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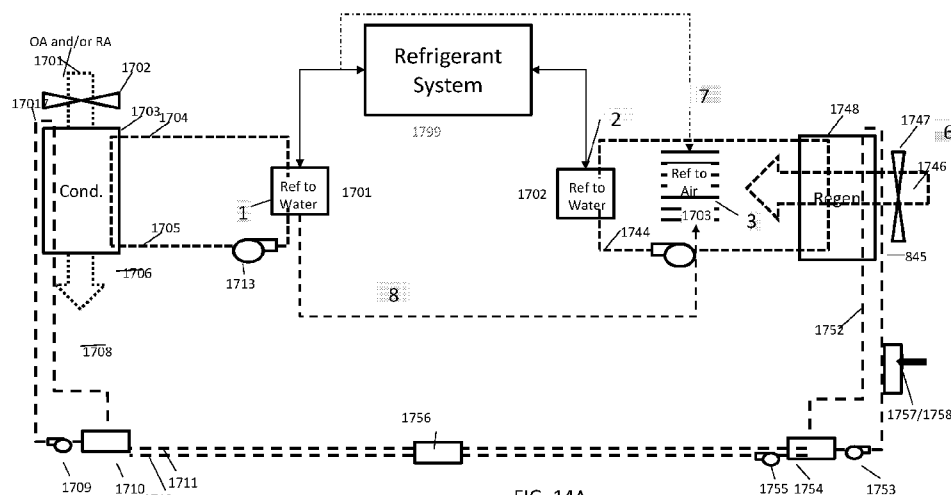


FIG. 14A

Table 14A

| HX | Operating modes |         |                 |             |
|----|-----------------|---------|-----------------|-------------|
|    | Cooling         | Heating | Defrost heating | Adv. Dehum. |
| 1  | Cold            | Hot     | Hot             | Cold        |
| 2  | Hot             | Cold    | Cold            | Hot         |
| 3  | Off             | Off     | Colder          | Cold        |

(57) Abstract: Methods and control systems are disclosed for operating a liquid desiccant air-conditioning system to efficiently maintain a target temperature and humidity level in a space.

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## **CONTROL SYSTEMS FOR LIQUID DESICCANT AIR CONDITIONING SYSTEMS**

### **CROSS REFERENCE TO RELATED APPLICATION**

**[0001]** This application claims priority from U.S. Provisional Patent Application No. 62/580,249 filed on November 1, 2017 entitled CONTROL SYSTEMS FOR LIQUID DESICCANT AIR CONDITIONING SYSTEMS, which is hereby incorporated by reference.

### **STATEMENT AS TO FEDERALLY SPONSORED RESEARCH**

**[0002]** The United States Government has rights in this invention under Contract No. DE-AC36-08GO28308 between the United States Department of Energy and the Alliance for Sustainable Energy, LLC, the Manager and Operator of the National Renewable Energy Laboratory.

### **BACKGROUND**

**[0003]** The present application relates generally to the use of liquid desiccants (LD) in combination with heat pumps, compressors, and chillers to condition the temperature and humidity of an air stream entering a space. Current control systems are designed for Direct eXpansion (DX) systems or solid desiccant wheel systems. Liquid desiccant air conditioning systems (LDAC) allow for significant independent control of humidity and temperature, while reducing the energy required to achieve specific supply air targets by up to 50%. The capability to Independently control of temperature and humidity supplied by LDAC systems not only improves comfort and health, but also simplifies building management controls and reduces the risk of humidity damage to the building. Depending on the latent and sensible loads that need to be managed, the system can either heat and simultaneously humidify air, or heat and simultaneously dehumidify or cool the air while humidifying or dehumidifying the air. This enables system managers to maintain more comfortable and healthier indoor air conditions than conventional systems can provide. Such independent control of humidity and temperature is critical for many applications, including but not limited to outside air supply to commercial buildings in monsoon regions or the use of air conditioners in buildings which have spaces with very different sensible heat ratio (SHR) requirements, like grocery stores with high humidity and relatively warm grocery/bakery sections and dry and cool refrigeration sections. Controlling systems and buildings to meet such load

characteristics requires appropriate control strategies for individual pieces of equipment and for the complete buildings.

**[0004]** Desiccant dehumidification systems – both liquid and solid desiccants – have been used in parallel to conventional vapor compression HVAC equipment to help reduce humidity in spaces, particularly in spaces that require large amounts of outdoor air or that have large humidity loads inside the building space itself. (ASHRAE 2012 Handbook of HVAC Systems and Equipment, Chapter 24, p. 24.10). Humid climates, such as Miami, FL, require a lot of energy to properly treat (dehumidify and cool) the fresh air that is required for a space's occupant comfort. Solid desiccant dehumidification systems have been used for many years and are generally quite efficient at removing moisture from the air stream. However, liquid desiccant systems generally use concentrated salt solutions such as ionic solutions of LiCl, LiBr, or CaCl<sub>2</sub> and water. Membrane based liquid desiccant systems have been primarily applied to unitary rooftop units for commercial buildings. However, in addition to rooftop units, commercial buildings also use air handlers located inside technical spaces in the building for the cooling and heating of both outside air and recirculated air. There is an additional substantial market for chillers that provide cold water to coils inside the building and use evaporative cooling for high efficiency condensers. Residential and small commercial buildings often use split air conditioners wherein the condenser (together with the compressor and control system) is located outside and one or more evaporator cooling coils is installed in the space than needs to be cooled. In Asia in particular (which is generally hot and humid) the split air conditioning system is the preferred method of cooling (and sometimes heating) a space. Each of those require different configurations and control mechanisms. Various configurations for managing humidity and temperature independently have been disclosed in U.S. Patent No. 9243810 and U.S. Patent Application No. 62/580270. They disclose configurations with liquid desiccant components, sensible coils, direct and indirect evaporative cooling and water addition. They disclose how performance for various modes of operation is optimized, including cooling and dehumidification, cooling only, cooling and humidification, heating only, heating and humidification and heating and dehumidification. Cooling and heating refer here to changing the DB condition only; changes in total enthalpy will be described as

net heating and net cooling. Existing control strategies need to be adjusted to optimize efficiency and comfort in each of these operating modes for the different configurations.

**[0005]** Liquid desiccant systems generally have two separate functions. The conditioning side of the system provides conditioning of air to the required conditions, which are typically set using thermostats or humidistats. The regeneration side of the system provides a reconditioning function of the liquid desiccant so that it can be re-used on the conditioning side. Liquid desiccant is typically pumped or moved between the two sides, and a control system helps to ensure that the flows and concentrations of the liquid desiccant is properly balanced between the two sides as conditions necessitate and that excess heat and moisture are properly dealt with, without leading to over-concentrating or under-concentrating of the desiccant.

**[0006]** Performance of the conditioner and regenerator is driven by the flow rates and temperatures the three fluids: air, water, and desiccant in the heat exchangers. The dehumidification potential is driven by the concentration of the desiccant, which can be controlled in a number of ways as disclosed, e.g., in U.S. Patent No. 9243810 and U.S. Patent Application No. 62/580270. The primary controls are compressor power and fan speed and water flow of the regenerator. Adding sensible cooling capacity with additional air-to-water or air-to-refrigerant coils can greatly broaden the amount work done by the system and the range of sensible heat ratios that can be supported by increasing the ability for independent control of temperature and humidity within the unit.

**[0007]** At a building level, overall effectiveness of the system is driven by the mix of systems used and how they are used. For example, the ASHRAE DOAS (Dedicated Outside Air Unit) design guide identifies how conditioning the outside air to handle the complete dehumidification load of the building can increase overall efficiency, due to the much improved efficiency of air cooled coils when only used for sensible cooling,.

**[0008]** The benefits of liquid desiccant systems have been described in various patents, e.g., U.S. Patent No. 9243810 and others. Such systems have been clearly demonstrated for hot and humid climates with a large latent load. As buildings get better insulated, these latent loads increase as a percentage of total cooling loads, making effective dehumidification more important. As internal sensible loads are

reduced in tighter, better insulated buildings, conditioning ventilation air becomes an even more significant part of total cooling and heating loads.

**[0009]** Extreme design conditions, including very humid and cool, very hot and dry, and very humid and cold require special cooling and heating solutions for which earlier liquid desiccant systems are not optimized.

**[0010]** At very high temperatures (>100F) and very low humidity (<20%RH), liquid desiccant systems can't operate efficiently and need special controls to avoid crystallization of the desiccant. Traditional evaporative cooling systems do well at low humidities and moderate cooling requirements, but are unable to deal with extreme heat or with more humid conditions that tend to occur at least part of the time in most locations.

**[0011]** Traditional cooling systems use refrigerant coils that are air cooled and are best suited for sensible cooling. Condense forming on the coil acts as an insulator that reduces its capacity. Thus multiple coils need to be used in series to fully dehumidify and cool the air. Four and six row coils are commonly used. Still, traditional systems often cannot handle the full latent load without significantly overcooling the air and then reheating it, or mixing high volumes of return air with small volumes of outside air to minimize the humidity level of the mix. Especially in times where only a small amount of sensible cooling is required, humidity control is compromised. When compressors are cycled to manage smaller loads, bursts of humidity enter the building as coils warm up and evaporate condense back into the air. Many split systems provide heating by operating as a reversible heat pump system. The liquid desiccant system is able to cool outside air without frost forming, significantly improving system efficiency by reducing or eliminating defrost cycles. The removal of humidity from the outside air also enables the humidification of the heated space, maintaining healthy RH levels between 30-60% RH. These tend to be most useful in moderate climates where cooling and heating loads are roughly in balance. Very cold climates like the Midwest and Northeast of the US still require additional heating, often from natural gas or oil. In more moderate climates, heat pump effectiveness is limited by humidity, which can lead to frost forming and the use of very inefficient defrost cycles. Using a liquid desiccant condenser coil prevents frost forming in a heat pump system.

**[0012]** Liquid desiccants can achieve effective dehumidification at higher temperatures of the compressors evaporator, during the cooling cycle. The regenerator fully rejects the condenser energy at lower temperatures than traditional air cooled systems. As a result, the compressor can move energy from the conditioned space to outside the space at a much lower temperature differential than traditional systems. This improves the efficiency of the compressor in proportion to the reduction in the temperature difference. The lower temperature difference between the condenser and evaporator is the lift of the compressor and drives the efficiency of the combination of compressor-based cooling and heating with liquid desiccant heat exchangers.

**[0013]** As disclosed in U.S. Patent No. 9243810, by actively diluting the desiccant, e.g., by using vapor transfer membrane modules, a liquid desiccant system can increase the ratio of sensible cooling to latent cooling. It can even starts to act like a direct evaporative cooler, which allows it to maintain a target minimum dewpoint (DP) under very dry conditions, without maintaining dry bulb (DB) targets at high wetbulb (WB) conditions, something traditional direct evaporators cannot do. While the conditioner operates at cooler temperatures the overall temperature differential over the compressor tends to be further reduced due to a much larger reduction in condenser temperatures, as diluted desiccant increases evaporation at the condenser transforming it into a de facto water cooled air conditioning system with comparable efficiency but with significantly reduced water consumption.

**[0014]** Existing control strategies for air conditioners can rely on bandwidth control, adaptive control or predictive control, they need to be modified for the various configurations of liquid desiccant systems and then optimized for each of the operating modes described before.

**[0015]** Additional building humidity “guidelines” are being developed to encourage maximum, and sometimes even minimum, humidity levels mostly driven by health considerations, especially the impact on respiratory disease and allergies.

**[0016]** In dry climates, water cooled chillers and evaporative coolers use the evaporation energy of water to cool spaces and/or improve compressor efficiency, but this uses potable water in substantial volumes. Managing the scaling effects and biological pollution of such water is a significant challenge. In locations where both humid and dry conditions occur, evaporative chillers are less effective. Standard liquid

desiccant solutions do not operate well under those conditions. We will disclose how water addition can be controlled while simplifying liquid desiccant systems, making them competitive in both dry and humid conditions. We will also disclose that using vapor transfer modules in liquid desiccant systems reduces water consumption significantly over the life time of an installation when compared to traditional evaporative coolers, a critical consideration in many climate zones.

**[0017]** Many buildings have to deal with a variety of conditions from very hot and dry to relatively cool and humid including high DP/high relative humidity (RH) and high DB/low DP design points. We will disclose how liquid desiccant systems can be controlled to handle these conditions effectively. This includes the use of a combined liquid desiccant system with direct evaporation of the air supplied to the regenerator and indirect evaporative cooling of supply air after dehumidification. Both significantly improve system performance in dry and hot climates. They are identical in their effect on the system to direct dilution of the liquid desiccant by adding demineralized water to the liquid desiccant tank or using membrane modules to transfer water from a feedstream to the highly concentrated liquid desiccant.

**[0018]** Since the concentration in the liquid desiccant system drives the RH of both the supply conditions in the conditioner as well as the output conditions of the regenerator, liquid desiccant systems operate as “constant RH machines.” This allows for different control strategies, including controls based RH and WB targets and measurements rather than DB and DP. These can be simpler and more effective than traditional control in independently controlling humidity and temperature.

**[0019]** Measuring the concentration can be done in a variety of ways. We will disclose how this can be used for different control strategies, using a combination of RH of the regenerator exhaust and the conditioner supply, tank levels, electrical resistance, defraction, specific weight of the desiccant, and measuring concentration based on viscosity and temperature.

**[0020]** Desiccant dilution through vapor transfer or forward osmosis can be done in a variety of ways.

**[0021]** Controlling for crystallization uses empirical data on crystallization points at different temperatures and humidity levels. Preventing crystallization which stop the system’s ability to manage humidity is critical. And since crystallization only occurs at

conditions that are too dry for comfort, avoiding those conditions tend to improve supply conditions that most benefit building occupants.

**[0022]** Energy recovery is a major factor in managing air quality and the efficiency of air conditioning systems. We will disclose how this can impact control strategies.

**[0023]** Concentrated liquid desiccants are a very efficient form of energy storage with more kWh per lb. than ice. We will disclose how control strategies can make optimal use of regeneration capacity due to waste heat, solar or other heat resources.

**[0024]** Frost control is a critical consideration in any heatpump system. Liquid desiccant systems can avoid them by proper sizing of system components, improving overall system efficiency. Frost prevention strategies will be disclosed

**[0025]** Liquid desiccant systems have distinct modes depending on the relationship between input conditions and target conditions. We will discuss the main modes, their impact on bandwidth/adaptive systems as well as predictive systems. Including transition between modes without flip flopping.

## **SUMMARY**

**[0026]** In accordance with one or more embodiments, a method is disclosed of operating a liquid desiccant air-conditioning system to maintain target temperature and humidity level in a space. The liquid desiccant air conditioning system comprises: (a) a conditioner for treating a first air stream flowing therethrough and provided to the space as a supplied air stream, said conditioner using a heat transfer fluid and a liquid desiccant to treat the first air stream; (b) a device for measuring temperature and a device for measuring humidity in the supplied air stream; (c) a regenerator connected to the conditioner such that the liquid desiccant can be circulated between the regenerator and the conditioner, the regenerator causing the liquid desiccant to desorb water vapor to a second air stream or to absorb water vapor from the second air stream depending on a selected mode of operation of the system; (d) a refrigerant system; (e) a first refrigerant-to-heat transfer fluid heat exchanger connected to the conditioner and the refrigerant system for exchanging heat between the refrigerant heated or cooled by the refrigerant system and the heat transfer fluid used in the conditioner; (f) a second refrigerant-to-heat transfer fluid heat exchanger connected to the regenerator and the

refrigerant system for exchanging heat between the refrigerant heated or cooled by the refrigerant system and the heat transfer fluid used in the regenerator; and (g) a system controller for controlling operation of the system. The method comprises the steps of:

- (i) measuring the temperature and humidity level in the supplied air stream;
- (ii) comparing the temperature measured in (a) to a target temperature to determining a temperature error, and comparing the humidity level measured in (a) to a target humidity level to determine a humidity error;
- (iii) comparing the humidity error and the temperature error on a common scale to determine the greater error;
- (iv) using the greater error to drive the system controller to control operation of the system to reduce the greater error;
- (v) repeat (i) through (iv) a plurality of times.

#### **BRIEF DESCRIPTION OF THE FIGURES**

**[0027]** FIG. 1 is a simplified diagram illustrating an exemplary 3-way liquid desiccant air conditioning system using a chiller or external heating or cooling sources.

**[0028]** FIG. 2 illustrates an exemplary single membrane plate in a liquid desiccant air conditioning system.

**[0029]** FIG. 3 shows the basic design of the membrane plate of FIG. 2.

**[0030]** FIG. 4A shows a basic version of a liquid desiccant air conditioning system in cooling mode.

**[0031]** FIG. 4B shows a wide range of options to enhance RH control in a liquid desiccant system.

**[0032]** FIG. 5 shows the psychrometric chart for the basic DOAS concentration with limited independent control of humidity and temperature.

**[0033]** FIG. 6 show an a system with all controls in the refrigerant system in cooling mode.

**[0034]** FIG. 7A shows a heat pump system for cooling and heating with refrigerant coils and one dual coil in cooling mode for hot and dry conditions.

**[0035]** FIG. 7B shows a heat pump system for cooling and heating with water cooled coils and a reversible compressor.

**[0036]** FIG. 8 shows a multi zone building with ceiling units, a DOAS unit and separate regeneration units.

**[0037]** FIG. 9 shows how in a split system a mid-unit can be used to improve control over pressure and flow in the heat transfer and desiccant channels for multi-floor and larger buildings.

**[0038]** FIG. 10 shows how in a multi zone building multiple tanks with different concentrations of desiccants can be used to optimize controls, including managing spaces with different humidity loads and target supply conditions, but also for rapid response to changing conditions and for energy conservations and peak shaving.

**[0039]** FIG. 11 shows multiple options for adding water to liquid desiccant including evaporator pads, vapor transfer units, forward osmosis systems and direct addition of demineralized water to a tank.

**[0040]** FIG. 12 shows a liquid desiccant energy recovery solution.

**[0041]** FIG. 13A shows how different configurations are driven by the change in temperature and humidity between input and target conditions. Eight zones are identified. System configurations are driven by the weather map for the location and the supply target.

**[0042]** FIG. 13B shows how typical south eastern US conditions are sub-tropical with mostly hot and humid during the middle of the day with cooler conditions at night and in the morning. Conditions in the south west of the US (e.g., Phoenix) represent a continental climate with conditions ranging from very hot and dry to hot and humid to very cold.

**[0043]** FIG. 13C shows how energy recovery reduces the range of input conditions, simplifying system requirements.

**[0044]** FIG. 14A shows how the basic configuration of a liquid desiccant system operates in different modes.

**[0045]** FIG. 14B shows water addition control settings for the different operating zones in the psychrometric chart.

**[0046]** FIG. 14C shows how a second cooled coil linked to the second heat exchanger operates in different operating zones in the psychrometric chart.

**[0047]** FIG. 14D shows how a dual use air cooled coil operates in different operating zones in the psychrometric chart.

**[0048]** FIG. 15A shows target supply conditions for a typical air conditioning systems with significant sensible and latent loads.

**[0049]** FIG. 15B shows how economizer modes in a direct outside air system depend on the function of the other components of the HVAC system.

**[0050]** FIG. 16 shows actual performance data of a liquid desiccant system with water addition.

**[0051]** FIG. 17 shows the main climate zones in the US as well as the annual temperature and humidity data for major US metropolitan areas.

**[0052]** FIG. 18 shows how different HVAC system reach a given humidity condition.

**[0053]** FIGS. 19A-19F shows how a system with a single evaporator air cooled coil can optimize all 4 920 conditions.

**[0054]** FIG. 20 shows how evaporative cooling and water addition can be used to manage the same conditions.

**[0055]** FIG. 21 shows how indirect or direct evaporative systems cannot match the supply conditions of a liquid desiccant system with water addition or combined with direct evaporative cooling.

**[0056]** FIG. 22 gives an overview of three primary heating modes.

**[0057]** FIG. 23A shows where a system without air cooled coils can outperform alternative systems.

**[0058]** FIG. 23B shows how a system with an air cooled “heat dump” coil can outperform alternative systems.

**[0059]** FIG. 23C shows how a system with water addition and an evaporator air cooled coil can outperform alternative systems.

**[0060]** FIG. 24 shows a psychrometric chart for T& RH sensors and a typical target range

**[0061]** FIG. 25 shows a psychrometric chart for a standard DOAS control system

**[0062]** FIG. 26 shows an advanced liquid desiccant control system to manage capacity and humidity directly

**[0063]** FIG. 27 shows how water addition is not required for a DOAS system that only controls for a maximum DP.

**[0064]** FIG. 28 shows a liquid desiccant tank level can be used to control concentration and thus RH.

**[0065]** FIG. 29 shows how such an adaptive algorithm can work.

**[0066]** FIG. 30A describes the liquid desiccant air conditioning control logic for an adaptive system.

**[0067]** FIG. 30B advanced adaptive controls using the largest error in humidity and temperature to from humidity and temperature.

**[0068]** FIG. 31 shows a control logic for the method of FIG. 30.

**[0069]** FIG. 32A shows a basic adaptive control structure using RH and DB.

**[0070]** FIG. 32B provides a more detailed description of the system of FIG. 32A.

**[0071]** FIG. 32C show the inclusion of some predictive controls.

**[0072]** FIG. 33 shows building and system sensors.

**[0073]** FIG. 34A shows the crystallization curve of a desiccant (LiCl).

**[0074]** FIG. 34B shows the implications of the crystallization curve of a desiccant (LiCl) for different operating modes and climates.

**[0075]** FIG. 35 shows an approach to crystallization alarms.

**[0076]** FIG. 36 shows how viscosity changes in a liquid desiccant system between conditioner and regenerator.

#### **DETAILED DESCRIPTION**

**[0077]** FIG. 1 depicts a new type of liquid desiccant system, as described in more detail in U.S. Patent No. 9243810, which is incorporated by reference herein. A conditioner 101 comprises a set of plate structures that are internally hollow. A cold heat transfer fluid is generated in cold source 107 and entered into the plates. Liquid desiccant solution at 114 runs down the outer surface of each of the plates. The liquid desiccant runs behind a thin membrane that is located between the air flow and the surface of the plates. Outside air at 103 is blown through the set of (wavy) conditioner plates. The liquid desiccant on the surface of the plates attracts the water vapor in the air flow and the cooling water inside the plates helps to inhibit the air temperature from rising. The treated air at 104 is put into a building space.

**[0078]** The liquid desiccant is collected at the bottom of the wavy conditioner plates at 111 and is transported through a heat exchanger 113, to the top of the regenerator 102, and to point 115 where the liquid desiccant is distributed across the wavy plates of the regenerator. Return air, or optionally outside air, at 105 is blown across the regenerator plate and water vapor is transported from the liquid desiccant into the leaving air stream at 106. An optional heat source 108 provides the driving force for the regeneration. The hot transfer fluid at 110 from the heat source can be put inside the wavy plates of the regenerator similar to the cold heat transfer fluid on the conditioner. Again, the liquid desiccant is collected at the bottom of the wavy plates 102 without the need for either a collection pan or bath, so the regenerator the air flow can be horizontal or vertical. An optional heat pump 116 can be used to provide cooling and heating of the liquid desiccant. It is also possible to connect a heat pump between the cold source 107 and the hot source 108, which is pumping heat from the cooling fluids rather than the desiccant.

**[0079]** FIG. 2 describes a 3-way heat exchanger as described in further detail in U.S. Patent No. 9308490, which is incorporated by reference herein. A liquid desiccant enters the structure through ports 204 and is directed behind a series of membranes as described in FIG. 1. The liquid desiccant is collected and removed through ports 205. A cooling or heating fluid is provided through ports 206 and runs counter to the air stream at 201 inside the hollow plate structures, again as described in FIG. 1 and in more detail in FIG. 3. The cooling or heating fluids exit through ports 207. The treated air at 202 is directed to a space in a building or is exhausted as the case may be. The figure illustrates a 3-way heat exchanger in which the air and heat transfer fluid are in a primarily vertical orientation. It is however also possible to flow the air and the heat transfer fluid in a horizontal aspect, which is not fundamental to the operation of the system.

**[0080]** FIG. 3 describes a 3-way heat exchanger as described in more detail in U.S. Patent No. 9631848, which is incorporated by reference herein. The air stream at 351 flows counter to a cooling fluid stream at 354. Membranes 352 contain a liquid desiccant at 353 that is flowing along the wall 355 that contains a heat transfer fluid at 354. Water vapor at 356 entrained in the air stream is able to transition through the membrane 352 and is absorbed into the liquid desiccant at 353. The heat of condensation of water at 358 that is released during the absorption is conducted

through the wall 355 into the heat transfer fluid at 354. Sensible heat at 357 from the air stream is also conducted through the membrane 352, liquid desiccant at 353 and wall 355 into the heat transfer fluid at 354.

**[0081]** FIG. 4A shows a liquid desiccant system with the basic configuration. FIG. 4B shows a system with additional means to regulate humidity independently from temperature as well for minimizing energy consumption. Conditioner 416 uses liquid desiccant 425 and hot water 418 to process a mixture of outside and return air 403 to supply air conditions at 419. The diluted liquid desiccant 420 is returned to a tank and then via a heat exchanger 413 to regenerator 423 where hot water 440 and a mixture of outside and exhaust air 406 is used to reconcentrate the liquid desiccant 420 to 425, while humidifying and heating exhaust air 406. The cold water 418 and hot water are supplied by evaporator coil 427 and condenser coil 428 of compressor 417 using pumps 460 and 461, respectively.

**[0082]** FIG. 4B shows additional coils and water addition options. For example, to reduce the concentration of the regenerated liquid desiccant part of the condenser heat can be diverted to air cooled coil 429 using outside air 430. To reduce the conditioner load, the process air 403 can be precooled using exhaust air 431 and heat exchanger 432. A second air cooled coil 433 is shown, which can be used to supply some of the cooling load by cooling exhaust air 406 from regenerator 423. Other coils 434 and 435 can be used to recool or post cool process air. The concentration of the liquid desiccant can be reduced using water addition modules 436, which can be positioned at different locations in the liquid desiccant loop 436a injects demineralized water 437a directly into the tank. 436b uses the heat from the regenerator 423 to maximize the efficiency of the heat transfer in a vapor unit. 436c is set up to further dilute the liquid desiccant before entering the regenerator thereby reducing the temperature of the heat transfer fluid 440 to condenser coil 428. This reduces lift and thus improves the efficiency of compressor 416. Water injection in 436D gives direct control over the RH of the supply air RH 404.

**[0083]** A valve system 441, 442, 443 and 444 is shown in the refrigerant circuit. For those skilled in the art it will be clear that a similar valve system in the heat transfer fluid circuit can be used to control the air cooled coils

**[0084]** The following fundamental approaches can be used to manage humidity and temperature independently:

1. Total cooling or heating capacity defined as the change in enthalpy of the system is driven by the capacity of the compressor system 416.
2. The liquid desiccant panels will use the available capacity to generate air with an RH that is significantly lower than the outside air used for regeneration. Fluid flows 418 and 440 for heat transfer fluids and 404 and 415 for liquid desiccant through the panels of 417 and 423 (air/heat transfer fluid/desiccant) can significantly adjust the ratio of latent versus sensible cooling.
3. Adding a sensible coil 429 to the condenser side of the compressor in parallel or in series to the regenerator 423 enables additional sensible cooling.
4. Adding a sensible coil 433 to the evaporator side 427 of the compressor with outside air or air from the regenerator 423 provides additional cooling power for deep dehumidification while maintaining or increasing the air temperature.
5. Adding a sensible coil 434 before the conditioner on the evaporator side of the compressor and after the conditioner on the airside maximizes sensible capacity.
6. Desiccant dilution 436 allows more sensible cooling. It can be used for net humidification of supply air and makes it possible to control humidity in a limited bandwidth.
7. Preconditioning the air 405 with direct evaporation has an identical effect to desiccant dilution, but uses existing components.
8. Exhaust air 431 can be used on the conditioner 427 to reduce the overall load, either sensible (Plate heat exchanger (HX)) 432 or sensible and latent (full enthalpy HX including wheels, plates, and LD plates).
9. Exhaust air 431 can be used at the regenerator 423 to provide deeper dehumidification.

**[0085]** The ability of the system in FIG. 4B to manage temperature and humidity independently is constrained by the need to balance the compressor. Therefore the cooling power available at the evaporator 427 is always less than the heat available for regeneration at 428. The difference results from the heat and friction losses in the compressor. Typically regeneration power is 20 to 30% higher than the conditioner power.

**[0086]** FIG. 5 shows the performance of the system in 4A. The psychrometric chart shows how starting from condition 501a, b and c, the conditioner will dehumidify and cool to condition 502 a, b, and c. The cooling energy 510a (delta enthalpy times airflow needs to balance with the regeneration energy 510b (delta enthalpy times the regeneration airflow), with the friction/heat losses available as extra regenerator power. The concentration at the conditioner and the concentration of LD at the regenerator are in balance with a 1-2% difference which is driven by the change in concentration within a panel at a given condition. As a result, the RH 541 at the regenerator and the RH at the conditioner have to be in balance with corrections for that small difference in concentration. The humidity absorbed 511a at the conditioner and the humidity desorbed 511b by the regenerator has to be in balance. The resulting outside air condition, supply conditions and regenerator out conditions are shown in FIG. 5. For condition A, a target condition 502A at 50% RH and can be reached, but for conditions B and C this may not be possible. Increasing regenerator air and water flows will change supply conditions for 501C. For example a higher regenerator flow 503 will result in a lower concentration of the LD and thus a higher supply RH 504. The wet bulb condition being the same this will result in a higher DP and lower DB. A lower water flow at the regenerator can have the opposite effect.

**[0087]** For greater flexibility in managing temperature and humidity independently, additional components can be added to the system (FIG. 4B) to improve control over concentration and thus over humidity. For example, to achieve a target supply condition similar to 502A from input condition 501c requires a lower concentration of liquid desiccant and more work. This can be achieved in several ways, among other by rejecting condenser heat through an air cooled coil 429, by direct dilution 436 or through the use of an evaporator coil 499 before the regenerator.

**[0088]** FIGS. 6, 7A, and 7B show several ways in which the liquid desiccant concentration can be changed, including direction and flow of the refrigerant and the heat transfer fluid through valves 617, 618, 619 in FIG. 6 and valves 717, 718, 762 and 763 in FIGS. 7A and 7B. Also, desiccant and heat transfer fluid flow rates can be set by pumps 609, 653, and 655 in FIG. 6 and 753, 755, 709 and 743 in FIGS. 7A and 7B with more flow through the regenerator and less through the conditioner increasing concentration in cooling mode. Airflow rates are driven by fans 602 and 642 and damper

660 in FIG. 6 and 702, 742 and damper 760 in FIGS. 7A and 7B, with concentrations becoming lower as more air is driven through coils 722, 733. Water addition at 652 and 752, 758 is the most direct way of reducing the concentration of the liquid desiccant. It also allows maintaining the concentration of liquid desiccant over a wide range of conditions. Since the size of the tank is driven by the ratio of the highest and lowest concentration used in the system, greater control over the concentration reduces the size required for the tank. Leading to a direct trade off in size and weight between liquid desiccant storage and concentration management components.

**[0089]** FIG. 6 shows how the additional coils 622 and 671 are on the refrigerant circuit. Heating and cooling mode are also realized by switching the evaporator and condenser 620 and 614 with four way switch 617. This results in a complex refrigerant circuit with accumulator 615b and 618b as well as a three way adjustable valve 618 and 619. Three expansion valves are shown at 624, 638 and 639. Airflows can be adjusted with fans 602 and 642 as well as the damper for 646B and a fan for airflow 672. Direct dilution 652 of liquid desiccant through module 657 can control concentration by controlling the flow rate at 652. Direct addition is also possible in tanks. Using outside air or exhaust air for 641 and 646B will significantly impact results. Using available exhaust air to preprocess 601 through energy recovery can improve efficiency. As does post processing of air 606 prior to it entering the space e.g. through an indirect evaporative cooler. All these options have been described in prior art. Several combinations give full flexibility of achieving target conditions for a full range of outside air conditions. Optimizing controls requires new approaches that use the dehumidification capability of the system and its capacity to achieve target conditions in a single step in conditioner 603.

**[0090]** The main challenge for the system shown in FIG. 6 is controlling the quality of refrigerant in the system and thus system performance over a full range of conditions.

**[0091]** FIG. 7A shows a simplification of the refrigerant circuit by eliminating coil 671 in FIG. 6 and using instead 722 as a dual fluid coil with water and refrigerant. The water circuit is set up to allow 722 to fulfill the function of 671 but using water rather than refrigerant. This avoids the need for a parallel refrigerant circuit on the condenser

side, which can be critical in a reversible system. The detailed operations are described in U.S. Patent No. 10024558.

**[0092]** FIG. 7B shows a comparable solution with a much simplified heat pump system with two refrigerant to heat transfer fluid heat exchanger 714 and 720. Four way switch reverses the desiccant flow. This type of heat pump is well understood and has few challenges. The functions of humidity control coils 672 and 622 in FIG. 6 are now replaced by 733 in combination with damper 760. A properly sized coil 733 can be used in heating and cooling mode, either as a condenser coil to reject heat from 720, which reduces the energy available for regeneration in 748 and thus reduces the concentration of liquid desiccant 752 coming out of the regenerator 748. A lower concentration of liquid desiccant will lead to a higher RH of air 706 leaving conditioner 703, thus shifting the air supply towards more sensible and less latent cooling. This “heat dump” function of coil 733 is critical when conditions shift from hot and humid to hot and dry. During cool and humid condition coil 733 is used as an additional evaporator side coil, providing an additional load to the compressor 715, which provides additional energy for regeneration to 748, increasing the concentration of the liquid desiccant 752. This mode is shown in FIG. 7B with switches 765, 762, 763 and 764 set such that conditioner 703 and coil 733 run in parallel. By diverting cooling fluid from conditioner 703 while increasing the concentration of LD 707 entering the conditioner 703, the air 706 is dehumidified while the DB temperature will start to rise as the flow of heat transfer fluid 704 goes to zero and all heat transfer fluid is diverted by valve 765 to 766.

**[0093]** For a system that needs to be able to deal with a very broad range of conditions a simplified refrigerant circuit would be preferred with more of the adjustment in the system configuration being done on the heat transfer fluid side. That does involve an efficiency loss driven by the efficiency of the LCE 714 and the LCC 720. But it eliminates the need for multiple refrigerant switches, multiple expansion valves and their controls and for additional design to balance refrigerant and oil with receivers and accumulators.

**[0094]** Managing the various coils on the refrigerant side of the system becomes more difficult as the system gets more complex. A multi zone system is shown in FIG. 8. To cool spaces 810 with desiccant conditioners 821 in low ceiling space 816 using a mixture of return air 817 and outside air 809 from DOAS to supply air 818 to the space.

The units 808 get cooling water from chiller 814 through piping 812 and 813. Liquid desiccant is regenerated in 801 using the condenser heat of chiller 814. The regenerator 801 can also be positioned close to available exhaust air to further improve the effectiveness of regeneration. The concentrated liquid desiccant is supplied via 802 and 805 to units 808 and 806. And returns via 803 and 804. When return air 817 is dry and the building latent load is fully covered by deeply drying air 809 by outside air unit 806, than units 808 can be sensible only cooling solutions, including highly efficient sensible solutions like chilled beams. Controlling such a system with a VRF \poses challenges in managing the refrigerant. Instead a chiller solution in combination of liquid desiccants heat exchangers and with heat transfer/water connections is more efficient and can be easier to install, maintain and operate.

**[0095]** FIG. 9 shows a different solution for residential or small commercial split system. Flexibility in locating various components of the system can simplify the system and improve performance depending on the building and its requirements. The configuration shown has multiple indoor units 903 inside the conditioned space. The conditioned spaces can be on different floors in the building and at different distances from the regenerator unit. Managing the flows of heat transfer fluid 904 and liquid desiccant 907 can be simplified by strategically locating tanks 910, 954, desiccant pumps 909, 954, 953, heat exchanger 956, and water pumps 913/944. In general the refrigerant system with heat exchangers 914, 920 and chiller 915 as well as the regenerator 948 and fan 947 the various air cooled coils like 922 will be located outside, however tanks and pumps as well as other features like the water addition module 0957 can be located in appropriate technical spaces in the home, where water and/or drains and/or power is available, where maintenance can be done easily and where there is sufficient space for tanks and components. Noise from the pumps can also be a consideration.

**[0096]** The heat pump system as shown in FIG. 9 is relatively complex. In practice, simpler versions of the refrigerant system can be used.

**[0097]** Alternatively the air coils 922 and 671 can be connected indirectly to the compressor via the first and second refrigerant to heat transfer fluid coils 914 and 920. This significantly simplifies the refrigerant circuit but requires additional valves in the heat transfer fluid system.

**[0098]** For example, the task of controlling the system becomes complex if target conditions and loads of the conditioned spaces differ significantly. Some of the potential issues include :

**[0099]** Spaces with different input conditions of outside and return air to the conditioner. Some spaces may have exhaust air for regeneration others may not. Also the temperature and humidity of return air from the spaces may differ and of course the proportion of outside air required for the space, which often depends on occupancy and potential requirements for over pressurization.

**[00100]** Different loads in the space including high humidity loads from plants, pools, kitchens and people and high sensible loads from outside walls in older buildings, lights, equipment etc.

**[00101]** Different targets, e.g., in stores high humidity is desirable in green /veggie sections and low humidity is desirable in refrigerant sections. As a result the required load per cfm and the required concentration of liquid desiccant for matching user requirements could differ significantly.

**[00102]** Liquid desiccant control systems need to be able to address this by adjusting the water temperature and the liquid desiccant concentration supplied to a specific space. This may require a more complex tank system that allows the regenerator to adjust the concentration of the liquid desiccant by using different airflows and a mix of outside and exhaust air. Another option for creating multiple concentrations of liquid desiccant is to vary the temperatures and flows rates of heat transfer fluids supplied to the regenerator and the air cooled coils discussed above.

**[00103]** FIG. 10A shows how multiple conditioners C1 through Cn (1009a, 1010a, 1011A) provide different conditions air (1009 through 1011) to spaces 1-N (1009c, 1010c and 1011c). Water flows 1003 through 1008 vary the supply of heat transfer fluid to the conditioners from evaporator coil 1001. The condenser coil 1002 is connected via heat transfer fluid flows 1020 through 1025 to conditions R1 through Rn 1026a, 1028a 1030a. Less exhaust air and higher overall airflows 1026, 1028 and 1030 will decrease the concentration of the regenerated liquid desiccant which can then be stored in one or more tanks shown as D1b, D2b and DnB. A lower water temperature provided to the regenerator further reduces the concentration of the liquid desiccant by adjusting airflows 1025, 1027 and 1029 to air cooled coils SC1, SCV2 through SCn. Obviously such a

complex system requires an appropriate building and system control system that uses these variables to ensure that target conditions in spaces or zones 1-n can be maintained with minimal effort by compressor 1000.

**[00104]** Referring to FIG. 11, direct dilution of liquid desiccant gives the most direct independent control of humidity and temperature supplied to the space to be air conditioned, while optimizing the efficiency of refrigerant system 1199. Direct dilution is especially important in hot and dry conditions when over drying of the air is not desirable and/or where the desiccant could be diluted until it crystalizes. It also helps maintain temperatures in the regenerator 1198 within the operational boundaries of the materials used in the regenerator. Direct dilution of the liquid desiccant can be done in a number of locations as shown in FIG. 11, including in tanks 1112/1154, in the lines to and from the conditioner 1157/1158 and as an integral part of heat exchanger 1156. As disclosed in U.S. Patent No. 9308490, dilution of desiccant can be done with a vapor transfer membrane unit 1100 or with a forward osmosis membrane unit, using feed streams that can use either potable water or water with a lower ionic content than the LiCl solution. This includes seawater. The vapor transition modules 1157 with feed stream 1158 use the high ionic content of liquid desiccant to drive the vapor transfer. The rate of water transfer is driven by the temperature of the liquid desiccant e.g. liquid desiccant coming from the regenerator. It is therefore energy efficient and avoids any addition of minerals or loss of liquid desiccant. Alternatively, a forward osmosis can be used to remove minerals in a similar unit. Alternatively, demineralized water can be added directly to tank 1110. Control of the volume of water added to the liquid desiccant can be done directly by controlling the tank level or by varying the feed flows to the water addition membrane modules.

**[00105]** Direct dilution ensures that the RH level of supply air 1106 cannot fall below a minimum level  $RH_{min}$  which can be calculated from the LD concentration LD % by a formula  $RH_{min} = (100\% - V \times LD\%) + \text{effectiveness factor}$ , where V is a factor driven by the vapor pressure of the liquid desiccant For LiCl V is about 2. The effectiveness factor is driven by the sensible and latent effectiveness of the panel and tends to be between 5-10% on average for cooling LD at a concentration of 25% will result in conditioned air at an RH of 55-60%. Minimum RH levels are critical for a wide range of applications. Optimal living and working conditions tend to have an RH between 40 to 70%. Maintaining a concentration of 20-30% ensures that those humidity conditions are always met. These results are based on

extensive modelling of liquid desiccant systems over a broad range of conditions as well as tests of liquid desiccant panels.

**[00106]** An alternative desiccant dilution method is shown in 1100. Placing a direct evaporative pad or cooler in the incoming air stream of the regenerator or the conditioner also dilutes the liquid desiccant. A nozzle that creates a fine mist in the incoming air stream that quickly vaporizes has the same effect. During very dry conditions outside air conditions 1101 a conditioner 1103 using fan 1102 can supply cool air 1106 with a humidity at or just below a target DP. In this situation the conditioner 1103 can actually desorb humidity from the liquid desiccant, increasing rather than decreasing the concentration and partially cooling the air. Evaporator coil 1100 in airstream 1146 humidifies and cools to a high DP. And a low temperature, The Regenerator 1148 will therefore absorb rather than desorb water vapor, diluting the liquid desiccant. In such extreme conditions the conditioner 1199 has a significantly lower load from evaporator 1114 which only has to provide the remaining sensible cooling to achieve target conditions. Therefore the heat load from condenser 1144 is low which results in a cool regenerator 1148 that absorbs water vapor, dehumidifying air 1146 to 1149 thus diluting the concentrated liquid desiccant from heat exchanger 1156 and pump 1153. Positioning an evaporator pad 1100 in airstream 1101 the same effect, but exposes the airstream to the conditioned space 1106 to the evaporator pad, while regenerator air 1149 is exhausted. The overall effect of humidifying the regenerator air has the same effect as directly diluting the liquid desiccant. Instead of adding water to the desiccant through a vapor module, the water is added indirectly via the air. The advantage of this approach is that evaporative pads are cheap and well understood. The water management including managing water quality and mineral content with appropriate bleed streams will differ depending on location and water conditions. Suppliers of evaporative coolers are familiar with the water management issues. The cost of pads is currently lower than that of vapor transition modules. From a control perspective these two mechanisms require similar solutions.

**[00107]** Controlling water addition in liquid desiccant systems can be done in a number of ways described below: by increasing the feed stream either cyclically or through a variable speed pump, maintaining direct tank level control with a level sensor, adjusting flow rates of feed water to the evaporator etc.

**[00108]** Often solid desiccant wheels are used to recover energy from an exhaust air stream. The same can be done with a “passive” liquid desiccant systems. FIG. 12 discloses how the liquid desiccant panels 703 and 704 can be used as an alternative for a full enthalpy desiccant wheel with comparable efficiency. Instead of using two different technologies two sets of panels are used. The main set of panels 702 and 903 conditions air 706 at 702 to supply air 101 and regenerates the desiccant 902 at 903 using hot and cold water 704 and 708 returning it via 705 and 709 to the evaporator and condenser of a separate chiller. The liquid desiccant 714 used to condition the air is pumped from tank 712 through the heat exchanger 718 to 702. The diluted liquid desiccant 902 is pumped by 901 to the regenerator 903 where the condenser heat 708 and the outside air 102 regenerate the liquid desiccant with humid air 707 leaving the unit.

**[00109]** A second set of panels does not use a compressor system or an external source of heat or cold. Instead it preconditions the incoming air 706, with water 801 and desiccant 717 which is pumped with 716 from tank 715. Panels 704 uses dry and cool exhaust air 102 to regenerate the diluted desiccant with the dry exhaust air 102, Water 802 has been warmed up by the absorption in 703 and is cooled by the low temperature of exhaust air 102. The desiccant regenerated in 903 uses the standard components including a liquid desiccant tank 712, a heat exchanger 718, and pumps 713 and 901. The tank allows for different concentrations of liquid desiccant as the exhaust air volume and outside air temperature and humidity change.

**[00110]** The liquid desiccant ERV in FIG. 12 uses additional panels in 703 and 704 for energy recovery, but since the humidity and latent loads for the panels 702 are significantly lowered by the energy recover process, fewer panels can be used for conditioner 702 and/or regenerator 903, resulting in cost effective solutions.

**[00111]** The prior art shows the following fundamental tools to manage humidity and temperature independently in the liquid desiccant systems described above.

1. Total cooling or heating is driven by the capacity of the compressor system.
2. The liquid desiccant panels will use the available cooling capacity to condition a combination of outside air, return air or air pre-conditioned by an energy recovery device or an evaporator unit. Higher airflows and/or higher heat transfer fluid flows through the regenerator panels can increase the ratio of sensible versus total cooling

or the sensible heat ratio (SHR). Low liquid desiccant flows can further increase the SHR.

3. Adding an air cooled coil to the condenser side of the compressor in parallel or in series with the regenerator on the airside enables additional sensible cooling.
4. Adding an air cooled coil to the evaporator side of the compressor that processes outside air or air from the regenerator provides additional cooling power for deep dehumidification in the conditioner increases latent cooling, until latent cooling is larger than the total cooling capacity resulting in heating of the air. If the air cooled coil provides all the load to the regenerator, the conditioner will dehumidify adiabatically, resulting in a negative SHR.
5. Adding a sensible coil before the conditioner on the evaporator side of the compressor and after the conditioner on the airside maximizes sensible capacity.
6. Desiccant dilution allows more sensible cooling and less dehumidification or even net humidification of supply air. Significantly increase the relative humidity of the supply air, while reducing the temperature resulting in an  $SHR > 1$ .
7. Maintaining a minimum level of liquid desiccant with demineralized water maintains a minimum RH level, e.g. 30% with LD of about 35% concentration. Supply conditions can exceed this minimum RH level, based on input and ambient air conditions and fluid flows through the coils.
8. Preconditioning the air with direct evaporation has an identical effect to desiccant dilution, but uses existing components.
9. Exhaust air can be used on the conditioner to reduce the overall load, either sensible (Plate HX) or sensible and latent (full enthalpy HX incl. wheels, plates, and LD plates).
10. Exhaust air with a lower RH than ambient conditions can be used at the regenerator to provide more efficient dehumidification, since less condenser heat is required to maintain a high concentration of liquid desiccant.

**[00112]** Air-cooled coils and liquid desiccant heat exchangers can be connected either directly via the heat transfer fluid system or indirectly via the liquid cooled refrigerant heat exchangers when the air cooled coils condition air directly with refrigerant.

**[00113]** The above focusses on the use of fluid flow rates, sensible coils, desiccant dilution and exhaust air for independent management of latent and sensible cooling. It shall

be clear to those skilled in the art that same options can be used for independent control of temperature and humidity with the refrigerant system in heating mode.

**[00114]** FIG. 13A shows the main modes of modes of operation in the psychrometric chart. The starting and target conditions are typical for a DOAS system. The zones are defined relative to a target supply condition 1699. Systems with recirculation have similar requirements but with different target conditions.

- a. Cooling and deep dehumidification is a typical requirement for many air conditioning systems in hot and humid climates (4).
- b. Dehumidification with deep sensible cooling (3) requires maintaining or increasing the relative humidity typically during the heat of the day.
- c. Warming the air while dehumidifying (zone 5, 6) is a requirement with very humid but cool air typically in the early morning and in spring or fall in moderate zones.
- d. Humidification during cooling ( 1, 2 ) can be a benefit during cooling of very dry air, e.g. desert air at a DP below 40F with a temperature of over 35C. Typically this is a requirement in areas where liquid desiccant systems are essential for monsoon period (1604) but where part of the year or even day can be very hot and dry.
- e. In heating mode the main distinction is between zone 7 where high humidity can lead to frost forming in heat pump mode and to over humidification of the space and zone 8 where humidity is too low and additional humidification is required.
- f. Cool and dry air needs no conditioning and the system can operate in economy mode. Other than traditional systems, liquid desiccant solutions do have the option to humidify the air while maintaining dry bulb conditions. In essence that is a heating operation since the enthalpy of the air increases.

**[00115]** Economizer mode 1699b is a set of conditions where further processing will not occur and the air is supplied to the space as is.

**[00116]** In DOAS applications, the economizer zone 1699b can include zones 1 and 2 if the liquid desiccant unit is used as a highly efficient dehumidifier with an SHR that optimizes compressor performance. This maximizes overall system savings if highly efficient

sensible-only systems are available for further cooling of the air, including but not limited to geothermal cooling and indirect evaporative cooling.

**[00117]** FIG. 13A shows the key characteristics of the 8 zones in relation to the desired target condition. In this disclosure, cooling refers to a decrease in DB and/or in WB condition and heating to an increase of DB and WB condition.

**[00118]** Dehumidification refers to a reduction in absolute humidity and/or relative RH. While humidification refer to an increase in absolute humidity.

**[00119]** FIG. 13B shows how in much of the eastern sea coast 1691 the cooling requirements for outside air are mostly in zones 4, 5 and 6, while in the southwest (Phoenix) 1692 conditions can differ from extremely hot and dry to hot and humid To cool and humid, covering the psychrometric zones 1 through 6. The eastern sea coast requires a relatively simple system with either only desiccant components or a single sensible coil on the evaporator side of the compressor and in series with the regenerator. Efficient operation in the south west requires some form of desiccant dilution /water addition.

**[00120]** FIG. 13C shows how the availability of exhaust air at condition 1693 reduces the range of conditions to be managed. With target condition 1699, exhaust air conditions 1693 and ambient condition 1695, 50% ERV would reduce the conditions “seen” by the system to a narrower envelop 1694. This allows for smaller sensible coils to be used for zone 2, 3 and may make an evaporator side coil for 5 and 6 unnecessary.

**[00121]** FIG. 4 showed how temperature and humidity can be managed independently using a number of coils. The following fundamental approaches can be used to manage humidity and temperature independently:

**[00122]** FIGS. 16A-16C show how fluid flow rates, sensible coils, desiccant dilution and exhaust air can be used for independent management of latent and sensible cooling. The same options can also be used for independent control of sensible and latent heating.

**[00123]** FIGS. 14A-14D summarize various control options and how they related to temperature and humidity control by describing how in each of the 8 zones of FIG. 13, heat exchangers and water additions should be used to achieve target conditions. It also shows when using these coils is not effective.

**[00124]** FIG. 14A shows a configuration with an air cooled coil connected to the evaporator side of the refrigerant system. The options with that coil off shows where a base unit without any coils or water addition can operate effectively.

**[00125]** FIG. 14B shows how dilution of the liquid desiccant can be used to further improve operations especially in zones 1, 2, 3, and 8. The degree of dilution in 3 and 8 depends on the humidity loads in the space.

**[00126]** FIG. 14C shows how a sensible coil connected to the condenser side of the refrigerant system can improve operations in the 8 zones.

**[00127]** FIG. 14D shows how a single air cooled coil can be used to manage the zones.

**[00128]** FIGS. 14A-14D show how the various coils are used in the main modes with HX1 being the refrigerant to water heat exchanger 1701, HX 2 the refrigerant to water HX 1702 and , HX 3 the air cooled coil in series with the regenerator 1743. HX 4 is the air-cooled coil 1770, which cools the condenser using outside air. HX 3 and HX4 can be combined but that requires a damper between the regenerator 1048 and air cooled coil 1043 to shift coil 1743 from exhaust air from the regenerator in mode 5/6 to outside air in mode 2, 3. It also requires either a dual fluid coil or a valve system to switch the coil from running on the evaporator side to being part of the condenser side of the refrigerant system.

**[00129]** FIG. 14B shows in Table 14B for each of the 8 zones of FIG. 6 disclosing how the air cooled coils and water addition can be used to achieve target conditions. The system shown in FIG. 14 has fan 1702 supplying air 1701 to conditioner 1703. The conditioner is cooled with heat transfer fluid 1704 from evaporator coil 1705 or HX1. Pump 1713 circulates the heat transfer fluid 1704 from the conditioner back to the refrigerant to water HX 1705. The desiccant 1707 is pumped by 1709 from tank 1710 to the conditioner 1703. The diluted desiccant 1708 returns to the tanks after processing supply air 1706. The diluted liquid desiccant 1711 is heated in heat exchanger 1756 before being supplied to the regenerator 1748 through pump 1753 as 1745. The concentrated liquid desiccant 1752 is returned via an optional high concentration tank 1754 by pump 1755 via heat exchanger 1756 where it is cooled by 1711 and returns to the tank 1710 as 1712.

**[00130]** Humidity control is driven by the temperature of the water 1744 and the air as 1746 as well as the humidity of the air 1746 which is supplied by fan 1747 to regenerator 1748. Pump 1740 rotates heat transfer fluid 1744 through refrigerant to water heat exchanger HX2 (1771) to regenerator 1748. Lower flows of water 1744 or lower flows of air 1746 result in a warmer regenerator and thus in more concentrated liquid desiccant which will result in dryer supply air at 1706. Air cooled coil HX3 1743 cools the hot and humid air

from regenerator 1748, providing a load to the compressor 1799 to maintain a high temperature at the condenser 1771 in order to concentrate the liquid desiccant 1745. This is used during times where outside air 1701 is humid requiring dehumidification but cool, requiring sensible heating.

**[00131]** Heat dump HX4 (1770) runs in series or in parallel with liquid condenser coil 1771, reducing the heat available for regeneration and thus enabling the conditioner to cool 1708 more deeply without over-drying it. HX 4 can be directly connected to the condenser 9 or via the refrigerant to water HX 2 1771 (link 8)

**[00132]** Water addition option 5 either with the membrane module 1757 or in a desiccant tank 171710 is critical in zones 1 and 2 of FIG. 14, and can make a major contribution in zones 603 and 608. With water addition, liquid desiccant systems become among the most competitive solutions for any supply condition, with comparable performance better control over supply conditions and using less water than existing evaporative system Evaporator Coil HX 6 at 1759 dilutes the desiccant indirectly by increasing the humidity of the incoming airstream at the regenerator resulting in significant absorption of vapor at regeneration 1748 which allows conditioner 1703 to operate at least partially as an evaporative cooler.

**[00133]** FIGS. 14A-14D show how the various coils are used in the main modes with HX1 being the refrigerant to water heat exchanger 1701, HX 2 the refrigerant to water HX 1702 and HX 3 the air cooled coil in series or in parallel with the regenerator 1743. HX 4 is the air cooled coil 1770 which cools the condenser using outside air. HX 5 and 6 describe the setting for the direct water addition and air humidification.

**[00134]** FIG. 14D shows how HX 3 and HX4 can be combined but that requires a damper 1710 between the regenerator 1048 and air cooled coil 1043 to shift coil 1743 from exhaust air from the regenerator in mode 5/6 to outside air in mode 2, 3. It also requires either a dual fluid coil or a valve system to switch the coil HX3 (1743) from a direct or indirect (8) connection to the evaporator side of the compressor to a direct or indirect via HX2 connection 9 on the condenser side of the compressor

**[00135]** For those skilled in the art it will be clear that the solutions shown in FIGS. 14A-D are not intended to be limitative. For example in zone 6, the conditioner 1706 will operate adiabatically to minimize cooling under conditions that already have an air enthalpy below that of the target. By accepting a somewhat dryer air condition, DB target conditions

can be realized without reversing the system. Coil 3 provides the extra cooling load needed to maintain the high concentration of liquid desiccant at 1752. Applying direct heat to heat transfer fluid 1744 going into regenerator 1748 is potentially a simpler but less efficient alternative. The solutions in the tables in FIG. 14 focus on maximizing system efficiency in term of EER and MRE, however at a cost in complexity and equipment costs. Zones 1 and 2 can also be served by using (in)direct evaporative coolers after the conditioner. Similarly zone 7 and 8 could be served by using gas, electric or waste heat sources when available. FIG. 14 does not seek to describe the impact of energy recovery from exhaust air supplied to the regenerator 1748 or the conditioner 1703 or to both. For those skilled in the art such combinations are clear.

**[00136]** FIG. 11 shows how the performance of the optimized liquid desiccant system outstrips that of existing dedicated outside air system by 50% or more.

**[00137]** FIG. 15A shows typical target supply conditions for heating and cooling systems on the Psychrometric chart in comparison to the standard cooling and heating comfort zones.

**[00138]** The target conditions are those supplied by the system to a space, which enable the space to maintain comfortable conditions. Most air conditioners supply air 1903 at a cooler and dryer condition than the comfort conditions 1902 in the space. This compensates for the sensible and latent loads due to occupancy, heat infiltration and the load of outside air. As described in ASHRAE's DOAS design guide, direct outside air systems can either supply room neutral conditions (1902) or ensure that all latent needs of the space are met (1904), plus at least some of the sensible cooling compared to input air condition 1901, with the remainder 1905 being cooled by sensible heat only coils, cold beam system etc.

**[00139]** Target condition 1906 is in a heating mode with sensible and humidity loads, i.e. warmer than comfort conditions.

**[00140]** Comfort zone conditions will be realized in the conditioned space after cooling and heating loads have been added at a given circulation rate.

**[00141]** The fan only or economizer mode is used during conditions at which no active air processing is required and the unit operates in economizer mode.

**[00142]** Economizer modes are a requirement for DOAS systems. ASHRAE recommends that the DOAS unit focusses on meeting the latent load requirements by

bringing the air to a required DP condition. In general that condition will be lower than the target humidity, since the building will have latent loads, which need to be compensated for. The economizer mode 1910 of a DX system is limited to outside air that is already at target 1912 DP and need no further dehumidification. Otherwise the DX system needs to overcool and then reheat the air. A solid desiccant system will dry the air at temperatures significantly higher than comfort levels, leaving the full load to be carried by the DX system that recirculate the air.

**[00143]** Liquid desiccant air conditioning systems dehumidify and cool simultaneously. With sufficient water addition/evaporative cooling capacity they can reach any supply condition from any input condition 1911. In that case the economizer mode is limited to the comfort zone conditions. Without water addition LDAC systems maintain a maximum DP.

**[00144]** Typical recirculation systems have an economizer zone with a maximum but no minimum humidity level. This is a problem, since low humidity levels can be harmful for health reasons, building quality and some kind of equipment. LDAC units can be combined with water addition or evaporation to maintain both minimum and maximum humidity levels.

**[00145]** We will propose how such conditions can be managed simultaneously using adaptive control systems without stability issues caused by conflicts between proportional–integral–derivative controller (PID) controller loops.

**[00146]** FIG. 16 shows actual performance data for a liquid desiccant system with water addition during a single day of operation. The water addition maintained only a minimum desiccant level in a tank, 2001 shows actual outside air conditions for a single day with 15 minutes intervals. 2002 shows how a maximum humidity and temperature are maintained during a these rapidly changing outside air conditions using a very simple desiccant dilution control that sets a maximum concentration level through a minimum tank level control. 2002 shows the regenerator exhaust conditions that correspond to the supply conditions 2003. These conditions were extreme and can be found in only a few locations like the red sea and the Arabian Gulf.

**[00147]** FIG. 17 shows the annual weather bin data for a number of major US cities. Apart from Phoenix all of these have RH levels above 30%. They correspond mostly to zones 4, 5, 6 and 7 of FIG 19.

**[00148]** FIG. 18 shows how the liquid desiccant system described here is able to achieve target conditions in a single step. This has major implication for the system control method which needs to control both temperature and (relative) humidity. Standard DX cooling systems first cool the air until the required humidity level is reached and then reheat the air to achieve the desired sensible condition. That means that each subsystem requires only a single control variable. For example in DOAS system the airflow requirements are driven by the ventilation needs of the space. The DP target is achieved by measuring air temperature of the air coming of the condensing coil until its equal to the target DP condition at 2202. A second system is then used to reheat the air to the target condition 2203. Systems using desiccant wheels have a similar two step approach. First the air is dehydrated to the target DP 2204. Then in a second step, it is cooled to the target condition. Most existing desiccant systems including spray liquid desiccant systems heat the air up less than the solid desiccant wheel, however they still required a second cooling coil to get air to the target DB condition. The liquid desiccant system described above cools and dehumidifies in a single step from start condition 2201 to target condition 2203. That requires a control that monitors both temperature and humidity and manages both compressor power and humidity controls.

**[00149]** FIG. 19A shows how a liquid desiccant system with a compressor is balanced in an equilibrium situation with an outside air system. The same control principals apply to systems using a mixture of outside air and return or exhaust air. Given input condition 2301 at the conditioner and regenerator, the total enthalpy difference 2204a between 2301 and the supply air 2302 needs to be identical to the heat rejected 2304b from the condenser to the air to regenerator exhaust air 2303 minus a correction factor for the heat and friction losses of the compressor (about 20-30% depending on conditions, turn down etc.). This is shown in FIG. 19A for equal air flows at the conditioner and the regenerator. This is of course true for any compressor driven system. However, typical for a single step liquid desiccant system is that two other variable have to be in balance as well to achieve a stable situation:

- a. The humidity absorbed at the conditioner 2305a needs to be identical to the humidity evaporated at the regenerator 2305b

b. The RH at the conditioner and the RH at the regenerator are both in balance with LiCL at a concentration that is nearly the same. Typically when the liquid desiccant has a concentration of 25% going into the conditioner it will come out at a concentration between 23 and 24% and will return to 25% at the regenerator. As a result the conditioners RH is likely to be 2 to 4% higher than the RH at the regen for the vapor pressures in the air to be equal to the partial vapor pressure at the surface of the liquid desiccant. The RH of the system is 1-2x concentration ensuring the RH of the conditioner and regenerator are closely correlated within 5%.

**[00150]** Such a system will always supply air at a concentration somewhat below that of the input condition, given that the energy available at the regenerator is always larger than the cooling power at the conditioner. Increasing the regenerator airflow will reduce the supply temperature at 2306b, which increases the RH of 2306B compared to 2304b. This will reduce the concentration of the liquid desiccant. Since the total cooling power remains the same, the WB condition will not change, but supply conditions will shift to a cooler, more humid condition 2306a. Increasing compressor power without changing the airflows will lead to a supply condition at a lower WB condition, but also to a lower RH at the regenerator and thus to deeper dehumidification at 2307a.

**[00151]** FIG. 19B shows the effect of adding sensible coils at the condenser side of the system. It allows for a greater delta enthalpy at the regenerator 2304B for a given regen out RH condition at 2303 and removed humidity of only 2305b. The lower rejection of humidity leads to a more humid, cooler supply condition 2302 for input condition 2301.

**[00152]** FIG. 19C shows the effects of three types of coils 2310. It shows the supply condition with equal airflows over regen and conditioner. Regenerator air out is than 2311. Adding a heat dump coil 2304 in parallel with the regenerator reduces the regenerator temperature to 2302 and increases the RH. This results in the lower concentration that causes supply conditions 2301. A heat dump fan 2303 in line with 2302 on the air side increases condenser temperature. It creates the same conditions 2301 but at a lower efficiency because the total lift increases from 23C (2330) to about 27C (2331). The fan 2322 in line with the regenerator, but connected to the evaporator has the net effect of increasing the condenser temperature and thus the concentration. This creates a supply condition that is warmer and dryer. This shows how evaporator coil 2322, heatdump coil

2304 using outside air and the same heat dump coil 2203 but using regen exhaust air can all be used to change the supply conditions.

**[00153]** FIG. 19D shows that using how the moisture removal efficiency or MRE is optimized when air is supplied at 2310. 2301 requires the unit to do additional work as the heat dump fan increases the RH of the regeneration exhaust air to 2302. While total work increases the additional work is done at a similar lift 2331 as both the condenser and regenerator temperatures decrease improving the overall efficiency (EER/COP). Using the heatdump fan to provide extra load for increased dehumidification and a lower RH increases the total load without increasing the amount of moisture removed. As a result, the moisture removal efficiency drops. The lift 2331 is again comparable to 2330 as both the supply temperature and the exhaust air temperature from the regenerator increase. Overall efficiency is reduced as the compressor does about the same load, but the total cooling effect is lower. In other words, a liquid desiccant unit has the highest MRE/moisture removal efficiency when it is used without coils. Using the heatdump coil increases EER/COP but at a lower MRE. Using the advanced dehumidification coil reduces both. The advanced dehumidification coil provides an advantage only when cool and humid conditions require a combination of latent heating and increased temperature. In that case the reduction in enthalpy because of the increase in temperature can be considered useful work. This is especially true when the building has high internal latent loads because of occupancy, pools, plants etc. The 920 standard recognizes this by penalizing a unit for supplying air at a temperature less than 70F. As a result the ISMRE at 70Fdb and 55Fdp conditions improves significantly when the advanced dehumidification coil is used. This is especially important for the selection of unit in maritime climates where cool and humid conditions requiring dehumidification together with heating is critical.

**[00154]** Ongoing discussion about the standard recognizes that in buildings with low latent loads and high sensible loads, e.g., from lights, supplying outside air at temperatures below comfort conditions could be justified. When selecting dehumidifiers for hot and humid climates the MRE may be more important than the ISMRE.

**[00155]** FIG. 19E show how the DB conditions can be managed independently from the target 55 DP using air cooled coils for each of the four 920 conditions a (2304), b (2344), C ( 2354) and D 2364 Using an air cooled coil in series with the regenerator allow the regenerator to take 920D air and regenerate it to 2365 enabling the adiabatic

dehumidification of supply of air 2364 to target condition 2361 at the same RH level. The same can be done for 920B 2354. 920D can supply condition 2311 with appropriate fluid flows in the regenerator. However, 920A will supply air at 2310, reflecting the regenerator performance without heat dump coil.

**[00156]** FIG. 19E shows the benefit of using an the air-cooled evaporator advanced dehumidification coil in line with the regenerator. 2365 is the input air condition of the coil, which cools it to 2366. This can be done without a need to remove condensation from the coil, which can be a significant benefit, especially if the unit is positioned inside a building with the outside air ducted in. It reduces maintenance and avoids any risk of water damage. The total lift of this combination is low. Alternatively, the coil could be used before the regenerator, providing it with dryer air 2371. The dryer air would allow the regenerator to regenerate at a lower temperature 2372. The lift over the regenerator is similar, suggesting a similar efficiency, however the additional condense removal required nullifies one of the key advantages of liquid desiccant systems that there is no need for condense management.

**[00157]** FIG 23F shows how air-cooled coils can be used to meet the supply air target of 70F DB and f55F DP (2310) for the 920 A, B and C conditions. For 920D, air is supplied at 70F adiabatically. In other words, the dewpoint while the wetbulb condition is the same as 920D 2360. FIG. 19F assumes that the advanced dehumidification coil uses outside air and that it is not in line with the regenerator. The effect is that adiabatic dehumidification is realized but with a higher lift then in FIG. 19E, and thus a lower efficiency. Also condense needs to be managed. But it does allow for a single air cooled coil to function as conditioner to running in parallel to the regenerator as is shown in FIG. 13. Damper 660 in FIG. 6 is a way to gain the efficiency benefits of FIG. 19E while only using a single air cooled coil. This makes such a system smaller, lighter and cheaper than a dual coil system, which is especially important when cool and humid conditions are rare. It also allows the coil to be sized for the larger "heatdump" load, which makes advanced dehumidification even more effective.

**[00158]** Still cooling outside air to 2359 and 2379 will lead to significant condensation on the coil while providing the additional load needed to reconcentrate desiccant for 920 D and C. 920B has been shown to balance to the 2310 supply condition without additional airflow over the sensible coil. 920A can match the 70/55 2310 condition by using the coil as a heat dump (2309)increasing the RH from 2310 in FIG. 19E to 2311 in FIG. 19F. It will be understood by those skilled in the art that this further increases the

flexibility by which humidity and temperature can be managed independently using sensible coils.

**[00159]** FIG 24 shows the impact of adding water to dilute the liquid desiccant as shown in FIG. 11, either through a vapor transfer unit as described in U.S. Patent No. 9308490, which is included by reference, through direct addition to a water tank or through evaporative pads as discussed in U.S. Patent Application No. 62/580270. In both cases the effect is more diluted Liquid desiccant and thus a higher DP for the supply condition. Diluted liquid desiccant can take very dry air just above the crystallization point of liquid desiccant (2400) and supply air 2401 at the same or even a higher DP supply target 2410. The regenerator will be evaporating strongly 2402 to exhaust air at 2411 maintaining and RH of about 50% RH e.g. using LiCl with a concentration of about 25%. The concentrations for other desiccants will differ.

**[00160]** Using evaporators 2403 prior to both the conditioner and the regenerator (2403) will provide both with input conditions 2413 at an RH of 75% or higher. The conditioner 2404 and the regenerator 2405 will be able to achieve the same supply and exhaust conditions 2410 and 2411 assuming a similar concentration of about 25% LiCl.

**[00161]** FIG. 20 shows how the system can be used to achieve humidity and temperature conditions independently for a wide variety of conditions. In particular the very dry and hot condition 2400 which is cooled and humidified 2401 with the regenerator conditions 2402 as described above. The same condition can be achieved using an evaporator pad 2403. From the dry but less hot condition 2420, the same system can adiabatically humidify and cool using either water dilution in combination with the conditioner or using a separate evaporator pad. While 2420 can be achieved using any direct evaporative cooler and 2410 in FIG. 20 using an indirect evaporative cooler, only a liquid desiccant system can handle both conditions while avoiding over humidification (2400) or overly dry air (2420). For locations like phoenix such independent control of humidity and temperature ensures that conditions can be managed year round with high efficiency and minimal water consumption. The same system can even be used to handle the cool and extremely humid condition 2430, which can occur during early morning conditions in the Phoenix wet season. They can dry the air again to 50% RH 2431, by regenerating the liquid desiccant at 2432, albeit with the support of an air cooled coil providing additional load at 2433. Now often multiple systems are needed and operators

may need to switch in the early morning and late evening between different systems. The controls described below seek to ensure that optimal conditions can be maintained around the clock. FIG 20 showed how conditions in a single location can change throughout the day from hot and humid to very hot and dry.

**[00162]** FIG. 21 shows the actual performance of an LDAC system for a range of outside air conditions and the target conditions 2500 the system was able to maintain. It compares that with the performance of two alternative systems. First an indirect evaporator cooler can be used to condition 2501 but how it will leave conditions too humid in 2502, for example. Conditions that are too humid. A direct evaporative cooler cannot achieve conditions 2500 from the same outside air conditions. Instead it can either supply air that is much too hot at 2504 or cool to 2503 but at too high humidity. For condition 2505 direct evaporative cooling is unsuitable. The Liquid desiccant system already demonstrated how conditions 2500 can be achieved reliably for the conditions shown in FIG. 21, with water addition as shown in FIG. 20. With more advanced control an even narrower range can be achieved.

**[00163]** FIG. 22 shows heating without frost forming by first dehumidifying and heating air adiabatically from 2601 to 2602 using the regenerator coil and then using the sensible coil to cool the dry air to 2603 with a DB condition that is still higher than the DP. When conditions are very dry outside as in 2604, water addition may be needed to achieve that target humidity and temperature and temperature 2605. In that case the regenerator outside should be bypassed. At cool but humid temperatures like 2610 where heating is required inside, the sensible coil can again be used with regenerator turned down or off. This will result in significant condensation at 2611 and to sensible heating only 2612 to 2605. Like above these heating examples are not limitative, but show how a very broad range of conditions can be achieved with independent control over humidity and temperature using a combination of a sensible coil, a valve system and/or desiccant dilution/water addition.

**[00164]** FIG. 23B shows how a DOAS system in the basic configuration of FIG. 4A can still operate under a broad range of input conditions. However it does results in a somewhat broader “extended” set of target conditions 2710. For the starting conditions 2711 the target T can be achieved and at performance will be significantly superior to existing DX and

solid desiccant wheel systems. For conditions 1712 the target condition T cannot be achieved and the range of supply conditions 2710 will be broader.

**[00165]** FIG. 23C shows how a DOAS system with the heat dump coil 429 in FIG. 4B has a narrower range of supply conditions 2710 and a broader set of starting conditions 2711 where those conditions can be maintained with superior performance to DX and solid desiccant wheel solutions.

**[00166]** FIG. 23C shows how a system with evaporative cooling and water addition has superior performance at all conditions and can compete directly with water cooled chillers and evaporative coolers in terms of efficiency and overall system cost.

**[00167]** FIG. 24 shows how typically temperature and relative humidity are measured. A widely used sensor in HVAC systems is the T&RH sensor. It measures dry bulb conditions and relative humidity. Controlling the system with DB and RH while possible, poses challenges since the conditions are not orthogonal, therefore RH conditions can be achieved at much too high DB conditions and DB can be achieved at very high or very low RH conditions, which tends to make controls less stable.

**[00168]** FIG. 25 shows how in most HVAC systems T&RH measurements are translated into a measurement of temperature (DB) and of absolute humidity (DP, grains/lb. etc. They are orthogonal which makes control more intuitive and direct. Simple bandwidth control than seeks to control supply conditions by sequentially achieving DP (3001a/b) and DB (3002 a/b) conditions. This is typically done by first achieving DP by either cooling the DX coil until the DP condition is achieved or by using the desiccant wheel to heat and dehumidify the air. A second step is needed to achieve DB supply conditions either by post cooling or post heating the process air using a separate coil.

**[00169]** FIG. 25 shows a typical control of HVAC system through a DB band 3002A to 3002B and a DP target band between 3001a and 3001B.

**[00170]** While traditional control sequences can also be used for such a system using bands of DB and DP conditions, the liquid desiccant also allows for an alternative approach show in FIG 31. Here the DB and RH are used to calculate the actual WB supply condition. The target WB condition 3101 shows the total amount of cooling required from the starting condition 3100 and shows the total compressor power required to get to the target condition 3104. At a given setting for the humidity controls the compressor will cool to

condition 3103. The humidity controls can be used to move the supply air from 3103 to 3104. Humidity controls can include:

- Condenser linked sensible coil or heat dump to reduce desiccant concentration and increase RH
- Evaporator linked sensible coil or advanced dehumidification coil to increase desiccant concentration and reduce RH
- Water addition or evaporative cooling pads to directly dilute the liquid desiccant and increase RH
- Increase regenerator airflow to cool the condenser at a lower temperature which reduces the concentration and increases RH
- Reduce water flow in the regenerator to increase the heat transfer fluid temperature and the condenser temperature which increases concentration and reduces RH
- Reducing heat transfer fluid flows at the conditioner to cool the air adiabatically and increase the DB temperature and reduce the sensible heat ratio

**[00171]** For those skilled in the art it will be clear that these examples are not limitative. Other options have been discussed including a post cooling coil linked to the evaporator to shift increase SHR.

**[00172]** Heating can be done similarly by allowing independent control of the compressor for overall heating effort and the humidity controls and frost prevention controls using the sensible coils, fluid flows and water addition/desiccant dilution.

**[00173]** FIG. 27 shows an alternative control strategy for DOAS systems with a DP target. That allows for a much broader range of conditions to be provided by a DOAS without additional sensible coils or water addition for conditions 3301, 3302 and 3303. A simple bandwidth control will work. Regenerator Fluid can be kept high to maximize conditioner performance. Only at very dry conditions 3304 and very wet condition 3305 is it than desirable to use additional humidity control coils, first by turning down the fluid flows

in the regenerator. Air cooled coils could be used in addition to prevent crystallization 3304 and overly humid air 3305

**[00174]** FIG. 28 shows an alternative approach to measuring and controlling the desiccant concentration by using the tank level 1401 rather than an RH driven approach. If the system aims to maintain an RH of 50% a concentration of about 25% needs to be maintained. That corresponds to a given level 3401 in the tank 3400. The tank level can be measured directly through a level sensor 3402. As the concentration of LD increases, the tank level drops. That can be done directly through dilution if the tank level fall below target. A solenoid valve 3403 triggered by a floater 3404 is a simple way of achieving this. As tank levels increase other controls are triggered to maintain the target level by increasing latent regeneration. As explained earlier additional loads from a sensible coil can be used to generate more condenser heat to maintain liquid desiccant concentration levels. Alternatively lower regenerator airflows and lower water flows will sensible coils need to be used to concentrate the desiccant, e.g. through the advanced dehumidification coil regenerator water and airflows.

**[00175]** In addition to using tank levels, desiccant concentration can be measured in a variety of ways. A simple measure is the RH out of the regenerator or conditioner. For LiCl the percentage RH is approximately  $1-2 \times \text{concentration LD}$ . Direct sensors are another options however the cost reliability and accuracy of these sensors needs to be further demonstrated. Multiple sensors in a system allow for more accurate supply conditions, more effective predictive controls with faster response times and system diagnosis e.g. detection of small leaks based on readings of different sensors.

**[00176]** Liquid desiccant system with RH sensors in the regenerator exhaust, where the RH is used to calculate the LD desiccant concentration using a formula based on a combination of basic physics (equal vapor pressures at equal RH at any temperature) and system know how (Latent effectiveness of the panels).  $1-X1(LD) \times C-LD + \Delta(\text{system}) = RH$ , where  $X1(LD)$  is determined by the type of liquid desiccant used (ca 2-2.1 for LiCl) and Delta is driven by the system dimension (1-5%).

**[00177]** A Liquid desiccant system with a ventricle (narrowing of the pipe) can measure concentration based on pressure drop over the ventricle at a given flow rate.

**[00178]** Specific weight can be used by using a floater with known specific weight in the LD, In that case the depth of the floaters determines the concentration.

**[00179]** Diffraction uses the changing optical properties of the LD to measure the diffraction of a known frequency laser beam to measure LD concentration.

**[00180]** Electrical resistance uses changing electrical properties of the LD to determine concentration. However for LiCl resistance plateaus in the critical 20-30% range which requires this measure to be used in combination with other system properties, e.g. RH out.

**[00181]** Alternative adaptive control algorithms that focus on the specific properties of the single step liquid desiccant cooling and dehumidification system include therefore the enthalpy /RH control, the tank level control. A potential problem with a single step adaptive control system is that controlling multiple variables at the same time can lead to conflicting PIC loops. This can lead to oscillations of the system control. A way to get to a single PID loop is to first determine whether temperature or humidity level is further from target. Most system measure Temperature and RH. The controller can covert that to enthalpy and absolute humidity conditions, i.e., WB and DP or RH. The system PID loop can approach the target conditions for temperature and humidity by determining which of the two variables is furthest of target. Whether that is the delta of actual versus target DB or WB or the target versus actual DP or RH.

**[00182]** FIG. 29 shows how such an adaptive algorithm can work. Given a relatively high WB condition with an RH already close to target (3501), the compressor first adjust total enthalpy until the delta in RH (DP) becomes larger than the delta in the WB (DB) condition 3502. A combination of water addition, heat dump/sensible coil fan, heat transfer flows or regenerator fan (in order of impact) can then be used to adjust the humidity level until the RH/DP is closer to target than the WB 3503 and then the compressor is adjusted 3504.

**[00183]** FIG. 30A shows how humidity DP or RH target 3601 and DB target 3605 can be used to calculate WB and RH targets 3620 and 3621. These are compared to the calculated measured WB 3622 based on the measured DB 3603 and RH 3605. The measured RH can be compared directly to the calculated RH target 3621 to be used by the controller to calculate the RH error 3624. The largest of the WB/ enthalpy delta 3623 and the RH delta 3624 is used to control the compressor through an adaptive PID loop. The PID loop it drives compressor settings 3606, which in turn drives the regenerator and conditioner flows 3611. A variable conditioner airflow will determine the relationship between the compressor

settings and the regenerator and conditioner fluid flows 3611. The RH or DP delta 3624 will drive the SHR setting 3630. This can be either the air cooled condenser coil or heat dump fluid flows and possible damper setting 3612 OR the water addition flows 3614 OR the advanced dehumidification (evaporator air cooled coil) flows 3613 OR evaporator coil settings (water flow/air. Multiple RH or DP and temperature settings can be used to drive crystallization settings 3630.

**[00184]** A similar structure can be used for the adaptive control method shown in FIG. 29, where the largest error of WB/ enthalpy and RH drives the PID loop. Or through a band with control where a WB bandwidth is used to control the compressor, while the RH is controls the humidity or SHR controls 3611 through 3615.

**[00185]** FIG 36B shows a similar process flow, but now with the errors in DB and DP driving the system, where the largest error again drives the capacity control and the humidity measure (DP) drives the SHR controls

**[00186]** Here the crystallization control 3631 is shown to be driven by the target DP and measured RH values for conditioner and regenerator.

**[00187]** FIG. 31 shows a simplified adaptive control logic based on actually implemented systems with DP and DB target. The actual DP 3703 is calculated from actual Temperature 3705 and the measured RH by the controller. Actual are compared to the DP target 3701 and the DB target 3502. Resulting in a DP error 3704 and a DB error 3705. The largest error controls the PID loop 3708, which controls compressor speed, and regenerator fluid flows. The humidity measure (DP) directly controls the air cooled coil airflows or water addition levels. Airflows are the primary fluid control. Hot water and cold water flows are driven by the compressor to match capacity. During very dry conditions or when the desiccant is over diluted, desiccant flows to the regenerator or conditioner can be turned off allowing the system to recover.

**[00188]** Such an approach can use any combination of WB/ RH/DB and DP to control the system. However WB/ RH, and DB/RH and DP controls are particularly important. The former can be used in a bandwidth controller. DB/RH is directly based on typical user settings. DP control is important when the Liquid desiccant unit is used as a dehumidifier with separate sensible cooling capacity.

**[00189]** FIG. 32A shows how such a system can work using a DB 3802 and DP setting 3801. These two can be used to calculate a target RH 3806, which when compared to actual RH in the supply air 3703 gives a RH error 3807.

**[00190]** The Actual DB condition in the supply airstream 3805 can be compared to the target 3802 to calculate DB error 3808. The controller 3809 identifies the larger of the two errors and uses that to drive the Compressor setting 3810. The regenerator fan speed 3810 is adjusted based on the RH error to approach the DB condition as much as possible than the Actual RH has to be used in combination with the actual DB to calculate the actual DP and thus the DP error. The largest error is again the driver of the compressor through the PID. While the RH error can adjust the regenerator fan to get closer to the target humidity condition. If the DB target is translated in an RH at the target DP, then DP/RH can be used, with RH driven by fluid controls for the various coils and the regenerator and the DP kept on target with the compressor speed.

**[00191]** FIG. 32B shows a more complete view of these controls using DB and RH error to control including protection against crystallization and algorithms that use input versus target conditions to set the right system configuration settings or system “value settings”. It also shows how Liquid desiccant dilution settings, heat transfer fluid flows and heat dump/sensible coil fan settings are alternatives to reduce the DP /RH error. User setting can be DP (3701), DB (705

**[00192]** FIG. 32B shows some of the key control options that can be used to optimize liquid desiccant performance depending on key considerations like overall efficiency, response time, system complexity and cost. It includes a crystallization controls 3812, high water temperature protection 3813 to ensure that the system doesn’t overheat, tank level controls to ensure that the tanks don’t overflow or dry out. Several of these can be used to adjust RH targets 3806 to maintain a single PID loop control. Direct control for of the compressor speed is necessary when the safety controls come within the safety margin. Full control that will require strong humidity control capabilities such as liquid desiccant dilution rates or heat dump fan (regenerator air cooled fan) settings.

**[00193]** The conditioner fan in can be variable or fixed but is set independently of the other variables within an allowed bandwidth. Especially in outside air unit, conditioner airflow 3818 should not be used to control conditions. This is of course different in recirculation units where airflow is one of the controls of the PID controller.

**[00194]** Input conditions to the conditioner and regenerator can be compared to target 3820 and can be used to accelerate the adaptive controls 3821. This can also be used as the main control esp. in outside air systems. The controller also has to automatically adjust valve settings 3821. In combination with ambient condition forecast or supply requirement forecast the controller can be optimized to anticipate future demand and capacity (Ambient air temperature and humidity) for dehumidification.

**[00195]** Controls for liquid desiccant systems are focused on humidity management. Using temperature as the primary driver can be necessary where the unit is the only option. However in that case additional sensible cooling capacity needs to be added. A coil in the supply air from the conditioner to the space heated by the condenser (hot gas reheat) is an effective alternative to the condenser coil and has comparable efficiency. Both the advanced dehumidification coil and the hot gas reheat coil operate at low compressor lifts during cool and humid conditions. Both provide additional load to the compressor respectively for concentrating the liquid desiccant and for overcooling air and then reheating it. The choice will be driven by the cpst of the more complex refrigerant systems with another hot gas coil and by the benefits that the advanced dehumidification coils have in heating mode, in particular the frost free heating avoiding or significantly reducing defrost cycles which is made possible by the additional coil.

**[00196]** FIG. 32C shows how latent 3850 and capacity 3851 control are driven by respectively the error in the actual humidity measures RH or DP (3807) and the largest of the two errors 3806 \*humidity and Temperature /energy level of the air (DB/WB) which drives the compressor speed which in turns drives the hot and cold water flows and the regenerator fan, Predictive algorithms can be integrated in these algorithms improve overall system efficiency in particular hourly, daily, weekly and annual information on weather and occupancy and load requirements.

**[00197]** Such continuous learning and improvement can be integrated in the system. Alternatively it requires monitoring of field systems to optimize controls based on actual usage. It also requires an ability to upgrade software controls remotely. This is a critical requirement for liquid desiccant systems. Users are not familiar with the capabilities of these systems and are used to the limitations of the traditional two step control systems. Remote monitoring and control will accelerate learning from the user community and

improve performance and acceptance of the systems. Wireless connections enable high reliability usage of the system.

**[00198]** Remote monitoring of the input and output conditions as well as the actual settings of the system is also a primary driver of preventive maintenance where deviations from historical performance indicate potential problems with system components.

**[00199]** Increasingly manufacturers use their pool of customers to learn what gives the best performance. An adaptive liquid desiccant system will combine direct feedback from field units with app based feedback from users. This can use a combination of smart algorithms with "wisdom of the crowd" based selection among those algorithms. For this to be useful a variety of settings for the units is needed. For example, humidity could be more of a problem when people are active or when they are trying to sleep than when they sit and relax. That could influence how much humidity levels fluctuate and how fast the system needs to respond. There are many crowd based systems that allow gathering that kind of info. Combining these systems with info on liquid desiccant air-conditioning solutions with a focus on humidity will lead to rapid acceptance of such systems.

**[00200]** Predictive controls shown in 38C can take different forms.

**[00201]** Predictive controls based on the existing detailed system models which can use input conditions to set compressor and humidity controls to quickly move to target

**[00202]** Learning models that actively perturbs (within limits) unit settings to measure effect on efficiency and learns the combinations of operating parameters that work best to minimize energy usage.

**[00203]** Predictive models are particularly important when there are large cycles in humidity during the day, or predictable swings in requirements (weekends) or time of year. For example a model can how dehumidification requirements and sensible loads will vary over the course of a day. Early in the day, outside air conditions will be humid and cool, requiring highly concentrated desiccant. During the middle of the day large sensible loads produce the heat required to deeply concentrate the liquid desiccant. Using a sufficient volume of liquid desiccant allows storage of concentrated liquid desiccant that can be used during cool and humid nights. Similarly at the end of the day, where cooling and dehumidification requirements are likely to remain large until well after sunset, loads can be reduced by using solar heat or solar power to store highly concentrated desiccant to reduce

demand in the early evening where the “duck shaped” network power demand curves could benefit from shifting some of the load from early evening to late afternoon.

**[00204]** Predictive models can use a combination historical data and last minute data, similar to the pricing models of airlines. Airlines set prices based on day of week and date, but then adjust based on actual buying behavior. A liquid desiccant predictive model can similarly use day of year / and hourly data to predict settings and then adjust based on last 24 hours and current data. For example this could allow a system to go into special dehumidification mode in September if temperatures the day before were 70F DP 60F+, but stay in standard cooling mode in July when that probably represents a rainy day, below seasonal averages.

**[00205]** Bandwidth control systems will show fluctuations between the upper and lower boundaries of the bandwidth. However given that the building changes all the air only once every 10-30 minutes fast response is often not required. Bandwidth control is simpler and can work effectively with a liquid desiccant system by using a combination of WB and RH bandwidth and measurements to drive the controls. Any input settings (DB/DP/RH) can be used to calculate the WB and RH bandwidth.

**[00206]** Referring to FIG. 33, control models can be equipment and building based. In building based systems, T&RH conditions of the space 4001 and airflow requirements are used to control the unit, rather than (only) the equipment supply conditions 4002 Return air conditions 4004 are a good proxy for 4001 assuming appropriate adjustments. The slowness of change in building conditions requires a slow response system to avoid fluctuations. In combination with ambient data 4003 and time of day information a predictive system can make optimal use of changing conditions to minimize energy use. Using the liquid desiccant system as to store the dehumidification capacity of concentrated liquid desiccant will significantly reduce energy use by reducing the need for using air cooled coils and their additional loads. In combination with water addition capacity the liquid desiccant system becomes highly efficient and responsive to expected changes in conditions. to use simpler controls that allow humidity and temperature to fluctuate in a narrow band.

**[00207]** Liquid desiccant systems do need to control for crystallization. FIG. 34A shows the conditions for crystallization for LiCl. Similar diagrams exist for CaCl and other desiccants.

**[00208]** Avoiding crystallization requires that nowhere in the panels the temperatures and humidities shown in FIG. 34A are achieved. The DP as function of temperature was calculated using the vapor pressure and solubility equations of the liquid desiccant. The system was then controlled by that function plus an offset reflecting the uncertainty about the precise dewpoint and temperatures realized in the unit given input and output DP and temperature. The DP of both the input and output conditions had to stay above the DP at the supply temperature. A similar equation was developed using RH as a function of Temperature which can be used in systems that control RH and temperature. Other combinations of T, RH, WB and DP can be realized using the same basic set of equations. The vapor pressure and solubility equations for LiCl were developed by among others Conde. FIG. 34 shows how the crystallization boundary is determined by concentration and temperature. Crystallization can occur in part of the panel where lower temperatures or lower RH conditions push the LiCl over the crystallization boundary 4101. For LiCl that zone ranges from 40% LiCl at 0C to about 50% LiCl at 60C (4102. Maintaining a minimum LiCl level in a storage tank corresponding to a concentration of about 35% prevents potential crystallization at any conditions.

**[00209]** FIG. 34B also shows how a typical east coast operating zone 4104 never gets close to crystallization conditions in cooling mode. That for operations in hot and dry climates like Phoenix 4105 desiccant dilution through evaporation or direct dilution is 4106 essential. In heating mode the chart suggest that an optimal LiCl concentration of about 25% to 35% is best to avoid frost forming on the outside unit, resulting in an RH of 35-45% for the heated air. The higher concentration is necessary to maintain dewpoints below 55F at supply temperatures of over 30C

**[00210]** FIG. 35 describes a sequence using water temperatures, dry bulb and dewpoint conditions as well as desiccant tank levels to identify when a crystallization warning is required. Protection includes water addition to the tank or by diluting the desiccant, turning down the compressor to reduce water temperatures, change regenerator airflows to decrease regenerator desorption. For those skilled in the art it is clear that a variety of controls and measures can be used depending on experience with the system, frequency of crystallization inducing conditions and performance requirements. For systems without water addition operating in hot and dry regions in the US, improving crystallization protection is important to reduce safety margins. Predictive controls are used with data

from modelling and testing. Direct concentration measurement can be used, e.g. by measuring the pressure loss across a restriction. The tank level is another predictor of concentration. RH out at high temperatures is a strong predictor of concentration, and will improve using system monitoring. Combinations are used to identify problems, e.g. flow restrictions, leaks and subsystem performance. Data history can be used on key performance metrics to identify reduced performance that may be due to lack of maintenance i.e. filter pressure drop too high, than filters need to be cleaned.

**[00211]** FIG. 36 describes how desiccant concentration and temperature determine viscosity of the desiccant. The chart describes LiCl, similar data are available on other desiccants. For a typical liquid desiccant system operating in cooling mode, dynamic viscosity in the regenerator is 2-5x larger than in the conditioner. As a result the conditioner requires higher pressure to maintain flows, or may have a lower flow rate than the regenerator. Modelled flow rates for different concentrations can be controlled in a number of ways.

- Use of fixed flow pumps to ensure that flow rates at conditioner and regenerator are equalized.
- Use of pressure controls through valves or air references to maintain a higher pressure for more viscose desiccants.
- Accept differences in flow rates on the conditioner and regenerator, the tanks or valves can be used to balance the regenerator and conditioner.

**[00212]** Having thus described several illustrative embodiments, it is to be appreciated that various alterations, modifications, and improvements will readily occur to those skilled in the art. Such alterations, modifications, and improvements are intended to form a part of this disclosure, and are intended to be within the spirit and scope of this disclosure. While some examples presented herein involve specific combinations of functions or structural elements, it should be understood that those functions and elements may be combined in other ways according to the present disclosure to accomplish the same or different objectives. In particular, acts, elements, and features discussed in connection with one embodiment are not intended to be excluded from similar or other roles in other embodiments. Additionally, elements and components described herein may be further divided into additional components or joined together to form fewer components for

performing the same functions. Accordingly, the foregoing description and attached drawings are by way of example only, and are not intended to be limiting.

**CLAIMS**

1. A method of operating a liquid desiccant air-conditioning system to maintain target temperature and humidity level in a space, the liquid desiccant air conditioning system comprising:
  - a conditioner for treating a first air stream flowing therethrough and provided to the space as a supplied air stream, said conditioner using a heat transfer fluid and a liquid desiccant to treat the first air stream;
  - a device for measuring temperature and a device for measuring humidity in the supplied air stream;
  - a regenerator connected to the conditioner such that the liquid desiccant can be circulated between the regenerator and the conditioner, the regenerator causing the liquid desiccant to desorb water vapor to a second air stream or to absorb water vapor from the second air stream depending on a selected mode of operation of the system;
  - a refrigerant system;
  - a first refrigerant-to-heat transfer fluid heat exchanger connected to the conditioner and the refrigerant system for exchanging heat between the refrigerant heated or cooled by the refrigerant system and the heat transfer fluid used in the conditioner;
  - a second refrigerant-to-heat transfer fluid heat exchanger connected to the regenerator and the refrigerant system for exchanging heat between the refrigerant heated or cooled by the refrigerant system and the heat transfer fluid used in the regenerator; and
  - a system controller for controlling operation of the system;wherein the method comprising the steps of:
  - (a) measuring the temperature and humidity level in the supplied air stream;
  - (b) comparing the temperature measured in (a) to a target temperature to determining a temperature error, and comparing the humidity level measured in (a) to a target humidity level to determine a humidity error;
  - (c) comparing the humidity error and the temperature error on a common scale to determine the greater error;

(d) using the greater error to drive the system controller to control operation of the system to reduce the greater error;

(e) repeat (a) through (d) a plurality of times.

2. The method of claim 1, wherein the temperature measured in (a) is a dry bulb temperature and the humidity level measured in (a) is a relative humidity level, and wherein the target temperature is a dry bulb temperature and the humidity target is a dew point target;

and the step (c) is based on errors in dry bulb and dew point.

3. The method of claim 2, wherein when the system operates as a dehumidifier, the system controller is operated to prevent the dew point based on measurements from being lower than the target dew point.

4. The method of claim 2, wherein the system controller controls the setting of a compressor in the refrigerant system to control the cooling capacity of the system, and

the system controller controls the liquid desiccant and heat transfer fluid flow rates in the regenerator, wherein higher liquid desiccant and heat transfer fluid flow rates in the regenerator increase the sensible cooling rate of the supply air stream.

5. The method of claim 4, wherein the liquid desiccant air conditioning system further comprises a liquid desiccant dilution device, and the method further comprises controlling the sensible cooling rate of the supply air stream by diluting the liquid desiccant using the liquid desiccant dilution device.

6. The method of claim 5, wherein the liquid desiccant dilution device is used to maintain minimum level of liquid desiccant in a liquid desiccant tank of the liquid desiccant air conditioning system to maintain a minimum relative humidity level in the supply air stream.

7. The method of claim 4, wherein the liquid desiccant air conditioning system further comprises an air-cooled coil associated with the second refrigerant-to-heat transfer fluid heat exchanger, the method further comprises increasing the sensible heat ratio by increasing fluid flow rates through the air-cooled coil.

8. The method of claim 4, wherein the liquid desiccant air conditioning system further comprises an air-cooled coil associated with the first refrigerant-to-

heat transfer fluid heat exchanger, the method further comprises reducing the sensible heat ratio by increasing fluid flow rates through the air-cooled coil.

9. The method of claim 1, wherein the temperature measured in (a) is a dry bulb temperature and the humidity level measured in (a) is a relative humidity level, and wherein the target temperature is a dry bulb temperature and the humidity target is a dew point target;

and the step (c) is based on errors in wet bulb and relative humidity.

10. The method of claim 9, wherein the wet bulb error controls the setting of a compressor in the refrigerant system to control the cooling capacity of the system, and

the relative humidity error controls the liquid desiccant and heat transfer fluid flow rates in the regenerator, wherein higher liquid desiccant and heat transfer fluid flow rates in the regenerator increase the sensible cooling rate of the supply air stream.

11. The method of claim 10, wherein the liquid desiccant air conditioning system further comprises a liquid desiccant dilution device, and the method further comprises controlling the sensible cooling rate of the supply air stream by diluting the liquid desiccant using the liquid desiccant dilution device.

12. The method of claim 11, wherein the liquid desiccant dilution device is used to maintain minimum level of liquid desiccant in a liquid desiccant tank of the liquid desiccant air conditioning system to maintain a minimum relative humidity level in the supply air stream.

13. The method of claim 9, wherein the liquid desiccant air conditioning system further comprises an air-cooled coil associated with the second refrigerant-to-heat transfer fluid heat exchanger, the method further comprises increasing the sensible heat ratio by increasing fluid flow rates through the air-cooled coil.

14. The method of claim 9, wherein the liquid desiccant air conditioning system further comprises an air-cooled coil associated with the first refrigerant-to-heat transfer fluid heat exchanger, the method further comprises reducing the sensible heat ratio by increasing fluid flow rates through the air-cooled coil.

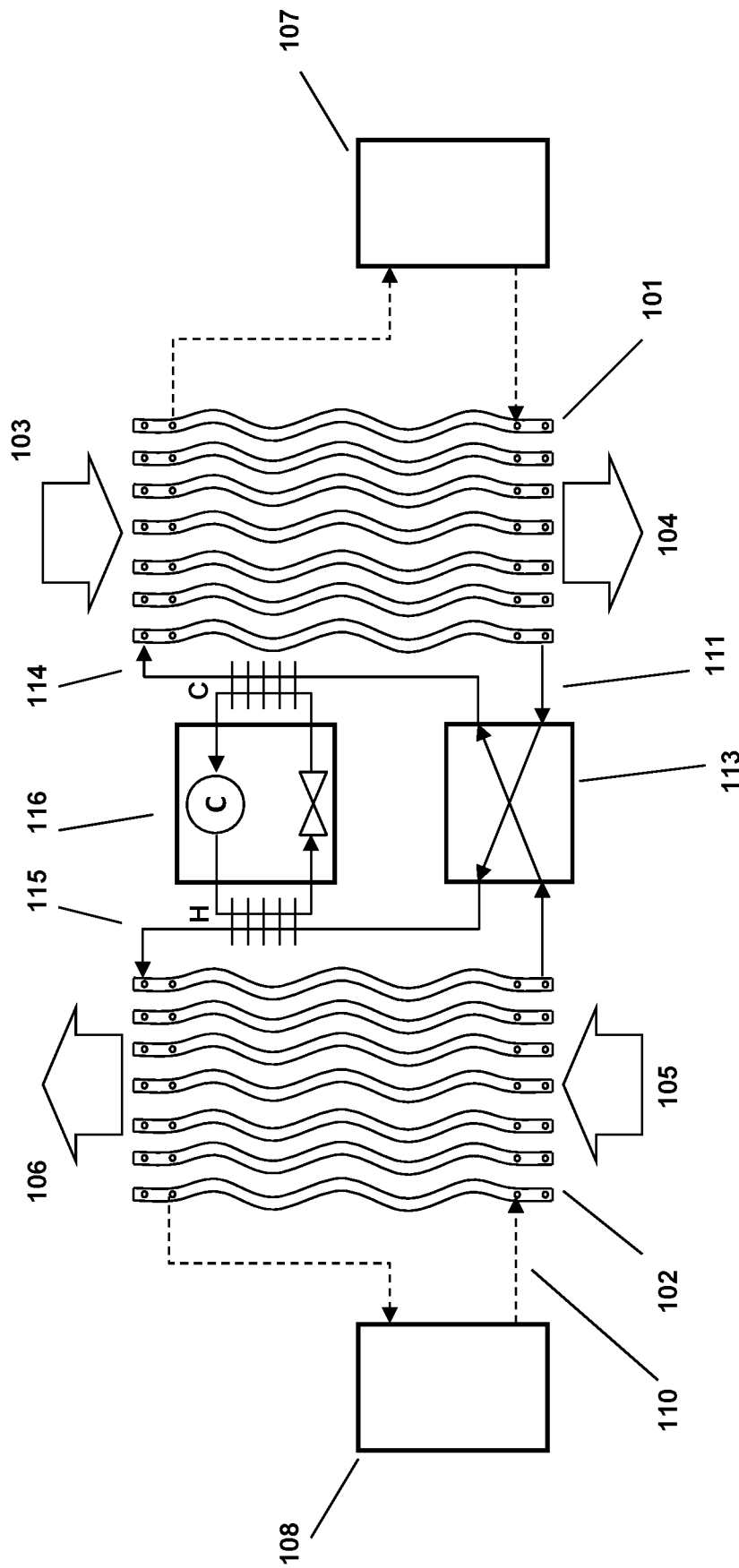


FIG. 1

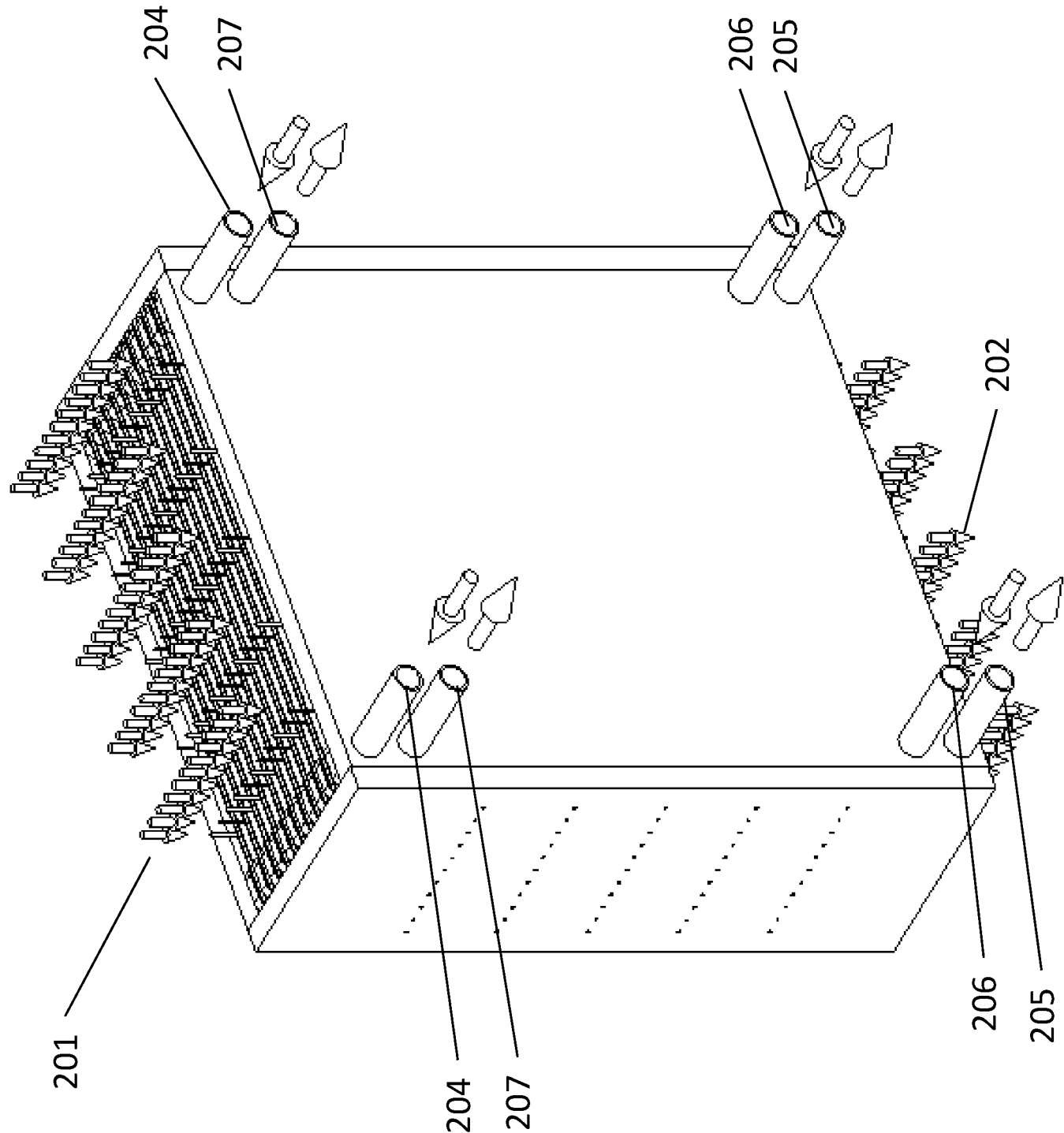


FIG. 2

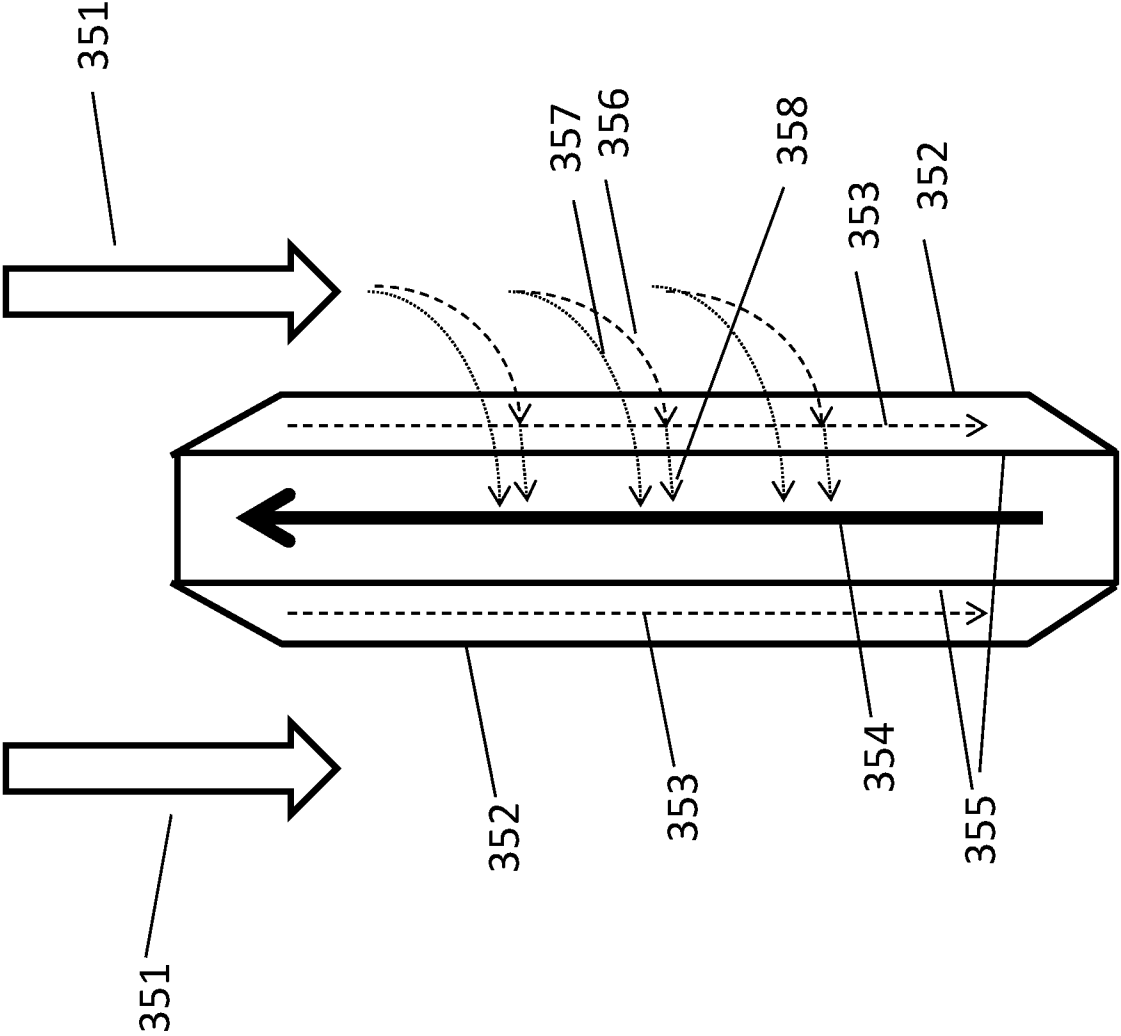


FIG. 3

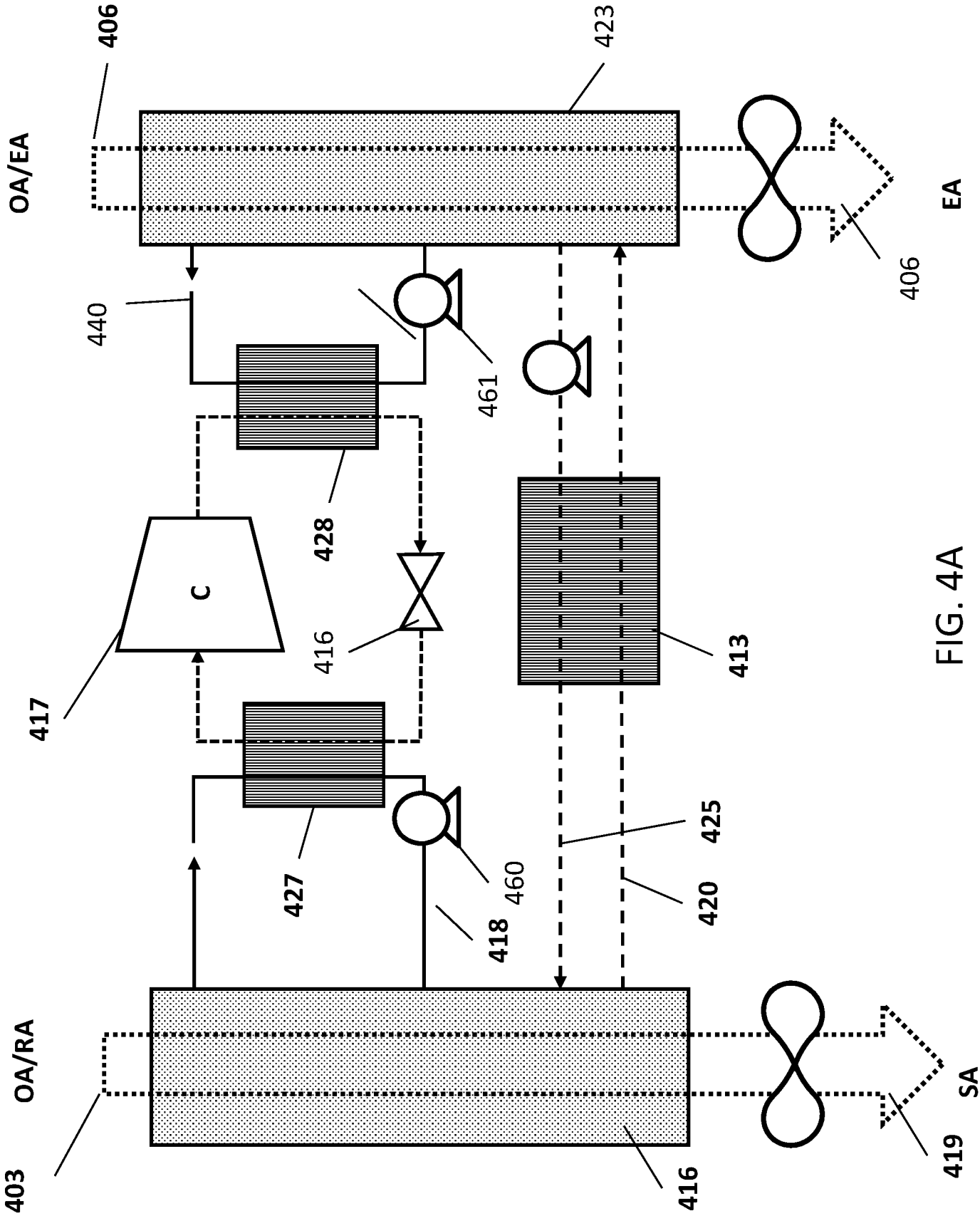


FIG. 4A

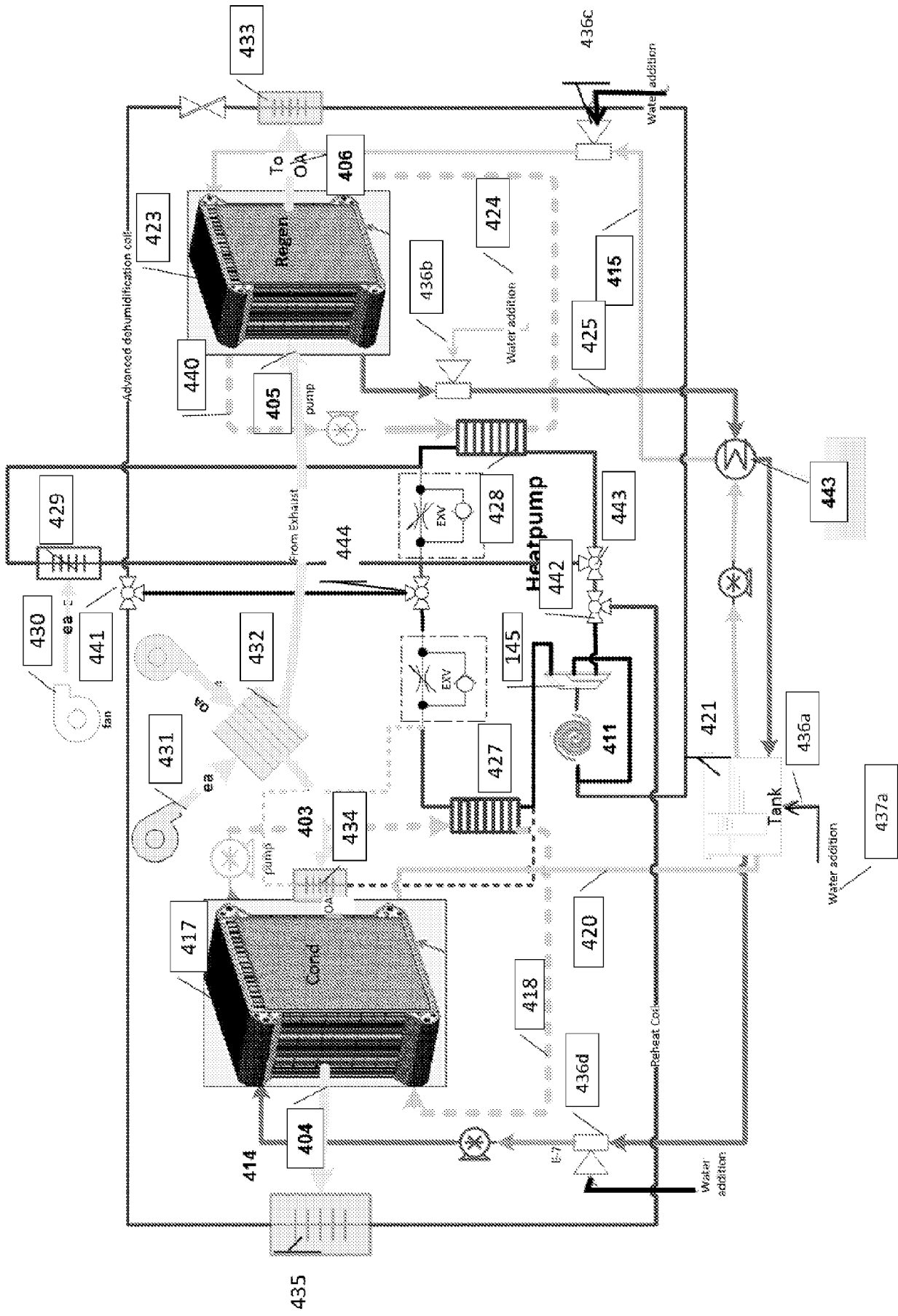


FIG. 4B

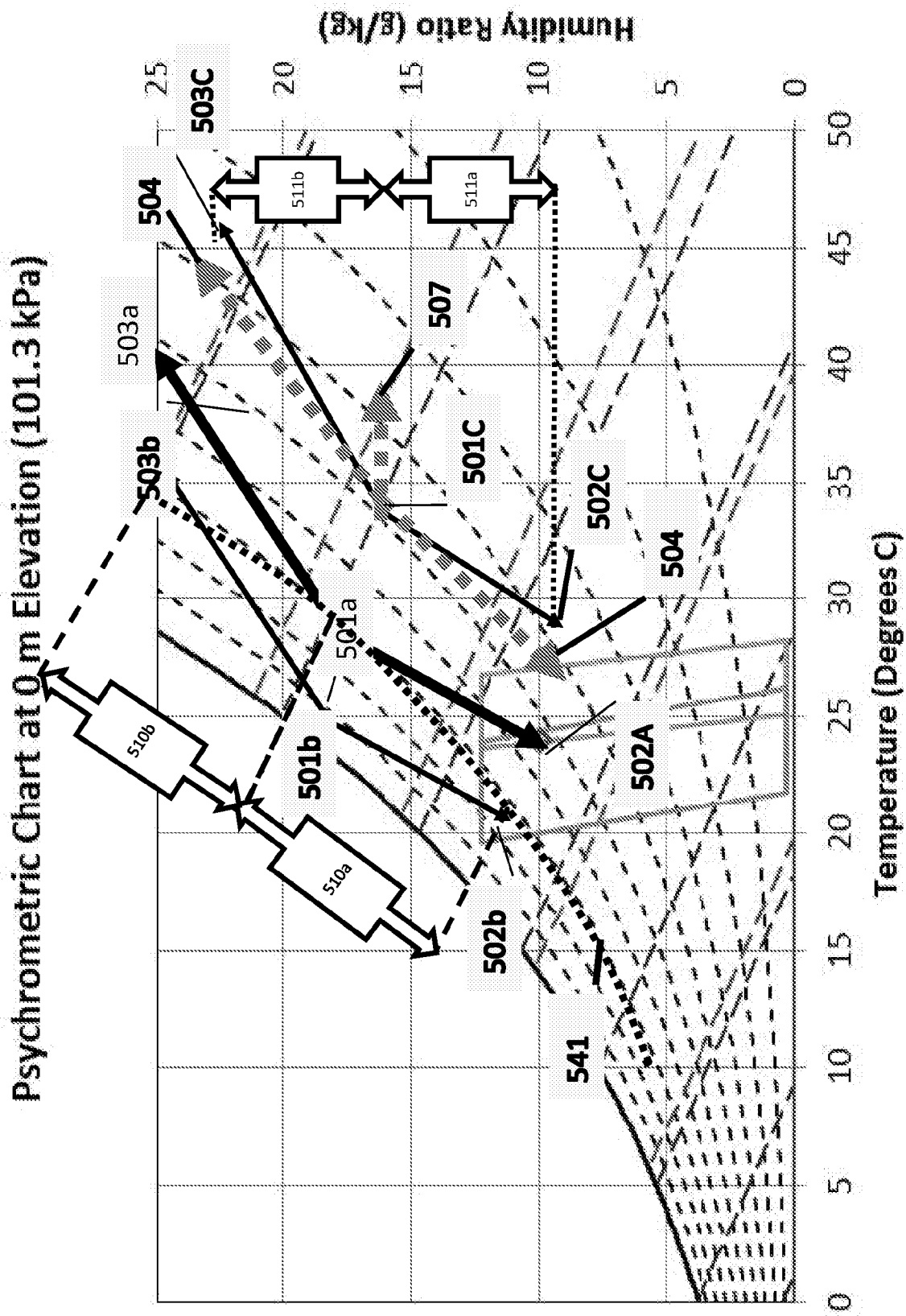


FIG. 5

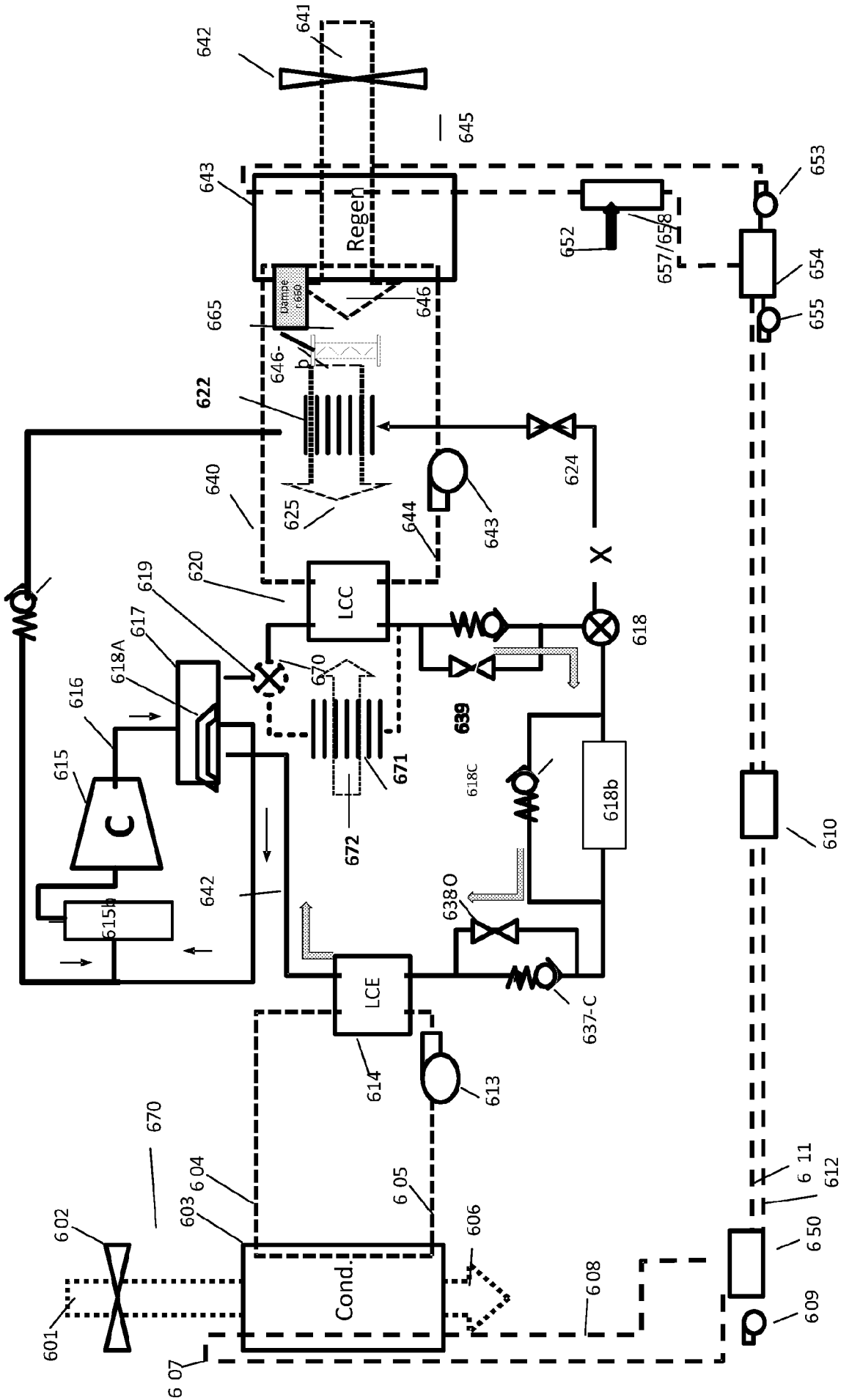
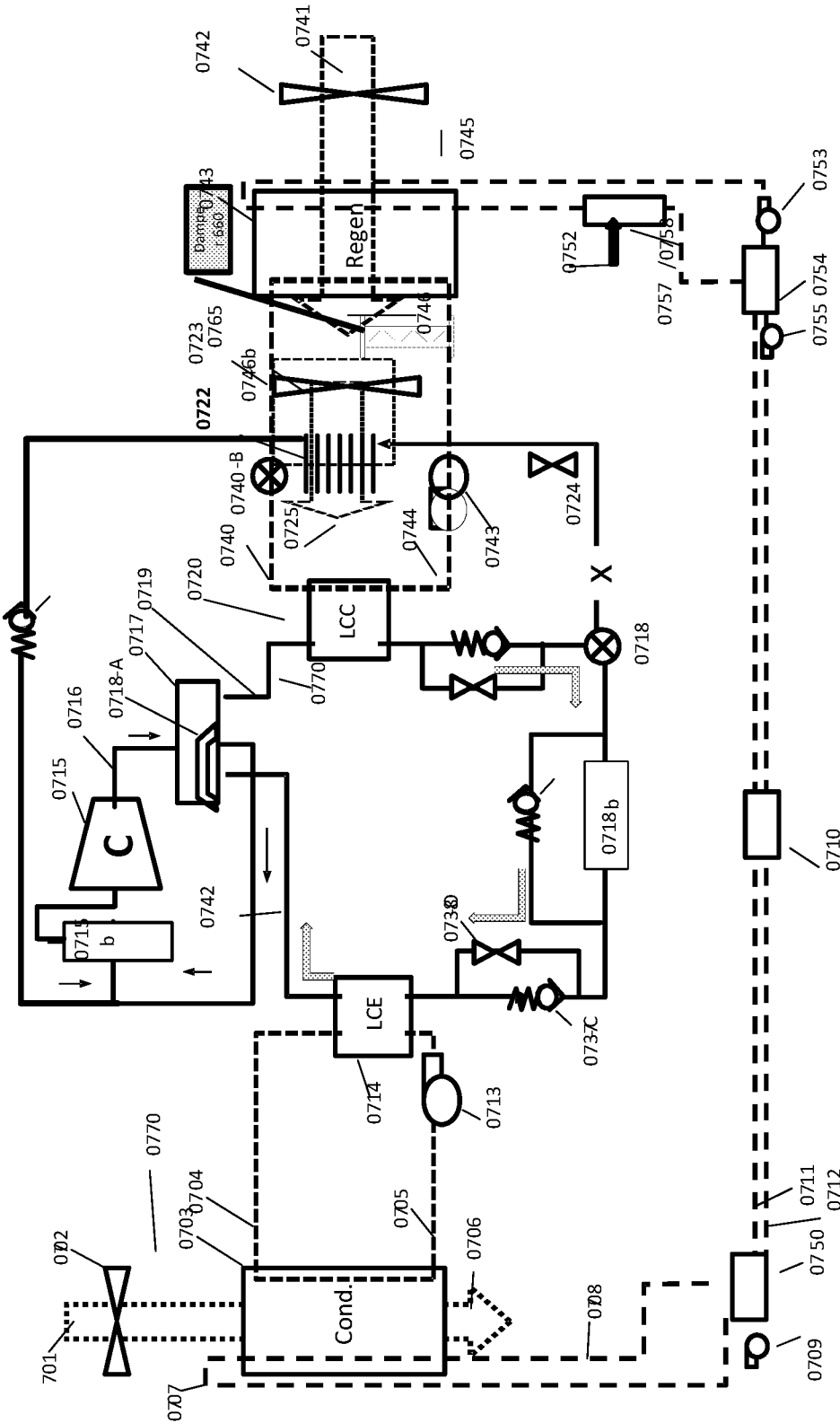


FIG. 6



Cooling mode

FIG. 7A

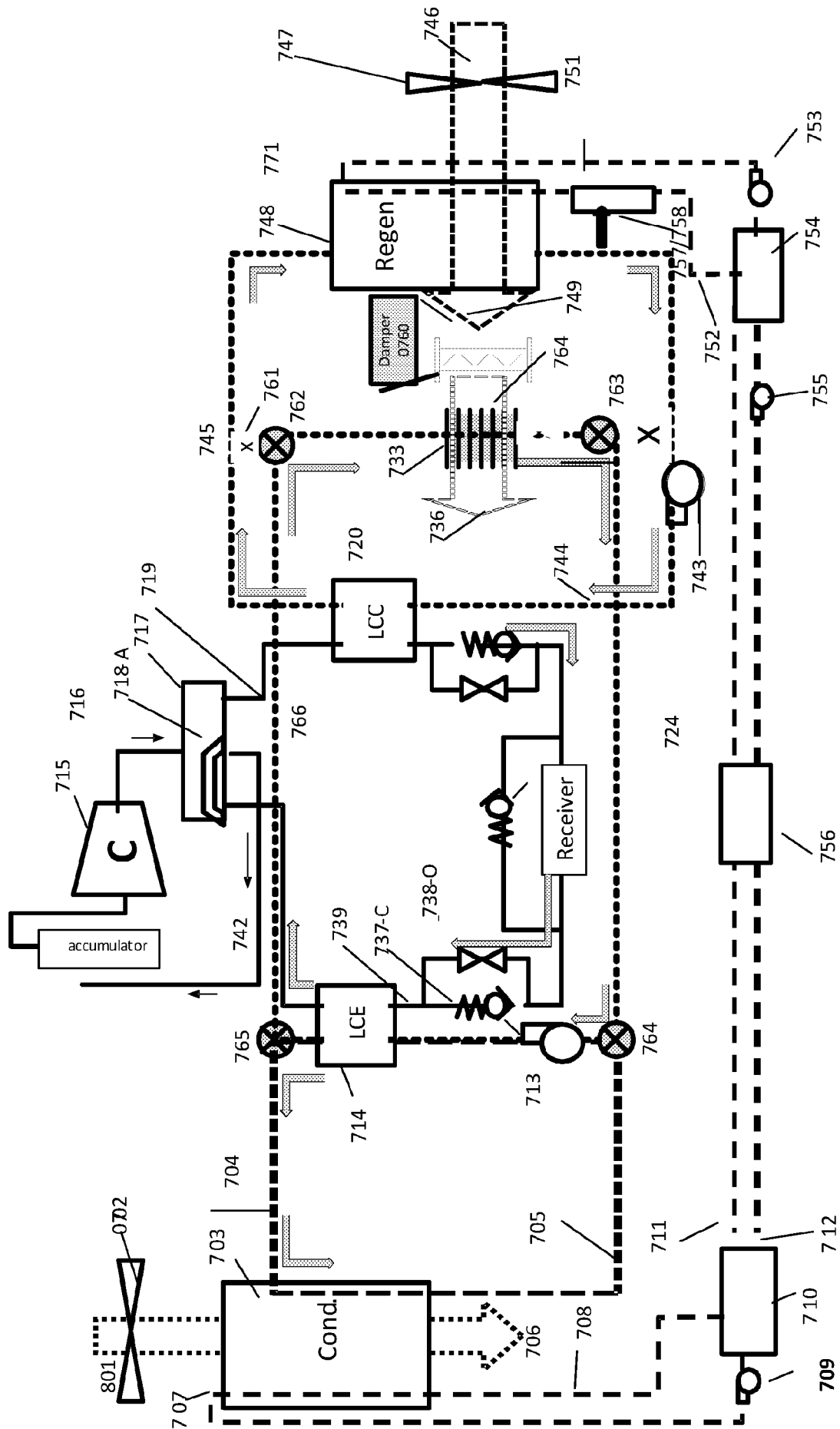


FIG. 7B

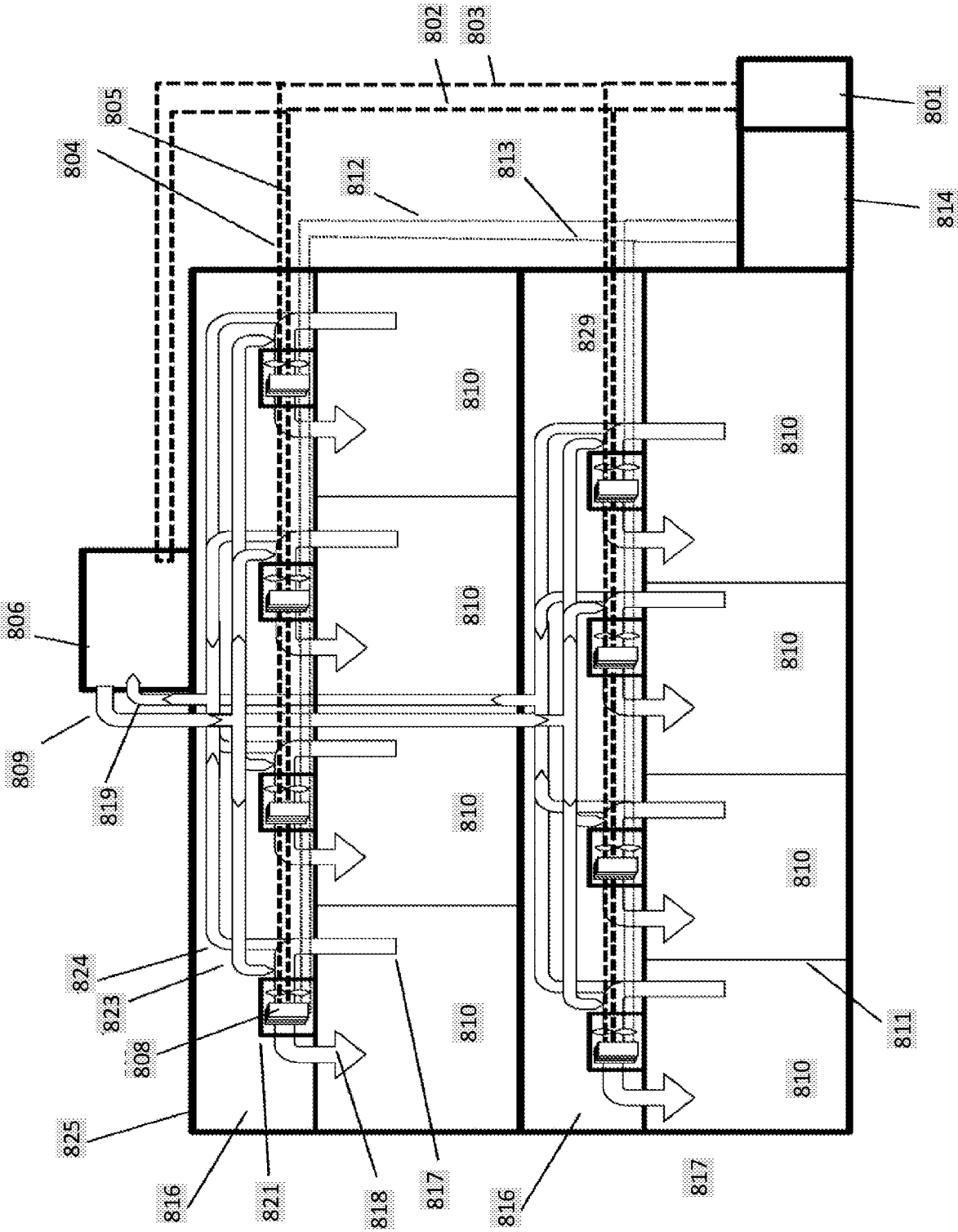


FIG. 8

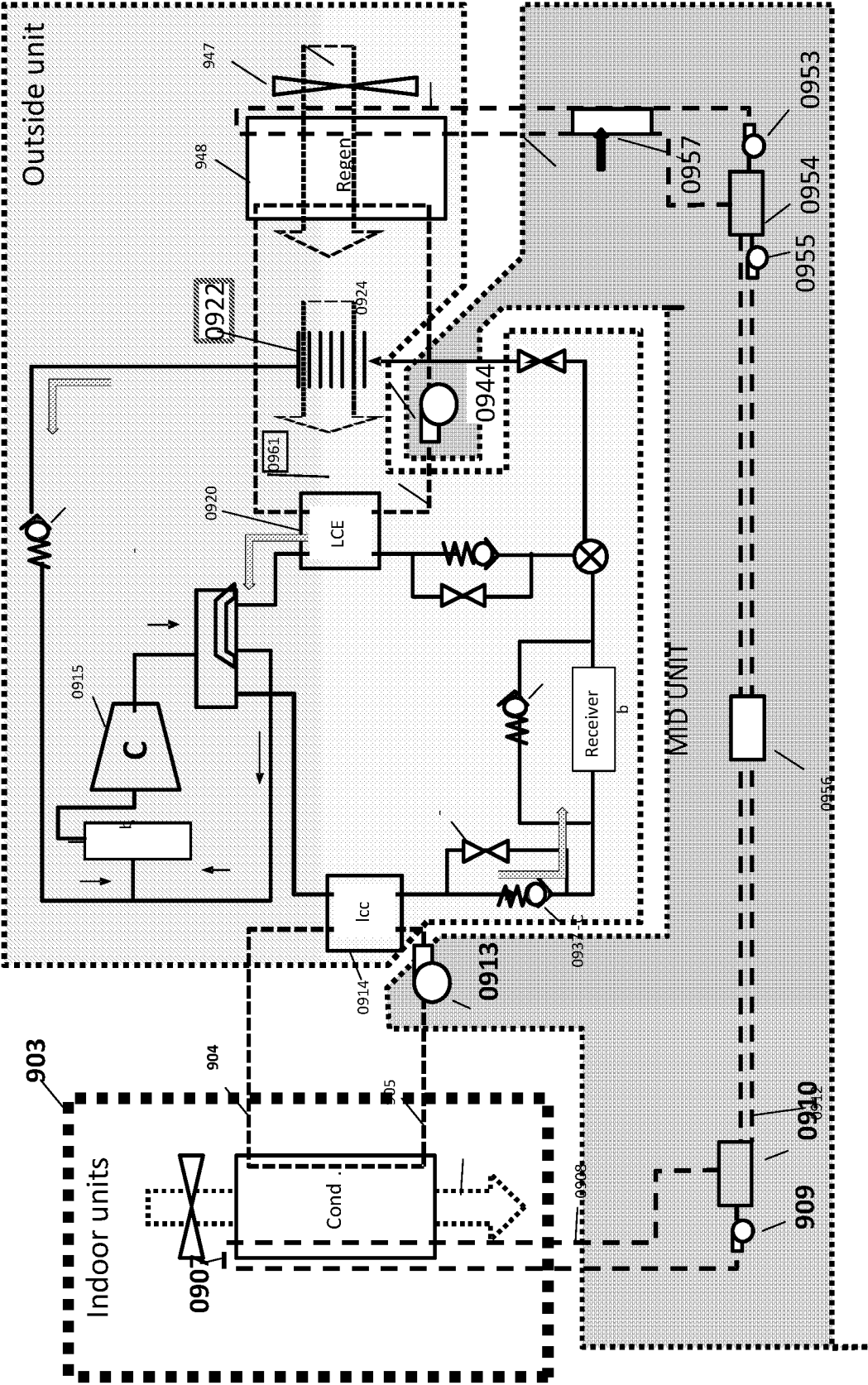


FIG. 9

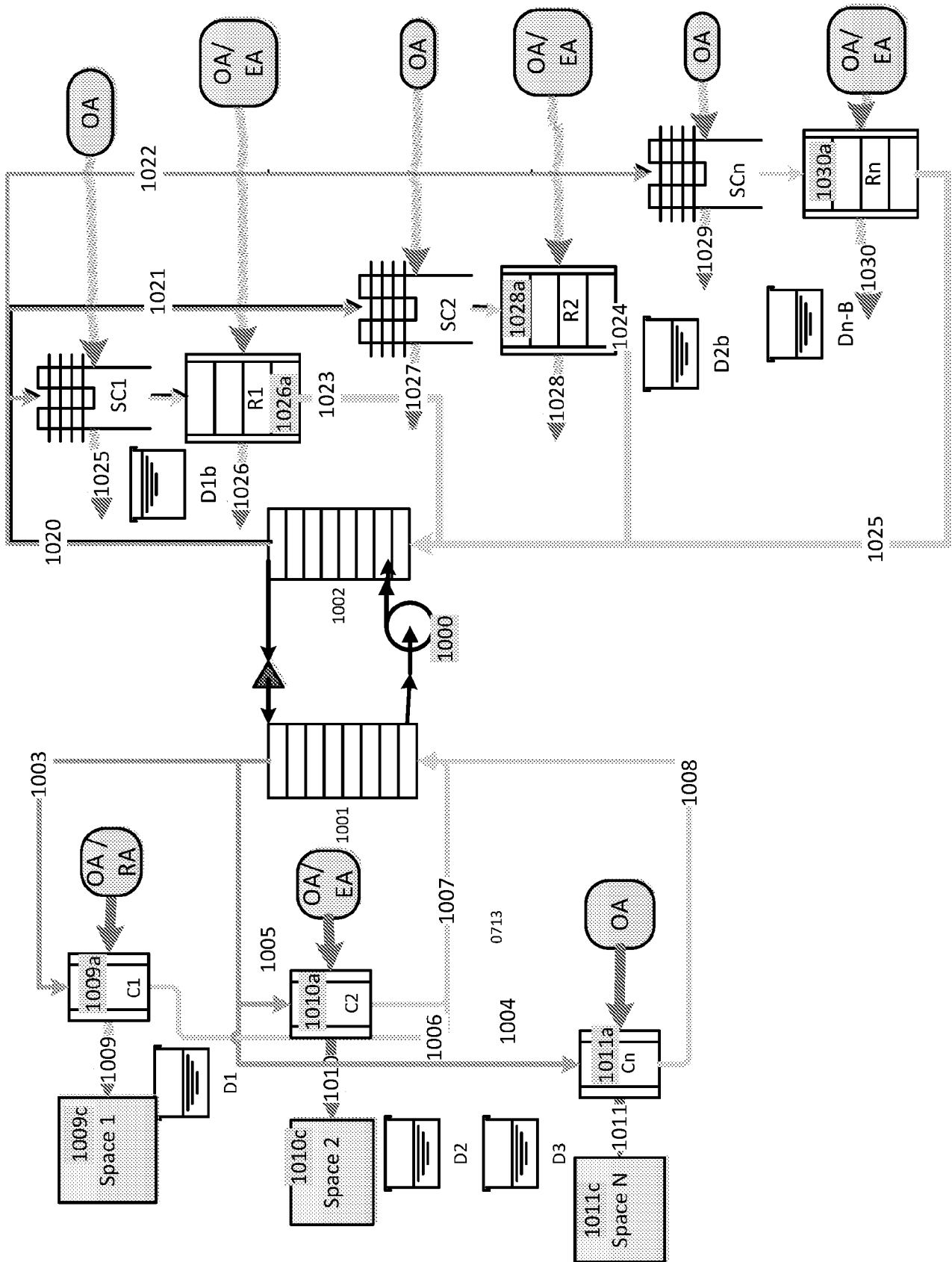


FIG. 10

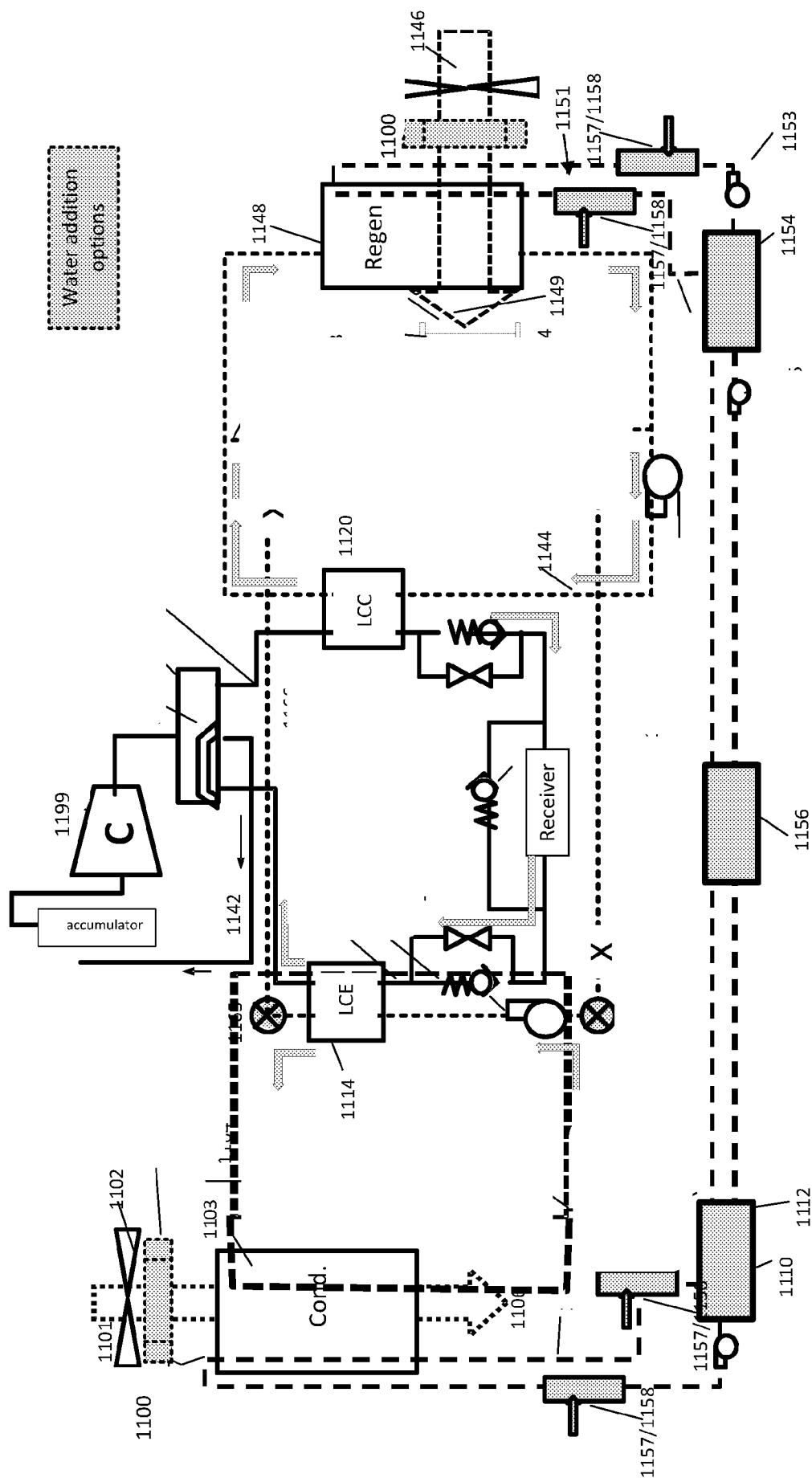


FIG. 11

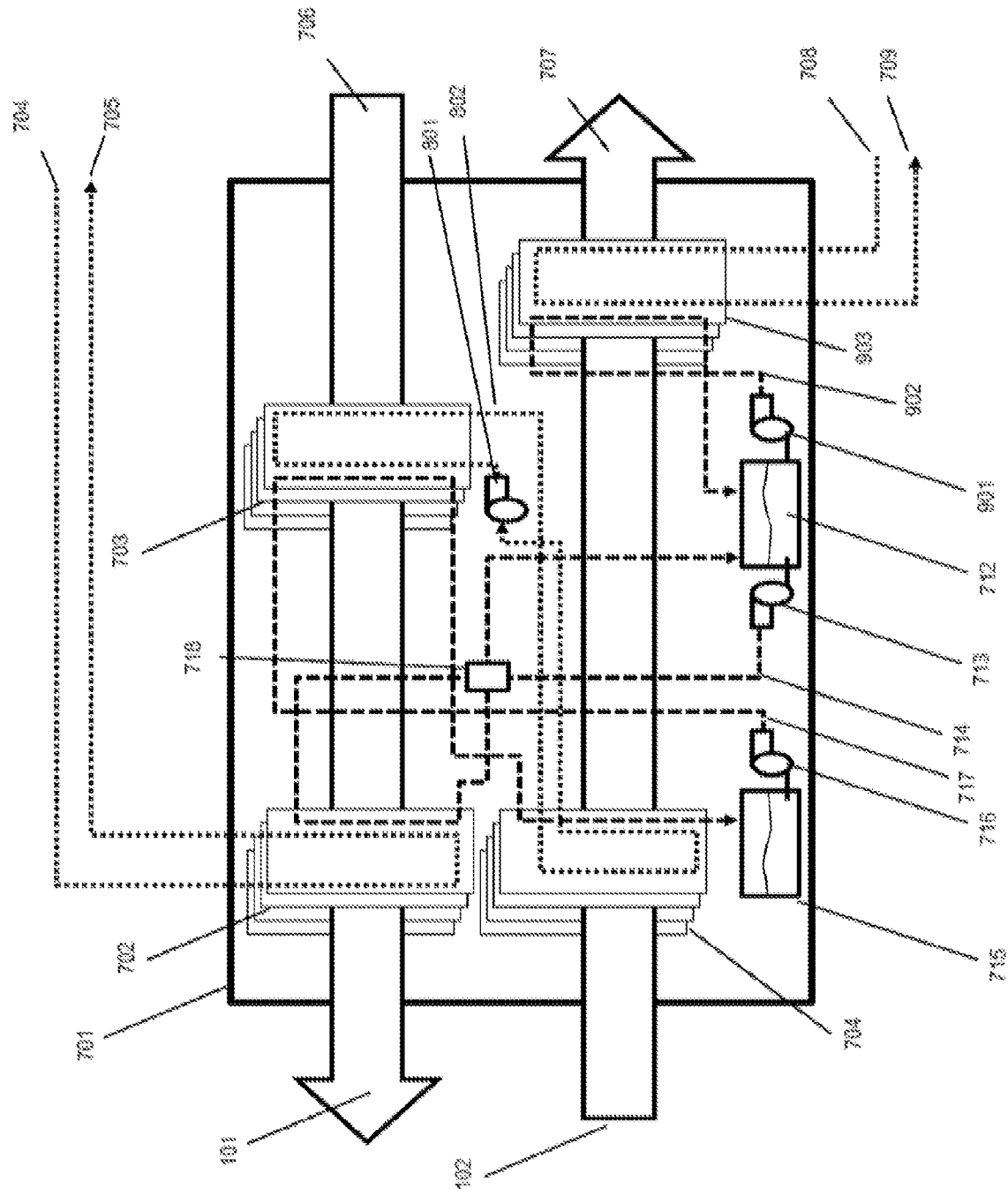


FIG. 12

FIG. 13A

- Modes :
- 1/2: Lower DB and higher DP

3/4: Lower DB and lower DP

5/6: Higher DB and lower DP

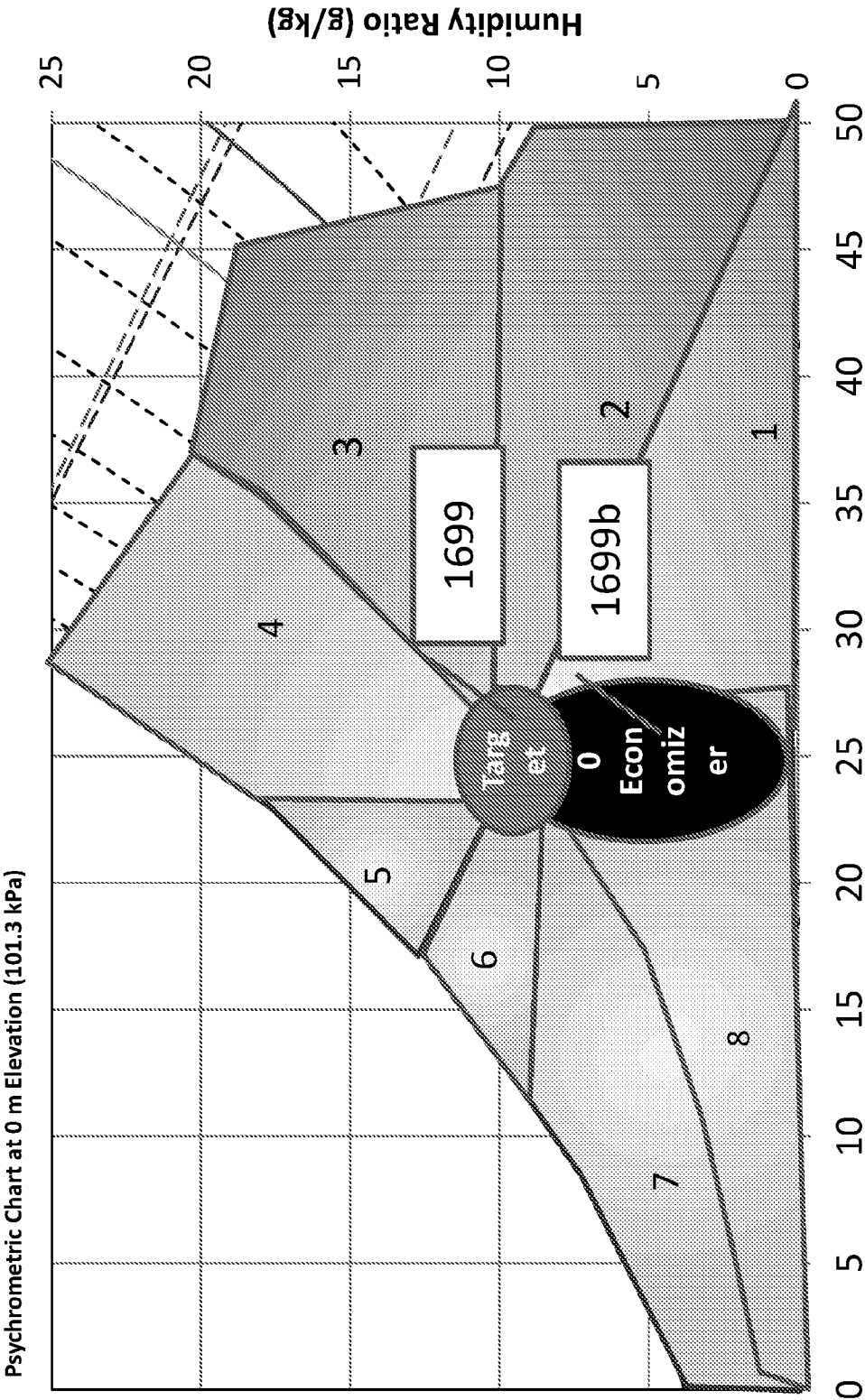
7/8: Higher DB and higher DP.

0- Economizer option
- 4/5/6/7: reduce RH

1/2/3/8: increase RH

2,3,4,5: reduce WB/Enthalpy

6/7/8/1 increase WB/enthalpy



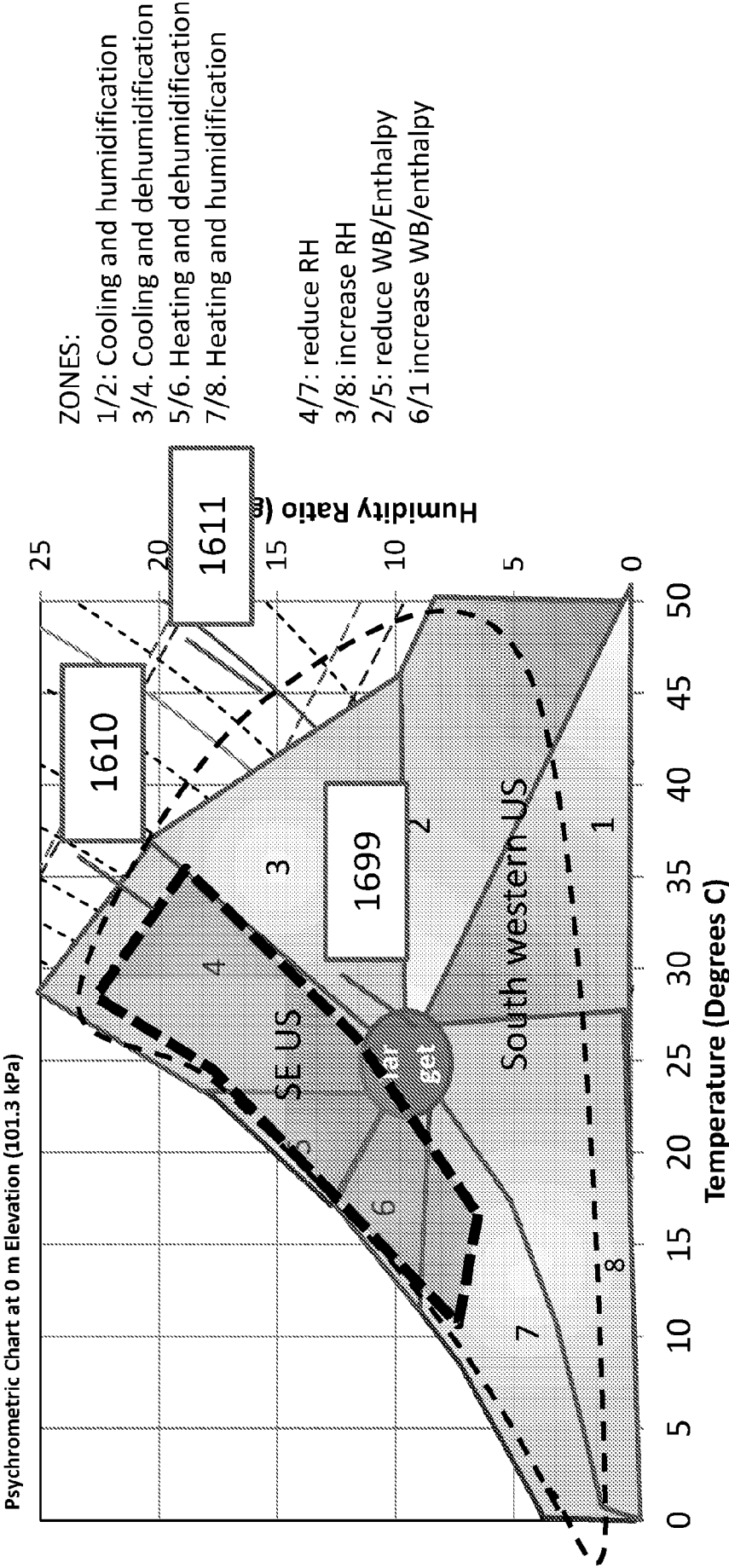


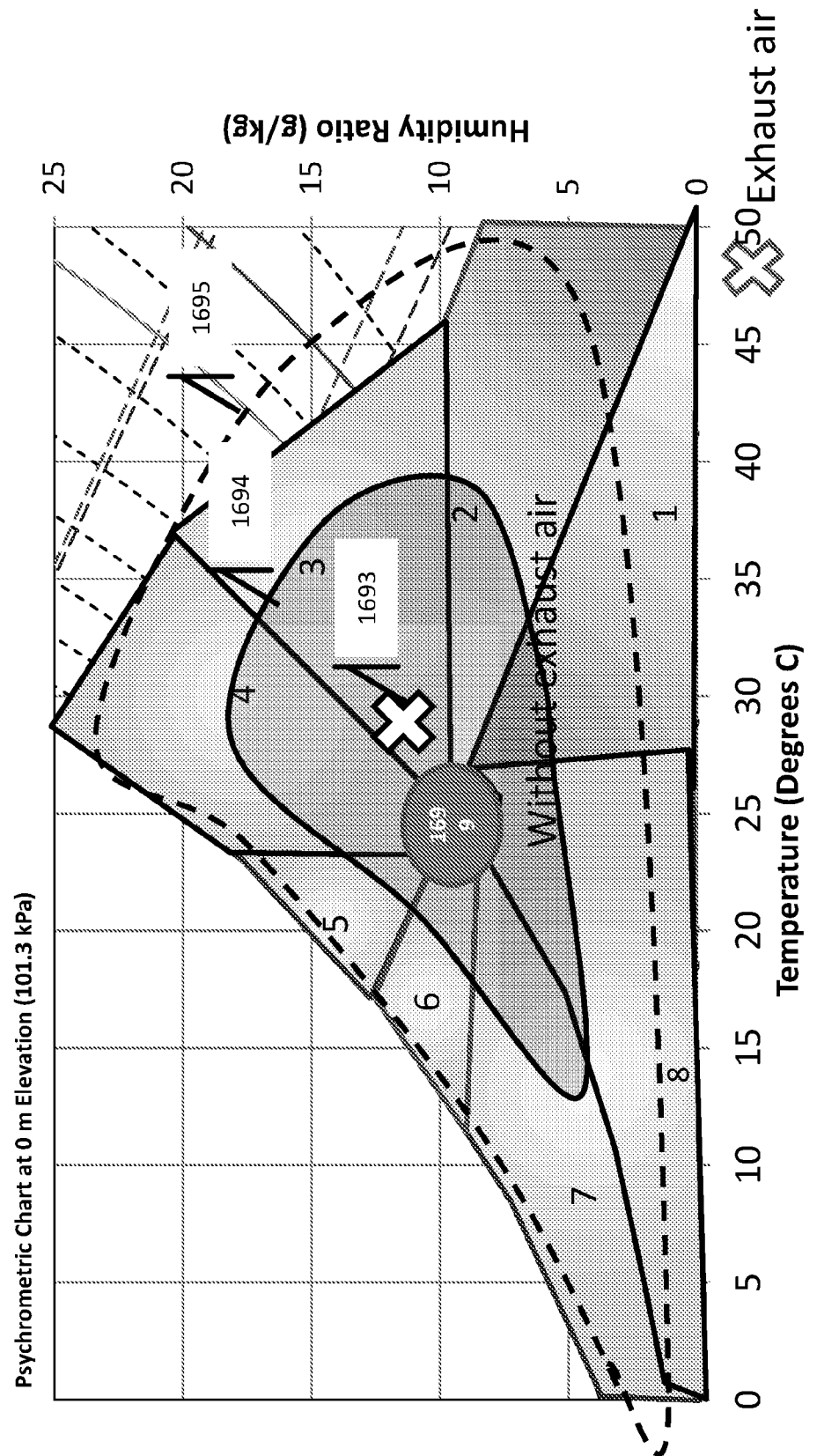
FIG. 13B

**FIG. 13C**

**ZONES:**

- 1/2: Cooling and humidification  
3/4: Cooling and dehumidification  
5/6: Heating and dehumidification  
7/8: Heating and humidification

- 4/7: reduce RH  
3/8: increase RH  
2/5: reduce WB/Enthalpy  
6/1 increase WB/enthalpy



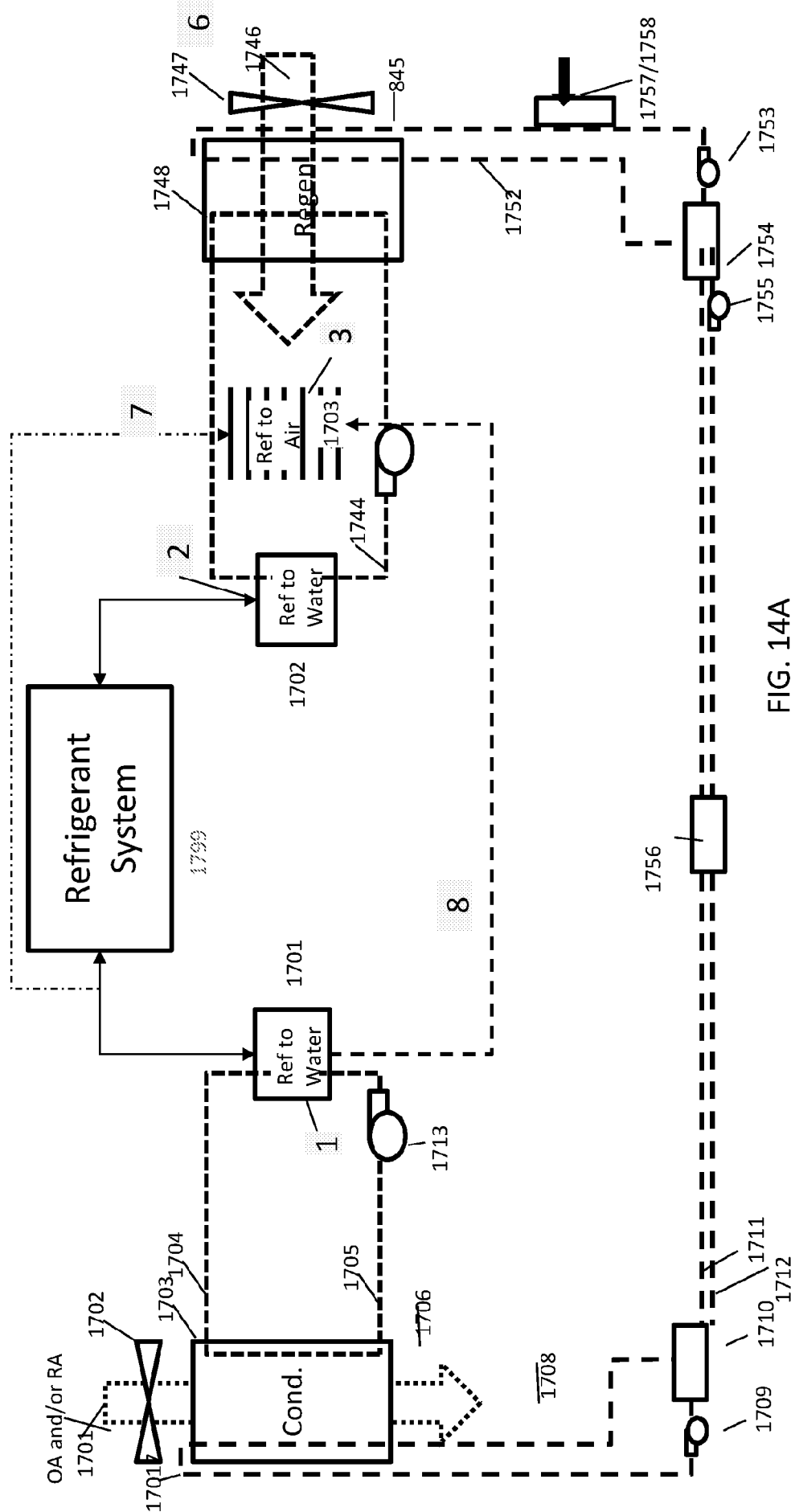


FIG. 14A

| Operating modes |         |         |                 |             |
|-----------------|---------|---------|-----------------|-------------|
| HX              | Cooling | Heating | Defrost heating | Adv. Dehum. |
| 1               | Cold    | Hot     | Hot             | Cold        |
| 2               | Hot     | Cold    | Cold            | Hot         |
| 3               | Off     | Off     | Colder          | Cold        |

Table 14A

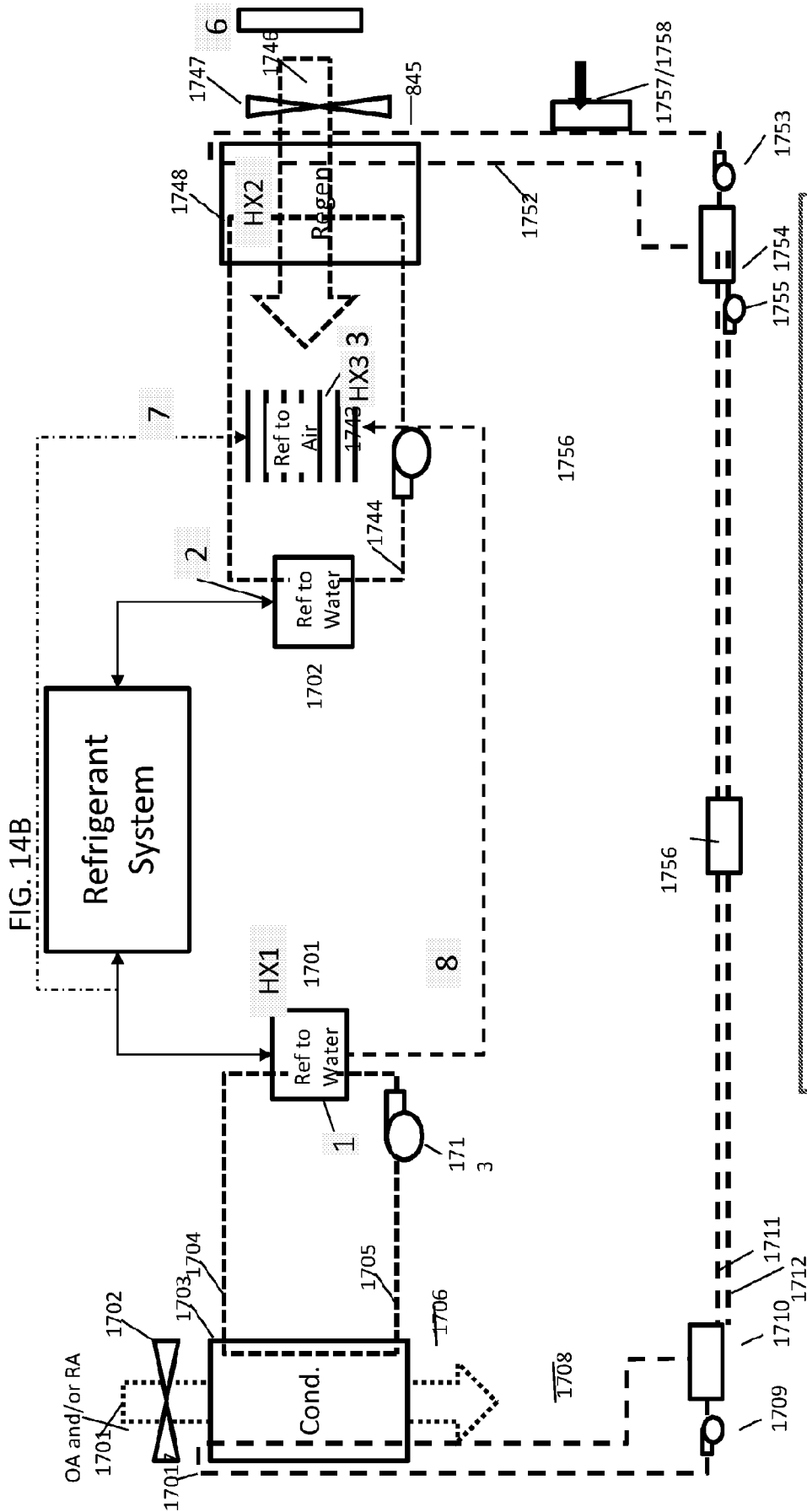


Table 14B

| Operating modes for Zones 1-8 |        |        |          |      |      |      |        |        |
|-------------------------------|--------|--------|----------|------|------|------|--------|--------|
| HX                            | 1      | 2      | 3        | 4    | 5    | 6    | 7      | 8      |
| 1                             | Cold   | Cold   | Cold     | Cold | Cold | Cold | Hot    | Hot    |
| 2                             | Off    | Hot    | Hot      | Hot  | Hot  | Hot  | Cold   | Cold   |
| 3                             | Hotter | Hotter | off      | off  | Cold | Cold | Colder | Colder |
| Humidification air (6)        | ON     | ON     | ON (low) | off  | off  | off  | off    | low    |
| Direct water add. (5)         | High   | High   | Low      | Off  | Off  | Off  | Off    | Low    |

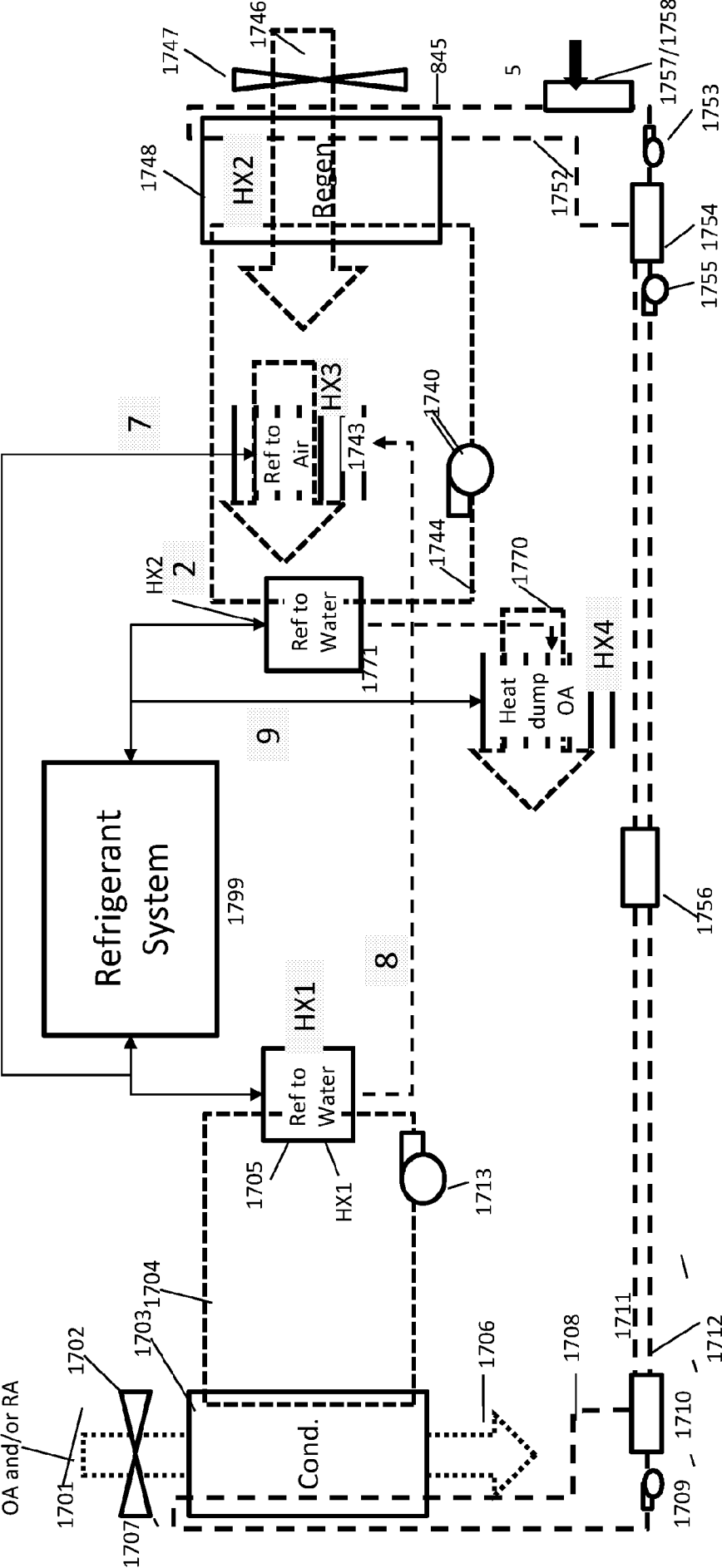
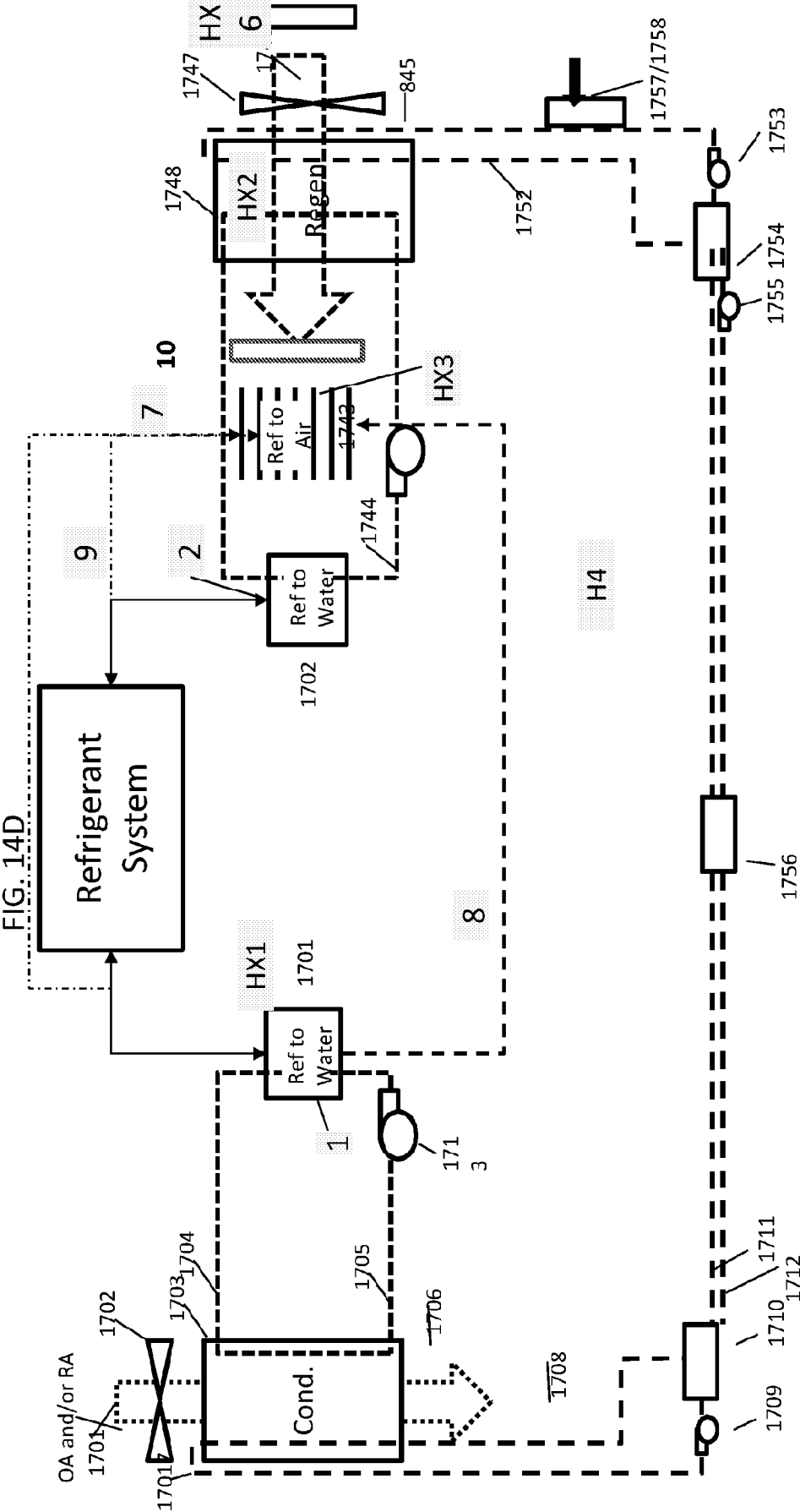


Table 17C  
Dual air  
cool coil  
system

| HX | 601    | 602    | 603  | 604  | 605  | 606  | 607    | 608    |
|----|--------|--------|------|------|------|------|--------|--------|
| 1  | Cold   | Cold   | Cold | Cold | Cold | Cold | Hot    | Hot    |
| 2  | Off    | Hot    | Hot  | Hot  | Hot  | Hot  | Cold   | Cold   |
| 3  | Hotter | Hotter | off  | off  | Cold | Cold | Colder | Colder |
| 4  | Hotter | Hotter | Hot  | off  | off  | off  | off    | off    |

Table 14C



Operating modes for Zones 1-8

| HX          | 1    | 2    | 3    | 4    | 5    | 6    | 7      | 8      |
|-------------|------|------|------|------|------|------|--------|--------|
| 1           | Cold | Cold | Cold | Cold | Cold | Cold | Hot    | Hot    |
| 2           | Off  | Hot  | Hot  | Hot  | Hot  | Hot  | Cold   | Cold   |
| 3           | OFF  | off  | off  | off  | Cold | Cold | Colder | Colder |
| 10 (damper) | OA   | OA   | OA   | OA   | RA   | RA   | RA     | RA     |

Table 14D

FIG. 15A

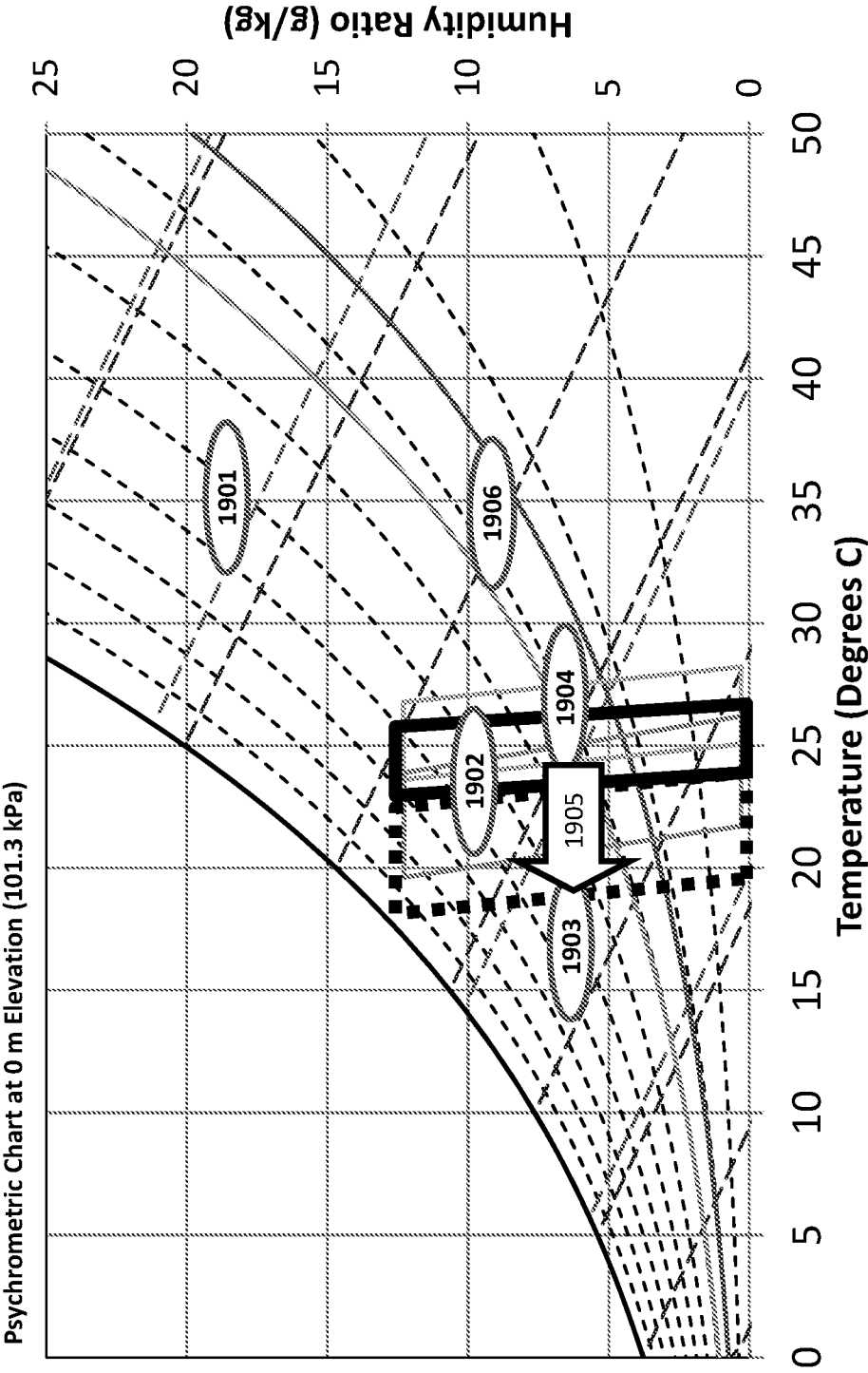
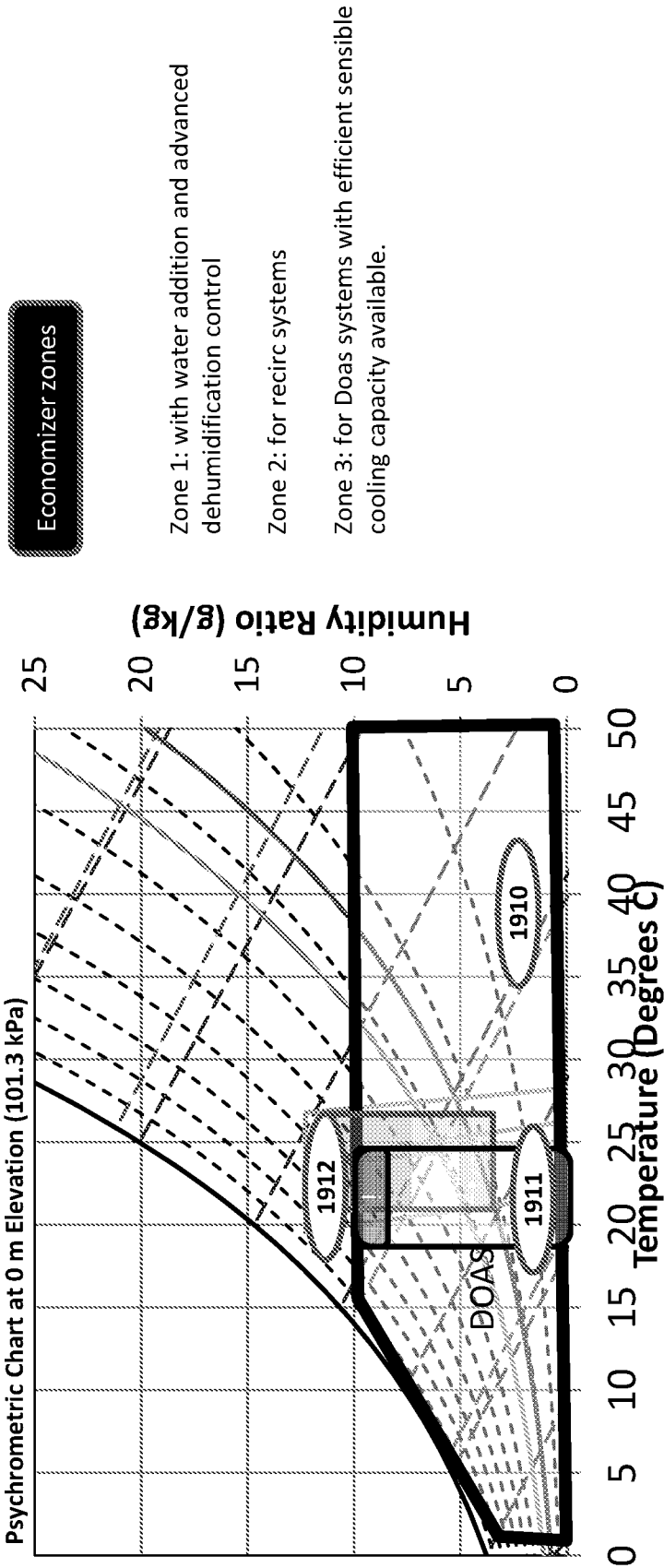


FIG. 15B

South east US has year round subtropical conditions with AC requirements for cooling and dehumidification  
South west can have a mix of desert and monsoon like conditions requiring all air conditioning capabilities



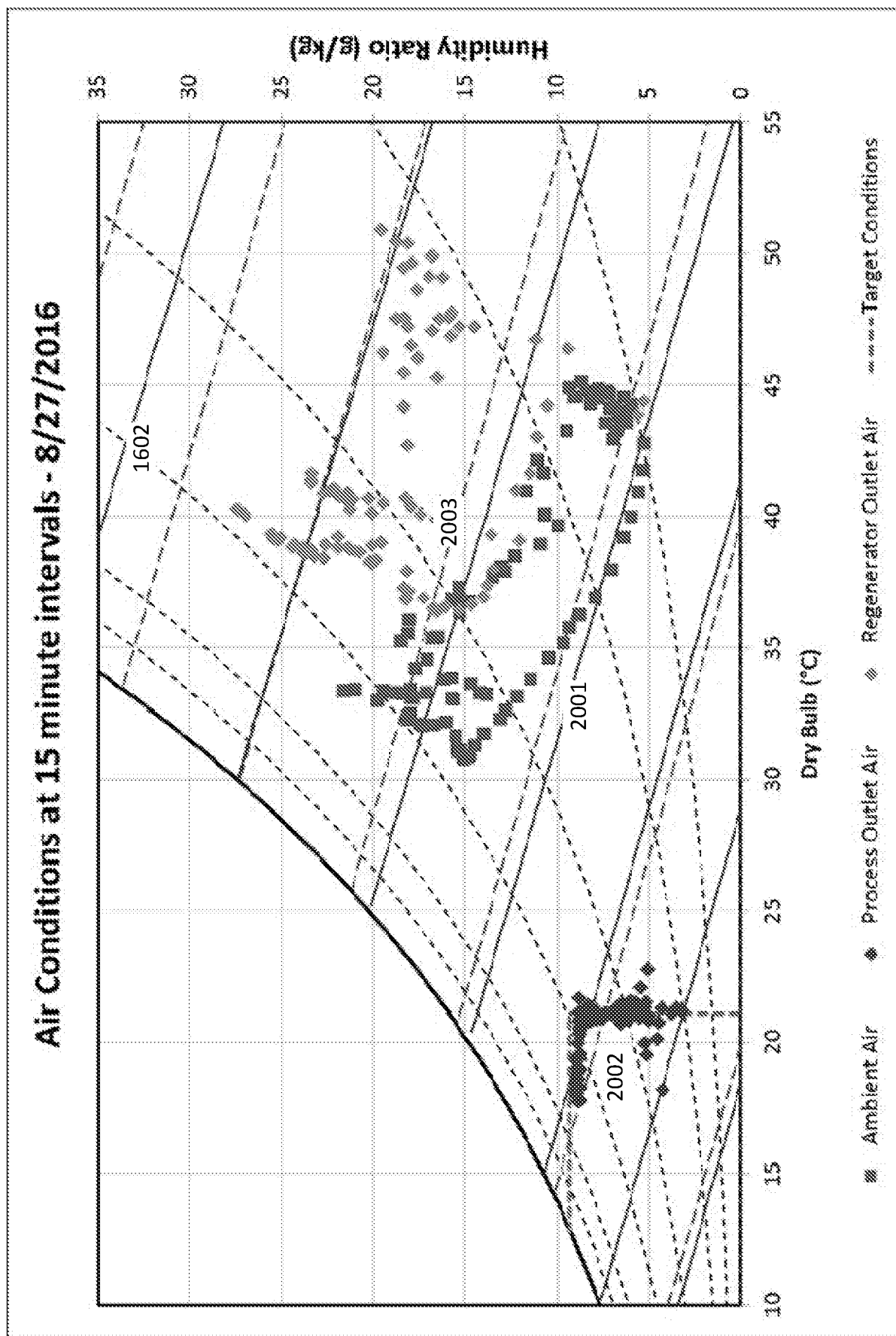
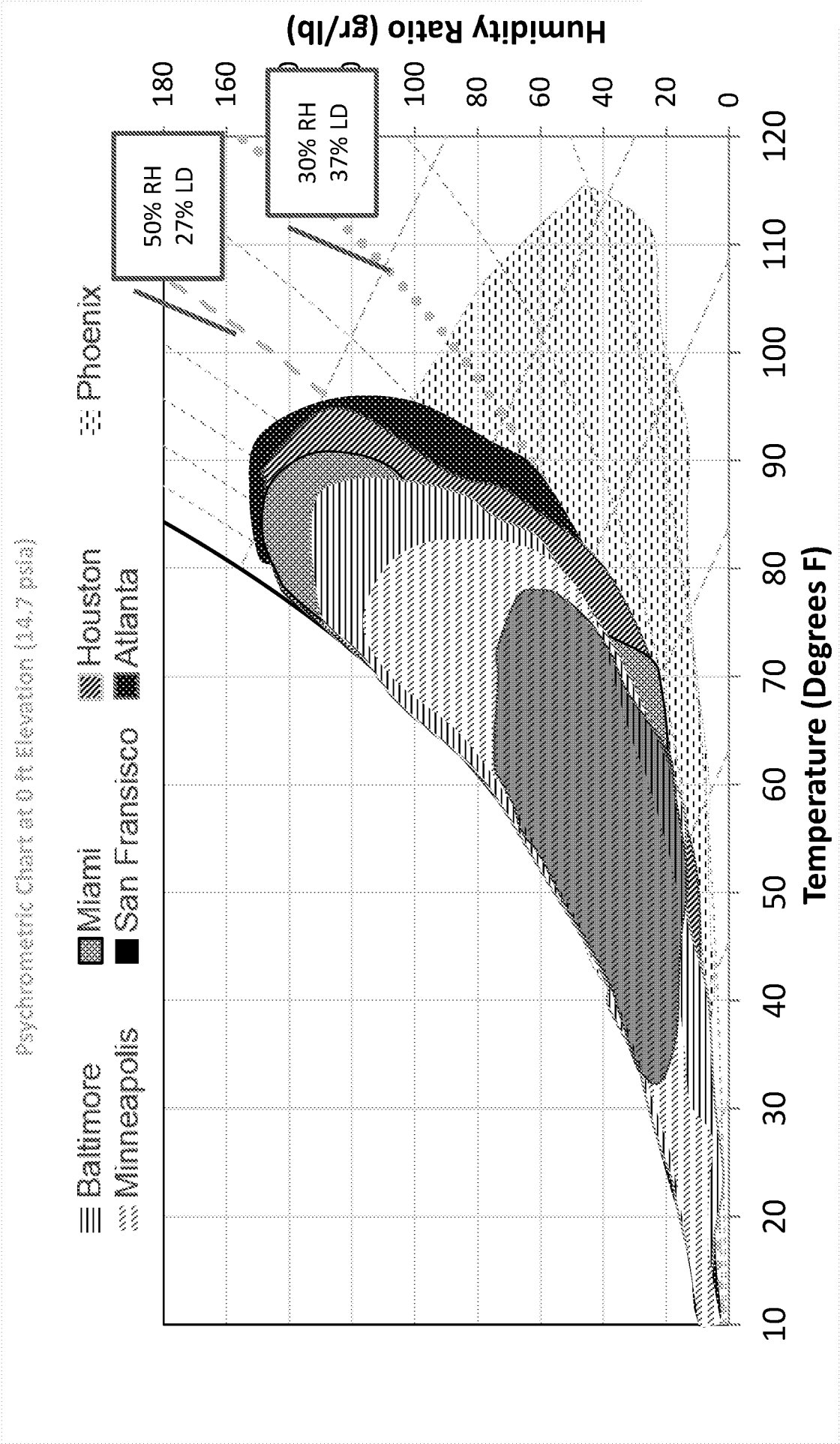


FIG. 16



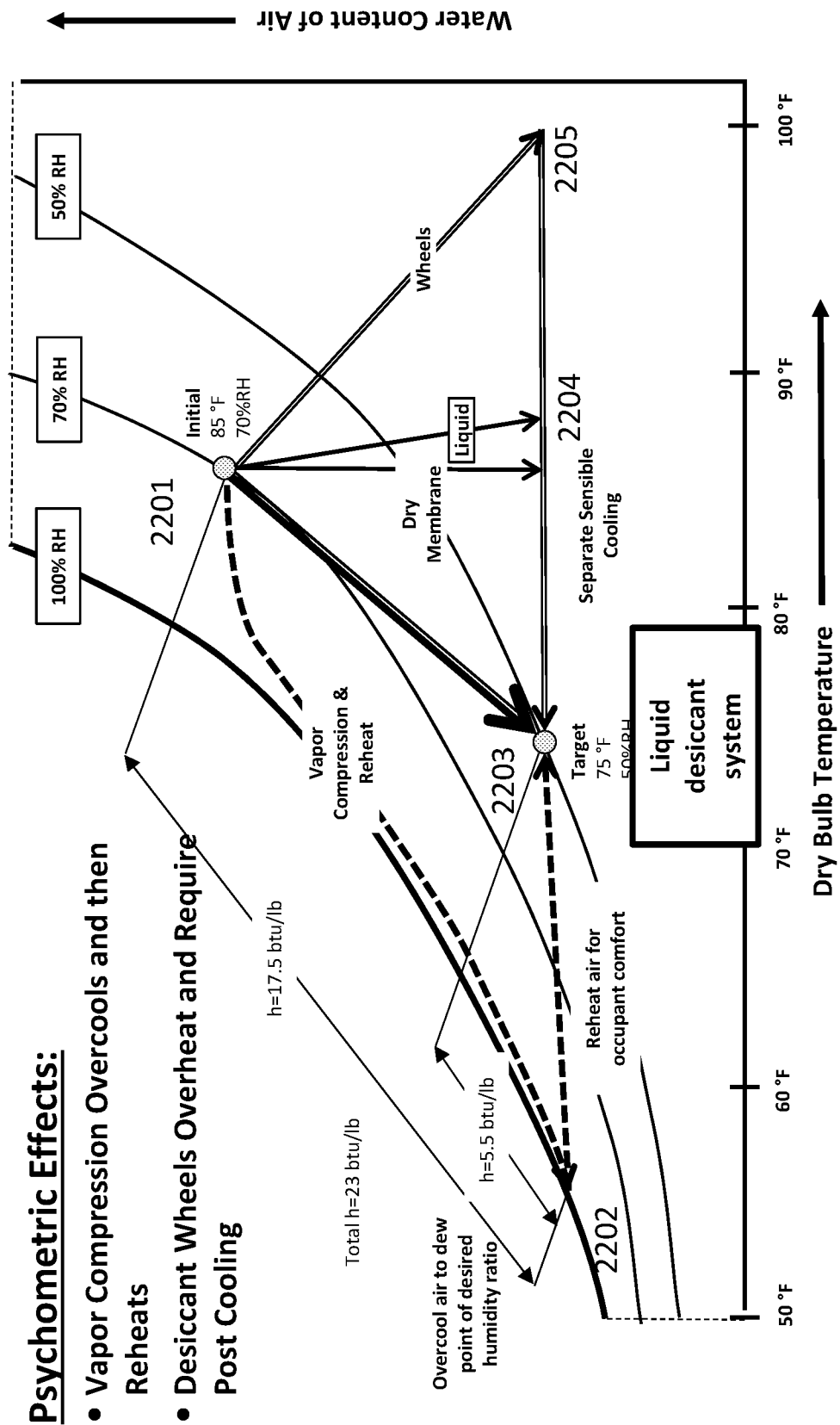


FIG. 18

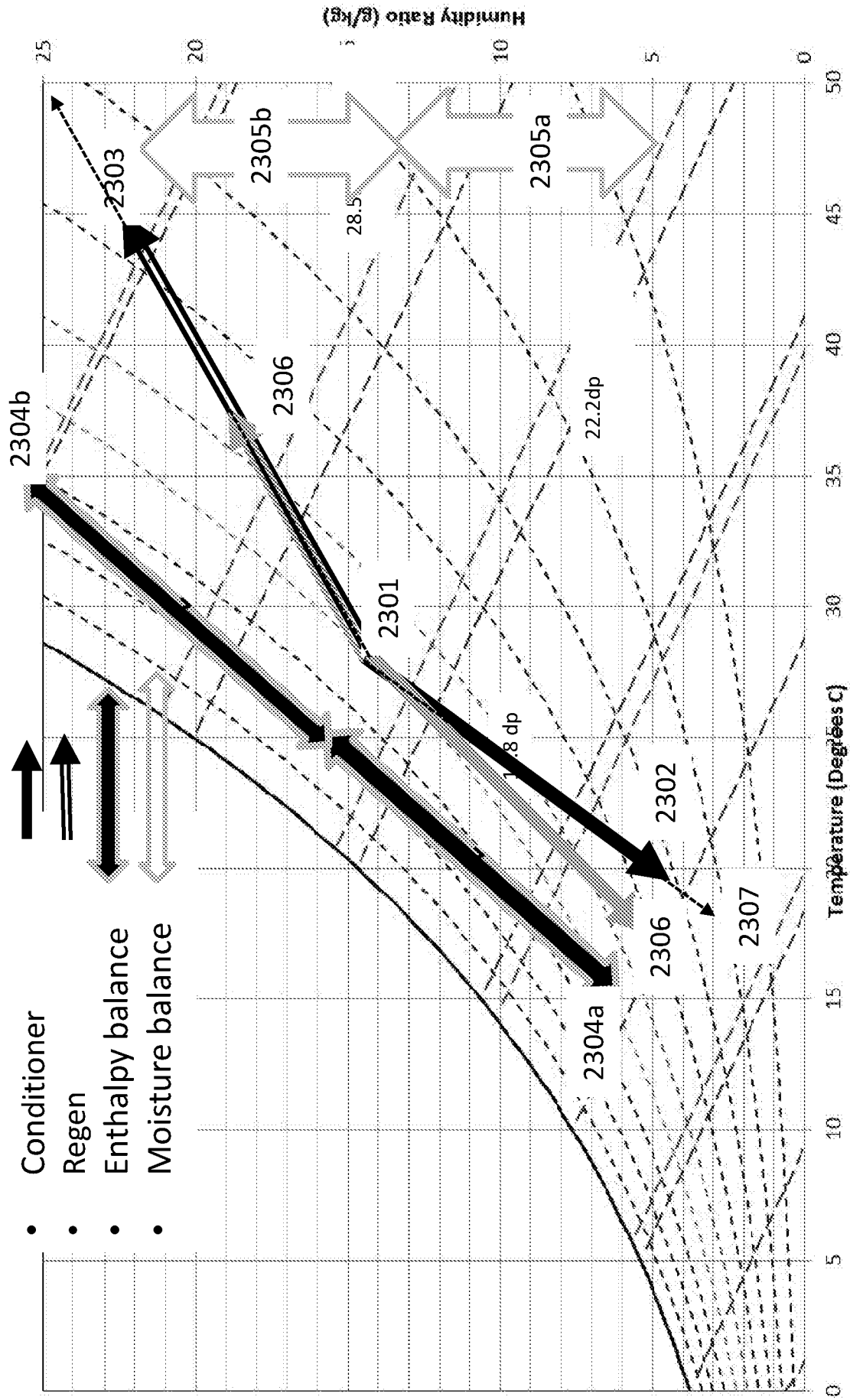


FIG. 19A

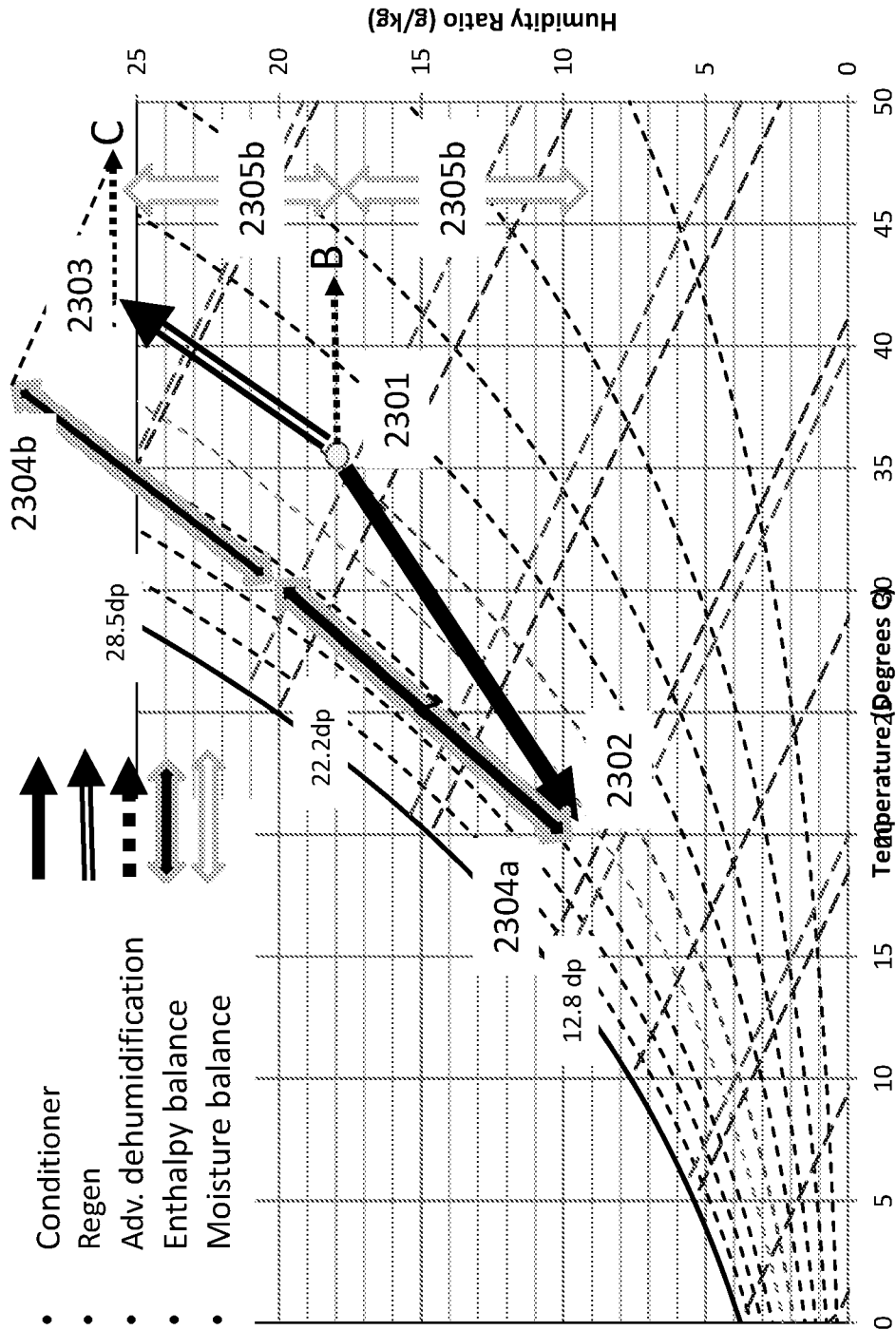


FIG. 19B

920 conditions optimization

Balance humidity and total enthalpy

Assumes higher airflow on the regen

Balanced humidity: Supply DP 12.8C, OA, 22.2C, exhaust 28.5C (blue arrows)

Balanced RH: RH exhaust (brown) = RH supply -5-10% (green)

Balanced Enthalpy: regen @ 120% enthalpy is balanced at 80% of the delta enthalpy (see red arrows)

Regen/conditioner at balanced humidity and RH are not balanced in enthalpy at A. Therefore additional heating is required using a heat dump (B) or a special dehum. Coil @

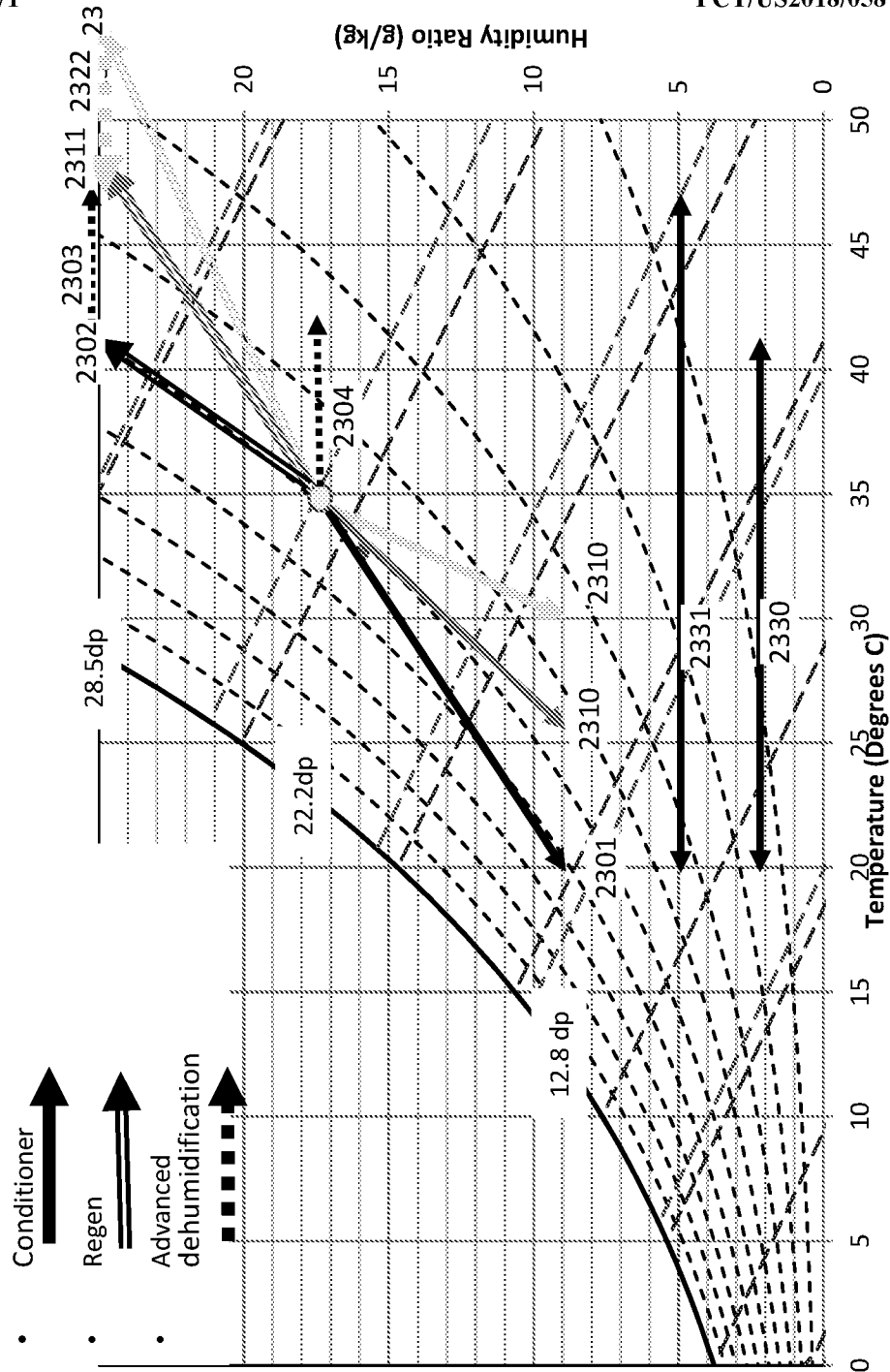


FIG. 19C

920A to 70F/21C requires a heat dump or the advanced dehumidification coil as extra heating coil.

Supply at 80F/26C is roughly in balance at 150% regen air.

Supply at 90F/32C requires the advanced dehumidification coil as extra cooling coil.

Total condenser heat > condenser heat in balance,

Therefore:

total cooling > total cooling in  
balance

**increasing DB above the balance point increases total power in exchange for a slightly lower lift.**

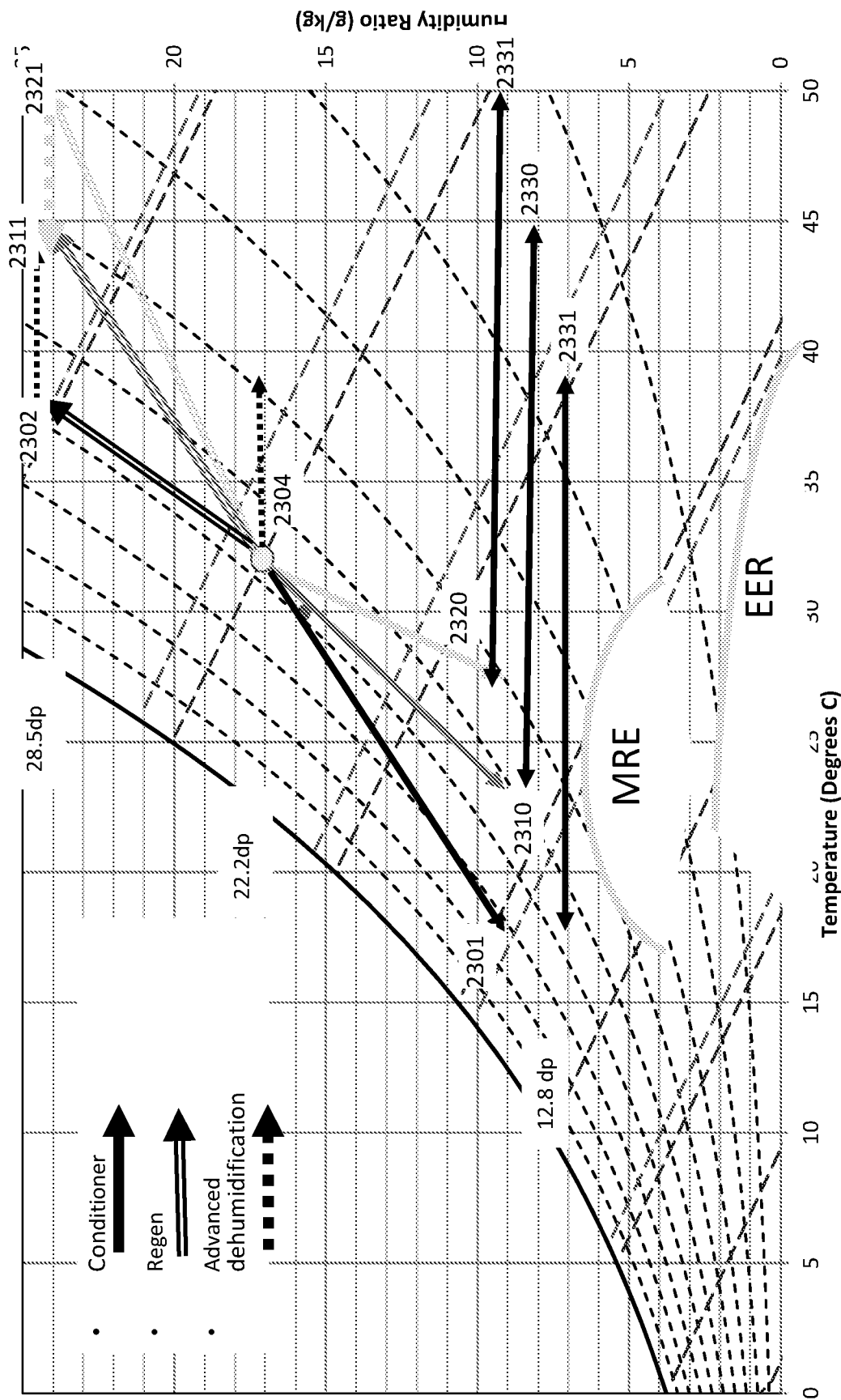


FIG. 19D

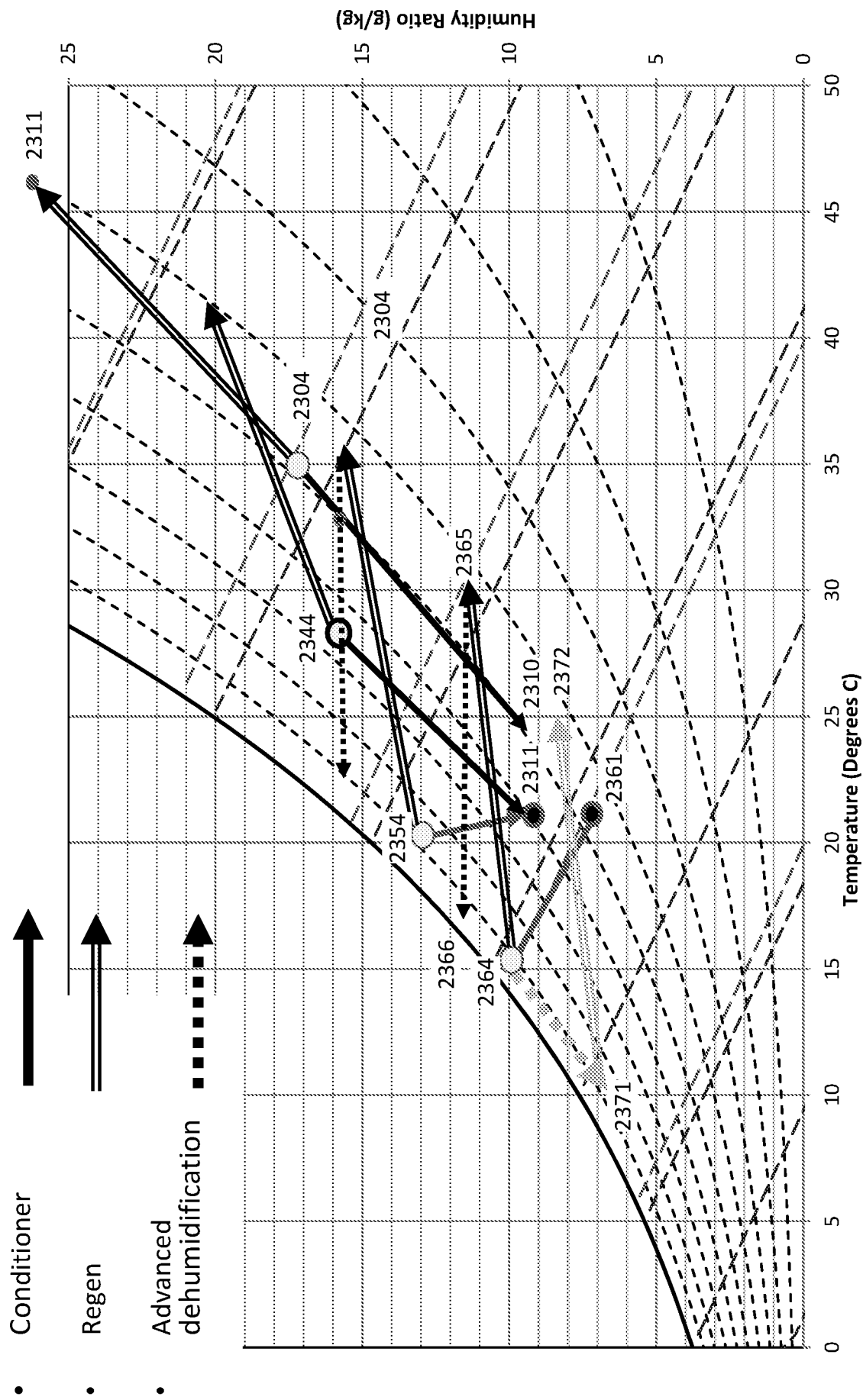


FIG. 19E

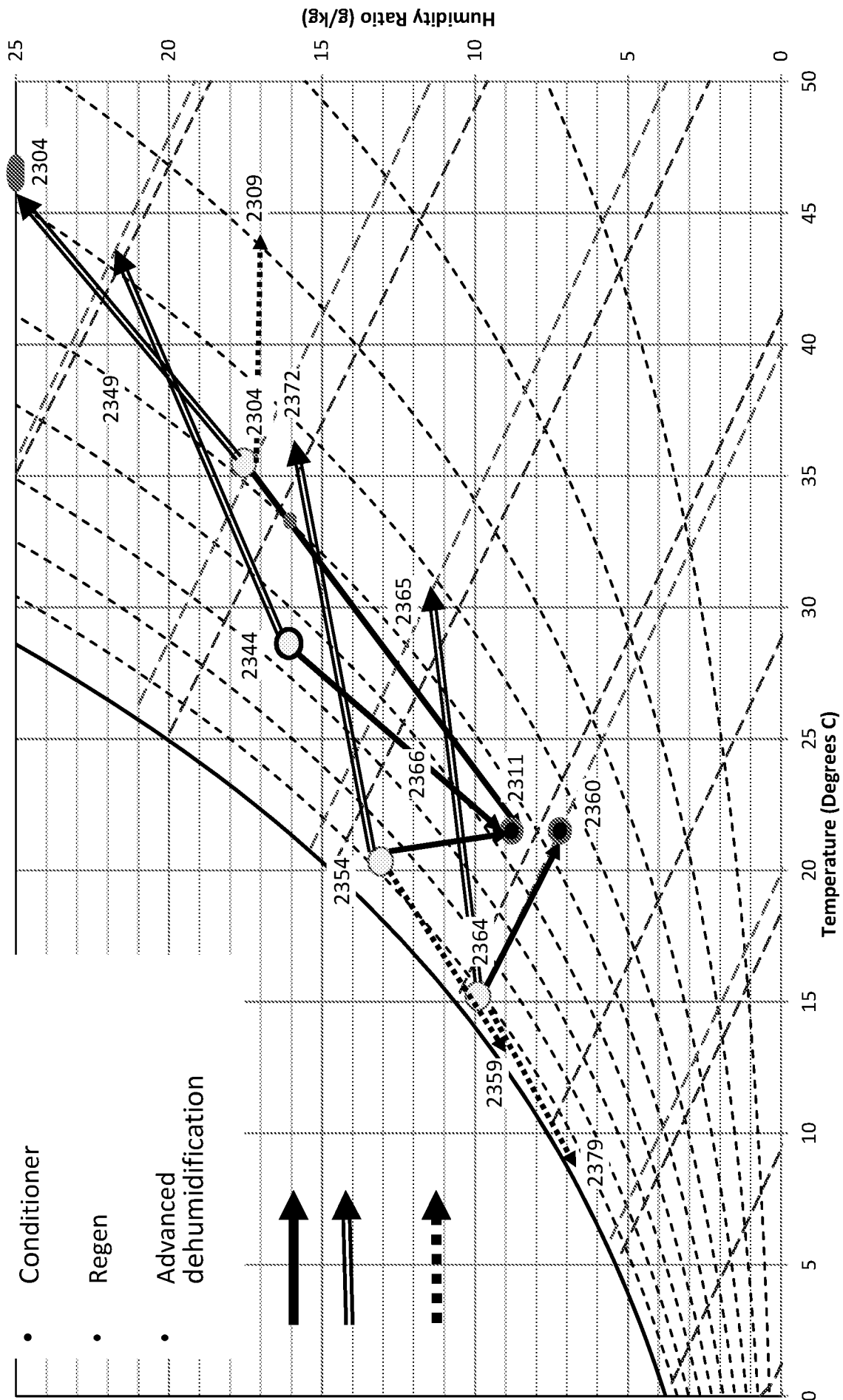


FIG. 19F

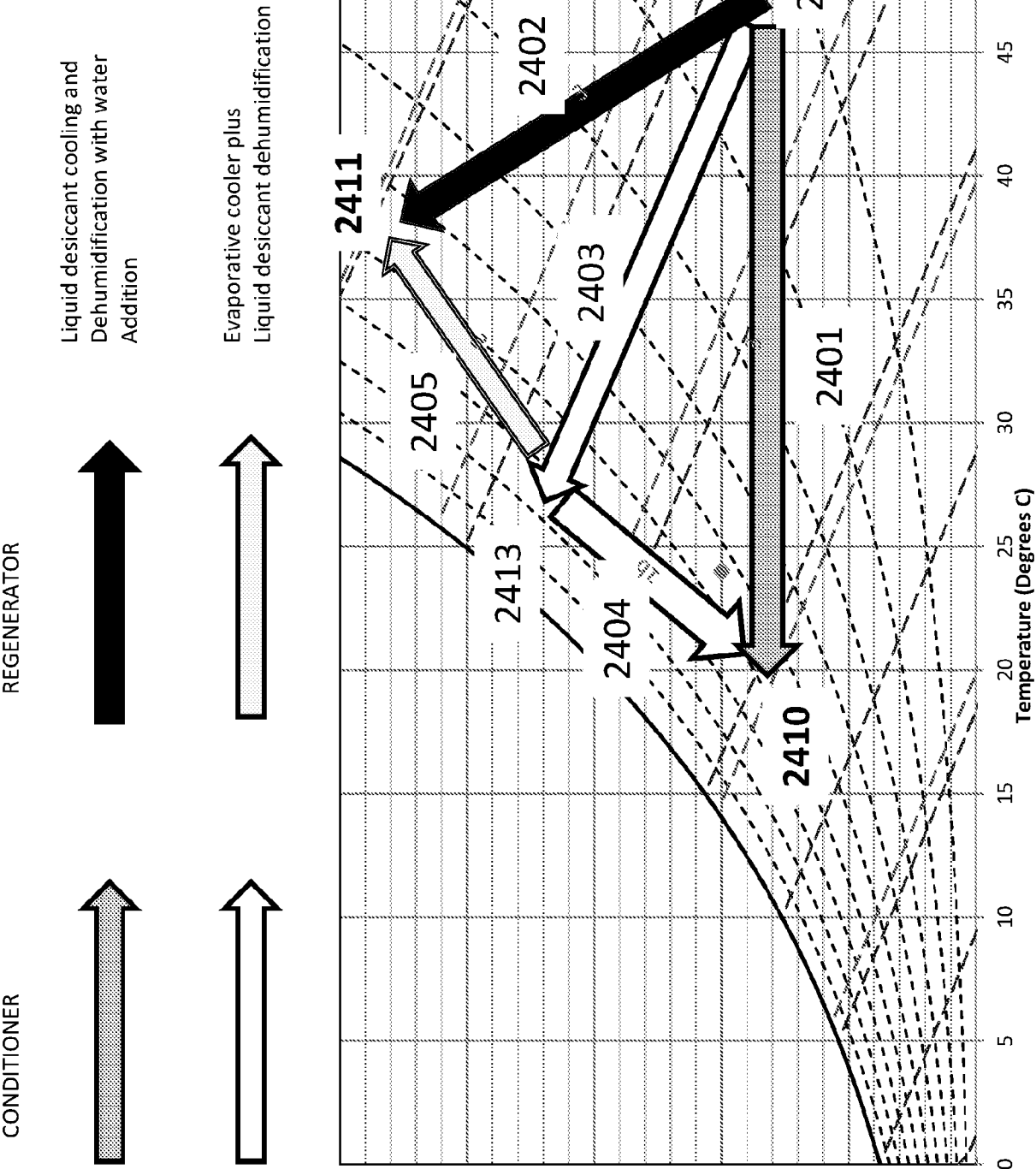


FIG. 20

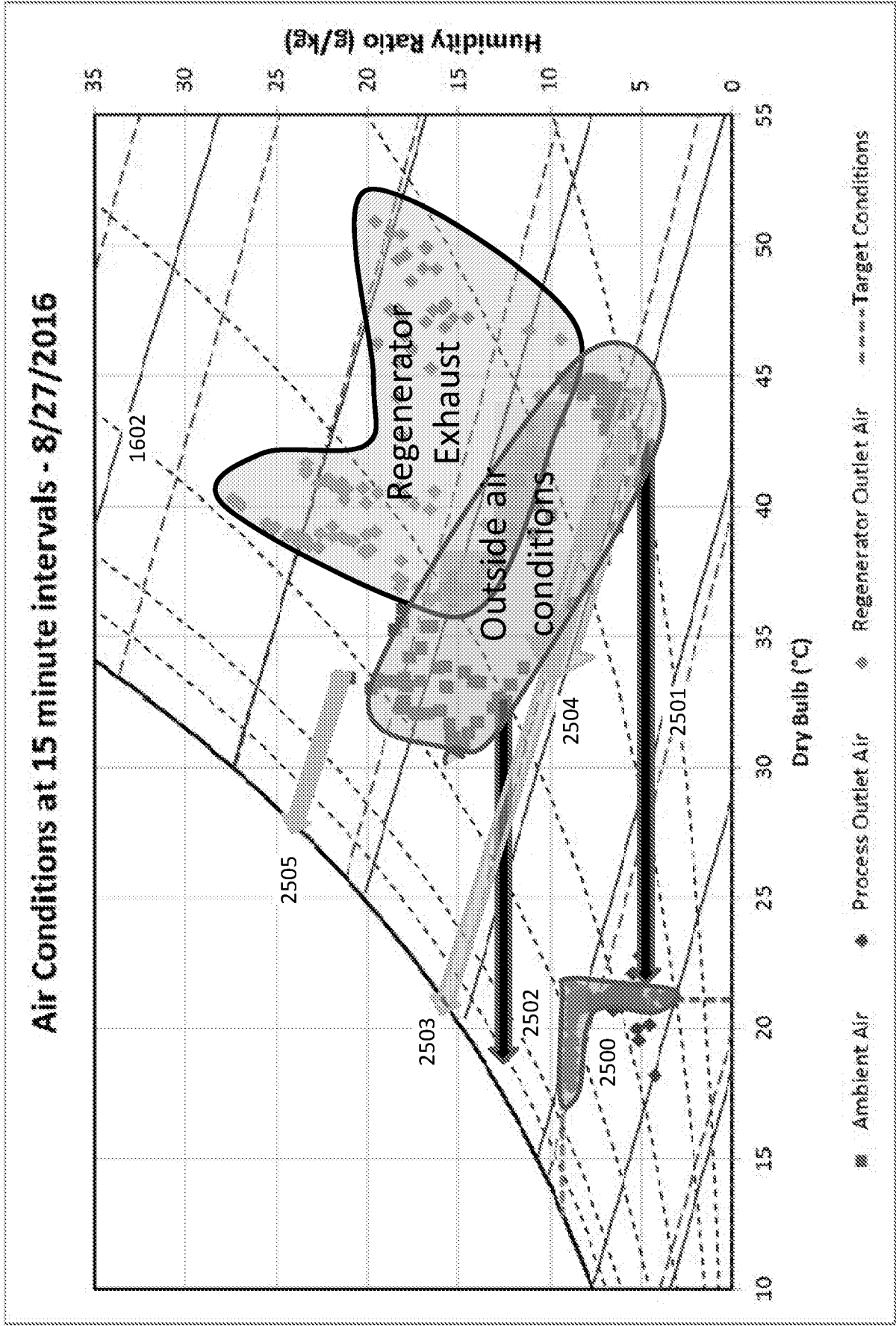


FIG. 21

FIG. 22

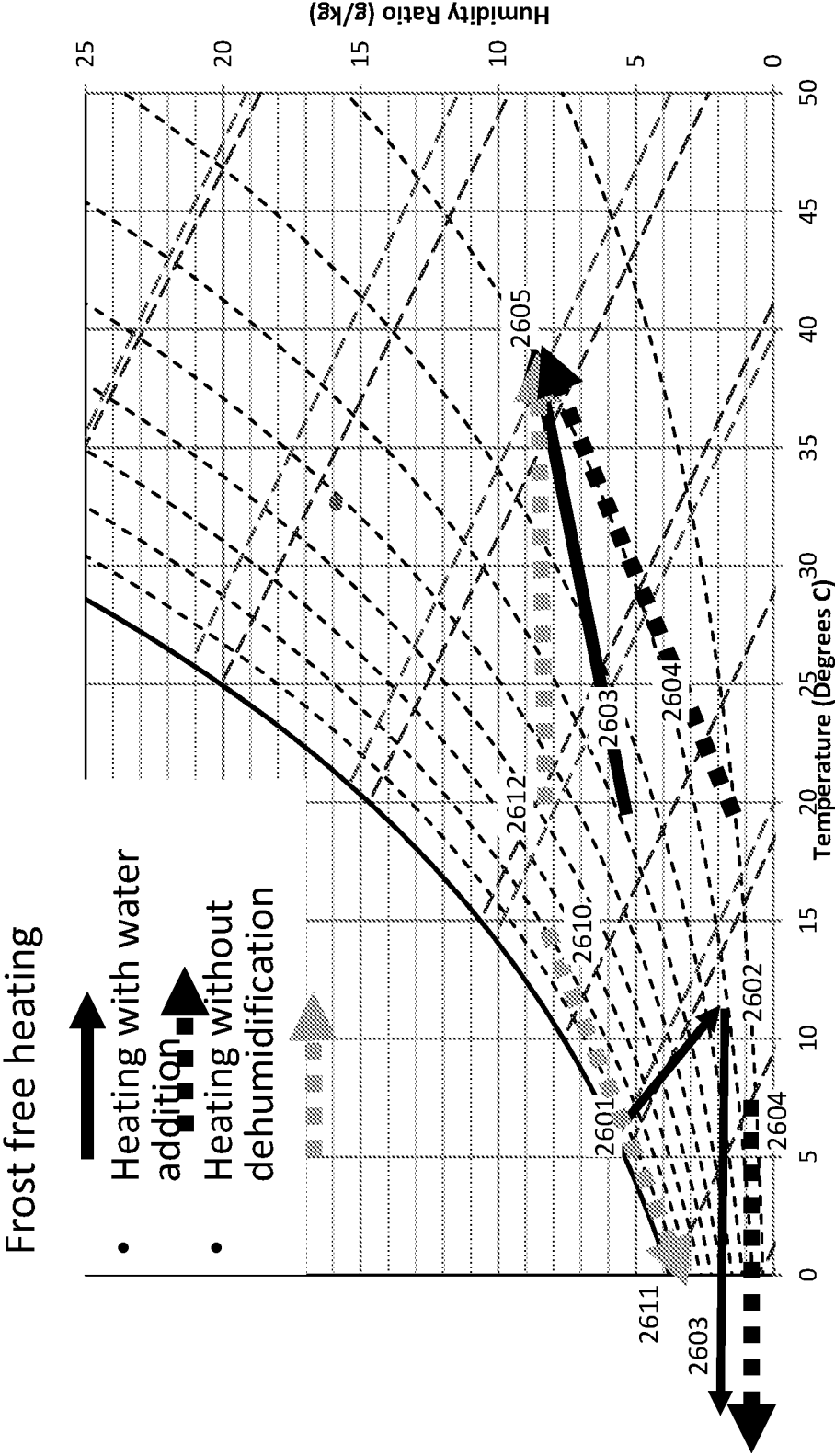


FIG. 23A

Full climate solution, including evaporative cooling option and deep defrost capacity to maximize heating potential in moderate but humid winters

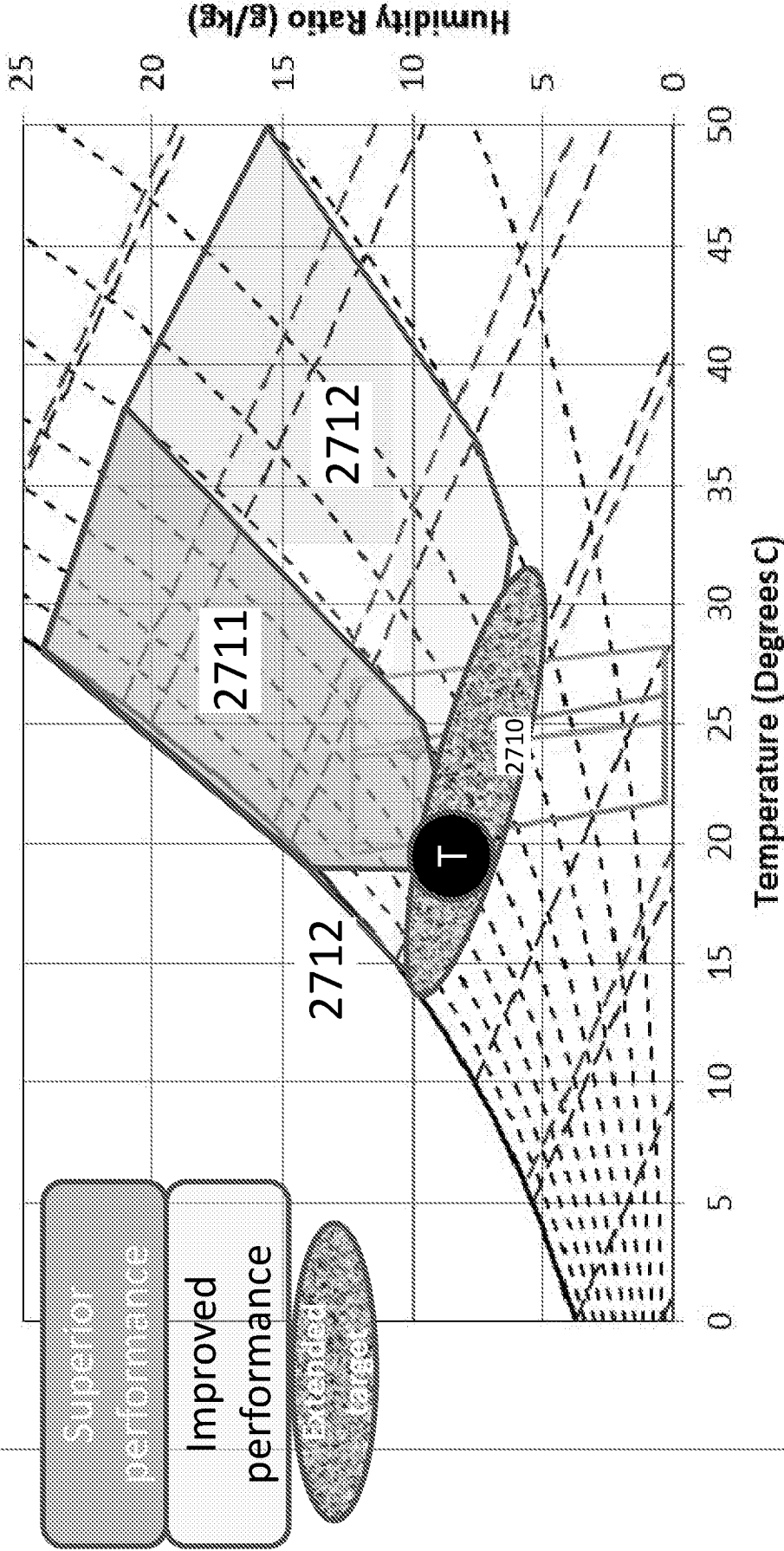


FIG. 23B

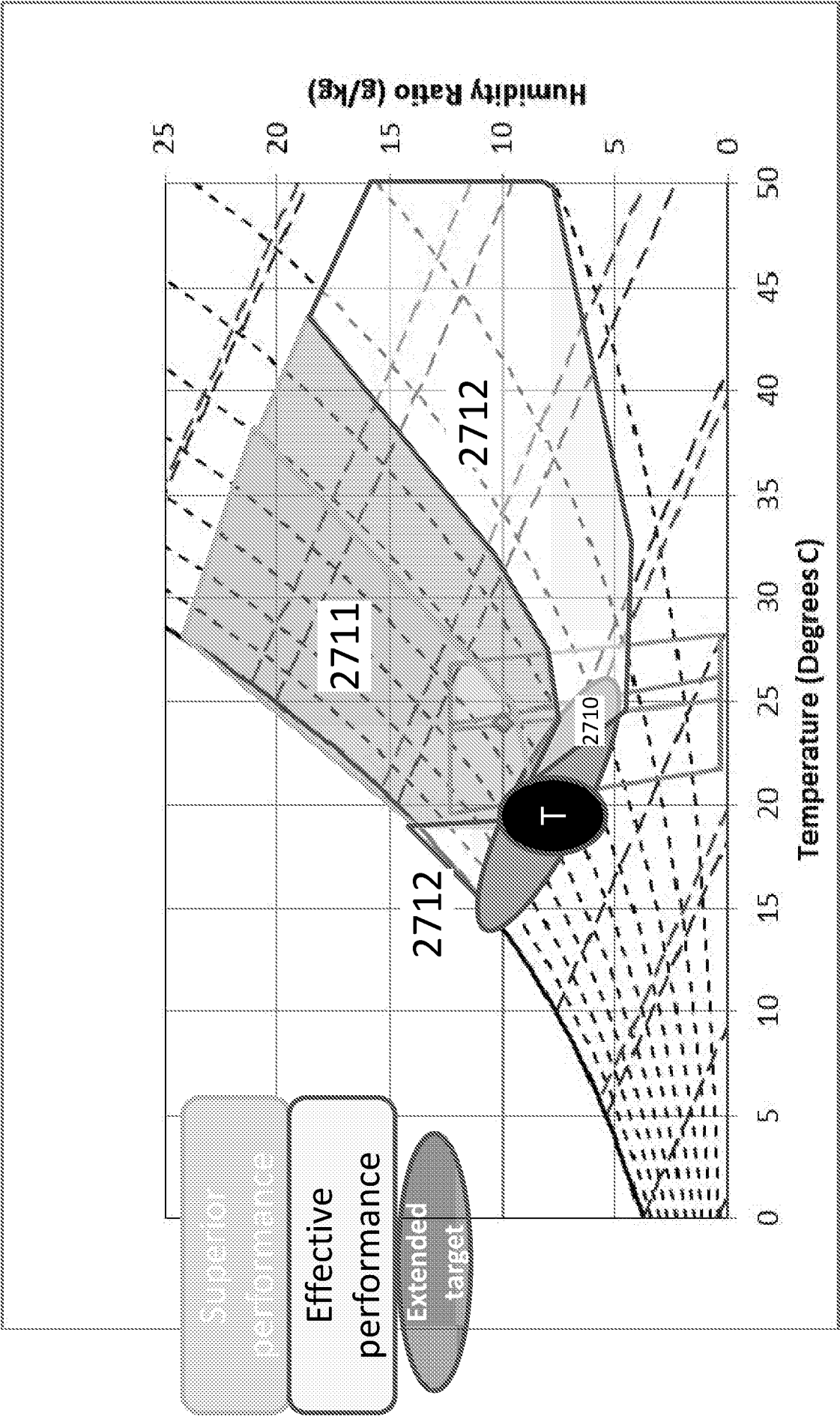


FIG. 23C

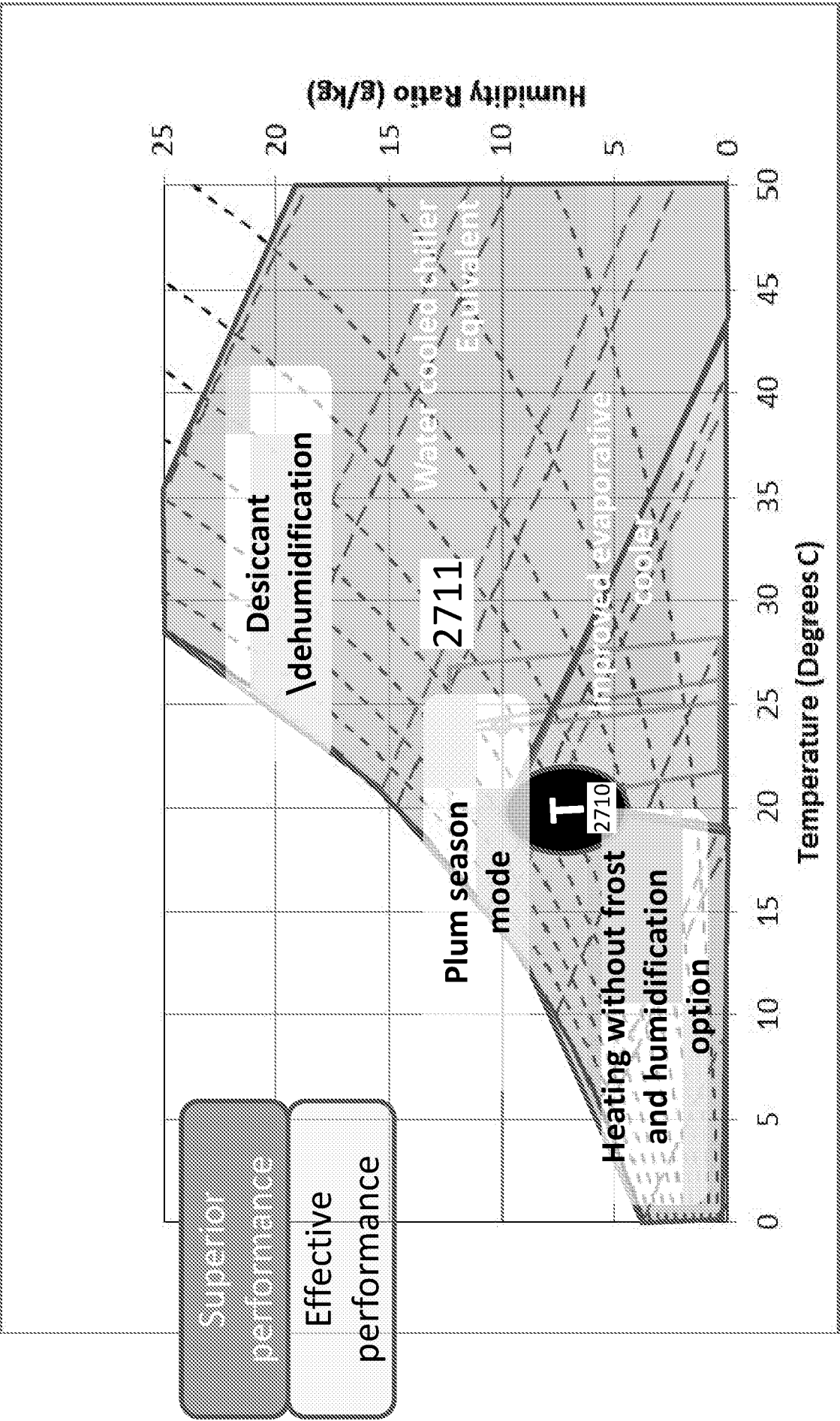


FIG. 24

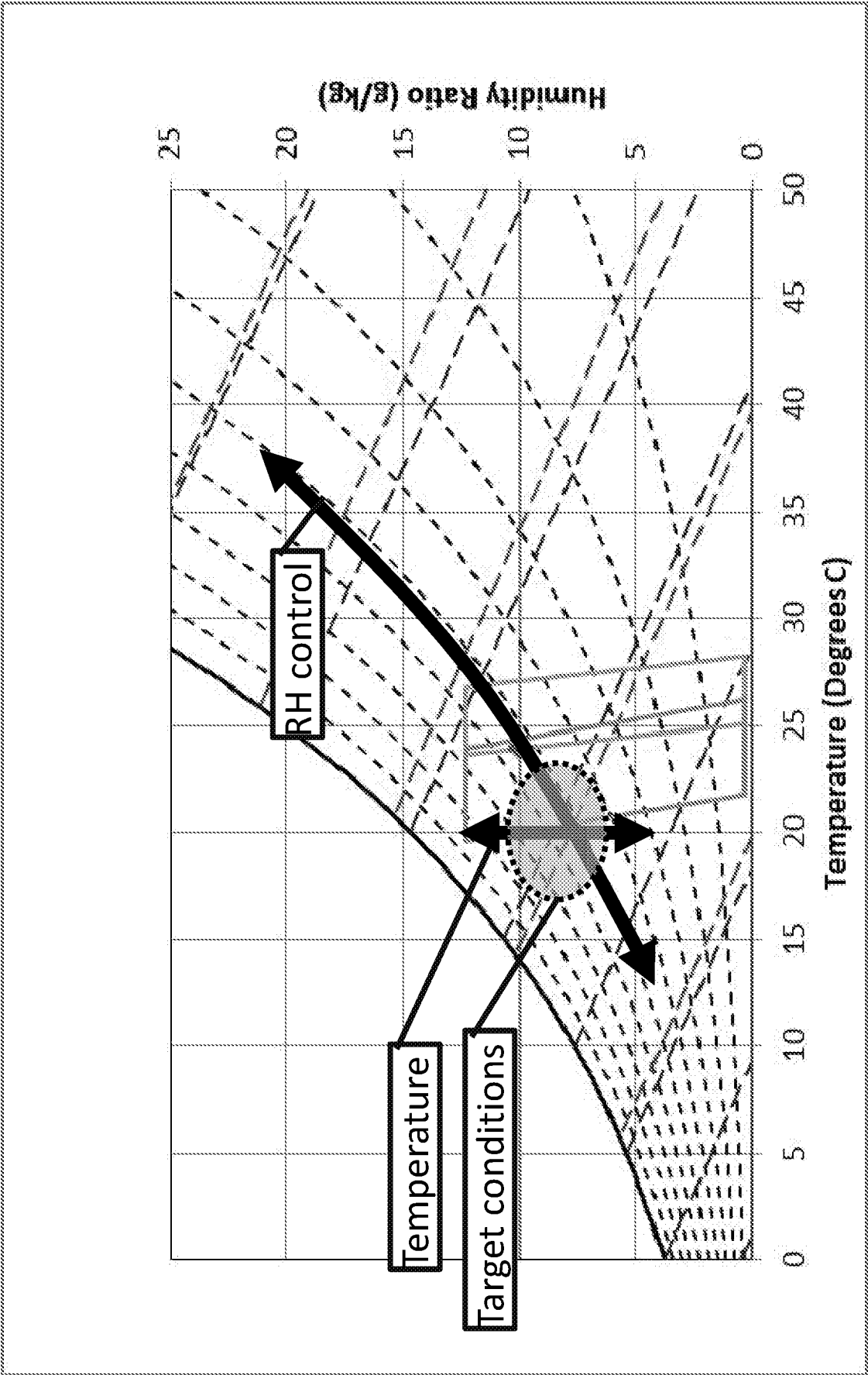


FIG. 25

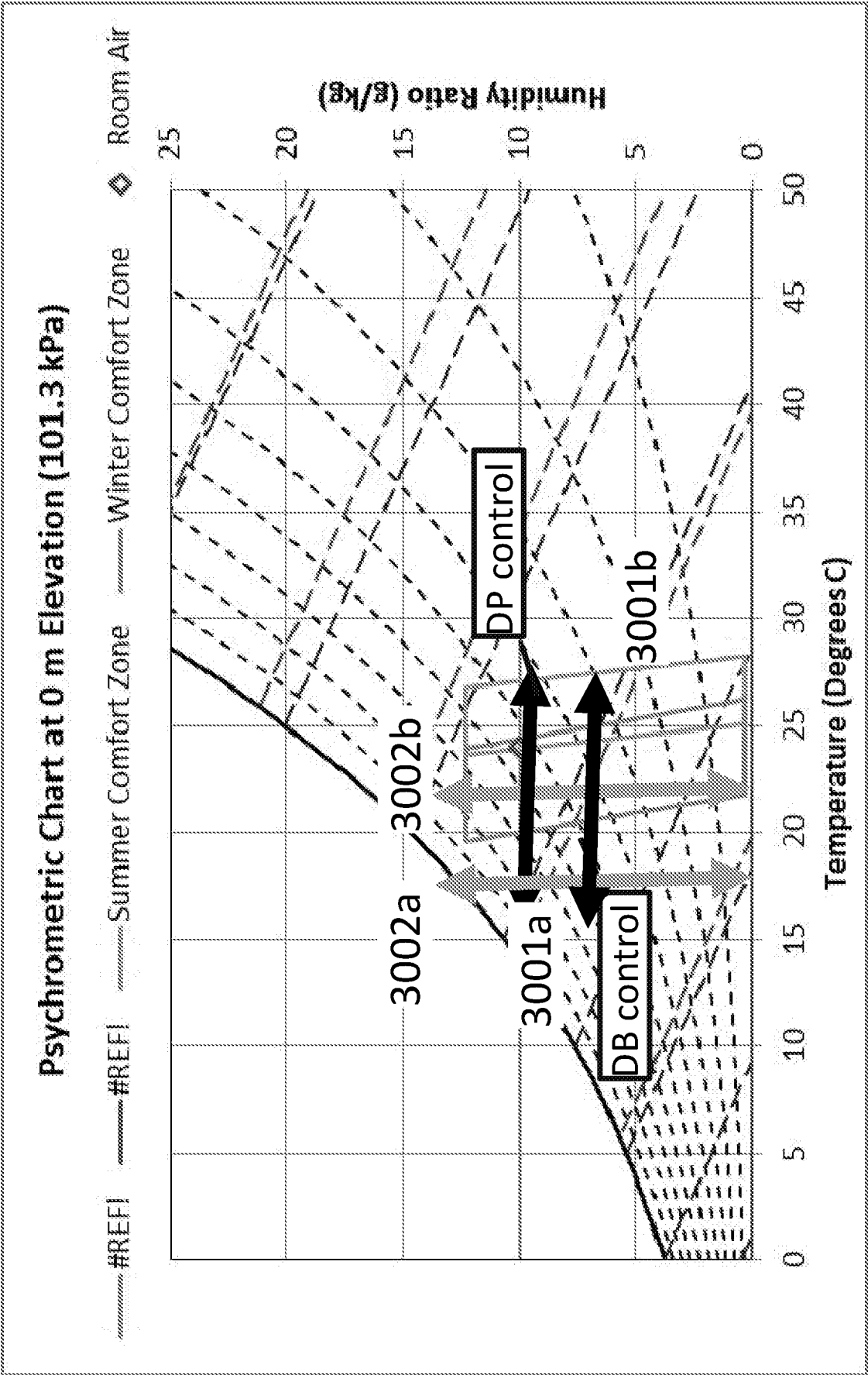
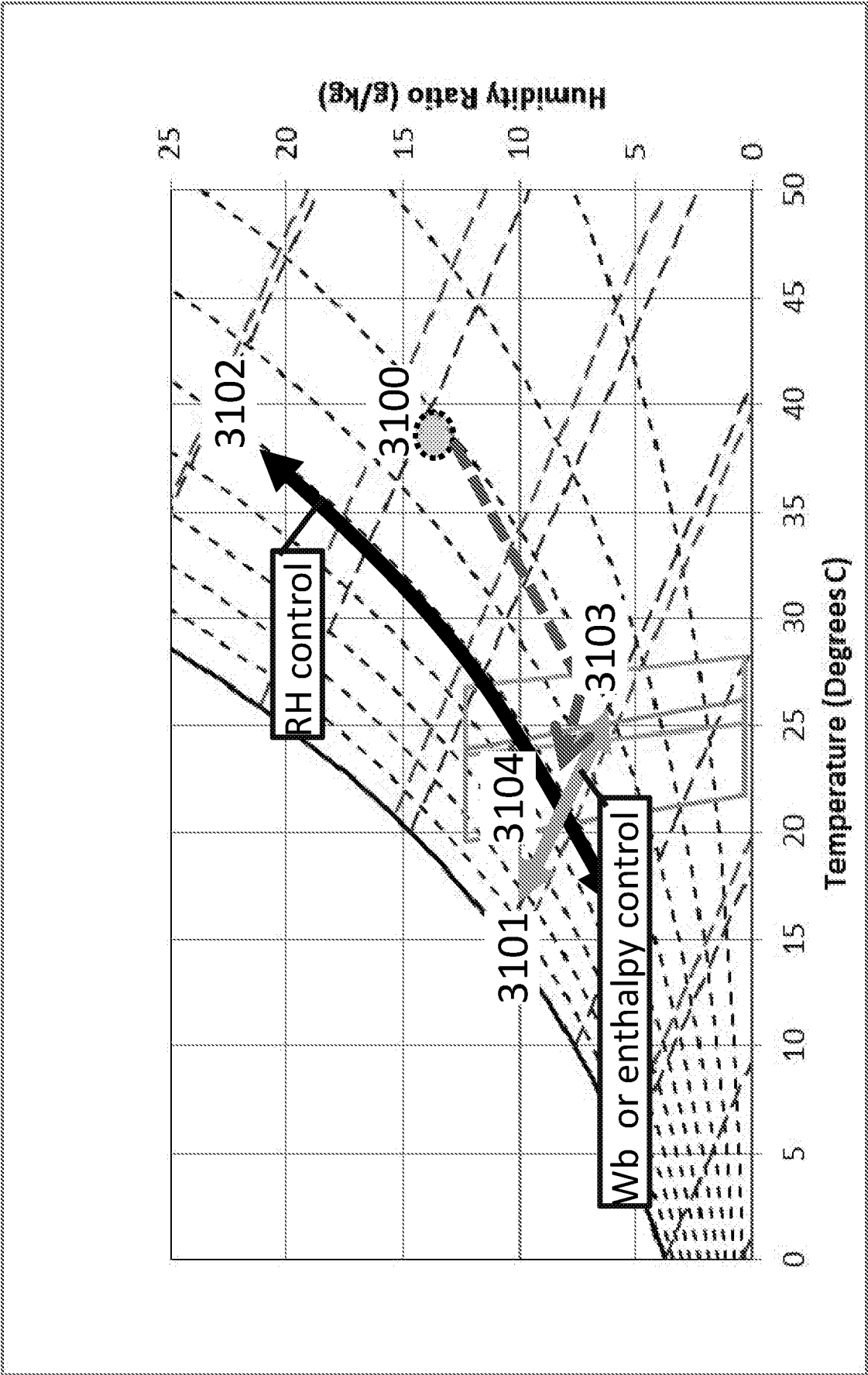


FIG. 26



**FIG. 27**

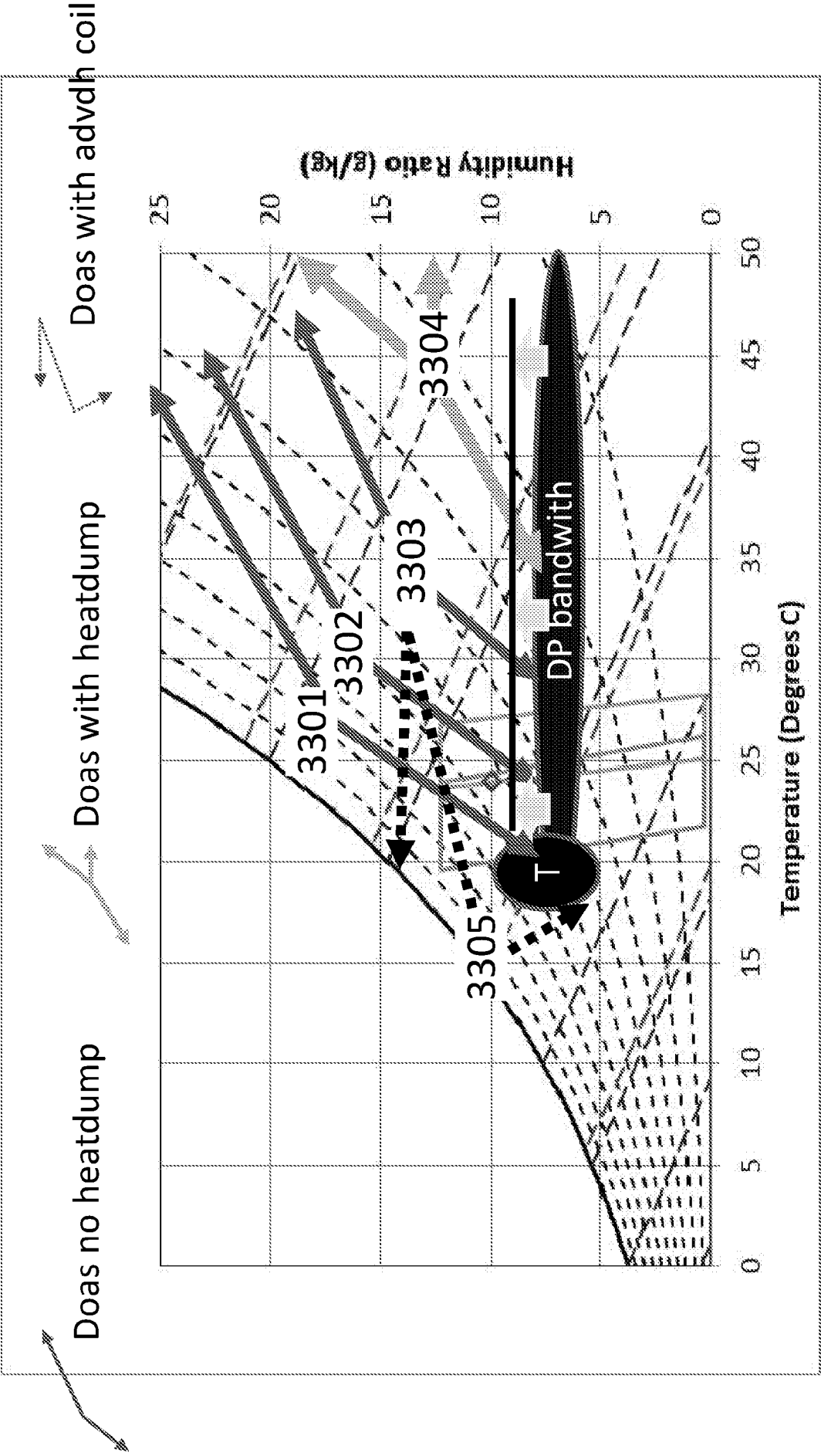
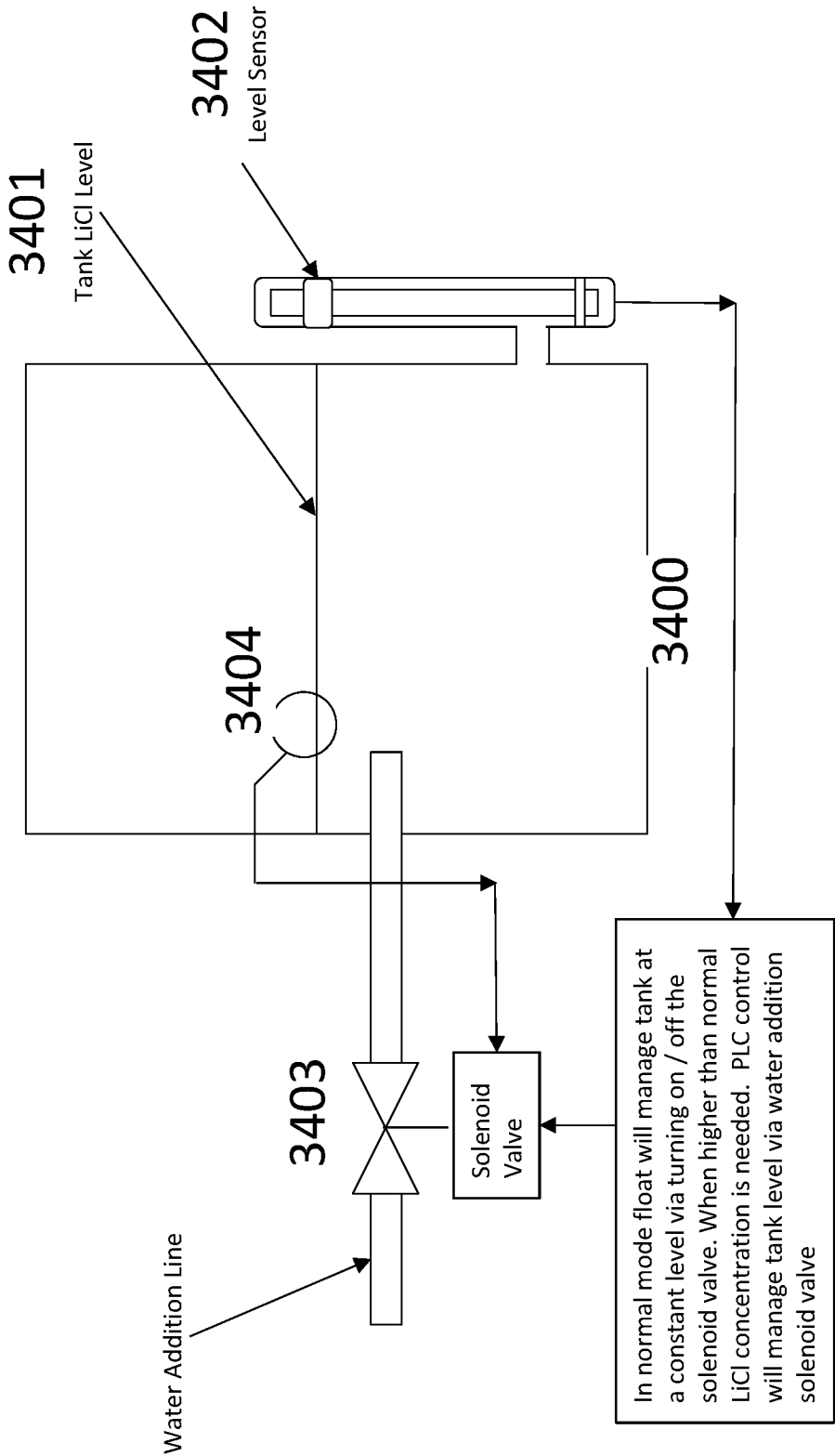
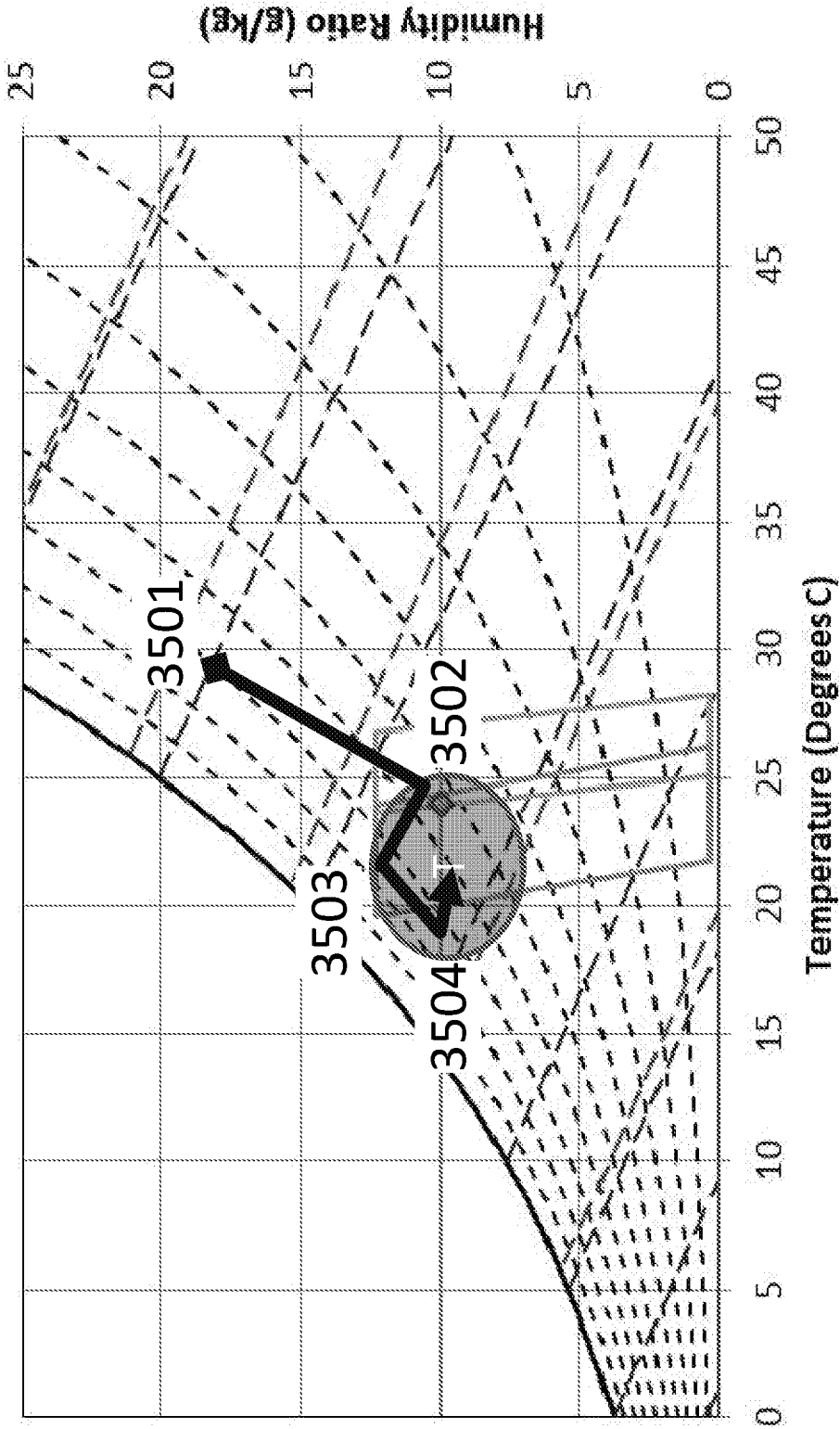


FIG. 28



**FIG. 29**

Compressor changes the total enthalpy  
Water addition, regen and acc fans change the SHR  
Concentration drives RH



**FIG. 30A**

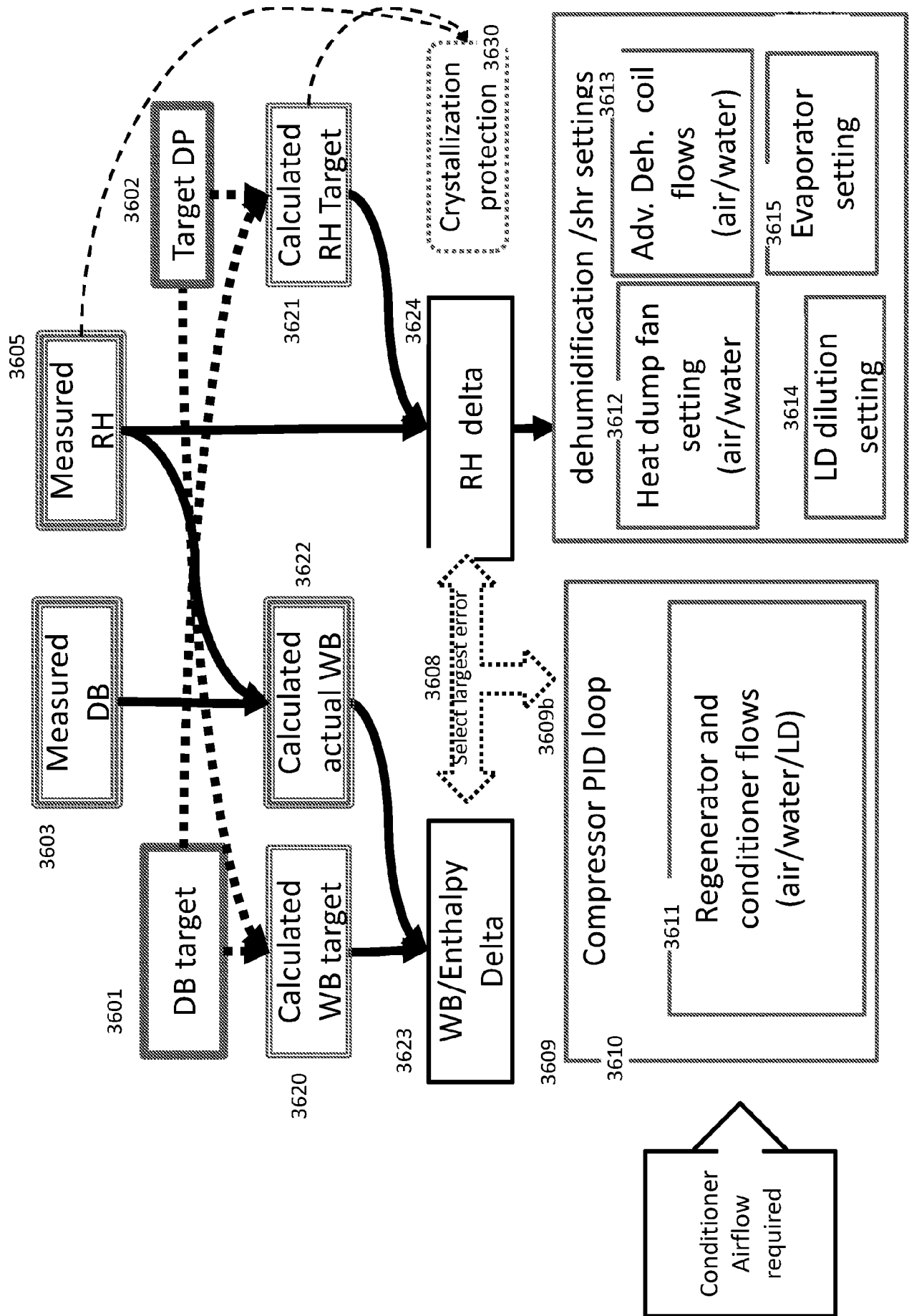
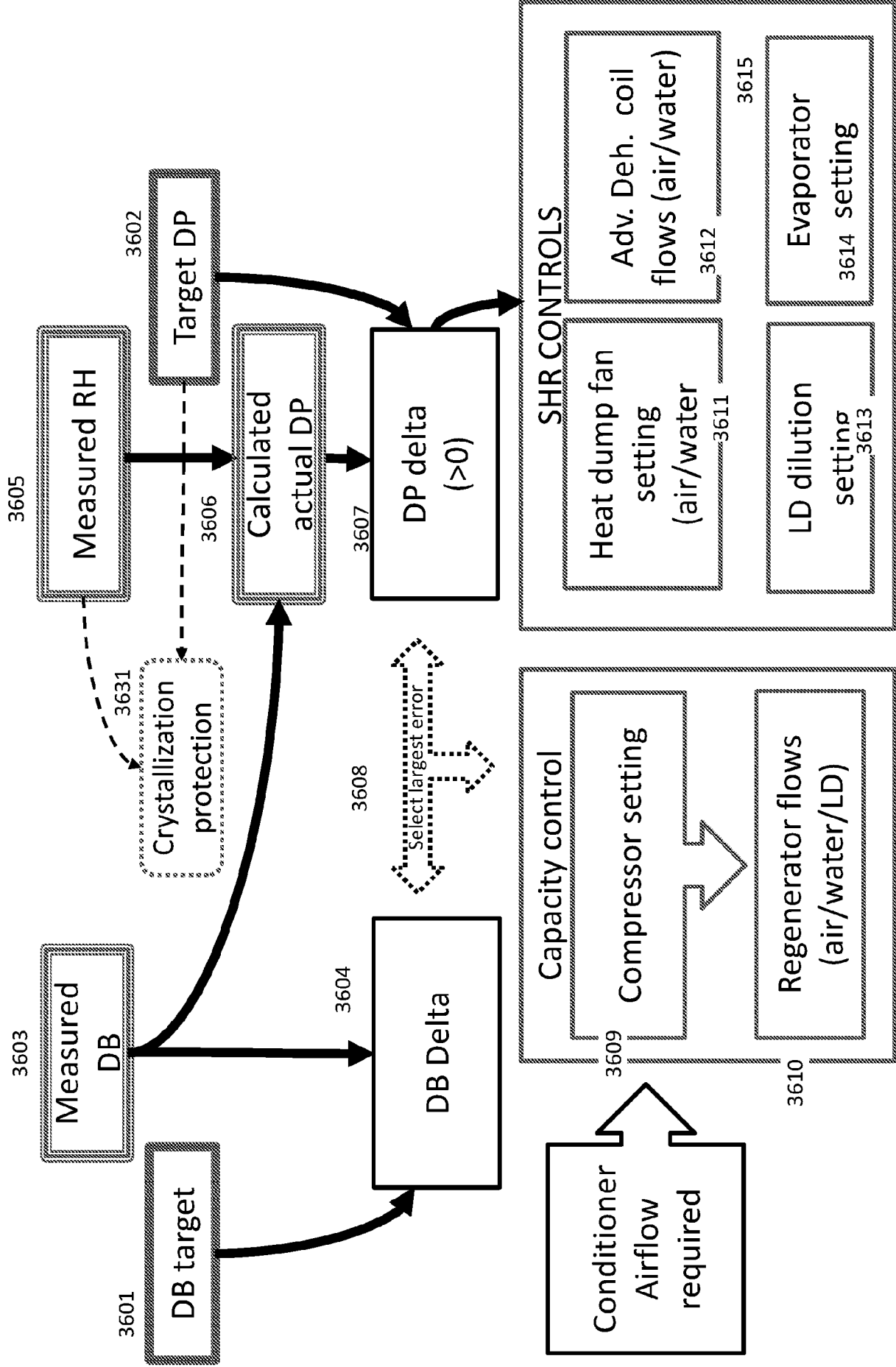


FIG. 30B



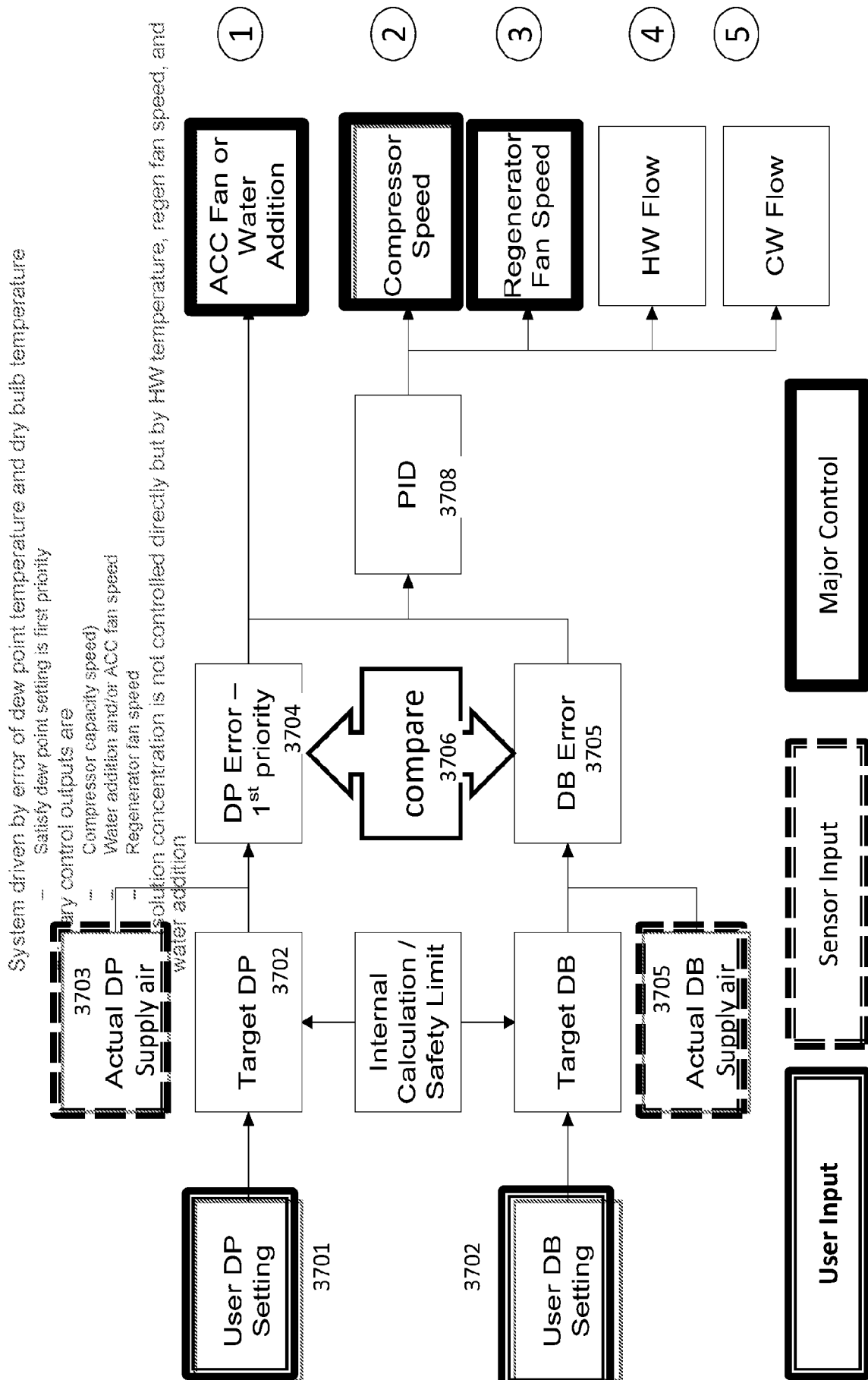
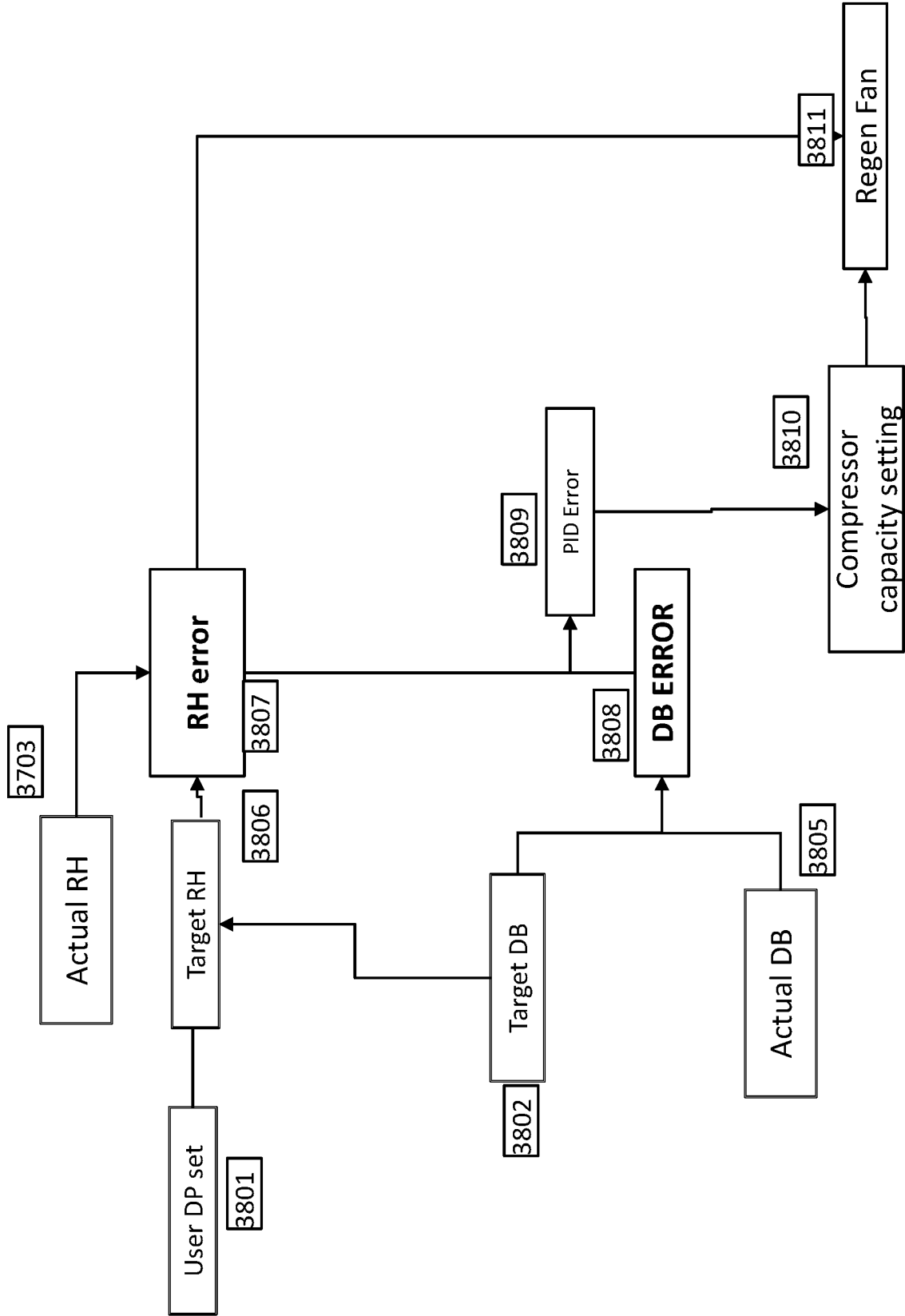
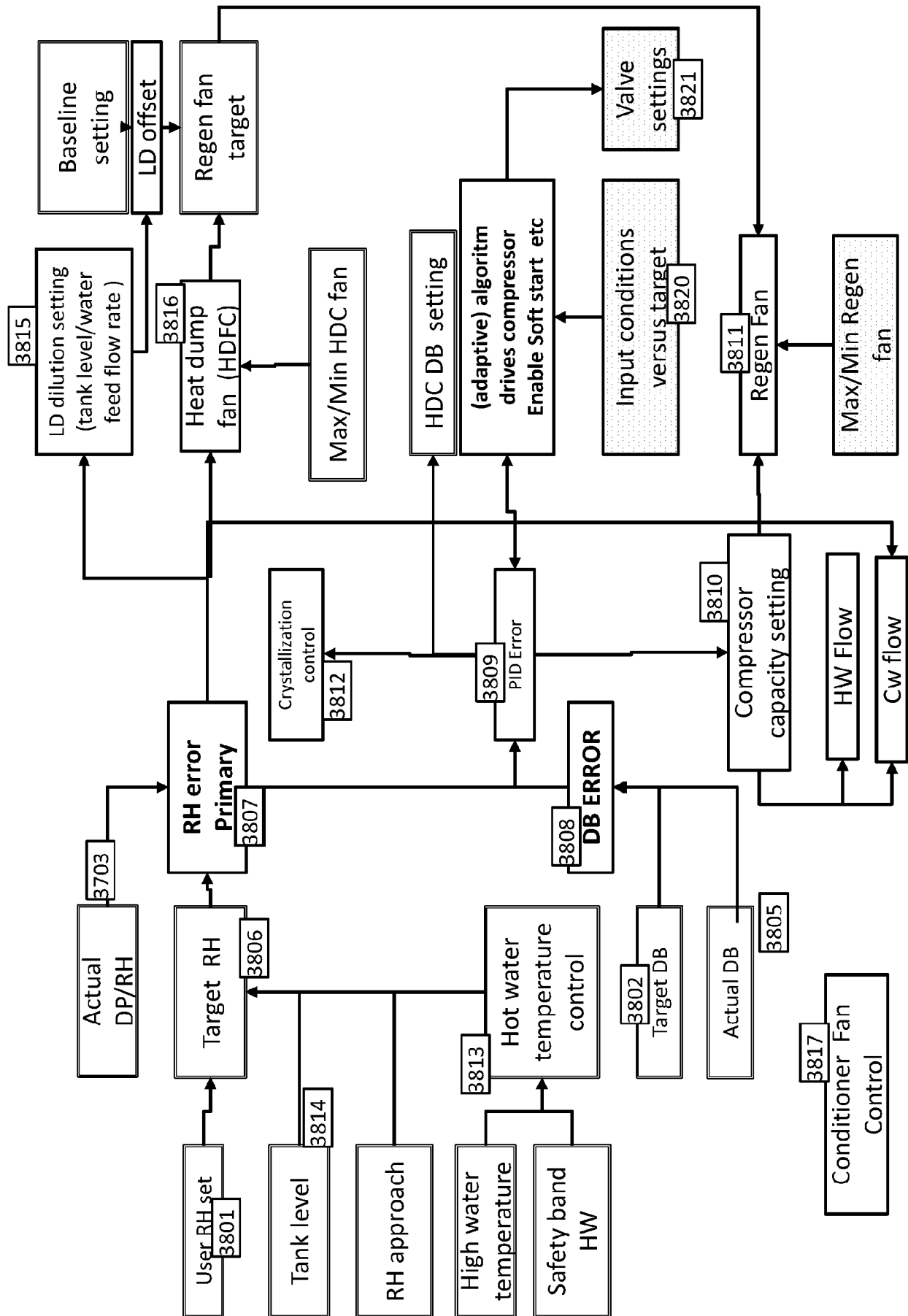
**FIG. 31**

FIG. 32A



**FIG. 32B**

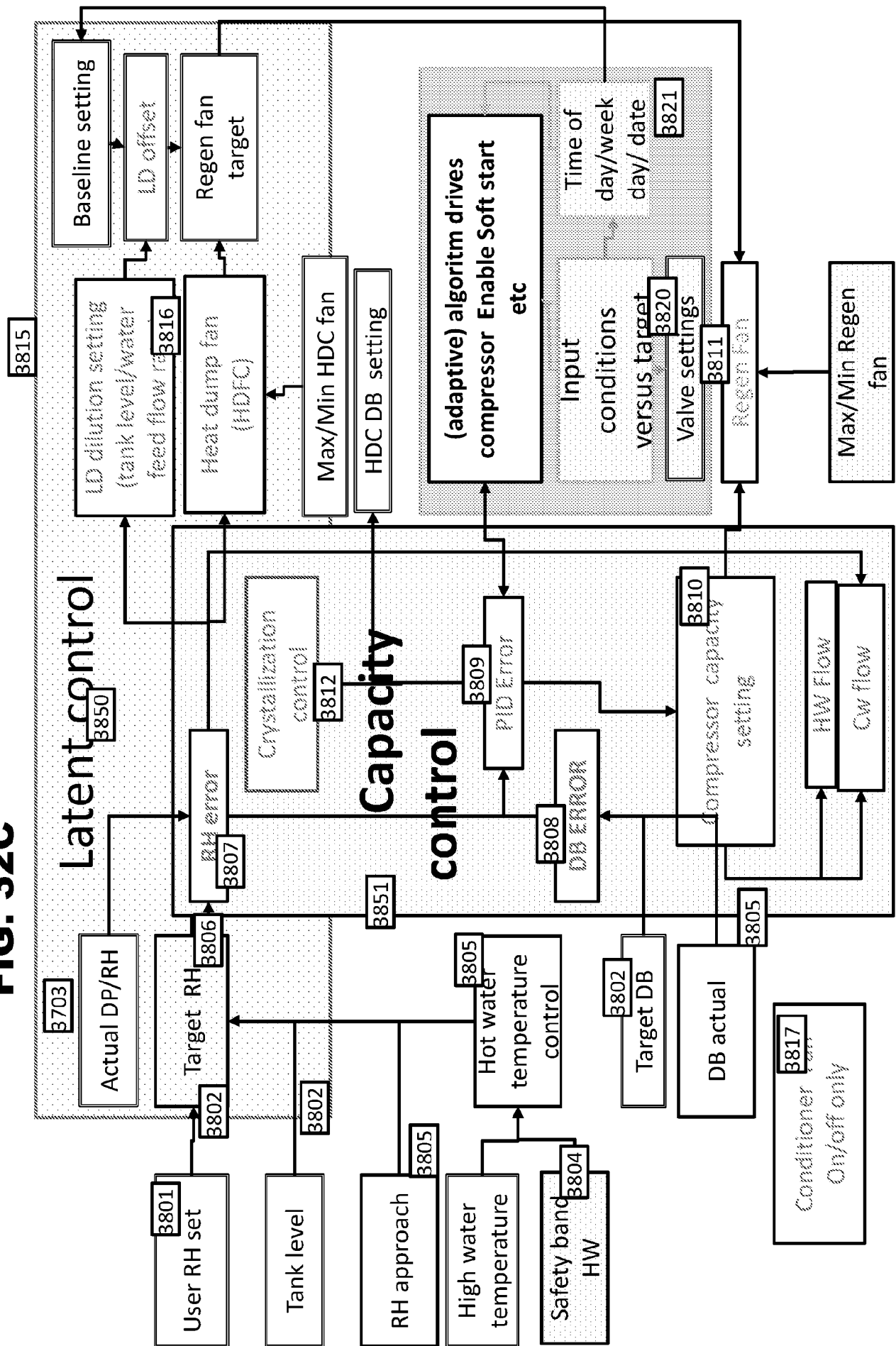
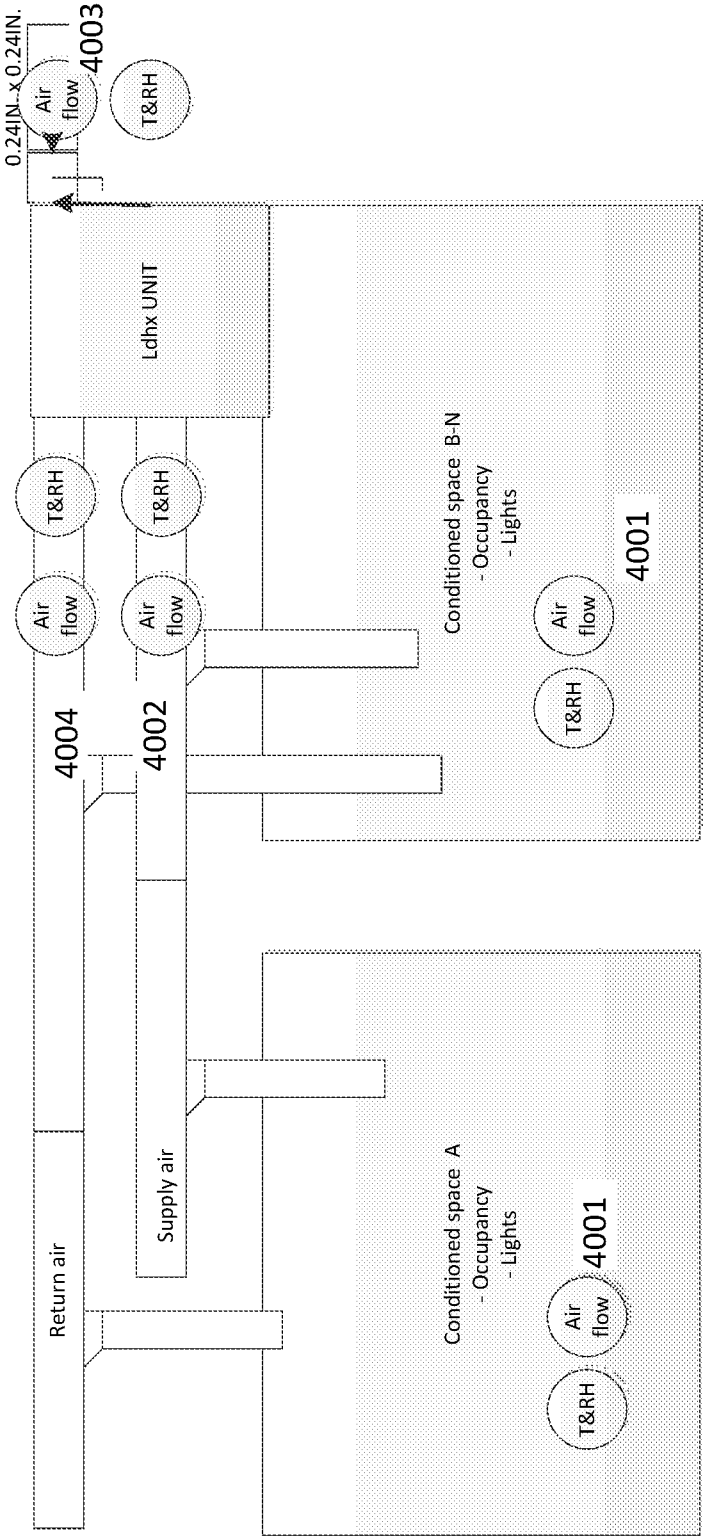
**FIG. 32C**

FIG. 33

Outside air only or outside air mixed with return air  
Driven by supply versus targets conditions Or space versus target conditions  
Including or excluding present input/ outside air conditions for predictive settings  
Including or excluding expected input/outside air conditions



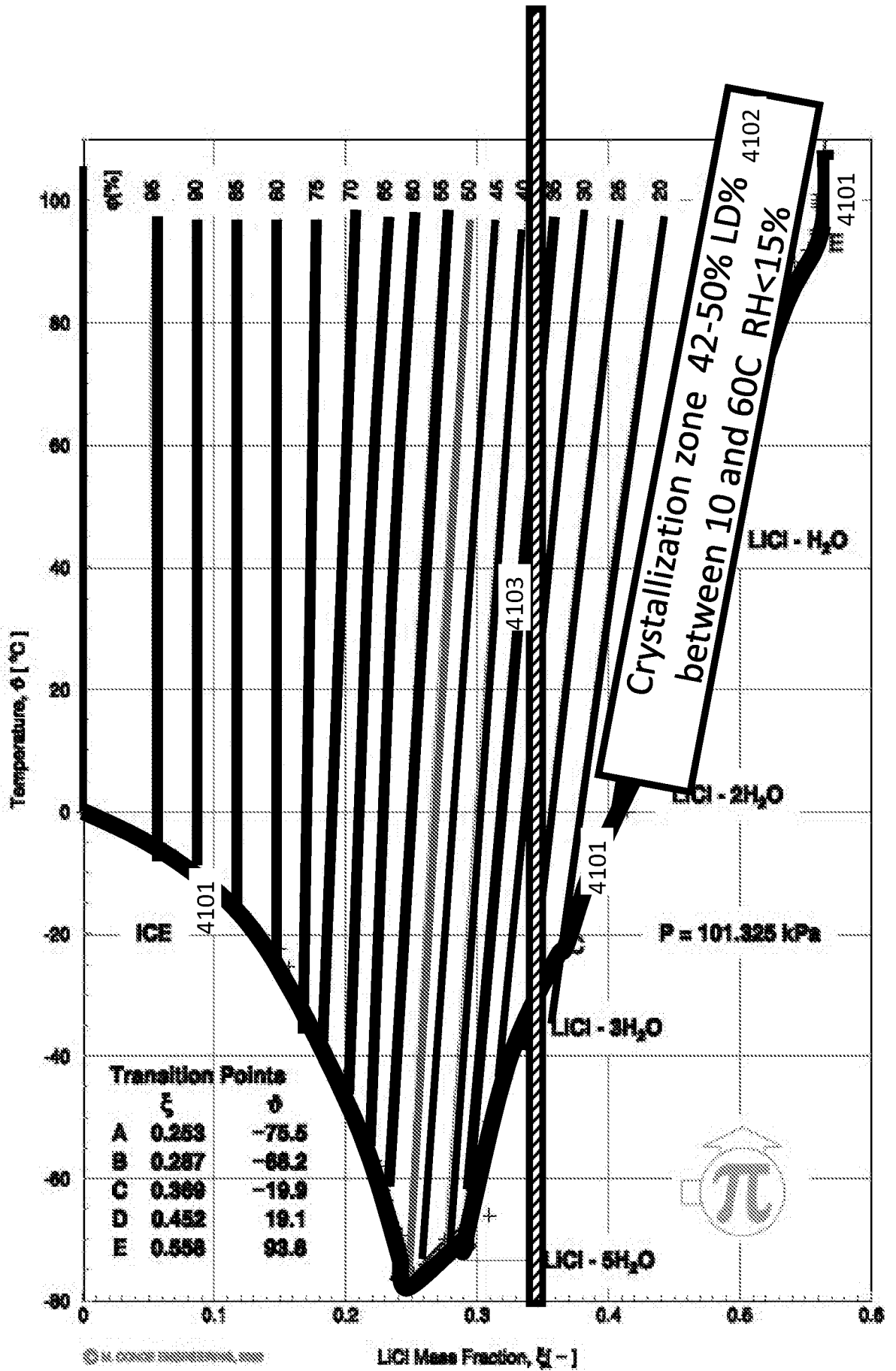
**FIG 34A**

FIG. 34B

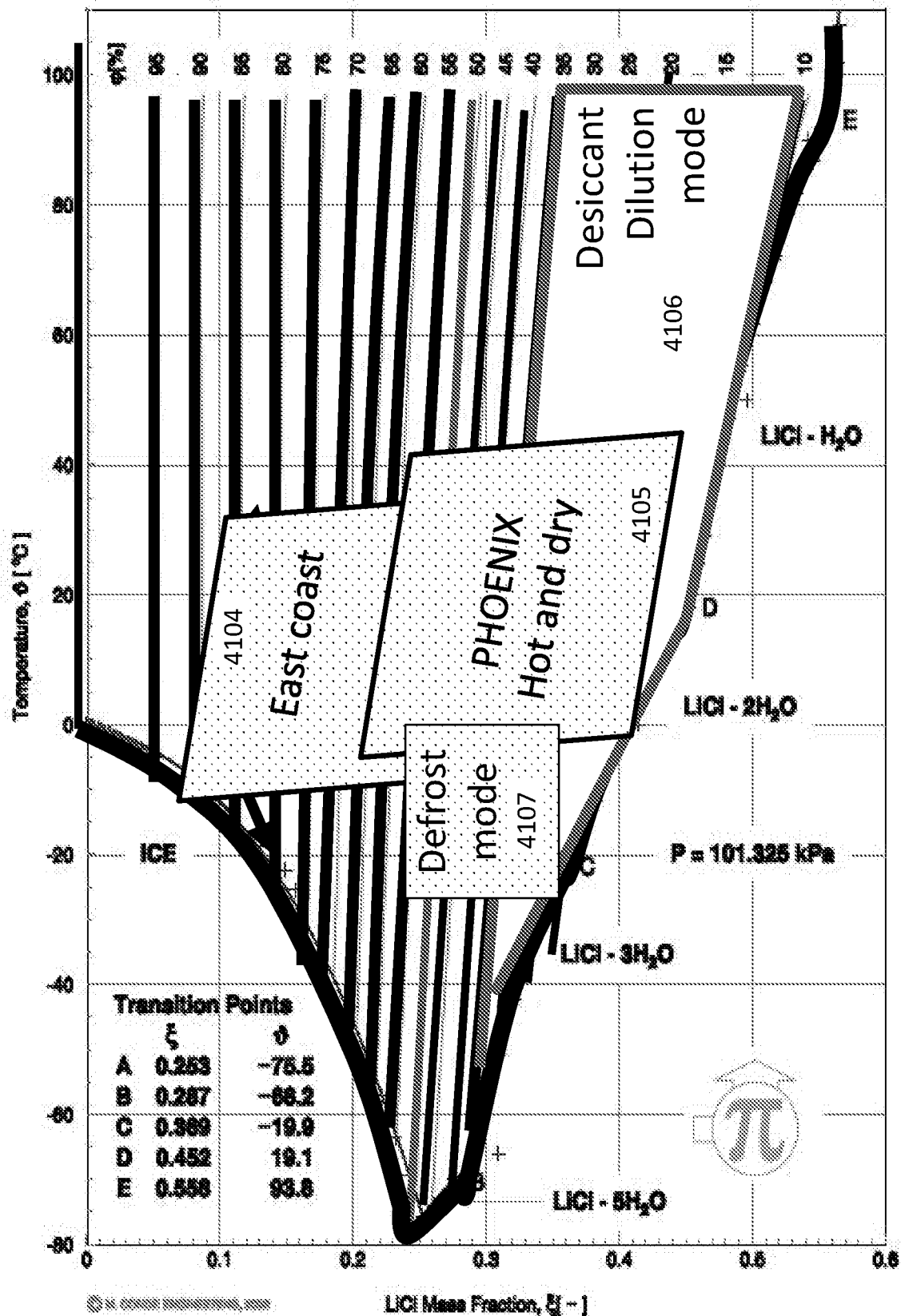


FIG. 35

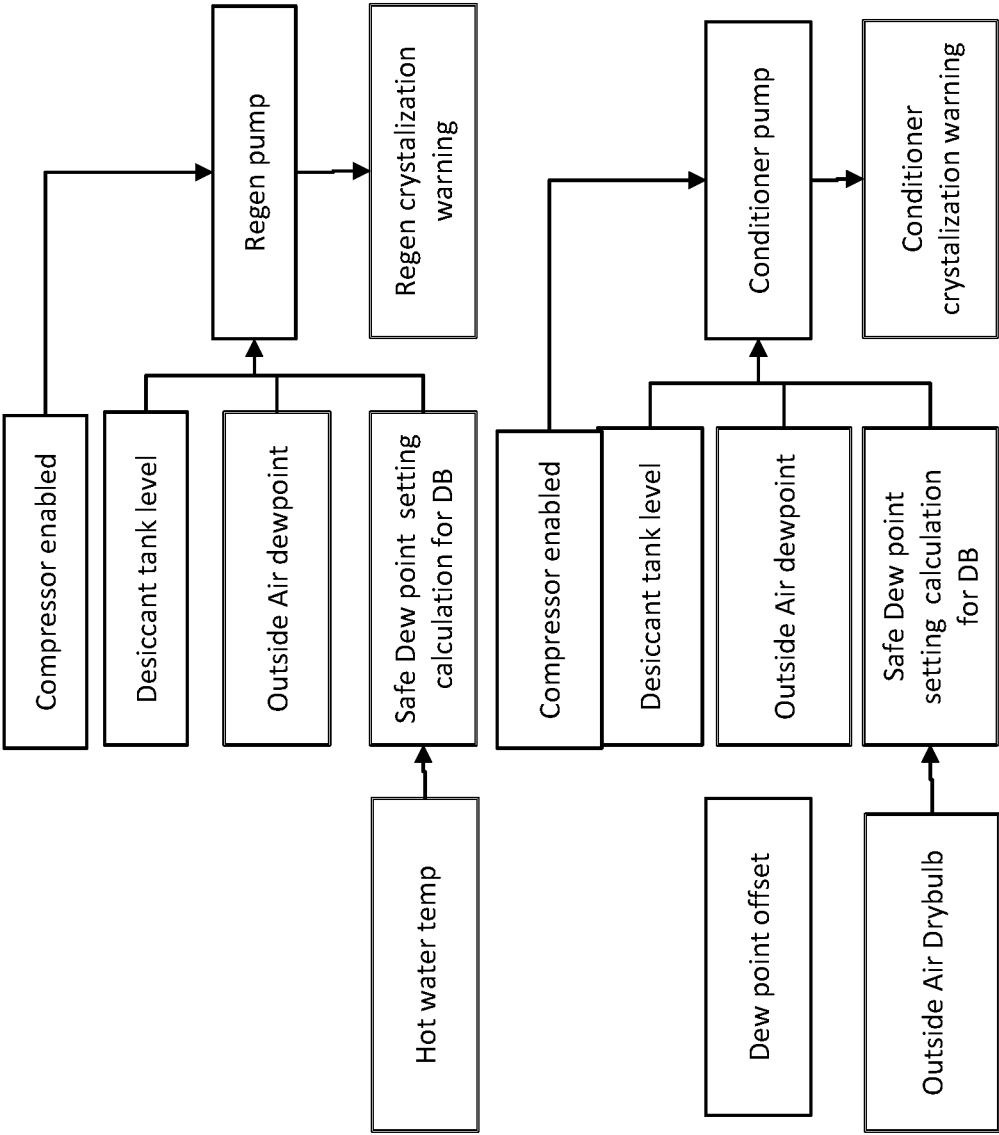
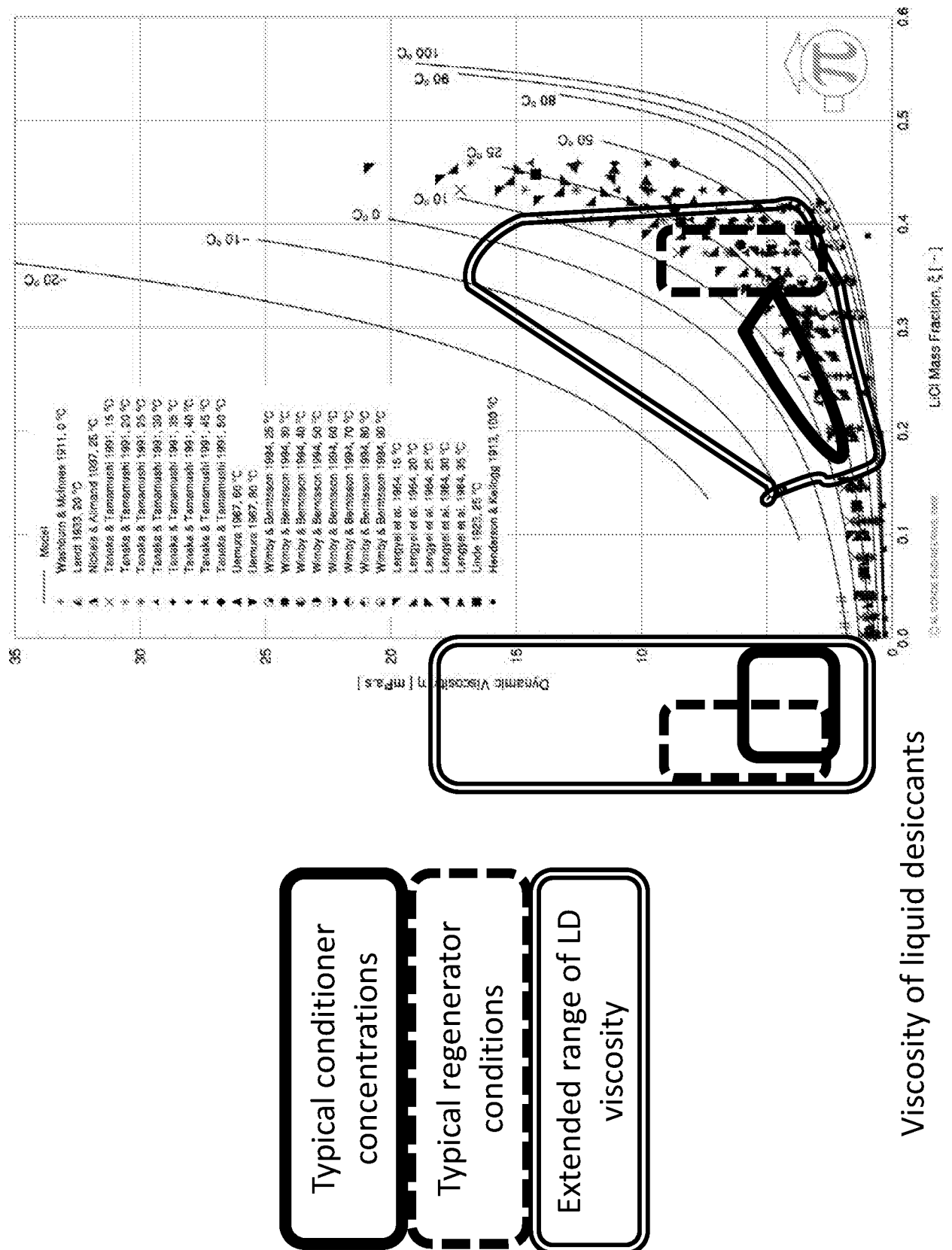


FIG. 36



**A. CLASSIFICATION OF SUBJECT MATTER****F24F 11/70(2018.01)i, F24F 3/14(2006.01)i, F24F 5/00(2006.01)i, F24F 110/10(2018.01)i, F24F 110/20(2018.01)i**

According to International Patent Classification (IPC) or to both national classification and IPC

**B. FIELDS SEARCHED**

Minimum documentation searched (classification system followed by classification symbols)

F24F 11/70; F24F 1/00; F24F 3/14; F24F 5/00; F25B 29/00; F24F 110/10; F24F 110/20

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Korean utility models and applications for utility models

Japanese utility models and applications for utility models

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

eKOMPASS(KIPO internal) &amp; Keywords: liquid desiccant, air conditioning system, conditioner, regenerator, heat exchanger, temperature, humidity, error, dew point

**C. DOCUMENTS CONSIDERED TO BE RELEVANT**

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Further documents are listed in the continuation of Box C.



See patent family annex.

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Date of the actual completion of the international search

11 March 2019 (11.03.2019)

Date of mailing of the international search report

**11 March 2019 (11.03.2019)**

Name and mailing address of the ISA/KR

International Application Division

Korean Intellectual Property Office

189 Cheongsa-ro, Seo-gu, Daejeon, 35208, Republic of Korea



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**INTERNATIONAL SEARCH REPORT**

Information on patent family members

International application No.

**PCT/US2018/058769**

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