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Osborne et al.

(54) **CONDENSER STAGING AND CIRCUITING FOR A MICRO COMBINED HEAT AND POWER SYSTEM**

(76) Inventors: **Rodney L. Osborne**, Hebron, OH (US); **William T. Hanna**, Gahanna, OH (US); **Donald Anson**, Worthington, OH (US)

Correspondence Address:
Killworth, Gottman, Hagan & Schaeff, L.L.P.
Suite 500
One Dayton Centre
Dayton, OH 45402-2023 (US)

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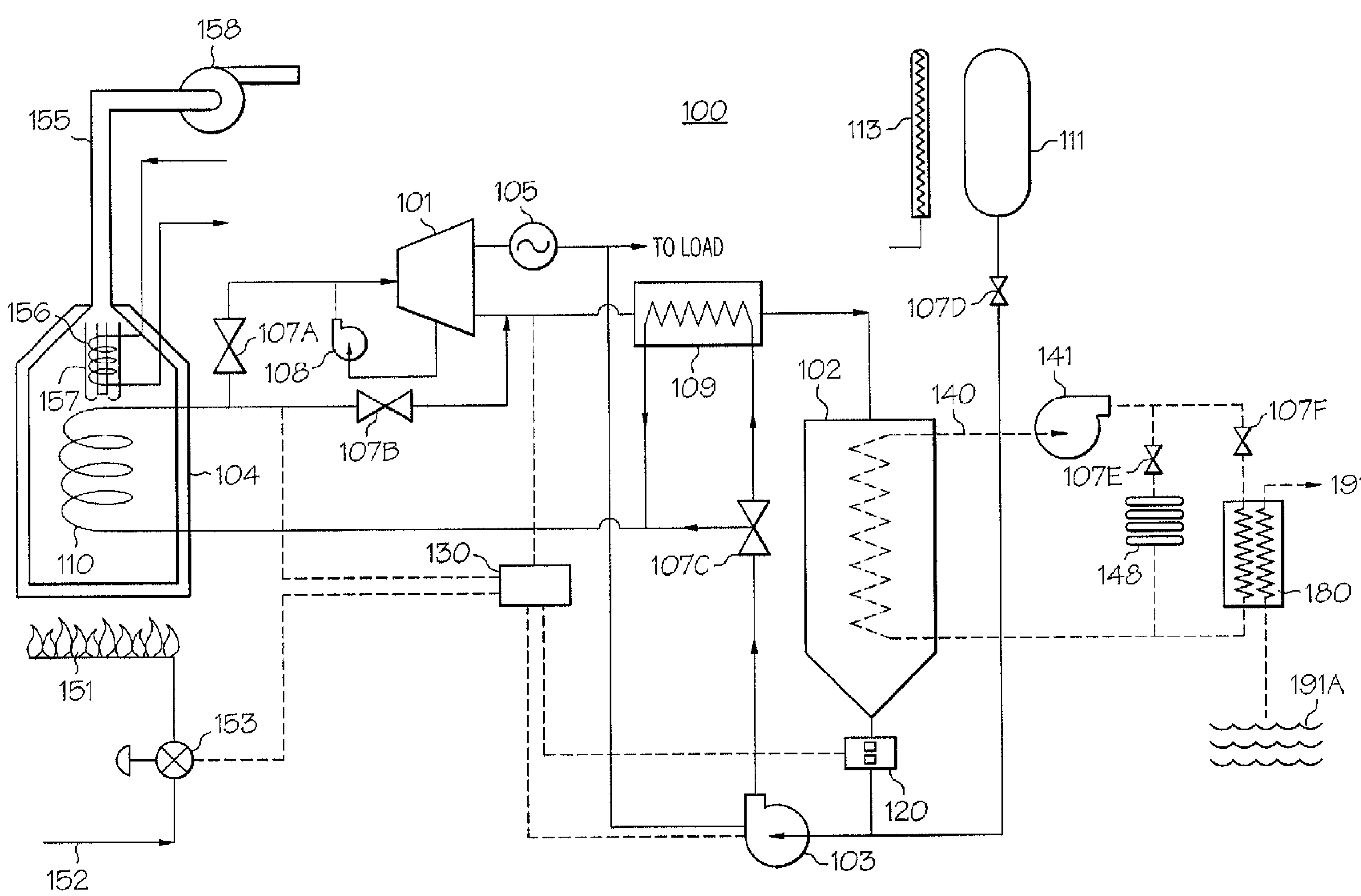
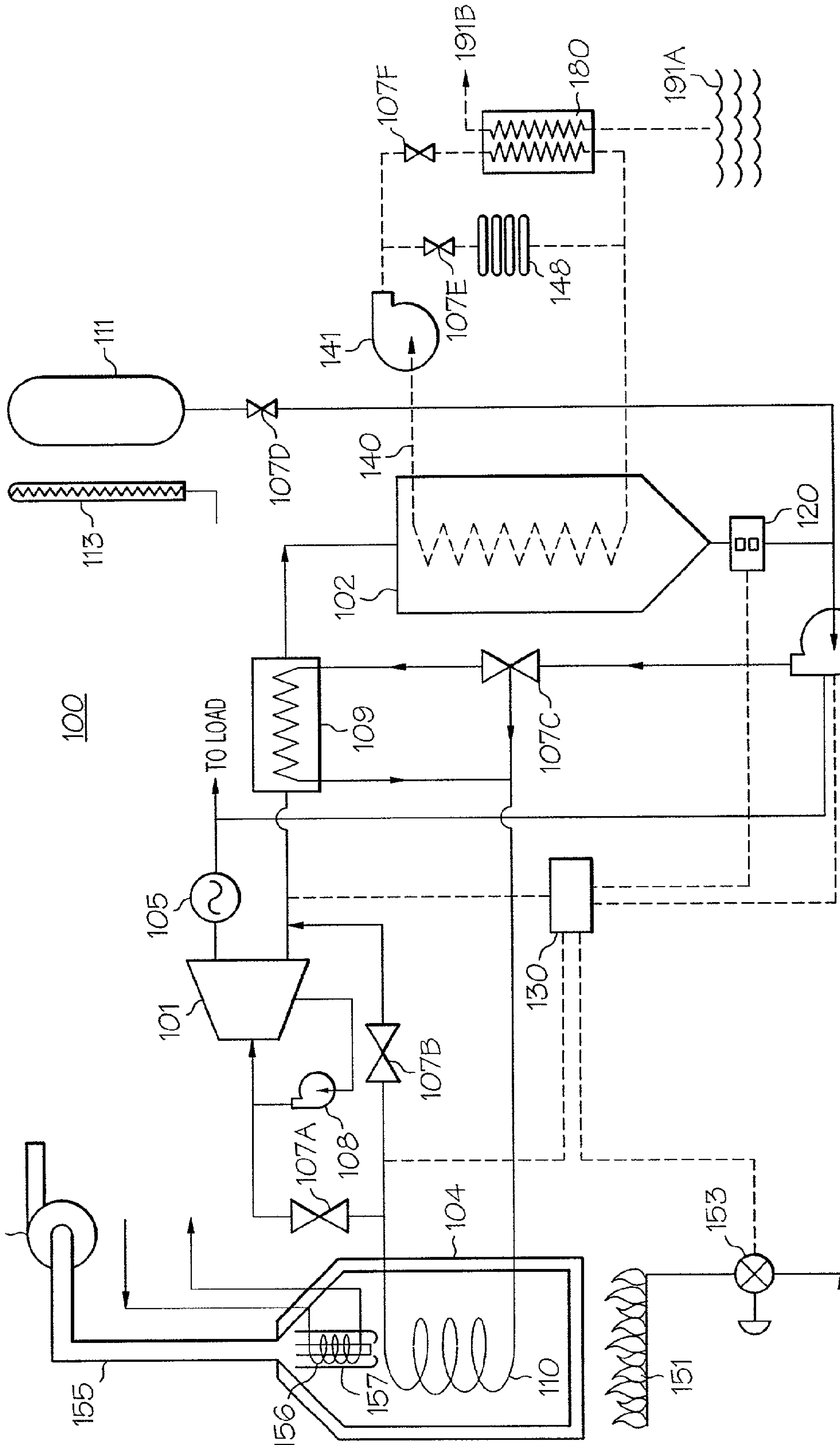
(57) **ABSTRACT**

A micro combined heat and power system includes at least a heat source, an expander, a pump, a staged, counterflow condenser and a conduit for transporting a working fluid. The heat source can be, for example, a burner, while the expander is preferably a scroll expander. The heat source superheats the working fluid, which is preferably an organic working fluid. The superheated organic working fluid passes through the expander, which is coupled to a generator to produce electricity. After the working fluid is expanded, its gives up at least a portion of its excess heat first to the condenser. By placing the condenser in a counterflow arrangement, the fluid receiving the heat from the condensing working fluid can be of a higher temperature, thus allowing more system variation in the heat to power output ratio. The fluid receiving heat from the condenser may include circulating air, water or related fluid to provide space heat or domestic hot water.




100
101
102
103
104
105
107A
107B
107C
107D
107E
107F
108
109
110
111
113
120
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140
141
148
151
152
153
155
156
157
158
180
191A
191B
TO LOAD

FIG. 1

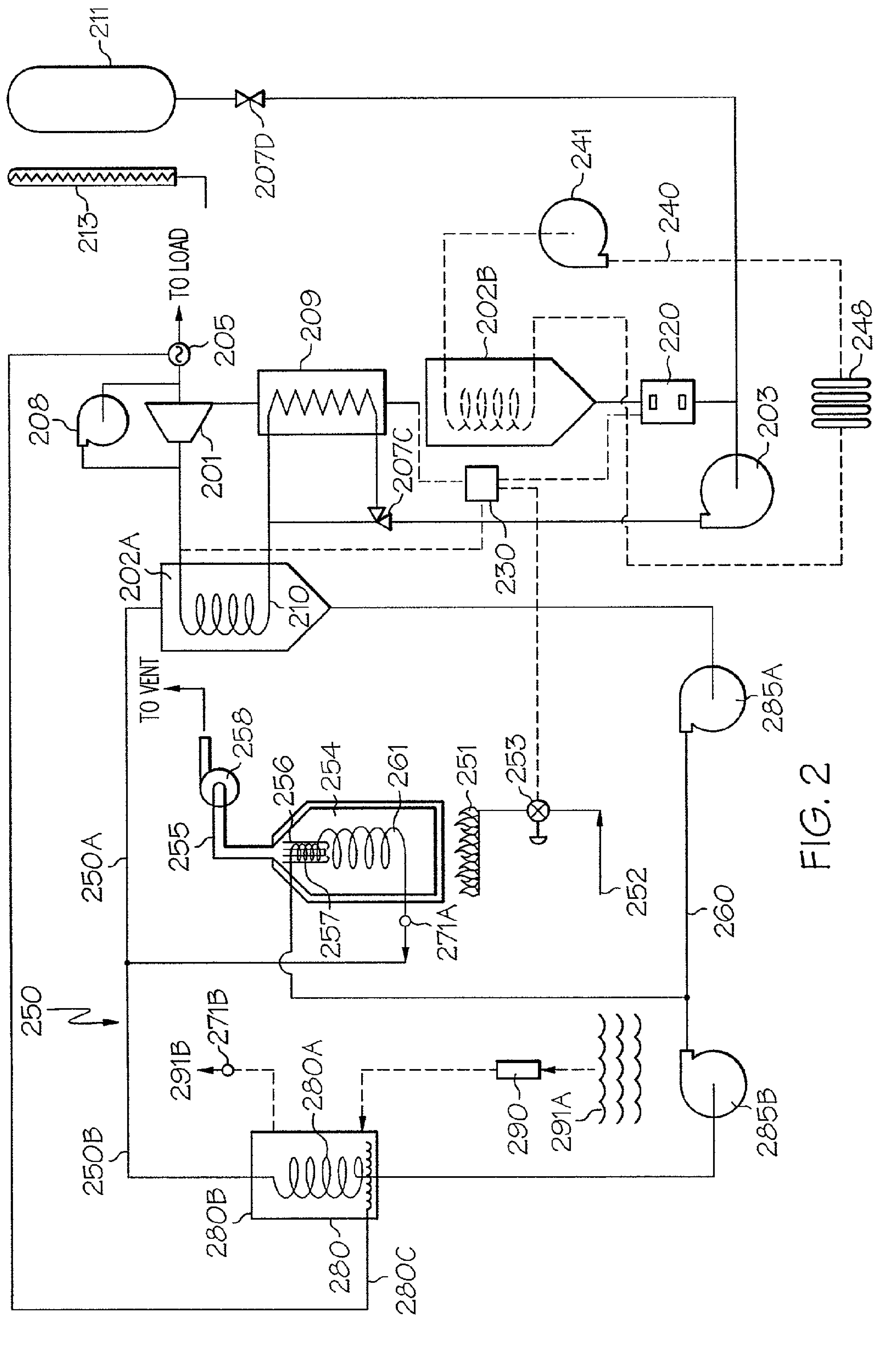
200
211
213
207D
241
240
TO LOAD
205
209
202B
220
248
208
201
207C
203
230
202A
210
TO VENT
258
256
254
261
251
253
255
257
271A
252
250A
285A
260
250
271B
291B
280A
290
291A
285B
250B
280B
280
280C

FIG. 2

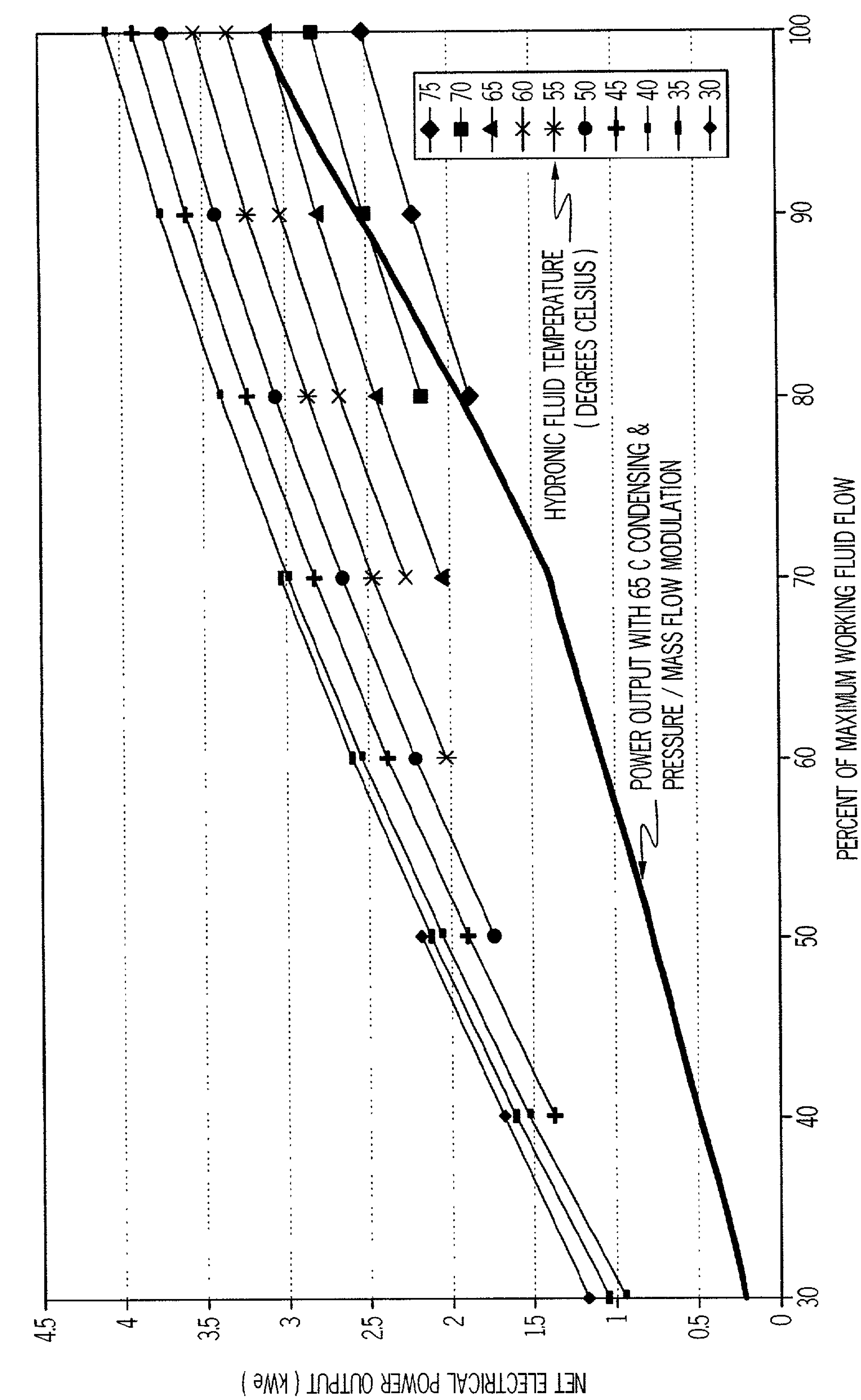
NET ELECTRICAL POWER OUTPUT (KWe)
PERCENT OF MAXIMUM WORKING FLUID FLOW
HYDRONIC FLUID TEMPERATURE (DEGREES CELSIUS)
POWER OUTPUT WITH 65 C CONDENSING & PRESSURE / MASS FLOW MODULATION
75
70
65
60
55
50
45
40
35
30
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
30
40
50
60
70
80
90
100

FIG. 3

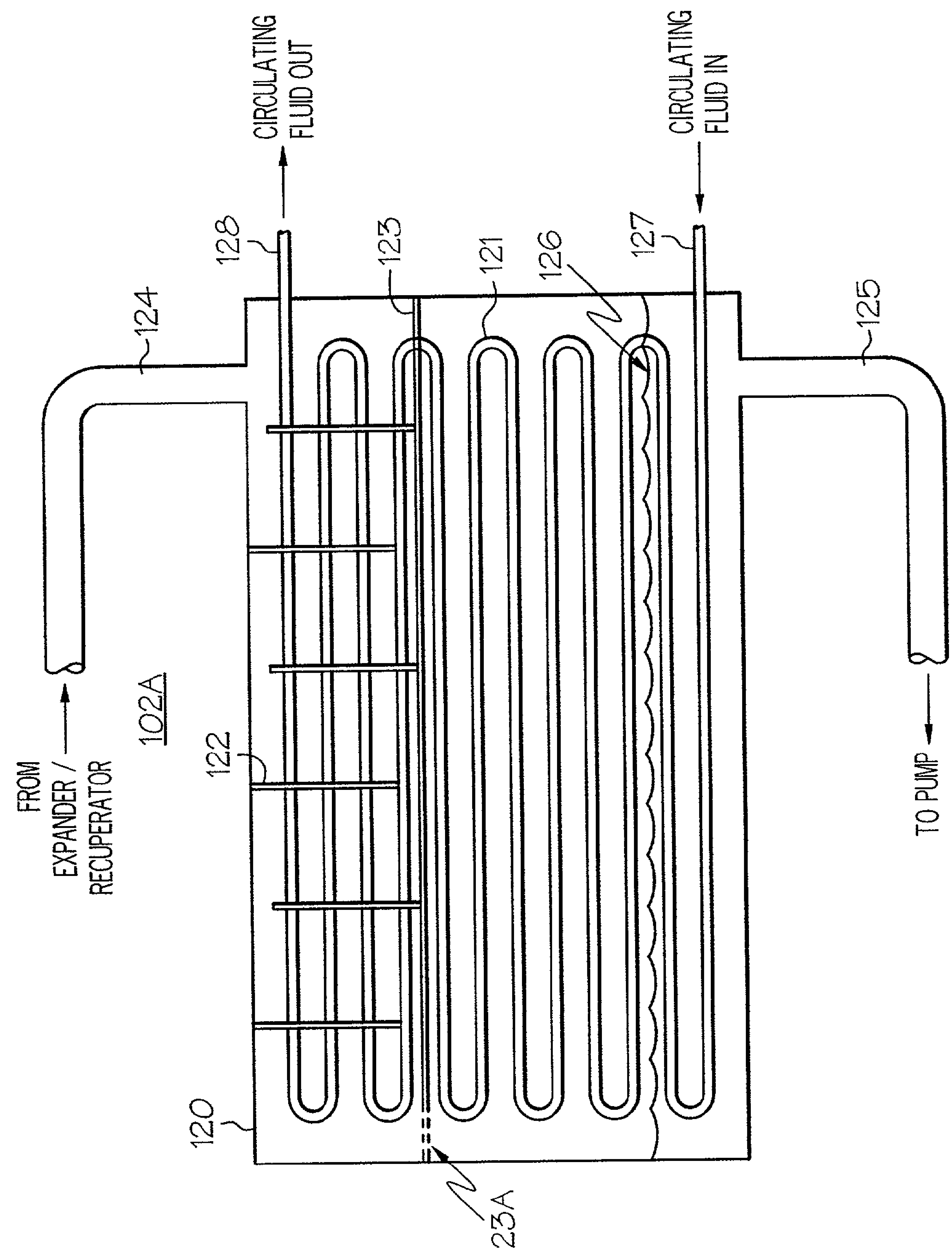
CIRCULATING FLUID OUT
CIRCULATING FLUID IN
128
123
121
126
127
124
125
FROM EXPANDER / RECUPERATOR
102A
122
TO PUMP
120
123A

FIG. 4A

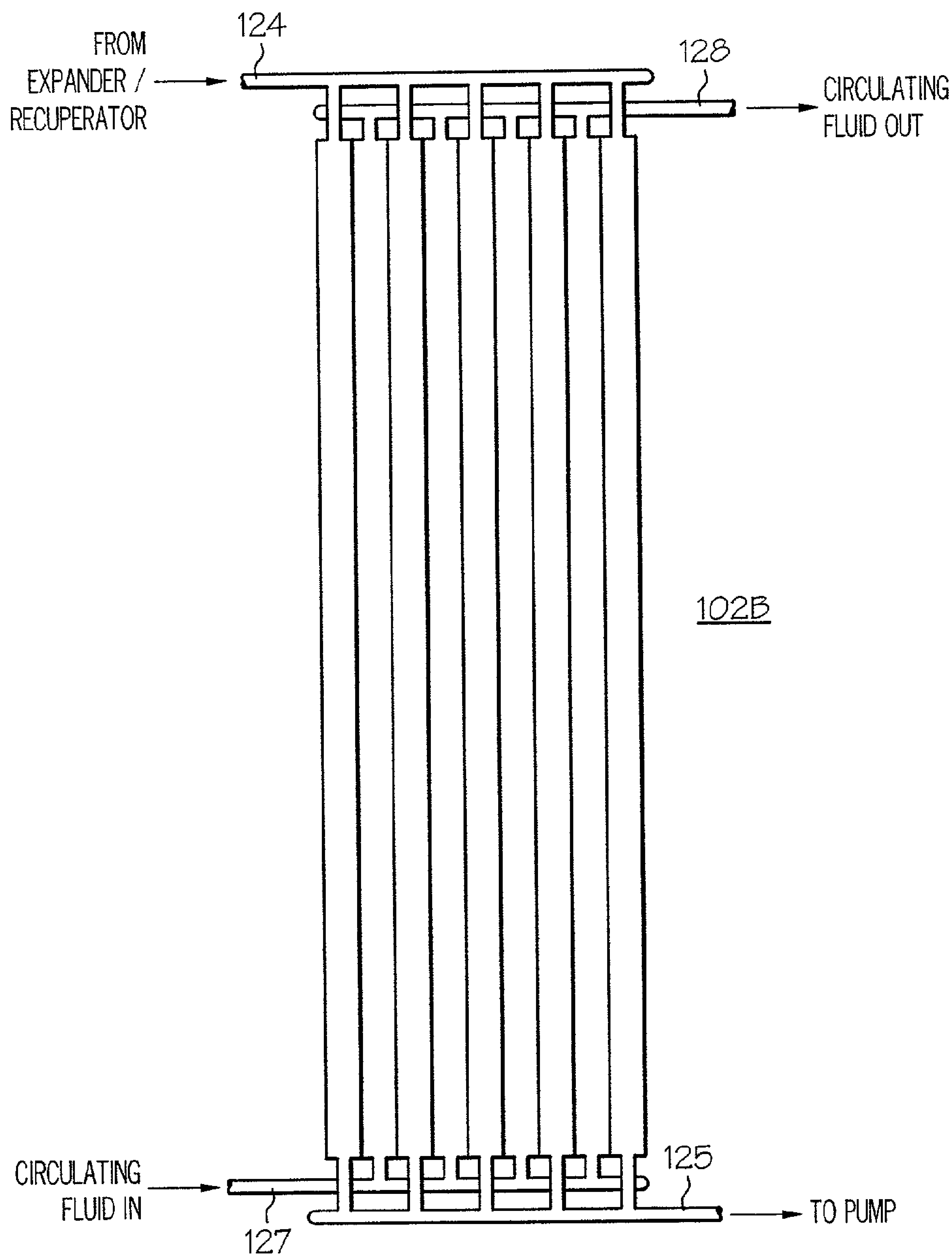
FROM
EXPANDER /
RECUPERATOR
124
128
CIRCULATING
FLUID OUT
102B
CIRCULATING
FLUID IN
127
125
TO PUMP

FIG. 4B

CONDENSER STAGING AND CIRCUITING FOR A MICRO COMBINED HEAT AND POWER SYSTEM

BACKGROUND OF THE INVENTION

[0001] The present invention generally relates to improvements in operability of a Rankine cycle cogeneration system using an organic working fluid, and more particularly to condenser configurations integrated into such a system to increase system operability via modulated cogeneration heat-to-power ratio.

[0002] The concept of cogeneration, or combined heat and power (CHP), has been known for some time as a way to improve overall efficiency in energy production systems. With a typical CHP system, heat (usually in the form of hot air or water) and electricity are the two forms of energy that are generated. In such a system, the heat produced from a combustion process can drive an electric generator, as well as heat up water, often turning it into steam for dwelling or process heat. Traditionally, CHP systems have been large, centrally-operated facilities under the control of the state or a large utility company, sized to provide energy for many thousands of users. If the region being served by the CHP has as part of its infrastructure adequate heat transporting capability, the centrally-generated heat and electric power model of the large CHP system can, within limits, function reasonably efficiently and reliably. In the absence of adequate heat transport capability, however, while the region's electric power needs would continue to be met by the central generating station, the heat needs would need to be fulfilled separately and remotely from the electricity production, often near or within the building housing the end-user. This latter configuration typically includes the presence of one or more boilers that could generate hot water or steam to provide most or all of the localized building heating requirements. While either configuration works well for its intended purpose, inefficiencies arise. In the former system, much of the heat generated at the central generating station is, after being transported over long distances, unavailable for remote use. In the latter system, the lack of CHP capability necessitates the consumption of additional energy at the remote location to satisfy heat requirements.

[0003] Recent trends in the deregulation of energy production and distribution have made viable the concept of distributed generation. With distributed generation, the large, central generating station is supplemented with, or replaced by numerous smaller autonomous or semi-autonomous units. These changes have led to the development of smaller CHP systems, called micro-CHP, which are distinguished from traditional CHP by the size of the system. By way of contrast, the electric output of a generating station-sized CHP could be in the tens, hundreds or thousands of megawatts (MW), where the electric output of a micro-CHP is fairly small, in the low kW_e or even sub-kW_e range. The inclusion of a distributed system into dwellings that already have fluid-carrying pipes for heat transport is especially promising, as little or no disturbance of the existing building structure to insert new piping is required. Similarly, a micro-CHP system's inherent multifunction capability can reduce structural redundancy. Accordingly, the market for localized heat generation capability in Europe and the United Kingdom (UK), as well as certain parts of the United States, dictates that a single unit for residential and small commercial sites provide heat for both space heat (SH), such as a hydronic system with radiator, and domestic hot water (DHW), such as a shower head or faucet in a sink or bathtub, via demand (instantaneous) or storage systems.

[0004] As with all energy production devices that rely on non-renewable sources, such as natural gas, coal or oil, a more efficient system consumes lower quantities of fuel to au generate the same energy output as its less efficient counterpart. A key factor in keeping micro-CHP system efficiency high over a wide range of operating conditions is how much thermal output is required at the heat source, such as a natural gas burner. Unfortunately, the nature of micro-CHP system operation, where both electric power and heat are generated from the same combustion process often under a fixed heat to power (Q/P) ratio, is such that when thermal output is reduced to minimize fuel consumption, the electric power production often drops even more quickly. As such, these systems cannot operate efficiently when climatic changes and user energy-consumption habits deviate significantly, over the course of a day or the year, from the rated Q/P. With a fixed Q/P heat-led system, because the electric power output follows heat production, a significant turndown in thermal load results in a concomitant loss in electric output, and because maximizing system efficiency is typically a corollary to maximizing electric output, such part power operation severely limits the benefits associated with cogeneration systems.

[0005] What is needed is a micro-CHP system that can operate at high efficiencies regardless of the Q/P requirements. The present inventors have recognized that with modulation, the system continues to operate over a longer period of time such that its duty cycle is relatively large and that the variable heat output need not encroach on maximizing electrical output. They have further recognized that by modulating the system, Q/P is improved at all conditions, especially at part power conditions. They have also recognized that a modulation approach that varies the mass flow of the working fluid to match the heat load while simultaneously varying the fuel flow to the heat source in order to keep the inlet temperature of the working fluid heated by the heat source at a constant temperature results in a thermal output that can be closely tailored to a user's needs while keeping electrical power output at a maximum. They have moreover recognized that the inclusion of various system componentry, such as staged condensers, can improve the efficiency of modulating a system by the aforementioned approach. They have additionally recognized that in countries where emission requirements are stringent, modulating can lower burner output at certain system operating conditions more effectively than by cycling the system such that fuel consumption to achieve the same amount of energy output is reduced.

BRIEF SUMMARY OF THE INVENTION

[0006] These needs are met by the present invention, where a micro-CHP system with staged condenser is described. According to a first aspect of the present invention, a cogeneration system includes a heat source, a working fluid circuit and at least one energy conversion circuit operatively responsive to the working fluid circuit such that upon operation of the cogeneration system, the energy conversion circuit provides useable energy. In the present context, the term "useable energy" includes that which a user can put to practical use, rather than waste or incidental energy. This is consistent with the concept of cogeneration, which is frequently considered to be the thermodynamically sequential production of two or more useful forms of energy from a single primary energy source. The most notable examples of useable energy arising out of the operation of a cogeneration system are electricity (preferably alternating current electricity, derived from the mechanical turning of a generator), and heat in the form of SH and DHW. The working fluid circuit is made up of at least conduit, an expander, condenser and pump. The conduit is configured to transport an organic working fluid, where at least a portion of the conduit is disposed adjacent the heat source such that during heat source operation, the heat transferred to the conduit is sufficient to superheat the organic working fluid disposed in that part of the conduit. The expander is in fluid communication with the conduit such that the organic working fluid remains superheated after expansion, while the condenser is in fluid communication with the expander to extract some of the excess heat still extant in the working fluid after the expansion process. The condenser is configured as a counterflow unit, in which a primary heat exchange loop (i.e., the loop carrying the fluid being cooled) thermally interacts with a secondary heat exchange loop such that the entrance of the primary loop is adjacent the exit of the secondary loop, while the exit of the primary loop is adjacent the entrance of the secondary loop. The counterflow arrangement enables the highest possible outlet temperature of the circulating fluid medium, which, as discussed below, could be DHW or hydronic fluid for SH. The pump is configured to circulate the organic working fluid through at least the conduit, expander and condenser.

[0007] The thermodynamic properties of some organic working fluids are ideally suited to the temperature and pressure regimes encountered in micro-CHP operation. Of particular interest is that such fluids remain superheated even after giving up a significant portion of their energy in the expansion process. This is in contrast to other working fluids, such as water, that typically condense to a saturated condition after expansion. Furthermore, the use of organic working fluid rather than water is important where shipping and even some end uses could subject portions of the system to freezing temperatures (below 32° Fahrenheit, 0° Celsius). With a water-filled system, damage and inoperability could ensue after prolonged exposure to sub-freezing temperatures, whereas an organic working fluid-based system would be impervious to temperature extremes encountered by dwellings and related buildings incorporating such a system. In addition, by using an organic working fluid rather than water, corrosion issues germane to water in the presence of oxygen are avoided. The organic working fluid is preferably either a halocarbon refrigerant or a naturally-occurring hydrocarbon. Examples of the former include the refrigerant known as R-245fa, while examples of the latter include some of the alkanes, such as isopentane. Another advantage associated with organic working fluids is that their high vapor density and heat transfer properties in the superheated state ensure that maximum heat and power can be extracted from the fluid without having to resort to a large expander.

[0008] Optionally, the energy conversion circuit comprises a generator coupled to the expander to produce electricity and a circulating fluid medium in thermal communication with the condenser such that at least a portion of the heat given up by the organic working fluid in the condenser provides increased thermal content to the circulating fluid medium. In the present context, the term "thermal communication" is meant to broadly cover all instances of thermal interchange brought about as a result of coupling between system components, whereas the more narrow "heat exchange communication" is meant to cover the more specific relationship between direct, adjacent heat exchange components designed specifically for that purpose. In addition, the expander is preferably a scroll expander, while the condenser construction facilitates the aforementioned counterflow arrangement. A flat plate heat exchanger is one example of a condenser that can be run in a counterflow arrangement, where each fluid enters a header from opposite directions and splits into alternately-spaced parallel streams. Another arrangement that can be run in a counterflow arrangement is a shell-and-tube heat exchanger, where the working fluid vapor enters the top of the shell and is directed by baffles past the circulating fluid, which traverses the condenser through a series of tubes, entering at or near the shell bottom and exiting at or near the top in close proximity to the incoming working fluid vapor.

[0009] A subcooling section in the condenser can be employed such that if the condensate level of the organic working fluid is maintained in the lower portion of the shell, the incoming circulating fluid can cool this condensate below the saturation temperature, which is necessary for pumping of the condensed organic working fluid. This helps to keep the condensed working fluid in liquid form which, if not for the additional subcooling, could experience enough of a pressure drop in the pump inlet section to flash back into vapor. This subcooling is also possible in the previously discussed flat plate heat exchanger. There, the superheated organic working fluid vapor enters the heat exchanger, desuperheats, and then condenses as it gives up its heat to the circulating fluid. As the circulating fluid is flowing counter to the vapor, it can approach the temperature of the incoming superheated vapor, which can be substantially higher than the vapor condensing temperature. Circulating fluid temperatures from this arrangement can therefore produce potable hot water. By keeping this water at or above a predetermined temperature, such as 140° F. (60° C.), the risk of growth of various pathogens, such as legionella, is reduced or eliminated.

[0010] Preferably, the circulating fluid medium is configured to transport either or both an SH fluid, such as water or forced air, or DHW. Preferably, the heat source is a burner in thermal communication with an evaporator such that heat provided by the burner causes the organic working fluid that flows through the conduit in the evaporator to become superheated. Also, the burner can be disposed within a container (of which the evaporator may form an integral part) which may include an exhaust duct to carry away combustion products (primarily exhaust gas), an exhaust fan to further facilitate such product removal, as well as an exhaust gas heat exchanger disposed adjacent (preferably within) the exhaust duct so that residual heat present in the exhaust gas can be used for supplemental heating in other parts of the cogeneration system. The exhaust gas heat exchanger can further include an exhaust gas recirculation device to further improve heat transfer from the exhaust gas.

[0011] The cogeneration system can be configured such that the organic working fluid is directly-fired or indirectly-fired. In the former configuration, the relationship between the burner and the organic working fluid-carrying evaporator is such that the flame from the combustion process in the burner directly impinges on either the conduit carrying the fluid or a container (alternately referred to as a combustion chamber) that houses at least a portion of the organic working fluid-carrying conduit such that the portion of the conduit where the organic working fluid becomes superheated is considered the evaporator. In the latter configuration, the flame from the combustion process in the burner gives up a portion of its heat to conduit making up a secondary circuit, which in turn conveys a heat exchange fluid to an interloop heat exchanger. The indirectly-fired system is advantageous in terms of system flexibility, due in part to its ability to minimize temperature excursions in the evaporator, and maintainability, as heat-sensitive components (such as the conduit used to carry the working fluid) are not directly exposed to the combustion process in the case of a burner for a heat source. The directly-fired system is advantageous in terms of system cost and simplicity.

[0012] According to still another aspect of the present invention, a Rankine cycle cogeneration system is disclosed. The system includes an organic working fluid, an evaporator made up of a burner and conduit adjacently spaced relative to the burner, a substantially closed-loop working fluid circuit in thermal communication with the burner, and at least one energy conversion circuit. Heat generated during burner operation is sufficient to superheat the organic working fluid disposed in the adjacent conduit. The closed-loop working fluid circuit includes an expander that in operation maintains the organic working fluid in a superheated state after expansion, a condenser in fluid communication with the expander, and a pump configured to circulate the organic working fluid. The energy conversion circuit includes a generator coupled to the expander to produce electricity, and a circulating fluid medium in thermal communication with the organic working fluid in the condenser such that at least a portion of the heat given up by the organic working fluid provides increased thermal content to the circulating fluid medium. The condenser is constructed such that the circulating fluid medium is in counterflow with the organic working fluid.

[0013] According to another aspect of the present invention, a dwelling configured to provide at least a portion of the heat and power needs of occupants therein is disclosed. The dwelling (which can be, for example, a house, apartment or commercial, industrial or office building) includes walls, a roof situated above the walls, at least one ingress/egress (such as a door) to facilitate passage into and out of the dwelling, and a cogeneration system in heat and power communication with at least one room formed within the dwelling. The cogeneration system includes, at a minimum, a heat source, a working fluid circuit, a generator and a circulating fluid medium both of which can be responsive to the operation of the working fluid circuit. The condenser is made up of at least a first loop to convey the organic working fluid, and a second loop to carry fluid from the circulating fluid medium. The second loop is in counterflow thermal communication with the first loop, and upon transfer of heat from the first to the second loop, the circulating fluid medium can provide at least SH or DHW to the dwelling. By way of example, the condenser could be a parallel plate condenser or a shell-and-tube condenser, as previously discussed. Optionally, the dwelling further comprises a controller (such as a thermostat) responsive to occupant input. Additionally, the condenser is configured such that a circulating fluid in the circulating fluid medium passes through the condenser in counterflow relationship to the organic working fluid.

[0014] According to yet another aspect of the present invention, a micro combined heat and power system is disclosed. The micro combined heat and power system includes an electric production subsystem and a heat production subsystem. The electric production subsystem includes an organic working fluid, a burner for superheating the organic working fluid, a scroll expander configured to receive and expand the organic working fluid in a superheated state, a generator operatively coupled to the scroll expander to produce electricity, a condenser in fluid communication with the scroll expander, and a pump to circulate the organic working fluid through the electricity generating loop. The heat production subsystem comprises a circulating fluid medium in thermal communication with the condenser such that DHW or hydronic SH can be produced from the heat exchanged in the condenser.

[0015] According to still another aspect of the present invention, a method of producing heat and electrical power from a cogeneration device is disclosed. A first working fluid circuit with conduit, expander, counterflow two loop condenser and pump, and an energy conversion circuit are used in the present method to achieve cogeneration. Steps in the method include providing a heat source, configuring the first circuit to transport an organic working fluid adjacent the heat source, superheating the organic working fluid in the first circuit, and expanding the superheated organic working fluid to generate electricity. The organic working fluid in the first circuit gives up at least a portion of its heat via counterflow heat exchange relationship with a fluid in the circulating fluid medium, where a first loop in the condenser defines the part of the working fluid circuit in the condenser and a second loop defines the part of the circulating fluid medium in the condenser. Optionally, the circulating fluid medium is configured to transport an SH fluid, such as water or forced air. Similarly, the circulating fluid medium can be configured to transport DHW.

BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

[0016] The following detailed description of the preferred embodiments of the present invention can be best understood when read in conjunction with the following drawings, where like structure is indicated with like reference numerals and in which:

[0017] **FIG. 1** shows a schematic diagram of a directly-fired cogeneration system according to an embodiment of the present invention having a connection to both SH and DHW capability;

[0018] **FIG. 2** shows a schematic diagram of an indirectly-fired cogeneration system configuration with connections to separate SH and DHW capability;

[0019] **FIG. 3** shows that electrical output is maximized when a cogeneration system is modulated according variable heat loads as compared to that of maintaining a constant heat load;

[0020] **FIG. 4A** shows a shell-and-tube counterflow condenser according to an embodiment of the present invention; and

[0021] **FIG. 4B** shows a parallel plate counterflow condenser according to an embodiment of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0022] Referring initially to **FIG. 1, a** micro-CHP system **100** capable of providing electric current and heated fluid is shown. The system **100** includes a working fluid circuit and an energy conversion circuit. The working fluid circuit includes an expander **101**, a condenser **102**, a pump **103** and an evaporator **104**. These four components define the major components that together approximate an ideal Rankine cycle system, where the evaporator **104** acts as a constant pressure heat addition, the expander **101** allows efficient, nearly isentropic expansion of the working fluid, the condenser **102** acts to reject heat at a constant pressure, and the pump **103** provides efficient, nearly isentropic compression. The evaporator **104**, details of which will be discussed at length below, functions as the primary heat generator in micro-CHP system **100**. In such a configuration, heat generated at the heat source (shown in the figure being produced by a combustion process where a fuel, such as natural gas, is transported via gas line **152** past a gas valve **153** to a burner **151**) in the evaporator **104** is transferred to an organic working fluid being transported through conduit **110** (alternately referred to as piping). In the present micro-CHP system **100**, the energy produced by the expansion of the organic working fluid is converted into electricity and heat as the two useable forms of energy. An optional fan **158** to pull away heat source byproducts is shown downstream of the heat source as an induced-draft fan, although it could also be a forced-draft fan if located upstream relative to the burner **151** and its ancillary componentry.

[0023] The energy conversion circuit takes the increased energy imparted to the working fluid in the working fluid circuit and converts it into useable form. The electrical form of the useable energy comes from a generator **105** (preferably induction type) that is coupled to expander **101**. The hot fluid form of the useable energy comes from a circulating fluid medium **140** (shown preferably as a combined SH and DHW loop) thermally coupled to condenser **102**. Hydronic fluid flowing through circulating fluid medium **140** is circulated with a conventional pump **141**, and can be supplied as space heat via radiator **148** or related device. As an example, hydronic fluid could exit the condenser **102** at about 112° F. (50° C.) and return to it as low as 86° F. (30° C.). The nature of the heat exchange process is preferably through either heat exchangers **180** (shown notionally for the DHW loop, but equally applicable to the SH loop), or through a conventional hot water storage tank (for a DHW loop). Isolation of either the SH or DHW loop within circulating fluid medium **140** is accomplished through valves **107**E and **107**F. It will be appreciated by those of ordinary skill in the art that while the embodiments depicted in the figures show DHW and SH heat exchangers in parallel (and in some circumstances being supplied from the same heat exchange device, shown later), it is within the spirit of the present disclosure that series or sequential heat exchange configurations could be used. It will also be appreciated that the heat exchanger **180** depicted in **FIG. 1** could be in the form of the aforementioned hot water storage tank, where the hot fluid circulating through circulating fluid medium **140** gives up at least a portion of its heat to incoming domestic cold water coming from water supply **191**A, which is typically from a municipal water source, well or the like. Once heated in the tank, the domestic water can then be routed to remote DHW locations, such as a shower, bath or hot water faucet, through DHW outlet **191**B.

[0024] The organic working fluid (such as naturally-occurring hydrocarbons or halocarbon refrigerants, not shown) circulates through the loop defined by the fluidly-connected expander **101**, condenser **102**, pump **103**, evaporator **104**, and conduit **110**. The embodiment of the micro-CHP system **100** shown in the figure is operated as a directly-fired system, where the fluid that passes adjacent the heat source through conduit **110** is also the working fluid passing through the expander **101**. While the expander **101** can be any type, it is preferable that it be a scroll device. For example, the scroll expander **101** can be based on a conventional single scroll device, as is known in the art. A scroll device exhibits numerous advantages over other positive-displacement systems. For example, since they are made in very high production volume in dedicated modem facilities, its cost is inherently low. Furthermore, the modification to an existing production line to convert from making scroll compressors to making scroll expanders is considerably simpler than to modify an existing reciprocating compressor production line, as the changes to valves and actuation are minimized. Additionally, by operating with very few moving parts, it can go long durations between service or component failure. Moreover, when operating in expansion mode, once the fixed volume of working fluid is captured, the nature of the working fluid-containing chamber is such that the volume of the chamber is always expanding. This also promotes long component life as it avoids the possibility of trapping and attempting to compress (such as upon a return stroke) a working fluid that could, under certain pressure and temperature regimes, include an incompressible liquid phase condensate. An optional oil pump **108** may be used to provide lubricant to the scroll. A generator **105** (preferably induction type) is coupled to expander **101** such that motion imparted to it by expander **101** generates electricity. The generator **105** is preferably an asynchronous device, thereby promoting simple, low-cost operation of the system **100**, and reducing reliance on complex generator speed controls and related grid interconnections. The optional recuperator **109** may be placed between expander **101** and condenser **102** in order to extract additional heat from the working fluid once the fluid has been expanded. An optional accumulator **111** may be connected intermediate condenser **102** and pump **103**, and can be isolated from the remainder of the organic working fluid circuit by valve **107**D. The accumulator **111** acts as a working fluid storage device in that during periods of low fluid flow rates (such as during system startup), it can provide an additional charge of fluid into the working fluid circuit to minimize, among other things, cavitation of pump **103**. An optional warming device **113** can be placed adjacent the accumulator **111** to keep it slightly warmer than the remaining circuit during periods of system inoperation. Heat for the warming device **113** can be, for example, from an electric (resistive) supply.

[0025] Referring next to **FIG. 3, a** comparison between two ways to mimic the modulation of a boiler to achieve maximum system efficiency is shown. In many conventional boiler applications, where the set point of the system **100** is determined by a single parameter, such as an outdoor temperature, controller **130** (not presently shown) can be used to provide primary control input to the evaporator **104** (not presently shown). By operating the evaporator in a variable-capacity mode, where the gas valve **153** on the burner **151** (neither of which are presently shown) can be modulated, the SH or DHW portions of the circulating fluid medium can be maintained at the desired set point. Such modulation permits quasi-steady state system operation that is responsive to heat needs that are keyed to a specified hydronic supply temperature set point, which is preferably the hydronic temperature coming off the condenser **102** (not presently shown). For example, the ambient outdoor temperature is measured and sets the desired hydronic supply temperature. A single measuring point is used, preferably positioned on the North side of the building (in the Northern hemisphere), to avoid the influence of direct sunlight on cold days. A linear variation of the hydronic set point is used, so that on very cold days the hydronic set point is at or near its maximum setting (shown in the figure as 75° C.), while on warm days the set point is at or near its minimum (shown in the figure as 25° C.). The hydronic pump **141** (not presently shown) operates continuously so there is always a flow through the system.

[**0026**] Similarly, for the micro-CHP, a single measurement of outdoor ambient temperature can be used to establish the hydronic supply set point temperature. The working fluid mass flow is then controlled by the controller to maintain the actual supply temperature at this set point. Either an inverter drive or a separate input on the pump **103** would be sufficient to adjust the displacement of the pump **103** at constant motor speed to vary flow rate. The gas valve **153** is modulated to maintain the desired set point for the evaporator **104** outlet temperature of the working fluid into the expander **101**. Properties of the working fluid, as well as of optional fluids, such as lubricants, may dictate maximum operating temperatures of the fluid coming out of the evaporator **104**. For example, if the working fluid is the refrigerant known as R-245fa, the temperature set point at the evaporator **104** exit is about 310° F. (154° C.).

[**0027**] By operating the system such that the temperature of the working fluid at the evaporator **104** outlet is at or near its maximum value, good overall system efficiency results, regardless of system load. This can include very low thermal loads; for example, if the thermal load falls much below about 30 to 40% of full load, it is appropriate to shutdown the system and cease making both heat and power. Since the hydronic pump is kept running at all times, even at a low flow rate, the controller **130** can continuously monitor the error signal between the hydronic actual and set point values. When this error is large enough, (i.e., the actual temperature is below the set point by a preselected value) the controller **130** can start the system for another on-cycle. As the system **100** operates it may find that even at the minimum system mass flow, the actual supply temperature begins to exceed the set point. When this occurs, the system **100** is again shut down. Under this approach, the system **100** will operate for as many hours as possible during the colder heating season by running just often enough to maintain the hydronic supply temperature at the right value for the nominal heating load. When the system **100** operates at less than the maximum hydronic supply temperature, more power is generated than at the maximum temperature, so the controller **130** automatically and passively maximizes the electric power, which can be produced. Thus, as shown in the figure, the net electrical output goes up (at the same working fluid mass flow rate) as hydronic fluid supply temperature requirements goes down, while variations in working fluid flow rate and can be used in conjunction to vary electric output under a given thermal load. This inherent flexibility promotes overall energy (electrical and heat) system efficiency.

[**0028**] Referring again to **FIG. 1**, the generator **105** is preferably an asynchronous device, thereby promoting simple, low-cost operation of the system **100**, and reducing reliance on complex generator speed controls and related grid interconnections. An asynchronous generator always supplies maximum possible power without controls, as its torque requirement increases rapidly when generator **105** exceeds system frequency. The generator **105** can be designed to provide commercial frequency power, for example, 50 or 60 Hz, while staying within close approximation (often **150** or fewer revolutions per minute (rpm)) of synchronous speed (3000 or 3600 rpm). Block valve **107**A and bypass valve **107**B are situated in the organic working fluid flow path defined by conduit **110**. These valves respond to a signal in controller **130** that would indicate if no load (such as a grid outage) were on the system, or if a high Q/P were desired, thus allowing the superheated vapor to bypass the expander, thereby transferring a majority of the excess heat to the heat exchange loop in the condenser **102** (for high Q/P operation), as well as additionally avoiding overspeed of expander **101**.

[**0029**] An optional level indicator switch **120** is placed at the discharge of condenser **102**, while controller **130** is used to regulate system operation. Sensors connected to controller **130** measure key parameters, such as fluid level information taken from the level indicator switch **120**, and organic working fluid temperatures at various points within the organic working fluid circuit. Through appropriate program logic, the controller **130** can be used to vary pump speed, gas flow rate and evaporator output temperature, as well as to open and close valves. In many applications, where the set point of the system is determined by a single parameter, such as an outdoor temperature, the controller **130** can then be used to provide primary control input to the evaporator **104**. By operating the evaporator **104** in a variable-capacity mode, where the gas valve **153** on the burner **151** can be modulated, optional SH or DHW portions of the circulating fluid medium **140** can be maintained at the desired set point. The circulating fluid medium **140** includes a pump **141** and block valves **107**E and **107**F that selectively permit flow to SH loop heat exchanger (indicated by radiator **148**) or DHW loop (indicated by DHW heat exchanger **180**, which preferably takes cold water from a source **191**A and after heating sends it to a DHW destination **191**B). Preferably, the fluid circulating through at least the SH portion of the circulating fluid medium **140** is a hydronic fluid. By way of example, hydronic fluid could exit the condenser **102** at about 50° C. (122° F.) and return to it as low as 300° C. (86° F.). Controller **130** may also be used to manipulate block valves **107**A and **107**C and bypass valve **107**B, which are situated in the organic working fluid flow path defined by conduit **110**. Valves **107**A and **107**B respond to a signal in controller **130** that would indicate if no load (such as a grid outage) were on the system, or if a high Q/P were desired, thus allowing the superheated vapor to bypass the expander, thereby transferring a majority of the excess heat to the heat exchange loop in the condenser **102** (for high Q/P operation), as well as additionally avoiding overspeed of expander **101**. Similarly, when recuperated operation is desired, optional block valve **107**C can be used to help tailor working fluid preheat needs of the system by selectively allowing a portion of the working fluid passing from the pump **103** to evaporator **104** to be routed through recuperator **109**.

[**0030**] Referring next to **FIG. 2**, an indirectly-fired cogeneration system **200** is shown. A second loop **250** in cogeneration system **200** includes two parallel sub-loops **250**A, **250**B, while a first loop, which includes an expander **201** coupled to generator **205**, condenser **202**B, pump **203**, recuperator **209**, conduit **210** and accumulator **211** with warming device **213**, is configured similarly to, although not necessarily identical to, the system shown in **FIG. 1**. The most significant difference of the first loop over the system of **FIG. 1** is that the evaporator **104** of the former system is now replaced with an interloop heat exchanger **202**A, thus acting as a heat source for the first loop. Controller **230** is similar to that of the previously described system, but now with enlarged functionality to additionally control some or all of the operations of the second loop **250**. It will be appreciated that circulating fluid medium **240** is, while notionally depicting only an SH component that includes a pump **241** and radiator **248**, is understood to be similar to that of **FIG. 1**. Also as before, valve **207**C can be used to bypass the recuperator **209** in order to achieve variable Q/P, while pump **208** is used to circulate oil or related lubricant through the expander.

[**0031**] Heat to the two parallel sub-loops **250**A, **250**B is provided by a burner **251**, which is supplied with fuel by a gas train **252** and variable flow gas valve **253**. Piping **260** (alternately referred to as conduit, and which makes up the parallel sub-loops) passes through a combustion chamber **254**, where the heat from the combustion of fuel at burner **251** is given up to the heat exchange fluid (not shown) that flows through piping **260**. Piping **260** branches out into the first parallel sub-loop **250**A, which transports the heat exchange fluid that has been heated in combustion chamber **254** to interloop heat exchanger **202**A in order to give up the heat to organic working fluid flowing through the first loop, which as previously described, save the presence of the interloop heat exchanger **202**A in place of the evaporator **104**, is similar in construction to the directly-fired cogeneration system **100** shown in **FIG. 1**. Block valves (not shown) could be used to regulate flow between the sub-loops **250**A and **250**B; however, by idling the pump of the inactive sub-loop, significant flow in that sub-loop is prevented without the need for additional valving. The second parallel sub-loop **250**B transports the heat exchange fluid to DHW heat exchanger **280** in order to heat up domestic hot water. One side of DHW heat exchanger **280** (which can be a water storage tank) includes coil **280**A configured to transport the heat exchange fluid, and another side, the shell **280**B, to transport DHW (not shown) from a cold water inlet **291**A, past coil **280**A and to DHW outlet **291**B. As with the system shown in **FIG. 1**, the cold water preferably comes from either a well or a city/municipal water supply. Similarly, temperature sensor **271**B can detect the temperature of the DHW coming out of the DHW heat exchanger **280**. This sensor can also be linked to a controller **230** (discussed in more detail below).

[**0032**] Combustion chamber **254** includes an exhaust duct **255**, an exhaust gas recirculation device **256** with exhaust duct heat exchanger **257**, and fan **258**. Temperature sensor **271**A is placed at the combustion chamber **254** outlet for the second loop **250** to measure the temperature conditions of the heat exchange fluid, in a manner similar to that of temperature sensor **271**B. Second loop pumps **285**A, **285**B are used to circulate heat exchange fluid through the second loop **250**, with pump **285**B circulating heat exchange fluid through DHW heater **280** and pump **285**A circulating heat exchange fluid through interloop heat exchanger **204**. The exhaust duct heat exchanger **257** and an EGR device **256** accept hot exhaust gas from the burner **251** and recirculate it in an internal heat exchange process, thereby lowering the temperature of the exhaust gas that is pulled away and vented to the atmosphere by fan **258**. The heat given up by the exhaust gas in the exhaust gas heat exchanger **257** can be used to provide additional heat to other parts of the system **200**. For example, this additional heat can be used to increase the temperature of the heat exchange fluid flowing in second loop **250**, or to increase the heat content of the organic working fluid in the first loop.

[**0033**] Referring next to **FIGS. 4A and 4B**, the condensers **102**A and **102**B are described in conjunction with the directly-fired cogeneration system **100** of **FIG. 1**, although they are equally applicable to the indirect system **200** of **FIG. 2**. Condensers **102**A and **102**B extract excess heat from the organic working fluid after the fluid has been expanded such that a circulating fluid medium **140** fluidly communicating with the condenser **102** can absorb and transfer the heat to remote locations. To achieve a Q/P that varies depending on the heat and electric needs, the burner **151** is capable of modulation, while either condenser **102**A, **102**B is simultaneously responsive to fluctuating thermal content in the working fluid and capable of transferring enough heat for hydronic SH needs (as well as DHW needs). Condenser staging is central to providing a balance between the often diverging requirements of high heat loads in the circulating fluid medium and the need for burner modulation. One of the basic approaches to controlling the temperature at the evaporator outlet is to vary the mass flow out of the expander. By way of example, when the working fluid is R-245fa, the outlet temperature can be fixed, or set, to about 310 F. (154° C.). By selecting a corresponding pump capacity and speed to control the mass flow of working fluid into the evaporator, coupled with the ability of the expander to accept a concomitant amount of vapor flow, the evaporator outlet pressure tends to be typically between 380 and 400 psia (2.62 MPa and 2.76 MPa, respectively), with a preferred pressure of 392 psia (2.70 MPa). The outlet temperature of 310° F. (154° C.) leaves about 30° F. (17° C.) of superheat above the saturation temperature for the fluid, thereby simplifying the control system and its ability to adjust the burner firing rate to maintain the set outlet temperature. Then, as the thermal output capacity to meet lower loads is reduced, which could be due to cool or warm weather, the pump flow is reduced and the pressure changes accordingly to a lesser value. In an alternative approach, the same superheat temperature (30° F.) (17° C.) can be maintained, while allowing the evaporator outlet temperature to fluctuate. Of the two approaches, it is preferable to run the evaporator outlet at a constant 310° F. (154° C.) from an efficiency stand point and for simplicity of the controller, as calculating the proper superheat temperature requires measuring, in some way, the evaporator outlet pressure.

[**0034**] Referring with particularity to **FIG. 4**A, shell-and-tube condenser **102**A is shown. Organic working fluid travels through a first, or primary, loop that passes through shell **120**, while circulating fluid travels through a second loop defined by tubing **121**. The organic working fluid first flows into a desuperheating section, then through a subcooling section situated below the desuperheating section. The two sections are separated by partition **123**, which includes perforations **123**A therein to permit vapor flow from the subcooling section to waft up into the desuperheating section. Baffles **122** are located in the desuperheating section of shell **120**, and create a tortuous path through which the incoming working fluid must flow, thus promoting increased thermal interchange between the working fluid and the circulating fluid in the portion of tubing **121** that is disposed in the desuperheating section. The shell **120** of condenser **102**A includes. After passing from the desuperheating section to the subcooling section through the partitions **123**A, the organic working fluid condenses, and is defined by a condensate free surface **126** above which the working fluid exists as a vapor, and below as a liquid. Outlet **125** is situated at or near the bottom of shell **120** to permit the condensed working fluid to continue on to the pump **103** (not presently shown) in the working fluid circuit. The circulating fluid enters through tubing inlet **127** and exits through tubing outlet **128**. In between, tubing **121** defines multiple passes through the shell **120** to maximize contact between the tubing **121** and the working fluid passing through the desuperheating and subcooling sections of shell **120**. By the present construction, the organic working fluid passing through shell **120** and the circulating fluid passing through tubing **121** are in a counterflow relationship to one another.

[**0035**] Referring with particularity to **FIG.** 4B, flat plate condenser **102**B is shown. The flow pattern between the organic working fluid and the circulating fluid is similar, in that working fluid enters an inlet **124** to a condenser inlet manifold and exits through an outlet **125** coming from condenser outlet manifold while circulating fluid enters through tubing inlet **127** and exits through tubing outlet **128**. As with the condenser **102**A shown in **FIG.** 4A, the two fluids exchange heat in a counterflow pattern, although, unlike condenser **102**A, the heat exchange process takes part without multiple passes or convoluted flow patterns, relying instead on the large ratio of surface area to flow volume made possible by the flat passages. Because of the absence of multiple passes, all of the working fluid leaving the condenser is in close heat transfer communication with only the incoming cooling fluid, hence all the exiting working fluid can more easily and more completely be cooled to a temperature closer to that of the incoming coolant.

[**0036**] Having described the invention in detail and by reference to preferred embodiments thereof, it will be apparent that modifications and variations are possible without departing from the scope of the invention defined in the appended claims. More specifically, although some aspects of the present invention are identified herein as preferred or particularly advantageous, it is contemplated that the present invention is not necessarily limited to these preferred aspects of the invention.

We claim:

1. A cogeneration system comprising:

a heat source;

a working fluid circuit comprising:

conduit configured to transport an organic working fluid through said working fluid circuit, at least a portion of said conduit disposed adjacent said heat source such that said organic working fluid disposed in said portion of said conduit is superheated during operation of said heat source;

an expander in fluid communication with said conduit such that said organic working fluid received therefrom remains superheated after expansion in said expander;

a condenser in fluid communication with said expander, said condenser configured such that a circulating fluid medium passing therethrough is in counterflow relationship to said organic working fluid; and

a pump configured to circulate said organic working fluid through at least said conduit, expander and condenser; and

at least one energy conversion circuit operatively responsive to said working fluid circuit such that upon operation of said cogeneration system, said at least one energy conversion circuit is configured to provide useable energy.

2. A cogeneration system according to claim 1, wherein said condenser is a flat plate condenser.

3. A cogeneration system according to claim 1, wherein said condenser is a shell-and-tube condenser.

4. A cogeneration system according to claim 3, wherein the shell of said shell-and-tube condenser is in fluid communication with said working fluid circuit, and the tube is configured to be in fluid communication with said circulating fluid medium.

5. A cogeneration system according to claim 1, wherein said at least one energy conversion circuit comprises:

a generator coupled to said expander to produce electricity; and a circulating fluid medium in thermal communication with said condenser such that at least a portion of the heat given up by said organic working fluid in said condenser provides increased thermal content to said circulating fluid medium.

6. A cogeneration system according to claim 5, wherein said expander is a scroll expander.

7. A cogeneration system according to claim 5, wherein said circulating fluid medium is configured to transport a space heating fluid.

8. A cogeneration system according to claim 7, wherein said space heating fluid is water.

9. A cogeneration system according to claim 7, wherein said space heating fluid is forced air.

10. A cogeneration system according to claim 5, wherein said circulating fluid medium is configured to transport domestic hot water.

11. A cogeneration system according to claim 1, wherein said heat source is a burner.

12. A Rankine cycle cogeneration system comprising:

an organic working fluid;

an evaporator capable of superheating said organic working fluid, said evaporator comprising:

a burner; and

conduit adjacently spaced relative to said burner such that during burner operation heat transferred therefrom is sufficient to superheat said organic working fluid disposed in said conduit;

a substantially closed-loop working fluid circuit in thermal communication with said burner, said substantially closed-loop working fluid circuit configured to transport said organic working fluid therethrough, said substantially closed-loop working fluid circuit comprising:

an expander in fluid communication with said conduit such that said organic working fluid received therefrom remains superheated after expansion in said expander;

a condenser in fluid communication with said expander;

a pump configured to circulate said organic working fluid through at least said conduit, expander and condenser; and

at least one energy conversion circuit comprising:

a generator coupled to said expander to produce electricity; and

a circulating fluid medium configured to pass through said condenser in counterflow thermal communication with said organic working fluid such that at least a portion of the heat given up by said organic working fluid in said condenser provides increased thermal content to said circulating fluid medium.

13. A dwelling configured to provide at least a portion of the heat and power needs of occupants therein, said dwelling comprising:

a plurality of walls defining at least one room therebetween;

a roof situated above said plurality of walls;

at least one ingress/egress to facilitate passage into and out of said dwelling; and

a cogeneration system in heat and power communication with said at least one room, said cogeneration system comprising:

a heat source;

a working fluid circuit comprising:

conduit configured to transport an organic working fluid through said working fluid circuit, at least a portion of said conduit disposed adjacent said heat source such that said organic working fluid disposed in said portion of said conduit disposed adjacent said heat source is superheated during operation of said heat source;

an expander in fluid communication with said conduit such that said organic working fluid received therefrom remains superheated after expansion in said expander;

a condenser in fluid communication with said expander, said condenser comprising:

a first loop configured to convey said organic working fluid therethrough; and

a second loop in counterflow thermal communication with said first loop; and

a pump configured to circulate said organic working fluid through at least said conduit, expander and condenser;

a generator responsive to said expander to provide electricity; and

a circulating fluid medium coupled to said second loop, said circulating fluid medium configured to provide at least space heat or domestic hot water to said dwelling.

14. A dwelling according to claim 13, further comprising a controller responsive to occupant input.

15. A dwelling according to claim 14, wherein said controller responsive to occupant input is a thermostat.

16. A dwelling according to claim 13, wherein said circulating fluid medium is configured to provide both space heat and domestic hot water to said dwelling.

17. A micro combined heat and power system comprising:

an electric production subsystem comprising:

an organic working fluid;

a burner for superheating said organic working fluid;

a scroll expander configured to receive and expand said organic working fluid in a superheated state;

a generator operatively coupled to said scroll expander to produce electricity;

a condenser in fluid communication with said scroll expander, said condenser comprising:

a first loop configured to convey said organic working fluid therethrough; and

a second loop in counterflow thermal communication with said first loop; and

a pump to circulate said organic working fluid through said electricity generating loop; and

a heat production subsystem including a circulating fluid medium coupled to said second loop, said circulating fluid medium configured to provide at least space heat or domestic hot water to said dwelling.

18. A method of producing heat and electrical power from a cogeneration device, the method comprising the steps of:

providing a heat source;

configuring a first circuit to transport an organic working fluid adjacent said heat source;

superheating said organic working fluid;

expanding said superheated organic working fluid to generate electricity;

exchanging at least a portion of the excess heat from said organic working fluid that has passed through said expander in a condenser with a circulating fluid medium such that after passing through said condenser, said organic working fluid is no longer in a superheated state, said condenser comprising a first loop configured to convey said organic working fluid therethrough, and a second loop in counterflow thermal communication with said first loop such that said second loop is in fluid communication with said circulating fluid medium; and returning said organic working fluid such that it is adjacent said heat source.

19. A method according to claim 18, wherein said circulating fluid medium is configured to transport a space heating fluid.

20. A method according to claim 19, wherein said space heating fluid is water.

21. A method according to claim 18, wherein said space heating fluid is forced air.

22. A method according to claim 18, wherein said circulating fluid medium is configured to transport domestic hot water.

23. A method according to claim 18, wherein said circulating fluid medium is configured to transport both space heat and domestic hot water.

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