention relates to a refrigeration method and apparatus.

Mechanical refrigeration is widely used for industrial purposes, for example the refrigeration or freezing of food. Conventional mechanical refrigeration plant operates a cycle in which refrigerant vapour is compressed, the resulting compressed gas is condensed by heat exchange with cold water or air, the condensed refrigerant is expanded through an expansion valve, and the resultant expanded fluid is evaporated, whereby heat is extracted from (and hence refrigeration provided for) a body or fluid to be refrigerated. The evaporated refrigerant provides the vapour for compression so that the mechanical refrigeration cycle is endless.

Such a refrigeration plant is conventionally operated with the inlet of the compressor receiving refrigerant vapour at its evaporation temperature under the prevailing pressure and with the expansion valve receiving condensed refrigerant at its condensation temperature. It is found that the volumic work of compression and hence the power required to drive the compressor increases markedly with increasing condensation temperature without there being a corresponding increase in the volumic refrigeration effect. Condensation is typically effected by heat exchanging the refrigerant vapour with water or air at ambient temperatures. In summer, therefore, when ambient temperatures rise considerably above average, difficulties are typically created in maintaining a given condensation temperature and pressure, and frequently the operator of the mechanical refrigeration plant has to allow the condensation temperature to rise at the expense of a markedly increased power consumption. Moreover, since an increased condensation pressure

entails an increased operating pressure in the compressor, compressor wear tends to be accelerated and as a result a more frequent servicing of the compressor is required to maintain pressure.

Another problem that arises with conventional mechanical refrigeration plant is that the plant is relatively inflexible in meeting varying refrigeration demands. It is found that the volumic refrigeration effect falls rapidly with diminishing evaporating temperatures. Accordingly, the refrigeration capacity produced by a compressor of the positive displacement type with a fixed swept volume rate becomes smaller with decreasing temperatures and this means that to produce a certain refrigeration capacity, more expensive compression plant is required for low evaporating temperature than for high ones. Where the refrigeration plant is, say, employed to freeze food, an attempt to accelerate the food freezing process or to increase the weight of food frozen per unit time by reducing the evaporating temperature will result in a loss of refrigeration capacity and thus the attempt will be self-defeating unless the plant is normally operated with the necessary spare capacity. One practical consequence of this lack of flexibility is that the operator of the plant is often unable to meet temporary increases in the demand for frozen food.

It has previously been proposed to enhance the performance of mechanical refrigeration plant by spraying liquid nitrogen into a chamber refrigerated by the evaporating refrigerant. This proposal has been found in practice to have a number of major drawbacks. First, there is tendency for the liquid nitrogen to take over entirely from the mechanical refrigeration plant so that substantially all the refrigeration load is met by the nitrogen with the result that the process becomes uneconomic and damage may be caused to the mechanical refrigeration plant. This phenomenon may at least in part be caused by ice being deposited on the external heat exchange surfaces of the evaporator, but in any case, difficulties do arise in controlling the liquid nitrogen introduction such that the refrigerative capacity of the mechanical refrigeration plant is fully used. Moreover, where the chamber is of the kind that permits or requires human beings to have access to

0208526

it, the creation of a nitrogen-enriched atmosphere as a result of the evaporation in the chamber of the liquid nitrogen sprays tends to give rise to a safety hazard. Accordingly, attempts to employ liquid nitrogen in such a way have ended in failure.

It is an aim of the present invention to provide a method and apparatus that makes it possible to enhance or augment refrigeration provided mechanically, the enhancement being provided by utilising a liquefied atmospheric gas.

In its broadest aspect the present invention comprises sub-cooling the condensed refrigerant by heat exchange with a second fluid typically comprising a liquefied atmospheric gas or cold gas evolved therefrom.

According to the present invention there is provided a mechanical refrigeration method comprising the steps of compressing a refrigerant in its vapour state, condensing the refrigerant, sub-cooling the condensed refrigerant to a chosen temperature, reducing the pressure to which said sub-cooled refrigerant is subjected, evaporating the refrigerant and thereby extracting heat from a heat load to be refrigerated, and returning the evaporated refrigerant for compression, wherein sub-cooling of the refrigerant is effected by direct or indirect heat exchange with a second fluid comprising a cryogenic medium or cold gas formed by evaporating a cryogenic liquid and said temperature is chosen such that at least 90% by volume of the sub-cooled liquid refrigerant remains in the liquid state upon completion of the pressure reduction.

The invention also provides mechanical refrigeration apparatus comprising means for compressing a vaporous refrigerant, a condenser for condensing the compressed refrigerant having an inlet in communication with the outlet of the compressor, means for sub-cooling the condensed refrigerant by direct or indirect heat exchange with a second fluid comprising a cryogenic medium or cold gas formed by evaporating a cryogenic liquid, means for reducing the pressure to which the sub-cooled refrigerant is subjected, an evaporator having an inlet in communication with said sub-cooling

0208526

means and an outlet in communication with the inlet of the compression means, and means for controlling the operation of the sub-cooling means such that in operation at least 90% by volume of the sub-cooled liquid remains in the liquid state upon completion of the pressure reduction.

The method and apparatus according to the present invention makes it possible to choose a relatively low evaporating temperature (say -40°C or below) of the refrigerant without loss of volumic refrigeration efficiency. Typically, the degree of sub-cooling is chosen such that any temperature drop undergone by the refrigerant on pressure reduction is less than 10°C (and more typically less than 5°C).

Preferably, said temperature is chosen such that all the sub-cooled liquid remains in a liquid state on completion of the pressure reduction, i.e. there is no temperature drop on pressure reduction.

The second fluid is preferably an atmospheric gas that is typically heat exchanged in its liquid state with the refrigerant in order to sub-cool the refrigerant to the desired temperature, or, alternatively, the atmospheric gas in its liquid state may be heat exchanged with an intermediate fluid which is in turn heat exchanged with the refrigerant so as to effect the necessary sub-cooling. In one example of the method according to the invention, the atmospheric gas in its liquid state is evaporated in a gaseous atmosphere, typically circulating, which is heat exchanged with the refrigerant so as to effect the desired sub-cooling. In another example, the liquid nitrogen is used to cool a bath of relatively low freezing point organic liquid (e.g. methanol).

The method according to the present method preferably additionally includes the steps of sensing the temperature of said second fluid at a location downstream of where heat exchange of said second fluid with said refrigerant takes place (or a parameter dependent upon such temperature), controlling the rate at which cryogenic medium or cryogenic liquid is brought into heat exchange relationship with

said refrigerant (or is evaporated in said second fluid) to maintain said temperature at a set chosen value, sensing the temperature of the refrigerant at a location downstream of where its sub-cooling is completed (or a parameter dependent upon such temperature), and adjusting the setting of said value in response to any deviations in the temperature of the sub-cooled refrigerant from said chosen temperature, whereby the temperature of the sub-cooled refrigerant is returned to said chosen temperature.

The method according to the present invention may be performed on an established mechanical refrigeration plant by retro-fitting a suitable heat exchanger. Alternatively, apparatus may be built to custom to perform the method according to the invention.

The method and apparatus according to the invention will now be described by way of example with reference to the accompanying drawings, in which:

Figure 1 is a schematic diagram illustrating the essential parts of a conventional mechanical refrigeration circuit;

Figure 2 is a graph illustrating the changes in pressure and the specific enthalpy that the refrigerant undergoes around the circuit shown in Figure 1;

Figure 3 is a simplified schematic drawing illustrating mechanical refrigeration apparatus in accordance with the invention.

Figure 4 is a graph illustrating the variation of pressure and specific enthalpy for the refrigerant as it flows round the apparatus shown in Figure 3;

Figure 5 is a schematic drawing illustrating a conventional mechanical refrigeration plant that has been converted to perform the method according to the invention by retro-fitting a heat exchanger for effecting sub-cooling of the refrigerant, and

Figure 6 is a schematic side elevation of the heat exchanger for use in the plant shown in Figure 5.

In the respective Figures of the drawings, like parts are indicated by the same reference numerals.

Figure 1 of the drawings illustrates a conventional mechanical refrigeration cycle of the vapour compression kind. A compressor 2 receives vapour of a refrigerant at its saturation temperature at the pressure prevailing on the suction side of the compressor 2. The compressor 2 compresses the vapour to a suitable elevated pressure. The resultant compressed, gaseous, refrigerant is then condensed in a condenser 4. The necessary cooling for the condenser 4 is typically provided by water or air. Both the refrigerant and the operating pressure are often chosen such that the condensation takes place at ambient temperature, or a temperature only slightly below ambient. The resulting condensate at the temperature of saturated vapour at the pressure prevailing in the condenser 4 is then passed through a throttle valve (or expansion valve) 6 and is thereby reduced in pressure. As a result of this step, there is a reduction in the temperature of the fluid. The refrigerant is then passed through an evaporator 8 in heat exchange relationship with a heat load (not shown) in the chamber 10. Liquid refrigerant is evaporated in the evaporator 8 (typically taking the form of one or more coils formed of metal having a high thermal conductivity) and thus provides refrigeration for the heat load. The refrigerant vapour that is thus formed is that which is compressed: the refrigerant thus passes around an endless circuit.

Figure 2 illustrates the enthalpy and pressure changes that take place as the refrigerant flows around the circuit shown in Figure 1. At the inlet to the compressor 2, the refrigerant has a specific enthalpy h and a pressure P that place it at point a on the graph shown in Figure 2. Compression increases its specific enthalpy and pressure and at the outlet of the compressor 2 the refrigerant has a specific enthalpy and pressure that place it at point b. The subsequent condensation is substantially isobaric but is accompanied by a reduction in the enthalpy of the refrigerant such that it is at point c when it reaches the outlet of the condenser 4. At point c a change of phase from saturated vapour to liquid takes place and this

change has been completed when the refrigerant leaves the condenser . Passage of the liquid refrigerant through the valve 6 causes an isenthalpic pressure reduction to take place and the condition of the refrigerant at the inlet to the evaporator 8 is represented by point d vertically below point c. Evaporation of liquid refrigerant in the evaporator 8 returns the refrigerant to point a. If desired, it can be arranged for point a to lie a little to the right of the position in which it is shown to ensure that no liquid enters the compressor 2. The specific refrigerating effect provided by the refrigerant is thus $h_a - h_c$ where h_a is the specific enthalpy at point a and h_c is the specific enthalpy at point c. The volumic refrigerating effect provided by the refrigerant is $(h_a - h_c)/V_a$ where V_a is the specific volume of the refrigerant at point a (i.e. the inlet of the compressor 2). The specific refrigerating effect is related to the amount of refrigeration provided per unit mass of refrigerant by the refrigeration circuit shown in Figure 1, whereas the volumic refrigeration effect is related to the refrigeration provided per unit power consumed by the compressor 2 and is thus a function of the efficiency with which the refrigeration is provided.

In practice with many conventional vapour compression refrigeration systems, as the heat load increases, so the rate of evaporating liquid refrigerant increases. This causes a rise in the evaporating pressure and hence a rise in the evaporating temperature which may therefore become unacceptably high. It should be appreciated, however, that an increase in the evaporating pressure also results in an increase in the mass of refrigerant vapour that is handled by the compressor per unit time and thus the refrigerant capacity of the apparatus may be increased with increasing evaporating temperature.

We believe that a key to optimising the performance of the refrigeration circuit resides in obtaining improved operating parameters at the inlet to the valve 6. In conventional operation, when the refrigerant liquid passes through the valve 6 a sizeable proportion of it flashes off as gas. The exact proportion will depend inter alia on the choice of refrigerant but it may typically

0208526

be in the order of 15 to 20% by volume of the liquid entering the expansion valve. Consider now the effect of this flash gas on the operation of the condenser 8. The liquid in the evaporator 8 makes a much greater contribution to the refrigeration of the heat load than the gas, as the former contributes its latent heat of condensation to the refrigeration. Accordingly, the greater the proportion of liquid in the fluid entering the evaporator, the greater will be the increase in the specific enthalpy of the refrigerant in the evaporator and thus the greater the specific refrigerating effect ($h_a - h_c$) will become (see Figure 2) assuming that there is a sufficient heat load in the chamber 10.

The method according to the present invention enables the refrigerant to enter the evaporator 8 entirely in the liquid state. This end can be achieved relatively simply without necessitating the provision of extra compression and without the need to reduce the flow rate of refrigerant through the circuit by diverting refrigerant therefrom. As shown in Figure 3, a sub-cooler 12 in the form of a heat exchanger is provided intermediate the condenser 4 and the valve 6. The heat exchanger 12 has an inlet 16 for liquid nitrogen communicating with a source 14 of liquid nitrogen and an outlet 18 for nitrogen vapour. The heat exchanger 12 may for example be of the shell and tube type. Alternatively, it may be of a kind in which cold gas is employed to effect the heat exchange, such cold gas being formed by evaporating liquid nitrogen or other liquid atmospheric gas (for example liquid air). Sub-cooled liquid refrigerant thus enters the expansion valve 6. The liquid refrigerant is sub-cooled to a temperature such that at the prevailing pressure downstream of the expansion valve 6 not more than 5% by volume of the liquid entering the expansion valve 6 is converted to flash gas, and preferably such that none of the liquid entering the expansion valve 6 is converted to flash gas.

Typically, in many food freezing processes it is desirable to evaporate the liquid refrigerant at a temperature substantially below 0°C . However, in a conventional system as shown in Figure 1 of the drawings the liquid entering the expansion valve 6 will

0208526

typically be at a temperature not substantially below 0°C . Thus, the passage through the expansion valve 6 is utilised to create the necessary evaporating temperature. Accordingly, flash gas is formed the proportion of flash gas that is formed increases with increasing temperature drop. However, by using the method according to the invention the proportion of flash gas that is formed can be kept to less than 10% by volume of the refrigerant entering the expansion valve even at low evaporating temperatures. Typically, in accordance with the invention, the temperature drop across the expansion valve 6 can be eliminated and thus no flash gas formed.

Referring now to Figure 4 of the accompanying drawings, this is a graph that represents the enthalpy and pressure changes that take place as the refrigerant flows around the circuit shown in Figure 3. At the inlet to the compressor 2, the refrigerant has a specific enthalpy h and a pressure P that places it at point a' on the graph shown in Figure 4. Compression increases its specific enthalpy and pressure and at the outlet of the compressor 2 the refrigerant has a specific enthalpy and pressure that place it at point b' . The subsequent condensation is substantially isobaric but is accompanied by a reduction in the enthalpy of the refrigerant such that it is at point x' when it reaches the outlet of the condenser 4. At point x' a change of phase from saturated vapour to liquid takes place and this change is completed before the refrigerant leaves the condenser. The liquid refrigerant is then sub-cooled in the heat exchanger 12. The enthalpy of the liquid refrigerant is thus reduced to a value represented by point c' . Passage of the sub-cooled liquid refrigerant through the expansion valve 6 causes an isenthalpic pressure reduction to take place. The pressure drop that takes place is represented by the line $c' d'$. Point d' represents the enthalpy and pressure of the liquid at the inlet to the evaporator 8. Evaporation of the liquid refrigerant in the evaporator 8 returns the refrigerant to Point a' . In comparison with the graph shown in Figure 2, in the event that the pressures at the inlet and outlet of the compressor 2 remain unaltered, the line $d' a'$ will be longer than the corresponding line $d a$. In other words, the increase in enthalpy and hence the increase in the

0208526

refrigeration provided by the refrigerant as it passes through the evaporator 8 will be greater in operation of the circuit shown in Figure 3 than in operation of the circuit shown in Figure 1. The specific refrigerating effect $(h_a' - h_c')$ associated with the Figure 3 circuit is therefore greater than the specific refrigerating effect $(h_a - h_c)$ associated with the Figure 1 circuit. Moreover, since the pressure at the inlet to the compressor 2 in Figure 3 remains unaltered as compared to that associated with the compressor 2 of the circuit shown in Figure 1, the volumic refrigerating effect $(h_a' - h_c')/V_a'$ associated with the Figure 3 circuit is similarly greater than the volumic refrigerating effect $(h_a - h_c)/V_a$ associated with the Figure 1 plant. The total increase in the specific refrigerating effect is therefore $h_c' - h_x'$ and the percentage increase in the specific refrigerating effect is $(h_c' - h_x')/(h_a' - h_x') \times 100\%$. Similarly when V_a equals V_a' , the total increase in the volumic refrigerating effect is $\frac{h_c' - h_x'}{V_a'}$ and

the percentage increase is $(h_c' - h_x')/(h_a' - h_x') \times 100\%$.

Typically, increases in the specific and volumic refrigerating effects of at least 10% may be achieved when ammonia is the refrigerant and greater increases can be achieved when a Freon (RTM) is the refrigerant. The percentage increase gained typically approximates to the percentage increase in the volume of liquid refrigerant leaving the valve when the liquid refrigerant is sub-cooled to the maximum temperature at which the liquid remains entirely in the liquid phase. Moreover, increased heat load may be met without an increase in evaporating temperature since the sub-cooling that is effected in accordance with the invention inherently provides extra refrigeration.

The condensed refrigerant may, if desired, be sub-cooled to a temperature below the maximum at which the refrigerant remains entirely in the liquid phase. Thus the temperature at which the

0208526

refrigerant leaves the expansion valve 6 may be lower than that at which it starts to evaporate. This extra sub-cooling makes available additional refrigeration for the chamber 10. The liquid refrigerant does not in such an example start to evaporate immediately upon its entry into the evaporator 8. As it flows through the evaporator 8 so the liquid refrigerant is raised in temperature until it begins to evaporate.

In the practical operation of the method according to the present invention, there will typically arise a point on the pressure-specific enthalpy graph in Figure 4 where any additional sub-cooling will have the effect of reducing the evaporating pressure in the evaporator 8 and the suction pressure of the compressor 2. Although such additional sub-cooling will produce an increase in the specific refrigerating effect over its value at the said point, there will be a decrease in the volumic refrigerating effect in consequence of the fall in the suction pressure of the compressor 2. Whether and to what extent such a fall in the volumic refrigerating effect is tolerable will vary according to circumstances, and any such fall may be offset against the increase in the volumic refrigerating effect gained by the sub-cooling of the liquid refrigerant to said point. It is desirable to control the operation of the compressor 2 in the event of a fall in evaporating pressure such that there is no corresponding fall in the pressure at the outlet of the compressor 2 since any such fall would lead to a concomitant reduction in the condensing temperature and thus a loss of efficiency.

It is possible to make various changes in the operation and layout of the refrigeration circuit shown in Figure 3. For example, it is possible to operate the condenser 4 such that some degree of sub-cooling takes place in the condenser 4, the sub-cooling being completed in the heat exchanger 12. Indeed, it is possible to combine the heat exchanger 12 with the condenser 4 such that condensation and sub-cooling to the desired temperature takes place in a single unit. Such an example of the method according to the invention is however not preferred. It is not essential to use liquid nitrogen in order to provide the necessary refrigeration for

the sub-cooler. A liquefied noble gas such as liquid argon or cold gas evolved from such liquid may be employed instead. Another alternative is to employ liquid air or liquid oxygen or cold gas evolved therefrom. If the refrigerant is ammonia, liquid carbon dioxide or its cold vapour may be employed to effect the necessary degree of sub-cooling.

It is not essential to use as the pressure reduction means a valve such as the expansion valve 6, that is a device which has a passage therethrough and means for opening and closing the passage. Instead, a simple orifice plate or other such means may be employed.

Referring now to Figure 5 of the drawings, there is illustrated a typical commercial plant for mechanically refrigerating a chamber 10 which has been fitted with a heat exchanger 12 for effecting sub-cooling in accordance with the invention. Many of the features of this plant are identical to those described with reference to Figure 3 and will not be described herein again. The plant has however a number of features not included in the circuit shown in Figure 3. Thus, the plant shown in Figure 5 is provided with an oil separator 20 intermediate the compressor 2 and condenser 4. The purpose of the oil separator 12 is to separate any oil entrained in the gas leaving the compressor 2 and to return such oil to the compressor 2. The plant shown in Figure 5 also has a liquid receiver 22 intermediate the sub-cooler 12 and the expansion valve 6. The liquid receiver enables a reservoir of liquid refrigerant to be stored in the circuit during periods in which the compressor 2 is not operated. The sub-cooler 12 may, instead of being provided intermediate the condenser 4 and the liquid receiver 22, be positioned upstream of the expansion valve 6 and downstream of the receiver 22. In addition, the plant shown in Figure 5 is provided with a surge drum 24 associated with the evaporator 8. The outlet of the expansion valve 6 communicates with a lower inlet to the surge drum 24 and the outlet of the evaporator 8 communicates with another inlet above the lower inlet. The surge drum 24 further has an outlet for vapour above said inlets, which outlet communicates with the inlet of the compressor 2. In operation, typically not all

0208526

of the liquid refrigerant is evaporated in the evaporator, unevaporated liquid being returned to the surge drum 24. Thus a level of liquid refrigerant is maintained in the surge drum 24. The surge drum 24 is provided with upper and lower level sensors 26 with which the expansion valve 6 is operatively associated. The arrangement is such that with normal heat loads in the chamber 10, and without operation of the sub-cooler 12, the liquid level in the surge drum 24 remains below that of the upper sensor 26 so that the expansion valve 6 remains open. In the event of the heat load in the chamber 10 becoming less, there is a tendency for more liquid refrigerant to be returned to the surge drum 24. The level of the liquid in the surge drum 24 therefore rises to that of the upper sensor 26. The sensor 26 then generates a signal effective to close the expansion valve 6. The surge drum 24 then becomes the only source of liquid refrigerant for the evaporator 8 so that the level of liquid refrigerant in the surge drum 24 falls to that of the lower sensor 26. At this juncture, the expansion valve 6 is caused to open and normal operation of the refrigeration circuit resumes.

When sub-cooling of the liquid refrigerant leaving the condenser 4 is employed in accordance with the present invention, a greater rate of liquid flow out of the expansion valve 6 takes place. This greater rate of liquid formation is utilised by employing a larger heat load in the chamber 10 than would otherwise be employed in efficient operation of a refrigeration plant (at a chosen evaporation temperature) operated conventionally, that is operated without the large degree of sub-cooling that is provided by the method according to the present invention.

It will be appreciated that the plant shown in Figure 5, and in particular the operation of the sub-cooler 12, will need to be able to cope with intermittent closures of the expansion valve 6 and thus care needs to be taken to avoid freezing the liquid refrigerant during such closure periods. It is in any event desirable that there should be no risk of the liquid refrigerant being cooled in the sub-cooler 12 to a temperature at which it begins to freeze. A heat exchange system able to be used without such risk in the plant shown in Figure 5 is illustrated in Figure 6.

Referring to Figure 6, the heat exchanger includes a chamber 30 provided with an inlet 32 for liquid refrigerant located at the top of the chamber 30 and communicating with two arrays of vertical, finned, heat exchange tubes 34 and 36, the array 36 being disposed vertically below the array 34. The arrangement is that feed from the inlet 32 to the first array 34 of heat exchange tubes is by way of a tubular heat exchange plate 38 and feed of liquid from the array 34 of heat exchange tubes to the array 36 of the heat exchange tubes is by way of an intermediate tubular heat exchange plate 40. The plates 38 and 40 are arranged generally perpendicularly to the tubes 34 and 36 and are effective to extend the flow path of the liquid thereby facilitating the heat exchange between it and cold gas that circulates within the chamber 30. This arrangement offers the advantage of reducing the overall space requirements for the heat exchanger. The outlets of the heat exchange tubes 36 forming the lower array communicate with a third tubular heat exchange plate 42, similar to the plates 38 and 40, and in turn communicating with an outlet 44 for sub-cooled liquid.

The heat exchanger shown in Figure 6 is provided with a conduit 46 having a valve 48 disposed therein at a location outside the chamber 30. The conduit 46 extends into the chamber 30 and terminates in a liquid nitrogen spray nozzle 50. The conduit 46 communicates with a source of liquid nitrogen (not shown in Figure 6).

A fan 52 is located within the chamber 30 and is able to be driven by means of an external motor 54. The chamber is provided with deflectors or baffles 56 which promote circulation of cold gas within the chamber 30 as shown by the arrows 58 when the fan 52 is operated.

In operation, liquid nitrogen is sprayed into the chamber 30 through spray nozzle 50. The resulting evaporated nitrogen merges with the circulating gas flow created by operation of the fan 52. The cold nitrogen gas that is thus formed passes over the heat exchange tubes 34 and 36 and over the heat exchanger plates 38, 40 and 42 thereby effecting the required sub-cooling of the liquid refrigerant

entering the inlet 32 from the condenser 4. Cold nitrogen gas leaves the chamber 30 through an outlet 60. It is then typically vented to the atmosphere at a location where no hazard to people is created. If desired, the cold nitrogen leaving the chamber 30 via the outlet 60 may be used to provide additional cooling for the condenser 4 so as to utilise any residual cold in the vented nitrogen.

The heat exchanger shown in Figure 6 is provided with temperature control means to enable the spraying of liquid nitrogen through the nozzle 50 to be controlled so as to enable the temperature of the sub-cooled liquid leaving the heat exchanger through the outlet 34 to be kept at or close to a chosen temperature. In order to facilitate the provision of a stable control system, it is preferred not to control the operation of the valve 48 and hence the spraying of the liquid nitrogen through the nozzle 50 directly by means of a temperature sensor located in the outlet 44. Instead, it is preferred to employ a cascade control system as illustrated in Figure 6. A first temperature sensor 64 is adapted to transmit a signal representative of the temperature of the liquid in the outlet 44 to a master controller 66 which generates a control signal and transmits it to a slave controller 68 so as to select a set point for the slave controller 68. The slave controller 68 also receive an input from a second temperature sensor 70 located in the outlet 60. The slave controller 68 generates a control signal for the valve 48. Any changes in the temperature of the liquid condensate entering the inlet 32 of the heat exchanger result in the master controller 66 receiving a signal indicative of this change from the first temperature sensor 64. The master controller 66 then calls for an appropriate change in the rate of introduction of liquid nitrogen into the chamber 30 so as to provide more or less refrigeration as the case demands to return the sensed temperature to the chosen value. This is effected by the master controller altering the set point of the slave controller appropriately and the slave controller 68 then adjusting the position of the valve 48 to bring the temperature sensed by the second temperature sensor 70 into conformity with that required by the set point. Changes that

take place in the liquid nitrogen flow rate independently of any change in the set point of the slave controller 68 will result in a variation in the temperature sensed by the temperature sensor 70 at the outlet of the chamber 30 and will be corrected for by the slave controller 68 producing an appropriately adjusted control signal to the valve 48. A suitable cascade control system of this type for the use in association with the heat exchanger 12 shown in Figure 6 is available from Gulton Limited of Brighton under the trade mark OPUS 72.

The temperature control system may also be provided with means effective to close the valve 48 once the expansion valve 6 shown in Figure 5 is shut for any reason. Similarly, the temperature control system may be provided with means for reopening the valve 48 upon the expansion valve 6 being re-opened.

CLAIMS

1. A mechanical refrigeration method comprising the steps of compressing a refrigerant in its vapour state, condensing the refrigerant, sub-cooling the condensed refrigerant to a chosen temperature, reducing the pressure to which said sub-cooled refrigerant is subjected, evaporating the refrigerant and thereby extracting heat from a heat load to be refrigerated, and returning the evaporated refrigerant for compression, wherein sub-cooling of the refrigerant is effected by direct or indirect heat exchange with a second fluid comprising a cryogenic medium or cold gas formed by evaporating a cryogenic liquid, and said temperature is chosen such that at least 90% by volume of the sub-cooled liquid refrigerant remains in the liquid state upon completion of the pressure reduction.
2. A method as claimed in claim 1, in which any temperature drop undergone by the refrigerant on said pressure reduction is less than 10°C .
3. A method as claimed in claim 1 or claim 2, in which any temperature drop undergone by the refrigerant on said pressure reduction is less than 5°C .
4. A method as claimed in any one of the preceding claims, wherein said temperature is chosen such that all the sub-cooled liquid refrigerant remains in the liquid state upon completion of the temperature reduction.
5. A method as claimed in any one of the preceding claims, in which said temperature is lower than the temperature at which the liquid refrigerant evaporates.

6. A method as claimed in any one of the preceding claims, in which the liquid refrigerant evaporates at a temperature of or below minus 40°C.
7. A method as claimed in any one of the preceding claims, in which the second fluid is an atmospheric gas that is heat exchanged in its liquid state with the refrigerant in order to effect sub-cooling of the refrigerant to the chosen temperature.
8. A method as claimed in any one of claims 1 to 7, in which the second fluid is formed by evaporating an atmospheric gas in its liquid state in a circulating gaseous atmosphere.
9. A method as claimed in any one of the preceding claims, additionally including the steps of sensing the temperature of said second fluid at a location downstream of where heat exchange of said second fluid with said refrigerant takes place (or a parameter dependent upon such temperature), controlling the rate at which cryogenic medium or cryogenic liquid is brought into heat exchange relationship with said refrigerant (or is evaporated in said second fluid) to maintain said temperature at a chosen set value, sensing the temperature of the refrigerant at a location downstream of where its sub-cooling is completed (or a parameter dependent upon such temperature), and adjusting the setting of said chosen value in response to any deviations in the temperature of the sub-cooled refrigerant from said chosen temperature, whereby the temperature of the sub-cooled refrigerant is returned to said chosen temperature.
10. A method as claimed in any one of the preceding claims, in which the cryogenic medium or cryogenic liquid is liquid nitrogen.

11. Mechanical refrigeration apparatus comprising means for compressing a vaporous refrigerant, a condenser for condensing the condensed refrigerant having an inlet in communication with the outlet of the compressor, means for sub-cooling the condensed refrigerant by direct or indirect heat exchange with a second fluid comprising a cryogenic medium or cold gas formed by evaporating a cryogenic liquid, means for reducing the pressure to which the sub-cooled refrigerant is subjected, an evaporator having an inlet in communication with said sub-cooling means and an outlet in communication with the inlet of the compression means, and means for controlling the operation of the sub-cooling means such that in operation at least 90% by volume of the sub-cooled liquid remains in the liquid state upon completion of the pressure reduction.
12. Mechanical refrigeration apparatus as claimed in Claim 11, in which said heat exchange means includes a chamber through which at least one heat exchange tube for the refrigerant extends, means for circulating atmosphere within said chamber, and means for introducing cryogenic liquid into said atmosphere, whereby in use said cryogenic liquid evaporates in said atmosphere.
13. Mechanical refrigeration apparatus as claimed in Claim 11 or Claim 12, wherein said controlling means comprises a temperature sensor for sensing the temperature of said second fluid at a location downstream of where heat exchange of said second fluid with said refrigerant takes place (or means for sensing a parameter dependent upon such temperature), valve means for controlling the rate at which, in use, cryogenic medium or cryogenic liquid is brought into heat exchange relationship with said refrigerant (or is evaporated in said second fluid), a slave controller for controlling the position of said valve

means in response to signals from said temperature sensor, said slave controller having means for providing a set point temperature whereby in operation the slave controller adjusts said valve as necessary to maintain the temperature sensed by the temperature sensor at said set point temperature, a master controller having means for adjusting said set point, and another temperature sensor at a location downstream of where the sub-cooling of the refrigerant is completed (or means for sensing a parameter dependent upon such temperature) operatively associated with said master controller, whereby in operation the master controller is able to adjust as necessary the set point temperature of the slave controller to maintain the temperature to which the refrigerant is sub-cooled at a chosen value.

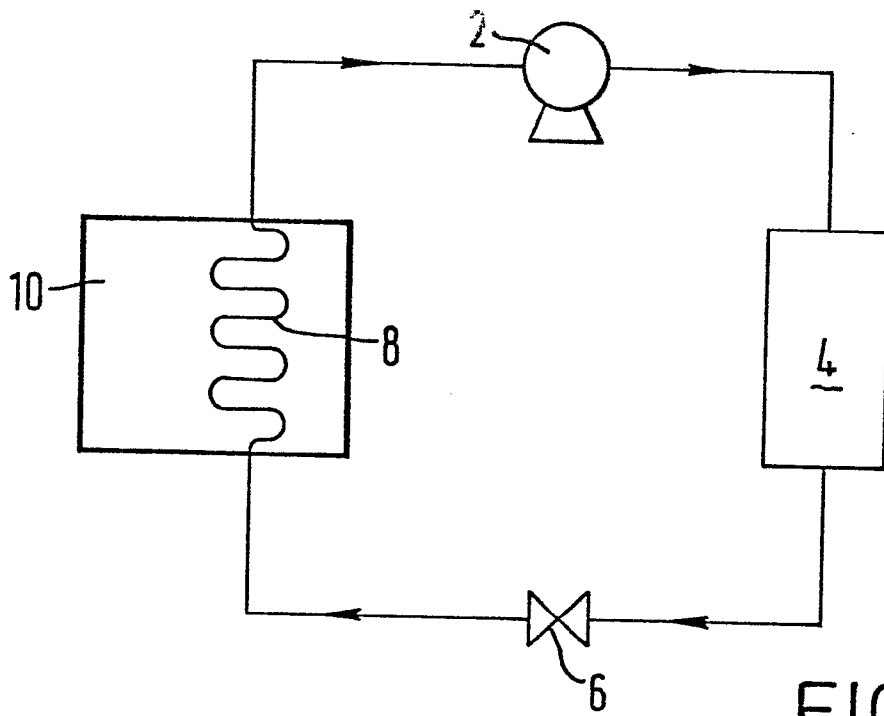


FIG. 1

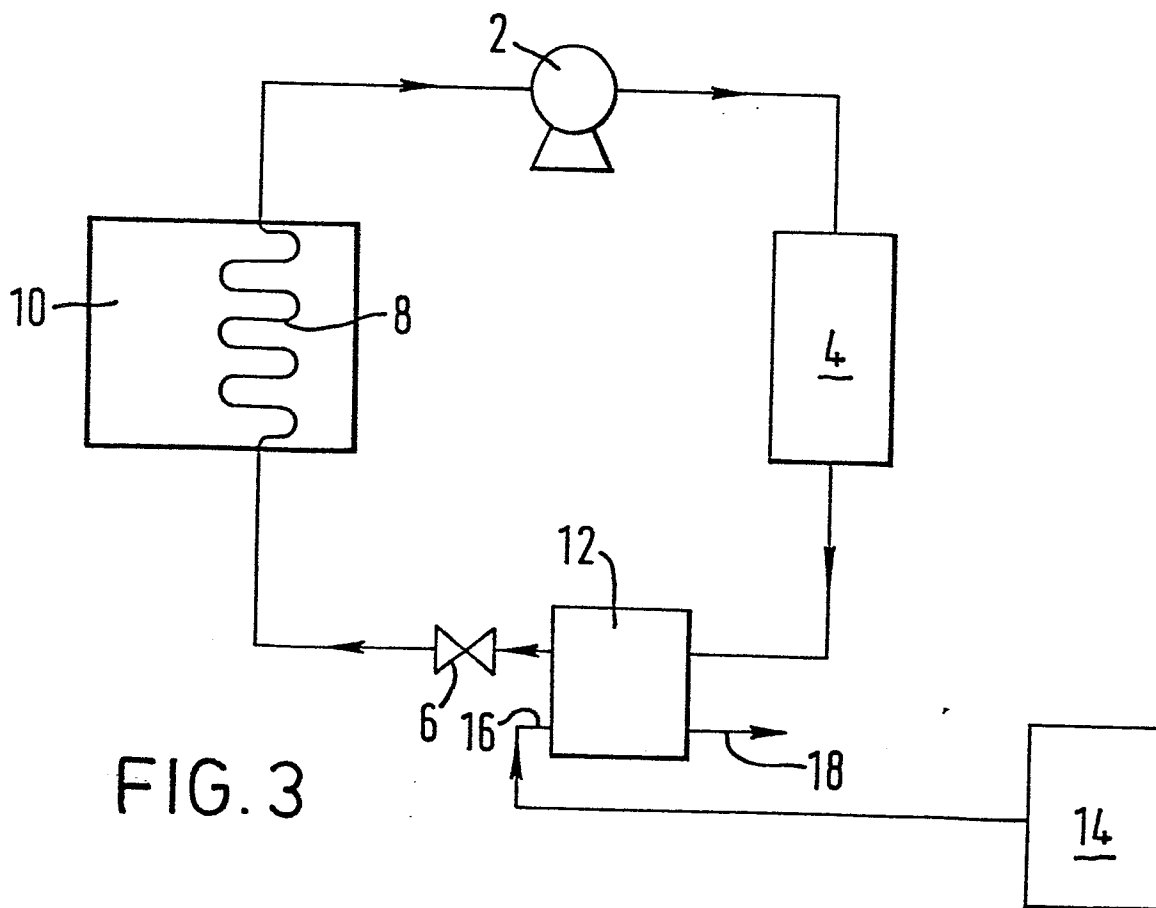


FIG. 3

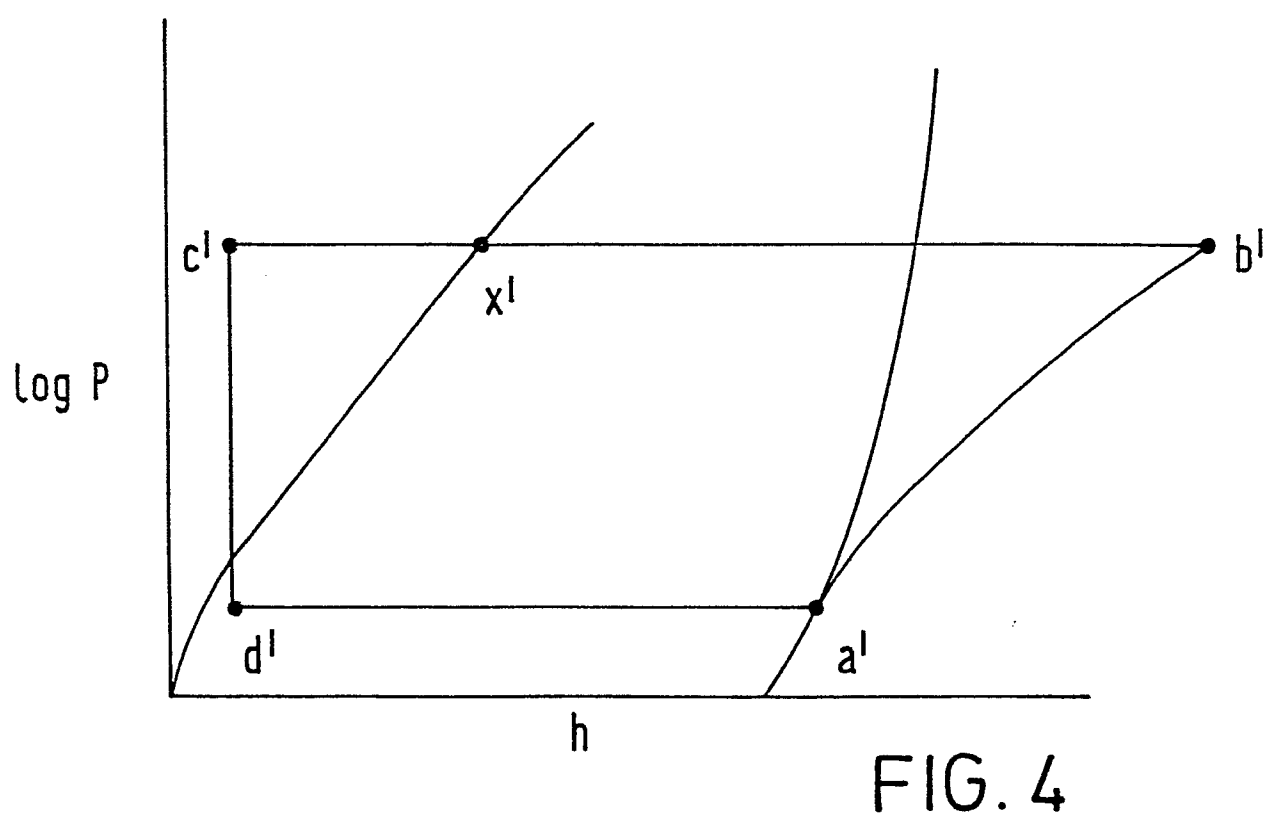
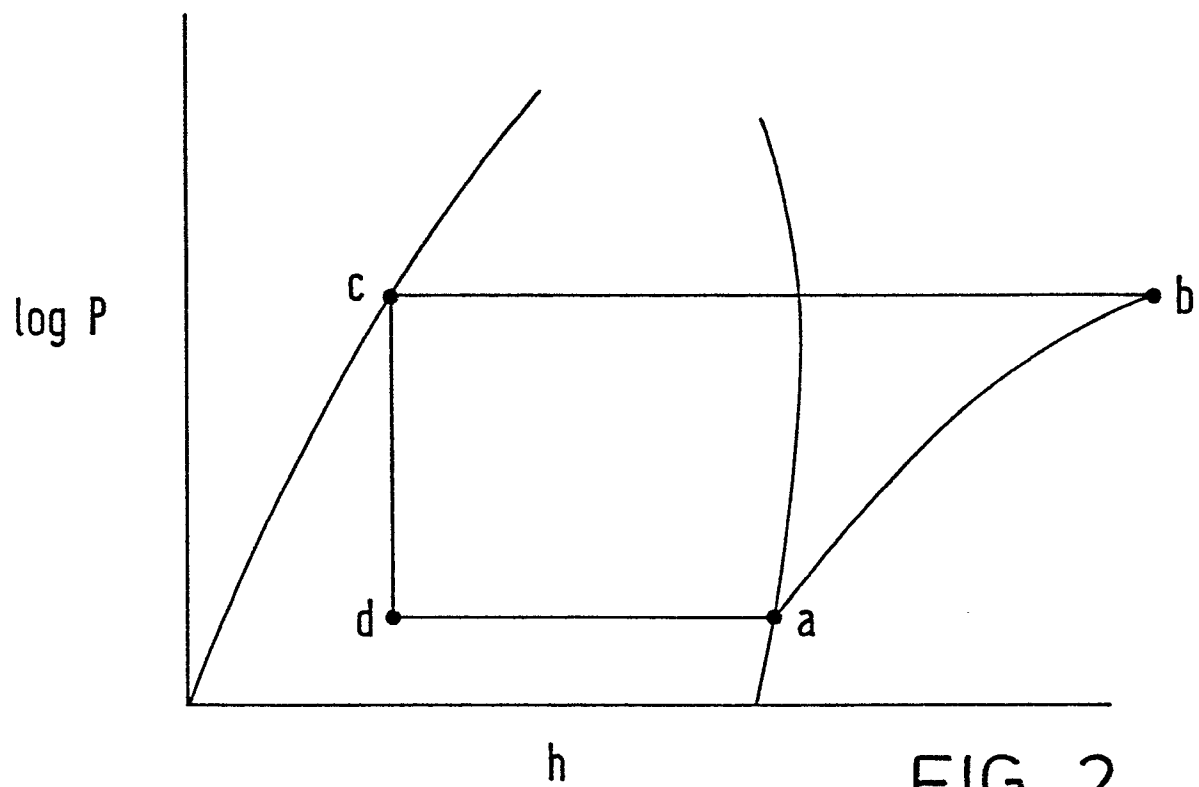
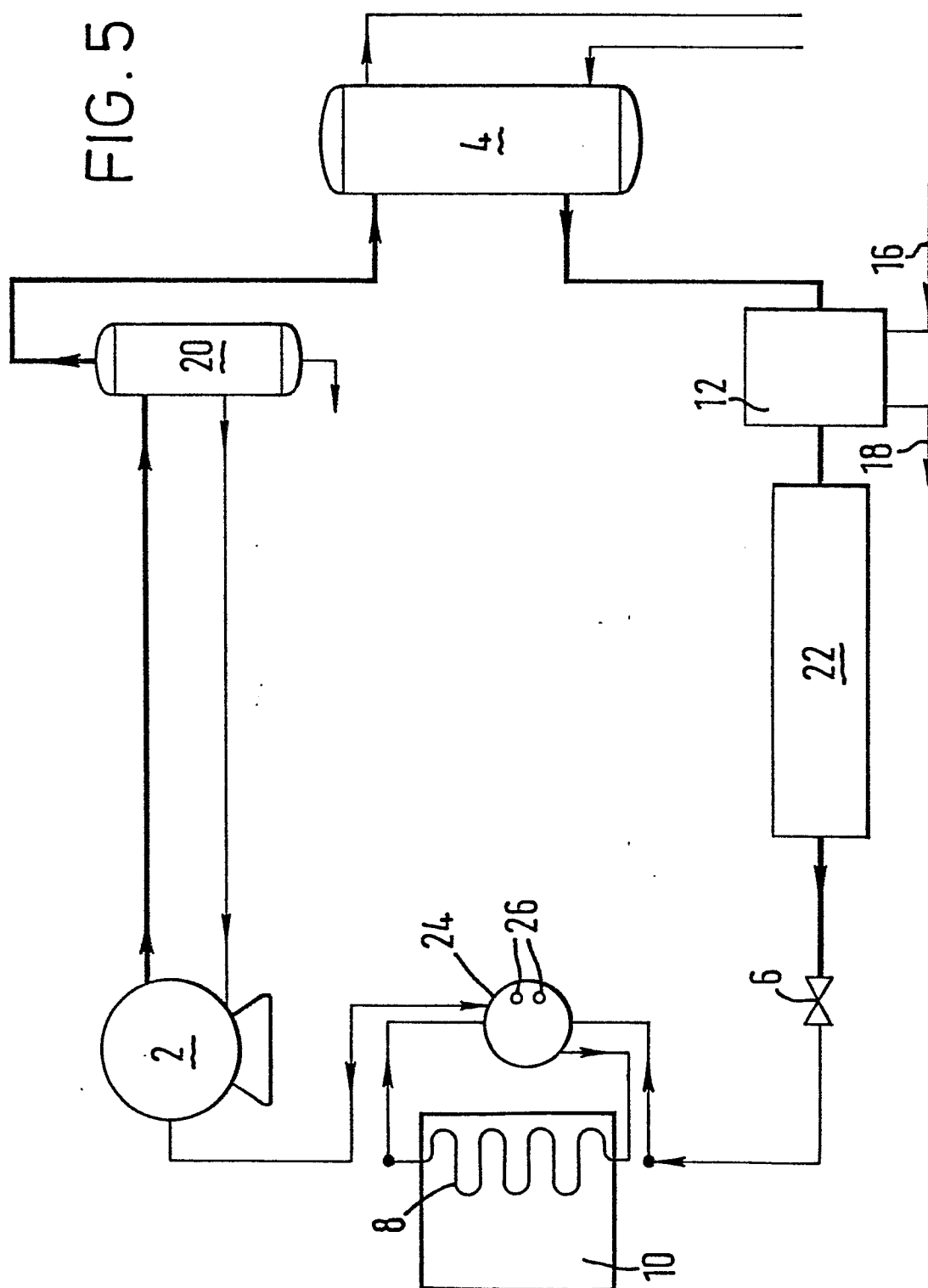


FIG. 5



0208526

FIG. 6

