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(54) PROCESS AND DEVICE FOR CONTROLLING THE THERMAL AND ELECTRICAL OUTPUT OF INTEGRATED MICRO COMBINED HEAT AND POWER GENERATION SYSTEMS

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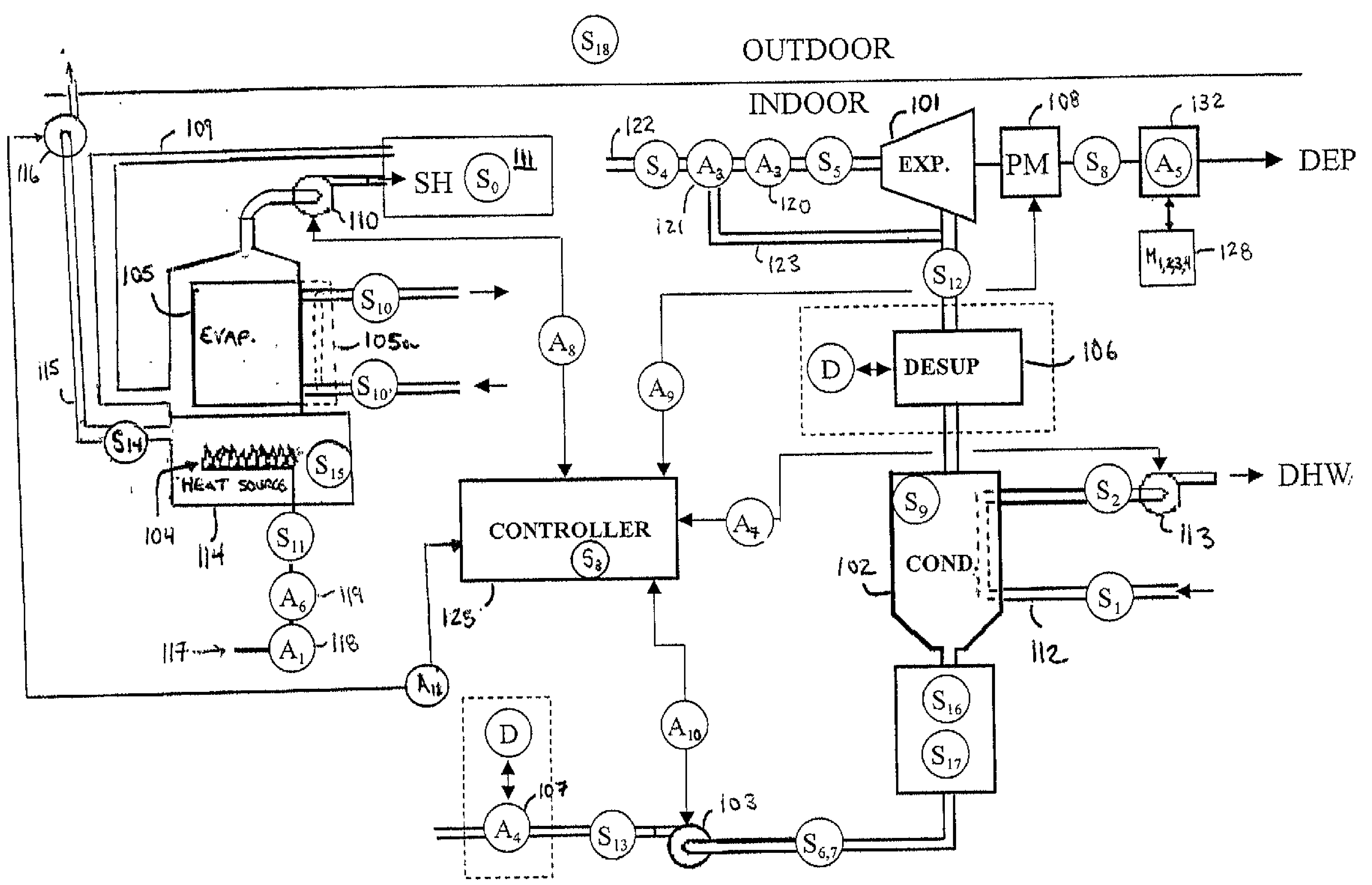
(57) ABSTRACT

Microprocessor-based control sub-systems which control the thermal and electrical output of integrated micro-combined heat and power generation (M-CHP) systems used to supply domestic electrical power, domestic space heating (SH) water, and domestic hot water (DHW). The M-CHP system uses a microprocessor controller to control the internal operating conditions, such as pump speeds, gas flow rate, and evaporator outlet temperature. Controlling these parameters enables setting the capacity of the system at any instant in time, thereby permitting load following, using a variable capacity operation. The controller also monitors through sensors a number of additional safety controls and system protection devices, such as relays/contactor of the alternator to grid, and electrical trips to the feed pump, the oil pump, the hydronic pump, the blower, the gas valve, the expander bypass valves, and other electrically powered devices in the system.



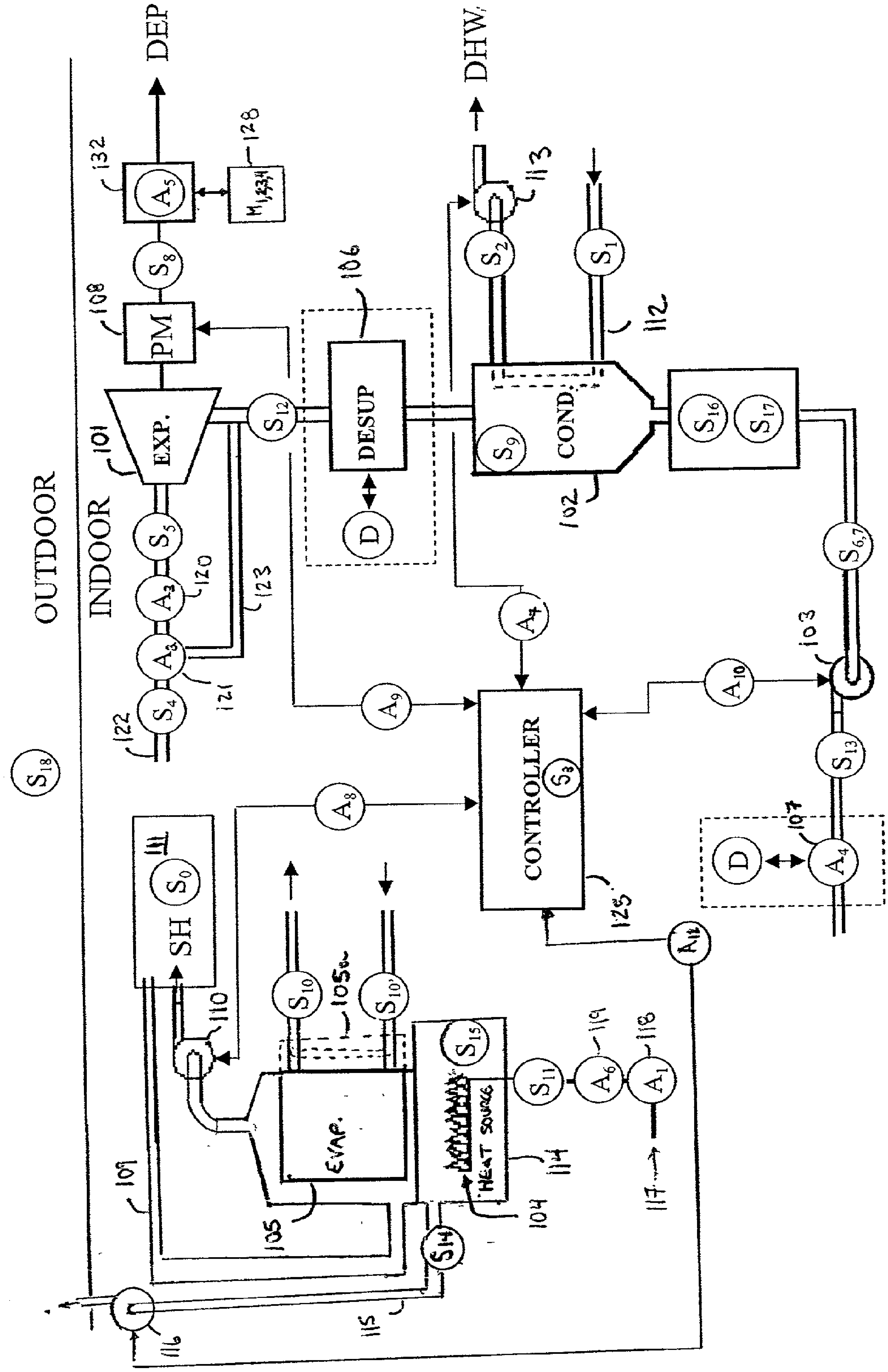

DEP
DHW
OUTDOOR
INDOOR
EXP.
PM
DESUP
COND.
CONTROLLER
SH
EVAP.
HEAT SOURCE
D
101
102
103
104
105
106
107
108
109
110
112
113
114
115
116
117
118
119
120
121
122
123
125
128
132

FIG. 1

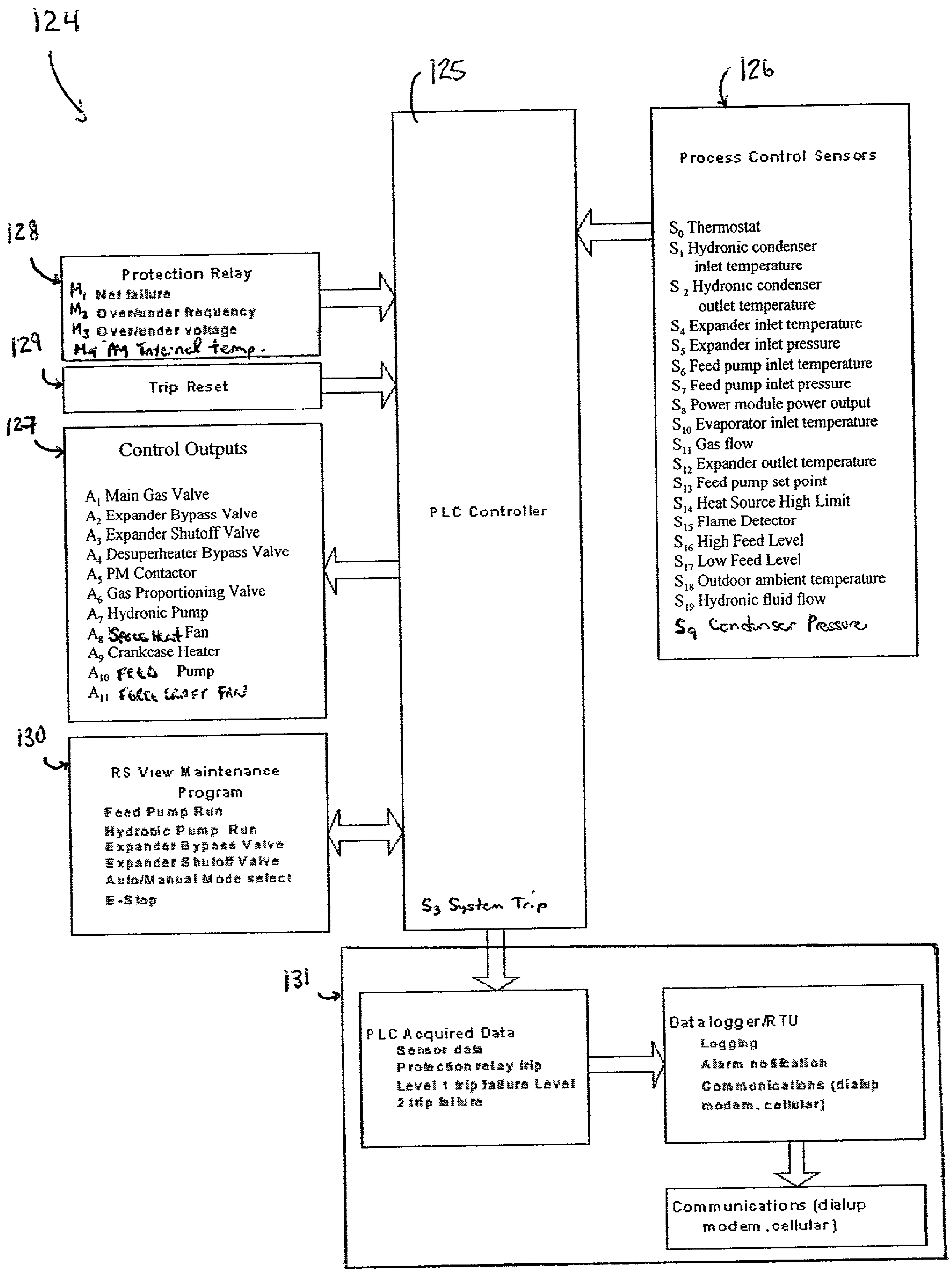

124
125
126
Process Control Sensors
S_0 Thermostat
S_1 Hydronic condenser inlet temperature
S_2 Hydronic condenser outlet temperature
S_4 Expander inlet temperature
S_5 Expander inlet pressure
S_6 Feed pump inlet temperature
S_7 Feed pump inlet pressure
S_8 Power module power output
S_{10} Evaporator inlet temperature
S_{11} Gas flow
S_{12} Expander outlet temperature
S_{13} Feed pump set point
S_{14} Heat Source High Limit
S_{15} Flame Detector
S_{16} High Feed Level
S_{17} Low Feed Level
S_{18} Outdoor ambient temperature
S_{19} Hydronic fluid flow
S_9 Condenser Pressure
128
Protection Relay
M_1 Net failure
M_2 Over/under frequency
M_3 Over/under voltage
M_4 PM Internal temp.
129
Trip Reset
127
Control Outputs
A_1 Main Gas Valve
A_2 Expander Bypass Valve
A_3 Expander Shutoff Valve
A_4 Desuperheater Bypass Valve
A_5 PM Contactor
A_6 Gas Proportioning Valve
A_7 Hydronic Pump
A_8 Space Heat Fan
A_9 Crankcase Heater
A_{10} FEED Pump
A_{11} FORCE DRAFT FAN
PLC Controller
S_3 System Trip
130
RS View Maintenance Program
Feed Pump Run
Hydronic Pump Run
Expander Bypass Valve
Expander Shutoff Valve
Auto/Manual Mode select
E-Stop
131
PLC Acquired Data
Sensor data
Protection relay trip
Level 1 trip failure Level 2 trip failure
Data logger/RTU
Logging
Alarm notification
Communications (dialup modem, cellular)
Communications (dialup modem, cellular)

Fig. 2

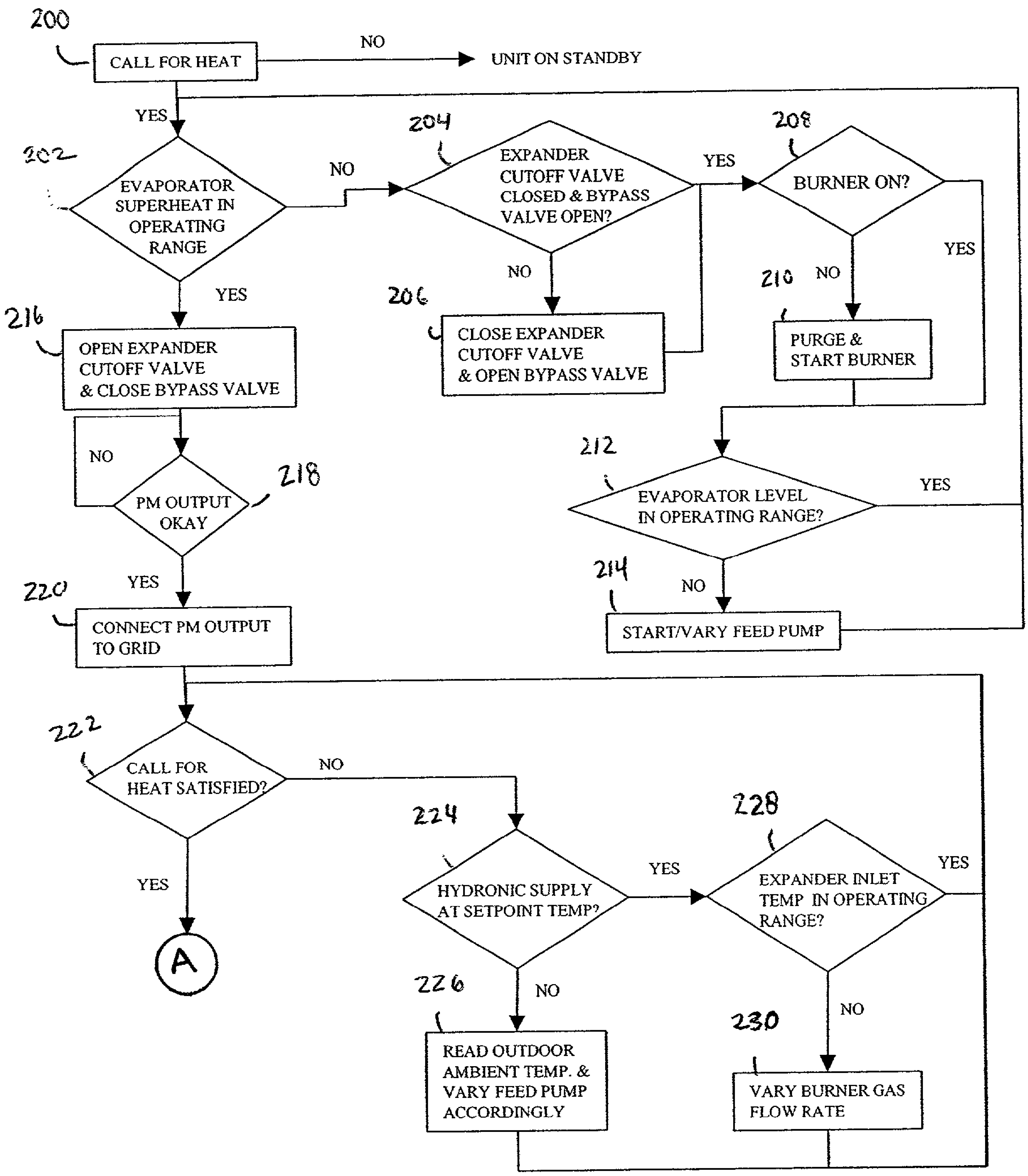

200
CALL FOR HEAT
NO
UNIT ON STANDBY
YES
202
EVAPORATOR SUPERHEAT IN OPERATING RANGE
NO
204
EXPANDER CUTOFF VALVE CLOSED & BYPASS VALVE OPEN?
YES
208
BURNER ON?
YES
NO
206
CLOSE EXPANDER CUTOFF VALVE & OPEN BYPASS VALVE
210
NO
PURGE & START BURNER
YES
216
OPEN EXPANDER CUTOFF VALVE & CLOSE BYPASS VALVE
NO
218
PM OUTPUT OKAY
212
EVAPORATOR LEVEL IN OPERATING RANGE?
YES
214
NO
START/VARY FEED PUMP
220
YES
CONNECT PM OUTPUT TO GRID
222
CALL FOR HEAT SATISFIED?
NO
YES
A
224
HYDRONIC SUPPLY AT SETPOINT TEMP?
YES
228
EXPANDER INLET TEMP IN OPERATING RANGE?
YES
226
NO
READ OUTDOOR AMBIENT TEMP. & VARY FEED PUMP ACCORDINGLY
230
NO
VARY BURNER GAS FLOW RATE

FIG. 3

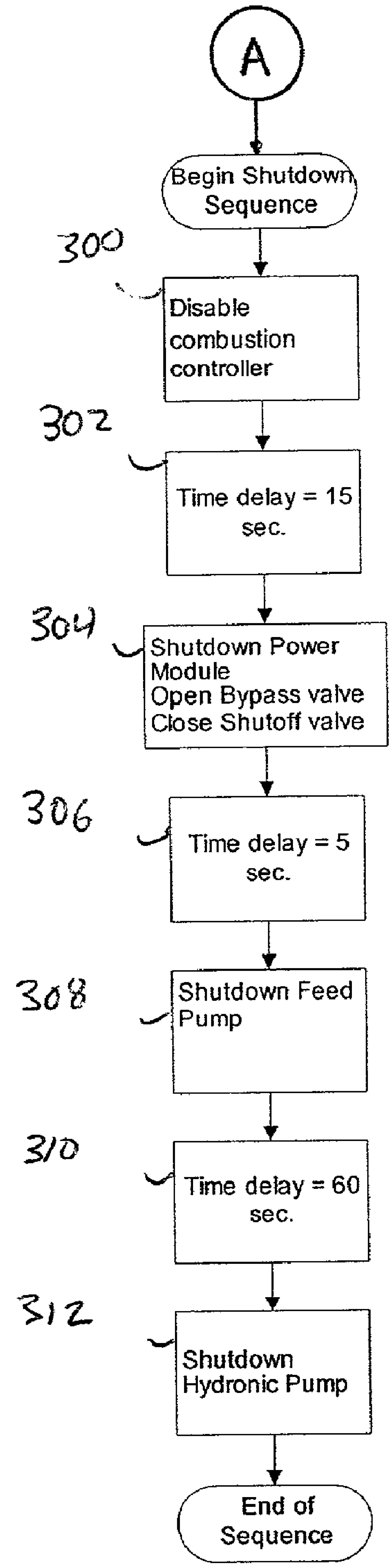

A
Begin Shutdown Sequence
300
Disable combustion controller
302
Time delay = 15 sec.
304
Shutdown Power Module
Open Bypass valve
Close Shutoff valve
306
Time delay = 5 sec.
308
Shutdown Feed Pump
310
Time delay = 60 sec.
312
Shutdown Hydronic Pump
End of Sequence

FIG. 4

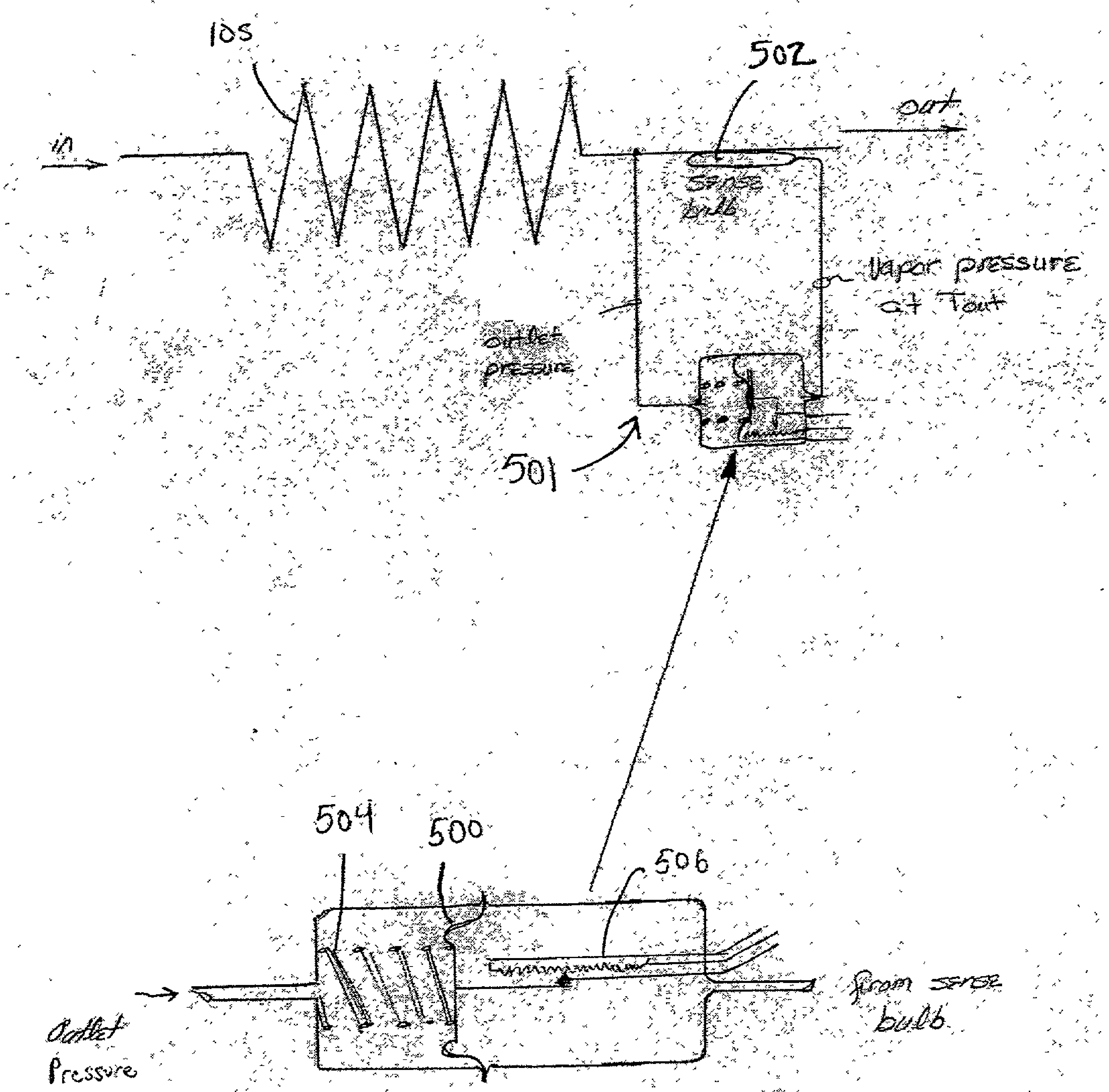

105
502
in
out
sense bulb
vapor pressure at Tout
outlet pressure
501
504
500
506
outlet Pressure
from sense bulb

FIG. 5

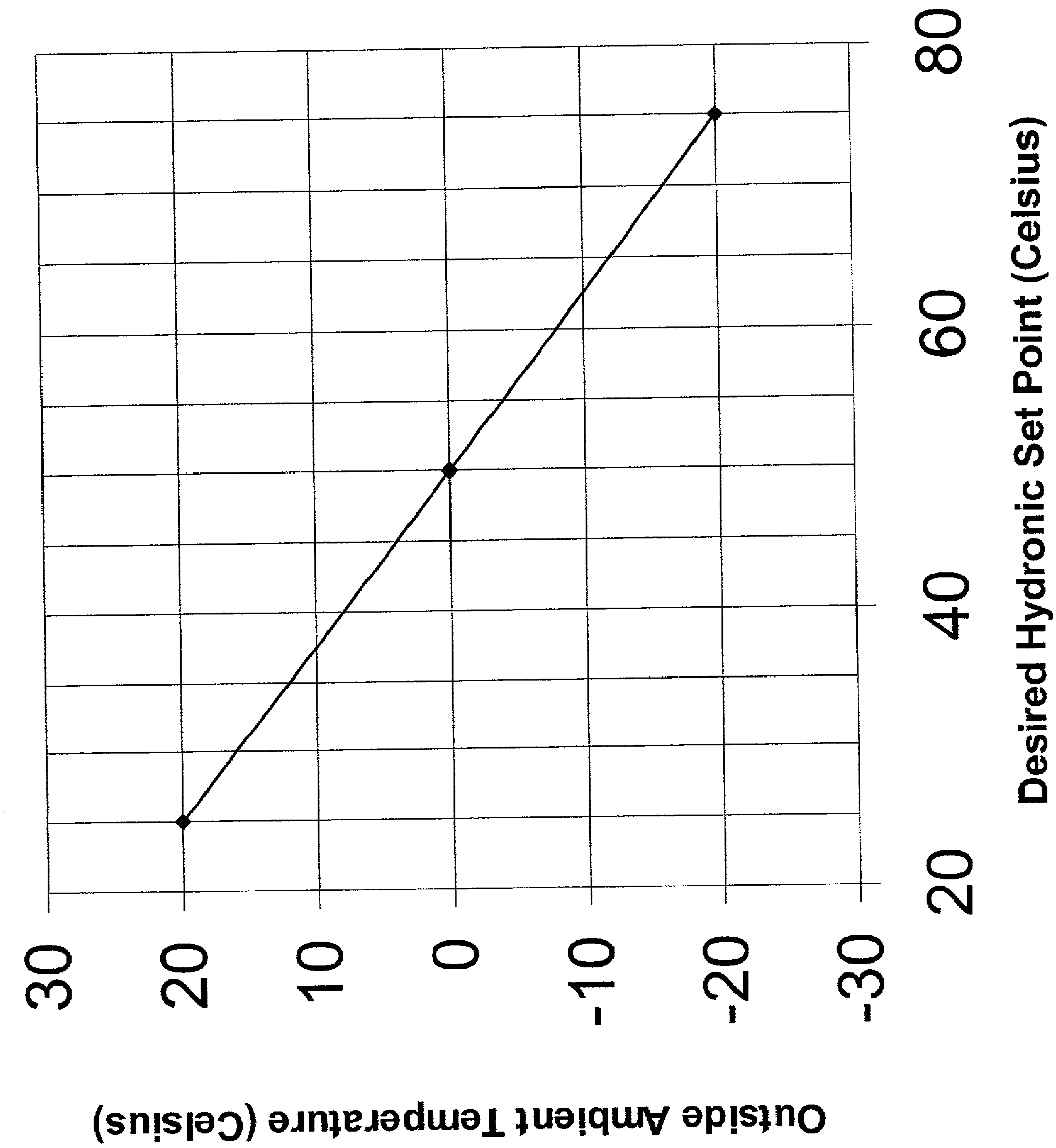

Desired Hydronic Set Point (Celsius)
20
40
60
80
Outside Ambient Temperature (Celsius)
30
20
10
0
-10
-20
-30

FIG. 6

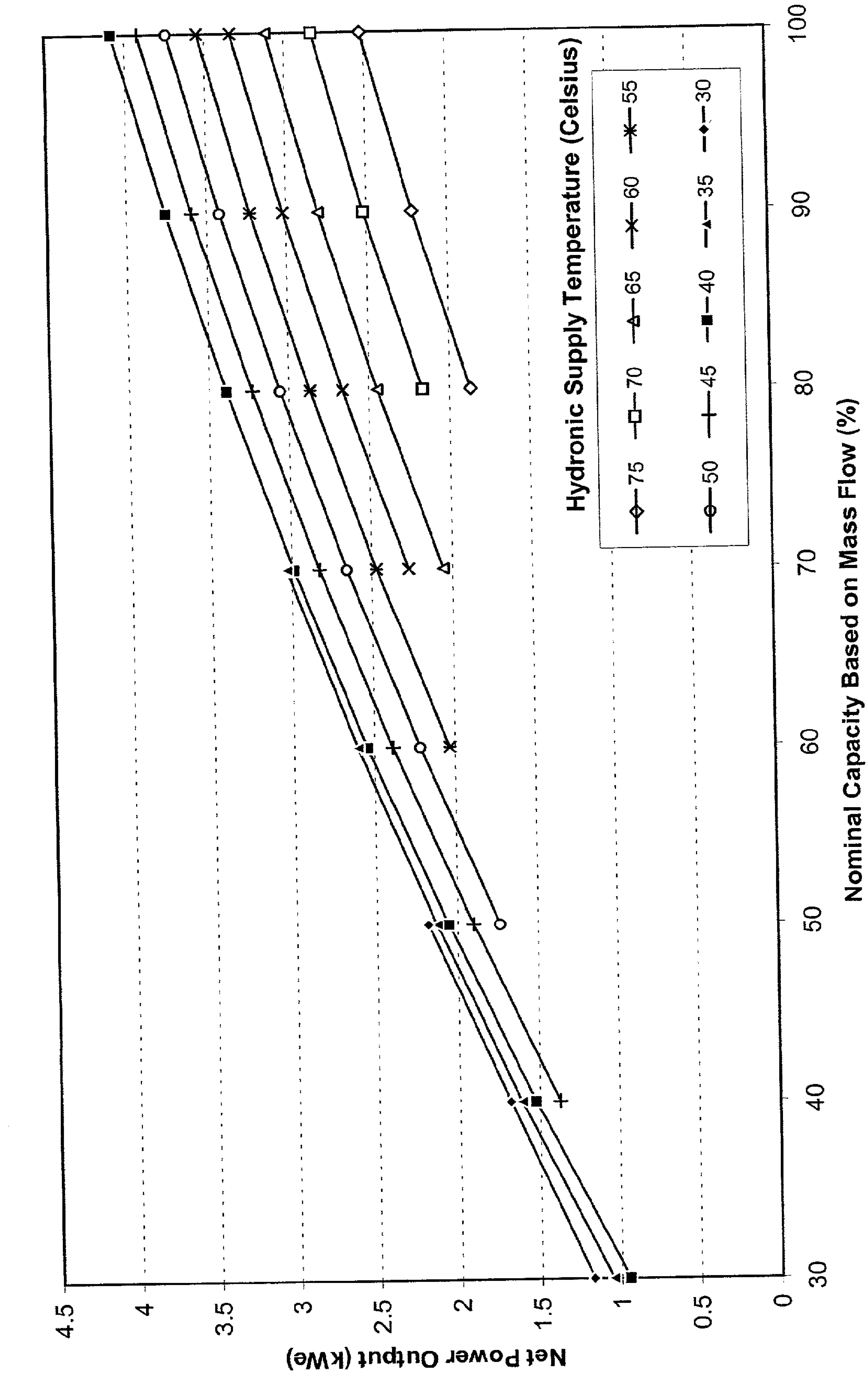

Hydronic Supply Temperature (Celsius)
75
70
65
60
55
50
45
40
35
30
Net Power Output (kWe)
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
Nominal Capacity Based on Mass Flow (%)
30
40
50
60
70
80
90
100

FIG. 7

PROCESS AND DEVICE FOR CONTROLLING THE THERMAL AND ELECTRICAL OUTPUT OF INTEGRATED MICRO COMBINED HEAT AND POWER GENERATION SYSTEMS

BACKGROUND OF THE INVENTION

[0001] The present invention relates to cogeneration systems and, more particularly, to microprocessor-based control sub-systems for controlling the thermal and electrical output of integrated micro-combined heat and power generation systems used to supply domestic electrical power, domestic space heating (SH) water, and domestic hot water (DHW).

[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. Most present-day CHP systems tend to be rather large, producing heat and power for either a vast number of consumers or large industrial concerns. Traditionally, the economies of scale have prevented such an approach from being extrapolated down to a single or discreet number of users. However, increases in fuel costs have diminished the benefits of centrally-generated power. Accordingly, there is a potentially great market where large numbers of relatively autonomous, distributed producers of heat and electricity can be utilized. For example, in older, existing heat transport infrastructure, where the presence of fluid-carrying pipes is pervasive, the inclusion of a system that can provide CHP would be especially promising, as no disturbance of the adjacent building structure to insert new piping is required. Similarly, a CHP system's inherent multifunction capability can reduce structural redundancy.

[0003] 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 single-family residential and small commercial sites provide heat for both SH (such as a hydronic system with radiator), and DHW (such as a shower head or faucet in a sink or bathtub), via demand or instantaneous system. Existing combination units are sometimes used, where heat for DHW is accumulated in a combination storage tank and boiler coil. In one configuration, SH water circulates through the boiler coil, which acts as the heating element for the water in the storage tank. By way of example, since the storage capacity required for instantaneous DHW supplying one to two showers in a single family residence (such as a detached house or a large apartment) is approximately 120 to 180 liters (roughly 30 to 50 gallons), it follows that the size of the storage tank needs to be fairly large, sometimes prohibitively so to satisfy thermal requirements of up to 25 kilowatts thermal (kWt) for stored hot water to meet such a peak shower demand. However, in newer and smaller homes there is often inadequate room to accommodate such storage tank volume. In addition to the need for instantaneous DHW capacity of up to 25 kWt, up to 10 kWt for SH is seasonally needed to heat an average-sized dwelling.

[0004] Furthermore, even in systems that employ SH and DHW in a single heating system to consolidate spacing, no provision for CHP is included. In the example given above, it is likely that the electrical requirements concomitant with the use of 35 kWt will be between 3 and 5 kilowatts electric (kWe). The traditional approach to providing both forms of power, as previously discussed, was to have a large central electricity generating station provide electricity on a common grid to thousands or even millions of users, with heat and hot water production capacity provided at or near the end-user on an individual or small group basis. Thus, with the traditional approach, the consumer has not only little control over the cost of power generation, as such cost is subject to prevailing rates and demand from other consumers, but also pays more due to the inherent inefficiency of a system that does not exploit the synergism of using otherwise waste heat to provide either additional electric generation or heating capacity.

[0005] Large-scale (in the megawatt (MW) range and up) cogeneration systems, while helpful in reducing the aforementioned inefficiencies of centrally-based power generation facilities, are not well-suited to providing small-scale (below a few hundred kW) heat and power, especially in the small-scale range of a few kWe and below (micro-based systems) to a few dozen kWe (mini-based systems). Much of this is due to the inability of the large prime mover systems to scale down, as reasonable electrical efficiency is often only achieved with varying load-responsive systems, tighter dimensional tolerances of key components, and attendant high capital cost. Representative of this class are gas turbines, which are expensive to build for small-scale applications and which sacrifice efficiency when operating over varying electrical load requirements. Efficiency-enhancing devices, such as recuperators, tend to reduce heat available to the DHW or SH loops, thus limiting their use in high heat-to-power ratio (hereinafter Q/P) applications.

[0006] A subclass of the gas turbine-based prime mover is the microturbine, which includes a high-speed generator coupled to power electronics. This microturbine could be a feasible approach to small-scale cogeneration systems. Other shortcomings associated with large-scale CHP systems stem from life-limited configurations that have high maintenance costs. This class includes prime movers incorporating conventional internal combustion engines, where noise, exhaust emissions, lubricating oil and spark plug changes and related maintenance and packaging requirements render use within a residential or light commercial dwelling objectionable. This class of prime mover also does not reject a sufficient amount of heat for situations requiring a high Q/P, such as may be expected to be encountered in a single family dwelling. Other prime mover configurations, such as steam turbines, while generally conducive to high Q/P, are even less adapted to fluctuating electrical requirements than gas turbines. In addition, the steam-based approach typically involves slow system start-up, and high initial system cost, both militating against small-scale applications.

[0007] In view of the limitations of the existing art, the inventors of the present invention have discovered that what is needed is an autonomous system that integrates electric and heat production into an affordable, compact, efficient and distributed power generator, and the ability to control the thermal and electrical output of such a system.

SUMMARY OF THE INVENTION

[0008] The above mentioned needs are meet by providing microprocessor-based control sub-systems which control the thermal and electrical output of integrated micro-combined heat and power generation (M-CHP) systems used to supply domestic electrical power, domestic space heating (SH) water, and domestic hot water (DHW). The M-CHP system uses a microprocessor controller to control the internal operating conditions, such as pump speeds, gas flow rate, and evaporator outlet temperature. Controlling these parameters enables setting the capacity of the system at any instant in time, thereby permitting load following, using a variable capacity operation. The controller also monitors a number of additional safety controls and system protection devices, such as relays/contactor of the alternator to grid, and electrical trips to the feed pump, the oil pump, the hydronic pump, the blower, the gas valve, the expander bypass valves, and other electrically powered devices in the system.

[0009] Overall, system capacity control emphasizes simplicity, reliability, and low cost. For the most part, the most basic control system is a single flow rate system with on/off control from a space thermostat, wherein the instantaneous capacity of a hydronic heating system of M-CHP system is fixed. In a more preferred system, working fluid flow rate and system thermal capacity are ultimately determined by the processor reading the outdoor ambient temperature. The outdoor ambient temperature is used to determine a set-point for the supply (hot-water) temperature in the hydronic heating system. The controller uses a look-up table or algorithm to establish the set-point according to a linear scale, such as varying the set-point linearly from 25° C. at an outdoor temperature of 20° C. to 75° C. at an outdoor temperature of −20° C. The controller uses this set-point to operate the M-CHP in a variable capacity mode by modulating the pump flow rate to maintain the actual hydronic supply temperature at the desired set-point.

[0010] Additionally, the controller coordinates the burner fuel flow rate (heat input rate) with the feed pump flow rate to provide optimal thermodynamic performance by preventing liquid from entering the expander. This is accomplished by maintaining a fixed evaporator exit temperature, such as, for example, 310° F. Furthermore, the controller controls the hydronic pump of the heating system at a speed that keeps the pressure difference between the supply and return headers of the condenser at a preselected value for optimal thermodynamic performance of the hydronic heating system.

[0011] In normal operation the M-CHP system capacity will vary as needed to maintain the supply temperature at its set-point, wherein the burner operates to provide the needed heat input. System mass flow varies essentially with pump speed, but since the expander operates at nearly a fixed speed, the volume flow of high-pressure vapor is essentially constant. Thus, the evaporator pressure varies with load, allowing the vapor density at the evaporator exit to vary to maintain a constant volume flow at each mass flow setting. Preferably, a variable frequency drive is used for the feed pump motor. However, a motor with discrete speed steps may also be used, as well as any other variable speed or flow approach. In normal operation the system operates at higher capacity to attain the higher hydronic supply temperatures required by colder weather. With the higher hydronic supply temperature, the capacity of the indoor radiators/convectors increases to handle the added heat flow. After M-CHP shut down, the hydronic pump may run on to deliver residual heat in the hydronic system to the radiators and maintain the flow of valid control data to the controller from the hydronic system.

[0012] In one embodiment, coordinating the flow rates consists of open loop speed control of the feed pump with an induction motor. Proportional integral differential (PID) control of the boiler gas flow to maintain an outlet superheat between about 20° F. and about 30° F. using a closed loop control system may also be provided. A liquid condensate reservoir ahead of the feed pump and the liquid level is maintained above a predetermined level to ensure adequate net positive suction head (NPSH) to the pump. Preferably, no active control is necessary as a sufficient minimum level can be maintained under all operating conditions by filling the loop with a minimum refrigerant quantity for most operating conditions. However, if for certain operating conditions, the liquid level at the pump inlet cannot be maintained within a desired range, then an active means of maintaining liquid level control may be used. For example, level sensing at the liquid reservoir along with some means of speed control of the feed pump motor may be used.

[0013] High and low side pressures float depending mainly on boiler heat input and hydronic loop return temperature to the condenser. Additionally, no active control of the power module speed is necessary. Connection of the generator to the grid loads the alternator sufficiently to limit the speed to about 3150 RPM (50 hz application). However, should the controller sense a power module overspeed (e.g., loss of grid connection, by sensing, for example, the rate of change of frequency), it will quickly command the opening of a bypass loop around the expander. Preferably, the sensor to detect expander overspeed is built into the power module.

[0014] Typically, it is unnecessary for the controller to control the generator output as long as the generator output is less than the dwelling load. However, the controller can be programmed to monitor and deal with the situation when that generator output does exceeds dwelling load. In particular, the controller can be programmed to switch the extra power on the public grid or use the extra power for a resistance heater to preheat boiler feed.

BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

[0015] 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:

[0016] **FIG. 1** shows a schematic diagram of an integrated micro-CHP system including a control subsystem according to the present invention;

[0017] **FIG. 2** shows a schematic diagram of a PLC control subsystem according to one aspect of the present invention;

[0018] **FIG. 3** is a start-up/operating flow chart of the operating logic of one embodiment of the integrated micro-CHP system according to the present invention;

[0019] **FIG. 4** is a shutdown flow chart of the operating logic of the integrated micro-CHP system according to one embodiment of the present invention;

[0020] **FIG. 5** shows a schematic diagram of a sensor that may be used by a control subsystem of one embodiment of the present invention to sense the saturation pressure of an organic working fluid of an integrated micro-CHP system;

[0021] **FIG. 6** is a chart showing measured outside ambient temperature versus desired hydronic set point; and

[0022] **FIG. 7** is a chart showing net power output at various hydronic supply temperatures, which varies approximately linearly with mass flow.

DESCRIPTION OF THE PREFERRED EMBODIMENT

[0023] With reference to the figures, an embodiment of the invention is illustrated incorporated into a number of the major components of an illustrated organic rankine cycle (ORC) micro-cogeneration system used for the supply of domestic electrical power, domestic space heating (SH) water, and domestic hot water (DHW). Fluid supply lines connecting these components together are not drawn for ease of illustration. However, it is to be understood that the present control subsystems and control method of the present invention may advantageously be used with a number of micro-cogeneration systems. Exemplary micro-cogeneration systems are disclosed by co-owned U.S. patent application Ser. No. 09/998,705 filed Nov. 30, 2001, the entire disclosure of which is herein incorporated fully by reference.

[0024] Referring initially to **FIG. 1, a** number of system components of an exemplary micro-CHP system **100** is schematically shown, such as an expander **101**, a condenser (heat exchanger) **102**, feed pump **103**, heat source (boiler) **104**, and evaporator **105**. The expander **101** may be of any design, such as for example, a positive displacement expander and a scroll expander. In a direct fired system, the feed pump **103** circulates an organic working fluid (such as naturally-occurring hydrocarbons or halocarbon refrigerants, not shown) through a loop at least defined by the fluidly-connected expander **101**, condenser **102**, feed pump **103** and evaporator **105**. In an indirectly fired system, the feed pump **103** circulates the working fluid through a first or inner loop, and at least defined by fluidly-connected expander **101**, condenser **102**, feed pump **103**, and a first loop portion of evaporator (heat exchanger) **105**, which is illustrated by dashed-line **105***a*. In such an arrangement, the first loop portion or interloop portion **105***a* receives heat from a second loop portion of the evaporator or heat pipe which is directly heated by heat source **104**. More detailed information is provided in co-pending U.S. patent application Ser. No. 09/998,705 regarding directly and indirectly micro-CHP system arrangements, to which reference is made.

[0025] Optionally, a desuperheater **106** may be included between the outlet of the expander **101** and the inlet of condenser/heat exchanger **102**. In this alternative embodiment, a regulated by-pass valve **107** is used to circulate the working fluid exiting feed pump **103** through the coil loops (not shown) of the desuperheater **106** extracting heat energy from the vaporized working fluid exiting expander **101** before being circulating into evaporator **105** or interloop portion **105***a,* if so configured. Since the size of the desuperheater is fixed, the amount of heat removed from the working fluid vapor exiting expander **101** will vary with working fluid flow rate and temperature as the system modulates. However, it is to be appreciated that controllable by-pass valve **107** permits the ability to control the flow of liquid working fluid through the desuperheater. By controlling the liquid working fluid flow, the amount of de-superheating is controllable, and therefore the heat-to-power ratio of the micro-CHP is controllable.

[0026] When maximum thermal output is required, bypassing the desuperheater **106** will divert all the thermal energy to the hydronic fluid in condenser **102**. When maximum electrical power output is desired, directing all the liquid refrigerant coolant first to the desuperheater **106** will remove the maximum amount of energy from the working fluid vapor in the desuperheater prior to it entering the condenser **102**. By controlling the liquid working fluid flow rate between zero and 100 percent mass flow to the desuperheater **106**, the thermal and electrical outputs can be tailored to better match the space heating load and the electrical power load of the house or building.

[0027] A generator **108** (preferably induction type) is coupled to expander **101** such that motion imparted to it by expander **101** generates electricity. While the expander **101** can be any type, it is preferable that it be a scroll device. The scroll expander can be a conventional single scroll device, as is known in the art. The generator **108** is preferably an asynchronous device, thereby promoting simple, low-cost operation of the system **100**, as complex generator speed controls and related grid interconnections are not required. An asynchronous generator always supplies maximum possible power without controls, as its torque requirement increases rapidly when generator **108** exceeds system frequency. The generator **108** can be designed to provide commercial frequency power, 50 or 60 Hz, while staying within close approximation (often 150 or fewer revolutions per minute (rpm)) of synchronous speed (3000 or 3600 rpm). The load on the expander **101** imposed by the grid ensures that mechanical speeds in the expander **101** are kept within its structural limits.

[0028] Inherent in a micro-CHP (cogeneration) system is the ability to provide heat in addition to electricity. Excess heat, from both the heat source **104** and the expanded working fluid, can be transferred to external DHW and SH loops. The nature of the heat exchange process is preferably through either counterflow heat exchangers (e.g., for either or both the DHW and SH loops), or through a conventional hot water storage tank (e.g., for a DHW loop). In one embodiment, for example, a simple SH loop **109** may include fan **110** which provides for the drawing and blowing of air heated by heat source **104** to a space **111**. For DHW services, an external heating loop **112** (shown partially) may be coupled to condenser **102**. As an option, a preheat coil (not shown) can be inserted into the external heating loop **112** such that the hydronic fluid (typically water) flowing therethrough can receive an additional temperature increase by virtue of its heat exchange relationship with the heat exchange fluid flowing through a second circuit (not shown).

[0029] The hydronic fluid flowing through external heating loop or DHW system **112** is circulated with a conventional hot water pump **113**, and is supplied as space heat via a radiator or related device (not shown) to, for example, space **111**. As an example, hydronic fluid could exit the condenser **102** at about 50° Celsius and return to it as low as 30° Celsius. It will be appreciated by those of ordinary skill in the art that while the embodiments depicted in the figures show separate DHW and SH heat exchangers, it is within the spirit of the present disclosure that series, parallel, sequential and/or the same heat exchange configurations could be used. Additionally, although the capacity of the system **100** is up to 60 kW_t if desired, larger or smaller capacity units could be controlled by the herein disclosed control subsystem and method of the present invention.

[**0030**] Combustion chamber **114**, enclosing heat source **104**, includes at least a flumed exhaust duct **115** to the outside of the building, and an exhaust gas fan **116**. Heat at least to the evaporator **105** is provided by heat source or burner **104**, which is supplied with fuel by a gas train **117** having a shut off valve **118** and variable flow gas valve **119**.

[**0031**] Other devices may be used with the exhaust duct **115**, such as an exhaust gas recirculation device with an exhaust duct heat exchanger (not shown) which can be used to improve the thermal efficiency of the system **100** by lowering the temperature of the exhaust gas that is pulled away and vented to the atmosphere by fan **116**. The heat given up by the exhaust gas in an exhaust gas heat exchanger may be used to provide additional heat to other parts of the system **100**. More detailed information is provided by copending U.S. patent application Ser. No. 09/998,705 regarding exhaust gas recirculation devices with or without an exhaust duct heat exchangers, to which reference is made.

[**0032**] Block valve **120** and bypass valve **121** are situated in the organic working fluid flow path defined by piping **122** (of which bypass conduit **123** is part). In response to a no load condition (such as a system start up/shut down or grid outage) on the system, the superheated vapor in piping **122** is permitted to bypass around expander **101** through bypass conduit **123**, thereby avoiding overspeed of expander **101**. In this condition, the rerouted superheated vapor is fed into the inlet of condenser **102** or if so configured, desuperheater **106**. Under normal operating conditions, where there is a load on the system, the superheated vapor enters the expander **101**, causing the orbiting involute spiral wrap to move relative to the intermeshed fixed involute spiral wrap. As the superheated vapor expands through the increasing volume crescent-shaped chambers of the expander **101**, the motion it induces in the orbiting wrap is transferred to the generator **108**, via a coupled shaft or an integral rotor/stator combination on the expander **101**.

[**0033**] Controller Sub-Systems

[**0034**] As illustrated by **FIG. 2**, provided are controller sub-systems, generally indicated by **124**, to control the thermal and electrical output of integrated micro-combined heat and power generation (M-CHP) systems, such as illustrated by **FIG. 1**. The controller sub-system **124** are microprocessor-based, wherein the M-CHP system **100** uses a microprocessor controller **125**, which is preferably a programmable logic controller (PLC) or alternatively, any other conventional microcomputer, to monitor and control the operating conditions of system **100**. It is to be appreciated that a PLC based subsystem simplifies setup and troubleshooting and may accommodate programming modifications from future developments and refinements to such micro-CHP systems. In a preferred embodiment, the controller is an Allen Bradley MicroLogix 1500 PLC with a 1764-LRP Processor and includes other components, such as for example, shown below in Table 1 for it operation, wherein the controller **125** is programmed using conventional programming software and techniques. In a preferred embodiment, the controller **125** ladder logic may be programmed with Allen Bradley RS Logix 500 software, wherein a graphical maintenance program running Allen Bradley RS View on a laptop PC can be used for troubleshooting and setup in the field.

[**0035**] The controller subsystems **124** further include a plurality of process control sensors **126**, a plurality of control outputs **127**, a protection relay subsystem **128**, a system trip reset **129**, and the necessary user and service personnel interfaces, such a maintenance subsystem **130** and a data acquisition and communications subsystem **131**. Since the user and service personnel interfaces are conventional, for brevity no further discussion is provided.

[**0036**] For the purpose of controlling and monitoring the operation of cogeneration system **100**, the plurality of process control sensors **126** and control outputs (I/O devices) **127** are individually called out in Table 2 and shown in **FIG. 1**, by sensors S_0-S_{19}, control points A_1-A_{11}, and protection relays M_1-M_4, for the system at various points therewithin. As illustrated in **FIG. 2**, the sensors and I/O devices shown in Table 2 are interfaced with the controller **125** for control of the micro-CHP system **100**. Additionally, the controlled and monitored parameter of sensors S_0-S_{19} and control points A_1-A_{12} are also listed in Table 2, as well as the preferred interface and type for each.

TABLE 1

Component	Mfg Part No	Description
Processor	1764LRP	MicroLogix 1500 Processor
Base	176424BWA	MicroLogix 1500 Base 24 I/O
Analog Voltage Input Module	17691F4	Analog Voltage Input Module
Analog Voltage Output Module	17690F2	Analog Voltage Output Module
Right end cap terminator	1769ECR	Right end cap terminator
Cable	1761CBLPM02	Cable
Thermocouple Input Module	1769-IT6	1769-IT6 Thermocouple Input Module
Real time clock	1764-RTC	Real time clock
Remote access modem	MICRORAD	Remote access modem

[**0037**] The information obtained from sensors S_1-S_{19} is used by the controller **125** to provide detailed system control through actuation and/or modulation of the various control points A_1-A_{11}. Such controlled and monitored parameters include pump speeds, gas flow rate, inlet and outlet temperatures and/or pressures to, from, and within the expander **101**, condenser **102**, evaporator **105**, combustion chamber **114**, desuperheater **106**, and throughout various points of the space heat, domestic hot water, and generation loops **109**, **112**, and **122**, respectively. Controlling and monitoring at least three of the above mentioned parameters (as will be discussed in later sections) enables setting the capacity of the system **100** at any instant in time, thereby permitting load following, using a variable capacity operation.

[0038] All of the pumps and fans **103**, **110**, **113**, and **116**, are responsive to input signals from controller **125** via the their associated control points, namely A_{10}, A_8, A_7, and A_{11}, respectively. The controller **125** uses appropriate program logic, as will be explained in a later section, to control the turning on/off the fans, running and varying the speeds of the pumps, and also opening and closing valves, such as for example, by-pass valves **107**, **121**, shut off valves **118** and **120**, and variable flow gas valve **119**, in response to predetermined conditions. Such predetermined conditions include the demand for maximum heat or power output and an electric grid outage. Valves **107**, **118**, **119**, **120**, and **121** are responsive to input signals from controller **125** via the their associated control points, namely A_4, A_1, A_6, A_3, and A_2, respectively.

[0039] The controller **125** also monitors a number of additional safety controls and system protection devices, such as relays/contactor **132** (**FIG. 1**) of the alternator to grid having monitors **M1-M4** listed in Table 2. Additionally, the controller **125** monitors and reports the electrical trips to the feed pump, an oil pump, the hydronic pump, the blower, the gas valve, the expander bypass valves, and any other electrically powered device in the system. Furthermore, the controller may monitor and activate a contactor of a crank-case heater for the power module via control point A_9 and switch the output of the generator **108** to the power grid via control point A_5.

[0040] In addition to controlling and monitoring system operations, the controller **125** possesses functions to optimize the performance, efficiency, and safety the system **100**. The functions of the controller **125** include automatic start up and shut down, modulation control of hydronic supply temperature, monitoring and tripping on safety and abnormal operating parameters, and electrical grid connection interface. With reference also to FIGS. **3-4**, these functions of the controller **125** are explained in further detail.

[0041] **FIG. 3** is a flow chart of the operating logic of the system **100**. In operation, the controller **125** in one embodiment is programmed to operate the system in a quasi-steady state in response to a need for heat that is keyed to a specified hydronic supply temperature set point. This automatic operation is a heat load following mode. In other embodiments, such a call for heat, for example, may be from a thermostat, such as sensor S_0, which demands heat when the space temperature falls below a user set-point, or according to an on-off timer, used for overnight shutdown. In the latter case, a timer would enable the thermostat to signal the controller **125** to initiate startup.

[0042] In the heat load following mode, the call for heat occurs whenever the controller **125** determines that the temperature differential between the actual hydronic supply temperature of the DHW system **112**, sensed via sensor S_2, and a desired hydronic supply temperature is greater the a predetermined value, such as, for example 0.5 to 5° Celsius.

TABLE 2

Sensor-I/O Device	Parameter	PLC Interface	Type
S_0	Space Temperature	Digital Input	Thermostat Or Analog Signal Thermometer
S_1	Hydronic Condenser Inlet Temperature	Thermocouple Input	Thermocouple
S_2	Hydronic Supply Temperature	Thermocouple Input	Thermocouple
S_3	System Trip	Data Logger	Error Signal
S_4	Expander Inlet Temperature	Thermocouple Input	Thermocouple
S_5	Expander Inlet Pressure	Analog Input	Pressure Transducer (0–500 Psi)
S_6	Feed Pump Inlet Temperature	Thermocouple Input	Thermocouple
S_7	Feed Pump Inlet Pressure	Analog Input	Pressure Transducer (0–200 Psi)
S_8	Power Module Power Output	Digital Input	Amp/Watt Meter
S_9	Condenser Internal Pressure	Analog Input	Pressure Transducer (0–200 Psi)
$S_{10\text{ or }10'}$	Evaporator Coil Temperature	Thermocouple Input	Thermocouple
S_{11}	Gas Flow	Digital Input	Flow Rate Meter
S_{12}	Expander Outlet Temperature	Thermocouple Input	Thermocouple
S_{13}	Feed Pump Set-point	Digital Input	Flow Rate Meter
S_{14}	Heat Source High Limit	Thermocouple Input	Thermocouple
S_{15}	Flame Detection	Digital Input	Electro Optical
S_{16}	High Feed Level	Digital Input	Level Indicator
S_{17}	Low Feed Level	Digital Input	Level Indicator
S_{18}	Outside Temperature	Digital Input	Thermostat Or Analog Signal Thermometer
S_{19}	Hydronic Fluid Flow	Digital Input	Flow Rate Meter
A_1	Main Gas Valve	Relay Output	Solenoid Valve Actuator
A_2	Expander Bypass	Relay Output	Solenoid Valve Actuator
A_3	Expander Shutoff	Relay Output	Solenoid Valve Actuator
A_4	Desuperheater Bypass	Relay Output	Solenoid Valve Actuator
A_5	Power Module Run/Stop	Relay Output	Power Module Contactor
A_6	Evaporator Bumer Firing Rate	Analog Output	Gas Proportioning Valve Actuator
A_7	Hydronic Pump Run And Speed	Relay Output Analog Output	Pump Variable Frequency Drive
A_8	Space Heat Fan Run/Stop	Relay Output	Motor Contactor
A_9	Crank Case Heater On/Off	Relay Output	Crank Case Heater Contactor
A_{10}	VFD - Feed Pump Run And Speed	Relay Output Analog Output	Pump Variable Frequency Drive
A_{11}	Forced Draft Fan Run/Stop	Relay Output	Motor Contactor
M_1	Net Failure	Digital Input	Protection Relay
M_2	Over/Under Voltage	Digital Input	Protection Relay
M_3	Over/Under Frequency	Digital Input	Protection Relay
M_4	Power Module Internal Temperature	Digital Input	Digital Thermometer

[0043] It is to be appreciated that the hydronic pump **113** operates continuously so there is always a flow through the DHW system **112** enabling the controller **125** to continuously monitor the actual supply temperature and return temperatures, via sensors S_2 and S_1, respectively. In other embodiments, the controller **125** may use the internal temperature of the condenser **102** and hydronic flow rate, via speed control of hydronic feed pump **113**, to determine a hydronic supply temperature using predetermined heat exchange values and/or algorithms.

[0044] The desired hydronic supply temperature is set by the controller **125** sensing the outdoor ambient temperature, via thermostat S_{18}, and correlating the sensed outdoor ambient temperature to a desired hydronic supply temperature for the boiler. The correlation between outdoor ambient temperature and desired hydronic supply temperature may be, for example, according to the illustrated linear relationship shown by **FIG. 6**. In this illustrative embodiment, for example, on cold days, say −20° C. ambient, the hydronic set point is 75° C., but on warm days, when only a little heat is needed, i.e., 20° C. ambient, the set point is 25° C., and at a 0° C. ambient the set point is 50° C. However, for other embodiments other scalar relationships between outdoor ambient temperature and the desired hydronic supply temperature may be used such as, for example, logarithmic, exponential, and other nonlinear functions. To avoid the influence of sunshine on cold days, a single measuring point on a north facing side of a building or home should be used for thermostat S_{18}.

[0045] Start-Up

[0046] In response to the call for heat in step **200**, the controller **125** is programmed to modulate the thermal output of the evaporator **105** in order to have the actual hydronic supply temperature match the desired hydronic set point for the given outdoor ambient temperature. Accordingly, the controller **125** in step **202** determines whether the evaporator **105** is already at its operating temperature via thermocouple sensor S_{10} or S_{10}, (**FIG. 1**). Assuming the temperature of the evaporator **105** is below the minimum operating temperature (e.g., cold start-up), the controller **125** verifies, and if necessary closes the expander shutoff valve **120** and opens the expander bypass valve **123** as shown in steps **204** and **206**, respectively. Once verified, the controller **125** checks to see if the burner is on via flame detector S_{15} and if not, purges the combustion chamber **114** with force draft blower **116** and activates the burner **104**, as shown at blocks **208** and **210**.

[0047] The burner **104** is activated by controller **125** opening the gas valve **118** and metering its flow with flow rate valve **119**, via actuator A_1, to nominally between 40 and 80% of full flow when ignition is expected. An igniter (not shown) is turned off by a timer and the gas valve is turned off when no flame is proven via flame detector S_{15}. The burner **104** will then typically be set to come on to about 50% of its capacity to warm up system **100**. If desired, an off-the-shelf combustion controller can initiate burner firing upon receiving an enabling signal from the controller **125**. If so arranged, the combustion controller will provide flame and combustion air detection, light the burner, and provide high temperature limit protection for the evaporator **105**. The controller also drives a variable speed draft fan to provide the approximately correct air flow rate for the burner.

[0048] Next, the controller **125** checks to see if the evaporator working fluid level is in the desired operating range, via level sensors S_{16} and S_{17} in step **212** and if not, in step **214** will activate feed pump **103**. It is to be appreciated that both burner **104** firing and feed pump **103** flow may be controlled in part, and conventionally by room temperature and its user determined set-point, as well as outdoor temperature, via sensor S_{18}. Additionally, feed pump **103** comes on to a speed predetermined by the controller **125** to coincide with the flow requirements established by the initial burner firing rate and the design response of the system **100**. In particular, the controller **125** runs feed pump **103** fast enough to keep the organic working fluid liquid level between level low sensor S_{17} and level high sensor S_{16}. Accordingly, the feed pump is turned on by the controller **125** at the appropriate time to keep the evaporator filled, but not over-filled with working fluid.

[0049] When the system is operating, superheated working fluid is moving past sensor S_4, which is able to provide a valid signal to the controller **125** so the heat source or burner **104** firing rate and feed pump **103** flow can be adjusted for both the safe operation and needed output. However, when the system is just starting, the controller **125** must be given some initialized state which can be used as a safe operating condition until such time as working fluid is flowing past temperature sensor S_4.

[0050] It is desirable to have a minimum amount of working fluid flow during startup, so that the fluid heats up as rapidly as possible. However, some flow is needed to prevent local overheating of the fluid in evaporator **105**, and to provide the controller **125** with an indication that the burner **104** is indeed firing. Accordingly, the burner gas rate is set to provide the longest possible run time for the system, consistent with measured outdoor temperature and rate of change of indoor temperature. Feed pump **103** operates to keep the evaporator **105** supplied with the working fluid at the factory-preset value for temperature sensor S_4. When temperature sensor S_4 gets to about 50% of the thermostat set-point, feed pump **103** speed is increased until the temperature reading in temperature sensor S_4 reaches its set-point, at which time the burner **104** modulates for constant values of at least temperature sensor S_4, and the feed pump speed is modulated to maintain the desired hydronic supply temperature.

[0051] The controller will abort the start up sequence of steps **202-214**, if burner **104** fails to heat the evaporator **105** or if any of the parameters listed in Table **3** are exceeded. If the startup sequence fails, the controller **125** will attempt to re-start again for a maximum of three re-start attempts. If three re-start attempts are completed with no successful re-start, the controller **125** will be locked out from re-starting until manually reset, via trip reset **129** (**FIG. 2**). The re-start attempt counter will reset after a successful start or a manual reset.

[0052] After the startup sequence is complete the system **100** will go into run mode, wherein working fluid pressure is allowed to build and feed pump **103**, burner **104**, and evaporator **105** are controlled by controller **125** via two separate feed back loops.

[0053] At the appropriate time, the expander **101** is connected to receive the now superheated vapors of the working fluid, wherein the controller **125** in step **216** opens the cutoff or block valve **120** to expander **101** and after a short delay (i.e., 1 second), closes the bypass valve **121**. It is to be appreciated that liquid is prevented from entering into the expander by maintaining the evaporator's working fluid exit temperature at a fixed value for all operating conditions. As mentioned previously, in one embodiment, the evaporator exit temperature is set to operate at about 154° C. (310° F.), which has been found to give good overall system efficiency regardless of system load.

TABLE 3

Parameter	Threshold	Fault Response
Expander inlet temperature	340° F. (171.1° C.)	Full shut down
Expander inlet pressure	420 psia (29.5 Kg/cm)	Full shut down
Feed pump inlet temperature	200° F. (93.3° C.)	Full shut down
Feed pump inlet pressure	200 psia (14 Kg/cm)	Full shut down
Protection relay trip	Suitable trip values	Power module shut down only
Power module temperature switch	300° F. (148.9° C.)	Power module shut down only
Startup sequence failure	3 attempts	Full shut down

[**0054**] Further protection to the scroll expander is provided during start-up and while running by controlling the displacement of the feed pump, which in one embodiment is by feed pump speed, wherein the rate of change in feed pump speed is limited by an output ramp in the controller logic which provides time for the evaporator PID control to modulate the evaporator burner for steadier operation and to prevent over temperature of the expander inlet. In this control arrangement, a negative feed forward value will be added to the feed pump speed control variable when ramping down to better control temperature of the expander inlet.

[**0055**] The system **100** will warm up and quickly come to a near-steady-state operating point for the feed pump flow setting. After a short start-up delay (i.e., 1 second) in step **218** wherein the controller verifies that the output of the generator is within set parameters via sensor S_8, the electrical output is then connected to the grid in step **220** via contactor **132** responding to control signal A_5 from the controller **125**. It is to be appreciated that there is no need to control the generator output as long as the generator output is less than the dwelling load. Preferably, connection of the generator **108** to the grid loads the alternator sufficiently to limit the speed to about 3150 RPM (50 hz application).

[**0056**] However, should the controller **125** sense a power module overspeed (e.g., loss of grid connection), it will quickly command the opening of a bypass loop **123** around the expander **101**. Preferably, the sensor to detect expander overspeed is built into the power module. Furthermore, the controller **125** can be programmed with at least two options for dealing with the case when the generator output exceeds the dwelling load. The controller **125** can put the extra power on the grid or the extra power can be used for a resistance heater at condenser **102**.

[**0057**] Additionally, it is to be appreciated that the power module **108** will trip on over/under voltage, over/under frequency and loss of mains conditions detected by the protection relay **128** (**FIG. 2**). To prevent nuisance tripping of the heating system, only the power module **108** will trip on voltage and frequency grid disturbances. The controller **125** will log the fault and re-start the power module **108** after the over/under voltage or over/under frequency condition has dissipated. If a loss of mains occurs, the entire system **100** will shut down with all latches/relays reset to start ready condition. The controller **125** will then automatically re-start on the restoration of power if heat is called for by the thermostat/timer conditions.

[**0058**] In step **222**, the controller checks to see if the call heat has been satisfied via continuously reading sensor S_0 and/or S_{18}. If the hydronic supply temperature is not at the desired step point as checked in step **224**, the controller **125** in step **226** reads the outdoor ambient temperature via sensor S_{18} and uses a look-up table or algorithm to establish the desired set-point according to the linear relationship illustrated by **FIG. 6**. The controller **125** will then use the continuously updated desire set-point to operate the evaporator **105** in a variable capacity mode by modulating the gas valve **119** on the burner **104** to maintain the actual hydronic supply temperature, sensed via sensor S_2, at the desired set-point.

[**0059**] In step **228**, the controller **125** checks to see whether the expander inlet temperature via sensor S_4 is in a desired operating range. If not, then in step **230** the controller **125** modulates the burner fuel flow rate (heat input rate) to provide optimal thermodynamic performance and to prevent liquid from entering the expander. Additionally, in this quasi-steady state the controller **125** may control the hydronic pump **113** of the heating system at a speed that maintains the pressure difference between the supply and return headers of the condenser at a preselected value for optimal thermodynamic performance of the hydronic heating system.

[**0060**] With the controller subsystems **124** operating the system **100** in the above described heat load following mode, it is to be appreciated that the system will operate for as many hours as possible during the heating season. The controller subsystems **124** will run the system **100** just hard enough to maintain the hydronic supply temperature at the correct value for the nominal heating load. When the system **100** operates at less than the maximum supply temperature, more power is generated than at the maximum temperature, the controller **125** can automatically and passively maximize the power which can be made and sold.

[**0061**] Further it is to be appreciated that in the above described heat load following mode, the greater the temperature differential error, the larger the thermal output response that will be initiated by controller **125** in order to achieve the desired hydronic supply temperature. As illustrated in **FIG. 7**, since the thermal output of the system varies approximately linearly with the mass flow of the system **100**, the controller **125** controls its thermal response by controlling both the displacement of the feed pump **103**, via a PID speed control, and the heat input to the evaporator **105**, via adjusting the gas flow rate to the burner **114** to maintain a 310° F. (154.4° C.) temperature into the expander.

[**0062**] In particular, the controller **125** is programmed to increase or decrease the mass flow rate in proportion to the system heating capacity required to match the actual and the desired set point for the hydronic supply temperature. For example, the desired set point of the hydronic supply temperature is 50° C. for a given sensed outdoor ambient temperature of 0° C., if while operating in a quasi-steady state, the actual hydronic supply temperature suddenly drops to 45° C. (e.g., a door or window of the space or building is open to the outdoors), the controller **125** will increase the percentage of system mass flow to meet this heating demand. Should in this example after a predetermined time period the temperature error between the actual and desired hydronic supply temperatures decrease, the controller **125** will further increase the percentage of system mass flow in order to meet this heating demand. Accordingly, it is to be appreciated that in normal operation the system **100** must operate at higher capacity to attain the higher hydronic supply temperatures required by colder weather and with the higher hydronic supply temperature the capacity of the indoor radiators/convectors increases to handle the added heat flow.

[**0063**] In run mode, the controller **125** monitors the operating parameters shown in Table 1 and performs a full or partial shut down if any of the parameters exceed preset thresholds. An individual trip event on the automatic trip settings is defined as a Level 1 trip. In the event of a Level 1 trip, the controller **125** will self reset and automatically attempt to re-start. If the re-start fails, the controller **125** will attempt to re-start again for a maximum of three re-start attempts. If the three re-start attempts are completed with no successful re-start, this case is defined as a Level 2 trip. In the event of a Level 2 trip, an alarm notification message will be sent by controller **125**, via the data acquisition and communications subsystem **131** and the controller **125** will be locked out from re-starting until manually reset, via trip reset **129** (**FIG. 2**). The re-start attempt counter will reset after a successful start or a manual reset.

[**0064**] Shutdown

[**0065**] Normal shutdown of the micro-CHP will occur if the outdoor thermostat signal indicates that the outside temperature has gone above the temperature that corresponds to the lowest hydronic supply temperature set-point and remains for 30 minutes. The system will re-start when the thermostat signal indicates that the outside temperature is below the normal startup temperature. The controller **125** will also perform the normal shutdown sequence when switched off manually by the user or service personnel.

[**0066**] In the former case, when the system load falls below about 30 to about 40% of a full load and 30 minutes has expired, the controller **125** is programmed to shutdown the system **100** and cease making both heat and power to ensure economical use of the m-CHP system. Since the hydronic pump is kept running at all times, even at a low flow rate, the controller **125** continuously monitors 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 starts the system for another on-cycle. Should the controller also sense that during operation of the system at the minimum system mass flow, the actual supply temperature begins to exceed the set point, it is programmed also to shutdown the system when this error exceeds a predetermined value. In either of these conditions of heat load following mode, or upon receiving a heat satisfied message from a space thermostat, if not configured in the heat load following mode, the controller **125** will follow a normal shutdown procedure according to the program control logic illustrated by **FIG. 4**.

[**0067**] In step **300**, normal shutdown begins with the controller **125** turning off burner **103**. After a time delay in step **303** (i.e., 15 seconds), the controller **125** will shutdown (disconnect) the power module **108**, open the bypass valve **121** and close the shutoff valve **120** to the expander **101** in step **304**. After another time delay (i.e., 5 seconds) in step **306**, the controller **125** stops the feed pump **103** in step **308**. After another time delay (i.e., 60 seconds) in step **310**, the controller **125** may slow the hydronic pump **113** to its minimum speed, completing a normal shutdown of the system **100**.

[**0068**] However, in step **300** the time delay may be conditioned on available reservoir heat energy in heating system **112**, wherein if desired, the hydronic pump **113** may run on to deliver residual heat in the hydronic system to the radiators until below a certain set-point. In such a case, the controller **125** will initiate a partial shutdown if the hydronic supply temperature exceeded the set-point by 5° C. for a period of 30 minutes when the hydronic set-point is at the minimum setting. The evaporator **105**, feed pump **103**, expander **101**, and power module **108** will shutdown while the hydronic pump will continues to run. When the hydronic supply temperature error signal reaches −0.5 to 5° C., the system will re-start.

[**0069**] Data will be logged using the logging capability of the controller **125**. Preferably, a remote access modem will interface with the controller **125** to download any system data. The modem may be self-dialing and use either land line or cellular service. In the event of a Level 2 trip, the controller may use the modem to send an alarm notification message to service personnel. Data points logged by controller **125** are shown in Table 4.

[**0070**] Evaporator Heat Control

[**0071**] In normal operation, no control of the evaporator heat rate is required as the ideal heat input rate is set by a given mass flow rate out of the feed pump. If the heat input rate is greater than the ideal rate, then the evaporator outlet fluid will be more superheated than desired. This will lead to increasing evaporator pressure until the density at the expander inlet is sufficient to provide a match between expander and feed pump mass flow rate. Thus, the results of excessive heat input rate will be excessive evaporator pressure and evaporator outlet superheat.

[**0072**] If there is too little evaporator heat input for a given feed pump flow rate, evaporator pressure will be reduced, and some liquid may be admitted to the expander **101**. Liquid admission to the expander **101** is not expected to result in damage to the expander because of the increasing chamber volume at all times. However, partial liquid admission is likely to result in reduced pressure ratio across the expander **101** and, therefore, less thermodynamic work by the expander. Accordingly, to obtain a better match between the evaporator heat input rate and the feed pump flow rate over all operating conditions, better control over the evaporator heat output is desired.

[**0073**] The control concept involves sensing the level of superheat at the evaporator outlet and maintaining this level in the range of 20° F. to 30° F. (−6° C. to −1° C.). This will tend to minimize the evaporator pressure while maintaining some margin from liquid entering the expander **101**. The evaporator pressure will then float until the density at the expander inlet is such that the expander and feed pump flow rates match.

TABLE 4

Channel	Parameter	Signal Type
1	Hydronic condenser inlet temperature	Thermocouple
2	Hydronic condenser outlet temperature	Thermocouple
3	Hydronic fluid flow	Analog
4	Expander inlet temperature	Thermocouple
5	Expander inlet pressure	Analog
6	Feed pump inlet temperature	Thermocouple
7	Feed pump inlet pressure	Analog
8	Power module power output	Analog
9	Evaporator inlet temperature	Thermocouple
10	Gas flow	Analog
11	Expander outlet temperature	Thermocouple
12	Feed pump speed (drive frequency)	Analog
13	Protection relay trip	Digital
14	Level 1 trip failure	Digital
15	Level 2 trip failure (failed to restart)	Digital

[0074] The superheat exiting the evaporator **105** may be sensed by either measuring the evaporator outlet temperature and saturation temperature, measuring the evaporator outlet temperature and outlet pressure, or sensing the saturation pressure. The first approach requires measuring the temperatures of the working fluid in its saturated state at the evaporator inlet with inlet temperature sensor S_{10} (indirectly fired) or S_{10}, (directly fired) and also in its vapor state such as with expander inlet temperature sensor S_4. The difference between the temperatures is then the superheat. For an improved superheat sensing, in addition to the temperature, expander inlet pressure sensor S_5 may be used to sense the evaporator outlet pressure. The controller **101** may then use a look-up table and/or calculation to determine the saturation temperature from the pressure and temperature readings, wherein the superheat is the difference between the measured temperature and the computed saturation temperature. Furthermore, pressure sensor S_5 can also provide a safety feature to protect both the evaporator **105** and the expander **101** from potentially harmful overpressure.

[0075] The third approach is illustrated by **FIG. 5**, wherein the evaporator outlet pressure is applied to one side of a diaphragm **500** of a superheat sensing device **501**. The other side of the diaphragm sees the same working fluid, but from a sensing bulb **502** at the same temperature as the evaporator outlet gas. The pressure of this latter side is the saturation pressure corresponding to the evaporator outlet temperature. If there is positive superheat, the pressure from the sensing bulb **502** will exceed the evaporator outlet pressure and the diaphragm **500** will compress a spring **504** by an amount proportional to the degree of superheat. A potentiometer (or other position sensor) **506** attached to the diaphragm **500** then provides an electrical output to the controller **125** proportional to the degree of superheat. The controller **125** can then use the output sign from the superheat sensing device **501** to minimize the evaporator pressure while maintaining some margin from liquid entering the expander **101**.

[0076] 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. For example, an alternative mode of providing heat following control may be as follows. As before, this alternative heat following mode uses an outdoor temperature sensor (i.e., sensor S_{18}) to determine the set point of the hydronic supply temperature. Controller **125** after sensing the actual hydronic supply temperature (i.e., via sensor S_2), adjusts the burner firing rate, via the gas valve **118** and/or **119**, via actuator A_1, to bring the actual hydronic supply temperature into line with the desired set point. If the actual hydronic supply temperature is too low, the controller **125** will increase the firing rate, and if the temperature is too low, the controller **125** will decrease the firing rate.

[0077] Further, in this alternative heat following mode, to maintain the evaporator outlet temperature of the superheated working fluid vapors within a desired operating parameter, such as, for example 310° F. (154.4° C.), the controller **125** will adjust the feed pump flow rate and/or speed of the organic working fluid past the burner in order to control the evaporator exit temperature of the superheated vapor. If the exit temperature of the superheated working fluid vapors from the evaporator is above its desire operating parameter, the controller **125** will increase the working fluid's flow rate. Should the exit temperature of the superheated working fluid vapors from the evaporator be less than its desired operating parameter, the controller **125** will decrease the working fluid's flow rate.

[0078] 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.

We claim:

1. A control subsystem for governing the operation of a cogeneration system configured to operate with an organic working fluid, said system having a plurality of functional devices including at least a heat source, an expander having operatively coupled to a generator to produce electricity, a condenser in fluid communication with said expander and adapted to heat a hydronic heating system, and a pump configured to circulate said organic working fluid from said condenser through piping in thermal communication with said heat source such that heat transferred therefrom superheats said organic working fluid to provide superheated organic working fluid vapors to said expander, said control subsystem comprising a programmed processor operatively coupled at least to said pump and said heat source, and adapted to operate said pump and said heat source in response to a call for heat, causing said organic working fluid in said piping in thermal communication with said heat source to be superheated and provided to said expander.

2. The control sub-system according to claim 1, wherein said functional devices of the cogeneration system further include a power grid contactor adapted to connect the electrical power output of said generator with a power grid and an electrical load, wherein said controller is further operatively coupled to said generator and said contactor for receiving and monitoring electrical power output from said generator, and connecting/disconnecting said electrical power output with said electrical load.

3. The control subsystem according to claim 1, further comprising a plurality of sensors each providing a sensor signal indicative of a parameter of the cogeneration system, said controller coupled to said plurality of sensors and adapted to at least monitor the operation of said functional devices of the cogeneration system.

4. The control subsystem according to claim 3, further comprising an operator interface coupled to said controller enabling modification of the operation of said controller in monitoring said functional devices and operating characteristics of the cogeneration system through entry of information via said operator interface.

5. The control subsystem according to claim 4, wherein said operator interface includes a data acquisition and communications subsystem enabling data logging and reporting of said operating characteristics of said functional devices of the cogeneration system.

6. The control subsystem according to claim 5, wherein said a data acquisition and communications subsystem includes a modem for reporting at least alarm conditions of said cogeneration system.

7. The control subsystem according to claim 3, wherein said parameters include at least three of condenser inlet temperature, condenser outlet temperature, hydronic fluid flow, hydronic supply temperature, expander inlet temperature, expander inlet pressure, feed pump inlet temperature, feed pump inlet pressure, power module power output, evaporator inlet temperature, gas flow, expander outlet temperature, outdoor ambient temperature, feed pump speed (drive frequency), protection relay trip, level 1 trip failure, and level 2 trip failure (failed to re-start).

8. The control sub-system according to claim 3, further comprising a plurality of control points, said controller being operatively coupled to said control points of said cogeneration system associated with functional devices thereof, said controller responding to said sensor signals to control said functional devices to thereby vary operating conditions of the cogeneration system.

9. The control sub-system according to claim 3, wherein said controller operates under program control for acquiring said sensor signals and generating control signals for application to said functional devices of said cogeneration system.

10. The control sub-system according to claim 9, wherein said controller operates according to said program control and at least acquires and takes into account an outdoor ambient temperature reading from one of said sensors before generating said control signals.

11. The control sub-system according to claim 10, wherein said controller determines a set-point for a hydronic supply temperature in the hydronic heating system from said outdoor ambient temperature reading.

12. The control sub-system according to claim 11, wherein said controller establishes said set-point according to a linear scale from **25**° C. at an outdoor temperature of 20° C. to 75° C. at an outdoor temperature of −20° C.

13. The control sub-system according to claim 11, wherein the heat source comprises a gas valve and a burner, and wherein said controller uses said set-point to operate the heat source in a variable capacity mode by modulating the gas valve on the burner to maintain actual hydronic supply temperature as sensed by one of said sensors at said set-point.

14. The control sub-system according to claim 13, wherein said controller further prevents working fluid in the liquid state from entering the expander by coordinating a fuel flow rate (heat input rate) to the burner with a feed flow rate of the working fluid exiting the feed pump.

15. The control sub-system according to claim 14, wherein the functional devices further include an evaporator fluidly connected to said expander by said piping and heated by said heat source, and said controller controls said fuel flow rate and said feed flow rate to maintain an exit temperature of the evaporator at 310° F. (154.4° C.).

16. The control sub-system according to claim 1, wherein the functional devices further include a hydronic pump circulating fluid in the hydronic heating system to and from heat exchange piping of the condenser, and wherein said controller controls the hydronic pump at a speed that maintains a pressure difference between the supply and return of the fluid to the heat exchanger piping of the condenser at a preselected value for optimal thermodynamic performance of the hydronic heating system.

17. The control sub-system according to claim 1, wherein the functional devices further include an evaporator fluidly connected between the feed pump and expander, and a desuperheater fluidly connected between the expander and condenser, said desuperheater having a return to the evaporator, and a switching valve provided in the piping for directing the organic working fluid either to the desuperheater or the evaporator directly, and wherein the controller controls the switching of the switching valve.

18. The control sub-system according to claim 1, wherein the expander is selected from a positive displacement expander and a scroll expander.

19. The control sub-system according to claim 1, wherein the functional devices further include a bypass valve and a shutoff valve in the piping to bypass and shutoff said superheat working fluid vapors from entering into the expander, the piping including a bypass loop connected between the bypass valve and condenser, and wherein said controller controls the opening/closing of the bypass valve and shutoff valve at least in response to startup and shutdown conditions of the cogeneration system.

20. The control sub-system according to claim 19, wherein said shutdown conditions include a heat call satisfied signal, a startup sequence failure, or exceeding a related preset value for an expander inlet temperature, an expander inlet pressure, a feed pump inlet temperature, a feed pump inlet pressure, a protection relay trip, or a power module temperature.

21. A control subsystem for governing the operation of a cogeneration system configured to operate with an organic working fluid, said system having a plurality of functional devices at least including a heat source, an expander having operatively coupled a generator to produce electricity, a condenser in fluid communication with said expander, and a pump configured to circulate said organic working fluid from said condenser through piping in thermal communication with said heat source such that heat transferred therefrom superheats said organic working fluid to provide superheated organic working fluid vapors to said expander, said control subsystem comprising:

a plurality of sensors each providing a sensor signal indicative of a parameter of the cogeneration system;

a plurality of control points; and

a programmable controller coupled to said plurality of sensors and to said control points of said cogeneration system associated with functional devices thereof, said controller responding to said sensor signals to control said functional devices of the cogeneration system to thereby vary the operating characteristics of the cogeneration system.

22. A control system in combination with a cogeneration system having a plurality of functional devices and using an organic working fluid to heat a hydronic heating system and produce electrical power, the combination comprising:

a plurality of sensors for providing electrical sensor signals indicative of operating parameters of the cogeneration system;

a plurality of control points associated with said functional devices to change the operating parameters of the cogeneration system; and

a programmable controller coupled to the electrical sensors and to said control points of the system associated with said functional devices, said controller responding to the sensor signals and generating a plurality of control signals to control the operation of the functional devices; said controller defining a plurality of interactive control loops each generating a control signal for controlling a different one of said functional devices as a function of variation in a sensor signal supplied to the controller relative to at least a set-point value for a control loop for the hydronic heating system, wherein the set-point is determined by said controller from receiving an outdoor ambient temperature sensor signal from one of said sensors.

23. A method of controlling the thermal and electrical output of integrated micro-combined heat and power generation systems used to supply domestic electrical power, domestic space heating (SH) water, and/or domestic hot water (DHW), and which converts heat energy contained in superheated vapors of an organic working fluid to mechanical energy, and distributes the superheated vapors under pressure to at least one functional device having a heating need which varies over time, comprising:

monitoring over a period of time an ambient outdoor temperature to determine said heating need by said at least one functional device; and

changing in response to said ambient outdoor temperature indicating at any given time that a different amount of said superheated vapors than that being delivered to said at least one functional device is so needed to satisfy said heating need.

24. The method according to claim 23, wherein there are a plurality of process control sensors communicating with a controller programmed to change operating parameters of said system in response to said ambient outdoor temperature in order to furnish said at least one functional device a supply of said superheated vapors.

25. The method according to claim 24, wherein said ambient outdoor temperature is used to determine a desired hydronic supply temperature for a hydronic fluid and said amount of superheat vapors is varied to heat said hydronic fluid to said desired hydronic supply temperature.

26. The method according to claim 25, further comprising sensing an actual hydronic supply temperature via at least one of said process control sensors, and adjusting a firing rate of a burner used to heat said organic working fluid in order to bring said actual hydronic supply temperature into line with said desired hydronic supply temperature.

27. The method according to claim 26, further comprising increasing said firing rate if said actual hydronic supply temperature is too low, and decreasing said firing rate if said actual hydronic supply temperature is too high.

28. The method according to claim 26, further comprising sensing via at least one of said process control sensors temperature of said superheated vapors and adjusting flow rate of said working fluid past said burner to maintain said temperature of said superheated vapors within a desired operating parameter.

29. The method according to claim 28, wherein said desired operating parameter for said temperature of said superheated vapors is 310° F. (154.4° C.).

30. The method according to claim 28, wherein said flow rate is increased if said temperature of said superheated vapors is above said desire operating parameter, and decreased if said temperature of said superheated vapors is below said desired operating parameter.

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