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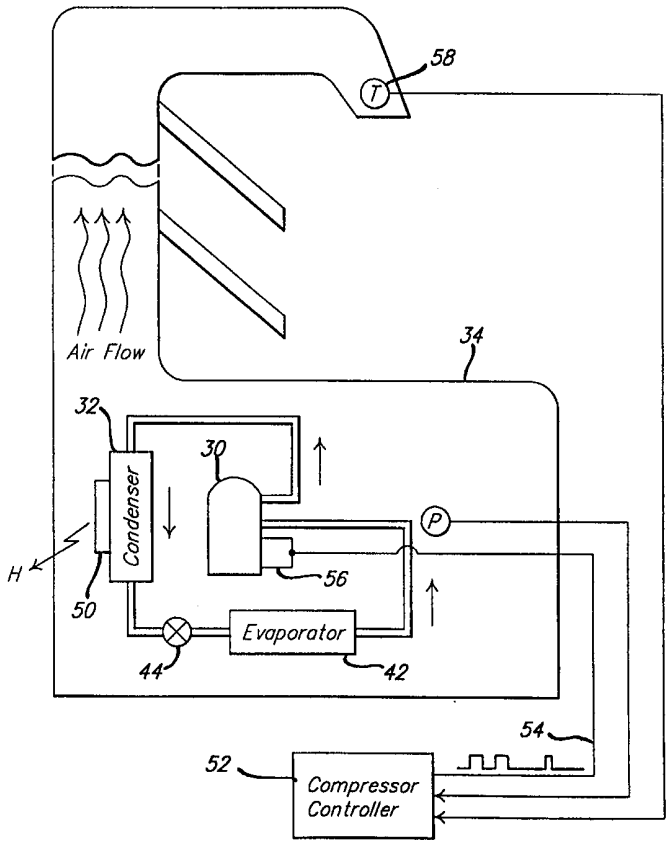
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(54) Title: AN ADAPTIVE CONTROL FOR A REFRIGERATION SYSTEM USING PULSE WIDTH MODULATED DUTY CYCLE SCROLL COMPRESSOR

(57) Abstract

A control for a refrigeration system serviced by a special pulse width modulated compressor (30) coupled to a condenser (32) and evaporator (42). The system can be of the distributed type wherein the condenser (32) and compressor (30) may be coupled to service a group of adjacent refrigeration cases, wherein each case would have its own evaporator (42). A controller (52) is coupled to a load sensor (58) for producing a variable duty cycle control signal (54) in which the duty cycle is a function of demand for cooling. Moreover, fuzzy logic technique may be utilized in order to provide self adaptive tuning control for the system.



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AN ADAPTIVE CONTROL FOR A REFRIGERATION SYSTEM USING PULSE WIDTH MODULATED DUTY CYCLE SCROLL COMPRESSOR

Cross-Reference to Related Application

This is a continuation-in-part application of application Serial No. 08/486,118, filed June 7, 1995, entitled Capacity Modulated Scroll Machine.

Background and Summary of the Invention:

The present invention relates generally to refrigeration systems, compressor control systems and refrigerant regulating valve control systems. More particularly, the invention relates to a refrigeration system employing a pulse width modulated compressor or evaporator stepper regulator controlled by a variable duty cycle signal derived from a load sensor. Preferably an adaptive controller generates the variable duty cycle signal. The compressor has two mechanical elements separated by a seal, and these mechanical elements are cyclically movable relative to one another to develop fluid pressure. The compressor includes a mechanism to selectively break the seal in response to the control signal, thereby modulating the capacity of the system.

The refrigeration system can be deployed as a distributed system in refrigeration cases and the like. The preferred arrangement allows the compressor and condenser subsystems to be disposed in or mounted on the refrigeration case, thereby greatly reducing the length of refrigerant conduit and refrigerant required.

Conventionally, refrigeration systems for supermarket refrigeration cases have employed air-cooled or water-cooled condensers fed by a rack of compressors. The compressors are coupled in parallel so that they may be switched on and off in stages to adjust the system cooling capacity to the demands of the load. Commonly, the condensers are located outside, on the roof, or in a machine room adjacent the shopping area where the refrigeration cases are located.

Within each refrigeration case is an evaporator fed by lines from the condensers through which the expanded refrigerant circulates to cool the case. Conventionally, a closed-loop control system regulates refrigerant flow through the evaporator to maintain the desired case temperature. Proportional-integral-

derivative (PID) closed loop control systems are popular for this purpose, with temperature sensors and/or pressure sensors providing the sensed condition inputs.

It is common practice within supermarkets to use separate systems to supply different individual cooling temperature ranges: low temperature (for frozen foods, ice cream, nominally -25°F); medium (for meat, dairy products, nominally + 20°F); high (for floral, produce, nominally +35 to +40°F). The separate low, medium and high temperature systems are each optimized to their respective temperature ranges. Normally, each will employ its own rack of compressors and its own set of refrigerant conduits to and from the compressors and condensers.

The conventional arrangement, described above, is very costly to construct and maintain. Much of the cost is associated with the long refrigerant conduit runs. Not only are long conduit runs expensive in terms of hardware and installation costs, but the quantity of refrigerant required to fill the conduits is also a significant factor. The longer the conduit run, the more refrigerant required. Adding to the cost are environmental factors. Eventually fittings leak, allowing the refrigerant to escape to atmosphere. Invariably, long conduit runs involve more pipefitting joints that may potentially leak. When a leak does occur, the longer the conduit run, the more refrigerant lost.

There is considerable interest today in environmentally friendly refrigeration systems. Shortening the conduit run is seen as one way to achieve a more environmentally friendly system. To achieve this, new condenser/compressor configurations and new control systems will need to be engineered.

Re-engineering condenser/compressor configurations for more environmentally friendly systems is not a simple task, because system efficiency should not be sacrificed. Generally, the conventional roof-mounted condenser system, supplied by condensers, benefits from economies of scale and is quite efficient. These systems serve as the benchmark against which more environmentally friendly systems of the future will need to be measured.

To appreciate why re-engineering an environmentally yet efficient system has proven so difficult, consider these thermodynamic issues. The typical refrigeration case operates in a very unpredictable environment. From a design

standpoint, the thermal mass being cooled is rarely constant. Within the supermarket environment, the temperature and humidity may vary widely at different times of day and over different seasons throughout the year. The product load (items in the refrigeration case) can also change unpredictably. Customers removing product and store clerks replenishing product rarely synchronize. Outside the supermarket environment, the outdoor air temperature and humidity may also vary quite widely between day and night and/or between summer and winter. The capacity of the system must be designed for the harshest conditions (when the condenser environment is the hottest). Thus systems may experience excess capacity in less harsh conditions, such as in the cool evenings or during the winter.

Periodic defrosting also introduces thermal fluctuations into the system. Unlike thermal fluctuations due to environmental conditions, the thermal fluctuations induced by the defrost cycle are cause by the control system itself and not by the surrounding environment.

In a similar fashion, the control system for handling multiple refrigeration cases can induce thermal fluctuations that are quite difficult to predict. If all cases within a multi-case system are suddenly turned on at once--to meet their respective cooling demands--the cooling capacity must rapidly be ramped up to maximum. Likewise, if all cases are suddenly switched off, the cooling capacity should be ramped down accordingly. However, given that individual refrigeration cases may operate independently of one another, the instantaneous demand for cooling capacity will tend to vary widely and unpredictably.

These are all problems that have made the engineering of environmentally friendly systems more difficult. Adding to these difficulties are user engineering/ergonomic problems. The present day PID controller can be difficult to adapt to distributed refrigeration systems. Experienced controls engineers know that a well-tuned PID controller can involve a degree of artistry in selecting the proper control constants used in the PID algorithm. In a large refrigeration system of the conventional architecture (non-distributed) the size of the system justifies having a controls engineer visit the site (perhaps repeatedly) to fine tune the control constant parameters.

This may not be practical for distributed systems in which the components are individually of a much smaller scale and far more numerous. By way of comparison, a conventional system might employ one controller for an entire multi-case, store-wide system. A distributed system for the same store might involve a controller for each case or adjacent group of cases within the store. Distributed systems need to be designed to minimize end user involvement. It would therefore be desirable if the controller were able to auto configure. Currently control systems lack this capability.

The present invention provides a distributed refrigeration system in which the condenser is disposed on the refrigeration case and serviced by a special pulse width modulated compressor that may be also disposed within the case. If desired, the condenser and compressor can be coupled to service a group of adjacent refrigeration cases, each case having its own evaporator.

The pulse width modulated compressor employs two mechanical elements, such as scroll members, that move rotationally relative to one another to develop fluid pressure for pumping the refrigerant. The compressor includes a mechanism that will selectively break the seal between the two mechanical elements, thereby altering the fluid pressure developed by the compressor while allowing the mechanical elements to maintain substantially constant relative movement with one another. The compressor can be pulse width modulated by making and breaking the fluid seal without the need to start and stop the electric motor driving the mechanical elements.

The pulse width modulated compressor is driven by a control system that supplies a variable duty cycle control signal based on measured system load. The controller may also regulate the frequency (or cycle time) of the control signal to minimize pressure fluctuations in the refrigerant system. The on time is thus equal to the duty cycle multiplied by the cycle time, where the cycle time is the inverse of the frequency.

The refrigeration system of the invention has a number of advantages. Because the instantaneous capacity of the system is easily regulated by variable duty cycle control, an oversized compressor can be used to achieve faster temperature pull down at startup and after defrost, without causing short cycling as

conventional compressor systems would. Another benefit of variable duty cycle control is that the system can respond quickly to sudden changes in condenser temperature or case temperature set point. The controller adjusts capacity in response to disturbances without producing unstable oscillations and without significant overshoot. Also, the ability to match instantaneous capacity to the demand allows the system to operate at higher evaporator temperatures. (Deep drops in temperature experienced by conventional systems at overcapacity are avoided.)

Operating at higher evaporator temperatures reduces the defrost energy required because the system develops frost more slowly at higher temperatures. Also, the time between defrosts can be lengthened by a percentage proportional to the accumulated runtime as dictated by the actual variable duty cycle control signal. For example, a sixty percent duty cycle would increase a standard three-hour time between defrosts to five hours (3/.60=5).

The pulse width modulated operation of the system yields improved oil return. The refrigerant flow pulsates between high capacity and low capacity (e.g. 100% and 0%), creating more turbulence which breaks down the oil boundary layer in the heat exchangers.

Another benefit of the variable duty cycle control system is its ability to operate with a variety of expansion devices, including the simple orifice, the thermal expansion valve (TXV) and the electronic expansion valve. A signal derived from the expansion device controller can be fed to the compressor controller of the invention. This signal allows the variable duty cycle control signal and/or its frequency to be adjusted to match the instantaneous operating conditions of the expansion device. A similar approach may be used to operate variable speed fans in air cooled condenser systems. In such case the controller of the invention may provide a signal to control fan speed based on the current operating duty cycle of the compressor.

Yet another benefit of the invention is its ability to detect when the system is low on refrigerant charge, an important environmental concern. Low refrigerant charge can indicate the presence of leaks in the system. Low charge may be detected by observing the change in error between actual temperature and set

point temperature as the system duty cycle is modulated. The control system may be configured to detect when the modulation in duty cycle does not have the desired effect on temperature maintenance. This can be due to a loss of refrigerant charge, a stuck thermal expansion valve or other malfunctions.

For a more complete understanding of the invention, its objects and advantages, refer to the following specification and to the accompanying drawings.

Brief Description of the Drawings:

Figure 1 is a system block diagram of a prior art refrigeration system configuration;

Figure 2 is a block diagram of a refrigeration system in accordance with the present invention;

Figure 3 is a cross-sectional view of an embodiment of the pulse width modulated compressor, shown in the loaded state;

Figure 4 is a cross-sectional view of the compressor of Figure 3, shown in the unloaded state;

Figure 5 is another embodiment of a refrigeration or cooling system in accordance with the present invention;

Figure 6 is a block diagram of the controller;

Figure 7 is a block diagram showing how the controller may be used to modulate an evaporator stepper regulator;

Figure 8 is a block diagram of the signal conditioning module of the controller of Figure 6;

Figure 9 is a block diagram of the control module of the controller of Figure 6;

Figure 10 is a state diagram depicting the operating states of the controller;

Figure 11 is a flowchart diagram illustrating the presently preferred PI control algorithm;

Figure 12 is a waveform diagram illustrating the variable duty cycle signal produced by the controller and illustrating the operation at a constant frequency;

Figure 13 is a waveform diagram of the variable duty cycle signal, illustrating variable frequency operation;

Figure 14 is a series of graphs comparing temperature and pressure dynamics of system employing the invention with a system of conventional design;

Figure 15 is a block diagram illustrating the adaptive tuning module of the invention;

Figure 16a is a flowchart diagram illustrating the presently preferred operation of the adaptive tuning module, specifically with respect to the decision whether to start tuning;

Figure 16b is a flowchart diagram illustrating the presently preferred process performed by the adaptive tuning module in the integration mode;

Figure 16c is a flowchart diagram illustrating the operation of the adaptive tuning module in the calculation mode;

Figure 17 is a state diagram illustrating the operative states of the adaptive tuning module;

Figure 18 is a block diagram illustrating the fuzzy logic block of the adaptive tuning loop;

Figure 19 is a membership function diagram for the fuzzy logic block of Figure 18;

Figure 20 is a truth table relating to the membership function of Figure 19 as used by the fuzzy logic block of Figure 18;

Figure 21 is an output membership function diagram for the fuzzy logic block of Figure 18; and

Figure 22 is a schematic illustrating exemplary sensor locations for control-related and diagnostic-related functions of the present invention.

Description of the Preferred Embodiments:

Figure 1 illustrates how a conventional supermarket refrigeration system is configured. As previously discussed, it is conventional practice to place the compressors **30** and the condenser **32** in a location remote from the refrigeration cases **34**. In this illustration, the compressors **30** are configured in a parallel bank located on the roof **36** of the building. The bank of compressors supply a large condenser **32**, which may be air cooled or water cooled. The condenser supplies liquid refrigerant to a receiver **38**. The receiver **38**, in turn, supplies the individual

refrigeration cases, which are connected in parallel, as illustrated. In most implementations a liquid line solenoid valve **40** is used to regulate flow to the associated evaporator **42**. The refrigerant is supplied to the evaporator through a suitable expansion device such as expansion valve **44**. The expansion valve **44** provides a restricted orifice that causes the liquid refrigerant to atomize into liquid droplets that are introduced into the inlet side of the evaporator **42**. The evaporator **42**, located within the refrigeration case **34**, extracts heat from the case and its contents by vaporization of the liquid refrigerant droplets into a gas. The compressors **30** extract this gas by suction and compress it back into the liquid state. The liquid refrigerant is then cooled in the condenser **32** and returned to the receiver **38**, whereupon the cycle continues.

To match cooling capacity to the load, the compressors **30** may be switched on and off individually or in groups, as required. In a typical supermarket arrangement there may be several independent systems, each configured as shown in Figure 1, to handle different operating temperature ranges. Note that the liquid line **46** and the suction line **48** may each need to be quite lengthy (e.g., up to 150 feet) to span the distance from refrigeration case to roof.

Figure 2 shows a refrigeration case **34** configured according to the principles of the present invention. The condenser **32** and compressor **30** are both disposed within case **34** or attached thereto. Evaporator **42** and the associated expansion valve **44** are likewise disposed within case **34**. The condenser **32** is provided with a heat removal mechanism **50** by which heat is transferred to ambient. The heat removal mechanism can be a water jacket connected to suitable plumbing for carrying waste heat to a water cooling tower located on the building roof. Alternatively, the heat removal mechanism can be a forced-air cooling system or a passive convection-air cooling system.

The refrigeration system of the invention employs a compressor controller **52** that supplies a pulse width modulated control signal on line **54** to a solenoid valve **56** on compressor **30**. The compressor controller adjusts the pulse width of the control signal using an algorithm described below. A suitable load sensor such as temperature sensor **58** supplies the input signal used by the controller to determine pulse width.

Figures 3 and 4 show the details of compressor **30**. Figure 3 shows the compressor in its loaded state and Figure 4 shows the compressor in its unloaded state. The solenoid valve **56** switches the compressor between these two states while the compressor motor remains energized. One important advantage of this configuration is that the compressor can be pulse width modulated very rapidly between the loaded and unloaded states without interrupting power to the compressor motor. This pulse width modulated cycling exerts less wear on the compressor, because the motor is not subjected to sudden changes in angular momentum.

Referring to Figures 3 and 4, there is shown an exemplary compressor **30**. Compressor **30** may be used within a hermetic scroll compressor such as generally of the type described in assignee's U.S. Patent No. 5,102,316.

The exemplary compressor **30** includes an outer shell **61** and an orbiting scroll member **64** supported on upper bearing housing **63** and drivingly connect to crankshaft **62** via crank pin **65** and drive bushing **60**. A second non-orbiting scroll member **67** is positioned in meshing engagement with scroll member **64** and axially movably secured to upper bearing housing **63**. A partition plate **69** is provided adjacent the upper end of shell **61** and serves to define a discharge chamber **70** at the upper end thereof.

In operation, as orbiting scroll member **64** orbits with respect to scroll member **67**, suction gas is drawn into shell **61** via suction inlet **71** and thence into compressor **30** through inlet **72** provided in non-orbiting scroll member **67**. The intermeshing wraps provided on scroll members **64** and **67** define moving fluid pockets which progressively decrease in size and move radially inwardly as a result of the orbiting motion of scroll member **64** thus compressing the suction gas entering via inlet **72**. The compressed gas is then discharged into discharge chamber **70** via discharge port **73** provided in scroll member **67** and passage **74**.

In order to unload compressor **30**, solenoid valve **56** will be actuated in response to a signal from control module **87** to interrupt fluid communication to increase the pressure within chamber **77** to that of the discharge gas. The biasing force resulting from this discharge pressure will overcome the sealing biasing force thereby causing scroll member **67** to move axially upwardly away from orbiting

scroll member **64**. This axial movement will result in the creation of a leakage path between the respective wrap tips and end plates of scroll members **64** and **67** thereby substantially eliminating continued compression of the suction gas.

A flexible fluid line **91** extends from the outer end of passage **90** to a fitting **92** extending through shell **61** with a second line **93** connecting fitting **92** to solenoid valve **56**. Solenoid valve **56** has fluid lines **82** and **84** connected to suction line **83** and discharge line **85** and is controlled by control module **87** in response to conditions sensed by sensor **88** to effect movement of non-orbiting scroll member **67** between the positions shown in Figures 3 and 4.

When compression of the suction gas is to be resumed, solenoid valve **56** will be actuated so as to move scroll member **67** into sealing engagement with scroll member **64**.

The refrigeration case embodiment of Figure 2 may be packaged as a self-contained unit. While that may be a desirable configuration for many applications, the invention is not restricted to stand alone, self-contained refrigeration case configurations. Rather, the invention lends itself to a variety of different distributed refrigeration systems. Figure 5 shows an example of such a distributed system.

Referring to Figure 5, a single compressor **30** and condenser **32** can service several distributed refrigeration cases or several distributed cooling units in a heating and cooling (HVAC) system. In Figure 5 the refrigeration cases or cooling system housings are shown as dashed boxes, designated **34a**, **34b**, and **34c**. Conveniently, the compressor **30** and condenser **32** may be disposed within or attached to one of the refrigeration cases or housings, such as refrigeration case or housing **34a**.

Each refrigeration case or housing has its own evaporator and associated expansion valve as illustrated at **42** (a, b, c) and **44** (a, b, c). In addition, each refrigeration case or housing may have its own temperature sensor **58** (a, b, c) supplying input information to the compressor controller **52**. Finally, a pressure sensor **60** monitors the pressure of the suction line **48** and supplies this information to compressor controller **52**. The compressor controller supplies a variable duty cycle signal to the solenoid valve **56** as previously described.

The multiple case or multiple cooling unit embodiment of Figure 5 shows how a single compressor can be pulse width modulated by compressor controller **52** to supply the instantaneous demand for cooling. The temperature sensors **58** (a, b, c) collectively provide an indication of the load on the system, as does pressure sensor **60**. The controller adjusts the pulse width of the control signal to modulate the compressor between its high capacity and low capacity states (100%, 0%) to meet the instantaneous demand for refrigerant.

As an alternate control technique, one or more of the suction lines exiting the evaporator can be equipped with an electrically controlled valve, such as an evaporator pressure regulator valve **45c**. Valve **45c** is coupled to controller **52**, as illustrated. It may be supplied with a suitable control signal, depending on the type of the valve. A stepper motor valve may be used for this purpose, in which case controller **30** would supply a suitable signal to increment or decrement the setting of the stepper motor to thereby adjust the orifice size of the valve. Alternatively, a pulse width modulated valve could be used, in which case it may be controlled with the same variable duty cycle signal as supplied to the compressor **30**.

Controller **52** is not limited to solely compressor control applications. The variable duty cycle control signal can also be used to control other types of refrigerant flow and pressure control devices, such as refrigerant regulating valves. Figure 7 shows such an application, where the output of controller **52** supplies control signals to evaporator stepper regulator **43.** This device is a fluid pressure regulator that is adjusted by stepper motor **45**. The evaporator stepper regulator (ESR) valve **43** adjusts the suction pressure to thereby adjust the capacity of the system.

A block diagram of the presently preferred compressor controller is illustrated in Figure 6. A description of the various signals and data values shown in this and successive figures is summarized in Table I below.

TABLE I

No.	Variable Name	Default Value	Description
1	**Signal Conditioning:**		
	- Sensor Alarm	- False	- Indicates Sensor Reading is not within expected range
	- Sensor Mode	- Min	- User configuration to indicate if Min/Max/Avg is performed for all temperature Sensors
	-Sampling Time (Ts)	- 0.5 sec	- Rate at which Signal conditioning block is executed
	- Control Type	- T/P Type	- if controlled by only Temp. or both Temp. & Pressure
2	**Control Block:**		
	- Sensor Alarm	- False	- Same as before
	- System Alarm	- False	- Generated by Adaptive Block indicative some system problem
	- SSL	- 0	- Steady state loading %
	- Defrost Status	- False	- Whether system is in defrost
	- Pull_Down_Time	- 0	- Time taken to pull down after defrost
	- Gain (K)	- 7	- Gain used in PI algorithm
	- Integral Time (Ti)	- 100	- used in PI
	- Control Time (Tc)	- 10 Sec	- used in PI
	- Control Set Pt. (St)	- 0 F	- used in PI
	- Operating State	- 1	- What state the machine is operating at
3	**Defrost Control**		
	- Defrost Status	- False	- If defrost status of the case
	- Defrost Type	- External	- If the defrost is from external timer or Internal clock of controller
	- Defrost Interval	- 8 hrs	- Time between defrost
	- Defrost Duration	- 1 hr	- Defrost Duration
	- Defrost Termination Temp.	- 50F	- Termination temperature for defrost

At the heart of the controller is control block module **102**. This module is responsible for supplying the variable duty cycle control signal on lead **104**. Module **102** also supplies the compressor ON/OFF signal on lead **106** and an operating state command signal on lead **108**. The compressor ON/OFF signal drives the contactors that supply operating current to the compressor motor. The operating state signal indicates what state the state machine (Fig 10) is in currently.

The control block module receives inputs from several sources, including temperature and pressure readings from the temperature and pressure sensors previously described. These temperature readings are passed through signal conditioning module **110**, the details of which are shown in the pseudocode Appendix. The control block module also receives a defrost status signal from defrost control module **112**. Defrost control module **112** contains logic to determine when defrost is performed. The present embodiment allows defrost to be controlled either by an external logic signal (supplied through lead **114**) or by an internal logic signal generated by the defrost control module itself. The choice of whether to use external or internal defrost control logic is user selectable through user input **116**. The internal defrost control uses user-supplied parameters supplied through user input **118**.

The preferred compressor controller in one form is autoconfigurable. The controller includes an optional adaptive tuning module **120** that automatically adjusts the control algorithm parameters (the proportional constant K) based on operating conditions of the system. The adaptive tuning module senses the percent loading (on lead **104**) and the operating state (on lead **108**) as well as the measured temperature after signal conditioning (on lead **122**). Module **120** supplies the adaptive tuning parameters to control block **102**, as illustrated. The current embodiment supplies proportional constant K on lead **124** and SSL parameter on lead **126**, indicative of steady-state loading percent. A system alarm signal on lead **126** alerts the control block module when the system is not responding as expected to changes in the adaptively tuned parameters. The alarm thus signals when there may be a system malfunction or loss of refrigerant charge. The alarm can trigger more sophisticated diagnostic routines, if desired. The

compressor controller provides a number of user interface points through which user-supplied settings are input. The defrost type (internal/external) input **116** and the internal defrost parameters on input **118** have already been discussed. A user input **128** allows the user to specify the temperature set point to the adaptive tuning module **120**. The same information is supplied on user input **130** to the control block module **102**. The user can also interact directly with the control block module in a number of ways. User input **132** allows the user to switch the compressor on or off during defrost mode. User input **134** allows the user to specify the initial controller parameters, including the initial proportional constant K. The proportional constant K may thereafter be modified by the adaptive tuning module **120**. User input **136** allows the user to specify the pressure differential (dP) that the system uses as a set point.

In addition to these user inputs, several user inputs are provided for interacting with the signal conditioning module **110**. User input **138** selects the sensor mode of operation for the signal conditioning module. This will be described in more detail below. User input **140** allows the user to specify the sampling time used by the signal conditioning module. User input **142** allows the user to specify whether the controller shall be operated using temperature sensors only (T) or temperature and pressure sensors (T/P).

Referring now to Figure 8, the signal conditioning module is shown in detail. The inputs (temperature and/or pressure sensors) are shown diagrammatically at **144**. These inputs are processed through analog to digital convertor **146** and then supplied to the control type selector **148**. Temperature readings from the temperature and/or pressure sensors are taken sequentially and supplied serially through the analog to digital convertor. The control type selector codes or stores the data so that pressure and temperature values are properly interpreted.

Digital filtering is then applied to the signal at **150** to remove spurious fluctuations and noise. Next, the data are checked in module **152** to ensure that all readings are within expected sensor range limits. This may be done by converting the digital count data to the corresponding temperature or pressure values and checking these values against the pre-stored sensor range limits. If the

readings are not within sensor range an alarm signal is generated for output on output **154**.

Next a data manipulation operation is performed at **156** to supply the temperature and/or pressure data in the form selected by the sensor mode user input **138**. The current embodiment will selectively average the data or determine the minimum or maximum of the data (Min/Max/Avg). The Min/Max/Avg mode can be used to calculate the swing in pressure differential, or a conditioned temperature value. The average mode can be used to supply a conditioned temperature value. These are shown as outputs **158** and **160**, respectively.

Figure 9 shows the control block module in greater detail. The conditioned temperature or pressure signal is fed to calculation module **162** that calculates the error between the actual temperature or pressure and the set point temperature or pressure. Module **162** also calculates the rate of change in those values.

The control block module is designed to update the operating state of the system on a periodic basis (every T_c seconds, nominally once every second). The Find Operating State module **164** performs this update function. The state diagram of Figure 10 provides the details on how this is performed. Essentially, the operating state advances, from state to state, based on whether there is a sensor alarm (SA) present, whether there is a defrost status signal (DS) present and what the calculated error value is. The Find Operating State module **164** supplies the operating state parameter and the Pull Down Time parameter to the decision logic module **166**.

Referring to Figure 10, the Find Operating State module **164** advances from state to state as follows. Beginning in the initial state **168** the module advances to the normal operating state **170** after initialization. It remains in that state until certain conditions are met. Figure 10 shows by label arrows what conditions are required to cycle from the normal operating state **170** to the defrost state **172**; to the pull down state **174**; to the sensor alarm pull down state **176**; to the sensor alarm operating state **178** and to the sensor alarm defrost state **180**.

The decision logic module **166** (Figure 9) determines the duty cycle of the variable duty cycle signal. This is output on lead **182**, designated % Loading. The decision logic module also generates the compressor ON/OFF signal on lead **184**.

The actual decision logic will be described below in connection with Figure 11. The decision logic module is form of proportional integral (PI) control that is based on an adaptively calculated cycle time T_{cyc}. This cycle time is calculated by the calculation module **186** based on a calculated error value generated by module **188**. Referring back to Figure 6, the conditioned pressure differential signal on lead **122** (Cond dP) is supplied to the Calculate Error module **188** (Figure 9) along with the pressure differential set point value as supplied through user input **136** (Figure 6). The difference between actual and set point pressure differentials is calculated by module **188** and fed to the calculation module **186**. The adaptive cycle time T_{cyc} is a function of the pressure differential error and the operating state as determined by the find operating state module **164** according to the following calculation:

$$T_{cyc(new)} = T_{cyc(old)} + K_c * \text{Error} \quad \text{.....(1)}$$

where:

K_c: proportional constant; and

Error: (actual - set point) suction pressure swing.

The presently preferred PI control algorithm implemented by the decision logic module **166** is illustrated in Figure 11. The routine begins at step **200** by reading the user supplied parameters K, T_i, T_c and S_t. See Figure 6 for a description of these user supplied values. The constant K_p is calculated as being equal to the initially supplied value K; and the constant K_i is calculated as the product of the initially supplied constant K and the ratio T_c/T_i.

Next, at step **202** a decision is made whether the absolute value of the error between set point temperature and conditioned temperature (on lead **190**, Figure 9) is greater than 5%F. If so, the constant K_p is set equal to zero in step **204**. If not, the routine simply proceeds to step **206** where a new loading percent value is calculated as described by the equation in step **206** of Figure 11. If the load percent is greater than 100 (step **208**), then the load percent is set equal to 100% at step **210**. If the load percent is not greater than 100% but is less than 0% (step **212**) the load percent is set equal to 0% at step **214**. If the load percent is between the 0% and 100% limits, the load percent is set equal to the new load percent at step **216**.

The variable duty cycle control signal generated by the controller can take several forms. Figures 12 and 13 give two examples. Figure 12 shows the variable duty cycle signal in which the duty cycle varies, but the frequency remains constant. In Figure 12, note that the cycle time, indicated by hash marks **220**, are equally spaced. By comparison, Figure 13 illustrates the variable duty cycle signal wherein the frequency is also varied. In Figure 13, note that the hash marks **220** are not equally spaced. Rather, the waveform exhibits regions of constant frequency, regions of increasing frequency and regions of decreasing frequency. The variable frequency illustrated in Figure 13 is the result of the adaptive modulation of the cycle time T_{cyc}.

Figure 14 graphically shows the benefits that the control system of the invention has in maintaining tighter temperature control and higher suction pressure with improved system efficiency. Note how the temperature curve **222** of the invention exhibits considerably less fluctuation than the corresponding temperature curve **224** of a conventional controller. Similarly, note that the pressure curve **226** of the invention has a baseline well above that of pressure curve **228** of the conventional controller. Also, the peak-to-peak fluctuation in pressure exhibited by the invention (curve **226**) is much smaller than that of the conventional controller (curve **228**).

The controller of the invention operates at a rate that is at least four times faster (typically on the order of at least eight times faster) than the thermal time constant of the load. In the presently preferred embodiment the cycle time of the variable duty cycle signal is about eight times shorter than the time constant of the load. By way of non-limiting example, the cycle time of the variable duty cycle signal might be on the order of 10 to 15 seconds, whereas the time constant of the system being cooled might be on the order of 1 to 3 minutes. The thermal time constant of a system being cooled is generally dictated by physical or thermodynamic properties of the system. Although various models can be used to describe the physical or thermodynamic response of a heating or cooling system, the following analysis will demonstrates the principle.

Modeling the thermal time constant of the system being cooled

One can model the temperature change across the evaporator coil of a refrigeration system or heat pump as a first order system, wherein the temperature change may be modeled according to the following equation:

$$\Delta T = \Delta T_{ss} [1-\exp(-t/\gamma)] + \Delta T_0 \exp(-\tau/\gamma).$$

where:

ΔT = air temperature change across coil

ΔT_{ss} = steady state air temperature change across coil

ΔT_0 = air temperature change across the coil at time zero

t = time

γ = time constant of coil.

The transient capacity of the unit can be obtained by multiplying the above equation by the air mass flow rate (m) and specific heat at constant pressure (C_p) and integrating with respect to time.

Generally, it is the removal of the refrigerant from the evaporator that controls the time required to reach steady state operating condition, and thus the steady state temperature change across the condenser coil. If desired, the system can be modeled using two time constants, one based on the mass of the coil and another based on the time required to get the excess refrigerant from the evaporator into the rest of the system. In addition, it may also be desirable to take into account, as a further time delay, the time lag due to the large physical distance between evaporator and condenser coils in some systems.

The thermal response of the evaporator coil may be modeled by the following equation:

$$\vartheta = \tfrac{1}{2} [(1-e^{-t/\gamma_1}) + (1-e^{-t/\gamma_2})]$$

where:

ϑ = temperature change across coil/steady state temperature change across coil

t = time

γ_1 = time constant based on mass of coil

γ_2 = time constant based on time required to remove excess refrigerant from evaporator

In practice, the controller of the invention cycles at a rate significantly faster than conventional controllers. This is because the conventional controller cycles on and off in direct response to the comparison of actual and set-point temperatures (or pressures). In other words, the conventional controller cycles on when there is demand for cooling, and cycles off when the error between actual and set-point temperature is below a predetermined limit. Thus the on-off cycle of the conventional controller is very highly dependent on the time constant of the system being cooled.

In contrast, the controller of the invention cycles on and off at a rate dictated by calculated values that are not directly tied to the instantaneous relation between actual and set-point temperatures or pressures. Rather, the cycle time is dictated by both the cycle rate and the duty cycle of the the variable duty cycle signal supplied by the controller. Notably, the point at which the controller cycles from on to off in each cycle is not necessarily the point at which the demand for cooling has been met, but rather the point dictated by the duty cycle needed to meet the demand.

Adaptive Tuning

The controller Geneva described above can be configured to perform a classic control algorithm, such as a conventional proportional-integral-derivative (PID) control algorithm. In the conventional configuration the user would typically need to set the control parameters through suitable programming. The controller may also be of an adaptive type, described here, to eliminate the need for the user to determine and program the proper control parameters.

Thus, one important advantage of the adaptive controller is its ability to perform adaptive tuning. In general, tuning involves selecting the appropriate control parameters so that the closed loop system is stable over a wide range of operating conditions, responds quickly to reduce the effect of disturbance on the control loop and does not cause excessive wear of mechanical components

through continuous cycling. These are often mutually exclusive criteria, and a compromise must generally be made. In Figure 18 (and also Figure 6) there are two basic control loops: the refrigeration control loop and the adaptive tuning loop. The refrigeration control loop is administered by control block module **102**; the adaptive loop is administered by adaptive tuning module **120**. Details of the adaptive tuning module **120** are shown in Figures 15, 16a-16c and 17. The presently preferred adaptive tuning module uses a fuzzy logic control algorithm that will be described in connection with Figures 18-20.

Referring to Figure 15, the adaptive tuning module performs basically three functions. First, it decides whether to perform adaptive tuning. This is handled by module **240**. Second, It gathers the needed parameters for performing adaptive tuning. This is handled by module **242**. Third, it calculates the adaptive gain used by the control loop. This is handled by module **244**.

Module **240** bases the decision on whether to start tuning upon two factors: the current operating state of the system and the control set point. The flowchart of Figure 16a shows the steps involved in this decision. Module **242** integrates key parameters needed for the calculations performed by module **244**. Essentially, module **242** inputs the percent loading, the temperature and pressure values and the set point temperature. It outputs the following data: S_ER (the total number of conditioned temperature and pressure data points that are within 0.5 degrees or 1Psig of the set point value), S_Close (the total number of percent loading data points that goes to zero percent during a given sampling interval, e.g. 30 min.), S_Open (the total number of percent loading data points that goes to 100% in the sampling interval) and SSLP (a moving average or rolling average of the percent loading during the sampling interval). Module **242** is responsive to a tuning flag that is set by module **240**. Module **242** performs the integration of these key parameters when signaled to do so by the tuning flag. Figure 16b shows the steps involved in performing integration of these key parameters.

Finally, the calculation block takes the data supplied by module **242** and calculates the adaptive gain using the process illustrated in Figure 16c.

The adaptive tuning module **120** will cycle through various operating states, depending on the state of a timer. Figure 17 is a state diagram showing how the

presently preferred embodiment will function. Note that the sequence transitions from the initialization mode to either the integration mode or the no tuning mode, depending on whether the tuning flag has been set. Once in the integration mode, the system performs integration until the timer lapses (nominally 30 minutes), whereupon the calculation mode is entered. Once the calculations are completed the timer is reset and the system returns to the initialization mode.

The block diagram of the adaptive scheme is shown in Figure 18. There are two basic loops -- The first one is the PID control loop **260** that runs every "dt" second and the second is the adaptive loop **262** that runs every "ta" second.

When the control system starts, the PID control loop **260** uses a default value of gain (K) to calculate the control output. The adaptive loop **262**, checks the error e(t) **264** every "ta" seconds **266** (preferably less than 0.2 * dt seconds). At module **268** if the absolute value of error, e(t), is less than desired offset (OS), a counter Er_new is incremented. The Offset (OS) is the acceptable steady-state error (e.g. for temperature control it may be +/1 °F). This checking process continues for "tsum" seconds **270** (preferably 200 to 500 times dt seconds). After "tsum" seconds **270**, the value Er_new is converted into percentage (Er_new% **272**). The parameter Er_new% **272** indicates the percentage of sampled e(t) that was within accepted offset (OS) for "tsum" time. In other words, it is a measure of how well the control variable was controlled for past *"tsum"* seconds. A value of 100% means "tight" control and 0% means "poor" control. Whenever Er_new% is 100%, the gain remains substantially unchanged as it indicates tighter control. However, if Er_new happens to be between 0 and 100%, adaptive fuzzy-logic algorithm module **274** calculates a new gain (K_new **276**) that is used for next "tsum" seconds by the control algorithm module **278**.

In the preferred embodiment, there is one output and two inputs to the fuzzy-logic algorithm module **274**. The output is the new gain (K_new) calculated using the input, Er_new%, and a variable, Dir, defined as follows:

$$Dir = Sign[(ER\text{-}new\% - ER_old\%) * (K_new - K\text{-}old)] \quad \text{.....(2)}$$

where:

Sign stands for the sign (+ve, -ve or zero) of the term inside the bracket;

Er_new% is the percentage of e(t) that is within the offset for past "tsum" seconds;

Er_old% is the value of Er_new% in "(tsum-1)" iteration;

K_new is the gain used in "tsum" time; and

K_old is the gain in (tsum-1) time.

For example, suppose the controller starts at 0 seconds with a default value of K=10 and, ta=1 seconds, tsum=1000 seconds and OS=1. Suppose 600 e(t) data out of a possible 1000 data was within the offset. Therefore, after 1000 sec, Er_new%=60 (i.e., 600/1000*100), K_new=10. Er_old% and K_old is set to zero when the adaptive fuzzy-logic algorithm module **274** is used the first time. Plugging these numbers in Eq.(2) gives the sign of the variable "Dir" as positive. Accordingly, the inputs to the adaptive fuzzy-logic module **274** for the first iteration are respectively, Er_new%=60 and Dir=+ve.

The next step is to perform fuzzification of these inputs into fuzzy inputs by using membership functions.

Fuzzification:

A membership function is a mapping between the universe of discourse (x-axis) and the grade space (y-axis). The universe of discourse is the range of possible values for the inputs or outputs. For ER_new% it is preferably from 0 to 100. The value in the grade space typically ranges from 0 to 1 and is called a fuzzy input, truth value, or a degree of membership. Figure 19 shows graph **300** which contains the membership functions for the input, Er_new%. Er_new% is divided into three linguistic variables - LARGE (**304**), MEDIUM (**306**) AND SMALL (**308**). For Er_new%=60, the fuzzy inputs (or degree of membership function) are - 0.25 of LARGE and 0.75 of MEDIUM. The input variable "Dir" is well defined (+ve, -ve or zero) and thus does not require a membership function in this application. The next step is to create the "Truth Table" or Rule Evaluation.

Rule Evaluation:

Rule evaluation takes the fuzzy inputs from the fuzzification step and the rules from the knowledge base and calculates fuzzy outputs. Figure 20 shows the rules as truth table. For the first column and first row, the rule is:

"IF ER_new% is LARGE AND Dir is NEGATIVE THEN New Gain is NO CHANGE (NC)" (i.e. if the percentage of e(t) data that is within the offset (OS) for last "tsum" seconds is LARGE and the direction (DIR) is NEGATIVE/ZERO then do not change the existing K value (NO CHANGE)).

In the example, because ER_new% has fuzzy inputs LARGE (0.25) AND MEDIUM (0.75) with POSITIVE Dir, the rules that will be used are:

IF ER_new% is LARGE (0.25) AND Dir is POSITIVE THEN New Gain is NO CHANGE (NC=1)

IF ER_new% is MEDIUM (0.75) AND Dir is POSITIVE THEN New Gain is POSITIVE SMALL CHANGE (PSC=1.2)

Defuzzification:

Finally, the defuzzification process converts the fuzzy outputs from the rule evaluation step into the final output by using Graph **310** of Figure 21. Graph **310**, uses the following labels = "NBC" for negative big change; "NSC" for negative small change; "NC" for no change; "PSC" for positive small change; and "PBC" for positive big change. The Center of Gravity or centroid method is used in the preferred embodiment for defuzzification. The output membership function for change in gain is shown in Figure 21.

The centroid (the Fuzzy-Logic Output) is calculated as:

$$Centroid = K_new \cdot \left[\frac{\sum_{all.x} \mu(x) \cdot x}{\sum_{all.x} \mu(x)} \right] \qquad \text{.....(3)}$$

where:

$\mu(x)$ is the fuzzy output value for universe of discourse value x.

In our example, the output (K_new) becomes

$$Output = 10 \cdot \left[\frac{0.25(1) + 0.75(1.2)}{0.25 + 0.75} \right] \approx 11.50 \qquad \text{.....(4)}$$

Once the three steps of fuzzification, rule evaluation, and defuzzification are finished and the output has been calculated, the process is repeated again for new set of Er_new%.

In the above example, after the first 1000 sec, the adaptive algorithm calculates a new gain of K_new=11.50. This new gain is used for the next 1000 sec (i.e. from t=1000 to 2000 sec in real time) by the PID control loop. At t=1001 sec, counter Er_new is set to zero to perform counting for the next 1000 seconds. At the end of another 1000 seconds (ie. at t=2000 seconds), Er_new% is calculated again.

Suppose this time, Er_new% happens to be 25. This means, by changing K from 10 to 11.5, the control became worse. Therefore, it would be better to change gain in the other direction i.e. decrease the gain rather than increase. Thus, at t=2000 sec, Er_new%=25, Er_old%=60 (previous value of Er_new%), K_new=11.5 and K_old=10 (previous value of K). Applying Eq.(2), a negative "Dir" is obtained. With Er_new% of 25 and Dir=Negative, the fuzzy-logic calculation is performed again to calculate a new gain for the next 1000 seconds. The new value of gain is K_new=7.76 and is used from t=2000 to 3000 seconds by the PID Loop.

Suppose for the third iteration i.e. from t=2000 to 3000 seconds, Er_new% comes out to be 95% (which represents tighter control). Performing the same fuzzy-logic operation gives the same value of K_new, and the gain remains unchanged until Er_new% again degrades.

Exemplary Applications

Both pulse width modulated (PWM) Compressors and electronic stepper regulator (ESR) Valves can be used to control evaporator temperature/pressure or evaporator cooling fluid (air or water) temperature. The former controls by modulating the refrigerant flow and the latter restricts the suction side to control the flow. Referring back to Figure 18, the block diagram of the control system for such

an actuator working in a refrigeration system **279** is shown. In Figure 18 one and preferably up to four temperatures of evaporator cooling fluid or one evaporator suction pressure (generally shown at **282**) is sampled every dt seconds. A sampling time of dt=10 seconds was found to be optimum for both the applications. After processing by the analog to digital module **284**, the sampled signal is then reduced to one number by taking the average or the minimum or the maximum of the four temperatures depending on the system configuration or the user preference at module **286**. Typically, in a single actuator (PWM/ESR) systems where the complete evaporator coil goes into defrost at one time, averaging of control signal is preferred. In a multiple evaporator-single actuator system where defrost of evaporator coils does not occur at the same time, minimum is the preferred mode. The value obtained after avg/min/max is called conditioned signal. At comparison module **288** this is compared with the desired set point to calculate the error, e(t).

The control algorithm used in the loop is a Proportion-Integral (PI) control technique (PID). The PI algorithm calculates the valve position (0-100%) in case of ESR or calculates the percentage loading (O to 100%) in case of PWM compressor. A typical integral reset time, Ti, for both the actuators is 60 seconds. The gain is tuned adaptively by the adaptive loop. The adaptive algorithm is turned off in the preferred embodiment whenever: the system is in defrost; is going through pull-down; there a big set point change; sensor failure has been detected; or any other system failure is detected.

Consequently, the adaptive algorithm is typically used when the system is working under normal mode. The time "ta" preferably used is about 1 seconds and "tsum" is about 1800 seconds (30 minutes).

Diagnostics Related to PWM Compressor/ESR Valves

Referring to Figure 22, a discharge cooling fluid temperature sensor **312** (Ta), an evaporator coil inlet temperature sensor **314** (Ti) and an evaporator coil exit temperature sensor **316** (To) can provide diagnostic features for the evaporator control using PWM/ESR. The Inlet temperature sensor **314** can be anywhere in

evaporator coil **318**. However, the preferred location is about one third of the total evaporator length from the evaporator coil distributor **320**.

Using these three temperature sensors, system learning can be performed that can be used for diagnostics. For example, diagnostics can be performed for ESR/PWM when it is used in a single evaporator along with an expansion valve. In this example, the following variables are tracked every "tsum" second in the adaptive loop. The variables can be integrated just after ER_new integration is done in the adaptive loop.

* N-Close: Number of times Valve position /PWM loading was 0%.
* N-Open: Number of times Valve position/PWM loading was 100%.
* MAVP: The moving average of the Valve position /PWM loading for "tsum" seconds.
* SSLP: The steady-state Valve position /PWM loading is set equal to MAVP if for the "tsum" duration ER_new% is greater than 50%.
* dT: Moving average of the difference between Ta and Ti (Ta-Ti).
* SH: Moving average of the difference between To and Ti (To-Ti) in the said duration. This is approximately the evaporator superheat.
* N_FL: Number of times To was less than Ti during the said duration i.e. "tsum" seconds. This number will indicate how much the expansion valve is flooding the evaporator.

In addition, Pull-down time after defrost, tpd, is also learnt. Based on these variables, the following diagnostics are performed: temperature sensor failure; degraded expansion valve; degraded ESR valve/PWM Compressor; oversized ESR/PWM; undersized ESR/PWM; and no air flow.

Temperature Sensor Failure:

Failures of temperature sensors are detected by checking whether the temperature reading falls within the expected range. If PWM/ESR is controlled using Ta as the control variable, then when it fails, the control is done as follows. The above said actuator is controlled based on Ti, or the Ta values are estimated using the learned dT (i.e., add dT to Ti value to estimate Ta). During pull down,

the valve/PWM can be set to full-open/load for the learned pull-down time (tpd). If Ti also fails at the same time or is not available, the actuator is opened 100% during pull down time and then set to steady-state loading percent (SSLP) after pull-down-time. An alarm is sent to the supervisor upon such a condition.

Degraded Expansion Valve:

If an expansion valve sticks or is off-tuned or is undersized/oversized, the following combinations of the tracked variable can be used to diagnose such problems. N_FL > 50% and ER_new% >10% indicate the expansion valve is stuck open or is off-tuned or may be even oversized and thus is flooding the evaporator coil. An alarm is sent upon such a condition. Moreover, SH>20 and N_FL=0% indicate an off-tuned expansion valve or an undersized valve or the valve is stuck closed.

Degraded ESR Valve/PWM Compressor

A degraded ESR is one that misses steps or is stuck. A degraded PWM Compressor is one whose solenoid is stuck closed or stuck open. These problems are detected in a configuration where defrost is performed by setting the ESR/PWM to 0%. The problem is detected as follows.

If ER_new% > 50% before defrost and during defrost Ti < 32°F and SH > 5° F, then the valve is determined to be missing steps. Accordingly, the valve is closed by another 100% and if Ti and SH remain the same then this is highly indicative that the valve is stuck.

If ER_new%=0 and N_Close is 100% and Ti < 32°F and SH > 5°F then PWM/ESR is determined to be stuck open. If ER_new%=0 and N_Open is 100% and Ti > 32° F and SH > 5° F then PWM/ESR is determined to be stuck closed.

Over-sized ESR/PWM:

If N_Close > 90% and 30% < ER_new% < 100%, then an alarm is sent for oversized valve/PWM Compressor.

Under-sized ESR/PWM:

If N_Open > 90% and ER_new% =0 and SH > 5, then an alarm is sent for undersized valve/PWM Compressor.

No Air Flow:

If N_Open =100%, ER_new%=0, SH < 5° F and Ti <25° F and N_FL > 50%, then either the air is blocked or the fans are not working properly.

Additionally, these diagnostic strategies can also be applied to an electronic expansion valve controller.

The embodiments which have been set forth above were for the purpose of illustration and were not intended to limit the invention. It will be appreciated by those skilled in the art that various changes and modifications may be made to the embodiments discussed in this specification without departing from the spirit and scope of the invention as defined by the appended claims.

Appendix

Pseudocode for performing signal conditioning

```
Repeat the following every Ts Seconds:
Read User Inputs:
        -Sampling Time (Ts)
        -Control Type (P or T)
        -Sensor Mode (Avg/Min/Max)
Perform Analog to Digital Conversion (ADC)
        -on all (four) Temp. Sensor Channels
        output data as Counts
Digitally Filter Counts
        -Ynew = 0.75 * Yold + 0.25 * Counts
        -output data as Filtered Counts
Convert Filtered Counts to Deg F.
Test if at least one Sensor is within normal operating limits
        -e.g. within -40 and +90F
        If none are within limit--Set Sensor Alarm to TRUE
        Else Perform Avg/Min/Max operation based on Sensor Mode
If Control Type is NOT a T/P Control Type
        Then End Signal Conditioning Routine (until next Ts cycle)
        Else (Control Type is T/P) Do the Following:
Perform ADC on Pressure Sensor Channel
        -output data as Counts
Digitally Filter Counts
        -Ynew = 0.75 * Yold + 0.25 * Counts
        -output data as Filtered Counts
Convert Filtered Counts to Psig
Test if pressure Sensor is within normal operating limits
        -e.g. within 0 and +200
        If not within limit:
                Set dP = dP Set Pt.
        Else :
                Calculate dP = Pmax-Pmin
Set Sensor Alarm to Conditioned T/dP
End Signal Conditioning Routine (until next Ts cycle)
```

We claim:

1. A compressor control system, comprising:

a compressor that is selectively operable while energized in at least two states, including a first state corresponding to a first capacity and a second state corresponding to a second capacity lower than said first capacity;

a load sensor for sensing a property indicative of demand for cooling;

a controller coupled to said load sensor for producing a variable duty cycle control signal in which the duty cycle is a function of demand for cooling;

said controller being coupled to said compressor for causing said compressor to selectively alternate between said first and second states in response to said variable duty cycle control signal, thereby adjusting the capacity of said compressor to the demand for cooling while said compressor is energized.

2. The control system of claim 1 wherein said compressor pumps a refrigerant under pressure and wherein said controller produces said variable duty cycle control signal with a variable cycle time that is a function of fluctuations in refrigerant pressure.

3. The control system of claim 1 wherein said load sensor is a temperature sensor.

4. The control system of claim 1 further comprising pressure sensor coupled to said controller.

5. The control system of claim 1 wherein said load sensor is a temperature sensor and further comprising pressure sensor coupled to said controller.

6. The control system of claim 1 wherein said load sensor senses a property having a first rate of change and wherein said control system further comprising second load sensor that senses a second property having a second rate of change, said second rate of change being substantially greater than said first rate of change.

7. The control system of claim 6 wherein said load sensor senses changes in temperature and wherein said second load sensor senses changes in pressure.

8. The control system of claim 1 wherein said compressor includes fluid compressing members and mechanism for selectively rendering said compressing members effective and ineffective to thereby assume said first state and said second state, respectively.

9. The control system of claim 1 wherein said compressor includes an electric motor and wherein said compressor is configured such that said electric motor remains energized in said first and second states.

10. The control system of claim 1 wherein said compressor has two mechanical elements separated by a seal, said mechanical elements being movable relative to one another to develop fluid pressure and wherein said compressor includes mechanism to selectively break said seal in response to said control signal to thereby alter said fluid pressure developed while allowing said mechanical elements to maintain substantially constant relative movement with one another.

11. The control system of claim 1 wherein said compressor is a scroll compressor.

12. The control system of claim 10 wherein said compressor is a scroll compressor and said two mechanical elements are scroll members.

13. A refrigeration system comprising,

a refrigeration case;

an evaporator disposed in said case;

a condenser and compressor coupled in fluid communication with said evaporator;

said condenser being disposed on said case and having means for transferring heat to ambient;

said compressor comprising a pulse width modulated variable capacity compressor;

a load sensor disposed in said case

a controller responsive to said load sensor and coupled to said compressor for providing a variable duty cycle control signal to said compressor whereby said compressor is modulated between first capacity and second capacity states while operating to thereby adjust the operating capacity of the refrigeration system to its thermal load.

14. The refrigeration system of claim 13 further comprising a second refrigeration case having a second evaporator disposed therein, said second evaporator being coupled in fluid communication with said condenser.

15. The refrigeration system of claim 13 further comprising a second refrigeration case having a second evaporator disposed therein, said second evaporator being coupled in fluid communication with said compressor.

16. The refrigeration system of claim 13 wherein said load sensor is a temperature sensor.

17. The refrigeration system of claim 13 wherein said controller produces said variable duty cycle control signal with a variable cycle time that is a function of fluctuations in pressure.

18. The refrigeration system of claim 13 further comprising pressure sensor coupled to said controller.

19. The refrigeration system of claim 13 further comprising refrigerant regulator valve coupled in fluid communication with said evaporator and being electrically controlled by said variable duty cycle control signal to further adjust the capacity of said refrigeration system.

20. The refrigeration system of claim 19 wherein said valve includes a stepper motor.

21. A cooling system comprising,

a housing;

an evaporator disposed in said housing;

a condenser and compressor coupled in fluid communication with said evaporator;

said condenser being disposed on said housing and having means for transferring heat to ambient;

said compressor comprising a pulse width modulated variable capacity compressor;

a load sensor disposed in said case

a controller responsive to said load sensor and coupled to said compressor for providing a variable duty cycle control signal to said compressor whereby said compressor is modulated between first capacity and second capacity states while operating to thereby adjust the operating capacity of the cooling system to its thermal load.

22. The cooling system of claim 21 further comprising a second housing having a second evaporator disposed therein, said second evaporator being coupled in fluid communication with said condenser.

23. The cooling system of claim 21 further comprising a second housing having a second evaporator disposed therein, said second evaporator being coupled in fluid communication with said compressor.

24. The cooling system of claim 21 wherein said load sensor is a temperature sensor.

25. The cooling system of claim 21 wherein said controller produces said variable duty cycle control signal with a variable cycle time that is a function of fluctuations in pressure.

26. The cooling system of claim 21 further comprising pressure sensor coupled to said controller.

27. The cooling system of claim 21 further comprising refrigerant regulator valve coupled in fluid communication with said evaporator and being electrically controlled by said variable duty cycle control signal to further adjust the capacity of said refrigeration system.

28. An adaptive self-tuning controller for a refrigerant pressure effecting component of refrigeration system, comprising:

a control processor coupled to said pressure effecting component for providing closed loop control over said component based on an error signal;

said control processor including memory for storing first processing instructions for performing a control algorithm that includes at least one programmably adjustable gain factor;

said control processor including memory for storing second processing instructions for adaptively changing said programmably adjustable gain factor by periodically generating a new gain factor;

said second processing instructions causing said control processor to:

(a) monitor fluctuation in said error signal and generate a numeric value indicative of the percent fluctuation in said error signal over a predetermined time;

(b) fuzzify said numeric value by applying a membership function to generate a set of fuzzy input values indicative of degree of membership;

(c) apply a predetermined set of rules upon said fuzzy input values to generate a set of fuzzy output values, the predetermined set of rules reflecting the degree to which the gain factor associated with a given member of said membership function is to be changed; and

(d) defuzzifying said fuzzy output values by a combining operation yielding said new gain factor.

29. The controller of claim 28 wherein said first processing instructions cause said control processor to effect proportional closed loop control and wherein said gain factor is a proportioning constant.

30. The controller of claim 29 wherein said first processing instructions further cause said control processor to effect integral closed loop control.

31. The controller of claim 28 wherein said first processing instructions cause said control processor to effect integral closed loop control and wherein said gain factor is an integrating constant.

32. The controller of claim 28 wherein said membership function defines a plurality of linguistic variables representing different ranges of error values.

33. The controller of claim 32 wherein said linguistic variables include at least three linguistic variables representing predefined, LARGE, MEDIUM and SMALL error value ranges.

34. The controller of claim 28 wherein said membership function defines at least three overlapping ranges of membership function values such that at least some of said numeric values indicative of percent fluctuation map onto more than one membership function value.

35. The controller of claim 28 wherein said second processing instructions further cause said control processor to monitor direction of fluctuation in said error signal and wherein said predetermined set of rules includes a first subset of rules for positive error signal fluctuations and a second subset of rules for negative error signal fluctuations.

36. The controller of claim 28 wherein said second processing instructions cause said controller to defuzzify said fuzzy output values by calculating the centroid of said fuzzy output values to derive a multiplier that is applied to said programmably adjustable gain factor to yield said new gain factor.

37. The controller of claim 28 wherein said pressure effecting component is a compressor.

38. The controller of claim 28 wherein said pressure effecting component is a valve.

39. The controller of claim 28 wherein said pressure effecting component is a stepper regulator valve.

40. A compressor control system, comprising:

a compressor that is selectively operable in at least two states, including a first state corresponding to a first capacity and a second state corresponding to a second capacity lower than said first capacity;

a load sensor for sensing a property indicative of demand for cooling;

a controller coupled to said load sensor for producing a control signal that cycles based on said sensed property and at a cycle time shorter than the time constant of the load;

said controller being coupled to said compressor for causing said compressor to selectively alternate between said first and second states in response to said control signal such that the cumulative effect of said selective alternation adjusts the capacity of said compressor to the demand for cooling.

41. The control system of claim 40 wherein said control signal is a variable duty cycle control signal and wherein said duty cycle is varied in response to said sensed property.

42. The control system of claim 40 wherein said compressor pumps a refrigerant under pressure and wherein said controller produces said control signal with a variable cycle time that is a function of fluctuations in refrigerant pressure.

43. The control system of claim 40 wherein said load sensor is a temperature sensor.

44. The control system of claim 40 further comprising pressure sensor coupled to said controller.

45. The control system of claim 40 wherein said load sensor is a temperature sensor and further comprising pressure sensor coupled to said controller.

46. The control system of claim 40 wherein said load sensor senses a property having a first rate of change and wherein said control system further comprising second load sensor that senses a second property having a second rate of change, said second rate of change being substantially greater than said first rate of change.

47. The control system of claim 46 wherein said load sensor senses changes in temperature and wherein said second load sensor senses changes in pressure.

48. The control system of claim 40 wherein said compressor includes fluid compressing members and mechanism for selectively rendering said compressing members effective and ineffective to thereby assume said first state and said second state, respectively.

49. The control system of claim 40 wherein said compressor includes an electric motor and wherein said compressor is configured such that said electric motor remains energized in said first and second states.

50. The control system of claim 40 wherein said compressor has two mechanical elements separated by a seal, said mechanical elements being movable relative to one another to develop fluid pressure and wherein said compressor includes mechanism to selectively break said seal in response to said control signal to thereby alter said fluid pressure developed while allowing said mechanical elements to maintain substantially constant relative movement with one another.

51. The control system of claim 40 wherein said compressor is a scroll compressor.

52. The control system of claim 50 wherein said compressor is a scroll compressor and said two mechanical elements are scroll members.

53. The control system of claim 40 wherein said controller is coupled to said compressor such that said compressor selectively alternates between said first and second states at said cycle time.

54. The control system of claim 40 wherein said controller produces a control signal that cycles at a cycle time at least four times shorter than the time constant of the load.

55. The compressor control system of claim 1 wherein controller provides said variable duty cycle control signal that cycles at a cycle time shorter than the time constant of the load.

56. The compressor control system of claim 13 wherein controller provides said variable duty cycle control signal that cycles at a cycle time shorter than the time constant of the load.

57. The compressor control system of claim 1 wherein said first state is substantially at one hundred percent capacity and wherein said second state is substantially at zero percent capacity.

58. The compressor control system of claim 13 wherein said first state is substantially at one hundred percent capacity and wherein said second state is substantially at zero percent capacity.

59. The compressor control system of claim 21 wherein said first state is substantially at one hundred percent capacity and wherein said second state is substantially at zero percent capacity.

60. A diagnostic system for a compressor controller, comprising:

a controller adapted for coupling to a compressor, said controller producing a variable duty cycle control signal for adjusting the capacity of said compressor, in which the duty cycle is a function of demand for cooling;

a diagnostic module coupled to said controller for monitoring and comparing said duty cycle with at least one predetermined fault value indicative of a system fault condition; and

an alert module responsive to said diagnostic module for issuing an alert signal when said duty cycle bears a predetermined relationship to said fault value.

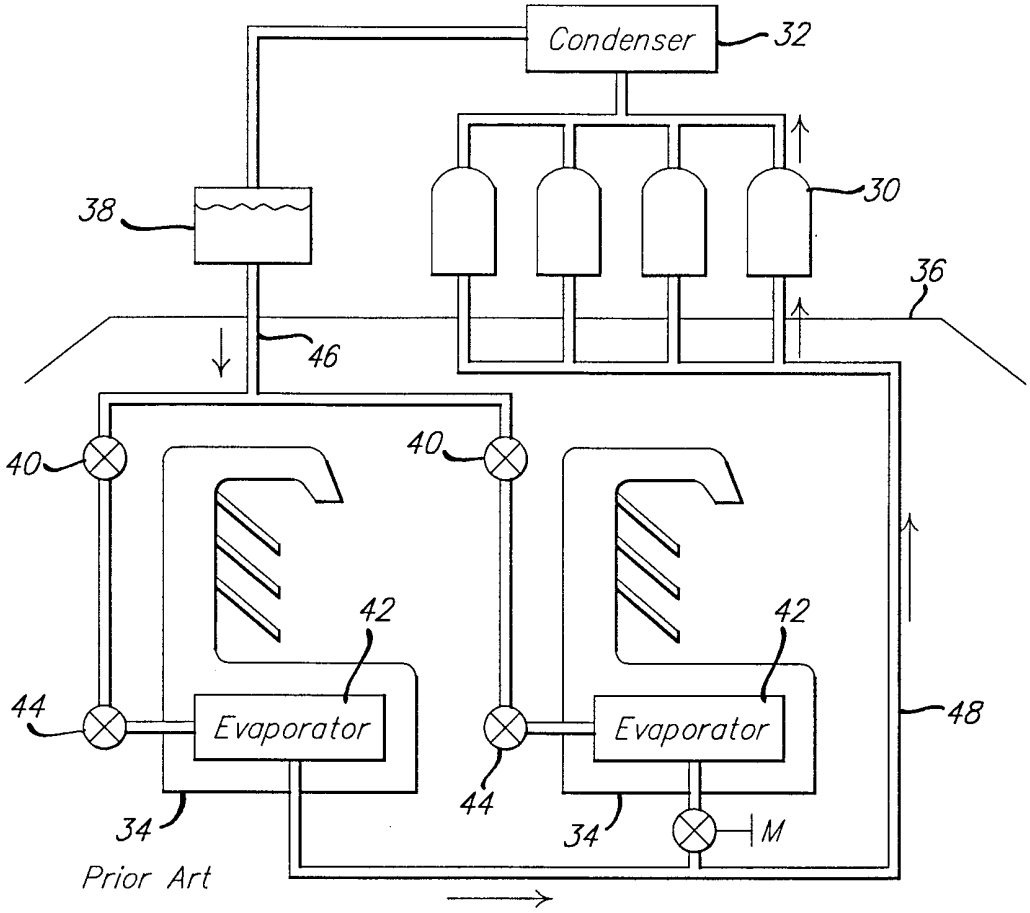

Condenser
32
38
30
36
46
40
40
42
42
44
Evaporator
Evaporator
48
44
34
34
M
Prior Art

Fig. 1.

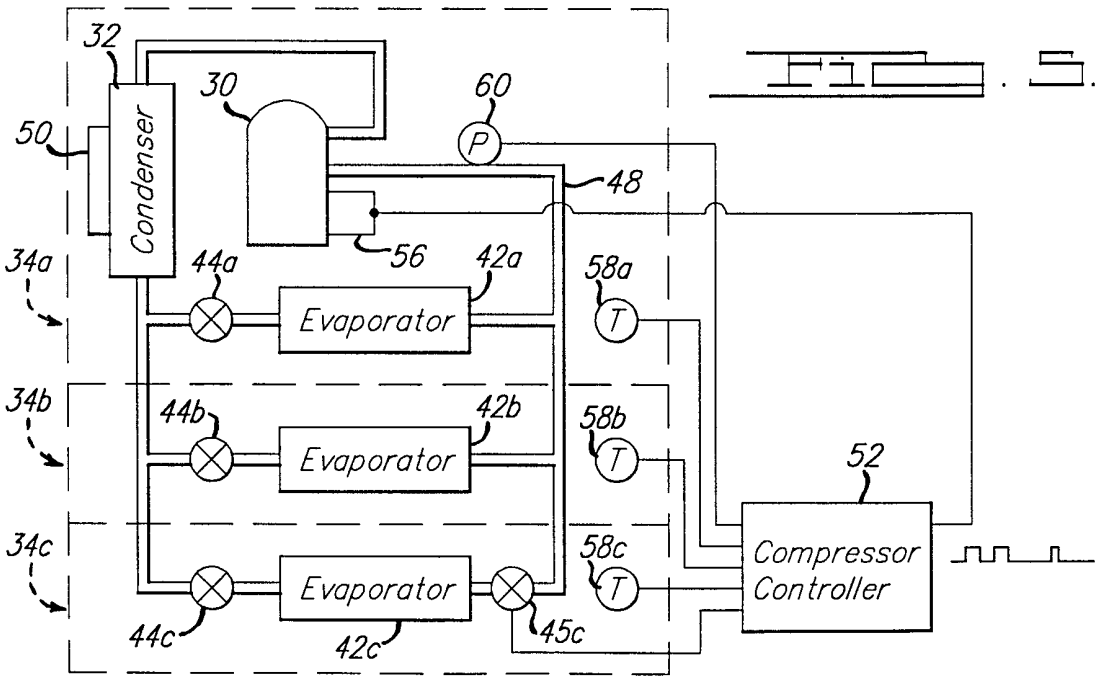

32
30
60
50
Condenser
P
48
56
34a
44a
42a
Evaporator
58a
T
34b
44b
42b
Evaporator
58b
T
52
34c
58c
Evaporator
T
Compressor
Controller
44c
42c
45c

Fig. 2.

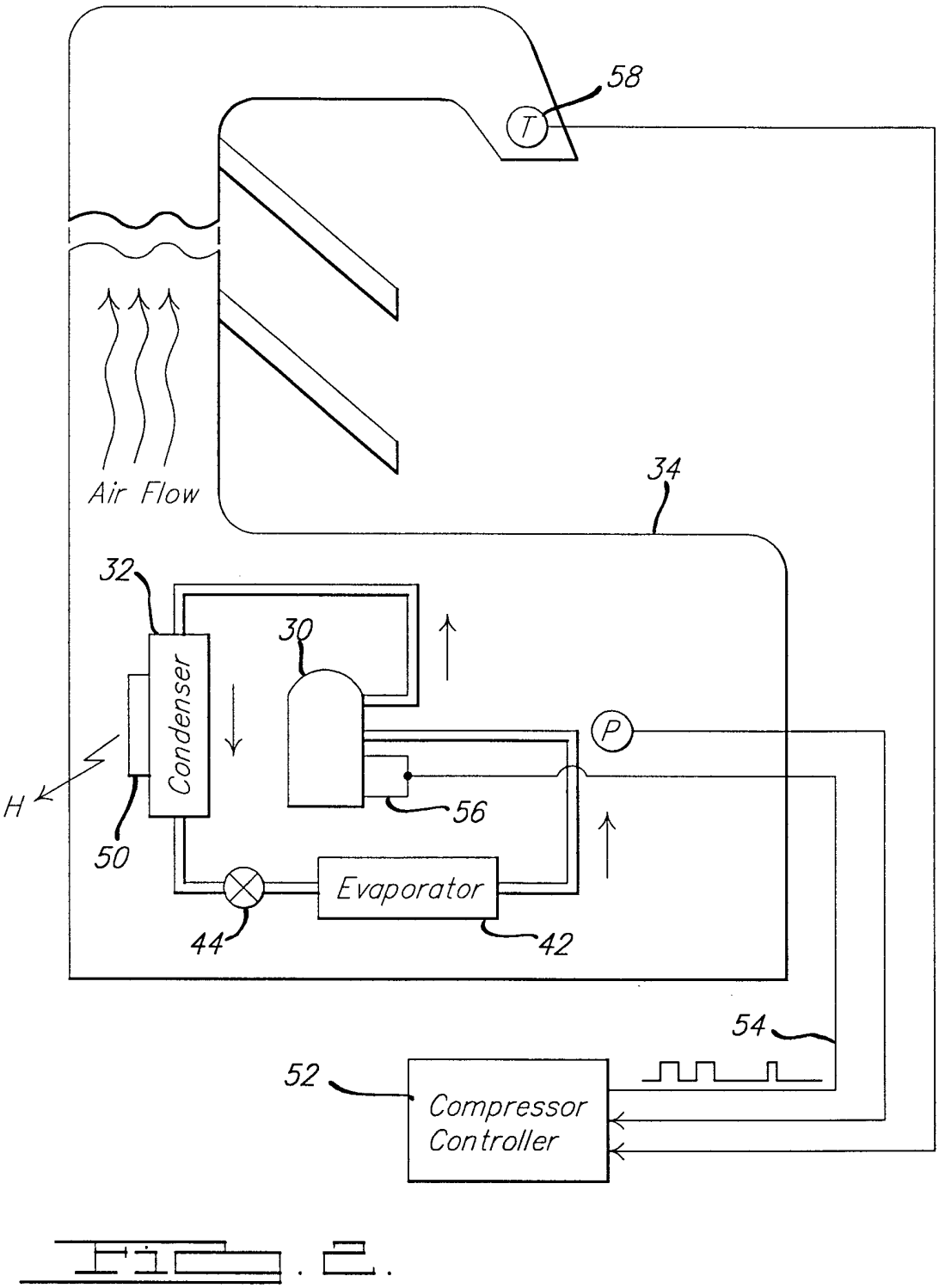

58
T
Air Flow
34
32
30
Condenser
P
H
56
50
Evaporator
44
42
54
52
Compressor
Controller

Fig. 2.

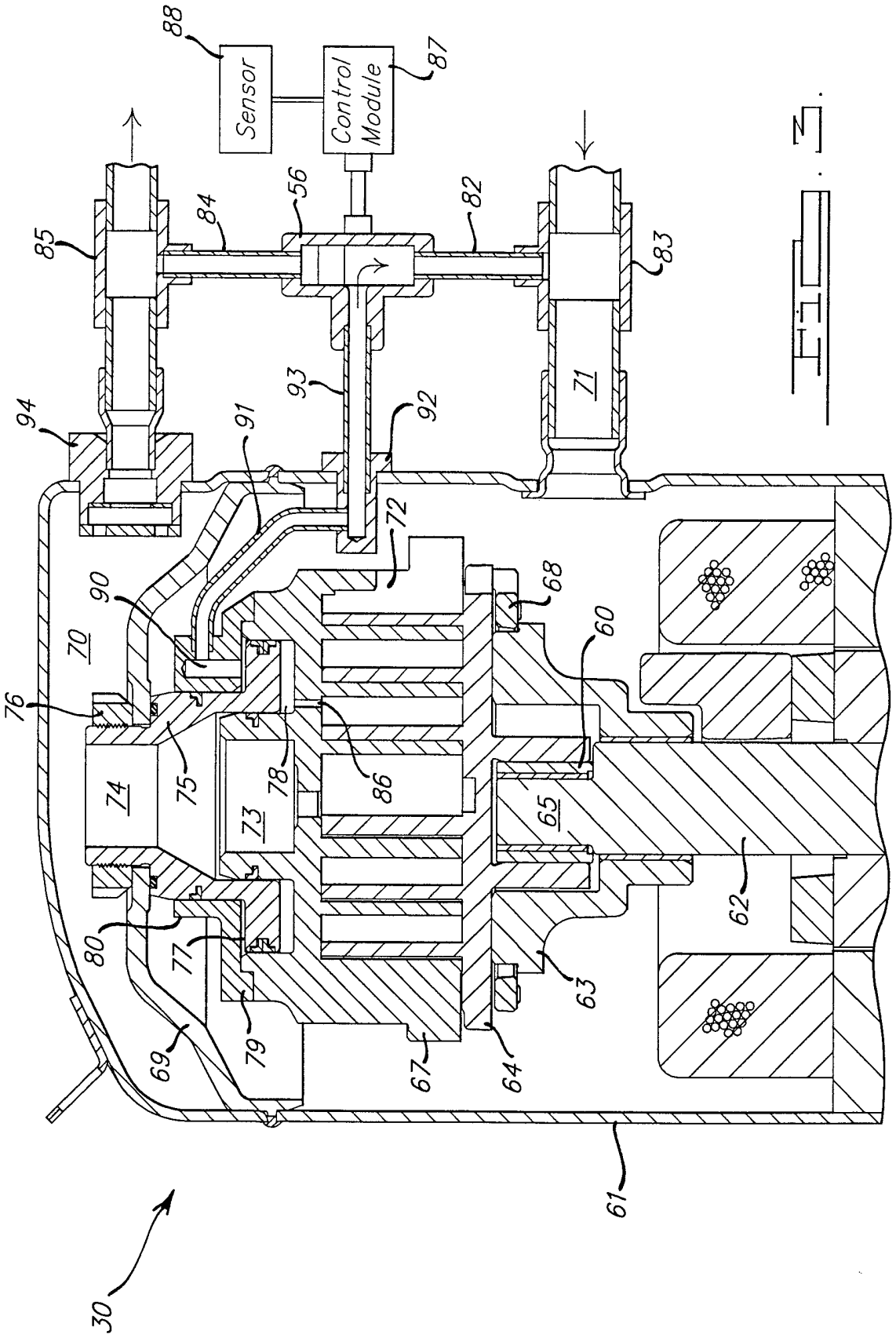

88
Sensor
Control Module
87
56
84
82
85
83
94
91
93
92
71
72
90
68
60
70
76
75
74
78
86
73
65
62
80
77
63
79
69
67
64
61
30

Fig. 3

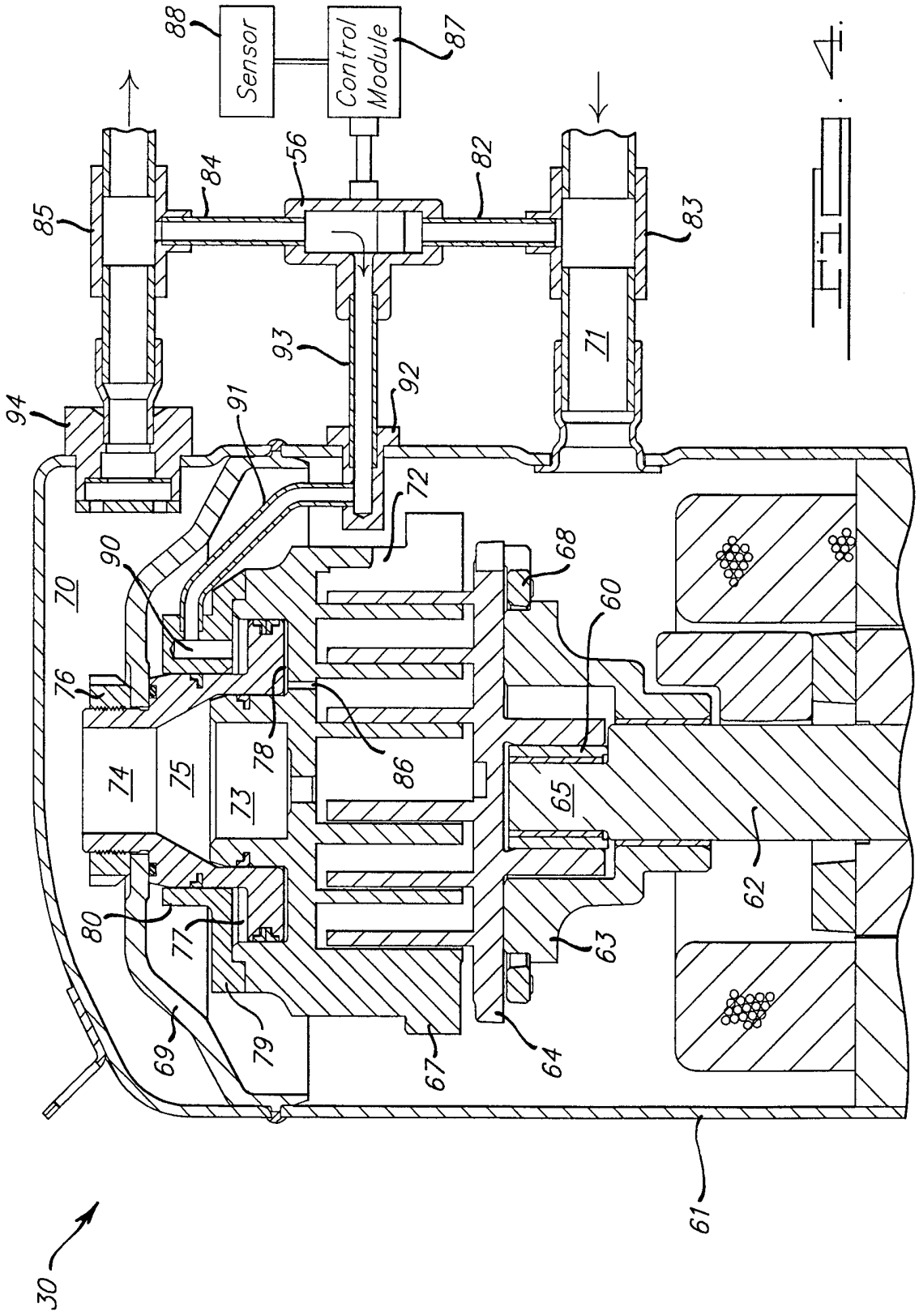
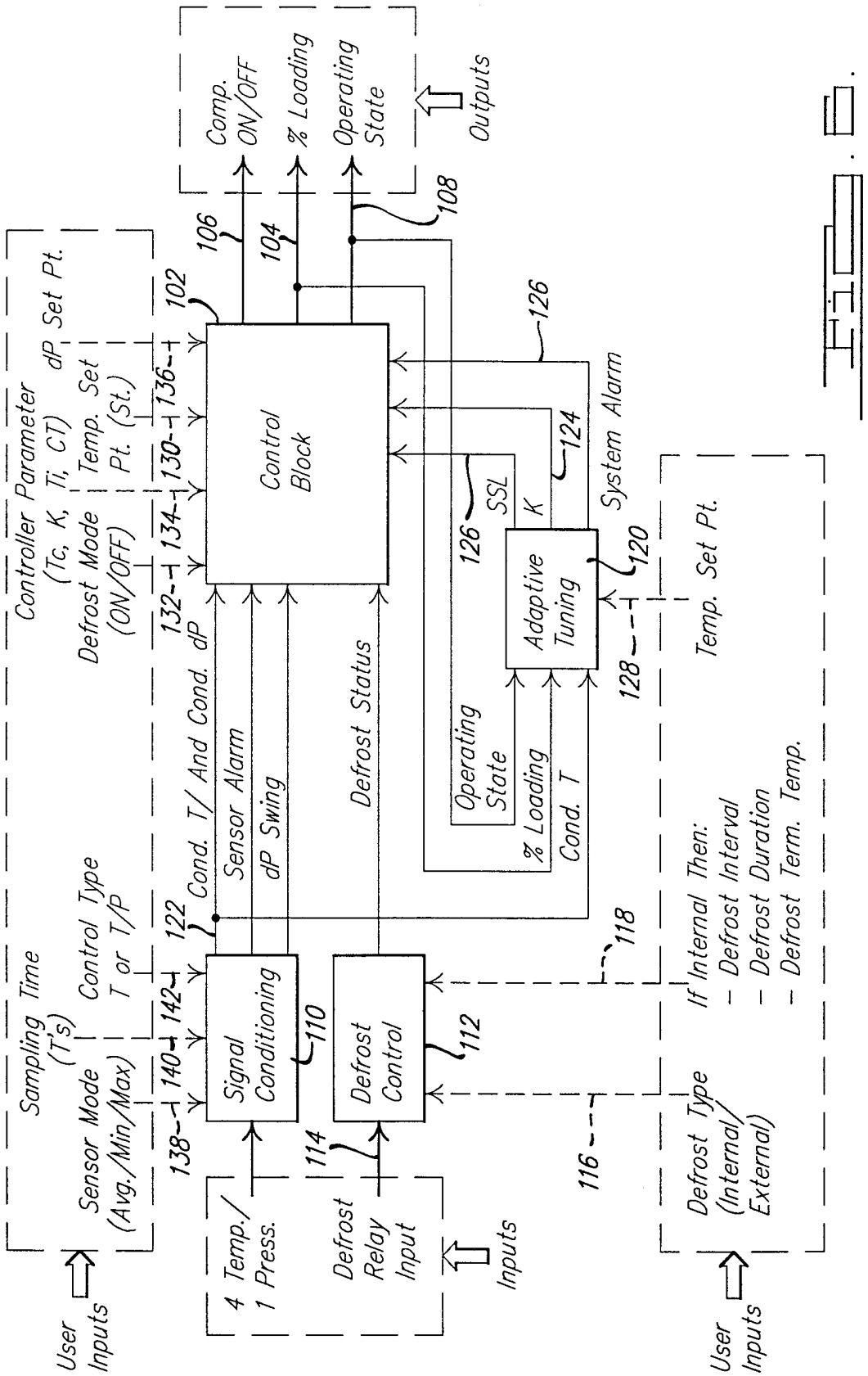

Sensor
Control Module
88
87
84
56
82
85
83
71
93
92
91
94
72
90
70
68
60
76
74
75
73
78
86
65
62
80
77
63
69
79
67
64
61
30

Fig. 4


User Inputs
Sampling Time (T's)
Sensor Mode (Avg./Min/Max)
Control Type T or T/P
Controller Parameter (Tc, K, Ti, CT)
Defrost Mode (ON/OFF)
dP Set Pt.
Temp. Set Pt. (St.)
Cond. T/ And Cond. dP
Sensor Alarm
dP Swing
Defrost Status
Operating State
% Loading
Cond. T
SSL
K
System Alarm
Temp. Set Pt.
If Internal Then:
- Defrost Interval
- Defrost Duration
- Defrost Term. Temp.
Defrost Type (Internal/ External)
4 Temp./ 1 Press.
Defrost Relay Input
Inputs
Signal Conditioning
Defrost Control
Control Block
Adaptive Tuning
Comp. ON/OFF
% Loading
Operating State
Outputs
102
104
106
108
110
112
114
116
118
120
122
124
126
128
130
132
134
136
138
140
142

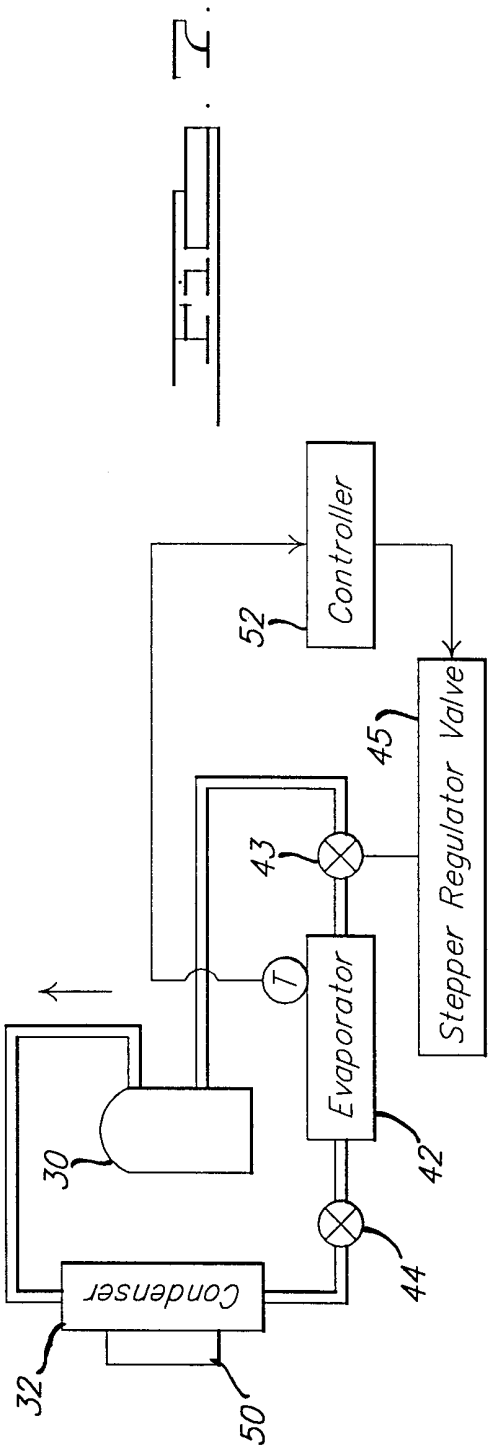

Controller
52
Stepper Regulator Valve
45
43
T
Evaporator
42
44
30
Condenser
32
50

Fig. 7.

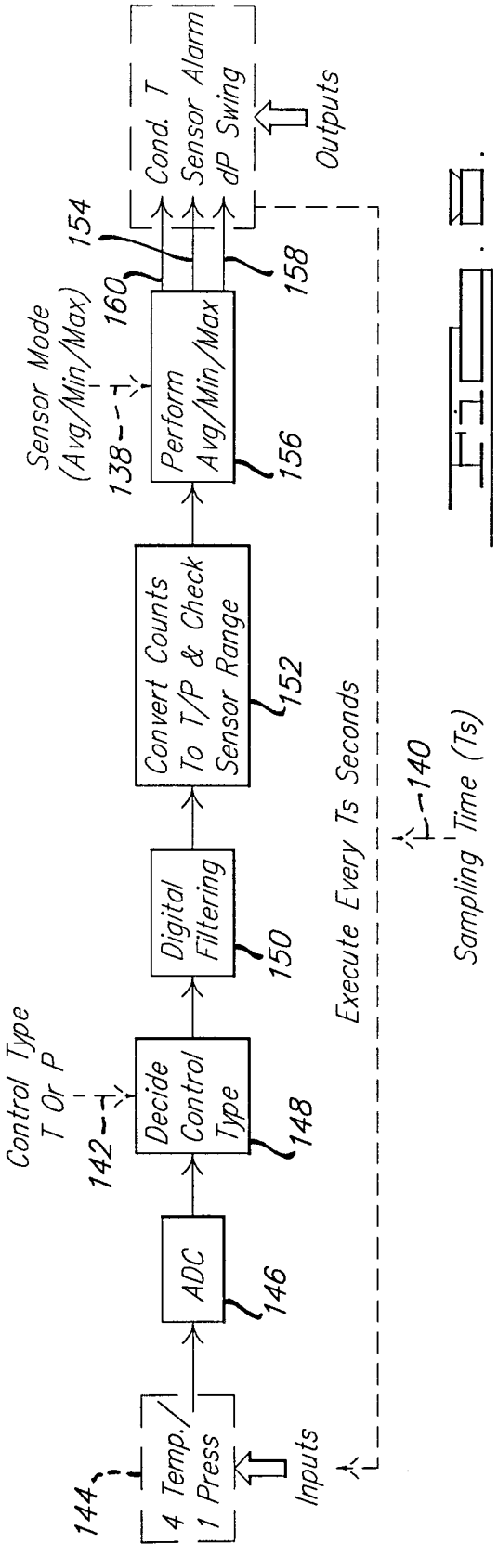

Sensor Mode
(Avg/Min/Max)
138
Control Type
T Or P
142
144
4 Temp./
1 Press
Inputs
ADC
146
Decide
Control
Type
148
Digital
Filtering
150
Convert Counts
To T/P & Check
Sensor Range
152
Perform
Avg/Min/Max
156
160
154
Cond. T
Sensor Alarm
dP Swing
158
Outputs
Execute Every Ts Seconds
140
Sampling Time (Ts)

Fig. 8.

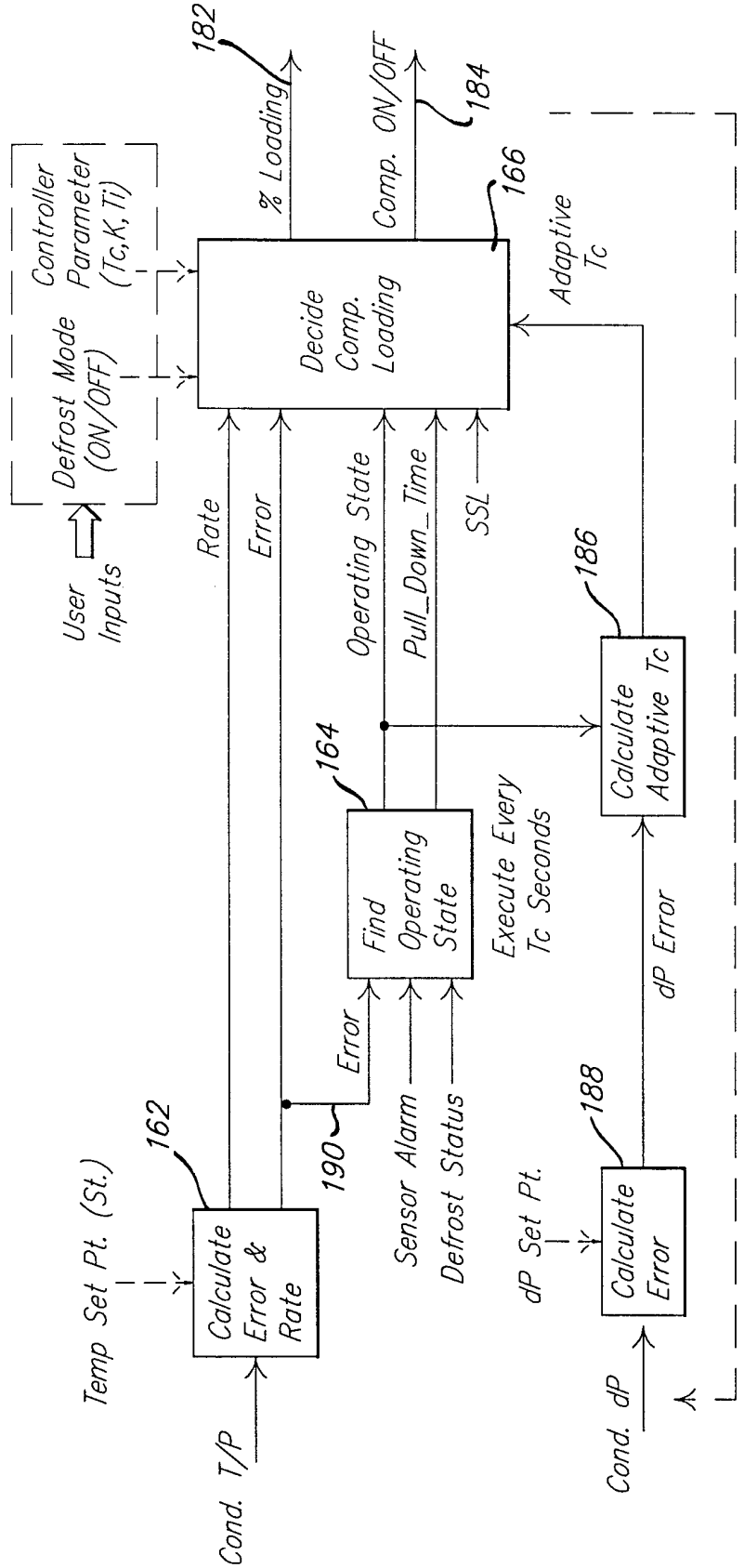

Controller Parameter (Tc,K,Ti)
Defrost Mode (ON/OFF)
User Inputs
182
% Loading
184
Comp. ON/OFF
166
Decide Comp. Loading
Adaptive Tc
Rate
Error
Operating State
Pull_Down_Time
SSL
186
Calculate Adaptive Tc
164
Find Operating State
Execute Every Tc Seconds
dP Error
Error
Sensor Alarm
Defrost Status
190
188
Calculate Error
dP Set Pt.
Cond. dP
162
Calculate Error & Rate
Temp Set Pt. (St.)
Cond. T/P

Fig. 9.

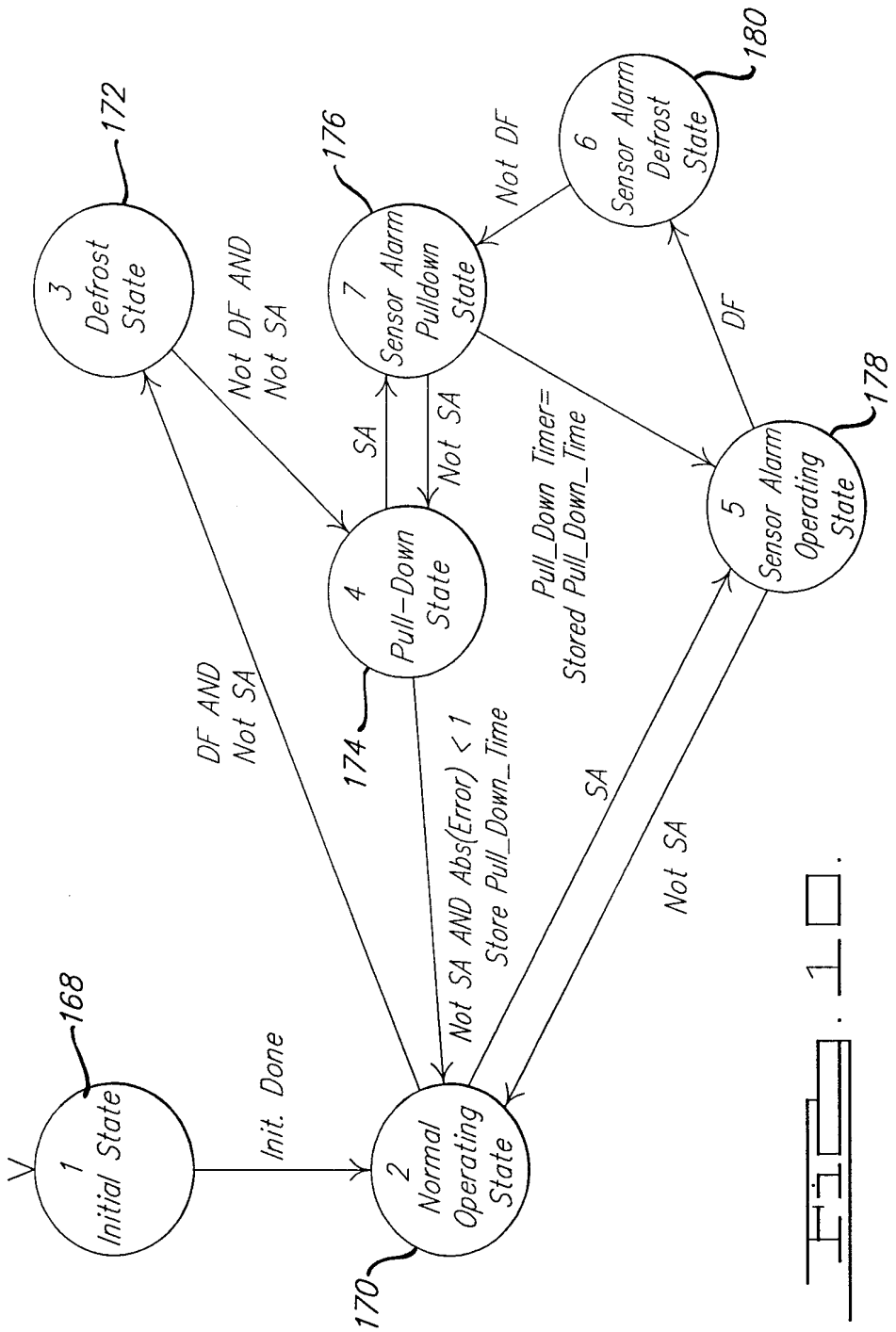

1 Initial State
168
2 Normal Operating State
170
3 Defrost State
172
4 Pull-Down State
174
5 Sensor Alarm Operating State
178
6 Sensor Alarm Defrost State
180
7 Sensor Alarm Pulldown State
176
Init. Done
DF AND Not SA
Not DF AND Not SA
SA
Not SA
Not SA AND Abs(Error) < 1 Store Pull_Down_Time
Pull_Down Timer= Stored Pull_Down_Time
SA
Not SA
Not DF
DF

Fig. 1

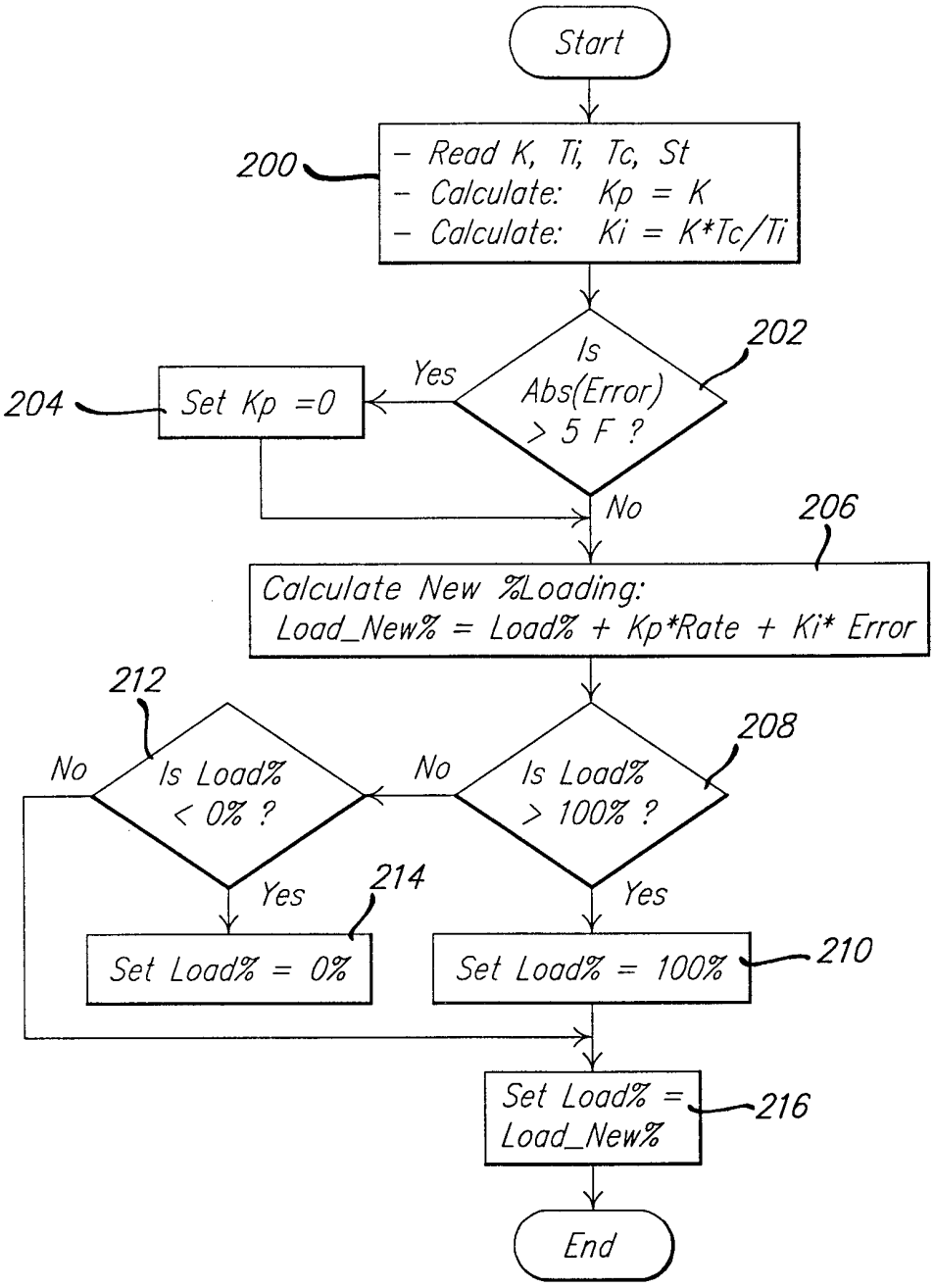

Start
200
- Read K, Ti, Tc, St
- Calculate: Kp = K
- Calculate: Ki = K*Tc/Ti
202
Is Abs(Error) > 5 F ?
Yes
204
Set Kp =0
No
206
Calculate New %Loading:
Load_New% = Load% + Kp*Rate + Ki* Error
212
No
Is Load% < 0% ?
208
No
Is Load% > 100% ?
Yes
214
Set Load% = 0%
Yes
Set Load% = 100%
210
Set Load% = Load_New%
216
End

Fig. 11.

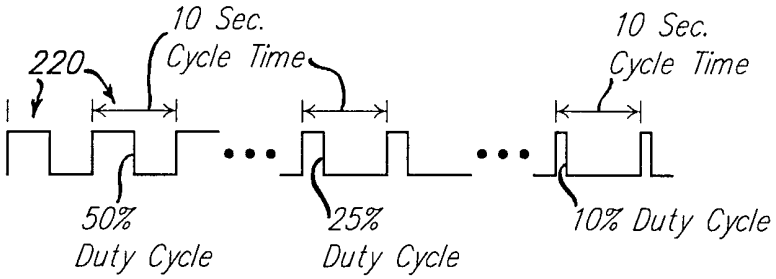


Fig. 12.

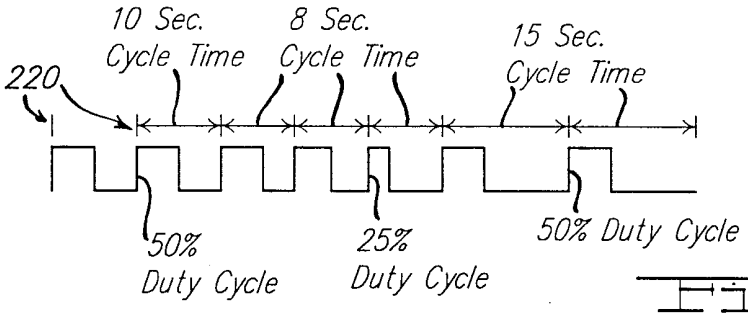


Fig. 13.

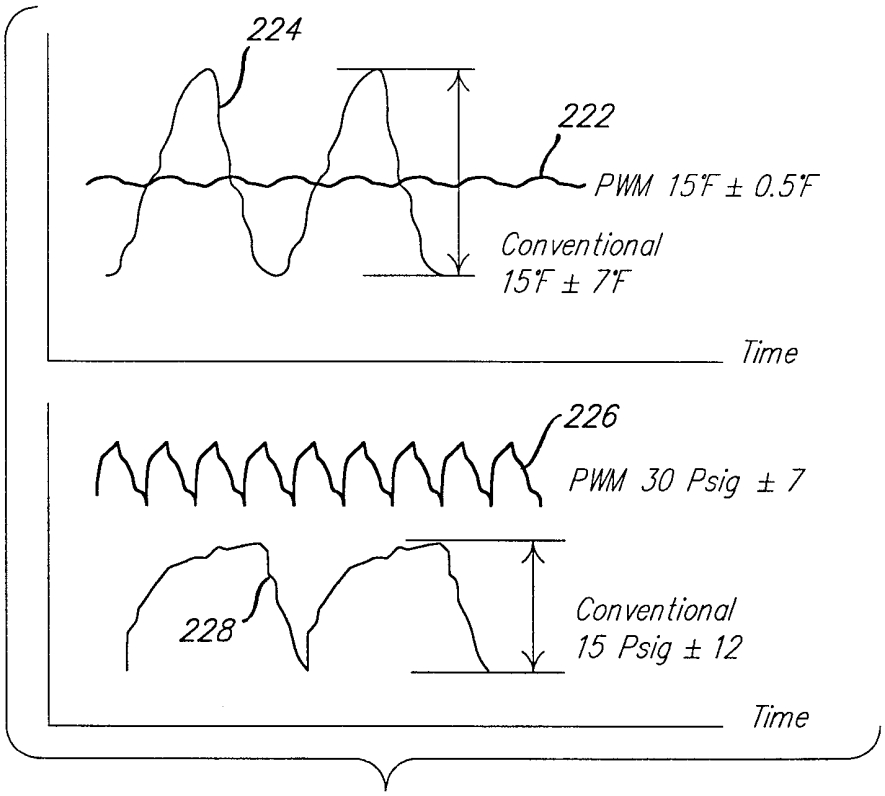


Fig. 14.

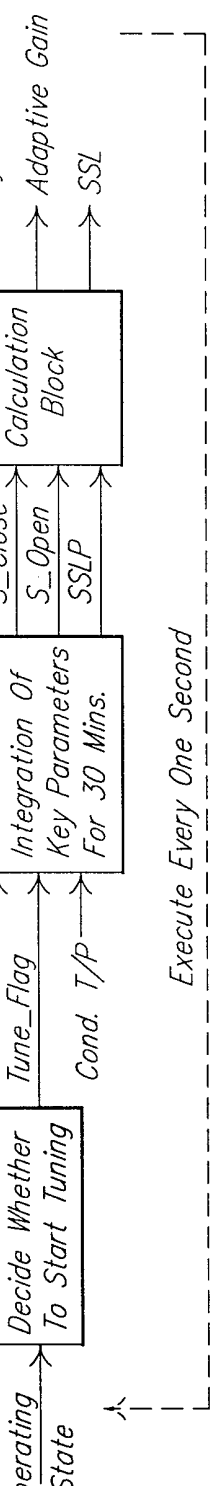

Control Set Pt. (St.)
Temp. Set Pt. (St.)
Operating State
240
Decide Whether To Start Tuning
242
Perform Integration Of Key Parameters For 30 Mins.
244
Calculation Block
% Loading
Tune_Flag
Cond. T/P
S_ER
S_Close
S_Open
SSLP
System_Alarm
Adaptive Gain
SSL
Execute Every One Second

Fig. 15.

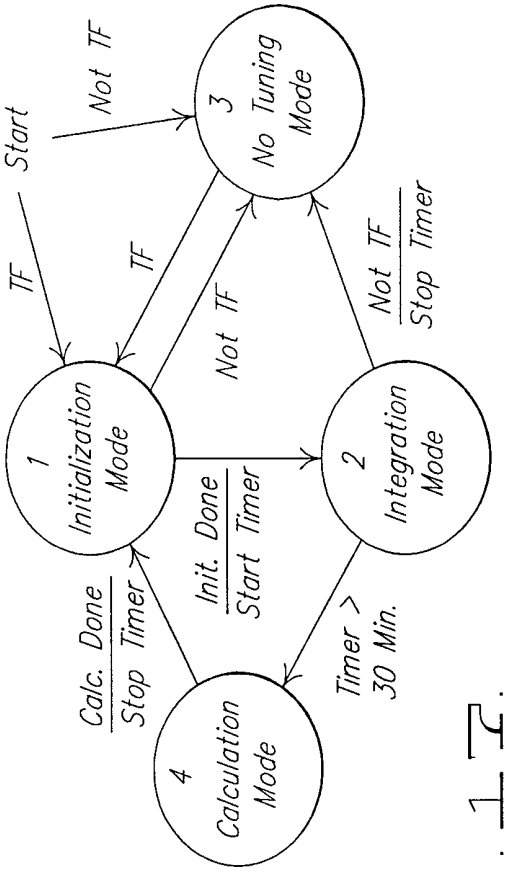

1 Initialization Mode
2 Integration Mode
3 No Tuning Mode
4 Calculation Mode
Start
TF
Not TF
Init. Done / Start Timer
Not TF / Stop Timer
TF
Not TF
Calc. Done / Stop Timer
Timer > 30 Min.

Fig. 17.

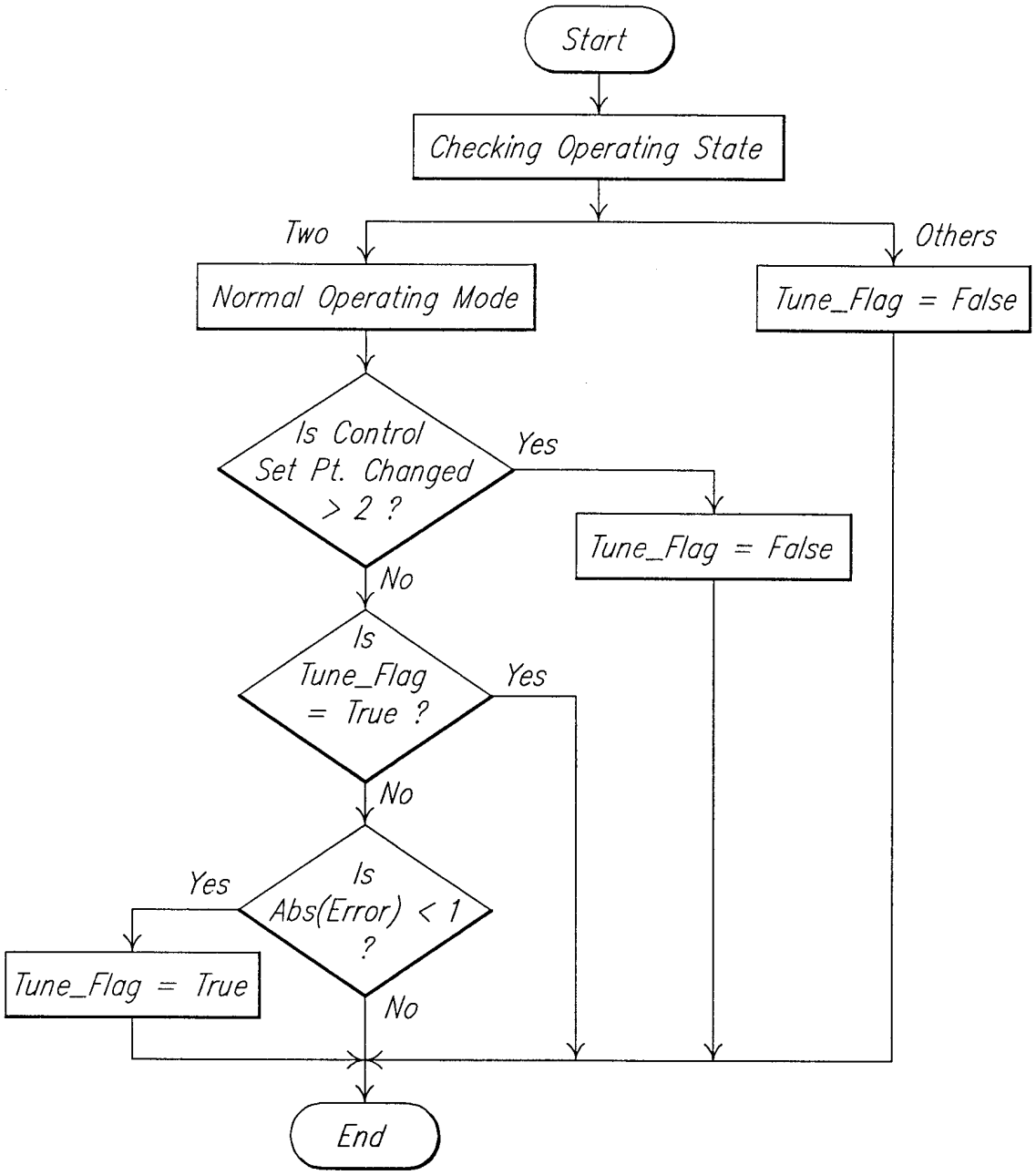

Start
Checking Operating State
Two
Others
Normal Operating Mode
Tune_Flag = False
Is Control Set Pt. Changed > 2 ?
Yes
Tune_Flag = False
No
Is Tune_Flag = True ?
Yes
No
Is Abs(Error) < 1 ?
Yes
Tune_Flag = True
No
End

FIG. 16A.

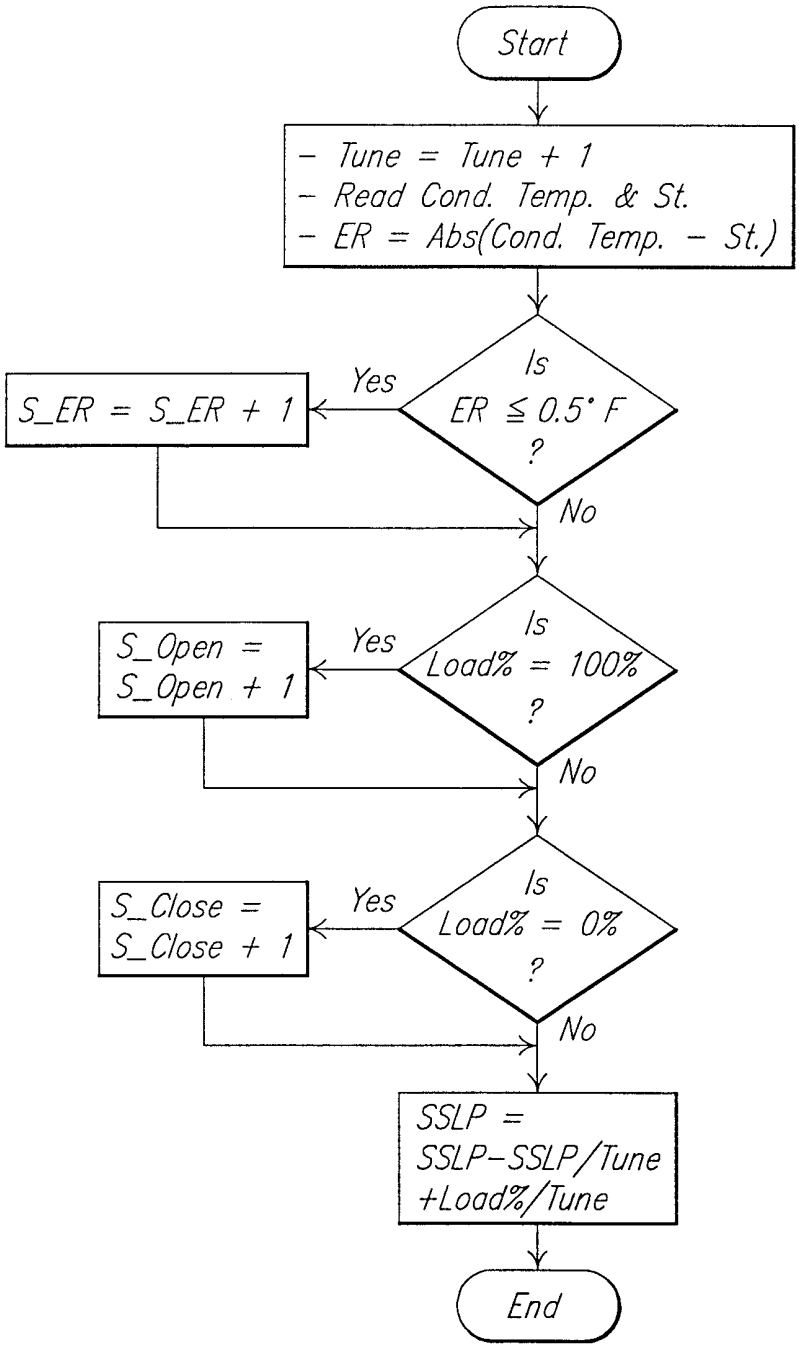

Start
- Tune = Tune + 1
- Read Cond. Temp. & St.
- ER = Abs(Cond. Temp. - St.)
Is ER ≤ 0.5° F ?
Yes
S_ER = S_ER + 1
No
Is Load% = 100% ?
Yes
S_Open = S_Open + 1
No
Is Load% = 0% ?
Yes
S_Close = S_Close + 1
No
SSLP = SSLP-SSLP/Tune +Load%/Tune
End

Fig. 16B.

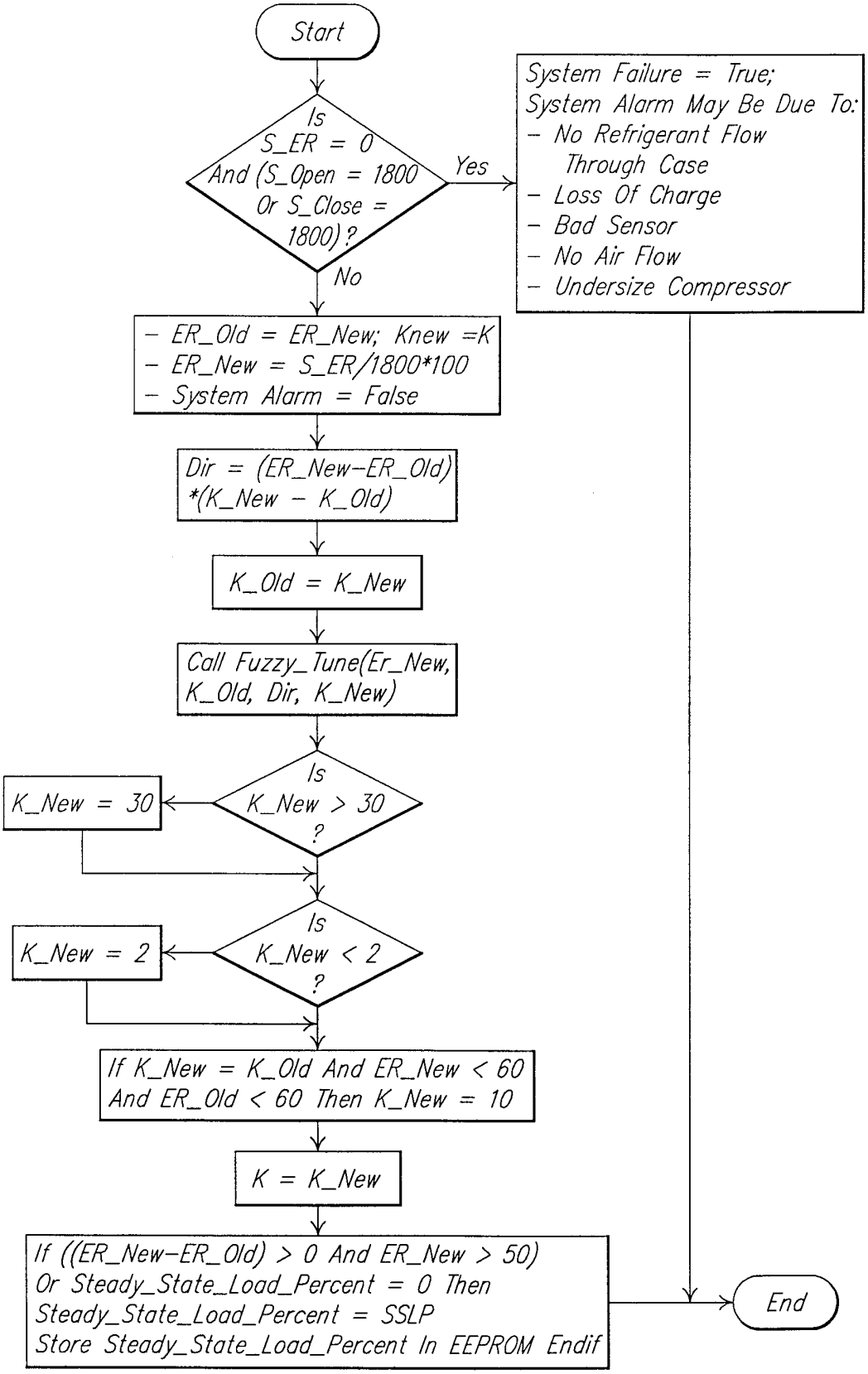
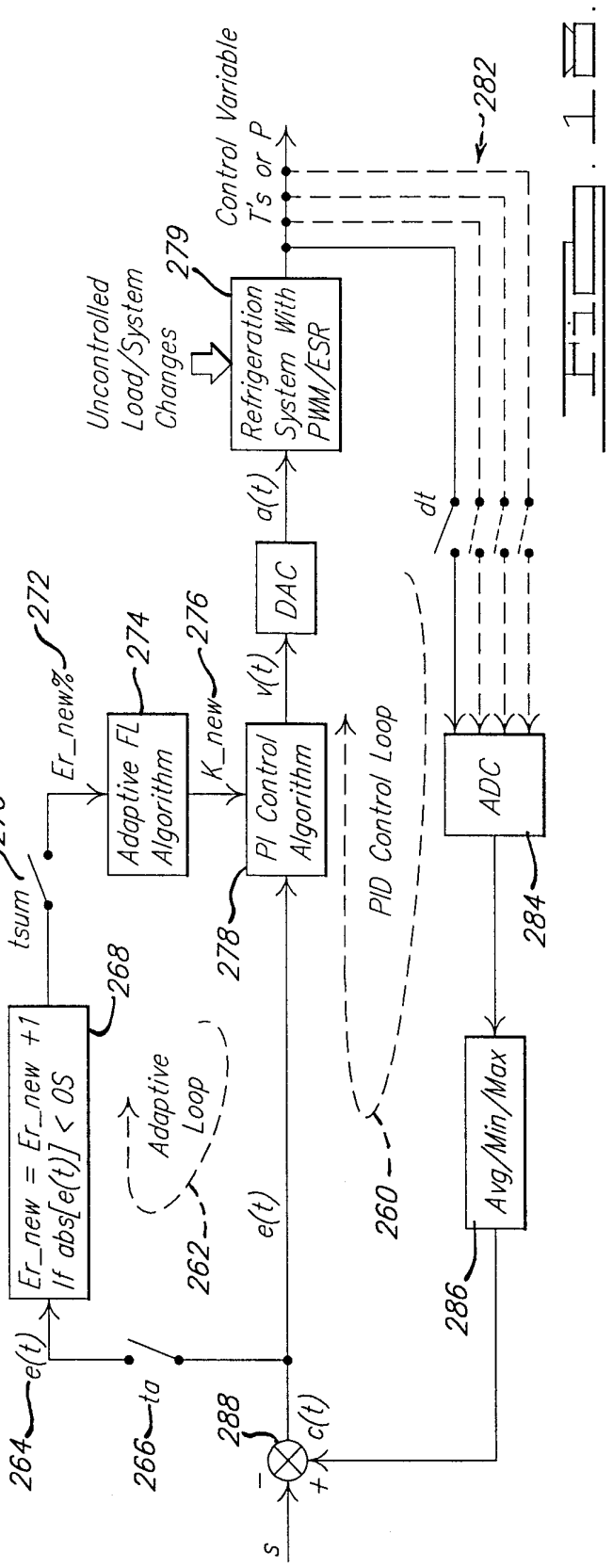
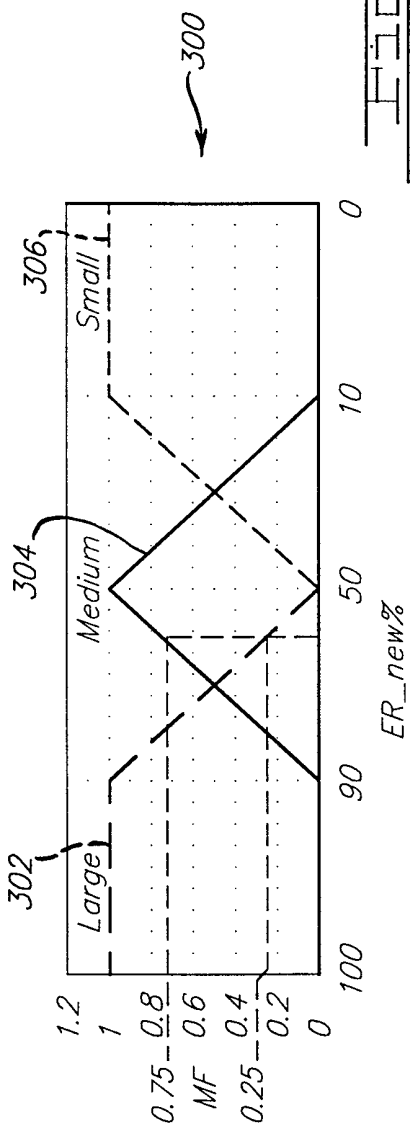

Start
Is S_ER = 0 And (S_Open = 1800 Or S_Close = 1800) ?
Yes
System Failure = True;
System Alarm May Be Due To:
- No Refrigerant Flow Through Case
- Loss Of Charge
- Bad Sensor
- No Air Flow
- Undersize Compressor
No
- ER_Old = ER_New; Knew =K
- ER_New = S_ER/1800*100
- System Alarm = False
Dir = (ER_New-ER_Old) *(K_New - K_Old)
K_Old = K_New
Call Fuzzy_Tune(Er_New, K_Old, Dir, K_New)
Is K_New > 30 ?
K_New = 30
Is K_New < 2 ?
K_New = 2
If K_New = K_Old And ER_New < 60 And ER_Old < 60 Then K_New = 10
K = K_New
If ((ER_New-ER_Old) > 0 And ER_New > 50) Or Steady_State_Load_Percent = 0 Then Steady_State_Load_Percent = SSLP
Store Steady_State_Load_Percent In EEPROM Endif
End

Fig. 16C.


Control Variable
T's or P
282
279
Uncontrolled
Load/System
Changes
Refrigeration
System With
PWM/ESR
a(t)
DAC
dt
v(t)
PI Control
Algorithm
K_new
276
Adaptive FL
Algorithm
274
Er_new%
272
270
tsum
268
Er_new = Er_new +1
If abs[e(t)] < OS
Adaptive
Loop
262
e(t)
260
PID Control Loop
278
ADC
284
Avg/Min/Max
286
264
e(t)
266
ta
288
c(t)
s


300
306
Small
304
Medium
302
Large
0
10
50
90
100
ER_new%
1.2
1
0.75
0.8
MF
0.6
0.25
0.4
0.2
0

	DIR(I)	1	2
Er_new% (J)		Negative/Zero	Positive
1	Large	NC	NC
2	Medium	NSC	PSC
3	Small	NBC	PBC

Fig. 20.

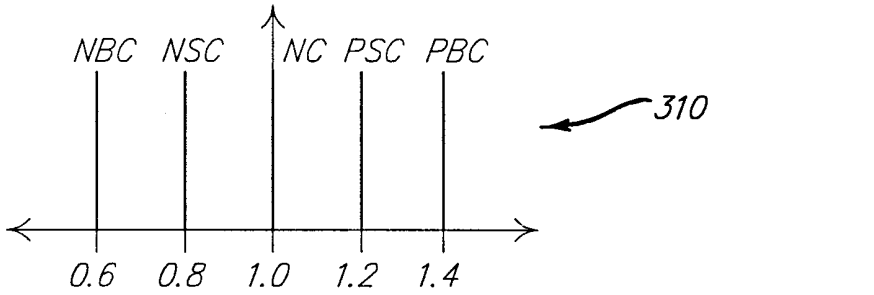


Fig. 21.

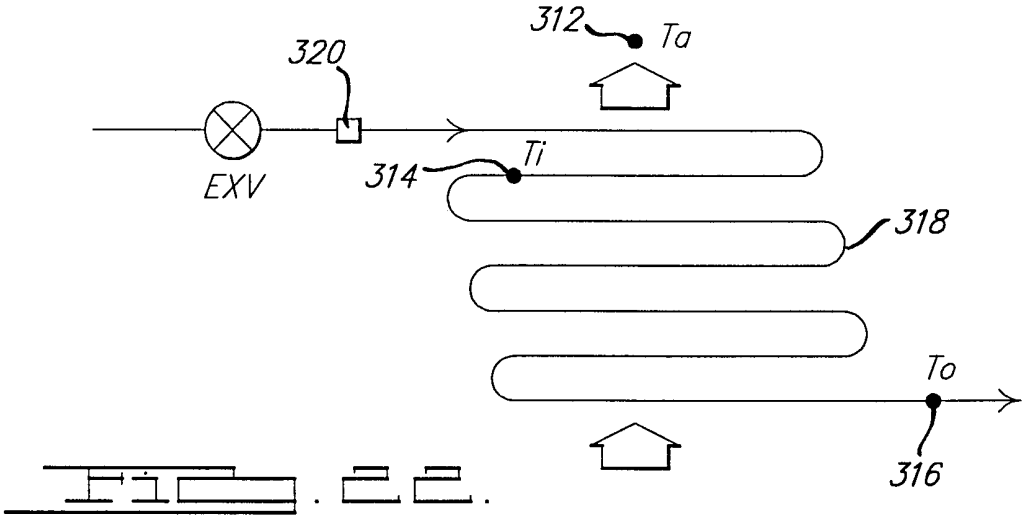


Fig. 22.

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US98/18710

A. CLASSIFICATION OF SUBJECT MATTER

IPC(6) : F25B 49/02, 41/04
US CL : 62/ 126, 204, 228.3, 229

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

U.S. : 62/ 126, 127, 129, 196.1, 196.2, 196.3, 203, 204, 228.3, 229; 236/1EA

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X ---- Y	US 4,494,383 A (NAGATOMO et al) 22 January 1985, col. 3, lines 20-44.	1,3,8,10,21,24,40,41,50,53-55,57,59 ---------- 2,4-7,9,11-20,22,23,25,27,51,52,56,58
X ---- Y	US 5,440,891 A (HINDMON, Jr. et al) 15 August 1995, col. 3, lines 20-68.	28-37,40,42-48 ---------- 2,4-7,13-20,22,23,25-27,38-39,56,58

[X] Further documents are listed in the continuation of Box C. [ ] See patent family annex.

*	Special categories of cited documents:	"T"	later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"A"	document defining the general state of the art which is not considered to be of particular relevance	"X"	document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"E"	earlier document published on or after the international filing date	"Y"	document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"L"	document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)	"&"	document member of the same patent family
"O"	document referring to an oral disclosure, use, exhibition or other means		
"P"	document published prior to the international filing date but later than the priority date claimed		

Date of the actual completion of the international search 02 DECEMBER 1998	Date of mailing of the international search report 26 JAN 1999
Name and mailing address of the ISA/US Commissioner of Patents and Trademarks Box PCT Washington, D.C. 20231 Facsimile No. (703) 305-3230	Authorized officer HARRY TANNER Telephone No. (703) 308-0861

Form PCT/ISA/210 (second sheet)(July 1992) ★

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US98/18710

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X ---- Y	US 5,243,827 A (HAGITA et al) 14 September 1993, col. 4, lines 59-68.	40, 43, 49 ---------- 9,11,12,51,52
Y	US 4,506,518 A (YOSHIKAWA et al) 26 March 1985, col. 4, lines 30-42.	19,20,27,38,39
Y	US 4,463,576 A (BURNETT et al) 07 August 1984, col. 2, lines 8-32.	60