844-348-9168

gcolip@myiaire.com

1240 S. Hoagland Blvd.
Kissimmee, FL 34741

Date: October 20, 2025

From: Gary Colip

Re: Solar Thermal Safe Harbor Information

This letter is the cover letter for the information that iAIRE is providing with our Solar Thermal HVAC units to document why and how it meets the Safe Harbor criteria of being able to run and HVAC system for an hour with a system that has stored thermal energy. We have included the following information:

1. US Patents that show that the iAIRE solar panel is insulated unlike normal solar panels and is capable of storing thermal for more than an hour. Patent numbers:
    a. 9810455
    b. 10900694
    c. 10982862
    d. 11143437

2. 3rd party testing done on iAIRE Solar panels for our OG-100 rating on the solar panels that are approved by ICC-SRCC (International Code Committee – Solar Rating & Certification Corporation) showing how much solar thermal energy is transmitted into the solar panels. Certification #:
    a. 10002196A 5-ton panel
    b. 10002196B 1.25-ton panel
    c. 10002196C 2.5-ton panel
    d. 10002196D 3.75-ton panel
    e. 10002196E 7.5-ton panel
    f. 10002196F 10-ton panel
    g. 10002196G 15-ton panel

3. Test information from testing a 5-ton Solar HVAC unit showing a Solar Thermal Panel being charged with thermal energy. Then the HVAC system was run for 1 hour showing that there was sufficient thermal energy to run the HVAC system for 1 hour.

US009810455B2

(12) **United States Patent**
Kaiser

(10) **Patent No.: US 9,810,455 B2**
(45) **Date of Patent: Nov. 7, 2017**

(54) **HEAT AND ENERGY RECOVERY AND REGENERATION ASSEMBLY, SYSTEM AND METHOD**

(71) Applicant: **GASN, LLC**, Boca Raton, FL (US)

(72) Inventor: **Stewart Kaiser**, Boca Raton, FL (US)

(73) Assignee: **GASN LLC**, Boca Raton, FL (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 1243 days.

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(51) **Int. Cl.**

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F25B 29/00	(2006.01)
F24D 19/10	(2006.01)
F24H 3/06	(2006.01)
F24D 5/04	(2006.01)

(52) **U.S. Cl.**

CPC ***F25B 29/00*** (2013.01); ***F24D 5/04*** (2013.01); ***F24D 19/1084*** (2013.01); ***F24H 3/065*** (2013.01); *F24D 2200/18* (2013.01); *F24D 2200/22* (2013.01); *F24D 2220/06* (2013.01)

(58) **Field of Classification Search**

CPC .. F25B 29/00; F24D 2200/01; F24D 2200/18; F24D 2200/22; F24D 2200/06

USPC 237/50

See application file for complete search history.

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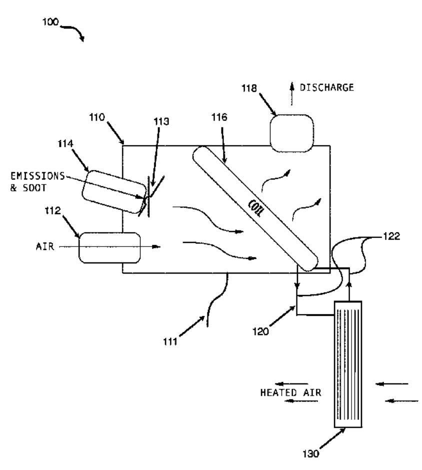
Primary Examiner — Avinash Savani
Assistant Examiner — Deepak Deean
(74) *Attorney, Agent, or Firm* — Allen Dyer Doppelt & Gilchrist

(57) **ABSTRACT**

The present invention is directed to a heat and energy recovery assembly, system and method. The heat and energy recovery assembly and system may include an insulated chamber for effectuating heat and energy exchange between a primary heat recovery exchanger and the reaction products of fossil fuel combustion gases, waste products, and air. The heat and energy recovery assembly and system are particularly useful on furnace systems.

15 Claims, 10 Drawing Sheets



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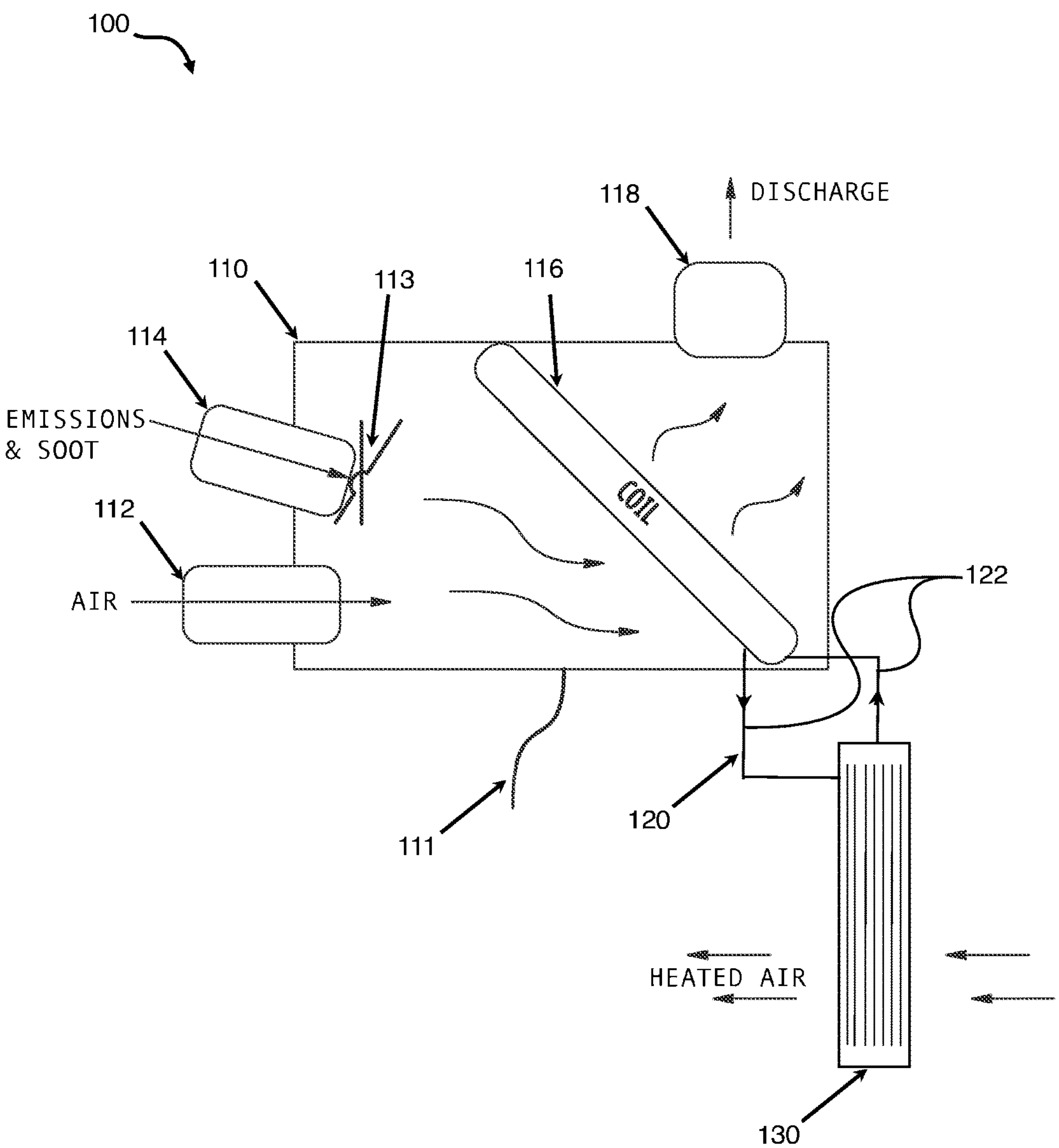
100
118
DISCHARGE
110
113
116
114
EMISSIONS
& SOOT
COIL
112
AIR
122
111
120
HEATED AIR
130

FIGURE 1

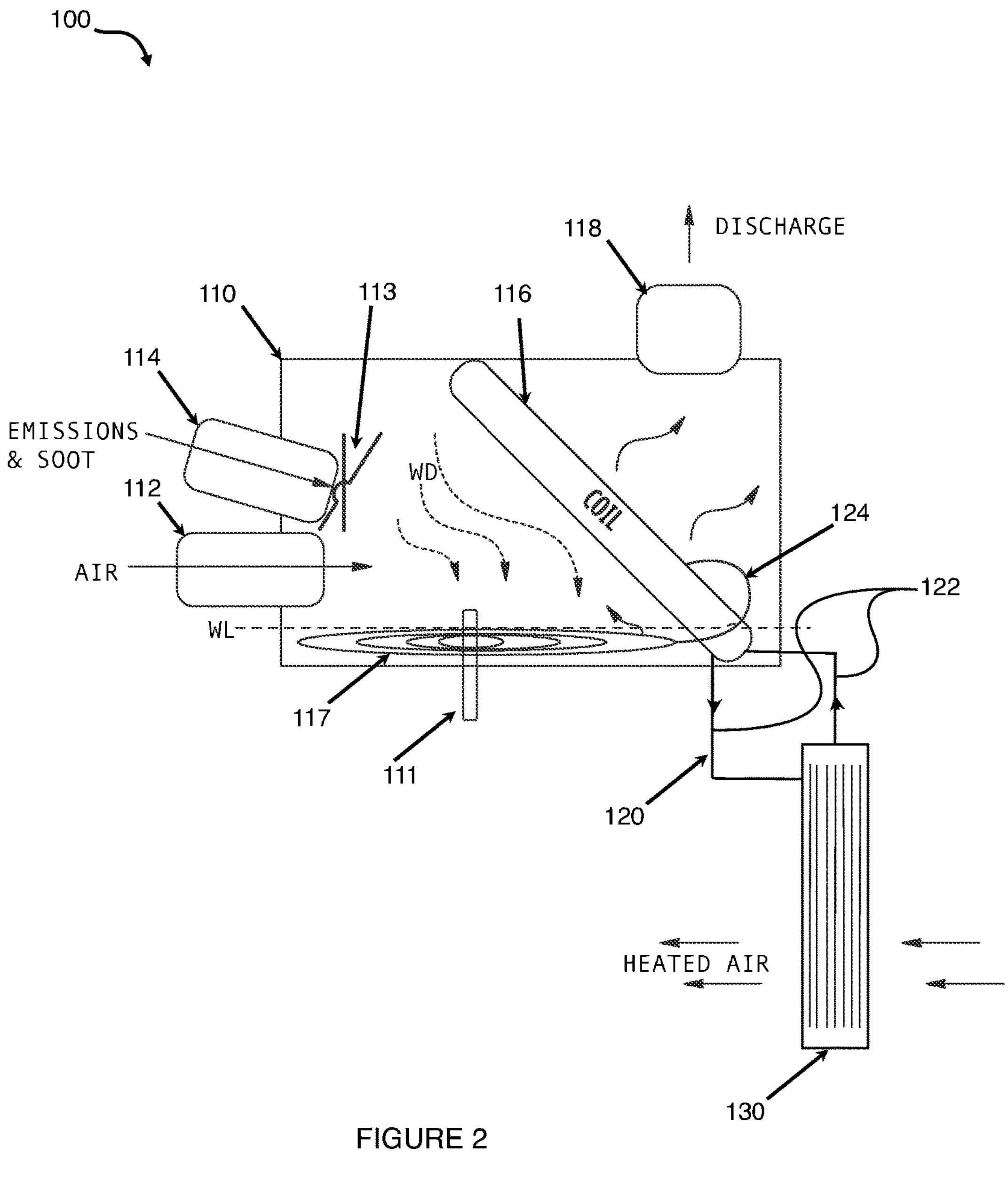
100
118
DISCHARGE
110
113
116
114
EMISSIONS
& SOOT
112
WD
COIL
124
AIR
122
WL
117
111
120
HEATED AIR
130

FIGURE 2

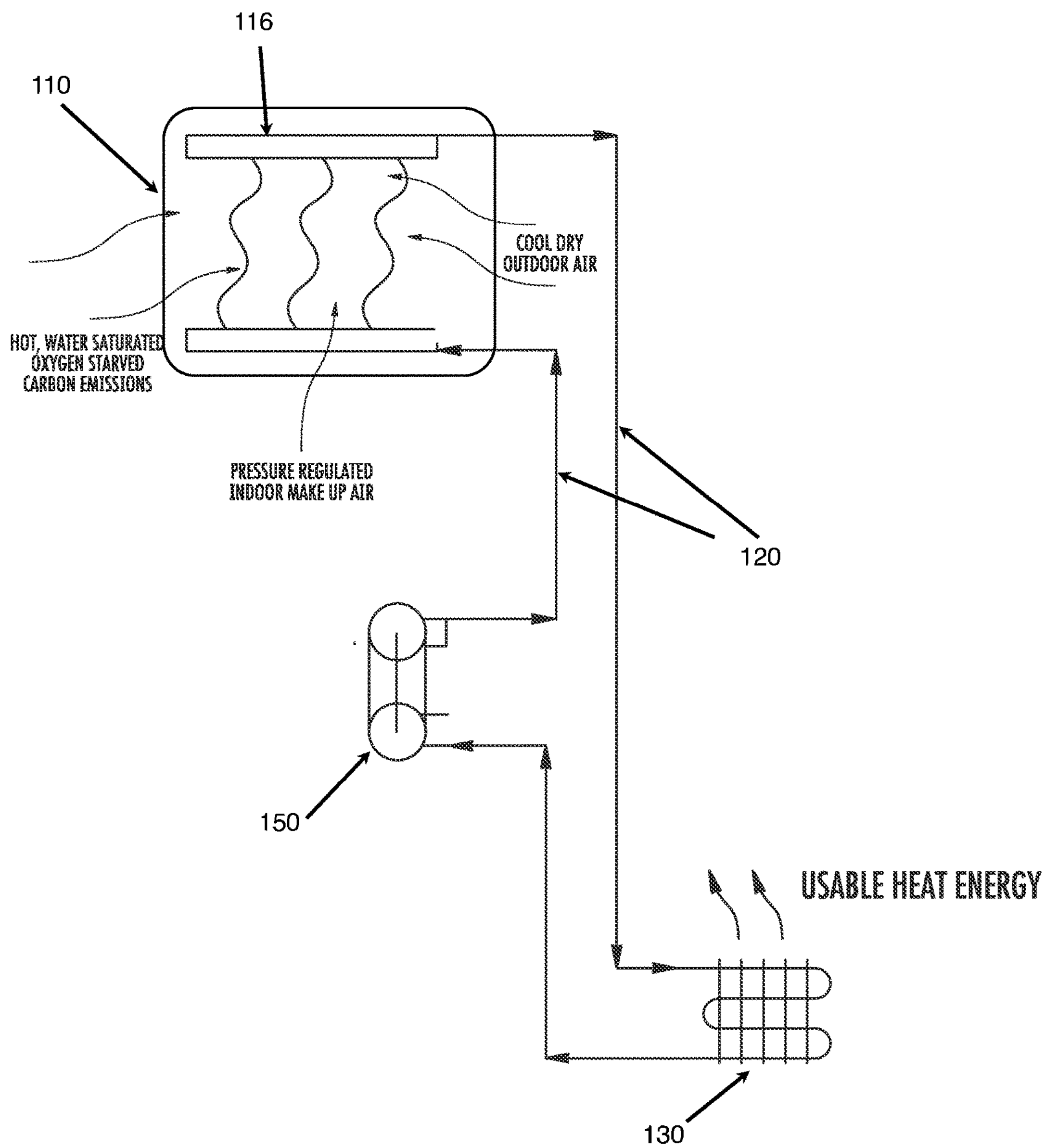
116
110
COOL DRY
OUTDOOR AIR
HOT, WATER SATURATED
OXYGEN STARVED
CARBON EMISSIONS
PRESSURE REGULATED
INDOOR MAKE UP AIR
120
150
USABLE HEAT ENERGY
130

FIGURE 3

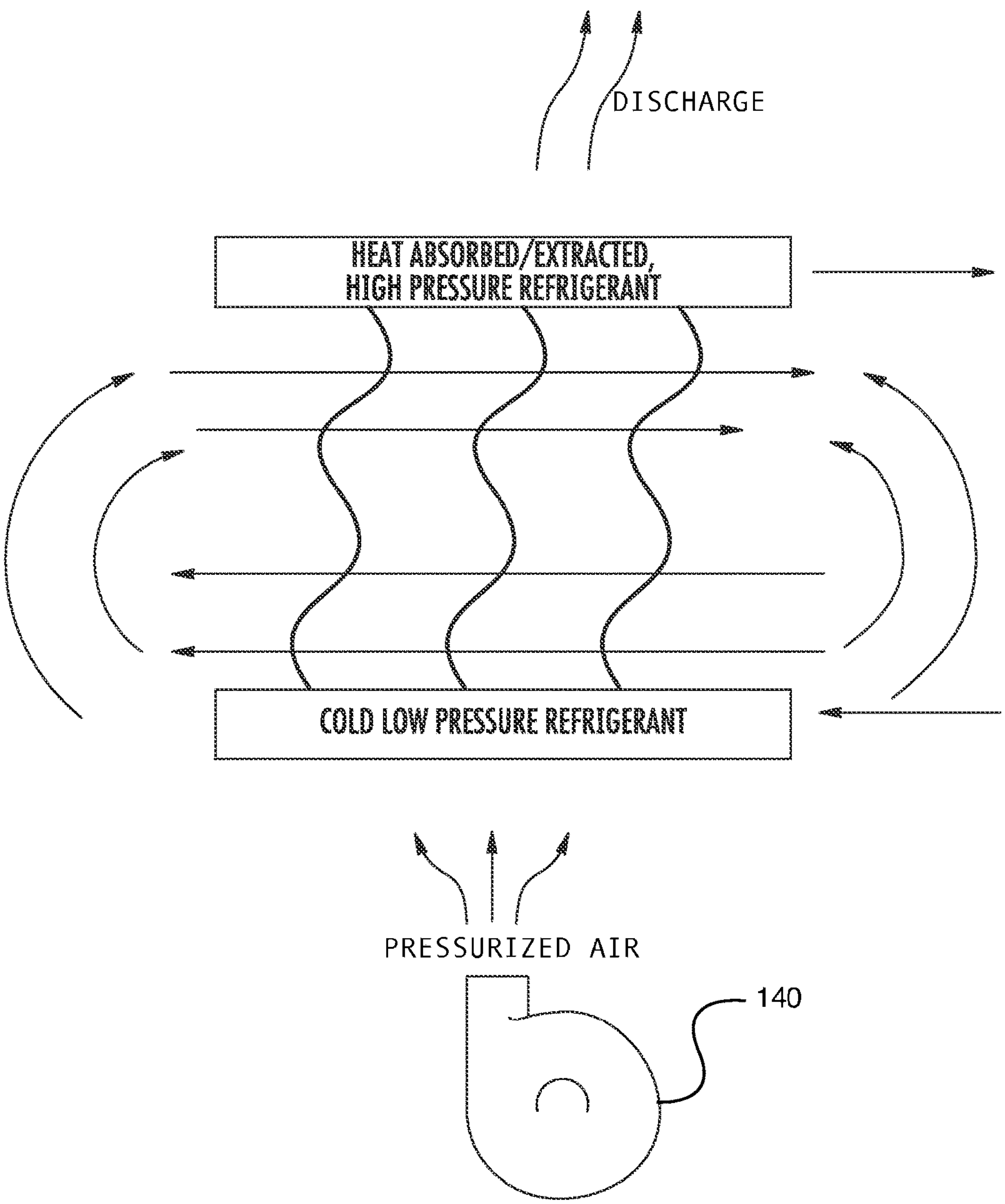
DISCHARGE
HEAT ABSORBED/EXTRACTED,
HIGH PRESSURE REFRIGERANT
COLD LOW PRESSURE REFRIGERANT
PRESSURIZED AIR
140

FIGURE 4

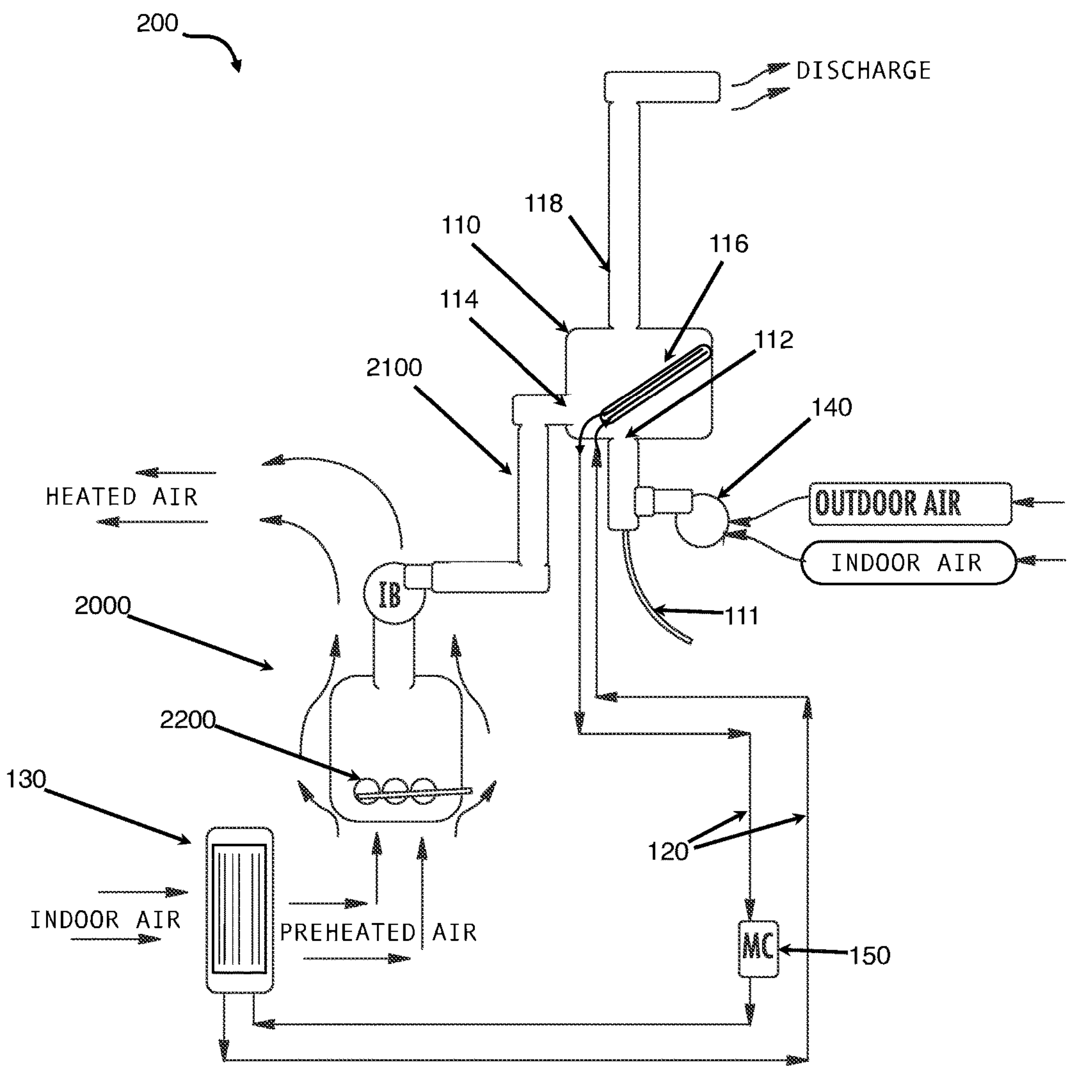
200
DISCHARGE
118
110
116
114
112
2100
140
HEATED AIR
OUTDOOR AIR
INDOOR AIR
IB
2000
111
2200
130
120
INDOOR AIR
PREHEATED AIR
MC
150

FIGURE 5

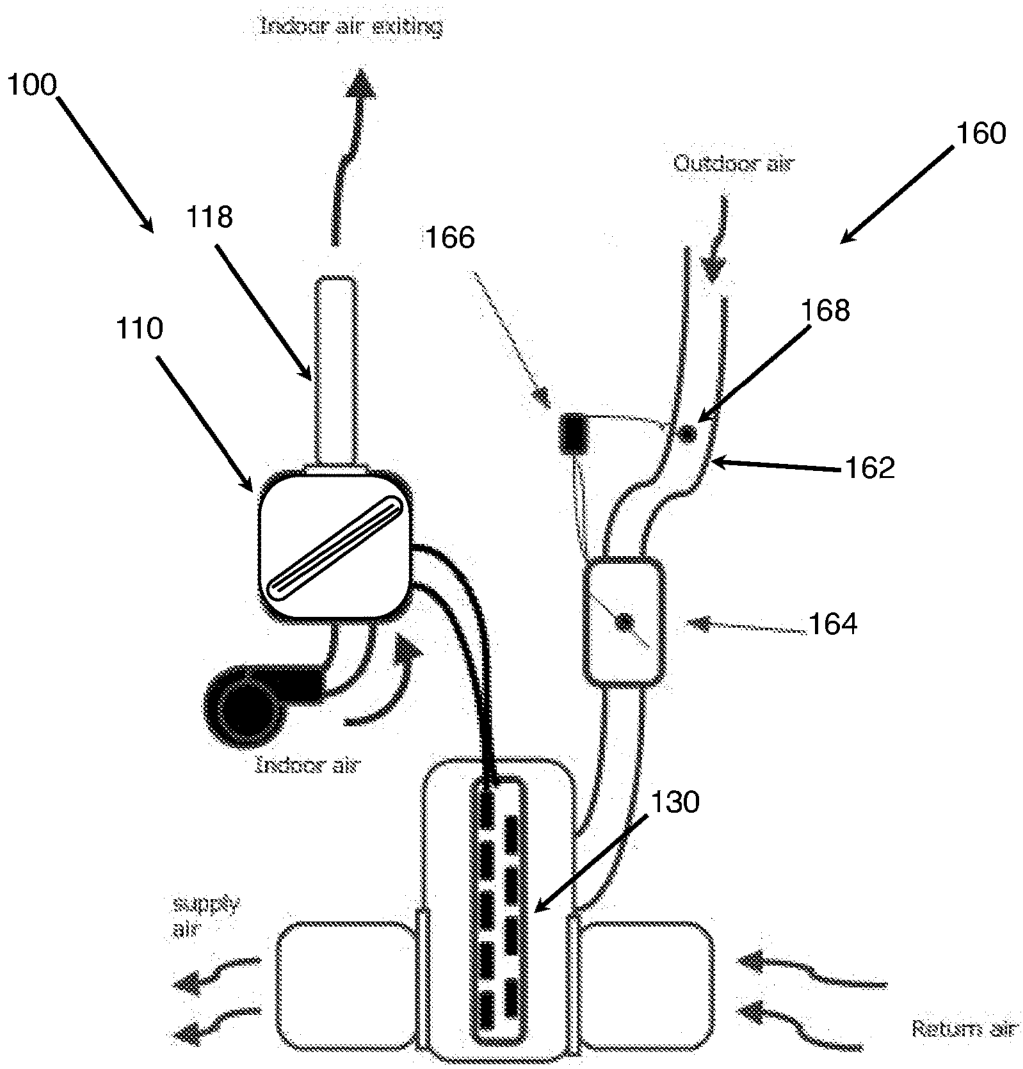
Indoor air exiting
100
118
110
166
160
Outdoor air
168
162
164
Indoor air
130
supply air
Return air

FIGURE 6

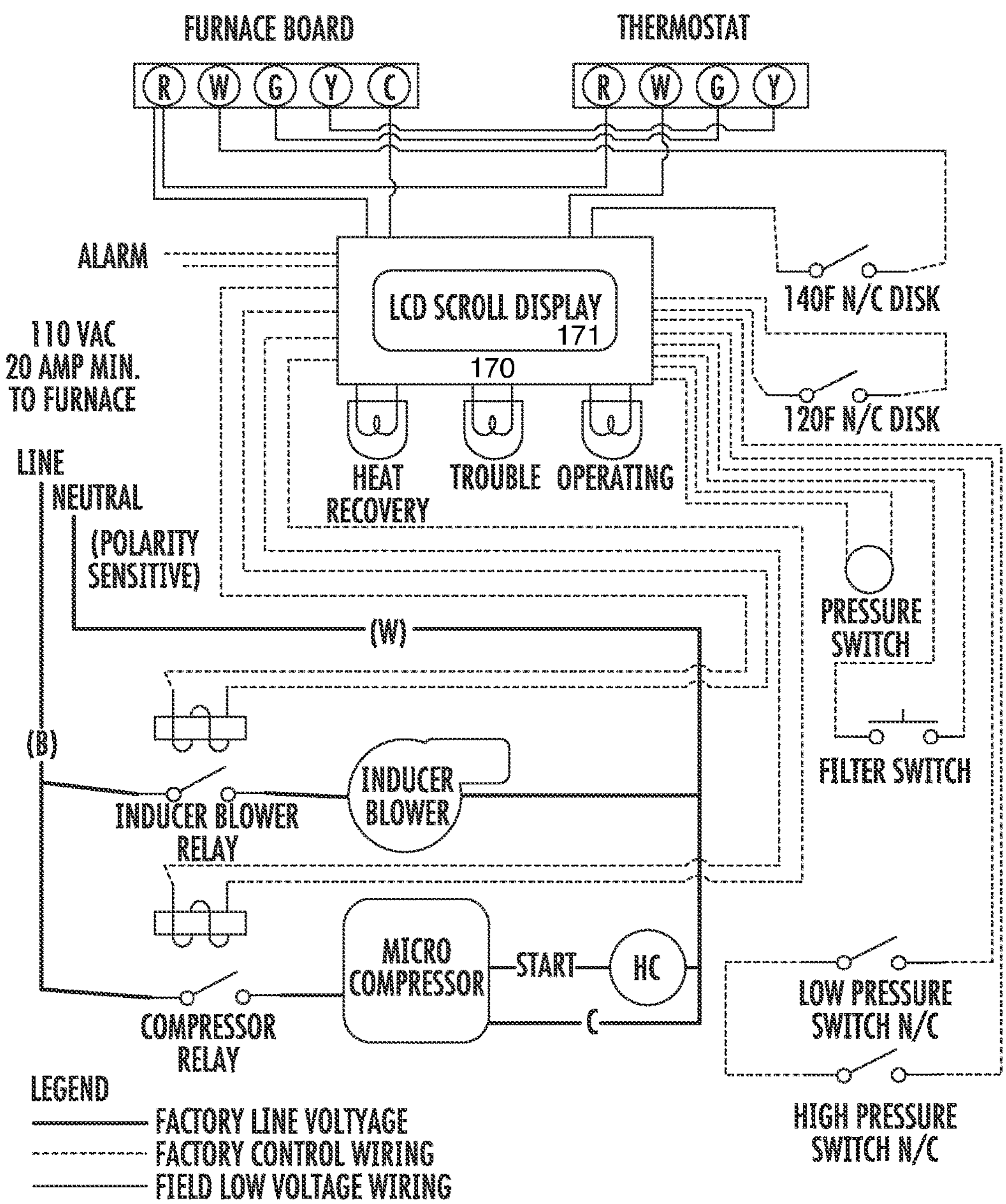
FURNACE BOARD
R W G Y C
THERMOSTAT
R W G Y
ALARM
LCD SCROLL DISPLAY
171
170
140F N/C DISK
120F N/C DISK
110 VAC
20 AMP MIN.
TO FURNACE
LINE
NEUTRAL
(POLARITY SENSITIVE)
HEAT RECOVERY
TROUBLE
OPERATING
PRESSURE SWITCH
(W)
FILTER SWITCH
(B)
INDUCER BLOWER
INDUCER BLOWER RELAY
MICRO COMPRESSOR
START
HC
C
COMPRESSOR RELAY
LOW PRESSURE SWITCH N/C
HIGH PRESSURE SWITCH N/C
LEGEND
FACTORY LINE VOLTYAGE
FACTORY CONTROL WIRING
FIELD LOW VOLTAGE WIRING

FIGURE 7

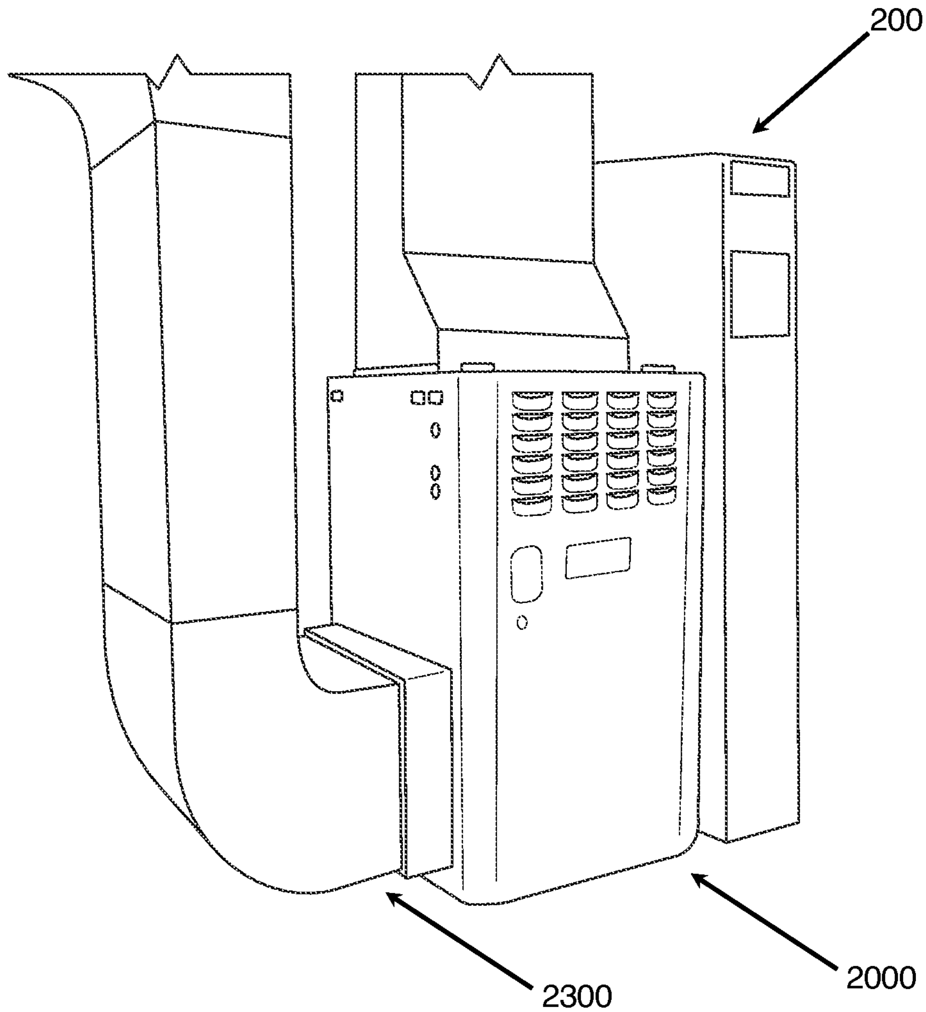

FIGURE 8

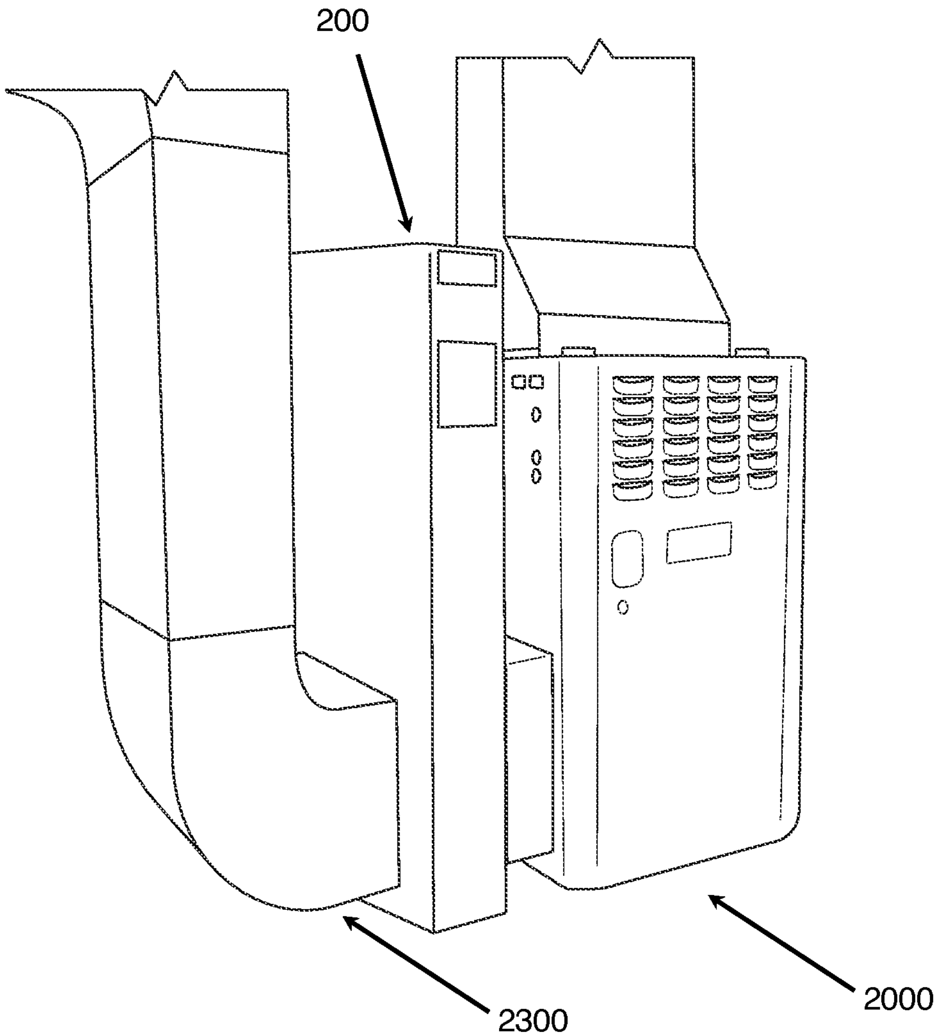

FIGURE 9

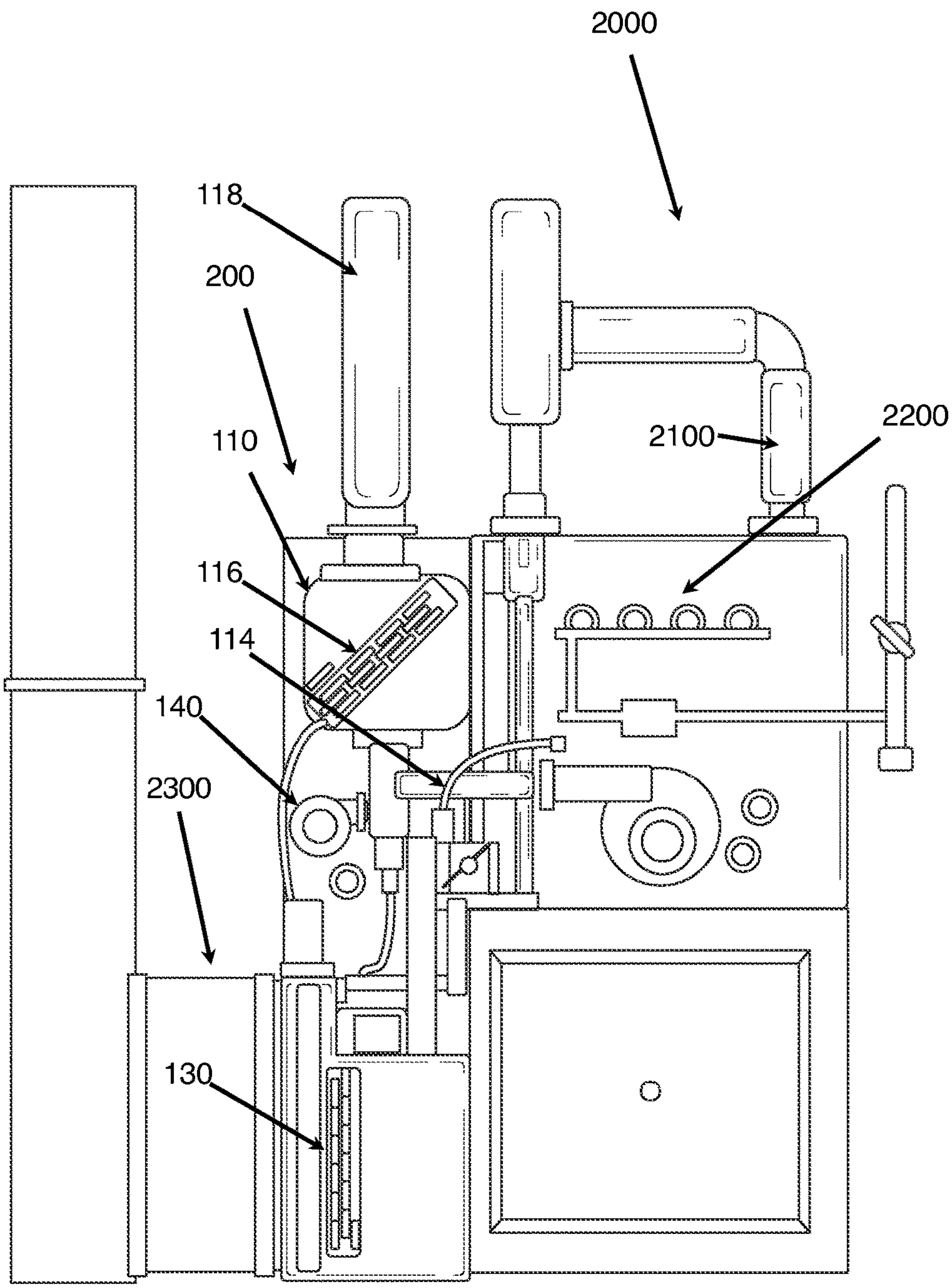

FIGURE 10

HEAT AND ENERGY RECOVERY AND REGENERATION ASSEMBLY, SYSTEM AND METHOD

FIELD OF THE INVENTION

This invention relates generally to the field of air conditioning and heating systems; more particularly, it concerns a system for efficiently combusting fossil fuels for heating a space.

BACKGROUND OF THE INVENTION

The standard methodology used in utilizing fossil fuels for heating is firing the fuel in a controlled heating chamber or heat exchanger. The heat created by the burning fuel is drawn away by air or water flowing around the outside of the heat exchanger. This can be accomplished by blower fans or pumps. The heat is transferred into the surrounding air or water, heating the conditioned space. The waste or emissions from the combustion reaction is allowed to flow outdoors usually utilizing flue piping to a chimney or stack. The efficiency of the furnace or boiler is calculated by the amount of heat which can be extracted from the heat exchanger and utilized to heat the conditioned space and the percentage of heat and by-products permitted to escape through the flue to be vented outside. This rating or efficiency quantification is placed on the furnace or boiler to depict how efficient it will be.

Releasing carbon and heat saturated emissions into the atmosphere contribute to environmental problems, such as global warming. Not only does carbon monoxide and carbon dioxide add to blanketing the release of heat into space, discharging heat through flue gas emissions adds to this issue by heat pollution. Just an average low to medium efficient residential natural gas, LPG or oil furnace can emit one million BTU's of heat waste into the atmosphere each day. Commercial and industrial units can discharge hundreds of millions, and occasionally billions, of BTU's per unit per day. In addition, these common and traditional methods of discharging the flue gas into the atmosphere are wasteful and inefficient.

SUMMARY OF THE INVENTION

In at least one embodiment, the invention is directed to a heat and energy recovery assembly. The present invention is advantageous over traditional HVAC systems in that it produces less greenhouse gases and further utilizes heat that is typically released into the environment. The assembly or apparatus may include a chamber, preferably insulated, comprising an air intake and an emissions intake. The emissions intake is structurally adapted to receive exhaust gas and waste products emitted as a result of fuel combustion. The assembly or apparatus may also include an exhaust for discharging remaining emissions from the chamber.

The chamber additionally includes a primary heat recovery exchanger contained within the chamber, which is in fluid communication with a fluid circuit that includes a primary conduit configured to convey a fluid therein. The primary heat recovery exchanger is disposed within the chamber such that during normal operation when exhaust gas and waste products and air are introduced, it is in thermal communication with the resulting mixture. As a result, heat exchange is effectuated with the fluid inside the exchanger and fluid circuit. A heat extraction exchanger is also in fluid communication with the fluid circuit and primary heat recovery exchanger and disposed in thermal communication with an airstream to be heated, such that heat is transferred from the heat extraction exchanger into the stream of air.

In another embodiment, the invention is directed to a heat and energy recovery system for a furnace. The system includes an insulated chamber comprising an air intake and an emissions intake. The emissions intake is in communication with the furnace exhaust to receive exhaust gas and waste products resulting from fuel combustion in the furnace. The air intake is configured for receiving air from a source of air, such as indoor or outdoor air. A primary heat recovery exchanger is contained within the insulated chamber and is in fluid communication with a fluid circuit that includes a conduit configured to convey a fluid therein. The primary heat exchanger is also configured such that during operation of the furnace it is in thermal communication with a mixture comprising air introduced via the air intake and exhaust gas and waste products introduced via the emissions intake, such that heat exchange is effectuated with the fluid. The system also includes a heat extraction exchanger in fluid communication with the fluid circuit and disposed in thermal communication with an airstream being drawn into the furnace for transferring heat energy from the exchanger to the airstream.

In at least one embodiment, the assembly and system of the present invention may further include a heat recovery ventilator assembly. The assembly provides an outdoor air intake in communication with the heat extraction exchanger such that outdoor air is drawn into the assembly and pushed across the heat extraction exchanger to heat the outdoor air as it is drawn into an air heating apparatus, such as a furnace.

The invention is further directed to a method of recovering heat and energy from fossil fuel combustion waste products. The method includes feeding excess heat and waste products emitted as a result of fuel combustion into an insulated chamber which contains a primary heat recovery exchanger, which contains fluid within, coupled with a fluid containing conduit circuit. The method further includes feeding air into the insulated chamber to initiate a reaction with the waste products that produces a reaction product with potential energy. Furthermore, the method includes effectuating heat energy exchange through the reaction product and excess heat interacting with the primary heat recovery exchanger. As a result, the temperature and reactive pressure within the first fluid-filled heat exchanger and fluid containing conduit circuit rises. Finally, the method includes releasing the heat energy by forced air blowing over a heat extraction exchanger that is in fluid communication with the fluid containing conduit circuit exteriorly of the insulated chamber.

These and other objects, features and advantages of the present invention will become clearer when the drawings as well as the detailed description are taken into consideration.

BRIEF DESCRIPTION OF THE DRAWINGS

For a fuller understanding of the nature of the present invention, reference should be had to the following detailed description taken in connection with the accompanying drawings in which:

FIG. **1** is an illustration of one embodiment of a heat recovery assembly of the present invention.

FIG. **2** is an illustration of another embodiment of a heat recovery assembly of the present invention.

FIG. **3** is an illustration of the functionality of the embodiments of a heat recovery assembly of FIGS. **1** and **2**.

FIG. **4** is an illustration of the heat exchange process utilized in the embodiments of a heat recovery assembly of FIGS. **1** and **2**.

FIG. **5** is an illustration of an embodiment of a heat recovery system of the present invention.

FIG. **6** is an illustration of another embodiment of a heat recovery system utilizing a heat recovery ventilator assembly.

FIG. **7** is an illustration of a wiring diagram of an embodiment of the heat recovery system illustrated in FIG. **5**.

FIG. **8** is a perspective view of an embodiment of the heat recovery system of the present invention.

FIG. **9** is a perspective view of another embodiment of the heat recovery system of the present invention.

FIG. **10** is an illustration of a cut-away view of the embodiment of a heat recovery system of the present invention illustrated in FIG. **9**.

Like reference numerals refer to like parts throughout the several views of the drawings.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

As illustrated in the accompanying drawings, the present invention is directed to a heat and energy recovery assembly and system, in addition to methods of using the same. Such heat recovery devices may be adapted for use in a furnace of an HVAC system or any other system where heat energy from fuel combustion is utilized for heating air spaces.

In one embodiment of the invention, a heat recovery assembly **100** is provided, as illustrated in FIG. **1**. The assembly **100** includes an insulated chamber **110**, or heat recovery box, which comprises an air intake **112** and an emissions intake **114** for receiving exhaust gas and waste products emitted as a result of fuel combustion. The insulated chamber may be made from a variety of metals or alloys. Preferably, the insulated chamber **110** is made of stainless steel and titanium alloy.

The assembly **100** further includes a primary heat recovery exchanger **116** contained within the insulated chamber. The primary heat recovery exchanger **116** is structured for contacting a mixture made up of air introduced via the air intake **112** and exhaust gas and waste products (made up of oxygen starved, carbon emissions) introduced via the emissions intake **114**. A coil sensor may also be in contact with the primary heat recovery exchanger **116** to relay any problems with the functionality of the exchanger to a central logic board (discussed later herein). The primary heat recovery exchanger **116** may be made from a variety of metals and alloys that are ideal for heat exchange, such as but not limited to, copper, aluminum and the like. The exchanger **116** may also be in the form of a hermetically sealed heat recovery coil.

The air intake **112** may be structured as a single intake or multiple intakes. The intake(s) may be adapted to introduce outdoor air, indoor air or both. Additionally, in some embodiments it may be desirable to generate a pressurized environment within the insulated chamber **110**; therefore, one or more of the air intake(s) **112** may connect to a pressure regulator inducer blower **140** (see FIG. **4**) that is part of a pressure equalization system to assist in pressurizing the air inside the insulated chamber **110**. The inducer blower **140** may also be a variable speed motor that is controlled by sensors that detect proper temperature and/or humidity and/or pressure of the air inside the insulated chamber **110**.

The primary heat recovery exchanger **116** is further interconnected to a fluid circuit **120** containing a primary conduit **122** for conveying fluid therein. The assembly **100** may also be interconnected to a heat extraction exchanger **130** exteriorly of the insulated chamber **110** such that the heat extraction exchanger **130** is in fluid communication with the fluid circuit **120** via the primary conduit **122**. The heat extraction exchanger **130** and the primary heat recovery exchanger **116** are collectively interconnected via the primary conduit **122** of the fluid circuit **120** such that the primary heat recovery exchanger **116** contacts (within the insulated chamber **110**) the mixture made up of air introduced via the air intake **112** and exhaust gas and waste products introduced via the emissions intake **114**, while the heat extraction exchanger **130** contacts air to be heated, outside of the insulated chamber **110**.

The insulated chamber **110** additionally comprises exhaust and drainage components. An exhaust **118** for discharging the remaining exhaust gas and waste products after heat exchange occurs is structured to interconnect the insulated chamber **110** to the outside environment. Furthermore, a drain **111** may be connected to the insulated chamber **110** to carry condensate with ash out of the chamber **110**. The drain **111** is particularly necessary when a mister **113** is included in the insulated chamber **110**. A mister **113** is utilized to saturate the air within the insulated chamber **110** with moisture and to help capture and remove particulates and ash soot from the exhaust gases by them becoming saturated with water from the flash heat steam from the super heated oil combustion emissions and falling to the bottom of the chamber to be discharged through the drain **111**. The mister **113** is typically connected to a pressurized water tube to provide water to the insulated chamber **110** to raise the dew point within the chamber **110** to raise the heat transfer potential.

The embodiment illustrated in FIG. **1** is typically designed for use when the input exhaust originates from the burning of cleaner burning propane or other natural gases, such as but not limited to, a natural gas-burning furnace component of a heating, ventilation and air conditioning (HVAC) unit. However, it would be understood by those skilled in the art that the assembly **100** could be utilized in other situations where the surrounding air is to be heating by fossil fuel combustion.

FIG. **2** illustrates an embodiment of the assembly **100** of the present invention that is particularly useful for when the input exhaust originates from an oil-burning furnace; however, this embodiment may be utilized in place of the embodiment illustrated in FIG. **1** for burning natural gas sources as well. The components and configuration of this embodiment are generally the same as in FIG. **1**; however, additional aspects are included for capturing the heat that is stored in the water condensate that accumulates at the bottom of the insulated chamber **110**. The embodiment illustrated in FIG. **2** includes a secondary heat recovery exchanger **117** in fluid communication with the primary heat recovery exchanger **116** via a secondary conduit **124** for absorbing the excess heat stored in the water as it accumulates from the condensate produced by the mister **113** reacting with the mixture of air and hot emissions and soot to produce super heated water droplets, WD. Since the secondary conduit **124** is in communication with the primary heat recovery exchanger **116**, the secondary heat recovery exchanger **117** is further in fluid communication with the fluid circuit **120** as a whole. The drain **111** in FIG. **2** is shown to be structured such that water and condensate ash/soot does not drain from the insulated chamber **110** until the water rises to a certain level, WL. This allows the secondary heat recovery exchanger **117** to remain underneath the surface of the water as it absorbs the excess heat energy stored in the condensate water to ensure that very little, or none, of the heat energy remains unabsorbed in the entire process.

During operation of the assembly and/or system of the present invention, hot emissions (carbon monoxide, carbon dioxide, H_2O, etc.) are exhausted into the insulated chamber **110**. Fresh, outdoor or indoor air is pressurized into the chamber to mix with the emissions. A large, cubic foot print of air is saturated and heated as a result. This mixture flows across the primary heat recovery exchanger **116** while the dew point rises, holding water and heat (saturation). The heat is then extracted from the mixture via the primary heat recovery exchanger **116** (including the secondary heat recovery exchanger **117** if the embodiment illustrated in FIG. **2** is utilized) and transferred to the heat extraction exchanger **130** such that heat transfer occurs to heat indoor air. Cooler, dry air is exported outdoors with a reduced heat, moisture and carbon content. This process allows heat energy to be pulled from the ambient air introduced into the insulated chamber such that it is compounded with the heat energy already being produced by the fossil fuel combustion process. This then gives the assembly and system the potential to achieve a higher efficiency of fuel burn.

By way of example and referring next to FIG. **3**, the overall aspects of an embodiment of the heat recovery assembly **100** are illustrated as utilized in an exemplary 80% annual fuel utilization efficiency (AFUE) furnace rated at 100,000 input/80,000 output. Hot, moist, oxygen-starved carbon emissions are extracted from the furnace at approximately 375° F./90% plus humidity/55 CFM. The hot, water-saturated, oxygen starved carbon emissions carry a large heat potential of at least 20,000 British Thermal Units per Hour (BTUH). In addition to heat energy, the water-saturation of the emissions (misting increases this water-saturation) contains high levels of potential energy for extraction. These emissions are then mixed with an oxygen rich, dry, cool fresh air of equal cubic feet per minute (CFM) using pressure regulation in the insulated chamber **110**. Within the insulated chamber **110** under controlled conditions, the dry, cool, oxygen rich air is saturated by the misted water within the emissions discharge, resulting in an increase in the dew point. The heat energy released in the emissions (approximately 375° F.) mixes with the cool, fresh air, resulting in a mean temperature of approximately 215° F. The oxygen starved emissions are also replenished with O_2, assisting in the heat transfer process. The end result is a warm 215° F./high dew point/high O_2/high-energy potential mixture ideal for high efficiency heat and energy extraction.

This mixture passes over the primary heat recovery exchanger **116**. The fluid, i.e., refrigerant, in the exchanger **116** is under controlled pressurized conditions and is able to extract a large amount of heat energy from the mixture and transfer the heat energy to the heat extraction exchanger **130** via the fluid circuit **120** such that it can be utilized to warm the indoor air. The flow of refrigerant in the fluid circuit **120** between each of the components of the assembly is illustrated by arrows in FIG. **3**. The discharge following the controlled and regulated reaction within the insulated chamber **110** is dry, cool, nearly carbon-free emissions. The average discharge of the resulting emissions is typically 49° F./10% humidity/0.05-0.00 PPM CO (carbon monoxide).

A compressor **150** may be utilized to assist in refrigerant flow between the primary heat recovery exchanger **116** and heat extraction exchanger **130** via the fluid circuit **120**. The heating of cooler refrigerant in the primary heat recovery exchanger **116** during the operation of the assembly **100** results in a pressure increase inside the exchanger and the fluid circuit **120**, resulting in heat-absorbed refrigerant being pushed to an area of lower pressure (see FIG. **4**). This pushing phenomena allows a large part of refrigerant flow in the circuit (approximately 50%) to be achieved without any compressor assistance, limiting the amount of electrical energy required; therefore, a large compressor may not be necessary in most embodiments of the assembly **100** to get sufficient refrigerant flow. As such, a micro-compressor is preferably utilized in embodiments of the invention to further provide energy conservation.

The assembly **100** could be adapted to attach to any age furnace with about 78% AFUE or higher efficiency, resulting in an increased efficiency of the system. Carbon discharge, emission temperature, and humidity may also be reduced if the assembly **100** is utilized with a furnace.

Referring next to FIG. **5** and FIG. **10**, a heat recovery system **200** is illustrated. The system **200** includes a furnace **2000** comprising an exhaust **2100** and a furnace intake **2300**. The system **200** further includes an insulated chamber **110** comprising an air intake **112** and an emissions intake **114**. The emissions intake **114** is adapted to be in communication with the exhaust **2100** of the furnace to receive exhaust gas and waste products resulting from fuel combustion in the furnace **2000**. A primary heat recovery exchanger **116** is contained within the insulated chamber **110** and is in fluid communication with a fluid circuit **120** that includes a primary conduit **122** configured to convey a fluid therein, such as a refrigerant. The primary heat recovery exchanger **116** is also configured such that during operation of the furnace **2000** it is in thermal communication with a mixture comprising air introduced via the air intake **112** and exhaust gas and waste products introduced via the emissions intake **114** that is connected to the furnace exhaust **2100**.

The system **200** also includes a heat extraction exchanger **130** in fluid communication with the fluid circuit **120** and disposed in thermal communication with an airstream being drawn into the furnace for heating (see INDOOR AIR passing through the heat extraction exchanger **130** in FIG. **5**). Refrigerant is heated in the primary heat recovery exchanger **116** and moved to the heat extraction exchanger **130** via the pressure gradient created by the heat exchange and, optionally, with assistance from a micro-compressor or the like, where heat exchange occurs between the airstream flowing from the indoor air source and the heat extraction exchanger **130**. The pre-heated air is directed into the heat exchanger **2200** of the furnace such that the air is further heated and then directed into the home or other structure in need of being heated.

Additionally, the system **200** further includes a drain **111** exiting the insulated chamber **110**. The drain **111** may be structured as in FIG. **1** or FIG. **2**, depending on the type of furnace being utilized in the system **200** (as explained previously herein). As such, a system **200** utilizing the drain **111** as illustrated in FIG. **2** would further include a secondary heat recovery exchanger **117** as previously described herein.

The system **200** may also utilize a compressor **150**, as previously described herein. It is preferable that the compressor is a micro-compressor to further aid in energy conservation. It is also contemplated that a furnace inducer blower, IB, may be in connection with the furnace exhaust **2100** to actively draw exhaust from the furnace **2000** into the emission intake **114** of the insulated chamber **110**.

The assembly **100** and system **200** of the present invention may further utilize a heat recovery ventilator. Heat recovery ventilators have been a known art in the HVAC industry for many years; however, the typical ventilator is much less efficient and structurally different than the embodiment disclosed in the present invention in combination with the assembly and system herein. A conventional Heat Recovery Ventilator (HRV) draws in fresh outdoor air to replace exhausted indoor air. The HRV helps create air exchanges within home or building structures which in turn helps to reduce pollutants, smoke, contaminants, airborne allergies, viruses, etc. from collecting within the home or building ventilation systems. During the air exchange process of a ventilator, fans and heat exchangers will pass heated or cooled indoor air over unconditioned outdoor air. The two air masses never combine but are separated by heat exchangers. This process can transfer as much as 85% of the heat energy from the conditioned air mass to the unconditioned air mass. About 15% of the energy is lost in this process, causing the home or building owner the expense of heating or air conditioning that loss to the newly introduced unconditioned air in order to maintain the same comfort level within the structure.

FIG. **6** shows a heat recovery ventilator (HRV) assembly **160** configured in relation to a heat recovery assembly **100** for providing fresh outdoor air to the interior environment. The HRV contains a ventilator outdoor air intake **162** that is structured to be in communication with the heat extraction exchanger **130** for heating outdoor air as it is drawn into the supply air intake of a heating apparatus or furnace. The HRV provides clean, outdoor air for circulation within the home or building. It directs the air into the airstream being drawn across the heat extraction exchanger **130** such that it can be heated by the energy efficient process utilized in the heat recovery assembly **100** or system **200**, as previously described herein. The HRV assembly **160** may further include a motorized damper **164** in communication with the outdoor air intake **162** such that the flow of outdoor air is regulated. A thermostat **166** may be in communication with the motorized damper **164** for controlling the opening and closing of the damper **164** based on the outdoor air temperature. Typically, the damper **164** allows air temperatures ranging from about 10 degrees Fahrenheit to about 70 degrees Fahrenheit to pass therethrough. The thermostat **166** utilizes a temperature sensor **168** to communicate the outside air temperature.

FIG. **7** illustrates a typical electrical wiring diagram of a heat and energy recovery system as provided in the present invention. The diagram illustrates the connections between a logic board **170** of the system and the furnace board and thermostat of a typical HVAC system. A LCD scroll display **171** is provided for a visual depiction of the operational parameters of the system. Heat recovery, troubleshooting, and normal operating conditions are indicated by LED lights. Various connections between sensors and switches (e.g., low and high pressure switches) are also depicted. The inducer blower and micro-compressor connections and requisite relays are also depicted. Connections between all components of the system are also wired with the logic board **170** to provide centralized control and functionality of the system.

FIGS. **8** and **9** illustrate typical operational embodiments of the heat and energy recovery system **200** with a furnace **2000**. As shown, the system **200** may be adapted to fit on the furnace unit either on a wall of the unit (FIG. **8**) or inline with the air intake of the furnace (FIG. **9**). A cut-away illustration is shown in FIG. **10** in an embodiment where the system **200** is structured to be inline with the furnace intake **2300** for receiving air as it is drawn into the furnace **2000**.

The invention is further directed to a method of recovering heat and energy from fuel combustion. The method includes feeding excess heat and waste products emitted as a result of fuel combustion into an insulated chamber which contains a primary heat recovery exchanger (fluid filled) coupled with a fluid containing conduit circuit. Typically, the fluid comprises a refrigerant. The method further includes feeding air into the insulated chamber to initiate a reaction with the waste products that produces a reaction product with potential energy. Furthermore, the method includes effectuating heat energy exchange through the reaction product and excess heat interacting with the primary heat recovery exchanger. As a result, the temperature and reactive pressure within the primary heat recovery exchanger and fluid containing conduit circuit rises. Finally, the method includes releasing the heat energy by forced air blowing over a heat extraction exchanger that is in fluid communication with the fluid containing conduit circuit exteriorly of the insulated chamber.

Since many modifications, variations and changes in detail can be made to the described embodiments of the invention, it is intended that all matters in the foregoing description and shown in the accompanying drawings be interpreted as illustrative and not in a limiting sense. Thus, the scope of the invention should be determined by the appended claims and their legal equivalents.

Now that the invention has been described,

What is claimed is:

1. A heat recovery assembly, the assembly comprising:

an insulated chamber comprising an air intake, an emissions intake, and an exhaust, the emissions intake for receiving exhaust gas and waste products emitted as a result of fuel combustion and the exhaust for discharging remaining emissions from the insulated chamber, the insulated chamber having a bottom surface over which residual water accumulates from a mist therein;

a primary heat recovery exchanger containing a fluid therein, the primary heat recovery exchanger contained within the insulated chamber for contacting a mixture comprising air introduced via the air intake and exhaust gas and waste products introduced via the emissions intake such that heat exchange is effectuated with the fluid;

a fluid circuit comprising a primary conduit in fluid communication with the primary heat recovery exchanger;

a heat extraction exchanger in fluid communication with the primary heat recovery exchanger via the fluid circuit and for effectuating heat exchange with an airstream running therethrough; and

a secondary heat recovery exchanger within the insulated chamber in fluid communication with the primary heat recovery exchanger, the secondary heat recovery exchanger extending over the bottom surface of the insulated chamber and configured to remain at least partially covered by the accumulated residual water.

2. The assembly of claim **1**, wherein the air intake is a pressure regulator inducer blower for providing pressurized air into the insulated chamber.

3. The assembly of claim **1**, further comprising a mister for providing the mist of water into the insulated chamber.

4. The assembly of claim **1**, wherein the fluid conveyed via the fluid circuit comprises a refrigerant.

5. The assembly of claim 1, wherein the primary heat recovery exchanger includes a hermetically sealed coil for effectuating heat exchange with the fluid therein.

6. The assembly of claim 1, further comprising a ventilator outdoor air intake in communication with the heat extraction exchanger for heating outdoor air that passes over the heat extraction exchanger.

7. A heat recovery system, comprising:

a furnace comprising an exhaust and a furnace intake;

an insulated chamber comprising an air intake and an emissions intake, the emissions intake in communication with the exhaust of the furnace for receiving exhaust gas and waste products resulting from fuel combustion and the air intake configured for receiving air from a source of air, the insulated chamber having a bottom surface over which residual water accumulates from a mist therein;

a fluid circuit including a primary conduit configured to convey a fluid therein;

a primary heat recovery exchanger contained within the insulated chamber, the primary heat recovery exchanger in fluid communication with the fluid circuit and configured for thermal communication with a mixture comprising air introduced via the air intake and exhaust gas and waste products introduced via the emissions intake such heat exchange is effectuated with the fluid;

a heat extraction exchanger in fluid communication with the fluid circuit and disposed in thermal communication with an airstream being drawn into the furnace intake for transferring heat energy from the heat extraction exchanger to the airstream; and

a secondary heat recovery exchanger within the insulated chamber in fluid communication with the primary heat recovery exchanger, the secondary heat recovery exchanger extending over the bottom surface of the insulated chamber and configured to remain at least partially covered by the accumulated residual water.

8. The heat recovery system of claim 7, wherein the air intake is a pressure regulator inducer blower for providing pressurized air into the insulated chamber.

9. The heat recovery system of claim 7, further comprising a mister for providing the mist of water into the insulated chamber.

10. The heat recovery system of claim 7, wherein the fluid conveyed via the fluid circuit comprises a refrigerant.

11. The heat recovery system of claim 7, wherein the primary heat recovery exchanger includes a hermetically sealed coil for effectuating heat exchange with the fluid therein.

12. The heat recovery system of claim 7, further comprising a ventilator outdoor air intake in communication with the heat extraction exchanger for heating outdoor air as it is drawn into the furnace.

13. The heat recovery system of claim 12, further comprising a motorized damper in communication with the ventilator outdoor air intake such that the flow of outdoor air is regulated.

14. The heat recovery system of claim 13, further comprising a thermostat for controlling the motorized damper based on the outdoor air temperature.

15. A method of recovering heat and energy using the assembly of claim 1, the method comprising:

feeding excess heat and waste products emitted as a result of fuel combustion into the insulated chamber;

feeding air into the insulated chamber for initiating a reaction with the waste products to produce a reaction product with potential energy;

effectuating heat energy exchange through the reaction product and excess heat interacting with the primary heat recovery exchanger, whereby the temperature and reactive pressure of the fluid within the primary heat recovery exchanger and conduit circuit rises; and

extracting heat from the residual water that accumulates from the mist via the secondary heat recovery exchanger within the insulated chamber;

releasing the heat energy by forcing air over a heat extraction exchanger that is in fluid communication with the fluid containing conduit circuit exteriorly of the insulated chamber.

* * * * *

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Kaiser

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(45) Date of Patent: Jan. 26, 2021

(54) **RECOVERABLE AND RENEWABLE HEAT RECOVERY SYSTEM AND RELATED METHODS**

(71) Applicant: **Commercial Energy Saving Plus, LLC**, Stroudsburg, PA (US)

(72) Inventor: **Stewart Kaiser**, Stroudsburg, PA (US)

(73) Assignee: **COMMERCIAL ENERGY SAVING PLUS, LLC**, Boca Raton, FL (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 145 days.

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(22) Filed: **Nov. 2, 2018**

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(51) **Int. Cl.**

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F24S 10/70	(2018.01)
F24S 70/20	(2018.01)
F24S 80/30	(2018.01)
F24S 20/20	(2018.01)
F24S 80/00	(2018.01)

(52) **U.S. Cl.**

CPC ***F24S 20/30*** (2018.05); ***F24S 10/70*** (2018.05); ***F24S 20/20*** (2018.05); ***F24S 70/20*** (2018.05); ***F24S 80/30*** (2018.05); *F24S 2080/03* (2018.05)

(58) **Field of Classification Search**

CPC F24S 20/30
USPC 126/714, 585, 569
See application file for complete search history.

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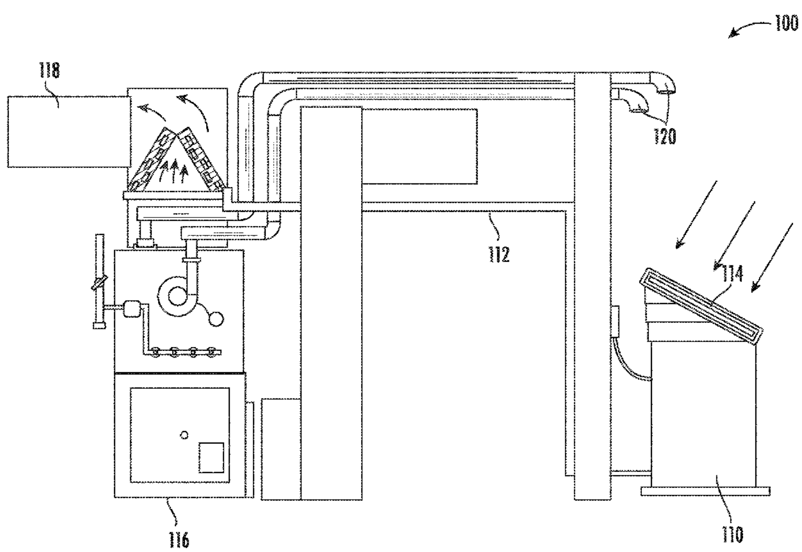
Primary Examiner — Avinash A Savani
(74) *Attorney, Agent, or Firm* — Allen Dyer Doppelt & Gilchrist, PA

(57) **ABSTRACT**

A recoverable and renewable heat recovery system includes a variable speed inverter compressor in fluid connection with a first heat exchanger and a second heat exchanger via a fluid circuit. The system further includes a solar thermal collection module positioned on top of the compressor and in fluid communication with the compressor, the first heat exchanger and the second heat exchanger via the fluid circuit. A light intensity sensor is configured to determine light intensity on the solar thermal collection module. The solar thermal collection module is configured to retain solar energy thermal energy to increase fluid pressure in the compressor.

21 Claims, 11 Drawing Sheets



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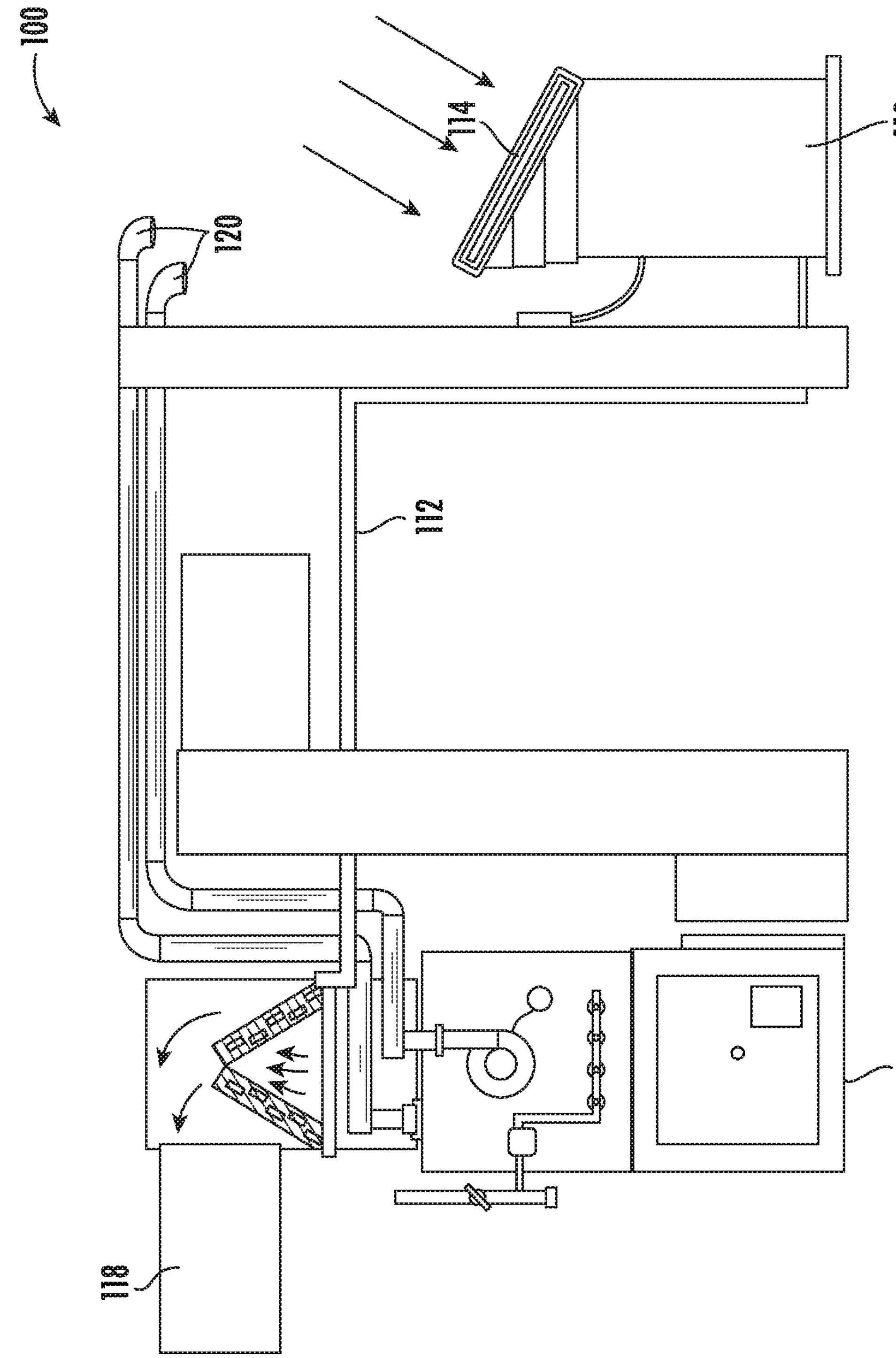


FIG. 1

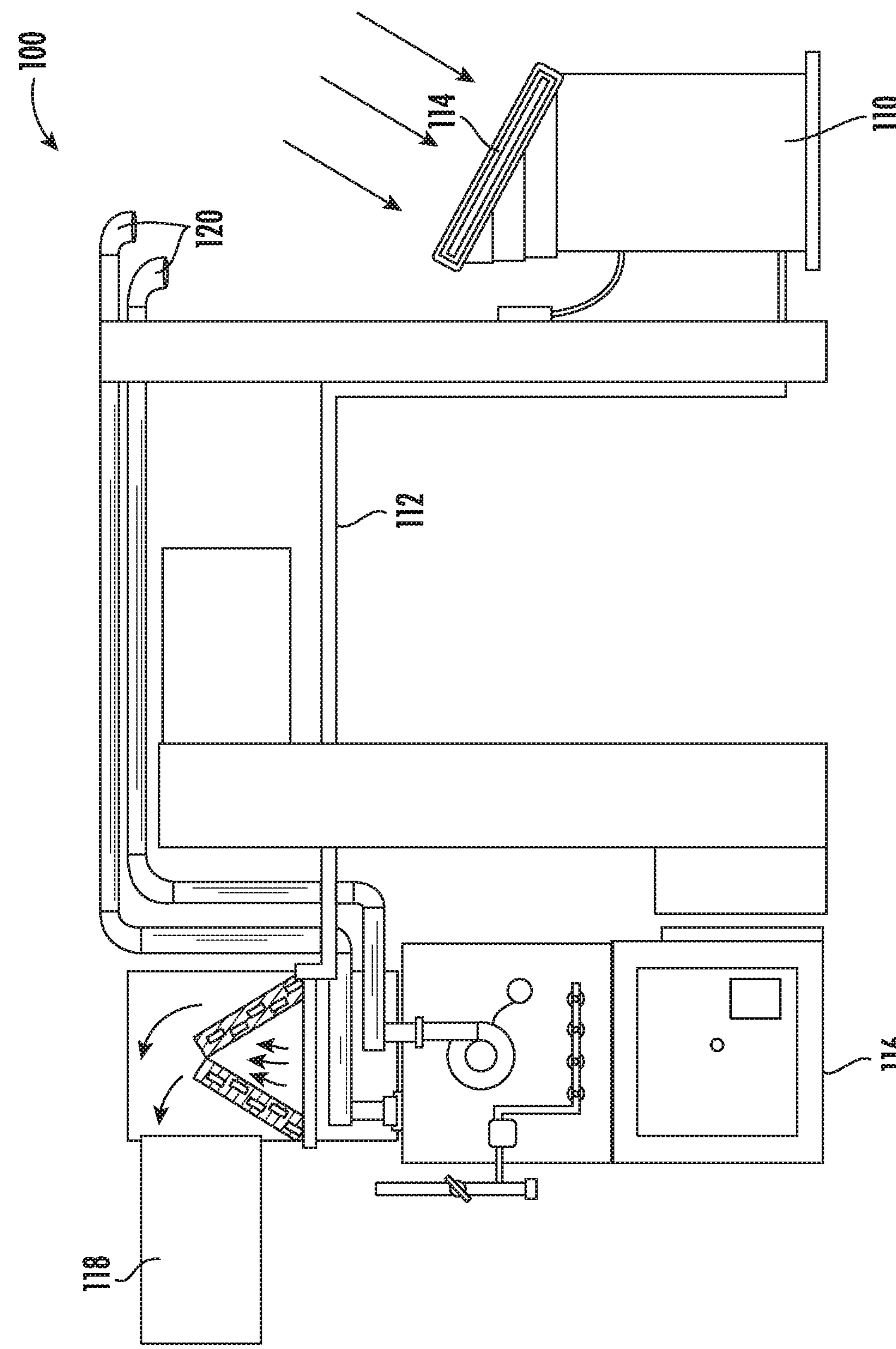


FIG. 2

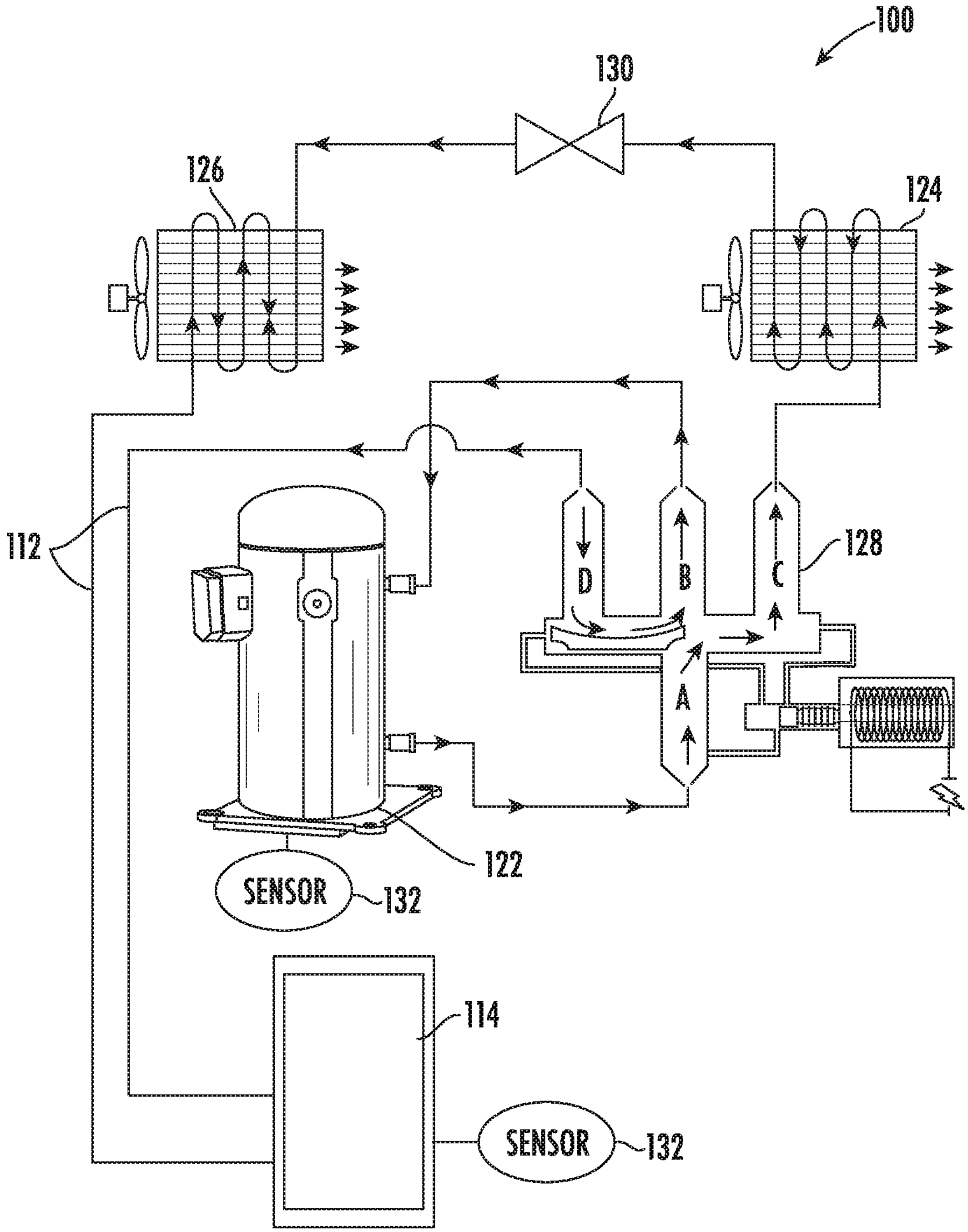


FIG. 3

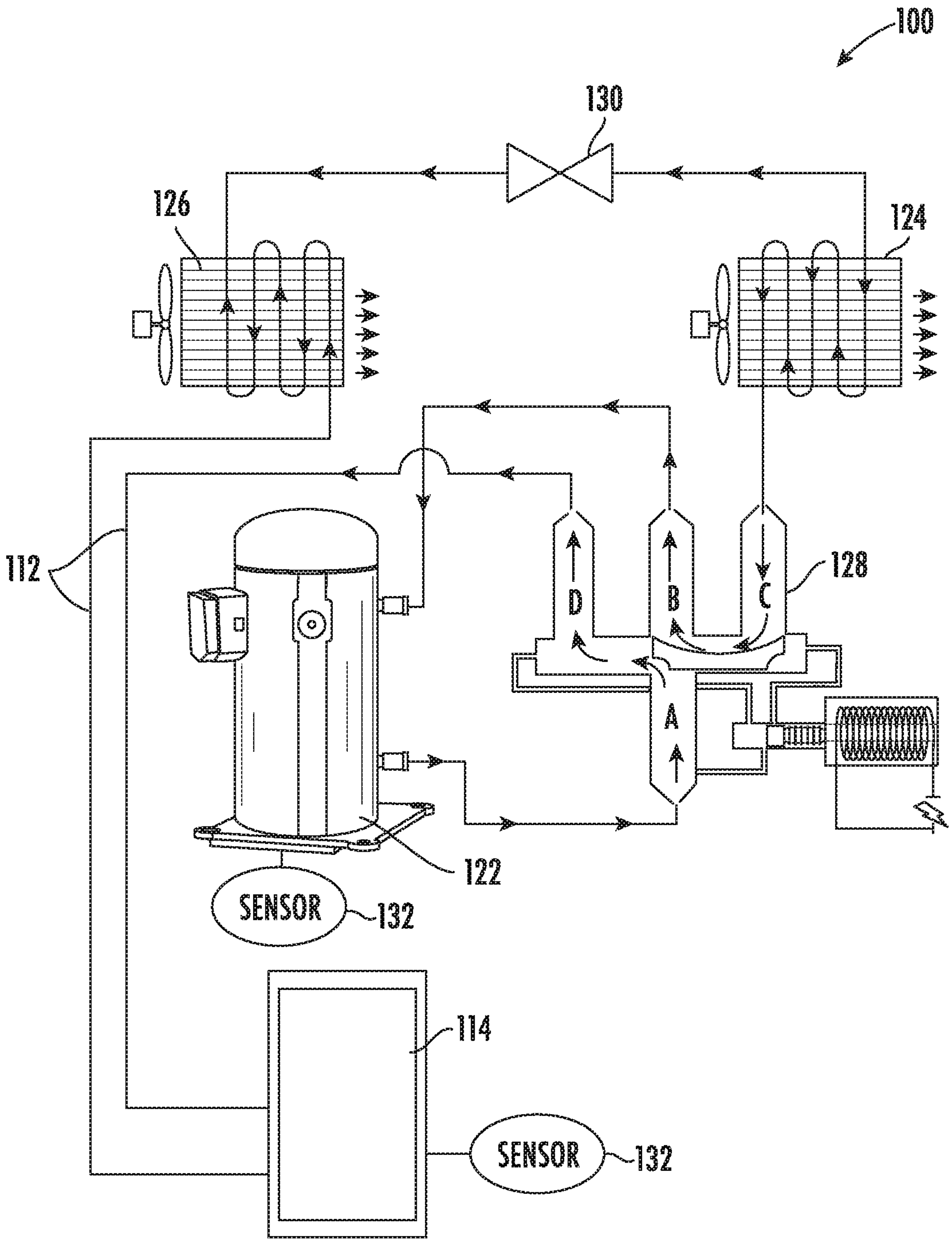
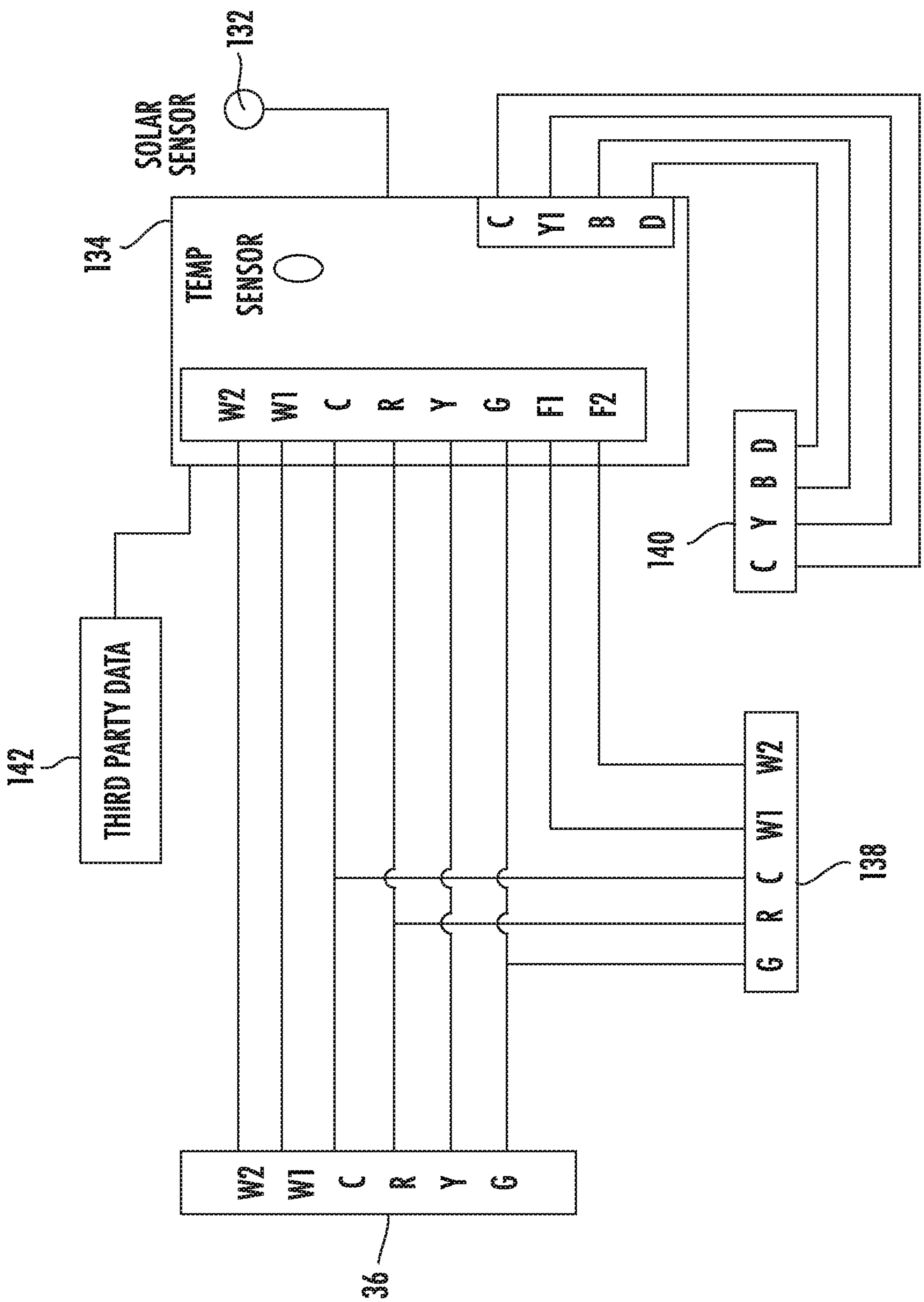


FIG. 4



SOLAR SENSOR
132
134
TEMP SENSOR
C Y1 B D
W2 W1 C R Y G F1 F2
140
C Y B D
THIRD PARTY DATA
142
W2 W1 C R G
138
W2 W1 C R Y G
136

FIG. 5

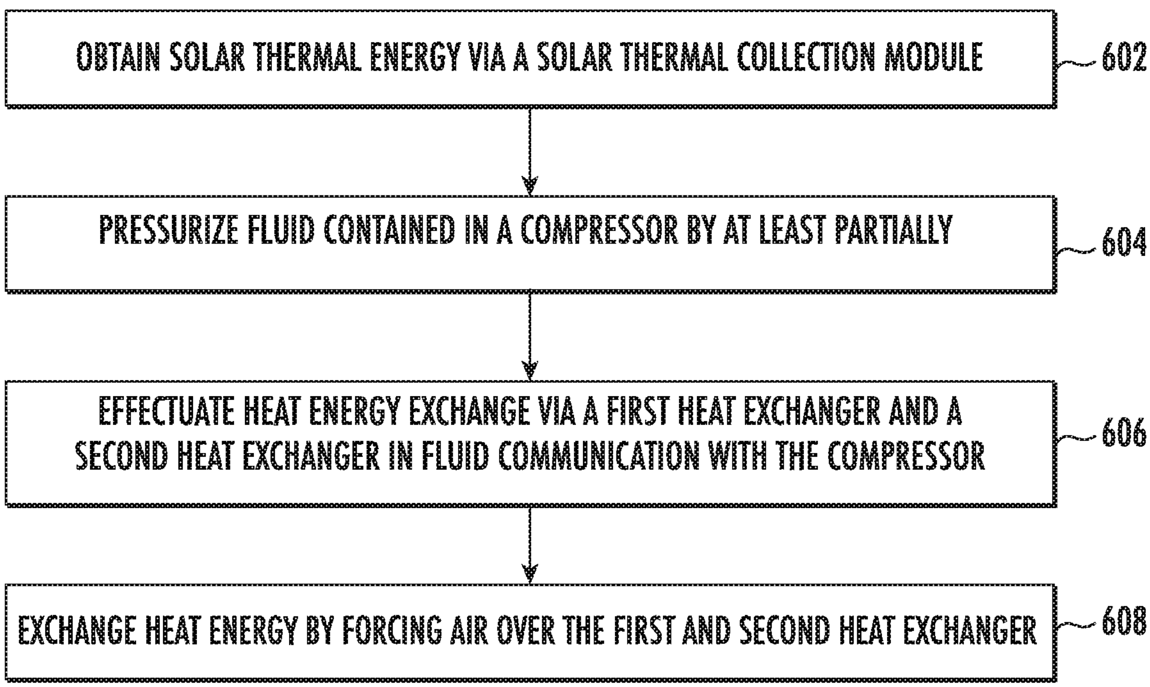

OBTAIN SOLAR THERMAL ENERGY VIA A SOLAR THERMAL COLLECTION MODULE
602
PRESSURIZE FLUID CONTAINED IN A COMPRESSOR BY AT LEAST PARTIALLY
604
EFFECTUATE HEAT ENERGY EXCHANGE VIA A FIRST HEAT EXCHANGER AND A SECOND HEAT EXCHANGER IN FLUID COMMUNICATION WITH THE COMPRESSOR
606
EXCHANGE HEAT ENERGY BY FORCING AIR OVER THE FIRST AND SECOND HEAT EXCHANGER
608

FIG. 6

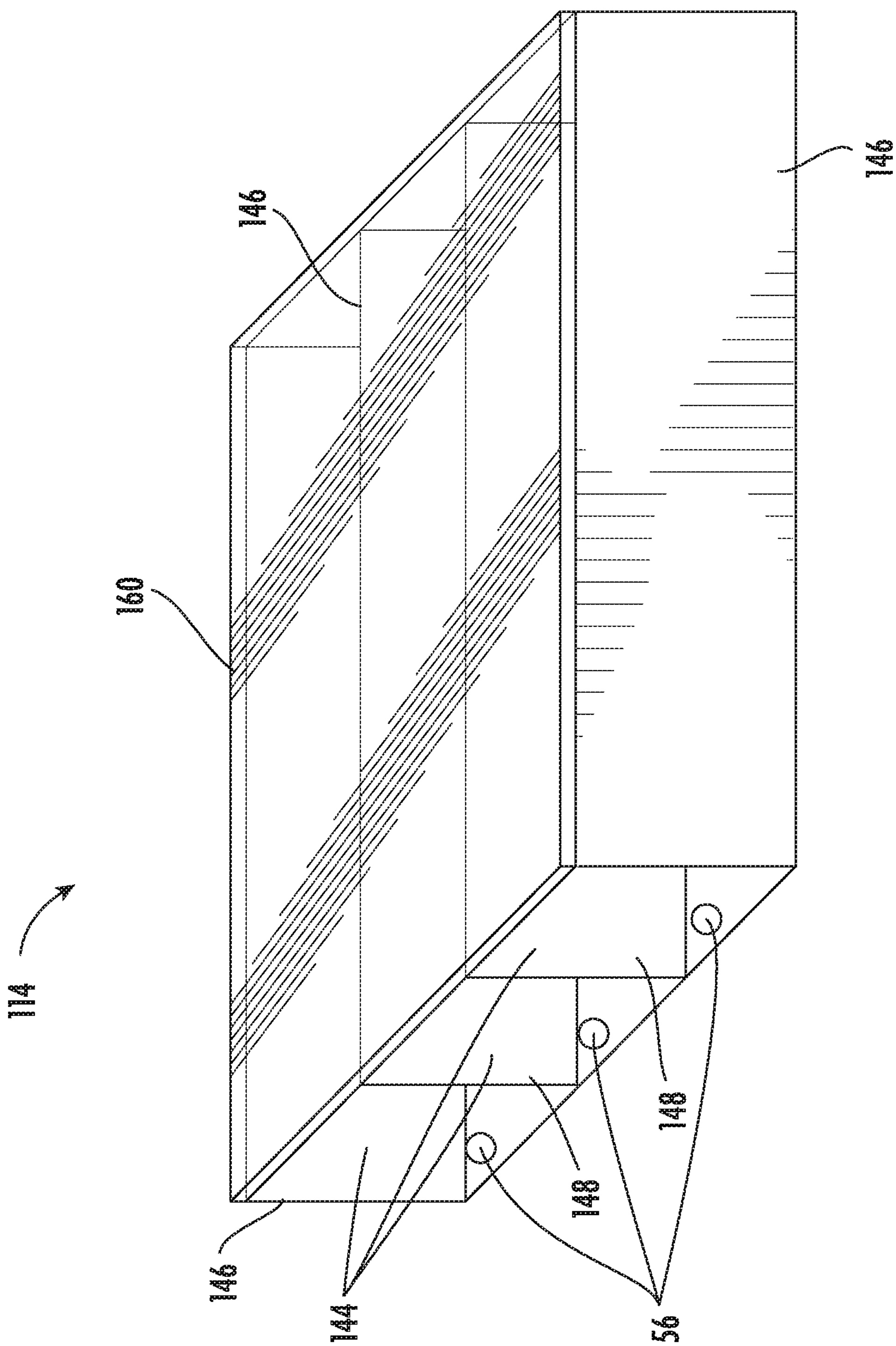


FIG. 7

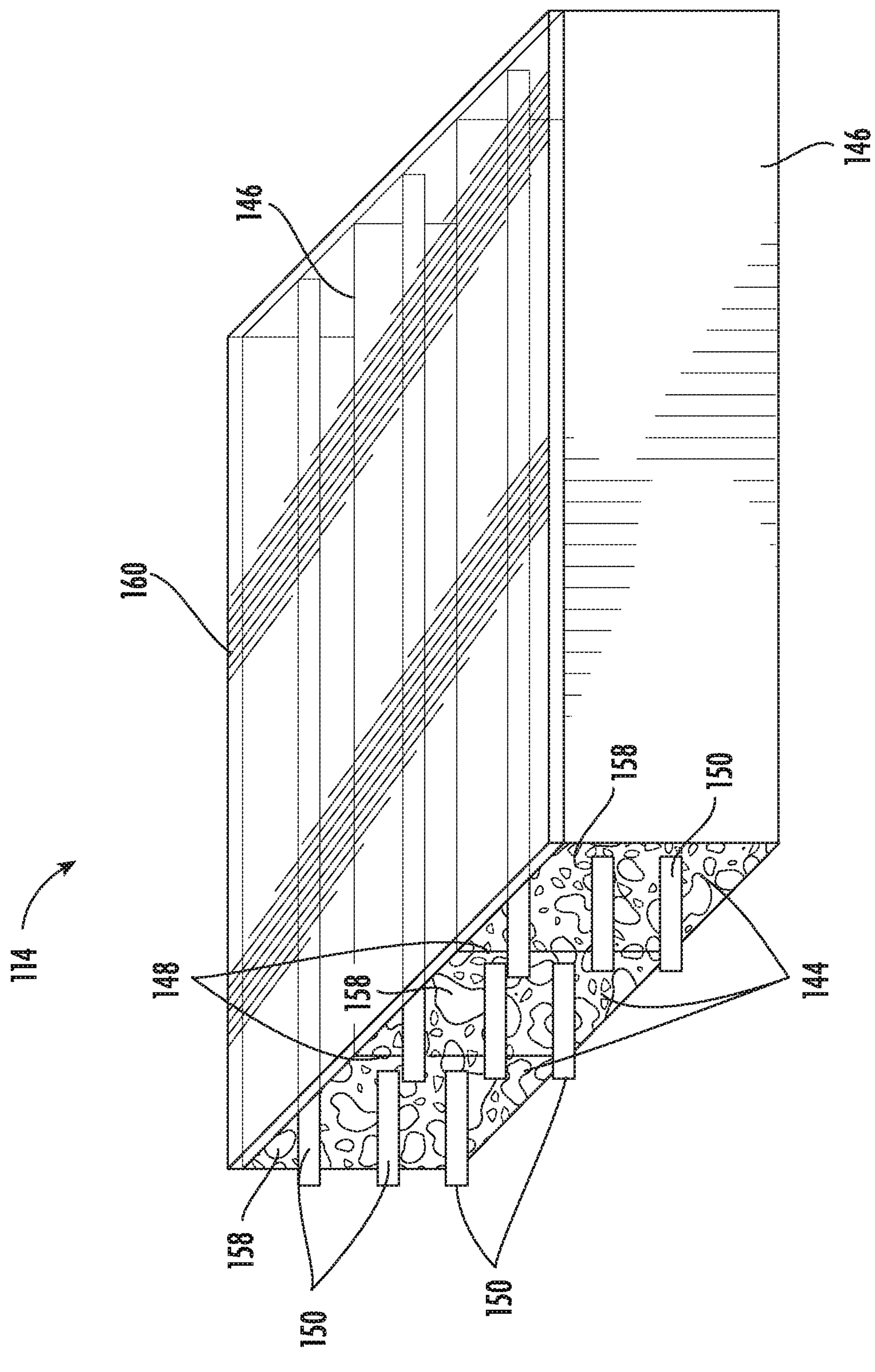


FIG. 8

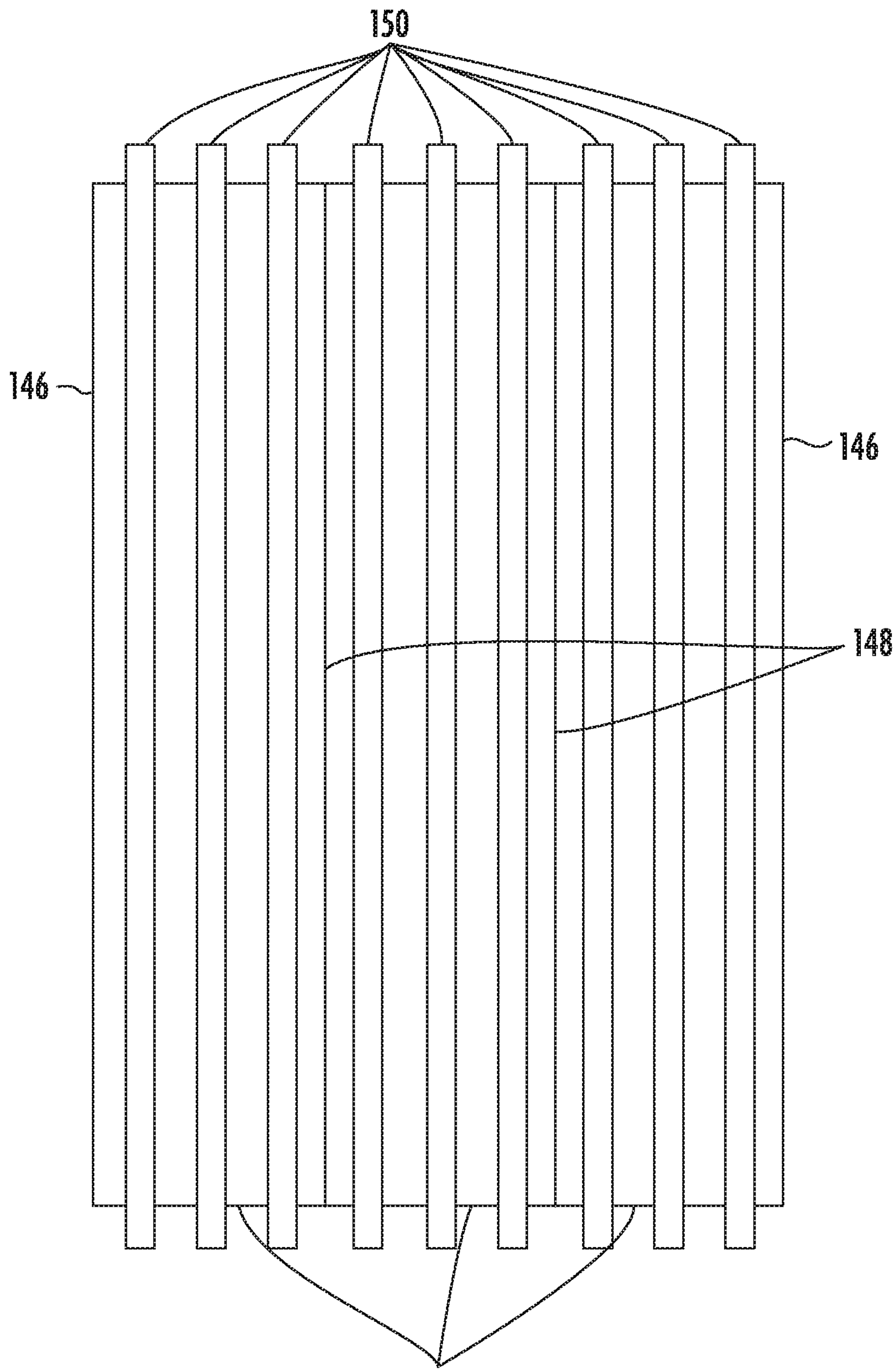


FIG. 9

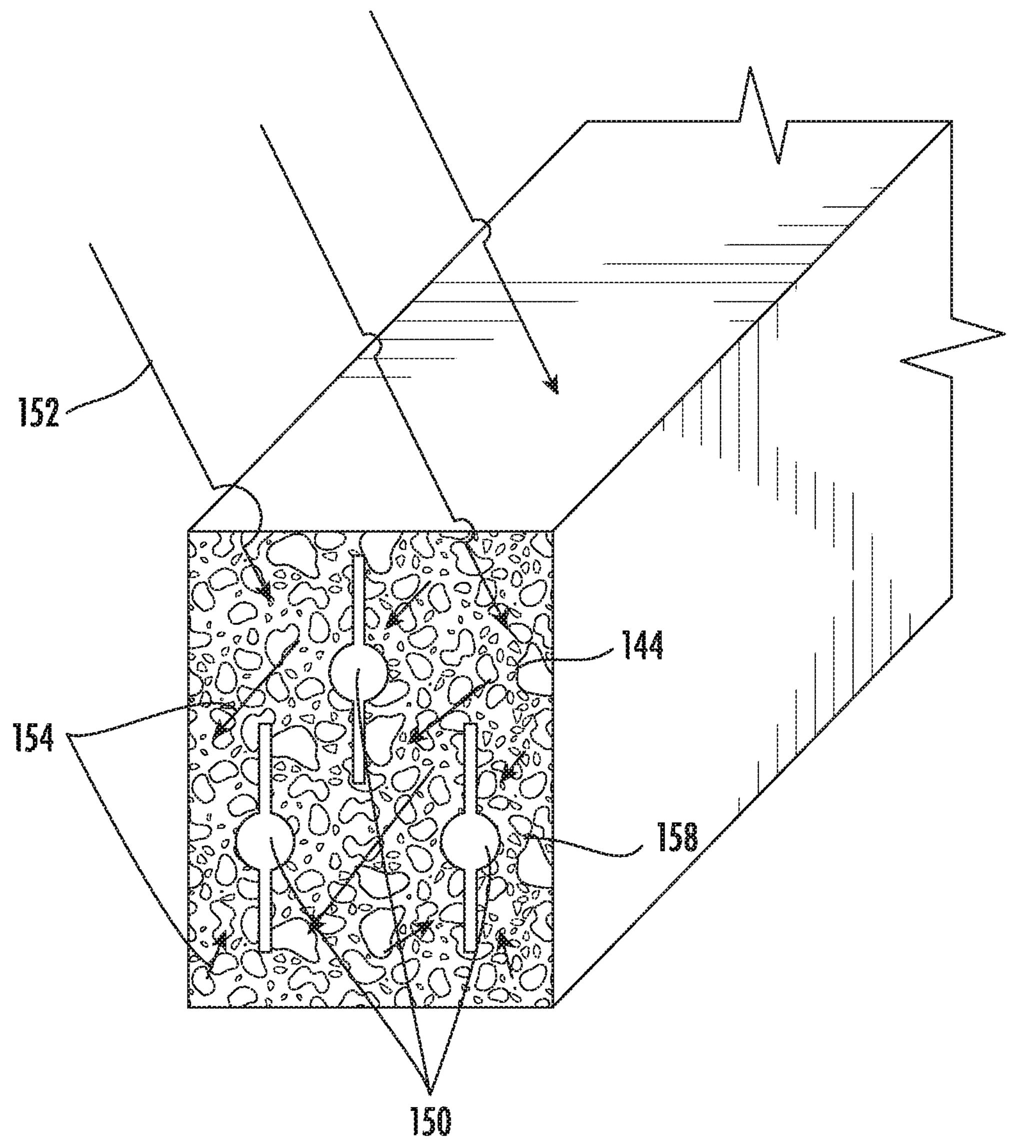


FIG. 10

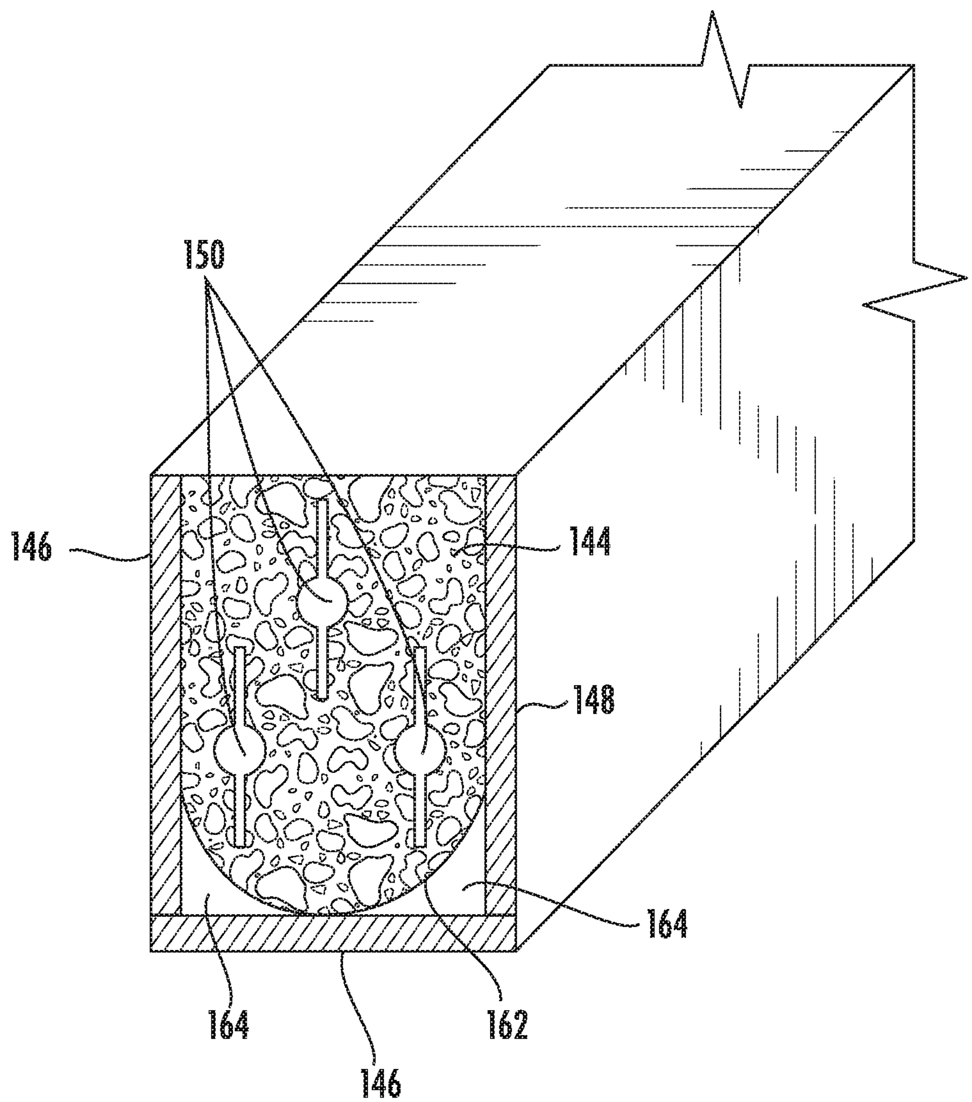


FIG. 11

RECOVERABLE AND RENEWABLE HEAT RECOVERY SYSTEM AND RELATED METHODS

CROSS-REFERENCE TO RELATED APPLICATION

This application claims the benefit of U.S. Provisional Patent Application Ser. No. 62/747,186, filed on Oct. 18, 2018, the contents of which are herein incorporated by reference in their entirety.

FIELD OF THE INVENTION

This invention relates generally to the field of air conditioning and heating systems; more particularly, a recovery and renewable heat exchange system using solar thermal energy.

BACKGROUND OF THE INVENTION

In a heat recovery system, a heat pump typically pulls heat out of the outdoor environment and transfers that thermal energy into a building, home or structure in a heating cycle. The efficiency of the heating cycle is too low to justify use when the outdoor temperature is below about 32° F. In this case, an auxiliary heating is needed to keep the building warm. This is typically achieved with electrical resistance coils, which are very inefficient in terms of energy conversion and expensive to operate.

A heat pump cannot take the place of a conventional gas-fired furnace if the temperature remains below freezing for long. In locations where this is common, gas-fired furnaces are used in a combustion cycle to produce large amounts of thermal energy. Such furnaces also exhaust greenhouse gases into the environment. Highly efficient condensing furnaces produce large amounts of acidic residue, which is pumped into drains and the soil.

Solar energy obtained from photovoltaic solar panels has been used to reduce electrical consumption in a heat recovery system. In the winter, however, ice and snow can cover solar panels, disabling them or making them much less efficient. Solar panels generally require a large surface area to accomplish certain goals. It can be difficult to have a large surface area of solar panels in an urban environment or wherever there is a high density of buildings. In addition, conventional solar panels are expensive and require many years of operation for the cost of materials and installation to be outweighed by savings on utility bills. Many conventional solar panels require direct sunlight for maximum performance, which is typically limited to a few hours each day of total sunshine, and on cloudy or partly sunny days, performance is drastically reduced. The deficiencies and limitations suggest that further improvements can be realized for a heat recovery system.

SUMMARY OF THE INVENTION

In view of the foregoing, it is an object of the present invention to provide a recoverable and renewable heat recovery system and related methods. According to one embodiment of the invention, a recoverable and renewable heat recovery system includes a variable speed inverter compressor in fluid connection with a first heat exchanger and a second heat exchanger via a fluid circuit. The system further includes a solar thermal collection module positioned on top of the compressor and in fluid communication with the compressor, the first heat exchanger and the second heat exchanger via the fluid circuit. A light intensity sensor is configured to determine light intensity on the solar thermal collection module. The solar thermal collection module is configured to retain solar energy thermal energy to increase fluid pressure in the compressor. Compressor operation is based on measurement of the light intensity sensor.

According to another embodiment of the invention, a method of recovering heat energy includes obtaining solar thermal energy via a solar thermal collection module and pressurizing fluid contained in a compressor by at least partially utilizing solar thermal energy from the thermal collection module. Heat energy exchange is effectuated via a first heat exchanger and a second heat exchanger in fluid communication with the compressor, and heat energy is exchanged by forcing air over the first and second heat exchanger.

According to another embodiment of the present invention, a solar thermal module includes a plurality of solar thermal cell chambers positioned in parallel and covered by tempered glass and a plurality of interconnected fluid pipes positioned through each solar thermal chamber. A layer of reflective materials covers the inner sidewall of each solar thermal cell chamber. Each cell chamber is filled with a foam material to retain solar thermal energy. One or more drain holes are located on a bottom surface of each cell chamber for draining condensing liquid and avoiding moisture buildup.

These and other objects, aspects and advantages of the present invention will be better appreciated in view of the drawings and following detailed description of preferred embodiments.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** illustrates an example heat recovery system in a cooling cycle utilizing solar thermal energy, according to an embodiment of the present invention;

FIG. **2** illustrates an example heat recovery system in a heating cycle utilizing solar thermal energy, according to another embodiment of the present invention;

FIG. **3** illustrates a detailed view of an example heat recovery system in a heating cycle, according to an embodiment of the present invention;

FIG. **4** illustrates another detailed view of an example heat recovery system in a cooling cycle, according to another embodiment of the present invention;

FIG. **5** is a central heat recovery unit incorporating a solar intensity sensor, according to another embodiment of the present invention;

FIG. **6** is a flow chart illustrating a method of heat exchange, according to another embodiment of the present invention;

FIG. **7** is a perspective view of an example solar thermal collection module, according to another embodiment of the present invention;

FIG. **8** is another perspective view of the solar collection module of FIG. **7**;

FIG. **9** is a front view of the solar collection module of FIG. **7**; and

FIG. **10** is a detailed view of a solar collection cell.

FIG. **11** is a cross sectional view of another embodiment of FIG. **10** of the present invention of a metal layer.

DETAILED DESCRIPTION OF A PREFERRED EMBODIMENT

According to an embodiment of the present invention, and referring to FIGS. **1-2**, the system **100** includes a heat pump

110 having a variable speed inverter compressor (not shown) in fluid connection with a first heat exchanger and a second heat exchanger (not shown) via a fluid circuit **112**. The compressor is configured to assist fluid (e.g., refrigerant) flow within the fluid circuit **112**. In the depicted embodiment, the heat pump **110** is placed outdoors and integrated with a solar thermal collection module **114** positioned thereon. A furnace **116** is used in addition to the heat pump **110** in under high heating demand. The furnace **116** can be an oil-burning furnace or other type of furnace, e.g. one that burns natural gas. Air is supplied to the inside of a building via a supply duct **118**. The exhaust gas from the burning furnace **116** is released to the external environment via the exhaust gas outlet **120**.

The solar thermal module **114** is configured to utilize solar thermal energy in the environment to increase fluid pressure inside the compressor of the heat pump **110**. Solar thermal energy can dramatically increase the fluid temperature within the compressor, and the temperature can exceed 500° F. under certain conditions. Capture of solar thermal energy can be used to achieve considerable fluid (e.g., refrigerant) pressurization with little mechanical work, limiting the electrical energy required. A high-powered compressor may not be necessary in most embodiments of the system **100** for sufficient fluid flow. The compressor of the heat pump **110** can operate only as needed and at a speed as needed. In the cooling mode, the solar thermal energy collected by the solar thermal collection module **114** can save up to 60% of the electrical energy that would be used in summer time. In essence the system allows the heat absorbed via the solar thermal module **114** to increase the pressure of the refrigerant instead of requiring the compressor of the heat pump **110** to achieve a same amounting of compressing utilizing mechanical performance. This allows the system **100** to maintain the same performance of heat transfer using much less electrical energy.

In a preferred embodiment, the solar thermal collection module **114** is placed on top of the heat pump **110** for minimum fluid transportation length and maximum exposure to sun light. The solar thermal collection module **114** is preferably installed facing south in the northern hemisphere for maximum sunlight absorption. The solar thermal collection module **114** and the heat pump **110** can be made as an integral piece.

FIGS. **3-4** illustrate detailed fluid communication between a compressor **122**, a solar thermal collection module **114**, a first heat exchanger **124** and a second heat exchanger **126**, and a reversing valve **128** via a fluid circuit **112**.

Referring to FIG. **3**, in the heating mode of a refrigeration cycle, refrigerant flows through the solar thermal collection module **114** after returning from an evaporator coil of the first heat exchanger **124**. When refrigerant flows through the solar thermal collection module **114**, refrigerant temperature and pressure increase. The pressurized vapor condenses at high pressure and temperature inside a condensing coil of the second heat exchanger **126**. During the condensation of refrigerant, the heat released is used to heat the air supplied to a HVAC system. Condensed refrigerant is then transported to the evaporation coil of the first heat exchanger **122**, which lowers the pressure. The refrigerant then goes on to another circle. Refrigerant flow in the fluid circuit **112** between each of the components of the system is illustrated by arrows in FIG. **2**.

Compressor **122** is configured to compress gas into a hot, high-pressure liquid and the high pressure liquid flows out of the bottom of the compressor **122** to the reversing valve **128**. The refrigerant enters the reversing valve **128** through port A and is diverted into port C and flow to the evaporating coil of the first heat exchanger **124**. A fan blows across the evaporating coil, releasing heat energy to an indoor area. Then the refrigerant flows out of evaporating coil to expansion valve **130**. The pressure of the refrigerant gas dramatically decreases on the opposite side of expansion valve **130** and the refrigerant cools downs to a cold vapor. The refrigerant flows from the expansion valve **130** to the condensing coil of the second heat exchanger **126** and a fan blows across the condensing coil and release cold air to an outdoor environment. The refrigerant flows out of condensing coil to solar thermal collection module **114** while the cold refrigerant absorbs the solar thermal heat, increasing in temperature and pressure. The refrigerant flows out of solar thermal module **114** back to reversing valve **128** and flows into port D of reversing valve **128**. The refrigerant then flows from port D into port B of the reversing valve **128** and flows from port B back to the compressor **122** to be recompressed into a high-pressure, warm liquid to repeat the cycle again.

The compressor **122** compress the refrigerant into a high-pressure liquid by supplementing solar thermal energy into the fluid circuit **112** and then directly delivery into a building. Even in the depth of winter, special properties of the solar thermal collection module **114** allow the temperature to rise to well over 400° F., which can still save a large of portion of electrical energy for the compressor **122**.

Referring to FIG. **4**, similar to the heating cycles depicted in FIG. **3**, the compressor **122** utilizes solar thermal heat to control the pressure and flow of refrigerant in a cooling cycle. The same benefits apply in the cooling cycle as mentioned in the heating cycle. In this scenario, the first heat exchanger **124** includes a condensing coil and the second heat exchanger **126** includes an evaporation coil. The compressor **122** pumps the refrigerant through the condensing coil of the first heat exchanger **124** and releases heat absorbed from an indoor environment. Fluid (i.e. refrigerant) in the fluid circuit **112** is then passed through an evaporation coil of the secondary heat exchanger **126** in which heat from a building is passed over to the refrigerant and raised its pressure on a high-pressure side of the system **100**. The refrigerant exiting the evaporation coil in the second heat exchanger **126** is then passed through the compressor **122** in which further increases the refrigerant pressure before entering the first heat exchanger **124** for another fluid moving cycle.

The compressor **122** is configured to compress cold gas into a hot high-pressure liquid, and the hot high-pressure liquid flows out of the bottom of the compressor **122** to the reversing valve **128**. The refrigerant enters the reversing valve **128** through port A and is diverted into port D. The refrigerant flows from port D to the solar thermal collection module **114**. The pressure of the refrigerant dramatically increases as it passes through the solar thermal collection module **114**. The increase in pressure of the refrigerant is detected by the light intensity sensors **132** within solar thermal collection module **114** and/or the compressor **122**. The compressor **122** is configured to decrease its speed and torque against the refrigerant, lowering the consumption of electrical energy. The refrigerant flows out of the solar thermal collection module to the condensing coil. The warm refrigerant flows through condensing coil of the second heat exchanger **126**. Fan blows air across the condensing coil, removing the excess heat to the outdoors and transferring the thermal energy from indoors to outdoors. The refrigerant flows from the condensing coil to the expansion valve **130**, and the refrigerant is released on the other side of expansion valve **130**, dropping the pressure of the refrigerant and therefore the temperature of the refrigerant. The refrigerant changes state from a hot, high-pressure liquid to a cold, low-pressure gas. The refrigerant flows from expansion valve **130** to the evaporating coil of the first heat exchanger **122**. A fan blows across the evaporating coil, blowing the cooler air indoors. The refrigerant flows out of the evaporating coil back to reversing valve **128** and through port C into port B of reversing valve **128**. The refrigerant flows out of port B back to compressor **122** to be recompressed into a high-pressure warm liquid to repeat the cycle again.

Similar to the heating cycle, the system **100** enables the solar thermal energy to increase the pressure within the refrigeration cycle, which in turn decreases the electrical consumption of the compressor **122** to achieve efficient heat transfer. As such, the sunnier and hotter it is outside, the more air conditioning will be required and the more solar thermal energy can be provided. Under these circumstances, the sunlight and heat in the environment can act as another source energy in addition of electrical energy to keep the compressor **122** working.

Referring to FIGS. **3**-**4**, at least one light intensity sensor **132** is configured to determine the light intensity of the environment. The light intensity sensor **132** can be positioned inside the solar thermal collection module **114** and/or the compressor **122**. The light intensity sensor **132** can be at least one of pressure and/or temperature sensor configured to measure pressure and/or temperature of fluid inside the solar thermal module **114**. The rise in pressure and/or temperature enabled by the solar thermal collection module **114** will reflect transfer of solar thermal energy. The more solar and thermal energy detected, the less the electricity used by the compressor **122**.

Referring to FIG. **5**, the system **100** further includes a central heat recovery unit **134** configured to receive data input from the light intensity sensor **132** and other components of the system. The central heat recovery unit **134** is also connected to one or more controllers operably linked to respective operating components of the heat recovery system **100**. For example, the central heat recovery unit **134** is connected to a furnace control board **136**, a thermostat board **138**, a heat pump controller board **140** and other functional components (e.g., fan, motors, etc.). Based on data inputs from the light intensity sensor **132**, the central intelligence unit **134** enables the system **100** to achieve highest efficiency by using solar thermal energy obtained via the solar thermal module **114** to replace a portion of electrical energy otherwise needed to increase fluid pressure of the compressor **122** of the system **100**.

According to another embodiment of the present invention, the central heat recovery unit **134** is configured to receive weather data from a third party **142** and enable the operation of compressor to work from a heating mode to a cooling mode (defrost cycle) when weather data indicate a certain threshold (e.g., temperature below freezing point) and/or certain conditions (e.g., snow, precipitation, etc.).

Buildup of ice and snow in the winter months dramatically reduce or completely nullify performance of the solar thermal collection module **114**. Converting from a heating mode to a cooling mode will enable the defrosting of ice or snow built up on the solar thermal collection module in a defrost cycle. Specifically, the central intelligence unit **134** will enable the evaporating coil of the second heat exchanger **126** inside the heat pump **110** to blow hot air onto the backside of the solar thermal collection module **114**, melting any ice or snow built up thereon and allowing the system to maintain maximum performance even in the depth of winter. The defrost cycle is a mechanical action in which the compressor **122** is configured to reverse back to a cooling mode from a heating mode and pull hot refrigerant through the outdoor section to defrost ice built up on the solar thermal collection module **114**. In this scenario, the hot air removed from the interior is drafted onto the solar thermal collection module **114** instead of being releasing into the environment. Even if there is no ice or snow built up, hot air is directed to the solar collection module **114** and transferred right back into the fluid circuit instead of blowing the thermal energy into the outdoor environment and wasting valuable heat.

Referring to FIG. **6**, a method of recovering heat and energy includes, at step **602**, obtaining solar thermal energy via a solar thermal collection module (e.g., solar thermal collection module **114**). The solar thermal collection module **114** is configured to utilize solar thermal energy to increase fluid pressure inside the heat pump **110**. The solar thermal energy can increase fluid temperature within a compressor (e.g., compressor **122**) in fluid communication with the solar thermal collection module **114**. By capturing solar thermal energy, a large part of fluid (e.g., refrigerant) pressurization can be achieved with little mechanical work, limiting the amount of electrical energy required.

At step **604**, fluid contained in a compressor is pressurized by at least partially utilizing the obtained solar thermal energy. Specifically, a light intensity sensor (e.g., light intensity sensor **132**) can measure pressure and/or temperature of fluid inside the solar thermal module **114** and/or the compressor. The rise in pressure and/or temperature enabled by the solar thermal collection module **114** reflects presence of the solar thermal energy. The more thermal and solar energy is detected to be present, the less the electricity will be used by the compressor **122** to pressurize the fluid in the compressor **122**. The amount of electricity needed to run the compressor **122** can be determined by a central intelligence unit (e.g., central intelligence unit **134**).

At step **606**, heat energy exchange is effectuated via a first heat exchanger and a second heat exchanger in fluid communication with the compressor. The central thermal recovery unit **134** is configured for determining an operating instruction based on the at least one environmental measurement (e.g., outdoor temperature) and system-related data received from the one or more components of the system (e.g., thermostat). For example, the system will start a heating mode or a cooling mode depending on the temperature of outdoor environment and the temperature set by a thermostat. The central thermal recovery unit **134** is further configured to determine a specific operation sequence of a series of operating components. For example, the central thermal recovery unit **134** is configured to ensure the compressor (e.g., compressor **122**) runs for a certain time period (e.g., 5 minutes, 10 minutes, etc.) based on the amount of solar thermal energy obtained via the solar thermal module **114**. As another example, when the central thermal recovery unit **134** detects the compressor and the solar thermal energy is not adequate to fulfill a heating need, a furnace (e.g., furnace **116**) is activated to input supplemental heat to the system.

In a heating mode, the pressurized vapor is condensed at high pressure and temperature inside a condensing coil of the second heat exchanger **124**. As the refrigerant condenses, heat is released, providing heated air to a heat recovery system. The liquid refrigerant is then transported to the evaporation coil of the first heat exchanger **124**, which lowers the pressure and goes on to another circle.

In a cooling cycle, a first heat exchanger (e.g., first heat exchanger **124**) includes a condensing coil and a second heat exchanger (e.g., second heat exchanger **126**) includes an evaporation coil. In this scenario, the compressor **122** pumps the refrigerant through the condensing coil of the first heat exchanger and releases heat absorbed from an indoor environment. Fluid (i.e. refrigerant) in the fluid circuit **112** is then passed through an evaporation coil of a second heat exchanger in which heat from a building is passed over to the refrigerant and raised its pressure on a high-pressure side of the system **100**.

At step **608**, heat energy is exchanged by forcing air over the first and second heat exchanger. Air was forced through the condensing coil and evaporating coil to achieve the heat exchange during a heating mode and cooling mode.

Referring to FIGS. **7-10**, a solar thermal collection module **114** includes a plurality of (e.g., three) cell chambers **144** positioned in parallel and formed by four exterior surfaces **146** (top surface, bottom surface, and two vertical surfaces) and a plurality of (e.g., two) vertical division panels **148**. The exterior surfaces **146** and the division panels are preferably made of materials having good insulating properties and a dark, radiation-absorbing coloring for obtaining the solar radiation. In the depicted embodiment, each chamber **140** is depicted as an elongated chamber having a rectangle cross section. Other suitable shapes can also be used. The interior walls of each chamber **144** are covered with a reflective material such as silver and other materials. The reflective material allows reflective properties up to one hundred times stronger than conventional reflective wraps. A plurality of interconnected fluid pipes **150** are positioned inside each chamber **144**. As a specific example, a plurality of interconnected fluid pipes **150** run through each chamber **144**, positioned approximately two inches apart and staggered at different heights so that no single pipe shadows another, as shown in FIGS. **8-10**. The plurality of interconnected fluid pipes **150** are covered with thermal absorbing coating material. Additionally, a layer of thermal coating can also be deposited onto the thermal absorbing coating material and/or the one or more fluid pipes **150**. This configuration allows direct and indirect sunlight to focus onto the one or more fluid pipes **150**, regardless of the angle of the sun for optimal solar and thermal heat transfer. Each solar thermal chamber **144** is insulated with foam or other high insulation factor materials **158** which will keep the heat within each chamber **144** long after the source of the thermal energy (e.g., direct sunlight **152** and indirect sunlight **154** or thermal energy) are no longer present. The solar collection module **114** is also covered by tempered glass **160** for maximum solar transfer and safety. Such design of the solar collection module **114** can trap heat within each cell chamber **144** and increase the pressure of the fluid after the source of the thermal energy gain is gone. One or more drain holes **156** are located on a bottom surface of each cell chamber **144** for draining condensing liquid and moisture buildup.

Referring to FIG. **11**, according to another embodiment of the present invention, a metal layer (e.g., a layer of galvanized steel) **162** is positioned inside each of the respective chambers and in close proximity to a bottom surface **146** of each chamber **144**. A respective air gap **164** is formed between the bottom surface **146** and the metal layer **162** as further insulation. The chamber **144** can be further covered by an outer metal layer (e.g., steel layer), which can facilitate heat absorbing into the plurality of chambers **144**.

The present invention allows heat pump to be used at extremely high efficiencies. This can reduce the need to use fossil fuels and electricity in the same HVAC system, in return, this can reduce greenhouse gas production and achieve the same comfort level within the building. This method will save significant amounts of electricity.

The present invention has advantages over traditional HVAC systems in that it produces less greenhouse gases and makes effective use of heat energy that is typically released into the environment. The present invention enables a dramatic reduction in electrical consumption by the compressors in the refrigeration circuit, which in turn transfer the thermal energy from the flue gas and solar energy mixture back into the HVAC airstream to the benefit of building heating efficiency.

In general, the foregoing description is provided for exemplary and illustrative purposes; the present invention is not necessarily limited thereto. Rather, those skilled in the art will appreciate that additional modifications, as well as adaptations for particular circumstances, will fall within the scope of the invention as herein shown and described and of the claims appended hereto.

What is claimed is:

1. A heat recovery system comprising:

a variable speed inverter compressor in fluid connection with a first heat exchanger and a second heat exchanger via a fluid circuit;

a solar thermal collection module positioned on top of the compressor and in fluid communication in the compressor, the first heat exchanger and the second heat exchanger;

a light intensity sensor configured to determine light intensity on the solar thermal collection module; and

a central heat recovery unit configured to receive input of the light intensity sensor and determine operation of the compressor based on the input from the light intensity sensor;

wherein the solar thermal collection module is configured to retain solar thermal energy to increase fluid pressure in the compressor; and

wherein the operation of the compressor is based on measurement of the light intensity sensor.

2. The heat recovery system of claim **1**, wherein the solar thermal collection module includes:

a plurality of solar thermal cell chambers positioned in parallel and covered by tempered glass; and

a plurality of interconnected fluid pipes positioned through the plurality of solar thermal cell chambers;

wherein each of the plurality of fluid pipes are covered by thermal absorbing coating material;

wherein a layer of reflective material is covered on inner sidewall of each solar thermal cell chamber;

wherein each cell chamber is filled with foam material to retain heat obtained from solar thermal energy; and

wherein one or more drain holes are located on a bottom surface of each cell chamber for draining condensing liquid and moisture buildup.

3. The heat recovery system of claim **1**, wherein the light intensity sensor is a pressure sensor configured to determine pressure of the fluid inside at least one of the solar thermal collection modules and the compressor.

4. The heat recovery system of claim **1**, wherein the light intensity sensor is a temperature sensor configured to determine temperature of the fluid inside at least one of the solar thermal collection modules and the compressor.

5. The heat recovery system of claim **1**, further comprising a reversing valve configured to switch a direction of the fluid movement between the compressor, the solar thermal collection module, the first heat exchanger and the second heat exchanger based on a cooling demand or a heating demand.

6. The heat recovery system of claim 1, wherein the central heat recovery unit is configured to achieve a highest efficiency of the system.

7. The heat recovery system of claim 1, wherein the central heat recovery unit is configured to determine the running time period for the compressor based on the input from the light intensity sensor.

8. The heat recovery system of claim 1, wherein the central heat recovery unit is further configured to receive weather data from a third party.

9. The heat recovery system of claim 8, wherein the central heat recovery unit is configured to activate a cooling cycle to heat the solar thermal collection module when weather indicates a certain condition.

10. The heat recovery system of claim 9, wherein the certain condition to activate a cooling cycle includes at least one of snow, precipitation and temperature below 32 degrees Fahrenheit.

11. A method of recovering heat and energy comprising:
- obtaining solar thermal energy via a solar thermal collection module;
- pressurizing fluid contained in a compressor by at least partially utilizing absorbed solar thermal energy;
- effectuating heat energy exchange via a first heat exchanger and a second heat exchanger in fluid communication with the compressor; and
- exchanging heat energy by forcing air over the first and second heat exchanger;
- wherein the solar thermal collection module is positioned on top of the compressor, forming an integral piece; and
- wherein pressurizing fluid contained in the compressor by at least partially utilizing thermal energy from the absorbed solar energy includes determining solar intensity on the solar thermal collection module and determining an amount of mechanical compression need to achieve a certain fluid pressure.

12. The method of claim 11, wherein the solar thermal collection module includes:
- a plurality of solar thermal cell chambers positioned in parallel and covered by tempered glass; and
- a plurality of interconnected fluid pipes positioned through the plurality of solar thermal cell chambers;
- wherein each of the plurality of fluid pipes are covered by thermal absorbing coating material;
- wherein a layer of reflective material is covered on inner sidewall of each solar thermal cell chamber;
- wherein each cell chamber is filled with foam material to retain heat obtained from solar thermal energy; and
- wherein one or more drain holes are located on a bottom surface of each cell chamber for draining condensing liquid and moisture buildup.

13. A solar thermal collection module comprising:
- a plurality of solar thermal cell chambers positioned in parallel and covered by tempered glass;
- a plurality of interconnected fluid pipes positioned through the plurality of solar thermal cell chambers; and
- a light intensity sensor positioned inside the solar thermal collection module;
- wherein each of the plurality of fluid pipes are covered by thermal absorbing coating material;
- wherein reflective film is covered on inner sidewall of each solar thermal cell chamber;
- wherein each cell chamber is filled with foam material to retain heat obtained from solar thermal energy; and
- wherein one or more drain holes are located on a bottom surface of each cell chamber for draining condensing liquid and moisture buildup.

14. The solar thermal collection module of claim 13, wherein a metal layer is positioned inside each of the plurality of solar thermal cell chambers and in close proximity to respective bottom surface of each chamber such that an air gap is formed between the respective bottom surface and the metal layer.

15. The solar thermal collection module of claim 13, wherein the solar light intensity sensor is a temperature sensor.

16. The solar thermal collection module of claim 13, wherein the solar light intensity sensor is a pressure sensor configured to measure pressure of fluid contained in the one or more of the plurality of fluid pipes.

17. The solar thermal collection module of claim 13, wherein the plurality of interconnected fluid pipes are positioned approximately at certain distance apart and staggered at different height in a manner in which no single pipe shadows another in each chamber.

18. A heat recovery system comprising:
- a variable speed inverter compressor in fluid connection with a first heat exchanger and a second heat exchanger via a fluid circuit;
- a solar thermal collection module positioned on top of the compressor and in fluid communication in the compressor, the first heat exchanger and the second heat exchanger;
- a light intensity sensor configured to determine light intensity on the solar thermal collection module; and
- a reversing valve configured to switch a direction of the fluid movement between the compressor, the solar thermal collection module, the first heat exchanger and the second heat exchanger based on a cooling demand or a heating demand;
- wherein the solar thermal collection module is configured to retain solar thermal energy to increase fluid pressure in the compressor; and
- wherein the operation of the compressor is based on measurement of the light intensity sensor.

19. A method of recovering heat and energy comprising:
- obtaining solar thermal energy via a solar thermal collection module;
- pressurizing fluid contained in a compressor by at least partially utilizing absorbed solar thermal energy;
- effectuating heat energy exchange via a first heat exchanger and a second heat exchanger in fluid communication with the compressor; and
- exchanging heat energy by forcing air over the first and second heat exchanger;
- wherein the solar thermal collection module is positioned on top of the compressor, forming an integral piece; and
- wherein the solar thermal collection module includes:
  - a plurality of solar thermal cell chambers positioned in parallel and covered by tempered glass; and
  - a plurality of interconnected fluid pipes positioned through the plurality of solar thermal cell chambers;
  - wherein each of the plurality of fluid pipes are covered by thermal absorbing coating material;
  - wherein a layer of reflective material is covered on inner sidewall of each solar thermal cell chamber;
  - wherein each cell chamber is filled with foam material to retain heat obtained from solar thermal energy; and

wherein one or more drain holes are located on a bottom surface of each cell chamber for draining condensing liquid and moisture buildup.

20. A solar thermal collection module comprising:

a plurality of solar thermal cell chambers positioned in parallel and covered by tempered glass; and

a plurality of interconnected fluid pipes positioned through the plurality of solar thermal cell chambers;

wherein each of the plurality of fluid pipes are covered by thermal absorbing coating material;

wherein reflective film is covered on inner sidewall of each solar thermal cell chamber;

wherein each cell chamber is filled with foam material to retain heat obtained from solar thermal energy;

wherein one or more drain holes are located on a bottom surface of each cell chamber for draining condensing liquid and moisture buildup; and

wherein a metal layer is positioned inside each of the plurality of solar thermal cell chambers and in close proximity to respective bottom surface of each chamber such that an air gap is formed between the respective bottom surface and the metal layer.

21. A solar thermal collection module comprising:

a plurality of solar thermal cell chambers positioned in parallel and covered by tempered glass; and

a plurality of interconnected fluid pipes positioned through the plurality of solar thermal cell chambers;

wherein each of the plurality of fluid pipes are covered by thermal absorbing coating material;

wherein reflective film is covered on inner sidewall of each solar thermal cell chamber;

wherein each cell chamber is filled with foam material to retain heat obtained from solar thermal energy;

wherein one or more drain holes are located on a bottom surface of each cell chamber for draining condensing liquid and moisture buildup; and

wherein the plurality of interconnected fluid pipes are positioned approximately at certain distance apart and staggered at different height in a manner in which no single pipe shadows another in each chamber.

* * * * *

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Kaiser

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(45) **Date of Patent: Apr. 20, 2021**

(54) **SYSTEM AND METHOD FOR HEAT AND ENERGY RECOVERY AND REGENERATION**

(71) Applicant: **Commercial Energy Saving Plus, LLC**, Boca Raton, FL (US)

(72) Inventor: **Stewart Kaiser**, Stroudsburg, PA (US)

(73) Assignee: **Commercial Energy Savings Plus, LLC**, Boca Raton, FL (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 258 days.

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F24D 19/10 (2006.01)
F24D 12/02 (2006.01)
(Continued)

(52) **U.S. Cl.**
CPC ***F24D 19/1084*** (2013.01); ***F24D 5/04*** (2013.01); ***F24D 12/02*** (2013.01); ***F24H 3/065*** (2013.01);
(Continued)

(58) **Field of Classification Search**
CPC F25B 27/02; F25B 27/005; F25B 27/002; F24F 2005/0064; F24D 11/007; F24D 11/003; F24D 11/0221; F24D 11/0264; F24D 19/1078; F24D 19/1093; F24D 19/1045; F24D 19/1057; F24D 19/106;
(Continued)

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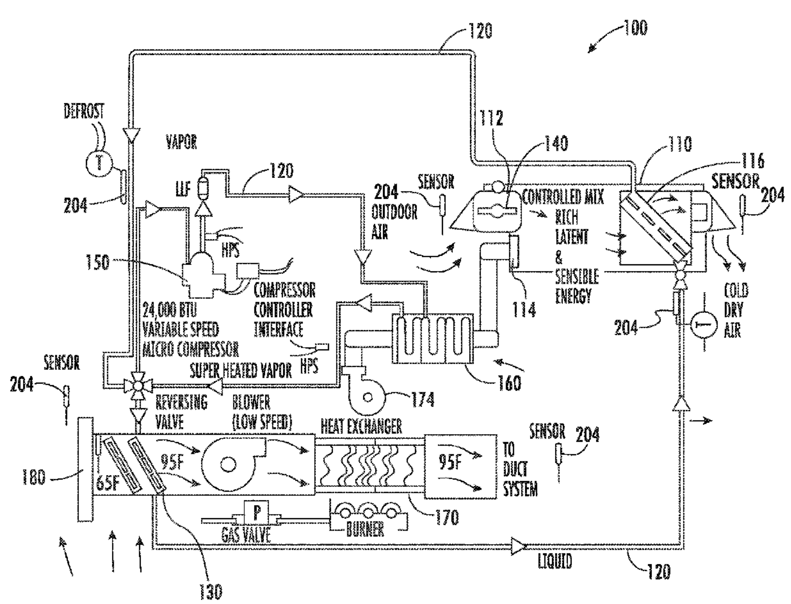
Primary Examiner — Tho V Duong
(74) *Attorney, Agent, or Firm* — Allen Dyer Doppelt & Gilchrist, PA

(57) **ABSTRACT**

A heat recovery system includes a compressor, a solar panel, and a first heat exchanger and a second heat exchanger in fluid connection to form a closed circuit. The compressor is configured to facilitate fluid movement in the fluid circuit between the solar panel, the first heat exchanger and the second heat exchanger. The solar panel includes a plurality of solar cells connected in parallel, and each solar cell includes a plurality of metal tubes for fluid to pass through. A temperature sensor is mounted within each of the solar cells and configured to measure temperature inside the respective solar cell. Each solar cell is connected to the circuit via a respective pressure valve, and the status of the pressure valve is configured to depend on the measurement of the temperature sensor in the respective solar cell.

10 Claims, 18 Drawing Sheets



Related U.S. Application Data

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(51) **Int. Cl.**

F24D 5/04	(2006.01)
F24H 3/06	(2006.01)
F24H 3/12	(2006.01)
F25B 29/00	(2006.01)
F25B 47/02	(2006.01)

(52) **U.S. Cl.**
CPC ***F24H 3/12*** (2013.01); ***F25B 29/00*** (2013.01); ***F25B 47/02*** (2013.01); *F24D 2200/04* (2013.01); *F24D 2200/18* (2013.01); *F24D 2200/22* (2013.01); *F24D 2220/06* (2013.01); *F25B 2700/11* (2013.01)

(58) **Field of Classification Search**
CPC F24S 40/55; F24S 50/201; F24S 50/40; F24S 70/65
See application file for complete search history.

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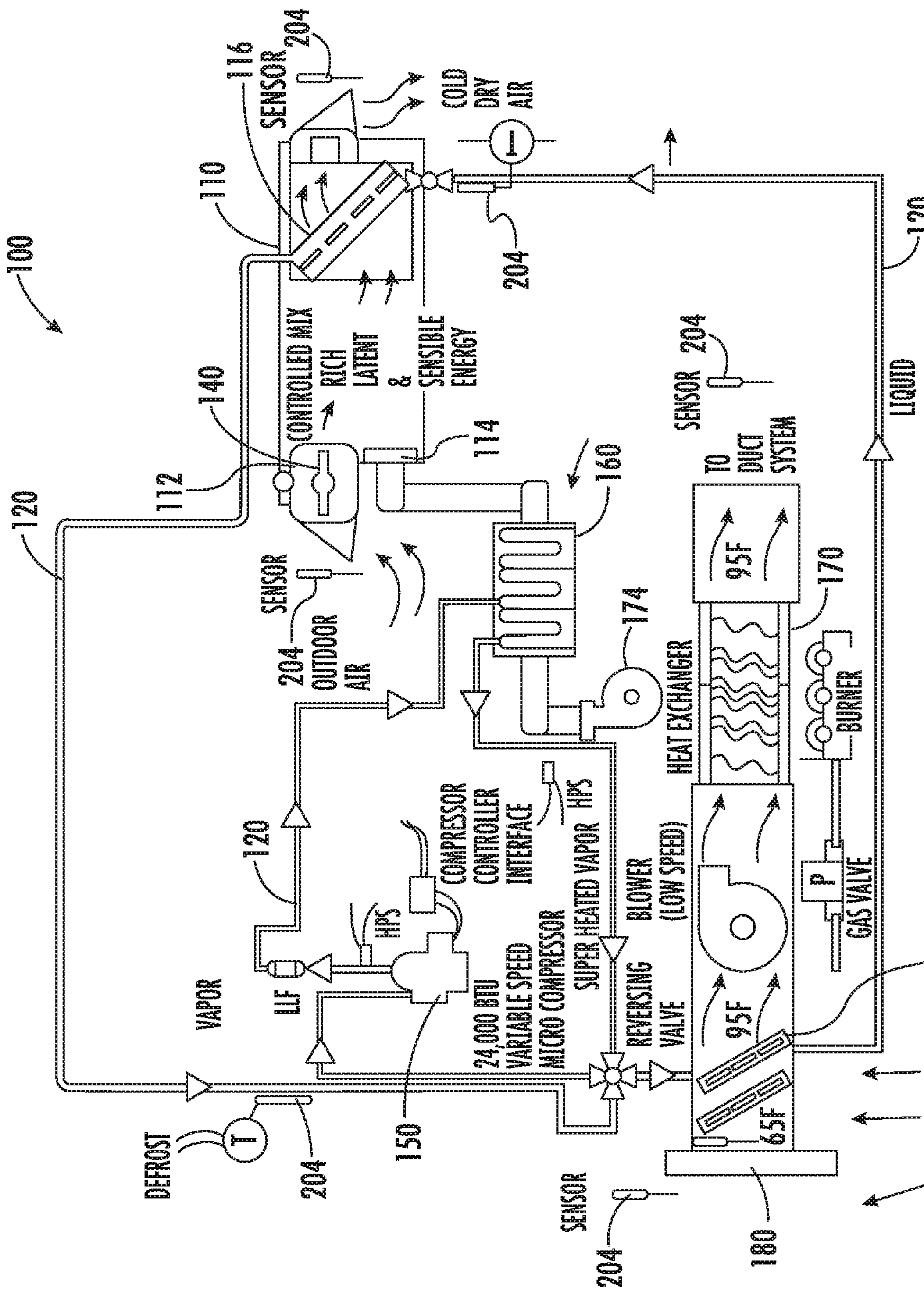
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FIG. 1

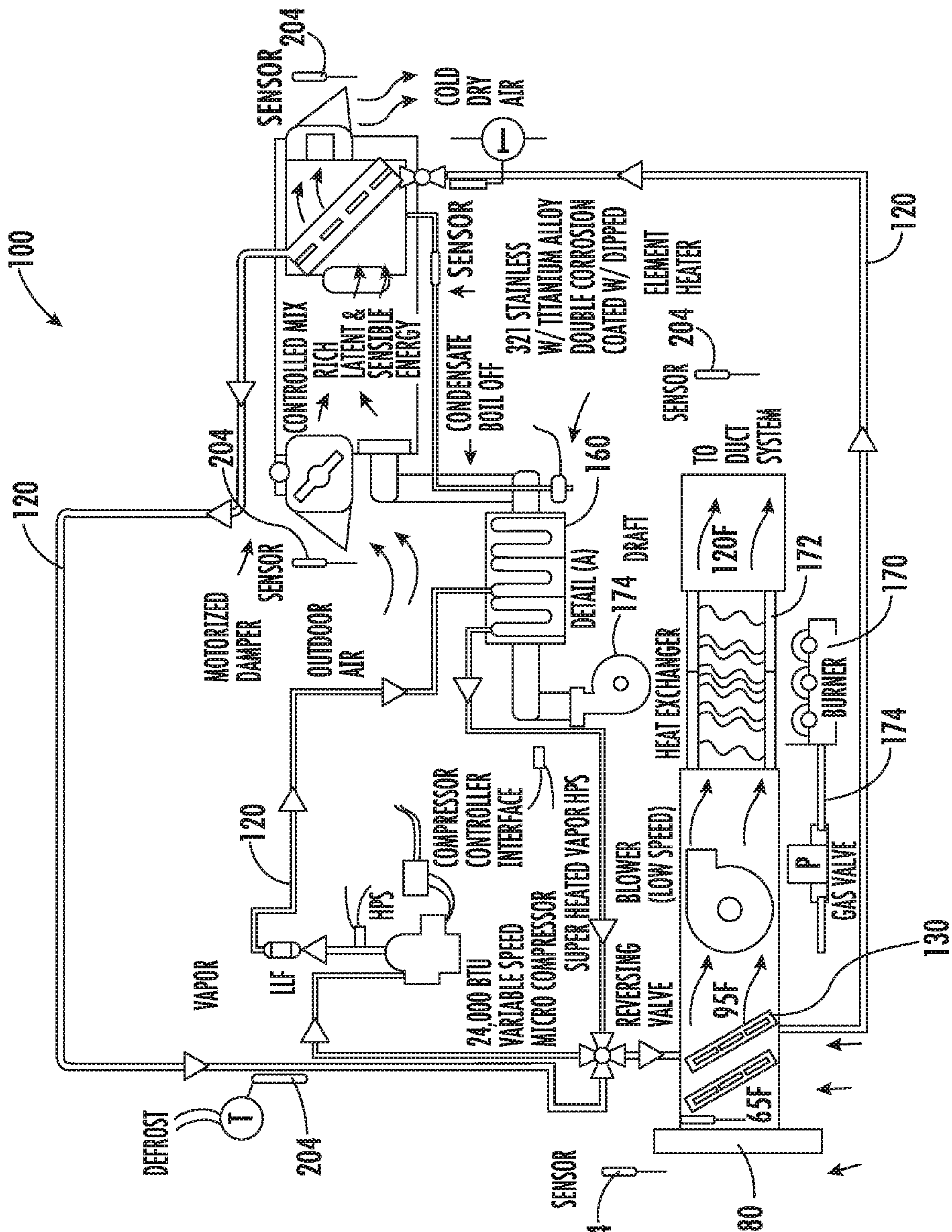
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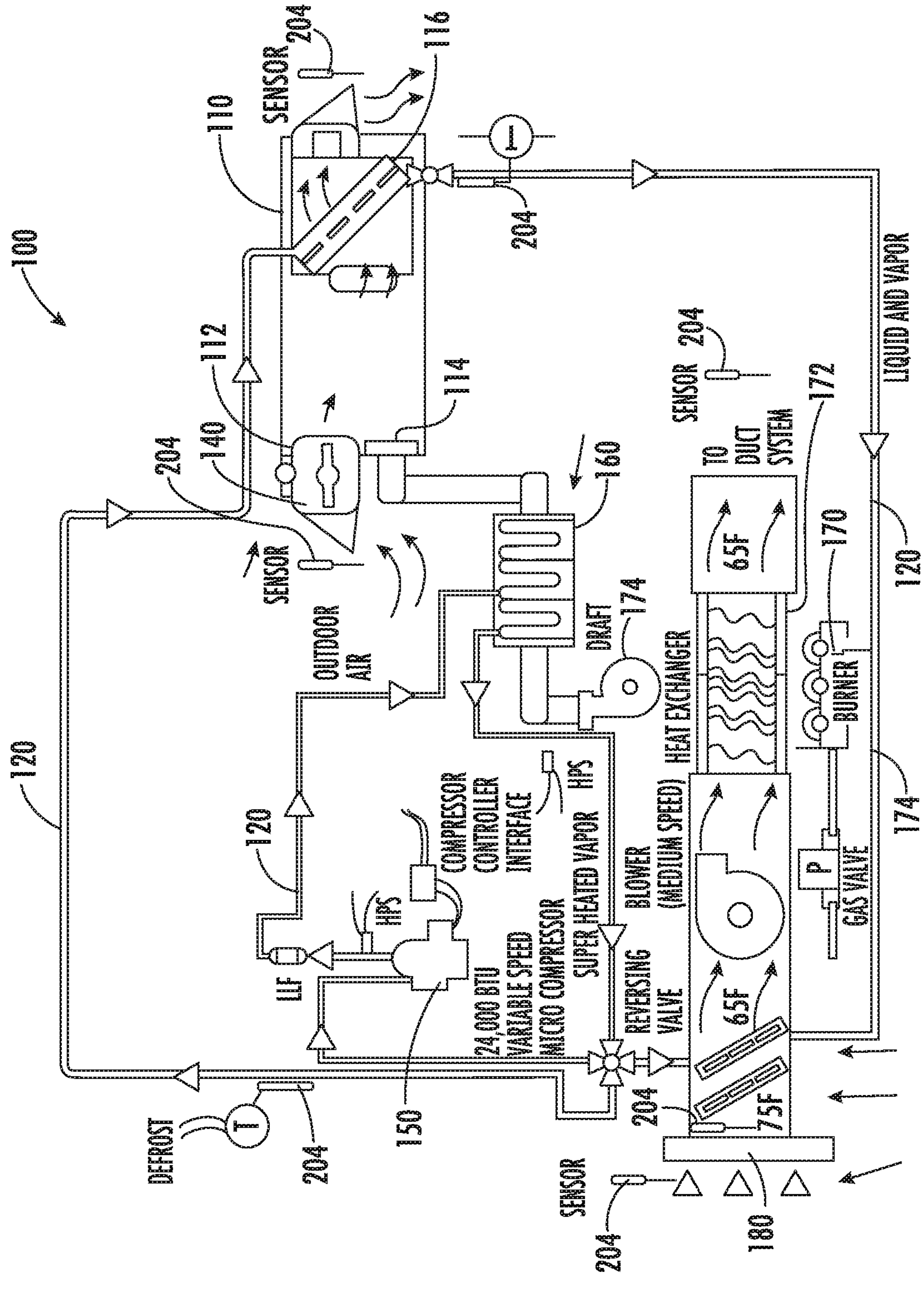
FIG. 2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SUPER HEATED VAPOR
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VARIABLE SPEED
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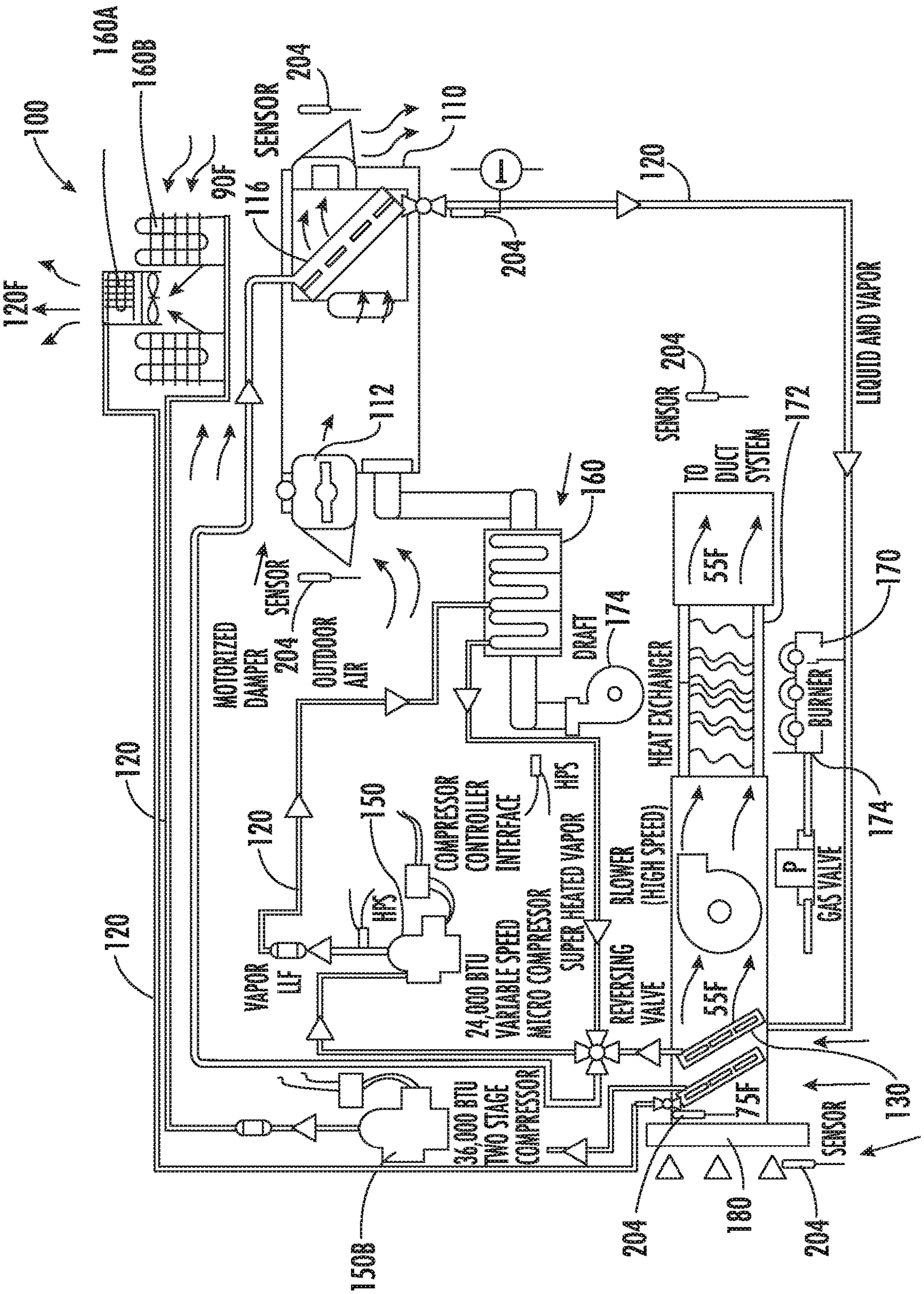
FIG. 3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LIQUID AND VAPOR
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TO DUCT SYSTEM
160
55F
170
MOTORIZED DAMPER
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204
OUTDOOR AIR
DRAFT
174
HEAT EXCHANGER
BURNER
174
120
150
COMPRESSOR CONTROLLER INTERFACE
HPS
HPS
SUPER HEATED VAPOR
BLOWER (HIGH SPEED)
P
GAS VALVE
VAPOR
LLF
24,000 BTU VARIABLE SPEED MICRO COMPRESSOR
REVERSING VALVE
55F
120
36,000 BTU TWO STAGE COMPRESSOR
75F
SENSOR
130
204
180
204
150B

FIG. 4

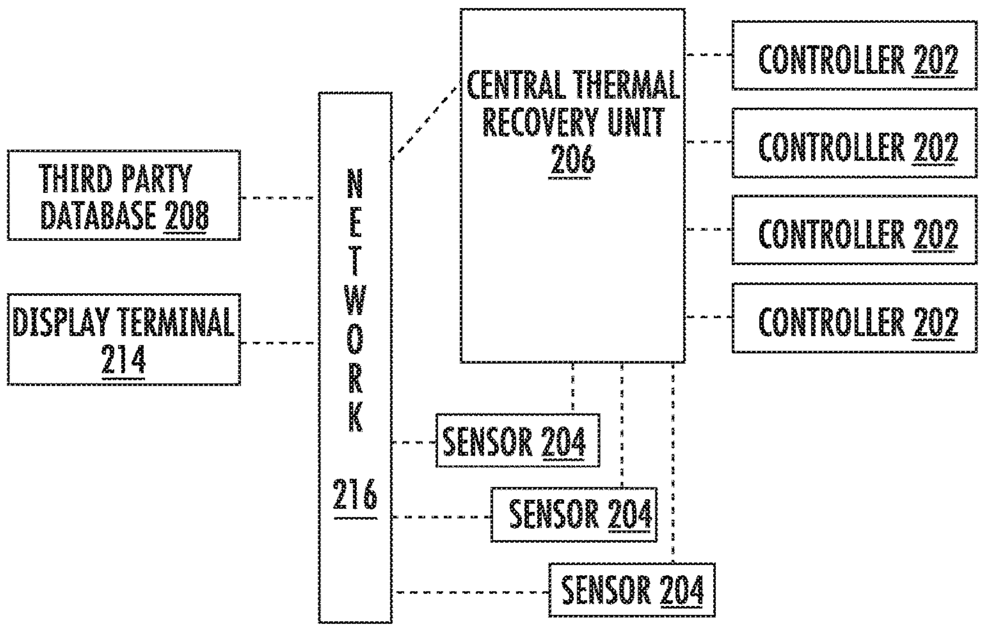
CENTRAL THERMAL RECOVERY UNIT 206
CONTROLLER 202
CONTROLLER 202
CONTROLLER 202
CONTROLLER 202
THIRD PARTY DATABASE 208
DISPLAY TERMINAL 214
NETWORK 216
SENSOR 204
SENSOR 204
SENSOR 204

FIG. 5

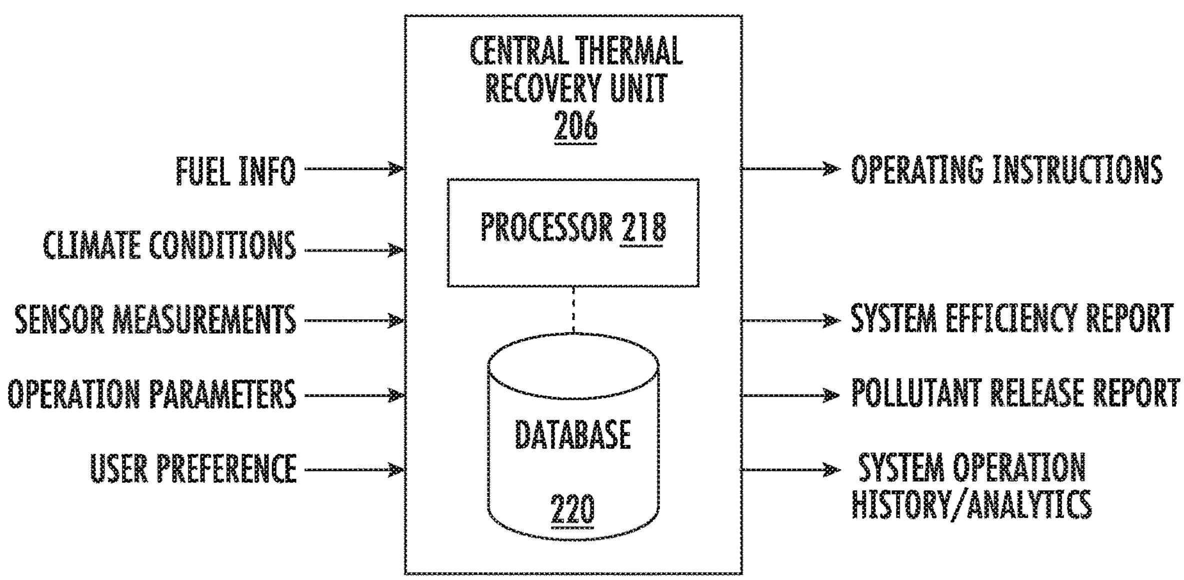
CENTRAL THERMAL RECOVERY UNIT 206
FUEL INFO
CLIMATE CONDITIONS
SENSOR MEASUREMENTS
OPERATION PARAMETERS
USER PREFERENCE
PROCESSOR 218
DATABASE 220
OPERATING INSTRUCTIONS
SYSTEM EFFICIENCY REPORT
POLLUTANT RELEASE REPORT
SYSTEM OPERATION HISTORY/ANALYTICS

FIG. 6

702 FEED EXCESS HEAT AND WASTE PRODUCTS EMITTED AS A RESULT OF FUEL COMBUSTION INTO A MIXING CHAMBER COMPRISING A FIRST HEAT EXCHANGER COUPLED WITH A FLUID CONTAINING CONDUIT CIRCUIT.

↓

704 FEED AIR INTO THE MIXING CHAMBER FOR INITIATING A REACTION WITH THE WASTE PRODUCTS TO PRODUCE A REACTION PRODUCT WITH POTENTIAL ENERGY

↓

706 EFFECTUATE HEAT ENERGY EXCHANGE THROUGH THE REACTION PRODUCT AND EXCESS HEAT INTERACTING WITH THE FIRST HEAT EXCHANGER, WHEREBY TEMPERATURE AND REACTIVE PRESSURE OF FLUID WITHIN THE FIRST HEAT EXCHANGER AND FLUID CIRCUIT

↓

708 PRESSURIZE FLUID CONTAINED IN THE FLUID CIRCUIT BY UTILIZING HEAT ENERGY FROM THE EXHAUST GAS AND WASTE PRODUCTS EMITTED AS THE RESULT OF FUEL COMBUSTION FIRST HEAT EXCHANGER AND FLUID CIRCUIT CONNECTED TO THE FIRST HEAT EXCHANGER RISES

↓

710 EXCHANGE HEAT ENERGY BY FORCING AIR OVER A SECOND HEAT EXCHANGER THAT IS IN FLUID COMMUNICATION WITH THE PRESSURIZED FLUID CONTAINING CONDUIT CIRCUIT EXTERIORLY OF THE MIXING CHAMBER

FIG. 7

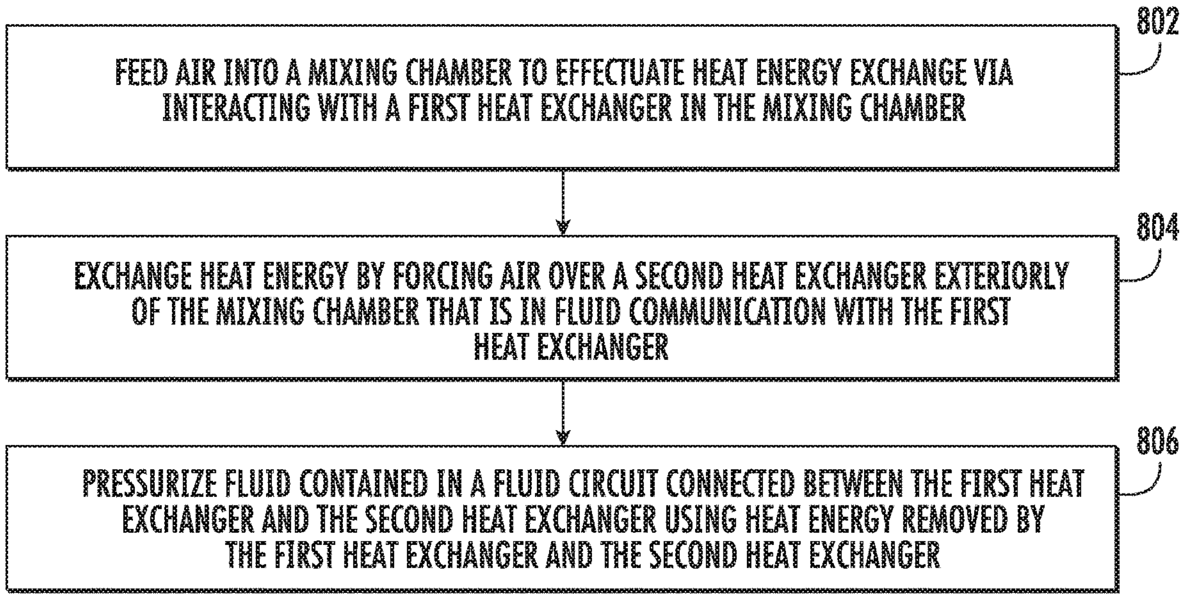
802
FEED AIR INTO A MIXING CHAMBER TO EFFECTUATE HEAT ENERGY EXCHANGE VIA INTERACTING WITH A FIRST HEAT EXCHANGER IN THE MIXING CHAMBER
804
EXCHANGE HEAT ENERGY BY FORCING AIR OVER A SECOND HEAT EXCHANGER EXTERIORLY OF THE MIXING CHAMBER THAT IS IN FLUID COMMUNICATION WITH THE FIRST HEAT EXCHANGER
806
PRESSURIZE FLUID CONTAINED IN A FLUID CIRCUIT CONNECTED BETWEEN THE FIRST HEAT EXCHANGER AND THE SECOND HEAT EXCHANGER USING HEAT ENERGY REMOVED BY THE FIRST HEAT EXCHANGER AND THE SECOND HEAT EXCHANGER

FIG. 8

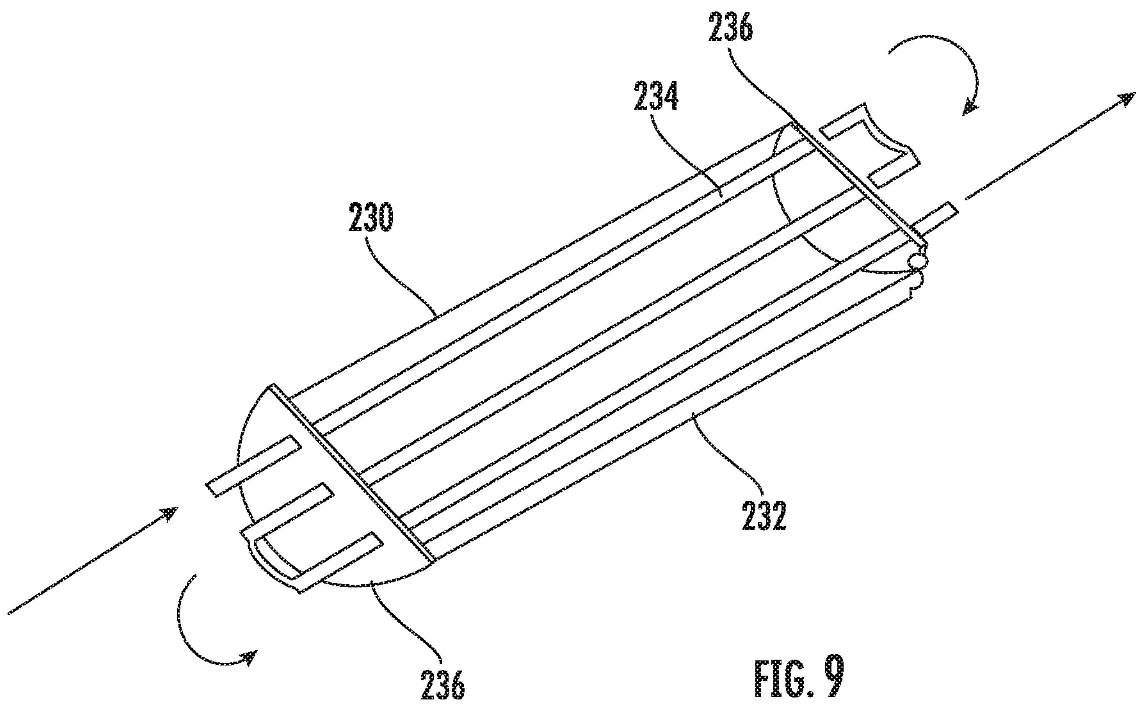

FIG. 9

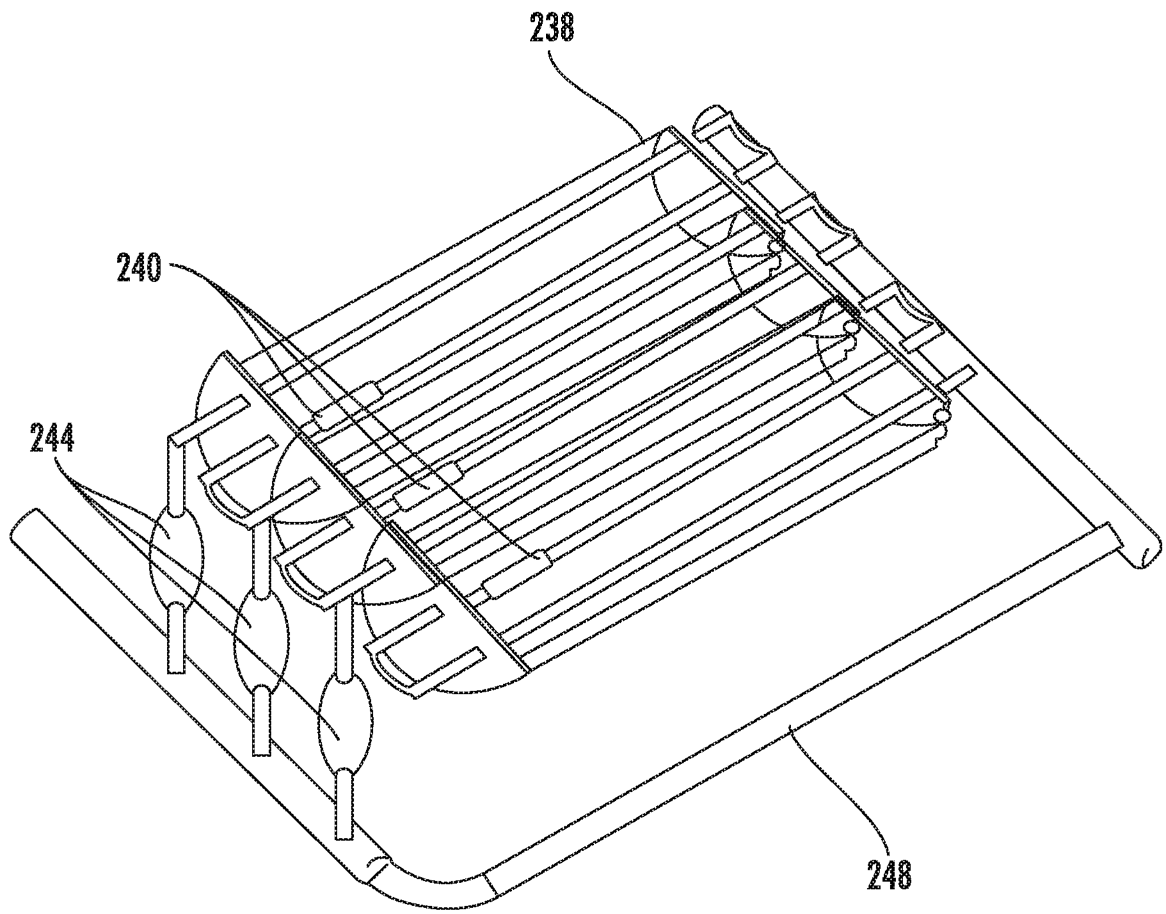

FIG. 10

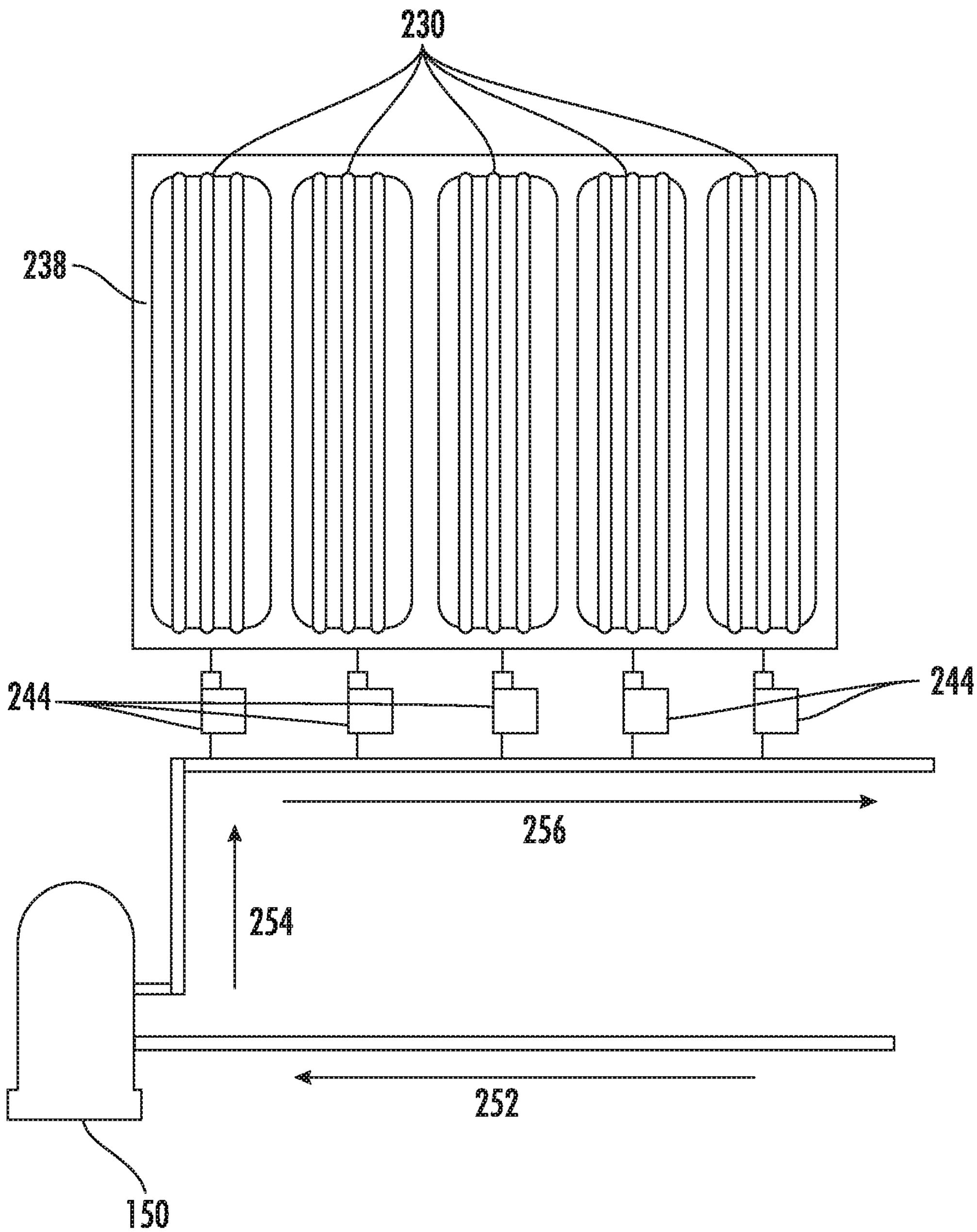

FIG. 11

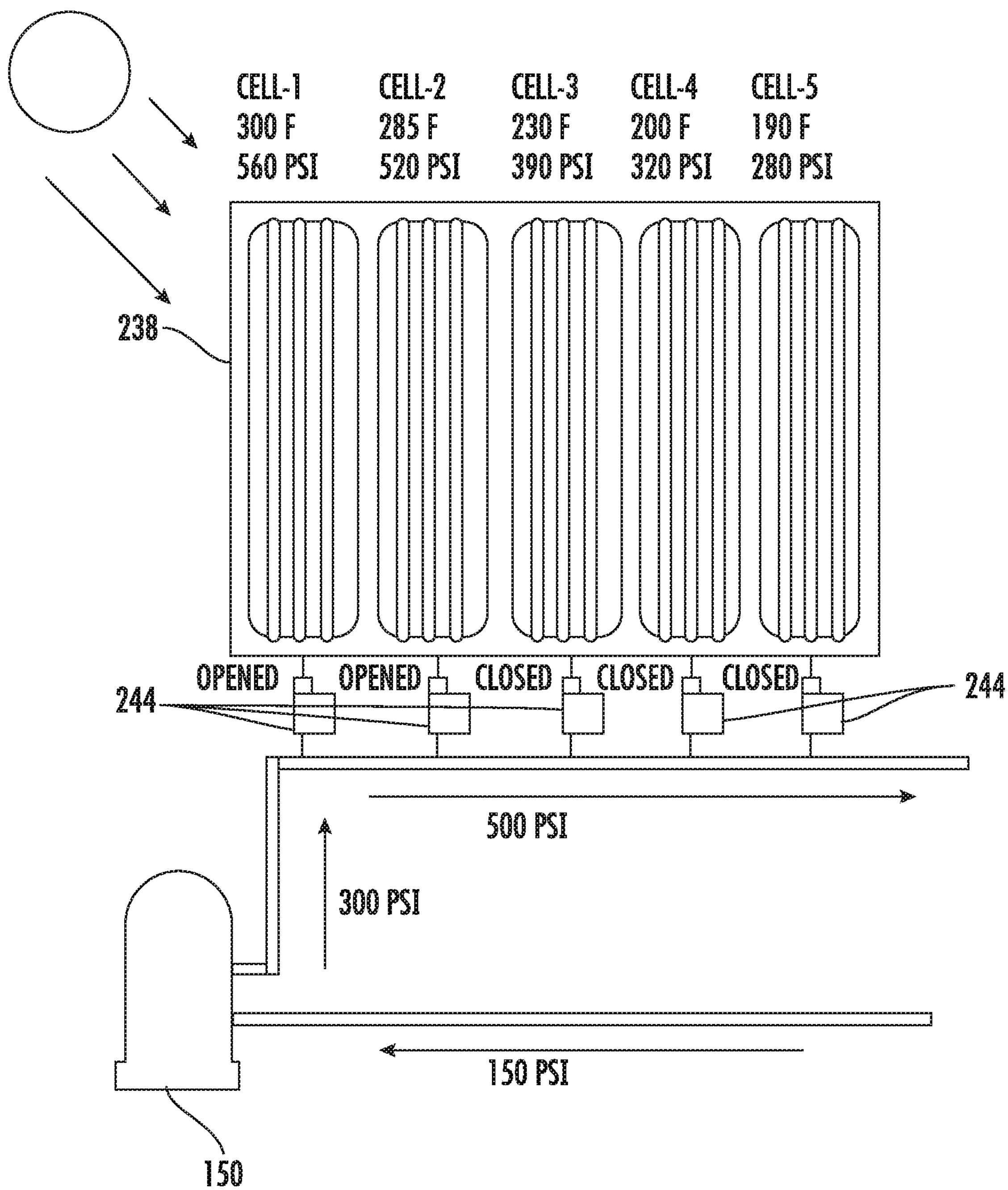
CELL-1
300 F
560 PSI
CELL-2
285 F
520 PSI
CELL-3
230 F
390 PSI
CELL-4
200 F
320 PSI
CELL-5
190 F
280 PSI
238
OPENED
OPENED
CLOSED
CLOSED
CLOSED
244
244
500 PSI
300 PSI
150 PSI
150

FIG. 12

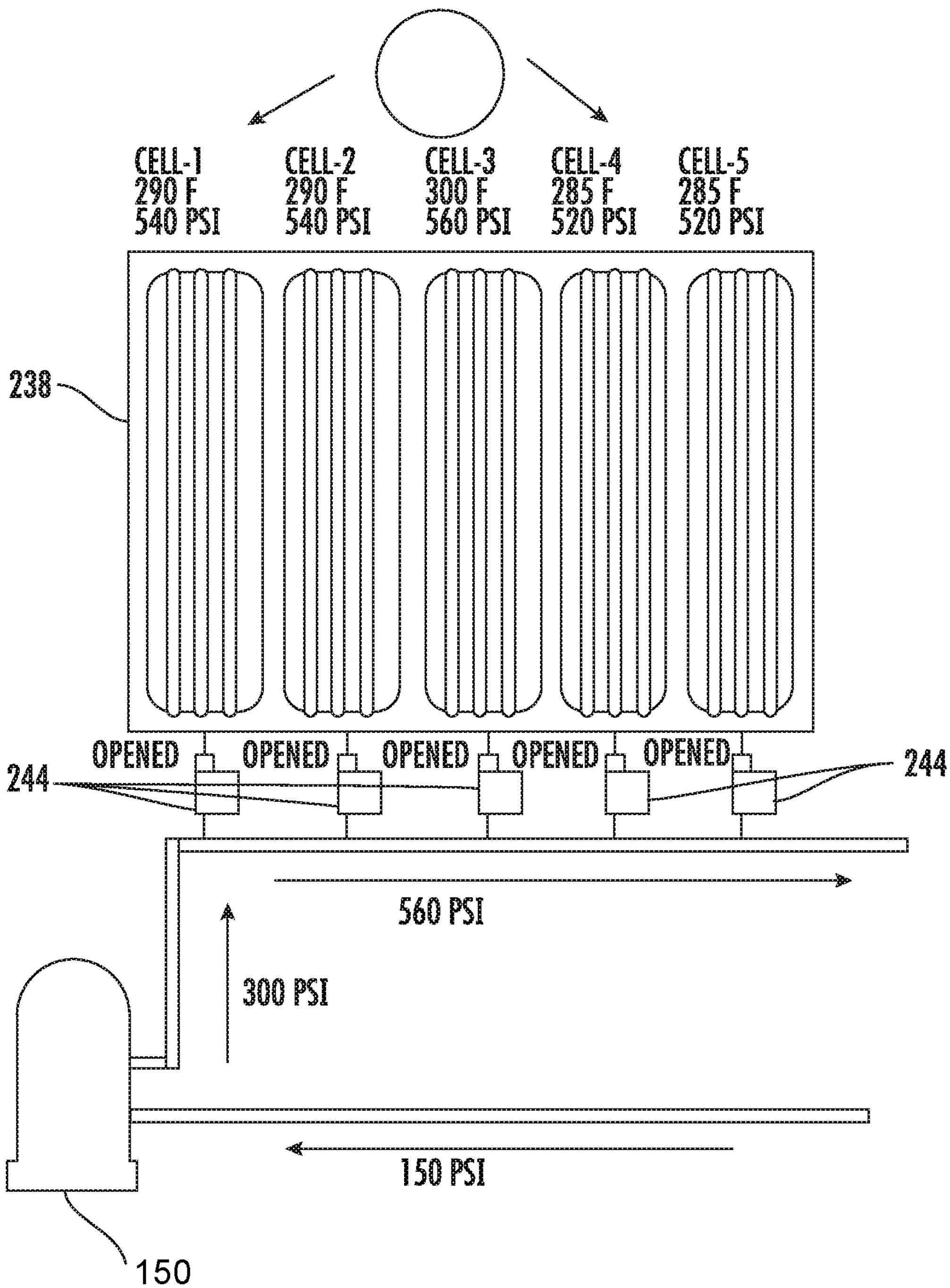
CELL-1
290 F
540 PSI
CELL-2
290 F
540 PSI
CELL-3
300 F
560 PSI
CELL-4
285 F
520 PSI
CELL-5
285 F
520 PSI
238
OPENED
OPENED
OPENED
OPENED
OPENED
244
244
560 PSI
300 PSI
150 PSI
150

FIG. 13

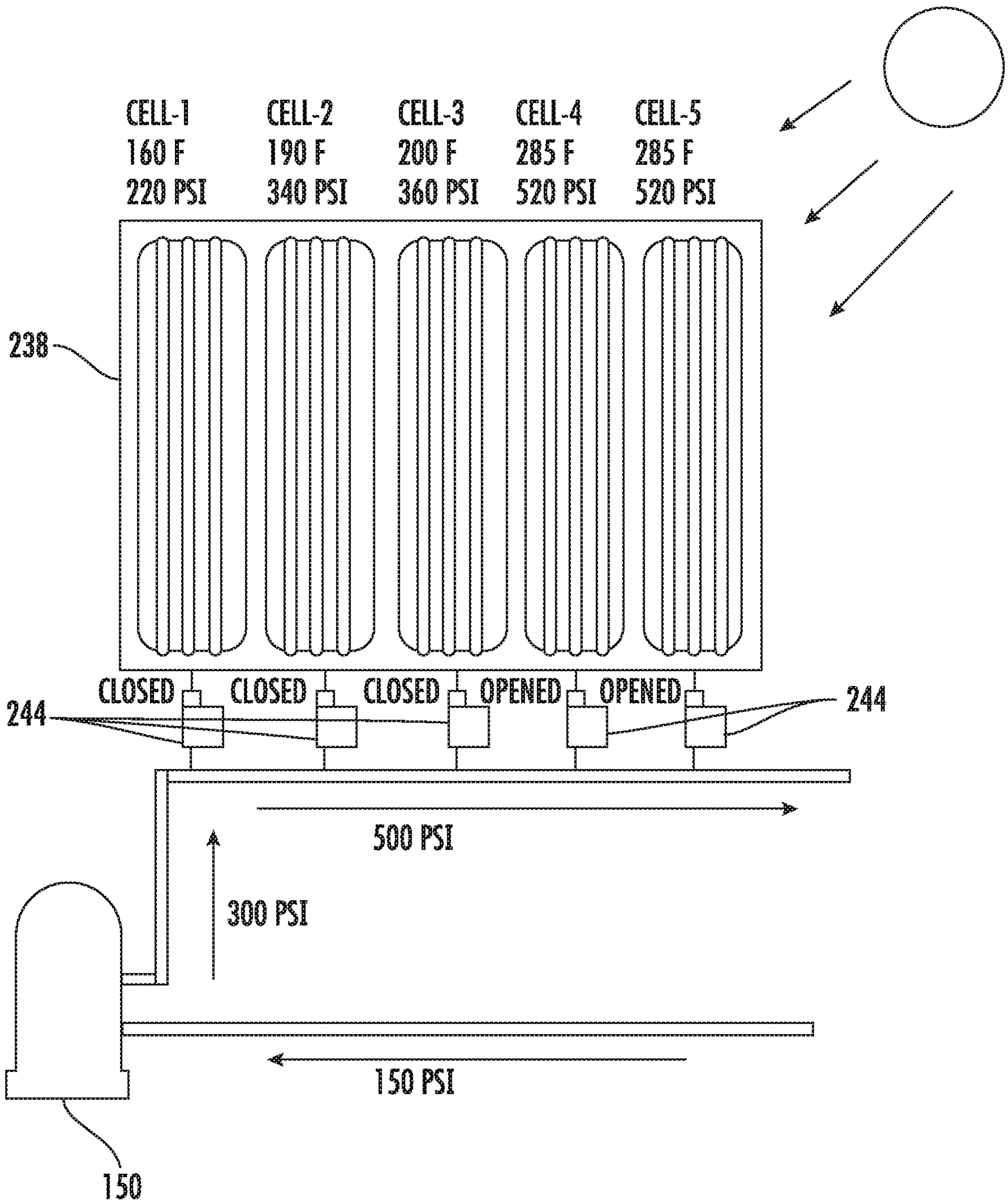
CELL-1
160 F
220 PSI
CELL-2
190 F
340 PSI
CELL-3
200 F
360 PSI
CELL-4
285 F
520 PSI
CELL-5
285 F
520 PSI
238
CLOSED
CLOSED
CLOSED
OPENED
OPENED
244
244
500 PSI
300 PSI
150 PSI
150

FIG. 14

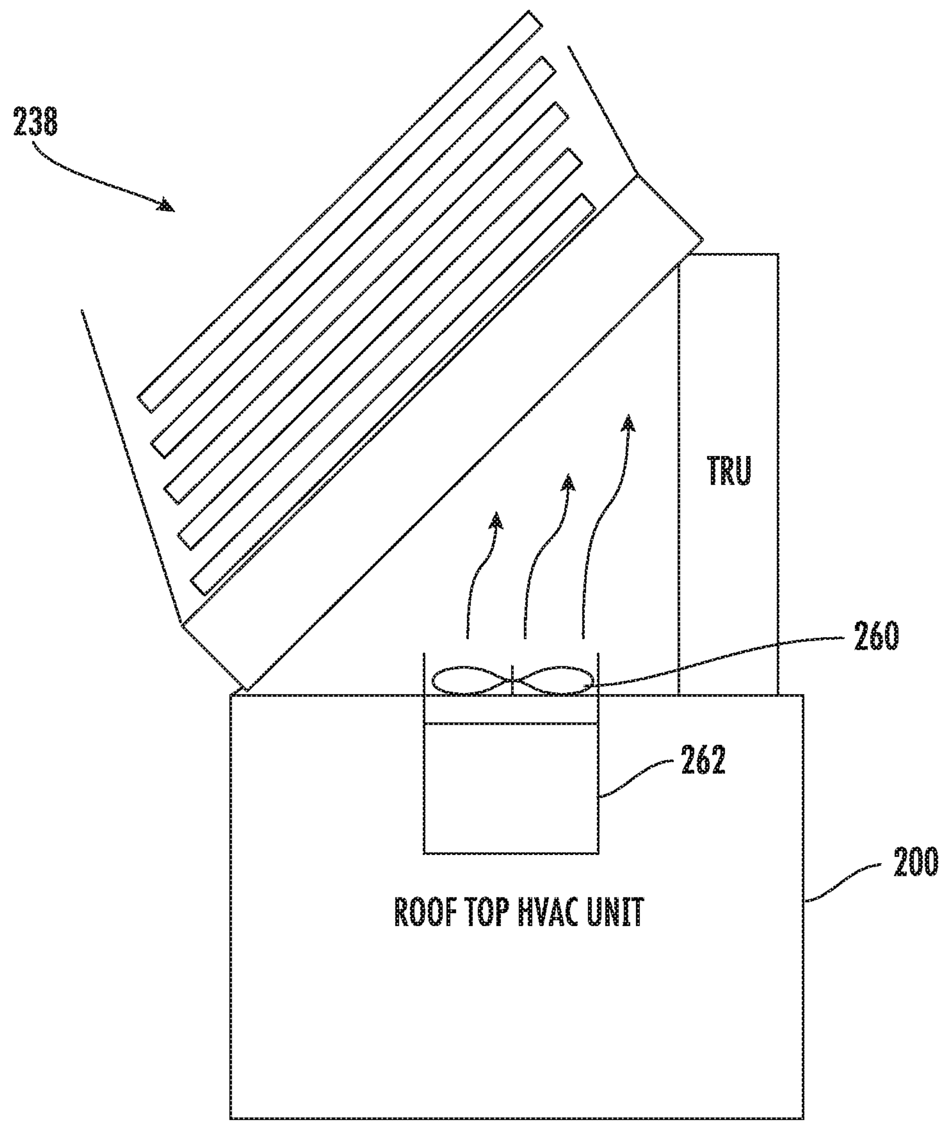

FIG. 15

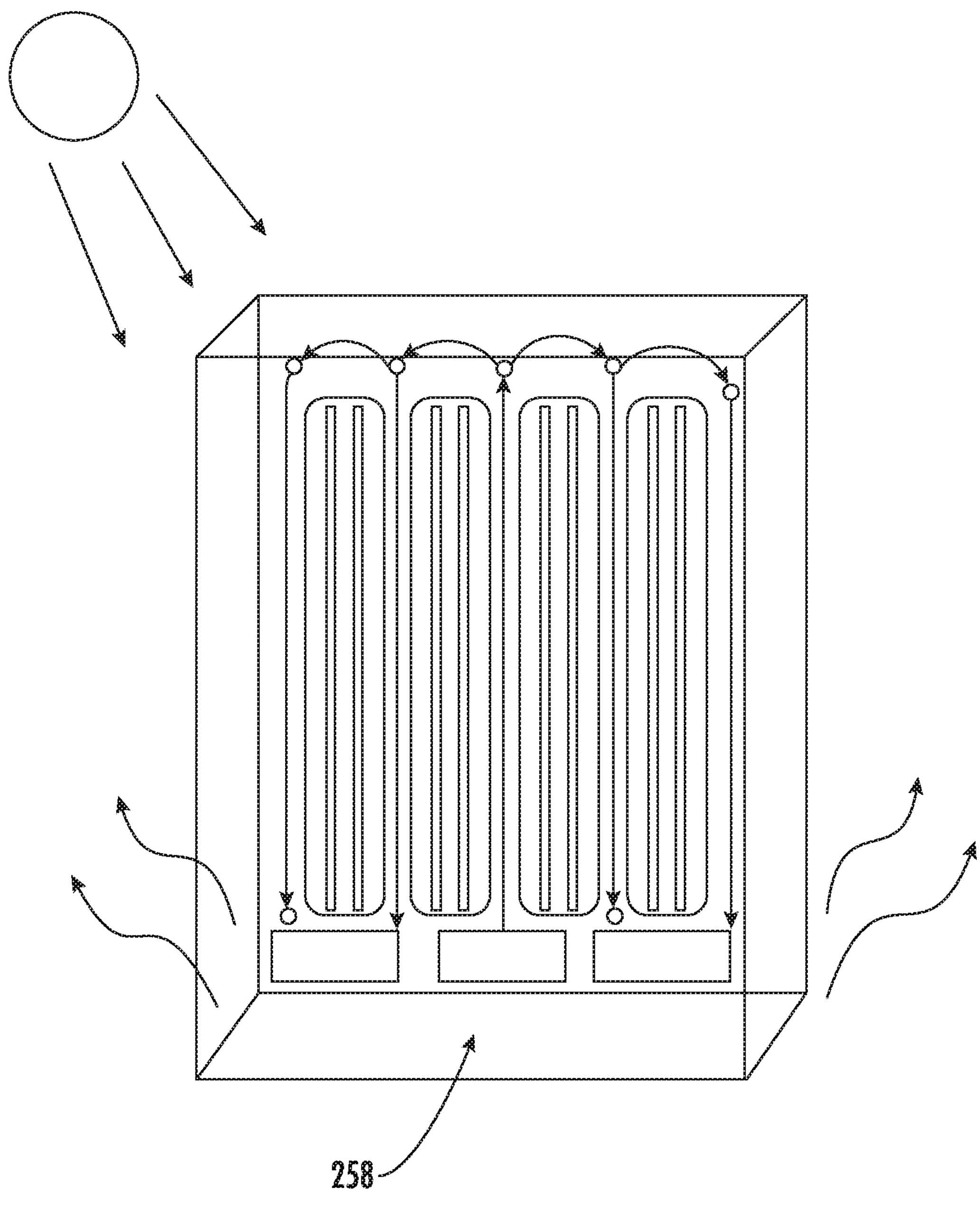

FIG. 16

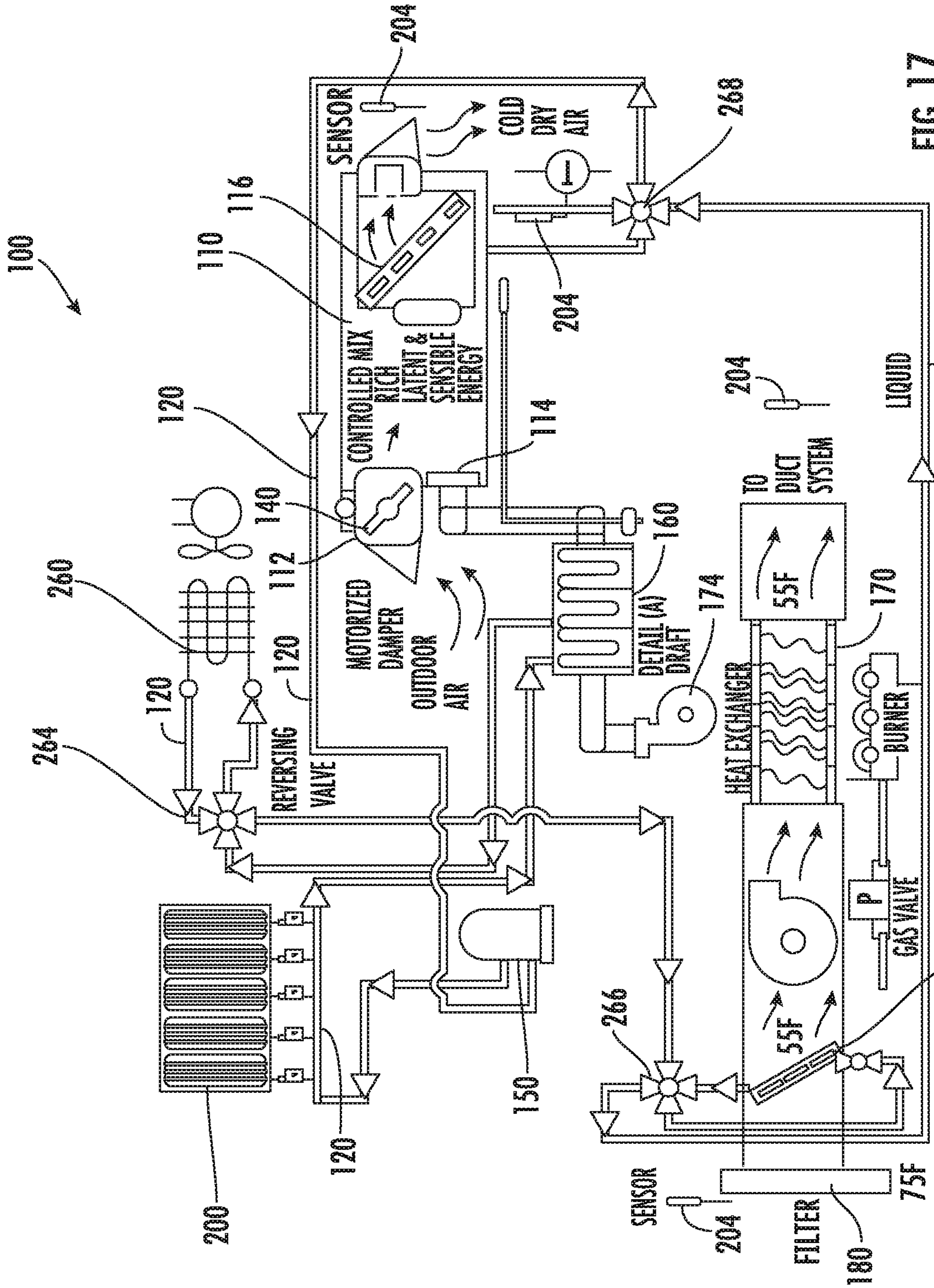
100
110
116
204
268
COLD DRY AIR
SENSOR
CONTROLLED MIX RICH LATENT & SENSIBLE ENERGY
120
114
140
112
MOTORIZED DAMPER
OUTDOOR AIR
260
120
REVERSING VALVE
264
160
DETAIL (A) DRAFT
174
TO DUCT SYSTEM
55F
170
HEAT EXCHANGER
BURNER
LIQUID
GAS VALVE
P
130
266
150
200
SENSOR
FILTER
180
75F

FIG. 17

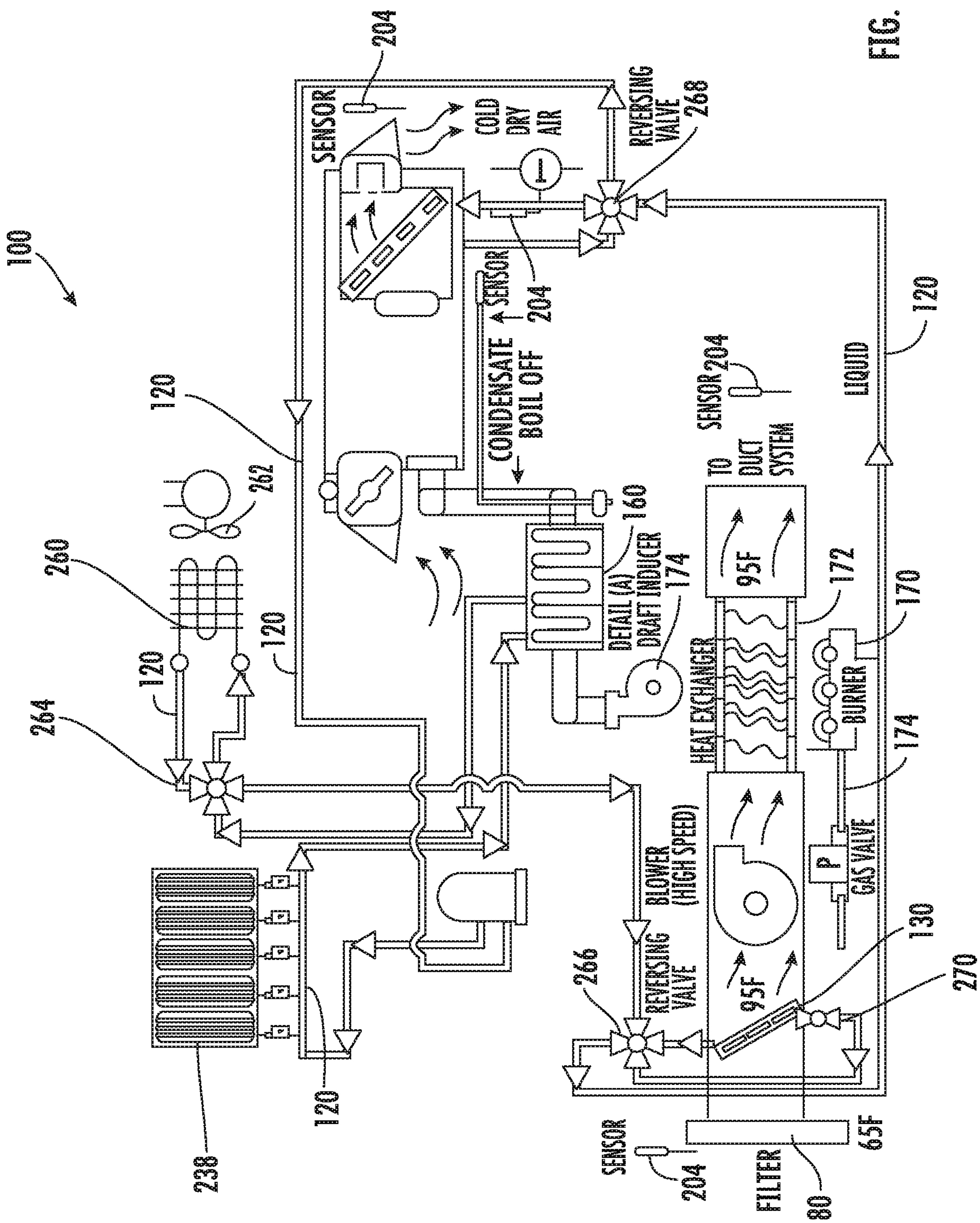
100
204
SENSOR
COLD
DRY
AIR
REVERSING
VALVE
268
204
SENSOR
CONDENSATE
BOIL OFF
120
262
260
120
120
264
SENSOR
204
TO
DUCT
SYSTEM
LIQUID
120
160
DETAIL (A)
DRAFT INDUCER
174
95F
172
170
HEAT EXCHANGER
BURNER
174
BLOWER
(HIGH SPEED)
GAS VALVE
P
266
REVERSING
VALVE
95F
130
270
238
120
SENSOR
204
FILTER
180
65F

FIG. 18

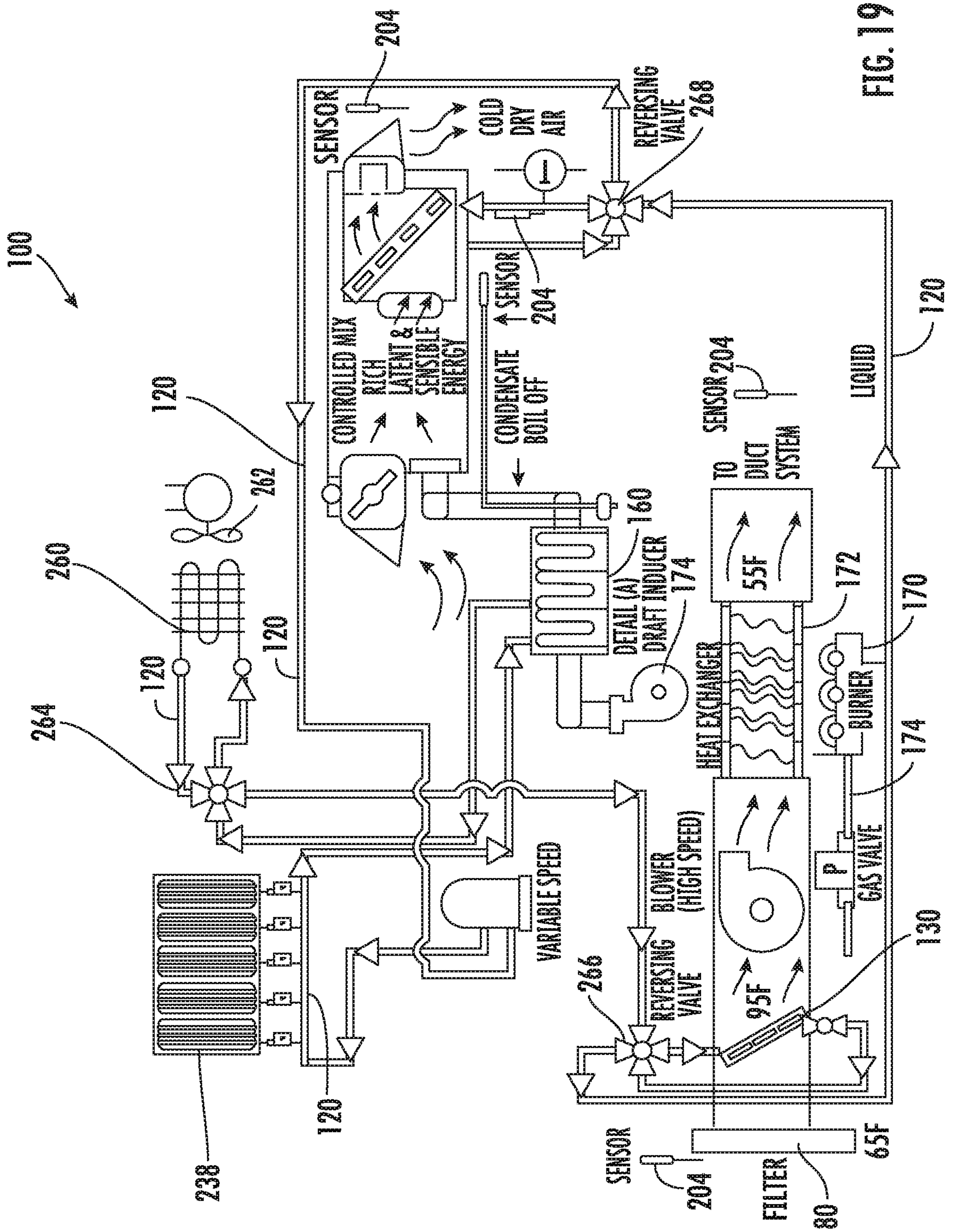
100
204
SENSOR
COLD
DRY
AIR
REVERSING
VALVE
268
CONTROLLED MIX
RICH
LATENT &
SENSIBLE
ENERGY
SENSOR
204
CONDENSATE
BOIL OFF
120
262
260
160
DETAIL (A)
DRAFT INDUCER
174
TO
DUCT
SYSTEM
SENSOR 204
LIQUID
55F
172
170
HEAT EXCHANGER
BURNER
174
264
120
VARIABLE SPEED
BLOWER
(HIGH SPEED)
P
GAS VALVE
266
REVERSING
VALVE
95F
130
238
SENSOR
204
FILTER
180
65F

FIG. 19

SYSTEM AND METHOD FOR HEAT AND ENERGY RECOVERY AND REGENERATION

CROSS-REFERENCE TO RELATED APPLICATION

This application is a continuation-in-part of U.S. patent application Ser. No. 15/897,692 filed Feb. 15, 2018 which claims the benefit of U.S. Provisional Patent Application Ser. No. 62/620,202, filed on Jan. 22, 2018, the contents of which are herein incorporated by reference in their entirety.

FIELD OF THE INVENTION

This invention relates generally to the field of air conditioning and heating systems, and more particularly, a heat and energy recovery system using solar energy for efficiently heating and cooling a space.

BACKGROUND OF THE INVENTION

The standard method of utilizing fossil fuels for heating involves firing the fuel in a controlled heating chamber or heat exchanger. The heat released on burning the fuel is drawn away by a fluid, often air or water, flowing around exterior surface of the heat exchanger. Heat transfer occurs spontaneously from a heat exchanger to the surrounding air or water in the conditioned space. Waste heat or emissions from the combustion reaction is allowed to flow outdoors, often via flue piping to a chimney or stack. The efficiency of the furnace or boiler is quantified in terms of the amount of heat extracted from the heat exchanger and used to heat the conditioned space and the amount of heat and by-products permitted to escape through the flue. Often, the efficiency of a furnace or boiler is indicated on the product at the point of sale.

Releasing carbon- and heat-saturated emissions into the atmosphere can impact the environment. Carbon dioxide in the atmosphere contributes to the greenhouse effect. An average residential furnace for natural gas, LPG or oil of low-to-medium efficiency will emit 1 million BTU's of waste heat into the atmosphere per day. Commercial and industrial units discharge hundreds of millions, and occasionally billions, of BTUs per unit per day. These common and traditional methods of discharging the flue gas into the atmosphere are wasteful and inefficient. Moreover, moving heat efficiently through a refrigeration circuit requires a significant amount of electrical power. Electrical consumption is deducted from the total thermal output to calculate the operating efficiency of a heating system.

In addition, conventional heat recovery system uses fossil fuels for a thermal chemical reaction in furnaces and boilers or electrical consumption to operate a refrigeration circuit to move thermal energy from one location to another. The fossil fuel burning furnace ignites fuel such as propane, natural gas or oil and allow flames to heat up a heat exchanger in which air or water is forced around the exchanger to absorb the heat and distribute the thermal energy within a conditioned space. Air conditioners or heat pumps use a refrigeration circuit to absorb heat energy from one location and release that thermal energy in another location to heat or cool a conditioned space.

Although recent developments intelligent or learning industry of thermostats has demonstrated some savings by knowing when a conditioned space is occupied at different times or patterns, the efficiency improvement is limited. Further improvements are possible to achieve a more efficient heat recovery system.

SUMMARY OF THE INVENTION

In view of the foregoing, it is an object of the present invention to provide an improved heat recovery system. In one embodiment, the invention is directed to a heat and energy recovery system. According to one embodiment of the present invention, a heat recovery system includes a compressor, a solar panel, and a first heat exchanger and a second heat exchanger in fluid connection to form a fluid circuit. The compressor is configured to facilitate fluid movement in the fluid circuit between the solar panel, the first heat exchanger and the second heat exchanger. The first exchanger and the second heat exchanger function as a condenser and evaporator respectively depending on a cooling cycle or a heating cycle. The solar panel includes a plurality of solar cells connected in parallel, and each solar cell includes a plurality of metal tubes for fluid to pass through. A temperature sensor is mounted within each of the solar cells and configured to measure temperature inside the respective solar cell. Each solar cell is connected to the fluid circuit via a respective pressure valve, and the status of the pressure valve is configured to depend on the measurement of the temperature sensor in the respective solar cell. The solar panel further includes a bypass channel for fluid to bypass the solar panel when all the pressure valves are closed.

According to another embodiment of the present invention, a method of recovering heat and energy includes feeding excess heat and waste products emitted as a result of fuel combustion into a mixing chamber comprising a first heat exchanger coupled with a fluid containing conduit circuit. Air is fed into the mixing chamber for initiating a reaction with the waste products to produce a reaction product with potential energy. Heat energy exchange is effectuated through the reaction product and excess heat interacting with the first heat exchanger, whereby the temperature and pressure of the fluid within the first heat recovery exchanger and conduit circuit increases. The fluid in the conduit circuit is further pressurized by utilizing heat energy from the exhaust gas and waste products emitted from fuel combustion and thermal energy obtained via a solar panel. Heat energy is exchanged by forcing air over a second heat exchanger that is in fluid communication with the pressurized fluid-containing conduit circuit outside the mixing chamber.

According to another embodiment of the present invention, a method of recovering heat and energy includes feeding air into a mixing chamber to effectuate heat energy exchange via interacting with a first heat exchanger in the mixing chamber. Heat energy is exchanged by forcing air over a second heat exchanger outside the mixing chamber that is in fluid communication with the first heat exchanger. Fluid contained in the fluid circuit connecting the first heat exchanger and the second heat exchanger is pressurized using heat energy removed by the first heat exchanger and the second heat exchanger and thermal energy obtained via a solar panel.

These and other objects, aspects and advantages of the present invention will be better appreciated in view of the drawings and following detailed description of preferred embodiments.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is an illustration of one embodiment of a heat recovery system of the present invention in a heating cycle;

FIG. 2 is an illustration of another embodiment of a heat recovery system of the present invention in a heating cycle;

FIG. 3 is an illustration of another embodiment of a heat recovery system of the present invention in a cooling cycle;

FIG. 4 is an illustration of another embodiment of a heat recovery system of the present invention in a cooling cycle;

FIG. 5 is a block diagram of a heat recovery system employing a central thermal recovery unit;

FIG. 6 is a block diagram illustrating the central thermal recovery unit of FIG. 5;

FIG. 7 is a flowchart illustrating a method of recovering heat and energy according to one embodiment of the present invention;

FIG. 8 is another flowchart illustrating a method of recovering heat and energy according to another embodiment of the present invention;

FIG. 9 is an example solar thermal cell of the present invention;

FIG. 10 is an example solar panel have a plurality of solar thermal cells;

FIG. 11 is an example solar panel connected to a compressor of a heat recovery system;

FIGS. 12-14 illustrate examples of exposure of a solar panel to the sun at different time of a day;

FIG. 15 illustrates a solar panel interacting with a roof top unit of a heat recovery system, according to one embodiment of the present invention;

FIG. 16 illustrate a detailed view of a solar panel interacting with a roof top unit of a heat recovery system, according to one embodiment of the present invention;

FIG. 17 illustrates an embodiment of a heat recovery system of the present invention incorporating a solar panel in a cooling cycle, according to one embodiment of the present invention;

FIG. 18 illustrates another embodiment of a heat recovery system of the present invention incorporating a solar panel in a heating cycle, according to one embodiment of the present invention; and

FIG. 19 illustrates another embodiment of a heat recovery system of the present invention incorporating a solar panel in a heating cycle, according to another embodiment of the present invention.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

As illustrated in the accompanying drawings, the present invention is directed to a heat and energy recovery system and methods of using the same. Such heat recovery devices may be adapted for use in a furnace of a heating, ventilation and air conditioning (HVAC) system or any other system where heat energy from fuel combustion is utilized for heating or cooling air spaces.

According to one embodiment of the invention, a heat recovery system **100** in a heating cycle is illustrated in FIG. **1**. The system **100** includes a mixing chamber **110**, preferably insulated, which includes an air intake **112** and an emissions intake **114** for receiving exhaust gas and waste products emitted from fuel combustion. The mixing chamber **110** may be made from a variety of metals or alloys. Preferably, the insulated chamber **110** is made of stainless steel and titanium alloy.

The system **100** further includes a first heat recovery exchanger **116** contained within the mixing chamber **110**. The first heat exchanger **116** is structured for contacting a mixture made up of air introduced via the air intake **112** and exhaust gas and waste products (made up of oxygen starved, carbon emissions) introduced via the emissions intake **114**. A coil sensor **204** can also be in contact with the primary heat recovery exchanger **116** to relay any problems with the functionality of the first heat exchanger **116** to a central processing unit (discussed later herein). The first heat exchanger **116** can be made from a variety of metals and alloys that are ideal for heat exchange, such as but not limited to copper, aluminum and the like. The first heat exchanger **116** can be in the form of a hermetically sealed heat recovery coil.

The air intake **112** can be structured as a single intake or multiple intakes. The intake(s) may be adapted to introduce outdoor air, indoor air or both. Additionally, in some embodiments it can be desirable to generate a pressurized environment within the insulated chamber **110**. As such, one or more of the air intake(s) **112** can connect to a pressure regulator inducer blower **140** that is part of a pressure equalization system to assist in pressurizing the air inside the mixing chamber **110**. The inducer blower **140** can include a variable speed motor that is controlled by sensors that detect proper temperature and/or humidity and/or pressure of the air inside the mixing chamber **110**.

The first heat recovery exchanger **116** is further interconnected to a fluid circuit **120** for conveying fluid therein. The system **100** is interconnected to a second heat exchanger **130** outside the mixing chamber **110** such that the second heat exchanger **130** is in fluid communication with the fluid circuit **120**. The second heat exchanger **130** and the first heat recovery exchanger **116** are collectively interconnected via the fluid circuit **120** such that the first heat recovery exchanger **116** contacts the mixture made up of air introduced via the air intake **112** and exhaust gas and waste products introduced via the emissions intake **114** within the insulated chamber **110**, while the heat extraction exchanger **130** contacts air to be heated outside of the insulated chamber **110**.

A compressor **150** is utilized to assist in fluid (e.g., refrigerant) flow between the first heat recovery exchanger **116** and the second heat exchanger **130** via the fluid circuit **120**. The heating of cooler refrigerant in the first heat recovery exchanger **116** during the operation of the system **100** results in a pressure increase inside the exchanger and the fluid circuit **120**, resulting in heat-absorbed refrigerant being pushed to an area of lower pressure. Such pushing allows a large part of refrigerant flow in the circuit (ca. 50%) to be achieved without any compressor assistance, limiting the amount of electrical energy required; therefore, a large compressor may not be necessary in most embodiments of the system **100** to get sufficient refrigerant flow. A micro-compressor is preferably utilized in embodiments of the invention to increase energy efficiency. The compressor **150** only operates at a speed as needed.

The system **100** can further include a ventilator outdoor air intake **180** that is structured to be in communication with the heat extraction exchanger **130** for heating outdoor air as it is drawn into the supply air intake of a heating apparatus or furnace **170**. The ventilator outdoor air intake **180** directs air into the airstream being drawn across the heat extraction exchanger **130** such that it can be heated by the energy efficient process utilized in the heat recovery system **100**.

The pressurized vapor is condensed at high pressure and temperature inside the condensing coil of the second heat exchanger **130**. During the condensation of refrigerant, heat is released and provided heated air to a HVAC system. The liquid refrigerant is then transported to the evaporation coil of the first heat exchanger **116**, which lowers the pressure and goes on to another circle. The flow of refrigerant in the fluid circuit **120** between each of the components of the system is illustrated by arrows in FIG. **1**.

Referring to FIG. **2**, a furnace **170** is used in addition to the compressor **150** described in FIG. **1** in a higher heating demand. In the depicted embodiment, the compressor **150** is configured to pass fluid (e.g., refrigerant) through a first thermal exchange coil **160** heated by exhaust gas and waste product generated by the furnace **170**, thereby further increasing refrigerant pressure in the fluid circuit **120**. The furnace **170** input exhaust originates from an oil-burning furnace or other type of furnaces such as furnaces burning natural gas sources. The furnace **170** includes a furnace exhaust **172** and a furnace intake **174**. The emissions intake **114** is adapted to be in communication with the furnace exhaust **172** of the furnace to receive exhaust gas and waste products resulting from fuel combustion in the furnace **170**. A furnace inducer blower **174** can be in connection with the furnace exhaust **172** to actively draw exhaust from the furnace **170** into the emission intake **114** of the mixing chamber **110**.

The compressor **150** pumps refrigerant into the second heat exchanger **130** while exhaust gas and waste product generated by furnace **170** pass through metal tubes of the thermal exchange coil **160**. The pressurized fluid (e.g., refrigerant) vapor passing through the thermal exchange coil **160** is fed into a condensing coil of the second thermal exchanger **130**. Waste heat or flue gas generated by furnace **170** is thus utilized to increase pressure of refrigerant in the fluid circuit **120** and directed into the mixing chamber **110**, wherein hot exhaust gas is combined with ambient air to create a rich 40° F. mixture for perfect extraction by the evaporator coil of the first heat exchanger **116**. The mixture can be adjusted so that is always around 40° F. by monitoring the entrance of the fresh air entering into the mixing chamber **110**.

This mixture passes over the first heat recovery exchanger **116**. The fluid (e.g., refrigerant) in the first exchanger **116** can extract a large amount of heat energy from the mixture and transfer the heat energy to the second heat exchanger **130** via the fluid circuit **120** to warm the indoor air.

The exhaust heat energy generated by the furnace **170** dramatically increases the pressure in the refrigerant, enabling the compressor to operate at 50% or less of capacity. In other words, only a fraction of the electricity is used to accomplish the same transfer of thermal energy further downstream in the refrigeration circuit. By using waste energy from furnace **170**, the electrical consumption of the compressor **150** can be greatly reduced. In addition, by controlling the outside air introduced to the evaporation coil of the first heat exchanger **116**, the efficiency of the compressor **150** can be maximized. The flow of refrigerant in the fluid circuit **120** between each of the components of the system is illustrated by arrows in FIGS. **1** and **2**.

The embodiment illustrated in FIGS. **1** and **2** are typically designed for use when the input exhaust originates from the burning of cleaner burning propane or other natural gases, such as but not limited to, a natural gas-burning furnace component of a HVAC unit.

The system **100** uses the waste heat of the furnace heater to boost the pressure in the refrigeration cycle reducing the need for mechanical energy by the compressor. The heat absorbed into the refrigeration cycle is transferred back into the buildings ventilation system. Waste flue gas are used several times in boosting efficiency before being fully absorbed and returning the thermal energy back into the building and greatly increases efficiencies over 100% by way of thermal and mechanical efficiency.

In addition, the system **100** significantly reduces fossil fuel use, as the condensing coil of the second heat exchanger utilizes the thermal energy absorbed from the flue gas and environmental air mixture. The condensing coil of the second thermal exchanger **130** releases heat into the ventilation system prior to the furnace heat exchanger. As a result, instead of heating the furnace from 65° F. and return air to 120° F., the furnace heat exchanger only needs to heat the air from 95° F. to 120° F., thanks to the condensing coil of the second heat exchanger **130**, which utilizes heat from waste heats in several respects.

The system **100** could be adapted to attach to any type of furnaces, resulting in an increased efficiency of the system. Carbon discharge, emission temperature, and humidity may also be reduced if the system **100** is utilized with a furnace.

Referring to FIG. **3**, similar to the heating cycles depicted in FIGS. **1** and **2**, a compressor **150** utilizes waste heat to pressure refrigerant flow in a cooling cycle. The same benefits apply in the cooling cycle as mentioned in the stage heating cycle. In this scenario, the first heat exchanger **116** includes a condensing coil and the second heat exchanger **130** includes an evaporation coil.

In this scenario, the compressor **150** pumps the refrigerant through the condensing coil of the first heat exchanger **116** in the mixing chamber **110** and releases heat absorbed from an indoor environment. Fluid (i.e. refrigerant) in the fluid circuit **120** is then passed through an evaporation coil of the secondary heat exchanger **130** in which heat from a building is passed over to the refrigerant and raised its pressure on a high-pressure side of the system. The refrigerant exiting the evaporation coil in the second heat exchanger **130** is then passed through the compressor **150** in which further increases the refrigerant pressure before entering the first heat exchanger **116** for another fluid moving cycle.

Referring to FIG. **4**, in a high demand period during a cooling cycle, a second compressor **150**B (e.g., a two-stage compressor) is used in addition to the compressor **150** configured to pump refrigerant through the condensing coil of the first heat exchanger **116** in the mixing chamber **110**, as described in FIG. **3**. The second compressor **150**B pumps the refrigerant through a second heat exchange coil **162**. In this scenario, the heat removed from the indoor environment is passed through a first portion **162**A of the second heat coil **162**. A second portion **162**B of the second heat exchange coil **162** is configured to pass through fluid exiting the second compressor **1506** to absorb heat obtained by the first portion **1626** of the second heat exchange coil **162**, via, for example, thermal radiation and/or convection. As such, the heat from the indoor environment is passed over to the refrigerant connected to the second compressor instead of being released to the outdoor environment. All the absorbed heat is then released in the first heat exchanger **116** though the refrigerant, the cooled refrigerant is passed through evaporation coils of the secondary heat exchanger **130**, and the next cycle commences.

The heat recovery system **100** utilizes what would be wasted or discharged heat in a conventional HVAC system to increase the pressure within the refrigeration cycle, which in turn decreases the electrical consumption of the compressors **150** and **1508** to achieve 100% heat transfer. The compressor **150** and **1508** can operate at 50% or less of capacity and still achieve 100% of heat transfer.

The system **100** can also employ artificial intelligence (AI) to monitor and control one or more components of the system. AI can be used to optimize control of the operation parameters of the variable-speed compressors, multi-speed blowers and fans, heat exchanger mixing components, motorized dampers and other components of the system.

Referring to FIG. **5**, in accordance with another embodiment of the present invention, the system includes at least one controller **202** operably linked to respective operating components (e.g., and/or operating zones) of the heat recovery system. For example, the operating components coupled to the at least one controller **202** includes a compressor (e.g., compressor **150**), a heat exchanger coil (e.g., a condensing coil, an evaporating coil), a fan, a motor, a valve and other functional components described in FIGS. **1-4**.

The system **100** also includes at least one sensor **204** configured to collect at least one environmental measurement and system-related data. The environment measurement and the system-related data can include internal and external temperature and pressure, humidity, barometric pressure, dew point, wind direction, sun peak and angle, annual precipitation, geographical location and elevation of the system, thermostats setting, chemical analysis at specific point of the system, carbon dioxide level, motion level, fuel consumption, electrical consumption, fuel price, and electrical energy price in real time.

The system **100** further includes a central thermal recovery unit **206** in signal communication with the at least one controller **202** and the at least one sensor **204**. The central thermal recovery unit **206** is configured for determining an operating instruction based on the at least one environmental measurement and system-related data received from the at least one sensor. The central thermal recovery unit **206** is further configured to transmit the operating instruction to the at least one controller **204**. The operating instructions includes a specific operation sequence of a series of operating components/zones controlled by the at least one controller.

The central thermal recovery unit **206** can also be configured for operating instructions to be based on environment measurements and system-related data retrieved from a third-party database **208**. For example, the third-party database **208** can include information such as weather conditions, user-preferred comfort level, fuel cost, air quality, and the like. The information can facilitate the central thermal recovery unit **206** to determine a set of operating instructions that will improve the efficiency and extend the life of the equipment and components of the system.

The at least one controller **202** and/or the at least one sensor **204** can also be used to detect potential issues concerning certain mechanical parts and/or zones of the system and transmit these issues to the central thermal recovery unit **206**. For example, the at least one controller **202** and/or the at least one sensor **204** can detect a depleted refrigerant and/or a leak at a specific location within the system. The central thermal recovery unit **206** can in turn determine parts in need of repair or replacement and repair or replacement sequences.

The central thermal recovery unit **206** can be configured to determine the operating instructions (e.g., temporal operating sequence) using an adaptive learning method. For example, the central thermal recovery unit **206** can record and analyze operation patterns, compare the efficiencies of each operation pattern, and on this basis predict the most efficient sequence under certain environmental/system conditions. The adaptive learning method will make the heat recovery system more efficient from the continuous determination and implementation of a more efficient operation pattern. The central thermal recovery unit **206** will enable a conventional HVAC system to achieve dramatically higher efficiency levels. As an example, the operation pattern can include motor running time, internal and external temperature and pressure, fuel combustion rate, fan speed and duration, inducer flow level, blower pressure and speed, ignition timing, and the like. The operation patterns that result in high efficiency can then be transmitted and shared with other thermal recovery units via a network.

The central thermal recovery unit **206** can be configured to achieve the highest system efficiency under given conditions. For example, if the price of fuel depends on the time of day, the central thermal recovery unit can account for fuel price to calculate system efficiency. As another example, for a system that can operate on certain cycles of either refrigeration or fossil fuels, if electric prices are more advantageous than natural gas at a certain time of the day, the system can favor operational cycles that use electricity over natural gas at that time of the day.

The central thermal recovery unit **206** can also be configured to achieve a balance between high system efficiency and low thermal pollutant release. For example, for a heat recovery system located in certain valleys in certain states, for instance, Simi Valley, Calif., the release of a certain pollutant will contribute to smog accumulation. In such cases, the system can be configured to monitor the release of CO/CO_2 and other system waste products and to balance energy consumption, system efficiency and materials release accordingly.

As another example, for a refrigeration system, the central thermal recovery unit **206** can be programmed to collect operating data and environmental data of the system on a periodic basis and respond with operating instructions. The operating instructions can include a sequence and duration for operating a compressor, an evaporator, a condenser and a pressure device. Slight changes in the operating times and pressures of specific components will increase the efficiency of heat transfer and decrease the stress on system components. Subtle changes in the operation of each component under specific internal and external conditions can lead to significant improvements in the ability of the system to extract and transfer thermal energy.

Referring to FIG. **6**, the central thermal recovery system **206** includes a processor **218** for receiving and processing system related data such as operation parameters, sensor measurements, climate conditions and fuel information (e.g., fuel price, fuel consumption, etc.) and user preferences. The processor **218** can also be configured to output information such as operating instructions, system efficiency report, system pollutant release report, system operation history and analysis, and the like. This information can be stored in the database **220**.

Referring to FIG. **7**, according to one embodiment of the present invention, a method for recovering heat and energy includes, at step **702**, feeding excess heat and waste products emitted as a result of fuel combustion into a mixing chamber (e.g., mixing chamber **110**) having a first heat exchanger (e.g., first heat exchanger **116**) coupled with a fluid containing conduit circuit (e.g., conduit circuit **120**).

At step **704**, air is fed into the mixing chamber for initiating a reaction with the waste products to produce a reaction product that includes heat energy.

At step **706**, heat energy exchange is effectuated through the reaction product and excess heat interacting with the first heat exchanger (e.g. first heat exchanger **116**), whereby the temperature and the pressure of fluid within the first heat exchanger and the fluid circuit connected to the first heat exchanger rise.

At step **708**, fluid contained in the fluid circuit is pressurized by utilizing heat energy from the exhaust gas and waste products emitted as the result of fuel combustion. Specifically, a compressor (e.g., compressor **150**) is configured to pass fluid (e.g., refrigerant) through a first thermal exchange coil (e.g., the first thermal **160**) heated by exhaust gas and waste product generated by the furnace **170**, thereby further increasing refrigerant pressure in the fluid circuit **120**.

At step **710**, heat energy is exchanged by forcing air over a second heat exchanger (e.g., second heat exchanger **130**) that is in fluid communication with the pressurized fluid containing conduit circuit outside the mixing chamber (e.g., mixing chamber **110**).

Referring to FIG. **8**, according to another embodiment of the present invention, a method for recovering heat and energy includes, at step **802**, feeding air into a mixing chamber to effectuate heat energy exchange via interacting with a first heat exchanger in the mixing chamber.

At step **804**, heat energy is exchanged by forcing air over a second heat exchanger exteriorly of the mixing chamber that is in fluid communication with the first heat exchanger.

At step **806**, fluid contained in a fluid circuit connected between the first heat exchanger and the second heat exchanger is pressurized using heat energy removed by the first heat exchanger and the second heat exchanger from an indoor environment. For example, a second compressor (e.g., compressor **150**B) and a second heat exchange coil (e.g., **162**) are used to recapture heat removed from the indoor environment. Specifically, heat removed from the indoor environment is configured to pass through the second heat exchange coil and in turn heating fluid flowing through the second compressor. As shown in FIG. **4**, a first portion (e.g., **162**A) of the second heat exchange coil is configured to be heated by the heat removed from the indoor environment, and a second portion (e.g., **162**B) of the second heat exchange coil is configured to pass through fluid exiting the second compressor to absorb heat obtained by the first portion (e.g., **162**A) of the second heat exchange coil.

Solar energy can be included in the HVAC system **100** and thus further increase system efficiency. FIG. **9** depicts an example of a solar cell that is useful for the purpose. The solar cell **230** includes a plurality of metal tubes (e.g., copper tubes) **234** arranged in parallel inside a casing **232**. Tubes **234** are designed for fluid (e.g., refrigerant) to pass through. The casing **232** is depicted as having a half-round shell base made of metal or plastic and a glass top. Reflective sheets are placed between the casing **232** and the plurality of tubes **234**, which are placed between grommets **236** mounted on both ends of the casing **232** to prevent the tubes **234** coming into direct contact with the casing **232** and thus eliminate impact from vibration. The plurality of tubes **234** can be sprayed with a dark blue coating to resist corrosion. Refrigerant pumped by a compressor (not shown) can pass through the plurality tubes **234** and absorb thermal energy from the Sun. Refrigerant flows into the plurality of tubes **234** at one end of the solar cell **230** and out of the tubes **234** at the other end of the solar cell **230**. The refrigerant absorbs thermal energy when moving along the tubes **234**, increasing the temperature and therefore the pressure of the refrigerant and reducing the electrical energy requirement of the compressor.

FIG. **10** illustrates a solar panel **238**, which comprises three solar cells **230** connected in parallel. When the solar panel **238** is placed in direct sunlight, solar energy is utilized when the temperature of the solar cells **230** exceeds a certain level. Specifically, each solar cell **230** has a temperature sensor **240** attached to its respective copper tubes **234**. Each temperature sensor **240** is configured to determine the temperature within the respective solar cell **230**. Respective pressure valves **244** are coupled between respective three solar cells and a compressor (not shown). The status (e.g., open and closed) of a pressure valve **244** is determined by the measurement of the respective temperature sensor **240**. When the temperature in a solar cell exceeds the thermal energy needed to increase the refrigerant pressure to above a pressure threshold, the respective pressure valve **244** connected to the specific solar cell will open, allowing refrigerant to flow inside and absorb solar thermal energy. The solar thermal energy will increase the pressure to supplement the compressor, using less mechanical energy and less electrical energy. A bypass **248** is also placed in parallel with the solar panel when none of the solar cells **230** of the solar panel can supplement solar energy to the heat recovery system. The solar cells **230** can be other shapes (e.g., square) and includes many more metal tubes than depicted. This would allow more refrigerant heating by the sun. Additionally, the solar cells would allow two or more small heat sinks running alongside of the respective solar cells to increase heat retention.

FIG. **11** shows a solar panel **238** connected to a compressor **150** of a heat recovery system. Arrow **252** shows cold, low-pressure refrigerant flowing into the compressor **150**. Arrow **254** shows warm, high-pressure refrigerant flowing out of the compressor **150**. Arrow **256** shows warm, high-pressure refrigerant absorbing solar thermal energy from solar cells **230** and further increasing refrigerant temperature and pressure, and thus decreasing the consumption of electrical and mechanical energy.

FIG. **12** depicts an example of a "morning" position. The Sun's rays are concentrated on a side angle of a solar panel **238**, causing more direct sunlight to fall on the two cells on the left side than the others. As a result, the temperature rises substantially on the left side of the solar cells **230** of the solar panel **238**, the two respective pressure valves **244** on the left side become open, and the pressure of the refrigerant flow therethrough rises, facilitating the compressor **150** to increase the pressure on the high side of the fluid circuit. The three pressure valves **244** on the right are closed.

FIG. **13** illustrates a typical day. The Sun has risen directly above the solar panel **238**, and each solar cell receives direct sunlight. The temperature in each cell is elevated, and all the respective pressure valves **244** are open. The thermal energy absorbed by each solar cell increases the pressure of the refrigerant, dramatically lowering the operation speed of the compressor **150** needed to achieve the same level of heat transfer when the compressor **150** is operated without the benefit of thermal energy. This significantly increases the efficiency of the entire system in both heating and cooling modes.

FIG. **14** shows a typical late afternoon. The angle of the Sun is such that certain pressure valves **244** are open, allowing interaction with a refrigeration circuit of a heat recovery system. The pressure valves **244** of three solar cells on the left side are closed, and two pressure valves **244** on the right side are open, assisting the compressor **150** to increase the pressure of the refrigerant.

FIG. **15** depicts the solar panel **238** interacting with a rooftop unit of a heat recovery system. In the depicted embodiment, a condenser fan **260** of the heat recovery system extracts heat from an outdoor condenser coil **262** and transfers it to the solar panel **238** via convection. As such, heat normally released to the surroundings and wasted is thus used to heat the solar panel **238**. The outdoor condensing fan would drastically reduce the necessity for the furnace to operate and produce the same amount of thermal delivery. This would reduce greenhouse gas emissions and reduce utility cost of operation.

FIG. **16** is a detailed view of air flow from the condenser fan **260** through the solar panel **238**. Warm air inducted by the condenser fan (not shown) is pushed into one side **258** in the solar panel **238**, where the air warmed by waste heat warms the cells and moves the heat collected by the solar cells evenly throughout the solar panels, including solar cells that are not directly in the rays of the Sun at that moment. The heat of each solar cell is thus evenly distributed among solar cells, in case exposure to the Sun results in heating one area more than another. This also helps to prevent any boiling of refrigerant due to concentrated heating in one area of the solar panel **238**. The heat from the condensing coil is then pushed through the sides of the system and out into the surrounding environment.

FIG. **17** illustrates a thermal recovery system incorporating a solar panel **238**. In a cooling process, the variable speed compressor **150** pumps refrigerant towards the solar panel **238**, and the thermal energy absorbed from the solar panel **238** will increase the pressure of the refrigerant, allowing the variable speed compressor **150** to reduce mechanical compression and thus reduce electricity consumption. The refrigerant then flows through a first thermal exchange coil **160** to a first reversing valve **264**, which directs the superheated refrigerant through the condenser coil **260**. The refrigerant then flows to a second reversing valve **266** and directs the refrigerant into the evaporator coil of a second heat exchanger **130**, in which high-pressure, warm refrigerant is rapidly released and changes state to a cold, low-pressure vapor. The refrigerant is then directed to a third reversing valve **268** in which it by-passes the mixing chamber **110** used in heating cycle and back to the compressor **150** to repeat the cycle. The increased pressure obtained by the solar panel can greatly reduce the electrical consumption of the compressor.

FIG. **18** illustrates another heat recovery system incorporating a solar panel **238**. In a heating cycle, heat is extracted from the air even in cold conditions and returned to a building by efficient methods of operation. Refrigerant is pumped from the variable speed compressor **150** towards the solar panel **238**. The solar panel **238** increase the pressure of refrigerant on the high-temperature side of the refrigeration cycle, lowering the speed of the variable speed compressor **150**. This in turn lowers electrical consumption and allows heat to be delivered on a cold day to a building using little electrical energy and no fossil fuel. The refrigerant is sent through the first thermal exchange coil **160** to a first reversing valve **264**, bypassing the condensing coil **260**. Warm refrigerant is directed to the in-door coil, which here functions as a condensing coil of the second heat exchanger **130** and releases the heat absorbed from the outside air into the building. The refrigerant then passes through the thermal expansion valve **270** and cool the refrigerant, and the refrigerant flows to the third reversing valve **268**, where it is directed to the mixing chamber **110** for release as cold air. The refrigerant is then directed back to the compressor **150** to repeat the cycle.

FIG. **19** depicts another heat recovery system incorporating a solar panel **238**. In a heating cycle of a heat and recovery system incorporating solar panel **238**, refrigerant is pumped from the variable speed compressor **150** toward the thermal solar panel **238**. Thermal energy obtained by the solar panel **238** will increase the pressure of the refrigerant and reduce the speed of the variable speed compressor **150**. The warm refrigerant will flow to the first thermal exchange coil **160**, which will significantly heat the refrigerant with the flue gas from the combustion process. This will raise the refrigerant pressure and enable the compressor **150** to operate at a lower speed. Refrigerant flow to the first reversing valve **264** will bypass the condensing coil **260** and be sent to the second reversing valve **266**, which will direct the refrigerant to the indoor coil of the second heat exchanger **130**, which will act as a condensing coil, release the heat into a building, and significantly reduce the amount of fossil fuel required to achieve the results of a conventional heat recovery system. The present method will also eliminate thermal pollution and significantly reduce greenhouse gas emissions. The refrigerant then passes through the thermal expansion valve **270**, becoming a cold, low-pressure vapor. The refrigerant is then directed to the third reversing valve **268**, which sends it through the mixing chamber **110**. The refrigerant can thus capture the waste heat and flue gases generated by the heating apparatus or furnace **170**. All the latent heat is removed from the flue gas and returned to the building via the refrigeration cycle. In addition, the flue gas is mixed with the cold dry outdoor air, resulting in a large volume of energy-rich dew droplets for the evaporator coil **116** to remove if all available heat from the cold outdoor air will be used to heat the building via the refrigeration cycle. The refrigerant is then directed back to the compressor **150** to repeat the cycle.

In the heating mode, the condensing fan **260** spin in reverse direction. This would draw heat away from the solar panel **238** on winter days and into the evaporating coil **116** of the heat pump in the heating mode. This would dramatically increase the ability to transfer and absorb heat on very cold days. Those coldest days are when a heat pump system struggles to operate effectively.

In the heating mode, instead of the refrigerant being directed through the solar panel **238** as soon as it leaves the compressor **150** as in the cooling mode, the refrigerant goes through a standard heat pump cycle and is re-directed through the solar panel **238** after it travels through the evaporator coils **116** outside of the building. As such, heat is absorbed when the refrigerant travels back into the building and to the indoor coil second heat exchanger **130** to increase the heat within the building more efficiently.

The thermal recovery unit (TRU) of the present invention can improve efficiency of energy recovery from the hot flue gas and function as a variable capacity cooling stage in cooling mode. In heating mode, the TRU can increases the overall heating efficiency over 100% annual fuel utilization efficiency. Machine learning algorithms can further adjust the performance to become more efficient each cycle.

The TRU of the present invention can be installed as a rooftop unit. The installation process does not require separate panels, labor, air conditioning and refrigeration (ACR) charging, wiring or piping. An installer can remove an old roof top unit and install a new one since all electrical and mechanical devices are internal. Each TRU should face south in the northern hemisphere to make the effective use of solar energy. An installer can easily match an old roof curb to accommodate the TRU of the present invention with little labor or cost by simply mounting an adapter to the old curb.

The TRU incorporated with solar panels can utilize solar energy to increase the efficiency of heat recovery system and reduce utility cost. The TRU of present invention has advantages over conventional systems in that it produces less greenhouse gas and makes effective use of solar energy and heat energy that is typically released into the environment. The present invention enables a dramatic reduction in electrical consumption by the compressors in the refrigeration circuit, which in turn transfer the thermal energy from the flue gas and environmental mixture flow back into the system to the improve system efficiency.

The current invention can be used in residential and/or commercial air conditioning and heat pumps. The residential condensing units and/or heat pumps can work in conjunction with a typical residential furnace.

It should be understood that the foregoing description is only illustrative of the aspects of the disclosed embodiment. Various alternatives and modifications can be devised by those skilled in the art without departing from the aspects of the disclosed embodiment. Accordingly, the aspects of the disclosed embodiment are intended to embrace all such alternatives, modifications and variances that fall within the scope of the appended claims. Further, the mere fact that different features are recited in mutually different dependent or independent claims does not indicate that a combination of these features cannot be advantageously used, such a combination remaining within the scope of the aspects of the invention.

The foregoing is provided for illustrative and exemplary purposes; the present invention is not necessarily limited thereto. Rather, those skilled in the art will appreciate that various modifications, as well as adaptations to particular circumstances, are possible within the scope of the invention as herein shown and described.

What is claimed is:

1. A heat recovery system, comprising:

a compressor, a solar panel, a first heat exchanger and a second heat exchanger in fluid connection to form a closed fluid circuit; and

wherein the compressor is configured to facilitate fluid movement in the closed fluid circuit between the solar panel, the first heat exchanger and the second heat exchanger;

wherein the first exchanger and the second heat exchanger function as a condenser and evaporator respectively depending on a cooling cycle or a heating cycle;

wherein the solar panel includes a plurality of solar cells connected in parallel, and each solar cell includes a plurality of metal tubes for fluid to pass through;

wherein a temperature sensor is mounted within each of the solar cells and configured to measure temperature inside the respective solar cell;

wherein each solar cell is connected to the circuit via a respective pressure valve, and the status of the pressure valve is configured to depend on the measurement of the temperature sensor in the respective solar cell;

wherein the solar panel further includes a bypass channel for fluid to bypass the solar panel when all the pressure valves are closed; and

wherein the heat recovery system further comprises an outdoor condenser and a condenser fan configured to push warm air induced from the condenser into the solar panel to increase the temperature of the plurality of solar cells via a convective effect in a cooling mode.

2. The heat recovery system of claim **1**, wherein a respective pressure valve is configured to open when the temperature in a respective solar cell is above a certain threshold.

3. The heat recovery system of claim **1**, wherein a respective pressure valve is configured to close when the temperature in a respective temperature sensor is below a certain threshold.

4. The heat recovery system of claim **1**, wherein metal tubes in the plurality of solar cells are made of copper.

5. The heat recovery system of claim **1**, wherein the compressor is a variable speed compressor, and wherein the speed of the compressor depends on thermal energy absorbed via the solar panel.

6. The heat recovery system of claim **1**, wherein the outdoor condenser fan is further configured to draw heat away from the solar panel and into the evaporator in a heating mode.

7. A heat recovery system, comprising:

a compressor, a solar panel, a first heat exchanger and a second heat exchanger in fluid connection to form a closed fluid circuit; and

wherein the compressor is configured to facilitate fluid movement in the closed fluid circuit between the solar panel, the first heat exchanger and the second heat exchanger;

wherein the first exchanger and the second heat exchanger function as a condenser and evaporator respectively depending on a cooling cycle or a heating cycle;

wherein the solar panel includes a plurality of solar cells connected in parallel, and each solar cell includes a plurality of metal tubes for fluid to pass through;

wherein a temperature sensor is mounted within each of the solar cells and configured to measure temperature inside the respective solar cell;

wherein each solar cell is connected to the circuit via a respective pressure valve, and the status of the pressure valve is configured to depend on the measurement of the temperature sensor in the respective solar cell;

wherein the solar panel further includes a bypass channel for fluid to bypass the solar panel when all the pressure valves are closed;

wherein the heat recovery system further comprises a mixing chamber configured for mixing an air intake and an emissions intake, the emissions intake receiving exhaust gas and waste products emitted as a result of fuel combustion;

wherein the first heat exchanger is contained within the mixing chamber for contacting a mixture of air introduced via the air intake and exhaust gas and waste products introduced via the emissions intake such that heat exchange is effectuated with the fluid in fluid circuit;

wherein the second heat exchanger is in fluid communication with the first heat exchanger via the compressor, the compressor is configured to facilitate fluid movement in a fluid circuit between the first and second heat exchanger; and

wherein the compressor is configured to flow fluid through a first thermal exchange coil heated by exhaust heat as a result of fuel combustion and solar energy obtained via the solar panel, thereby increasing fluid pressure in the fluid circuit between the first and second heat exchanger and decreasing electrical consumption of the compressor in a heating cycle.

8. The system of claim **7**, further comprising a reversing valve configured to switch a direction of the fluid movement between the first heat exchanger and the second heat exchanger based on a cooling demand or a heating demand.

9. The heat recovery system of claim **7**, wherein in a heating cycle, the exhaust heat is generated by a furnace and passed through the first thermal exchanger coil and fed into the emission intake of the mixing chamber and the second thermal exchanger is functioned as a condensing coil and releases the heat into a building.

10. The heat recovery system of claim **7**, wherein in a cooling cycle, heat removed from an indoor environment and heat obtained from the solar panel are configured to pass through the second heat exchange coil and thereby increasing fluid temperature and increasing fluid pressure flowing through the compressor.

* * * * *

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Kaiser

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(45) **Date of Patent: Oct. 12, 2021**

(54) **THERMAL CELL PANEL SYSTEM FOR HEATING AND COOLING AND ASSOCIATED METHODS**

(71) Applicant: **Commercial Energy Saving Plus, LLC**, Boca Raton, FL (US)

(72) Inventor: **Stewart Kaiser**, Palm Beach Gardens, FL (US)

(73) Assignee: **Commercial Energy Saving Plus, LLC**, Boca Raton, FL (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 91 days.

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(22) Filed: **May 28, 2019**

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F25B 29/00	(2006.01)

(Continued)

(52) **U.S. Cl.**

CPC ***F24S 90/00*** (2018.05); ***F24S 10/70*** (2018.05); ***F24S 50/00*** (2018.05); ***F24S 70/20*** (2018.05); ***F24S 80/30*** (2018.05); ***F24S 80/50*** (2018.05); ***F25B 29/003*** (2013.01); ***F25B 49/022*** (2013.01); *F25B 2700/19* (2013.01)

(58) **Field of Classification Search**

CPC .. F24S 90/00; F24S 10/70; F24S 50/00; F24S 70/20; F25B 29/00; F25B 49/02

See application file for complete search history.

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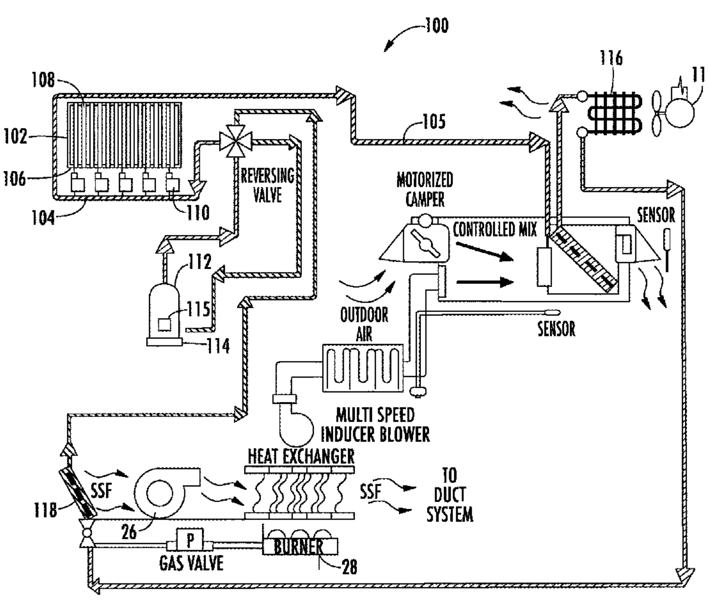
Primary Examiner — David J Laux
Assistant Examiner — Nikhil P Mashruwala
(74) *Attorney, Agent, or Firm* — Allen, Dyer, Doppelt & Gilchrist, P.A.

(57) **ABSTRACT**

A thermal cell panel system for heating and cooling using a refrigerant includes a plurality of solar thermal cell chambers, and a piping network for a flow of the refrigerant through the plurality of solar thermal cell chambers. In addition, the system includes a compressor having a motor coupled to a variable frequency drive ("VFD"), where the compressor is coupled to the piping network upstream of the plurality of solar thermal cell chambers and the VFD is configured to adjust a speed of the motor in response to the pressure of the refrigerant within the plurality of solar thermal cell chambers. The piping network includes an inlet manifold coupled to the inlet of each solar thermal cell chamber, and an outlet manifold coupled to the outlet of each solar thermal cell chamber.

16 Claims, 8 Drawing Sheets



(51) **Int. Cl.**

F25B 49/02	(2006.01)
F24S 10/70	(2018.01)
F24S 70/20	(2018.01)
F24S 80/30	(2018.01)
F24S 80/50	(2018.01)
F24S 50/00	(2018.01)

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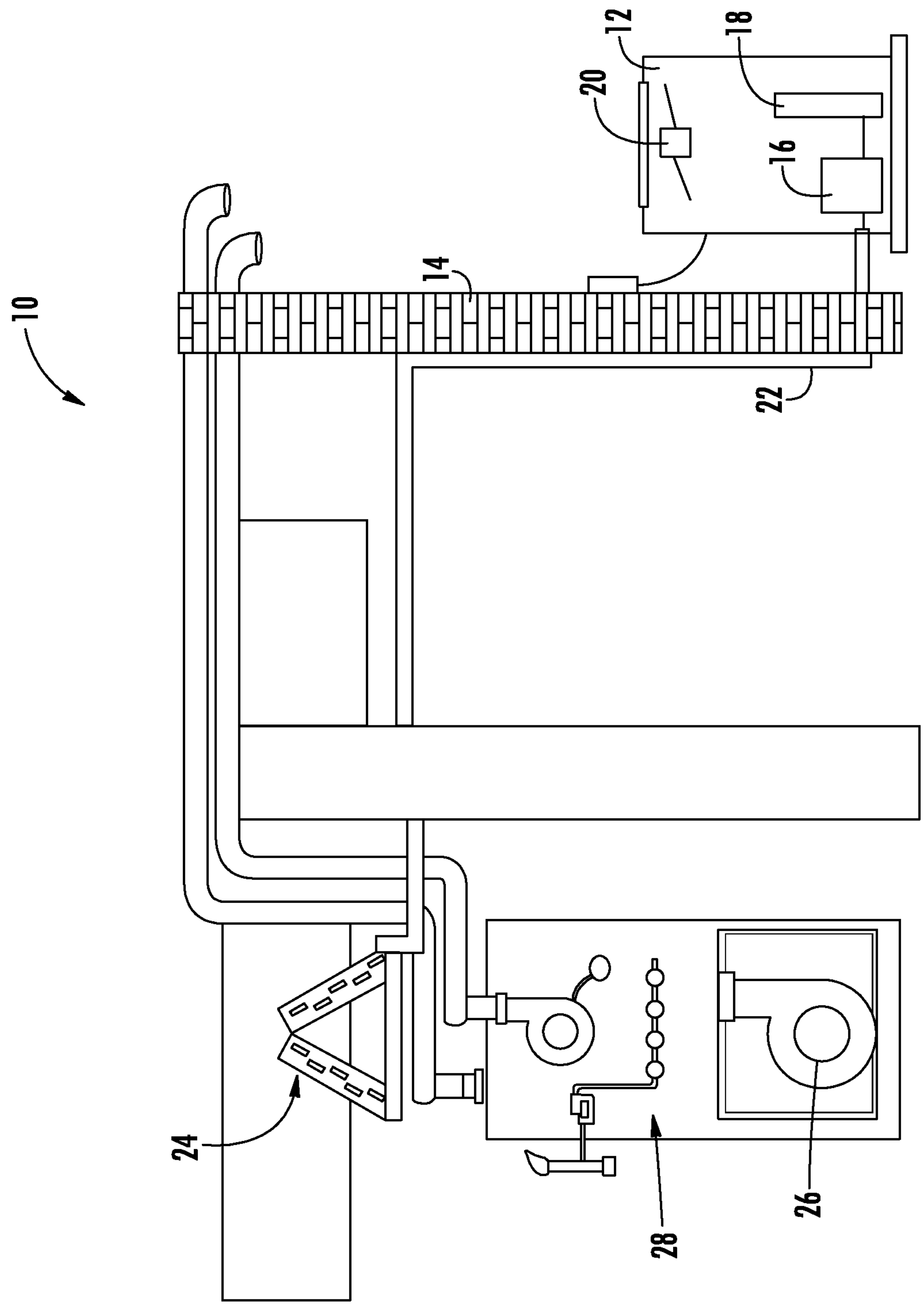


FIG. 1
(PRIOR ART)

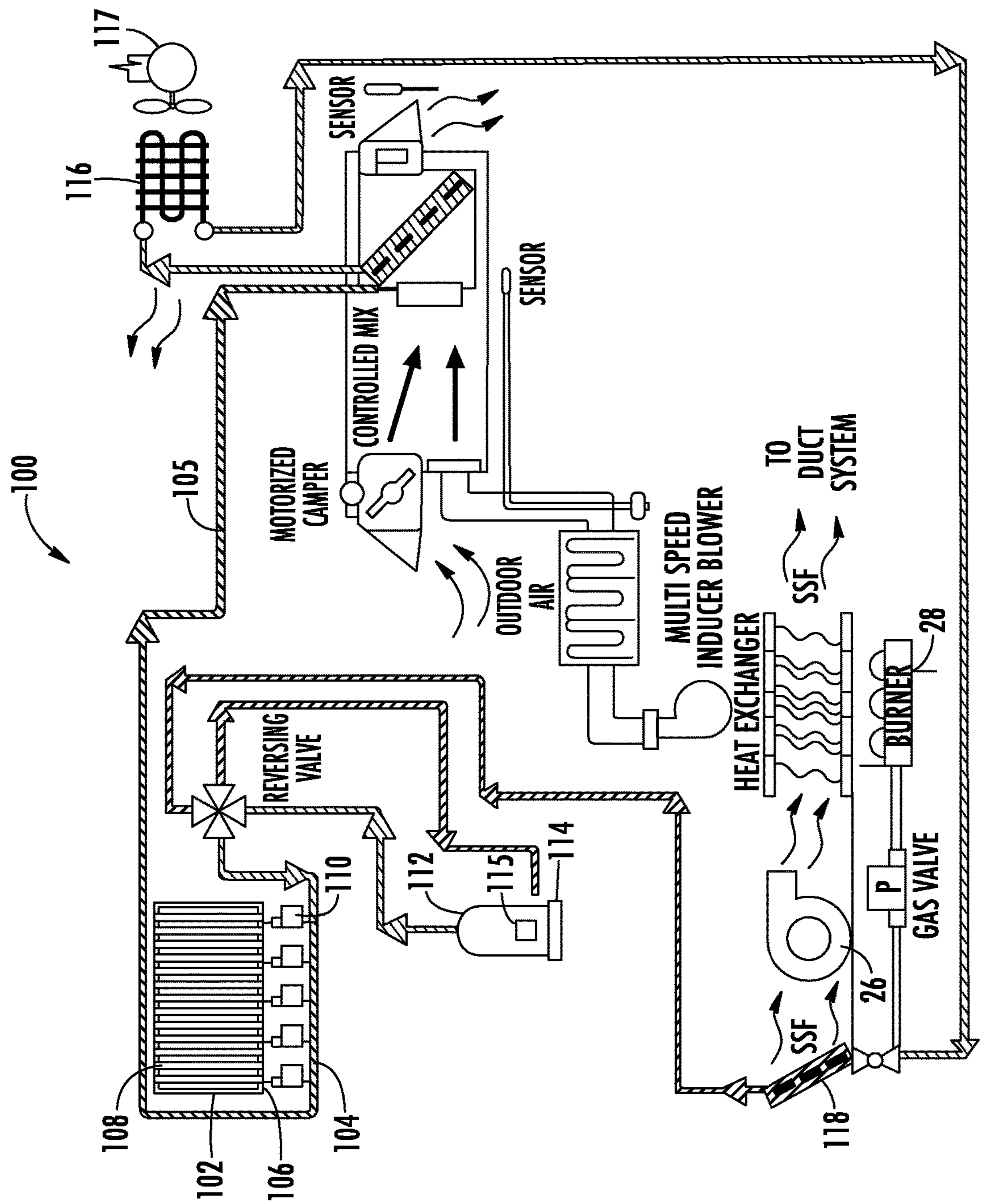

100
105
117
116
SENSOR
SENSOR
CONTROLLED MIX
MOTORIZED CAMPER
OUTDOOR AIR
MULTI SPEED INDUCER BLOWER
TO DUCT SYSTEM
SSF
HEAT EXCHANGER
BURNER
28
REVERSING VALVE
110
112
115
114
P
GAS VALVE
26
SSF
118
108
102
106
104

FIG. 2

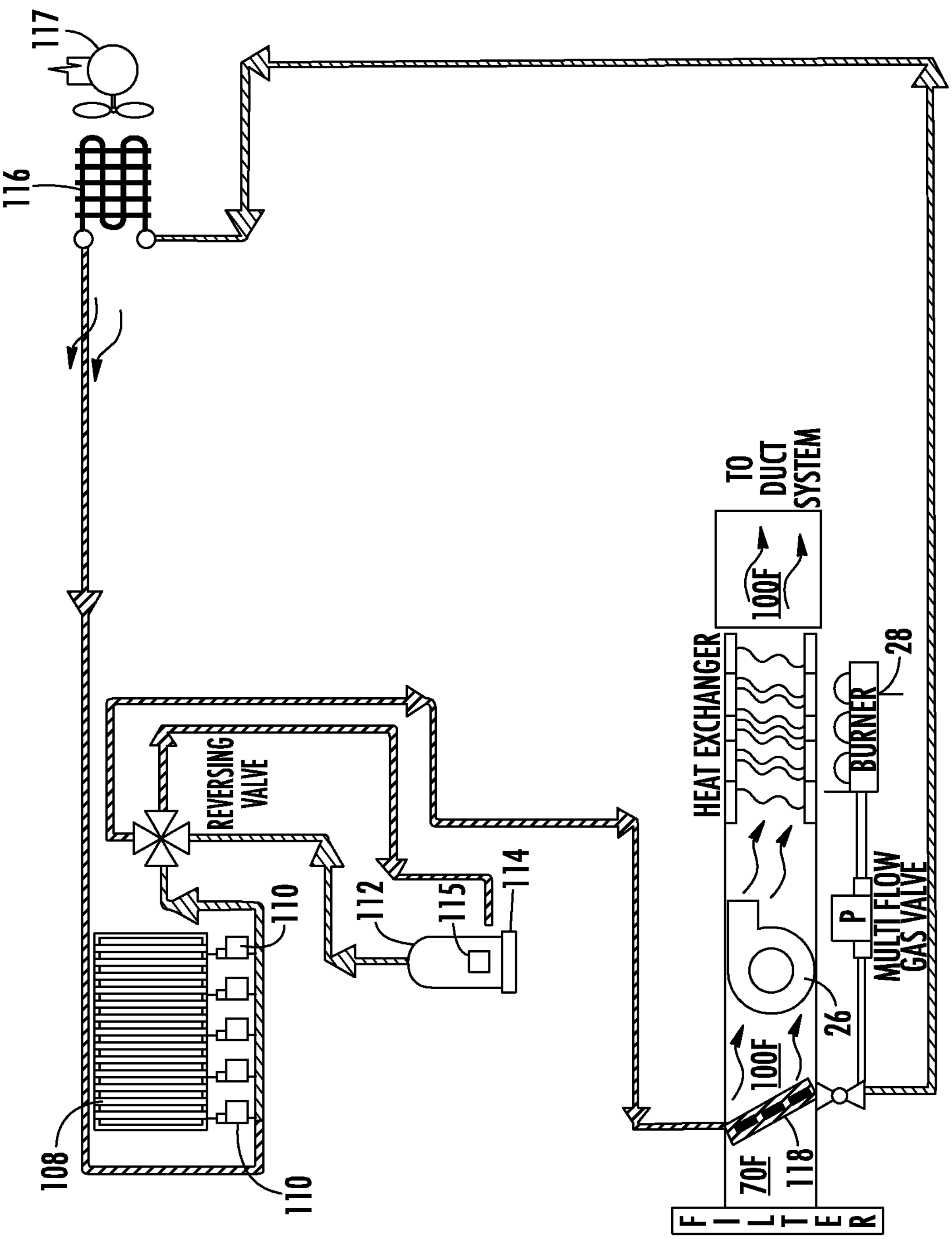

117
116
REVERSING VALVE
110
112
115
114
108
110
TO DUCT SYSTEM
100F
HEAT EXCHANGER
28
BURNER
P
MULTI FLOW GAS VALVE
26
100F
70F
118
FILTER

FIG. 3A

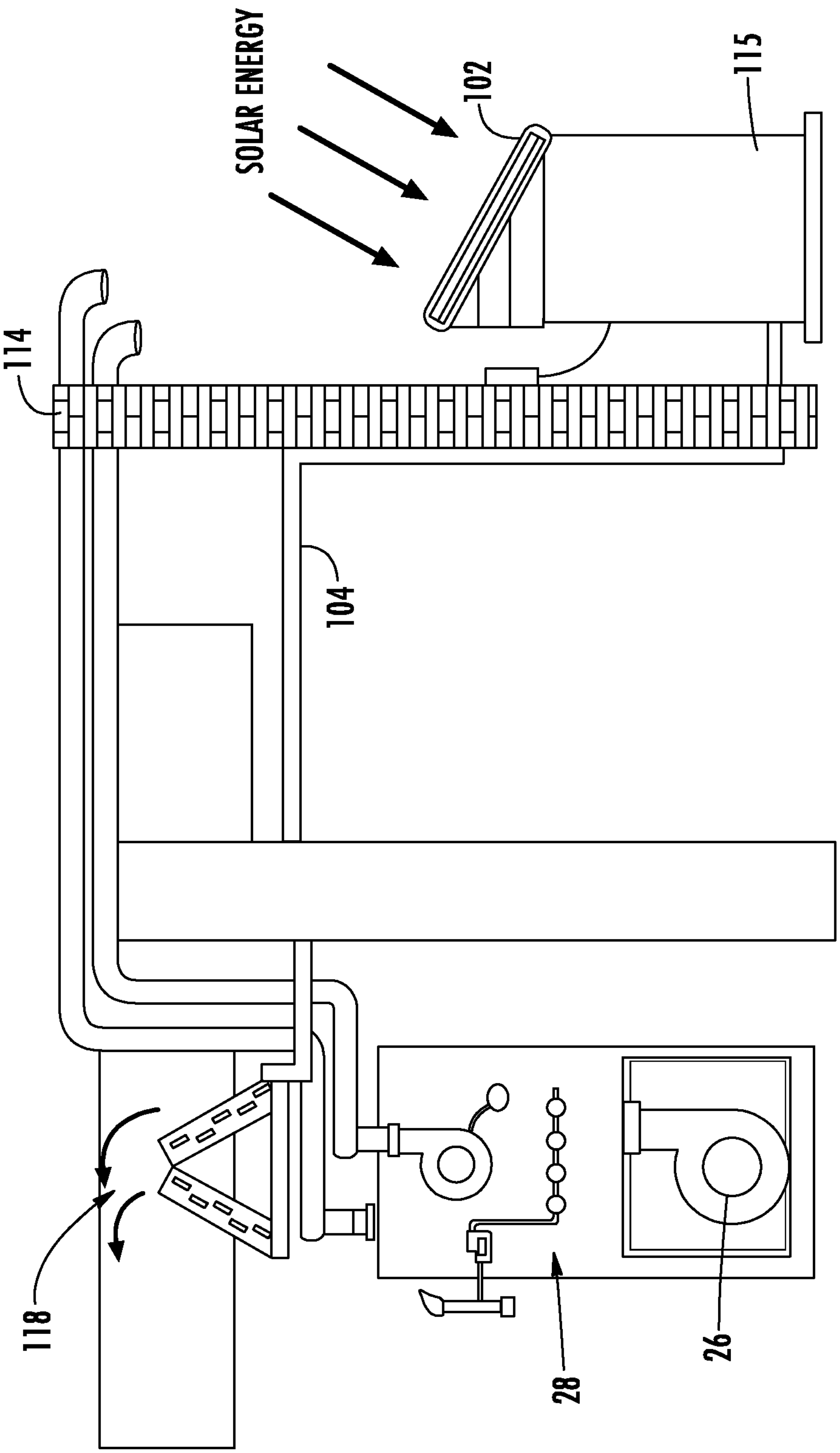


FIG. 3B

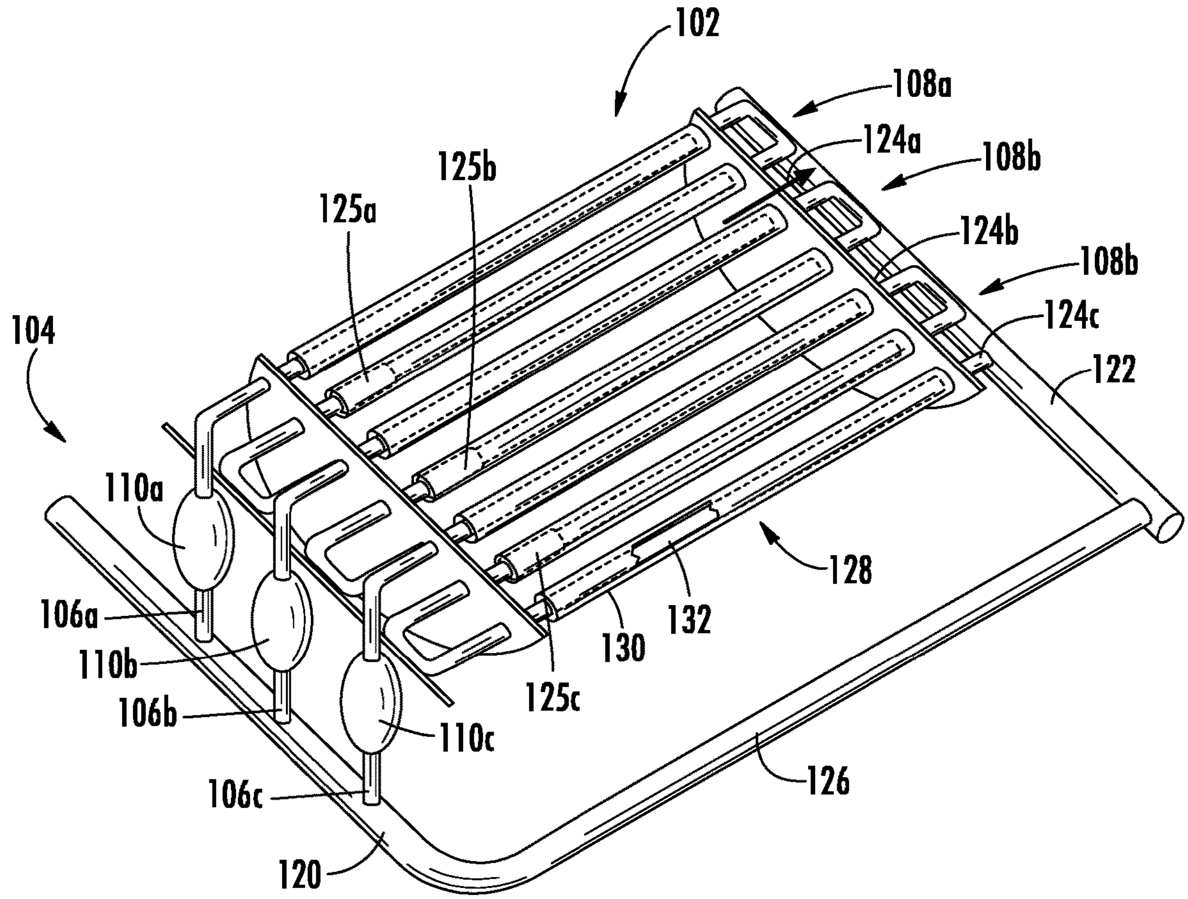

102
108a
124a
108b
125b
125a
124b
108b
124c
104
122
110a
128
106a
132
110b
130
125c
106b
110c
106c
126
120

FIG. 4

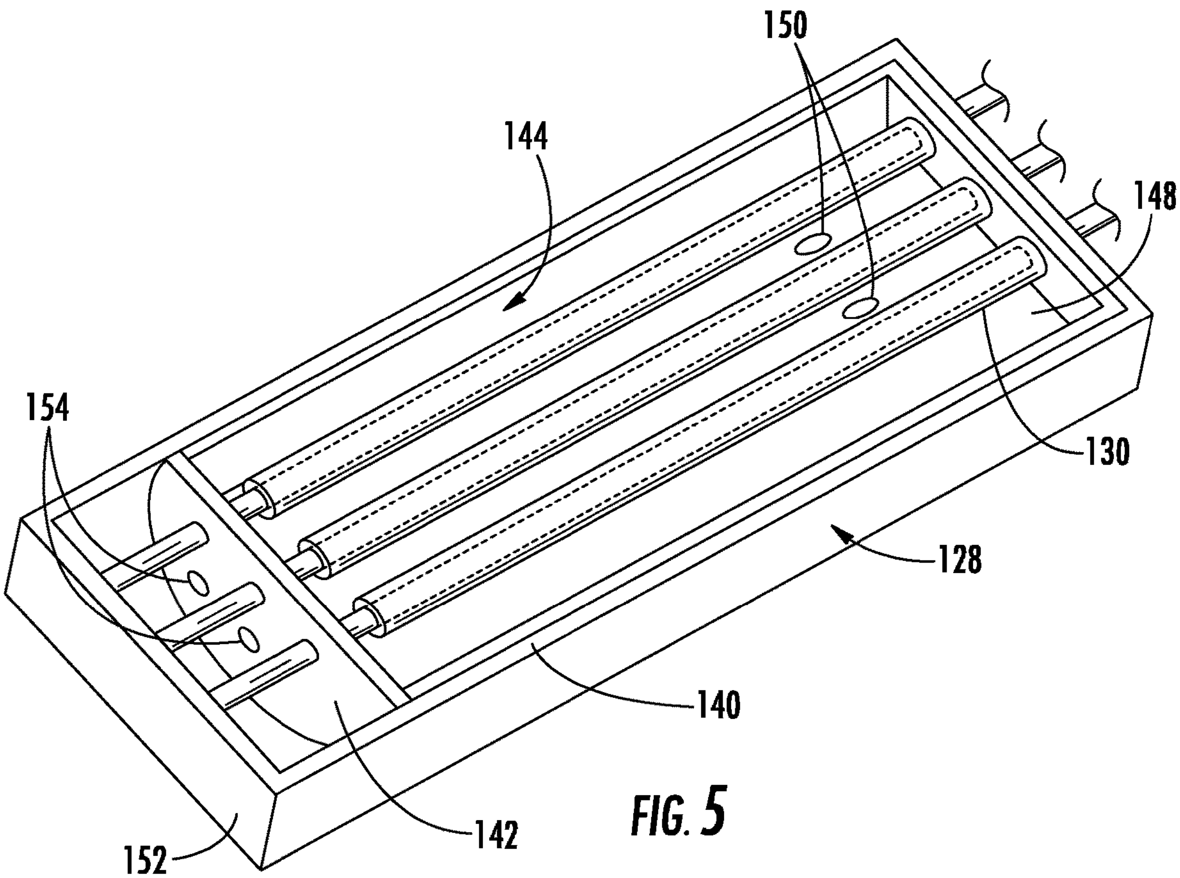


FIG. 5

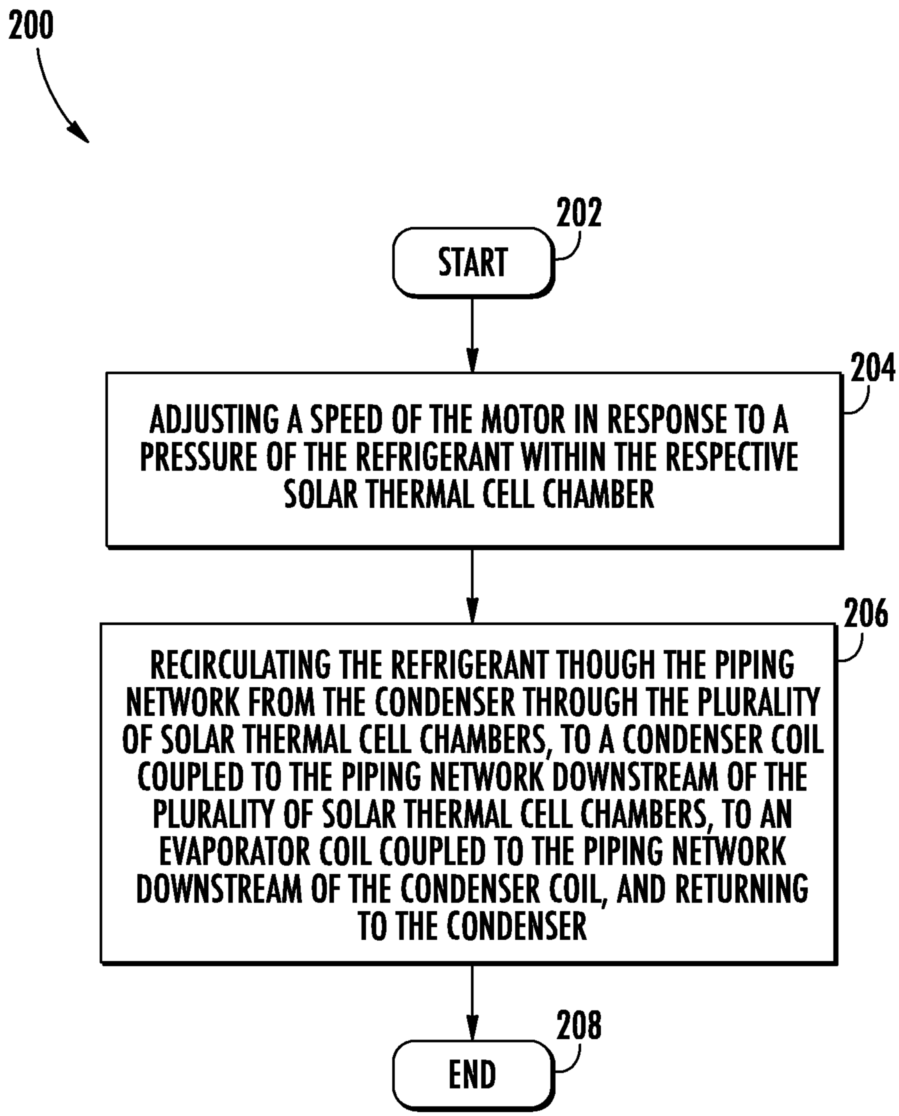

200
202
START
204
ADJUSTING A SPEED OF THE MOTOR IN RESPONSE TO A PRESSURE OF THE REFRIGERANT WITHIN THE RESPECTIVE SOLAR THERMAL CELL CHAMBER
206
RECIRCULATING THE REFRIGERANT THOUGH THE PIPING NETWORK FROM THE CONDENSER THROUGH THE PLURALITY OF SOLAR THERMAL CELL CHAMBERS, TO A CONDENSER COIL COUPLED TO THE PIPING NETWORK DOWNSTREAM OF THE PLURALITY OF SOLAR THERMAL CELL CHAMBERS, TO AN EVAPORATOR COIL COUPLED TO THE PIPING NETWORK DOWNSTREAM OF THE CONDENSER COIL, AND RETURNING TO THE CONDENSER
208
END

FIG. 6

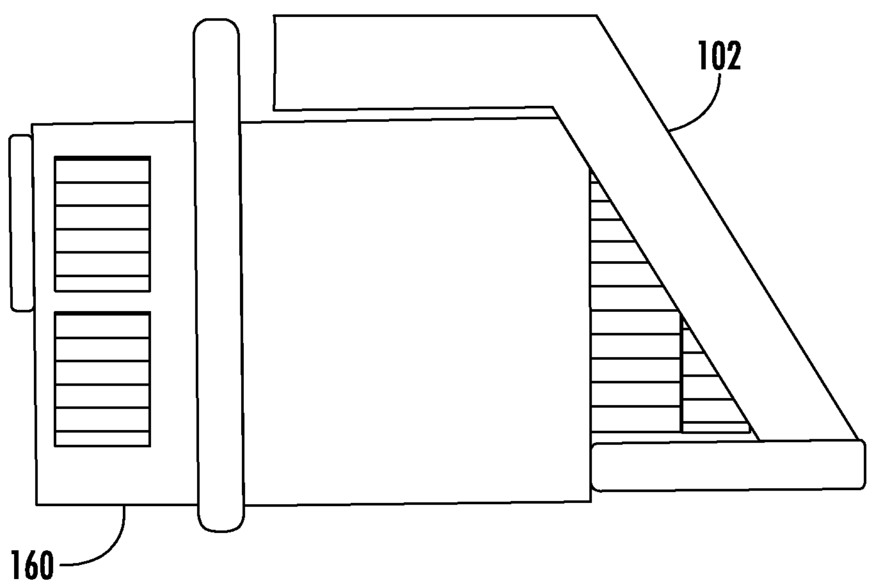


FIG. 7

THERMAL CELL PANEL SYSTEM FOR HEATING AND COOLING AND ASSOCIATED METHODS

FIELD OF THE INVENTION

The present invention relates to heating and cooling systems, and more particularly, to a thermal cell panel system for heating and cooling and associated methods.

BACKGROUND OF THE INVENTION

Various heating and air conditioning systems have been used in buildings for many years. Typical systems pertain to heating a building utilizing a method of combusting a fossil fuel such as home heating oil, natural gas, propane, coal, wood, etc. and using the thermal energy or heat from that combustion process to heat the building. These systems also include diverting the emissions or flue gas from the combustion process outside the building and into the environment.

In the new carbon aware world we now live in, these types of heating systems are rather primitive and destructive. Not only does burning these fossil fuels release harmful emissions including carbon dioxide, but these heating systems also release thermal waste or heat into the environment. As scientist are well aware, major factors that cause global warming include carbon dioxide acting as a greenhouse blanket in the upper layers of the atmosphere and heat being trapped by this blanket. Burning fossil fuels increases both of these factors in their greatest and most damaging form.

Another type of system for heating and cooling a building involves utilizing a refrigerant to absorb heat from one location and transfer it to another. A compressor moves refrigerant in one state to a device which changes the state and temperature of the refrigerant during a cooling cycle. This type of system allows for the absorption of heat from one location and transfers it to another. For example, heat can be absorbed from an outdoor area and released inside the building during a reverse cycle of this type of system often referred to as the heat pump cycle.

In more detail, refrigerants absorb and move thermal energy or heat from one location to another. The chemical formulation of the refrigerants specific to certain applications allow for a state and/or chemical reaction of the refrigerant to allow these properties. For example, a refrigerant under high pressure will take the form of a liquid, while under low pressure that same refrigerant will take the form of a vapor or gas which has a much lower temperature and the ability to absorb heat. Furthermore, in order for the cooling cycle to continue the refrigerant must be compressed using mechanical energy after an expansion process. Compressing the refrigerant will increase the pressure of the refrigerant in order to repeat the compression and expansion of the cooling cycle.

Solar technology has also been used in heating systems. For example, photovoltaics use thin silicone layers and compounds to convert solar rays into DC electric current. Another type of solar technology is a solar thermal collector. The solar thermal collector may use pipes running through a panel usually lined with reflective material or dark backgrounds to direct solar rays into the pipes in a method of heating liquids flowing through these pipes. This method is used to heat pools, and water heaters, for example.

However, none of the existing heating and cooling systems that minimize environmental impacts such as current solar technology, do not perform as well as conventional

fossil fuel systems or systems powered by fossil fuels. Accordingly, there is a need in the art for an improved heating and cooling system.

SUMMARY OF THE INVENTION

It is an object of the invention to provide a heating and cooling system that is environmentally friendly but performs as well as, or better than, heating and cooling systems that are powered by fossil fuels.

A thermal cell panel system for heating and cooling using a refrigerant is disclosed. The system includes a plurality of solar thermal cell chambers, and a piping network for a flow of the refrigerant through the plurality of solar thermal cell chambers. The piping network includes an inlet and outlet to each solar thermal cell chamber. In addition, the system includes a compressor having a motor coupled to a variable frequency drive ("VFD"), where the compressor is coupled to the piping network upstream of the plurality of solar thermal cell chambers and the VFD is configured to adjust a speed of the motor in response to the pressure of the refrigerant within the plurality of solar thermal cell chambers.

The piping network may include an inlet manifold coupled to the inlet of each solar thermal cell chamber, an outlet manifold coupled to the outlet of each solar thermal cell chamber, and a bypass in fluid communication with the inlet manifold and outlet manifold. The bypass is configured to direct a fluid flow from the inlet manifold to the outlet manifold in order to bypass the plurality of solar thermal cell chambers.

The piping network includes pipes through each of the thermal cell chambers and the pipes have heat sink aluminum sleeves. In addition, the pipes and the heat sink aluminum sleeves may be coated with a thermal absorbing material. Each of the solar thermal cell chambers may be insulated and have an interior surface coated with a reflective material.

The solar thermal cell chambers may have a U-shaped bottom portion to reflect solar energy to the pipes and each of the solar thermal cell chambers may have tempered glass secured over a top portion to retain heat therein. Each of the solar thermal cell chambers may have a plurality of drain holes and the pipes may comprises copper piping.

The system may also include a plurality of pressure valves in fluid communication with each inlet and outlet, where each of the pressure valves is configured to selectively open and close the flow of refrigerant through a respective solar thermal cell chamber in response to a pressure of the refrigerant within the respective solar thermal cell chamber.

The system may include a condenser coil coupled to the piping network downstream of the plurality of solar thermal cell chambers, and an evaporator coil coupled to the piping network downstream of the condenser coil and upstream of the compressor.

In another particular aspect, a solar thermal cell chamber for heating and cooling using a refrigerant is disclosed. The solar thermal cell chamber includes a housing, and piping within the housing for a flow of the refrigerant. The piping has an inlet and outlet to the housing. The solar thermal chamber also includes a heat sink aluminum sleeve over the piping within the housing, and a pressure sensor configured to determine a pressure of the refrigerant within the housing and the pressure sensor is configured to be coupled to a compressor in order to adjust a speed of the compressor in response to the pressure of the refrigerant within the housing.

In yet another particular aspect, a method of operating a thermal cell panel system comprising a plurality of solar thermal cell chambers, a piping network for a flow of a refrigerant through the plurality of solar thermal cell chambers for heating and cooling, and a compressor having a motor coupled to the piping network upstream of the plurality of solar thermal cell chambers, is disclosed. The method includes adjusting a speed of the motor in response to a pressure of the refrigerant within the respective solar thermal cell chamber, and selectively opening and closing a pressure valve in fluid communication with the piping network in order to open and close the flow of refrigerant through the housing in response to the pressure of the refrigerant within the housing.

The method may also include recirculating the refrigerant though the piping network from the condenser through the plurality of solar thermal cell chambers, to a condenser coil coupled to the piping network downstream of the plurality of solar thermal cell chambers, to an evaporator coil coupled to the piping network downstream of the condenser coil, and returning to the condenser.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic of a typical heating and cooling system.

FIG. **2** is a schematic process diagram of a thermal cell panel system for heating and cooling in which various aspects of the disclosure may be implemented.

FIG. **3**A is a schematic of a heating stage process diagram in which various aspects of the disclosure may be implemented.

FIG. **3**B is a schematic of the thermal cell panel system of FIG. **2** having a heat pump.

FIG. **4** is a perspective schematic view of a plurality of the solar thermal cell chambers.

FIG. **5** is a perspective schematic view of the solar thermal cell chamber shown without various components for clarity.

FIG. **6** is a flowchart illustrating a method of operating the thermal cell panel system illustrated in FIG. **1**.

FIG. **7** is an example of a window unit installed with the thermal solar thermal cell panel system.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

In the summary of the invention, provided above, and in the descriptions of certain preferred embodiments of the invention, reference is made to particular features of the invention, for example, method steps. It is to be understood that the disclosure of the invention in this specification includes all possible combinations of such particular features, regardless of whether a combination is explicitly described. For instance, where a particular feature is disclosed in the context of a particular aspect or embodiment of the invention, that feature can also be used, to the extent possible, in combination with and/or in the context of other particular aspects and embodiments of the invention, and in the invention generally.

This invention stretches across two well-known industries, the heating and cooling industry and the solar industry. This invention describes new systems and methods of solar thermal reactions with refrigerants in order to dramatically reduce the need of mechanical and electrical energy to move along the refrigeration circuit. Accordingly, this will significantly reduce the electrical energy required for the heating and cooling cycles of a refrigeration circuit or heat pump.

Also, the present systems and methods described herein will reduce the need for fossil fuel combustion as a form of heating a structure. The systems and methods, which utilize sun rays and heat as a sort of fuel to cool a building, overcome the obstacles which cause a building temperature to rise and will instead be used be utilized to cool the building. The warmer and sunnier it is outside, the more efficient the present cooling system will be which is the opposite of the operation of conventional cooling systems.

With respect to heating a building, the present systems and methods use the sun rays and thermal energy in the environment, no matter how cold, to heat a building utilizing a refrigeration cycle or heat pump process. The conventional systems are enhanced by the present improvements described herein which assist the heat pump cycle to heat a building without the need for fossil fuels. This in turn reduces greenhouse gases caused by the combustion of fossil fuels that are used to typically heat a building. Furthermore, the present systems and methods significantly reduce thermal pollution caused by the same combustion processes.

The thermal cell panel system for heating and cooling and associated methods disclosed herein use the sun and environmental thermal energy along with a typical heat pump or air conditioning condensing unit with a variable speed compressor and associated sensors. The system and method detects the outside temperatures, humidity, and weather conditions and interfaces those to the system's internal pressures, temperatures etc., in order to achieve maximum efficiencies. This process resembles artificial environmental intelligence. In addition, the heat pump and the thermal cell panel system are fabricated in one complete and enclosed unit.

Referring now to FIG. **1**, a schematic of a typical heating and cooling system is illustrated and designated **10**. The system **10** includes a condenser unit **10** positioned outside the building **14**. The condenser unit **10** includes a compressor **16**, a condenser coil **18**, and a fan **20**. The compressor **16** comprises a pump that moves refrigerant between an evaporator coil **24** and the condenser coil **18** to chill the indoor air. The condenser coil **18** releases collected heat into the outside air. The fan **20** blows air over the condenser coil **18** to help dissipate the heat outside the building **14**. Coolant lines **22** run from the condenser unit **10** to the evaporator coil **24** inside the building **14** and back to the condenser unit **10**. A blower **26** is positioned to circulate air over the evaporator coil **24** in order to disperse the chilled air inside the building **14**. A typical gas furnace **28** is also illustrated in FIG. **1** that is used for heating the building **14** through the combustion of natural gas and the blower **26** can be used to disperse the heated air inside the building **14**.

A schematic of a thermal cell panel system for heating and cooling in accordance with the present invention is illustrated in FIG. **2** and generally designated **100**. The system **100** includes a plurality of solar thermal cell chambers **102** and a piping network **104** for a flow of the refrigerant through the plurality of solar thermal cell chambers **102**. The piping network **104** includes an inlet **106** and outlet **108** to each of the solar thermal cell chambers **108**.

A plurality of pressure valves **110**, that are optional and are not required for maximum performance, are in fluid communication with each inlet **106**, and each of the pressure valves **110** may be configured to selectively open and close the flow of refrigerant **105** through a respective solar thermal cell chamber **108** in response to a pressure of the refrigerant **105** within the respective solar thermal cell chamber **108**.

The system includes a compressor **112** having a motor **114** coupled to a variable frequency drive ("VFD") **115**. The compressor **112** is coupled to the piping network **104** upstream of the plurality of solar thermal cell chambers **108** and the VFD **115** is configured to adjust a speed of the motor **114** in response to the pressure of the refrigerant **105** within the plurality of solar thermal cell chambers **108**.

The system **100** also includes a condenser coil **116** coupled to the piping network **104** downstream of the plurality of solar thermal cell chambers **108**. A fan **117** blows air over the condenser coil **116** to help dissipate the heat of the refrigerant **105**. In addition, an evaporator coil **118** is coupled to the piping network **104** downstream of the condenser coil **116** and upstream of the compressor **112**.

Referring now to FIGS. **3**A and **3**B, a reverse flow of refrigerant **105** through the system **100** is known as the heat pump cycle. The refrigerant **105** is allowed to absorb heat from the cold air and then flow through the pipes **128** of the plurality of solar thermal cell chambers **102**. This results in increasing the temperature of the refrigerant **105** even further from external forces allowing a heat pump **115** to absorb heat efficiently during the coldest winter conditions. The system **100** reduces the need to use fossil fuels as a combustion source to heat a structure **114** and releases virtually no thermal pollution or cause greenhouse gas emissions from operation.

Furthermore, the system **100** is dramatically more efficient than any source of combustion and will significantly reduce the cost of heating a building **114**. The system **100** is a relatively simple and inexpensive retro fit to a conventional heat pump **115** or air conditioning condensing unit **12**. In addition, the system **100** installs virtually identically to a conventional heat pump **115** or condensing unit **12** and works with practically any furnace or air handler. The system **100** reduces the need for the furnace **28** to use combusted fossil fuels but utilizes the blower section **26** of these components making the combustion chambers obsolete.

Referring now to FIG. **4**, the piping network **104** may also include an inlet manifold **120** coupled to the inlet **106***a*, **106***b*, **106***c* of the respective solar thermal cell chamber **108***a*, **108***b*, **108***c*. In addition, an outlet manifold **122** may be coupled to the outlet **124***a*, **124***b*, **124***c* of the respective solar thermal cell chamber **108***a*, **108***b*, **108***c*, and a bypass **126** in fluid communication with the inlet manifold **120** and outlet manifold **122**. The bypass **126** is configured to direct a fluid flow from the inlet manifold **120** to the outlet manifold **122** to bypass the plurality of solar thermal cell chambers **108** when the pressure valves **110** are closed. As stated above, the pressure valves **110** are optional and are not required for maximum performance. The solar thermal cell chambers **108***a*, **108***b*, **108***c*, which hold gained thermal energy for extended and prolonged periods of time 12-24 hours in many cases with no direct solar or thermal gain.

The piping network **104** includes pipes **128** through each of the thermal cell chambers **108** and the pipes **128** may have heat sink aluminum sleeves **130**. The pipes **128** and the heat sink aluminum sleeves **130** may also be coated with a thermal absorbing material **132**. Sensors **125***a*, **125***b*, **125***c* having pressure and/or temperature capabilities are within each solar thermal cell chamber **108***a*, **108***b*, **108***c*, and may be in communication with the respective pressure valve **110***a*, **110***b*, **110***c*. As stated above, the pressure valves **110***a*, **110***b*, **110***c* are optional and are not required for maximum performance. Once the temperature exceeds the corresponding refrigerant pressure, the respective pressure valve **110***a*, **110***b*, **110***c* are opened. In addition, the sensors **125***a*, **125***b*, **125***c* are in communication with the compressor **112** in order to adjust a speed of the motor **114** in response to the pressure of the refrigerant.

Referring now to FIG. **5**, a housing **152** for each of the solar thermal cell chambers **108** is insulated **140** and has an interior surface **144** coated with a reflective material **146**. In a particular aspect, the solar thermal cell chambers **108** have a U-shaped bottom portion **142** to reflect solar energy to the pipes **128** and tempered glass **148** secured over a top portion to retain heat with the solar thermal cell chamber **108**. The solar thermal cell chambers may also have a plurality of drain holes **150** and the piping **128** may comprises copper piping.

As explained above, the thermal cell chambers **108** include a series of pipes **128** running a circuit through the housings **152** which may be lined with highly reflective material **146**, which is many times more reflective mirrored film than any other ever developed. Each pipe **128** running through the thermal cell chambers **108** may be fitted with a heat sink aluminum sleeve **130**. The sleeve **130** may be made of any material with high thermal transfer properties.

The heat sink sleeves **130** and pipes **128** may also be coated with a compound or coating **132**, which has been specifically developed to absorb many spectrums of solar rays and absorb energy from those rays into thermal energy and into the pipes **128**. The U-shaped bottom portion **142** and pipes **128** may be encased in a highly insulated housing **152** to collect and maintain the thermal energy collected. The tempered glass covering **148** may be comprised of carbon filtered ultra, clear glass with little or no light refraction properties. The high temperature double wall insulated housing **152** may also have drain holes **150** and thermal expansion release holes **154**. The use of specialized reflective films and coatings applied to the heat sinks and pipes directs all phases and rays of solar activity into thermal heat gain on the specialized coated materials, even in direct sunshine and through clouds in which certain rays are found.

Referring now to FIG. **6**, a method of operating a thermal cell panel system comprising a plurality of solar thermal cell chambers, a piping network for a flow of a refrigerant through the plurality of solar thermal cell chambers for heating and cooling, and a compressor having a motor coupled to the piping network upstream of the plurality of solar thermal cell chambers, is disclosed. The method is generally designated **200** and begins at **202**. The method includes adjusting a speed of the motor, at **204**, in response to a pressure of the refrigerant within the respective solar thermal cell chamber.

The method also includes, at **206**, recirculating the refrigerant though the piping network from the condenser through the plurality of solar thermal cell chambers, to a condenser coil coupled to the piping network downstream of the plurality of solar thermal cell chambers, to an evaporator coil coupled to the piping network downstream of the condenser coil, and returning to the condenser. The method ends at **208**.

As explained above, the refrigerant **105** from the heat pump or condensing coil **116** is diverted from the compressor **112** prior to the condensing coils **116** through the panel and piping circuit described above. The refrigerant **105** flows through the series of pipes **128** and after it is compressed by variable speed compressor **112**, the refrigerant **105** flows through the solar thermal cell chamber **108** described above. The warm high pressure refrigerant **105** flows through the hot pipes **128** and the pressure in the refrigerant **105** dramatically rises. The reaction by the gas causes an increase in pressure ahead of the compressor **112** is detected and the compressor **112** can significantly reduce its speed and compression which in turn reduces the electrical use.

The system **100** and method **200** described herein has little or nothing to do with a transfer of heat, which is the key component to conventional solar collection. The present invention describes the warm refrigerant **105** flowing through an even warmer chamber (i.e., the solar thermal cell chamber **108**), in order to cause a state change reaction forcing an increase in the pressure of the refrigerant **105** but pulling very little thermal energy away from the heated solar thermal cell chamber **108**.

In addition, the system **100** and method **200** can be applied to many forms of HVAC equipment including but not limited to P-TAC units, window units, roof top units, residential and commercial units, industrial units, etc. For example, a window unit **160** is shown in FIG. **7** with the solar thermal cell chambers **102**. Further, the system **100** and method can be used in pool heaters, water heaters, fluid and solid heating systems etc.

Accordingly, this allows the reaction to continue much longer than a conventional solar collection system in which the heat is transferred to a colder liquid in an effort to transfer the collected thermal energy. This new system and method described herein in which there is a pressure increase but little if any thermal transfer lasts much longer than any solar based unit because the heat is stored inside the insulated solar thermal cell chamber **108** allowing for optimal use well after the sun is no longer shining on the system.

Even on cloudy days, certain spectrums of the sun are absorbed to increase the pressure and collect the thermal energy within the solar thermal cell chamber **108**. This system **100** can reduce the electrical consumption dramatically when compared to conventional cooling methods. Using this system **100** when it is sunnier and the outdoor conditions are warmer can have even more dramatic savings as the pressure from external forces will be greater. This is specifically when cooling capacity is at its greatest. One of the important factors of the system **100** is being able to exploit the characteristic that the rising pressure of the refrigerant **105** causes little or no absorption of heat. This factor is significant as a refrigeration cycle is an equilibrium of heat absorbed and heat released. In other words, heat cannot be added to a refrigeration cycle or the equilibrium will be altered. The system **100** operates on using thermal energy for a pressure increase and not thermal transfer.

In general, the foregoing description is provided for exemplary and illustrative purposes; the present invention is not necessarily limited thereto. Rather, those skilled in the art will appreciate that additional modifications, as well as adaptations for particular circumstances, will fall within the scope of the invention as herein shown and described and of the claims appended hereto.

What is claimed is:

1. A thermal cell panel system for heating and cooling using a refrigerant, the system comprising:

a plurality of solar thermal cell chambers;

a piping network for a flow of the refrigerant through the plurality of solar thermal cell chambers, the piping network having an inlet and outlet to each solar thermal cell chamber; and

a compressor having a motor coupled to a variable frequency drive ("VFD"), the compressor coupled to the piping network upstream of the plurality of solar thermal cell chambers and the VFD configured to adjust a speed of the motor in response to the pressure of the refrigerant within the plurality of solar thermal cell chambers;

wherein the piping network comprises:

an inlet manifold coupled to the inlet of each solar thermal cell chamber;

an outlet manifold coupled to the outlet of each solar thermal cell chamber; and

a bypass in fluid communication with the inlet manifold and outlet manifold, the bypass configured to direct a fluid flow from the inlet manifold to the outlet manifold to bypass the plurality of solar thermal cell chambers.

2. The thermal cell panel system for heating and cooling using a refrigerant of claim **1**, wherein the piping network comprises pipes through each of the thermal cell chambers and the pipes have heat sink aluminum sleeves.

3. The thermal cell panel system for heating and cooling using a refrigerant of claim **2**, wherein the pipes and the heat sink aluminum sleeves are coated with a thermal absorbing material.

4. The thermal cell panel system for heating and cooling using a refrigerant of claim **1**, wherein each of the solar thermal cell chambers is insulated and has an interior surface coated with a reflective material.

5. The thermal cell panel system for heating and cooling using a refrigerant of claim **1**, wherein each of the solar thermal cell chambers has a U-shaped bottom portion to reflect solar energy to the pipes.

6. The thermal cell panel system for heating and cooling using a refrigerant of claim **1**, wherein each of the solar thermal cell chambers has tempered glass secured over a top portion to retain heat therein.

7. A thermal cell panel system for heating and cooling using a refrigerant comprising:

a plurality of solar thermal cell chambers;

a piping network for a flow of the refrigerant through the plurality of solar thermal cell chambers, the piping network having an inlet and outlet to each solar thermal cell chamber; and

a compressor having a motor coupled to a variable frequency drive ("VFD"), the compressor coupled to the piping network upstream of the plurality of solar thermal cell chambers and the VFD configured to adjust a speed of the motor in response to the pressure of the refrigerant within the plurality of solar thermal cell chambers;

wherein each of the solar thermal cell chambers has a plurality of drain holes.

8. The thermal cell panel system for heating and cooling using a refrigerant of claim **1**, further comprising a plurality of pressure valves in fluid communication with each inlet, each of the pressure valves configured to selectively open and close the flow of refrigerant through a respective solar thermal cell chamber in response to a pressure of the refrigerant within the respective solar thermal cell chamber.

9. The thermal cell panel system for heating and cooling using a refrigerant of claim **1**, further comprising:

a condenser coil coupled to the piping network downstream of the plurality of solar thermal cell chambers; and

an evaporator coil coupled to the piping network downstream of the condenser coil and upstream of the compressor.

10. A solar thermal cell chamber for heating and cooling using a refrigerant, the solar thermal cell chamber comprising:

a housing;

piping within the housing for a flow of the refrigerant, the piping having an inlet and outlet to the housing;

a heat sink aluminum sleeve over the piping within the housing; and

a pressure sensor configured to determine a pressure of the refrigerant within the housing and the pressure sensor configured to be coupled to a compressor in order to adjust a speed of the compressor in response to the pressure of the refrigerant within the housing;

wherein the housing has a plurality of drain holes.

11. The solar thermal cell chamber for heating and cooling using a refrigerant of claim **10**, further comprising a pressure valve in fluid communication with the inlet and configured to selectively open and close the flow of refrigerant through the housing in response to the pressure of the refrigerant within the housing.

12. The solar thermal cell chamber for heating and cooling using a refrigerant of claim **10**, wherein the pipes and the heat sink aluminum sleeves are coated with a thermal absorbing material.

13. The solar thermal cell chamber for heating and cooling using a refrigerant of claim **10**, wherein the housing is insulated and has an interior surface coated with a reflective material.

14. The solar thermal cell chamber for heating and cooling using a refrigerant of claim **10**, wherein the housing has a U-shaped bottom portion to reflect solar energy to the piping.

15. The solar thermal cell chamber for heating and cooling using a refrigerant of claim **10**, wherein the housing has tempered glass secured over a top portion to retain heat therein.

16. The solar thermal cell chamber for heating and cooling using a refrigerant of claim **10**, wherein the piping comprises copper piping.

* * * * *

SOLAR RATING & CERTIFICATION CORPORATION

An ICC Evaluation Service Program

OG-100 Solar Thermal Collector Certification

No./10002196B

Issued: August 20, 2024

Expiration Date: June 01, 2026

www.solar-rating.org | **(800) 423-6587** | **(562) 699-0543**

CERTIFICATION HOLDER:	**EVALUATION SUBJECT**	
Commercial Energy Savings Plus 2604 NW 53rd Drive Boca Raton, FL 33496, USA. www.commercialesp.com	**BRAND:**	CESP
	MODEL:	SLR-BX-02C15.75
	TYPE:	Glazed Flat Plate

PRODUCT CERTIFICATION SYSTEM:

The ICC-SRCC OG-100 certification program includes evaluation and performance ratings for solar thermal collectors as established in the *ICC-SRCC Rules for Solar Heating & Cooling Product Listing Reports.* The program also includes periodic factory inspections and surveillance of the manufacturer's quality management system.

COMPLIANCE WITH THE FOLLOWING STANDARD(S): *ICC 901/SRCC 100 - 2020, Solar Thermal Collectors Standard*

OG-100 THERMAL PERFORMANCE RATINGS:

ICC-SRCC OG-100 thermal performance ratings provided for the collector are calculated for a 24-hour period using OG-100 standard conditions using collector parameters measured through laboratory testing. Actual performance will vary with local conditions, installation details and usage.

OG-100 STANDARD DAILY PRODUCTION TABLE							
Kilowatt-hours (thermal) per Collector per Day				**Thousands of BTU per Collector per Day**			
Climate → **Category (T_i-T_a)**	High Radiation (6.3 kWh/m²•day)	Medium Radiation (4.7 kWh/m²•day)	Low Radiation (3.1 kWh/m²•day)	**Climate →** **Category (T_i-T_a)**	High Radiation (2 kBTU/ft²•day)	Medium Radiation (1.5 kBTU/ft²•day)	Low Radiation (1 kBTU/ft²•day)
A (-5°C)	0.52	0.40	0.29	**A (-9°F)**	1.77	1.38	0.99
B (5°C)	0.39	0.28	0.17	**B (9°F)**	1.35	0.96	0.57
C (20°C)	0.22	0.12	0.03	**C (36°F)**	0.76	0.41	0.11
D (60°C)	0.00	0.00	0.00	**D (90°F)**	0.00	0.00	0.00
E (80°C)	0.00	0.00	0.00	**E (144°F)**	0.00	0.00	0.00

A – Pool Heating (Warm Climate) **B** – Pool Heating (Cool Climate) **C** – Water Heating (Warm Climate) **D** – Space & Water Heating (Cool Climate) **E** – Commercial Hot Water & Cooling

THERMAL EFFICIENCY:

The efficiency of solar thermal collectors is determined using test methods set in ICC 901/SRCC 100, based on ISO 9806 procedures. Results are processed to provide unique coefficients ($\eta_{0,hem}$, a_1, a_2…) for efficiency equations, provided in several forms below. For the simplified equations, instantaneous power at normal incidence is given by $\dot{Q} = K_\theta \eta_{hem} A_G G$. Incident Angle Modifiers (IAMs) are provided to indicate the change in output as the incidence angle of solar irradiance changes. The three forms should provide similar results, but the extended form is considered the most accurate. Users should select the equation to be used based the application and available data.

T_i: Temperature of the fluid entering the collector.
T_m: Average temperature of the fluid within the collector (between the inlet and outlet).
T_a: Temperature of the ambient air around the collector.
G: Hemispherical solar irradiance with sub-types beam (b) and diffuse (d)
E_L: Longwave (infrared) irradiance.
ε/α: Collector emissivity/absorptance ratio.

RATING CONDITIONS	
Gross Collector Area (A_G)	0.196 m^2
Fluid Mass Flowrate ($\dot{m}$)	0.090 kg/(m^{2}s)
Test Fluid	Water
Performance Test Standard	ISO 9806-2017

SIMPLIFIED THERMAL PERFORMANCE EQUATIONS (ISO 9806-2013, T_i, A_G)

Second Order Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G} - a_2 G \left(\frac{T_i - T_a}{G}\right)^2$$

Linearized Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G}$$

UNITS	$\eta_{0,hem}$	a_1	a_2	$\eta_{0,hem}$ ("Intercept")	a_1 ("Slope")
SI	0.379	5.425 W/(m^{2}K)	0.029 W/(m^{2}K^2)	0.392	6.995 W/(m^{2}K)
IP	0.379	0.956 BTU/(h ft^2°F)	0.003 BTU/(h ft^2°F^2)	0.392	1.233 BTU/(hr ft^2°F)

** Thermal efficiency equations per ISO 9806-2013 using inlet (Ti) fluid temperature, provided in second and first order (linearized) forms for normal incidence. The second order efficiency equation is a more accurate representation of the collector performance than the linearized equation. The linearized efficiency equation is provided for use with incentive programs, regulations and software that require the simplified "slope" and "intercept" coefficients to describe collector performance.*

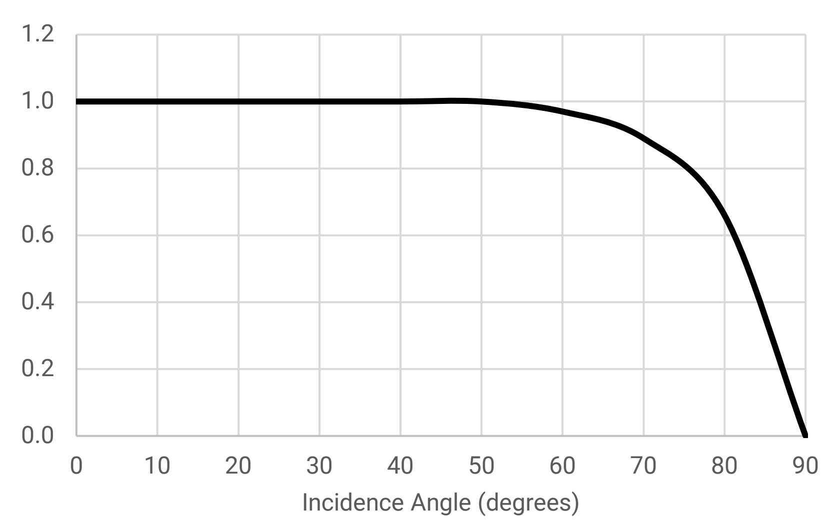
GENERAL THERMAL PERFORMANCE EQUATION (ISO 9806-2017, T_m, A_G)

Extended Thermal Efficiency Equation**

$$\dot{Q} = A_G(\eta_{0,b}K_b(\theta_L,\theta_T)G_b + \eta_{0,b}K_dG_d - a_1(T_m - T_a) - a_2(T_m - T_a)^2 - a_3u'(T_m - T_a) + a_4(E_L - \sigma T_a^4) - a_5\left(\frac{dT_m}{dt}\right) - a_6u'G - a_7u'(E_L - \sigma T_a^4) - a_8(T_m - T_a)^4$$

	$\eta_{0,hem}$	$\eta_{0,b}$	K_d	a_1	a_2	a_3	a_4	a_5	a_6	a_8
VALUE	0.382	0.384	0.97	5.498	0.027	0.000	0.000	49567	0.000	0.000
UNITS	-	-	-	W/(m^{2}K)	W/(m^{2}K^2)	J/(m^{3}K)	-	J/(m^{2}K)	s/m	W/(m^{2}K^4)

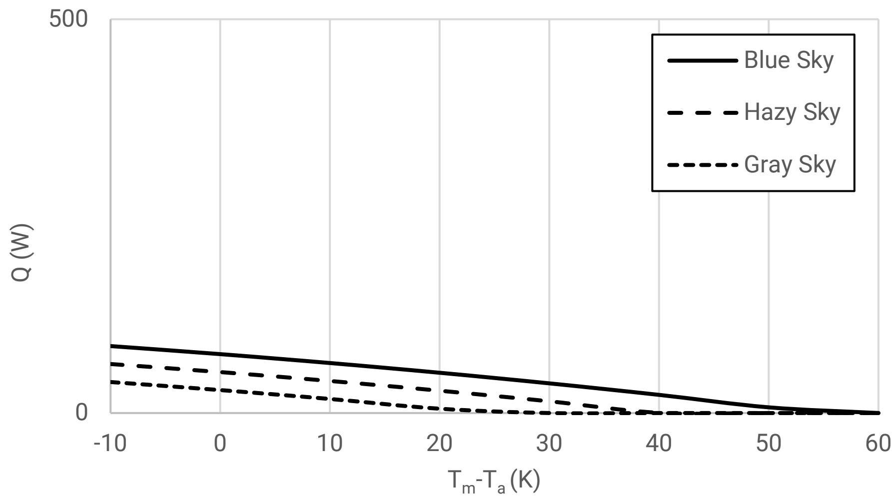
*** General thermal efficiency equation for mean (T_m) fluid temperature provided in accordance with ISO 9806-2017. Where data measured using a standard other than ISO 9806-2017, additional assumptions may be applied to determine extended equation coefficients. Reduced surrounding air speed (u') is defined as u'=u-3 m/s. See ISO 9806-2017 for other coefficient definitions.*



Angle (θ)	Total IAM $K_b(\theta)$
0°	1.00
10°	1.00
20°	1.00
30°	1.00
40°	1.00
50°	1.00
60°	0.97
70°	0.89
80°	0.66
90°	0.00

POWER OUTPUT:

The instantaneous power output of the collector under different conditions is calculated at the Standard Reporting Conditions (SRC) defined by ISO 9806-2017 using the measured performance coefficients above.



STANDARD COLLECTOR POWER OUTPUT (W)			
Based on Standard Rating Conditions (SRC) and mean temperature (T_m) in accordance with ISO 9806-2017			
T_m-T_a (°C)	**Blue sky** G_b = 850, G_d = 150 (W/m²)	**Hazy sky** G_b= 440, G_d= 260 (W/m²)	**Grey sky** G_b=0, G_d= 400 (W/m²)
-10	85	62	39
0	75*	52	29
10	64	41	18
20	51	28	6
30	38	15	0
40	23	0	0
50	7	0	0
60	0	0	0

**Peak Power is defined by ISO 9806 as the Blue Sky irradiance condition at T_m-T_a =0 and normal incidence.*

TEST SAMPLE SPECIFICATIONS:

The specifications of the collector sample submitted for testing are provided below.

TEST & SAMPLE SPECIFICATIONS					
Gross Length:	0.60 m	2.0 ft	**Maximum Design Temperature:**	57 °C	135 °F
Gross Width:	0.32 m	1.1 ft	**Maximum Design Pressure:**	3500 kPa	508 psi
Gross Depth:	224.0 mm	8.8 in	**Standard Stagnation Temperature:**	110 °C	230 °F
Empty Weight:	11.3 kg	25 lb			
Fluid Capacity:	1.20 L	0.3 gal			
Notes:	Standard stagnation at 1000 W/m² and 30 °C as defined by ISO 9806 – 2017. **Note: The standard stagnation temperature of this collector exceeds the maximum design temperature specified by the manufacturer. Therefore, it must be installed in systems where the temperature of the collector is prevented from exceeding the maximum design temperature to prevent damage to the collector and hazardous conditions.**				

IDENTIFICATION:

Certified systems must be identified with the OG-100 certification mark below in accordance with the *Rules for Certification Mark and Certificate Use* and labeled in with the information below per *ICC 901/SRCC 100*:

1. Manufacturer's name and model number.
2. OG-100 collector certification number
3. Maximum operating pressure
4. Dry weight
5. Fluid volume
6. Compatible heat transfer fluids
7. Standard stagnation temperature
8. Year of manufacture and/or serial number.

CONDITIONS:

1. Collector must be installed and operated in accordance with the manufacturer's published instructions and local codes and regulations.
2. OG-100 Standard Performance Ratings and Standard Collector Power Output have been calculated for the tested components using standardized conditions established by the OG-100 program and associated test standards. Actual performance will vary based on the specific usage, installation and local environmental conditions.
3. The collector listed in this ICC-SRCC OG-100 certification must be labeled in accordance with the *ICC-SRCC Rules for Mark and Certificate Use*.
4. OG-100 certifications do not include mounting hardware and fixtures.
5. Solar thermal collectors and mounting hardware and appurtenances must comply with all applicable local requirements for fire resistance. Solar thermal collectors must be mounted in accordance with the requirements of the collector and mounting hardware manufacturers to comply with local codes for structural loading for wind, seismic, snow and other loads.
6. Solar thermal collectors must be used with the heat transfer fluids listed in this document.
7. Solar thermal collector manufactured under a quality control program subject to periodic evaluation in accordance with the requirements of ICC-SRCC.
8. This document must be reproduced in its entirety.
9. Certification status should be confirmed on the ICC-SRCC Directory at www.solar-rating.org

OG-100 Solar Thermal Collector Certification

No./10002196C

Issued: August 20, 2024

Expiration Date: June 01, 2026

www.solar-rating.org | **(800) 423-6587** | **(562) 699-0543**

CERTIFICATION HOLDER:	**EVALUATION SUBJECT**	
Commercial Energy Savings Plus 2604 NW 53rd Drive Boca Raton, FL 33496, USA. www.commercialesp.com	**BRAND:**	CESP
	MODEL:	SLR-BX-02C31.50
	TYPE:	Glazed Flat Plate

PRODUCT CERTIFICATION SYSTEM:

The ICC-SRCC OG-100 certification program includes evaluation and performance ratings for solar thermal collectors as established in the *ICC-SRCC Rules for Solar Heating & Cooling Product Listing Reports.* The program also includes periodic factory inspections and surveillance of the manufacturer's quality management system.

COMPLIANCE WITH THE FOLLOWING STANDARD(S): *ICC 901/SRCC 100 - 2020, Solar Thermal Collectors Standard*

OG-100 THERMAL PERFORMANCE RATINGS:

ICC-SRCC OG-100 thermal performance ratings provided for the collector are calculated for a 24-hour period using OG-100 standard conditions using collector parameters measured through laboratory testing.   Actual performance will vary with local conditions, installation details and usage.

OG-100 STANDARD DAILY PRODUCTION TABLE							
Kilowatt-hours (thermal) per Collector per Day				**Thousands of BTU per Collector per Day**			
Climate → **Category** **(T_i-T_a)**	High Radiation (6.3 kWh/m²•day)	Medium Radiation (4.7 kWh/m²•day)	Low Radiation (3.1 kWh/m²•day)	**Climate →** **Category** **(T_i-T_a)**	High Radiation (2 kBTU/ft²•day)	Medium Radiation (1.5 kBTU/ft²•day)	Low Radiation (1 kBTU/ft²•day)
A (-5°C)	1.06	0.82	0.59	**A (-9°F)**	3.60	2.81	2.01
B (5°C)	0.81	0.58	0.34	**B (9°F)**	2.76	1.97	1.17
C (20°C)	0.46	0.25	0.07	**C (36°F)**	1.58	0.86	0.23
D (60°C)	0.00	0.00	0.00	**D (90°F)**	0.00	0.00	0.00
E (80°C)	0.00	0.00	0.00	**E (144°F)**	0.00	0.00	0.00

A – Pool Heating (Warm Climate) **B** – Pool Heating (Cool Climate) **C** – Water Heating (Warm Climate) **D** – Space & Water Heating (Cool Climate) **E** – Commercial Hot Water & Cooling

THERMAL EFFICIENCY:

The efficiency of solar thermal collectors is determined using test methods set in ICC 901/SRCC 100, based on ISO 9806 procedures. Results are processed to provide unique coefficients ($\eta_{0,hem}$, a_1, a_2…) for efficiency equations, provided in several forms below. For the simplified equations, instantaneous power at normal incidence is given by $\dot{Q} = K_\theta \eta_{hem} A_G G$. Incident Angle Modifiers (IAMs) are provided to indicate the change in output as the incidence angle of solar irradiance changes. The three forms should provide similar results, but the extended form is considered the most accurate. Users should select the equation to be used based the application and available data.

T_i: Temperature of the fluid entering the collector.
T_m: Average temperature of the fluid within the collector (between the inlet and outlet).
T_a: Temperature of the ambient air around the collector.
G: Hemispherical solar irradiance with sub-types beam (b) and diffuse (d)
E_L: Longwave (infrared) irradiance.
ε/α: Collector emissivity/absorptance ratio.

RATING CONDITIONS	
Gross Collector Area (A_G)	0.325 m^2
Fluid Mass Flowrate ($\dot{m}$)	0.090 kg/(m^{2}s)
Test Fluid	Water
Performance Test Standard	ISO 9806-2017

SIMPLIFIED THERMAL PERFORMANCE EQUATIONS (ISO 9806-2013, T_i, A_G)

Second Order Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G} - a_2 G \left(\frac{T_i - T_a}{G}\right)^2$$

Linearized Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G}$$

UNITS	$\eta_{0,hem}$	a_1	a_2	$\eta_{0,hem}$ ("Intercept")	a_1 ("Slope")
SI	0.467	6.572 W/(m^{2}K)	0.035 W/(m^{2}K^2)	0.483	8.503 W/(m^{2}K)
IP	0.467	1.499 BTU/(h ft^2°F)	0.003 BTU/(h ft^2°F^2)	0.483	1.499 BTU/(hr ft^2°F)

** Thermal efficiency equations per ISO 9806-2013 using inlet (Ti) fluid temperature, provided in second and first order (linearized) forms for normal incidence. The second order efficiency equation is a more accurate representation of the collector performance than the linearized equation. The linearized efficiency equation is provided for use with incentive programs, regulations and software that require the simplified "slope" and "intercept" coefficients to describe collector performance.*

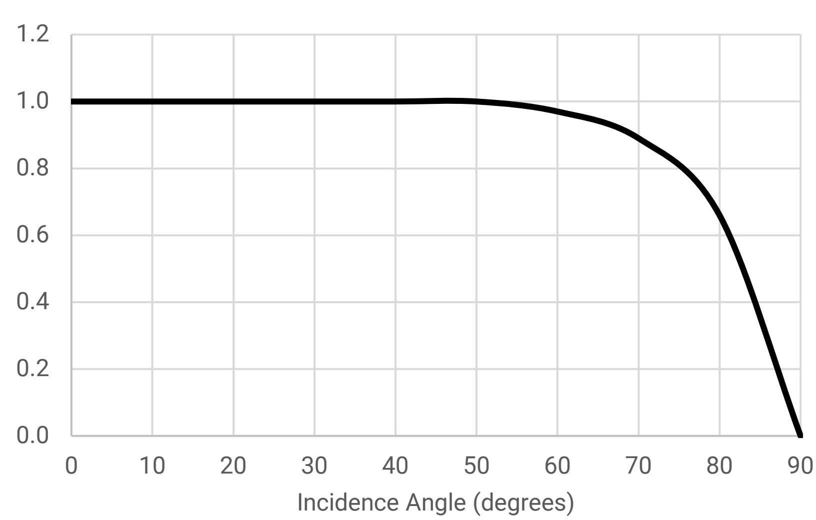
GENERAL THERMAL PERFORMANCE EQUATION (ISO 9806-2017, T_m, A_G)

Extended Thermal Efficiency Equation**

$$\dot{Q} = A_G(\eta_{0,b} K_b(\theta_L, \theta_T) G_b + \eta_{0,b} K_d G_d - a_1(T_m - T_a) - a_2(T_m - T_a)^2 - a_3 u'(T_m - T_a) + a_4(E_L - \sigma T_a^4) - a_5\left(\frac{dT_m}{dt}\right) - a_6 u' G - a_7 u'(E_L - \sigma T_a^4) - a_8(T_m - T_a)^4$$

	$\eta_{0,hem}$	$\eta_{0,b}$	K_d	a_1	a_2	a_3	a_4	a_5	a_6	a_8
VALUE	0.471	0.473	0.97	6.663	0.033	0.000	0.000	29819	0.000	0.000
UNITS	-	-	-	W/(m^{2}K)	W/(m^{2}K^2)	J/(m^{3}K)	-	J/(m^{2}K)	s/m	W/(m^{2}K^4)

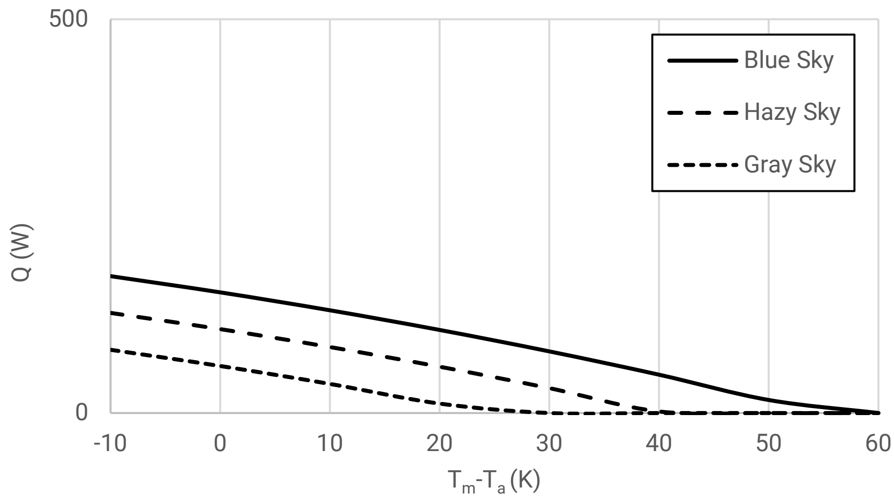
*** General thermal efficiency equation for mean (T_m) fluid temperature provided in accordance with ISO 9806-2017. Where data measured using a standard other than ISO 9806-2017, additional assumptions may be applied to determine extended equation coefficients. Reduced surrounding air speed (u') is defined as u'=u-3 m/s. See ISO 9806-2017 for other coefficient definitions.*



Angle (θ)	Total IAM $K_b(\theta)$
0°	1.00
10°	1.00
20°	1.00
30°	1.00
40°	1.00
50°	1.00
60°	0.97
70°	0.89
80°	0.66
90°	0.00

POWER OUTPUT:

The instantaneous power output of the collector under different conditions is calculated at the Standard Reporting Conditions (SRC) defined by ISO 9806-2017 using the measured performance coefficients above.



STANDARD COLLECTOR POWER OUTPUT (W)			
Based on Standard Rating Conditions (SRC) and mean temperature (T_m) in accordance with ISO 9806-2017			
T_m-T_a (°C)	**Blue sky** G_b = 850, G_d = 150 (W/m^2)	**Hazy sky** G_b= 440, G_d= 260 (W/m^2)	**Grey sky** G_b=0, G_d= 400 (W/m^2)
-10	174	127	80
0	153*	107	60
10	130	84	37
20	105	59	12
30	78	32	0
40	49	2	0
50	17	0	0
60	0	0	0

**Peak Power is defined by ISO 9806 as the Blue Sky irradiance condition at T_m-T_a =0 and normal incidence.*

TEST SAMPLE SPECIFICATIONS:

The specifications of the collector sample submitted for testing are provided below.

TEST & SAMPLE SPECIFICATIONS					
Gross Length:	1.00 m	3.3 ft	**Maximum Design Temperature:**	57 °C	135 °F
Gross Width:	0.32 m	1.1 ft	**Maximum Design Pressure:**	3500 kPa	508 psi
Gross Depth:	224.0 mm	8.8 in	**Standard Stagnation Temperature:**	110 °C	230 °F
Empty Weight:	22.7 kg	50 lb			
Fluid Capacity:	2.33 L	0.6 gal			
Notes:	Standard stagnation at 1000 W/m^2 and 30 °C as defined by ISO 9806 – 2017. **Note: The standard stagnation temperature of this collector exceeds the maximum design temperature specified by the manufacturer. Therefore, it must be installed in systems where the temperature of the collector is prevented from exceeding the maximum design temperature to prevent damage to the collector and hazardous conditions.**				

IDENTIFICATION:

Certified systems must be identified with the OG-100 certification mark below in accordance with the *Rules for Certification Mark and Certificate Use* and labeled in with the information below per *ICC 901/SRCC 100*:

1. Manufacturer's name and model number.
2. OG-100 collector certification number
3. Maximum operating pressure
4. Dry weight
5. Fluid volume
6. Compatible heat transfer fluids
7. Standard stagnation temperature
8. Year of manufacture and/or serial number.

CONDITIONS:

1. Collector must be installed and operated in accordance with the manufacturer's published instructions and local codes and regulations.
2. OG-100 Standard Performance Ratings and Standard Collector Power Output have been calculated for the tested components using standardized conditions established by the OG-100 program and associated test standards. Actual performance will vary based on the specific usage, installation and local environmental conditions.
3. The collector listed in this ICC-SRCC OG-100 certification must be labeled in accordance with the *ICC-SRCC Rules for Mark and Certificate Use*.
4. OG-100 certifications do not include mounting hardware and fixtures.
5. Solar thermal collectors and mounting hardware and appurtenances must comply with all applicable local requirements for fire resistance. Solar thermal collectors must be mounted in accordance with the requirements of the collector and mounting hardware manufacturers to comply with local codes for structural loading for wind, seismic, snow and other loads.
6. Solar thermal collectors must be used with the heat transfer fluids listed in this document.
7. Solar thermal collector manufactured under a quality control program subject to periodic evaluation in accordance with the requirements of ICC-SRCC.
8. This document must be reproduced in its entirety.
9. Certification status should be confirmed on the ICC-SRCC Directory at www.solar-rating.org

ACCREDITED
CERTIFICATE # 3299.01, 3299.03

SOLAR RATING & CERTIFICATION CORPORATION

An ICC Evaluation Service Program

OG-100 Solar Thermal Collector Certification

No./10002196D

Issued: August 20, 2024

Expiration Date: June 01, 2026

www.solar-rating.org | **(800) 423-6587** | **(562) 699-0543**

CERTIFICATION HOLDER:	**EVALUATION SUBJECT**	
Commercial Energy Savings Plus 2604 NW 53rd Drive Boca Raton, FL 33496, USA. www.commercialesp.com	**BRAND:**	CESP
	MODEL:	SLR-BX-03C31.50
	TYPE:	Glazed Flat Plate

PRODUCT CERTIFICATION SYSTEM:

The ICC-SRCC OG-100 certification program includes evaluation and performance ratings for solar thermal collectors as established in the *ICC-SRCC Rules for Solar Heating & Cooling Product Listing Reports.* The program also includes periodic factory inspections and surveillance of the manufacturer's quality management system.

COMPLIANCE WITH THE FOLLOWING STANDARD(S): *ICC 901/SRCC 100 - 2020, Solar Thermal Collectors Standard*

OG-100 THERMAL PERFORMANCE RATINGS:

ICC-SRCC OG-100 thermal performance ratings provided for the collector are calculated for a 24-hour period using OG-100 standard conditions using collector parameters measured through laboratory testing. Actual performance will vary with local conditions, installation details and usage.

OG-100 STANDARD DAILY PRODUCTION TABLE							
Kilowatt-hours (thermal) per Collector per Day				**Thousands of BTU per Collector per Day**			
Climate → **Category (T_i-T_a)**	High Radiation (6.3 kWh/m²•day)	Medium Radiation (4.7 kWh/m²•day)	Low Radiation (3.1 kWh/m²•day)	**Climate →** **Category (T_i-T_a)**	High Radiation (2 kBTU/ft²•day)	Medium Radiation (1.5 kBTU/ft²•day)	Low Radiation (1 kBTU/ft²•day)
A (-5˚C)	1.61	1.25	0.90	**A (-9˚F)**	5.50	4.28	3.06
B (5˚C)	1.25	0.89	0.53	**B (9˚F)**	4.25	3.04	1.82
C (20˚C)	0.73	0.40	0.12	**C (36˚F)**	2.49	1.38	0.40
D (60˚C)	0.01	0.00	0.00	**D (90˚F)**	0.03	0.00	0.00
E (80˚C)	0.00	0.00	0.00	**E (144˚F)**	0.00	0.00	0.00

A – Pool Heating (Warm Climate) **B** – Pool Heating (Cool Climate) **C** – Water Heating (Warm Climate) **D** – Space & Water Heating (Cool Climate) **E** – Commercial Hot Water & Cooling

THERMAL EFFICIENCY:

The efficiency of solar thermal collectors is determined using test methods set in ICC 901/SRCC 100, based on ISO 9806 procedures. Results are processed to provide unique coefficients ($\eta_{0,hem}$, a_1, a_2…) for efficiency equations, provided in several forms below. For the simplified equations, instantaneous power at normal incidence is given by $\dot{Q} = K_\theta \eta_{hem} A_G G$. Incident Angle Modifiers (IAMs) are provided to indicate the change in output as the incidence angle of solar irradiance changes. The three forms should provide similar results, but the extended form is considered the most accurate. Users should select the equation to be used based the application and available data.

T_i: Temperature of the fluid entering the collector.
T_m: Average temperature of the fluid within the collector (between the inlet and outlet).
T_a: Temperature of the ambient air around the collector.
G: Hemispherical solar irradiance with sub-types beam (b) and diffuse (d)
E_L: Longwave (infrared) irradiance.
ε/α: Collector emissivity/absorptance ratio.

RATING CONDITIONS	
Gross Collector Area (A_G)	0.492 m^2
Fluid Mass Flowrate ($\dot{m}$)	0.090 kg/(m^2s)
Test Fluid	Water
Performance Test Standard	ISO 9806-2017

SIMPLIFIED THERMAL PERFORMANCE EQUATIONS (ISO 9806-2013, T_i, A_G)

Second Order Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G} - a_2 G \left(\frac{T_i - T_a}{G}\right)^2$$

Linearized Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G}$$

UNITS	$\eta_{0,hem}$	a_1	a_2	$\eta_{0,hem}$ ("Intercept")	a_1 ("Slope")
SI	0.473	6.419 W/(m^2K)	0.036 W/(m^2K^2)	0.489	8.372 W/(m^2K)
IP	0.473	1.131 BTU/(h ft²°F)	0.003 BTU/(h ft²°F²)	0.489	1.475 BTU/(hr ft²°F)

** Thermal efficiency equations per ISO 9806-2013 using inlet (Ti) fluid temperature, provided in second and first order (linearized) forms for normal incidence. The second order efficiency equation is a more accurate representation of the collector performance than the linearized equation. The linearized efficiency equation is provided for use with incentive programs, regulations and software that require the simplified "slope" and "intercept" coefficients to describe collector performance.*

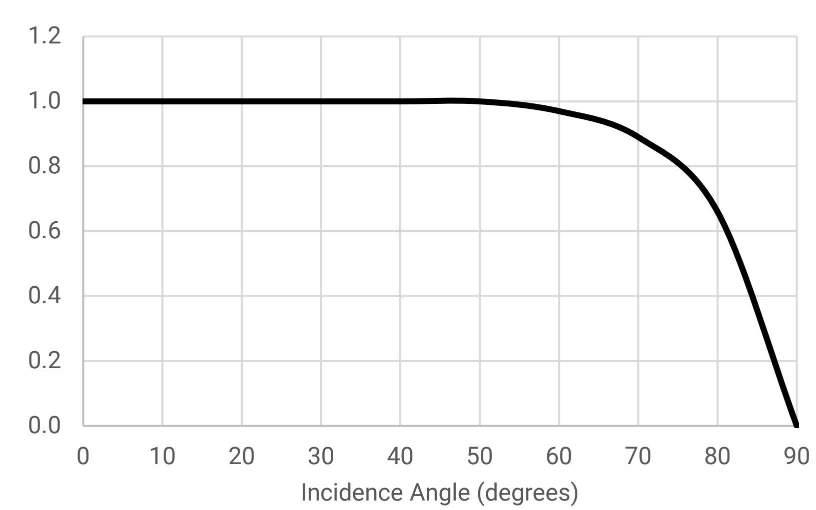
GENERAL THERMAL PERFORMANCE EQUATION (ISO 9806-2017, T_m, A_G)

Extended Thermal Efficiency Equation**

$$\dot{Q} = A_G(\eta_{0,b} K_b(\theta_L, \theta_T) G_b + \eta_{0,b} K_d G_d - a_1(T_m - T_a) - a_2(T_m - T_a)^2 - a_3 u'(T_m - T_a) + a_4(E_L - \sigma T_a^4) - a_5\left(\frac{dT_m}{dt}\right) - a_6 u' G - a_7 u'(E_L - \sigma T_a^4) - a_8(T_m - T_a)^4$$

	$\eta_{0,hem}$	$\eta_{0,b}$	K_d	a_1	a_2	a_3	a_4	a_5	a_6	a_8
VALUE	0.478	0.480	0.97	6.503	0.034	0.000	0.000	19724	0.000	0.000
UNITS	-	-	-	W/(m^2K)	W/(m^2K^2)	J/(m^3K)	-	J/(m^2K)	s/m	W/(m^2K^4)

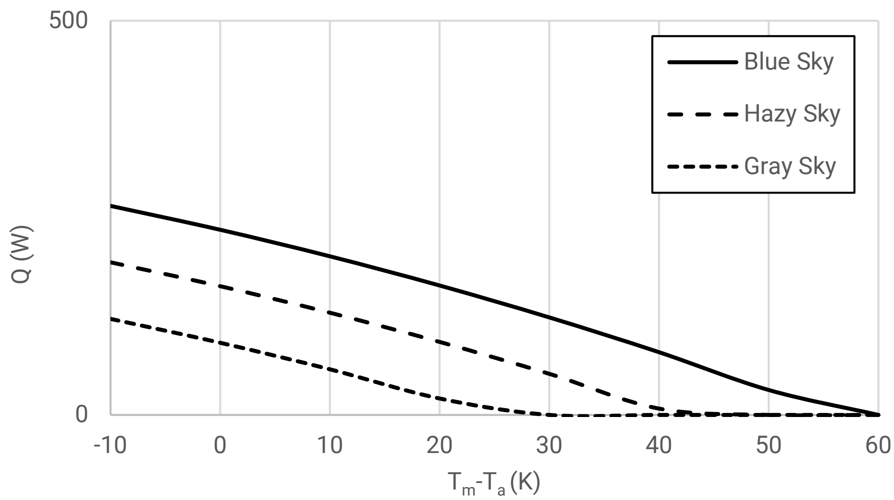
*** General thermal efficiency equation for mean (T_m) fluid temperature provided in accordance with ISO 9806-2017. Where data measured using a standard other than ISO 9806-2017, additional assumptions may be applied to determine extended equation coefficients. Reduced surrounding air speed (u') is defined as u'=u-3 m/s. See ISO 9806-2017 for other coefficient definitions.*



Angle (θ)	Total IAM $K_b(\theta)$
0°	1.00
10°	1.00
20°	1.00
30°	1.00
40°	1.00
50°	1.00
60°	0.97
70°	0.89
80°	0.66
90°	0.00

POWER OUTPUT:

The instantaneous power output of the collector under different conditions is calculated at the Standard Reporting Conditions (SRC) defined by ISO 9806-2017 using the measured performance coefficients above.



STANDARD COLLECTOR POWER OUTPUT (W)			
Based on Standard Rating Conditions (SRC) and mean temperature (T_m) in accordance with ISO 9806-2017			
T_m-T_a (°C)	**Blue sky** G_b = 850, G_d = 150 (W/m^2)	**Hazy sky** G_b= 440, G_d= 260 (W/m^2)	**Grey sky** G_b=0, G_d= 400 (W/m^2)
-10	265	194	122
0	235*	163	92
10	201	130	58
20	164	93	21
30	124	52	0
40	80	8	0
50	32	0	0
60	0	0	0

**Peak Power is defined by ISO 9806 as the Blue Sky irradiance condition at T_m-T_a =0 and normal incidence.*

TEST SAMPLE SPECIFICATIONS:

The specifications of the collector sample submitted for testing are provided below.

TEST & SAMPLE SPECIFICATIONS					
Gross Length:	1.03 m	3.4 ft	**Maximum Design Temperature:**	57 °C	135 °F
Gross Width:	0.48 m	1.6 ft	**Maximum Design Pressure:**	3500 kPa	508 psi
Gross Depth:	223.0 mm	8.8 in	**Standard Stagnation Temperature:**	110 °C	230 °F
Empty Weight:	34.0 kg	75 lb			
Fluid Capacity:	1.96 L	0.5 gal			
Notes:	Standard stagnation at 1000 W/m^2 and 30 °C as defined by ISO 9806 – 2017. **Note: The standard stagnation temperature of this collector exceeds the maximum design temperature specified by the manufacturer. Therefore, it must be installed in systems where the temperature of the collector is prevented from exceeding the maximum design temperature to prevent damage to the collector and hazardous conditions.**				

IDENTIFICATION:

Certified systems must be identified with the OG-100 certification mark below in accordance with the *Rules for Certification Mark and Certificate Use* and labeled in with the information below per *ICC 901/SRCC 100*:

1. Manufacturer's name and model number.
2. OG-100 collector certification number
3. Maximum operating pressure
4. Dry weight
5. Fluid volume
6. Compatible heat transfer fluids
7. Standard stagnation temperature
8. Year of manufacture and/or serial number.

CONDITIONS:

1. Collector must be installed and operated in accordance with the manufacturer's published instructions and local codes and regulations.
2. OG-100 Standard Performance Ratings and Standard Collector Power Output have been calculated for the tested components using standardized conditions established by the OG-100 program and associated test standards. Actual performance will vary based on the specific usage, installation and local environmental conditions.
3. The collector listed in this ICC-SRCC OG-100 certification must be labeled in accordance with the *ICC-SRCC Rules for Mark and Certificate Use*.
4. OG-100 certifications do not include mounting hardware and fixtures.
5. Solar thermal collectors and mounting hardware and appurtenances must comply with all applicable local requirements for fire resistance. Solar thermal collectors must be mounted in accordance with the requirements of the collector and mounting hardware manufacturers to comply with local codes for structural loading for wind, seismic, snow and other loads.
6. Solar thermal collectors must be used with the heat transfer fluids listed in this document.
7. Solar thermal collector manufactured under a quality control program subject to periodic evaluation in accordance with the requirements of ICC-SRCC.
8. This document must be reproduced in its entirety.
9. Certification status should be confirmed on the ICC-SRCC Directory at www.solar-rating.org

ACCREDITED
CERTIFICATE # 3299.01, 3299.03

OG-100 Solar Thermal Collector Certification

No./10002196A

Issued: August 20, 2024

Expiration Date: June 01, 2026

www.solar-rating.org | **(800) 423-6587** | **(562) 699-0543**

CERTIFICATION HOLDER:	**EVALUATION SUBJECT**	
Commercial Energy Savings Plus 2604 NW 53rd Drive Boca Raton, FL 33496, USA. www.commercialesp.com	**BRAND:**	CESP
	MODEL:	SLR-BX-04C31.50
	TYPE:	Glazed Flat Plate

PRODUCT CERTIFICATION SYSTEM:

The ICC-SRCC OG-100 certification program includes evaluation and performance ratings for solar thermal collectors as established in the *ICC-SRCC Rules for Solar Heating & Cooling Product Listing Reports.* The program also includes periodic factory inspections and surveillance of the manufacturer's quality management system.

COMPLIANCE WITH THE FOLLOWING STANDARD(S): *ICC 901/SRCC 100 - 2020, Solar Thermal Collectors Standard*

OG-100 THERMAL PERFORMANCE RATINGS:

ICC-SRCC OG-100 thermal performance ratings provided for the collector are calculated for a 24-hour period using OG-100 standard conditions using collector parameters measured through laboratory testing. Actual performance will vary with local conditions, installation details and usage.

OG-100 STANDARD DAILY PRODUCTION TABLE							
Kilowatt-hours (thermal) per Collector per Day				**Thousands of BTU per Collector per Day**			
Climate → **Category** **(T_i-T_a)**	High Radiation (6.3 kWh/m²•day)	Medium Radiation (4.7 kWh/m²•day)	Low Radiation (3.1 kWh/m²•day)	**Climate →** **Category** **(T_i-T_a)**	High Radiation (2 kBTU/ft²•day)	Medium Radiation (1.5 kBTU/ft²•day)	Low Radiation (1 kBTU/ft²•day)
A (-5°C)	2.15	1.68	1.20	**A (-9°F)**	7.35	5.72	4.10
B (5°C)	1.66	1.18	0.71	**B (9°F)**	5.66	4.04	2.42
C (20°C)	0.96	0.53	0.15	**C (36°F)**	3.28	1.81	0.51
D (60°C)	0.01	0.00	0.00	**D (90°F)**	0.02	0.00	0.00
E (80°C)	0.00	0.00	0.00	**E (144°F)**	0.00	0.00	0.00

A – Pool Heating (Warm Climate) **B** – Pool Heating (Cool Climate) **C** – Water Heating (Warm Climate) **D** – Space & Water Heating (Cool Climate) **E** – Commercial Hot Water & Cooling

THERMAL EFFICIENCY:

The efficiency of solar thermal collectors is determined using test methods set in ICC 901/SRCC 100, based on ISO 9806 procedures. Results are processed to provide unique coefficients ($\eta_{0,hem}$, a_1, a_2…) for efficiency equations, provided in several forms below. For the simplified equations, instantaneous power at normal incidence is given by $\dot{Q} = K_\theta \eta_{hem} A_G G$. Incident Angle Modifiers (IAMs) are provided to indicate the change in output as the incidence angle of solar irradiance changes. The three forms should provide similar results, but the extended form is considered the most accurate. Users should select the equation to be used based the application and available data.

T_i: Temperature of the fluid entering the collector.
T_m: Average temperature of the fluid within the collector (between the inlet and outlet).
T_a: Temperature of the ambient air around the collector.
G: Hemispherical solar irradiance with sub-types beam (b) and diffuse (d)
E_L: Longwave (infrared) irradiance.
ε/α: Collector emissivity/absorptance ratio.

RATING CONDITIONS	
Gross Collector Area (A_G)	0.653 m^2
Fluid Mass Flowrate ($\dot{m}$)	0.090 kg/(m^2s)
Test Fluid	Water
Performance Test Standard	ISO 9806-2017

SIMPLIFIED THERMAL PERFORMANCE EQUATIONS (ISO 9806-2013, T_i, A_G)

Second Order Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G} - a_2 G \left(\frac{T_i - T_a}{G}\right)^2$$

Linearized Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G}$$

UNITS	$\eta_{0,hem}$	a_1	a_2	$\eta_{0,hem}$ ("Intercept")	a_1 ("Slope")
SI	0.475	6.541 W/(m^2K)	0.036 W/(m^2K^2)	0.491	8.502 W/(m^2K)
IP	0.475	1.153 BTU/(h ft^2°F)	0.003 BTU/(h ft^2°F^2)	0.491	1.498 BTU/(hr ft^2°F)

** Thermal efficiency equations per ISO 9806-2013 using inlet (Ti) fluid temperature, provided in second and first order (linearized) forms for normal incidence. The second order efficiency equation is a more accurate representation of the collector performance than the linearized equation. The linearized efficiency equation is provided for use with incentive programs, regulations and software that require the simplified "slope" and "intercept" coefficients to describe collector performance.*

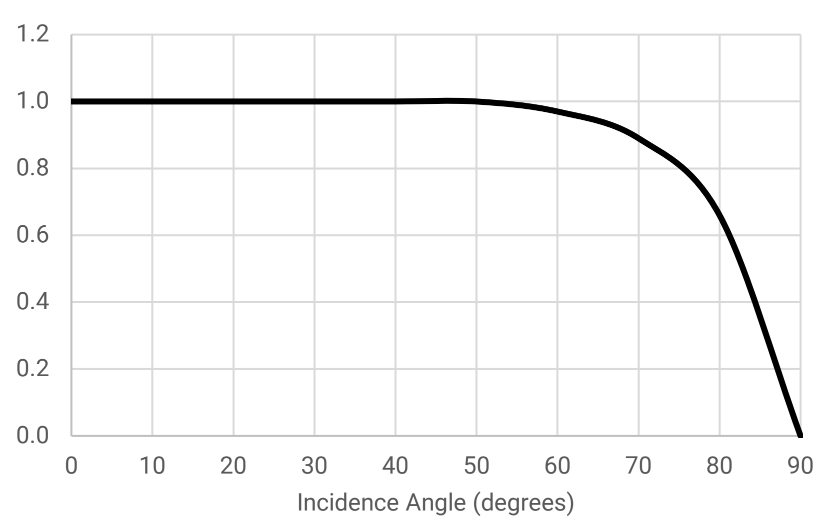
GENERAL THERMAL PERFORMANCE EQUATION (ISO 9806-2017, T_m, A_G)

Extended Thermal Efficiency Equation**

$$\dot{Q} = A_G(\eta_{0,b} K_b(\theta_L,\theta_T) G_b + \eta_{0,b} K_d G_d - a_1(T_m - T_a) - a_2(T_m - T_a)^2 - a_3 u'(T_m - T_a) + a_4(E_L - \sigma T_a^4) - a_5\left(\frac{dT_m}{dt}\right) - a_6 u' G - a_7 u'(E_L - \sigma T_a^4) - a_8(T_m - T_a)^4$$

	$\eta_{0,hem}$	$\eta_{0,b}$	K_d	a_1	a_2	a_3	a_4	a_5	a_6	a_8
VALUE	0.479	0.477	0.97	6.550	0.037	0.000	0.000	14900	0.000	0.000
UNITS	-	-	-	W/(m^2K)	W/(m^2K^2)	J/(m^3K)	-	J/(m^2K)	s/m	W/(m^2K^4)

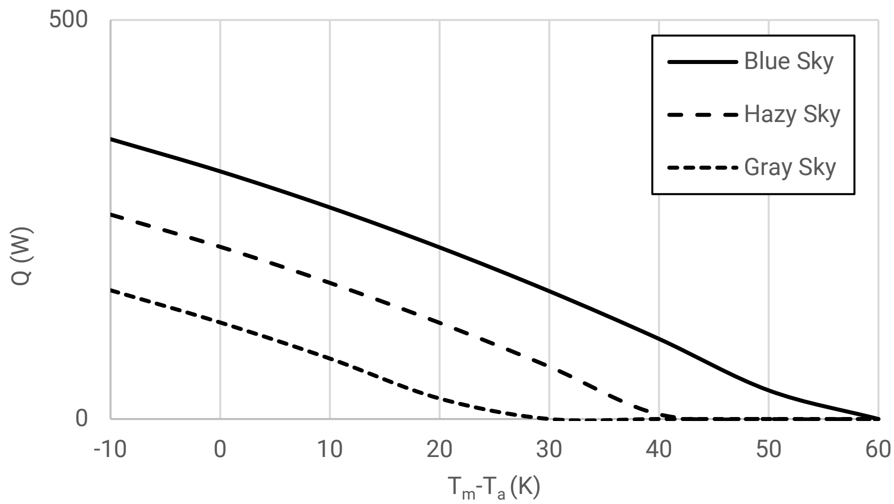
*** General thermal efficiency equation for mean (T_m) fluid temperature provided in accordance with ISO 9806-2017. Where data measured using a standard other than ISO 9806-2017, additional assumptions may be applied to determine extended equation coefficients. Reduced surrounding air speed (u') is defined as u'=u-3 m/s. See ISO 9806-2017 for other coefficient definitions.*



Angle (θ)	Total IAM K_b(θ)
0°	1.00
10°	1.00
20°	1.00
30°	1.00
40°	1.00
50°	1.00
60°	0.97
70°	0.89
80°	0.66
90°	0.00

POWER OUTPUT:

The instantaneous power output of the collector under different conditions is calculated at the Standard Reporting Conditions (SRC) defined by ISO 9806-2017 using the measured performance coefficients above.



STANDARD COLLECTOR POWER OUTPUT (W)			
Based on Standard Rating Conditions (SRC) and mean temperature (T_m) in accordance with ISO 9806-2017			
T_m-T_a (°C)	**Blue sky** G_b = 850, G_d = 150 (W/m^2)	**Hazy sky** G_b= 440, G_d= 260 (W/m^2)	**Grey sky** G_b=0, G_d= 400 (W/m^2)
-10	351	256	161
0	310*	216	121
10	265	171	76
20	215	120	26
30	160	66	0
40	100	6	0
50	36	0	0
60	0	0	0

**Peak Power is defined by ISO 9806 as the Blue Sky irradiance condition at T_m-T_a =0 and normal incidence.*

TEST SAMPLE SPECIFICATIONS:

The specifications of the collector sample submitted for testing are provided below.

TEST & SAMPLE SPECIFICATIONS					
Gross Length:	1.03 m	3.4 ft	**Maximum Design Temperature:**	57 °C	135 °F
Gross Width:	0.63 m	2.1 ft	**Maximum Design Pressure:**	3500 kPa	508 psi
Gross Depth:	277.0 mm	10.9 in	**Standard Stagnation Temperature:**	110 °C	230 °F
Empty Weight:	50.0 kg	110 lb			
Fluid Capacity:	2.22 L	0.6 gal			
Notes:	Standard stagnation at 1000 W/m^2 and 30 °C as defined by ISO 9806 – 2017. **Note: The standard stagnation temperature of this collector exceeds the maximum design temperature specified by the manufacturer. Therefore, it must be installed in systems where the temperature of the collector is prevented from exceeding the maximum design temperature to prevent damage to the collector and hazardous conditions.**				

IDENTIFICATION:

Certified systems must be identified with the OG-100 certification mark below in accordance with the *Rules for Certification Mark and Certificate Use* and labeled in with the information below per *ICC 901/SRCC 100*:

1. Manufacturer's name and model number.
2. OG-100 collector certification number
3. Maximum operating pressure
4. Dry weight
5. Fluid volume
6. Compatible heat transfer fluids
7. Standard stagnation temperature
8. Year of manufacture and/or serial number.

CONDITIONS:

1. Collector must be installed and operated in accordance with the manufacturer's published instructions and local codes and regulations.
2. OG-100 Standard Performance Ratings and Standard Collector Power Output have been calculated for the tested components using standardized conditions established by the OG-100 program and associated test standards. Actual performance will vary based on the specific usage, installation and local environmental conditions.
3. The collector listed in this ICC-SRCC OG-100 certification must be labeled in accordance with the *ICC-SRCC Rules for Mark and Certificate Use*.
4. OG-100 certifications do not include mounting hardware and fixtures.
5. Solar thermal collectors and mounting hardware and appurtenances must comply with all applicable local requirements for fire resistance. Solar thermal collectors must be mounted in accordance with the requirements of the collector and mounting hardware manufacturers to comply with local codes for structural loading for wind, seismic, snow and other loads.
6. Solar thermal collectors must be used with the heat transfer fluids listed in this document.
7. Solar thermal collector manufactured under a quality control program subject to periodic evaluation in accordance with the requirements of ICC-SRCC.
8. This document must be reproduced in its entirety.
9. Certification status should be confirmed on the ICC-SRCC Directory at www.solar-rating.org

ACCREDITED
CERTIFICATE # 3299.01, 3299.03

SOLAR RATING & CERTIFICATION CORPORATION

An ICC Evaluation Service Program

OG-100 Solar Thermal Collector Certification

No./10002196E

Issued: August 20, 2025

Expiration Date: June 01, 2026

www.solar-rating.org | **(800) 423-6587** | **(562) 699-0543**

CERTIFICATION HOLDER:	**EVALUATION SUBJECT**	
Commercial Energy Savings Plus 2604 NW 53rd Drive Boca Raton, FL 33496, USA. www.commercialesp.com	**BRAND:**	CESP
	MODEL:	SLR-BX-05C37.50
	TYPE:	Glazed Flat Plate

PRODUCT CERTIFICATION SYSTEM:

The ICC-SRCC OG-100 certification program includes evaluation and performance ratings for solar thermal collectors as established in the *ICC-SRCC Rules for Solar Heating & Cooling Product Listing Reports.* The program also includes periodic factory inspections and surveillance of the manufacturer's quality management system.

COMPLIANCE WITH THE FOLLOWING STANDARD(S): *ICC 901/SRCC 100 - 2020, Solar Thermal Collectors Standard*

OG-100 THERMAL PERFORMANCE RATINGS:

ICC-SRCC OG-100 thermal performance ratings provided for the collector are calculated for a 24-hour period using OG-100 standard conditions using collector parameters measured through laboratory testing. Actual performance will vary with local conditions, installation details and usage.

OG-100 STANDARD DAILY PRODUCTION TABLE							
Kilowatt-hours (thermal) per Collector per Day				**Thousands of BTU per Collector per Day**			
Climate → **Category** **(T_i-T_a)**	High Radiation (6.3 kWh/m²•day)	Medium Radiation (4.7 kWh/m²•day)	Low Radiation (3.1 kWh/m²•day)	**Climate →** **Category** **(T_i-T_a)**	High Radiation (2 kBTU/ft²•day)	Medium Radiation (1.5 kBTU/ft²•day)	Low Radiation (1 kBTU/ft²•day)
A (-5°C)	3.26	2.53	1.81	**A (-9°F)**	11.11	8.64	6.18
B (5°C)	2.52	1.80	1.08	**B (9°F)**	8.61	6.15	3.69
C (20°C)	1.48	0.83	0.24	**C (36°F)**	5.05	2.82	0.83
D (60°C)	0.02	0.00	0.00	**D (90°F)**	0.07	0.00	0.00
E (80°C)	0.00	0.00	0.00	**E (144°F)**	0.00	0.00	0.00

A – Pool Heating (Warm Climate) **B** – Pool Heating (Cool Climate) **C** – Water Heating (Warm Climate) **D** – Space & Water Heating (Cool Climate) **E** – Commercial Hot Water & Cooling

THERMAL EFFICIENCY:

The efficiency of solar thermal collectors is determined using test methods set in ICC 901/SRCC 100, based on ISO 9806 procedures. Results are processed to provide unique coefficients ($\eta_{0,hem}$, a_1, a_2…) for efficiency equations, provided in several forms below. For the simplified equations, instantaneous power at normal incidence is given by $\dot{Q} = K_\theta \eta_{hem} A_G G$. Incident Angle Modifiers (IAMs) are provided to indicate the change in output as the incidence angle of solar irradiance changes. The three forms should provide similar results, but the extended form is considered the most accurate. Users should select the equation to be used based the application and available data.

T_i: Temperature of the fluid entering the collector.
T_m: Average temperature of the fluid within the collector (between the inlet and outlet).
T_a: Temperature of the ambient air around the collector.
G: Hemispherical solar irradiance with sub-types beam (b) and diffuse (d)
E_L: Longwave (infrared) irradiance.
ε/α: Collector emissivity/absorptance ratio.

RATING CONDITIONS	
Gross Collector Area (A_G)	0.904 m^2
Fluid Mass Flowrate ($\dot{m}$)	0.090 kg/(m^{2}s)
Test Fluid	Water
Performance Test Standard	ISO 9806-2017

SIMPLIFIED THERMAL PERFORMANCE EQUATIONS (ISO 9806-2013, T_i, A_G)

Second Order Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G} - a_2 G \left(\frac{T_i - T_a}{G}\right)^2$$

Linearized Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G}$$

UNITS	$\eta_{0,hem}$	a_1	a_2	$\eta_{0,hem}$ ("Intercept")	a_1 ("Slope")
SI	0.520	6.995 W/(m^{2}K)	0.039 W/(m^{2}K^2)	0.538	9.141 W/(m^{2}K)
IP	0.520	1.233 BTU/(h ft^2°F)	0.004 BTU/(h ft^2°F^2)	0.538	1.611 BTU/(hr ft^2°F)

** Thermal efficiency equations per ISO 9806-2013 using inlet (Ti) fluid temperature, provided in second and first order (linearized) forms for normal incidence. The second order efficiency equation is a more accurate representation of the collector performance than the linearized equation. The linearized efficiency equation is provided for use with incentive programs, regulations and software that require the simplified "slope" and "intercept" coefficients to describe collector performance.*

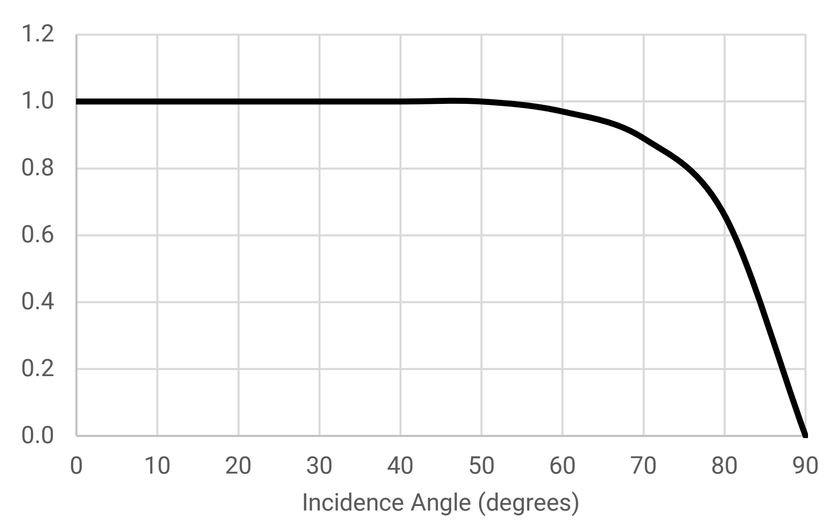
GENERAL THERMAL PERFORMANCE EQUATION (ISO 9806-2017, T_m, A_G)

Extended Thermal Efficiency Equation**

$$\dot{Q} = A_G(\eta_{0,b}K_b(\theta_L,\theta_T)G_b + \eta_{0,b}K_dG_d - a_1(T_m - T_a) - a_2(T_m - T_a)^2 - a_3u'(T_m - T_a) + a_4(E_L - \sigma T_a^4) - a_5\left(\frac{dT_m}{dt}\right) - a_6u'G - a_7u'(E_L - \sigma T_a^4) - a_8(T_m - T_a)^4$$

	$\eta_{0,hem}$	$\eta_{0,b}$	K_d	a_1	a_2	a_3	a_4	a_5	a_6	a_8
VALUE	0.525	0.528	0.97	7.087	0.037	0.000	0.000	10730	0.000	0.000
UNITS	-	-	-	W/(m^{2}K)	W/(m^{2}K^2)	J/(m^{3}K)	-	J/(m^{2}K)	s/m	W/(m^{2}K^4)

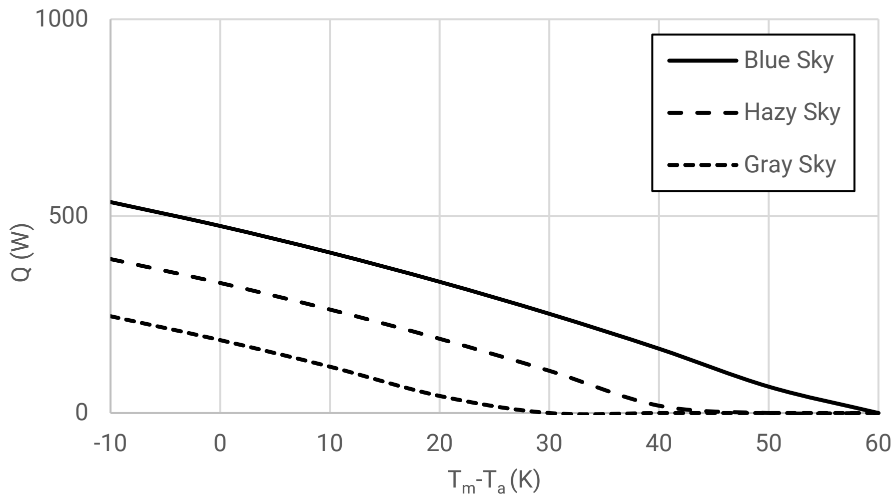
*** General thermal efficiency equation for mean (T_m) fluid temperature provided in accordance with ISO 9806-2017. Where data measured using a standard other than ISO 9806-2017, additional assumptions may be applied to determine extended equation coefficients. Reduced surrounding air speed (u') is defined as u'=u-3 m/s. See ISO 9806-2017 for other coefficient definitions.*



Angle (θ)	Total IAM $K_b(\theta)$
0°	1.00
10°	1.00
20°	1.00
30°	1.00
40°	1.00
50°	1.00
60°	0.97
70°	0.89
80°	0.66
90°	0.00

POWER OUTPUT:

The instantaneous power output of the collector under different conditions is calculated at the Standard Reporting Conditions (SRC) defined by ISO 9806-2017 using the measured performance coefficients above.



STANDARD COLLECTOR POWER OUTPUT (W)			
Based on Standard Rating Conditions (SRC) and mean temperature (T_m) in accordance with ISO 9806-2017			
T_m-T_a (°C)	**Blue sky** G_b = 850, G_d = 150 (W/m^2)	**Hazy sky** G_b= 440, G_d= 260 (W/m^2)	**Grey sky** G_b=0, G_d= 400 (W/m^2)
-10	536	391	246
0	475*	330	185
10	408	263	118
20	333	189	43
30	252	107	0
40	163	19	0
50	67	0	0
60	0	0	0

**Peak Power is defined by ISO 9806 as the Blue Sky irradiance condition at T_m-T_a =0 and normal incidence.*

TEST SAMPLE SPECIFICATIONS:

The specifications of the collector sample submitted for testing are provided below.

TEST & SAMPLE SPECIFICATIONS					
Gross Length:	1.16 m	3.8 ft	**Maximum Design Temperature:**	57 °C	135 °F
Gross Width:	0.78 m	2.6 ft	**Maximum Design Pressure:**	3500 kPa	508 psi
Gross Depth:	224.0 mm	8.8 in	**Standard Stagnation Temperature:**	110 °C	230 °F
Empty Weight:	56.7 kg	125 lb			
Fluid Capacity:	4.12 L	1.1 gal			
Notes:	Standard stagnation at 1000 W/m^2 and 30 °C as defined by ISO 9806 – 2017. **Note: The standard stagnation temperature of this collector exceeds the maximum design temperature specified by the manufacturer. Therefore, it must be installed in systems where the temperature of the collector is prevented from exceeding the maximum design temperature to prevent damage to the collector and hazardous conditions.**				

IDENTIFICATION:

Certified systems must be identified with the OG-100 certification mark below in accordance with the *Rules for Certification Mark and Certificate Use* and labeled in with the information below per *ICC 901/SRCC 100*:

1. Manufacturer's name and model number.
2. OG-100 collector certification number
3. Maximum operating pressure
4. Dry weight
5. Fluid volume
6. Compatible heat transfer fluids
7. Standard stagnation temperature
8. Year of manufacture and/or serial number.

CONDITIONS:

1. Collector must be installed and operated in accordance with the manufacturer's published instructions and local codes and regulations.
2. OG-100 Standard Performance Ratings and Standard Collector Power Output have been calculated for the tested components using standardized conditions established by the OG-100 program and associated test standards. Actual performance will vary based on the specific usage, installation and local environmental conditions.
3. The collector listed in this ICC-SRCC OG-100 certification must be labeled in accordance with the *ICC-SRCC Rules for Mark and Certificate Use*.
4. OG-100 certifications do not include mounting hardware and fixtures.
5. Solar thermal collectors and mounting hardware and appurtenances must comply with all applicable local requirements for fire resistance. Solar thermal collectors must be mounted in accordance with the requirements of the collector and mounting hardware manufacturers to comply with local codes for structural loading for wind, seismic, snow and other loads.
6. Solar thermal collectors must be used with the heat transfer fluids listed in this document.
7. Solar thermal collector manufactured under a quality control program subject to periodic evaluation in accordance with the requirements of ICC-SRCC.
8. This document must be reproduced in its entirety.
9. Certification status should be confirmed on the ICC-SRCC Directory at www.solar-rating.org

CERTIFICATE # 3299.01, 3299.03

OG-100 Solar Thermal Collector Certification

No./10002196F

Issued: August 20, 2025

Expiration Date: June 01, 2026

www.solar-rating.org | **(800) 423-6587 | (562) 699-0543**

CERTIFICATION HOLDER:	**EVALUATION SUBJECT**	
Commercial Energy Savings Plus 2604 NW 53rd Drive Boca Raton, FL 33496, USA. www.commercialesp.com	**BRAND:**	CESP
	MODEL:	SLR-BX-05C50.00
	TYPE:	Glazed Flat Plate

PRODUCT CERTIFICATION SYSTEM:

The ICC-SRCC OG-100 certification program includes evaluation and performance ratings for solar thermal collectors as established in the *ICC-SRCC Rules for Solar Heating & Cooling Product Listing Reports.* The program also includes periodic factory inspections and surveillance of the manufacturer's quality management system.

COMPLIANCE WITH THE FOLLOWING STANDARD(S): *ICC 901/SRCC 100 - 2020, Solar Thermal Collectors Standard*

OG-100 THERMAL PERFORMANCE RATINGS:

ICC-SRCC OG-100 thermal performance ratings provided for the collector are calculated for a 24-hour period using OG-100 standard conditions using collector parameters measured through laboratory testing. Actual performance will vary with local conditions, installation details and usage.

OG-100 STANDARD DAILY PRODUCTION TABLE							
Kilowatt-hours (thermal) per Collector per Day				Thousands of BTU per Collector per Day			
Climate → **Category (T_i-T_a)**	High Radiation (6.3 kWh/m²•day)	Medium Radiation (4.7 kWh/m²•day)	Low Radiation (3.1 kWh/m²•day)	**Climate →** **Category (T_i-T_a)**	High Radiation (2 kBTU/ft²•day)	Medium Radiation (1.5 kBTU/ft²•day)	Low Radiation (1 kBTU/ft²•day)
A (-5°C)	4.35	3.39	2.42	**A (-9°F)**	14.85	11.55	8.26
B (5°C)	3.38	2.41	1.45	**B (9°F)**	11.52	8.24	4.95
C (20°C)	1.99	1.11	0.33	**C (36°F)**	6.79	3.80	1.13
D (60°C)	0.03	0.00	0.00	**D (90°F)**	0.11	0.00	0.00
E (80°C)	0.00	0.00	0.00	**E (144°F)**	0.00	0.00	0.00

A – Pool Heating (Warm Climate) **B** – Pool Heating (Cool Climate) **C** – Water Heating (Warm Climate) **D** – Space & Water Heating (Cool Climate) **E** – Commercial Hot Water & Cooling

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T_i: Temperature of the fluid entering the collector.
T_m: Average temperature of the fluid within the collector (between the inlet and outlet).
T_a: Temperature of the ambient air around the collector.
G: Hemispherical solar irradiance with sub-types beam (b) and diffuse (d)
E_L: Longwave (infrared) irradiance.
ε/α: Collector emissivity/absorptance ratio.

RATING CONDITIONS	
Gross Collector Area (A_G)	1.159 m^2
Fluid Mass Flowrate ($\dot{m}$)	0.090 kg/(m^{2}s)
Test Fluid	Water
Performance Test Standard	ISO 9806-2017

SIMPLIFIED THERMAL PERFORMANCE EQUATIONS (ISO 9806-2013, T_i, A_G)

Second Order Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1\frac{T_i - T_a}{G} - a_2 G\left(\frac{T_i - T_a}{G}\right)^2$$

Linearized Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1\frac{T_i - T_a}{G}$$

UNITS	$\eta_{0,hem}$	a_1	a_2	$\eta_{0,hem}$ ("Intercept")	a_1 ("Slope")
SI	0.543	7.264 W/(m^{2}K)	0.041 W/(m^{2}K^2)	0.562	9.504 W/(m^{2}K)
IP	0.543	1.280 BTU/(h ft^2°F)	0.004 BTU/(h ft^2°F^2)	0.562	1.675 BTU/(hr ft^2°F)

** Thermal efficiency equations per ISO 9806-2013 using inlet (Ti) fluid temperature, provided in second and first order (linearized) forms for normal incidence. The second order efficiency equation is a more accurate representation of the collector performance than the linearized equation. The linearized efficiency equation is provided for use with incentive programs, regulations and software that require the simplified "slope" and "intercept" coefficients to describe collector performance.*

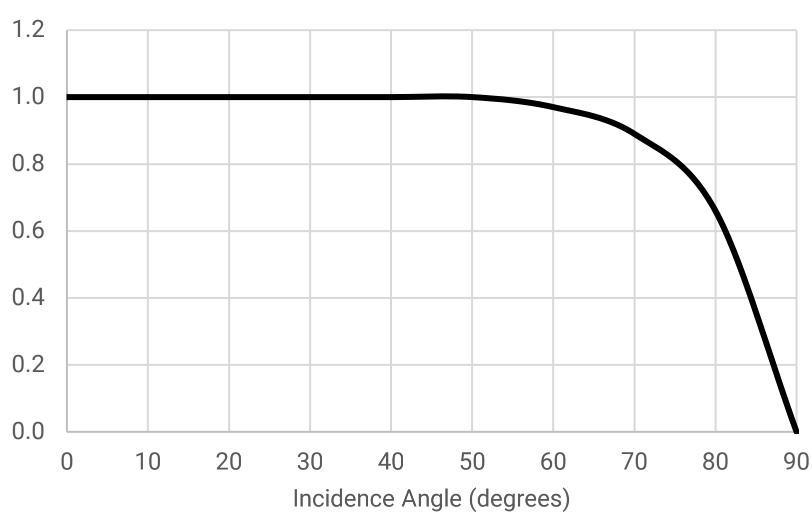
GENERAL THERMAL PERFORMANCE EQUATION (ISO 9806-2017, T_m, A_G)

Extended Thermal Efficiency Equation**

$$\dot{Q} = A_G(\eta_{0,b}K_b(\theta_L,\theta_T)G_b + \eta_{0,b}K_dG_d - a_1(T_m - T_a) - a_2(T_m - T_a)^2 - a_3u'(T_m - T_a) + a_4(E_L - \sigma T_a^4) - a_5\left(\frac{dT_m}{dt}\right) - a_6u'G - a_7u'(E_L - \sigma T_a^4) - a_8(T_m - T_a)^4$$

	$\eta_{0,hem}$	$\eta_{0,b}$	K_d	a_1	a_2	a_3	a_4	a_5	a_6	a_8
VALUE	0.549	0.551	0.97	7.359	0.039	0.000	0.000	8372	0.000	0.000
UNITS	-	-	-	W/(m^{2}K)	W/(m^{2}K^2)	J/(m^{3}K)	-	J/(m^{2}K)	s/m	W/(m^{2}K^4)

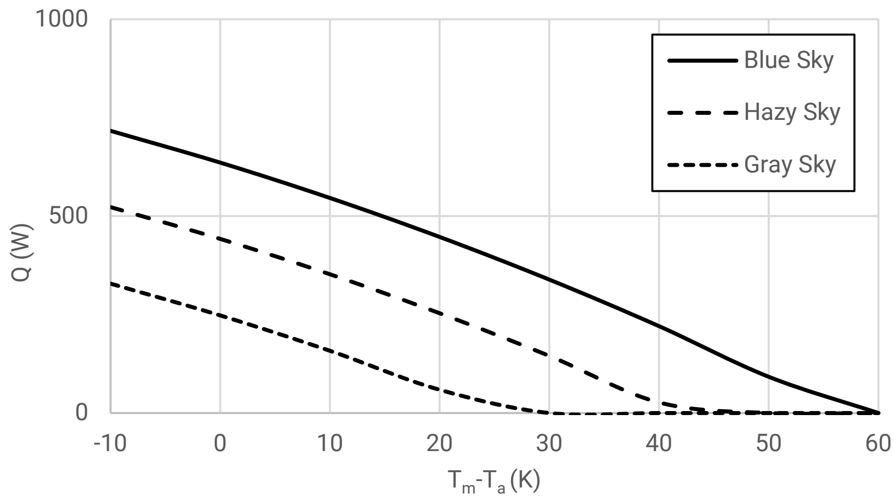
*** General thermal efficiency equation for mean (T_m) fluid temperature provided in accordance with ISO 9806-2017. Where data measured using a standard other than ISO 9806-2017, additional assumptions may be applied to determine extended equation coefficients. Reduced surrounding air speed (u') is defined as u'=u-3 m/s. See ISO 9806-2017 for other coefficient definitions.*



Angle (θ)	Total IAM K_b(θ)
0°	1.00
10°	1.00
20°	1.00
30°	1.00
40°	1.00
50°	1.00
60°	0.97
70°	0.89
80°	0.66
90°	0.00

POWER OUTPUT:

The instantaneous power output of the collector under different conditions is calculated at the Standard Reporting Conditions (SRC) defined by ISO 9806-2017 using the measured performance coefficients above.



STANDARD COLLECTOR POWER OUTPUT (W)			
Based on Standard Rating Conditions (SRC) and mean temperature (T_m) in accordance with ISO 9806-2017			
T_m-T_a (°C)	**Blue sky** G_b = 850, G_d = 150 (W/m^2)	**Hazy sky** G_b= 440, G_d= 260 (W/m^2)	**Grey sky** G_b=0, G_d= 400 (W/m^2)
-10	717	523	329
0	636*	442	248
10	546	352	158
20	447	253	59
30	339	145	0
40	221	27	0
50	92	0	0
60	0	0	0

**Peak Power is defined by ISO 9806 as the Blue Sky irradiance condition at T_m-T_a =0 and normal incidence.*

TEST SAMPLE SPECIFICATIONS:

The specifications of the collector sample submitted for testing are provided below.

TEST & SAMPLE SPECIFICATIONS					
Gross Length:	1.47 m	4.8 ft	**Maximum Design Temperature:**	57 °C	135 °F
Gross Width:	0.79 m	2.6 ft	**Maximum Design Pressure:**	3500 kPa	508 psi
Gross Depth:	224.0 mm	8.8 in	**Standard Stagnation Temperature:**	110 °C	230 °F
Empty Weight:	68.0 kg	150 lb			
Fluid Capacity:	5.19 L	1.4 gal			
Notes:	Standard stagnation at 1000 W/m^2 and 30 °C as defined by ISO 9806 – 2017. **Note: The standard stagnation temperature of this collector exceeds the maximum design temperature specified by the manufacturer. Therefore, it must be installed in systems where the temperature of the collector is prevented from exceeding the maximum design temperature to prevent damage to the collector and hazardous conditions.**				

IDENTIFICATION:

Certified systems must be identified with the OG-100 certification mark below in accordance with the *Rules for Certification Mark and Certificate Use* and labeled in with the information below per *ICC 901/SRCC 100*:

1. Manufacturer's name and model number.
2. OG-100 collector certification number
3. Maximum operating pressure
4. Dry weight
5. Fluid volume
6. Compatible heat transfer fluids
7. Standard stagnation temperature
8. Year of manufacture and/or serial number.

CONDITIONS:

1. Collector must be installed and operated in accordance with the manufacturer's published instructions and local codes and regulations.
2. OG-100 Standard Performance Ratings and Standard Collector Power Output have been calculated for the tested components using standardized conditions established by the OG-100 program and associated test standards. Actual performance will vary based on the specific usage, installation and local environmental conditions.
3. The collector listed in this ICC-SRCC OG-100 certification must be labeled in accordance with the *ICC-SRCC Rules for Mark and Certificate Use*.
4. OG-100 certifications do not include mounting hardware and fixtures.
5. Solar thermal collectors and mounting hardware and appurtenances must comply with all applicable local requirements for fire resistance. Solar thermal collectors must be mounted in accordance with the requirements of the collector and mounting hardware manufacturers to comply with local codes for structural loading for wind, seismic, snow and other loads.
6. Solar thermal collectors must be used with the heat transfer fluids listed in this document.
7. Solar thermal collector manufactured under a quality control program subject to periodic evaluation in accordance with the requirements of ICC-SRCC.
8. This document must be reproduced in its entirety.
9. Certification status should be confirmed on the ICC-SRCC Directory at www.solar-rating.org

CERTIFICATE # 3299.01, 3299.03

OG-100 Solar Thermal Collector Certification

No./10002196G

Issued: August 20, 2025

Expiration Date: June 01, 2026

www.solar-rating.org | **(800) 423-6587** | **(562) 699-0543**

CERTIFICATION HOLDER:	**EVALUATION SUBJECT**	
Commercial Energy Savings Plus 2604 NW 53rd Drive Boca Raton, FL 33496, USA. www.commercialesp.com	**BRAND:**	CESP
	MODEL:	SLR-BX-05C75.00
	TYPE:	Glazed Flat Plate

PRODUCT CERTIFICATION SYSTEM:

The ICC-SRCC OG-100 certification program includes evaluation and performance ratings for solar thermal collectors as established in the *ICC-SRCC Rules for Solar Heating & Cooling Product Listing Reports.* The program also includes periodic factory inspections and surveillance of the manufacturer's quality management system.

COMPLIANCE WITH THE FOLLOWING STANDARD(S): *ICC 901/SRCC 100 - 2020, Solar Thermal Collectors Standard*

OG-100 THERMAL PERFORMANCE RATINGS:

ICC-SRCC OG-100 thermal performance ratings provided for the collector are calculated for a 24-hour period using OG-100 standard conditions using collector parameters measured through laboratory testing. Actual performance will vary with local conditions, installation details and usage.

OG-100 STANDARD DAILY PRODUCTION TABLE

Kilowatt-hours (thermal) per Collector per Day				**Thousands of BTU per Collector per Day**			
Climate → **Category (T_i-T_a)**	High Radiation (6.3 kWh/m²•day)	Medium Radiation (4.7 kWh/m²•day)	Low Radiation (3.1 kWh/m²•day)	**Climate →** **Category (T_i-T_a)**	High Radiation (2 kBTU/ft²•day)	Medium Radiation (1.5 kBTU/ft²•day)	Low Radiation (1 kBTU/ft²•day)
A (-5°C)	6.86	5.33	3.81	**A (-9°F)**	23.39	18.19	12.99
B (5°C)	5.33	3.81	2.29	**B (9°F)**	18.17	12.99	7.82
C (20°C)	3.15	1.77	0.53	**C (36°F)**	10.74	6.03	1.81
D (60°C)	0.06	0.00	0.00	**D (90°F)**	0.19	0.00	0.00
E (80°C)	0.00	0.00	0.00	**E (144°F)**	0.00	0.00	0.00

A – Pool Heating (Warm Climate) **B** – Pool Heating (Cool Climate) **C** – Water Heating (Warm Climate) **D** – Space & Water Heating (Cool Climate) **E** – Commercial Hot Water & Cooling

THERMAL EFFICIENCY:

The efficiency of solar thermal collectors is determined using test methods set in ICC 901/SRCC 100, based on ISO 9806 procedures. Results are processed to provide unique coefficients ($\eta_{0,hem}$, a_1, a_2…) for efficiency equations, provided in several forms below. For the simplified equations, instantaneous power at normal incidence is given by $\dot{Q} = K_\theta \eta_{hem} A_G G$. Incident Angle Modifiers (IAMs) are provided to indicate the change in output as the incidence angle of solar irradiance changes. The three forms should provide similar results, but the extended form is considered the most accurate. Users should select the equation to be used based the application and available data.

T_i: Temperature of the fluid entering the collector.
T_m: Average temperature of the fluid within the collector (between the inlet and outlet).
T_a: Temperature of the ambient air around the collector.
G: Hemispherical solar irradiance with sub-types beam (b) and diffuse (d)
E_L: Longwave (infrared) irradiance.
ε/α: Collector emissivity/absorptance ratio.

RATING CONDITIONS	
Gross Collector Area (A_G)	1.690 m^2
Fluid Mass Flowrate ($\dot{m}$)	0.090 kg/(m^2s)
Test Fluid	Water
Performance Test Standard	ISO 9806-2017

SIMPLIFIED THERMAL PERFORMANCE EQUATIONS (ISO 9806-2013, T_i, A_G)

Second Order Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G} - a_2 G \left(\frac{T_i - T_a}{G}\right)^2$$

Linearized Thermal Efficiency Equation*

$$\eta_{hem} = \eta_{0,hem} - a_1 \frac{T_i - T_a}{G}$$

UNITS	$\eta_{0,hem}$	a_1	a_2	$\eta_{0,hem}$ ("Intercept")	a_1 ("Slope")
SI	0.587	7.800 W/(m^2K)	0.044 W/(m^2K^2)	0.607	10.220 W/(m^2K)
IP	0.587	1.375 BTU/(h ft^2°F)	0.004 BTU/(h ft^2°F^2)	0.607	1.801 BTU/(hr ft^2°F)

** Thermal efficiency equations per ISO 9806-2013 using inlet (Ti) fluid temperature, provided in second and first order (linearized) forms for normal incidence. The second order efficiency equation is a more accurate representation of the collector performance than the linearized equation. The linearized efficiency equation is provided for use with incentive programs, regulations and software that require the simplified "slope" and "intercept" coefficients to describe collector performance.*

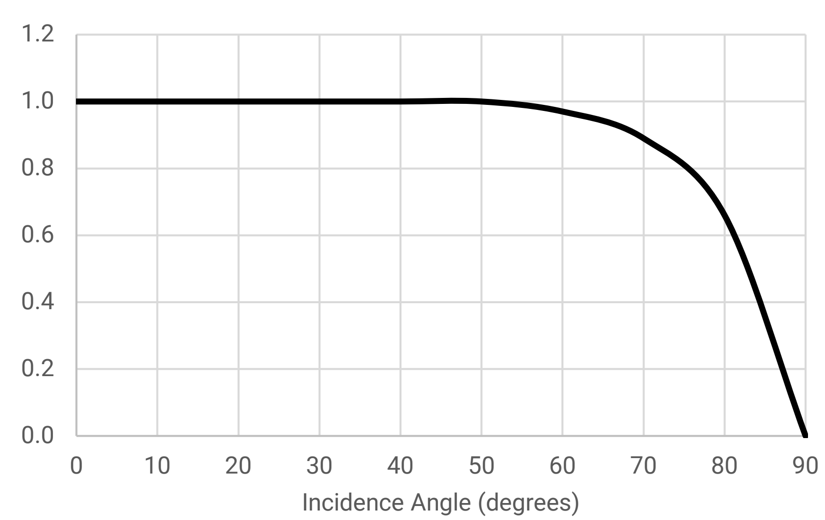
GENERAL THERMAL PERFORMANCE EQUATION (ISO 9806-2017, T_m, A_G)

Extended Thermal Efficiency Equation**

$$\dot{Q} = A_G(\eta_{0,b} K_b(\theta_L, \theta_T) G_b + \eta_{0,b} K_d G_d - a_1(T_m - T_a) - a_2(T_m - T_a)^2 - a_3 u'(T_m - T_a) + a_4(E_L - \sigma T_a^4) - a_5\left(\frac{dT_m}{dt}\right) - a_6 u' G - a_7 u'(E_L - \sigma T_a^4) - a_8(T_m - T_a)^4$$

	$\eta_{0,hem}$	$\eta_{0,b}$	K_d	a_1	a_2	a_3	a_4	a_5	a_6	a_8
VALUE	0.593	0.596	0.97	7.903	0.042	0.000	0.000	5740	0.000	0.000
UNITS	-	-	-	W/(m^2K)	W/(m^2K^2)	J/(m^3K)	-	J/(m^2K)	s/m	W/(m^2K^4)

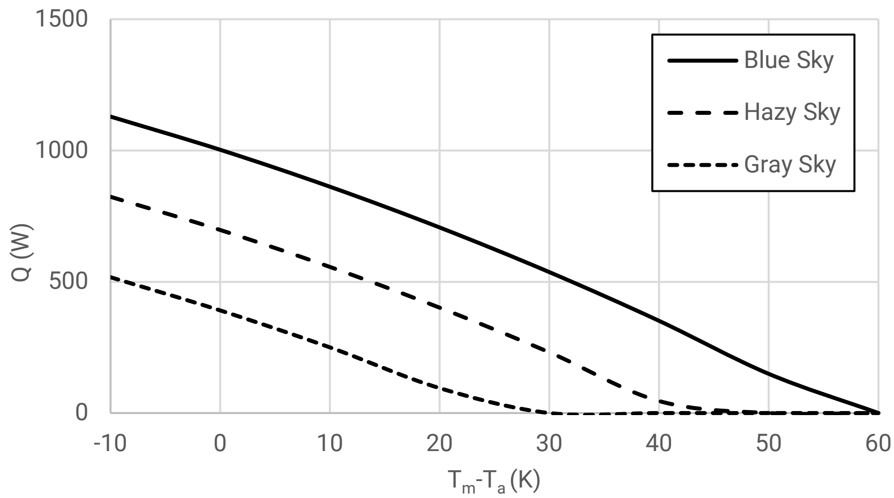
*** General thermal efficiency equation for mean (T_m) fluid temperature provided in accordance with ISO 9806-2017. Where data measured using a standard other than ISO 9806-2017, additional assumptions may be applied to determine extended equation coefficients. Reduced surrounding air speed (u') is defined as u'=u-3 m/s. See ISO 9806-2017 for other coefficient definitions.*



Angle (θ)	Total IAM $K_b(\theta)$
0°	1.00
10°	1.00
20°	1.00
30°	1.00
40°	1.00
50°	1.00
60°	0.97
70°	0.89
80°	0.66
90°	0.00

POWER OUTPUT:

The instantaneous power output of the collector under different conditions is calculated at the Standard Reporting Conditions (SRC) defined by ISO 9806-2017 using the measured performance coefficients above.



STANDARD COLLECTOR POWER OUTPUT (W)			
Based on Standard Rating Conditions (SRC) and mean temperature (T_m) in accordance with ISO 9806-2017			
T_m-T_a (°C)	**Blue sky** G_b = 850, G_d = 150 (W/m^2)	**Hazy sky** G_b= 440, G_d= 260 (W/m^2)	**Grey sky** G_b=0, G_d= 400 (W/m^2)
-10	1129	824	517
0	1003*	697	391
10	862	557	250
20	707	401	95
30	537	231	0
40	351	46	0
50	149	0	0
60	0	0	0

**Peak Power is defined by ISO 9806 as the Blue Sky irradiance condition at T_m-T_a =0 and normal incidence.*

TEST SAMPLE SPECIFICATIONS:

The specifications of the collector sample submitted for testing are provided below.

TEST & SAMPLE SPECIFICATIONS					
Gross Length:	2.15 m	7.1 ft	**Maximum Design Temperature:**	57 °C	135 °F
Gross Width:	0.79 m	2.6 ft	**Maximum Design Pressure:**	3500 kPa	508 psi
Gross Depth:	221.0 mm	8.7 in	**Standard Stagnation Temperature:**	110 °C	230 °F
Empty Weight:	113.4 kg	250 lb			
Fluid Capacity:	7.03 L	1.9 gal			
Notes:	Standard stagnation at 1000 W/m^2 and 30 °C as defined by ISO 9806 – 2017. **Note: The standard stagnation temperature of this collector exceeds the maximum design temperature specified by the manufacturer. Therefore, it must be installed in systems where the temperature of the collector is prevented from exceeding the maximum design temperature to prevent damage to the collector and hazardous conditions.**				

IDENTIFICATION:

Certified systems must be identified with the OG-100 certification mark below in accordance with the *Rules for Certification Mark and Certificate Use* and labeled in with the information below per *ICC 901/SRCC 100*:

1. Manufacturer's name and model number.
2. OG-100 collector certification number
3. Maximum operating pressure
4. Dry weight
5. Fluid volume
6. Compatible heat transfer fluids
7. Standard stagnation temperature
8. Year of manufacture and/or serial number.

CONDITIONS:

1. Collector must be installed and operated in accordance with the manufacturer's published instructions and local codes and regulations.
2. OG-100 Standard Performance Ratings and Standard Collector Power Output have been calculated for the tested components using standardized conditions established by the OG-100 program and associated test standards. Actual performance will vary based on the specific usage, installation and local environmental conditions.
3. The collector listed in this ICC-SRCC OG-100 certification must be labeled in accordance with the *ICC-SRCC Rules for Mark and Certificate Use*.
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ACCREDITED
CERTIFICATE # 3299.01, 3299.03

Date: October 17, 2025

From: Joseph E. Finkam, P.E.

Re: Solar Thermal HVAC test

This test report will document the test conducted on the iAIRE Solar Thermal HVAC to demonstrate the ability of the equipment to be able to store thermal energy and then utilize this thermal energy to improve the operation of the iAIRE Solar Thermal HVAC system for at least 1 hour.

<u>Equipment utilized in the test</u>

The test was conducted utilizing an iAIRE Model SHRPZ-60LH00A00*-F 5-ton residential Solar Thermal HVAC system. This system utilized 208/230V, 1 phase power to operate both the solar condensing unit and the air handling unit (AHU). The condensing unit and AHU were plumbed together on a skid.

<u>Test Method</u>

The solar panel on the condensing unit was covered to prevent the solar panel absorbing thermal energy (being charged) and allow any heat to dissipate prior to the start of the test. The Solar HVAC unit did not operate during this time. The unit was not exposed to any sunlight for more than 1 day. At the start of the test, the cover was removed from the solar panel, and the temperature of the solar panel was measured with an infrared temperature gun. This temperature would be the temperature of the solar panel in the "uncharged" state. The solar panel was then exposed to the sun to allow the solar panel to be charged. Temperatures were taken every 10 minutes to record the temperature rising in the solar panel. After approximately 1 hour of exposure to the sun to put the thermal energy into the solar panel, the Solar HVAC unit was turned on to operate and forced to operate for 1 hour. Temperatures of the solar panel were taken every 10 minutes showing the decay in the temperature in the solar panel as the thermal energy was being transferred into the system. At the end of 1 hour, the solar panel should still be slightly above the initial temperature of the solar panel to demonstrate the stored thermal energy could help improve the operation of the system for at least 1 hour.

<u>Test Results</u>

The table below shows the temperature of the solar panel during the course of the test. The first temperature is the solar panel prior to being exposed to the sun. The next set of data is the solar panel during the 1 hour and 20-minute period where the thermal energy was added to the solar panel without the Solar HVAC system operating to charge the thermal storage device. The last set of data is the temperature in the solar panel as the Solar HVAC system was operating.

Item	Temperature (F)	Time
Unit Not exposed to Sun to Charge Solar Thermal Panel	81.1	12:40
Sun exposure, no unit operation	82.5	12:45
Sun exposure, no unit operation	88.5	12:56
Sun exposure, no unit operation	91.5	13:06
Sun exposure, no unit operation	101.3	13:16
Sun exposure, no unit operation	108.1	13:26
Sun exposure, no unit operation	105.2	13:36
Sun exposure, no unit operation	108.2	13:46
Sun exposure, no unit operation	108.6	13:56
Sun exposure, no unit operation	111.5	14:04
Unit Operational	111.5	14:05
Unit Operational	93.7	14:15
Unit Operational	93.1	14:25
Unit Operational	90.5	14:35
Unit Operational	88.3	14:45
Unit Operational	85.8	14:55
Unit Operational	83.1	15:05

As seen from the above data, the solar panel started from the initial temperature of 81.1 degrees Fahrenheit. After the solar panel is charged to get the thermal energy into the solar panel without the Solar HVAC unit operating, the solar panel got to 111.5 degrees Fahrenheit. The Solar HVAC was operated for 1 hour and temperatures recorded every 10 minutes. The temperature in the solar panel at the end of 1 hour was 83.1 degrees. The temperature at the end of 1 hour of operation was more than the initial starting temperature (reference). The test showed that the solar panel's stored energy helped improve the operational efficiency of the Solar HVAC system for at least 1 hour.

Summary

iAIRE's Solar Thermal HVAC system meets the requirement of having the ability to store thermal energy and utilize the stored thermal energy to improve operation of an HVAC system for at least 1 hour.