US 20130340449A1

(19) **United States**

(12) **Patent Application Publication**

KOZUBAL et al.

(10) **Pub. No.: US 2013/0340449 A1**

(43) **Pub. Date: Dec. 26, 2013**

(54) INDIRECT EVAPORATIVE COOLER USING MEMBRANE-CONTAINED LIQUID DESICCANT FOR DEHUMIDIFICATION AND FLOCKED SURFACES TO PROVIDE COOLANT FLOW

(71) Applicant: **Alliance For Sustainable Energy, LLC**, (US)

(72) Inventors: **Eric J. KOZUBAL**, Superior, CO (US); **Jason D. WOODS**, Boulder, CO (US)

(73) Assignee: **ALLIANCE FOR SUSTAINABLE ENERGY, LLC**, Golden, CO (US)

(21) Appl. No.: **13/886,131**

(22) Filed: **May 2, 2013**

Related U.S. Application Data

(60) Provisional application No. 61/662,146, filed on Jun. 20, 2012.

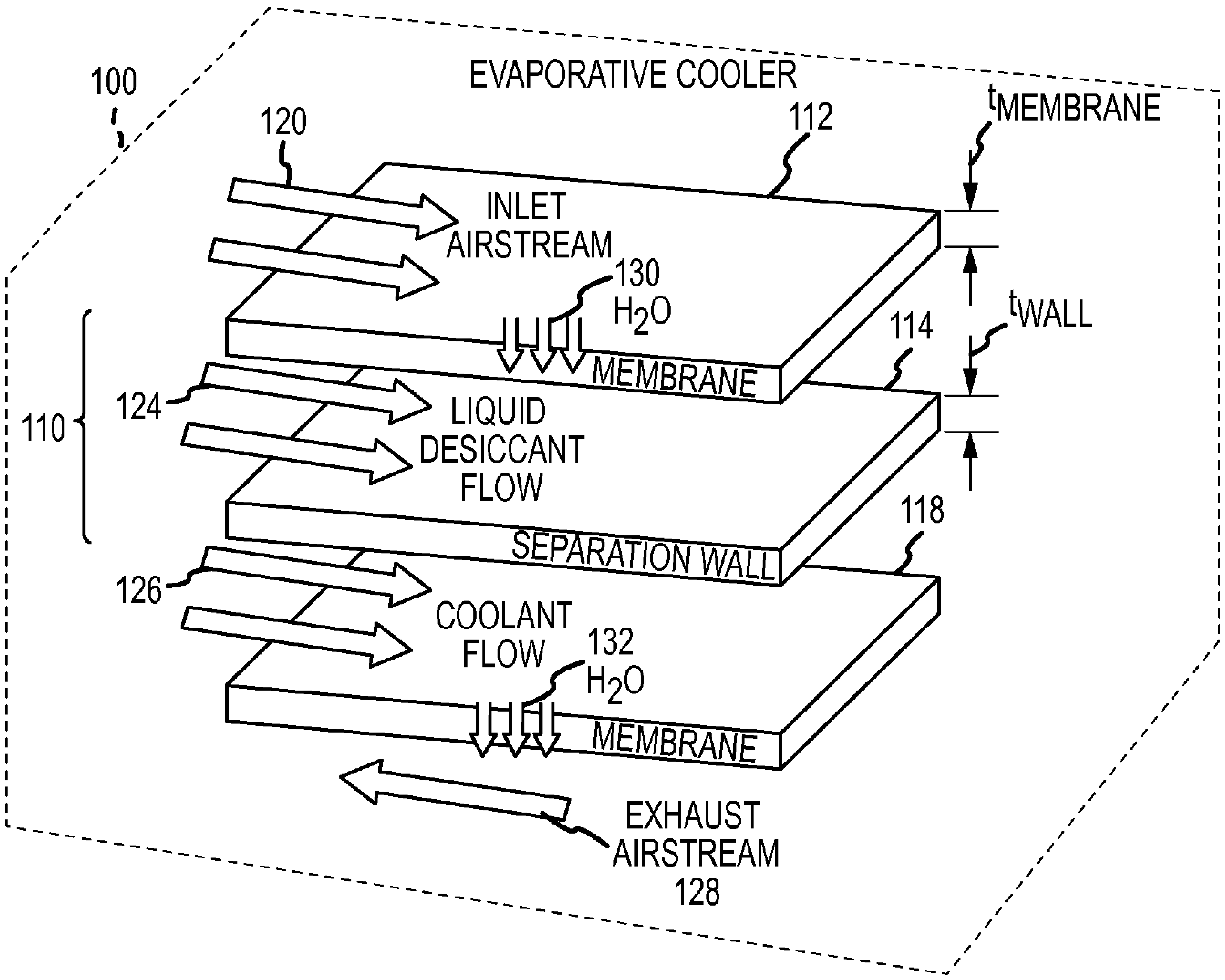
Publication Classification

(51) **Int. Cl.**
F25B 15/00 (2006.01)

(52) **U.S. Cl.**
CPC ***F25B 15/00*** (2013.01)
USPC **62/92**; 62/271; 62/94

(57) **ABSTRACT**

An apparatus for conditioning an inlet air stream. A first stage is provided with a dehumidifier cooling an air stream input by absorption of water vapor from the input air stream. A second stage is provided with an indirect evaporative cooler to receive a cooled portion of the input air stream and sensibly cool the received portion of the input air stream to a temperature range near the dew point temperature. A first portion of the sensibly cooled air stream is exhausted to a cooled space while a second portion is directed to a wet side of the indirect evaporative cooler and receives heat to sensibly cool the input air stream. A flow channel for the second portion of the sensibly cooled air stream in the indirect evaporative cooler is defined by a surface of a separation wall covered with wicking material acting to wick a stream of liquid coolant.



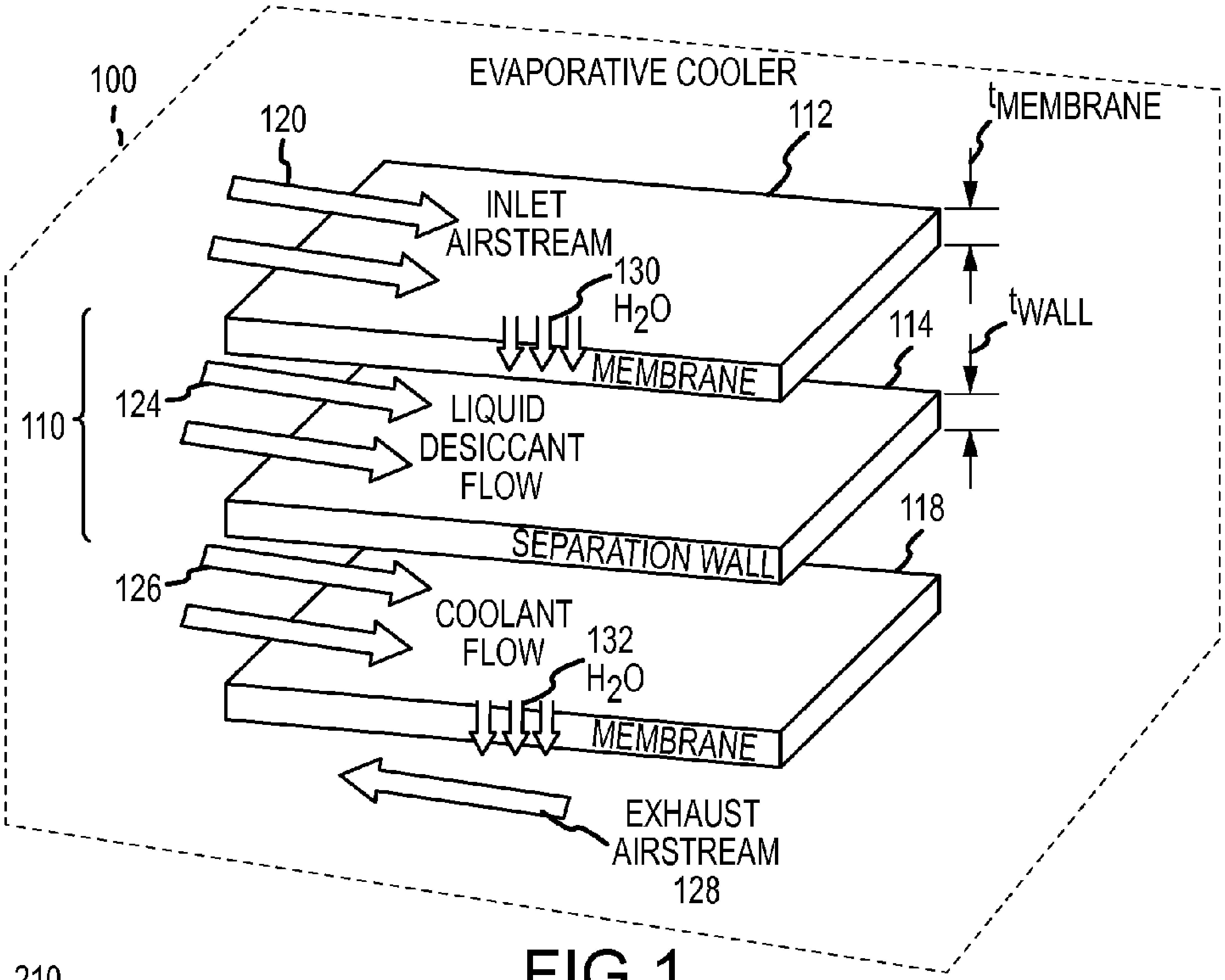

100
EVAPORATIVE COOLER
120
112
$t_{MEMBRANE}$
INLET AIRSTREAM
130 H_2O
t_{WALL}
114
MEMBRANE
110
124
LIQUID DESICCANT FLOW
SEPARATION WALL
118
126
COOLANT FLOW
132 H_2O
MEMBRANE
EXHAUST AIRSTREAM 128

FIG.1

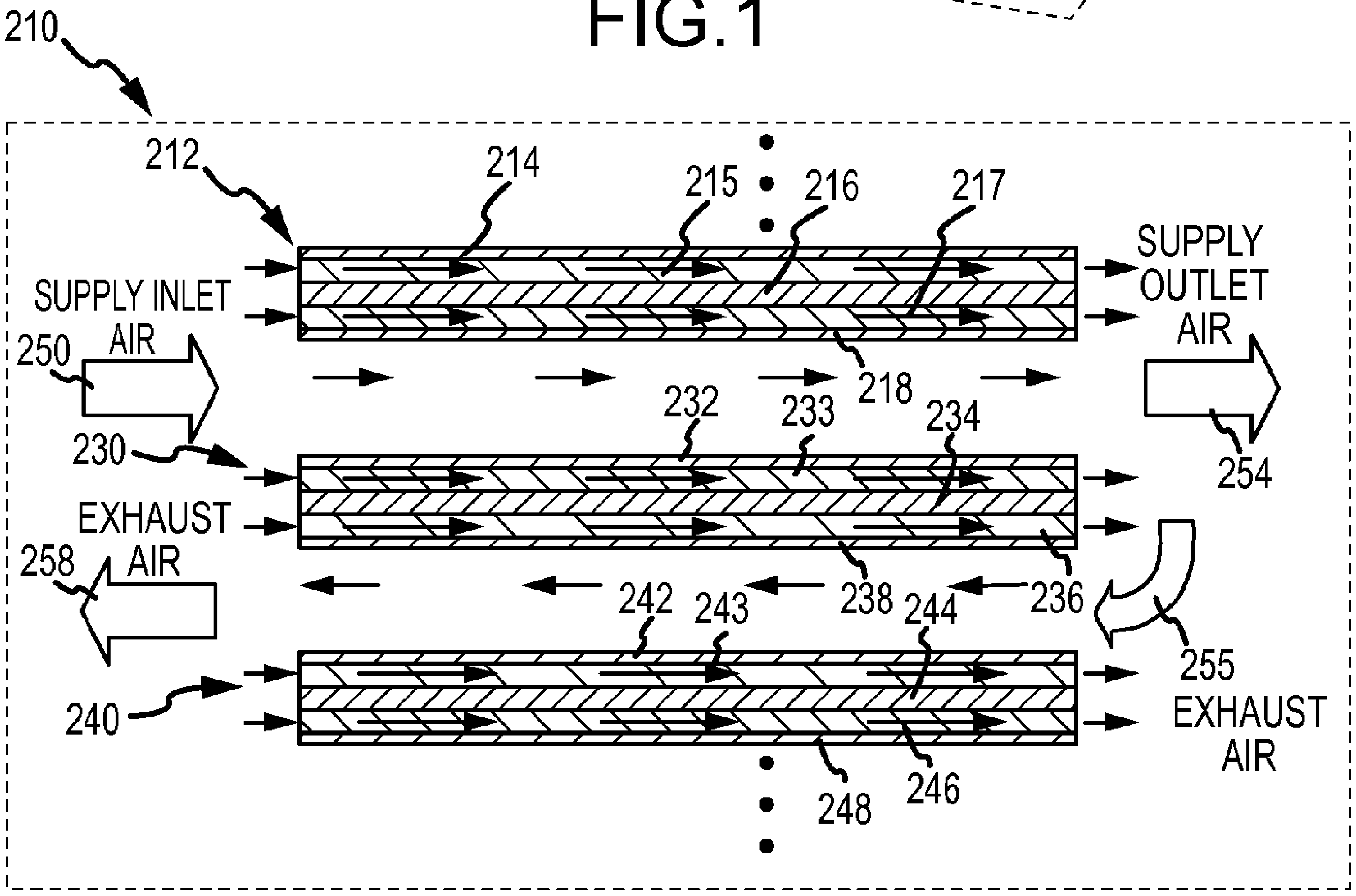

210
212
214
215
216
217
SUPPLY INLET AIR
250
SUPPLY OUTLET AIR
218
254
232
233
234
230
EXHAUST AIR
258
238
236
242
243
244
255
240
EXHAUST AIR
248
246

FIG.2

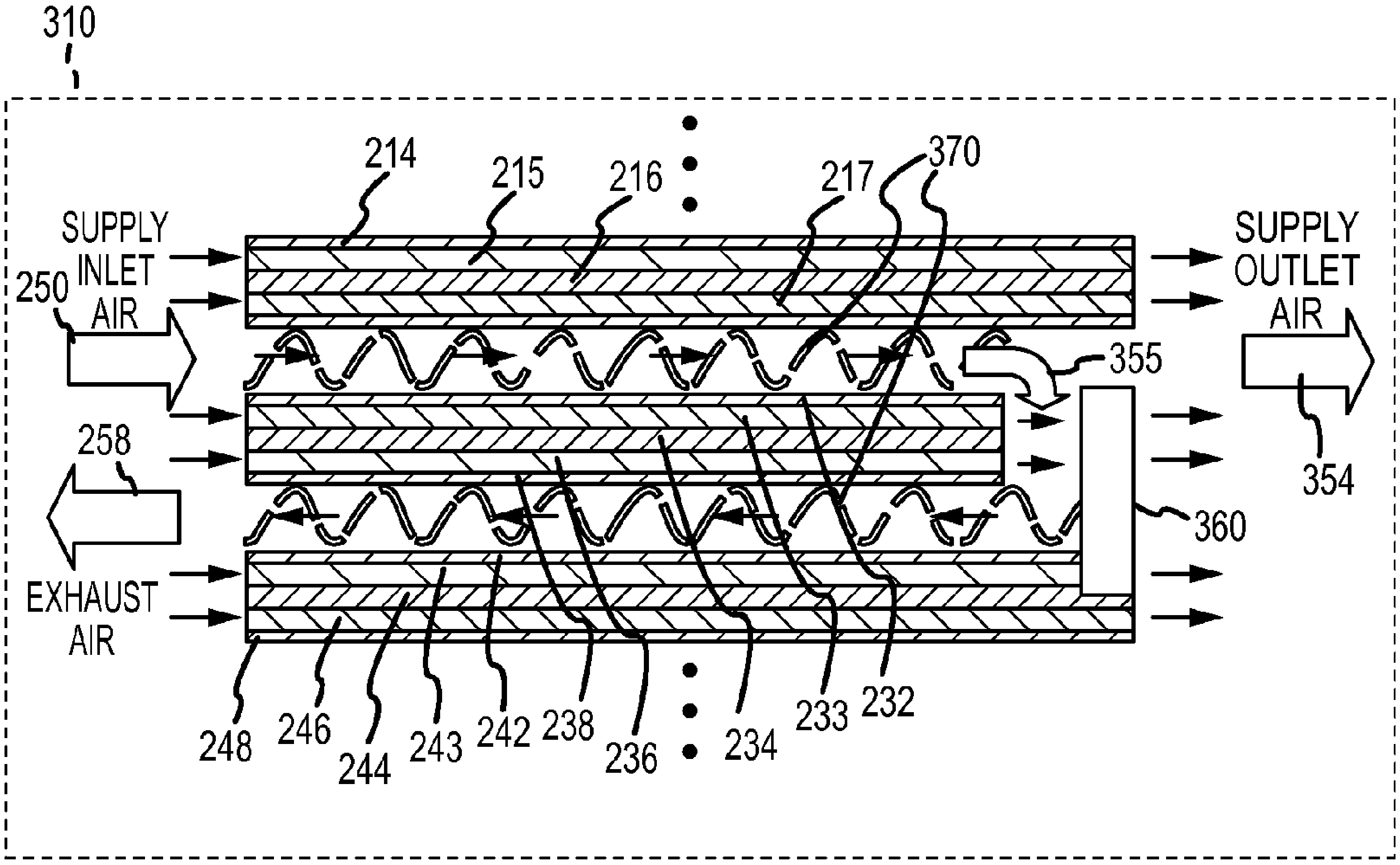

310
214
215
216
217
370
SUPPLY INLET AIR
250
SUPPLY OUTLET AIR
355
354
258
360
EXHAUST AIR
248
246
244
243
242
238
236
234
233
232

FIG.3

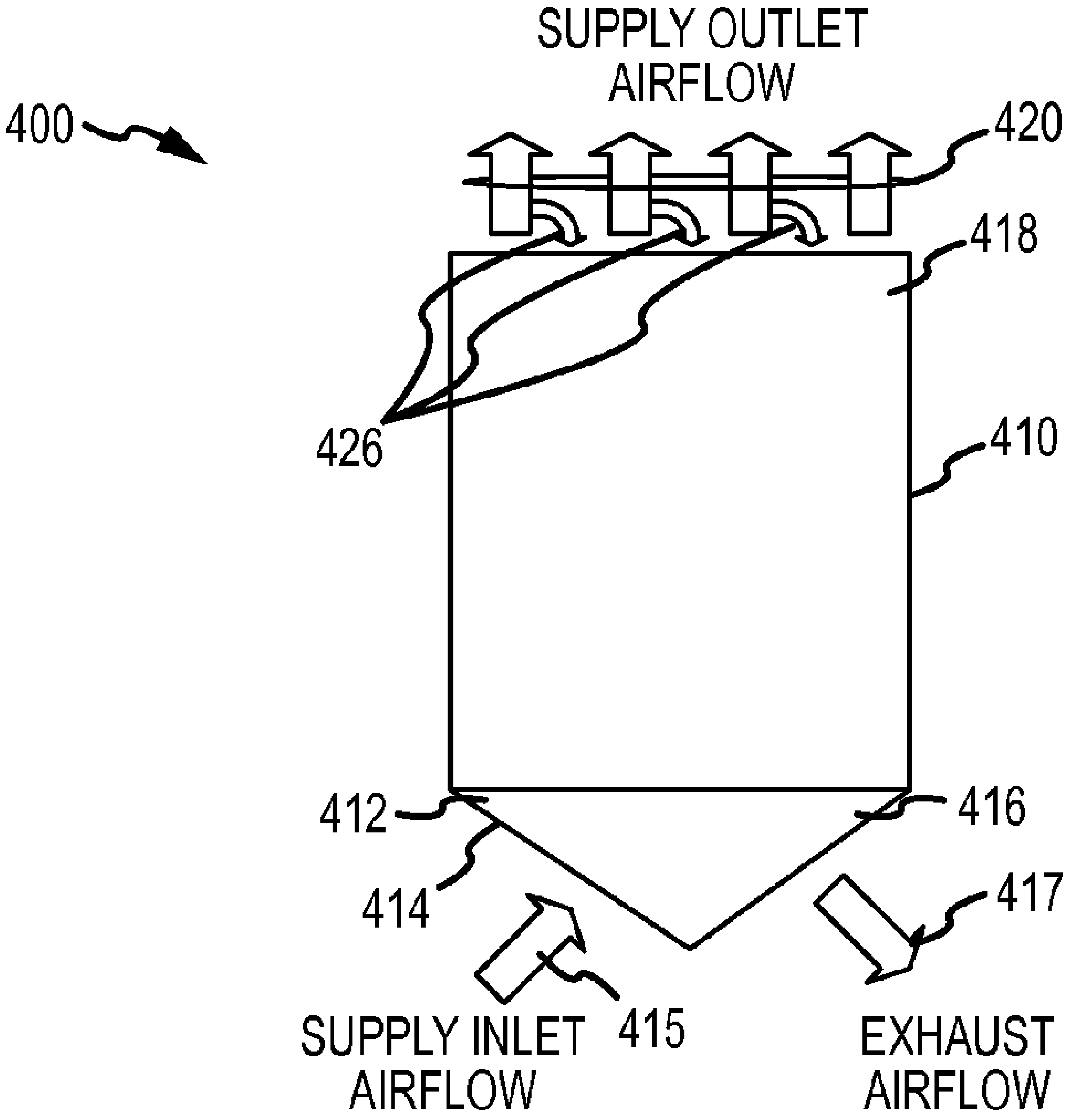

SUPPLY OUTLET AIRFLOW
400
420
418
426
410
412
416
414
417
SUPPLY INLET AIRFLOW
415
EXHAUST AIRFLOW

FIG.4

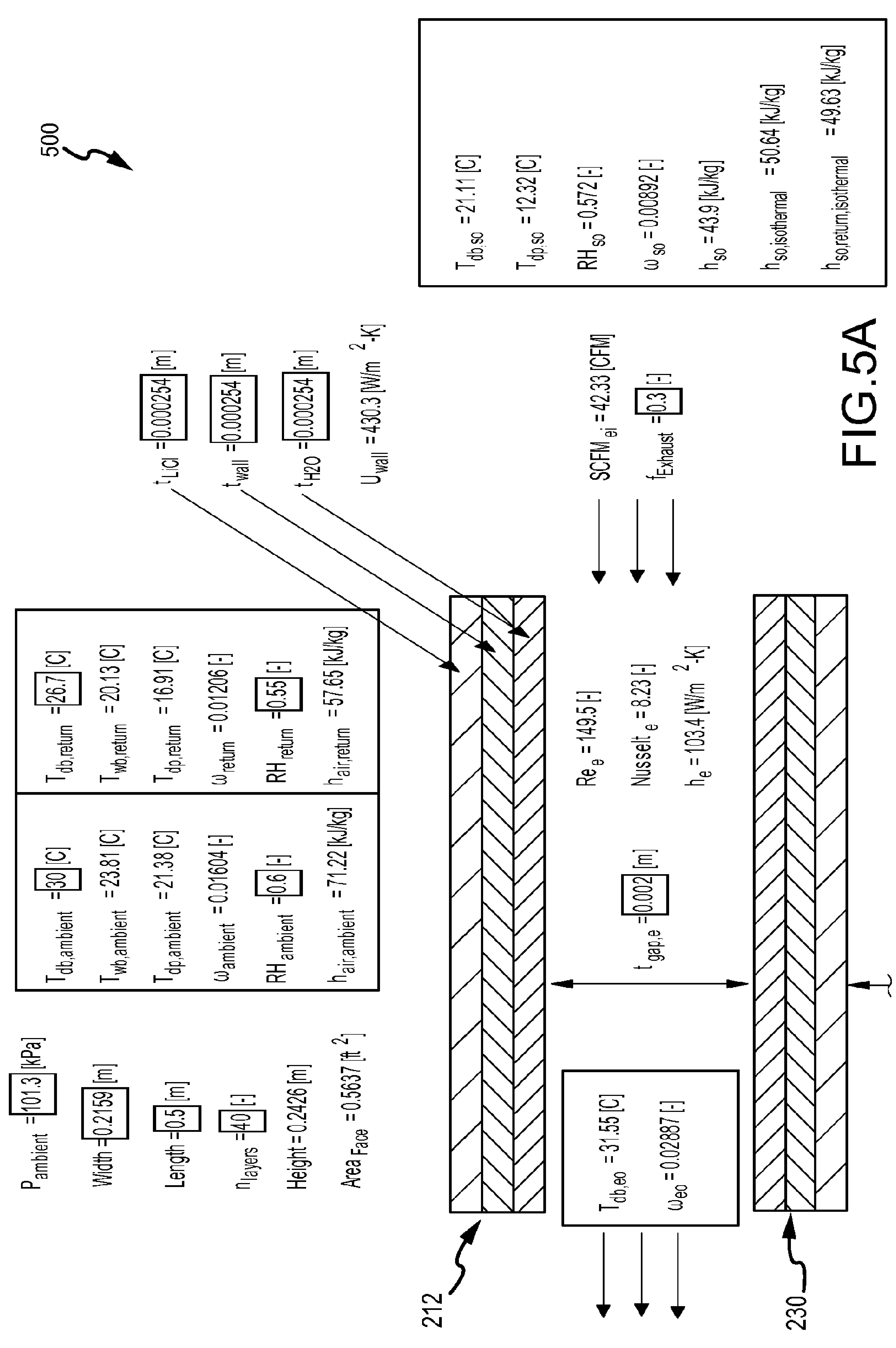


FIG.5A

CONTINUED ON FIG.5B

CONTINUED FROM FIG.5A

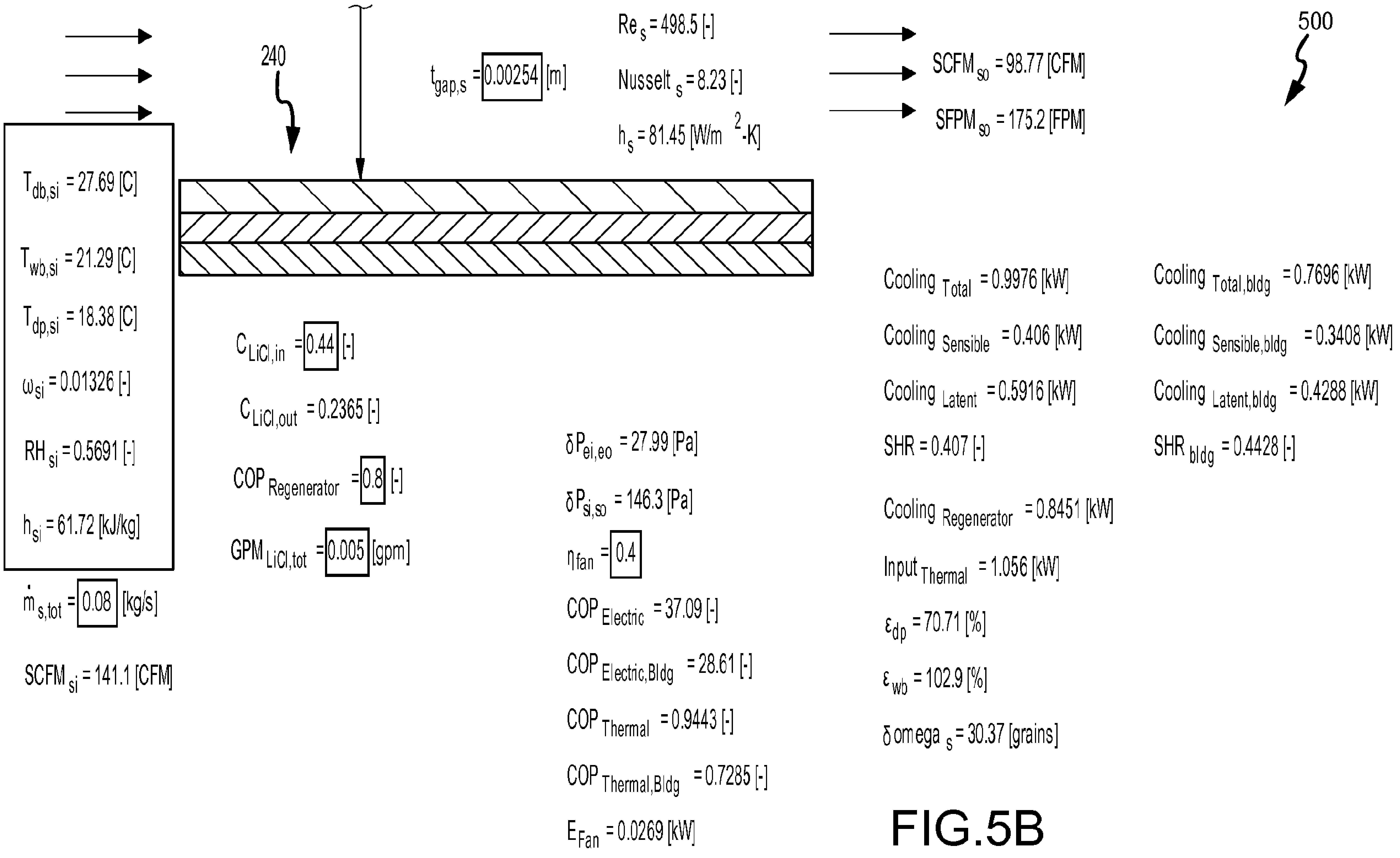


FIG.5B

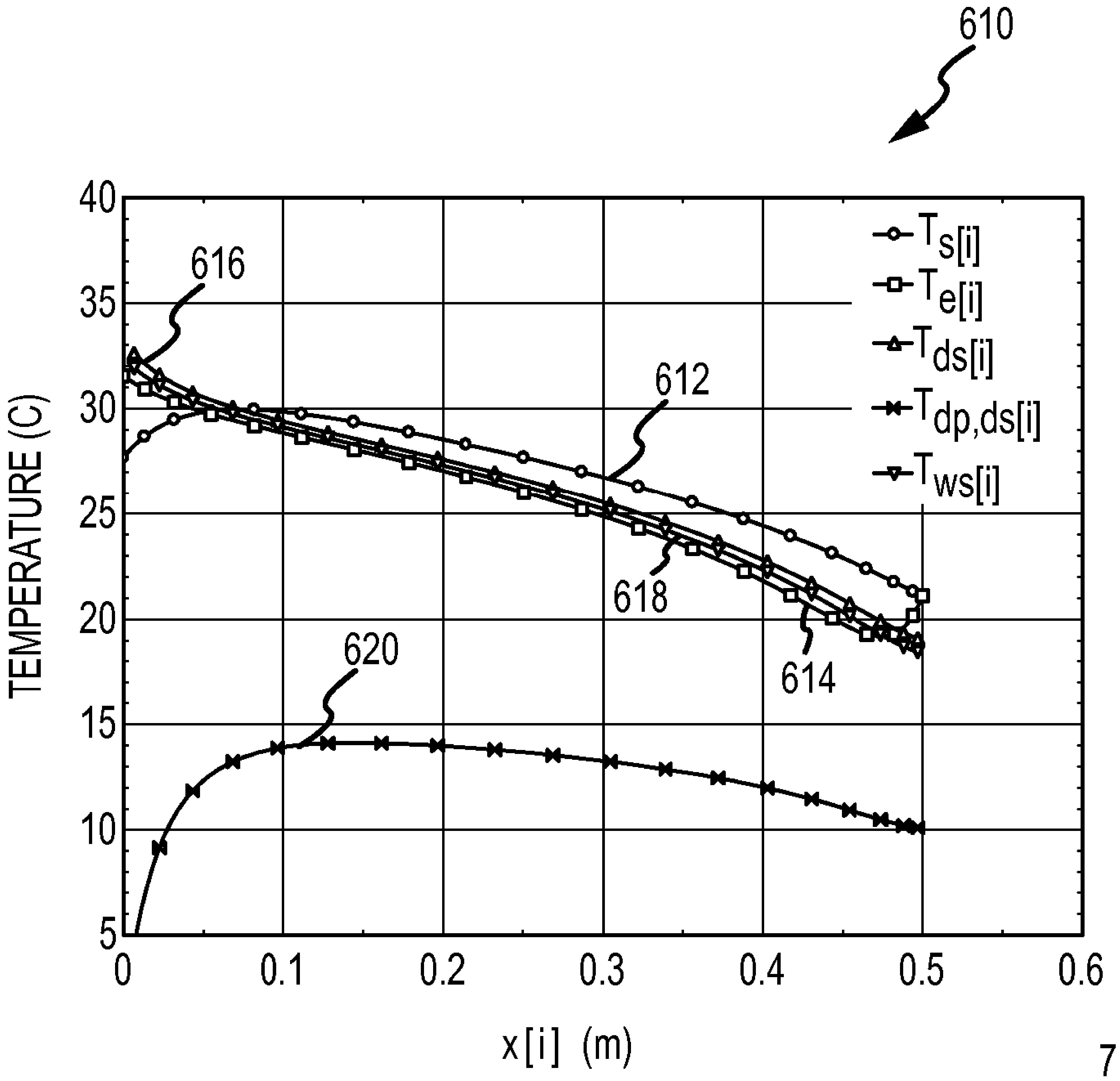

610
616
612
618
614
620
$T_{s[i]}$
$T_{e[i]}$
$T_{ds[i]}$
$T_{dp,ds[i]}$
$T_{ws[i]}$
TEMPERATURE (C)
40
35
30
25
20
15
10
5
0
0.1
0.2
0.3
0.4
0.5
0.6
x[i] (m)

FIG.6

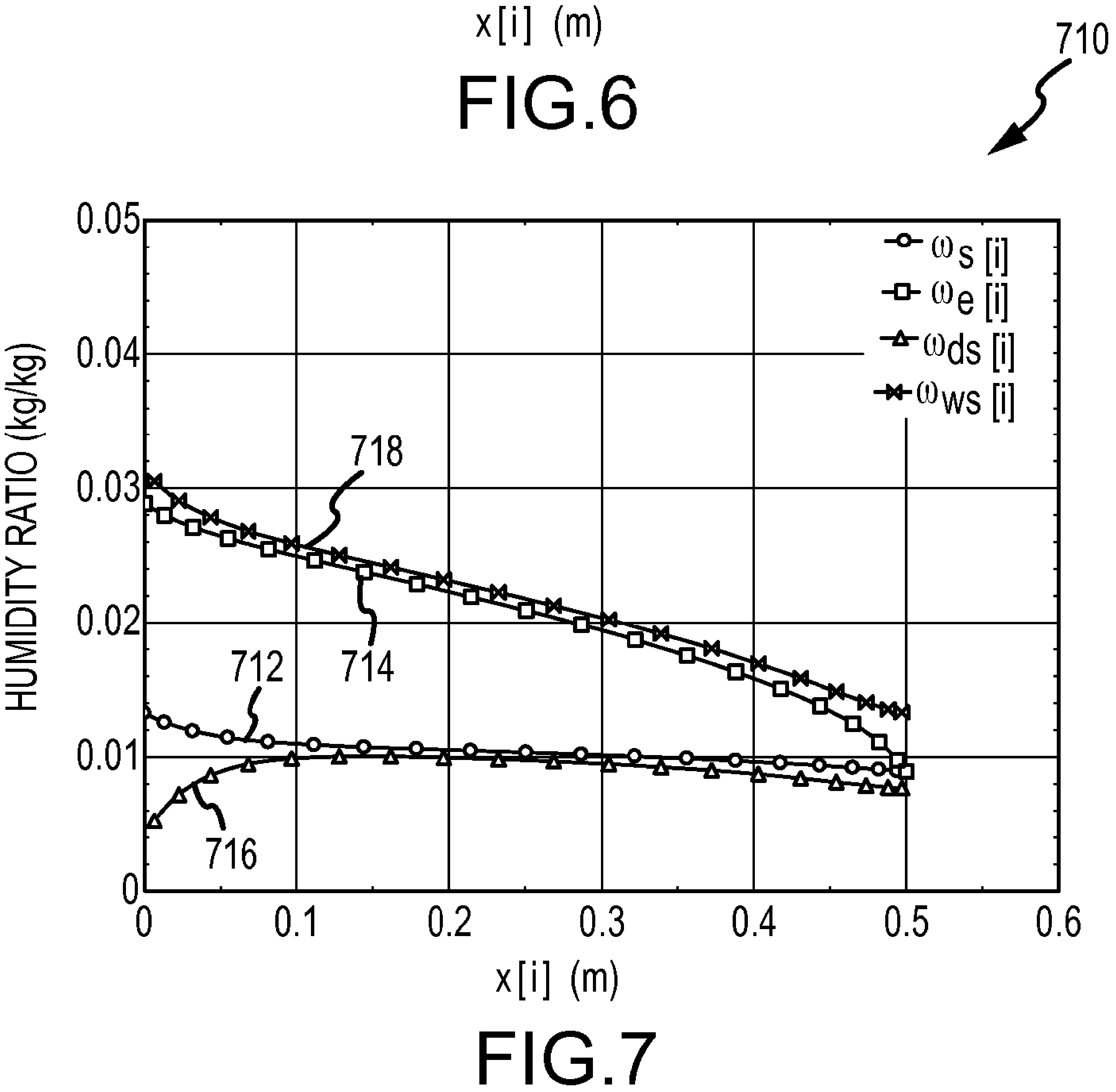

710
718
714
712
716
ω_s [i]
ω_e [i]
ω_{ds} [i]
ω_{ws} [i]
HUMIDITY RATIO (kg/kg)
0.05
0.04
0.03
0.02
0.01
0
0
0.1
0.2
0.3
0.4
0.5
0.6
x[i] (m)

FIG.7

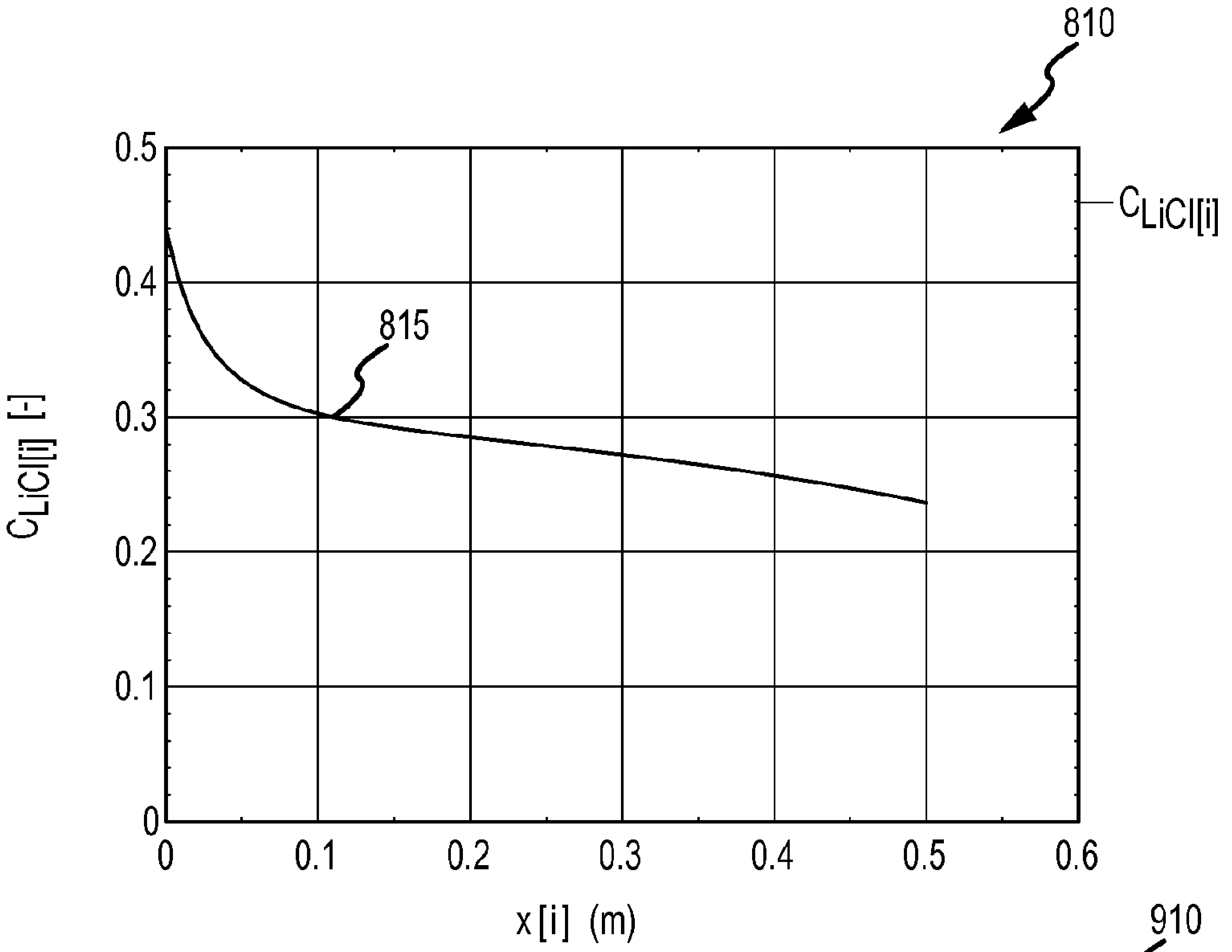

810
815
$C_{LiCl[i]}$
$C_{LiCl[i]}$ [-]
0.5
0.4
0.3
0.2
0.1
0
0
0.1
0.2
0.3
0.4
0.5
0.6
x[i] (m)

FIG.8

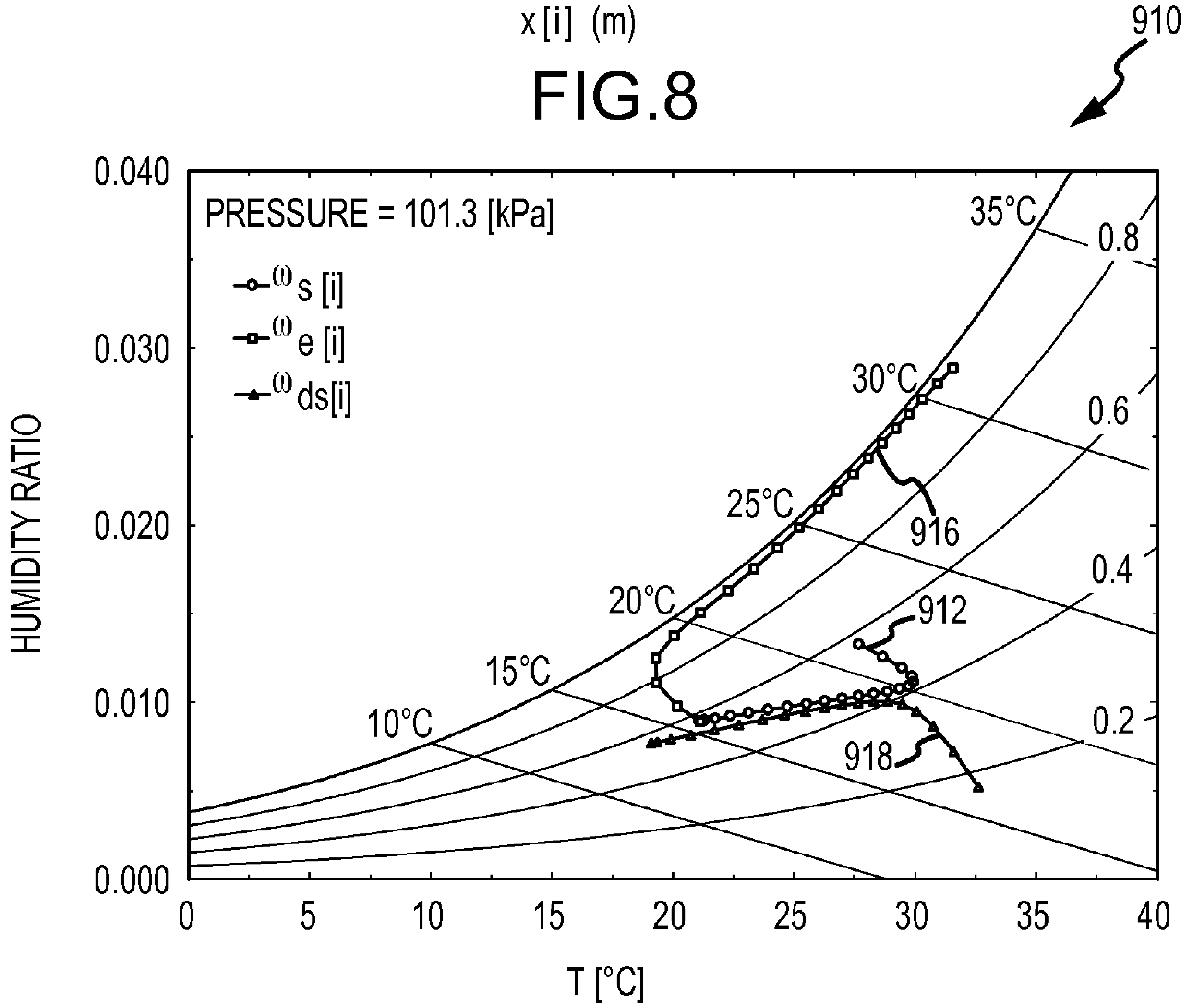

910
PRESSURE = 101.3 [kPa]
ω_s [i]
ω_e [i]
ω_{ds}[i]
35°C
30°C
25°C
20°C
15°C
10°C
0.8
0.6
0.4
0.2
916
912
918
HUMIDITY RATIO
0.040
0.030
0.020
0.010
0.000
0
5
10
15
20
25
30
35
40
T [°C]

FIG.9

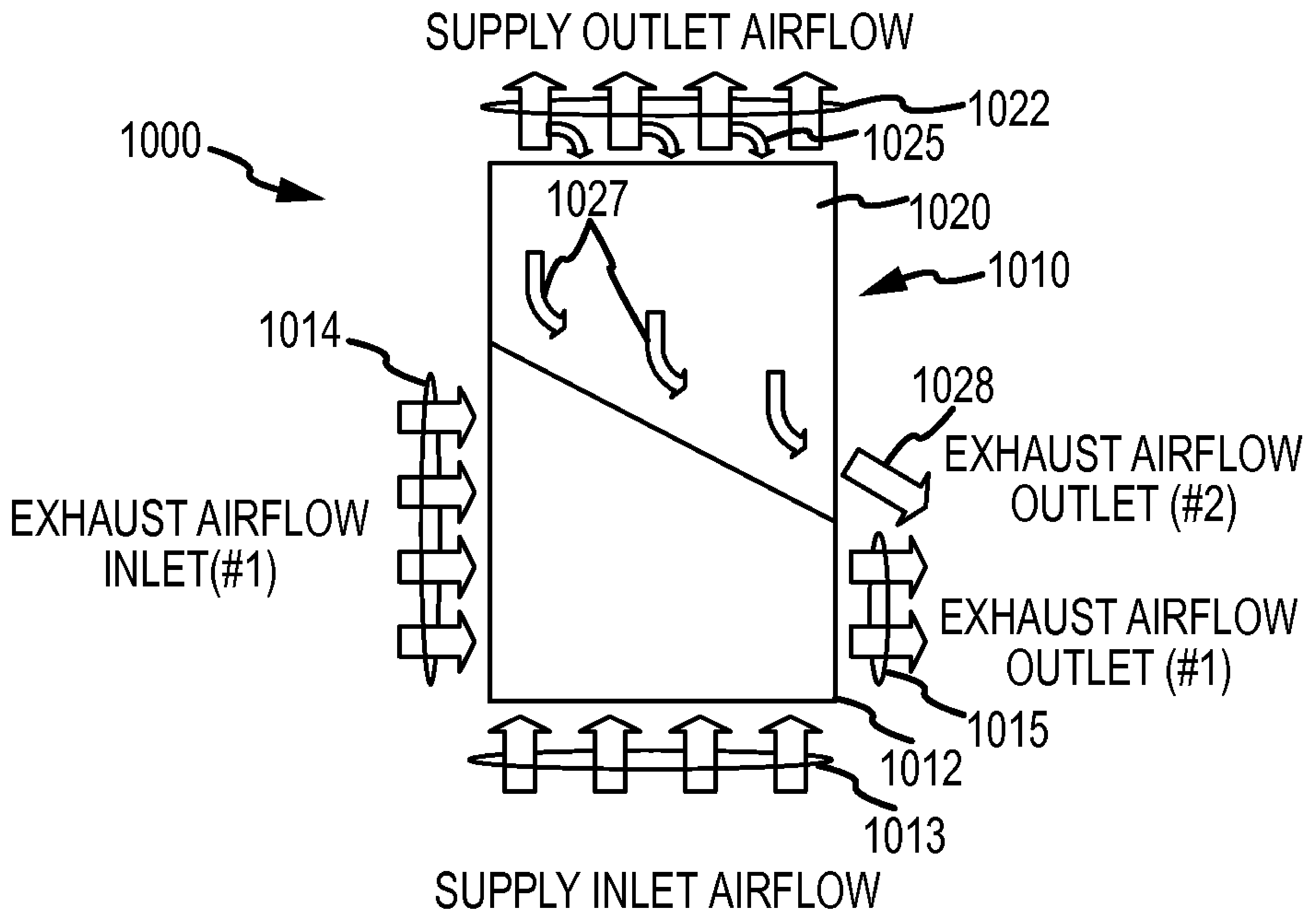

SUPPLY OUTLET AIRFLOW
1022
1025
1000
1027
1020
1010
1014
1028
EXHAUST AIRFLOW OUTLET (#2)
EXHAUST AIRFLOW INLET(#1)
EXHAUST AIRFLOW OUTLET (#1)
1015
1012
1013
SUPPLY INLET AIRFLOW

FIG.10

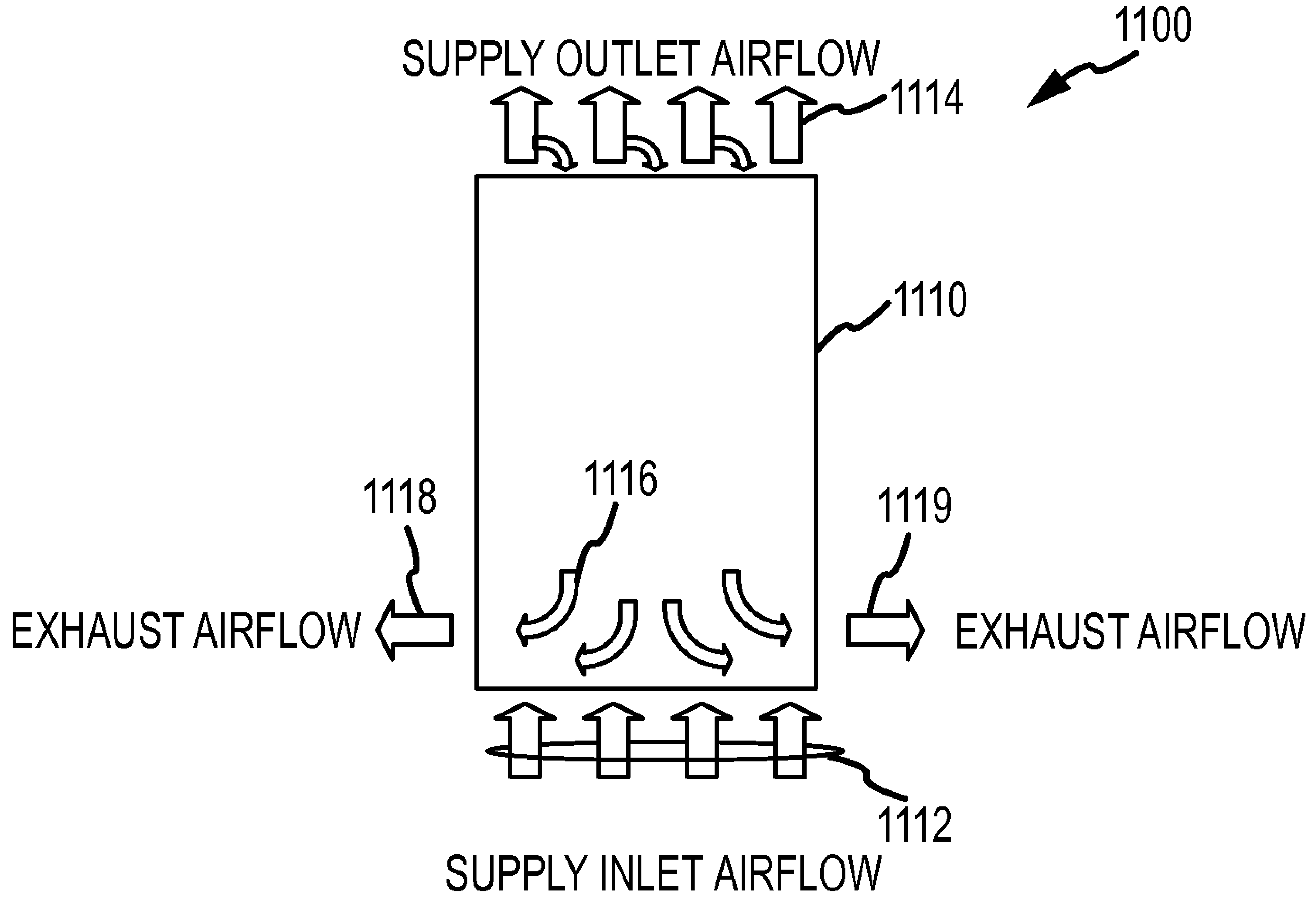

1100
SUPPLY OUTLET AIRFLOW
1114
1110
1118
1116
1119
EXHAUST AIRFLOW
EXHAUST AIRFLOW
1112
SUPPLY INLET AIRFLOW

FIG.11

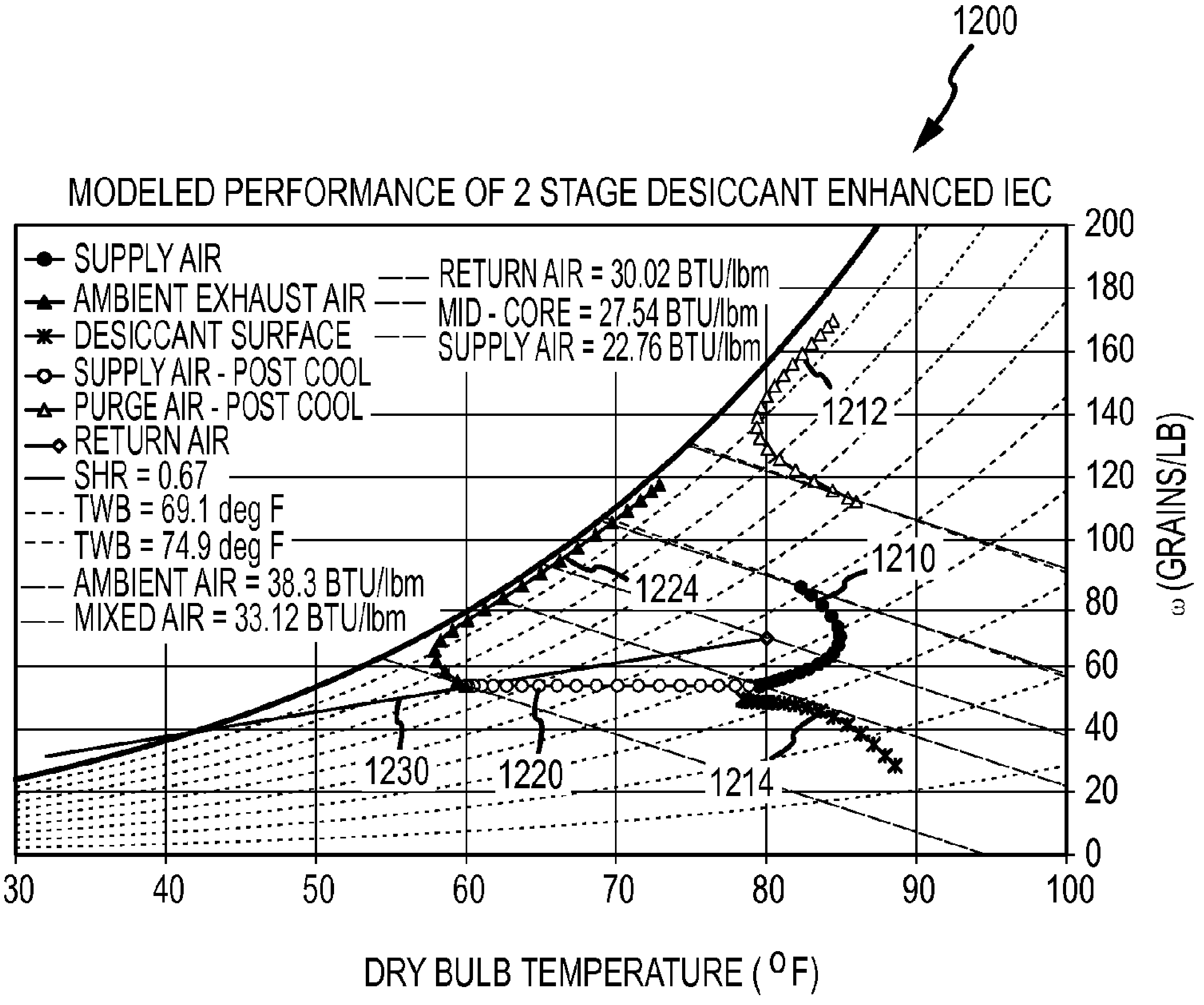

1200
MODELED PERFORMANCE OF 2 STAGE DESICCANT ENHANCED IEC
SUPPLY AIR
AMBIENT EXHAUST AIR
DESICCANT SURFACE
SUPPLY AIR - POST COOL
PURGE AIR - POST COOL
RETURN AIR
SHR = 0.67
TWB = 69.1 deg F
TWB = 74.9 deg F
AMBIENT AIR = 38.3 BTU/lbm
MIXED AIR = 33.12 BTU/lbm
RETURN AIR = 30.02 BTU/lbm
MID - CORE = 27.54 BTU/lbm
SUPPLY AIR = 22.76 BTU/lbm
1212
1210
1224
1230
1220
1214
200
180
160
140
120
100
80
60
40
20
0
ω (GRAINS/LB)
30
40
50
60
70
80
90
100
DRY BULB TEMPERATURE (°F)

FIG.12

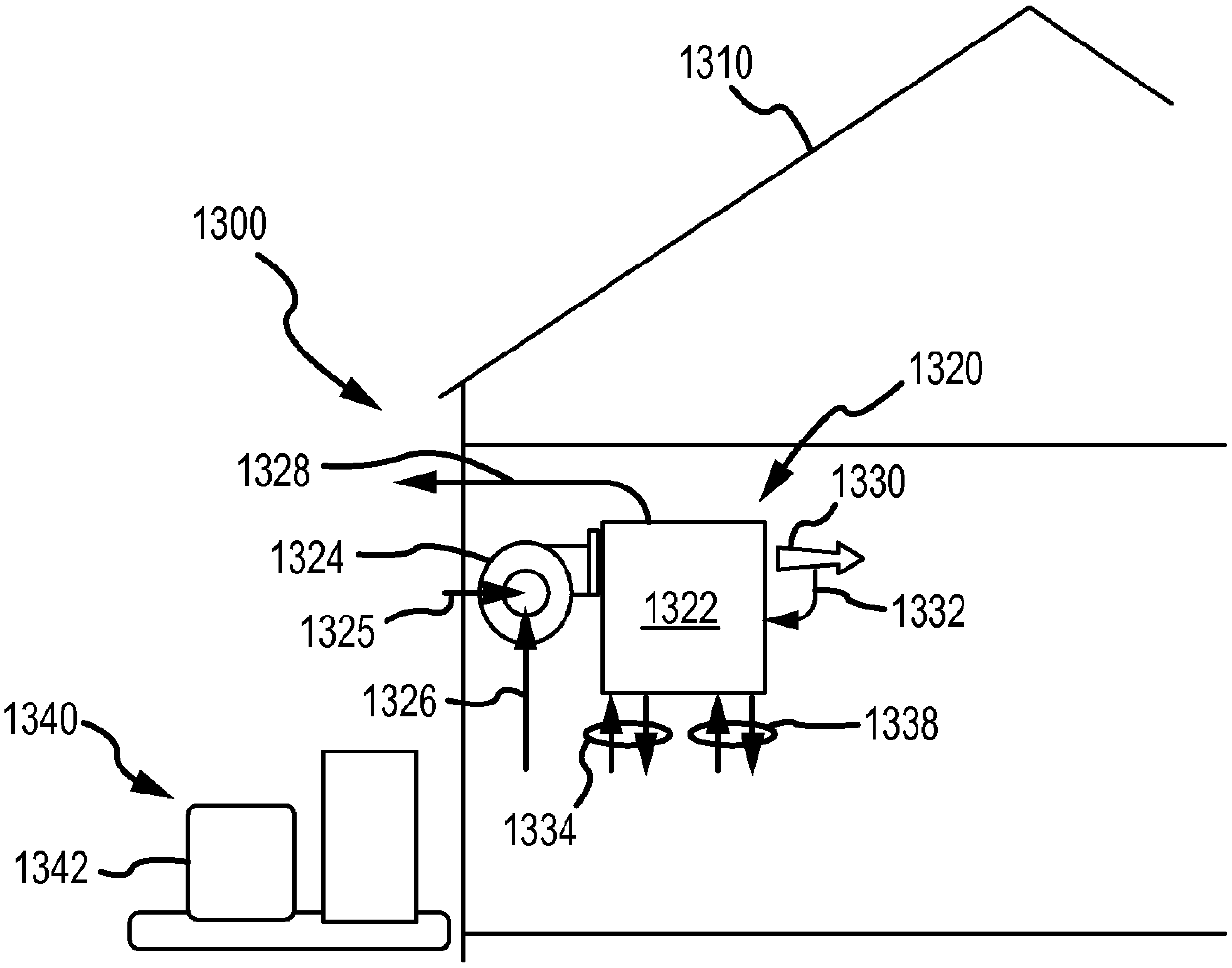

1310
1300
1320
1328
1330
1324
1322
1325
1332
1326
1340
1338
1334
1342

FIG.13

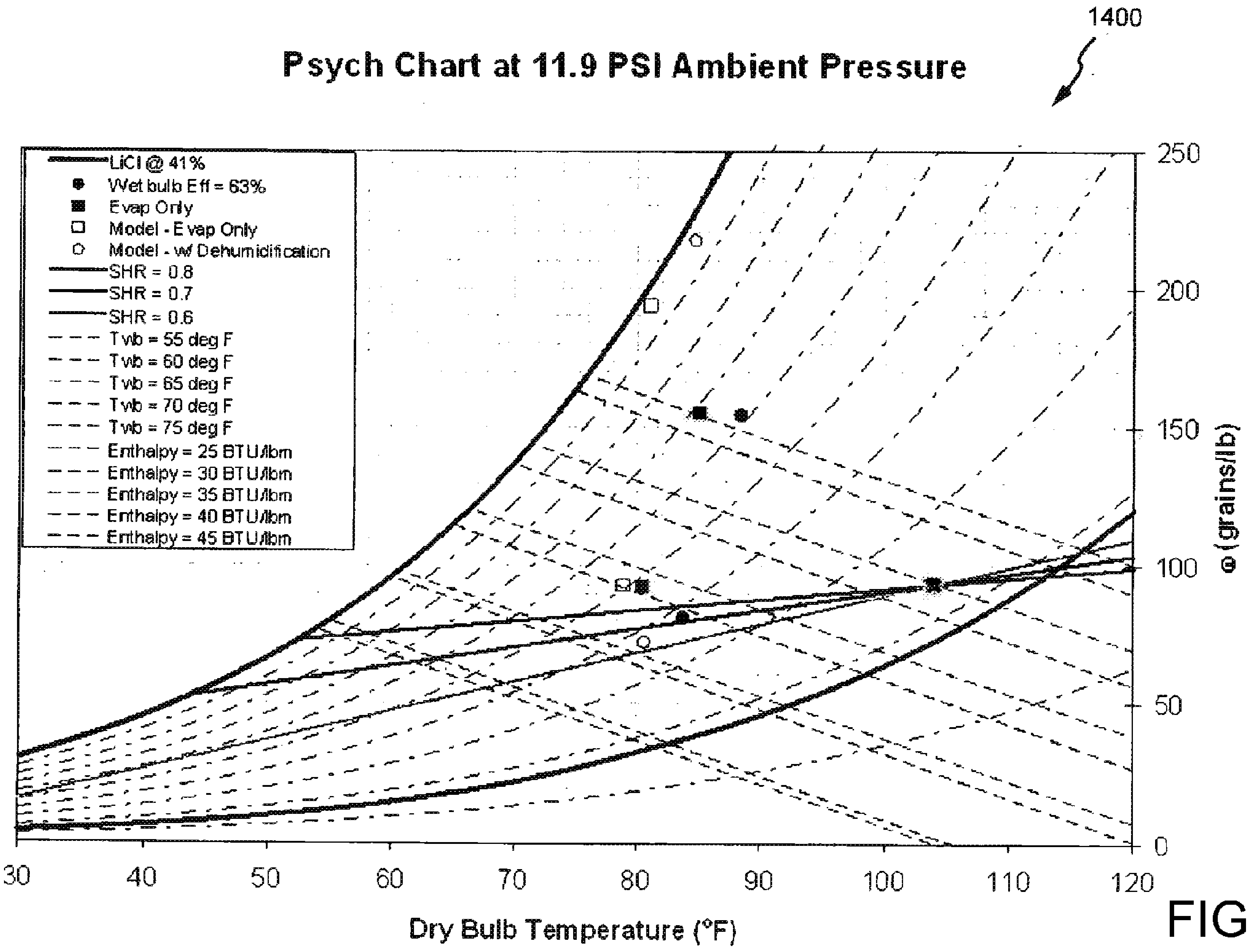

1400
Psych Chart at 11.9 PSI Ambient Pressure
LiCl @ 41%
Wet bulb Eff = 63%
Evap Only
Model - Evap Only
Model - w/ Dehumidification
SHR = 0.8
SHR = 0.7
SHR = 0.6
Twb = 55 deg F
Twb = 60 deg F
Twb = 65 deg F
Twb = 70 deg F
Twb = 75 deg F
Enthalpy = 25 BTU/lbm
Enthalpy = 30 BTU/lbm
Enthalpy = 35 BTU/lbm
Enthalpy = 40 BTU/lbm
Enthalpy = 45 BTU/lbm
250
200
150
100
50
0
ω (grains/lb)
30
40
50
60
70
80
90
100
110
120
Dry Bulb Temperature (°F)

FIG.14

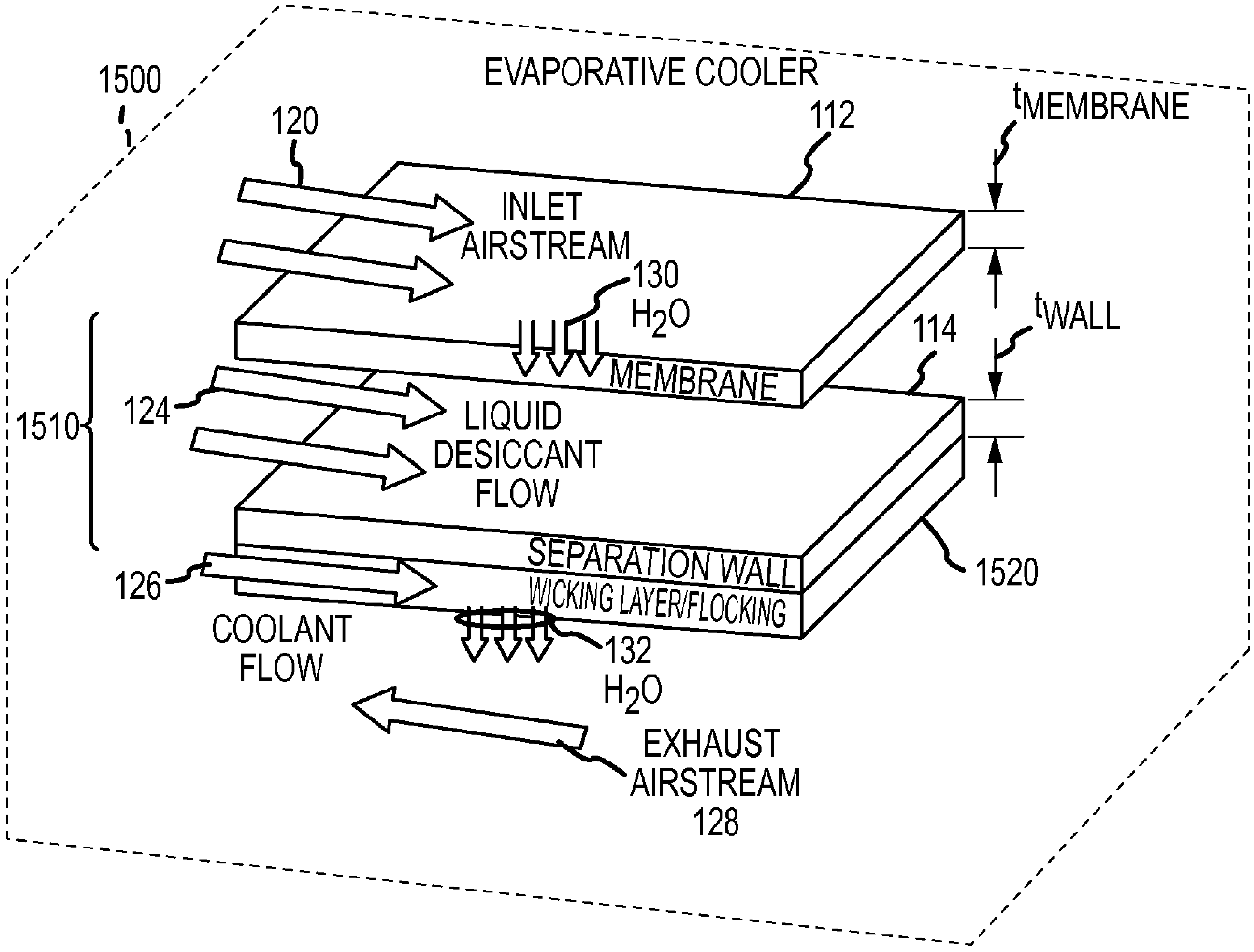

1500
EVAPORATIVE COOLER
120
112
$t_{MEMBRANE}$
INLET
AIRSTREAM
130
H_2O
t_{WALL}
114
MEMBRANE
1510
124
LIQUID
DESICCANT
FLOW
SEPARATION WALL
WICKING LAYER/FLOCKING
1520
126
COOLANT
FLOW
132
H_2O
EXHAUST
AIRSTREAM
128

FIG.15

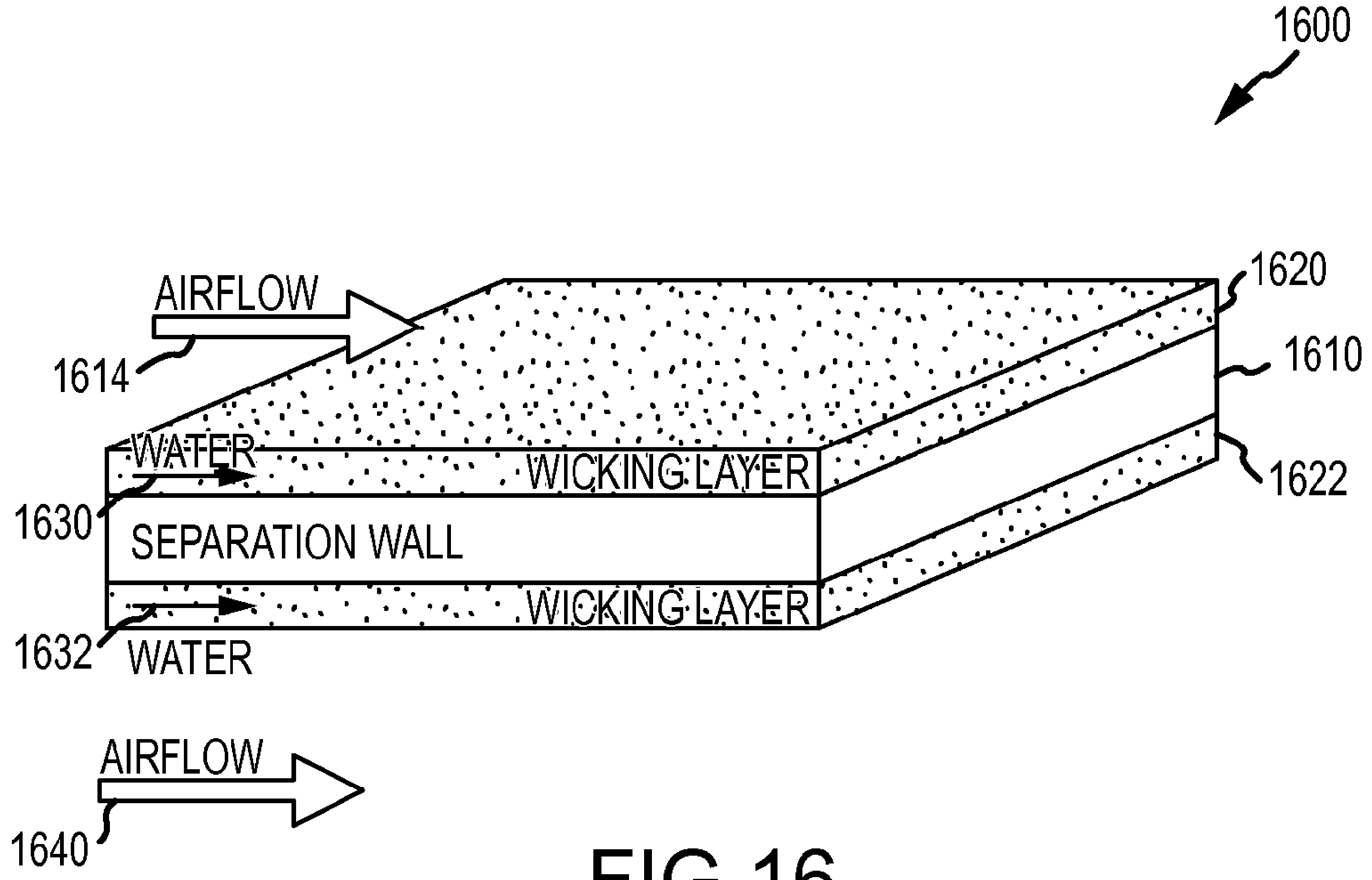

1600
AIRFLOW
1614
1620
1610
1622
WATER
WICKING LAYER
1630
SEPARATION WALL
WICKING LAYER
1632
WATER
AIRFLOW
1640

FIG.16

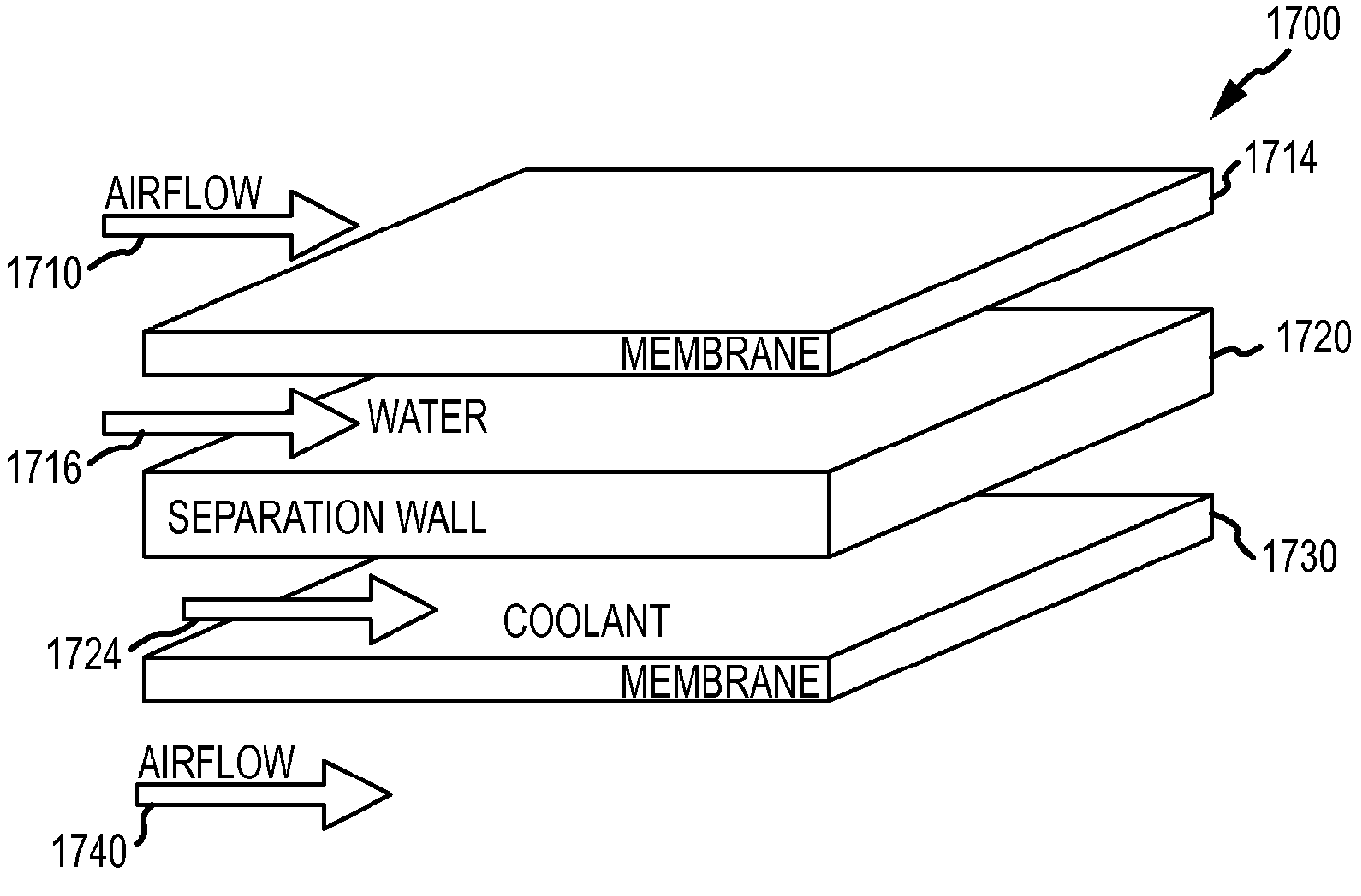

1700
AIRFLOW
1710
1714
MEMBRANE
1720
WATER
1716
SEPARATION WALL
1730
COOLANT
1724
MEMBRANE
AIRFLOW
1740

FIG.17

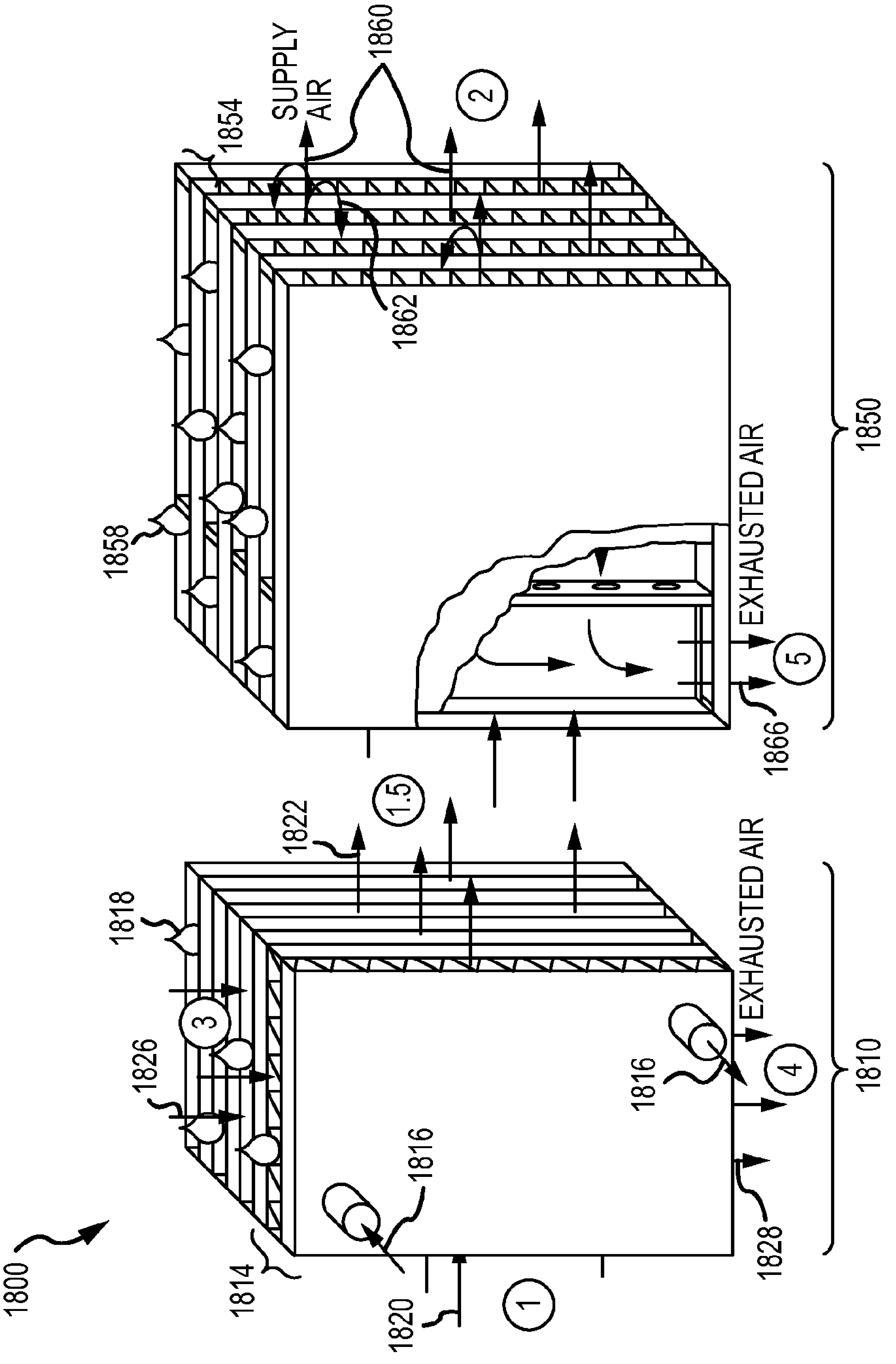

1800
1814
1820
①
1816
1816
1828
1826
③
1818
④
EXHAUSTED AIR
1810
1822
1.5
1858
1854
SUPPLY AIR
1860
②
1862
EXHAUSTED AIR
⑤
1866
1850

FIG. 18

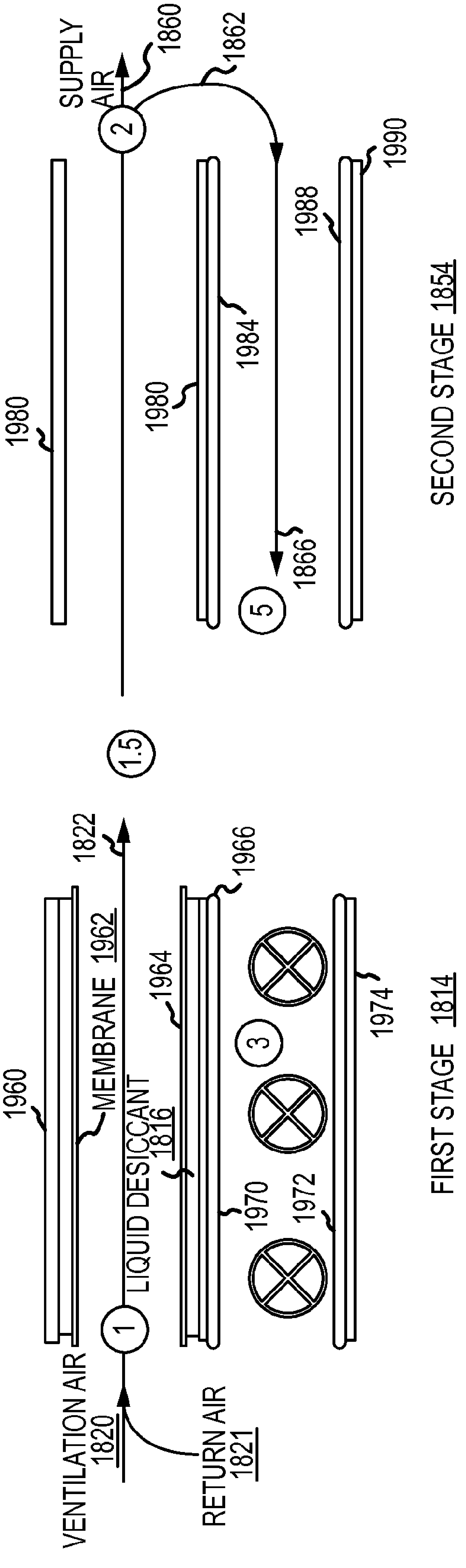

VENTILATION AIR
1820
RETURN AIR
1821
1
1960
MEMBRANE 1962
LIQUID DESICCANT
1816
1964
1970
1972
3
1974
1966
1822
FIRST STAGE 1814
1.5
1980
1980
1984
1988
1990
5
1866
2
SUPPLY AIR
1860
1862
SECOND STAGE 1854

FIG.19

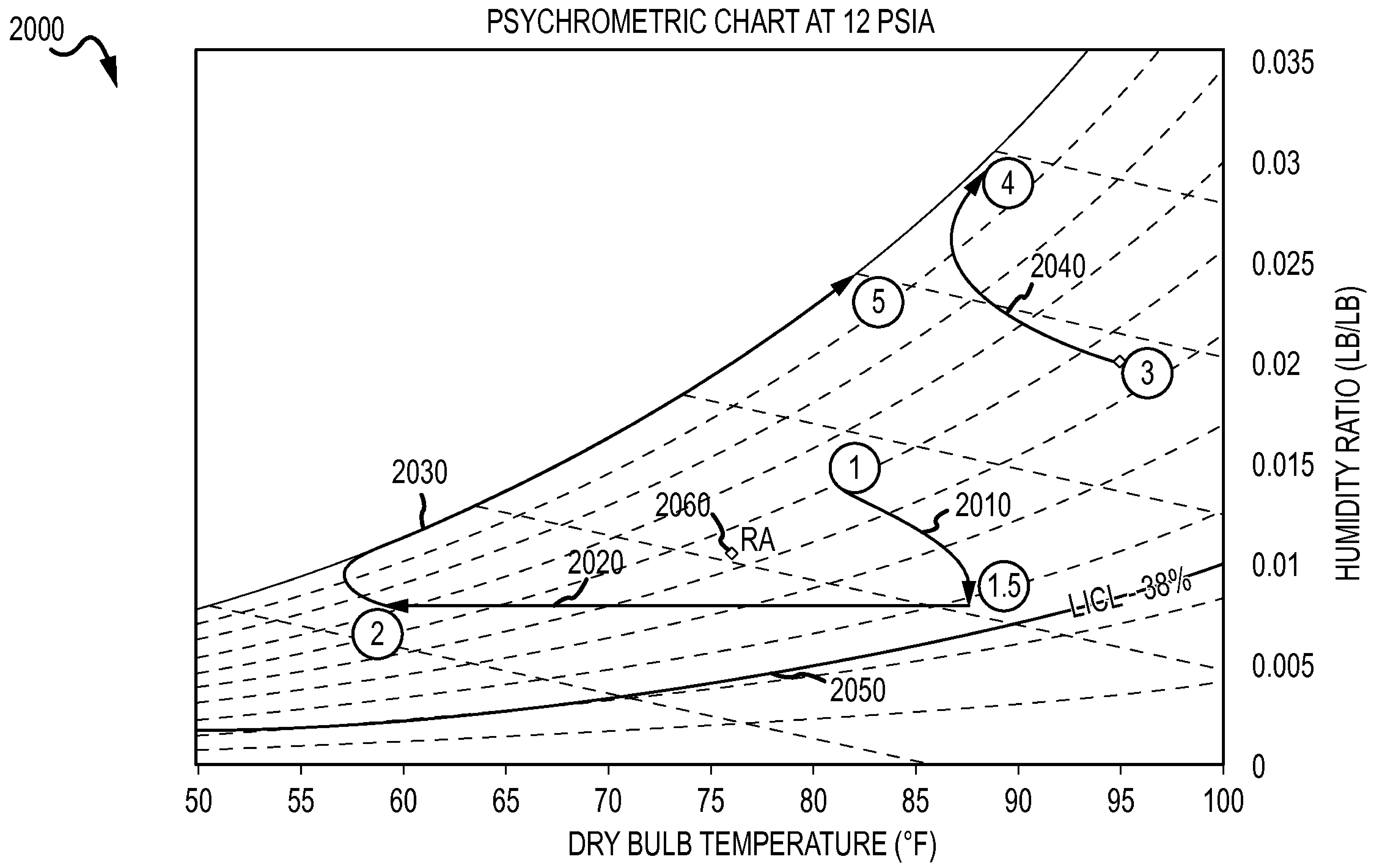

2000
PSYCHROMETRIC CHART AT 12 PSIA
0.035
0.03
0.025
0.02
0.015
0.01
0.005
0
HUMIDITY RATIO (LB/LB)
50
55
60
65
70
75
80
85
90
95
100
DRY BULB TEMPERATURE (°F)
4
2040
5
3
2030
1
2010
2060
RA
2020
1.5
LICL--38%
2
2050

FIG.20

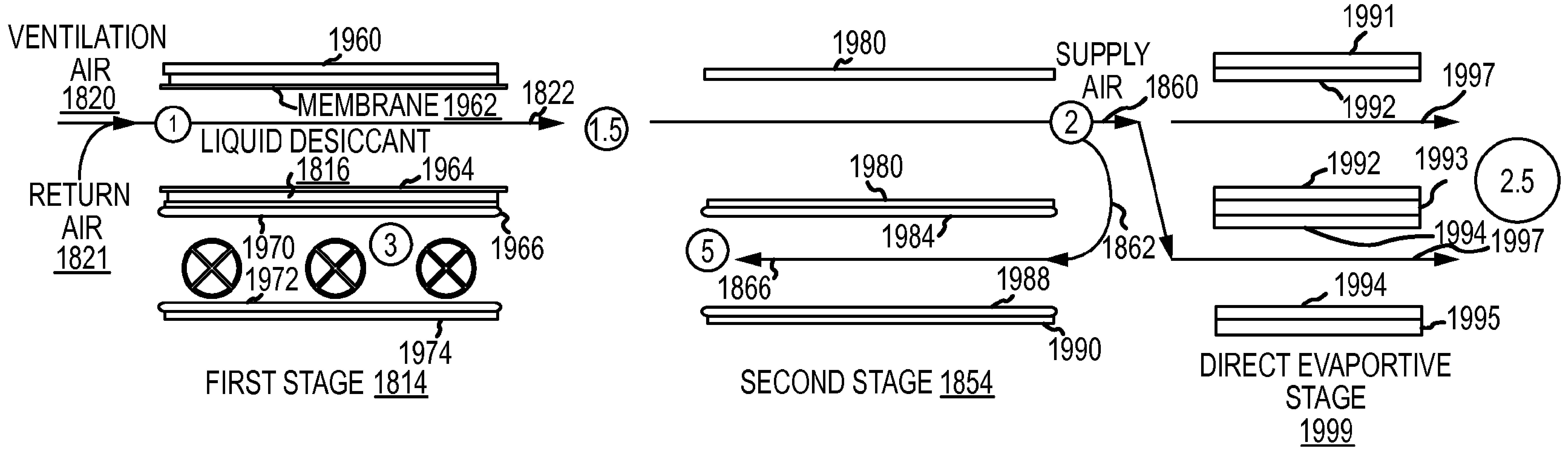

VENTILATION
AIR
1820
RETURN
AIR
1821
1
1960
MEMBRANE 1962
LIQUID DESICCANT
1816
1964
1822
1.5
1966
1970
1972
3
1974
FIRST STAGE 1814
1980
SUPPLY
AIR
1860
2
1980
1984
1862
5
1866
1988
1990
SECOND STAGE 1854
1991
1992
1997
1992
1993
2.5
1994
1997
1994
1995
DIRECT EVAPORTIVE
STAGE
1999

FIG.21

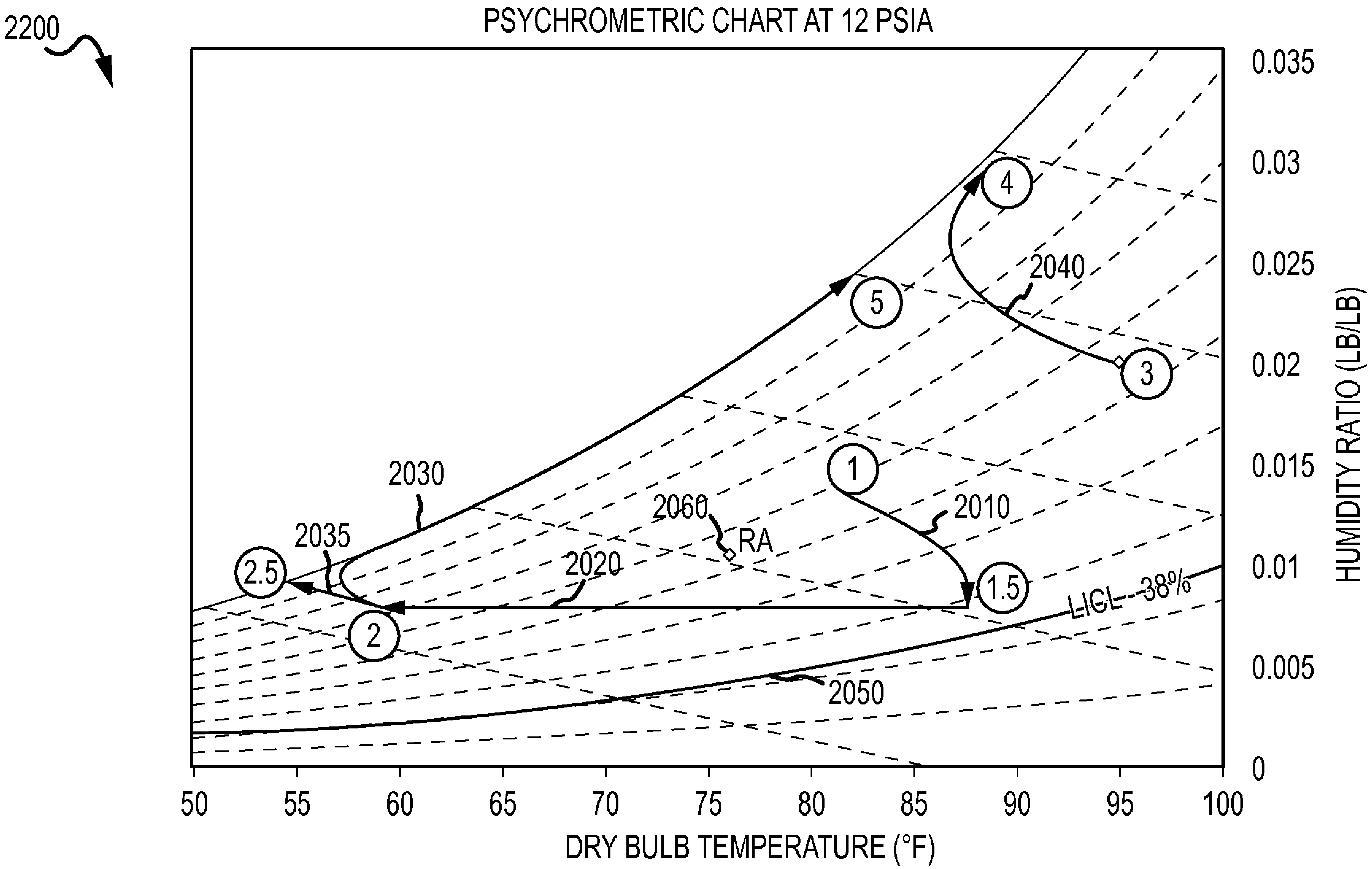

2200
PSYCHROMETRIC CHART AT 12 PSIA
HUMIDITY RATIO (LB/LB)
0.035
0.03
0.025
0.02
0.015
0.01
0.005
0
DRY BULB TEMPERATURE (°F)
50
55
60
65
70
75
80
85
90
95
100
4
2040
5
3
2030
1
2010
2060
RA
2035
2020
2.5
1.5
LICL--38%
2
2050

FIG.22

INDIRECT EVAPORATIVE COOLER USING MEMBRANE-CONTAINED LIQUID DESICCANT FOR DEHUMIDIFICATION AND FLOCKED SURFACES TO PROVIDE COOLANT FLOW

RELATED APPLICATIONS

[0001] This application claims the benefit of U.S. Provisional Application No. 61/662,146, filed Jun. 20, 2012, which is incorporated herein in its entirety.

CONTRACTUAL ORIGIN

[0002] The United States Government has rights in this invention under Contract No. DE-AC36-08G028308 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] Air conditioning is used worldwide to provide comfortable and healthy indoor environments that are properly ventilated and cooled and that have adequate humidity control. While being useful for conditioning supply air, conventional air conditioning systems are costly to operate as they use large amounts of energy (e.g., electricity). With the growing demand for energy, the cost of air conditioning is expected to increase, and there is a growing demand for more efficient air conditioning methods and technologies. Additionally, there are increasing demands for cooling technologies that do not use chemicals and materials, such as many conventional refrigerants, that may damage the environment if released or leaked. Maintenance is also a concern with many air conditioning technologies, and, as a result, any new technology that is perceived as having increased maintenance requirements, especially for residential use, will be resisted by the marketplace.

[0004] Evaporative coolers are used in some cases to address air conditioning demands or needs, but, due to a number of limitations, conventional evaporative coolers have not been widely adopted for use in commercial or residential buildings. Evaporative coolers, which are often called swamp coolers, are devices that use simple evaporation of water in air to provide cooling in contrast to conventional air conditioners that use refrigeration or absorption devices using the vapor-compression or absorption refrigeration cycles. The use of evaporative cooling has typically been limited to climates where the air is hot and humidity is low such as in the western United States. In such dry climates, the installation and operating costs of a conventional evaporative cooler can be lower than refrigerative air conditioning. Residential and industrial evaporative coolers typically use direct evaporative cooling with warm dry air being mixed with water to change the water to vapor and using the latent heat of evaporation to create cool moist air (e.g., cool air with a relative humidity of 50 to 70 percent). For example, the evaporative cooler may be provided in an enclosed metal or plastic box with vented sides containing a fan or blower, an electric motor to operate the fan, and a water pump to wet evaporative cooling pads. To provide cooling, the fan draws ambient air through vents on the unit's sides and through the dampened pads. Heat in the air evaporates water from the pads, which are continually moistened to continue the cooling process. The cooled, moist air is then delivered to the building via a vent in the roof or a wall.

[0005] While having an operation cost of about one fourth of refrigerated air conditioning, evaporative coolers have not been widely used to address needs for higher efficiency and lower cost conditioning technologies. One problem with many sump coolers is that in certain conditions these evaporative coolers cannot operate to provide adequately cooled air. For example, air may only be cooled to about 75° F. when the input air is 90° F. and 50 percent relative humidity, and such cooling may not be adequate to cool a particular space. The problem may get worse as temperatures increase well over 100° F. as found in many locations in the southwest portion of the United States and elsewhere. As a result, the air conditioning system may need to include refrigerated air conditioning to cool the outlet air from the evaporative cooler, which results in a system that is more expensive to purchase, operate, and maintain.

[0006] Additionally, conventional evaporative coolers provide no dehumidification of the air and, in fact, often output air at 80 to 90 percent relative humidity, which may only be acceptable in very dry environments as very humid air reduces the rate of evaporation for occupants of the building (e.g., reduces comfort levels) and can cause condensation resulting in corrosion or other problems. Dehumidification is provided as a second or later stage in some evaporative coolers such as by wicking a liquid desiccant along a wall of the air flow channel or chamber, but such systems have not been widely adopted due to increased operating and maintenance costs and concerns of having the desiccant expelled with the conditioned air. In general, maintenance is a concern with evaporative coolers as the evaporation process can result in mineral deposits on the cooling pads and other surfaces of the cooler that need to be cleaned or replaced to maintain the efficiency of the system, and the water supply line needs to be protected against freezing during the off season such as by draining the system. Due to these and other concerns, evaporative cooling is unlikely to be widely used to provide an energy efficient, air conditioning alternative for commercial and residential applications until significant improvements are made that address maintenance concerns while improving achievable cooling (e.g., providing adequately cooled output air for direct use in a building).

[0007] The foregoing examples of the related art and limitations related therewith are intended to be illustrative and not exclusive. Other limitations of the related art will become apparent to those of skill in the art upon a reading of the specification and a study of the drawings.

SUMMARY

[0008] The following embodiments and aspects thereof are described and illustrated in conjunction with systems, tools and methods that are meant to be exemplary and illustrative, not limiting in scope. In various embodiments, one or more of the above-described problems have been reduced or eliminated, while other embodiments are directed to other improvements.

[0009] This is achieved, in part in some applications, by providing a mass/heat transfer assembly for use in indirect evaporative coolers or heat exchangers. The assembly is formed of alternating stacks each including a first (or upper) layer or sheet of membrane material, a separation wall, and a second (or lower) layer or sheet of membrane material. The membrane or membrane material for each layer is permeable to water molecules in the vapor state while the separation wall is impermeable to water but allows heat transfer (e.g., is a thin layer and/or is made of materials that conduct heat). In a first one of adjacent pairs of stacks, coolant such as water flows between the first membrane layer and the separation wall and liquid desiccant flows between the separation wall and the second membrane layer while in the second or next one of the adjacent pairs of stacks the flow order is reversed. This ordering is repeated throughout the mass/heat transfer assembly to form alternating supply and exhaust air flow channels or chambers. Supply air (or air to be conditioned) is directed through a channel between a first pair of stacks while a portion of the pre-cooled exhaust air (e.g., a fraction of the supply air that is cooled by flowing through the stacks) is directed through a chamber between a second or next pair of stacks (e.g., typically in a counterflow arrangement relative to the flow of the incoming supply air). Liquid desiccant is provided proximate to the supply inlet airflow while coolant such as water is provided proximate to the exhaust airflow (i.e., a fraction of supply outlet airflow directed to be exhausted) with the air only being separated from these flowing liquids by the water permeable membrane. The supply air inlet airflow, supply outlet airflow, exhaust airflow, liquid desiccant flow, and coolant flow are plumbed such as via one or more manifold assemblies to the mass/heat transfer assembly, which can be provided in a housing as a single unit (e.g., an indirect evaporative cooler).

[0010] In a typical embodiment, dehumidification and evaporative cooling are accomplished by separation of the air to be processed and the liquid and/or gas substances (e.g., liquid desiccant, water, desiccated air, and the like) by a membrane. The membrane is formed of one or more substances or materials to be permeable to water molecules in the vapor state. The permeation of the water molecules through the membrane is a driving force behind (or enables) dehumidification (or dehumidification in some implementations) and evaporative cooling of one or more process air streams. As described above, multiple air streams can be arranged to flow through chambers in the mass/heat transfer assembly such that a secondary (purge) air stream, such as the exhaust airflow of pre-cooled supply air, is humidified and absorbs enthalpy from a primary (process) air stream, such as the supply inlet airflow that can then be directed to a building as supply outlet airflow (e.g., make up air for a residential or commercial buildings or the like). The process air stream is sensibly cooled and is, in some embodiments, simultaneously dehumidified by providing a liquid desiccant flow contained by membranes defining the sidewalls of the supply inlet airflow channel or chamber.

[0011] The membrane is also used in some embodiments to define sidewalls of the exhaust (e.g., counter) airflow channel or chamber such that the membrane controls or separates coolant liquid from the exhaust air stream. Wicking materials/surfaces or other devices may be used to contain or control water flow (e.g., direct-contact wicking surfaces could be used in combination with the use of the liquid desiccant containment by a membrane), but membrane liquid control facilitates fabrication of the stacks or manifold structure useful for heat and mass exchanger/assembly configurations described herein that provide cooling, dehumidification, and/or humidification. In such configurations, the air streams can be arranged in counter-flow, counter-flow with pre-cooled exhaust air, cross-flow, parallel flow, and impinging flow to perform desired simultaneous heat and mass transfer in the evaporative cooling units.

[0012] By way of example, but not limitation, an embodiment includes an indirect evaporative cooler for cooling a stream of inlet supply air from a first temperature to a second, lower temperature using a stream of liquid coolant and a stream of exhaust or purge air. The cooler includes a first flow channel through which the stream of inlet supply air flows and a second flow channel adjacent the first flow channel through which the stream of exhaust air, at a lower temperature than the inlet or first temperature of the supply air, flows. The second flow channel is formed or defined in part by a sheet of a membrane or membrane material that is permeable to water vapor but that otherwise contains the liquid coolant. In this manner, the coolant flows on a side of the membrane (and not in direct contact with) the air in the second flow channel but mass is transferred as a vapor through the membrane to the exhaust air when or in response to heat being transferred from the inlet supply air to the liquid coolant. In some cases or configurations, as will become clear, the supply air stream (or inlet supply air) is cooled and dehumidified in this first stage. A second stage may be provided to sensibly cool the air stream to a very cool temperature, which could be below the dewpoint of the original supply inlet air as it was dehumidified initially or in the first state to allow this.

[0013] A separation wall that is spaced apart from the sheet of membrane is used to define a flow channel for the liquid coolant, with the wall being formed from a material (such as plastic) that is impermeable to the liquid coolant but that conducts or allows the heat to be transferred from the inlet air supply to the coolant. A second sheet of membrane may be spaced apart from the opposite side of this separation wall to define a flow channel for a liquid desiccant, and during operation, water vapor is transferred from the stream of inlet supply air through the membrane to the liquid desiccant, which results in the inlet supply air being concurrently cooled and dehumidified. The membrane is effective for resisting or even fully blocking flow of the liquid coolant and the liquid desiccant while allowing flow of water vapor, and, in some embodiments, the coolant is water and the desiccant is a halide salt solution (e.g., a weak desiccant such as CaCl or the like). The exhaust air in some cases is a redirected portion of the stream of inlet supply air after it has been cooled to the second, lower temperature (e.g., as it is exiting the first flow channel), and the exhaust air may flow in a direction through the second flow channel that is cross, counter, or a combination of these relative to the supply air flowing in the first flow channel.

[0014] In another exemplary embodiment, a method is provided for conditioning a process or return air for a residential or commercial building. The method includes first directing the process air through a first flow channel and second directing a stream or volume of liquid desiccant adjacent one or more walls defining the first flow channel, the liquid desiccant is separated from the process air by a membrane (e.g., the membrane provides the walls) that contains the liquid desiccant and also allows water vapor from the process air to flow into and be absorbed by the liquid desiccant, which dehumidifies the process air. The method further includes concurrent with the first and second directing, third directing a stream of purge air through a second flow channel proximate to the first flow channel (e.g., parallel and adjacent). The purge air is at a temperature lower than all or at least a substantial portion of the process air in the first flow channel, and in some cases, the purge air is a fraction of the dehumidified process air exiting the first flow channel that is directed in a counter flow direction relative to the process air through the second flow channel. The method also includes fourth directing a stream of liquid coolant adjacent a wall of the second flow channel. The liquid coolant is also separated from the air by a membrane that is permeable to vapor from the coolant such that mass is transferred from the coolant to the purge air. The method provides for concurrent (or single stage) dehumidification and cooling of the process air.

[0015] According to another aspect, a mass and heat transfer assembly is provided for use in an indirect evaporative cooler or exchanger device. The assembly includes a first stack including an upper membrane, a lower membrane, and a separation wall between the upper and lower membranes. The upper and lower membranes are permeable to water in vapor form and the separation wall is substantially impermeable to liquid and vapor. Second and third stacks are provided that also each includes an upper membrane, a lower membrane, and a separation wall positioned therebetween. In the assembly, the first stack and second stacks are spaced apart (such as less than about 0.25 to 0.5 inches apart) to define a flow channel for receiving a first stream of air (e.g., air to be conditioned) and the second and third stacks are spaced apart to define a flow channel for a second stream of air (e.g., purge or exhaust air directed in cross or counter flow relative to the first stream of air). In some configurations and/or operating modes, the device does only evaporative cooling and no dehumidification. Such that the membranes are only used on the purge side and the other side of the wall is left bare for the supply air to exchange heat.

[0016] The first, second, and third stacks may be considered a set of stacks, and the assembly includes a plurality of such sets of stacks to define a plurality of air flow channels spaced apart by the stacks or layers of membranes and separation walls. A divider or separator may be provided in the flow channels to maintain spacing of the membranes while allowing flow of the air streams in the channels. The assembly may further include in the first stack a liquid coolant flowing between the upper membrane and the separation wall and a liquid desiccant flowing between the separation wall and the lower membrane. In the second stack, a liquid desiccant flows between the upper membrane and the separation wall while a liquid coolant flows between the separation wall and the lower membrane. In the third stack, liquid desiccant flows between the upper membrane and the separation wall while liquid coolant flows between the separation wall and the lower membrane. The liquid coolant may be water and during operation water vapor may be transferred from the coolant through the membrane to the second stream of air. The liquid desiccant may be a salt solution (such as weak desiccant such as CaCl, or the like) and during operation or use of the assembly water vapor may be transferred from the first stream of air through the membrane to the liquid desiccant, whereby the first stream of air is simultaneously dehumidified and cooled to a lower temperature.

[0017] In some cases, the mass and heat transfer assembly may be configured with no membrane on the coolant (e.g., water) side of the device. In such a mass transfer assembly, liquid desiccant is contained by a vapor permeable membrane in the combined stacks, as discussed above. However, the coolant, which in many cases is water, is allowed to flow without membrane containment. To this end, the coolant is maintained or attached on a surface or side of the separator or separation wall through the use of surface tension forces on a wicked or flocked surface (e.g., a wicking layer is attached to the separation wall surface). The flocked surface or layer of wicking material is attached to the separation wall, and, thus, there is direct thermal contact between the separation wall and the liquid coolant (e.g., water flowing through the wicking material). Water evaporation occurs freely between this coolant-soaked/containing surface on the separation wall and the purge or exhaust air stream.

[0018] Further, the mass and heat transfer assembly may include a humidification stage. The heat and mass transfer assembly may include an assembly or section where water adjacent to a supply air stream is membrane contained with a vapor permeable membrane or in a layer of wicking material. The supply air would be in contact with the membrane and allow for humidification of the supply air stream prior to discharge from the mass and heat transfer assembly (e.g., a humidifier stage provided downstream from the sensible or indirect evaporative cooler stage).

[0019] In addition to the exemplary aspects and embodiments described above, further aspects and embodiments will become apparent by reference to the drawings and by study of the following descriptions.

BRIEF DESCRIPTION OF THE DRAWINGS

[0020] Exemplary embodiments are illustrated in referenced figures of the drawings. It is intended that the embodiments and figures disclosed herein are to be considered illustrative rather than limiting.

[0021] FIG. **1** illustrates in schematic form an evaporative cooler or heat exchanger including an exemplary representative of a permeable membrane stack or assembly for use in providing indirect evaporative cooling concurrently with dehumidification in an integral unit or single stage;

[0022] FIG. **2** illustrates another exemplary representation of an evaporative cooler showing an assembly of membrane/wall/membrane stacks used in combination to direct the supply and exhaust airflows relative to membrane-contained liquid desiccant and coolant (e.g., cooling water) to achieve cooling and dehumidification;

[0023] FIG. **3** illustrates an evaporative cooler similar to that shown in FIG. **2** but being configured with integral counterflow passages for exhaust/cooled air;

[0024] FIG. **4** is a top view of an exemplary heat exchanger illustrating air flows through a plurality of channels or chambers provided by membrane-based assemblies such as those shown in FIGS. **1**-**3** or other embodiments shown or described herein;

[0025] FIG. **5** (specifically, **5**A and **5**B) illustrates an exemplary modeling of an evaporative cooler or counterflow heat/mass exchanger such as one with the stack assembly shown in FIG. **2** and flow arrangement shown in FIG. **4**;

[0026] FIG. **6** is a graph of air flow and surface temperatures along the length of the exchanger modeled as shown in FIG. **5**;

[0027] FIG. **7** is a graph of humidity ratios of the air along the length of the exchanger modeled as shown in FIG. **5**;

[0028] FIG. **8** is a graph showing concentration of liquid desiccant flowing through the modeled heat exchanger of FIG. **5**;

[0029] FIG. **9** is a psychrometric chart showing the cooling and dehumidifying process modeled as shown in FIG. **5**;

[0030] FIG. **10** is a top view of another exemplary heat exchanger illustrating air flows through a plurality of channels or chambers provided by membrane-based assemblies such as those shown in FIGS. **1**-**3**, or other embodiments shown or described herein;

[0031] FIG. **11** is a top view of another exemplary heat exchanger similar to those shown in FIGS. **4** and **10** showing a differing unit arrangement with differing exhaust airflows;

[0032] FIG. **12** is a psychrometric chart showing the cooling and dehumidifying process modeled similar to the modeling shown in FIG. **5** for the configuration of a heat exchanger shown in FIG. **10**;

[0033] FIG. **13** illustrates a HVAC system using an indirect evaporative cooler to provide conditioned air to a building;

[0034] FIG. **14** is a psychrometric chart providing results of one test of a prototype fabricated similar to the embodiment of FIG. **4** with the stack assembly of FIG. **2**;

[0035] FIG. **15** illustrates in schematic form an evaporative cooler or heat exchanger similar to that shown in FIG. **1** including another representative permeable membrane stack or assembly;

[0036] FIGS. **16** and **17** illustrate in schematic form two humidification sections (or portions of a stack that may be provided in such a section), each of which makes use of a wicking layer wetted with water or other humidification fluids/sources;

[0037] FIGS. **18**, **19**, and **20** provide, respectively, a schematic side view of a two-stage evaporative cooler, a top view of a pair of first and second stage stacks used to form the cooler, and a psychrometric chart of the cooling process provided during operation of the two-stage evaporative cooler; and

[0038] FIGS. **21** and **22** provide, respectively, a top view of a cooler similar to that of FIG. **19** but with an added direct evaporative stage and a psychrometric chart of the cooling process during operation of the cooler.

DETAILED DESCRIPTION

[0039] The following provides a description of exemplary indirect evaporative coolers with dehumidification and mass/heat transfer assemblies for such coolers that provide inlet air stream chambers with sidewalls defined by permeable membrane sheets containing liquid desiccant. The assemblies also include outlet or exhaust air stream chambers (such as in counterflow to the inlet air streams) with sidewalls defined by permeable membrane sheets containing coolant such as water. In embodiments described below, the membrane is "permeable" in the sense that moisture in the form of a vapor (e.g., water in the vapor state) generally can permeate readily through the membrane such as from an inlet supply air and from liquid coolant via evaporation. However, the membrane generally contains or blocks moisture in the form of a liquid from flowing through as it is instead directed to flow within the channel or chamber. In some cases, water in the liquid state is contained by the membrane at pressures less than about 20 psi and more typically less than about 5 psi. The coolant and the liquid desiccant in some embodiments are maintained at pressures below about 2 psi, and the permeable membrane contains moisture such as water in the liquid state while water vapor permeates the membrane.

[0040] As will become clear from the following description, use of the assemblies such as for evaporative coolers or mass/heat exchangers provides a number of benefits. The inlet or process air stream can be cooled and dehumidified simultaneously or in a single chamber/stage, and this combined action reduces system size and cost as well as the number of required components and equipment (e.g., do not require a multi-stage unit or device to cool and then to dehumidify and/or further cool with refrigerant or the like). The combination of liquid desiccant dehumidification with indirect evaporative cooling provides very high energy transfer rates due to evaporation and absorption. The design creates a liquid desiccant system that does not require separate equipment for liquid desiccant cooling (e.g., a separate cooling tower or chiller). The stacked arrangements or multi-layered mass/heat transfer assemblies (or manifolded flow chambers/channels) enable ultra-low flow liquid desiccant designs. This is due in part to the enhanced geometry of the assembly and its ability to decrease the liquid desiccant's temperature to a lower temperature than achievable with traditional cooling tower technologies. Hence, in the cooler, there are higher concentration gradients of liquid desiccant (e.g., more than 20 percentage points of lithium chloride (LiCl) and similar gradients for other desiccants), which provides the following advantages: (a) a higher thermal coefficient of performance (COP) to regenerate the desiccant (i.e., to remove water from the desiccant) for reuse in the cooler; (b) less desiccant storage requirements due to better utilization; and (c) the ability to use desiccants that are less expensive than LiCl such as calcium chloride (CaCl), which may not be used in conventional systems because their absorption properties are not as favorable as LiCl but lower temperature operation provided by the cooler embodiments described herein makes the properties of this and other "weaker" desiccants more acceptable or favorable.

[0041] The use of membranes as chamber sidewalls facilitates fabrication of counter-flow and counter-flow with precooled exhaust air embodiments. Liquid desiccant containment with water molecule-permeable membranes eliminates liquid desiccant "carry over" in which small droplets of desiccant are passed into the air stream as is a concern with direct contact arrangements. The embodiments described herein also provide considerable reduction or even elimination of deposited solids during the process of water evaporation or adsorption (and liquid flow rates can be maintained at levels that are high enough to further control potential deposits) whereas fouling leads to increased maintenance and operating costs with prior evaporative coolers.

[0042] FIG. **1** illustrates in a schematic an evaporative cooler (or mass/heat exchanger) **100** that is useful for providing concurrent or simultaneous dehumidifying and cooling of a process or inlet air stream **120** (e.g., outdoor or process air to be cooled and conditioned prior to being fed into a building ventilation system). The cooler **100** is shown in simplified form with a housing shown in dashed lines, without inlet and outlet ducts, plumbing, and/or manifolds. Also, the cooler **100** is shown with a single mass/heat transfer stack **110** whereas in a typical cooler **100** there would be numerous stacks **110** provided by repeating the configuration shown (e.g. by alternating the liquid passed through the chamber defined by the membrane and wall) to provide an assembly with a plurality of air and liquid flow channels or chambers to provide the desired mass and heat transfer functions described for the stack **110**.

[0043] As shown, an inlet air stream **120** is directed in a chamber or channel defined in part by a sheet or layer of a membrane **112**. Liquid desiccant **124** flows in an adjacent chamber or channel on the other side of the membrane **112**.

The liquid desiccant **124** is contained by the membrane **112**, which is permeable to water molecules in a liquid or vapor state but generally not to the components of the liquid desiccant **124**. The chamber for the desiccant flow **124** is also defined by a sheet or layer of material that is impermeable to fluid flow (i.e., a separation wall) **114** so as to contain the liquid desiccant **124** in the chamber or flow path. The chamber for stream **120** is also defined by an opposing membrane (not shown) that is used to contain another flow of liquid desiccant. In this manner, heat is passed or removed from the inlet air stream **120** and transferred to the liquid desiccant flow **124** (and the desiccant behind the opposite sidewall/membrane (not shown)). Concurrently, the inlet air stream **120** is dehumidified as water **130** is removed by passing through the permeable membrane **112** into liquid desiccant **124**.

[0044] The liquid (or gas) desiccant **124** may take many forms to act to dehumidify and cool the air stream **120** as it passes over the membrane **112**. Desiccant **124** is generally any hygroscopic liquid used to remove or absorb water and water vapor from an air stream such as stream **120**. Preferably, the desiccant **124** chosen would be a regenerable desiccant (e.g., a desiccant that can have the absorbed water separated and/or removed) such as a glycol (diethylene, triethylene, tetraethylene, or the like), a salt concentrate or ionic salt solution such as LiCl, CaCl, or the like, or other desiccants. The membrane **112** may be formed of any material that functions to contain liquid desiccant **124** and, typically, coolant **126** (e.g., water or the like) while also being permeable to molecules of water in liquid or vapor state. For example, polymer membranes may be used that have pores that are about the size or just bigger than a water molecule and, in some cases, that are also adapted to provide water molecules with high mobility through the membrane **112**. In one particular embodiment, the membrane **112** is formed from a membrane material as described in detail U.S. Pat. No. 6,413,298 to Wnek, which is incorporated in its entirety herein by reference. The membrane material may also be obtained from a number of distributors or manufacturers such as, but not limited to, Dias-Analytic Corporation, Odessa, Fla., U.S.A. The membranes **112**, **118** and separation wall **114** preferably also are formed from materials that are resistive to the corrosive effects of the desiccant, and in this regard, may be fabricated from a polymer or plastic with the wall **114** in some cases being formed of a corrosion resistant metal or alloy, which provides a higher thermal conductivity compared with a plastic.

[0045] The embodiment **100** shown is configured for counter-flow of the pre-cooled exhaust air stream **128** (relative to the inlet air stream **120**). Other embodiments may use cross (at about a 90 degree flow path) or quasi-counter flow (e.g., not directly counter or opposite in direction but transverse such as a greater than 90 degree angle flow path relative to air stream **120**). The exhaust air stream **128** flows in a channel or chamber defined by a sheet or layer of membrane (e.g., second or lower membrane) **118** and an upper membrane of another stack (not shown). The separation wall **114** and membrane **118** define a flow chamber or channel for coolant flow **126**, which is typically a flow of water or the like. Heat is transferred from the liquid desiccant **124** to the coolant **126** through the separation wall, and the coolant **126** is cooled as heat and mass (e.g., water or other moisture **132**) is transferred to the exhaust stream **128** via membrane **118**. Heat transfer is not shown but generally is flowing through the membrane **112** to the liquid desiccant **124**, through the separation wall **114** from the liquid desiccant **124** to the coolant **126**, and through the membrane **118** from the coolant **126** to the exhaust air stream **128**. The membranes **112**, **118** are relatively thin with a thickness, t_{mem}, that typically is less than 0.25 inches and more typically less than about 0.1 inches such as 100 to 130 microns or the like. The membrane **112**, **118** may have a tendency to expand outward if unrestrained, and, in some embodiments, such as that shown in FIG. **3**, a divider or “flow field” support is provided in the inlet air stream **120** and exhaust air stream **128** (i.e., in the airflow chambers) to maintain the separation of the adjacent membranes (e.g., a plastic or metallic mesh with holes or openings for air flow and a zig-zag, S or W-shaped, or other cross section (or side view) that provides many relatively small contact points with the membranes **112**, **118**). The separation wall **114** also typically is relatively thin to facilitate heat transfer between the desiccant **124** and coolant **126** such as with a thickness, t_{wall}, of less than 0.125 inches or the like. The flow chambers for the air, desiccant, and coolant are also generally relatively thin with some applications using chambers less than 1 inch thick (or in depth) while others use chambers less than about 0.5 inches, such as about 0.25 inches or less.

[0046] FIG. **2** illustrates an indirect evaporative cooler **210** utilizing the membrane/separation wall/membrane stack or assembly configuration to provide a mass/heat transfer exchanger device in which dehumidification and cooling occur within a single stage and, therefore, an integral or unitary device. In some embodiments (not shown), there is no desiccant side membrane or desiccant flow. Thus, these embodiments are useful for providing an indirect evaporative cooler in which the membrane contains liquid coolant but not liquid desiccant and the membrane typically would not be provided on the supply air side (or in these channels) to provide better heat transfer surfaces with the separation wall. As shown in FIG. **2**, the cooler **210** includes a mass/heat transfer assembly formed from stacks or devices **212**, **230**, **240** and such an assembly of stack would typically be repeated to provide a plurality of inlet and exhaust air, coolant, and desiccant flow channels or chambers in the cooler **210**. As shown, each set of stacks (or layered assemblies or devices) **212**, **230**, **240** is formed similarly to include a membrane, a separation wall, and a membrane, with the membrane being permeable to water on the molecular level to allow mass and heat transfer and the wall being impermeable (or nearly so) to only allow heat transfer and not mass transfer.

[0047] Specifically, the stack **212** includes an upper membrane layer **214**, a separation wall **216**, and a lower membrane layer **218**. Dividers or spacers (not shown) would typically be provided to space these layers apart to define flow channels for coolant **215** and for liquid desiccant **217**. For example, the separators may be configured to also provide a connection to a supply line for coolant and for regenerated desiccant, provide a manifold(s) to direct flow through the various stacks **212**, **230**, **240**, and provide a connection to a return line for the coolant and diluted desiccant. The stacks **230** and **240** likewise include an upper membrane layer **232**, **242**, a separation wall **234**, **244**, and a lower membrane layer **238**, **248**. The stack **240** has coolant (such as water) **243** directed in the chamber between the upper membrane **242** and wall **244** and desiccant **246** flowing between the wall **244** and lower membrane layer **248** similar to stack **212**. In contrast, the stack **230** has liquid desiccant **233** directed to flow in the chamber defined by the upper membrane layer **232** and wall **234** and has coolant **236** directed to flow in the chamber or channel defined by the wall **234** and lower membrane layer **238**.

[0048] The cooler **210** includes ducting and the like (not shown) to direct supply inlet air **250** through the channel or flow path between the stack **212** and the stack **230**. The arrangement of the stacks **212**, **230**, **240** and contained fluids results in the supply inlet air **250** being passed over the surfaces of the membranes **218**, **232** that are containing liquid desiccant **217**, **233**. As a result, supply outlet air **254** is output that is dehumidified as moisture in the air **250** is absorbed by the desiccant **217**, **233** via permeable membrane **218**, **232**, and the air **254** is also cooled by the interaction with desiccant **217**, **233**. The cooling effect in the cooler **210** is in part effected by a fraction of supply outlet air **254** being redirected in the cooler **210** by ducting/manifolds (not shown) to flow as pre-cooled exhaust air **255** through the channel or flow path between stacks **230**, **240** to be output as warmer and moister air **258**. Heat passes from desiccant **233** through wall **234** to coolant **236** (with similar heat transfer occurring in stacks **212**, **240**), and the coolant **236** is able to transfer heat and mass (e.g., water molecules) via membrane **238** to the incoming exhaust air **255**. As discussed above, the stack pattern or set provided by **212**, **230**, **240** would typically be repeated within the cooler **210** to create a mass/heat transfer assembly with numerous, parallel flow channels for air, coolant, and desiccant.

[0049] The cooler **210** is shown as a counter flow exchanger, but other flow patterns may be used to practice the desiccant-based dehumidification and cooling described herein. For example, cross flow patterns may readily be established as well as quasi (or not fully opposite) counter flow patterns. These patterns may be achieved by altering the manifolding and/or ducting/plumbing of the cooler as well as the dividers provided between the stacks. Additionally, the counter flow passages may be provided integral to the stack assembly rather than externally as is the case in the cooler **210**. For example, the cooler **310** has a similar stack arrangement as shown in the cooler **210** of FIG. **2** except that it includes a counterflow baffle or dividing wall **360** (FIG. **3**) on the end of the flow channels for inlet air **250** and exhaust air **258**. The counterflow divider **360** allows a majority of the cooled air to exit the stacks as supply outlet air **354** (e.g., more than about 50 percent and more typically 60 to 90 percent or more of the air flow **250**). A smaller portion (e.g., a volume equal to the make up outdoor air or the like) is directed by divider **360** to flow between stacks **230**, **240** as pre-cooled exhaust air **355**. FIG. **3** also illustrates the use of a divider or flow field baffle **370** that functions to maintain a separation of membranes in the stacks **212**, **230**, **240** (or at about their original thickness rather than puffed out or expanded as may occur with some permeable membranes). The dividers **370** may take many forms such as a mesh with a wavy pattern (e.g., an S or W-shaped side or cross sectional view), with the mesh selected to provide as little resistance to air flow as practical while still providing adequate strength. Also, it is desirable to limit the number of contact points or areas with the membranes as these can block moisture transfer from the air **250** and to the air **355**.

[0050] FIG. **4** illustrates an indirect evaporative cooler **400** of one embodiment. A housing **410** is provided for supporting a mass/heat transfer assembly such as one formed with the stack sets shown in FIGS. **1-3** and **15-19**. As shown, the housing **410** includes a first end **412** with an inlet **414** for supply inlet airflow **415** and an outlet **416** for exhaust airflow **417**. The cooler **410** further includes a second end **418** opposite the first end **412** that provides an outlet or vent for directing supply outlet airflow **420** to an end-use device or system (e.g., an inlet or supply for return air to a building). The second end **418** is also configured to redirect a portion **426** of the cooled (and, in some operating modes, dehumidified) air **426** for use in counter flow cooling of the supply inlet airflow **415**. A prototype of the cooler **400** was fabricated with a stack assembly as shown in FIG. **2** with 32 desiccant channels. The prototype was tested with 10 liters per minute (LPM) flow (or about 0.3 LPM per desiccant channel). Coolant was provided as water at a water flow rate of about 1.25 to 2.00 times the evaporation rate. The evaporation rate for this prototype was about 1.33 gallons/ton-hr or about 5 liters/ton-hr, which provides a water or coolant flow rate of about 6-10 liters/ton-hr of cooling. Of course, these are exemplary and not limiting flow rates, and it is expected that the flow rates of liquid desiccant and coolant will depend on numerous factors and will be matched to a particular channel design and cooling need as well as other considerations.

[0051] An indirect evaporative cooler such as the cooler **400** using stack sets as shown in FIG. **2** (or FIGS. **15-19**) may be modeled to determine the effectiveness of the use of a permeable membrane to contain coolant and liquid desiccant. FIGS. **5**A and **5**B provide a diagram **500** of such modeling showing use of stacks **212**, **230**, and **240** as discussed with reference to FIG. **2** to cool inlet or process air and to also dehumidify this air in the same stage or process. The inputs to the model **500** are shown, and results for a typical inlet air condition are provided, with results and modeling being performed in this case with Engineering Equation Solver (EES). The numeric values shown in boxes or with squares around them are input values (or assumed typical operating conditions), and the values outside or without boxes are outputs or results of the modeling. The modeling results shown in the diagram **500** are believed to be self-explanatory to those skilled in the heating, ventilation, and air conditioning (HVAC) arts and do not require detailed explanation to understand the achieved effectiveness of the embodiments using membrane containment in indirect evaporative coolers; however, the following provides a graphical description of some of the results in the diagram **500**.

[0052] FIG. **6** illustrates a graph or diagram **610** showing the temperatures of the air flows in the channels between the stacks (e.g., in an evaporative cooler using such mass/heat transfer assembly described herein). The graph **610** also shows surface temperatures along the length of the counterflow mass/heat exchanger (e.g., exchanger **400** with stack arrangements as shown in FIG. **2**). Specifically, the graph **610** shows the temperature of supply air with line **612**, the temperature of exhaust/purge air with line **614**, the temperature of the desiccant side membrane surface (e.g., at the interface of the membrane and the supply air) with line **616**, the dewpoint temperature of the desiccant side membrane surface (e.g., at the interface of the membrane and the supply air) with line **620**, and the temperature of the water side membrane surface (e.g., at the interface of the membrane and the exhaust/purge air) with line **618**.

[0053] FIG. **7** is a graph or diagram **710** showing the humidity ratios of the air along the length of the counterflow heat/mass exchanger. Specifically, the graph **710** shows the bulk humidity ratio of the supply air with line **712**, the bulk humidity ratio of exhaust/purge air with line **714**, the humidity ratio of the air in close proximity to the desiccant side membrane surface (e.g., at the interface of the membrane and the supply air) with line **716**, and the humidity ratio of the air in close proximity to the water side membrane surface (e.g., at the interface of the membrane and the exhaust/purge air) with line **718**.

[0054] FIG. **8** illustrates a graph **810** showing with line **815** the concentration of desiccant (in this particular modeling the desiccant is LiCl) as it flows concurrent with the supply air flow down the length of the counterflow mass/heat exchanger. As shown with line **815**, the desiccant is getting weaker as it flows through the channel between the membrane and the separation wall as it absorbs water molecules from the air, e.g., the concentration of the desiccant is dropping from about 44 percent down to about 24 percent in this particular modeling example (which results from the membrane being characterized as permeable (at a particular input rate or setting) to water molecules in the flowing air at these operating conditions).

[0055] FIG. **9** shows the process of model **500** of FIG. **5** in a psychrometric chart or diagram **910**. The supply air shown with line **912** can be seen to be gradually losing humidity (in kilograms water vapor/kilograms dry air or kg_v/kg_{da}). The supply air **912** has its temperature initially rise slightly due to the large heat flow of vapor sorption into the desiccant. As the supply air **912** continues down the length of the exchanger (or flow channel or chamber between membrane layers or walls of adjacent stacks containing liquid desiccant), the temperature then drops to a cooler/drier condition than at the inlet. At the exit of the exchanger, the supply air **912** is split into two streams. The majority of the air is supplied to the cooled space, and the minority of the air (such as less than about 50 percent and more typically less than about 30 percent of the volume) gets funneled into the exhaust/purge side (or exhaust/counterflow channels between the membrane walls containing coolant) of the heat/mass exchanger or cooler, which is shown with the line **916**. The exhaust air **916** has a low dewpoint, and, thus, it can pick up a large amount of heat evaporatively. The pre-cooled exhaust or purge air **916** picks up water vapor (and associated heat of vaporization) from the wet side channel. The air **916** exits out of the unit with a much higher enthalpy than either the supply inlet or exit shown with line **912**. The diagram **910** also shows the humidity ratio and temperature of the supply air in close proximity to the desiccant side membrane surface (ds) with line **918**.

[0056] The following table shows results in tabulated form for modeling of FIG. **5** for inlet and outlet air flows. As shown, a wide range of temperatures and humidity levels can be chosen and input into the model **500**. In the configuration whose results are shown in the table, the equivalent wet bulb effectiveness with the desiccant flow turned off (e.g., in some operating modes it may not be required or useful to utilize the desiccant to dehumidify the air) would be 113 percent, which means the cooler is able to cool the supply air below the inlet wet bulb temperature.

TABLE

Inlet and outlet conditions from model runs (° C. and kg/kg)

Run #	$T_{supply,in}$	$T_{supply,out}$	$T_{exhaust,out}$	$\omega_{supply,in}$	$\omega_{supply,out}$	$\omega_{exhaust,out}$
1	27.7	21.11	31.55	0.0133	0.00892	0.0289
2	50.0	33.7	50.7	0.0319	0.0179	0.0834
3	50.0	20.7	41.0	0.0077	0.00406	0.0494
4	30.0	13.1	27.2	0.00262	0.00158	0.0226
5	30.0	18.9	42.55	0.0269	0.0137	0.0547
6	15.0	16.9	25.4	0.0105	0.00418	0.0207
7	15.0	11.9	20.0	0.00528	0.00203	0.0147

[0057] where LiCl Inlet Concentration=44%; flow ratio (flow exhaust/(flow exhaust+flow supply)=0.3; supply outlet face velocity=175 SCFM; and ambient pressure=101.3 kPa.

[0058] The cooler **210** of FIG. **2** (or FIGS. **15-19**) may be thought of as a desiccant-enhanced, indirect evaporative cooler that utilizes a membranes or layers of membrane material that is permeable to water molecules to provide desired liquid containment. A standard psychrometric chart (such as one at **14.7** psi ambient pressure and other typical parameters) may be used to view lines of equal sensible heat ratios (SHRs) originating at a typical room setpoint. For vapor compression dehumidification, a SHR of less than about 0.7 is difficult to attain without reheat (e.g., given reasonable evaporator temperatures). Also, it is psychrometrically impossible to attain a SHR of less than about 0.6 without reheat, and attempting such a SHR often leads to frozen evaporator coils that require defrost cycles. The desiccant-enhanced, indirect evaporative cooler, such as shown in FIG. **2** at **210**, addresses this problem with a unique, new process (as has been described above and is presented in more detail below).

[0059] It may be useful at this point to review the process with reference to FIGS. **2** and **3**. FIGS. **2** and **3** show diagrams describing the inner flow channels of the unit or assembly for use in an evaporative cooler **210**, **310**. The mixed return/outdoor air is shown by the arrow **250** (e.g., return air from a conditioned space along with outdoor make up air such as 400 cfm/ton supply and 175 cfm/ton outdoor air or the like). The air **250** is dehumidified by the desiccant **217**, **233** through the membrane **218**, **232**. This lowers both the dew point and temperature of this air stream until it is output at **254** or **354**. At the exit of the supply air passage (between the liquid desiccant-containing membranes), a portion of the air is fractioned off as shown with arrows **255** and **355** and sent through an adjacent passage (between the coolant-containing membranes **238**, **242**) which picks up moisture from the water layer **236**, **243** through the membrane **238**, **242**. The heat of evaporation is a source of cooling that acts to remove the sensible heat and heat of absorption from the supply air stream **250**. This air is then exhausted (purged) out at **254**, **354**.

[0060] The heat exchanger configuration shown at **400** in FIG. **4** has been built in the laboratory and was modeled as shown in FIG. **5**. Other options for flow/housing designs are shown in configuration with the cooler **1000** of FIG. **10** and the cooler **1100** of FIG. **11**. The cooler **1000** is shown to have a housing **1010** with a first portion or end **1012** and a second portion or end **1020**. The first portion **1012** is configured with inlets or vents for receiving supply inlet airflow **1013** as well as input exhaust airflow **1014**, and the first portion **1012** also includes vents or outlets for outputting exhaust airflow **1015** from the unit **1000**. The second portion **1020** is configured (e.g., with manifolds and other components to direct air flow) with outlets for supply outlet airflow **1022** with a portion **1025** being redirected back into the housing **1010** as shown at arrows **1027** to provide counterflow for a fraction of the channel provided for supply inlet airflow **1013** (with exhaust airflow **1014** provided as a cross flow in the other or initial portion of the channel) and then this air is exhausted from the housing portion **1020** at **1028**. The input exhaust airflow **1014** may be return air to be exhausted or outdoor air (e.g., from the building space). This approach 1000 improves the efficiency by utilizing a smaller purge airflow **1025**, **1027**, and it is typically preferred to limiting purge air flow to increase or maintain desirable efficiency.

[0061] Referring again to FIG. **4**, operation of the cooler **400** is expected to have the cooling process shown in the psychrometric chart **910** of FIG. **9**. As shown, line **912** represents the supply air flow while line **916** represents the purge air flow stream. The desiccant side air boundary layer is represented with line **918**. The chart shows graphically how the dehumidification driver for the cooler **400** is advantageously utilized to provide a more effective cooler. The cooler **400** may use even a weak desiccant such as CaCl solution to provide significant dehumidification, and this is due in part to the cold temperatures that are achieved with the configuration of the cooler **400** that allow weak desiccants to attain high dehumidification potential.

[0062] The configuration shown with cooler **1000** of FIG. **10** was modeled to determine the desirability of its performance, and the results are provided in psychrometric chart **1200** of FIG. **12**. In the chart **1200**, line **1210** represents supply air, line **1212** represents ambient exhaust air, line **1214** represents desiccant side surface temperatures, line **1220** represents the supply air post cooling, line **1224** represents the purge air post cooling, and line **1230** is the sensible heat ratio (SHR) line in which the load on the building follows. So, for example, a building will have 0.67 units of sensible heat and 0.33 units of latent heat added to the space to arrive at the return air condition, which is the middle diamond at 80° F. and about 70 grains/lb, and that point may be considered the return air condition. The first point of line **1210** is the "mixed air" condition, which is a 30/70 mixture of outdoor air and return air. The two-stage approach to cooling provided by cooler **1000** allows the process to be split into two distinct sections of dehumidification plus a post cooling stage (e.g., sensible cooling only stage in which, for example, there is no desiccant layer and dehumidification and only evaporative cooling is provided). The cooler **1000** is, of course, only one example of numerous configurations that may be implemented to provide two or more stage cooling using the membrane containment features described herein, and it shows the possibility of attaining nearly any SHR desired (e.g., in this case, a SHR of about 0.67). In the modeling to provide the chart **1200** (FIG. **12**), a 1 cubic foot core (or mass/heat transfer assembly) was used with 176 SCFM, and a flow ratio of about 0.3 (e.g., 30 percent purge and 70 percent supply air). Also, the return air was at 80° F. and 40 percent relative humidity, ambient air was at 86° F. and 60 percent relative humidity, and the liquid desiccant fed into the assembly was 44 percent LiCl (but other desiccants such as solutions of salt (such as, but not limited to, halide salts) and water that are about 20 to 40 percent salt by weight may be used). The assembly was able to provide 0.5 tons of building cooling with just this 1 cubic foot at about 7 Btu/lb. As can be appreciated from this example and modeling, the use of membranes to contain desiccant and coolant (e.g., to contain liquids) enable indirect evaporative coolers to be produced that are much more compact than prior designs, that are easier to maintain (e.g., have less or no fouling issues), and that are more efficient in producing cooling (e.g., with simultaneous dehumidification and cooling to provide an evaporative cooler that can condition as well as cool process air).

[0063] FIG. **11** illustrates an evaporative cooler **1100** providing another counterflow arrangement in which the counterflow cooling air (or pre-cooled supply air) is directly opposite in direction but only for a selected length (such as half to 80 or 90 percent or more of the length) of the stacks or flow chambers (e.g., when full counterflow is not required or desired). As shown, the cooler **1100** includes a housing **1110** containing a plurality of stacks or sets of stacks configured as a mass/heat transfer assembly (as discussed above) with alternating flow channels for supply inlet airflow **1112** and for counterflow air (e.g., redirected supply outlet airflow **1114**). The housing **1110** includes venting and/or manifolding for directing the supply inlet airflow **1112** (e.g., outdoor make up air and return air) into channels between desiccant containing membranes and to output the cooled and, often, dehumidified supply outlet airflow **1114**. The cooler **1100** further includes ducting, manifolding, and the like for redirecting a fraction of the supply outlet airflow back into the housing **1110** to provide cooling counterflow air as shown at **1116** (e.g., into flow channels between coolant containing membranes). The counterflow air **1116** typically does not travel along the entire length of the housing **1110** but is, instead, discharged out a side vent at some point along a channel length (e.g., at a distance about 60 to 80 percent of the length). Such a configuration is useful to tune a cooler **1100** for particular operating environments (e.g., to provide a desired amount of cooling to the supply outlet airflow based on outside air temperatures and humidities and other operating parameters).

[0064] The stack and membrane technology described herein are readily applicable to a number of indirect evaporative cooler designs (with and without use of liquid desiccant for dehumidification) and applications. However, it may be useful to discuss the use of the technology within an air conditioning or HVAC system with the belief that those skilled in the art will readily understand that the technology is useful in many other such systems. FIG. **13** illustrates a simplified air conditioning system **1300** in which the membrane technology may be provided to disclose desiccant dehumidification and evaporative cooling to condition air within a building **1310** (e.g., a residential or commercial building or other structure requiring conditioned and cooled air). As shown, the system **1300** includes a cooler **1320** with a housing **1322** that is used to house a membrane stack assembly, such as described above with reference to FIGS. **1-12** and below with reference to FIGS. **15-20**. A fan or blower **1324** is provided to draw in outside or make up air **1325** and move return air **1326** from the building **1310**. The fan **1324** pushes these two air streams as inlet supply air through the stacks as described above (e.g., adjacent liquid desiccant contained in membrane in embodiments providing dehumidification or adjacent separation walls in embodiments with just evaporative cooling). The cooled (and, typically, conditioned air is output at **1330** as supply to the building **1310** and a portion is returned **1332** as purge or pre-cooled exhaust air that passes on the coolant or evaporative cooling side of the stacks in housing **1322** and then out as exhaust **1328**. Coolant is provided in the form of a water supply and drain **1334** to the housing (and through the stack assembly), and liquid desiccant is provided at **1338** as supply and drain. The desiccant **1338** is regenerated with a regenerator system **1340** including, in this example, a desiccant boiler **1342**.

[0065] The desiccant enhanced indirect evaporative cooler (DE-IDEC) **1320** is the portion of the system **1300** that takes strong desiccant and water to provide cooling to building **1310**. The system **1300** provides both sensible and latent cooling to building **1310** on demand and in proportion to the demand, e.g., the system **1300** can provide cooling in the form of 100 percent sensible, 100 percent latent, or any combination thereof. The DE-IDEC **1320** uses some portion of outdoor air **1325** with equal exhaust air **1328** to reject the heat load outside of the building **1310**. The DE-IDEC **1320** itself can sit inside or outside of the building envelope because it has no wet surfaces and the liquid streams **1334**, **1338** are closed loop. This makes system **1300** acceptable for indoor use and for placement of cooler **1320** inside the building **1310**. The water source (or coolant source, not shown) for water or coolant **1334** is not required to be potable, and the system **1300** is compact enough to be acceptable by building managers. The electricity usage is much less than that of typical vapor compression systems or units (e.g., less than 0.2 kW/ton peak compared with 1.2 kW/ton typical for conventional compression units).

[0066] The regenerator **1340** is another of the significant components to the operation of the system **1300**. This unit **1340** takes the weakened desiccant from the DE-IDEC **1320** and applies heat with boiler **1342** (see list of heat sources below) to drive off the moisture contained in the desiccant **1338**. The result is a desiccant **1338** that has higher salt concentration and can be re-used by the DE-IDEC **1320** (e.g., in the membrane contained/defined flow channels adjacent to supply inlet air **1325**, **1326**). A list of heat sources suitable for desiccant regeneration may include: (a) gas or other fossil fuel; (b) solar heat; (c) waste heat from any waste heat stream such as combine heat and power plant; and (d) waste heat from a condenser unit originating from a vapor compression cycle.

[0067] The inventors performed a test of a prototype fabricated similar to the cooler shown in FIG. **4** with a stack assembly such as shown in FIG. **2**. FIG. **14** provides results of the testing for this proof of concept prototype that was constructed and tested at 104° F. and 93 grains/lb inlet air. The prototype was tested with and without desiccant flow, but with membranes provided to define liquid desiccant flow channels. Without the desiccant flow, the indirect evaporative cooler had a wet-bulb effectiveness of 73%. When desiccant was turned on (with 41% LiCl solution as the desiccant), the effectiveness was 63% and had 12 grains/lb of dehumidification. This resulted in a sensible heat ratio of 0.73. The prototype did not reach model expectations as explained above, and this was likely due to prototype defects creating non-uniform air, water, and desiccant flow distribution.

[0068] It was recognized that use of the membrane to contain the liquid desiccant and separate it from air flow is desirable in most if not all mass transfer/heat exchanger assemblies. For example, with reference to the indirect evaporative cooler **100** of FIG. **1**, the membrane **112** is used to block flow of the liquid desiccant **124** into the inlet air stream **120** while concurrently allowing water molecules **130** to flow from the inlet air stream **120** to the desiccant **124** to dehumidify and cool the inlet or process air **120**.

[0069] However, it was further determined that the second membrane **118** is not needed to practice many aspects of the evaporative coolers described herein. Particularly, an indirect evaporative cooler may be provided in which each stack only includes a single water-permeable membrane (such as membrane **112**) while coolant flow is provided on the opposite side of a separation wall (such as wall **114**) through other techniques such as by providing a flocking sheet or layer (or wicking element) on the separation wall **114** opposite the side of the wall defining the liquid desiccant flow chamber/channel. The stack may be arranged vertically in such embodiments of the evaporative cooler to make use of gravity to encourage coolant flow from the top to the bottom of the stack in the wicking layer. In other cases, though, the wicking layer or flocking may be provided on a top or bottom side of a separation wall (a horizontal stack arrangement) with capillary action (or other mechanisms) used to obtain a desired coolant flow through the stack.

[0070] FIG. **15** schematically illustrates an indirect evaporative cooler (or mass/heat exchanger) **1500**, which may be used in place of the evaporative cooler **100** shown in FIG. **1**. The cooler **1500** may be thought of as a modification of the cooler **100** with retained components or elements having like reference numerals in FIGS. **1** and **15**. Particularly, the evaporative cooler **1500** is useful for providing concurrent dehumidifying and cooling of a process or inlet air stream **120**. This is achieved with one or more mass/heat transfer stacks **1510**. As shown, the inlet air stream **120** is directed to flow in a chamber or channel defined in part by a sheet or layer of a membrane **112**, which may take the form described above for stack **110**. Liquid desiccant **124** flows in an adjacent chamber or channel on the other side of the membrane **112**. The chamber for the desiccant **124** flow is also defined by a separation wall **114**, which, as described above, is impermeable to fluid flow so as to contain the liquid desiccant **124**. The chamber for air stream **120** is also defined by an opposing membrane (not shown) that is used to contain another flow of liquid desiccant (e.g., a membrane of another stack configured similar to stack **1510**).

[0071] As with cooler **100** of FIG. **1**, the evaporative cooler **1500** is configured for counter-flow of the pre-cooled exhaust air stream **128** (relative to the inlet air stream **120**). In contrast to the cooler **100**, though, the exhaust air stream **128** flows in a channel or chamber defined on one side by a wicking layer or flocking element **1520** and on another side by an upper element of another stack (not shown, but may be another wicking layer or a membrane).

[0072] Significantly, the wicking layer or flocking **1520** is attached to a side of the separation wall **114** and acts to wick or guide flow of a volume of coolant **126** in the stack **1510**. In other words, the second membrane **118** of cooler **100** is removed as it is not needed to define a coolant flow channel/chamber. Instead, the wicking layer **1520** may be thought of as defining a channel or flow path for the coolant **126**, which is shown to be counter to the exhaust air stream **128**. The air in stream **128** is in contact with the wicking layer **1520** and the coolant **126**.

[0073] The coolant **126** may be a flow of water or the like, and heat is transferred from the liquid desiccant **124** to the coolant **126** through the separation wall **114**. The coolant **126** flowing or being wicked by wicking layer **1520** is cooled as heat and mass (e.g., water or other moisture **132**) is transferred to the exhaust air stream **128** directly rather than through a membrane as in cooler **100** of FIG. **1**. Heat transfer is not shown in FIG. **15** but generally heat is flowing through the membrane **112** to the liquid desiccant **124** via water **130** and then through the separation wall **114** from the liquid desiccant **124** to the coolant **126**, and then from the coolant **126** to the exhaust air stream **128**.

[0074] Capillary action may support flow of coolant **126** in wicking layer **1520** when the stack **1510** is arranged in a horizontal configuration, but some embodiments will position the stack **1510** including the separation wall **114** and attached/contacting wicking layer **1520** to be vertical such that gravity facilitated coolant flow **126** from the top to the bottom of the evaporative cooler **1500**. As with the stack **110**, the stack **1510** may be provided in multi-stack assemblies/coolers such as the cooler **210** with the stack **1510** being used to provide, or in place of, stack **230** (and/or other stacks **212**, **240**). In such an arrangement, the flow channel for the exhaust air stream **128** typically would be defined by facing but spaced apart wicking layers **1520** on separation walls **114** (e.g., spaced apart, flocked surfaces of two separation walls).

[0075] A variety of flocking materials may be used to implement the wicking layer **1520** on separation wall **114**. The wicking layer **1520** acts to spread out or disperse the flowing coolant **126**, e.g., to avoid rivulets of flowing coolant, which enhances heat transfer from the wall **114** and also mass/heat transfer to exhaust air stream **128** in the adjacent flow chamber/channel of stack **1510**. The flocking material of the wicking layer **1520** also acts to impede gravity to get a slower flow in vertical configurations. The thickness of the layer **1520** may vary but in some cases may be approximately 0.015 inches thick while other useful implementations may use flocking in the range of 0.005 to 0.05 inches in thickness. Exemplary flocking for the wicking layer **1520** include: (a) knitted nylon fabric; (b) polypropylene woven or non-woven fabric; and (c) adhesive-backed flocking fibers (typically polyester or polypropylene), e.g., the layer **1520** may include fibers standing up along (or arranged transverse to) planar surface of wall **114** and may have lengths of 0.01 to 0.05 inches or more. In some embodiments, the wicking layer **1520** may be provided by one or more fabrics coated with a hydrophilic coating. While in other cases, the layer of wicking material **1520** is created with a hydrophilic coating on a surface of the separation wall **114**.

[0076] While a wide variety of materials may be used in layer **1520**, there are a number of wicking or flocking characteristics that may be desirable for operation of the cooler **1500**. The wicking surface of layer **1520** provides a method or mechanism to evenly spread either desiccant or water (as shown in FIG. **15**) over a surface (e.g., surface or side of wall **114**). The wicking surface impedes the forces of gravity on the flowing liquid to slow the flow rate down to a range of about 5 to 50 inches per minute, with some useful implementations using a flow of about 20 in/min. The flocking also enables low total flow rate of water to be applied. The total flow rate of water or other coolant enables flow rates that are between 1.2 to 4.0 times the evaporation rate of water (or other coolant). Typically, this flow would be set based on water quality that is being used and would be 1.2 to 2 times the evaporation rate. In another embodiment, the flow rate of water may be set higher than in the above examples by use of re-circulating the water. In this case, the water flow rate may typically be 4 times the evaporation rate and could be in the range of 3-10 times the coolant evaporation rate.

[0077] As shown in FIG. **15**, indirect evaporative cooler **1500** provides a channel pair where a first airflow **120** is cooled and dehumidified by water absorption **130** through the vapor permeable membrane **112** to the liquid desiccant **124**. The second airflow (in the second channel of the channel pair provided by stack **1510**) **128** removes heat from the first airflow **120** by the evaporation of water **132**. The water/coolant **132** is contained within a flocked or wicked surface (which provides layer of flocking **1520**) on wall **114** opposite the flow channel for liquid desiccant **124**. The evaporation of water/coolant **132** from the flocked or wicked surface of wall **114** removes heat from the first airflow **120** by heat conduction and convection through the membrane-desiccant-separation wall assembly or stack **1510**.

[0078] Generally, the cooling process or method provided by operation of an evaporative cooler (such as cooler **1500**) involves receiving an input or process air stream. This process air stream undergoes dehumidification in a first section or portion of the evaporative cooler (i.e., the desiccant-contained dehumidification section), and this is followed by sensible cooling in a second section (i.e., indirect evaporative cooling section). As shown herein, though, dehumidification and sensible cooling may occur in a single or integral section or portion of the cooler to occur concurrently. The process air is then delivered to a work space or indoor area for use in cooling a space while the purge/exhaust air is used to remove heat from the coolant and is output/discharged from the cooler.

[0079] In some cases, it may be desirable for an indirect evaporative cooler to be provided with a humidification section. This would allow the above cooling method/process to be modified to include a step after sensible cooling in which the process air is humidified adiabatically to further drop the temperature of the air prior to output from the indirect evaporative cooler into a work space or building space. In some embodiments, humidification is provided by having the sensibly cooled air flowing in channels/chambers with one or both sidewalls defined by vapor permeable membranes. Particularly, the indirect section (indirect evaporative cooler) may be followed by a section that provides direct evaporative cooling, which also humidifies. This acts to further reduce the temperature of the outlet stream to provide higher sensible cooling, but such higher cooling comes at the expense of providing less latent cooling (dehumidification). Such additional cooling is shown with line **2025** in the psychrometric chart **2200** of FIG. **22**, where the air is moved from an air state "2" to an air state "2.5" (with this chart **2000** explained in more detail in the following description). The particular methods or mechanisms used to provide direct evaporative cooling may be performed in many ways to practice such a cooler.

[0080] In other cases, though, a flocked surface may be used in the humidification section. For example, FIG. **16** illustrates a humidification section (or portion of such a stack/assembly) **1600** in which a separator or separation wall **1610** is provided to define sidewalls of two adjacent flow channels for process air **1614** (i.e., air that has been sensibly cooled in an upstream section of an evaporative cooler). Both sides of the wall **1610** have been covered with flocking or wicking material to provide a top wicking layer/element **1620** and a bottom wicking layer/element **1622** that are wetted (such as with water) to provide a moisture source or coolant **1630**, **1632** for humidification as the air **1614** flows over the wetted surfaces of layer **1620** and to provide a heat/mass transfer to exhaust air stream **1640** (but the bottom flocking surface/layer **1622** may be omitted in some embodiments). Note, the air streams **1614** and **1640** (and **1710**, **1740** below) may both be supply air.

[0081] FIG. **17** illustrates another humidification section **1700** that may be used in an indirect evaporative cooler (downstream from the sensible cooling section). In the humidification section **1700**, a sensibly cooled air stream **1710** flows over a vapor permeable membrane **1714** separating the air flow **1710** (or the channel it flows within) from a flow of water or the like **1716** (or the channel in which it flows). The water flow channel or humidification source is defined on the other side by a first side/surface of a separation wall **1720**. The other side of the separation wall **1720** along with a vapor permeable membrane **1730** defines a channel or chamber for flow of a coolant **1724**. The humidification section **1700** further includes another or second channel or chamber in which exhaust air **1740** flows along the other side of the vapor permeable membrane **1730** and to remove heat from the evaporative cooler containing humidification section **1700**.

[0082] At this point, it may be useful to describe a two-stage indirect evaporative cooler **1800** with reference to FIGS. **18**-**20**. These figures show graphically how the cooler **1800** works on three levels: (1) FIG. **18** illustrates a heat exchange schematic showing general air, water, and desiccant flows; (2) FIG. **19** illustrates a channel pair graphic or schematic that shows an air channel pair and location of membranes and wicked water surfaces in the first stage and in the second stage heat and mass exchangers; and (3) FIG. **20** provides a psychrometric chart **2000** showing each air state in the cooler **1800**.

[0083] FIG. **18** schematically illustrates a two-stage indirect evaporative cooler **1800** and its air flow pattern during operation. Air states are numbered in FIG. **18**, and these air state numbers are repeated in FIGS. **19** and **20** (as are reference numerals to components shown in both FIGS. **18** and **19**). In this discussion, air streams may be referred to or described as moving from one state to the next such as air stream "1" to "1.5" is the stream of air moving from a first air state to a second air state as the air is dehumidified.

[0084] The cooler **1800** is configured in two distinct stages or assemblies **1810** and **1850** providing a first-stage dehumidifier and a second-stage indirect evaporative cooler. As shown, the dehumidifier **1810** is made up of a number of stacks **1814** (as discussed above and shown in FIG. **19**). Each stack **1814** defines a flow channel or chamber for inlet or process air **1820** to flow through the dehumidifier **1810** and be output to the second stage **1850** as dehumidified air **1822**. The stacks **1814** also define flow paths for and act to contain liquid desiccant **1816** in the dehumidifier **1810** (e.g., LiCl, CaCl or the like at 35 to 40 percent by weight at a flow rate of about 0.34 gallons/minute per space cooling ton). Further each stack **1814** defines, with a pair of spaced apart wicking layers or surfaces on separation walls wicking or flowing water/coolant **1818**, flow channels or pathways for exhaust air **1826** (input at air state "3") to flow through the dehumidifier **1810** and remove heat from the liquid desiccant **1816** and be output at **1828** (at air state "4").

[0085] The first-stage dehumidifier **1810** is a cross-flow heat and mass exchanger between two air streams **1820/1822** and **1826/1828**. Desiccant **1816** and water **1818** flow vertically and are gravity driven. The liquid desiccant **1816** is contained by a polypropylene microporous membrane or other vapor permeable membrane (e.g., a Z-series from Celgard LLC or another distributor/manufacturer). In some implementations of cooler **1800**, nozzles may be used to spray a high water flow rate (water **1818**) that creates a two-phase flow of water and outdoor air in air stream **1826/1828** (air states "3" to "4"). The dehumidifier **1810** may be designed to provide a low water flow rate that is spread by wicked surfaces in contact with the air stream **1826/1828**. In some embodiments, a waterside membrane may be used for controlling biological growth because it creates a barrier that blocks organisms from implanting or growing onto wet surfaces.

[0086] The second-stage or indirect evaporative cooler **1850** is formed with an assembly or number of stacks **1854** (as shown in FIG. **19**). Each stack **1854** defines a flow path or channel for dehumidified air **1822** to flow through the evaporative cooler **1850** to be output as cooled/dehumidified supply air **1860**. Further, the stacks **1854** and/or manifolds or other portions of evaporative cooler **1850** define flow paths/channels for a portion of the supply air **1862** to be returned to flow through the cooler **1850** and be exhausted at **1866** after removing heat from the air stream **1822/1860**. Further, the stacks **1854** provide flow paths or channels for coolant (e.g., water) **1858** such as via gravity flow in wicking layers on separation walls. The second stage **1850** is designed as a counterflow indirect evaporative cooler. In testing of some embodiments, the stage **1850** has a wet bulb effectiveness measured at 120 to 128 percent at the design mass flow rate. For both stages **1810**, **1850** the water **1818**, **1858** was gravity driven and provided at a low flow rate distributed across the heat transfer surfaces of stacks **1814**, **1854** by a wicking material or thickness of flocking.

[0087] Top views of exemplary implementations of the stacks **1814** and **1854** of the stages **1810**, **1850** are shown in FIG. **19** (with repeated components and flows labeled with like reference numbers). As shown, the first stage stack **1814** provides a pair of air flow channels: a first channel/chamber for ventilation or input air **1820** (that typically includes a volume of return air **1821** from the cooled space) and a second channel/chamber for exhaust air **1826** flowing into the page (cross flow in this example). The first channel is defined by a first wall assembly formed of a separation wall **1960** (e.g., a plastic or metal sheet) and a vapor permeable membrane **1962**, which faces the air stream **1820**, **1821**. A flow of liquid desiccant **1816** is contained within the wall assembly provided by separator **1960** and membrane **1962**. The first channel is further defined by a second wall assembly formed of a separation wall **1966** and another vapor permeable membrane **1964**. Again, the membrane **1964** faces or is exposed to the air stream **1820**, **1821** and a flow of liquid desiccant **1816** is provided and contained between a side/surface of separation wall **1966** and the membrane **1964**.

[0088] The second or paired air flow channel of first stage stack **1814** is defined by the other/opposite side of the separation wall **1966** upon which a wicking layer **1970** is provided. The wicking layer **1970** wicks coolant/water that is directly in contact with flowing exhaust air to allow heat to be released from liquid desiccant **1816** and air stream **1820**, **1821**. The second air flow channel is further defined by another separation wall **1974** (which may be a top wall of a next stack), and another wicking layer **1972** of flocking or wicking material is provided on the surface/side of the separation wall **1974** facing the wicking layer **1970**. Coolant such as water is wicked or gravity fed through the wicking layer **1972** as the exhaust air flows through the stack **1814**.

[0089] With regard to the second stage stack **1854** of the indirect evaporative cooler **1850**, a flow channel is provided for air stream **1822**. This channel is provided by a side/surface of a separation wall **1980** and a spaced apart second separation wall **1982**. A second flow channel is provided in stack **1854** into which a portion **1862** of the supply air **1860** is returned into the stack **1854** to remove heat and be exhausted at **1866**. A second air flow channel/chamber is defined by the opposite side of separation wall **1982**, which is covered with flocking/wicking material to provide a wicking layer **1984**. Water or coolant is gravity fed through this layer **1984** during use of the stack **1854** in a cooler assembly. The second flow channel for air stream **1862** is further defined by a second wicking layer **1988** provided on a facing side or surface of an additional separation wall **1990**. As discussed throughout, numerous first and second stage stacks **1814**, **1854** would be assembled or stacked upon each other to form a two-stage cooler **1800**.

[0090] FIG. **20** is a psychrometric chart **2000** illustrating the thermodynamics of the cooling processes provided by operation of the cooler **1800**. The return air state is shown at **2060** while the state of the liquid desiccant is provided with line **2050** in the chart **2000**. Line **2010** shows the thermodynamics as the incoming or supply air moves from air state "1" to air state "1.5" (as shown in FIGS. **18** and **19**) and is dehumidified using the liquid desiccant contained in the vapor permeable membranes in the first stage dehumidifier **1810**. Line **2020** illustrates thermodynamics of the dehumidified air as it passes through the second stage indirect evaporative cooler **1850** and moves from air state "1.5" to air state "2" and is subject to sensible cooling. Line **2030** shows the thermodynamic properties of the return air **1862** that is passed back through the second-stage cooler **1850** and is then output as purge or exhaust air **1866**. Line **2040** illustrates thermodynamic properties of exhaust air stream **1826** to **1828** (e.g., outside air) as it passes through the first-stage dehumidifier **1810**. As shown in the chart **2000**, the air to be supplied to a building space was dehumidified and was also reduced from an original temperature between 80 and 85° F. to about 60° F., which is useful for cooling many residential and commercial spaces.

[0091] The cooler **1800** may be modified by adding a direct evaporative section or stage as shown in FIG. **21**. In the cooler of FIG. **21**, the cooled supply air **1860** from the second stage **1854** is output to the direct evaporative stage **1999** where the air **1860** undergoes humidification and further cooling before being discharged at **1997** in air state "2.5." As shown with the psychrometric chart **2200** of FIG. **22**, direct evaporative cooling may be provided as shown with line **2035** to further reduce the temperature of the outlet air stream (but, as shown, the air stream is also humidified) as the air moves from air state "2" to air state "2.5." The direct evaporative stage may be integrated into the second stage device **1854** or provided as a separate device (e.g., with reference to FIG. **18**, a separate heat and mass exchanger in the cooler **1800** downstream of evaporative cooler **1850** or be integrated into evaporative cooler **1850**). As shown in FIGS. **20** and **22**, the air stream is cooled at least 15° F. such as to a temperature below 60° F. (e.g., an outlet air stream is produced with a temperature between 50 and 60° F. or the like). Also, as shown, dehumidifying is achieved with a relatively small increase in air stream temperature, e.g., an increase of less than about 5° F.

[0092] As shown, the supply air **1860** flows in channels defined by separation walls **1991**, **1993**, and **1995** with wicking material or flocked surfaces **1992**, **1994** facing into each channel. In this way, water may be caused to flow next to the air **1860** to provide humidification to the output supply air **1997** (cooled and humidified to air state "2.5" as shown in FIG. **22**). Air **1997** is colder than air **1860** from the second stage **1854**, therefore less energy is required to provide a desired level of cooling.

[0093] The cooler **1800** (FIG. **18**) may be assembled and implemented in a variety of ways to practice the cooling methods and techniques described herein, but it may be useful to describe one tested assembly or cooler. In the first-stage, flutes were created by extrusion to form the coolant airstream **1826** to **1828** (state "3" to state "4"). Water **1818** was distributed via flow nozzles at the top of the dehumidifier **1810** (e.g., in the airstream **1826** plenum) and mixed with airstream **1826** to **1828**, which ran vertically downward. Some water evaporated as it traveled through the dehumidifier **1810**, but most was collected at the bottom of the airstream **1828** plenum. Louvers in this plenum were used to separate the water droplets from the airstream. Because this design did not have a mechanism to hold up the water internal to the flutes (e.g., wicked surfaces), this configuration used a water flow rate that was significantly higher than the water evaporation rate. Thus, a water reservoir and pump were used to return the water from the collection sump to the top flow nozzles.

[0094] The unbacked vapor permeable membrane was welded to the flutes/extrusions. A liquid manifold distributed desiccant to the space between the membrane and the flutes/extrusions. Air gaps on airstream **1820** to **1822** (air state "1" to air state "1.5") were maintained by strips of spacers with the extruded flutes oriented parallel to the airflow. The design also incorporated spacers that mixed the airstream to enhance heat and mass transfer. Flutes were used to form the channels for airstream **1822** to **1860** (air state "1.5" to air state "2"). A nylon wick was applied to the outer walls of the separation wall/plastic sheets. These subassemblies were then stacked with spacers between each to form the channels for air flow **1862** to **1866** (air state "2" to air state "5"). A low flow of water **1858** was distributed into the second-stage channels from the top. The nylon wick had sufficient water upkeep to allow this flow rate to be marginally above the water evaporation rate. Thus, a solenoid valve controlling domestic cold water may be used to distribute water. Purge water was collected at the bottom of the plenum of air stream **1866**, at which point it was directed to a drain.

[0095] Wicked surfaces provide a number of advantages for the indirect evaporative coolers described herein. The wicking ensures that the walls are fully wetted and that there is no lost evaporation area. The water feed rate can be held to a factor of 1.25 to 2 times that of the evaporation rate. This technique allows for "once-through" water use. The water that drains off the heat and mass exchanger is concentrated with minerals and can then be drained away. A sump and pumping system are not required, which improves energy performance and eliminates sump-borne biological growth. A simple controller can periodically use fresh (low concentration) water to rinse the heat and mass exchanger (such as cooler **1800**) and clear any built-up minerals. Air streams **1822** to **1860** and **1862** to **1866** are in counterflow in the second-stage **1850**. A sensitivity analysis showed that the cooling effectiveness could be reduced by as much as 20 percent if proper counterflow was not achieved. Air stream **1822** to **1860** flowed straight, through extruded flutes, but airstream **1862** to **1866** used a 90-degree turn before exiting the second stage **1850**. Computational fluid dynamics software may be used design an air restrictor to ensure proper counterflow of air stream **1862** to **1866**.

[0096] Likewise, the stacks including the membranes and wicking material may be formed in a variety of ways to implement a mass and heat exchanger of the present description (such as cooler **1800**). The construction of one prototype revolved around laminated layers of polyethylene terephthalate (PET) plastic that were adhered with layers of acrylic pressure-sensitive adhesive. Although this assembly method may not easily be scaled to high-volume manufacturing, the achievable geometries are nearly ideal and, therefore, appropriate for prototypes. This enabled the creation of a prototype with parallel plate geometry that included airside turbulators to enhance heat and mass transfer on airstreams. Another prototype was built using layers of extruded polypropylene (PP). It is likely that formed aluminum sheets may be used to create a parallel plate structure to implement a cooler described herein. For example, the aluminum sheets may be corrugated to form a wavy flow channel, which would increase heat transfer by the waviness of the channel (which promotes mixing of the air stream and impingement of the air into the separator plate wall) and also act to reinforce the structure by giving the sheets/plates increased rigidity. Such an arrangement may work better in the second stage where there is no desiccant (since the desiccant may corrode the aluminum).

[0097] For the first-stage **1810**, the laminated layers enabled the use of wicked surfaces in the air stream **1826** to **1828** channels. For the spacer, an off-the-shelf expanded aluminum grating was used, and the spacer was used in channels for air stream **1820** to **1822** and air stream **1826** to **1828**. The design of the stacks such as stack **1814** used expanded polypropylene hydrophobic membrane backed with a non-woven polypropylene fabric to add strength. The backing reduces vapor diffusion through the membrane but increases tear resistance. The backing was oriented to the airside gap, where tears can originate from abrasion by foreign objects or the aluminum spacer. A desiccant manifold was developed that used laminated layers of plastic and adhesive to effectively and evenly distribute liquid desiccant behind the membrane. The second stage **1850** used laminated construction but, with minimal spacers to create laminar flow, used parallel plate air channels. The design used strips as airflow spacers and wicked surfaces on the wet side of the heat and mass exchanger **1800**.

[0098] The above description concentrated or stressed designs of heat/mass transfer assemblies for use in providing unique indirect evaporative coolers. Those skilled in the art will recognize that the coolers described can readily be included in more complete HVAC systems for residential and commercial use. Such HVAC systems would include plumbing and components to circulate liquid desiccant to and from the cooler at desirable and controllable flow rates. These systems would also include a regenerator for the desiccant (e.g., one that heats the liquid desiccant to remove absorbed water such as heat provided by solar panels, electrical heaters, or the like). The regenerator also may include a sump and lines for recovering potable water from the desiccant, and storage would be provided for the desiccant prior to it being pumped or fed to the cooler. Portions of the system that come into contact with the desiccant typically would be fabricated of corrosion resistant materials such as certain metals or, more typically, plastics. The HVAC system would also include ducting and other components such as fans or blowers for moving the return air from the building through the cooler and back to the cooled spaces, for moving make up air through the cooler and into the cooled spaces, and for discharging any purge or exhaust air. A coolant supply system with piping and pumps/valving (as necessary) would also provide coolant such as potable water to the cooler stacks (e.g., channels between membranes and separation walls).

[0099] The embodiments shown typically discussed ongoing use of the liquid desiccant to dehumidify the supply or process air. However, in many operating conditions, the cooler may be operated without desiccant flow, and these operating conditions may be considered "free evaporative cooling" conditions (or zones on a psychrometric chart). "Free cooling" is exemplified by cooling efficiency so high that the cost of energy to run the system is of no consequence. For example, cooling without drying/dehumidifying may be performed by coolers described herein when the humidity ratio is below about 80 (and the dry bulb temperatures are above 60° F.) while cooling and drying may be required above this humidity ratio at which point the cooler can be operated with flowing liquid desiccant. Such "free" cooling is practical for relatively large numbers of days in less humid areas of the world (such as the southwest portion of the United States).

[0100] Embodiments of an indirect evaporative cooler according to the above description and attached figures can be provided as a single unit that provides an integral heat and a mass transfer device utilizing a number of separation walls. The transfer device or assembly uses membrane containment and air flows do not come in direct contact with desiccant or water (coolant). The coolers use evaporative cooling to drive heat and mass exchange, with heat being transferred through the separation walls between the liquid desiccant and the coolant. The heat exchange is between two counter and/or cross flowing air streams. The mass exchange, such as during dehumidification, is generally the transfer of water vapor from the inlet supply air or process air through a water molecule-permeable membrane to a liquid state (e.g., to absorption by the liquid desiccant). The evaporative section of the coolers drives heat through the separation wall and expels that heat by evaporation from the coolant/water to an air stream.

[0101] It should also be kept in mind that the first and second stages (dehumidifier and indirect evaporative cooler) may be provided in a single system or machine and be packaged to be within a single housing or two or more housings within a single system. In some cases, the first, second, and third (when included) stages are provided in a single system with one, two, or more housings or machines. In other cases, though, the first and second stages are not packaged into the same machine but are configured to cool the same building space. The third stage if present may be provided with the second stage or in a separate machine/housing. Particularly, the dehumidifier can be packaged into a machine and take a mixture of indoor and outdoor air, dehumidify that air, then deliver it to the space. A second machine may pull air from the space (and maybe some outdoor air) and send it through the indirect evaporative cooler as described, then deliver colder supply air to the space. The second machine could exhaust some air so the dehumidifier machine would supply some make-up air from the outside. Essentially, the same process takes place as shown for single systems and/or machines but in two separate machines, one for each heat and mass exchanger.

[0102] In some embodiments, a method has been described for conditioning a supply air stream. The method includes the step of dehumidifying the supply air stream to provide a dehumidified air stream. Such dehumidifying includes directing the supply air stream through a first channel defined by a surface of a vapor permeable membrane containing liquid desiccant. The method further includes transferring heat from the liquid desiccant to a layer of coolant flowing in a second channel adjacent to the liquid desiccant.

[0103] In some implementations of the method, the layer of coolant comprises coolant flowing in a layer of wicking material positioned for heat transfer from the liquid desiccant. Then, the method may further include evaporating a portion of the coolant flowing in the layer of wicking material into an air stream flowing adjacent to the layer of wicking material. It may be useful in the method for the coolant in the layer to be flowing at a flow rate of less than about 50 inches per minute. In some cases, a temperature of the supply air stream is increased during the dehumidifying less than about 5° F. In the same or other cases, the dewpoint temperature of the supply air stream is decreased during the dehumidifying to less than about 55° F.

[0104] In some embodiments of the indirect evaporative cooler or supply air conditioning apparatus, a dehumidifier is included in a first stage, and the dehumidifier performs cooling of an air stream input to an inlet of the dehumidifier by absorption of water vapor from the input air stream. Such cooling may be performed using a membrane-contained liquid desiccant that is liquid cooled. The phrase "liquid cooled" may take a number of meanings including, but not limited to, flowing a liquid coolant such as water adjacent the channel containing the liquid desiccant, and the liquid coolant flows such that the temperature of the liquid is raised as it passes through the dehumidifier to carry away heat from the liquid desiccant.

[0105] The apparatus may also include a second stage comprising an indirect evaporative cooler in fluid communication with an outlet of the first stage to receive a cooled portion of the input air stream. The indirect evaporative cooler may be operable to sensibly cool the received and cooled portion of the input air stream to a temperature (or to be within a temperature range) less than the wet bulb temperature and greater than the dew point temperature of the input air stream. During operation of the apparatus, a first portion of the sensibly cooled air stream is supplied to a cooled space and a second portion of the sensibly cooled air stream is directed to a wet side of the indirect evaporative cooler and receives heat from the received and cooled portion of the input air stream.

[0106] In some implementations of such an apparatus, the dehumidifier performs the cooling using a membrane-contained liquid desiccant cooled by an indirect evaporative channel that includes a wicking layer containing a flow of coolant. In other implementations, the dehumidifier performs the cooling using a membrane-contained liquid desiccant that is liquid cooled. The apparatus may be operated such that the second portion comprises 10 to 40 percent of the received and cooled portion of the input air. It may be useful for a flow channel for the second portion of the sensibly cooled air stream in the indirect evaporative cooler to be defined in part by a surface of a separation wall. Then, a layer of wicking material may be provided on the surface acting to wick a stream of liquid coolant adjacent the flow channel.

[0107] In some settings, the apparatus includes a direct evaporative stage between the second stage and the cooled space. Then, during operations, the direct evaporative stage receives the first portion of the sensibly cooled air stream and provides additional cooling via direct evaporative cooling prior to supplying the first portion of the sensibly cooled air stream to the cooled space. Further, the direct evaporative stage may be adapted to contain flow of water with a vapor permeable membrane or within a layer of wicking material (e.g., the layer of wicking material is provided as a surface with a hydrophilic coating).

[0108] While a number of exemplary aspects and embodiments have been discussed above, those of skill in the art will recognize certain modifications, permutations, additions, and sub-combinations thereof. It is therefore intended that the following appended claims and claims hereafter introduced are interpreted to include modifications, permutations, additions, and sub-combinations to the exemplary aspects and embodiments discussed above as are within their true spirit and scope.

1. An apparatus for dehumidifying air supplied to a space, comprising:

- a first flow channel for receiving a stream of inlet supply air, wherein the first flow channel is defined in part by a vapor permeable membrane;
- a second flow channel adjacent to the first flow channel receiving a stream of liquid desiccant; and
- a third flow channel adjacent to the second flow channel receiving a stream of exhaust air, whereby heat is transferred from the stream of inlet supply air to the stream of exhaust air in the third flow channel via the stream of liquid desiccant,
- wherein the third flow channel is defined in part by a first surface of a separation wall, wherein the separation wall is spaced apart a distance from the vapor permeable membrane, and wherein a layer of wicking material is provided on the first surface acting to wick a stream of liquid coolant.

2. The apparatus of claim **1**, wherein the second flow channel is defined by the vapor permeable membrane and a second surface of a separation wall, opposite the first surface, that is impermeable to fluid.

3. The apparatus of claim **1**, wherein the stream of liquid coolant flow through the wicking material layer is gravity driven.

4. The apparatus of claim **1**, wherein the wicking material is knitted nylon fabric, polypropylene woven fabric, polypropylene non-woven fabric, adhesive-backed flocking fibers, or one or more fabrics coated with a hydrophilic coating and wherein the liquid desiccant comprises a salt solution and the liquid coolant comprises water.

5. The apparatus of claim **1**, wherein the layer of wicking material comprises a hydrophilic coating on the first surface of the separation wall.

6. The apparatus of claim **1**, wherein the exhaust air comprises a portion of the stream of inlet supply air exiting the first flow channel at about the second, lower enthalpy.

7. The apparatus of claim **1**, wherein the stream of inlet supply air flows in a first direction in the first flow channel and the stream of exhaust air flows in a second direction in the third flow channel, the second direction being in at least one of cross or counter to the first direction, and wherein the third flow channel is arranged such that the exhaust air stream flows in an at least a partially cross or counter flow direction relative to the stream of inlet supply air in the first flow channel.

8. A method of conditioning a supply air stream, comprising:

dehumidifying the supply air stream to provide a dehumidified air stream, wherein the dehumidifying includes directing the supply air stream through a first channel defined by a surface of a vapor permeable membrane containing liquid desiccant; and

transferring heat from the liquid desiccant to a layer of coolant flowing in a second channel adjacent to the liquid desiccant.

9. The method of claim **8**, wherein the layer of coolant comprises coolant flowing in a layer of wicking material positioned for heat transfer from the liquid desiccant and wherein the method further includes evaporating a portion of the coolant flowing in the layer of wicking material into an air stream flowing adjacent to the layer of wicking material.

10. The method of claim **9**, wherein the coolant in the layer is flowing at a flow rate of less than about 50 inches per minute.

11. The method of claim **8**, wherein a temperature of the supply air stream is increased during the dehumidifying less than about 5° F.

12. The method of claim **8**, wherein the dewpoint temperature of the supply air stream is decreased during the dehumidifying to less than about 55° F.

13. An apparatus for conditioning an inlet air stream, comprising:

a first stage comprising a dehumidifier operable for cooling an air stream input to an inlet of the dehumidifier by absorption of water vapor from the input air stream; and

a second stage comprising an indirect evaporative cooler in fluid communication with an outlet of the first stage to receive a cooled portion of the input air stream, wherein the indirect evaporative cooler is operable to sensibly cool the received and cooled portion of the input air stream to a temperature range less than the wet bulb temperature and greater than the dew point temperature of the input air stream;

wherein a first portion of the sensibly cooled air stream is supplied to a cooled space, and

wherein a second portion of the sensibly cooled air stream is directed to a wet side of the indirect evaporative cooler and receives heat from the received and cooled portion of the input air stream.

14. The apparatus of claim **13**, wherein the dehumidifier performs the cooling using a membrane-contained liquid desiccant cooled by an indirect evaporative channel that includes a wicking layer containing a flow of coolant.

15. The apparatus of claim **13**, wherein the dehumidifier performs the cooling using a membrane-contained liquid desiccant that is liquid cooled.

16. The apparatus of claim **13**, wherein the second portion comprises 10 to 40 percent of the received and cooled portion of the input air.

17. The apparatus of claim **13**, wherein a flow channel for the second portion of the sensibly cooled air stream in the indirect evaporative cooler is defined in part by a surface of a separation wall and wherein a layer of wicking material is provided on the surface acting to wick a stream of liquid coolant adjacent the flow channel.

18. The apparatus of claim **13**, further comprising a direct evaporative stage between the second stage and the cooled space, wherein the direct evaporative stage receives the first portion of the sensibly cooled air stream and provides additional cooling via direct evaporative cooling prior to supplying the first portion of the sensibly cooled air stream to the cooled space.

19. The apparatus of claim **18**, wherein the direct evaporative stage is adapted to contain flow of water with a vapor permeable membrane or within a layer of wicking material.

20. The apparatus of claim **19**, wherein the layer of wicking material is provided as a surface with a hydrophilic coating.

21. A method for conditioning a supply air stream, comprising:

cooling an air stream by absorption of water vapor from the air stream to a first wet bulb temperature and first dewpoint temperature;

after the cooling by absorption, using indirect evaporative cooling to sensibly cool the air stream to a second temperature; and

supplying a first portion of the sensibly cooled air stream to a space, wherein a second portion of the sensibly cooled air stream is used to perform the indirect evaporative cooling of the air stream.

22. The method of claim **21**, wherein the second temperature is less than the first wet bulb temperature and greater than the first dewpoint temperature and the cooling is performed using membrane-contained liquid desiccant or with a liquid coolant flowing on a separation wall covered with a layer of wicking material or a hydrophilic coating.

23. The method of claim **21**, further comprising, prior to the supplying the first portion to the space, humidifying the sensibly cooled air stream by providing a flow of water adjacent to the sensibly cooled air stream, wherein the flow of water with a vapor permeable membrane or within a layer of wicking material.

24. The method of claim **21**, further comprising, prior to the supplying the first portion to the space, providing additional cooling to the sensibly cooled air stream via direct evaporative cooling.

* * * * *