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(54) **DESICCANT AIR CONDITIONER WITH THERMAL STORAGE**

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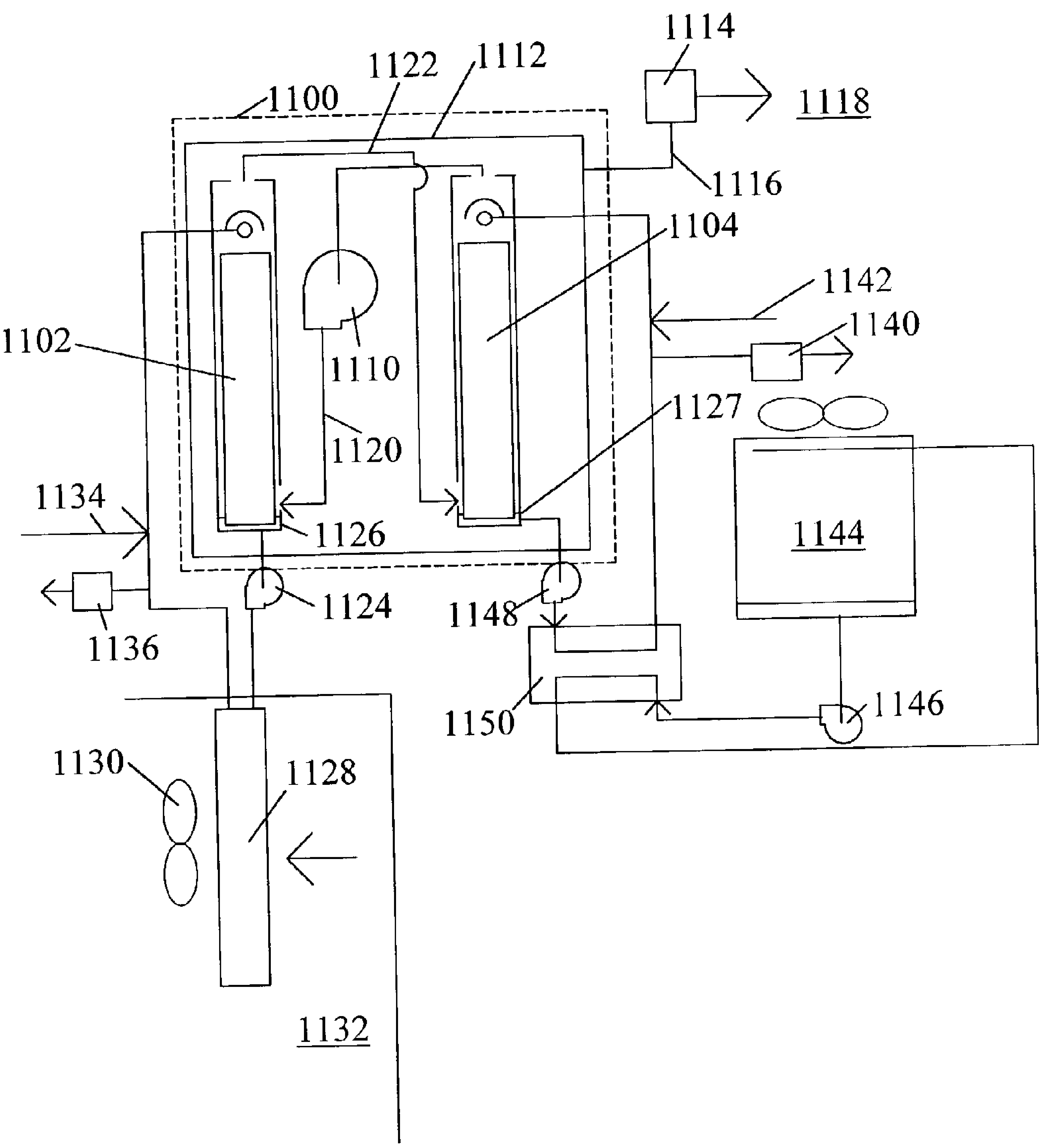
(52) **U.S. Cl.** **62/271**

(57) **ABSTRACT**

An air conditioning system with thermal storage that is suitable for using solar energy. A solar collector used energy from the sun to evaporate water from a desiccant fluid. The desiccant fluid is then flows into a mass transfer device, which removes moisture from an air stream. Calcium chloride is the preferred desiccant material and can serve as an energy-storage medium. Electric or fuel backup can be used with this system to regenerate the desiccant material. In some embodiments an indirect evaporative cooler is added to provide sensible cooling. A new desiccant cooling system that is specially designed to work with the properties of this desiccant and meet comfort requirements of conventional air conditioning.



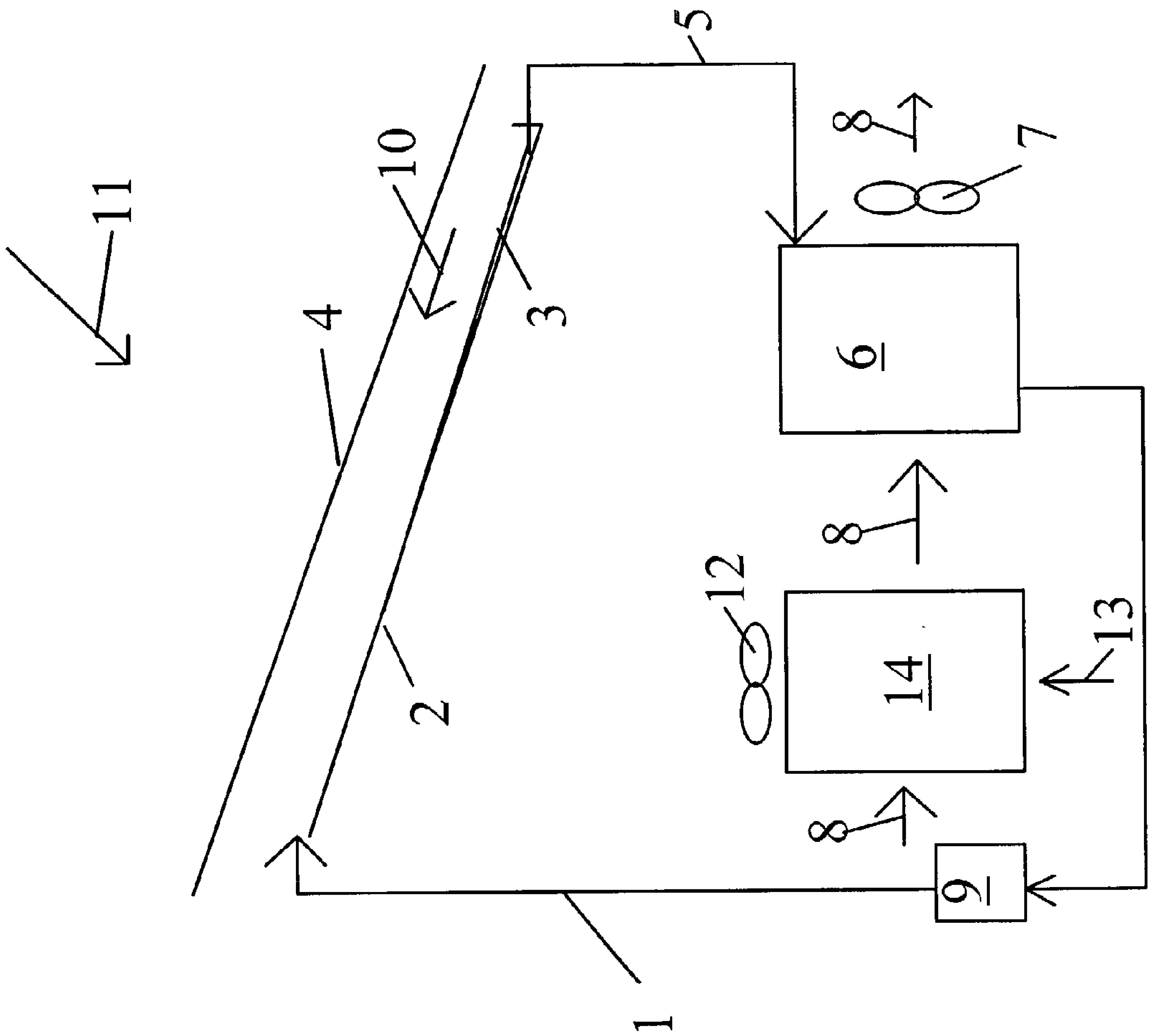


Figure 1

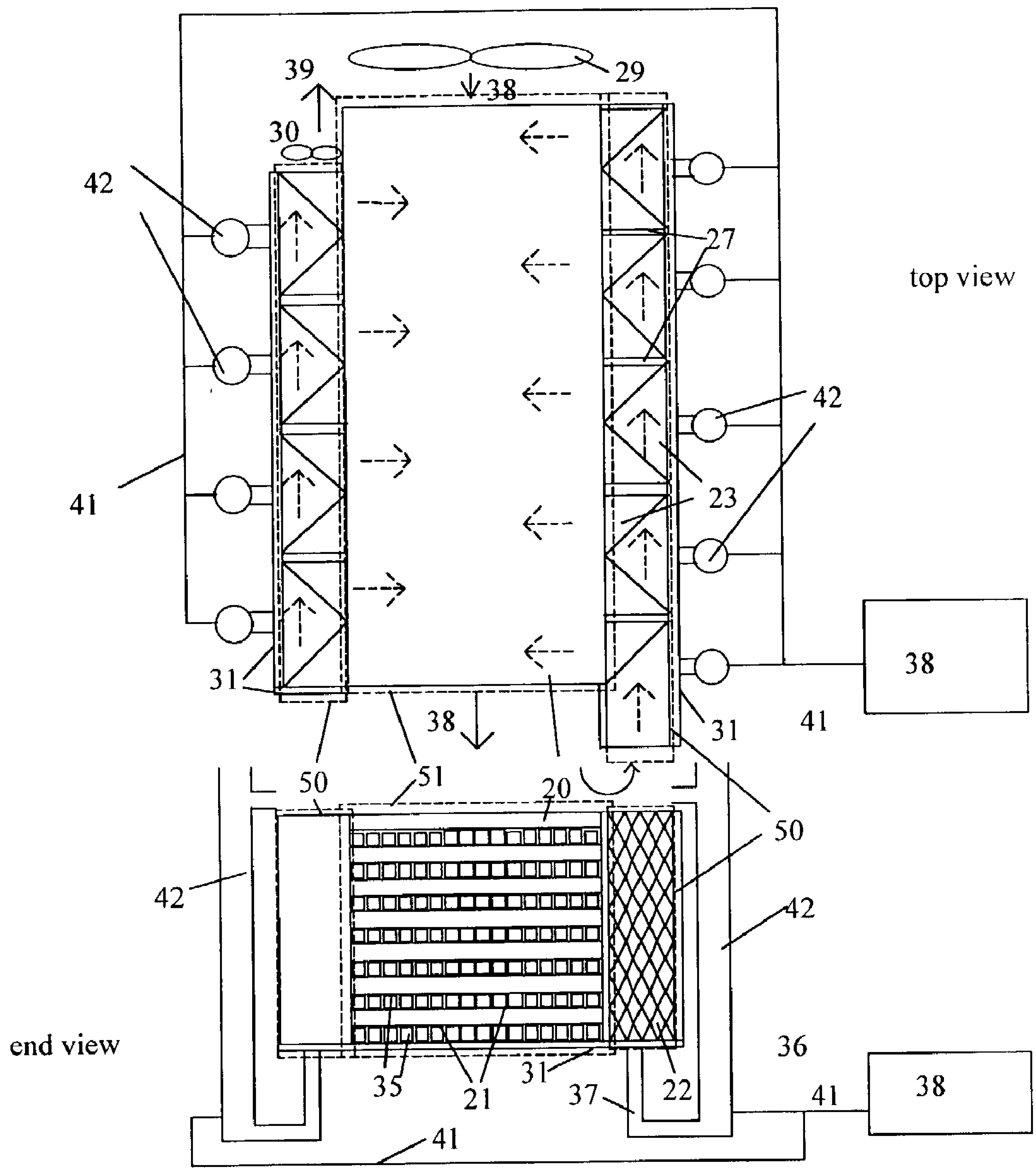

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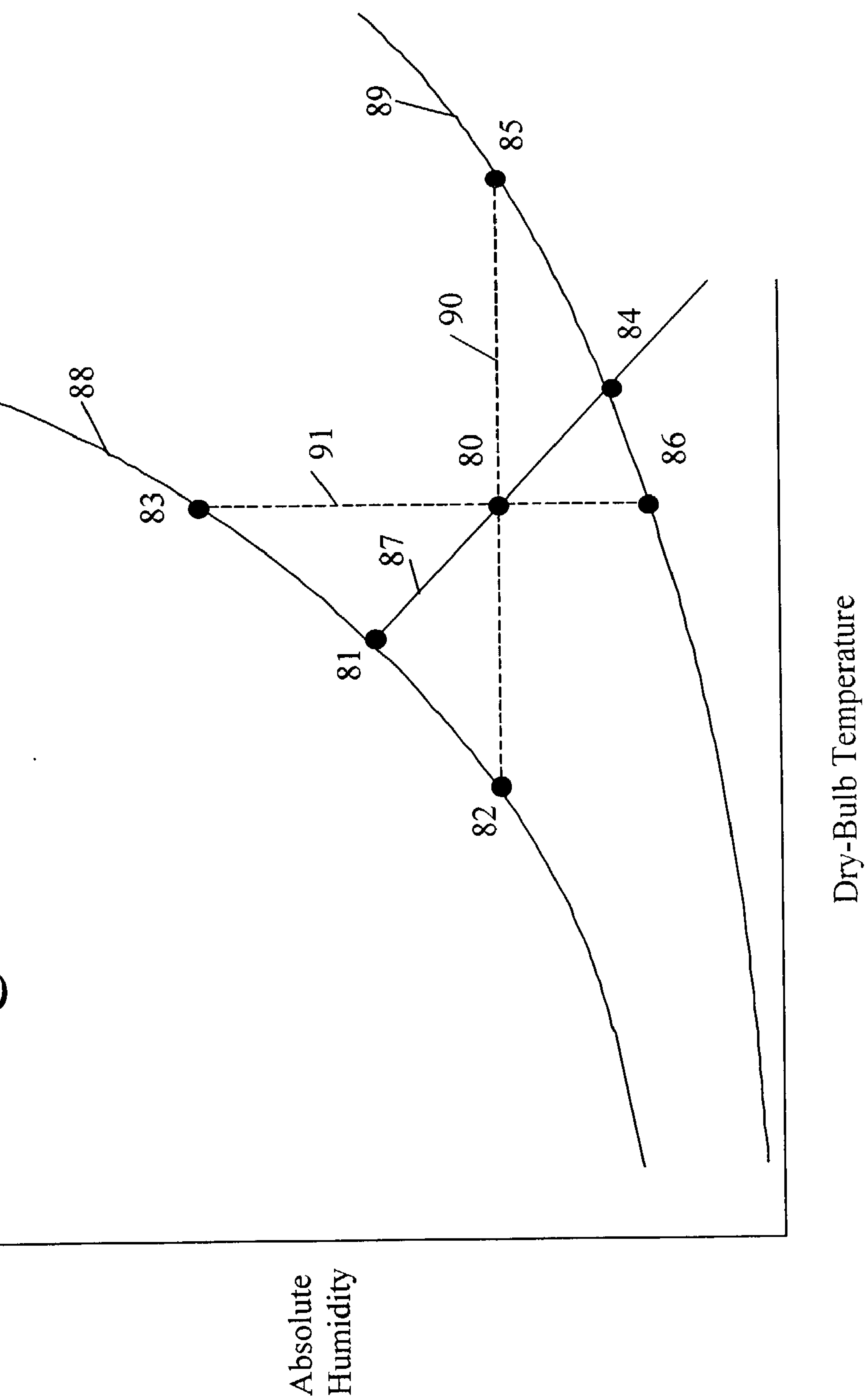
Figure 2

Figure 3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Absolute Humidity
Dry-Bulb Temperature

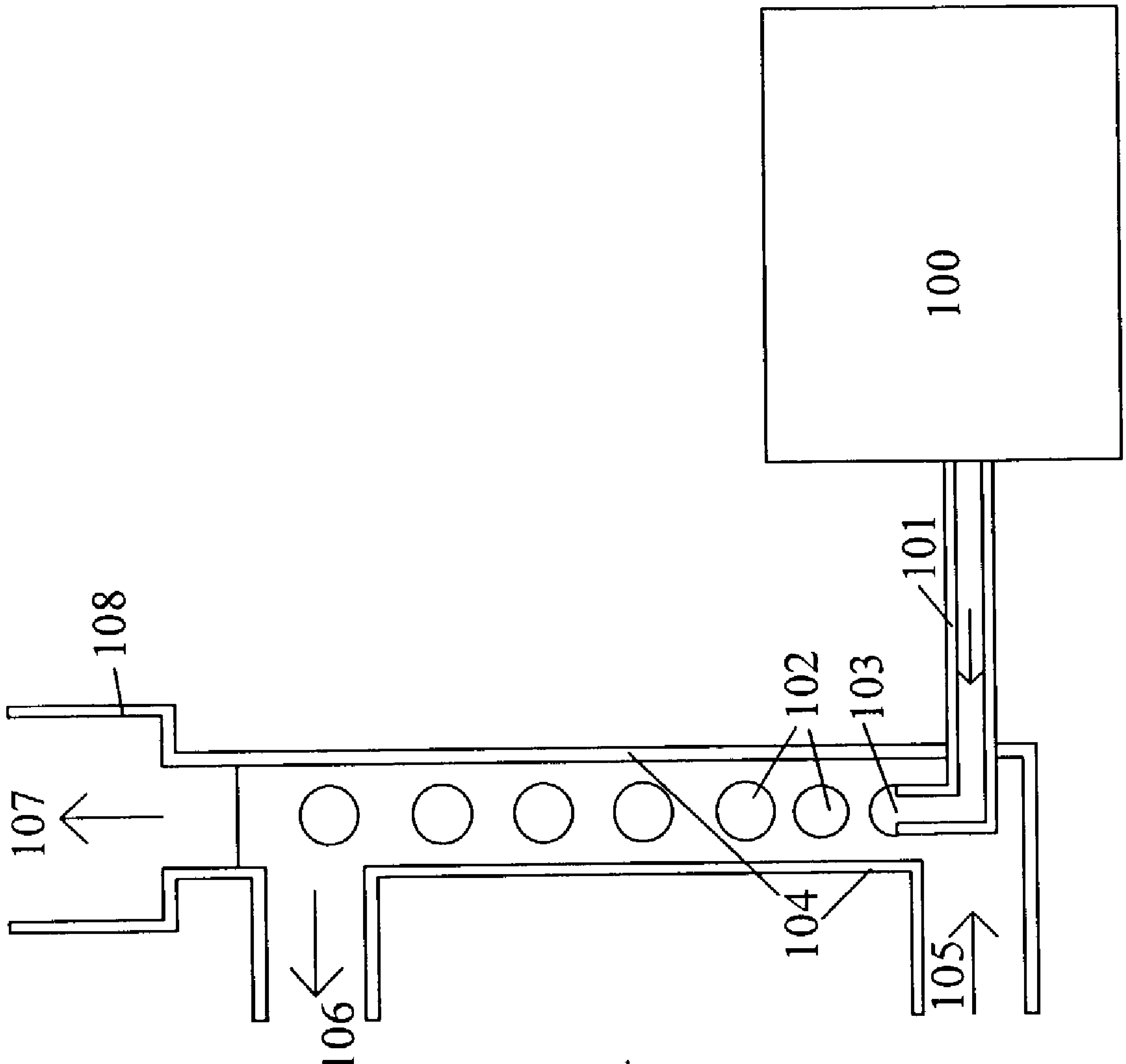

Figure 4

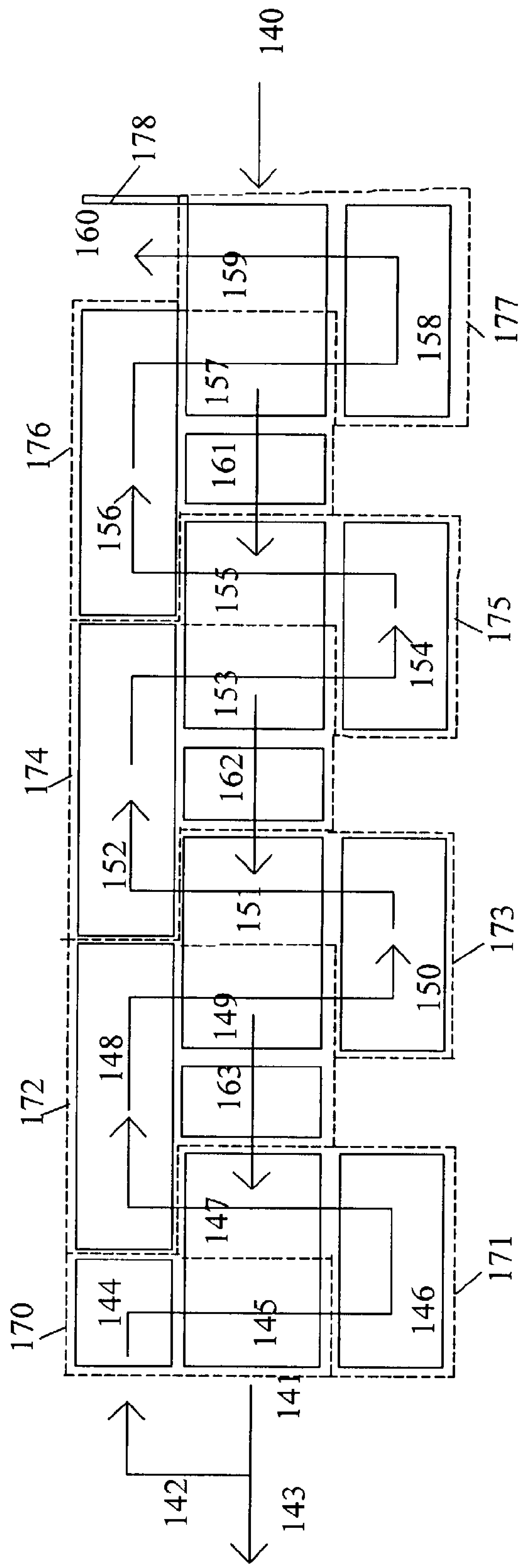
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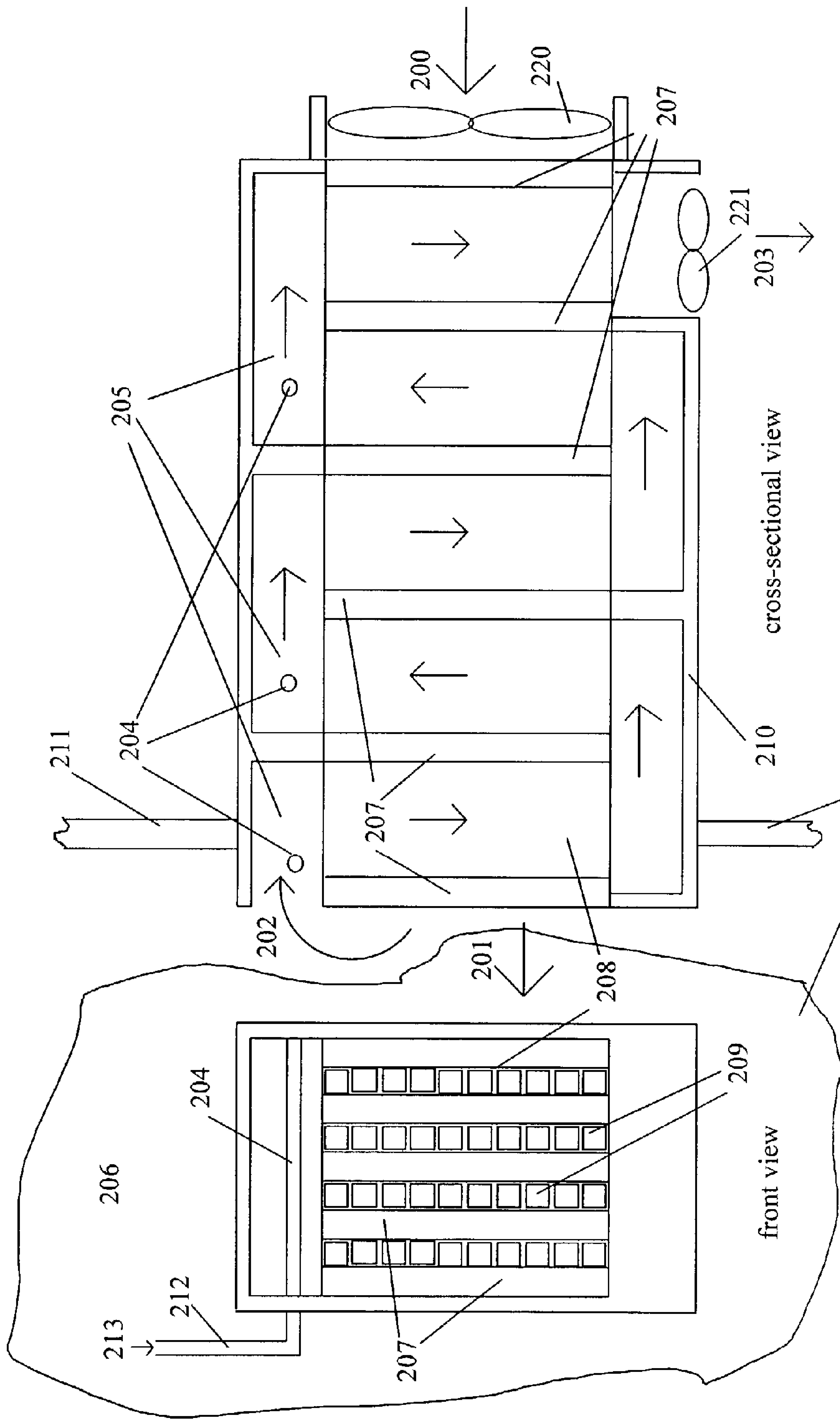
Figure 5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cross-sectional view
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front view
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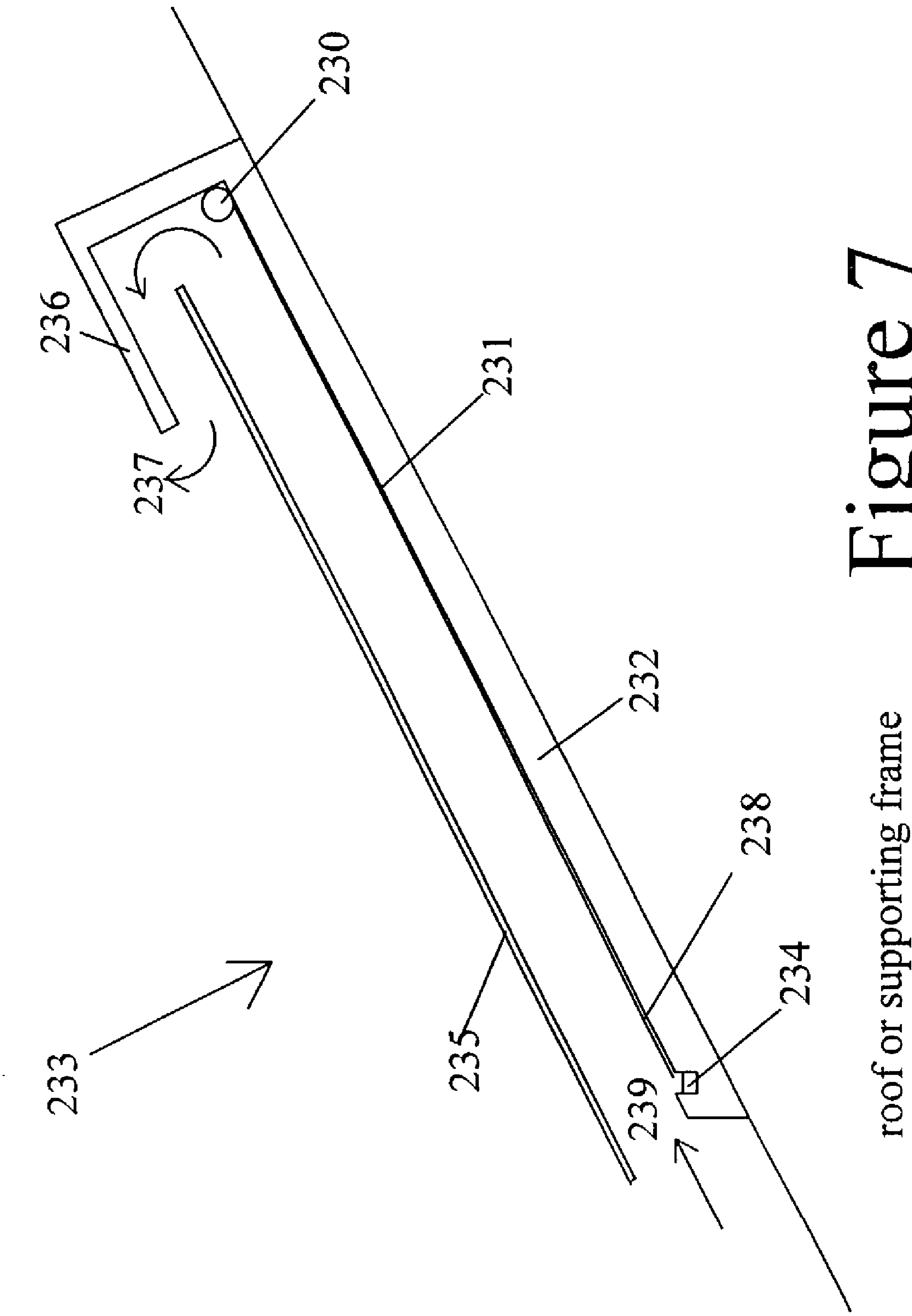
Figure 6


230
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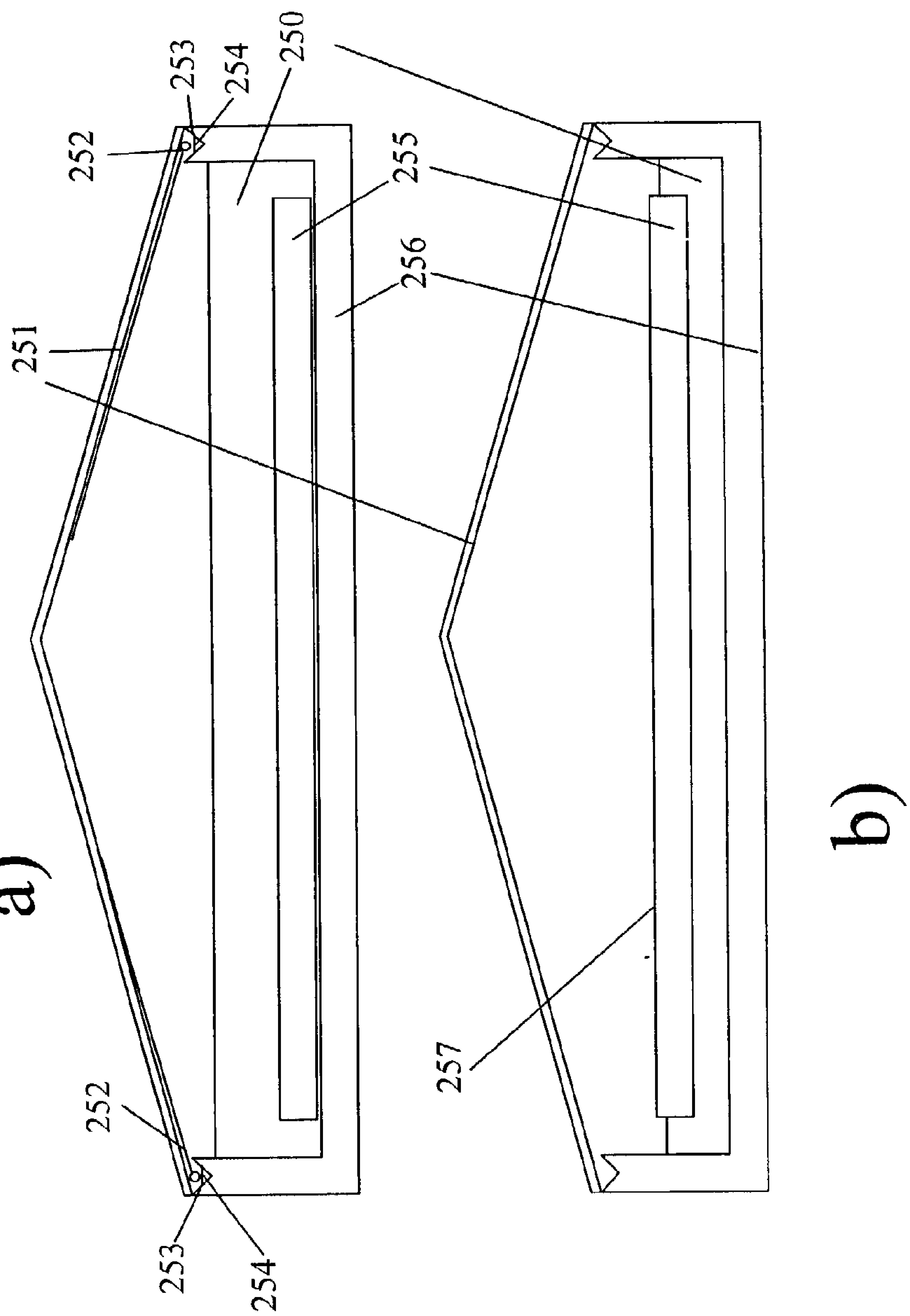
roof or supporting frame

Figure 7

Figure 8



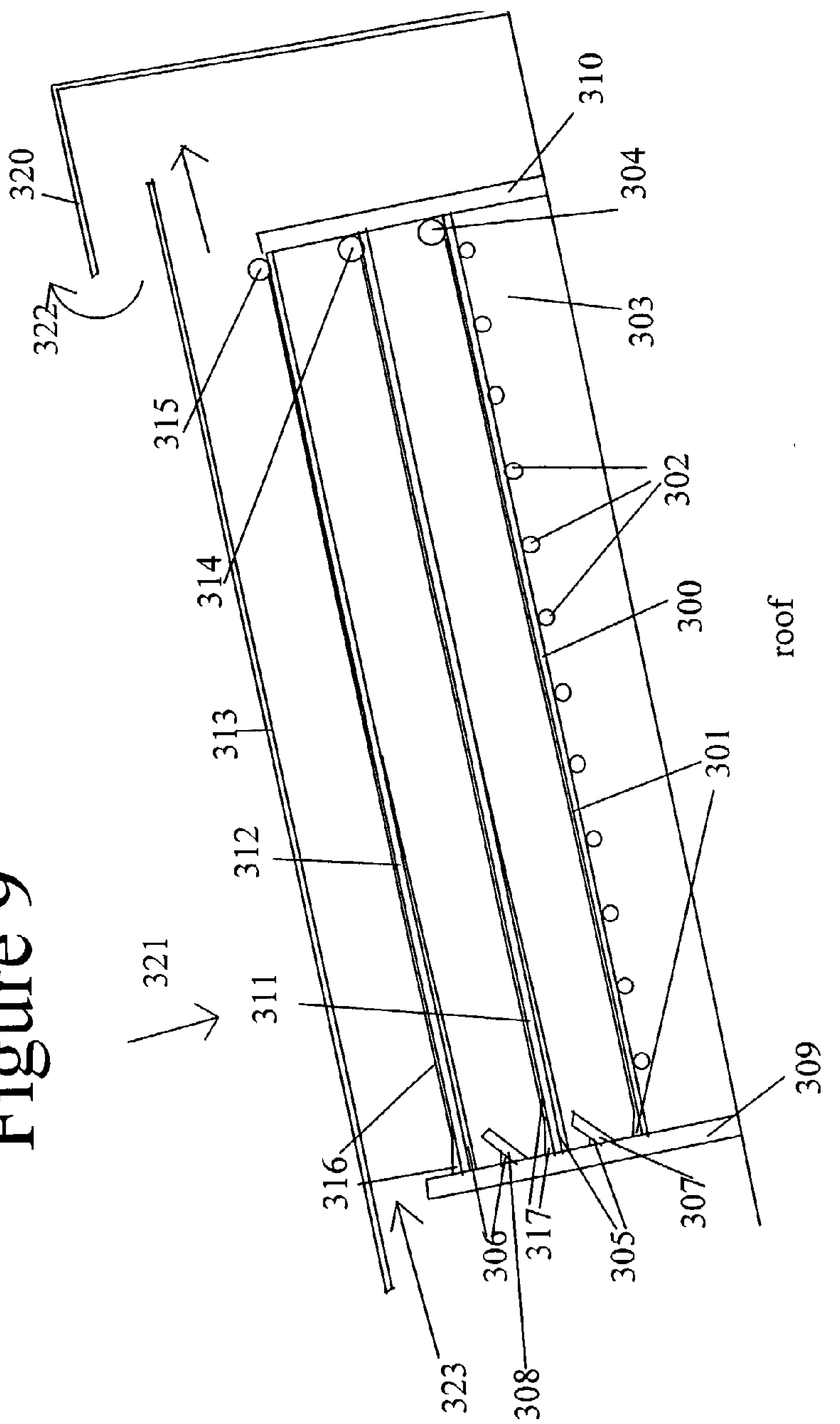

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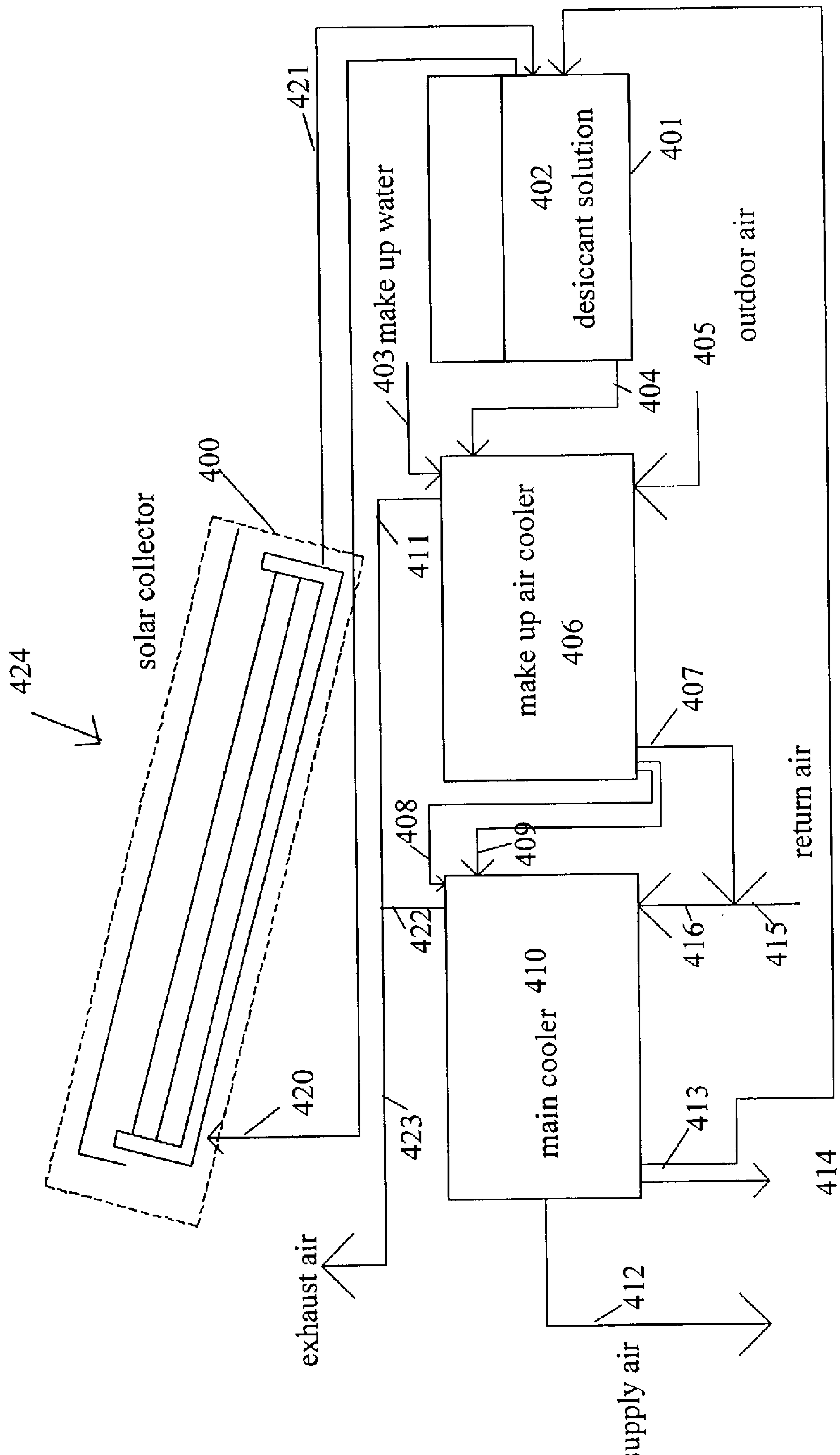
Figure 9

Figure 10



424
solar collector
400
420
421
403 make up water
402
desiccant solution
401
404
405
outdoor air
make up air cooler
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409
return air
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416
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main cooler 410
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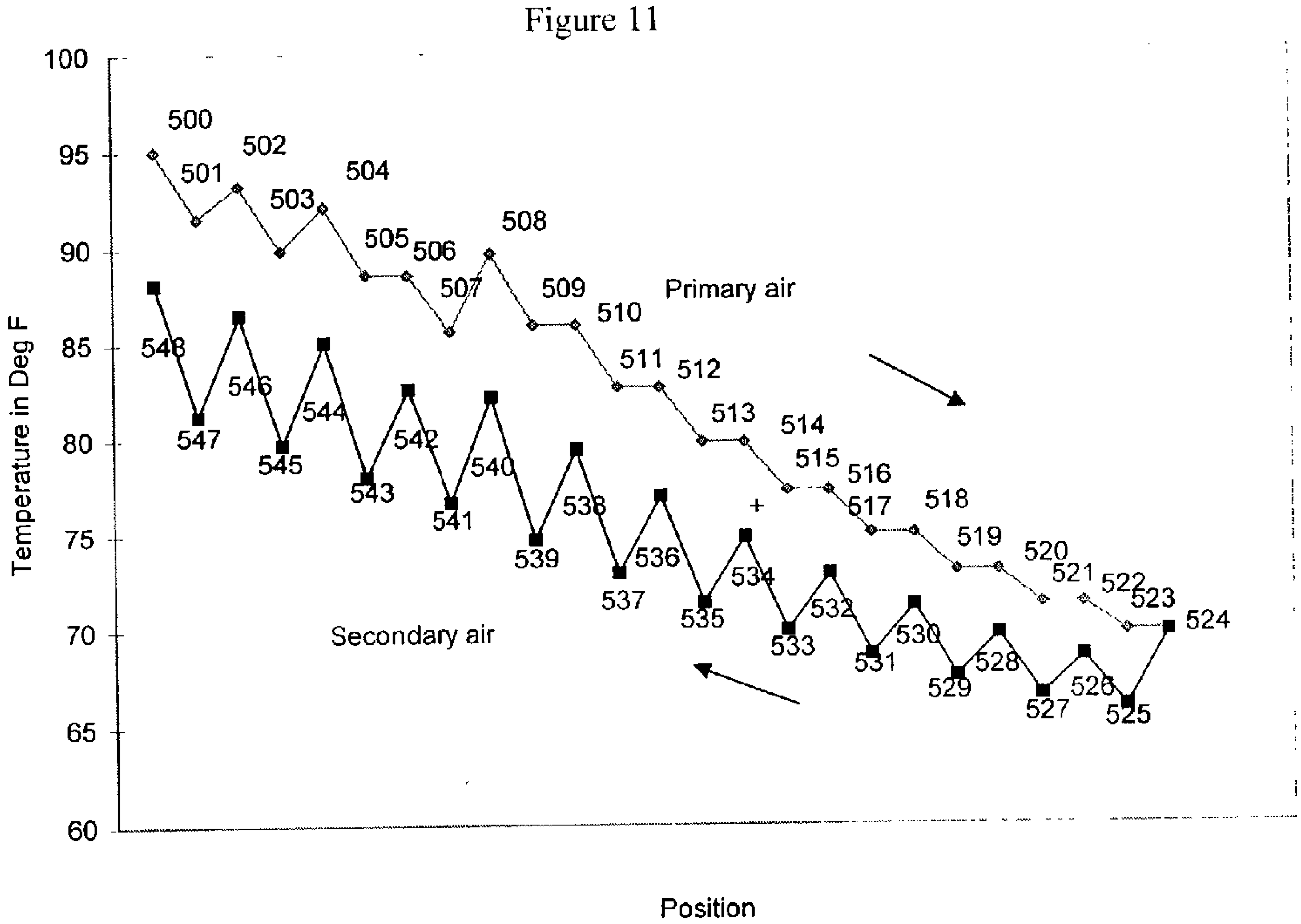
414
423
exhaust air
412
supply air

Figure 11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Temperature in Deg F
Position
Primary air
Secondary air
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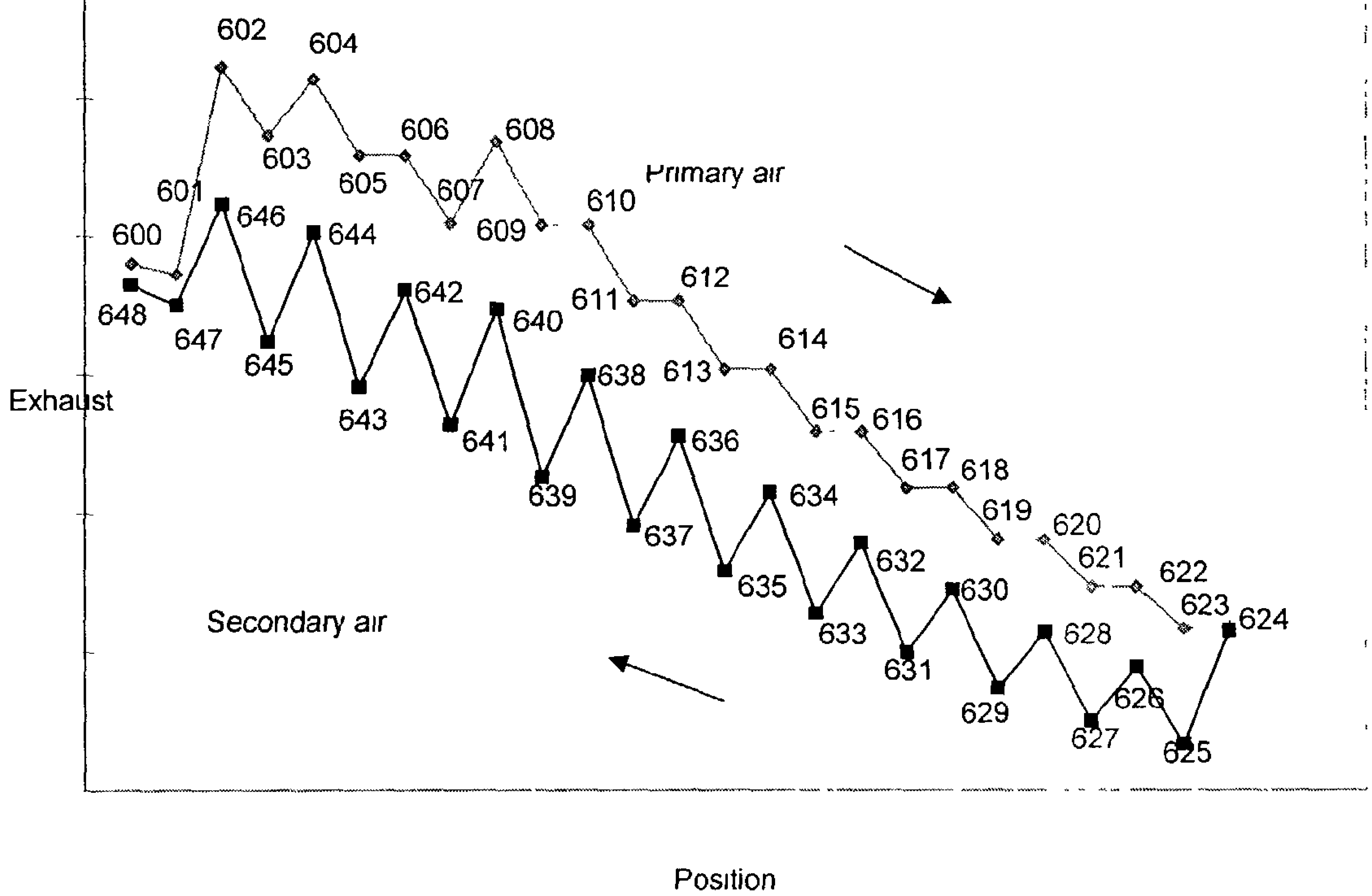
Figure 12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Exhaust
Secondary air
Position

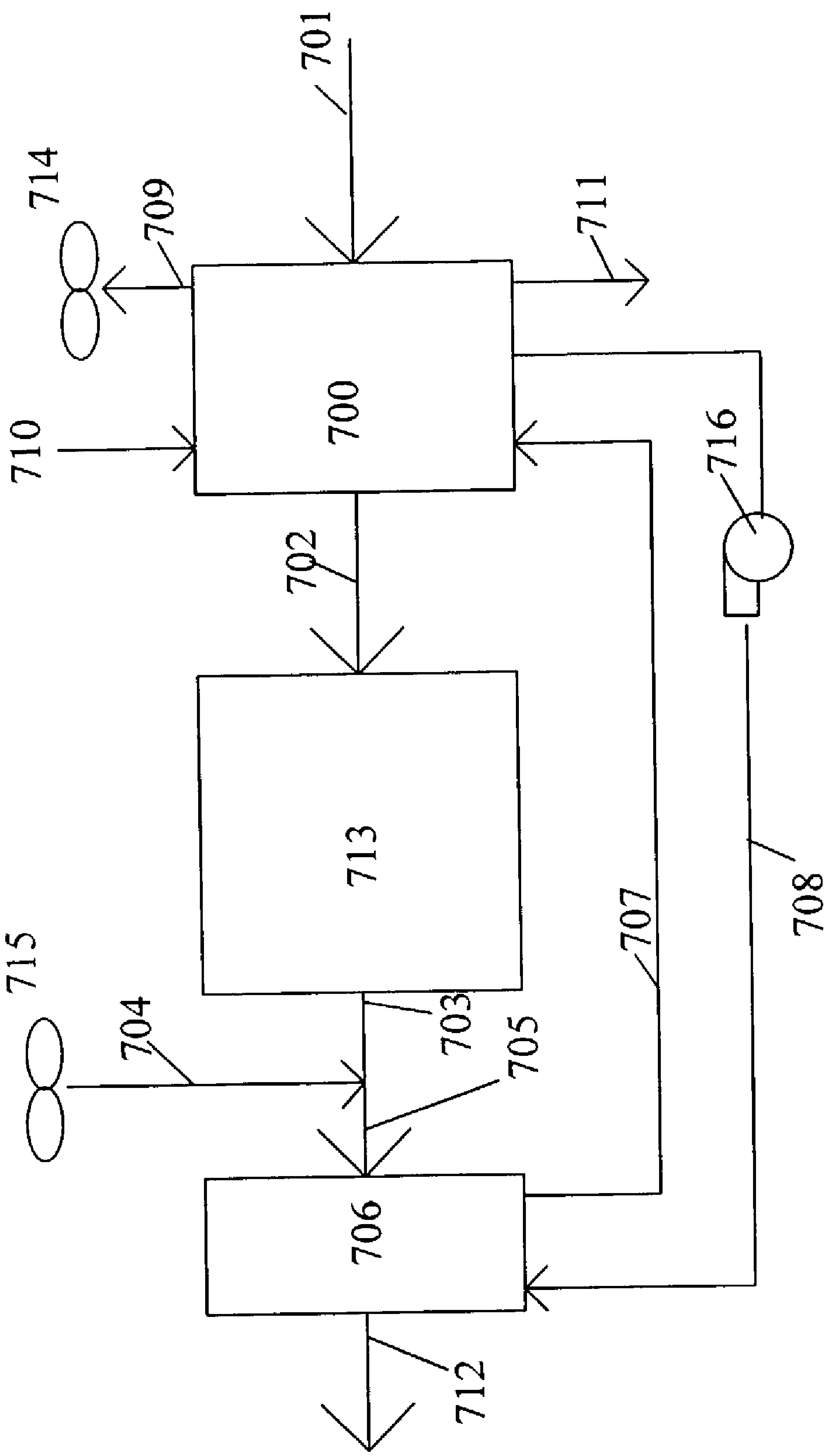
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Figure 13

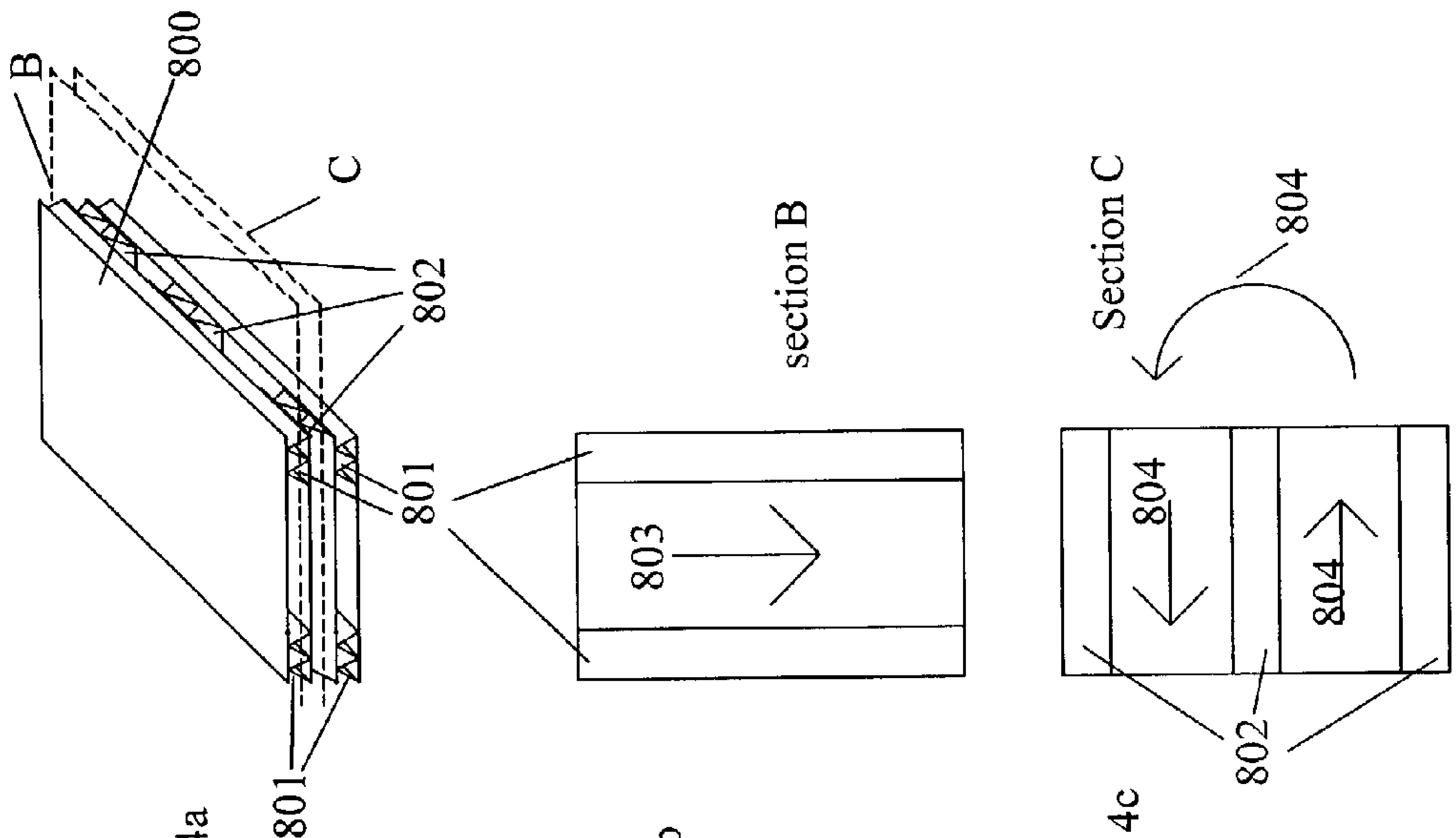

Figure 14a

Figure 14b

Figure 14c

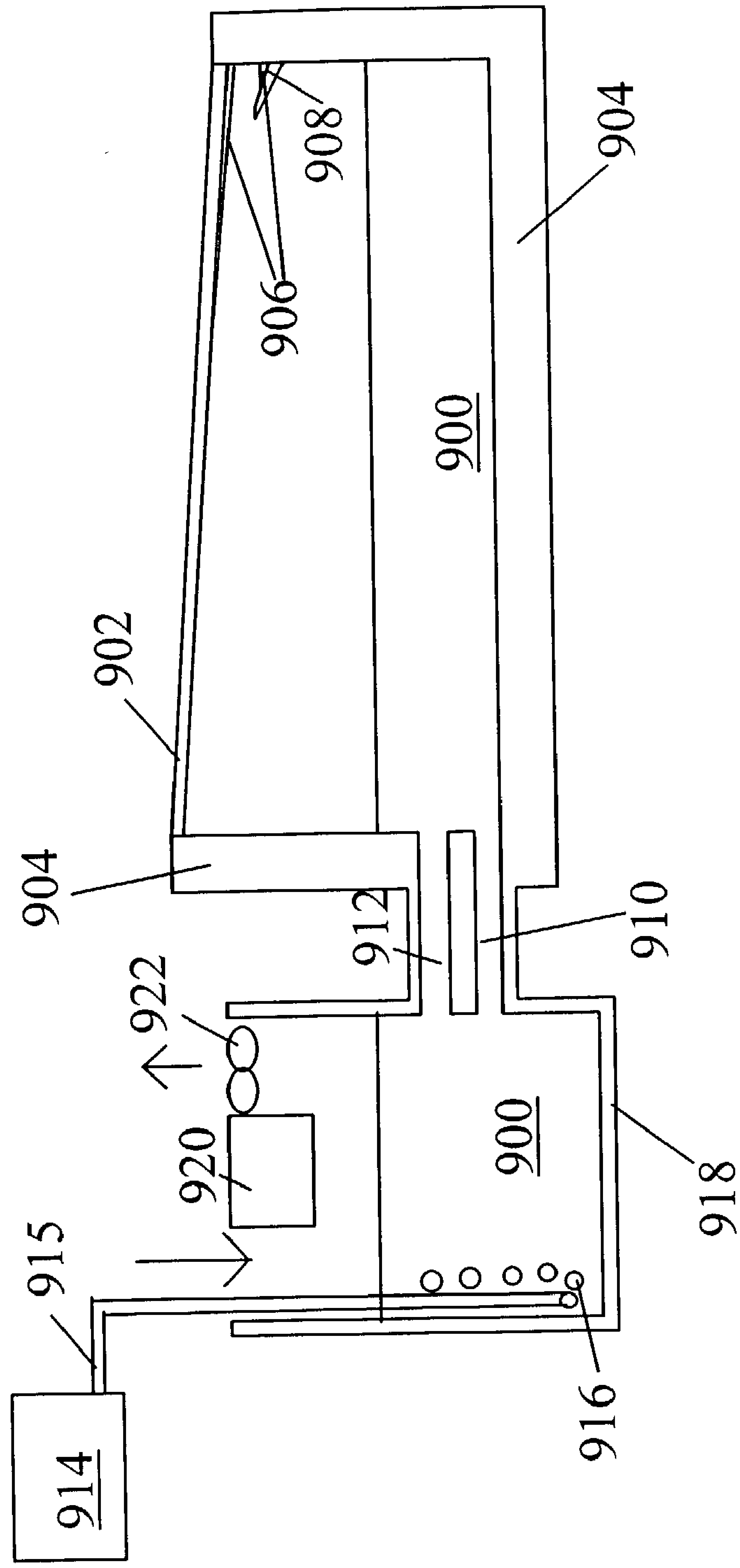

Figure 15

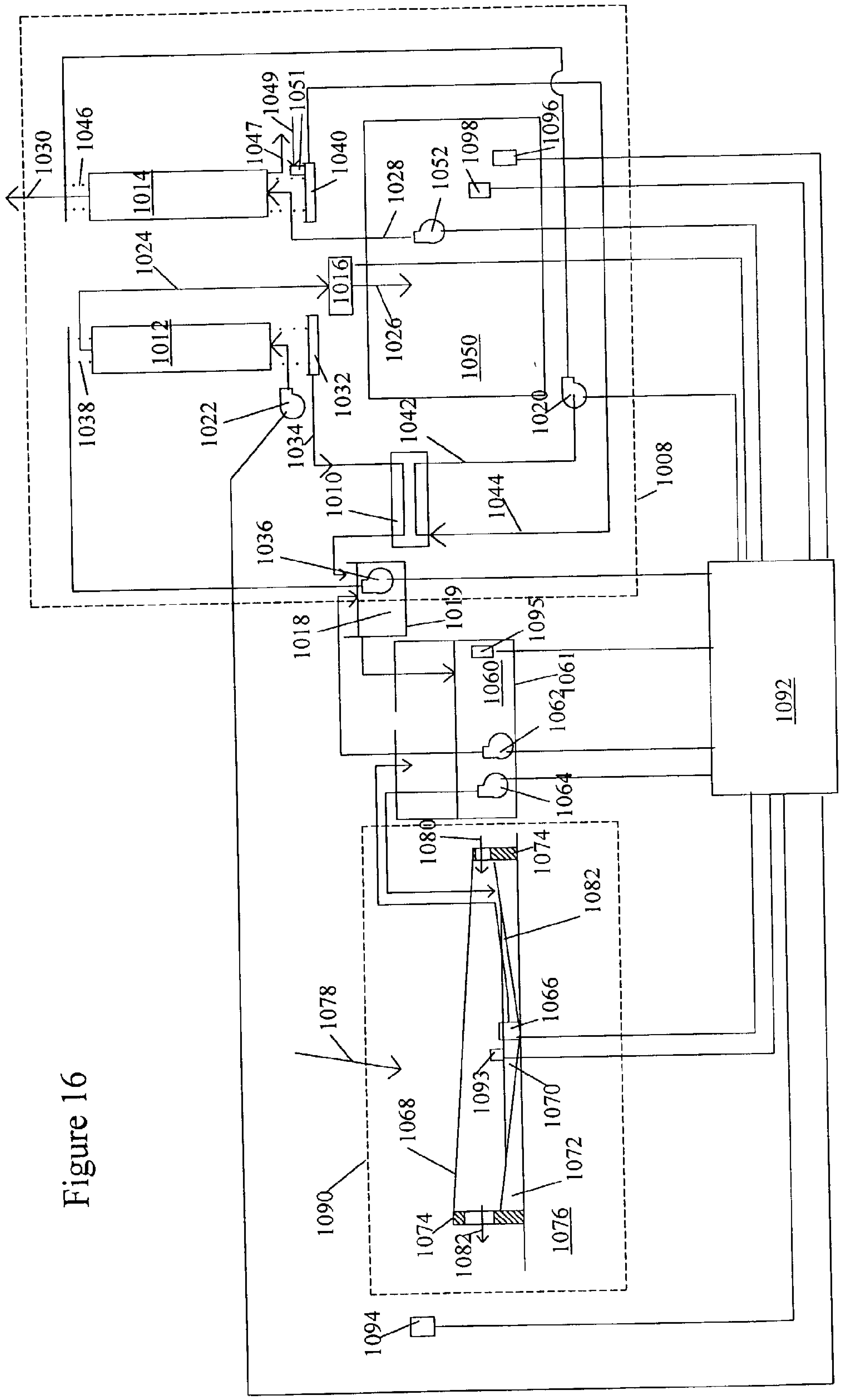
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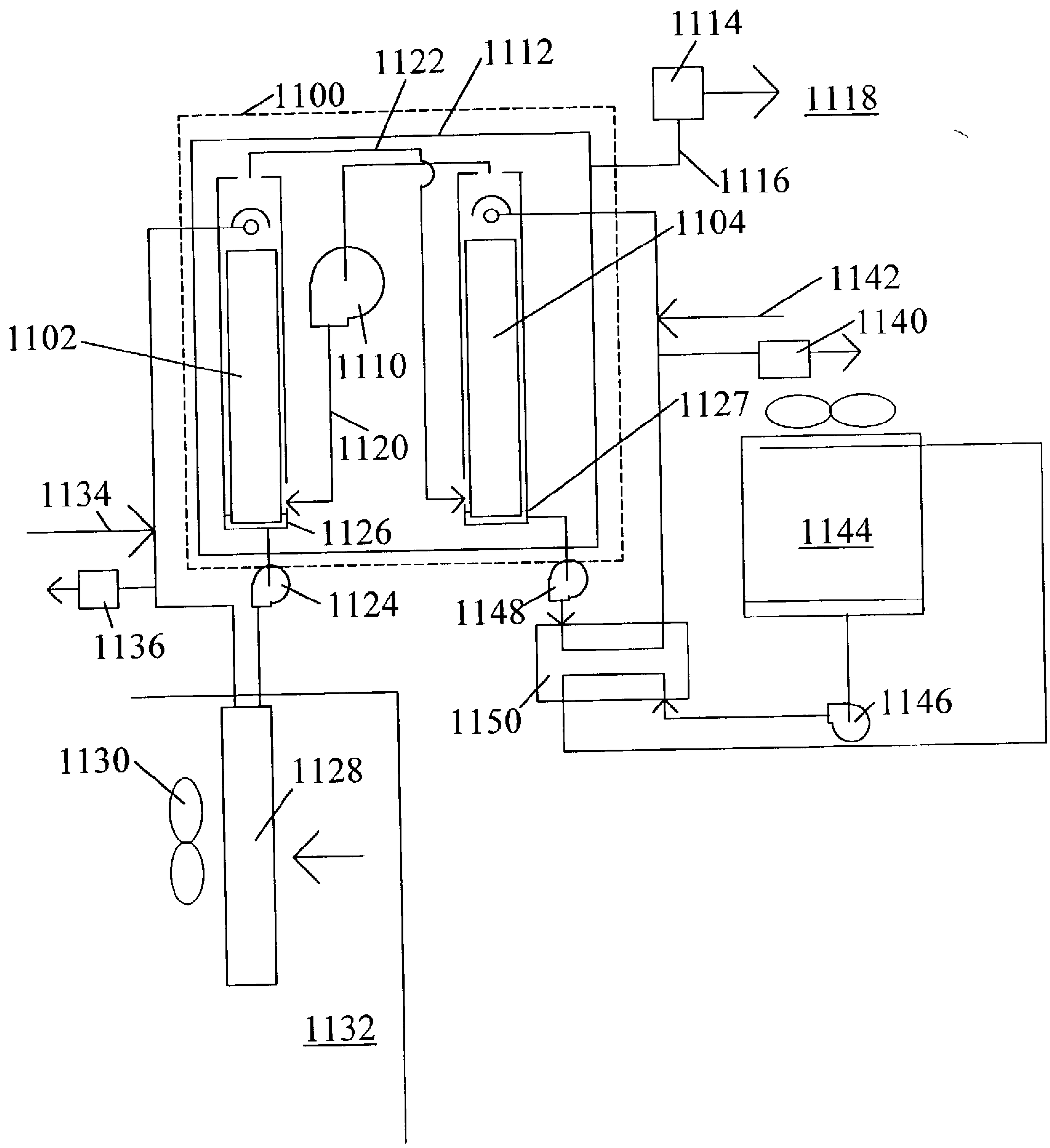
Figure 16

Figure 17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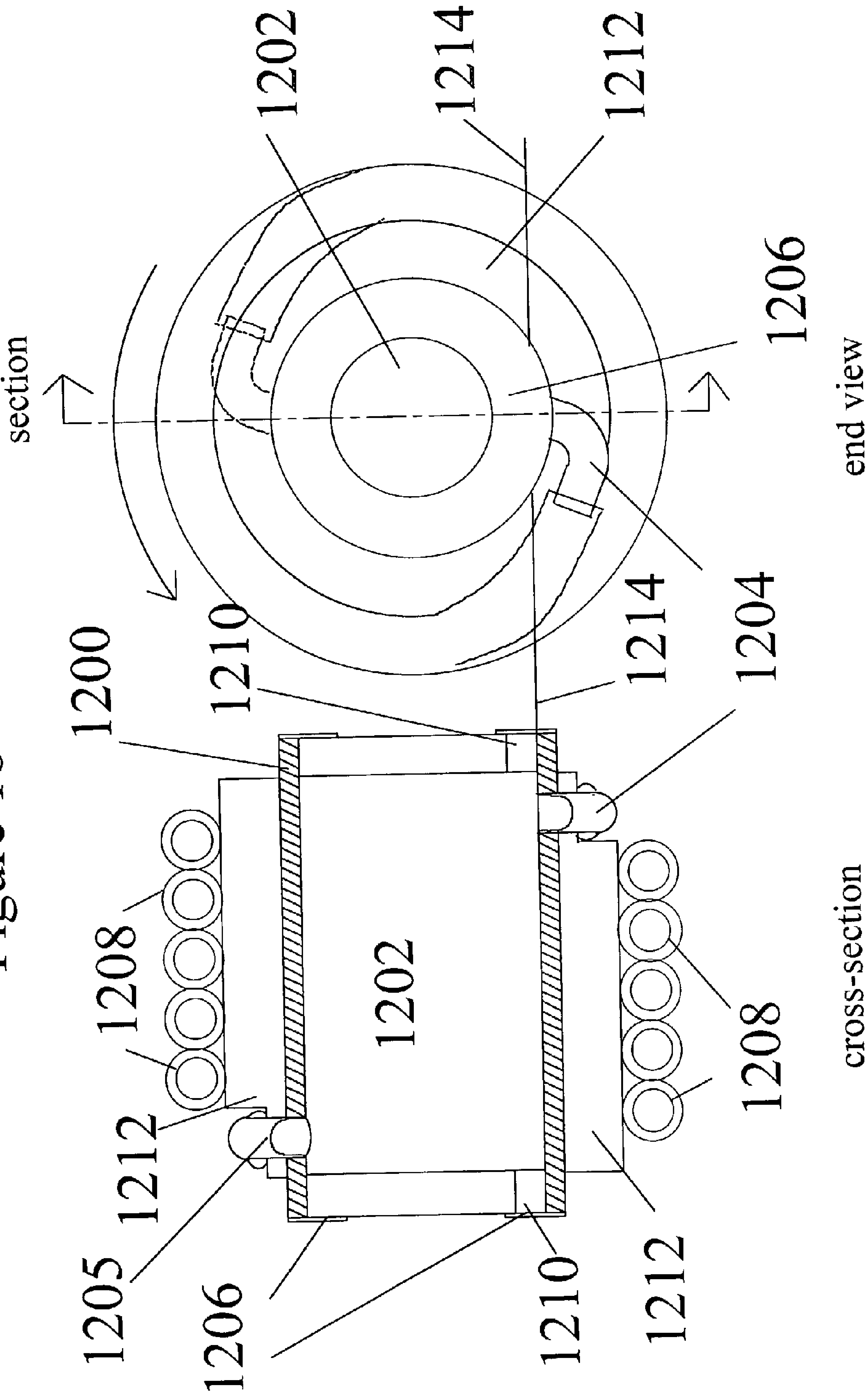
1202
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section
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end view
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1200
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cross-section
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Figure 18

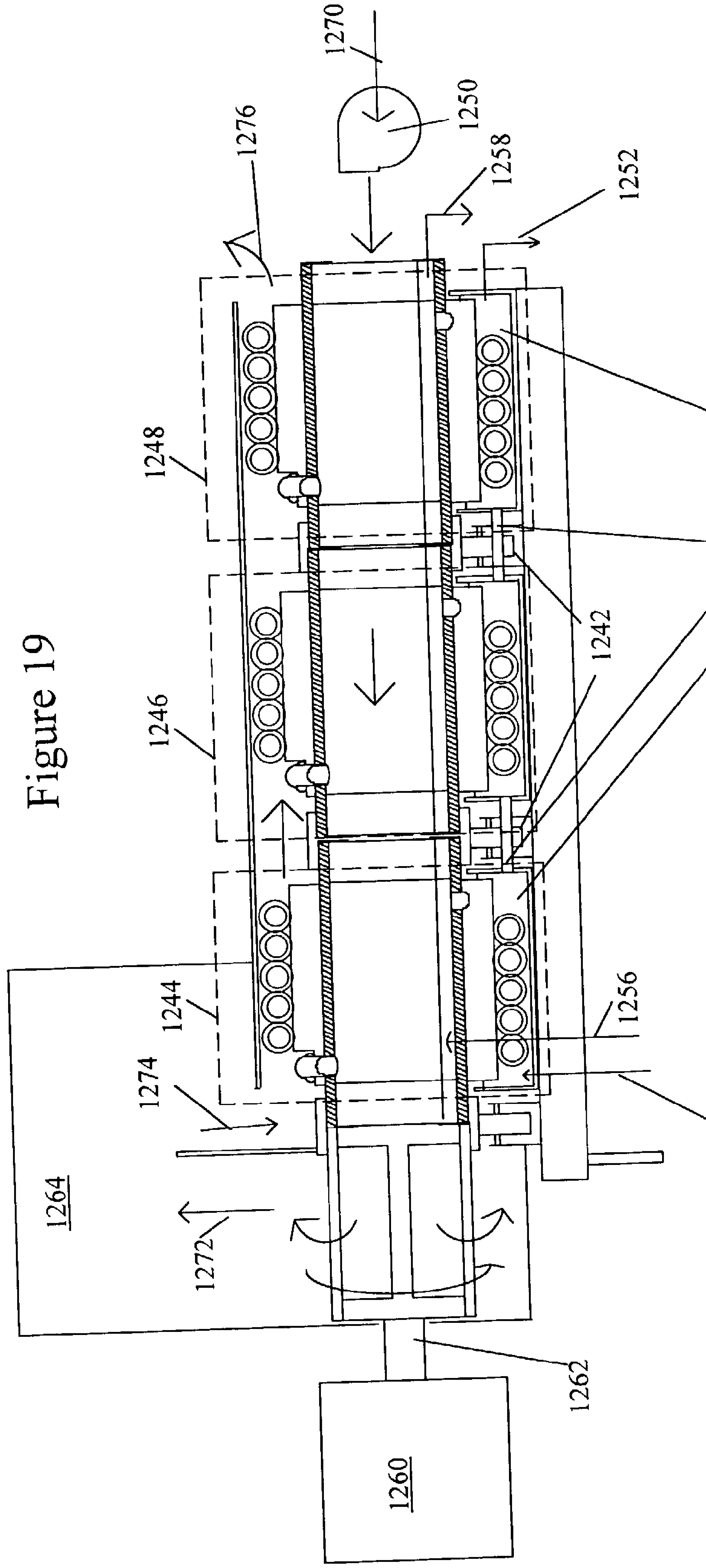
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Figure 19

DESICCANT AIR CONDITIONER WITH THERMAL STORAGE

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application is a continuation-in-part of application Ser. No. 09/549329 entitled "Solar Air Conditioner" filed on Apr. 14, 2000.

FIELD OF THE INVENTION

[0002] This invention is in the field of air conditioning, specifically thermally driven air-conditioners with thermal that are capable of accepting a thermal input from solar or off-peak electricity.

BACKGROUND OF THE INVENTION

[0003] Solar air conditioning has great potential to reduce energy use from air conditioning. Sunlight is most plentiful in the summer when air conditioning is required.

[0004] The problem is that existing solar technologies have not produced systems that are economically competitive with conventional electrically driven systems. Prior work with solar air conditioning has not produced practical systems. Solar air conditioning systems have used two basic approaches in an attempt to capture the sun's energy for cooling—thermal and photovoltaic.

[0005] The photovoltaic systems use photovoltaic panels to convert solar radiation directly into DC electricity. Photovoltaic systems have two major advantageous attributes: they can use conventional electrically driven air-conditioning equipment which is widely available and inexpensive with the addition of the solar panels that use an inverter to produce AC power, and they can use the utility grid for backup power during dark or cloudy periods.

[0006] Unfortunately there are other attributes: the high cost of manufacturing, the low conversion efficiencies, and the need for a continual stream of photons to produce power, create three major disadvantages. First electricity from solar cells is very expensive because of the high cost of the solar panels. (Panels for a residential air conditioner can cost well over $10,000.) Second the space needed for powering the air conditioning units is large. And third the panels provide no energy storage, which creates a need for use of grid based electricity at night and on cloudy day. In fact, the peak output from the solar panels occurs around solar noon, while peak air-conditioning load occurs several hours later, resulting in a significant mismatch between supply of needed power and demand. This mismatch greatly reduces the value of the system in reducing peak power demand to the utility, demands which recently deregulated markets is demonstrating are much more expensive to meet than had heretofore been obvious. For off-grid locations, the only viable energy storage system to match the provision of power to times when demand is high (later in afternoon and at night) is batteries. Batteries have a high first cost, require periodic replacement, and normally use toxic and/or corrosive materials. These problems have prevented the use of photovoltaic systems in other than a few high-cost demonstration systems.

[0007] Thermal systems use heat from the sun to drive an air conditioner. Typical approaches use a high-temperature flat-plate collector to supply heat to an absorption system. Systems with concentrating collectors and steam turbines have also been proposed. Natural gas or other fuel is used for backup heat. While thermal systems have the advantage of eliminating the need for expensive photovoltaic panels, they have attributes that produce major disadvantages.

[0008] One problem is the high cost and large size of the solar collectors. Flat-plate collectors running at about 190° F. (90° C.) require double-glazing and selective surface to achieve reasonable efficiency levels, which greatly increases the collector cost. This high collector cost reduces the comparative attractiveness of such systems to standard vapor compression systems driven by grid electricity. Large collector size also reduces the potential market size by eliminating many locations from possible use of the systems.

[0009] Furthermore, existing thermal technologies also suffer from the poor COP of absorption systems, typically around 0.5. When combined with a typical collector efficiency of 20 to 50%, this inefficiency, besides creating a need for large collector areas, makes the whole system much less economically and environmentally attractive.

[0010] Another important problem introduced by the performance attributes of current solar thermal air-conditioning concepts is the high-cost and large size of high-temperature thermal storage. Large thermal storage is required to reduce backup energy (typically gas) that would be used much of the time when their was a mismatch between demand for cooling and solar inputs. This mismatch is the discrepancy between high solar input at noon and large demands for cooling during late afternoon, at night, and on cloudy days. A related problem with existing concepts for thermally driven solar cooling is the need for significant power input for circulating pumps and fans, which further reduces the possible energy savings.

[0011] Together these attributes for current concepts for thermally driven solar cooling imply that the large majority of their energy input would come from the backup fuel and electrical input for fans and pumps. In essence, these various problems mean that these solar systems are effectively very expensive gas-driven systems.

[0012] No commercially available or conceptually proposed system has been demonstrated that has the attributes that would be needed for commercially viable solar air conditioner. Commercial success will require the system to have the following attributes: low first cost (The market tends to be first cost driven so it is critical that the cost and thus ultimate selling price not be too high.); small collector area (critical to cost and to finding many locations in which installation is practical); small storage size (The mismatch between solar supply and cooling demand requires storage if the system is not to become a glorified means for using fossil energy and if it is to be practical to install in many locations—as well as low cost to manufacture.); easy to incorporate backup capability (Regardless of storage capacity, the ability of the system to meet demand in extreme and unusual circumstances will be critical to market acceptance; customers demand perfection and then some.).

[0013] Evaporative coolers are a related technology with a long history. Direct evaporative coolers are the simplest and most common. They consist of a means for moving air over a wet pad. Water evaporates from the pad and thereby cools and humidifies the air. They are commonly used for comfort cooling in warm, dry climates such as those found in the southwest U.S.

[0014] Indirect evaporative coolers are more sophisticated. An indirect evaporative cooler means that air is cooled by contact with a dry surface that is in turn cooled evaporatively.

[0015] Desiccant systems dry air for air-conditioning purposes. A typical system uses a solid desiccant impregnated on a wheel of corrugated metal or plastic.

[0016] Some more obscure systems appear in the patent literature, but each has its own problems. Patents RE 20,469; 4,660,390; 4,854,129 describe regenerative indirect evaporative coolers that use a portion of the air exiting the dry cooler as inlet air to the wet side. RE 20,469 describes a cumbersome arrangement of coils and cooling towers this complicated and expensive. 4,660,390 describes another system that uses tubes in a crossflow configuration to transfer heat between a wet side and a dry side. 4,854,129 also uses a system that uses a cooling coil with water from a cooling tower.

[0017] Patent 5,050,391describes another option for the desiccant system. This system uses solid desiccant material and a true counterflow arrangement for the heat exchangers. It also has essentially a single stage of cooling which limits it performance and its ability to use inexpensive desiccant materials.

DESCRIPTION OF THE FIGURES

[0018] **FIG. 1** shows a basic embodiment of the invention.

[0019] **FIG. 2** shows a design of an indirect evaporative cooler used in this invention.

[0020] **FIG. 3** is a schematic psychrometric chart that shows how this cooler works.

[0021] **FIG. 4** is drawing of a simple air-lift pump that is preferred for use in the invention.

[0022] **FIG. 5** shows a diagram of a combination evaporative-desiccant cooler that is a component of this system.

[0023] **FIG. 6** shows another cooler configuration that uses a water mist for evaporative cooling.

[0024] **FIGS. 7, 8**, and **9** shows solar collectors that may be used in the invention.

[0025] **FIG. 10** shows another embodiment of the invention.

[0026] **FIGS. 11 and 12** are plots of temperatures through the coolers used in the invention.

[0027] **FIG. 13** is an embodiment that can use exhaust heat from a gas turbine for cooling inlet air to the turbine.

[0028] **FIGS. 14***a*, **14***b*, and **14***c* show details of heat exchanger design that may be used in the invention.

[0029] **FIG. 15** shows another preferred embodiment that is suitable for use as a dehumidifier.

[0030] **FIG. 16** is a preferred embodiment of the invention that uses a counterflow liquid-to-liquid heat exchanger and counterflow liquid-to-air heat exchangers.

[0031] **FIG. 17** is a preferred embodiment of the invention that is suitable for producing cooled water.

[0032] **FIG. 18** is an alternate embodiment showing a single-stage cooler that rotates.

[0033] **FIG. 19** is a multistage embodiment that is a assembled from the stages as shown in **FIG. 18**.

SUMMARY OF THE INVENTION

[0034] The present invention is a liquid desiccant cooling system with thermal storage capability.

DESCRIPTION OF THE INVENTION

[0035] Description a Preferred Embodiment: **FIG. 1** shows a preferred embodiment of the invention. A flow of desiccant fluid **1** is pumped by pump **9** to a solar collector **15** that acts as a regenerator for the desiccant fluid. The fluid trickles over a collector surface **2** in the form of a thin sheet **3**. A cover **4** transmits solar radiation **11**, which warms the desiccant fluid as it flows over the collector surface. A flow of air **10** removes water vapor that evaporates from the desiccant fluid. The concentrated desiccant **5** leaves the collector and flows to a mass-transfer device **6** that allows the desiccant to absorb moisture from an air stream **8**. The mass-transfer device is preferably a direct-contact exchanger similar to those used for direct evaporative coolers and may also include a pump for recirculating the desiccant liquid through the device to ensure good mass transfer. A supply air fan **8** moves the moves the air stream through the mass-transfer device.

[0036] An indirect evaporative cooler **14**, cools the air stream **8** without adding moisture to it. A fan **12** draws a secondary air stream **13** through the cooler. The secondary air stream may be exhaust air from a building, ambient air, or a portion of the conditioned air leaving the evaporative cooler or mass-transfer device. This indirect evaporative cooler is optional and may be eliminated in cases where no sensible cooling is required.

[0037] Indirect Evaporative Cooler Design: **FIG. 2** shows the one heat exchanger system that is suitable for use as an indirect evaporative cooler for this system. The cooler has two basic parts—mass transfer means **50** and an air-to-air heat exchanger **51**. The cooler is shown without a top cover for clarity. Corrugated panels **20** for secondary air are oriented so that corrugations run from side to side while corrugated panels **21** for primary air have corrugations that run from end to end. Channels **35** formed by the corrugations in panels **21** allow for free flow of air through the panels. Likewise similar channels run in a perpendicular direction through panels **20**. The panels **20** are stacked alternately with panels **21** so that the channels for each panel are perpendicular to the channels for the adjacent panels. The outside surface of the sheets may be covered with an adhesive or filler material to ensure good contact between the sheets. For maximum durability the panels are preferably made of polypropylene or polyethylene plastic. Metals, such as aluminum, are also possible materials.

[0038] Another option is to use corrugated cardboard and paper. Waterproof adhesive material, such as one based on acrylic or linseed oil, can coat the paper or cardboard and joins the layers together to form a single unit. The advantage of cardboard or paper is its very low cost. The disadvantage is that it is may be less durable. One advantage of this system is that it is not possible to create condensation within the heat exchanger, which allows the possible use of cardboard in some applications. This is especially true in desiccant applications since it is possible to keep both air streams above their respective dew point temperatures even when outdoor conditions are at 100% relative humidity (such as rain or fog conditions).

[**0039**] The main fan **29** moves primary air stream **38** through a single pass through channels **35**. The stacked panels **20** and **21** form a heat exchanger that cools the primary air without addition of humidity. A portion of the cooled primary air splits off and forms a secondary air stream **39** which is moved by secondary fan **30**. The secondary air first flows through multiple passes of the air-to-air heat exchanger to cool the primary air stream. The direction of the secondary air flow through the heat exchanger is shown by the dashed arrows.

[**0040**] The passes of the secondary air stream are preferably arranged in a counter crossflow configuration with a mass transfer means ahead of each pass of the heat exchanger. The mass transfer means is preferably a direct evaporative cooler. The direct evaporative cooling sections **23** form U-bends that direct the secondary air through each pass. As shown in the figure three triangular pieces fit together to form two mitered elbows which make each U-bend. Pass dividers **27** would normally be included to prevent excessive leakage between passes in the wet media in each pass. Housing **31** ensures that excessive air does not leak in or out of the heat exchanger.

[**0041**] The chief use of this system is as an evaporative air cooler, but many other applications are possible. In addition to air, this system can work equally well with any number of nonreactive gasses such as nitrogen, carbon dioxide, inert gasses, etc. This system can also be used as a heater. For example if a desiccant liquid is substituted for water and the entering gas stream has a high relative humidity, the system would act to heat the gas stream. Volatile liquids other than water-based solutions can be used in the system, but they are very expensive and may pose risks with flammability or toxicity.

[**0042**] The direct evaporative cooling sections need to be thoroughly wetted to ensure good evaporation while minimizing mineral deposits. In addition there is normally a large change in the wet-bulb temperature from one end of the heat exchanger to the other, so that water circulation between passes needs to be minimized to reduce undesirable heat exchange. These factors make it desirable to use multiple water circuits with multiple pumps.

[**0043**] For large systems using multiple pumps does not introduce a significant cost penalty, but for small systems multiple pumps can add greatly to the cost. One possible solution is to have multiple pumps that share a common shaft and motor. Seals separate the pumps from each other to minimize leakage and heat transfer.

[**0044**] **FIG. 2** shows another possible option for circulating liquid using air-lift pumps **42**. Air pump **38** supplies pressurized air through air line **36**. Drain **37** removes water from the bottom of the direct evaporative cooling sections **23**. Air bubbles into the water to create a pumping action. Extra water can be supplied to the pumps to make up for that lost to evaporation or blow down.

[**0045**] Other configurations of the air-to-air heat exchanger are possible. For example instead of stacking corrugated panels on top of each other, it may be possible on use spacers between the panels that are oriented in the same direction. The spacer could separate the passes of the secondary air and allow free flow of the secondary air over the panels. In this configuration the primary air would flow inside the channels of the panels. This alternative configuration should reduce material cost and reduces thermal resistance of the walls between the two air streams.

[**0046**] Another configuration would simply stack sheets with spacers to direct air flow. For example sheets of paper can be separated by corrugated cardboard spacers. The spacers would be on the order of 0.1 inches thick to form a flow channel for air. The orientation of the spacers would alternate so that the air flow for the secondary air is perpendicular to the that for the primary air. This arrangement would use a minimum amount of material and is a simple design and would be the preferred configuration for materials, such as paper, that are easily glued together.

[**0047**] Indirect Cooler Theory of Operation: **FIG. 3** is a psychrometric diagram showing how the idealized behavior of the system. For the case of conventional direct evaporative cooler, the process start at entering air **80** and follow the constant wet-bulb temperature line **87** (which is also essentially a line of constant enthalpy) and approach ideal exit condition **81** which is along saturation curve **88**. For the new system used as a cooler there are two exit conditions, the supply air **82** and exhaust air **83**. The primary air stream follows the line of constant absolute humidity **90** and approaches the saturation condition at point **82**. A portion of this air exits system as supply air and the rest moves along the saturation line **88** as it is heated and humidified until it approaches the ideal exhaust condition **83**. This exhaust condition is ideally at the intersection of the saturation line **88** and the constant dry-bulb temperature line **91**. The result is a colder supply air temperature than is possible with a simple direct evaporative cooler.

[**0048**] For the case of a heater, the conventional direct contact system would again follow constant wet-bulb line **87**. The process would start at the entering air condition **80** and approach ideal equilibrium point **84**, which is on the desiccant equilibrium curve **89**. For the new system, there are again two ideal exit conditions, the supply air condition **85** and the exhaust condition **86**. As with the cooler, the ideal supply air temperature is the point along the constant absolute humidity line **90** that is in equilibrium with the liquid. In both cases only a fraction of the primary air stream needs to be exhausted, typically 30 to 50%, which leaves the rest as supply air.

[**0049**] Air-Lift Pump: **FIG. 4** shows detailed drawing of an air-lift pump that is suitable for pumping liquid desiccant and water. Air pump **100** supplies pressurized air **103** to air line **101**. The air pump is preferably an aquarium pump or similar design. The air line **101** discharges inside water pipe **104** and creates a flow of air bubbles **102**. The bubbles lower the average density of the fluid column which causes the air and water mixture to move upward. This upward movement draws intake water **105**. On the top end a separator **108** allows outlet air **107** and outlet water **106** to discharge from the pump in separate flows. The advantages of this pump include low cost, simple design, reliability, no moving parts.

This pump is excellent for handling small liquid flows with a small head requirement. A single air pump can drive many air-lift pumps an thus create many liquid circuits.

[0050] Evaporative-Desiccant Heat Exchanger: **FIG. 5** shows a heat exchanger that adds the use of a desiccant. This arrangement has eight stages of cooling, **170, 171, 172, 173, 174, 175, 176**, and **177**. The incoming primary air **140** enters the rightmost stage **177**. It then flows through the eight stages in a straight path where it is cooled and dehumidified. The exiting primary air **141** splits into two flows. A secondary flow **142** goes back through the heat exchanger in a counter-crossflow arrangement. The remaining primary air **143** is supplied to the load.

[0051] As shown in this figure each stage of cooling includes an evaporative pad for cooling and humidifying the secondary air and an air-to-air heat exchanger that transfers heat between the two air streams. In addition some of the stages include desiccant, which dries the primary air stream. The desiccant is preferably a liquid, such as an aqueous solution of calcium chloride, lithium chloride, lithium bromide, glycol, or similar material. Materials such as sodium hydroxide and sulfuric acid have excellent physical properties but are very corrosive and dangerous to handle. Calcium chloride is very inexpensive, has acceptable thermodynamic properties, has relatively low toxicity, and is generally the preferred desiccant material for this system.

[0052] Starting with the rightmost stage **177** secondary air flows over evaporative pad **158** and through air-to-air heat exchanger **159**. The primary air flows through the other side of the air-to-air heat exchanger **159**. The flow directions of the two air streams are perpendicular to each other with the primary air going is a straight line through the heat exchanger.

[0053] Next to the left is stage **176**, which includes a desiccant **161** that is in the primary air stream. The desiccant is preferably provides a surface that is wetted by a liquid desiccant that is in direct contact with the primary air stream. The evaporative pad **56** cools and humidifies the secondary air stream. The secondary air stream cools the primary air stream in air-to-air heat exchanger **157**.

[0054] The stages **170, 175, 173**, and **171** are similar to stage **177** with evaporative pads **154, 150, 146**, and **144** in the secondary air stream and air-to-air heat exchangers **155, 151, 147**, and **145** transferring heat between the two air streams.

[0055] The stages **174** and **172** are similar to stage **176**. They include desiccants **162** and **163** which dehumidify and increase the temperature of the primary air stream. The evaporative pads **148** and **152** cool and humidify the secondary air which cools the primary air in air-to-air heat exchangers **149** and **153**.

[0056] Mist Cooling Option: **FIG. 6** is another heat exchanger configuration that uses a mist cooling system. Fan **220** draws in the main air stream **200**. The air moves in a straight line through interior channels **209** in panels **208** and is cooled by evaporating water mist **205** on the outside of panels **208**. The mist is supplied by nozzles **204** that are connected by way of pipe **212** to a source of pressurized water **213**. The water is preferably demineralized and filtered to prevent clogging and fouling of the nozzles and the heat exchanger surfaces. As the main air stream leaves the heat exchanger, a portion of the air forms a secondary air stream **202**, which returns on the wet side of the heat exchanger. Dividers **207** direct the secondary air in multipass counter crossflow arrangement as shown by the arrows. Housing **210** and wall **211** prevent undesirable air leakage. Fan **221** moves the secondary air out of the heat exchanger in exhaust stream **203**. Drains **222** may be included at the bottom of the housing to remove excess water.

[0057] Simple Solar Collector: **FIG. 7** is a simple solar collector for regenerating a desiccant solution. Liquid header, **230** trickles desiccant liquid **238** over collector surface **231** which is tilted at an angle to allow drainage through trough **234**. Solar radiation **233** is transmitted through cover **235** and warms the desiccant liquid **238**. The collector surface is preferably black and is backed by thermal insulation **232** to maximize energy collection.

[0058] The cover is preferably of a transparent material such as polycarbonate, polyvinyl chloride (pvc), fluoropolymers (such as Tedlar), acrylic, or other plastic. For rigid matericals, the cover may be flat or corrugated. Flexible films such as Tedlar would normally be held taught in a frame. Glass is another option for a cover material. The selection of optimum collector material depends on the cost and durability of the different materials. The duty is similar to that for greenhouses, windows, skylights, etc. with temperatures that are much lower than those for most other types of solar collectors.

[0059] Incoming air **239** flows by natural convection between the cover and the collector surface and absorbs moisture that evaporates from the desiccant liquid. The leaving air **237** exits the collector between end piece **236** and cover **235**. The end piece is shaped so as to prevent rain from entering the collector.

[0060] The desiccant solution **238** preferably flows in a sheet that wets the entire collector surface. One way of achieving this flow is to use a screen, cloth, or other roughness on collector surface **231**. Another option is to use a relatively large flow of liquid to create a continuous film of liquid. A third option is to add a detergent or other wetting agent to the solution to enhance wetting. Combinations of these three alternatives are also possible.

[0061] The orientation of the collector is preferably such that the rays of midday summer sun is approximately normal to the collector surface. For the most of the US this corresponds to a tilt of angle of 5 to 30 degrees from horizontal towards the south. In tropical areas the collector surface can be nearly horizontal with only a few degrees of tilt to allow adequate drainage. In the Southern Hemisphere the collector is preferably tilted to face north.

[0062] The operation of this collector is quite simple. When solar radiation is available to raise the collector surface to a temperature that is sufficiently high, desiccant liquid is allowed to flow through the collector. At other times no liquid would flow. A simple thermostat that controls the circulating pump can accomplish this control.

[0063] This collector has several advantages. First the temperatures necessary to regenerate the desiccant liquid are quite low, in the range of about 110 to 140 degrees Fahrenheit, which allows the use of inexpensive materials such as plastic, wood, asphalt roofing material, etc. Second the operation is very simple with no moving parts. Third the collector can be mounted on an existing roof or other surface.

[0064] While the preferred embodiment of the solar collector includes a cover, the collector would also function without a cover. The main advantages eliminating the cover are reduced cost and complexity. The collector can, in fact, be as simple as a section of dark roof or other surface with the addition of a system desiccant liquid over the surface. The chief problem with operation without a cover is that rain would tend to wash away any residual desiccant solution. The resulting diluted desiccant would have to be discarded or else it would dilute the solution in storage. Wind or leaves may also carry desiccant solution away when no cover is present. Loss of large quantities of desiccant solution is costly, may damage nearby plants or metals, and may create unsightly salt deposits on surrounding surfaces. A simple cover should greatly reduce or eliminate these problems, but it is not absolutely necessary for operation.

[0065] In dry climates an evaporation pond is an alternative to a solar collector. A pond is an inexpensive way of regenerating a desiccant solution. The chief problems are related to control over the salt concentration. An extended rainy period can dilute the solution excessively, while long periods of dry, sunny conditions can result in crystallization. Another issue is the possibility of high winds blowing droplets of desiccant solution onto surrounding surfaces which may create problems with corrosion, plant damage, etc.

[0066] Solar Still with Automatic Shut-Off Feature: **FIGS. 8***a* and **8***b* shows a solar still with an automatic shutdown feature that can regenerate the desiccant liquid. **FIG. 8***a* shows the still in normal operation. Solar radiation **260** warms desiccant liquid **250**. Water vapor evaporates from the desiccant liquid and form condensate **252** on cover **251**. The condensate trickles down the inside of the cover and collects in troughs **253** and **254**. An insulated tank **256** forms the bottom and the sides of the collector and holds the desiccant liquid.

[0067] Float **255** provides a simple control mechanism. As shown in **7***a*, the desiccant solution is relatively dilute, which reduces its density and causes the float to sink to the bottom of the tank **256**. **FIG. 8***b* shows a situation where the desiccant solution is quite concentrated, which increases its density and causes the float to rise to the top of the pool of desiccant liquid. Water evaporates out of any remaining desiccant liquid on the surface of the float and eventually creates a thin layer of salt crystals **257**, which helps to reflect solar energy and controls the temperature inside the collector. The action of the float thus provides an automatic shutdown feature that prevents excessive crystallization of the desiccant solution.

[0068] The float should be of nearly neutral buoyancy with respect to the desiccant solution, so that the change in solution density is enough determine whether the float rises or sinks. The float materials should be resistant to high temperatures and compatible with the desiccant solution. Foam glass, ceramics, high-temperature plastics, and metals that are compatible with the desiccant are likely choices. The float may be divided into smaller pieces to simplify handling.

[0069] The sealed cover has the advantage of keeping ambient moisture out of the desiccant during dark or cloudy periods. A cover without a seal would allow free movement of humid air, which can add undesirable moisture to the desiccant solution.

[0070] High-Performance Solar Collector with Electric Back-Up: **FIG. 9** is a high-performance solar collector with electric backup for use in regenerating the desiccant liquid. This collector provides three stages of regeneration and would operate with peak temperatures of around 200 to 240° F. The three stages of regeneration are arranged so the waste heat from higher-temperature stage drives a lower temperature stage. Desiccant liquid **301** flows from bottom header **304** over collector surface **300**. Electric heater elements **302** are located just under the collector surface and provide an auxiliary source of heat. Insulation **303** prevents excessive heat loss through the back of the collector.

[0071] The collector has three covers. The bottom cover **311** and middle cover **312** fit tightly with frames **309** and **310** to minimize air leakage. The top cover **313** has large gaps at each end, which allow for air movement under the cover. The bottom and middle covers **311** and **312** are preferably of glass or other heat resistant material. The top cover is preferably made of a tough plastic material to minimize risks of hail damage or other hazards. The top cover experiences much lower temperatures, so heat resistance is not an important issue.

[0072] Covers **311**, **312**, and **313** transmit solar radiation **321** which warms collector surface **300**. At night or during cloudy periods, electric heater elements **302** provide an auxiliary heat source for warming the collector surface. The warm temperatures cause moisture in the bottom stream of desiccant liquid **301** to evaporate. The water vapor thus produced is moved by convection and/or diffusion to bottom cover **311** where it condenses to form condensate **305**. The condensate flows down the undersurface of bottom cover and collects in a trough formed by catch member **307** and frame **309**. The condensate then flows out of the collector and can be used in an evaporative heat exchanger, as distilled drinking water, or discarded. Likewise a portion of the desiccant liquid that collects at the bottom of the collector can be returned to storage and the rest can be recirculated.

[0073] The middle stream of desiccant liquid **317** flows from middle header **314** over the top of the bottom cover **311**. The heat transmitted to the bottom cover from below evaporates moisture from the middle stream of desiccant liquid **317**. The moisture condenses on the bottom surface of the middle cover **312** to form the middle condensate stream **306** which drains through the trough formed by catch piece **308**.

[0074] The top header **315** supplies the top desiccant liquid stream **316** that flows across the top surface of the middle cover **312**. Moisture that evaporates from the top desiccant liquid stream **316** is removed by natural convection of air and does not normally condense on the top cover **313**. Entering air **323** flows through the collector and receives the evaporating water vapor. End piece **320** prevents excessive amounts of rain from entering the collector and air leaves the collector as exhaust stream **322**.

[0075] While this figure shows an electric heater as the backup heating system other heat sources are possible. Hot water, steam, and direct heating with a fuel are also possible.

The surface may be heated directly or the desiccant liquid can be heated in a separate heat exchanger. For small systems, a gas water heater may provide the heat source. The selection of the heat source would be determined by fuel cost and availability, installed cost and other factors. If electric power is used it would preferably used at night to take advantage of lower off-peak electric rates.

[**0076**] The optimum collector temperatures depend on the desiccant concentration, the ambient conditions, and other factors. For calcium chloride a minimum temperature difference of about 30° F. is necessary to evaporate a desiccant solution and condense the resulting water vapor. Assuming a temperature difference of about 40° F. in each stage, temperature of the middle cover would be about 120° F., the bottom cover would be 160° F. and the collector surface would be 200° F. If the peak temperature is a problem, a two-stage system can be used instead but with a performance penalty. Of course four or more stages are also possible, but the collector temperature is normally limited to the boiling point of the desiccant liquid which would be roughly 230° F. or somewhat higher.

[**0077**] This collector requires a means for circulating the desiccant liquid. Air-lift pumps or conventional pumps are possible alternatives. The desiccant liquid would normally be recirculated several times through the collector before returning to the storage tank. This recirculation ensures sufficient movement of liquid for adequately wetting the surfaces inside the collector without creating excessive heat loss. A heat exchanger between liquid entering and leaving the collector would further reduce heat losses, but this feature is not required for operation of the system.

[**0078**] The covers need to be tilted by roughly 10 degrees or more to ensure proper drainage of condensate. Smaller angles could result in excessive drainage of condensate back into the desiccant, which would create a large performance penalty.

[**0079**] The actual collector surface can be horizontal, which can allow desiccant to pool inside the collector. This arrangement allows the collector to also function as storage tank. Using a float as a shut-off control as shown earlier would be desirable for this system. This configuration may be especially desirable in tropical areas where the sun's rays are nearly vertical during much of the day.

[**0080**] Preferred Embodiment: **FIG. 10** is a schematic drawing of a complete solar air-conditioning system. Solar collector **400** concentrates desiccant solution using heat input from solar radiation **424** or auxiliary heat source as explained in the description of **FIGS. 8A and 8B**. A tank **401** stores a large quantity of desiccant solution **402**. The weak desiccant solution **420** leaves tank **401**and flows through the solar collector **400** and returns as a concentrated desiccant solution **421**.

[**0081**] The amount of desiccant solution in the tank depends on the size and efficiency of the system and the length of operation required. A reasonable objective would be to achieve one to three days of storage capability. For a system with a rated capacity of 12,000 Btu/hr system with a 50% duty cycle over two days, corresponds to storage requirement of 288,000 Btu of cooling (24 ton-hours). For a cooling COP of 1 and heat of vaporization of 1000 Btu/lbm mean that this cooling requirement could be met by the ability to absorb 288 lbm of water vapor. For calcium chloride solution with a starting concentration of 50% and an ending concentration of 40% $CaCl_2$ by weight, requires two pounds of calcium chloride to absorb one pound of water. This analysis means that 576 lbm of calcium chloride is required to store the required cooling. For a 40% final concentration, this corresponds to a tank capacity to handle 1440 pounds of solution or about 150 gallons.

[**0082**] The energy storage density per unit volume is almost twice that of ice and requires no special insulation. With proper sealing, the tank can store this cooling capacity indefinitely with essentially zero loss. The cost of calcium chloride is on the order of $0.20/lbm so that the cost of the salt for the above example is a little over $100. The cost of storage tank is similar. These costs work out to be roughly $10 per ton-hour, which is roughly 10 to 20% of the cost of conventional ice storage or cold-water storage. This storage system thus has major cost and performance advantages compared to other systems. This inexpensive, compact storage capability combined with simple, efficient solar recharging is a tremendous improvement over the prior art.

[**0083**] Theory of Operation of the Coolers: **FIG. 11** shows how temperatures vary through a system for supplying outside air. The layout of the modeled system is similar to that shown in **FIG. 4**. This system has 12 stages. Each stage has an air-to-air heat exchanger between primary and secondary, a direct evaporative cooling section on the secondary side, and a desiccant section on the primary side. In this analysis only stages **1**, **2**, and **4** have active desiccant sections. The mass flow rate on the secondary air stream is approximately half of that of the primary air stream for this analysis. The entering air conditions are 95° F. dry bulb and 75° F. wet-bulb temperature, which is a typical design condition for the eastern US.

[**0084**] The system in **FIG. 11** can be used to supply outside air to laboratories or other applications that have a large outside air requirement and limitations on heat recovery or other energy-saving technologies. It has the advantage of conditioning outside air with extremely high efficiency and requires no access to exhaust air. Changes in the details of the design can give different supply-air conditions as required for a particular application.

TABLE 1

Location	Temperature (degrees F.)	Enthalpy (Btu/Ibm)	Stage Location
		Primary Air Stream:	
500	95.0	38.6	1 Inlet
501	91.6	37.8	1 After heat exchanger
502	93.3	37.8	1 After desiccant
503	89.9	37.0	2 After heat exchanger
504	92.1	37.0	2 After desiccant
505	88.6	36.1	3 After heat exchanger
506	88.6	36.1	3 After desiccant
507	85.6	35.4	4 After heat exchanger
508	89.7	35.4	4 After desiccant
509	85.9	34.5	5 After heat exchanger
510	85.9	34.5	5 After desiccant
511	82.6	33.7	6 After heat exchanger
512	82.6	33.7	6 After desiccant
513	79.8	33.0	7 After heat exchanger
514	79.8	33.0	7 After desiccant

TABLE 1-continued

Location	Temperature (degrees F.)	Enthalpy (Btu/lbm)	Stage Location
515	77.2	32.3	8 After heat exchanger
516	77.2	32.3	8 After desiccant
517	75.0	31.8	9 After heat exchanger
518	75.0	31.8	9 After desiccant
519	73.0	31.3	10 After heat exchanger
520	73.0	31.3	10 After desiccant
521	71.3	30.8	11 After heat exchanger
522	71.3	30.8	11 After desiccant
523	69.9	30.5	12 After heat exchanger
524	69.9	30.5	12 After desiccant (supply air)
		Secondary Air Stream:	
525	88.1	45.9	12 After direct evaporative cooler
526	81.2	44.3	12 After heat exchanger
527	86.5	44.3	11 After direct evaporative cooler
528	79.7	42.7	11 After heat exchanger
529	85.0	42.7	10 After direct evaporative cooler
530	78.0	41.0	10 After heat exchanger
531	82.6	41.0	9 After direct evaporative cooler
532	76.7	39.6	9 After heat exchanger
533	82.2	39.6	8 After direct evaporative cooler
534	74.7	37.8	8 After heat exchanger
535	79.4	37.8	7 After direct evaporative cooler
536	72.9	36.2	7 After heat exchanger
537	77.0	36.2	6 After direct evaporative cooler
538	71.4	34.9	6 After heat exchanger
539	74.8	34.9	5 After direct evaporative cooler
540	69.9	33.7	5 After heat exchanger
541	72.9	33.7	4 After direct evaporative cooler
542	68.7	32.7	4 After heat exchanger
543	71.2	32.7	3 After direct evaporative cooler
544	67.5	31.8	3 After heat exchanger
545	69.7	31.8	2 After direct evaporative cooler
546	66.6	31.0	2 After heat exchanger
547	68.6	31.0	1 After direct evaporative cooler
548	66.0	30.4	1 After heat exchanger (exhaust)

[0085]

TABLE 2

Location	Temperature (degrees F.)	Enthalpy (Btu/lbm)	Stage Location
		Primary Air Stream:	
600	74.0	30.2	1 Inlet
601	73.6	30.1	1 After heat exchanger
602	81.1	30.1	1 After desiccant
603	78.6	29.5	2 After heat exchanger
604	80.7	29.5	2 After desiccant
605	77.9	28.8	3 After heat exchanger
606	77.9	28.8	3 After desiccant
607	75.5	28.2	4 After heat exchanger
608	78.4	28.2	4 After desiccant
609	75.4	27.5	5 After heat exchanger
610	75.4	27.5	5 After desiccant
611	72.6	26.9	6 After heat exchanger
612	72.6	26.9	6 After desiccant
613	70.2	26.3	7 After heat exchanger
614	70.2	26.3	7 After desiccant
615	68.0	25.7	8 After heat exchanger
616	68.0	25.7	8 After desiccant
617	65.9	25.2	9 After heat exchanger
618	65.9	25.2	9 After desiccant
619	64.1	24.8	10 After heat exchanger
620	64.1	24.8	10 After desiccant
621	62.4	24.3	11 After heat exchanger
622	62.4	24.3	11 After desiccant
623	60.9	24.0	12 After heat exchanger
624	60.9	24.0	12 After desiccant (supply air)

TABLE 2-continued

Location	Temperature (degrees F.)	Enthalpy (Btu/lbm)	Stage Location
		Secondary Air Stream:	
625	73.2	36.0	12 After direct evaporative cooler
626	72.5	35.8	12 After heat exchanger
627	76.1	35.8	11 After direct evaporative cooler
628	71.2	34.7	11 After heat exchanger
629	75.1	34.7	10 After direct evaporative cooler
630	69.5	33.3	10 After heat exchanger
631	73.0	33.3	9 After direct evaporative cooler
632	68.2	32.1	9 After heat exchanger
633	72.3	32.1	8 After direct evaporative cooler
634	66.3	30.7	8 After heat exchanger
635	69.9	30.7	7 After direct evaporative cooler
636	64.5	29.4	7 After heat exchanger
637	67.8	29.4	6 After direct evaporative cooler
638	62.9	28.2	6 After heat exchanger
639	65.8	28.2	5 After direct evaporative cooler
640	61.4	27.2	5 After heat exchanger
641	63.9	27.2	4 After direct evaporative cooler
642	60.0	26.2	4 After heat exchanger
643	62.3	26.2	3 After direct evaporative cooler
644	58.7	25.4	3 After heat exchanger
645	60.7	25.4	2 After direct evaporative cooler
646	57.5	24.6	2 After heat exchanger
647	59.5	24.6	1 After direct evaporative cooler
648	56.7	23.9	1 After heat exchanger (exhaust)

[0086] **FIG. 12** shows air temperatures for a system for cooling return air. This system is similar to that of **FIG. 11** except for the different entering-air conditions. Table 2 describes each location. This system is capable of providing a supply air temperature of 61° F., which is sufficiently low to be compatible with most conventional air-distribution systems. The supply air is at about 75% relative humidity, which is sufficiently low to prevent mold growth in ducts and maintain comfortable space humidity.

[0087] These supply-air conditions are illustrative of what is possible with this system. Changes in the details of the design can change the supply air conditions to whatever is required. For example this system is suitable for use in an application which uses a 65 to 70° F. supply air temperature such as is described in my co-pending application entitled "High Efficiency Air Conditioning System with High Volume Air Distribution." Low temperature air is also possible, but with reduced performance.

[0088] A complete cooling system is a combination of the system in **FIGS. 10 and 11**. The system in **FIG. 12** exhausts approximately half of the return air. The system in **FIG. 11** supplies the make-up air required to replace this exhaust air.

[0089] High System Efficiency: The efficiency of these systems is quite high. For these systems the coefficient of performance (COP) is defined as cooling output divided by latent heat absorbed by the desiccant. For the system in **FIG. 11**, the COP is approximately 2.1. For the combined system including ventilation load the efficiency is about 1.5.

[0090] Note that this configuration requires the introduction of outside air. If the basis of comparison is system with no outside air, then there should be no credit for the load associated with cooling the outside air to the building conditions. On this basis the system COP is approximately 0.8.

[0091] If the desiccant is recharged using a solar collector, the collector efficiency must be considered. For the three-stage collector shown if **FIG. 9**, the waste heat from one stage is used to drive the next, which theoretically can triple the output of concentrated desiccant solution. In real life, the collector loss would reduce the effect. Assuming a 50% collector loss, the total system efficiency based on solar input to cooling output can exceed 2.0. This performance is much better than what is possible with expensive absorption chillers with high-performance solar collectors, which gives a system efficiency of 0.2 to 0.5 at best. This efficiency advantage translates into a massive reduction in collector cost and area required to drive the new system compared to the prior art. Even with the use of low-cost collectors such shown in **FIG. 7**, the new system has a massive efficiency advantage compared with the prior art.

[0092] This high performance allows the use electric resistance as a back-up heat source. The efficiency with electric backup should be higher than that for a solar input, since transmission and reflection losses are not a factor. This means that system efficiency in the range of 2 to 3 based on electric input is possible. This efficiency is comparable to that of conventional electric air conditioners. If combined with a suitable storage system, electric back up can take advantage of inexpensive, off-peak electric rates. These rates can be as much as a factor of 10 lower than peak rates. The combination of solar input and off-peak rates can result in a massive reduction in energy cost compared to conventional systems.

[0093] Gas Turbine Inlet Cooler: **FIG. 13** shows a system for cooling inlet air to a gas turbine that uses heat from the turbine for regenerating the desiccant solution. Ambient air **701** enters a regenerative desiccant cooler **700**, which cools the air. The cooler includes pumps for circulating water and desiccant solution inside the cooler. The cooler receives make up water **710** and used water **711** drains from the cooler. Exhaust fan **714** draws the exhaust air **709** from the cooler and discharges away from the turbine to prevent recirculation. The turbine inlet air **702** leaves the cooler **700** and enters the gas turbine **713**. Fan **715** adds ambient air **704** to the turbine exhaust air **703** to form mixed air **705**. The mixed air **705** enters regenerator **706** where it evaporates water from the desiccant solution. The regenerator comprises an extended surface that is wetted with desiccant liquid. It may be made of materials similar to that used in direct evaporative coolers. Mixing ambient air with turbine exhaust lowers the temperature of air entering the regenerator, which allows the use of inexpensive low-temperature materials. Pump **716** moves diluted desiccant solution **708** from the cooler **700** to the regenerator **706**. The regenerator can include a pump or circulating desiccant liquid inside the regenerator. Concentrated desiccant **707** returns to the cooler from the regenerator. Outlet air **712** exits from the regenerator.

[0094] This system can increase turbine output power by roughly 20 percent at summer peak conditions. The capacity gas turbine declines by about 0.4 percent per degree Fahrenheit. A 20% improvement in capacity corresponds to cooling the inlet temperature is reduced from 100 to 50° F. at peak conditions. The system can also control relative humidity to the turbine. Turbine efficiency improves by roughly 0.1%/° F. which corresponds to as much as 5% improvement at peak conditions. Input power for fans and pumps needed to operate the desiccant system is small and should not significantly effect these figures.

[0095] Heat-Exchanger Details: **FIGS. 14*a*, 14*b*,** and **14*c*** show an alternate gas-to-gas heat exchanger design using paper and cardboard. Sheets of paper **800** are supported between first spacers **801** and second spacers **802**. The spacers are preferably made of corrugated cardboard or similar material. As shown in **FIG. 14*b***, first spacers **801** are oriented to allow primary air stream **803** to flow the length of the paper in a single pass. Second spacers **802** are set to form multiple passes of secondary air stream **804**. The whole heat exchanger is coated with a material such as linseed oil, acrylic, wax, etc. which serves as both an adhesive and a protective coating. While this drawing shows a two-pass arrangement on the secondary side, similar geometries can accommodate any number of passes. This heat exchanger construction has many applications including exhaust-air heat recovery in addition to use in evaporative and desiccant systems.

[0096] Dehumidifer Embodiment: **FIG. 15** shows another preferred embodiment that is acts as a dehumidifier. Desiccant fluid **900** is contained in an insulated container **904** and is heated by solar radiation that is transmitted through a cover **902**. Moisture evaporates from the fluid and forms condensate **906** on the bottom side of the cover. The condensate flows down the underside of the cover and collects in a channel **908**. The desiccant fluid moves by natural convection through channels **910** and **912** to a tank **918**. An air pump **914** blows air through a tube **915** into the desiccant fluid **900** forming bubbles **916**. The flow of air mixes the desiccant liquid in the tank. A fan **922** draws air over the desiccant fluid, which dehumidifies the air stream. A baffle **920** directs the air to toward the surface of the liquid. The baffle also acts to cut off air flow if the liquid level gets too high, thus preventing and overflow of desiccant liquid. A level switch may also be included to turn off the air pump and fan at high liquid levels.

[0097] This embodiment may be useful as a small dehumidification system that can fit in a window. It may be especially useful for bathrooms or basements in homes.

[0098] **FIG. 16**—Preferred embodiment with liquid-to-liquid heat exchange: **FIG. 16** shows a preferred embodiment desiccant cooler with a heat exchanger between desiccant liquid and water. The system comprises a desiccant cooler **1008**, a desiccant tank **1061** and a solar collector **1090**. A fan **1022** draws ambient air and moves an air stream **1024** through desiccant-gas heat and mass exchanger **1012** that cools and dehumidifies the air. Air stream **1024** moves through an evaporative cooler to form a supply air stream **1026**, which is enters a conditioned space **1050**. A fan **1052** moves air from the conditioned space as an air stream **1028**, which then moves through a water-gas heat and mass exchanger **1014**. The water-gas heat and mass exchanger humidifies and adds thermal energy to the air stream to create an exhaust air stream **1030**, which is discharged to the ambient. It also acts to cool water **1046**, which flows through the exchanger.

[0099] There are two liquid loops that supply these two heat and mass exchangers. A desiccant loop includes a pump **1036**, which draws liquid desiccant **1018** from a reservoir **1019** and supplies it at the top of the desiccant-gas heat and mass exchanger **1012** as a drops of water **1038** that wets media that forms the exchanger. The air and desiccant are preferably in a counterflow configuration with the desiccant flowing down while the air flows upward. Desiccant collects in a drain pan **1032** and then flows through a liquid-to-liquid heat exchanger **1010**. The heat exchanger **1010** cools the desiccant, which then flows back to reservoir **1019**.

[**0100**] The water loop is formed by a pump **1020** which draws warm Water **1042** from liquid-to-liquid heat exchanger **1010**, which preferably has a counterflow arrangement. Water **1046** is distributed at the top of the water-to-gas heat and mass exchanger **1014** and accumulates in a drain pan **1040** and returns to the heat exchanger **1010** as cooled water **1044**. As with the desiccant side, the water and air are preferably arranged in a counterflow configuration to maximize thermal performance. Make-up water **1049** enters through a valve **1051** to replace blow-down water **1047** which drains from the system and water that evaporates. The valve **1051** may be a float valve or other device to maintain an approximately constant volume of water in the water loop.

[**0101**] The solar collector **1090** comprises a cover **1068** that transmits solar radiation **1078** to warm a pool of desiccant liquid **1070**. The pool of desiccant liquid is preferably is contained by a liner **1082** which is supported by a layer of support material **1072** which sits on top of a roof **1076** or other flat surface. The support material is preferably stone or ceramic granular material such as pea-sized gravel, pearlite, sand, or polystyrene foam beads. Closed-cell foam in another option. The liner is preferably made of a black plastic material such polyvinyl chloride, polyethylene, rubber such as is used in pools liners or landfill liners. It preferably includes a cloth or foam underlay for improved life and reduced heat transfer. These liners are typically about 20 to 40 mils thick and are designed to withstand years of ultraviolet radiation from the sun. A frame **1074** supports the cover **1068**. The frame may be made of pressure-treated lumber or other material suitable for outdoor use. Ambient air **1080** enters the collector through holes in the frame **1074** and flow through the collector to remove moisture evaporated from the desiccant and leaves as an exhaust stream **1082**. The walls formed by the liner **1082** slope slightly to allow liquid to drain to a pump **1066**.

[**0102**] A desiccant storage tank **1061** contains a quantity of desiccant **1060** sufficient for at least two hours of operation of the cooler **1008**. The preferred quantity depends on the climate conditions, but would normally be at least sufficient to allow operation from late afternoon to the next morning when solar input is limited or not existent. A pump **1064** moves desiccant liquid to the solar collector during period when solar energy is available and the desiccant solution needs to be further concentrated. A pump **1062** moves liquid to the cooler **1008** as required to maintain a proper desiccant concentration in the cooler. An optional heat exchanger may be included between the desiccant entering the reservoir **1019** and leaving the reservoir so as to reduce thermal losses associated with the fluid transfer.

[**0103**] The system preferably includes a controller **1092** that controls operations of the fans and pumps. The controller is in communication with a solar sensor **1094** and liquid-level sensor **1093**. The solar sensor may comprise a black thermistor or other temperature sensor that is exposed to solar radiation. The liquid-level sensor is preferably a simple liquid-level switch. The controller also receives input from a desiccant concentration sensor **1095** that preferably comprises a float switch that closes when the desiccant density reaches a predetermined value. Other options include more sophisticated sensors such as density sensor or electrical conductivity measurements or a simple liquid level sensor.

[**0104**] The controller uses the input from these two sensor to determine when operate the solar collector. When the solar sensor **1094** senses a sufficiently high temperature (about 100 to 130 F) and the concentration sensor shows that the desiccant is sufficiently dilute, the controller turns on pump **1064** to move a quantity of desiccant to the collector **1090**. Once the liquid level reaches a predetermined limit, the liquid-level sensor **1093** communicates to the controller and the control turns off the pump **1064**. The collector preferably has sufficient liquid holding capacity, that in the event of a failure of the liquid level sensor would not result in overflow of desiccant liquid. The controller activates the pump **1066** and pump **1062** to periodically move concentrated desiccant back to the tank **1061**, which accumulates concentrated liquid desiccant. A heat exchanger may be included between the desiccant entering and leaving the collector to improve thermal performance.

[**0105**] During extended periods of very dry weather, it may be necessary to include a means for adding water to the desiccant to prevent crystallization of salt. This situation may occur in desert climates where ambient dewpoint temperatures are lower than about 55 F, but sensible cooling is still required. The water addition is preferably accomplished by diverting circulating a portion of the exhaust stream **1030** into the air entering fan **1022**, which raises the dewpoint of air entering the desiccant-gas heat and mass exchanger **1012**. Alternatively, the evaporative cooler **1016** may be operated without the rest of the cooler **1008** during periods of sufficiently low humidity without diverting exhaust air.

[**0106**] A humidity sensor **1096** and a temperature sensor **1098**, which are located in the conditioned space **1050**, provide input to control the operation of the cooler **1008**. The humidity sensor may be a humidistat and the temperature sensor may be a thermostat. If the temperature or humidity exceeds predetermined limits, then the controller turns on the cooler **1008**. If the humidity is sufficiently low but the space is too warm, then the controller may activate the evaporative cooler **1016** to provide a lower temperature for the supply air stream **1026**, otherwise the evaporative cooler is normally off.

[**0107**] For good performance of the cooler **1008**, it is necessary to maintain proper flow for the fluids. In a typical application, the flow rate of the air stream **1028** should be close to that for the supply air stream **1026**. In addition the flows of the water and the desiccant liquid should be adjusted to maintain close the same temperature change across the heat exchanger **1010**. The liquid temperature change should also be close the airside temperature changes for good performance of the heat exchangers. For simplicity, these adjustments are preferably made manually and set at a design value. Alternatively, for optimum performance, the adjustments can be made continuously with an automatic controller that receives input from appropriate temperature sensors.

[0108] **FIG. 17**—Chiller embodiment: **FIG. 17** shows an embodiment that is suitable for producing chilled water for air conditioning. A cooler **1100** comprises a desiccant-gas heat and mass exchanger **1104** and water-gas heat and mass exchanger **1102** in an enclosure **1112**. The enclosure is preferably gas-tight and capable of withstanding atmospheric pressure when a partial vacuum is created inside. A fan **1110** circulates gas between the two exchangers. A vacuum pump **1114** draws gas **1116** from the space and discharges it to the atmosphere **1118**.

[0109] The operating pressure inside the enclosure **1112** is preferably about 1 to 5 psia. Gas filling the enclosure is mixture a mixture of air and water vapor. Lowering the atmospheric pressure has several advantages. First it greatly improves the heat- and mass-transfer coefficients. These coefficients are approximately inversely proportional to the partial pressure of the air, which means they get very large as the pressure of the gas mixture approaches the vapor pressure of the water. A second advantage is that the lower pressure reduces the thermal losses associated with circulating the gas between the two exchangers. A third advantage is that the lower pressure reduces the fan energy required to circulate the gas.

[0110] While air is preferred for simplicity and low-cost, low-molecular-weight gas such as hydrogen or helium may be used instead to improve heat transfer, in which case gas should be recovered from the exit of the vacuum pump. While operation at below atmospheric pressure is preferred, the system can work at atmospheric pressure, but with a large performance penalty.

[0111] Dry gas **1120** leaves the top of the desiccant-gas exchanger **1104** and enters the bottom of the water-gas exchanger **1102**. The exchangers are arranged in a counter-flow configuration with liquid entering at the top and gas entering as the bottom. The temperature of water drops as it flows through the water-gas exchanger **1102** and the exit temperature approaches the wet-bulb temperature of the gas entering the exchanger. Humid gas **1122** leaves the top of the water-gas exchanger **1102** and enters the bottom of the desiccant-gas exchanger **1104**.

[0112] The cooler include provisions for changing the working liquids. A pump **1136** removes a small portion of the circulating water to prevent excessive accumulation of salts. Make-up water **1134** replaces water withdrawn by the pump along with water evaporated in the water-gas exchanger **1102**. Likewise a pump **1140** removes a quantity of the circulating desiccant solution, which is replaced by concentrated desiccant **1142** to ensure a proper concentration of desiccant.

[0113] Pumped fluid transfers heat outside of the cooler. Cooled water **1126** is circulated by pump **1124** through a coil **1128**. A fan **1130** moves air over the coil **1128** to cool a conditioned space **1132**. A pump **1148** moves desiccant **1126** through a heat exchanger that is cooled by water from a cooling tower **1144**. A pump **1146** circulates water through heat exchanger **1150** to the cooling tower **1144**.

[0114] The cooler **100** differs from an conventional absorption chiller in that it is designed to handle a gas-vapor mixture. In conventional chillers, the systems are designed to operate with extremely low levels of non-condensable gases. Any appreciable quantity of non-condensable gas creates large heat transfer penalties because there is no fan for moving the gas across the heat exchange surfaces. By comparison the present invention uses fan and an extended heat/mass exchange surface to allow operation with a large amount of non-condensable. This ability to handle large quantities of non-condensable gas greatly improves the ability to use storage of desiccant and regeneration of desiccant at atmospheric pressure without special concerns about contact with air.

[0115] The present cooler is designed to take advantage of a temperature glide inherent in a gas-vapor mixture. In a conventional absorption chiller, the vapor pressure of the evaporator and the absorber is a single value. In contrast, air or other non-condensable gas allows the vapor pressure to vary, while maintaining close to a constant total pressure. This difference increases the available temperature lift from the desiccant by an amount that roughly corresponds to the temperature change of the cooling water across the absorber.

[0116] The optimum performance of the cooler **1100** occurs when the temperature change of each fluid is approximately the same value. This setup minimizes the temperature drop through each heat exchanger and improves the temperature lift capability.

[0117] For calcium chloride the maximum available temperature lift (entering desiccant temperature minus leaving chilled water temperature) is about 25 F. For temperatures lifts greater than this amount a two-stage system or a different desiccant is required.

[0118] For climates with relatively low design wet-bulb temperatures (such as those in California), the cooler should be able to provide sensible cooling in a single-stage configuration. For example, for a design wet-bulb temperature of 70 F, a leaving cooling-tower water temperature of 75 F is reasonable (5 F approach temperature). A temperature of 77 F for the desiccant leaving heat exchanger **1148** should allow a water temperature to the coil of about **60** F, which should allow a supply air temperature of about 68 F. This setup is especially suitable for use with the air conditioning system described in U.S. Pat. Nos. 6,405,543 and 6,185,943, which use high-temperature air distribution system with separate dehumidification.

[0119] **FIGS. 18 and 19**—Rotating Embodiment: **FIGS. 18 and 19** show an alternate embodiment of a cooler with rotating heat exchangers, which preferably comprises multiple stages. **FIG. 18** shows a detail of a single stage. A pipe **1200** encloses a first direct-contact heat exchanger **1202**. The interior of the pipe also contains a liquid, preferably the desiccant **1210** with end pieces **1206** that prevent leakage from the ends.

[0120] A second direct-contact heat exchanger **1212** is located around the outside of pipe **1202**. Tubing **1208** is wrapped around the second direct contact heat exchanger and is connect at each end through fittings **1204** and **1205** to the inside of pipe **1202**. The bottom portion of the tubing and the second direct-contact heat exchanger sit in pool of liquid, which is preferably water, which is located below liquid level line **1214**.

[0121] The whole assembly rotates as a unit, which provides a means for circulating the liquids. Liquid desiccant **1210** enters into tube **1208** through fitting **1204**. The rotation and the force of gravity move the desiccant through the tubing and it eventually returns to the inside of pipe **1200** through fitting **1205**. The turning action also submerges portions of the first direct-contact heat exchanger in desiccant **1210** and portions of the second direct-contact heat exchanger **1212** into water located below liquid line **1214**. These setup allows circulation of liquid for heat and mass transfer without the use of a pump.

[**0122**] **FIG. 19** shows a multi-stage assembly for this cooler. A first, second, and third stage **1244**, **1246**, and **1248** are all connected together a rotate as a unit. Each stage has a geometry that is similar to that in **FIG. 18**. Pans **1241** are connected together with tubes **1242** and are filled with water. Make up water **1254** enters the pan for the first stage **1244** and blow-down water **1252** exits the pan for the third stage. Rollers **1242** support the stages and allow them to rotate freely. A motor **1260** that is connected to the rotating stages by a shaft **1262** turns the assembly. Ambient air **1270** is drawn by fan **1250** and moves through the desiccant side of the stages. Supply air **1272** exits the first stage and cools a conditioned space **1264**. Return air **1274** flow through the waterside of the assembly and exits as exhaust air **1276**. While three stages are shown the cooler may use twelve or more stages, depending on the design requirements.

[**0123**] Alternate thermal-energy input: While solar energy input is preferred in many applications, there are situations where solar is not practical because of space limitations, climate, or other factors. In those situations the preferred embodiment uses an alternate source of thermal energy. While natural gas or other fuel is one alternative, availability and/or cost may limit its use. Another alternative is to use waste heat from a conventional vapor-compressor or absorption refrigeration system.

[**0124**] Resistance heat from off-peak electricity is yet another alternative. The system can regenerate the desiccant at night or on weekends during periods of low electric prices for use during periods of high electric prices.

[**0125**] For these systems without solar input, no collector is required. Instead a regenerator, preferably with multiple stages of regeneration, may be included. Various arrangements similar to those found in the prior art for distillation of sea water, absorption chillers, etc. are possible. For a regenerator that is limited to atmospheric pressure and peak temperatures of about **200** to **250** F, three stages of regeneration should be possible with calcium chloride. This temperature and pressure is achievable using relatively inexpensive and corrosion-resistant material such as plastic and ceramic in the construction of the regenerator.

[**0126**] For more stages, higher pressures and temperatures are required. Theoretically ten stages or more of regeneration are possible. More stages of regeneration increase cost and complexity of the regenerator, but improve the efficiency of the system. The optimum design depends on material costs, pressure-vessel code considerations, cost of the thermal input, temperature limits of available materials, and so on.

[**0127**] Advantages Summary: Overall this invention has many advantages over the prior art:

[**0128**] 1) Simple, low-cost, reliable designs

[**0129**] 2) Counter cross flow configuration reduces cost an achieves good performance

[**0130**] 3) Evaporative system with the ability to approach the dewpoint temperature

[**0131**] 4) Desiccant system can act as a heat pump to raise supply air temperatures

[**0132**] 5) Ability to use low-cost, safe, desiccants such as calcium chloride for air-conditioning

[**0133**] 6) Compact energy, inexpensive, low-cost, energy-storage capability in the form of a concentrated desiccant solution

[**0134**] 7) Low-cost solar regeneration

[**0135**] 8) Ability to use off-peak electricity for back-up or alternate to solar

[**0136**] 9) Ability to separate the wet portions of the system so as to allow easy replacement

[**0137**] 10) Use of reliable, low-cost air-lift pumps which allow the use of many different liquid circuits

[**0138**] 11) Extremely high efficiency possible

[**0139**] 12) Simple controls

[**0140**] 13) Ability to use readily available components

[**0141**] 14) Low-cost paper-based or plastic heat exchangers

1) A desiccant cooling system comprising:

a. A desiccant cooler capable of cooling a fluid to a temperature below the ambient wet-bulb temperature,

b. A quantity of desiccant liquid sufficient for providing at least about two hours of operation of said desiccant cooler at design capacity,

c. Means for regenerating said desiccant liquid,

d. Means for storing said desiccant liquid.

2) A desiccant cooling system of claim 1 wherein said desiccant cooler comprises:

a. A desiccant-gas heat and mass exchanger which acts to cool and dehumidify said gas,

b. Means for cooling said desiccant liquid,

c. Means for circulating said desiccant liquid through said desiccant-gas heat and mass exchanger,

d. A water-gas heat and mass exchanger which humidifies said gas, and

e. Means of circulating gas that has been previously dehumidified by said desiccant-gas heat and mass exchanger to said water-gas heat and mass exchanger so as to cool said water below the ambient wet-bulb temperature.

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