US012405034B2

(12) **United States Patent**
Rossi et al.

(10) **Patent No.: US 12,405,034 B2**
(45) **Date of Patent: Sep. 2, 2025**

(54) **COMPACT HEAT PUMP FOR COLD CLIMATE APPLICATIONS**

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(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 126 days.

(21) Appl. No.: **18/319,927**

(22) Filed: **May 18, 2023**

(65) **Prior Publication Data**

US 2024/0003595 A1 Jan. 4, 2024

Related U.S. Application Data

(60) Provisional application No. 63/365,008, filed on May 19, 2022.

(51) **Int. Cl.**
F25B 7/00 (2006.01)
F24F 1/14 (2011.01)
F25B 30/02 (2006.01)

(52) **U.S. Cl.**
CPC ***F25B 7/00*** (2013.01); ***F24F 1/14*** (2013.01); ***F25B 30/02*** (2013.01)

(58) **Field of Classification Search**
CPC F25B 7/00; F25B 1/10; F25B 30/02; F24F 1/14
See application file for complete search history.

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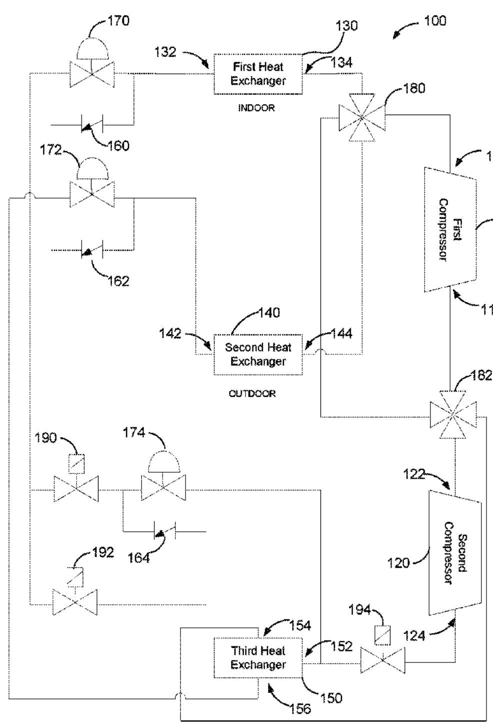
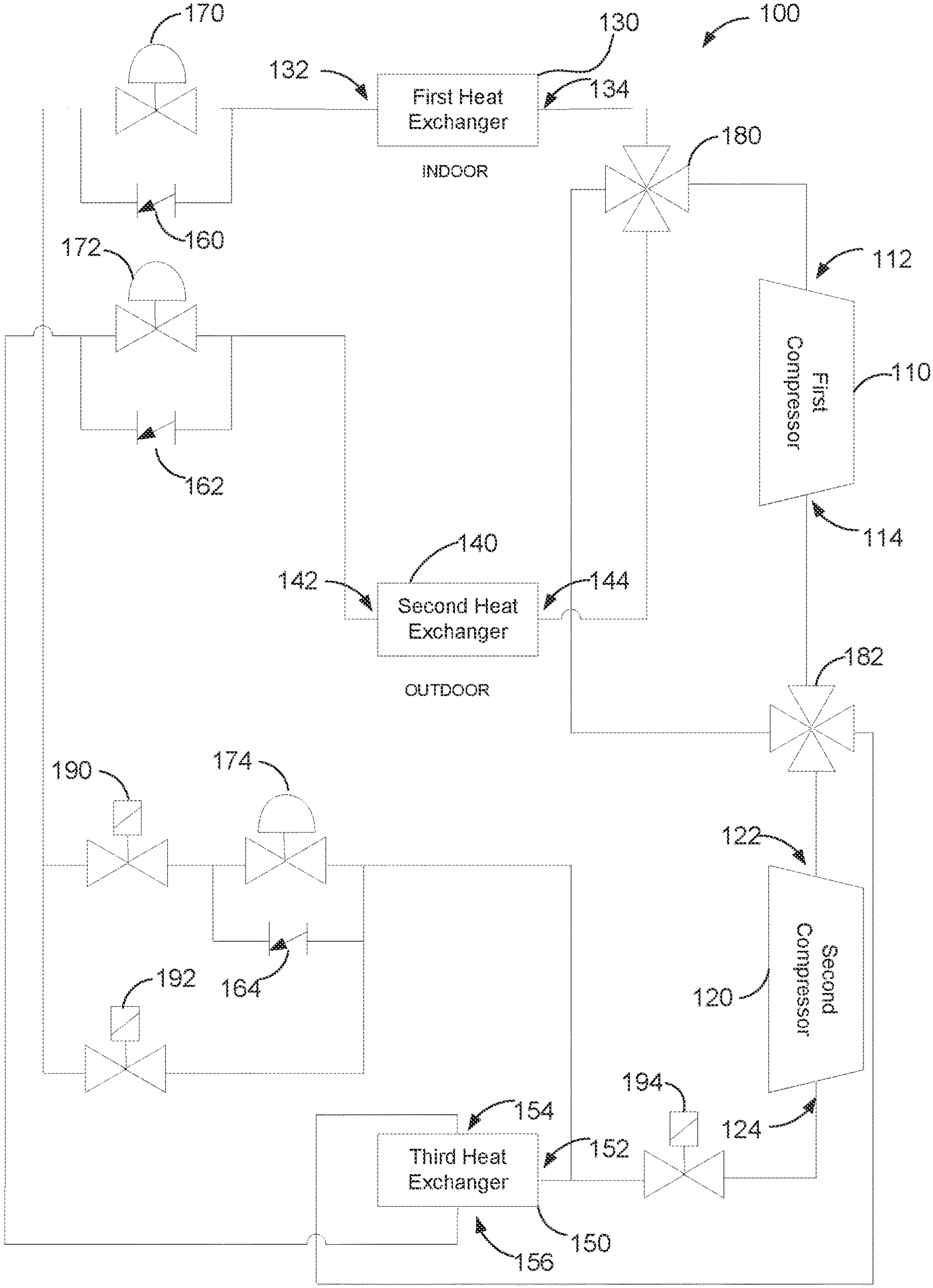
(57) **ABSTRACT**

A heat pump system is provided. The heat pump system includes a first compressor and a second compressor and a first heat exchanger, a second heat exchanger, and a third heat exchanger. The system further includes a first expansion device, a second expansion device, and a third expansion device. When the system operates in a cascade heating mode, the third heat exchanger is configured to: receive a first portion of refrigerant from the first compressor and a second portion of refrigerant from the second compressor, and provide a first consistent flow of refrigerant to the first compressor and a second consistent flow of refrigerant to the second compressor so as to improve an efficiency of the system.

16 Claims, 3 Drawing Sheets





170
132
130
134
100
First Heat Exchanger
INDOOR
180
160
172
112
First Compressor
110
162
114
140
142
144
Second Heat Exchanger
OUTDOOR
182
190
174
122
164
192
Second Compressor
120
194
154
152
124
Third Heat Exchanger
150
156

FIG. 1

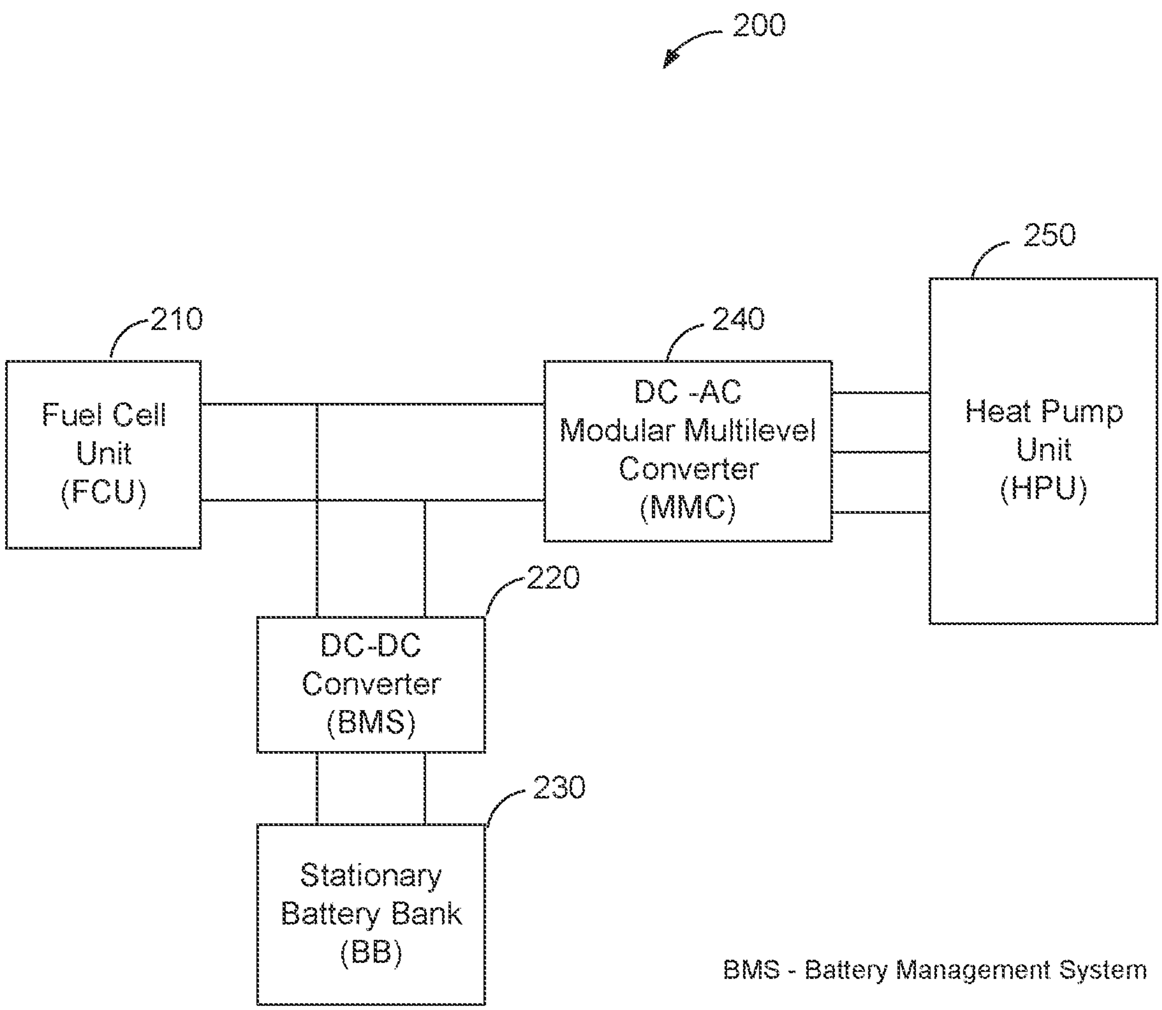
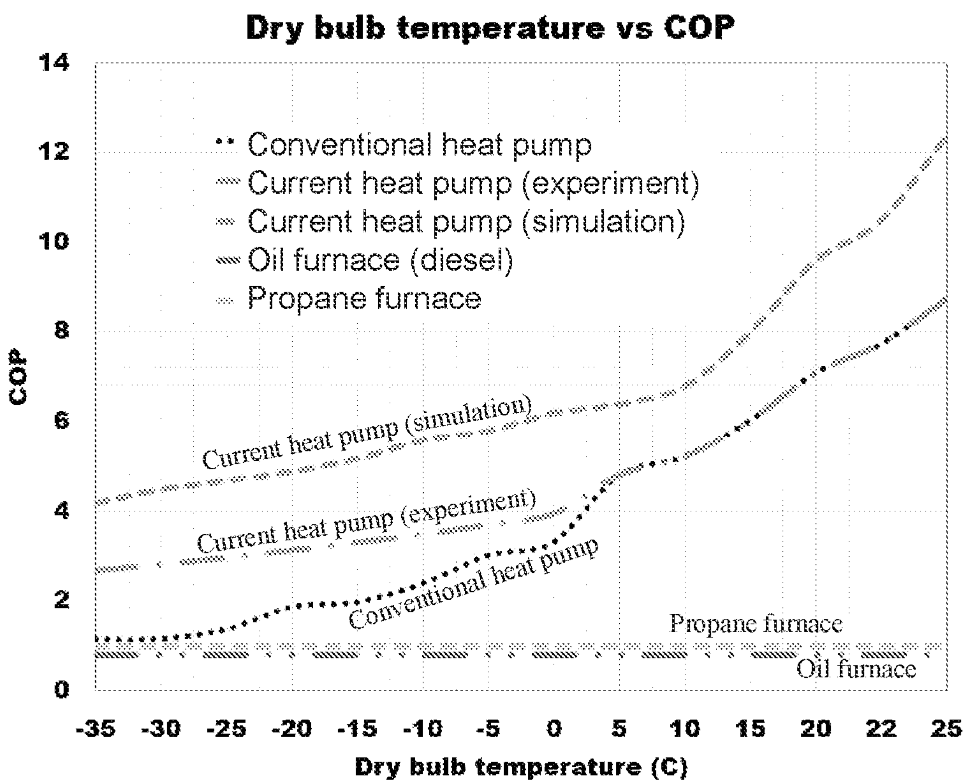

200
210
Fuel Cell Unit (FCU)
240
DC -AC Modular Multilevel Converter (MMC)
250
Heat Pump Unit (HPU)
220
DC-DC Converter (BMS)
230
Stationary Battery Bank (BB)
BMS - Battery Management System

FIG. 2

FIG. 3


Dry bulb temperature vs COP
Conventional heat pump
Current heat pump (experiment)
Current heat pump (simulation)
Oil furnace (diesel)
Propane furnace
Current heat pump (simulation)
Current heat pump (experiment)
Conventional heat pump
Propane furnace
Oil furnace
COP
14
12
10
8
6
4
2
0
-35
-30
-25
-20
-15
-10
-5
0
5
10
15
20
22
25
Dry bulb temperature (C)

COMPACT HEAT PUMP FOR COLD CLIMATE APPLICATIONS

CROSS-REFERENCE TO RELATED APPLICATION

This application claims priority under 35 U.S.C. § 119(e) to U.S. Provisional Application 63/365,008, filed on May 19, 2022. The disclosure of this prior application is considered part of the disclosure of this application and is hereby incorporated by reference in its entirety.

TECHNICAL FIELD

This disclosure relates to a heat pump system that is configured to operate efficiently in cold climate conditions.

BACKGROUND

A heat pump, which is a thermal energy transferring system, is able to heat or cool an enclosed space by transferring thermal energy. When used to cool a building, a heat pump works like an air conditioner by transferring heat from inside the building to the outdoors. When used to heat a building, the heat pump operates in reverse, such that heat is transferred from the outdoors to the inside of the building.

In an air source heat pump system, the efficiency of the heat pump system (e.g., Coefficient of Performance (COP)) is directly dependent on the temperature of the ambient air, which is the source of the thermal energy for heating indoor air. Accordingly, the COP of the heat pump is high when the ambient temperature is high. Likewise, the COP of the heat pump is low when the ambient temperature is low. As a result, in cold climates where the ambient air temperature often drops below −10° C., the COP of the heat pump is very poor, such that the air source heat pump system is not practical to use. Further, when the ambient air temperature is low, not only is the COP of the heat pump poor as discussed, but the heating capability of the heat pump is also poor since it is difficult to extract or absorb thermal energy from the cold ambient air.

SUMMARY

In one aspect, a heat pump system configured to provide a consistent flow of refrigerant to one or more compressors in the system is provided. The heat pump system includes a plurality of compressors including a first compressor and a second compressor; a plurality of heat exchangers including a first heat exchanger, a second heat exchanger, and a third heat exchanger; a plurality of expansion devices including a first expansion device, a second expansion device, and a third expansion device. When the system operates in a cascade heating mode, the third heat exchanger is configured to: receive a first portion of refrigerant from the first compressor and a second portion of refrigerant from the second compressor, and provide a first consistent flow of refrigerant to the first compressor and a second consistent flow of refrigerant to the second compressor.

In one aspect, the third heat exchanger includes a flash-tank-heat-exchanger.

In another aspect, the third heat exchanger is configured to contain a mixture of liquid refrigerant and vapor refrigerant.

In another aspect of the system, before received by the third exchanger, the first portion of refrigerant passes

through the first heat exchanger and the third expansion device. In such an aspect, the first heat exchanger is located indoors.

In another aspect, the third heat exchanger is configured to output a vapor portion of a mixture of the first portion and second portion of the refrigerant via a first output and a liquid portion of mixture of the first portion and second portion of the refrigerant via a second output. In such an aspect, the first output of the third heat exchanger is in fluid communication with an input of the first compressor when the system operates in the cascade heating mode. Further, the second output of the third heat exchanger may be in fluid communication with the second expansion device when the system operates in the cascade heating mode.

In another aspect, the second expansion device is in fluid communication with the second heat exchanger when the system operates in the cascade heating mode. The second heat exchanger may be in fluid communication with an input port of the second compressor when the system operates in the cascade heating mode. In yet another aspect, the second heat exchanger is located outdoors.

The system may include one or more of the following aspects: the first compressor includes a high-pressure compressor; the second compressor includes a low-pressure compressor; the first compressor and the second compressor are in operation when the system operates in the cascade heating mode; and when the system operates in a heating or a cooling mode, the first compressor is in operation and the second compressor is not in operation.

In another aspect of the disclosure relates to a heating and cooling system. The heating and cooling system includes a fuel cell unit; a backup battery unit including a battery management circuitry; a direct current (DC) to alternate current (AC) converter; and a heat pump system, wherein the modular multi-level converter is configured to: receive direct current from the fuel cell or the backup battery unit; convert the direct current to alternate current; and supply the alternate current to the heat pump system. The heat pump system includes a plurality of compressors including a first compressor and a second compressor; a plurality of heat exchangers including a first heat exchanger, a second heat exchanger, and a third heat exchanger; a plurality of expansion devices including a first expansion device, a second expansion device, and a third expansion device. When the system operates in a cascade heating mode, the third heat exchanger is configured to: receive a first portion of refrigerant from the first compressor and a second portion of refrigerant from the second compressor, and provide a first consistent flow of refrigerant to the first compressor and a second consistent flow of refrigerant to the second compressor.

In one aspect, the DC to AC converter includes a nine-level modular multi-level convert.

In another aspect, the fuel cell unit includes a hydrogen storage tank, a fuel cell, and a condenser.

In yet another aspect, wherein the battery management circuitry is configured to maintain the backup battery.

DESCRIPTION OF DRAWINGS

FIG. **1** is a schematic diagram of a heat pump system configured to operate in various operation modes, according to certain embodiments of the current invention.

FIG. **2** is a schematic diagram of a heat pump system integrated with a modular multi-level converter.

FIG. 3 is a comparison graph that shows the efficiency of a heat pump system in terms of the Coefficient of Performance at various temperatures.

Like reference symbols in the various drawings indicate like elements.

DETAILED DESCRIPTION

Implementations herein are directed toward a heat pump system (e.g., air source heat pump system) including a liquid receiver (e.g., third heat exchanger **150** in FIG. **1**) to overcome the inefficiency associated with current heat pump systems. In particular, the liquid receiver is configured to provide a consistent flow of refrigerant to one or more compressors in the system. The consistent flow also supplies an outdoor heat exchanger (e.g., second heat exchanger **140** in FIG. **1**) with consistent flow of the refrigerant which is colder than ambient air temperature. Thus, the heating capability and the COP of the heat pump are maintained even when the ambient air temperature is very low, e.g., −30° C. As will be described in the present disclosure, the heat pump system operates in various modes including cooling mode, heating mode for normal climate conditions, and cascaded heating mode for cold climate conditions. The efficiency of the heat pump system in all modes of operation is improved by the liquid receiver which provides a consistent pressure to the heat exchangers.

Referring to FIG. **1**, in some implementations, an example system **100** includes a plurality of compressors (e.g., first compressor **110** and second compressor **120**) that are configured to work with a plurality of heat exchangers (e.g., first heat exchanger **130**, second heat exchanger **140**, and third heat exchanger **150**) and a plurality of expansion devices (e.g., first expansion device **170**, second expansion device **172**, and third expansion device **174**).

In some implementations, the first compressor **110** is a high-pressure compressor which generates high-pressure refrigerant, and the second compressor **120** is a low-pressure compressor which generates medium-pressure refrigerant. As used herein a high-pressure refrigerant means a pressure between 300-500 psi, a medium-pressure refrigerant means a pressure between 200-300 psi, and a low-pressure refrigerant means a pressure less than 200 psi. It is contemplated herein that multiple refrigerants may be utilized within the same system.

As the compressor **110**, **120** compresses the one refrigerant or respective refrigerants, the temperature of the compressed refrigerant (i.e., high-pressure refrigerant) increases. As the refrigerant decompresses or expands at the expansion device **170**, **172**, **174** (e.g., expansion valves), the refrigerant becomes cooler. Expansion devices **170**, **172**, **174** for heat pumps are known, and any expansion device currently known or used may be adapted for use herein, illustratively including an expansion valve commonly referred to as a thermal expansion valve.

As shown, various components are implemented to direct the circulation of the refrigerant in the system **100** based on the operation mode. For example, a first check valve **160**, a second check valve **162**, and a third check valve **164** are to allow the refrigerant to bypass the first expansion device **170**, the second expansion device **172**, and the third expansion device **174**, respectively. In some implementations, as shown, the system **100** includes a first reversing valve **180** and a second reversing valve **182** that are configured to direct the circulation of the refrigerant based on the operation mode. As shown, in some implementations, the system **100** utilizes a plurality of solenoid valves (e.g., first solenoid valve **190**, second solenoid valve **192**, and third solenoid valve **194**) to direct the circulation of the refrigerant based on the operation mode. In some implementations, the plurality of solenoid valves are bi-directional solenoid valves.

As discussed, the system **100** is configured to operate in various modes: cooling mode; heating mode for normal climate conditions; and cascaded heating mode for cold climate conditions.

Cascaded Heating Mode

Referring to FIG. **1**, when the system **100** operates in the cascaded heating mode for cold climate conditions (e.g., −30° C. outdoor ambient air), the first compressor **110** (e.g., high-pressure compressor) and the second compressor **120** (e.g., low-pressure compressor) are energized (i.e., in operation). As will be described later in the present disclosure, a first stream of the refrigerant from the first compressor **110** and a second stream of the refrigerant from the second compressor **120** cycle within the system **100** separately until the two streams are intersected (i.e., mixed) at the third heat exchanger **150**. The first stream of the refrigerant provides medium-pressure cold liquid/gas refrigerant and the second stream of the refrigerant provides medium-pressure hot gas refrigerant. The third heat exchanger **150** is configured to output the mixture of the first stream of the refrigerant and the second stream of the refrigerant into medium-pressure saturated vapor and medium-pressure liquid refrigerant.

The medium-pressure saturated vapor (e.g., hot gas refrigerant) from the mixture (located at the third heat exchanger **150**) is separately output from the third heat exchanger **150** and flows to the first compressor **110**, and the medium-pressure saturated vapor supplied to the first compressor **110** will become high-pressure hot gas refrigerant (hotter than the saturated vapor from the third heat exchanger **150**) at the first compressor **110**. Then, the high-pressure hot gas refrigerant flows to the first heat exchanger **130** located indoors and releases its thermal energy to the ambient indoor air—heating the indoor air. The medium-pressure liquid refrigerant from the mixture (located at the third heat exchanger **150**) is separately output from the third heat exchanger **150** and flows to the second expansion device **172**. At the second expansion device **172**, the refrigerant becomes colder (or significantly colder) than the temperature of the outdoor ambient air and flows to the second heat exchanger **140** where the cold refrigerant absorbs the thermal energy from the outdoor ambient air. As shown in FIG. **1**, the refrigerant which captured the outdoor thermal energy flows to the second compressor **120** where the medium-pressure hot gas refrigerant is generated using the refrigerant.

As described above, the system **100** is configured to utilize the medium-pressure saturated vapor and the medium medium-pressure liquid refrigerant from the third heat exchanger **150** to capture more thermal energy from outdoor ambient air in cold climate and to provide more thermal energy to heat the indoor air. In some implementations, the same type of the refrigerant is used for the first compressor **110** and the second compressor **120** to reduce the overall cost of ownership (including costs related to manufacturing, operating, maintaining, and installing the system).

As stated above, the first compressor **110** (e.g., high-pressure compressor) and the second compressor **120** (e.g., low-pressure compressor) are energized (i.e., in operation) when the system **100** is in the cascaded heating mode.

In the cascaded heating mode, the third solenoid valve **194**, in fluid communication with the second compressor **120**, is energized (i.e., open) to utilize the second compressor **120**. In the cascaded heating mode, the first reversing valve **180** is also energized to provide a conduit (i.e., fluid communication channel) between a first port **112** of the first compressor **110** and a second port **134** of the first heat exchanger **130**. By the conduit formed between the first port **112** of the first compressor **110** and the second port **134** of the first heat exchanger **130**, high-pressure high temperature gas refrigerant flows from the first compressor **110** to the first heat exchanger **130** located indoors. As the indoor air, which is relatively colder than the refrigerant from the first compressor **110**, passes over the first heat exchanger **130**, the energy (e.g., thermal energy) from the refrigerant is transferred to the indoor air. Thus, the indoor air is warmed. As a result, after passing the first heat exchanger **130**, the refrigerant cools down into liquid state (i.e., high-pressure "slightly cooler" liquid refrigerant).

As shown, the first expansion device **170** is in parallel with the first check valve **160**, and the first expansion device **170** and the first check valve **160** are in fluid communication with a first port **132** of the first heat exchanger **130**. When the system **100** operates in the cascaded heating mode, the first check valve **160** is energized (i.e. open or in bypass mode) so that the high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** bypasses the first expansion device **170**. As a result, the pressure of the high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** is maintained.

As shown, the first expansion device **170** is fluidly coupled with the first check valve **160**. The first solenoid valve **190** and the second solenoid valve **192** control the flow of refrigerant to and from the first expansion device **170** and the first check valve **160**. When the system **100** operates in the cascaded heating mode, the first solenoid valve **190** is energized (i.e., open) and the second solenoid valve **192** is not energized (i.e., closed). The high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** flows to the first solenoid valve **190**.

As shown, the first solenoid valve **190** is also in fluid communication with the third expansion device **174** and the third check valve **164**. When the system **100** operates in the cascaded heating mode, the third check valve **164** is not energized (i.e., closed or non-bypass mode). Since the third check valve **164** is not energized, the high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** flows into the third expansion device **174** and expands and the pressure of the refrigerant drops. As a result, the high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** expands at the third expansion device **174** and becomes medium-pressure cold liquid/gas refrigerant.

As shown, the third expansion device **174** is in fluid communication with a first port **152** of the third heat exchanger **150**. As a result, the medium-pressure cold liquid/gas refrigerant from the third expansion device **174** (i.e., the first stream of the refrigerant from the first compressor **110**) flows to the third heat exchanger **150**.

When the system **100** operates in the cascaded heating mode, the third solenoid valve **194** is energized (i.e., open) so that the first port **124** of the second compressor **120** and the first port **152** of the third heat exchanger **150** are in fluid communication. As a result, medium-pressure hot gas refrigerant from the second compressor **120** (i.e., the second stream of the refrigerant from the second compressor **120**) flows to the third heat exchanger **150** and is mixed with the medium-pressure cold liquid/gas refrigerant from the third expansion device **174** at the third heat exchanger **150**.

In some implementations, the third heat exchanger **150** includes a flash-tank-heat-exchanger. As shown, a second port **154** of the third heat exchanger **150** is in fluid communication with the second reversing valve **182**. When the system **100** operates in the cascaded heating mode, the second reversing valve **182** is energized to provide a conduit (i.e., fluid communication channel) between the second port **154** of the third heat exchanger **150** and a second port **114** of the first compressor **110**. As a result, medium-pressure saturated gas refrigerant of the mixture in the third heat exchanger **150** flows to the first compressor **110**.

As shown, a third port **156** of the third heat exchanger **150** is in fluid communication with the second expansion device **172**. The second expansion device **172** is in parallel with the second check valve **162**. When the system **100** operates in the cascaded heating mode, the second check valve **162** is not energized (i.e., closed or non-bypass mode). As a result, medium-pressure liquid refrigerant flows from the third port **156** of the third heat exchanger **150** to the second expansion device **172** and expands at the second expansion device **172**. As a result, the temperature of the medium-pressure liquid refrigerant and the pressure of the medium-pressure liquid refrigerant drop.

As shown, the second expansion device **172** is in fluid communication with a first port **142** of the second heat exchanger **140**. As a result, the low-pressure cooler liquid refrigerant output from the second expansion device **172** flows to a first port **142** of the second heat exchanger **140** located outdoors. Since the low-pressure cooler liquid refrigerant from the second expansion device **172** is colder (or significant colder) than outdoor ambient air temperature, the low-pressure cooler liquid refrigerant absorbs the thermal energy from the outdoor ambient air and becomes low-pressure hot gas refrigerant at the second heat exchanger **140**. The low-pressure hot gas refrigerant flows through a second port **144** to the first reversing valve **180** and is directed by the first reversing valve **180** (in an energized state) and the second reversing valve **182** (in an energized state) to a second port **122** of the second compressor **120**. The refrigerant circulates in accordance with the loops described above when the system **100** is in the cascade heating mode.

As discussed, the system **100** is includes a liquid receiver (e.g., the third heat exchanger **150**). The third heat exchanger **150** includes one input port (first port **152**) and two output ports (second port **154** and third port **156** as shown) in the cascade heating mode for the cold climate applications. As discussed above, both the medium-pressure hot refrigerant from the second compressor **120** and the medium-pressure cold liquid/gas refrigerant from the third expansion device **174** flow into the third heat exchanger **150** via the first port **152** of the third heat exchanger **150**. As discussed above, at the third heat exchanger **150**, both of the refrigerants are mixed. After mixing, a first portion of the mixture (i.e., medium-pressure saturated gas refrigerant) is cycled back to the first compressor **110** and a second portion of the mixture (i.e., medium-pressure liquid refrigerant) is cycled back to the second compressor **120** as the low-pressure hot gas refrigerant.

In the present disclosure, with the third heat exchanger **150** which is capable of supporting two different refrigerant cycles (i.e., liquid refrigerant cycle and gas refrigerant cycle) using the one input port (i.e., first port **152**) and two output ports (i.e., second port **154** and third port **156**), a consistent rate of refrigerant is provided to the first compressor **110** and the second compressor **120**—resulting an efficient cascade heat pump system for cold climate applications.

Heating Mode

Referring to FIG. **1**, when the system **100** operates in the heating mode for normal climate conditions (e.g., temperatures at or above 8° C.), the first compressor **110** is energized (in operation) and the second compressor **120** is not energized (not in operation). In the heating mode, the third solenoid valve **194** is not energized (i.e., closed) as well. In the heating mode, the first reversing valve **180** is energized to provide a conduit (i.e., fluid communication channel) between the first port **112** of the first compressor **110** and the second port **134** of the first heat exchanger **130**. By the conduit formed between the first port **112** of the first compressor **110** and the second port **134** of the first heat exchanger **130**, high-pressure hot gas refrigerant flows from the first compressor **110** to the first heat exchanger **130** located indoors. As the indoor air, which is relatively colder than the refrigerant from the first compressor **110**, passes over the first heat exchanger **130**, the energy (e.g., thermal energy) from the refrigerant is transferred to the indoor air. Thus, the indoor air becomes warm. As a result, after passing the first heat exchanger **130**, the refrigerant cools down into liquid state (i.e., high-pressure "slightly cooler" liquid refrigerant).

As shown, the first expansion device **170** is in parallel with the first check valve **160**. The first expansion device **170** and the first check valve **160** are in fluid communication with a first port **132** of the first heat exchanger **130**. When the system **100** operates in the heating mode, the first check valve **160** is energized (i.e., open or bypass mode) so that the high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** bypasses the first expansion device **170**. As a result, the pressure of the high-pressure slightly cooler liquid refrigerant from the first heat exchanger **130** is maintained.

As shown, the first expansion device **170** and the first check valve **160** is in fluid communication with the first solenoid valve **190** and the second solenoid valve **192**. When the system **100** operates in the heating mode, the first solenoid valve **190** is not energized (i.e., closed) and the second solenoid valve **192** is energized (i.e., open). The high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** flows to the first solenoid valve **190** and the second solenoid valve **192**.

As shown, the first solenoid valve **190** is in fluid communication with the third expansion device **174** that is connected in parallel with the third check valve **164**. When the system **100** operates in the heating mode, the third check valve **164** is energized (i.e., open or bypass mode). Since the third check valve **164** is energized, the high-pressure "slightly cooler" liquid refrigerant bypasses the third expansion device **174**. As a result, the pressure of the high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** is maintained.

As shown, the third expansion device **174** is in fluid communication with the first port **152** of the third heat exchanger **150**. As a result, the high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** flows to the third heat exchanger **150** via the first solenoid valve **190**.

As shown, the second solenoid valve **192** in fluid communication with the first port **152** of the third heat exchanger **150**. As a result, the high-pressure "slightly cooler" liquid refrigerant from the first heat exchanger **130** also flows to the third heat exchanger **150** via the second solenoid valve **192**.

As shown, the third port **156** of the third heat exchanger **150** is in fluid communication with the second expansion device **172** and the second check valve **162**. When the system **100** operates in the heating mode, the second check valve **162** is not energized (i.e., closed or non-bypass mode). As a result, high-pressure liquid refrigerant flows from the third port **156** of the third heat exchanger **150** to the second expansion device **172** and expands at the second expansion device **172**. Accordingly, the pressure of the refrigerant drops.

As shown, the second expansion device **172** is in fluid communication with the first port **142** of the second heat exchanger **140** located outdoors. As a result, the low-pressure cooler liquid refrigerant output from the second expansion device **172** flows to a first port **142** of the second heat exchanger **140** located outdoors. Since the low-pressure cooler liquid refrigerant from the second expansion device **172** is colder than outdoor ambient air temperature, the low-pressure cooler liquid refrigerant absorbs the thermal energy from the outdoor ambient air and becomes low-pressure hot gas refrigerant at the second heat exchanger **140**. When the system **100** operates in the heating mode, the first reversing valve **180** is energized, and the second reversing valve **182** is not energized. Accordingly, the low-pressure hot gas refrigerant from the second heat exchanger **140** is directed by the first reversing valve **180** and the second reversing valve **182** to a second port **122** of the first compressor **110**. The refrigerant circulates in accordance with the loop described above when the system **100** is in the heating mode.

Cooling Mode

Referring to FIG. **1**, when the system **100** operates in the cooling mode for normal climate conditions, the first compressor is energized (i.e. in operation), but the second compressor is not energized (i.e., not in operation). In the cooling mode, the third solenoid valve **194** is not energized (i.e., closed) as well. In the cooling mode, the first reversing valve **180** is not energized to provide a conduit (i.e., fluid communication channel) between the first port **112** of the first compressor **110** and the second port **143** of the second heat exchanger **140** located outdoors. By the conduit formed between the first port **112** of the first compressor **110** and the second port **143** of the second heat exchanger **140**, high-pressure hot gas refrigerant flows from the first compressor **110** to the second heat exchanger **140** located outdoors. As the outdoor air, which is relatively colder than the refrigerant from the first compressor **110**, passes over the second heat exchanger **140**, the energy (e.g., thermal energy) from the refrigerant is transferred to the outdoor air. As a result, after passing the second heat exchanger **140**, the refrigerant cools down into liquid state (i.e., high-pressure "slightly cooler" liquid refrigerant).

As shown, the second expansion device **172** is in parallel with the second check valve **162**. The second expansion device **172** and the second check valve **162** are in fluid communication with a first port **142** of the second heat exchanger **140**. When the system **100** operates in the cooling mode, the second check valve **162** is energized (i.e., open or in bypass mode) so that the high-pressure slightly cooler liquid refrigerant from the second heat exchanger **140** bypasses the second expansion device **172**. As a result, the pressure of the high-pressure "slightly cooler" liquid refrigerant from the second heat exchanger **140** is maintained.

As shown, the second expansion device **172** is in parallel with the second check valve **162** and is in fluid communication with the third port **156** of the third heat exchanger **150**. When the system **100** operates in the cooling mode, the first solenoid valve **190** and the third solenoid valve **194** are not energized (i.e., close) and the second solenoid valve **192** is energized (i.e., open).

As shown, the first port **152** of the third heat exchanger **150** is in fluid communication with the second solenoid valve **192** and the second solenoid valve **192** is in fluid communication with the first expansion device **170** and the first check valve **160**. As a result, the high-pressure "slightly cooler" liquid refrigerant from the second heat exchanger **140** flows to the first expansion device **170** and the first check valve **160**. When the system **100** operates in the cooling mode, the first check valve **160** is not energized (i.e., closed or non-bypass mode). The high-pressure "slightly cooler" liquid refrigerant from the second heat exchanger **140** flows to the first expansion device **170** and expands at the first expansion device **170**. The expanded refrigerant becomes low-pressure cold liquid refrigerant.

As shown, the first expansion device **170** is in fluid communication with the first port **132** of the first heat exchanger **130** located indoor. The low-pressure cold liquid refrigerant will absorb heat from indoor air and become low-pressure hot gas refrigerant. When the system **100** operates in the cooling mode, the first reversing valve **180** and the second reversing valve **182** are not energized to provide a conduit between the second port **134** of the first heat exchanger **130** and the second port **114** of the first compressor **110**. The low-pressure hot gas refrigerant flows back to the first compressor **110** by the conduit. The refrigerant circulates in accordance with the loop described above when the system **100** is in the cooling mode.

Switching Between Modes

As discussed, the system **100** is configured to operate in various modes: cooling mode; heating mode for normal climate conditions; and cascaded heating mode for cold climate conditions. As shown above, the operation mode of the system **100** is capable of seamlessly switching from one of the three modes to another by simply applying corresponding settings for respective components. Table 1 below shows combinations of component settings based on the operation mode.

As shown, the first compressor **110** (Comp1), the second compressor **120** (Comp2), the first reversing valve **180** (RV1), the second reversing valve **182** (RV2), the first solenoid valve **190** (SV1), and the third solenoid valve **194** (SV3) are energized when the system **100** is in the cascade heating mode (Cascade HP). While other components are energized, the second solenoid valve **192** (SV2) is not energized when the system **100** is in the cascade heating mode (Cascade HP).

As shown, the first compressor **110** (Comp1), the first reversing valve **180** (RV1), the first solenoid valve **190** (SV1), and the second solenoid valve **192** (SV2) are energized when the system **100** is in the heating mode for normal climate conditions (Single Stage HP). While other components are energized, the second compressor **120** (Comp2), the second reversing valve **182** (RV2), and the third solenoid valve **194** (SV3) are not energized when the system **100** is in the heating mode for normal climate conditions.

As shown, the first compressor **110** (Comp1), the first reversing valve **180** (RV1), and the second solenoid valve **192** (SV2) are energized when the system **100** is in the cooling mode (Single Stage AC). While other components are energized, the second compressor **120** (Comp2), the second reversing valve **182** (RV2), and the first solenoid valve **190** (SV1), and the third solenoid valve **194** (SV3) are not energized when the system **100** is in the cooling mode.

TABLE 1

Component settings based on the operation mode.

	Component						
Mode	Comp1	Comp2	RV1	RV2	SV1	SV2	SV3
Cascade HP	ON	ON	ON	ON	ON	OFF	ON
Single Stage HP	ON	OFF	ON	OFF	OFF	ON	OFF
Single Stage AC	ON	OFF	ON	OFF	OFF	ON	OFF

Exemplary Implementation with Multi-Level Converter

FIG. **2** is a schematic diagram of a heat pump system integrated with a modular multi-level converter.

Referring to FIG. **2**, in some implementations, an example system **200**, for off-grid applications, includes a fuel cell unit **210**, a battery management system **220**, a stationary battery bank **230**, a DC-AC modular multi-level converter **240**, and a heat pump unit **250**. The fuel cell unit **210** includes various components (including a hydrogen storage tank, a fuel cell, and condenser) to provide electrical power (e.g., direct current) derived from the fuel cell via its output. The battery management system **220** coupled to the output is configured to control the charging and discharging of the stationary battery bank **230**, so the stationary battery bank **230** (e.g., backup battery) is in the optimal condition.

As shown, the modular multi-level converter **240** is coupled to the fuel cell unit **210** and the stationary battery bank **230** (via the battery management system **220**). The modular multi-level converter **240** (e.g., nine-level modular multi-level converter) is configured to regulate power from the fuel cell unit **210** to power the heat pump unit **250**, such as stepping up power, preventing power peaks, or stepping down power. In one aspect, where the heat pump unit **250** operates off of AC electrical power, the modular multi-level converter **240** is configured to generate AC electrical power (e.g., three-phase AC electrical power) by converting DC electrical power from the fuel cell unit **210** and/or the stationary battery bank **230**. In some implementations, the modular multi-level converter **240** includes Gallium Nitride (GaN) controlling components that increases efficiency of the DC-AC conversion. The AC electrical power generated by the modular multi-level converter **240** is provided for operating the heat pump unit **250**. In some implementations, the heat pump unit **250** includes the system **100**.

Testing and Evaluation

FIG. **3** is a comparison graph **300** that shows the results obtained from a prototype of the heat pump system **100** under different conditions (e.g., at various ambient air temperatures). The graph **300** also compares the efficiency of the heat pump system **100** with an existing heat pump system and other various traditional space heating systems, such as oil furnace (diesel) and propane furnace in terms of the coefficient of performance.

As discussed, the heat pump system **100** includes the liquid receiver (e.g., third heat exchanger **150** in FIG. **1**) to overcome the inefficiency associated with the existing heat pump systems. In particular, the liquid receiver is configured to provide a consistent flow of refrigerant to one or more compressors (e.g., the first compressor **110** and the second compressor **120** in FIG. **1**) in the heat pump system **100**. The consistent flow also supplies an outdoor heat exchanger (e.g., second heat exchanger **140** in FIG. **1**) consistent flow of the refrigerant which is colder than ambient air temperature. Thus, the heating capability and the COP of the heat pump system **100** are maintained under cold climate conditions.

The COP is calculated using Equations (1) and (2).

$$COP = \frac{Q_H}{W_c} \tag{1}$$

where Q_H is the heat leaving a condenser (into the house) and W_c is the total energy required to power the system (e.g., for the compressors and sensors). The COP is a measure of how much heat will be generated out of the pump for every unit of electrical energy needed to power the compressors (and other equipment). The higher the COP, the more heat is generated in the system for every unit of electricity put into the system.

$$m = \rho Q \tag{2}$$

where m is the mass flow, ρ is the density at the given measured temperature and pressure, and Q is the measured volumetric flow rate. The volumetric flow rate of refrigerant is likely the easiest to measure (using a flow meter), and can be converted to a mass flow rate using Equation 1. The mass flow rate throughout the system is constant, as no mass should be created or destroyed anywhere as the system is closed.

As shown in FIG. **3**, it is clear from the results that the prototype based on the heat pump system **100** has a COP of 3.0 at −25° C. which is significantly higher than COPs of other heating systems including existing heat pump systems at −25° C. This indicates that the heat pump system **100** including the liquid receiver actually works under cold climate conditions and is capable of providing cost efficient heating for cold climate conditions. Also, as shown, with further optimization of the prototype, the COP of the heat pump system **100** can be improved to the point where the actual results from the heat pump system **100** are close to simulation results of the heat pump system **100** shown in FIG. **3** (e.g., a COP of 4.7 at −25° C.).

A number of implementations have been described. Nevertheless, it will be understood that various modifications may be made without departing from the spirit and scope of the disclosure. Accordingly, other implementations are within the scope of the following claims.

What is claimed is:

1. A heat pump system comprising:

a plurality of compressors including a first compressor and a second compressor, each compressor having a first port and a second port;

a plurality of heat exchangers including a first heat exchanger in communication with the first port of the first compressor, a second heat exchanger, and a third heat exchanger,

wherein the third heat exchanger comprises an input port and a plurality of output ports, such that:

the first port of the second compressor is in fluid communication with the input port of the third heat exchanger,

the second port of the first compressor is in fluid communication with a first of the plurality of output ports of the third heat exchanger;

a plurality of expansion devices including a first expansion device in communication with the first heat exchanger, a second expansion device in communication with the second heat exchanger and with a second of the plurality of output ports of the third heat exchanger, and a third expansion device in communication with the input port of the third heat exchanger,

wherein, when the system operates in a cascade heating mode, the third heat exchanger is configured to:

receive a first portion of refrigerant from the first compressor and a second portion of refrigerant from the second compressor via the input port of the third heat exchanger,

provide a first consistent flow of refrigerant from the second of the plurality of output ports to the first compressor, and

provide a second consistent flow of refrigerant from the first of the plurality of output ports to the second compressor.

2. The heat pump system of claim **1**, wherein the third heat exchanger includes a flash-tank-heat-exchanger.

3. The heat pump system of claim **1**, wherein the third heat exchanger is configured to contain a mixture of liquid refrigerant and vapor refrigerant.

4. The heat pump system of claim **1**, wherein the third heat exchanger is configured to output a vapor portion of mixture of the first portion and second portion of the refrigerant via the first of the plurality of outputs and a liquid portion of mixture of the first portion and second portion of the refrigerant via the second of the plurality of outputs.

5. The heat pump system of claim **1**, wherein the second heat exchanger is in fluid communication with an input port of the second compressor when the system operates in the cascade heating mode.

6. The heat pump system of claim **1**, wherein the second heat exchanger is located outdoors.

7. The heat pump system of claim **1**, wherein the first compressor includes a high-pressure compressor.

8. The heat pump system of claim **1**, wherein the second compressor includes a low-pressure compressor.

9. The heat pump system of claim **1**, wherein the first compressor and the second compressor are in operation when the system operates in the cascade heating mode.

10. The heat pump system of claim **1**, wherein when system operates in a heating or a cooling mode, the first compressor is in operation and the second compressor is not in operation.

11. The heat pump system of claim **1**, wherein before received by the third exchanger, the first portion of refrigerant passes through the first heat exchanger and the third expansion device.

12. The heat pump system of claim **11**, wherein the first heat exchanger is located indoors.

13. A heating and cooling system comprising:

a fuel cell unit;

a backup battery unit including a battery management circuitry;

a direct current (DC) to alternate current (AC) converter; and

the heat pump system of claim **1**,

wherein the direct current (DC) to alternate current (AC) converter is configured to:

receive direct current from the fuel cell or the backup battery unit;

convert the direct current to alternate current; and

supply the alternate current to the heat pump system of claim **1**.

14. The system of claim **13**, wherein the DC to AC converter includes a nine-level modular multi-level converter.

15. The system of claim **13**, wherein fuel cell unit includes a hydrogen storage tank, a fuel cell, and a condenser.

16. The system of claim **13**, wherein the battery management circuitry is configured to maintain the backup battery.

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