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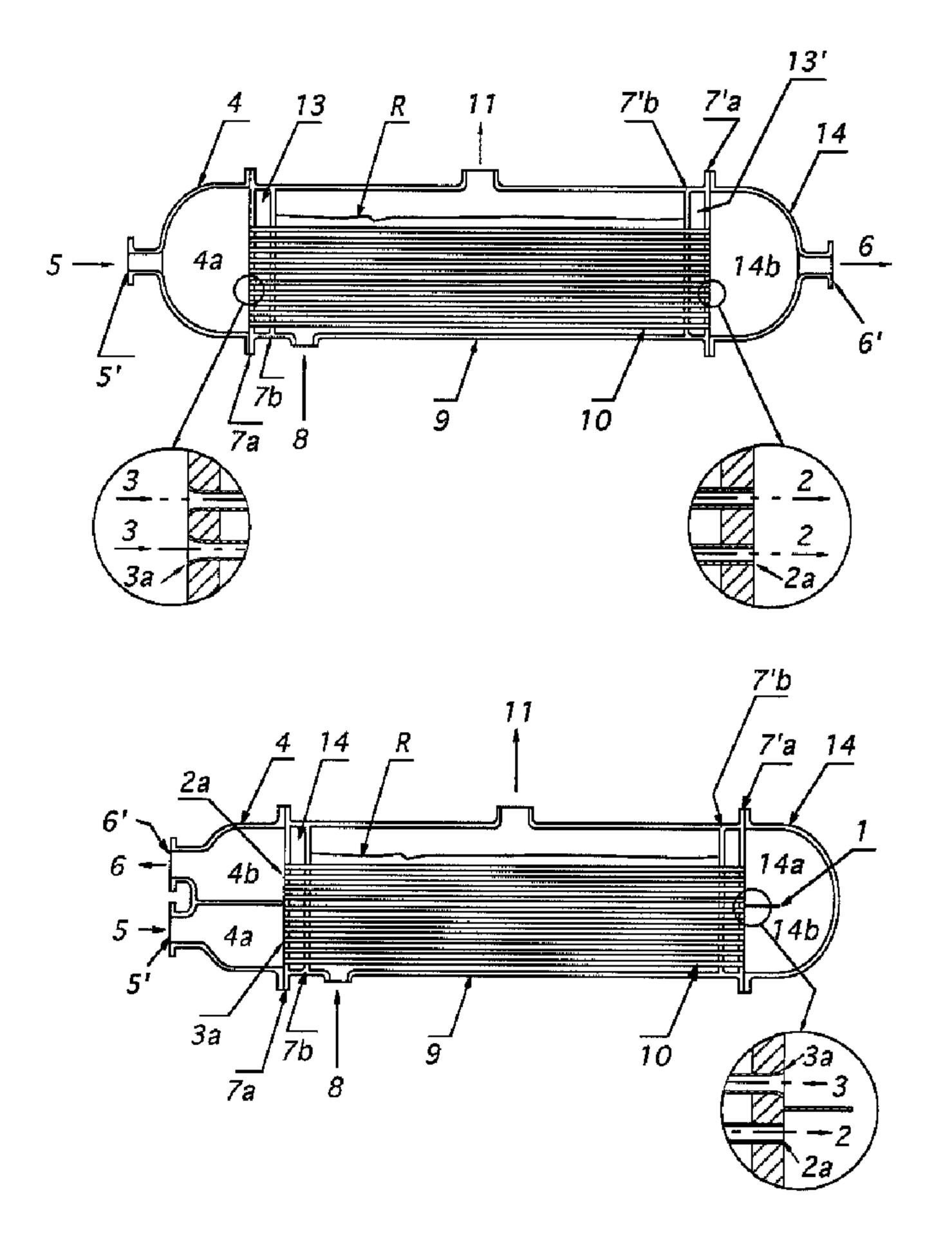
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#### (57) Abrégé/Abstract:

A liquid cooling system for generating very cold liquid flows, comprising sending said liquid to a heat pump circuit; said circuit including a hot-side heat exchanger (HSHE) and a cold-side heat exchanger (CSHE); said CSHE being provided with conduit





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#### (57) Abrégé(suite)/Abstract(continued):

means through which said liquid flows; said conduit means being in contact with and being cooled externally by cold medium. The improvement in the system comprising the installation of proper thermal barriers between said cold medium and said liquid in all areas where the velocity of said liquid is much lower than the average velocity found inside said conduit means; said improvement also comprising proper design of the entrance of said conduit means to prevent flow detachment; said design preventing any freeze-up of said liquid as well as improving efficiency and durability. An alternative to the above said improvement is simply the elimination of all areas where the velocity of said liquid is much lower than the average velocity found inside said conduit means. Other alternatives are proposed. The improvement in the system also comprising the capability to generate and manipulate supercooled liquids in a way that prevents liquid line and cooler freeze-ups. The improvement in the system also comprising new vapor compression cycles as well as new absorption cycle arrangements for better practicing the described method. Applications in very diversified fields are described. The technology is given the name "SUPERPAC", which stands for "SUPER Pompe A Chaleur", or super heat pump.

# A FULL-RANGE, HIGH-EFFICIENCY LIQUID CHILLER

#### **ABSTRACT**

A liquid cooling system for generating very cold liquid flows, comprising sending said liquid to a heat pump circuit; said circuit including a hot-side heat exchanger (HSHE) and a cold-side heat exchanger (CSHE); said CSHE being provided with conduit means through which said liquid flows; said conduit means being in contact with and being cooled externally by cold medium. The improvement in the system comprising the installation of proper thermal barriers between said cold medium and said liquid in all areas where the velocity of said liquid is much lower than the average velocity found inside said conduit means; said improvement also comprising proper design of the entrance of said conduit means to prevent flow detachment; said design preventing any freeze-up of said liquid as well as improving efficiency and durability. An alternative to the above said improvement is simply the elimination of all areas where the velocity of said liquid is much lower than the average velocity found inside said conduit means. Other alternatives are proposed. The improvement in the system also comprising the capability to generate and manipulate supercooled liquids in a way that prevents liquid line and cooler freeze-ups. The improvement in the system also comprising new vapor compression cycles as well as new absorption cycle arrangements for better practicing the described method. Applications in very diversified fields are described. The technology is given the name "SUPERPAC", which stands for "SUPER Pompe A Chaleur", or super heat pump.

### **BACKGROUND OF THE INVENTION**

This invention relates to a liquid cooling system for cooling any liquid, said cooling being performed for any purpose, including the extraction of energy from said liquid for heating purposes. More particularly, this invention is concerned with a simple and efficient system for obtaining and handling very cold liquids, down to and below their solid-liquid equilibrium temperature of said liquid.

A cooler is a component of a liquid-chilling system in which a liquid, usually an aqueous liquid (v.g. water or brine) is cooled by a refrigerant. In a conventional system, the cooling effect comes from the evaporation of the said refrigerant. The most common cooler types are the direct expansion and the flooded (1990 ASHRAE Handbook: Refrigeration Systems and Applications).

In a typical <u>direct expansion</u> shell-and-tube cooler, the refrigerant evaporates inside tubes, while the fluid to be cooled is channeled throughout the shell side by a series of baffles. In a typical <u>flooded</u> shell-and-tube cooler, the fluid flowing through the tubes gives its heat to the boiling liquid refrigerant on the shell side. These coolers are usually mounted horizontally. They are normally found in larger systems using screw or centrifugal compressors. In very large diameter shells, the heat transfer at the bottom can be adversely affected by the head of the refrigerant. This problem can be solved by spraying the liquid refrigerant all over the tubes instead of flooding them.

Freezing of the fluid can cause considerable damage to coolers and some freeze protection must be provided. Two methods can be used. The suction pressure can be held above the one corresponding to the freezing point of the fluid; or the system can be shut down when the fluid temperature gets too close to the freezing point. For pure water, a temperature of 5,5 °C is normally considered as a lower limit.

# Limitations of present systems

# a) The freeze-up problem

If one tries to obtain low fluid temperatures from a conventional cooler, a freeze-up will occur: the freezing process will start at a location where the wall temperature is well below phase equilibrium temperature and where the fluid velocity is low. In a <u>flooded</u> type cooler, the flow of water in the tubes does not exhibit any low velocity region, except maybe at the inlet, where the flow might be detached for a certain distance inside: a low velocity bubble is created where freezing might start. The real problem is in the cooler heads, where very low velocity water is exposed to the cold tubesheets. The freezing process will start there if the tubesheet is at a temperature below freezing. In a direct expansion cooler, several low velocity regions exist in the shell, which is crossed by cold tubes containing the refrigerant.

### b) The small $\Delta T$ limitation.

In a conventional chiller used in air conditioning, the cooler outlet on the fluid side will usually be kept at 7 °C. Since the inlet is at about 13 °C, a  $\Delta T$  (temperature differential) of only 6 °C is thus available for energy absorption. This is an important limitation. If the  $\Delta T$  could be doubled (e.g. by using a cooler capable of generating 1°C fluid outlet temperatures), the system capacity could be doubled without even having to increase the size of the distribution system (pumps, piping and heat exchangers).

# c) The constant-temperature phase change problem.

If a high system COP (Coefficient Of Performance) is to be obtained, it is important to have a cycle with the minimum source-sink  $\Delta T$  possible: this is possible only with an efficient heat transfer in the CSHE and in the HSHE, which supposes a large heat transfer area and small  $\Delta Ts$  in both heat exchangers (HX). The latter condition is quite difficult to obtain in conventional systems. Indeed, if the fluid had to be cooled from 13 °C to 1 °C in a conventional cooler, the refrigerant would have to evaporate at about -5 °C. The  $\Delta T$  between the entering fluid and the refrigerant would then be about 18 °C: this large value of the  $\Delta T$  increases the irreversibility of the heat transfer and causes a significant decrease in the COP of the chilling system.

# d) The COP / capacity dilemma.

As explained by Didion and Bivens ("The Role of Refrigerant Mixtures as Alternatives". Didion, D.A. and Bivens, D.B.. Proceedings, A.S.H.R.A.E. 1989 C.F.C. Conference, Washington, Sept. 1989), with the usual one-component refrigerants, one has to choose between a large volumetric capacity and a large COP. With non-azeotropic mixtures, on the other hand, one could theoretically get the best of both worlds.

#### **SUMMARY OF THE INVENTION**

The above-described drawbacks can all be eliminated.

Indeed, in one of its aspects, the invention provides a liquid cooler design for cooling liquids inside a liquid cooler, said cooler being connected to a liquid supply circuit including one or more conduit means through which said liquid flows; said conduit means passing through and being in contact with a cold medium in said cooler, said cooler and/or said conduit means being straight or smoothly curved, said cooler being characterized by the fact that it is capable of cooling liquids down to very low temperatures, including the phase-equilibrium temperature as well as supercooling temperatures, without inducing freezing of said liquid in said conduit means; said capability being obtained by one, two or all of the following three ways; the first of said ways being preventing the contact of the said liquid with a cold surface in locations where said liquid is stagnant or has a low velocity as compared to the average liquid velocity inside said conduit means passing through said cooler; the second of the said ways being the elimination of all said stagnant regions or said low velocity regions near said cold surfaces; the third of said ways being the elimination of the cold surfaces themselves.

In another aspect, the invention also provides a system for handling supercooled liquids generated by a cooler in which outlet ends of individual conduit means extensions, or outlet ends of said main conduit means, from which the supercooled liquid flows, are equipped in such a way as to prevent the formation of ice crystals at said outlet ends, said formation provoking a rapid blockage of said extensions or said main by inducing a phase change of part of said supercooled water, formation of said ice crystals caused by said outlet ends being colder than ambiance, said ambiance being any liquid, including water, or any humid atmosphere, said atmosphere containing

any gas or mixture of gas, including air, said humidity condensing on said outlet ends and eventually changing to ice crystals.

In one of its other aspects, the invention also provides a liquid cooling system comprising a cooler, said cooler being the CSHE of a heat pump, said heat pump being used for heating or cooling purposes, said heat pump being of the vapor-compression (VCHP) type, the refrigerant circulating in said heat-pump circuit being of the non-azeotropic mixture type, said heat pump being characterized by the presence in the cycle of a reformer LPR, said reformer being a reservoir capable of containing said refrigerant in liquid and vapor phases, said vapor from the top of said reformer being entrained, compressed, condensed, and expanded in the hot-side portion of said circuit, said liquid from the bottom part of said reformer being entrained, circulated and evaporated in the cold-side portion of said heat-pump circuit, said circulation in said cold-side being produced by a refrigerant pump P, said pump being capable of producing an overfeeding of said cooler EV; said cycle being an improvement over conventional cycles in that the composition of said refrigerant vapor circulating in said hot side is rich in the more volatile, higher density, component(s) of said mixture while the composition of said refrigerant liquid, circulating in said cold side is rich in the less volatile, higher latent heat, component(s) of said mixture, said modified cycle thus providing at the same time higher system efficiency and larger system capacity.

In another aspect, the invention also provides a system for making artificial snow in which a continuous flow of supercooled liquid water is changed to a flow of supercooled droplets using any method, each of said supercooled droplets then rapidly changing to droplets containing a mixture of dendritic ice and water at 0 °C, said droplets then being sent into cold air where they grow to become a snow flake or are used as seeders for larger droplets which will then grow into larger snow flakes.

In still another aspect, the invention provides a system for generating ice with a liquid cooler, said cooler possibly being the CSHE of a liquid chiller, said chiller being capable of cooling any liquid, said liquid being cooled by said cooler to supercooling temperatures, said exiting supercooled liquid being made to partially change phase, said phase change providing a mixture of solid and cold liquid at the phase-equilibrium temperature, said change of phase being provoked by any convenient method, said mixture, and/or said solid, and/or said cold liquid being used for any purpose.

In still another one of its aspects, the invention provides a system for removing a slurry from within a reservoir and bringing said slurry, via conduit means, towards another location where it is needed, said system using a Coanda-effect wall-jet ejector, said ejector being characterized by the fact that said wall jet, having a high speed, is capable, because of the Coanda effect, of following bellmouth-shaped entrance to said conduit means, its turbulent eddies thus being capable of breaking agglomerations of solid particles near said entrance, said wall jet being also capable of entraining said slurry into said conduit means, thus moving said slurry to said other location.

In a further aspect, the invention also provides a system for breaking up crystal agglomerations inside a reservoir containing a liquid and said crystals, said breaking up being performed for any reason, including improving entrainment of said crystals into an outlet towards slurry conduit means, said system being characterized by the installation, in said outlet, of a "self-propelled" rotor, said rotor being made in such a way that it can be partly located inside conduit means and partly inside reservoir; the part of said rotor located inside said conduit means acting as a turbine capable of driving any device, said device being the part of said rotor protruding inside said reservoir; the purpose of said device being mainly the breaking-up of ice-crystal agglomerations and also the entrainment of crystals towards said outlet; said turbine being designed approximately like one of the well known types of hydraulic turbines (e.g. axial type; or mixed flow type); said device being made approximately like a propeller or an axial flow pump or any component that can at the same time break-up said agglomerations and entrain said crystals; said system using relatively little external energy.

In still another one of its aspects, the invention provides another system for breaking up crystal agglomerations inside a reservoir for the purpose of improving entrainment of said crystals into an outlet, said breakup being performed by the action of several liquid jets, said jets generating a vortex inside said reservoir, axis of rotation of said vortex passing through (or close to) said outlet, said vortex being generated because axes of said jets are not parallel with one another and do not meet at a point, path of every individual jet crossing said axis of said vortex at approximately identical angles smaller than 90°, preferably of the order of 30°, said crossings being made at a small and preferably identical distance from said axis of said vortex, said crossings all being approximately in the same plane, said plane being approximately perpendicular to said axis of said vortex, said jets creating entrainment of said agglomerations towards said outlet, said jets also creating rotation of said agglomerations, said jets then acting as so many jet saws capable of cutting through said agglomerations and moving broken-up pieces towards said outlet.

In one other aspect, the invention also provides still another system for breaking up crystal agglomerations inside a reservoir, said breakup being performed by the action of a giant "eggbeater" type of device, said device being installed close to the reservoir outlet in such a way that the resulting slurry is directed towards said outlet.

In still another one of its aspects, the invention provides a system for joining flows of supercooled liquid from two or more conduit means in order to create a larger flow of supercooled liquid, inside a larger conduit means, by making sure that said flows of supercooled liquid, between the said small and larger conduit means, do not endure any important mechanical or thermal perturbation, including large changes in velocity (i.e. speed and direction).

In a further important aspect, the invention provides a system for cooling liquids down to or below their phase equilibrium temperature using a hybrid multiple-cascade heat-pumping system comprising preferably 3 absorption heat pumps (AHP<sub>1</sub> to AHP<sub>3</sub>), said AHPs using any working pair, including LiBr-H<sub>2</sub>O, said system also using preferably 2 vapor-compression heat pumps (VCHP2 and VCHP3), said AHPs being normally unable to attain the very low temperatures necessary for providing desired final state, said AHPs being helped here by VCHPs in cascade; warm liquid coming from load first being cooled by AHP1; then chilled liquid out of AHP1 entering VCHP2 where it is cooled a few more degrees, said VCHP2 being installed in series with AHP<sub>1</sub>; said liquid finally entering VCHP<sub>3</sub>, also installed in series with VCHP<sub>2</sub>; fully cooled liquid out of VCHP3 then going back to said load; AHP2, installed in cascade with VCHP2, being used as a source of cooling water for the condenser of VCHP2; AHP3, installed in cascade with VCHP3, also being used as a source of cooling water for the condenser of VCHP3; this system for generating deeply cooled or supercooled liquids being characterized by the fact that the system, while remaining essentially an absorption system (most of its driving power still coming in as heat energy), permits the use, as main components, of AHPs which would normally be unable by themselves of deeply cooling liquids; said system also characterized by the fact that, despite the low working temperatures, the resulting COP is almost as good as the one obtained normally with AHPs working at higher temperatures.

## BRIEF DESCRIPTION OF THE DRAWINGS

Preferred embodiments of the present invention will now be described and schematic drawings presented. It will be assumed here for the sake of simplicity that the type of heat pump cycle used is the one found in the common vapor-compression heat pump (VCHP). Figures 1 and 2 show schematic diagrams of one-pass and two-pass flooded shell-and-tube coolers based on the present invention. Figures 3, 4, 5 and 6 show several alternative designs also based on the present invention. Figures 8 and 10 show true counterflow coolers based on the present invention and capable of using to the fullest the non-azeotropic refrigerant mixtures. Figures 7 and 9 show schematic diagrams of improved refrigeration cycles permitting an efficient generation of very cold water flows. Fig. 11 shows a simple atomizer for supercooled-liquids, the main application of which could be the generation of "artificial" snow.

### DESCRIPTION OF THE INVENTION

The invention provides a simple system for yielding very cold liquid flows, down to temperatures heretofore impossible to attain with any machine. It also discloses improved thermodynamic cycles permitting to attain these low temperatures in a very efficient manner.

# a) A low temperature cooler

The problems described above can be solved by first modifying the coolers to make them practically freeze-proof. The concept applies to most types of coolers. We will first take as an example the case of the flooded shell-and-tube cooler featuring conventional circular cross-section fluid tubes. Other types of HX will be discussed later. Figure 1 shows a modified flooded cooler (9) designed according to the present invention. It features double tubesheets (7a,7b and 7'a, 7'b), the spaces (13, 13') between the latter being filled with insulation, e.g. polyurethane foam, air, etc. The width of that space (13, 13') should be sufficient to provide a good heat barrier, e.g. about 10 to 20 tube diameters (O.D.) . Said spaces (13, 13') can also be partly or fully open to atmosphere. The improvement here is that the low velocity liquid in the cooler heads (4), (14), is nowhere in contact with a cold surface (7b, 7'b). Alternate designs will be shown later where low liquid velocity regions (4a,14b) are simply eliminated.

Preventing the cold medium (e.g. refrigerant R) from getting into the spaces (13, 13') between the tubesheets (7a, 7b and 7'a, 7'b) is easily done by expanding the tube (10) diameter into said tubesheets by means of an expander. Many expansion methods are available (see "Heat Exchangers; Selection, Design and construction", a book by E.A.D. Saunders; John Wiley and Sons, New York).

Another minor modification that will improve considerably on the cooler's reliability is a smooth entrance (3a) at the inlet end of the fluid tubes (10), as shown in figure 1. This:

- will decrease the pressure drop on the fluid side
- will prevent gradual erosion of the tubes at the inlet end (3a)
- will further reduce the chances of freezing inside the tubes (10).

The last statement can be explained by the fact that a detached flow at the abrupt entrance to a conduit (10) creates a "bubble" of reduced velocity where fluid freezing could possibly start. Preventing erosion is also important because said erosion can cause surface pitting, thus creating minute pockets of stagnant liquid which might start the freeze-up process.

Figure 2 shows another cooler (9) designed according to the present invention. It features two tubeside passes (that is, the fluid goes twice through the shell, once from left to right, in the lower part of the shell, and back from right to left in the higher part), which reduces the length of said cooler (9) for a given fluid temperature difference between inlet (5) and outlet (6). It also features the above described double tubesheets and smooth fluid-tube inlets (3a). An optional baffle (1) is also added for separating the outlet (2) and inlet (3) flows, said baffle extending far enough in the cooler right-hand head (14) to effectively prevent one fluid flow (2) from disrupting the other (3). A baffle length equivalent to about 10 tube diameters should be sufficient.

One could obviously think of many other ways of creating a thermal barrier between the cold medium and the liquid to be cooled. Instead of installing a second tubesheet (7b), one could, for example, install a piece of rigid or semi-rigid insulating material (16) between the cold medium (R) and the tubesheet (7a) (fig. 3). Expansion of the tubes (10) into the insulator (16) and the installation of O-rings (15) between the insulator (16) and the shell (9) would prevent the cold medium (R) from leaking to the tubesheets (7a).

The insulating material (16) could also be installed on the liquid side of the tubesheets, i.e. between the liquid to be cooled and the tubesheet (7a), thus eliminating the cold surface itself. One could also add extensions (20) to the conduit means (10) and even eliminate the cooler head (14) at

the exit end (fig. 4a), or even at both ends, if desired, thus <u>eliminating</u> all the <u>low fluid velocity</u> regions close to the tubesheets (7a,7'a): in this case the inner tubesheets (7b, 7b') would become superfluous.

Instead of being passively warmed up by means of insulation (16 or 13), tubesheet (7a) could also be actively heated in some way. Although this would not be done normally, it is possible to imagine instances in which active heating of said tubesheets could become desirable, thermodynamically speaking. For example, in a VCHP system, subcooling of the liquid refrigerant (from condenser) before its entrance into the expansion device provides improved cycle efficiency. Said subcooling could be performed by sending said liquid refrigerant into the space (13) between tubesheets (7a, 7b). Care should be taken so that the final temperature of said subcooled liquid refrigerant should not fall below entering-liquid (5) temperature; this might necessitate the presence of some thermal insulation on the surface of tubesheet (7b), preferably within space (13). Said subcooling within space (13) would save the cost of a refrigerant-to-refrigerant HX. Of course, said subcooling could be performed just as well within space 13<sup>t</sup>. Many other methods of actively protecting the stagnant liquid inside cooler head (4a) from freezing could be imagined.

When fed with a liquid at 3 °C, a cooler (9) like the one in figure **4a**, having the proper length and the proper coolant-to-liquid ΔT, will give an outlet temperature of 0 °C. When fed with a liquid at 0 °C instead, the same cooler (9) would have the capacity to provide an outlet temperature of about -3 °C. If the liquid was pure water, that outlet temperature would correspond to a supercooled state. A liquid is said to be supercooled if it is at a temperature well below the phase equilibrium temperature while still in the liquid phase. The supercooled state is a metastable state. In such a state, a substance has a certain phase stability; but if disturbed, it will try to acquire a greater stability, i.e. it will try to go up in temperature (up to the phase equilibrium temperature, e.g. 0 °C for pure water) by partially changing phase. For a metastable flow to continue to exist, sources of perturbations (mechanical, thermal, etc.) must be eliminated.

Looking at figure 4a and 4b, it is seen that such designs are capable of generating supercooled liquids (23): indeed, the liquid (23) at the outlets is not subjected to any perturbation (abrupt changes in direction, etc.) and no stagnant or low velocity regions exist near a cold surface (7'a, 7b). Supercooled liquid (23) out of extensions (20) can be discharged directly into some reservoir or large body of water. When fitted with longer conduit extensions (21), cooler (9) will also be capable of delivering said supercooled liquids to a nearby reservoir (26) (fig. 4b). Note that the reservoir (26) does not have to be open to atmosphere: the system works just as well, for example,

with reservoir (26) under high inert gas pressure. It should be stressed also that supercooled liquids can be stored in a reservoir (26) almost as easily as any conventional liquid, the only condition being a careful handling (minimum of perturbations). Supercooled liquids can also be delivered at locations far from the cooler (9), provided said liquids are handled according to the principles given in the present invention.

It should be noted that flows of supercooled fluids inside a conduit means can tolerate a certain degree of large scale turbulence; indeed, tests with flows of water inside a tube have shown that increasing abruptly said tube diameter by a factor of 3 (and decreasing it as abruptly a few cm further downstream) did not always provoke a freeze-up in the line. The problem is that it sometimes did! Small changes in diameter never were a problem; for example, connecting a plastic tube extension onto a metal tube, downstream of the cooler, never provoked a freeze-up, at least at normal operating conditions. Problems can arise at borderline conditions, e.g. at -3,5°C with pure water.

The design shown in figure 1, is not capable of generating supercooled liquids: the liquid (2) flowing in the outlet head (14) is subjected to too many perturbations (abrupt changes in directions, in velocity, in flow cross-sections, etc.). It can easily be modified for that purpose, however. Figure 5a shows one of several possible designs for a one-tubeside-pass cooler capable of generating supercooled liquids: connectors (31) are used to connect the tube extensions (20), inside the cooler head (14), to the main conduit means (33). Depending on the material used for making said connectors (e.g. metal or plastic) said connectors (31) can be glued or soldered to one another, side by side, and press fitted inside an extension (32) of the head (14). Said head (14) could be made in 2 parts, one of which could be removed for inspection. Of course, one can think of an infinite number of variations on the same theme, the fundamental idea being simply to eliminate low velocity regions near cold surfaces (e.g. internal tubesheets) and provide smooth flow inside the cooler (9), between said cooler (9) and main conduit (33), and also inside said main conduit means (33), large scale turbulence being reduced to a minimum. Figure 5b shows a 3-tubeside-pass cooler designed according to the present invention, perfectly smooth flow being maintained between inlet (5') and outlet (32) of said cooler (9). It should stressed that the 180° elbows (34) in figure 5b could be located inside the shell (9), if desired, instead of inside the heads (4, 14): this, however, would prevent easy cleaning of the inside of tubes (10).

Once generated and sent through a line (33) of some sort, a supercooled fluid must still be handled with special care. All perturbations must be eliminated from said line; particularly the

presence of ice crystals at any point in the flow. Even solid particles that might resemble ice crystals should be eliminated from the flow, for example by filtering the liquid at a location upstream of the cooler (9). The presence of ice crystals must be prevented especially at the final outlet ends, (22', 22, 36), where the flow leaves the conduit means (20 or 21 or 33) and enters the ambiance (said ambiance being any space having a characteristic dimension sensibly larger than said conduit means, said space containing any liquid, vapor or gas). Said outlet ends or tips (22', 22, 36) are often in a very humid atmosphere, for example in the space above liquid level in a reservoir (26); humidity will thus condense on said cold tips. Being at a temperature below phase equilibrium temperature, this condensed humidity will eventually freeze and ice crystals will touch the flow of supercooled water (23). Ice will then rapidly build up, jam said tips and prevent the liquid from flowing. This in turn will provoke freezing in the conduit means (10).

The problem can be solved by preventing any condensation on the tips (22', 22, 36). There are many ways to do it. For example, by depositing on said tips an anti-fogging material that "repels" the water vapor. Another method is simply the local heating of the surfaces of said tips. Said heating can be done by any of the known heating means, using conduction and/or convection and/or radiation as the basic heat transfer methods. Local surface heating using conductive paints as a resistive element is a possibility. Using chemical (exothermic) paints is another. Circulation of a warm liquid in a small conduit means (36') at the very tip (36) of the main conduit means (33) is another possibility (fig. 5a). Many other methods could be imagined. Very little surface heating is needed in practice since the area involved is small and since the  $\Delta T$  is also small. Indeed, for preventing condensation, the surface temperature only has to be kept above the local dew point.

Any liquid droplet splashed onto or near the tip (22', 22, 36) of the conduits (20, 21, 33) will have the same disastrous effects on the flow of supercooled liquid: said droplets will freeze on said tips and eventually provoke said conduit means blockage. The problem can be solved the same way as above (e.g. localized heating).

Local freezing and blockage at the tips (22', 22, 36) can also occur if tips are located below liquid level of a reservoir (26) or large body of water. Being at a temperature below phase equilibrium temperature, said tips will tend to induce the formation of ice at the very tips of the outlet ends. Localized heating can be used here also, but in this case, more power will be needed. Warm liquid circulation at said tips, inside a small-diameter conduit (36') is a valuable option (fig. 5a). It has also been found that making said tips out of an insulating material (v.g. plastic) and/or keeping

ambiant liquid moving relative to said tips (convection) helped in preventing freeze-up. Outlet designs favoring ambiant liquid entrainment (convection) will also work well.

The modifications described here were applied to flooded shell-and-tube coolers (9). As mentioned earlier, the invention applies also to other types of coolers. What is fundamental is the following: the presence of the thermal barrier (active or passive) between cold refrigerant and liquid in all regions inside the cooler where the liquid velocity is low compared to the average velocity inside the conduit means (10) and/or the elimination of low-velocity regions in the liquid path inside the cooler as well as immediately upstream and downstream of the cooler. In the case of supercooled liquids, other conditions have to be met: elimination of mechanical and thermal perturbations and elimination of ice crystals (and of anything that resembles an ice crystal) at any point in the flow, even at the very tip (36). Several individual flows (2) of supercooled liquid can be joined in a single flow, as long as the transition between the many individual conduit means (10) and the main line (33) is smooth and free of perturbations.

It should also be obvious that the system should never be operated while the liquid (5) is not circulating inside the cooler (9).

Figure 6a shows a coil-in-shell type cooler where the cold medium (R) is inside the shell (9) and the liquid (5) being cooled circulates in the conduit means (10), the top of said shell (9) acting as the tubesheet (7) for said conduit means (10). Figure 6b shows an element (60) of a plate type cooler featuring internal channels (61). Both types (fig. 6a, 6b) can be designed to cool liquids down to and below the freezing point if the fundamental concepts explained above are applied. In both cases, regions of low velocities near cold surfaces have simply been eliminated. In the latter case, the plates (e.g. roll-bonded aluminium) could be stacked-up, the space (62) between plates (60) being used for the circulation of the cold medium, the individual flows (2) of cold liquid merging in a manifold designed according to the present invention. By properly directing the fluid and the cold medium flows (in a fashion similar to what is done in the case of shell-and-tube coolers), it is possible to design plate-type coolers capable of counter-current heat exchange. Many other designs of plate-type HX capable of supercooling liquid are possible.

In a previous Canadian patent (#1 253 353), the applicant described an apparatus for generating ice slurries based on the supercooling phenomenon. It was then mentioned that in order to be capable of generating supercooled water, the cooler had to comprise water tubes having a certain geometry, the length-to-diameter ratio being a controlling parameter. Since then, and to the

applicant's surprise however, it has been found that length-to-diameter ratios of liquid conduit means are not an important parameter; neither is pressure, type of flow (laminar or turbulent), exact value of velocity, absolute value of diameter (or length) of said conduit means, Reynolds number, smoothness of the inside of the conduit means, etc. The important factor here is that the liquid be kept moving when touching a cold surface: the liquid must not be stagnant when it is in contact with a cooled wall. If the liquid has to be stagnant (e.g. in the head of a cooler, where the liquid is distributed to the different conduit means), said liquid must not touch any surface significantly colder than itself. Said cooled surfaces must thus be warmed up. As already mentioned, passive or active methods of warming up said cold surfaces are possible.

# b) A large ΔT chilling system

One of the above mentioned limitations of the present systems, namely the freeze-up problem, can thus be solved, as we have just seen. Let us now explain how the other 3 limitations can also be eliminated. To simplify the explanations, let us first continue using the simple case of the VCHP chiller. It has been shown above that with a conventional one-component refrigerant, the constant temperature phase change eliminates the possibility of having a large fluid-side  $\Delta T$  (e.g. from 13 °C to 0 °C) inside a single shell (9) while keeping a good system COP. There are at least three solutions to this problem. Two will make use of multiple subsystems in series; the other ones will use another type of refrigerant, the non-azeotropic mixture, and will feature improved thermodynamic cycles.

#### -- Multiple cooler systems

An example of such a system would be composed of 4 subsystems of equal capacities installed in series, each one being capable of cooling the liquid by 3 °C, this small fluid  $\Delta T$  across the cooler helping in maintaining good efficiency. The fluid would enter the lead subsystem at 12 °C, go through each subsystem, one after the other, and leave the lag unit at 0 °C. The condenser water would enter the lag unit at, say, 26 °C, go in turn through each condenser, and leave the lead unit at 38 °C. Since the  $\Delta T$  between the fluid and the refrigerant would be everywhere small, both in the CSHE and in the HSHE, the overall heat transfer efficiency would be good and the system COP would be nearer to its theoretical maximum. The arrangement would also have the advantage that if the load decreases, one or more of the subsystems could be shut down.

The same type of arrangement could theoretically be used with other types of heat pumps, for example the absorption heat pump (AHP) or the chemical heat pump (CHP). The problem here,

however, is that <u>large</u> AHPs or CHPs capable of providing low temperature liquids (e.g. 0 °C water, or ice slurries) are not available on the market and will probably not be for many years to come. The large machines that are normally available, use the pair H<sub>2</sub>0-LiBr and can only provide water at approximately 6°C, at the minimum.

Since there is a need for such low-temperature systems in the 0,1 to 30 MW capacity range, especially in space-cooling applications, the following arrangement is suggested. Figure 7a shows an example of what we could call a hybrid multiple-cascade system, capable of generating very cold liquids (e.g. 0 °C water) efficiently if large quantities of cheap, low-grade energy are available. An infinity of arrangements are possible. The important feature here is the fact that: one or more AHPs are arranged in series fashion with a VCHP, in order to bring, in two or more steps, the liquid temperature down to its desired final value; and that one (or more) AHP (or CHP) be coupled in cascade fashion with one (or more) VCHP heat pump.

Here is how such a hypothetical hybrid water-chilling system for space cooling might work. AHP<sub>1</sub>, with a capacity of 4 MW (about 1100 tons) is capable of bringing the 12 °C water (80), coming from load, down to 6 °C (81). The cascade subsystem AHP<sub>2</sub> / VCHP<sub>2</sub> (2 MW capacity) is installed in series with AHP<sub>1</sub> in order to bring water (81) from 6 °C to 3 °C (82). By itself, AHP<sub>2</sub> is not capable of bringing the water (81) temperature from 6 °C down to 3 °C. It is merely used here as a source of cooling water (CW<sub>2</sub>) at about 6 °C for the condenser of VCHP<sub>2</sub>. In a similar fashion, VCHP<sub>3</sub> (2 MW), with the help of AHP<sub>3</sub> (about 2,2 MW) as a source of cooling water (CW<sub>3</sub>), is capable of bringing the water (82) to its final temperature of 0 °C (83).

The advantage of this arrangement is the fact that most of the power (about 96%) for cooling still comes from low grade heat sources. Some mechanical power is necessary to drive the compressors of the VCHPs. But the quantities involved are very small (3% of total). The other 1% is used to drive the pumps of the AHPs. Indeed, said VCHPs have a very high COP since they are optimized to work between (almost) fixed source and sink temperatures that are very close to one another (about 6 °C apart, here). Said COPs are calculated to be around 12, if all the components as well as the refrigerant are well chosen. In this example, we would thus have a system having an overall capacity of 8 MW. The sources of power would be about 12 MW of low grade heat and less than 0,5 MW of mechanical power, assuming a typical COP of 0,75 for the AHPs.

Several strategies could be imagined for operating the system at part load conditions, the large number of subsystems in this arrangement offering all the flexibility needed. For example, one

could choose to operate the VCHPs at full load all the time, the variations in load being taken over completely by the AHPs; this would save the cost of a variable speed drive on the motors and improve on the overall COP.

Of course, many variations of the above arrangement are possible. For example, (fig. 7b) the evaporator of AHP<sub>2</sub> could be used directly as the condenser of VCHP<sub>2</sub>: the refrigerant in the VCHP<sub>2</sub> circuit would then condense, in indirect contact with the refrigerant evaporating in AHP<sub>2</sub>. Same thing for AHP<sub>3</sub> and VCHP<sub>3</sub>. Two HX would thus be saved and the COP of the VCHPs would be even greater than in the above described case.

One could even go a few steps further and keep only one AHP and one VCHP (fig. 7c), the chilled water out of the larger AHP providing all the water needed by the smaller VCHP, i.e. the condenser cooling water (entering at 6 °C and leaving at 12 °C) as well as the cooler water (entering at 6 °C and leaving at 0 °C). Warm water from load (at 12 °C) and condenser cooling water (also at 12 °C) would join and both go back to the inlet of the AHP to be chilled again. Such an arrangement still features the original idea of a VCHP in cascade with a AHP, but has a much smaller number of components. It would thus be cheaper but would be somewhat less versatile than the previous versions, as far as the number of part-load control strategies are concerned.

The above described multiple-cascade arrangements are of course capable of cooling almost any liquid, down to temperatures well below 0 °C. Moreover, it should be stressed that the working pair in the AHPs is not limited to LiBr-H<sub>2</sub>O.

#### --Non-azeotropic refrigerants

The multiple-subsystem arrangements described above are rather expensive. Using non-azeotropic refrigerants inside a single chiller will help decrease costs and further improve on the operating efficiency. Indeed, with this type of refrigerant, phase changes are not constant temperature processes; it is thus possible to keep a small and constant ΔT between the fluid and the refrigerant. Depending on the composition, the evaporation in CSHE can induce a 10, 20 or even 30 °C temperature increase (or glide). A corresponding decrease in temperature will happen in the HSHE.

There are two main criteria for an efficient machine of this type. The first is that the temperature glide of the refrigerant during phase change be the same as the temperature glide of the fluid, in

the evaporator as well as in the condenser. The other one is that both the evaporator and the condenser be counter-flow heat exchangers. The conventional flooded shell-and-tube exchanger is thus inappropriate for such an application. In very small systems, the best type of HX for real counterflow heat exchange is the basic double-pipe or tube-in-tube element. For large systems, however, other arrangements have to be found. A real counterflow HX suitable both for secondary refrigerants and for non-azeotropic phase change fluids and designed according to the present invention is shown in figure 8: it is what we might call a one-tubeside-pass "shell-and-multi-double-tube" cooler featuring three tubesheets (7a, 7b, 7c) at each end. It provides full counterflow without presenting any limit to the maximum capacity. It can be built with any convenient number of passes both on the fluid side and on the refrigerant side. If modified according to the general principles explained above, it can also be used to generate supercooled liquids.

A simplified counter-current cooler will be described later. As already mentioned, it is also possible to build counter-current plate-type coolers designed according to the present invention.

It has also been mentioned that for maximum system efficiency, the fluid-side temperature glide must match exactly the one on the refrigerant side. In practice, the perfect refrigerant mixture with a perfectly linear temperature glide, might never be found. Fluid-side and/or refrigerant-side parameters will then have to be adjusted accordingly to try and keep the refrigerant-to-fluid  $\Delta T$  approximately uniform from cooler inlet to exit, and also from condenser inlet to exit.

### --- Improved cycles

Figure 9a shows an improved VCHP cycle featuring a low-pressure receiver, LPR, acting as a reformer (i.e. a partial distiller), capable of partially separating the components of the non-azeotropic refrigerant mixture: the liquid at the bottom, rich in high-boiling-temperature (less volatile, more dense) component(s) (70) leaves towards the pump P and the evaporator EV, while the vapor (71) that is rich in the low-boiling-temperature component(s) (more volatile) leaves towards the compressor C. A fresh flow of partially evaporated refrigerant mixture (72) comes from an overfed evaporator EV, appearing here as a two-shell HX (EV<sub>1</sub>, EV<sub>2</sub>). Said fresh flow (72) provides the reservoir LPR with vapor (71) for the compressor (C) as well as with new liquid (70) for the evaporator, said liquid and vapor having roughly the needed composition. Liquid refrigerant also comes at a smaller rate through valve FL from condenser CR (appearing here as a two-part HX, CR<sub>1</sub> and CR<sub>2</sub>) and high-pressure accumulator HPA. The liquid to be chilled (5) flows

counter-current to the flow of refrigerant in evaporator EV and leaves at (6). Condenser cooling water also flows counter-current to the refrigerant in condenser CR.

The choice of refrigerant mixture is partly dependant on the temperature differential needed in the evaporator, on the fluid side. Possible choices of refrigerant components are the pairs R13B1/R152a and R22/R114: the high density of the first component provides a high system capacity while the high latent heat of the second provides superior efficiency. Many other binary or ternary mixtures are possible. It should be noted that if the overfeed rate is large, the refrigerant (R) will behave in the cooler (9) almost like a (non-evaporating) highly subcooled fluid. The design of the coolers (9) can then be simplified, as will be seen below. The pump P is capable of providing us with that large flow. In a way, the cycle is thus similar to the well known liquid-overfeed system, which has many well known advantages: smaller size of the components, high system efficiency and much reduced operating costs (ASHRAE Handbook: op. cit.).

This basic cycle can be modified and/or improved in different ways; what is important is the presence here of a reformer LPR which permits sending mixtures of different composition to the hot side and the cold side of the heat pump circuit. How different these compositions will be, depends in a large part on the overfeed rate through the evaporator. It also depends on the pair of refrigerants chosen. To give an idea, the denser component can represent more than 75% of the total mass flow in compressor C, but less than 50% in evaporator EV; i.e. the concentration of the dense component can thus go from 1:1 in the evaporator to 3:1 in the compressor (C). The higher the density of the vapor going to the compressor (C), the greater the capacity of the system.

The capacity of such a system can also be modulated by using methods similar to those used in more "conventional" systems. For example, the extraction system shown in figure 9b can change the average level of pressure and density in the system, permitting a modulation of the capacity (dayly, weekly or seasonal modulation). This is done by removing from circulation (extracting and temporarily storing) some refrigerant extracted from either one of two points: at or near the condenser exit, for example at the top (73) of the high pressure receiver HPA; or along the condenser itself, immediately after the point where the desuperheating is complete (about one fourth to one third of the way along the condenser), using separator reservoir SR<sub>1</sub>.

Vapor at the top of HPA is rich in the low-boiler, volatile, higher-density component(s): removing this vapor from the circuit (via valve V<sub>3</sub>) by condensing it and storing it in the extraction receiver ER will give a decrease in system capacity. Extracting the liquid (rich in the high-boiler, non-

volatile components) from the bottom of  $SR_1$  (via valve  $V_2$ ) will have the opposite effect. The valves  $V_1$ to  $V_3$ , which control the extraction process, can be made to respond automatically to a change in demand.

At part load conditions, there will be changes in temperature glide of the fluid along the cooler. The refrigerant temperature glide in the evaporator EV should thus be changed to keep a good match. This in turn will change the glide in the condenser CR: a change in condenser fluid flow (74) will keep a good temperature-glide match there. This change in the refrigerant temperature glide along the cooler EV can be accomplished in different ways, for example, by varying the refrigerant mass-flow rate through said cooler: a variable-speed drive on the pump P can do that.

Other methods can be used which will at the same time improve the cycle performance. For example, a subcooler S can be installed: the liquid (75) from condenser CR and/or high pressure accumulator HPA is subcooled by giving up heat to the cold liquid (76) coming from pump P. Said subcooling of said liquid refrigerant (75) can also be obtained by putting it in indirect contact with cool liquid about to enter (5) the cooler EV (fig. 9c).

Removing the already vaporized refrigerant at the exit of  $EV_1$  (77) (fig. 9c) and sending it (78) directly to reformer LPR will also ensure a still larger concentration of non-volatile components in cooler  $EV_2$  and more of the dense vapor at the top of reformer LPR, thus improving cycle capacity and efficiency. In a similar fashion, extracting some liquid from the bottom of  $SR_2$  and sending it to the reformer LPR (via valve  $V_5$  and expansion device CP) will ensure that the mixture entering  $CR_2$  is rich in volatile component(s), thus improving the heat transfer properties on the refrigerant side in  $CR_2$ .

Another interesting modification to the cycle is the overfeeding of the condenser, the idea being to send into the condenser CR more vapor than it is capable of condensing in one pass. This will have at least three positive effects. First, said overfeeding will permit sending a denser mixture to the compressor C; this, as we have seen, is very desirable. It will also improve on the heat transfer coefficient (HTC) on the refrigerant side, said coefficient being known to be quite low for all condensing mixtures. A large value of said HTC will reduce the size and cost of said condenser CR. It will also decrease the  $\Delta T$  between refrigerant and coolant in said CR, thus improving system COP. Different circuits could be imagined for doing said overfeeding; what is new here is the condenser overfeeding itself.

For example, figure 9d shows a cycle featuring a low-pressure reformer LPR, an overfed evaporator EV within a low pressure circuit L<sub>1</sub>, an overfed condenser CR within a high pressure circuit L<sub>3</sub>, and in between, a compression circuit L<sub>2</sub>. Said high-pressure circuit L<sub>3</sub> also comprises a high-pressure reformer HPR for partially separating the refrigerant mixture (79) into its components. L<sub>1</sub> and L<sub>2</sub> are thus connected via LPR; L<sub>2</sub> and L<sub>3</sub> are connected via HPR. Said mixture (79) out of the condenser CR is not completely condensed: its liquid part, poor in dense component(s), will accumulate at the bottom of HPR, from where it will go towards LPR, via expansion device FL; the vapor part in (79), rich in dense component(s), will be re-routed to the condenser CR by the blower B. Said blower B is capable of overfeeding said condenser CR, the overfeed rate having any value, including 1. Said blower B and said pump P will thus have very similar functions.

The cycle in figure 9d can also be used with advantage with pure refrigerants or with azeotropic mixtures of refrigerants. In this case, HPR and LPR will simply become liquid-vapor separators. Said overfeeding in CR and/or EV will still be very useful because of its potential for improving system COP and decreasing costs. Said condenser overfeeding can be used independently of said evaporator overfeeding. For example, a condenser overfeeding system would be perfect in an application like the one shown in figure 7b, where calculations show that AHP<sub>2</sub>'s evaporator will normally be unable to completely condense the refrigerant circulated in circuit VCHP<sub>2</sub>; the same thing can be said for AHP<sub>3</sub>.

More than one of the above described cycle modifications (fig. 9b to 9d) could be implemented at the same time, if desired.

Figure 10 shows schematically a SUPERPAC cooler having one shell-side pass and one tube-side pass. The number of liquid conduit means (10) in the shell (9) can be as large as needed. When used with one of the systems shown in figure 9, such a cooler is capable of providing whatever fluid  $\Delta T$  is needed, assuming the shell (9) is long enough. With a large refrigerant flow rate (high overfeed rate), the relative volume flow rate of vapor generated will be small and the cooler will behave almost like a true counter-current liquid-to-liquid HX, the closeness of the fluid tubes providing the guiding walls needed by the refrigerant for an orderly counter-current flow: the tube-in-tube arrangement of figure 8 is thus not necessary here. Still more orderly flow could be obtained with conduit means (10) featuring longitudinal fins. Figure 10 shows quite clearly how the mere presence of 2 longitudinal fins per conduit means (10) could provide very good flow guidance as

well as improved heat transfer. Of course, many other fin arrangements (e.g. with 3 fins) could be used.

The cooler (9) of figure 10 could be further improved by the installation of a tubesheet (37) made of a porous material or any material providing a small pressure drop, for the purpose of improving the cross-wise evenness of the refrigerant flow in the cooler. A second porous tubesheet (37') could also be installed in a similar fashion at the other end of the cooler. Such a cooler (9) will give the same good results as the one shown in figure 8, being particularly useful with liquid-overfeed systems and/or with non-azeotropic refrigerants, but with reduced size, complexity and cost. In the case of more conventional cycles (without a reformer LPR), the said porous tubesheet (37) could also perform part of the expansion of the liquid refrigerant exiting from the hot side of the heat pump.

Liquid expansion from high to low pressure inside the cooler (9) itself could be performed in many other ways. For example, one of the fluid conduit means (10) inside said cooler could be replaced by a capillary tube extending between tubesheet 7'b and tubesheet 7b; the liquid refrigerant under pressure would, for example, enter space 13', where it would be partly subcooled, go through said capillary tube while being cooled (counter-current) by neighbouring evaporating refrigerant mixture R. After going through cooler EV, said liquid refrigerant would have a reduced pressure and would be sent directly to reservoir LPR (valve FL then being superfluous).

For a fluid  $\Delta T$  of 13 °C, the single shell of cooler (9) in figure 10 would be quite long. Its length could be cut in half by installing two shorter shells in series. Building each one of said shorter shells with two shell-side passes and two tube-side passes would further reduce the length of said shorter shells for a given fluid  $\Delta T$ . Many such combinations could be imagined. Of course, the counter-flow cooler (9) of figure 10 and all of its possible adaptations can be made to handle supercooled liquids by applying the general principles explained above.

#### APPLICATIONS OF THE FULL-RANGE CHILLER

The full-range chiller has obvious applications in the field of space cooling. It also has many other applications in very diversified fields. Following are more details on some of these applications.

# ---Space cooling

The liquid chillers described above can be used in a large number of cooling applications, especially space cooling. For example, in retrofit applications, the full-range (SUPERPAC) chiller can be installed in series with a conventional chiller to obtain water at any temperature between 6 °C and 0 °C (using one SUPERPAC unit or, preferably, two SUPERPAC units in series, for better efficiency). A final temperature of 0 °C would give an increase in system capacity of more than 100% without any change in the size of the main chilled-water line.

If a further increase in capacity is ever needed, another cooler capable of generating supercooled water can be installed in series: when sent to a reservoir, this supercooled water can be made to partially change phase in a reservoir: the ice crystals are accumulated while the liquid is recirculated to obtain more ice. The floating ice crystals can be used for short or long term energy storage. The water at the bottom of the reservoir can be circulated and used as the energy transport fluid. The slurry itself can also be used as the energy transport fluid.

Another possible arrangement for obtaining an ice slurry for energy storage purposes would be the following: during off-peak hours, the lag unit (used normally for generating very cold water, e.g. at 0 °C), could be used to obtain supercooled water and ice crystals. During day time, this ice could be used to shave the peak load, while said lag unit would come back to its normal use, e.g. the cooling of water from 3 °C to 0 °C.

The crystallization of supercooled liquids (e.g. inside the reservoir) can be obtained by a mechanical or thermal perturbation, for example an abrupt change in direction, a vibration of any frequency (e.g. from a sonic or ultrasonic generator); by a cooling element (e.g. of the thermo-electric type) touching the liquid (e.g. partly under and partly above liquid level), etc.; or by seeding the supercooled liquid with ice crystals or a piece of ice obtained elsewhere.

The ice slurry accumulating in the reservoir is probably the ideal energy transport fluid, in new installations as well as in retrofit situations: at the proper solids concentration (e.g. 20%), it will circulate in pipes, pumps and heat exchangers. Its energy absorption capability being about 4 times that of liquid water at 6 °C, mass flows in main lines will be 4 times smaller, pumping power will be about 5 times smaller, line diameters about 50% smaller, etc. Different storage strategies can be utilized depending on the application, the main advantage always being that pure water can be

used instead of solutions containing glycol or other environmentally dangerous freezing-point depressants. If desired, however, additives can be added to the slurry to further improve on its characteristics: anti-friction and anti-corrosion additives, dispersants that will prevent the agglomeration of the crystals, etc.

If the ice slurry (24') (a mixture of ice crystals (24) and cold liquid (25)) is to circulate well between reservoir (26) and the location where it is needed, one can add dispersants into said reservoir (26). Said dispersants will prevent crystal agglomeration and ensuing blockage of outlet (42) with ice crystals (24). A special design of the reservoir outlet (42) can also help ice crystals (24) in entering the ice-slurry conduit (28). Figure 4c shows one possible design, based on the ejector principle, with the added feature of a "Coanda" inlet. The pump (29) removes liquid (25) from reservoir (26) and sends it (43) to distribution chamber (44) from where it is pushed through thin slot (45); because of the "Coanda effect", the resulting wall jet (46) (having, in this example, a circular symmetry) "sticks" to the wall of bellmouth-shaped outlet (27) (see any good book on fluid mechanics for an explanation) and enters into conduit (28). Liquid (25) and ice crystals (24) are entrained very efficiently by said wall-jet (46) into conduit (28), agglomerations close to outlet (27) being easily broken-up by turbulent wall jet (46). Many variations on the same theme are possible; what is new here is the efficient combination ejector/wall jet/"Coanda effect".

The ice crystals (24) would move more easily towards said outlet (42) if large ice-crystal agglomerations were broken down into smaller agglomerations by one or more jets (49) of liquid, said liquid (47) flowing through pipe (48) having been pumped from the bottom of reservoir (26) by a pump similar to pump (29). Several jets (49) could be arranged in such a way as to produce a vortex having its axis going through outlet (42), all axes of said jets (49) being at an approximately identical angle (e.g. 30°) with said axis of said vortex, said axis of said jets (49) all passing at a small and approximately identical distance from said axis of said vortex. Said large ice agglomerations would then be broken down by said multiple jets (49), said jets acting like so many water-jet saws. Outlets of pipes (48) should preferably be located above liquid level. Entrainment of small agglomerations towards said outlet (42) would also be very efficient. Other physical arrangements are possible. What is new here is the combination of sawing action and entrainment action, resulting in moving and cutting said large agglomerations at the same time.

A simpler way of destroying ice-crystal agglomerations, using relatively little external energy, is the installation, inside said reservoir outlet (42), of a "self-propelled" rotor, said rotor being made in such a way that it can be partly located inside conduit means (28) and partly inside reservoir

(26), the part of said rotor located at least partly inside said conduit means (28) acting as a turbine capable of driving any device, said device being the part of said rotor protruding inside said reservoir (26), the purpose of said device being mainly the breaking-up of ice-crystal agglomerations and also the entrainment of crystals towards said outlet (42); said turbine being designed approximately like one of the well known types (e.g. axial (Kaplan) type; or mixed flow type); said device being made approximately like a propeller, an axial flow pump or any component that can at the same time breakup said agglomerations and entrain said crystals.

Another way of destroying ice-crystal agglomerations very efficiently, without using much external energy, will be the installation, inside said reservoir (26), of slowly-rotating giant "egg beeters". Properly located in front of the outlet (42), such a device will insure a well-behaved slurry flow into inlet (42) and conduit means (28).

# ---Heat pumps

The coolers constructed according to the present invention can also be used as CSHE for liquid-source heat pumps, using conventional cycles as well as the new improved cycles described above. Indeed, with a CSHE design such as the one in fig.1, for example, cold water down to about 2 or 3 °C can be used as a source of heat, for space heating as well as for many other uses. In the case of saline water, the temperature of the source can be as low as 0 °C. This cannot be done with other types of CSHE now on the market. With CSHE designs such as the ones shown in fig. 4, 5, etc., liquids at the freezing point (e.g. from a lake during the winter period) can be used as a heat source. The supercooled water coming out of the machine is simply returned to the large body of water where it will tend, being lighter than water at 4 °C (and at 0 °C), to move up to the surface and mix with the ambiant water; if ice is generated by this mixing process, it will tend to float on the surface.

Such heat pumps can be used for space heating. They can also be used in many special situations, e.g. snowmelting plants: indeed, in certain northern cities, disposing of the polluted snow removed from city streets can become a problem. With the invention, energy from a partly frozen lake or the sea can be transferred to the hot side of the heat pump and to the snow to be melted. A possible arrangement could be the following: a large pit similar to a very large swimming pool is dug in the ground. Its walls are then covered with a large number of pipe loops in which the warm fluid (e.g. a brine heated with the energy from the heat-pump HSHE) will circulate. The loops are then

covered with several cm of earth. When trucks (transporting the snow) dump their load into the pit, the bottom and the sidewalls of the pit being warm, the snow will melt and drain down into the soil. If needed, the melted snow can also be sent to a water treatment plant; in this case, an impermeable sheet (e.g. plastic sheet) would have to be layed down over the bottom and sidewalls of the pit (e.g. after covering said pipe loops with earth). Many improvements to such an installation could be imagined.

# ---Deep mines cooling

Deep mines must be cooled because the earth gets warmer as one digs deeper: at 4 000 m, it is at a temperature of 60 °C approximately. Obviously, the ambiant air down below must be cooled. The above described full-range chiller being capable of generating very cold water down to about -2 °C, it becomes possible to send cold or supercooled water into the mine from ground level (or any underground level), being sure that it will arrive at the bottom still very cold. This could even be a more economical proposition than the circulation of ice slurries being studied for certain very deep mines. If desired, such ice slurries can also be generated by the above described invention and circulated inside the mine.

### ---Snow-making machine

Most ski slopes are now equiped with one type or another of snow-making machines. The machine-made snow, however, is of poor quality: depending on the ambiant conditions, it will sometimes have the look and feel of hard ice pebbles, sometimes of wet slush, rarely that of real snow. The difference in quality comes from the fact that real snow is made from supercooled water droplets (i.e. the clouds). With the full-range chiller described above, it is possible to obtain all the supercooled water needed. When a flow of supercooled water from a conduit is changed into a flow of small droplets, the supercooled droplets will rapidly start freezing in a dendritic fashion, the droplet becoming a mixture of dendritic ice and water which, in cold air (e.g. ambiant), will grow to a full size snow flake.

There is no limitation as to the type of atomizer used for delivering the liquid in droplet form, the basic parameter being the fact that the snow is made here from supercooled liquid. For example, the atomizer shown in figure 11 is capable of generating very small droplets: supercooled liquid

(90) flows out of a conduit (91) as a thin, wide film (93); compressed air (94) going through tube (95) and through wide thin slot (96) is then blown through said thin liquid film (93) of supercooled liquid, changing the film flow to a small droplet flow (97). As mentioned earlier, tip (92) of said tube (91) must be modified (e.g. said tip being heated) for reliable operation. When the partly frozen droplets generated by an atomizer are very small in size, like the ones generated by an atomizer similar to that of figure 11, they can also be used as seeders for larger droplets of cold liquid generated by more conventional methods: when the small dendritic crystal touches a larger drop of liquid (that has been cooled, e.g. by ambiant air), it provokes, in said larger drop, the beginning of the phase change process.

The supercooled water necessary for the snow-making atomizer (e.g. that of fig. 11) can also be obtained by simply circulating cold water in a properly designed air-to-water HX (v.g. a pipe loop or a plate HX located above ground in cold ambiant air). Such a system would be more difficult to control but would save enormous amounts of energy. The HX would have to be designed following the guidelines given above.

Instead of using ambiant air for providing further cooling of water droplets and/or for completing the phase change process, a flow of cold air could be used, said cold air having been cooled by some type of HX, e.g. the CSHE of a refrigerating system, or some other device, e.g. a vortex tube. It will then be possible to generate snow in any weather, even during summer.

### ---Freeze concentration

Freeze concentration (FC) applications are divided into two categories. In the first one, the desired product is the solvent and the degree of concentration is low because the concentrate is easily disposed of. Water desalination is an example: the concentrate (e.g. 50% or more of the original solution) is discharged back into the sea. In the second category, the desired product is the concentrate. A relatively high concentration is needed; what is disposed of is the solvent. Examples are the concentration of acids, alkali, milk, salts, coffee extract, etc. Being formed without any physical constraint, the ice crystals formed from a supercooled aqueous liquid flow are absolutely pure solid water. Supercooling can thus become the basic process in a new freeze concentration method. An example is obtaining potable water from polluted water. When separated from the original solution, washed clean and melted, the crystals become pure water.

The separation process can be very simple: indeed, <u>crystallization and separation</u> can be made in a <u>single operation</u>. For example, sending a jet of supercooled water (from a reservoir or directly from a cooler) onto a conveyor belt (made of any convenient material, including thin perforated sheet metal) creates a mechanical shock which will initiate the crystalization process, the crystals remaining on the belt while the liquid is drained back into the reservoir. The accumulated crystals can then be moved elsewhere (v.g. into a wash column) by the belt. Other crystallization methods are possible, including vibrations from a sonic or an ultrasonic generator. Other separation methods are possible, including centrifugation.

The cleaning of crystals can be done in wash columns. This is a relatively mature technology and two types of systems are used: the gravity column and the pressurized column. The former is very tall, the height then being sufficient to squeeze the ice bed, helping in the final cleaning process, which is done by introducing wash liquid at the top of the column. The said wash liquid is obtained from melted pure crystals. In pressurized columns, hydraulic pressure forces the wash liquid to flow down.

An important advantage of the invention in FC applications is the fact that by modifying the operating conditions, the crystal size can easily be changed. "Deeply" supercooled liquids (e.g. -1,5 °C water) will produce soft and fragile dendritic ice. Slightly supercooled, low velocity liquid will give large hard ice crystals: blocks of pure ice having a mass of several kg have even been obtained during lab experiments with low-velocity supercooled water at -0,5 °C. Such blocks are obviously very easy to clean, without the expense of wash columns. Different installations can be imagined permitting the fabrication of much larger blocks of pure ice. For example, one can imagine that a large flow of supercooled water at -0,5 °C has been generated; directing one or more low-velocity jets of such water along a large plate will permit hard ice to build up to unlimited thicknesses on the plate.

In such an installation, it would be better (but not necessary) for the plate to be slightly cooled by some type of cooling system for two reasons. First, it would be easier to induce crystallization at the beginning of the process (crystallization of -0,5 °C water is relatively difficult to induce; but once started, it is self propagating); second, it would prevent the ice from slipping away from the plate, which would preferably be slanted a few degrees from horizontal to facilitate the draining of the liquid. Stopping temporarily the plate-cooling process would permit the block of ice to slip and be transported to a nearby reservoir where it could be melted. All the fabrication process could be

easily automated. It is important to realize here that the slight cooling of the said plate is not performed for freezing the supercooled water but to help in the handling process.

Melting can be done with the energy from any source, including that from the HSHE of the machine; or with the energy from a building to be air conditioned: one then has a dual-purpose system capable of purifying polluted water (or desalinating sea water, or treating waste water, e.g. industrial) and cooling a building. Other dual-purpose systems could be imagined.

# --- Miscellaneous applications

Cold or supercooled water from a full-range chiller can be used in the fishing industry. In fishing boats, it will keep the fish fresh, these being sprayed directly with said cold or supercooled water. In the case of a supercooled water spray, some ice crystals will normally form onto and around the fish. In the fish plant, said water can be used for washing the fish. Said water can also be sprayed onto the fish before or after it is prepared by the workers to make sure that it stays cold and fresh. A full-range chiller can also be used in agriculture, the cold or supercooled water being sprayed onto fruits or vegetable before or after they have been picked up: this cleans them and increases their shelf—life appreciably. An ice slurry can also be generated—with the invention, said slurry being injected, for example, into the boxes containing—fruits and/or vegetable, said boxes being normally used for transport purposes.

Said cold or supercooled water can also be used in the computer industry. Indeed, mainframe computers have to be cooled. Using said cold or supercooled water will permit the use of much smaller conduit means and HX, which will save space and money. Circulating an ice slurry is another possible solution.

Said cold or supercooled water (or the ice slurry formed with the latter) can also be used for the mixing of concrete. Indeed, problems arise during the construction of dams, bridges, etc. because of the heat generated during the hardening of the thick concrete. A solution is the mixing of the concrete using very cold water, supercooled water or an ice slurry, all of which can be obtained with the above described technology.

Finally, an important application can be found in the kitchens of hotels, hospitals, etc. Indeed, the trend now is to prepare meals days in advance, and vacuum pack them in plastic bags. Said plastic

bags should then be cooled very rapidly to keep the meat in good condition. The problem is in the cooling process: if it is cooled by a cold air stream, the meat will freeze on the surface and prevent cooling deep underneath: dangerous bacteria will then develop, especially in the case of poultry. The best method is to quickly dip said plastic bags into water at 0 °C or into an ice slurry (also at 0 °C). Both can be easily obtained with the invention. Putting the bags under a shower of 0 °C water is another possible method.

The new cooler designs presented above have often been described as parts of VCHP systems. Absorption and chemical heat pump systems were also briefly mentioned. It is worth mentioning that the new cooler designs can be used, if desired, within any of the known (and future) heat pumping cycles, even as part of far-fetched cooling "cycles" like the sound-wave cooling method. Two or more very different cycles can even be used in a cascade arrangement. There is no limitation as to the phase of the cold medium used in the cooler (e.g. liquid, gas). There is no limitation as to the method of circulation of the refrigerant in the cooler (e.g. spray type, overfeed type, etc.). There is no limitation as to the type of liquid refrigerant used in the system: primary (halocarbons, etc.) or secondary (e.g. brine, etc). There is no limitation as to the number of components in the liquid refrigerant: single component refrigerants or multi-component mixtures of refrigerants can be used. There is no limitation as to the type of liquid chilled or supercooled: acids, alkalis, coffee extracts, water, etc. When the term ice crystal is mentioned, it can actually be a crystal made out of any liquid, e.g. pivalic acid crystalizing from a methanol solution. When water is mentioned in the above text, there is no limitation as to the purity of said water.

As we have seen, the basic ideas relating to cooler design presented above can also be applied to practically any cooler type, even to the very unusual: plate type, coil in shell, flooded shell-and-tube, shell-and-multi-double-tube, etc. All of these can be designed according to the general principles outlined. There is no limitation as to the position of the cooler, e.g. horizontal or vertical. The counter-flow HXs described in this invention are not only for using with non-azeotropic refrigerants. When the term "tubesheet" is mentioned in the text, there is no limitation as to the number and type of "tubes" going through said tubesheets. There is no limitation as to the material used for making the conduit means and/or the shell and/or parts thereof: metals, plastics, etc. There is no limitation either as to the shape of the conduit means cross-section: circular, oval, flattened, etc. Said conduit means can be provided with internal and/or external fins. There is no limitation as to the exact fin size, shape, number and arrangement, except that in the case of supercooled liquid flows, the internal fins should respect the general principles stated in the invention, e.g. smooth flow, etc. There is no limitation either as to the fluid velocity and pressure

inside said conduit means: velocities and pressures compatible with common HX design practice are acceptable. There is no limitation either as to the method of construction and assembly of the CSHE: all welded, expanded conduit means, etc. When double or triple tubesheets are used, there is no limitation as to the distance between said tubesheets. The space between said tubesheets can be partially or completely open to atmosphere. There is no limitation as to the nature of the insulation used for filling the said space: urethane foam, air, etc. Said insulation, in thin sheet form, could also be fixed to one or both tubesheets, on the cold or on the warmer side, depending on the application. There is no limitation as to the temperature of the fluid at the outlet of the CSHE: liquids can be cooled above, at or below their phase equilibrium temperature (i.e. supercooled).

When supercooled, the exiting fluid can be changed partly to ice; there is no limitation as to the method of crystallization: any mechanical or thermal perturbation can be used, including vibrations, etc. Seeding with a crystal or a block of ice is another method. There is no limitation as to the type of solid-liquid separation method: conveyor belts, gravity draining, centrifugal drives, etc. There is no limitation as to the type of ice generated: depending on operating conditions, soft and fragile dendritic ice as well as hard ice can be obtained. There is no limitation as to the method of preventing ice from forming at the outlet of supercooled liquid conduits; if local heating is used, above or below liquid level, there is no limitation as to the heat transfer method used for heating: conduction, convection or radiation.

In spite of what has been said above, coolers built according to the present invention do not have to be used within any particular heat pumping cycle. Actually, said coolers can be used to obtain very cold or supercooled liquids or an ice slurry without using any thermodynamic cycle; for example, said cooler can be simply an air-to-liquid HX, said air being ambiant air. Other arrangements could be imagined.

This new technology is given the name SUPERPAC, which stands for "SUPER Pompe A Chaleur". Supercooled water generated with said invention will be called SUPER WATER. Ice obtained from a supercooled aqueous liquids will be called SUPER ICE. Snow obtained from the invention will be called SUPER SNOW. The thermodynamic cycles of figure 9 will be called SUPER CYCLES, even when said cycles are not being used to produce supercooled liquids.

### <u>Claims</u>

# 1. A liquid-cooling system comprising:

a cooler (9) containing a cold medium (R), said cooler (9) having an inlet end provided with an inlet (5') through which a liquid (5) to be cooled arrives, said cooler (9) also having an outlet end provided with an outlet (6') through which the liquid (6) cooled within the cooler (9) exits,

at least one conduit means (10) mounted within the cooler (9) and through which said liquid (5) flows at an average velocity in order to get cooled by said cold medium (R), each of said at least one conduit means (10) being straight or smoothly curved and extending within said cooler (9) between one cooler head (4) located at the inlet end of said cooler (9) and an other cooler head (14) located at the outlet end of said cooler (9), said one cooler head (4) acting as a flow distributor (4a) for the liquid (5) arriving through said inlet (5') whereas said other cooler head (14) acts as a flow receiver (14b) for the liquid (2) having passed through each of said at least one conduit means (10),

each one of said one and other cooler heads being spaced apart from said cold medium (R) by a pair of spaced-apart tubesheets (7a, 7b and 7'a, 7'b) through which each of said at least one conduit means (10) extends, each of said pair of spaced-apart tubesheets (7a, 7b and 7'a, 7'b) comprising an inner tubesheet (7b, 7'b) and an outer tubesheet (7a, 7'a), said pair of tubesheets preventing said liquid (5) which is stagnant within the adjacent cooler head (4, 14) or which flows therein at a velocity lower than said average velocity, from contacting the cold inner tubesheets (7b, 7'b).

2. A liquid-cooling system according to claim 1 in which said cooler (9) also comprises a cold-medium inlet (8) through which said cold medium (R) arrives, and a cold-medium outlet (11) through which said cold medium (R) exits, said cold-medium inlet (8) being located near said outlet end through which said cooled liquid (6) exits, said cold-medium outlet (11) being located near said inlet end through which said liquid (5) to be cooled arrives,

and in which said at least one conduit means (10) of said cooler (9) is equipped with at least two longitudinal fins (10'), said fins spanning along said at least one conduit means (10) between positions (37, 37'), positions (37, 37') located between the two said pairs of spaced-apart tubesheets (7a, 7b and 7'a, 7'b), one position (37) of said set of positions (37, 37') being located near said pair of spaced-apart tubesheets (7a, 7b) adjacent to said one cooler head (4) at said inlet end, the other position (37') of said set of positions (37, 37') being located near said pair of spaced-apart tubesheets (7'a, 7'b) adjacent to said other cooler head (14) at said outlet end,

said at least one conduit means (10) and another conduit means (10) and their respective fins (10') forming at least one formed conduit means through which said cold medium (R) flows counter-current to said liquid (5) to be cooled flowing inside said at least one conduit means (10),

the space between said one position (37) of said set of positions (37, 37') and adjacent inner tubesheet (7b) of said pair of spaced-apart tubesheets (7a, 7b) located near said inlet end acting as a cold-medium receiver for said cold medium (R) having passed through each of said formed conduit means, said cold medium (R) in said receiver then exiting through said cold-medium outlet end (11),

the space between said other position (37') of said set of positions (37, 37') and adjacent inner tubesheet (7'b) of said pair of spaced-apart tubesheets (7'a, 7'b) located near said outlet end acting as a flow distributor for said cold medium (R) arriving through said cold-medium inlet (8) and thus being distributed into each of said formed conduit means,

said cooler (9) thus being converted into a cooler providing true counter-current heat exchange between said cold medium and said liquid.

3. A liquid-cooling system according to claim 1 in which said cooler (9) also comprises a cold-medium inlet (8) through which said cold medium (R) arrives, and a cold-medium outlet (11) through which said cold medium (R) exits, said cold-medium inlet (8) being located near said outlet end through which said cooled liquid (6) exits, said cold-medium outlet (11) being located near said inlet end through which said liquid (5) to be cooled arrives, said cooler (9) also comprising a set of innermost tubesheets (7c, 7'c) located between the two said pairs of spaced-apart tubesheets (7a, 7b and 7'a, 7'b), one tubesheet (7c) of said set of innermost tubesheets being located near said pair of spaced-apart tubesheets (7a, 7b) adjacent to said one cooler head (4) at said inlet end, the other tubesheet (7'c) of said set of innermost tubesheets being located near said pair of spaced-apart tubesheets (7'a, 7'b) adjacent to said other coler head (14) at said outlet end,

and in which said cooler (9) also comprises, around each of said at least one conduit means (10), an outer conduit means (30) of a larger diameter, each of said outer conduit means (30) only extending, within said cooler, between said one (7c) and said other (7c) tubesheet of said set of innermost tubesheets,

the annular space between said at least one conduit means (10) and said outer conduit means (30) becoming an annular conduit means through which said cold medium (R) flows countercurrent to said liquid (5) to be cooled flowing inside said at least one conduit means (10), the space (41) between said one tubesheet (7c) of said set of innermost tubesheets and adjacent inner tubesheet (7b) of said pair of spaced-apart tubesheets (7a, 7b), located near said inlet end,

acting as a cold-medium receiver for said cold medium (R) having passed through each of said annular conduit means (30), said cold medium (R) in said receiver then exiting through said cold-medium outlet (11),

the space (41') between said other tubesheet (7'c) of said set of innermost tubesheets and adjacent inner tubesheet (7'b) of said pair of spaced-apart tubesheets (7'a, 7'b) located near said outlet end acting as a flow distributor for said cold medium (R) arriving through said cold-medium inlet (8) and thus being distributed into each of said annular conduit means (30),

said cooler (9) thus being converted into a shell-and-multi-double-tube cooler providing true counter-current heat exchange between said cold medium and said liquid.

4. A liquid-cooling system according to claim 3 in which said cold medium is a refrigerant mixture of the non-azeotropic type, said liquid-cooling system being formed as a heat pump having a hot-side portion and a cold-side portion including said cooler (9) and also comprising a low-pressure reformer (LPR), said reformer (LPR) being a reservoir capable of containing said refrigerant in its liquid phase and in its vapor phase,

said vapor phase occupying upper part of said reformer (LPR), said liquid phase occupying the bottom part of said reformer (LPR),

said vapor phase being entrained, compressed, condensed, and expanded inside the hot-side portion of said liquid-cooling system, the composition of said vapor phase being rich in the more volatile, higher density components of said mixture,

said liquid phase being entrained, circulated and evaporated inside the cold-side portion of said liquid-cooling system, said circulation being carried out by a refrigerant pump (P), said pump (P) being capable of overfeeding said cooler, the composition of said liquid phase being rich in the less volatile, higher latent heat components of said mixture.

5. A liquid-cooling system according to claim 4 in which the condensation of said refrigerant mixture in said hot-side portion is performed using an overfed condenser (CR), said liquid-cooling system comprising three different interconnected circuits,

the first of said interconnected circuits being a low-pressure circuit (L<sub>1</sub>) featuring said cooler, said refrigerant pump (P) and said low-pressure reformer (LPR),

the third of said interconnected circuits being a high-pressure circuit (L3) comprising a blower (B), said overfed condenser (CR) and a high-pressure reformer (HPR), said high-pressure reformer (HPR) being a high-pressure reservoir capable of containing said refrigerant in its liquid phase and in its vapor phase,

the second (L2) of said interconnected circuits, linking said first (L1) and said third (L3) interconnected circuits and also comprising said low-pressure reformer (LPR), a compressor (C), said high-pressure reformer (HPR) and an expansion device (FL),

said blower (B) being capable of overfeeding said condenser (CR) in such a way that said refrigerant (79) exiting from said condenser has a liquid part and a vapor part, said liquid part accumulating at the bottom of said high-pressure reformer (HPR), said vapor part occupying the upper portion of said high-pressure reformer (HPR),

said vapor part being recirculated by said blower (B) into said condenser (CR), said liquid part being sent to said low-pressure reformer (LPR) via said expansion device (FL), said high-pressure reformer (HPR) also being fed with refrigerant vapor by said compressor (CR),

said condensing system thus being capable of partially separating the dense, volatile components from the light, non-volatile components of said non-azeotropic refrigerant mixtures,

said dense, volatile components in said vapor part going back into said high-pressure circuit (L3), resulting in a better heat-transfer coefficient in said condenser,

said light, non-volatile components in said liquid part going towards said low-pressure reformer (LPR) via said expansion device (FL).

- 6. A liquid-cooling system according to claim 3 in which said inlet head and said outlet head of said cooler (9) are redesigned in such way that said cooler (9) becomes a multitubeside-pass cooler in which said liquid (5) being cooled can pass a plurality of times.
- 7. A liquid-cooling system according to claim 3 in which said cooler (9) is the cold-side heat exchanger of a heat pump, energy extracted from said liquid (23) being transferred to the hot-side of said heat pump, said energy then being rejected at said hot side of said heat pump and used for heating purposes.
- 8. A liquid-cooling system according to claim 3 in which said at least one conduit means (10) of said cooler (9) are provided with individual extensions (20, 21), said at least one conduit means (10) of said cooler (9) thus extending past said outer tubesheet (7'a) of said pair of spaced-apart tubesheets (7'a, 7'b) at said outlet end, cover of said cooler head (14) at said outlet end of said cooler (9) thus becoming superfluous,

said liquid (23) exiting from said cooler (9) being fed by said individual extensions (20, 21) to a location where it is needed, said individual extensions (20, 21) eliminating regions of said stagnant liquid and eliminating regions where said cooled liquid flows at a velocity lower than said

average velocity, thus rendering unnecessary inner tubesheet (7'b) of said pair of spaced-apart tubesheets (7'a, 7'b) at said outlet end of said cooler (9),

the outer surface of the extremity (22, 22') of said individual extensions (20, 21) where said cooled liquid (23) exits being constructed of an anti-fogging material and/or being heated by heating means to prevent the formation of ice crystals at said extremity (22, 22'), said extremity (22, 22') being colder than its ambiance,

said cooler (9) thus being capable of safely bringing said liquid to a temperature below its solid-liquid equilibrium temperature without provoking said freeze-ups, said liquid (5) thus being transformed into a supercooled liquid (23), said cooler (9) thus being transformed into a superchiller.

- 9. A liquid-cooling system according to claim 8 in which said cooler (9) is the cold-side heat exchanger of a heat pump, energy extracted from said liquid (23) then being rejected by the hot-side heat exchanger of said heat pump and used for heating purposes, said heat pump thus being capable of using, as a source of heat, liquids at their solid-liquid equilibrium temperature.
- 10. A liquid-cooling system according to claim 8 in which said supercooled liquid (23) exiting from said supercooler (9) is made to partially change phase, said phase change providing a slurry of cold liquid (25) and crystals (24), said slurry being at its phase-equilibrium temperature,

said crystals (24) being accumulated (e.g. in a reservoir 26) until the desired solid/liquid ratio is achieved, said slurry being kept homogeneous and in constant motion by a mixing device, said slurry then being easily pumpable towards an other location.

11. A liquid-cooling system according to claim 1 in which said spaced-apart tubesheets (7a, 7b) at said inlet end of said cooler (9) are located far enough from one another, so as to form a thermal barrier between said liquid (5) within said one cooler head (4) and said cold medium (R), the space (13) between said spaced-apart tubesheets (7a, 7b) being filled with an insulating material,

said spaced-apart tubesheets (7'a, 7'b) at said outlet end of said cooler (9) being located far enough from one another, so as to form a thermal barrier between said liquid (2) within said other cooler head (14) and said cold medium (R), the space (13') between said spaced-apart tubesheets (7'a, 7'b) also being filled with an insulating material.

12. A liquid-cooling system according to claim 11 in which said thermal barrier provided by said spaced-apart tubesheets (7a, 7b and/or 7'a, 7'b) is provided by solid insulating

material (16) fixed to said outer tubesheets (7a and/or 7'a) of said pairs of spaced-apart tubesheets (7a, 7b and/or 7'a, 7'b), said inner tubesheets (7b and/or 7'b) of said pairs of spaced apart tubesheets (7a, 7b and/or 7'a, 7'b) thus being eliminated.

- 13. A liquid-cooling system according to claim 1 in which said at least one conduit means (10) of said cooler (9) are provided with individual extensions (35), said at least one conduit means (10) of said cooler (9) thus extending past outer tubesheet (7a) of said pair of spaced-apart tubesheets at said inlet end, said individual extensions (35) permitting the feeding of said at least one conduit means (10) individually, using a distributor located at some distance upstream of said cooler (9), said individual extensions (35) and location of said distributor preventing said liquid which is stagnant or which flows therein at a velocity lower than said average velocity from contacting a cold surface, thus rendering unnecessary the presence of said inner tubesheet (7b) of said pair of spaced-apart tubesheets (7a, 7b) at said inlet end.
- 14. A liquid-cooling system according to claim 1 in which said at least one conduit means (10) of said cooler (9) are provided with individual extensions (20), said at least one conduit means (10) of said cooler (9) thus extending past said outer tubesheet (7'a) of said pair of spaced-apart tubesheets (7'a, 7'b) at said outlet end, said individual extensions (20) thus protruding from said outside tubesheet (7'a) into said other cooler head (14) in such a way that said cooler can be converted, when desired, into a supercooler capable of supercooling said liquid (5).
- 15. A liquid-cooling system according to claim 1 in which said inlet cooler head (4) and said outlet cooler head (14) of said cooler (9) are redesigned in such a way that said cooler (9) becomes a multi-tubeside-pass cooler in which said liquid (5) can pass a plurality of times.
- 16. A liquid-cooling system according to claim 1 in which said at least one conduit means (10) of said cooler (9) are provided with individual extensions (20), said at least one conduit means (10) of said cooler (9) thus extending past said outer tubesheet (7'a) of said pair of spacedapart tubesheets (7'a, 7'b) at said outlet end, cover of said cooler head (14) at said outlet end of said cooler (9) thus becoming superfluous,
- said liquid (23) exiting from said cooler (9) being fed by said individual extensions (20, 21) to a location where it is needed, said individual extensions (20, 21) eliminating regions of said stagnant liquid and eliminating regions where said cooled liquid flows at a velocity lower than said average velocity, thus rendering unnecessary inner tubesheet (7'b) of said pair of spaced-apart tubesheets (7'a, 7'b) at said outlet end of said cooler (9),

extremity of said individual extensions where said cooled liquid (23) exits being constructed of an anti-fogging material and/or being heated by heating means to prevent the formation of ice crystals at said extremity, said extremity being colder than its ambiance,

said cooler thus being capable of safely bringing said liquid to a temperature below its solid-liquid equilibrium temperature without provoking said freeze-ups, said liquid (5) thus being transformed into a supercooler and said liquid-cooling system thus being transformed into a superchiller.

17. A liquid-cooling system according to claim 16 comprising means for partially changing phase of said supercooled liquid (23) exiting from said supercooler (9), said phase change providing a slurry of solid and cold liquid, said slurry being at its phase-equilibrium temperature,

the operating conditions of said liquid-cooling system being modulated in order to modify the type of solid material generated during said phase change, said conditions being mainly the temperature of supercooled liquid,

a slight supercooling of less than about 0.5°C favouring the generation of harder and larger crystals, their shape depending mainly on the way said supercooled liquid and said slurry is handled,

a "deep" supercooling of more than about 1.2°C tending to generate finer crystals (24), a medium supercooling tending to generate medium-size crystals.

18. A liquid-cooling system according to claim 16 comprising means for partially changing phase of said supercooled liquid (23) exiting from said supercooler (9), said phase change providing a slurry of cold liquid and crystals, said slurry being at its phase-equilibrium temperature,

said crystals, when separated from said slurry and when cleaned, becoming solids which are purer than said supercooled liquid (23) from which they were made, said cooling system thus being capable of purifying liquids and concentrating solutions.

19. A liquid-cooling system according to claim 16 comprising means for partially changing phase of said supercooled liquid (23) exiting from said supercooler (9), said phase change providing a slurry of cold liquid (25) and crystals (24), said slurry being at its phase-equilibrium temperature,

said crystals being accumulated (24) (e.g. in a reservoir (26) until the desired solid/liquid ratio is achieved, said slurry being kept homogeneous and in constant motion by a mixing device, said slurry then being easily pumpable towards an other location.

- 20. A liquid-cooling system according to claim 19 in which said supercooled liquid (23) exiting from said cooler (9) is supercooled water at about -0.8°C, said ice crystals (24), when separated from said mixture, having the look and feel of natural snow, said liquid-cooling system thus becoming a snow-making machine capable of operating independently of weather conditions.
- 21. A liquid-cooling system according to claim 16 in which said supercooled liquid (23) exiting from said supercooler (9) is fed to a reservoir where it is made to partially change phase, said change of phase providing a slurry of cold liquid and crystals at its phase-equilibrium temperature inside said reservoir,

said supercooled liquid being fed to said reservoir under liquid level, said ambiance being said slurry.

22. A liquid-cooling system according to claim 16 in which said supercooled liquid (23) exiting from said supercooler (9) is fed to a reservoir where it is made to partially change phase, said change of phase providing a slurry of cold liquid and crystals at its phase-equilibrium temperature inside said reservoir,

said supercooled liquid being fed to said reservoir above liquid level, said ambiance being the humid atmosphere above said reservoir.

- 23. A liquid-cooling system according to claim 1 in which said cooler (9) is the cold-side heat exchanger (CSHE) of a heat pump, energy extracted from said liquid being transferred to the hot-side of said heat pump, said energy then being rejected at said hot side of said heat pump and used for heating purposes.
- 24. A liquid-cooling system according to claim 16 in which the flows from said at least one conduit means (20 or 31) are joined in order to create a larger flow of supercooled liquid inside a larger conduit means (33).
- 25. A liquid-cooling system according to claim 24 in which said joining is performed by bringing said individual extensions (31) close to one another, side by side, in such a way that said flows, which are then practically parallel, can be introduced into said other larger conduit means (33) witout creating any significant perturbation.

26. A liquid-cooling system according to claim 16 in which said inlet head (4', 4) and said outlet head (14) of said cooler (9) are redesigned in such a way that said cooler (9) becomes a multi-tubeside-pass cooler in which said liquid (5) can pass a plurality of times,

passage from one pass to the other pass by said supercooled liquid being performed without causing any significant perturbation to said supercooled liquid by using individual U-bend connections (34) between said at least one conduit means (10) forming said one pass and said at least one conduit means (10) forming said other pass.

- 27. A liquid-cooling system according to claim 16 in which said cooler (9) is the cold-side heat exchanger of a heat pump, energy extracted from said liquid then being rejected by the hot-side heat exchanger of said heat pump and used for heating purposes, said heat pump thus being capable of using, as a source of heat, liquids at their solid-liquid equilibrium temperature.
- 28. A liquid-cooling system according to claim 1 in which said cooled liquid (6) is used for cooling bags of hot prepared food products.
- 29. A liquid-cooling system according to claim 16 in which said supercooled liquid (23) is supercooled water, the flow of said supercooled water being changed into a flow of supercooled droplets (98) by means for causing a major perturbation, all of said supercooled droplets (98) thus rapidly becoming droplets containing a mixture of dendritic ice and 0°C water,

proportion of said dendritic ice in said droplets (98) then growing because of cold ambiant air, until said droplets become practically 100% ice crystals (i.e. snow flakes), said liquid-cooling system thus becoming a snow-making machine.

- 30. A liquid-cooling system according to claim 29 in which said droplets (98) containing a mixture of dendritic ice and 0°C water are instead used as seeders which facilitate the crystallization of other cold water droplets.
- 31. A liquid-cooling system according to claim 16, operating as part of a vapor-compression chiller, the liquid circuit of said vapor-compression chiller being arranged in cascade fashion with and downstream of a conventional absorption chiller, thus forming a hybrid-cascade system,

the capacity of said conventional absorption chiller being sufficiently large to generate enough chilled liquid both for the hot side and for the cold side of said vapor-compression chiller,

said chilled liquid normally exiting from said conventional absorption chiller at a temperature of approximately 6°C,

said cooler of said vapor-compression chiller then being capable of supercooling said chilled liquid from approximately 6°C down to temperatures below its liquid-solid equilibrium temperature.

32. A liquid-cooling system according to claim 16 in which said at least one conduit means (10) of said supercooler (9) are provided with individual extensions (35), said at least one conduit means (10) of said cooler (9) thus extending past said outer tubesheet (7a) of said pair of spaced-apart tubesheets (7a, 7b) at said inlet end,

said individual extensions (35) permitting the feeding of said at least one conduit means (10) individually, using a distributor located at some distance upstream of said cooler (9),

said extensions (35) and said distance preventing said liquid which is stagnant or which flows therein at a velocity lower than said average velocity from contacting a cold surface, thus rendering unnecessary the presence of said inner tubesheet (7b) of said pair of spaced-apart tubesheets (7a, 7b) at said inlet end.

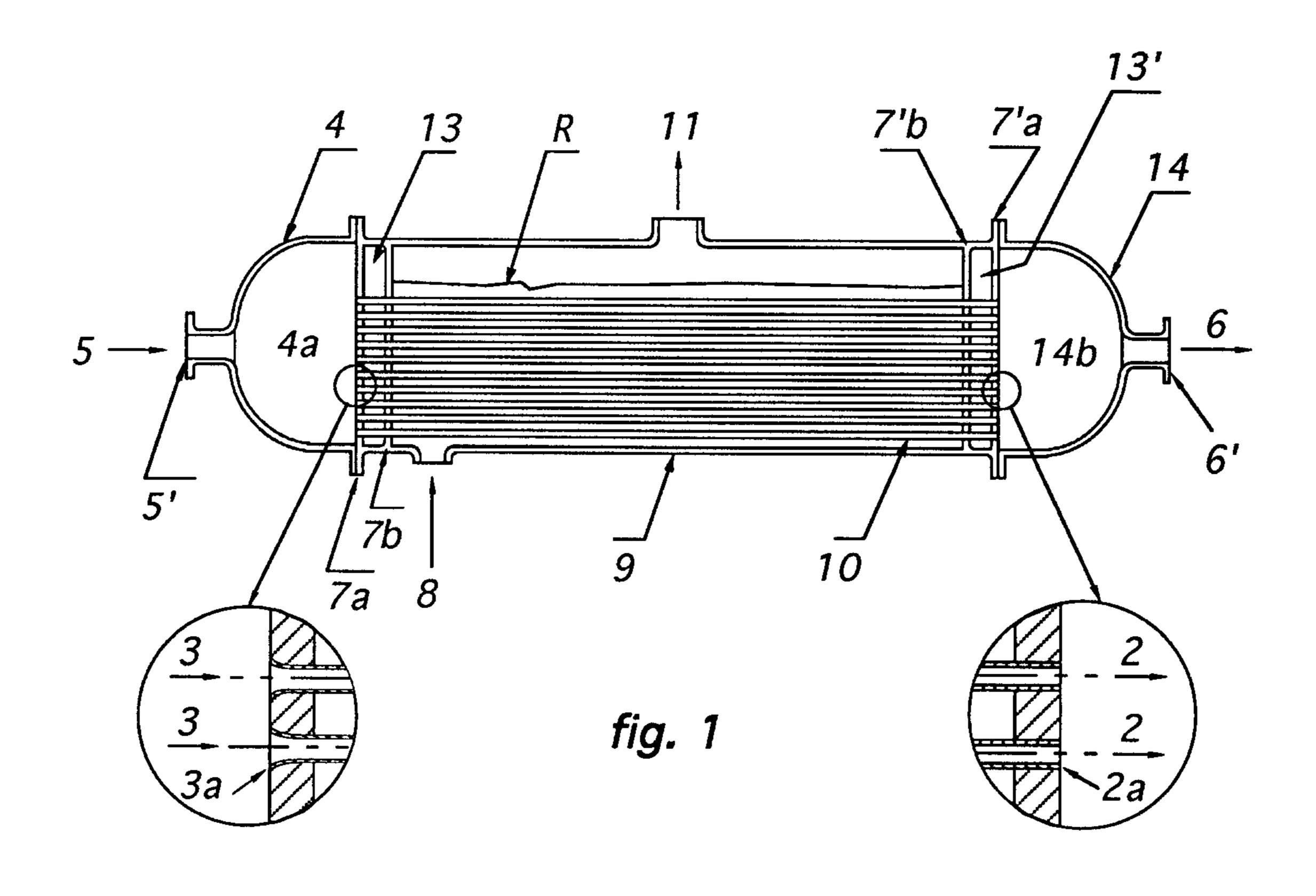
33. A liquid-cooling system according to claim 1, operating as part of a vapor-compression chiller, the liquid circuit of said vapor-compression chiller being arranged in cascade fashion with and downstream of a conventional absorption chiller, thus forming a hybrid-cascade system,

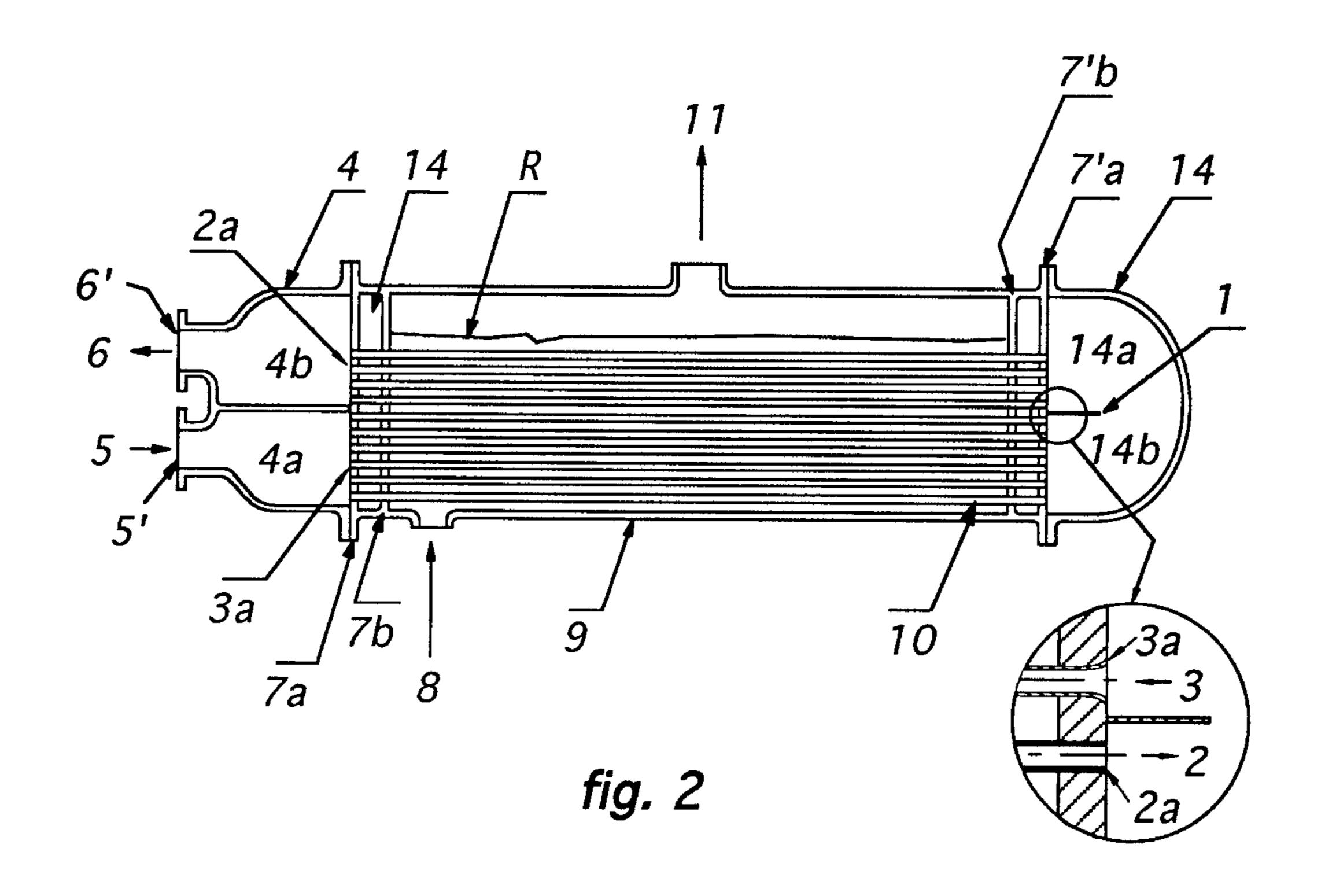
the capacity of said absorption chiller being sufficiently large to provide enough chilled liquid both for the hot side and for the cold side of said vapor-compression chiller, said chilled liquid normally exiting at a temperature of approximately 6°C,

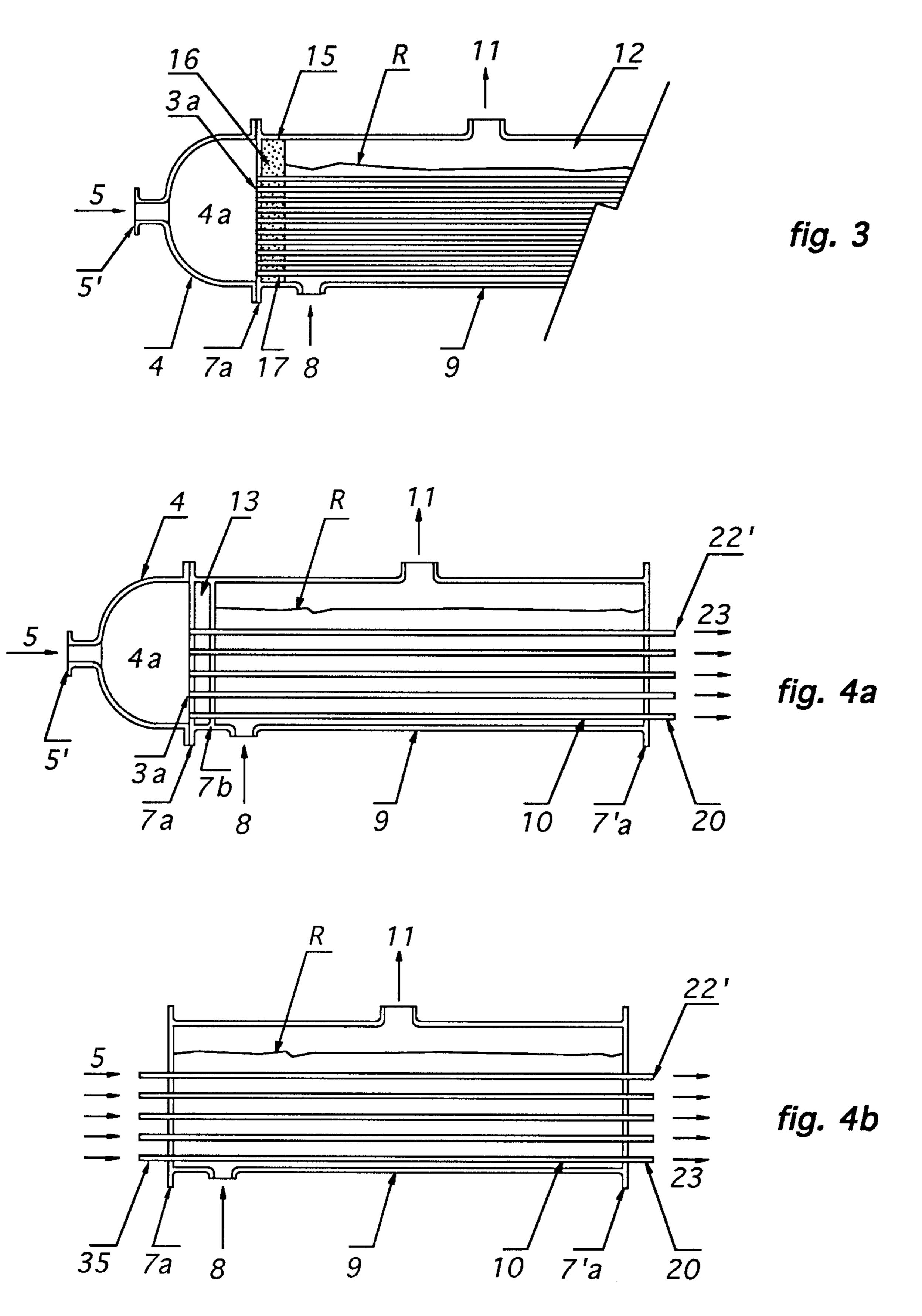
said cooler of said vapor-compression chiller then being capable of cooling said chilled liquid from approximately 6°C down to its liquid-solid equilibrium temperature.

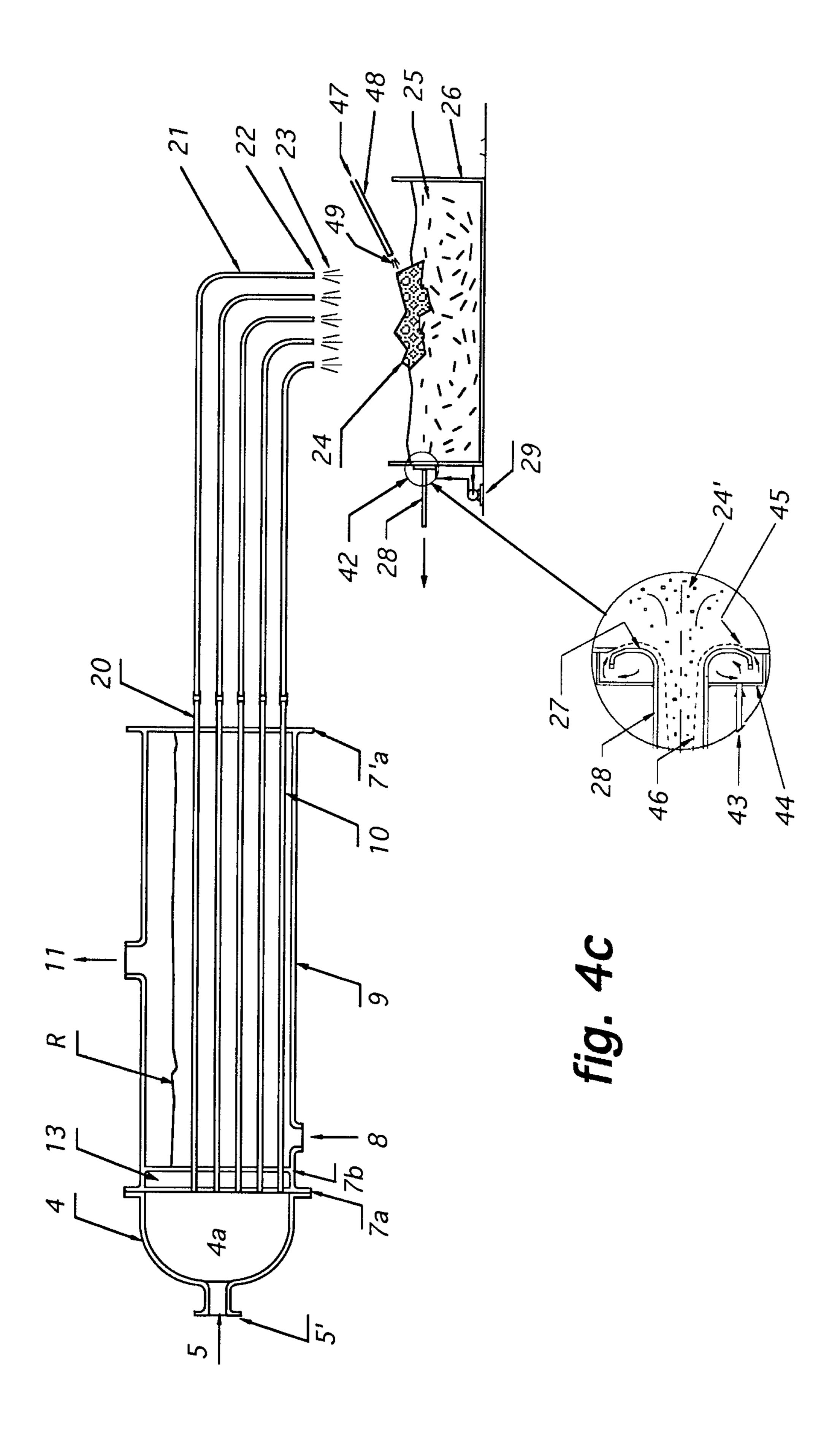
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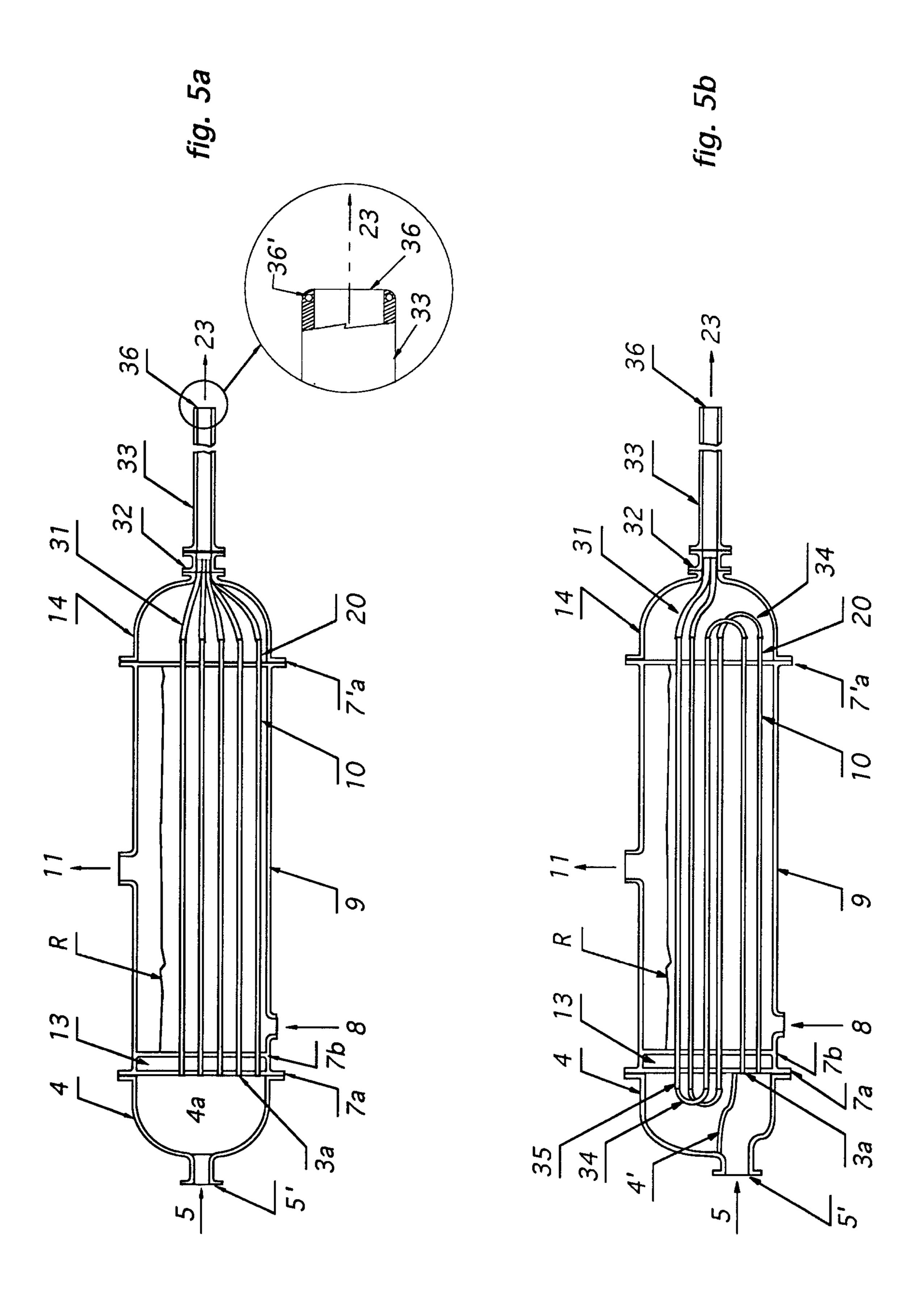
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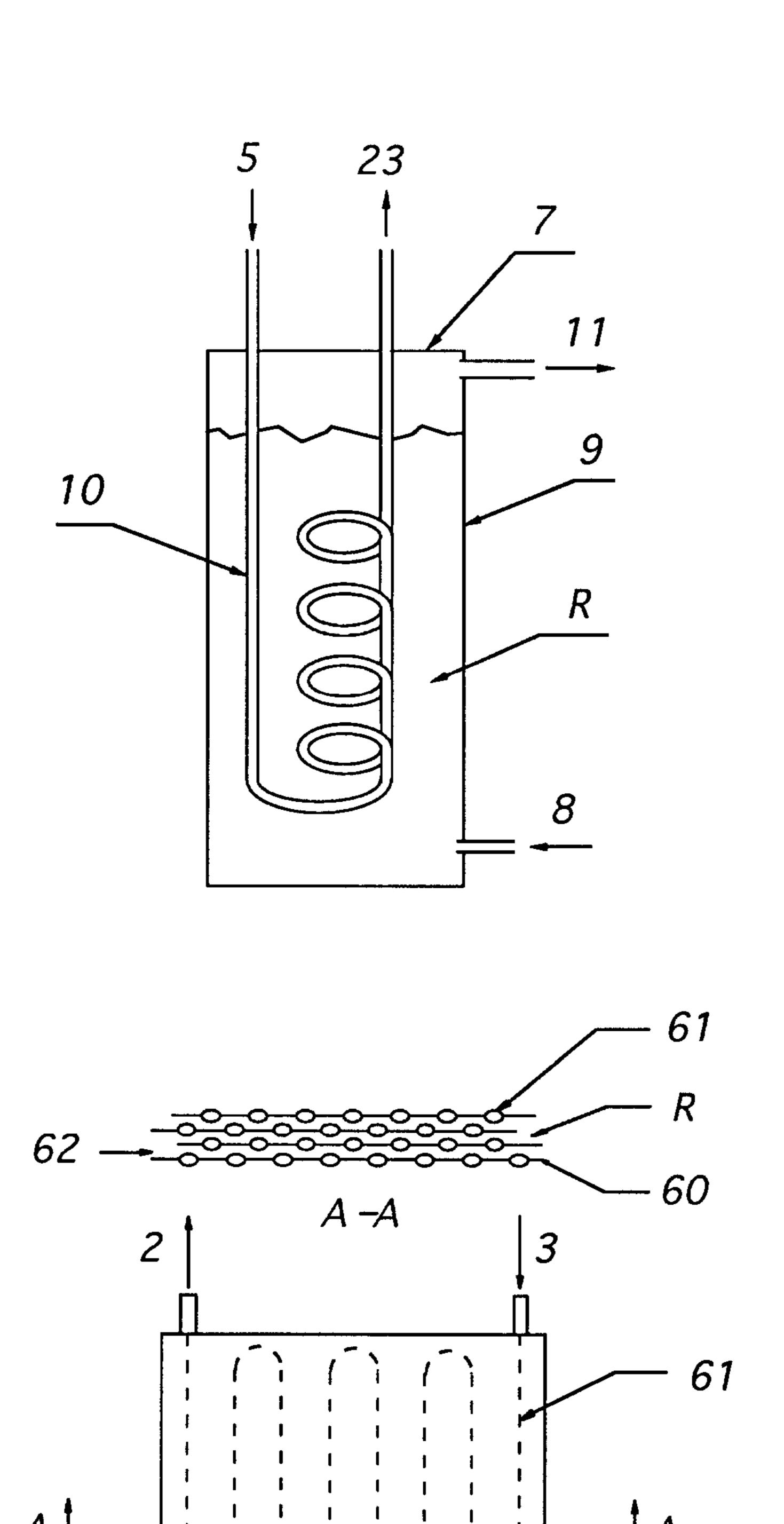


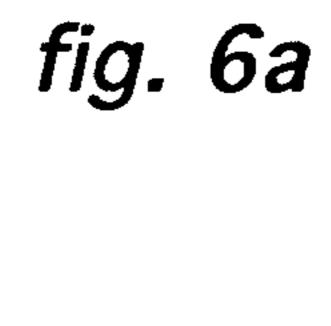


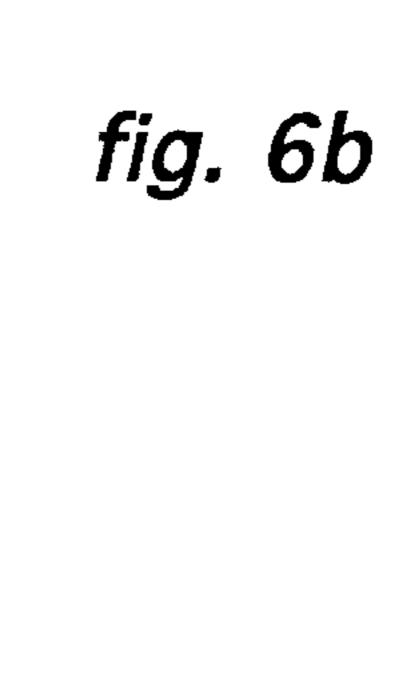


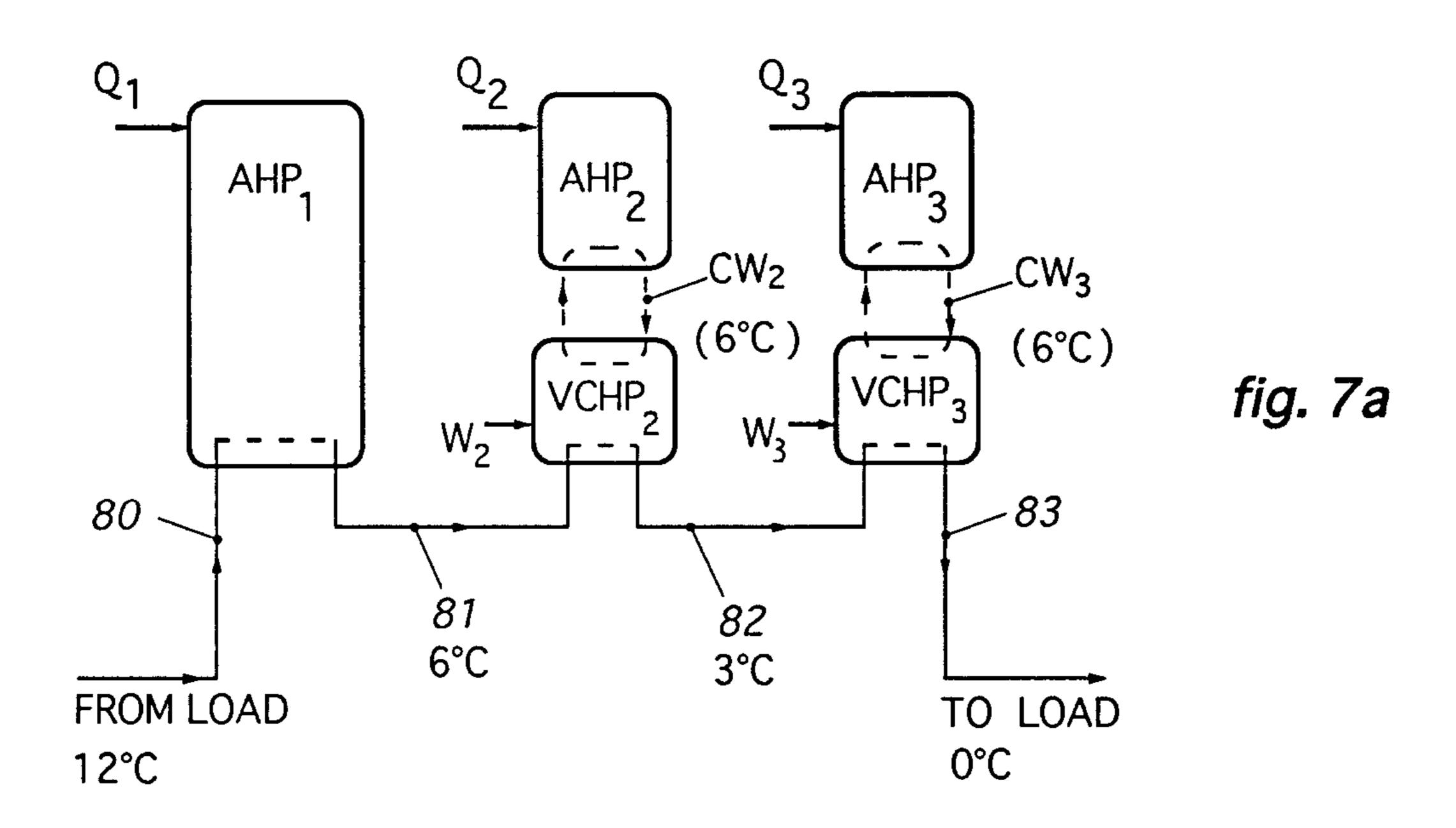


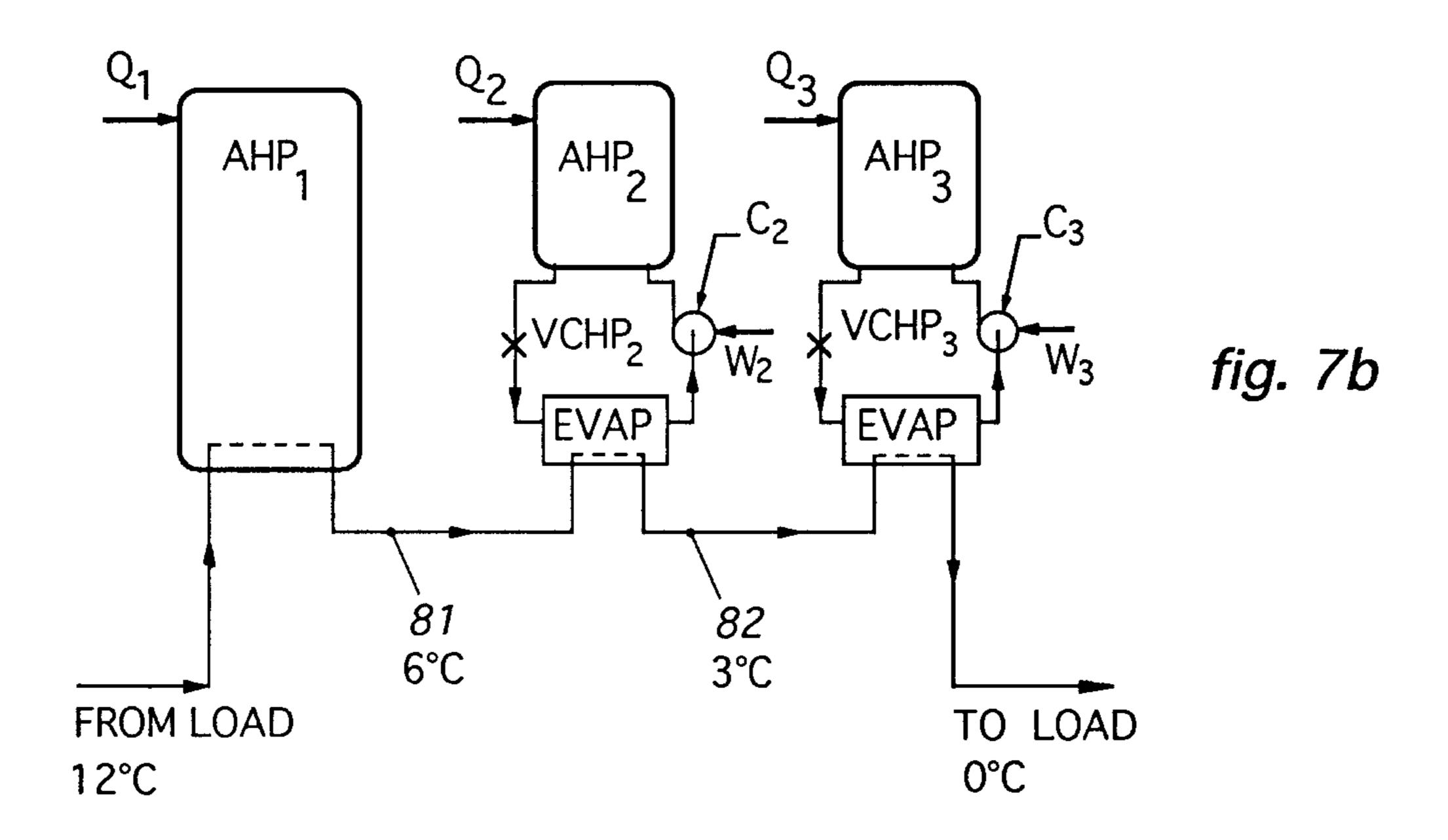












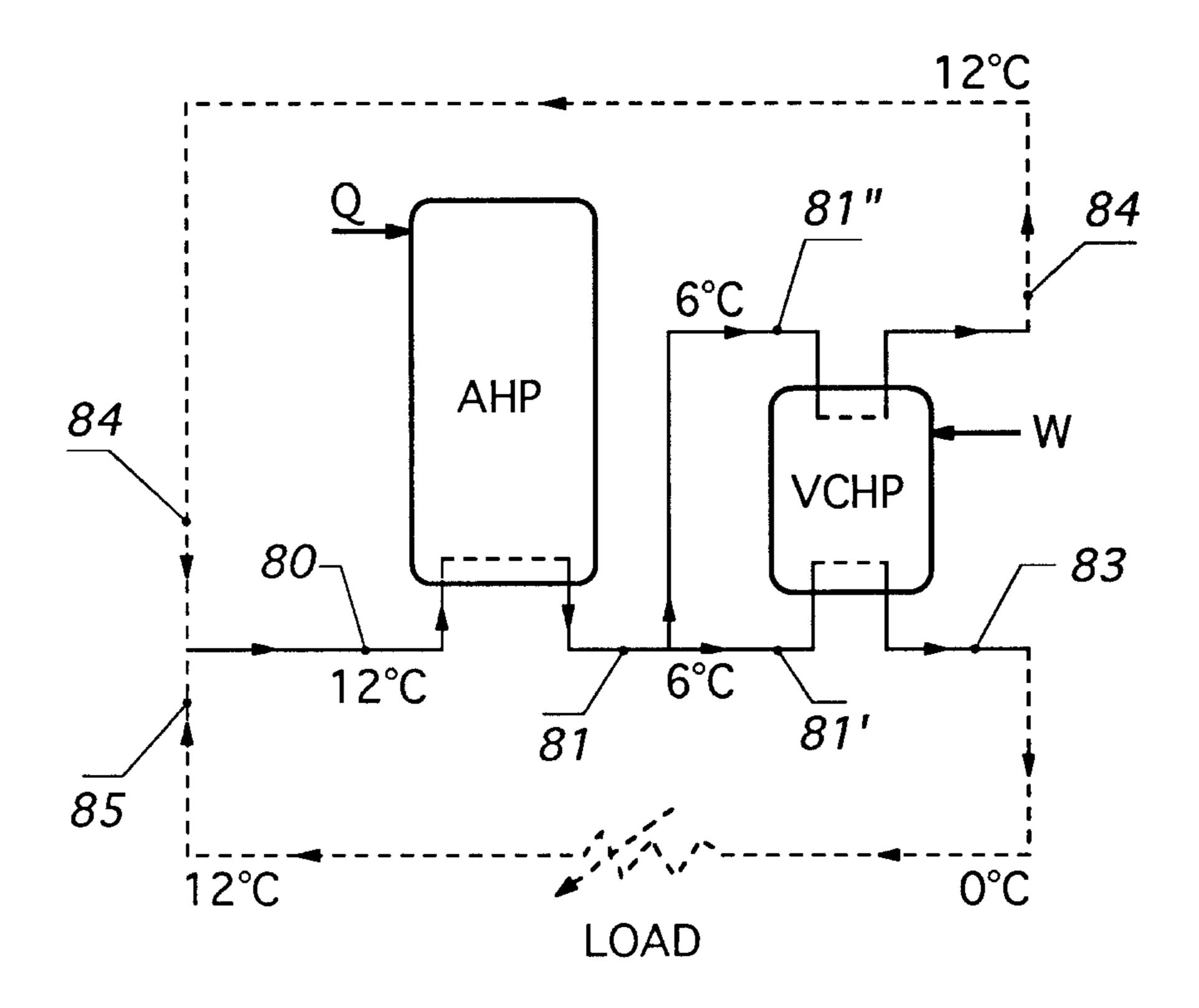


fig. 7c

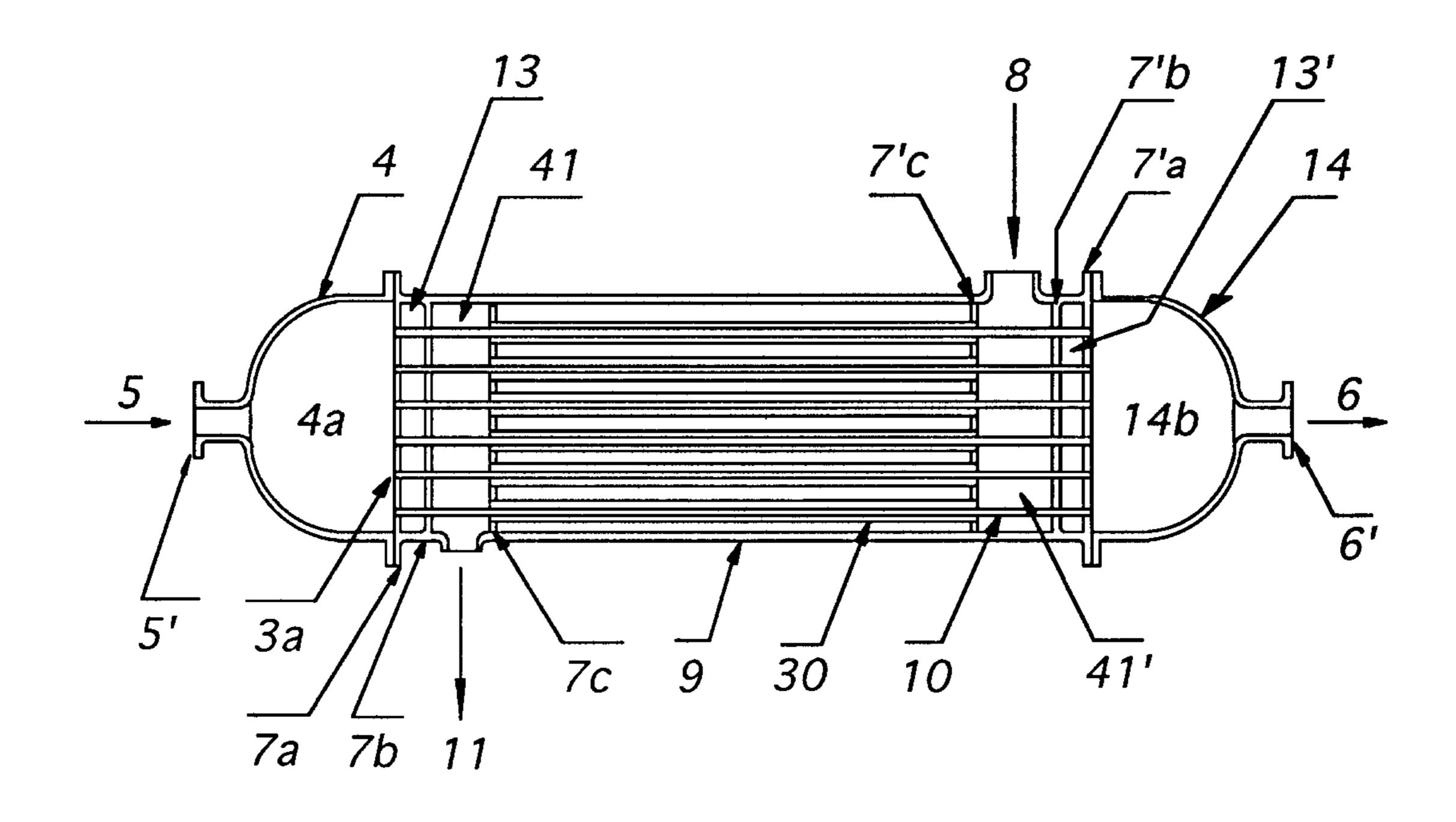


fig. 8

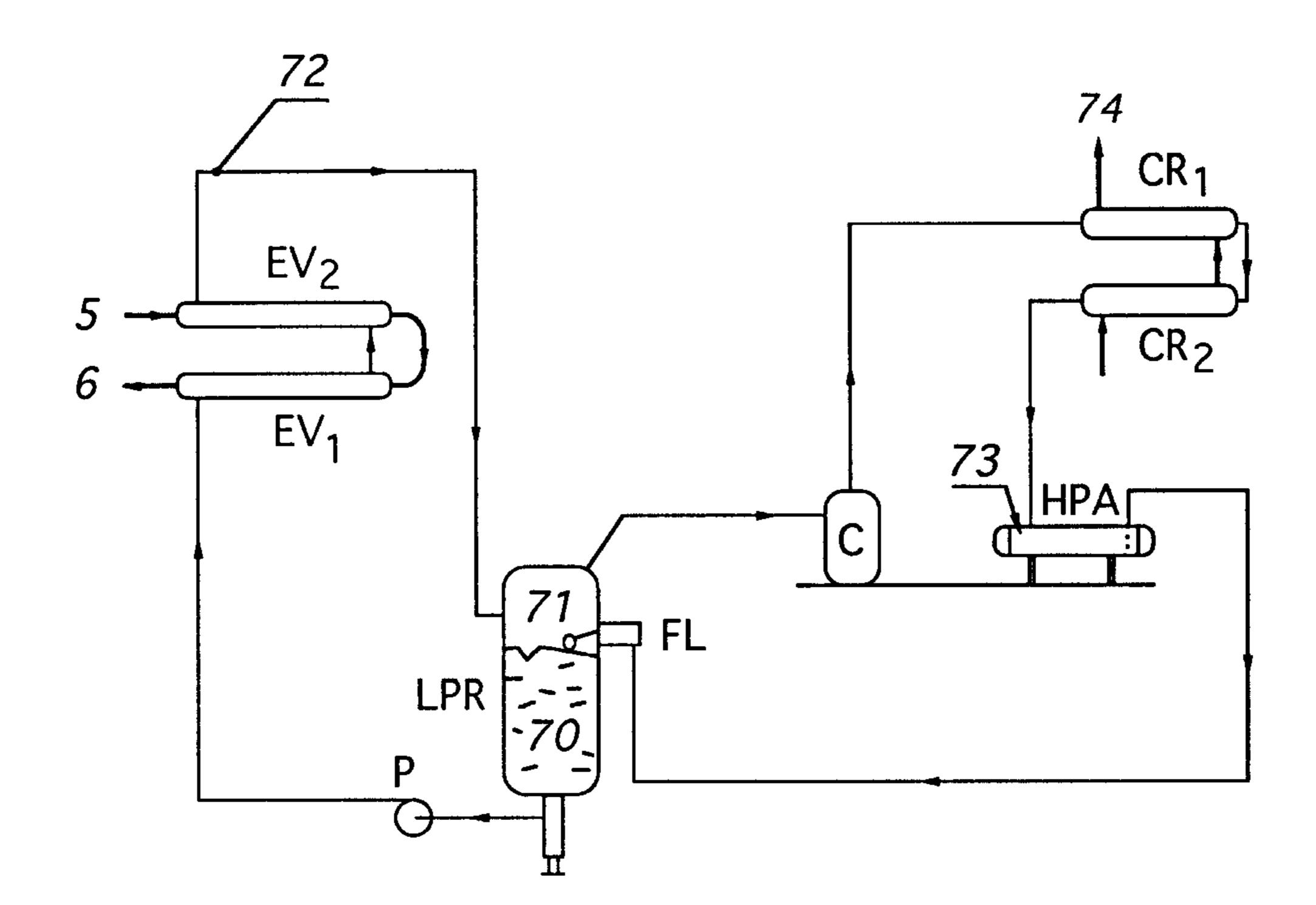


fig. 9a

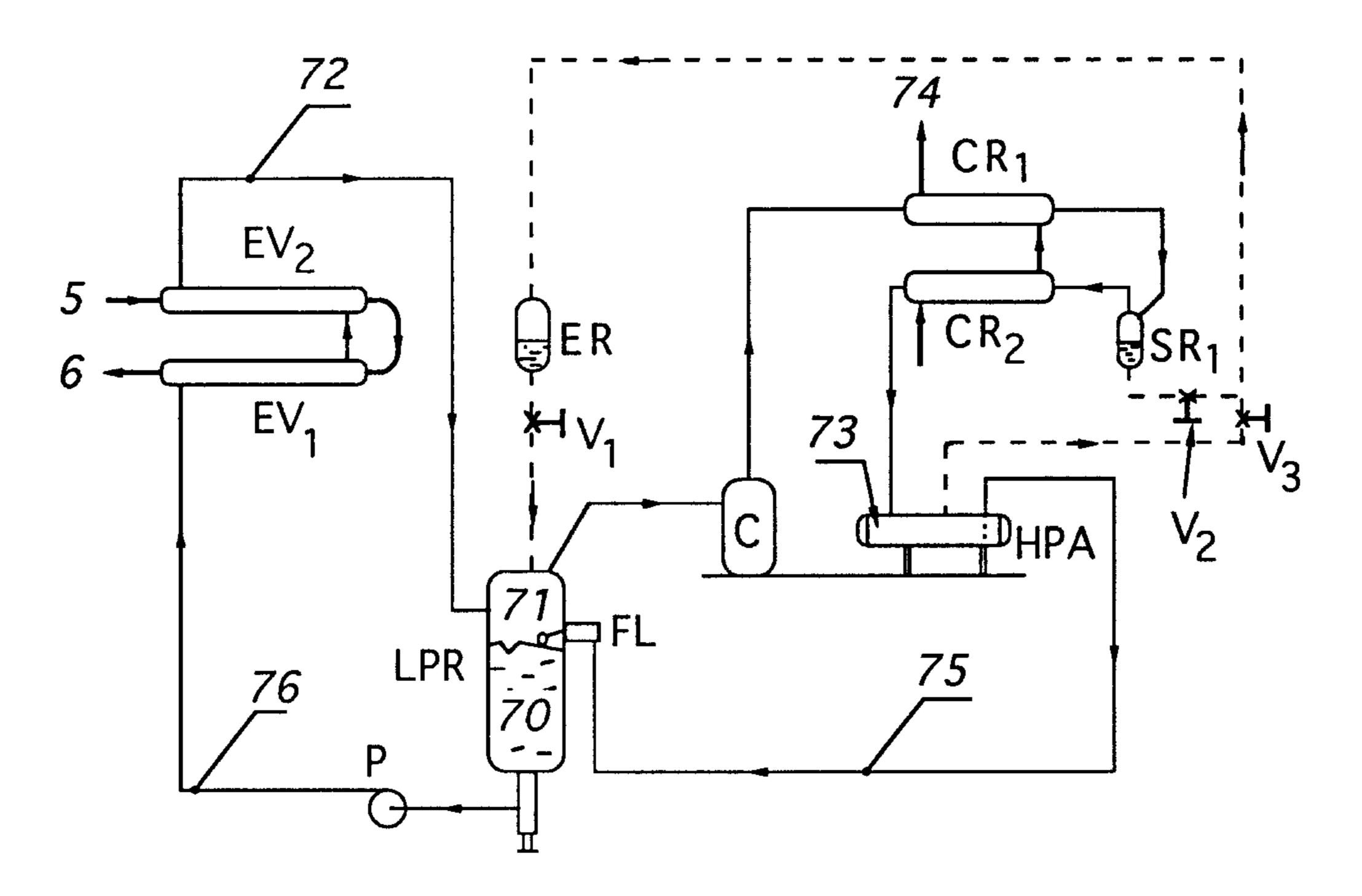


fig. 9b

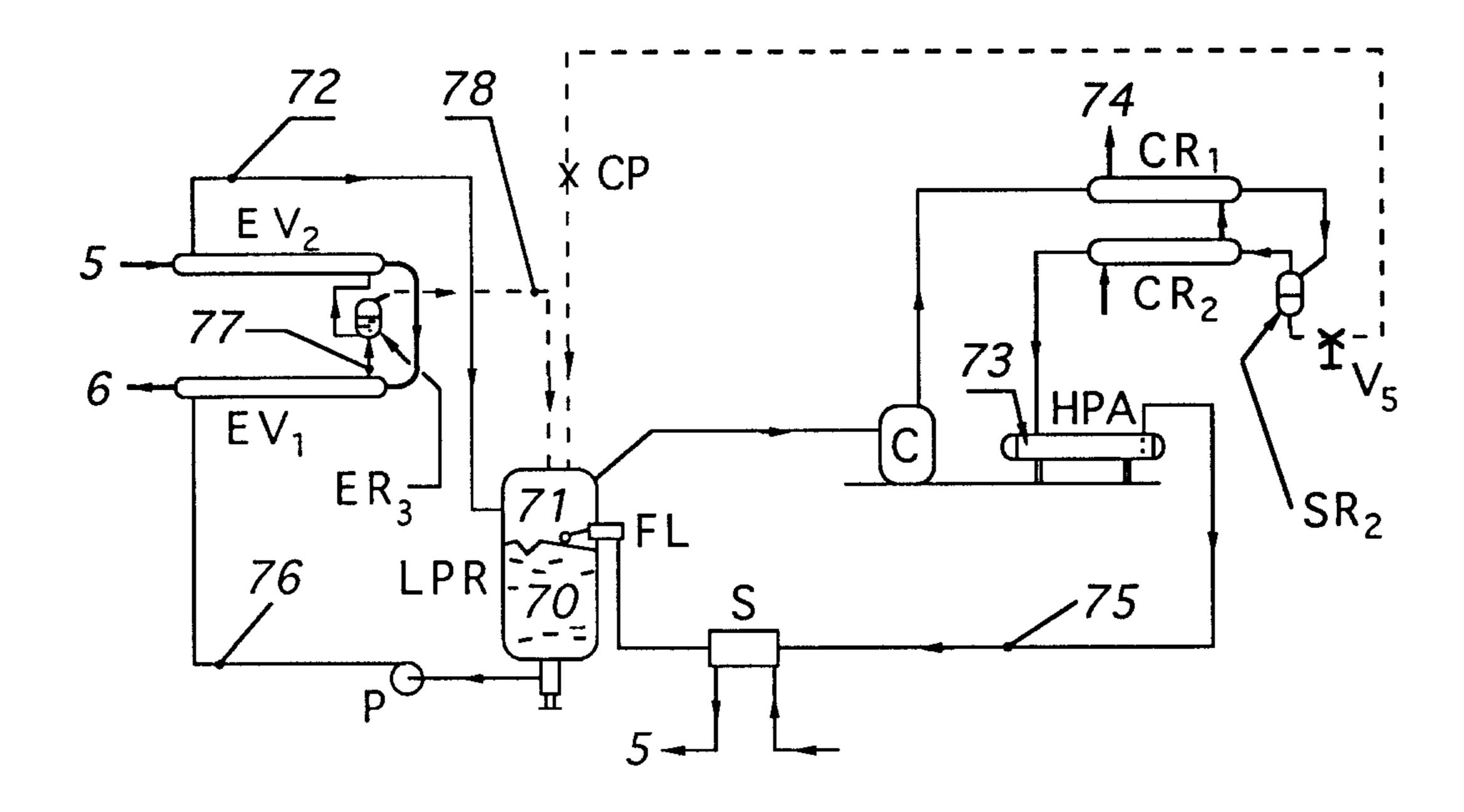
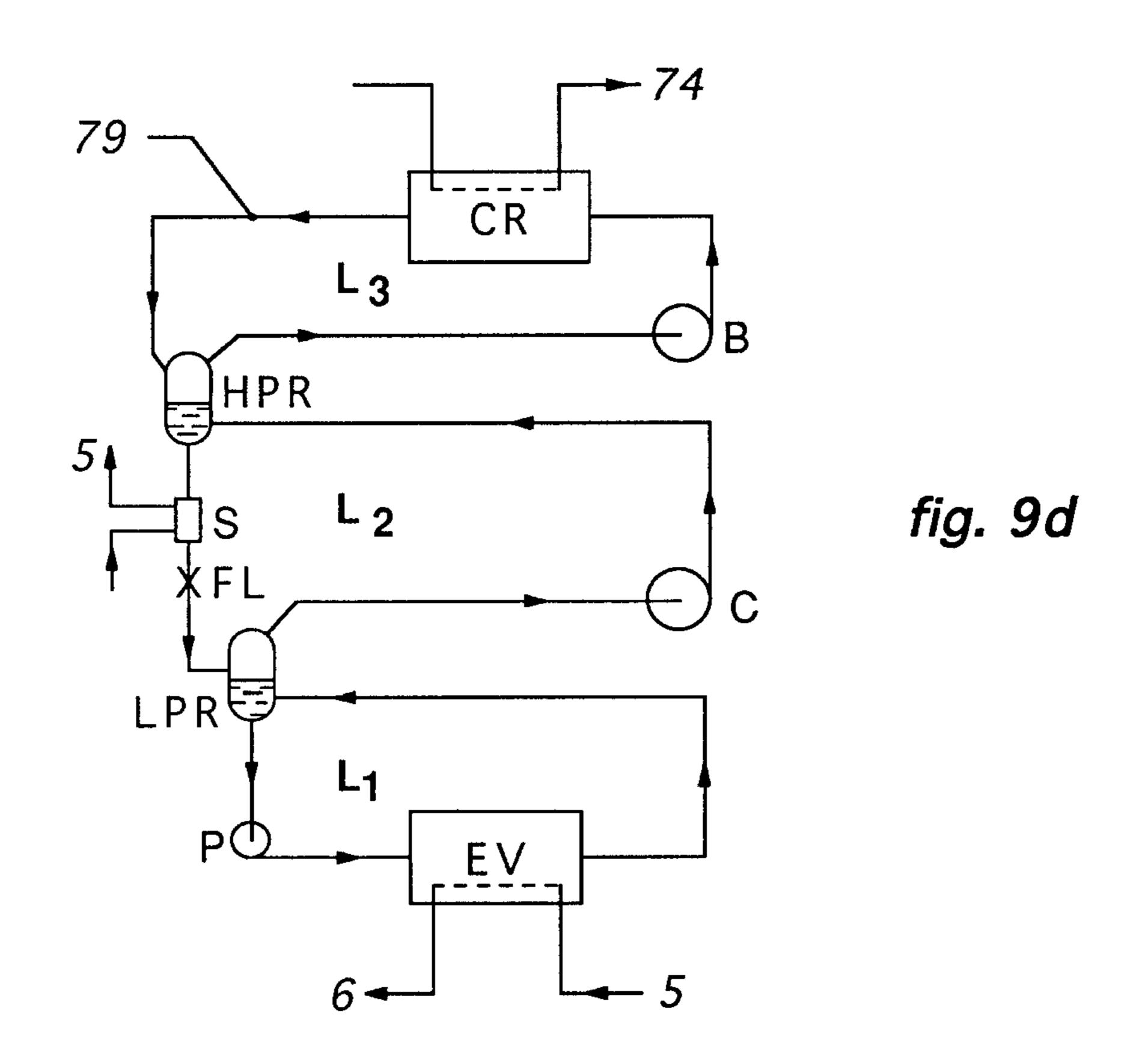


fig. 9c



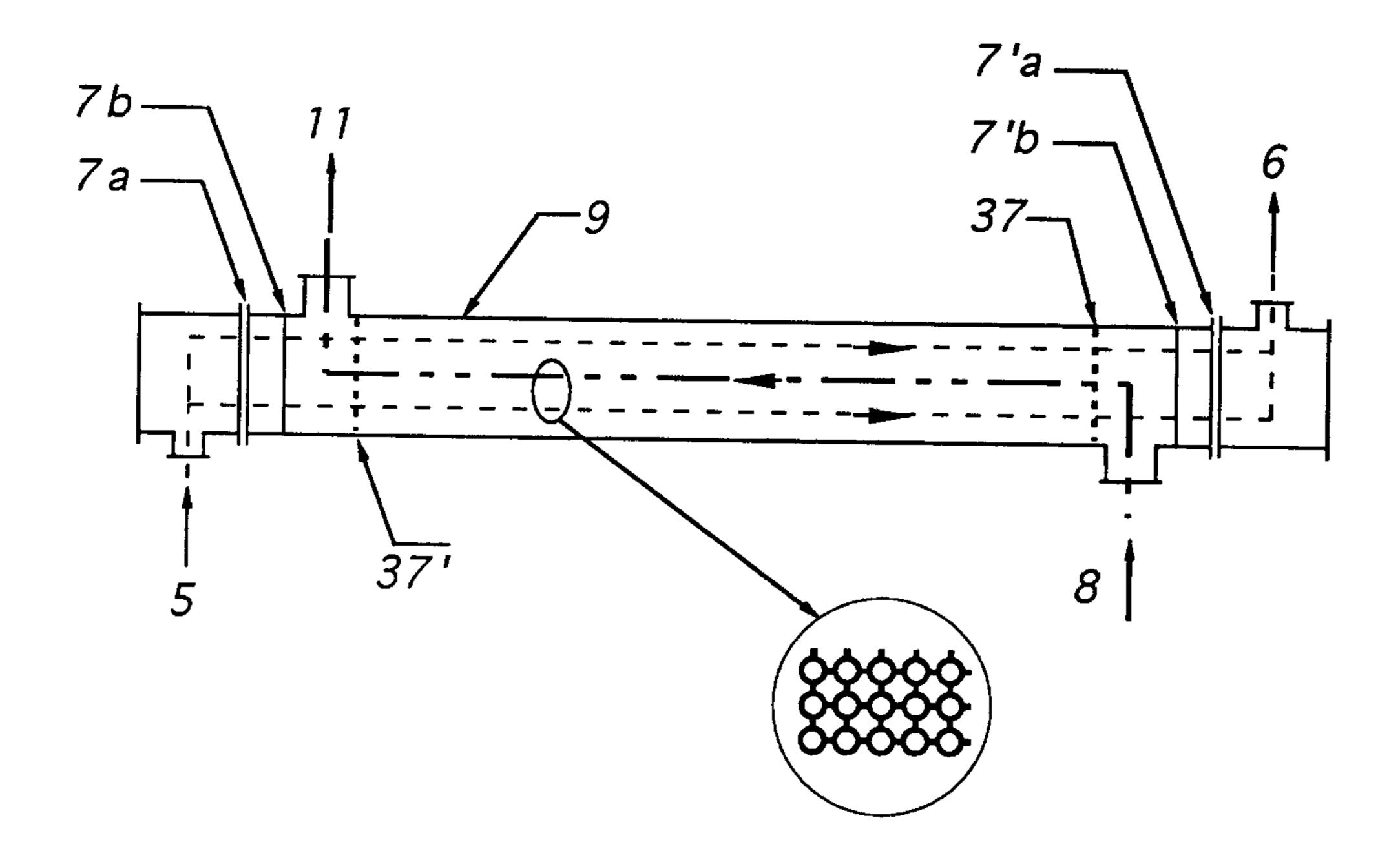


fig. 10

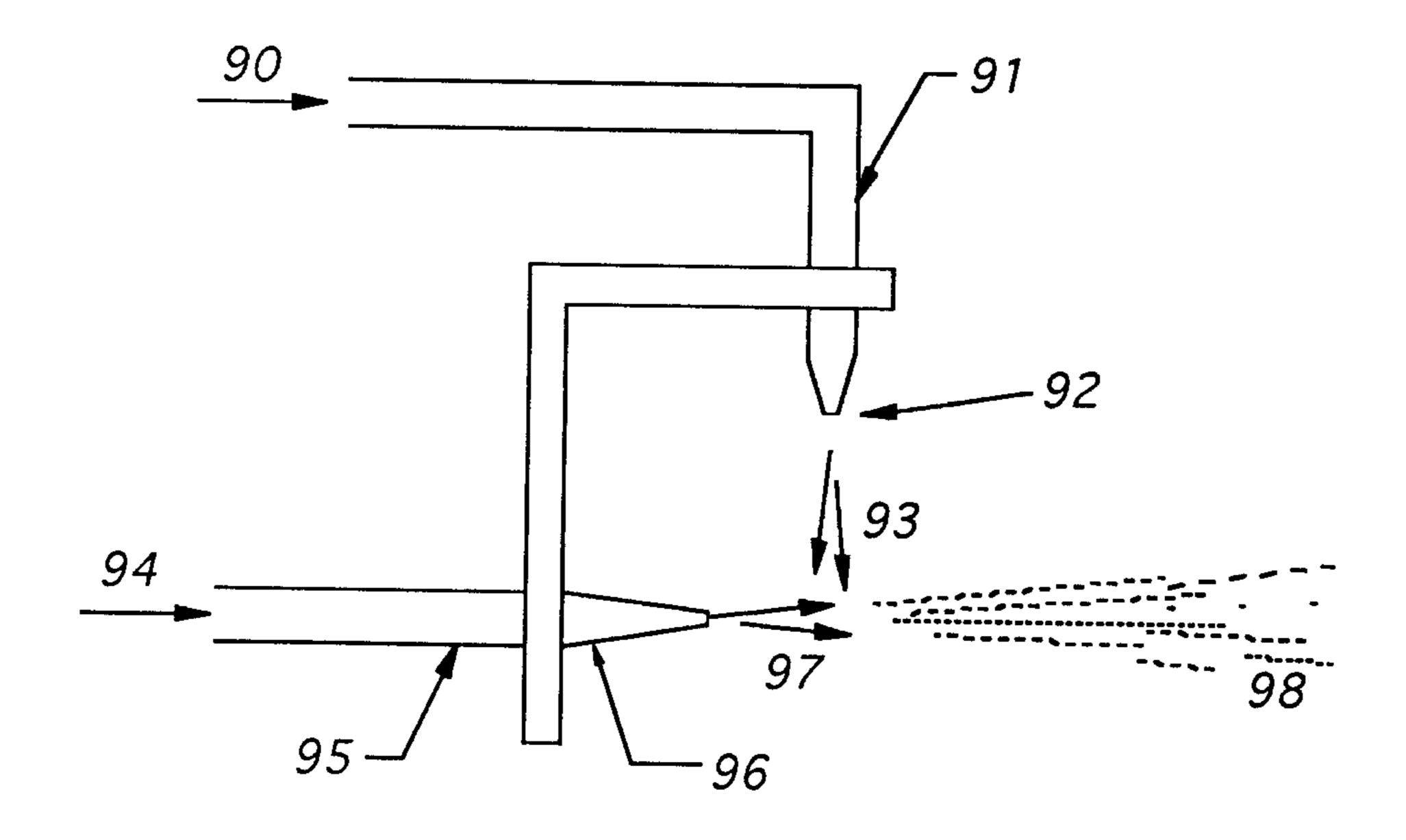


fig. 11

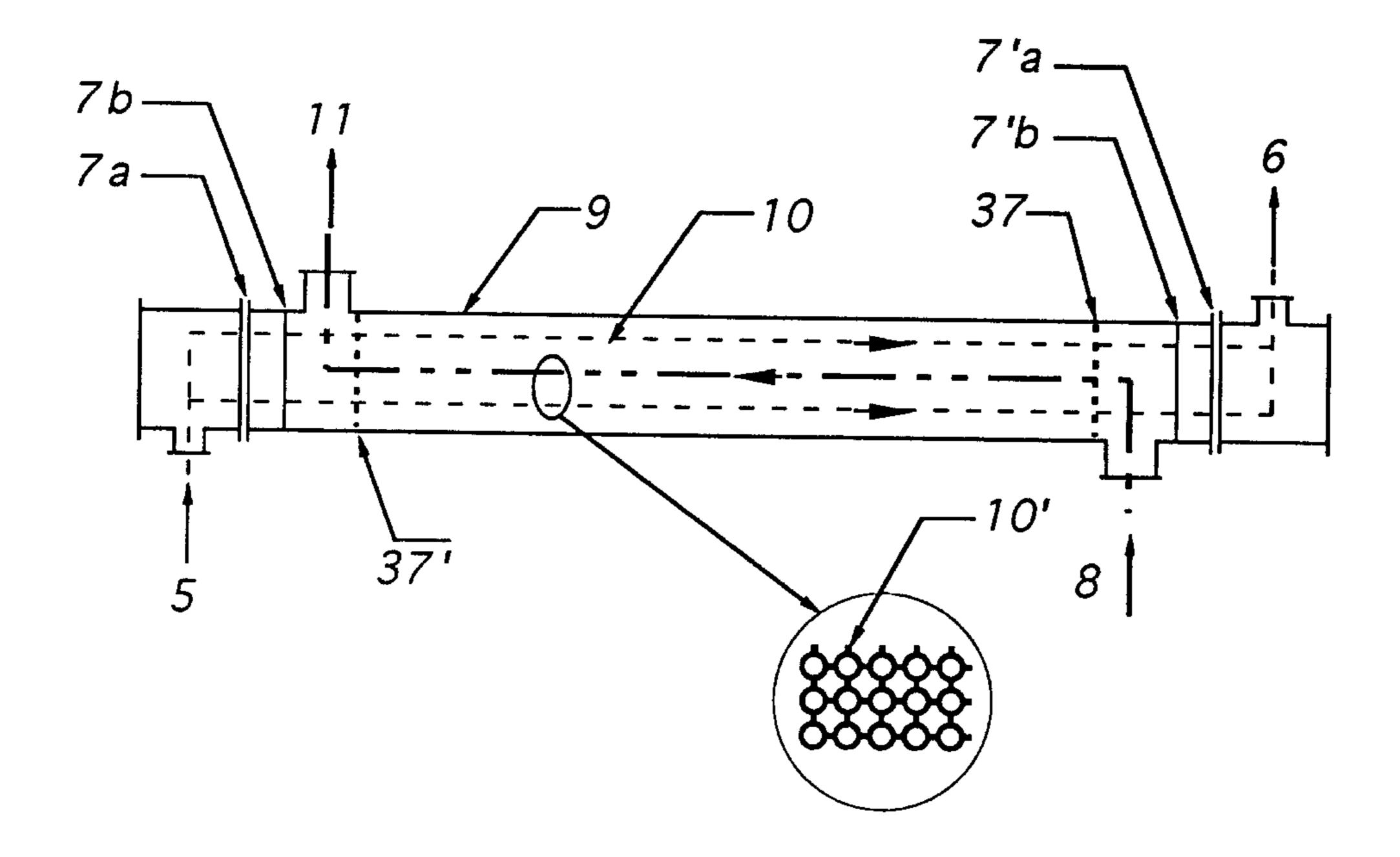


fig. 10

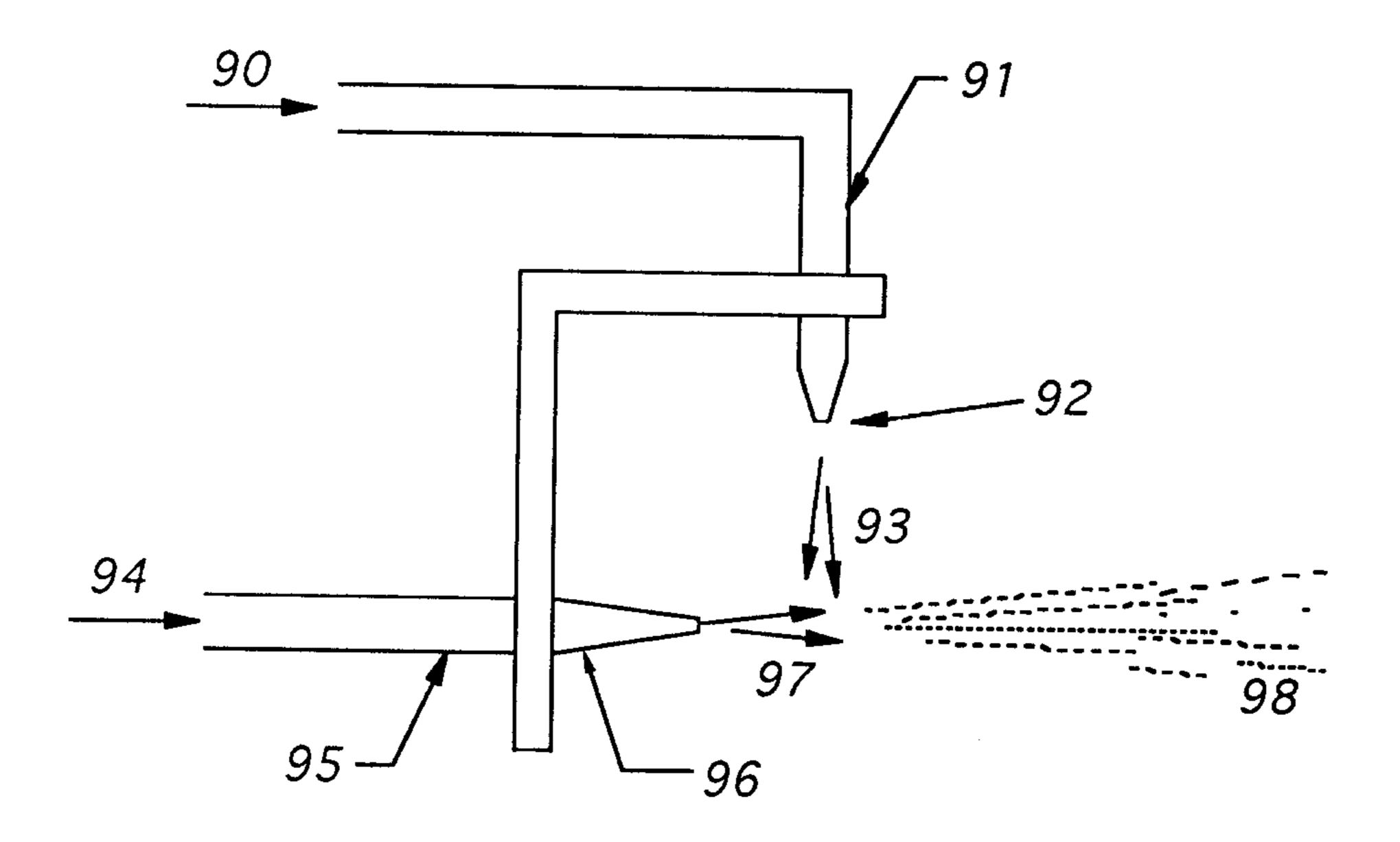


fig. 11



