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Girardin

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[54] **EXPANSION AND EVAPORATION APPARATUS FOR REFRIGERATING MACHINES**

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[52] **U.S. Cl.** **62/512**
[51] **Int. Cl.** **F25b 43/00**
[58] **Field of Search** 62/498, 512, 500, 62/218

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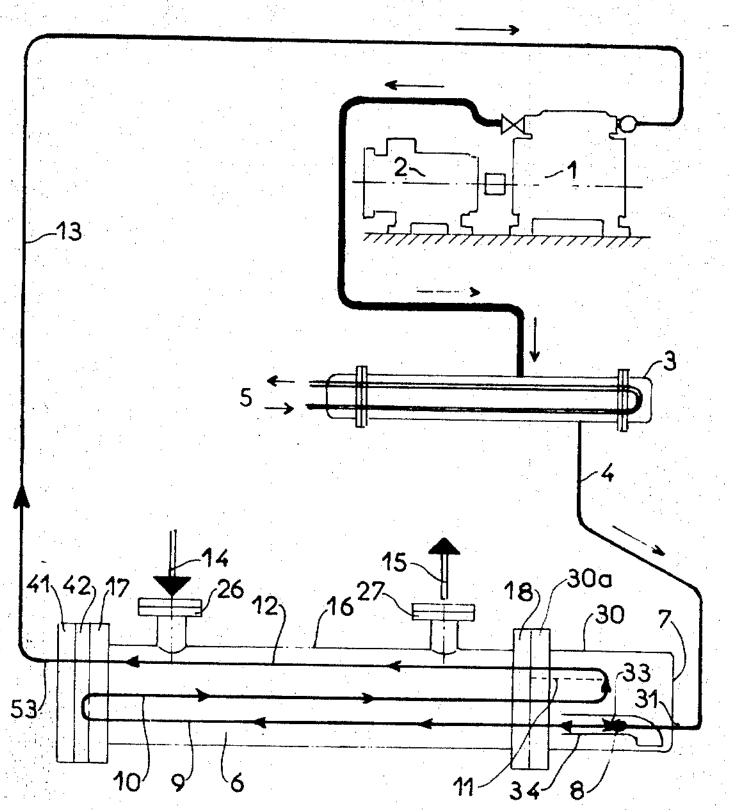
[57] **ABSTRACT**

Apparatus for expanding and evaporating refrigerating fluid in a compression-type refrigerating machine for cooling a fluid such as water, brine, air, moist air, circulating outside the tubes of a tube bundle, said apparatus being characterized in that it comprises (i) a tube bundle part of which is fed, inside the tubes, by forced circulation of refrigerating fluid with a flow exceeding the flow vaporized, the other part of the said bundle serving to dry and to superheat the vapour produced in the first part, (ii) a device, for separating excess liquid, positioned between the first and second parts of said tube bundle, (iii) a separation chamber arranged to collect said excess liquid, and (iv) an ejector taking up said excess liquid to recirculate it, said ejector operating with a motive fluid consisting of the high pressure liquid refrigerating fluid coming from the condenser, said motive fluid being expanded in the neck of the primary nozzle of the ejector.

4 Claims, 6 Drawing Figures



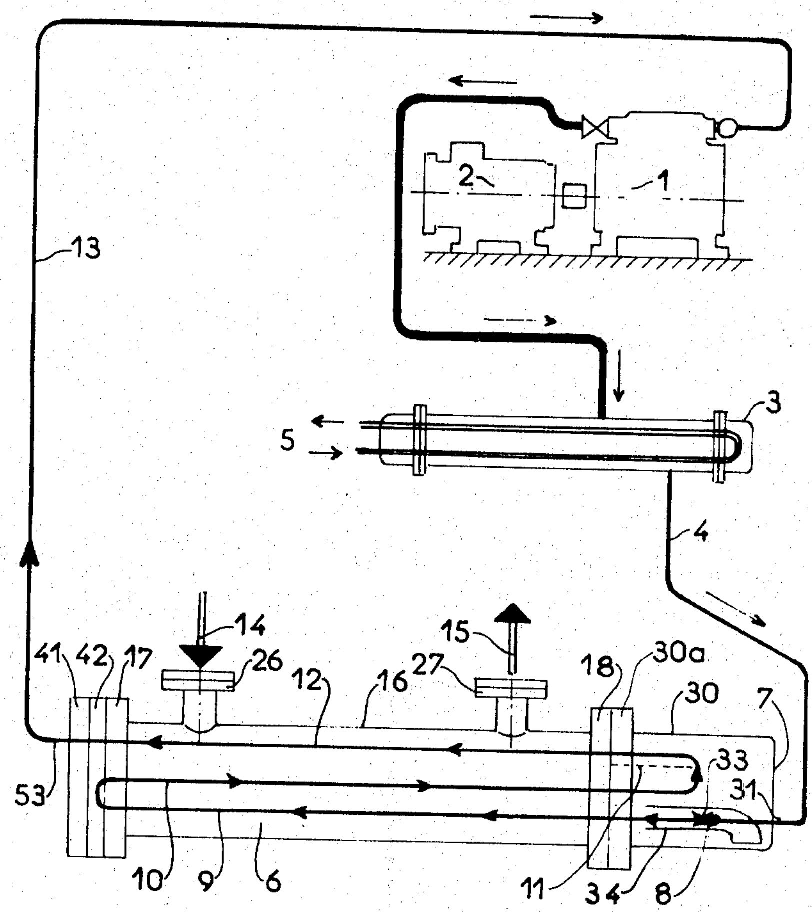


fig.1

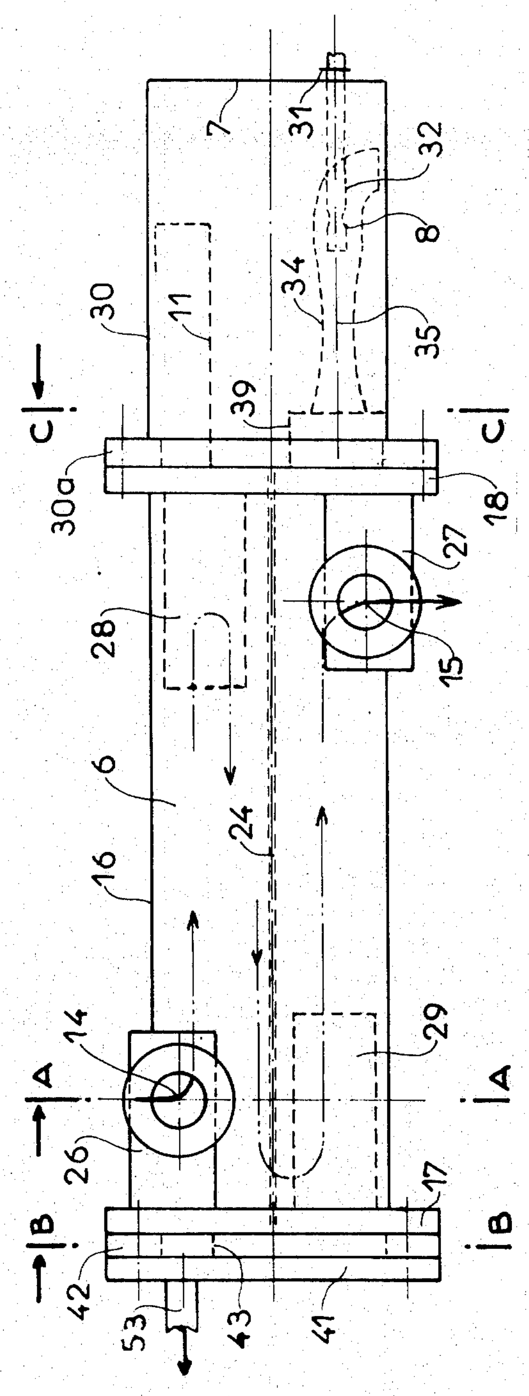

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fig. 2

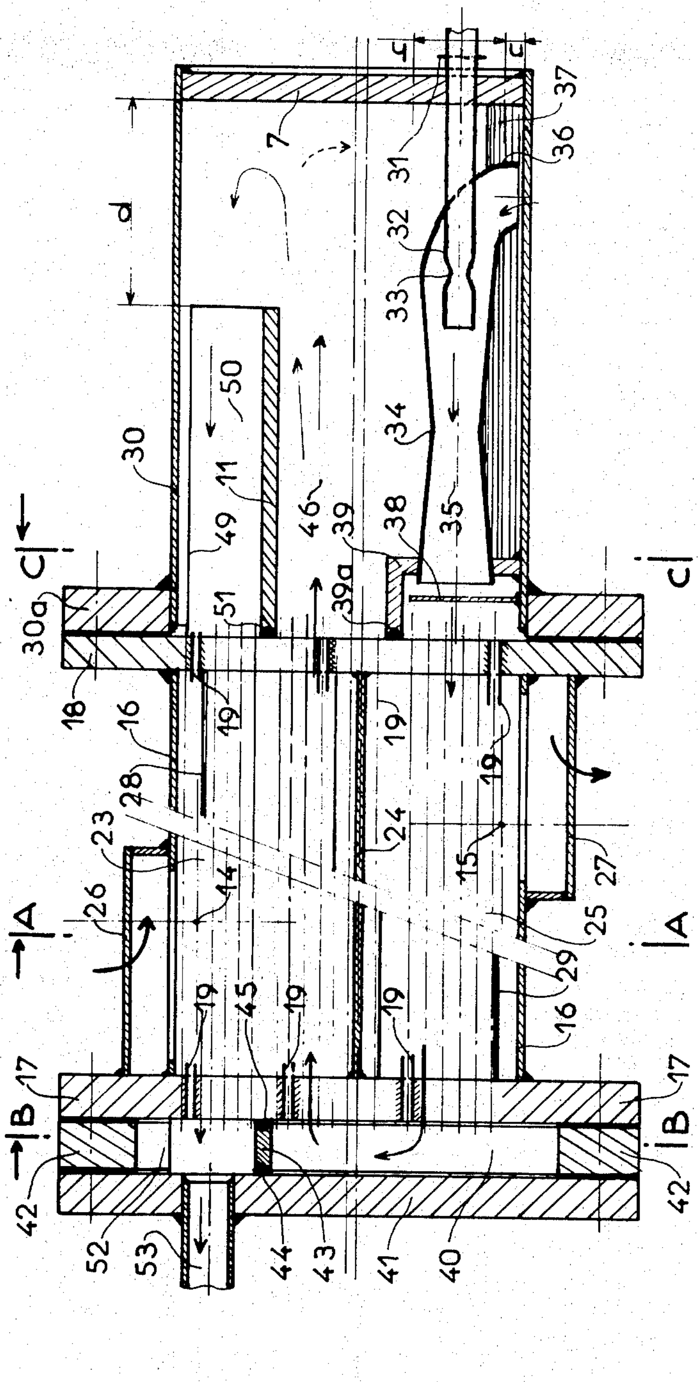
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fig. 3

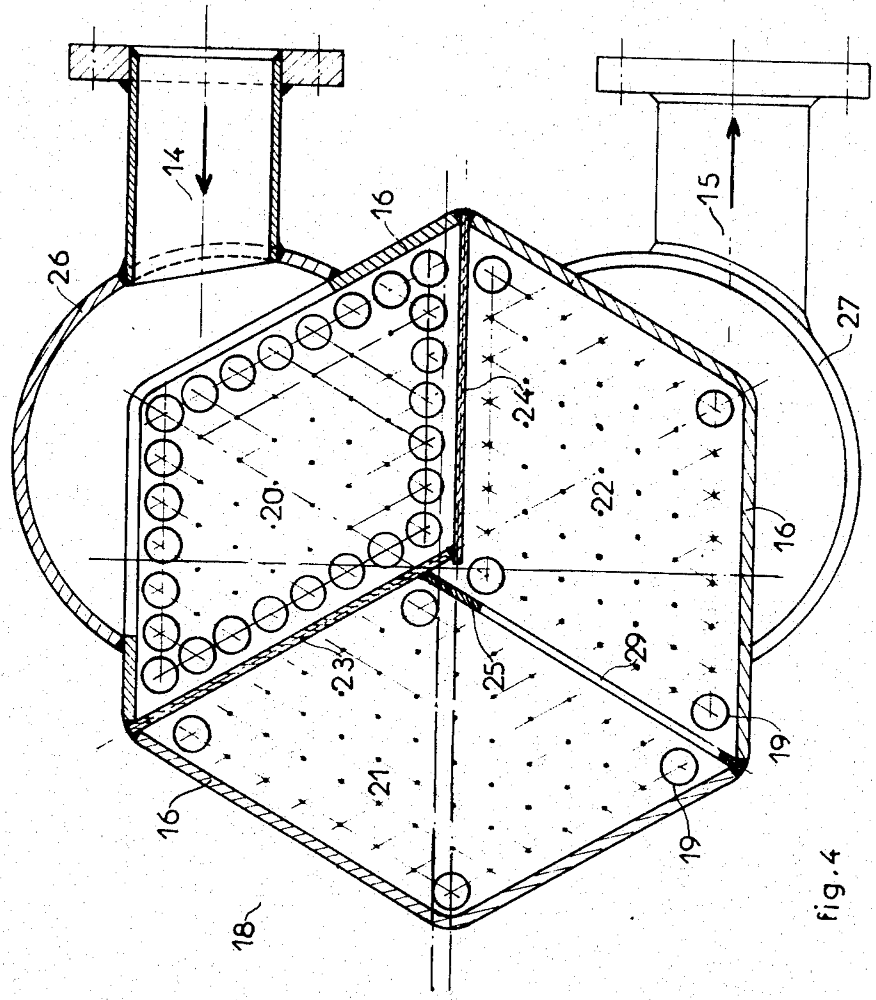

fig.4

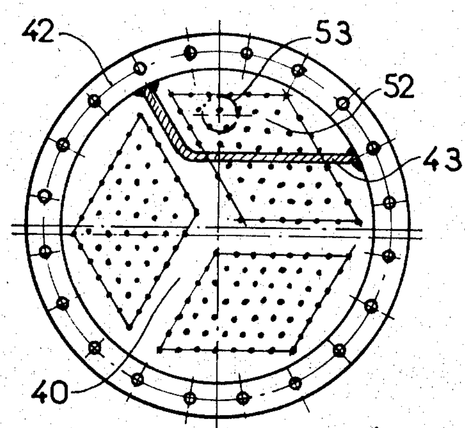

fig. 5

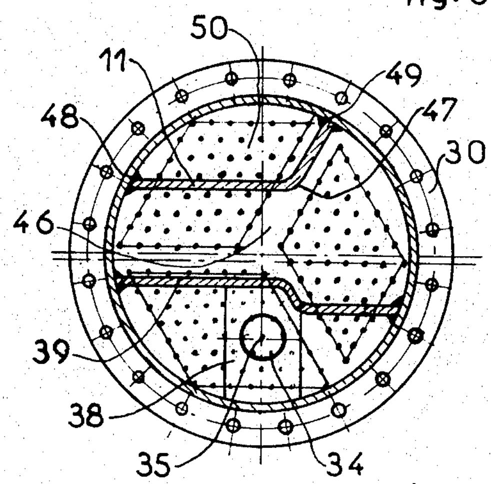

fig. 6

EXPANSION AND EVAPORATION APPARATUS FOR REFRIGERATING MACHINES

The present invention relates to compression-type refrigerating machines. In particular, it refers to the evaporator and the expansion system thereof which forms one of the parts of these machines. This evaporator is the type where the refrigerating fluid recirculates inside the tubes, the fluid to be cooled — e.g. water — being outside.

In conventional installations of this type, two kinds of evaporator are mainly used:

Submerged evaporators where the fluid to be cooled circulates inside the tubes and the refrigerating fluid outside. The high pressure liquid is expanded in the evaporator either by a high pressure valve (in this case, all the liquid in the circuit is wholly in the evaporator) or by a float valve which maintains a constant level in the evaporator.

Dry expansion evaporators where the fluid to be cooled is outside the tubes and where the refrigerating fluid is inside the tubes. In this case, the refrigerating fluid is completely vaporized in the tubes and at the outlet the gas is slightly superheated. The quantity of liquid injected into the evaporator is regulated by a valve termed "thermostatic," which measures the superheating of the gas when it emerges.

Despite the constructional simplicity of submerged evaporators and the possibility they afford of replacing the regulating valve with a simple orifice, appliances of this type have certain disadvantages when the working rates vary within certain limits:

Risk of the cooled fluid freezing, which may cause the exchanger tube to burst;

Large quantity of refrigerating fluid in the evaporator;

Trapping of oil driven into the evaporator by the compressor (reciprocating type in particular) which necessitates providing a more or less complicated device to ensure regular oil return to the machine.

As for dry expansion evaporators, while they use a small quantity of refrigerating fluid, although allowing continuous oil return and not being sensitive to freezing (the liquid to be cooled being outside the tubes), they nevertheless have the following disadvantages:

The need to regulate precisely the quantity of refrigerating liquid injected by means of a thermostatic valve which is a fragile and expensive component, sometimes requiring filters.

High power loss due to the cooled fluid being circulated perpendicularly to the tubes, with numerous direction changes (the coefficient being equal, the power loss of the fluid to be cooled is higher than in a submerged evaporator). A large number of baffle plates is needed to ensure the correct speed perpendicularly to the bundle.

On the refrigerating fluid side, a poor coefficient of internal thermal exchange in the part of the tubes which is superheated or where there is a small quantity of liquid in the vapour.

It is a known fact that forced circulation of the refrigerating circuit improves thermal exchange in the tubes. This is the object of recirculating, i.e. forced circulation of refrigerating liquid injected into the vapour current.

However, up to now, this recirculation has been complicated by, inter alia, supplementary devices to main-

tain a suitable level at the intake of the liquid injection device.

The apparatus of the present invention make it possible to eliminate the disadvantages mentioned above, which apply the plant using either one of the two main types of evaporator. Moreover, by a highly characteristic combination of certian components, the apparatus makes it possible to retain the useful industrial results of each of these two types. As an example, a few of these results are mentioned below.

The fact of associating recirculation (by ejector) of the refrigerating fluid with a dry expansion type evaporator makes it possible to do without regulation by the thermostatic valve and, moreover, allows the whole of the refrigerant charge to be in the low-pressure part of the circuit (evaporator). This permits expansion, if necessary, to be effected via a fixed orifice.

The combination of the superheater associated with a separating chamber enables the unit to work like a dry expansion ovaporator, i.e., with regular oil return to the compressor and a slightly superheated gas outlet.

Recirculation of the refrigerating fluid, associated with longitudinal circulation of the fluid to be cooled, substantially improves the performance of the apparatus.

The possibility of eliminating all moving parts and the grouping of all components into a single appliance makes working very reliable and is a considerable simplification as compared with the conventional system.

Finally, the apparatus permits improved working of the machine as a whole, at slow rates, elininating in particular the use of a by-pass when it is desired to prevent frequent starting of the compressor sets.

The combined expansion and evaporation apparatus of the invention is made up by associating the following components:

a. A multitibe exchanger whse internally flanged (or possibly smooth) tubes are positioned in parallel bundles separated by longitudinal partitions and having a flow path forming as many successive passes outside the tubes as are necessary to achieve a suitable flow rate in the fluid to be cooled. A single pass may also be sufficient. Preferably, no transversal intermediate baffle plate alters the direction of the current, which is parallel to the tubes. The refrigerating fluid circulates inside the tubes, in at least two successive passes. Partitions spaced over the forward and rear plates limit the number of tubes involved in each pass as a function of the power losses. The first pass(es) is (are) travelled by the recirculated refrigerating fluid (forced circulation). The last pass(es) act(s) as an exchanger-super-heater drying the gas. The recirculation pass(es) has (have) a refrigerating fluid flow exceeding the flow of vaporized fluid. Recirculation is obtained by means of a device described below (ejector).

The bundles are confined in a casing which exactly follows their outside shape after assembly. For example, in the case of three bundles, each section is lozenge-shaped and this casing itself has a hexagonal section.

b. A separating device, e.g. baffle plate, between the last or only recirculating pass and the first or only superheating pass over the refrigerating fluid. Suitably shaped, the said baffle plate leaves an adequate and necessary flow section for proper separation of the liquid to be recirculated and for admission of the gas entering the superheater. The excess refrigerating liquid is collected under the separating device.

c. A separating chamber, collecting the refrigerating liquid, wherein the level is maintained without an complementary device.

d. An ejector feeding refrigerating fluid to the first pass. This consists of a primary nozzle fed by the motive fluid (high pressure refrigerating fluid) the neck of which is carefully calibrated. A diaphragm may possibly be mounted upstream of the primary nozzle, to alter its rated flow and thus adapt it to different working conditions. The body of the ejector consists of a convergent-divergent tube forming the mixing chamber and the diffuser. This tube is preceded by a bent part enabling the recirculated liquid to be sucked from the bottom of the separating chamber.

e. The head (or chamber, properly so called), the volume and shape of which afford the possibility of associating the ejector, the separating baffle plate and the possible partitions dividing the tubes into each of the passes, as well as the supply of liquid.

By a special geometrical arrangement, the gases are separated from the liquid before entry into the superheater. This arrangement enables a slight impulsion to be given to the drops of liquid, ensuring continuous oil return to the compressor. This arrangement is highly characteristic because, if the impulsion of the drops is excessive, the superheating part will be insufficient to vaporize this excess liquid and the compressor will be working wet, which is detrimental to it. Conversely, if the impulsion is inadequate, the oil will not be properly returned. Moreover, the percentage of liquid impelled must remain within clearly defined limits when the output decreases and when the quantity of refrigerating fluid contained in the vaporator varies. In fact, all the refrigerating fluid being in the evaporator, it is essential for the working of the unit to remain correct even if there is an accidental leak. This arrangement described below, as an example, allows a loading variation of very roughly 30 percent. When working at low output (e.g., by creating a vacuum in a certain number of cylinders in the reciprocating compressor) tests have shown that a fixed orifice calculated for full output allowed a small flow of gas to pass, coming directly from the condenser (by-pass effect). This gas flow creates a slight decrease of effective refrigerating output, but increases the expansion energy in the ejector, which improves the coefficient of heat transmission. This creates an incresase in the evaporation temperature and the efficiency of th cycle thereby increases. This increase compensates for the loss of power due to the passage of the gas, up to a power which may be less than 40 percent of the rated power. On the other hand, at very low power (e.g. one-quarter of the compressor power) the by-pass effect becomes predominant, which reduces the effective refrigerating power to a low value. This enables the refrigerating machine to work at very low power and to avoid all-or-nothing working of the compressor. These features, ascertained during testing, and already briefly mentioned as a new industrial result, make it possible to avoid the use of an automatic by-pass which is essential when it is desired to avoid frequent starting of the machines.

A better understanding of the invention will be gained from the following description of a from of embodiment, given by way of limiting example. In the accompanying drawings:

FIG. **1** is a diagram of the complete circuit of the refrigerating machine with an embodiment of evaporator comprising three passes on the water side and three passes on the refrigerating fluid side.

FIG. **2** is an external view of the evaporator;

FIG. **3** is a partial section through the longitudinal axis of the evaporator.

FIG. **4** is a cross section along the line A—A of FIG. **2**;

FIG. **5** is a cross section along the line B—B of FIG. **2**;

FIG. **6** is a cross section of the chamber along the line C—C of FIG. **2**.

Schematically, the plant in which the apparatus of the invention is incorporated is shown in FIG. **1**. The compressor **1**, driven by the motor **2**, drives the refrigerating fluid in a gaseous state into the condenser **3** where it liquefies. The condensed liquid comes out at **4**. The condenser is cooled by water circulation **5**.

The refrigerating fluid goes into the evaporator **6** via its chamber **7** and expands in the neck of the ejector **8**. The refrigerating fluid stored in the bottom of the separator chamber is sucked in by the ejector and mingles with the primary fluid, which imparts its energy to it. The ejector sends the mixture into the pass **9** and then the pass **10** of the evaporator where it partially vaporizes. The liquid contained in the mixture coming out of the pass **10** encounters the baffle plate **11** and falls back to the bottom of the chamber where it is again sucked in by the ejector while the gases are taken into the pass **12**. The change of direction imposed on the fluid by the baffle plate **11**, associated with the effect of impact on the bottom of the chamber **7** effeciently separates the liquid. Due to the precise geometry of the chamber and the baffle plate, there is a slight residual impulsion of liquid drops which returns the oil to the compressor. The wet gas is superheated in the pass **12** and returns to the compressor via the pipe **13**. The evaporator is traversed, outside the tubes, by the fluid to be cooled entering at **14** and coming out at **15**, after having yielded its calories to the refrigerating fluid. The notable fact is that, on the piping **4** in the particular, no automatic expansion or dividing device is needed. The evaporator **6** (FIGS. **2, 3** and **4**) consists of a body **16**, hexagonal in section, closed by two tube plates **17** and **18**. The exchanger tubes **19** welded into the latter are arranged in three bundles **20, 21** and **22**, separated by the longitudinal partitions **23, 24** and **25** connnecting one edge of the hexagonal casing to the evaporator axis. These partitions are at an angle of about 120° to one another. The fluid to be cooled enters through the header **26** and leaves through the header **27**. The fluid first passes over the bundle **20** (FIG. **4**), towards the plate **18**. It then reaches the bundle **21** through an opening **28** in the partition **23** (FIG. **2**). It then turns back towards the plate **17**, close to which it passes into the bundle **22** through the opening **29**. Then it goes towards the outlet header **27**. Of course, opposite each header **26** or **27**, welded onto the body, the latter is cut away. This arrangement enables the fluid to circulate longitudinally between the tubes. This arrangement is preferred, but does not exclude a mounting with conventional baffle plates perpendicular to the tubes. The longitudinal circulation allows better coefficients of thermal exchange than transversal circulation to be obtained with the same loss of power. Moreover, it simplifies the construction.

A vessel (including a chamber) disposed at one end of the evaporator simultaneously serves as a support for the ejector, for a container for liquid refrigerating fluid, for a separating device **11**, and for a separation chamber, collecting the liquid. It consists of a sleeve **30**, preferably cylindrical, closed by the head **7** (the sleeve and head forming the vessel) and is connected to the evaporator by a flange **30***a*. In the example selected, it contains the ejector (fed with motive fluid at **31**) made up of the injection nozzle **32** with a neck **33** and the body of the ejector proper **34** made up of a convergentdivergent tube **35** preceded by a bend **36** enabling the liquid contained in the bottom of the chamber at **37** to be sucked up. A deflector **38** correctly distributes the liquid. The body of the ejector is connected to the loop **39** which delimites the number of tubes in the first pass (see FIG. **6**). A joint **39***a* forms a seal between the partition and the tube plate **18**.

At the forward end (FIG. **3**), the tubes open into a space **40**, formed between the forward plate **17** and the forward head **41** by means of a bracing flange **42**. The space **40** is bounded by a partition **43** (see also FIG. **5**) sealed by joints **44** and **45** compressed between the plate and the head. The flow therefore returns by other tubes into the part **46** of the chamber **6**. These two first passes, termed recirculation passes, improve the coefficients of thermal exchange, due to the internal wetting of the tubes.

Before the flow goes into the tubes of the final pass, the baffle plate **11** separates the gas and the liquid. It serves simultaneously as a deflector, a screen and to fix the section of the pass (see FIGS. **3** and **6**). Its purpose is to cause the drops of refrigerating liquid to fall back into the supply **35** in the chamber, restocking this supply, in closed circuit; it enables the third and last pass to dry the gases. But if the gas - drop separation is excessive, it will also stop the oil, which must necessarily return to the compressor intake; this oil will pollute the evaporator or dissolve in the supply of liquid to the detriment of the oil level in the compressor. This is why the functional shape of this baffle plate is important and is specified later in the description.

The quantity of fluid stored at **35** has been calculated by experiment for each type of cold-producing machine. According to the variations in the rate of working and fluid losses, there may be a variation in the level from a minimum of n to a maximum of $n+h$. These levels are ascertained by experiment, the low level still allowing the bend **36** in the ejector to dip in the liquid supply and the high level, at the most, not approaching the baffle plate **11** in the example selected. The geometry and the volume of the chamber in this part are such that temporary losses of fluid at **37** or transfers to this supply **37** do not hinder the working of the unit. In principle, the quantity (main circuit and recirculation circuit) of refrigerating fluid is constant. A working equilibrium is established, despite variations in working, between the effects of the ejector, the recirculation and the separating baffle plate **11**. Experiments and bench tests confirmed this point.

The baffle plate **11** consists of a horizontal metal sheet bent at **47** and welded along two generatrices on the chamber, at **48** and **49**. The bend **47** is partly gov-

erned by the number of tubes enclosed in **50**, for the third and last pass. Its profile and its position in relation to the bundles are, moreover, the same as those of the partition **43** between the plate and the forward head (see FIG. **5**).

The horizontal part and the sloping part of the said baffle plate are parallel to the axis of the chamber. They bear on the rear plate **18** via its field tube **51**, which may be fitted with a joint. The other tube is parallel to the rear plate of the chamber. The distance d (FIG. **3**) between the end of the baffle plate and the rear plate is governed by the passing section in the opening so formed as to give access to the passage **50**, formed by the said baffle plate and the corresponding sector of the cylindrical casing of the chamber. The description of the said buffle plate is given as an example. It may also be made up of several metal sheets forming efficient screens. The metal sheets may be solid or holed or perforated.

After passing through the tubes in the zone **50**, the dried refrigerating fluid goes into the part **52** of the forward head. It comes out of the evaporator at **53**, towards the compressor intake.

It goes without saying, and is evident from the foregoing, that the invention is in no way limited to th form of embodiment described above. On the contrary, it covers all possible variations within the scope of the appendent claims. Thus, the tubes may or may not be imporved by inside flanges in order to achieve better thermal exchange. Similarly, the number of passes on the side of the fluid to be cooled can be greater or lesser to suit the appliance to the flow of fluid to be cooled. Similarly, the number of passes on the refrigerating fluid side may differ so as to suit the appliance to different refrigerating fluids and to the different outputs to be transmitted. The same applies to the superheater pass.

The nature of the refrigerating fluid is immaterial to the working principle of the appliance.

The apparatus of the invention can be used in refrigerating machines when it is desired to increase the coefficient of thermal exchange, to simplify the mounting, and consequently to reduce the price of the apparatus and provide greater working reliability.

I claim:

1. An expansion and evaporation apparatus for a refrigerating system adapted to cool a fluid comprising:

a an elongated evaporator body having an inlet and outlet whereby fluid to be cooled flows through said evaporator body, said evaporator body having a wall at each end;

b a tube bundle in said evaporator body, said tube bundle including a plurality of tubes through which a refrigerant flows, said tube bundle having first and second portions;

c a vessel disposed at one end of said evaporator body in contiguous relationship with one wall;

d said one wall having openings through which said tubes communicate with said vessel;

e an ejector disposed in said vessel so that the longitudinal axis thereof is parallel with the longitudinal axis of said tube bundle;

f a partition positioned in said vessel and interposed between said first and second portions of said tube bundle, said partition having at least one surface disposed parallel to the longitudinal axes of said evaporator body and said ejector;

g whereby liquid refrigerant is removed by said partition to be recirculated by said ejector so that said fluid flowing into said second tube bundle may be superheated.

2. An apparatus as defined in claim 1 including means in said evaporator body for directing the initial flow of fluid to be cooled from said inlet and along outer surfaces of the tubes of said second portion of said tube bundle to effect superheating of refrigerant in the tubes of said second portion.

3. An apparatus as defined in claim 1, wherein said ejector includes a curved portion therein and an inlet

opening in said curved portion, the axis of said inlet opening being disposed at approximately 90° with respect to the longitudinal axis of said ejector, whereby said ejector may draw liquid refrigerant which has been collected in said vessel regardless of the level of the liquid refrigerant.

4. An apparatus as defined in claim 3 including a deflector in said vessel disposed approximately perpendicularly with respect to said ejector to distribute the flow from said ejector to the tubes of said first portion of said tube bundle.

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