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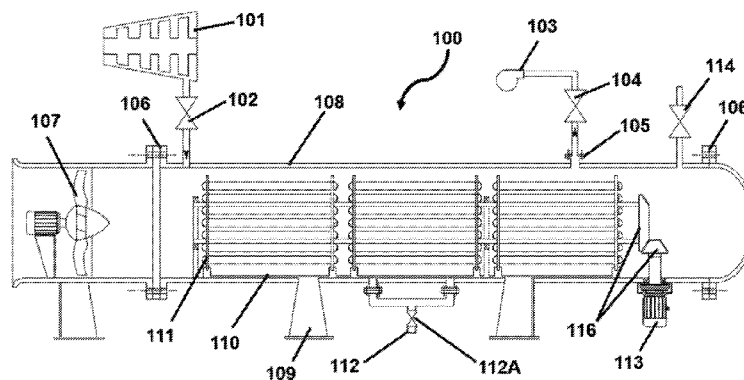


Fig. 1

(57) Abstract: In various embodiments devices and methods are provided for an improved dry-cooling condensation system. In certain embodiments the methods involve receiving steam from a source of steam (e.g., a power plant); condensing the steam into water while transferring the latent heat of the steam into the latent heat of a thermal storage material; and dissipating the latent heat from the thermal storage material at a later time when the ambient temperature is lower than the ambient temperature at the time the steam was condensed into water.

**METHOD AND APPARATUS FOR A DELAYED AND PROLONGED AIR
COOLING CONDENSATION SYSTEM**

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application claims priority to and benefit of USSN 61/533,551, filed
5 September 12, 2011, which is incorporated herein by reference in its entirety for all
purposes.

STATEMENT OF GOVERNMENTAL SUPPORT

[Not Applicable]

BACKGROUND

10 [0002] In electrical power plants, large amounts of latent heat carried by low
temperature steam from the steam turbine exhausts needs to be removed and condensed into
water to complete the Rankine cycle for electricity generation. Similar situations exist in
other heat engines and appliances, such as central air-conditioning systems. Typically, the
low quality (temperature) latent heat was removed by the use of a large amount of cooling
15 water, which is a method used by the majority of coal fired power and nuclear power plants.
However, this so called “wet-cooling” method consumes extremely large amounts of
water, accounting for almost 50% of total industrial water usage and 20% of total water
consumption.

[0003] For water resource scarce areas, “dry-cooling” methods were developed to
20 directly dissipate the latent heat into ambient air without using a lot of water. There are two
major types of air-cooling condensation methods, one is direct air-cooling condensation and
the other is indirect air-cooling condensation. These two methods are distinguished from
each other by whether the heat transfer process occurs between low temperature steam and
ambient air directly or through other heat transfer media.

25 [0004] The heat transfer efficiency of air-cooling condensation strongly depends on
the temperature difference between the exhaust steam and the environment ambient air, and
the speed of the air that blows onto the condensation heat exchanger surfaces. To dissipate
the latent heat of exhausted steam instantaneously requires a larger temperature difference
between the exhausted steam and the ambient air, typically as much as 30°C to 40°C in
30 “dry-cooling” methods, and 10°C to 15°C in “wet-cooling” methods. Moreover, since the
volume heat capacity of the air is almost a thousand times smaller than that of the water, an

extremely large amount of cooling air is blown upon the heat exchanger surfaces for the cooling. The large temperature difference between the exhaust steam and the ambient air results in higher exhaust steam pressure. For example, in a typical dry-cooling system, the exhaust steam pressure is typically about 25 to 50 kPa instead of 5 to 7 kPa for a typical
5 wet-cooling system. This exhaust steam pressure increase results in a drop in heat to electricity conversion efficiency for the steam turbine generator. For example, the heat to electricity conversion efficiency would decrease from 40% to 35% when “dry-cooling” is used instead of a “wet-cooling” for a typical 600 MW coal-fire power generator.

[0005] On the other hand, during a “dry-cooling” process, the heat dissipation rate
10 depends on the surface area of the condenser’s heat exchanger and the wind blowing speed that passes through the heat exchanger surfaces. Due to cost considerations, the heat exchanger’s surfaces cannot be too large otherwise the wind speed has to be very large to achieve effective cooling. According to the aerodynamics, the power consumption of a wind-blowing machine used for dry-cooling varies as the 3rd power of the wind speed it
15 generates. The higher the wind speed it requires, the more electricity it consumes. Therefore, the dry-cooling approach may consume a few percent up to more than 10 percent of the electricity that the same power plant generates.

SUMMARY

[0006] To overcome above mentioned shortcomings of an air-cooling condensation
20 process, it was discovered that it is beneficial to use an indirect air-cooling approach while dividing the cooling process into two time periods or "cycles": 1) A condensation period where the low temperature exhaust steam transfers its latent heat to a low temperature thermal storage material while being condensed into water, *e.g.*, during a time period when ambient air temperature is relatively high; and 2) A dissipation period where the heat is
25 dissipated from the thermal storage material to the ambient air by air-cooling, *e.g.*, during a time period when ambient air temperature is relatively low. If these two stages are separated in time it is possible to reduce the shortcomings of the above mentioned dry-cooling methods. This two stage (time period) method can also be applied partially, *e.g.*, in certain embodiments, during the high temperature period, only a part of the generated latent
30 heat is stored in thermal storage materials, and dissipated into ambient air during the cool time period; while other part is still dissipated using conventional cooling methods. This is especially useful to modify and improve existing power plants.

[0007] It is even more beneficial for a concentration solar thermal power (CSP) plant to use a dry-cooling method because in most cases, CSP power plants are constructed in desert areas where water resources are very limited. In this case, a separated two-stage dry-cooling can bring even more advantages compared to the traditional dry-cooling methods because the temperature differences between day and night are normally larger in desert areas than in other locations.

[0008] The apparatus and methods described herein have been developed in view of the above points. Accordingly, in various embodiments, methods and apparatus are described that provide delayed and prolonged dry-cooling condensation methods, that take the advantage of different ambient air temperatures during the day and night to lower the energy consumption for cooling (*e.g.*, for a wind blower) while decreasing the condensation temperature of the exhaust steam so that the overall thermal to electricity conversion efficiency can be improved for dry-cooling.

[0009] Let the starting time of the first condensation stage to be t_1 , and the duration of the stage 1 to be Δt_1 ; the starting time of the second heat dissipation stage from the thermal storage media to be t_2 , and the duration of stage 2 to be Δt_2 . If t_1 equals t_2 and Δt_1 equals to Δt_2 , the entire cooling process becomes a normal indirect air-cooling condensation process. For a delayed and prolonged indirect air-cooling condensation, t_2 (the starting time of the dissipation stage) is typically later than t_1 (*e.g.*, at a cooler time of a given day), and in certain embodiments, Δt_2 is not equal to Δt_1 . In this way, effectiveness of the condensation process can be optimized independently from the subsequent heat dissipation process, thereby improving the effectiveness of the air-cooling process.

[0010] In certain embodiments by taking advantage of the fact that the ambient air temperature during the night is lower than that during the day, one can use low temperature thermal storage materials to store the latent heat of exhausted steam during the day while condensing the turbine exhaust steam into water, and then dissipate the stored thermal energy into the ambient air during the night when the ambient temperature is lower, using a wind blower to drive the ambient air through the packaged thermal storage material surface to release the latent heat stored therein into the environment. In certain embodiments low temperature phase change materials (PCMs) are used to store the latent heat from the exhaust steam (during Δt_1 time period). The stored thermal energy is then dissipated into the ambient air during the night when the ambient air temperature is lower than temperature during the day via wind blower in a pre-determined time duration Δt_2 .

[0011] In addition, when circumstances require it, one can choose time durations for the condensation stage Δt_1 and the heat dissipation stage Δt_2 so that the heat dissipation rate can be slower than that of a typical indirect dry-cooling system. In certain embodiments one can choose a time duration $\Delta t_2 > \Delta t_1$. Thus, the exhaust steam condensation temperature
5 can be significantly decreased. It should be noted that for every 10°C of higher condensation temperature for a steam turbine generator, the heat to electricity conversion efficiency would be lower by 4.5%. This is a very significant loss of efficiency because normally a “dry-cooling” approach raises the condensation temperature up to 30°C higher than that of a “wet-cooling” approach.

10 [0012] In certain embodiments if the delay and prolonged indirect dry-cooling approach described herein is used, in the stage of condensation for the exhausted steam from the steam turbine or other heat appliances, the low temperature exhausted steam transfers its latent heat to the low temperature thermal storage media via an efficient heat exchanger while condensing the steam into water. The exhaust steam condensation
15 temperature (typically a few degrees higher than the low temperature thermal storage material’s phase change temperature) can be set at only a few degrees higher than the ambient air temperature at night. During the heat dissipation stage, the phase change material would have enough temperature difference relative to the ambient air temperature so that the stored latent heat can be dissipated into the environment, as described in detail
20 below. This is how the new approach described herein can lead to an improvement for the heat to electricity conversion efficiency.

[0013] As described in detail below, the large amount of heat exchange area of our apparatus described herein ensures that the steam can be condensed into water at close to phase change temperature during the condensation stage, and that the stored latent heat can
25 be dissipated into the environment during the night, resulting in a drop of wind speed required for effective cooling. This can reduce the electricity consumption significantly due to the third power relationship between the power needed to drive a wind blower and the wind speed that is created by that blower.

[0014] Accordingly, in various embodiments, device for steam condensation and
30 delayed dissipation of the heat produced by the condensation is provided. In certain embodiments the device comprises a combined condensation/thermal storage chamber, the chamber comprising: one or more valved ports for receiving steam from a steam source; one or more containers containing a thermal storage material; one or more valved ports for

applying a vacuum to the thermal storage chamber; one or more valved ports for introducing ambient pressure air into the chamber; one or more valved ports for removing condensed water from the chamber; and a valve system operably coupling the chamber to a source of ambient temperature air. In this context a valved port indicates that flow of fluid or gas through the port is under the control of one or more valves, however, the valves need not be located at the site of the port and can be remote (*e.g.*, disposed between a vacuum source and a vacuum port, between a steam source and a steam port, and the like). In certain embodiments the one or more containers containing a thermal storage material is a plurality of containers each containing a thermal storage material. In certain embodiments the thermal storage material is a phase change thermal storage material (PCM). In certain embodiments the thermal storage material is a liquid/solid phase change thermal storage material. In certain embodiments the thermal storage material comprises a material selected from the thermal storage materials shown in Table 2 (*e.g.*, $\text{Na}_2\text{CO}_3 \cdot 10\text{H}_2\text{O}$, and the like). In certain embodiments the phase change material contains glass microfibers or nanofibers. In certain embodiments the device further comprises an apparatus to cause mixing of liquid phase thermal storage material in the containers containing the thermal storage material. In certain embodiments the containers (containing the thermal storage material(s)) are attached to a frame structure (*see, e.g.*, Figure 3) and the apparatus comprises a motor configured to cause rotation of the structure and the attached containers. In certain embodiments the one or more valved ports for applying a vacuum to the thermal storage chamber are operably coupled to a vacuum pump. In certain embodiments the one or more valved ports for applying a vacuum to the thermal storage chamber and the one or more valved ports for introducing ambient pressure air into the chamber are controlled by separate valves. In certain embodiments the one or more valved ports for applying a vacuum to the thermal storage chamber and the one or more valved ports for introducing ambient pressure air into the chamber are controlled by the same valve(s). In certain embodiments the valve system operably coupling the chamber to a source of ambient temperature air comprises a butterfly valve (or a flange cover, or other functionally equivalent structure). In certain embodiments the source of ambient air is a fan and/or blower and/or one or more ducts configured to receive ambient wind. In certain embodiments the device is one of a plurality of the devices configured in a parallel configuration. In certain embodiments the one or more valved ports for receiving steam from a steam source are operably coupled to the low temperature steam output from a turbine (*e.g.*, a turbine in a power plant selected from the group consisting of a coal-fired power plant, a gas-fired power plant, a nuclear power plant, and a solar thermal

power plant). In certain embodiments the device is sited in a desert or a non-desert region with limited water availability.

[0015] Also provided are systems for delayed heat dissipation from the condensation of waste steam, said system comprising a plurality of devices for steam
5 condensation and delayed dissipation of the heat produced by the condensation, *e.g.*, as described herein. In certain embodiments the devices are configured in a parallel configuration, *e.g.*, as illustrated in Figure 2. In certain embodiments the system comprises at least 10, or at least 20, or at least 30, or at least 40, or at least 50, or at least 60, or at least 70, or at least 80, or at least 90, or at least 100, or at least 150, or at least 200 of said
10 devices. In certain embodiments the system is operably coupled to the low temperature steam output from a turbine. In certain embodiments the turbine is a turbine in a power plant (*e.g.*, a coal-fired power plant, a gas-fired power plant, a nuclear power plant, a solar thermal power plant, *etc.*). In certain embodiments the system is sited in a desert or a region with limited water availability.

15 [0016] In various embodiments a dry-cooling methods are provided. It will be recognized that the methods, as described herein provide methods of condensing waste steam while delaying heat dissipation from said condensation; and can be used to complete a Rankine cycle at enhanced efficiency. Accordingly, in certain embodiments a dry-cooling condensation method is provided where the method comprises: receiving steam from a
20 source of steam; condensing the steam into water while transferring the latent heat of said steam into the latent heat of a thermal storage material; and dissipating the latent heat from said thermal storage material at a later time when the ambient temperature is lower than the ambient temperature at the time the steam was condensed into water. In certain embodiments the thermal storage material is a phase change thermal storage material
25 (PCM). In certain embodiments the thermal storage material is a liquid/solid phase change thermal storage material. In certain embodiments the thermal storage material comprises a material selected from the thermal storage materials shown in Table 2 (*e.g.*, $\text{Na}_2\text{CO}_3 \cdot 10\text{H}_2\text{O}$, and the like). In certain embodiments the source of steam is the steam output from a turbine. In certain embodiments the turbine is in a power plant selected from the group
30 consisting of a coal-fired power plant, a gas-fired power plant, a nuclear power plant, and a solar thermal power plant. In certain embodiments the receiving and condensing comprises receiving and condensing during daylight hours. In certain embodiments the receiving and condensing comprises receiving and condensing peak temperature hours (*e.g.*, between noon and 3:00 pm). In certain embodiments the dissipating comprises dissipating the latent

heat from the thermal storage material during cooler hours (*e.g.*, the late afternoon, and/or evening, and/or night). In certain embodiments the method is performed using one or more devices for steam condensation and delayed dissipation of the heat produced by the condensation, *e.g.*, as described herein. In certain embodiments the receiving and

5 condensing comprises: opening said one or more valved ports for applying a vacuum to a thermal storage chamber to reduce the ambient pressure in the thermal storage chamber; opening one or more valved ports for receiving steam to introduce steam from a steam source (*e.g.*, a turbine) into the chamber, whereby the steam condenses transferring latent heat of steam into a thermal storage material; and operating one or more valved ports for

10 removing condensed water from the chamber to return the condensed water to the system providing the steam. In certain embodiments the dissipating comprises: restoring the pressure in the thermal storage chamber to atmospheric pressure; operating a valve system operably coupling the chamber to a source of ambient temperature air to pass ambient temperature air through the thermal storage chamber to transfer heat from the thermal

15 storage material to the air. In certain embodiments the passing ambient temperature air comprising operating a fan and/or blower or to force air through the chamber, and/or coupling the chamber to a duct system that channels wind through the chamber. In certain embodiments the dissipating further comprises operating an apparatus to provide mixing of fluid thermal storage material in chambers containing the fluid thermal storage material. In

20 certain embodiments the method comprises operating a motor to rotate a structure frame to which the chambers containing the thermal storage material are attached. In certain embodiments the method is performed using a system comprising a plurality of In certain embodiments the method is performed using one or more devices for steam condensation and delayed dissipation of the heat produced by the condensation, *e.g.*, as described herein.

25 In certain embodiments the devices are configured in a parallel configuration. In certain embodiments substantially all of the devices in said system perform said receiving and condensing at the same time. In certain embodiments substantially all of the devices in said system perform said dissipating at the same time. In certain embodiments some of the devices perform said receiving and condensing at the same time that other devices are

30 dissipating.

BRIEF DESCRIPTION OF THE DRAWINGS

[0017] Figure 1 shows a schematic diagram of the apparatus **100** operably linked to a steam source (*e.g.*, a steam turbine **101**). An efficient condensing heat exchanger is used to convert the latent heat of exhausted steam (*e.g.*, from a power plant) into the latent heat

of a thermal storage medium while the exhausted steam is condensed into water during a “condensation stage”. During a second “heat dissipation stage”, the valves (*e.g.*, butterfly valves, or flange covers) are opened and the stored heat in thermal storage medium is dissipated into the ambient air by an air stream (wind) by a blower or other wind source.

5 The labeled parts are described as follows: steam source (*e.g.*, turbine) **101**; steam exhaust valve **102** that connects the steam turbine and the condensing heat exchanger/thermal storage tank; vacuum pump **103** that is used to pump the air inside the tank out to create a vacuum (reduced pressure) in the tank; a vacuum valve **104** that connects the vacuum pump and the tank; micro pole **105** for the vacuum pathway; valves **106** (*e.g.*, butterfly valves,
10 flange covers, or other functionally comparable structure(s)) that seal the vacuum during the “condensation stage”; wind blower **107** (or other source of an ambient temperature air stream); a thermal storage tank **108**; a supporting base **109** for the thermal storage tank; packaging containers **110** (*e.g.*, pipes) for the thermal storage materials (*e.g.*, phase change materials (PCMs)); supporting structure **111** for the phase change material packaging pipes
15 (*e.g.*, as illustrated in Figure 3) that, in certain embodiments, is supported by an rotates on bearing **117**; condensed water passage way **112** from the tank; shutoff valve **112A** for use during a “heat dissipation stage”; an apparatus (*e.g.*, a motor) **113** that rotates the entire phase change packaging system (*e.g.*, comprising a structure frame **115** attached to the packaging containers containing the thermal storage material), *e.g.*, via a helical drive
20 system **116** to mix the phase change thermal storage materials inside the thermal storage packaging pipe to prevent the possible phase separation inside the pipes; air release valve **114** to vent the air into the tank until the valve **106** can be opened at the beginning of the “heat dissipation stage”.

[0018] Figure 2 schematically illustrates another embodiment of the device, where
25 multiple steam condensation tank/heat dissipation devices (*e.g.*, as illustrated in Figure 1) are configured in a parallel system/structure **200** operably coupled to a steam source **201**. The labeled parts are described as follows: steam source (*e.g.*, turbine) **201**; N number of similar modules **202** as described in Figure 1 (illustrated with N=5); wind path pipes **207** that connect the condensing tanks together to the source of the air stream; the wind blower
30 **204** (or other source of a cooling air stream); the valve **205** (*e.g.*, a butterfly valve, flange cover or other functionally equivalent structure) that allows air from the blower to enter the thermal storage chamber **108** (when operating as a heat dissipation apparatus); valve **206** (*e.g.*, a butterfly valve, flange cover or other functionally equivalent structure) that allows the air to blow out of the heat dissipation apparatus; the parallel wind path pipes **203** to

allow the air stream to flow out of system; valve **208** that connects the steam source (*e.g.*, steam turbine exhaust) with the condensation/heat dissipation system. Vacuum pump **213** connects (*e.g.*, is operably coupled) via vacuum valve **214** and pipe **215** connects with the main exit pipe **203**. Its function is to reduce pressure in the entire parallel condensation/heat
5 dissipation system before the turbine exhaust steam is released into the system for condensation.

[0019] Figure 3 illustrates a structure frame **115** in a cross section view for the packaged thermal storage pipe fixture and arrangement. **110** is the cross section of the packaged thermal storage pipes; **116** is the structure frame to hold the storage pipes in the
10 storage tank. The center distance between packaged pipes can be optimized to allow the air stream to pass through the pipes along the length direction of the storage tank.

DETAILED DESCRIPTION

[0020] In various embodiments devices are described that provide delayed and prolonged dry-cooling condensation methods that take advantage of different ambient air
15 temperatures during the day and night to reduce energy consumption for cooling (*e.g.*, for operating a wind blower in a dry-cooling system) while decreasing the condensation temperature of the exhaust steam so that the thermal to electricity conversion efficiency can be improved for dry-cooling.

[0021] Methods are described in here in detail with reference to devices examples of
20 which are illustrated in the accompanying drawings. The methods and devices are intended to be illustrative and not limiting. Using the teaching provided herein variations of the illustrated devices and methods will be readily available to one of skill in the art.

[0022] Aspects of the innovations, such as those set forth in some of the implementations below, may relate to systems and methods of air-cooling. However, it
25 should be understood that the inventions herein are not limited to any such specific illustrations, but are defined by the scope of the claims and full disclosure.

[0023] As illustrated in Figure 1 a vacuum can be applied to the chamber **108** by running a vacuum pump **103** (or other source of vacuum) after opening of vacuum valve
30 **104** of the thermal storage tank **108** via a micro pole **105**. Low temperature exhaust steam from a steam source **101** (*e.g.*, from a power plant turbine) enters thermal storage tank **108** via opened valve **102**, to release the latent heat of the stem to containers **110** (*e.g.*, packaged pipes) holding thermal storage materials (*e.g.*, phase change thermal storage materials

(PCMs)). This transfer of latent heat from the steam to the thermal storage material is accompanied by condensation of the steam into water. The water drains out from the tank 108 via water path (e.g., optionally valved ports) 112 to complete the Rankin cycle for the power (e.g., electricity) generation. Where the thermal storage material is a phase change material, the phase change material (PCM) thermal storage material inside containers (e.g. pipes) 110 absorbs the latent heat of steam and changes its phase from solid form into liquid form, which means the phase change material (PCM) melts and keeps (stores) the latent heat. This process comprises the “condensation stage” .

[0024] The maximum temperature difference required for the exhaust steam and the phase change temperature of the thermal storage material can be described by the following equation:

$$T_w - T_m = \frac{\rho_{cs} \frac{d_o^2}{4} \Phi_1^2 \Delta t_1}{2 \gamma_c \lambda_{cs} (\rho_{cs} V_c)^2} = \frac{\rho_{cs} \gamma_c d_o^2}{8 \lambda_{cs} \Delta t_1}$$

where

$$V_c = \frac{Q_1}{\rho_c \gamma_c} = \frac{\Delta t_1 \Phi_1}{\rho_c \gamma_c}$$

where T_w is the wall temperature for the packaging pipe, which is also very close to the exhaust steam temperature for the thermal storage material, T_m is the phase change temperature, ρ_{cs} (ρ_c) is the phase change material density in the solid state, γ_c is the heat of fusion of the phase change material (or latent heat during phase change), λ_{cs} is the thermal conducting coefficient for the solid state phase change material, V_c is the total volume; d_o is the diameter for the phase change material’s packaging container (e.g., pipe), Φ_1 is the amount of latent heat for the turbine exhaust steam that needs to be condensed per hour; as defined before, and Δt_1 is the time duration for the “condensation stage”, and Q_1 is the total latent heat condensed during the condensation stage. One can find that if the time duration for the “condensation stage” Δt_1 is larger, the maximum temperature difference is smaller because longer time is used to transfer the steam latent heat to the PCMs.

[0025] The following illustrates the use of this equation for a real world example: Consider small scale CSP power plant with 10 MW_e peak capacity. With the help of the delay and prolonged dry-cooling apparatus described herein, the exhaust steam condensation temperature for the steam turbine can be 35°C or below. In this case, its heat to electricity conversion efficiency would be 35%. Therefore, about 65% of the total

thermal energy produced by a solar thermal field needs to be eventually condensed into ambient air during power generation. Assume such a CSP system would work six hours to produce electricity per day. The amount of latent heat for the turbine exhaust would be 15MW_{th}. To apply the delay and prolonged dry-cooling approach described herein 26
5 individual storage/dissipation units, similar to those illustrated in Figure 1 can be used to complete the task. In one illustrative embodiment, each storage unit comprises a cylindrical container about 5 meters in diameter and 16 meters in length, made from fiber-glass material with a steel enhancement frame structure. Inside each container, 5200 x 3 thin
10 glass tubes (with 25 mm diameter and 0.5 mm wall thickness, 5 meter in length with 3 identical sections to fill the 16 meter long tank, as illustrated in Figure 1) are placed in parallel with the long dimension of the container in a supporting a structure such as the one illustrated in Figure 3, where the frame structure is designed to hold the thin glass tubes in such a way that the distance between glass tube centers is about 62.5 mm. This distance is an optimized value for the storage tank design after considering the density of the storage
15 pipes in the tank and the wind resistance when the wind is blown along the pipes to dissipate the heat from the storage pipe. An illustrative thermal storage material is Na₂CO₃·10H₂O compound, whose phase change temperature is at 32°C with the latent heat (heat of fusion) of 267 kJ/kg. The total amount of thermal storage material that is packaged inside the glass tubes in the container is 50 ton, with total latent heat capacity about 3.6
20 MWh. To store all the turbine exhaust steam latent heat of the CSP plant into the phase change material during the “condensation stage”, which is 6 hours in this case, applying the equation above, the temperature difference of T_w and T_m would be 2.6°C. As mentioned before, 26 such storage units configured in parallel, *e.g.*, as illustrated in Figure 2, would be sufficient to condense all the exhaust steam from the 10MW CSP plant into water during 6
25 hours of “condensation stage”, store all the latent heat into these storage tanks, and wait until in the night time to be dissipated into ambient air.

[0026] During the “heat dissipation stage” , *e.g.*, during the nighttime or cooler daylight periods, when the ambient air temperature is significantly lower than the phase change temperature of the thermal storage materials in the storage tank, a vent valve **114** is
30 opened to release the air into the tank **108** until the pressure inside the tank reaches atmospheric pressure, the two butterfly valves **106** are opened, and a fan (wind blower) **107** is started to drive the ambient cooling-air flow through the thermal storage package pipe surfaces via the spaces between the package pipes **110**. The ambient cooling air-flow carries the latent heat of the phase change material (PCM) from the package pipes **110**

surfaces into the environment. After releasing its latent heat the phase change material (PCM) of the thermal storage medium returns to its solid form again.

[0027] In certain embodiments, in order to avoid the nucleation of the phase change material (PCM) inside its container (*e.g.*, pipe) an agitator or mixer can be provided to facilitate mixing of the PCM inside the container(s). In certain embodiments the agitator/mixer can be fixed at the bottom of the thermal storage tank **108**. In certain embodiments the agitator/mixer comprises a motor **113** that can drive rotation of the heat pipes attached to structure frame **111** via for example, a helical drive **116** when the thermal storage materials are in liquid phase. As a result, this motion effectively mixes the liquid phase change thermal storage materials inside the pipes to avoid the possible nucleation and phase separation of the thermal storage medium. In various embodiments other agitators/agitation systems can be contemplated. Such systems include, for example acoustic/ultrasound agitation, mechanical vibrators (*e.g.*, piezoelectric vibrators), and the like.

[0028] An alternative method to avoid the possible nucleation and phase separation of the thermal storage medium is to mix super thin (*e.g.*, 5-50 μm , or 10-30 μm , or 10-20 μm , *etc.*) glass fibers (*e.g.*, about 0.1 to about 10%, or 0.5% to about 5%, or about 1% in volume) into the phase change material inside the packing containers (*e.g.*, pipes). Other suitable materials will be recognized by one of skill in the art. Typically, any matrix materials with low density to prevent solids from precipitating to the bottom are suitable.

[0029] As noted above, the methods and devices described herein are particularly advantageous because of the separation of the “dry-cooling” process into the “condensation stage” and the “heat dissipation stage”. During the “condensation stage”, the turbine exhaust steam enters the storage tank and convert its latent heat (from vapor to water) into thermal storage material’s latent heat with phase change process.

[0030] However, the heat dissipation process involves entirely different heat exchange mechanism because it utilizes a forced convection process with air to dissipate the heat inside the packaged thermal storage pipes into the environment.

[0031] Consider the process of “heat dissipation stage” in the following illustrative example. Taking the temperature variation in a typical day into account, assume that the lowest temperature of that day equals to (a °C) and the highest temperature equals to ($a+b$ °C). Furthermore we propose a hypothesis that the temperature variation within one day can be described as a Sine function, and the summit of this curve (the highest temperature)

appears at 14:00 o'clock. We can then use the following formula to describe the ambient air temperature:

$$T_f = a + b \sin \left[\frac{\pi}{24} (t - 2) \right] \quad (1)$$

Suppose that the starting time of heat dissipation stage t_2 is later than 2:00 pm, so the
5 duration of the heat dissipation stage Δt_2 is given by:

$$\Delta t_2 = 52 - 2t_2 \quad (2)$$

By integration of the temperature variation function over the entire time duration of stage 2, we obtain the average ambient temperature during the heat dissipation stage:

$$\bar{T}_f = a + b \frac{\sin^2\left(\frac{\pi}{96} \Delta t_2\right)}{\frac{\pi}{96} \Delta t_2} \quad (3)$$

10 According to the relevant aerodynamics, the Nusselt number Nu, Reynolds Number Re and the heat exchange coefficient h are related by:

$$h = Nu \frac{\lambda}{d_e} \quad (4)$$

where Nu is given by:

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.33}$$

15 while Reynolds Number Re is in the range of $Re = 10^4 - 1.2 \times 10^5$; d_e is the equivalent diameter of the thermal storage tank cylinder, λ is the thermal conductivity of the air flow, which is 2.63×10^{-2} W/(m K) for air at temperature of 25°C Reynolds Number Re is given by:

$$Re = \frac{u d_e}{\nu} \quad (5)$$

where μ is the velocity of the air flow and ν is the coefficient of viscosity of air, which is $15.53 \times 10^{-6} \text{ m}^2/\text{s}$ for air at temperature of 25°C . Substituting equation (5) into equation (4), we have:

$$h = 4.04u^{0.613} d_e^{-0.382} \text{ (W / m}^2\text{ K)} \quad (6)$$

5 Moreover, the equivalent diameter of cylinder d_e is expressed by:

$$d_e = \frac{4A_c}{P} = \frac{D_i^2 - Nd_o^2}{D_i + Nd_o} \quad (7)$$

where D_i is the internal diameter of the cylinder tank, while d_o is the external diameter of the N pipes.

[0032] If the heat transfer rate Φ_1 by the exhausted steam from turbine and the time
10 duration of “condensation stage” Δt_1 are known, the requirement of the phase change material (PCM) is then determined. The total volume of the phase change material (PCM) is given by:

$$V_{PCM} = \frac{\Delta t_1 \Phi_1}{\rho_{PCM} \gamma_{PCM}} \quad (8)$$

where ρ_{PCM} is the density of the phase change material (PCM) in units of kg/m^3 ; and γ_{PCM}
15 is the latent heat of the phase change material (PCM), in unite of J/kg .

[0033] From the fundamental heat transfer equation:

$$\frac{\Delta t_1 \Phi_1}{\Delta t_2} = hA\Delta T_m \quad (9)$$

where h is the convection heat transfer coefficient between air flow and outer wall of pipes, in units of $\text{W}/(\text{m}^2 \text{ K})$; A is the heat exchange area for heat dissipation processes, in units of
20 m^2 ; ΔT_m is the average convection heat exchange temperature difference, i.e. temperature difference between the outer diameter of the main tank surface and cooling air temperature, K ; Substitute (6) into (9):

$$\frac{\Delta t_1 \Phi_1}{\Delta t_2} = AhT_m = 4.04u^{0.613} d_e^{-0.382} A\Delta T_m \quad (10)$$

[0034] When we consider the formula of air specific heat we have:

$$\Phi_2 = C_p q_m \Delta T_a \tag{11}$$

where Φ_2 is the heat dissipation power for the proposed apparatus, C_p is the specific heat at constant pressure of the air flow, in units of J/(kg K); q_m is the mass flow rate of the air in units of kg/s, and ΔT_a is the temperature different of the air flow between inlet and outlet of thermal storage tank **108**. In addition, it is noted that $\Delta t_1 \Phi_1 = \Delta t_2 \Phi_2$ which means that equation (10) equals equation (11).

[0035] Another temperature difference that can be taken into account is the difference between the temperature of the phase change material (PCM) and the outer wall of package pipe containing the PCM. This temperature difference relates to the properties of the phase change material (PCM), the external diameter of pipe d_o and the time duration of stage Δt_1 or Δt_2 :

$$|T_m - T_w| = \frac{\rho_{PCM} \gamma_{PCM} d_o^2}{8 \lambda_{PCM} \Delta t} \tag{12}$$

In order to keep the temperature difference between the phase change material (PCM) and the outer wall of the pipes suitable small, the pipes (PCM containers) can be selected such that the outer diameter is less than:

$$d_o \leq \sqrt{\frac{8 \lambda_{PCM} \Delta t |T_m - T_w|}{\rho_{PCM} \gamma_{PCM}}} \tag{13}$$

[0036] The Power of the blower motor generating the cooling airflow (e.g., wind) is given by:

$$N = \frac{169.6S}{T} u^3 \tag{6}$$

using equation (5), we have:

$$N = 0.019 S T^{-1} d^{1.76} \left(\frac{\Delta_1 \Phi}{\Delta_2 \Delta T_m A} \right)^{4.76} \tag{7}$$

where N is the power of the air blower in units of W ; S is the cross sectional area of the air duct in units of m^2 ; T is the temperature of the ambient air in units of K ; A is the heat transfer area in units of m^2 which is the sum of all the surface areas for the thermal storage package pipe (container) outside surfaces.

5 [0037] In certain embodiments, as indicated above the devices described herein can be configured in a "parallel" architecture comprising multiple devices, *e.g.*, as illustrated in Figure 2. In certain embodiments N , the number of devices in such a configuration is greater than 2, more preferably greater than 3, still more preferably greater than 4, 5, or 6, in certain embodiments, greater than 10, 20, 30, 40, or 50, and in certain embodiments, greater
10 than 75, 100, 150 or greater than 200.

[0038] In this way, the system can handle larger amounts of exhausted steam (for example, larger than 3.6 MWh heat capacity, as in the example storage unit described above) or sustain 24 hours of continuing operation by alternating "condensing/storage" and "heat dissipation" process with different containers. Another advantage with this
15 configuration is that the number of valves that allow ambient air to enter the tank during the dissipation cycle can be reduced significantly to lower the system cost. For example, in certain embodiments, the N containers can share two butterfly valves, one for air entering path and the other for the air exhaust path through the main pipe and parallel pipes configured as described in Figure 2. In addition, one wind blower with larger operation
20 power can be used for the entire system, further simplifying system construction and reducing cost.

[0039] Continuing with the example described above, and considering the heat dissipation stage. Table 1 lists typical heat dissipation time periods required for each thermal storage tank given the ambient air temperature for a typical year for the said
25 MW CSP power plant. The average ambient air temperatures for each month are for the application location in northern China. It should be noted, however, that because the ambient air temperature during the night is much lower than the phase change temperature relative to the temperature difference of T_w and T_m , the required Δt_2 is actually much shorter than Δt_1 . In other words, the "delay" is much more effective than the "prolonged",
30 especially during the winter season.

Table 1. Illustrates typical heat dissipation time periods given the ambient temperature for a typical year and realistic application of the system.

Phase change material	Phase change temperature	density (kg/m ³)	Heat of fusion (kJ/kg)	Thermal conductivity (W/m · K)
Na ₂ CO ₃ · 10H ₂ O	32 °C	1349 (l) 1447 (s)	267	0.54
	Ambient temperature	Δt ₂	T _w - T _m	Dissipation temperature difference
January	-16.1 / -7.2 °C	2.37 h	2.6 °C	26.8 °C
February	-14.3 / -3.4 °C	2.45 h		25.7 °C
March	-1.9 / 9.9 °C	3.18 h		18.6 °C
April	5.1 / 17.7 °C	3.81 h		14.6 °C
May	9.2 / 23.1 °C	4.37 h		12.1 °C
June	14.7 / 26.3 °C	5.08 h		9.1 °C
July	17.7 / 29.6 °C	5.58 h		7.4 °C
August	15.0 / 25.2 °C	5.09 h		9.1 °C
September	10.5 / 21.9 °C	4.44 h		11.6 °C
October	2.4 / 15.1 °C	3.54 h		16.1 °C
November	-4.6 / 5.3 °C	2.97 h		20.3 °C
December	-13.0 / -3.4 °C	2.51 h		25.0 °C

[0040] It is very easy to control the system operation. During the “condensation stage”, because the heat exchange area is sufficiently large and the required heat exchange temperature difference is relatively small, as described in the above examples, no specific control is needed except to open the control valve **102** to start the “condensation stage” and to close this valve when the condensation stage” is finished. During this stage, the exhaust steam will first condensed onto the packaged thermal storage pipe surface while transferring the latent heat into the phase changing material with very small temperature difference. As time passes, this heat transfer temperature difference gradually increases. However, even at the end of the stage, this temperature difference is still relatively small. In the above described example, this value is only 2.6°C. For the “heat dissipation stage”, as described above, valve **102** is closed, and valve **114** is opened to let air into the storage tank until the pressure inside the tank is at or close to outside pressure. The two valves **106** in Figure 1 or valves **205** and **206** in Figure 2 are opened. The blower is turned on, or where a cooling air stream is provided by ducted ambient wind, the ducts are open. When used, the power setting of the blower depends on the ambient temperature, as illustrated in the above example and Table 1.

[0041] In various embodiments low temperature phase change materials are preferred as the thermal storage material for the delay and prolonged dry-cooling applications. Means of determining suitable parameters for the phase change materials are provided above. Illustrative low temperature phase change materials believed to be suitable for the heat storage material include, but are not limited to those shown in Table 2.

Table 2. Illustrative low temperature phase change materials suitable for exhaust steam condensation/thermal storage applications.

Phase Change Thermal Storage Material (molecular formula)	Melting point °C	Heat of Fusion (KWh/ton)
LiClO ₃ ·3H ₂ O	8.1	70.3
ZnCl ₂ ·3H ₂ O	10	/
K ₂ HPO ₄ ·6H ₂ O	13/14	30.3
45%CaCl ₂ ·6H ₂ O+ 55%CaBr ₂ ·6H ₂ O	14.7	38.9
NaOH 3 1/2 H ₂ O	15/15.4	/
51-55%Cu(NO ₃) ₂ ·6H ₂ O +45-49% LiNO ₃ ·3H ₂ O	16.5	69.4
45-52%LiNO ₃ ·3H ₂ O+48- 55% Zn(NO ₃) ₂ ·6H ₂ O	17.2	61.1
Na ₂ CrO ₄ ·10H ₂ O	18	/
KF·4H ₂ O	18.5-19.0	64.2
FeBr ₃ ·6H ₂ O	21/27	29.2
55-65%LiNO ₃ ·3H ₂ O +35-45% Ni(NO ₃) ₂ ·6H ₂ O	24.2	63.9
66.6%CaCl ₂ ·6H ₂ O+33.4% MgCl ₂ ·6H ₂ O	25	35.3
45%Ca(NO ₃) ₂ ·6H ₂ O +55%Zn(NO ₃) ₂ ·6H ₂ O	25	36.1
50%CaCl ₂ ·6H ₂ O +50% MgCl ₂ ·6H ₂ O	25	26.4
Mn(NO ₃) ₂ ·6H ₂ O	25.8/25.5	38.9
48%CaCl ₂ +4.3%NaCl +0.4%KCl+47.3%H ₂ O	26.8	52.2
50%CH ₃ CONH ₂ + 50%NH ₂ CONH ₂	27	45.3
CaCl ₂ ·6H ₂ O	29	53.0
CaCl ₂ ·6H ₂ O	29/29.2/ 29.6/29.7/30/ 29.9	50.0
LiNO ₃ ·3H ₂ O	30	82.2

LiNO ₃ ·3H ₂ O	30	52.5
67%Ca(NO ₃) ₂ ·4H ₂ O +33%Mg(NO ₃) ₂ ·6H ₂ O	30	37.8
47%Ca(NO ₃) ₂ ·4H ₂ O +53%Mg(NO ₃) ₂ ·6H ₂ O	30	37.8
60%CH ₃ COONa·3H ₂ O +40%CO(NH ₂) ₂	31.5	62.8
Na ₂ CO ₃ ·10H ₂ O	32	74.2
Na ₂ SO ₄ ·10H ₂ O	32.4	66.9
KFe(SO ₄) ₂ ·12H ₂ O	33	48.1
Na ₂ CO ₃ ·10H ₂ O	33/32	74.2
LiBr ₂ ·2H ₂ O	34	34.4
CaBr ₂ ·6H ₂ O	34	33.3
Na ₂ HPO ₄ ·12H ₂ O	35.5/36/35/40	77.8
0.925 mol Ca(NO ₃) ₂ ·4H ₂ O+0.075 mol CaCl ₂ ·6H ₂ O	35.6	43.1
Zn(NO ₃) ₂ ·6H ₂ O	36/36.4/36.1	40.8
FeCl ₃ ·6H ₂ O	37	61.9
Mn(NO ₃) ₂ ·4H ₂ O	37.1	31.9
Na ₂ HPO ₄ ·12H ₂ O	40	77.5
50%CH ₃ COONa·3H ₂ O +50%HCONH ₂	40.5	70.8
CoSO ₄ ·7H ₂ O	40.7	47.2
KF·2H ₂ O	41.4/42	45.0
CH ₃ COOK·1 1/2 H ₂ O	42	/
MgI ₂ ·8H ₂ O	42	36.9
CaI ₂ ·6H ₂ O	42	45.0
K ₂ HPO ₄ ·7H ₂ O	45	40.3
K ₃ PO ₄ ·7H ₂ O	45	/
Zn(NO ₃) ₂ ·4H ₂ O	45/45.5	30.6
Ca(NO ₃) ₂ ·4H ₂ O	42.7/47	42.5
Mg(NO ₃) ₂ ·4H ₂ O	47	39.4
Fe(NO ₃) ₃ ·9H ₂ O	47	43.1
Na ₂ HPO ₄ ·7H ₂ O	48	/
Na ₂ SiO ₃ ·4H ₂ O	48	46.7
K ₂ HPO ₄ ·3H ₂ O	48	27.5
MgSO ₄ ·7H ₂ O	48.5	56.1

$\text{Na}_2\text{S}_2\text{O}_3 \cdot 5\text{H}_2\text{O}$	48/48-49/48.5	58.1
$\text{Ca}(\text{NO}_3)_2 \cdot 3\text{H}_2\text{O}$	51	28.9
61.5% $\text{Mg}(\text{NO}_3)_2 \cdot 6\text{H}_2\text{O}$ +38.5% NH_4NO_3	52	36.5

[0042] The foregoing embodiments are intended to be illustrative and not limiting. Using teachings provided herein, other variations will be available to those of skill in the art. For example, while the system in Figure 1 is shown with a fan **107** to drive ambient air through the thermal storage system during the heat dissipation period in various

5 embodiments, a fan is not required. For example, where the system is sited in locations having significant winds, particular at night, the system can comprise ducts to channel ambient winds through the thermal storage system thereby reducing or avoiding the power usage required by a fan.

[0043] Similarly, a valve **104** is shown to control application of vacuum to chamber **108**, while a separate valve **114** is shown to introduce atmospheric pressure air into chamber **108**. However, in certain embodiments, these need not be separate valves. Thus, for example valve **104** can be configured to switch between a vacuum source and atmospheric pressure air thereby obviating valve **114**.

[0044] Figure 1 also shows that in certain embodiments, the device can comprise an
15 agitator/mixer configured to rotate perforated plates to which the chambers containing the thermal storage material are attached. However, numerous other configurations to achieve such mixing will be recognized and available to one of skill in the art.

[0045] Thus, it is understood that the examples and embodiments described herein are for illustrative purposes only and that various modifications or changes in light thereof
20 will be suggested to persons skilled in the art and are to be included within the spirit and purview of this application and scope of the appended claims. All publications, patents, and patent applications cited herein are hereby incorporated by reference in their entirety for all purposes.

CLAIMS

What is claimed is:

- 5 1. A device for steam condensation and delayed dissipation of the heat produced by said condensation, said device comprising:
- a combined condensation/thermal storage chamber, said chamber comprising:
- 10 one or more valved ports for receiving steam from a steam source;
- one or more containers containing a thermal storage material;
- 15 one or more valved ports for applying a vacuum to said thermal storage chamber;
- one or more valved ports for introducing ambient pressure air into said chamber;
- one or more valved ports for removing condensed water from said chamber; and
- a valve system operably coupling said chamber to a source of ambient temperature air.
- 20 2. The device of claim 1, wherein said one or more containers containing a thermal storage material is a plurality of containers each containing a thermal storage material.
3. The device according to any one of claims 1-2, wherein said thermal storage material is a phase change thermal storage material (PCM).
- 25 4. The device of claim 3, wherein said thermal storage material is a liquid/solid phase change thermal storage material.
5. The device of claim 3, wherein said thermal storage material comprises a material selected from the thermal storage materials shown in Table 2.
6. The device of claim 3, wherein said thermal storage material comprises $\text{Na}_2\text{CO}_3 \cdot 10\text{H}_2\text{O}$.

7. The device according to any one of claims 3-6, wherein said PCM contains glass microfibers or nanofibers.

8. The device according to any one of claims 1-7, wherein said device further comprises an apparatus to cause mixing of liquid phase thermal storage material in the containers containing the thermal storage material.

9. The device of claim 8, wherein said containers are attached to a structure frame and said apparatus comprises a motor configured to cause rotation of said structure frame and the attached containers.

10. The device according to any one of claims 1-9, wherein said one or more valved ports for applying a vacuum to said thermal storage chamber are operably coupled to a vacuum pump.

11. The device according to any one of claims 1-10, wherein said one or more valved ports for applying a vacuum to said thermal storage chamber and said one or more valved ports for introducing ambient pressure air into said chamber are controlled by separate valves.

12. The device according to any one of claims 1-10, wherein said one or more valved ports for applying a vacuum to said thermal storage chamber and said one or more valved ports for introducing ambient pressure air into said chamber are controlled by the same valve(s).

13. The device according to any one of claims 1-12, wherein said valve system operably coupling said chamber to a source of ambient temperature air comprises a butterfly valve.

14. The device according to any one of claims 1-13, wherein said source of ambient air is a fan and/or blower.

15. The device according to any one of claims 1-13, wherein said source of ambient air comprises one or more ducts configured to receive ambient wind.

16. The device according to any one of claims 1-15, wherein said device is one of a plurality of said devices configured in a parallel configuration.

17. The device according to any one of claims 1-16, wherein said one or more valved ports for receiving steam from a steam source are operably coupled to the low temperature steam output from a turbine.

18. The device of claim 17, wherein said turbine is a turbine in a power plant selected from the group consisting of a coal-fired power plant, a gas-fired power plant, a nuclear power plant, and a solar thermal power plant.

19. The device in claim 18, wherein said device is sited in a desert or a region with limited water availability.

20. A system for delayed heat dissipation from the condensation of waste steam, said system comprising a plurality of devices according to any one of claims 1-15, configured in a parallel configuration.

21. The system of claim 20, wherein said system comprises at least 10, or at least 20, or at least 30, or at least 40, or at least 50, or at least 60, or at least 70, or at least 80, or at least 90, or at least 100, or at least 150, or at least 200 of said devices.

22. The system according to any one of claims 20-21, wherein said system is operably coupled to the low temperature steam output from a turbine.

23. The system of claim 22, wherein said turbine is a turbine in a power plant selected from the group consisting of a coal-fired power plant, a gas-fired power plant, a nuclear power plant, and a solar thermal power plant.

24. The system in claim 23, wherein said device is sited in a desert or a region with limited water availability.

25. A dry-cooling condensation method, said method comprising receiving steam from a source of steam; condensing the steam into water while transferring the latent heat of said steam into the latent heat of a thermal storage material; and dissipating the latent heat from said thermal storage material at a later time when the ambient temperature is lower than the ambient temperature at the time the steam was condensed into water.

26. The method of claim 25, wherein said thermal storage material is a phase change thermal storage material (PCM).

27. The method of claim 25, wherein said thermal storage material is a liquid/solid phase change thermal storage material.

5 28. The method of claim 25, wherein said thermal storage material comprises a material selected from the thermal storage materials shown in Table 2.

29. The method of claim 25, wherein said thermal storage material comprises $\text{Na}_2\text{CO}_3 \cdot 10\text{H}_2\text{O}$.

10 30. The method according to any one of claims 25-29, wherein said source of steam is the steam output from a turbine.

31. The method of claim 30, wherein said turbine is in a power plant selected from the group consisting of a coal-fired power plant, a gas-fired power plant, a nuclear power plant, and a solar thermal power plant.

15 32. The method according to any one of claims 25-31, wherein said receiving and condensing comprises receiving and condensing during daylight hours.

33. The method of claim 32, wherein said receiving and condensing comprises receiving and condensing between noon and 3:00 pm.

20 34. The method according to any one of claims 25-33, wherein said dissipating comprise dissipating the latent heat from said thermal storage material during the late afternoon, and/or evening, and/or night.

35. The method according to any one of claims 25-34, wherein said method is performed using a device according to any one of claims 1-19.

36. The method of claim 35, wherein said receiving and condensing comprises:

25 opening said one or more valved ports for applying a vacuum to said thermal storage chamber to reduce the ambient pressure in said thermal storage chamber;
opening said one or more valved ports for receiving steam to introduce steam from said steam source into said chamber, whereby said steam condenses transferring latent heat of steam into said thermal storage material; and

operating said one or more valved ports for removing condensed water from said chamber to return the condensed water to the system providing said steam source.

37. The method according to any one of claims of claims 35-36, wherein
5 said dissipating comprises:

restoring the pressure in said thermal storage chamber to atmospheric pressure;

operating said valve system operably coupling said chamber to a source of ambient temperature air to pass ambient temperature air through said thermal
10 storage chamber to transfer heat from said thermal storage material to said air.

38. The method of claim 37, wherein passing ambient temperature air comprising operating a fan/blower to force air through said chamber.

39. The method of claim 37, wherein passing ambient temperature air comprising coupling said chamber to a duct system that channels wind through said
15 chamber.

40. The method according to any one of claims 37-39, wherein said dissipating further comprises operating an apparatus to provide mixing of fluid thermal storage material in the chambers containing said fluid thermal storage material.

41. The method of claim 40, wherein said method comprises operating a
20 motor to rotate a structure frame to which said chambers containing said thermal storage material are attached.

42. The method according to any one of claim 35-41, wherein said method is performed using a system comprising a plurality of said devices.

43. The method of claim 42, wherein said devices are configured in a
25 parallel configuration.

44. The method according to any one of claims 42-43, wherein substantially all of the devices in said system perform said receiving and condensing at the same time.

45. The method according to any one of claims 42-43, wherein substantially all of the devices in said system perform said dissipating at the same time.

46. The method of claim 42, wherein a plurality of the devices perform said receiving and condensing at the same time others of the devices are dissipating.

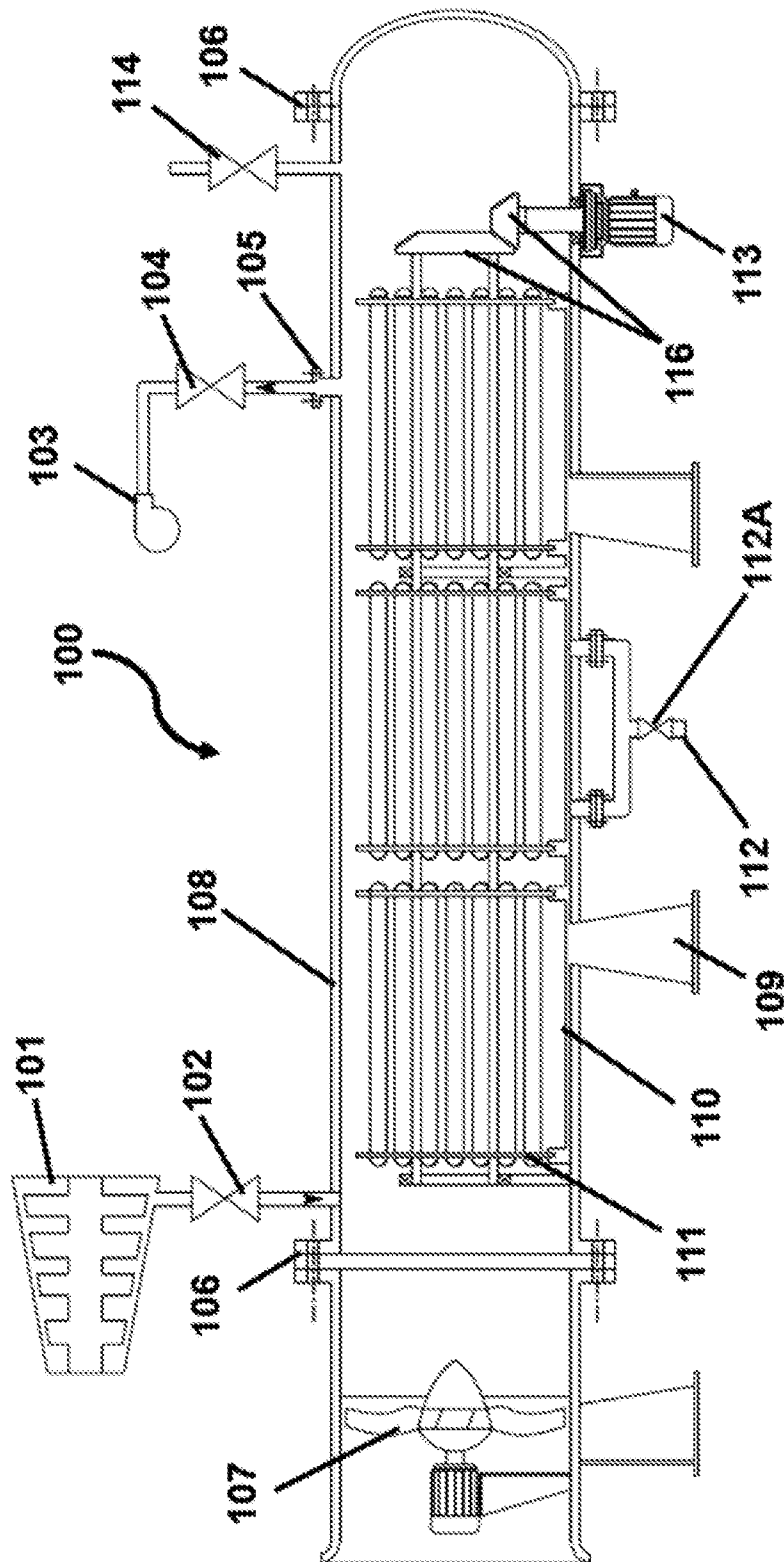


Fig. 1

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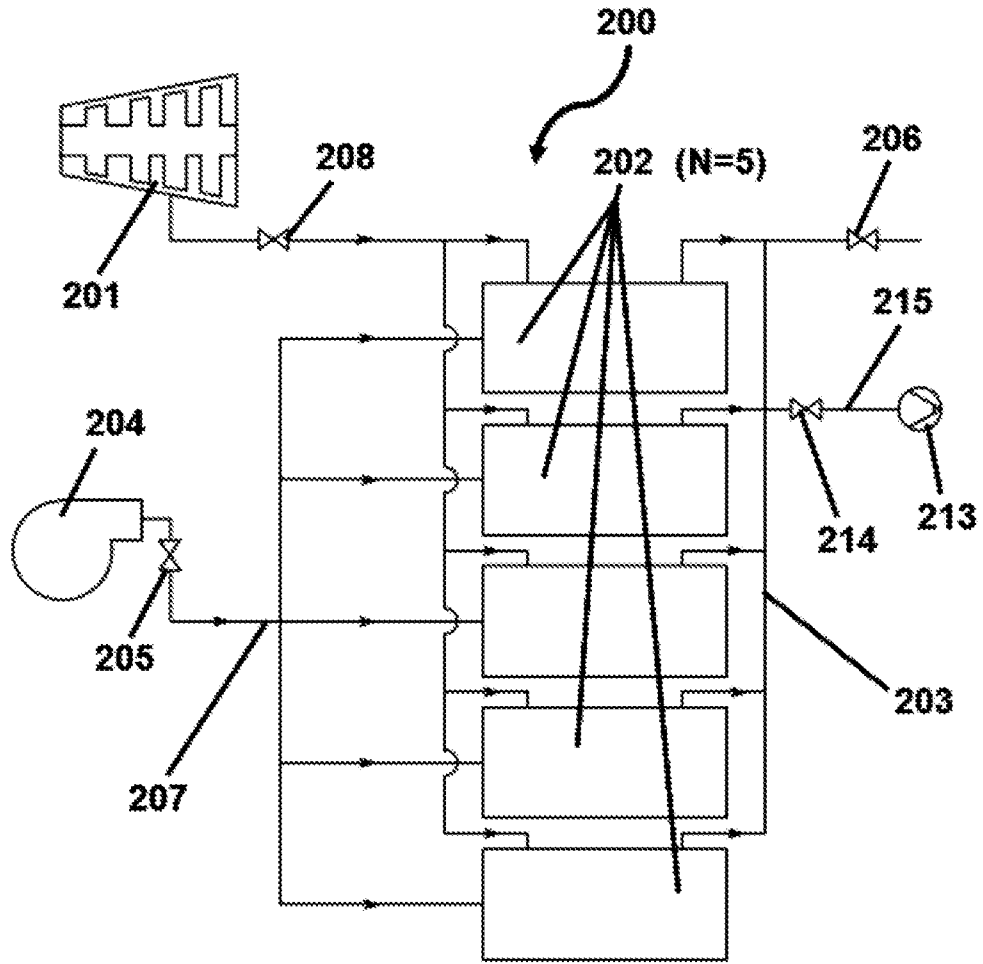


Fig. 2

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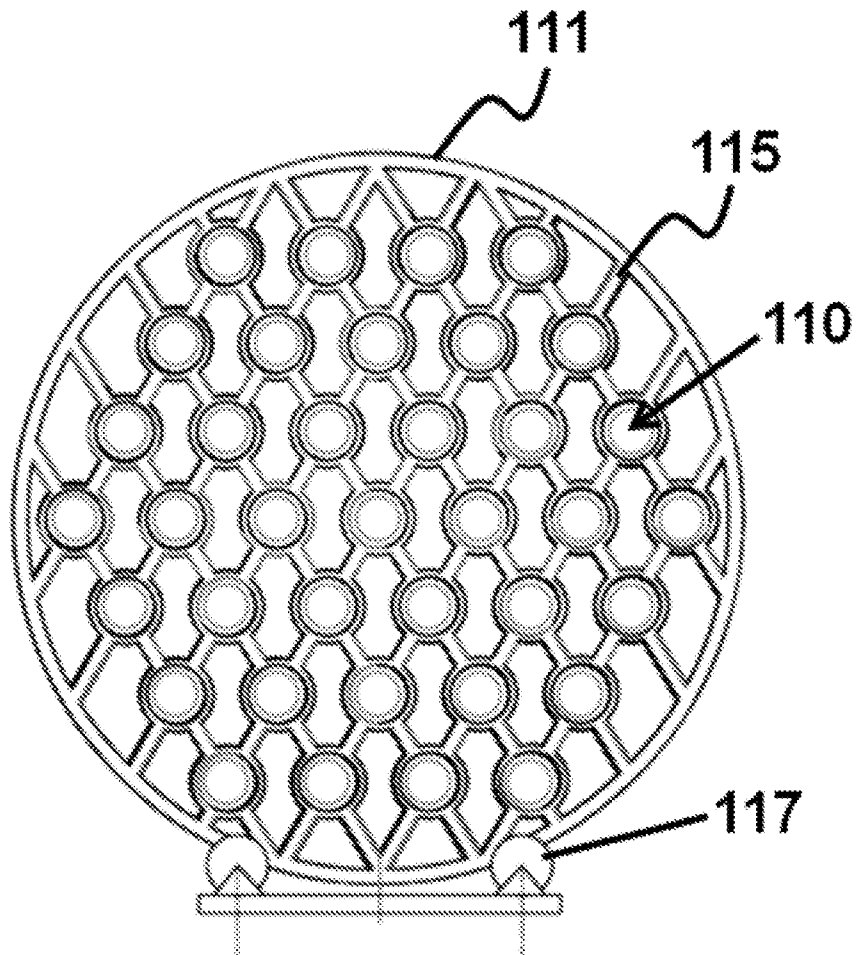


Fig. 3

INTERNATIONAL SEARCH REPORT

International application No.

PCT/US2012/054688

<p>A. CLASSIFICATION OF SUBJECT MATTER IPC(8) - F28D 20/02 (2012.01) USPC - 165/10 According to International Patent Classification (IPC) or to both national classification and IPC</p>																				
<p>B. FIELDS SEARCHED</p> <p>Minimum documentation searched (classification system followed by classification symbols) IPC(8) - F25B 27/00, 27/02; F28B 1/06, 9/00, 9/04, 9/10; F28C 1/08; F22D 3/04; F28D 15/00, 15/02, 20/00, 20/02; F01K 3/00 (2012.01) USPC - 165/4, 10, 104.11, 104.19, 104.21, 104.22, 112</p> <p>Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched</p> <p>Electronic data base consulted during the international search (name of data base and, where practicable, search terms used) PatBase, Google Patents</p>																				
<p>C. DOCUMENTS CONSIDERED TO BE RELEVANT</p> <table border="1"> <thead> <tr> <th>Category*</th> <th>Citation of document, with indication, where appropriate, of the relevant passages</th> <th>Relevant to claim No.</th> </tr> </thead> <tbody> <tr> <td>Y</td> <td>US 2011/0120673 A1 (XIANG et al) 26 May 2011 (26.05.2011) entire document</td> <td>25-31</td> </tr> <tr> <td>Y</td> <td>US 7,891,575 B2 (SAMI) 22 February 2011 (22.02.2011) entire document</td> <td>25-31</td> </tr> <tr> <td>A</td> <td>US 6,588,499 B1 (FAHLSING) 08 July 2003 (08.07.2003) entire document</td> <td>1-6, 25-31</td> </tr> <tr> <td>A</td> <td>US 3,175,960 A (KASSAT) 30 March 1965 (30.03.1965) entire document</td> <td>1-6, 25-31</td> </tr> <tr> <td>A</td> <td>US 5,082,534 A (BREU) 21 January 1992 (21.01.1992) entire document</td> <td>1-6, 25-31</td> </tr> </tbody> </table>			Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.	Y	US 2011/0120673 A1 (XIANG et al) 26 May 2011 (26.05.2011) entire document	25-31	Y	US 7,891,575 B2 (SAMI) 22 February 2011 (22.02.2011) entire document	25-31	A	US 6,588,499 B1 (FAHLSING) 08 July 2003 (08.07.2003) entire document	1-6, 25-31	A	US 3,175,960 A (KASSAT) 30 March 1965 (30.03.1965) entire document	1-6, 25-31	A	US 5,082,534 A (BREU) 21 January 1992 (21.01.1992) entire document	1-6, 25-31
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Y	US 2011/0120673 A1 (XIANG et al) 26 May 2011 (26.05.2011) entire document	25-31																		
Y	US 7,891,575 B2 (SAMI) 22 February 2011 (22.02.2011) entire document	25-31																		
A	US 6,588,499 B1 (FAHLSING) 08 July 2003 (08.07.2003) entire document	1-6, 25-31																		
A	US 3,175,960 A (KASSAT) 30 March 1965 (30.03.1965) entire document	1-6, 25-31																		
A	US 5,082,534 A (BREU) 21 January 1992 (21.01.1992) entire document	1-6, 25-31																		
<p><input type="checkbox"/> Further documents are listed in the continuation of Box C. <input type="checkbox"/></p>																				
<p>* Special categories of cited documents:</p> <table border="0"> <tr> <td>“A” document defining the general state of the art which is not considered to be of particular relevance</td> <td>“T” later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention</td> </tr> <tr> <td>“E” earlier application or patent but published on or after the international filing date</td> <td>“X” document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone</td> </tr> <tr> <td>“L” document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)</td> <td>“Y” document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art</td> </tr> <tr> <td>“O” document referring to an oral disclosure, use, exhibition or other means</td> <td>“&” document member of the same patent family</td> </tr> <tr> <td>“P” document published prior to the international filing date but later than the priority date claimed</td> <td></td> </tr> </table>			“A” document defining the general state of the art which is not considered to be of particular relevance	“T” later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention	“E” earlier application or patent but published on or after the international filing date	“X” document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone	“L” document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)	“Y” document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art	“O” document referring to an oral disclosure, use, exhibition or other means	“&” document member of the same patent family	“P” document published prior to the international filing date but later than the priority date claimed									
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<p>Date of the actual completion of the international search 14 November 2012</p>		<p>Date of mailing of the international search report 07 DEC 2012</p>																		
<p>Name and mailing address of the ISA/US Mail Stop PCT, Attn: ISA/US, Commissioner for Patents P.O. Box 1450, Alexandria, Virginia 22313-1450 Facsimile No. 571-273-3201</p>		<p>Authorized officer: Blaine R. Copenheaver PCT Helpdesk: 571-272-4300 PCT OSP: 571-272-7774</p>																		

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US2012/054688

Box No. II Observations where certain claims were found unsearchable (Continuation of item 2 of first sheet)

This international search report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

- 1. Claims Nos.:
because they relate to subject matter not required to be searched by this Authority, namely:

- 2. Claims Nos.:
because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out, specifically:

- 3. Claims Nos.: 7-24, 32-46
because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a).

Box No. III Observations where unity of invention is lacking (Continuation of item 3 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:

- 1. As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims.
- 2. As all searchable claims could be searched without effort justifying additional fees, this Authority did not invite payment of additional fees.
- 3. As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:

- 4. No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

- Remark on Protest**
- The additional search fees were accompanied by the applicant's protest and, where applicable, the payment of a protest fee.
 - The additional search fees were accompanied by the applicant's protest but the applicable protest fee was not paid within the time limit specified in the invitation.
 - No protest accompanied the payment of additional search fees.