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(71) Demandeur/Applicant:
ATTRIDGE, RUSSELL G., JR., US

(72) Inventeur/Inventor:
ATTRIDGE, RUSSELL G., JR., US

(74) Agent: PIASETZKI & NENNIGER

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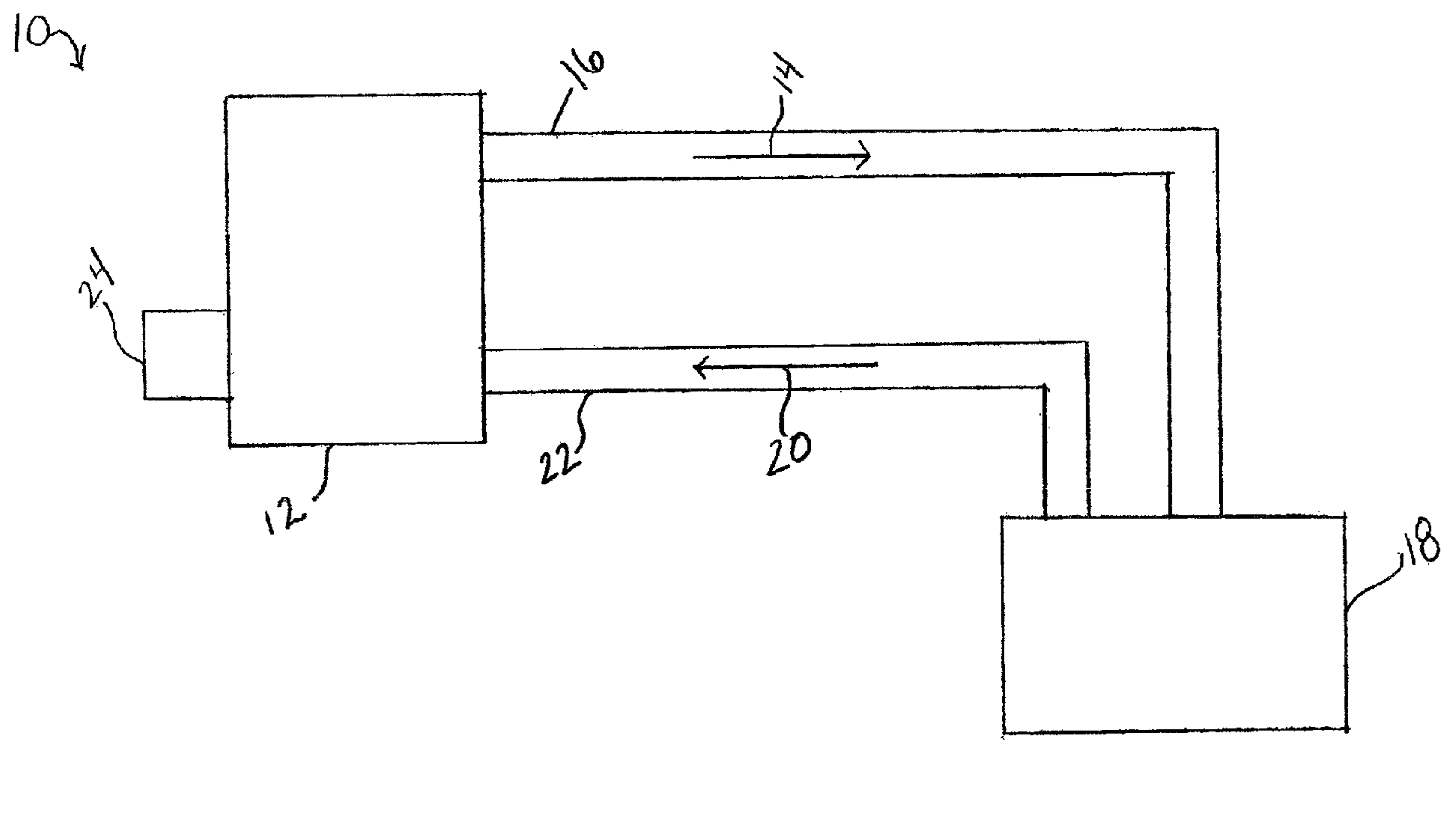


(57) Abrégé/Abstract:
A method and apparatus for controlling a temperature-regulated zone utilizing a thermal balance temperature control system. The thermal balance control system is a dynamic real time control system that measures the sensible thermal load in the zone, and directly regulates the BTU output of the HVAC package to balance such output with the measured sensible thermal load.

ABSTRACT

A method and apparatus for controlling a temperature-regulated zone utilizing a thermal balance temperature control system. The thermal balance control system is a dynamic real time control system that measures the sensible thermal load in the zone, and directly regulates the BTU output of the HVAC package to balance such output with the measured sensible thermal load.

Title: THERMAL BALANCE TEMPERATURE CONTROL SYSTEM

BACKGROUND OF THE INVENTION

The present invention relates to a temperature control system and, more particularly, to a system which directly regulates the system output to balance such output with the sensible thermal load.

Heating, ventilating and air-conditioning (HVAC) systems are used to both heat and cool the air within an enclosure, e.g., a building or zone within such building. An HVAC system typically includes a heating unit, a cooling unit, a supply air fan, a supply duct for directing air into the enclosure, and a return duct for removing air from the enclosure. It will be appreciated by those skilled in the art that HVAC systems are generally designed to operate in one of three modes: a heating mode to heat the enclosure, a cooling mode to cool the enclosure and a economizer mode to ventilate the enclosure. The economizer mode typically utilizes both an outdoor air damper and a return air damper, commonly referred to as an economizer, that can be selectively modulated opened to allow the return air to mix with fresh outside air.

There is typically a control system associated with an HVAC system, such control system including a thermostat (typically located within the enclosure) and associated hardware/software for controlling the components of the particular HVAC system in response to pre-programmed instructions. Typically, the control system allows a user to pre-select one of the three operating modes, as well as selecting a desired temperature for the enclosure. Thereafter, the control system activates either the heating or cooling portion of the HVAC system to maintain the pre-selected temperature within the enclosure. Under certain conditions the economizer mode may be able to maintain the enclosure at the pre-selected temperature.

When set in the cooling mode, the control system will provide cold air to the enclosure when the temperature of the enclosure exceeds the pre-selected temperature. The control system accomplishes this task by activating the cooling unit (or stage of a multi-stage cooling unit) and the supply air fan. The supply air fan blows the air through the cooling unit and into the enclosure. As a result of the cold air entering the enclosure, the temperature in the enclosure is lowered. Once the temperature in the enclosure falls below the pre-selected temperature, the thermostat in the enclosure provides a signal to the control system which either turns off the cooling unit, or turns off a stage of cooling (if part of a multi-stage unit).

Similarly, when set in the heating mode, the control system will provide hot air to the enclosure when the temperature of the enclosure falls below the pre-selected temperature. The control system accomplishes this task by activating the heating unit (or stage of a multi-stage heating unit) and the supply air fan. The supply air fan blows the air through the heating unit into the enclosure. As a result of the hot air entering the enclosure, the temperature in the enclosure is raised. Once the temperature in the enclosure rises above the pre-selected temperature, the thermostat in the enclosure provides a signal to the control unit which either turns off the heating unit, or turns off a stage of heating (as part of the multi-stage unit).

As mentioned, the economizer mode may be able to maintain the enclosure at the pre-selected temperature under certain conditions. Particularly, during times when the outside air temperature is low (e.g., 50° F), and the control system needs to provide cold air to the enclosure to cool such enclosure, the system can utilize the economizer mode to provide the desired cold air to the enclosure. In the economizer mode, the control system will selectively modulate open and close both an outside air damper and a return air damper to mix the cool outside air with the warmer return air. In this manner, the air being supplied to the enclosure is cooled to the desired temperature without the need for activating the cooling unit. Of course, if the outside air temperature is too high and/or too humid, the cooling unit will need to be activated.

The above-described temperature control systems are typically designed to allow "time cycling" of the heating/cooling components, which of course limit/preclude these known systems from regulating the BTU output of the HVAC to balance such output with the measured sensible thermal load.

More to the point, those skilled in the art will appreciate that "time cycling" prevents a system from operating in a "real time" mode, and often allows undesirable temperature swings, as well as inefficient operation of the individual components. This inefficient operation can include the operation of excess cooling/heating capacity (resulting in unneeded energy costs) and excess cycling of the systems components (resulting in the shortening of the life of the unit and/or an increase in maintenance of such unit). In fact, the prior art has generally believed that real time temperature control systems which attempt to directly regulate BTU output to balance such output with the system load are inherently unstable, and will produce excessive and potentially damaging "short cycling" of the heating/cooling components.

Moreover, the prior art systems are generally inefficient because the supply air is often colder/hotter than necessary to satisfy the measured sensible thermal load. Finally, such systems are generally incapable of satisfying an unmet cooling/heating load.

There is therefore a need in the art for a dynamic real time temperature control system which directly regulates the BTU output of an HVAC package to balance such output with the sensible thermal load being measured in the temperature-regulated enclosure, thereby eliminating/reducing undesirable temperature swings in the regulated environment, reducing excess cycling of components and eliminating/reducing utilization of unneeded excess capacity.

SUMMARY OF THE INVENTION

The present invention, which addresses the needs of the prior art, relates to a method of controlling room temperature within a zone of a temperature control system. The method generally includes the steps of defining a thermal demand set point temperature curve for the temperature control system, measuring a sensible thermal load within the zone, calculating a thermal demand set point temperature based upon the sensible thermal load, defining at least one load band for the temperature control system corresponding to an equilibrium condition, and operating the temperature control system to maintain individual components of the system in a constant operating condition for as long as the system operates within the load band.

The present invention further relates to a thermal balance temperature control system for controlling room temperature within a predefined zone. The system includes at least one air handling unit for providing supply air at a preselected temperature, the air handling unit includes at least one unit stage. The system further includes a supply duct for transporting supply air from the air handling unit to the predefined zone. Finally, the system includes at least one controller for controlling room temperature within the predefined zone. The controller comprises at least one processor circuit for measuring a sensible thermal load within the zone and for calculating a thermal demand set point temperature based upon the sensible thermal load in accordance with a predefined thermal demand set point temperature curve. The processor circuit operates the temperature control system to maintain the unit stage in an energized condition for as long as the system operates within a predefined load band corresponding to an equilibrium condition.

Finally, the present invention relates to a controller for controlling room temperature within a zone of a temperature control system. The controller includes at least one processor circuit for measuring a sensible thermal load within the zone and for calculating a thermal demand set point temperature based upon the sensible thermal load in accordance with a predefined thermal demand set point temperature curve. The processor circuit operates the temperature control system to maintain individual system components in a constant operating condition for as long as the system operates within a predefined load band corresponding to an equilibrium condition.

As a result, the present invention provides a dynamic real time temperature control system which directly regulates the BTU output of an HVAC package to balance such output with the sensible thermal load being measured in a temperature-regulated enclosure, thereby eliminating/reducing undesirable temperature swings in the regulated environment, reducing excess cycling of components and eliminating/reducing utilization of unneeded excess capacity.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 is a schematical representation of a heating, ventilating and air conditioning system including the thermal balance temperature control system of the present invention;

Figure 2 is a schematical representation of the components of an HVAC package used in accordance with the present invention;

Figure 3 is a graphical representation of the thermal demand set point temperature curve for the thermal balance temperature control system of the present invention;

Figure 4 is a graphical representation of a cooling load band curve for the thermal balance temperature control system of the present invention;

Figure 5 is a graphical representation of an economizer load band curve superimposed on the curve of figure 4; and

Figure 6 is a schematical representation of the controller used in accordance with the present invention.

DETAILED DESCRIPTION OF THE INVENTION

As discussed more fully hereinbelow, the present invention is directed to a method and apparatus for controlling a temperature-regulated zone utilizing a thermal balance temperature control system. The thermal balance control system is a dynamic real time control system that constantly measures the sensible thermal load in the mentioned zone, and directly regulates the BTU output of the HVAC package to balance such output with the measured sensible thermal load, thus providing a state of system equilibrium. The system will continue to operate in this equilibrium state (without time cycling of any heating/cooling components) until the system measures a change in the sensible thermal load within the mentioned zone.

The sensible thermal load is the amount of deviation (measured in degrees) between the set point temperature for the zone and the actual zone temperature. When the actual room temperature is above the set point temperature, the sensible thermal load is a cooling load, and the system must therefore reduce the supply air temperature to balance the BTU output of the HVAC package with such load. If the actual room temperature is below the set point temperature, then the sensible thermal load is a heating load, and it is necessary for the system to increase the supply air temperature to balance the BTU output with such load.

The thermal balance control system of the present invention utilizes the formula: Thermal Transfer Rate (BTU/HR) = Supply Air Volume (cubic feet per minute) x 1.08 x (Room Temperature-Supply Air Temperature). As will be appreciated from the foregoing formula, the thermal transfer rate is equal to 0 when the room temperature is equal to the supply air temperature.

As discussed herein, the thermal balance control system of the present invention operates in a "load cycling" manner, in contrast to the "time cycling" manner of conventional units. It will be appreciated that available HVAC units which operate in an on/off function (e.g., direct expansion (DX) cooling, electric heat, etc.) are typically utilized in a time-cycled manner. Particularly, if the prior art system requires supply air at 55° and stage 1 of a DX cooling system only reduces the temperature to 60°, the second stage of such system will be cycled on and off to reduce the temperature of the supply air to below 55°. Every time a unit cycles on and off the system can experience wide and comfortable temperature swings. With respect to the cycling on and off of a DX cooling unit, condensation caught in a coil will evaporate back into the supply air when such unit is cycled off. This increase in humidity of the supply air can cause discomfort to the occupants in the building, and also decreases the overall efficiency of the unit (in that the unit must again remove the vapor from the air when cycled back on). For example, the cycling of a stage of DX cooling on a rainy summer day may cause such an undesirable condition.

Referring to Figure 1, a thermal balance temperature control system 10 in accordance with the present invention includes a heating, ventilating and air conditioning (HVAC) package 12 for supplying cold or heated supply air 14 (as well as fresh outside air) into a supply air duct 16, which communicates with an interior enclosure, i.e., zone 18. Return air 20 is thereafter removed from zone 18 via return air duct 22. Temperature control system 10 also includes a thermal balance controller 24, which is a dynamic real time controller that measures the sensible thermal load in zone 18, and regulates the output capacity of HVAC package 12 to balance such output with this measured load.

As shown in Figure 2, HVAC package 12 includes a supply air fan 26 for moving supply air into zone 18 and a return air fan 28 for removing return air from zone 18. HVAC package 12 further includes an economizer section 30, a heating unit 32, a cooling unit 34 and a supply air temperature sensor 36. Package 12 may also include a filter 38, a low temperature alarm 40 and low limit temperature sensor 42.

Economizer section 30 preferably includes an exhaust damper 44, an outside air damper 46 and a return air damper 48. Return air damper 48, together with outside air damper 46, control the percent mixture of return air/fresh air being fed into supply air duct 16. Those skilled in the art will understand that exhaust damper 44, outside air damper 46 and return air damper 48 are preferably operated to meet at least some of the following goals: 1) to operate in economizer mode when conditions permit; 2) to take maximum advantage of the temperature of the return air; and 3) to mix sufficient fresh air into the supply air.

In one preferred embodiment, HVAC package 12 includes an economizer section, a two-stage gas heating section, a three-stage direct expansion (DX) cooling unit, a constant volume supply fan and a constant volume return air fan. One preferred package is rated at 25 tons at 10,000 cubic feet per minute. This design capacity is based on approximately 400 cubic feet per minute per ton, and 5-6 air changes per hour. The operation sequence of HVAC package 12 preferably follows an ASHRAE Cycle II.

Thermal balance temperature control system 10 can be used in a constant volume system or in a variable air volume (VAV) system. It will be recognized by those skilled in the art that a VAV system would utilize variable speed supply and return fans (in contrast to the constant speed fans used in a constant volume system). Unlike the constant volume system, the VAV system will typically include a differential pressure gauge located in the supply air duct downstream from the supply air fan.

Thermal balance temperature control system 10 may operate in either the heating, economizer or cooling mode, depending on the sensible thermal load measured within zone 18. More particularly, the heating mode is preferably controlled by cycling (in sequence) the two gas valves to maintain a desired supply air temperature. The heating mode is generally not initiated until outside air damper 46 is at its minimum open setting. Preferably, morning warm-up will be accomplished with both outside air damper 46 and exhaust damper 44 fully closed, and return damper 48 fully opened. The economizer cooling mode is preferably controlled by modulating exhaust damper 44, outside air damper 46 and return air damper 48 to maintain the desired supply air temperature. The economizer cooling mode is preferably limited by an outside air temperature sensor set at 60° that reduces the intake of fresh outside air (for ventilation) to a minimum value at temperatures exceeding 60°. Of course, this 60° setting is adjustable, depending on system criteria. Finally, the cooling mode is preferably controlled by cycling the stages of cooling in direct relation to the sensible thermal load measured within zone 18. Because temperature control system 10 seeks to balance the BTU output of HVAC package 12 with the sensible thermal load measured within zone 18, the stages of heating and cooling do not experience short cycling (i.e., excessive cycling on and cycling off of the individual stages). Rather, such stages remain activated until such time as the system measures a change in the sensible thermal load.

It will be appreciated by those skilled in the art that a multi-stage heating/cooling unit generally provides better overall efficiency. For example, in a multi-stage cooling unit having three stages, each stage providing approximately 33% of the total cooling capacity of the unit. When maximum cooling is required, all three stages can be activated. However, when maximum system output is not needed, one or more stages can be deactivated, thus allowing the system to operate in a more energy-efficient mode. Similarly, each stage in a two-stage unit provides 50% of the total capacity of the unit, while each stage of a four-stage unit provides 25% of the total capacity. In one embodiment, the relay differential of a stage of cooling is made greater then the temperature change which results from that stage being energized or deenergized. This prevents the cooling stage from short cycling due to the action of the discharge sensor. Preferably, the relays should be set up to provide Vernier controls.

It will be understood by those skilled in the art that resetting the temperature of the supply air in response to certain system measurements can improve the performance and operation of the overall system. Although prior art systems utilize reset schedules, such schedules generally consist of a standard fixed ratio which does not directly correlate to the operating characteristics of the system and does not allow the system to reach a state of equilibrium. In contrast, the thermal demand set point temperature curve for the system of the present invention (as shown in Figure 3) is established to directly correlate with the operating characteristics of HVAC package 12 and to allow the system to reach a state of equilibrium (i.e., the BTU output is balanced with the measured sensible thermal load).

Referring now to Figure 3, the illustrated thermal demand set point temperature curve for HVAC package 12 includes a heating portion and a cooling portion. For example, if the particular heating unit is capable of providing a maximum temperature rise of 50°, then the heating portion of the curve is drawn to extend between a minimum thermal demand set point P_0 (wherein 0 heat is required) and a maximum thermal demand set point P_1 (wherein maximum heat, i.e., plus 50° F) is required. This maximum heat condition corresponds to a measured sensible thermal load of -2° F. The cooling portion of the curve is drawn in accordance with the particular cooling unit installed in the system. For example, if the system is capable of reducing the supply air temperature by a maximum of 25°, then the curve is drawn between a minimum thermal demand set point P_0 (wherein 0 cooling is required) and a maximum thermal demand set point P_2 (wherein maximum cooling, i.e., minus

25°F,) is required. This maximum cooling condition corresponds to a measured sensible thermal load of +2°F.

The thermal demand set point temperature curve of Figure 3 is based upon a temperature band of plus and minus 2°F. On a drop in space temperature of 2°F, the supply air temperature will be reset from set point temperature P_0 to P_0 plus 50°F. On a rise in space temperature of 2°F, the supply air temperature will be reset from set point temperature P_0 to P_0 minus 25°F. This band can, of course, be widened (although widening the band may cause the temperature in zone 18 to move into an uncomfortable region), may be narrowed (which may increase the cost of operating such system) or may include integral control action for improved responsiveness.

The method of the current system will now be described with respect to Figures 3 and 4. As described, Figure 3 is used to calculate the thermal demand set point temperature of the supply air during operation of the system. To begin, the sensible thermal load in zone 18 is measured. If, for example, the room set point is 73°F and the actual measured room temperature is 74°F, the deviation from set point (i.e., the sensible thermal load) is +1°. Referring to the thermal demand set point temperature curve of Figure 3, a +1 temperature deviation is within the cooling portion of the curve and corresponds to approximately -12.5° on the Y axis. The set point P_0 of Figure 3 corresponds to the set point temperature of zone 18. Thus, the thermal demand set point temperature for the supply air would be calculated to be 73° - 12.5°= 60.5°. This is the temperature at which the system is balanced, i.e., providing supply air at 60.5°F to zone 18 will maintain zone 18 in a state of equilibrium at 74°F.

In certain applications, as described in commonly-owned co-pending U.S. application Serial No. 10/704,251 filed November 7, 2003, the disclosure of which is incorporated herein by reference, the system can be designed to recognize this unmet cooling load (i.e., the +1°F in zone 18). Thereafter, the system would calculate and supply the additional cooling necessary to move the actual room temperature towards the room set point.

Figure 4 illustrates the novel load band curve of the present invention, which is preferably a proportional curve having preselected parameters which correspond to the components of the system. The particular graph shown in Figure 4 represents a plot for a multi-stage DX cooling system having three stages wherein the maximum cooling is approximately 20°. A 40% allowance (i.e., 8°) may be designed into the system such that the X axis extends from 0° to 28° (20° + (40% of 20°)). The X axis of the load band is 10° wide (i.e., it extends from 9° to 19°). It will be appreciated that each stage of the three stage DX cooling system is capable of approximately a 7° temperature drop. Again, a 40% allowance may be designed into the system to provide a total of approximately 10° (7° + (40% of 7) = 9.8, which is approximately 10°).

If the desired supply air temperature is calculated to be 60.5° (as discussed hereinabove), the set point S of the graph of Figure 4 will be set to 60.5°. The value of this point will remain fixed until the system measures a change in the sensible thermal load in zone 18 and recalculates the thermal demand set point temperature from Figure 3. The actual supply air temperature (as measured by sensor 36) is then plotted along the curve. With set point S set at 60.5° F, point S_1 will correspond to 55.5° F and point S_2 will correspond to 65.5° F.

The first stage of cooling will be turned on, resulting in a 7° drop of temperature. If this is sufficient to bring the supply air temperature within the load band which, in this example, will extend from 55.5° to 65.5° (5° on either side of the set point), then no additional stages will be turned on. As long as the supply air temperature remains within this load band, the first stage of the compressor will remain on. Unlike conventional systems which would automatically begin time cycling this stage of the compressor, the system of the present invention will allow this stage of the compressor to stay on as long as the supply air temperature remains within in such load band. In other words, the thermal balance control of the present invention has reached a state of system equilibrium, and may remain in this state until a change in the sensible thermal load is measured.

The portion of the curve of Figure 4 extending from point S_1 to S_2 is referred to herein as the load band. Once the supply air temperature moves outside of the load band, it moves into one of two integrating regions. For example, if two stages of the three stage compressor are on and the supply air temperature continues to decrease such that it moves down the curve into the lower integral region, an integral factor will increase the speed at which the supply air temperature moves towards the stage-off point. Once, the supply air temperature hits this point, the particular stage is turned off, thereby raising the supply air temperature and pushing such supply air temperature back towards the load band. Likewise, if the supply air temperature increases such that it moves up the curve into the upper integral region, eventually additional stages of cooling will be turned on. Again, integral action decreases the time necessary to reach the point where an additional stage of cooling is turned on. Thus, the system anticipates overcooling and undercooling through the integral action portions of the control system.

More particularly, the system anticipates a change in the sensible thermal load. If the load is increasing (thus indicating the need for an extra stage of cooling), the thermal demand set point temperature will decrease (thus providing a lower set point to the cooling control module). The supply air temperature will now be higher than the thermal demand set point temperature, and will begin to move up the curve into the upper integral region. An integral factor will increase the speed at which the supply air temperature moves towards the stage-on point. If the sensible thermal load is decreasing, the reverse action will occur. As a result, the system provides load change anticipation.

Stated differently, the present invention anticipates gain in the wrong direction, and corrects this unwanted gain prior to the regulated enclosure experiencing an uncomfortable temperature swing. It will be appreciated by those skilled in the art that although a conventional system would eventually compensate for the change in the temperature of the supply air, because of the inherent time delays and time constants associated with HVAC systems, the conventional system cannot respond until “after the fact”. In other words, the regulated enclosure has already undergone the unwanted temperature swing before it begins to react to the temperature swing due the change in the temperature of the supply air.

Figure 5 illustrates an economizer load band curve superimposed on the cooling load band curve of Figure 4. In this particular example the economizer load band will extend plus and minus 1.5° from set point S. Once the supply air temperature has increased 1.5° above set point S, the system will begin to modulate open the outside air damper. Similarly, once the supply air temperature decreases 1.5° below set point S, the system will begin to modulate closed the outside air damper. While the supply air temperature is within the economizer load band, the outside air damper will be maintained in a constant position.

Referring to Figure 6, the control system of the present invention, i.e., controller 24, uses three individual control modules, namely a first control module 50 for the heating unit, a second control module 52 for the economizer unit and a third control module 54 for the cooling unit. The control system is designed so that each one of the individual control modules operates its respective unit depending on whether the supply air temperature is above or below the thermal demand set point temperature calculated from Figure 3.

The system calculations and operations described hereinabove are preferably performed by controller 24, and particularly by the individual control modules. More particularly, the controller and/or control modules preferably include hardware/software which is capable of performing the mentioned calculations, and of utilizing predefined thermal demand set point temperature and load band curves to control the operations of system 10 in accordance with the parameters described herein.

It should be noted that each control module receives two sets of numbers. Specifically, each module receives the thermal demand set point temperature T_P for the supply air (from Figure 3), and the actual temperature of the supply air T_{SA} (as measured by sensor 36). Moreover, each control module has a specific temperature set point that is used to determine which of three individual modules is activated. The specific temperature set point for each module is based on the thermal demand set point temperature, as well as a predefined bias setting.

In a preferred embodiment, the modules are all biased to control at a different temperature based on the thermal demand set point temperature for the supply air so that only a single module will activate at any one time. Depending on whether the supply air is above or below each one of the module's specific temperature set points determines which unit will be activated, and thus controlling the system. For example, should the actual supply air temperature (as measured by sensor 36) be below the thermal demand set point temperature, the heating control module would be activated to raise the temperature of the supply air. During this time, the cooling control module and economizer control module are not activated since the supply air temperature is below their specific temperature set points. As mentioned, the heating, economizer and cooling control modules are set up with a predefined bias setting. The heating control module has a bias setting of -3°F, the economizer control module has a bias setting of 0°F, and the cooling control module has a bias setting of +2° F. These bias set point are of course adjustable.

Referring back to the example set forth above wherein the thermal demand set point temperature for the supply air was calculated to be 60.5°F, the local set point of the heating control module would be calculated to be 60.5° - 3° = 57.5°F. The local set point for the economizer control module would be calculated to be 60.5°F + 0° = 60.5°F, while the local set point for the cooling control module would be calculated to be 60.5°F + 2.5°F = 63°F.

The local set point separates the control action of the individual control modules. If the supply air temperature (as measured by sensor 36) is below 57.5°F (the local set point of the heating control module) the system will add heat to satisfy the demand. If the supply air temperature (as measured by sensor 36) is above 60.5°F (the local set point of the economizer control module) and cool outside air is available the economizer control module will modulate damper 46 satisfy the demand. If the outside air temperature is above a predefined temperature limit, the cooling control module will cycle the cooling to satisfy the demand. Finally, if the supply air temperature (as measured by sensor 36) is above 63°F (the local set point of the cooling control module), the system will cool the supply air to satisfy the demand.

The set point of each control module is 50. Each control module defines a load band and upper and lower integrating regions (for load anticipation). The heating control module is reverse acting, and the economizer and cooling control modules are direct acting.

The control modules are set up to stabilize whenever the supply air temperature is within the load band. The system then stabilizes at that level of BTU output, i.e., it will stay there until there is a change in the sensible thermal load in the zone. The load band is set up to match the BTU output to the measured sensible thermal load. The load anticipation feature operates when the sensible thermal load changes, indicating a required increase or decrease in the BTU output of the HVAC package.

For heating control applications, the heating control module can be set up for single control, multiple-stage control, or modulating control. For economizer control applications, the economizer control module can be set up for mixing damper control with minimum damper position or modulating a free cooling valve with a high temperature limit. For DX cooling control applications, the cooling control module can be set up to utilize the load band and load anticipation adjustments to provide load cycling. When a stage of DX cooling is energized the stage will stay ON until there is a decrease in the measure sensible thermal load. The system provides load cycling of the DX stages, not time cycling. The control module will lengthen the ON time of a stage of cooling if there is an increase in the latent load on the unit, internal or external.

In accordance with the present invention, control system 10 can eliminate droop, overshoot and mechanical lag by providing the optimum cycle rate of any stage for efficient operation under all load conditions. Control system 10 can respond immediately to a change in the measured sensible thermal load by optimizing the cycle rate of the heating or DX cooling stages or repositioning the mixed air dampers. Control system 10 can also respond immediately to the measured change in the BTU output of the HVAC package (due to changes in the outdoor air temperatures) by optimizing the cycle rate of the heating or DX cooling stages or repositioning the mixed air dampers.

Control system 10 can dynamically optimizes the cycling rate of the heating or cooling stages based on the BTU output of the HVAC package by measuring the supply air temperature and adjusting the cycle rate to match the BTU output to the measured sensible thermal load. The cycle rate can be adjusted real time to match the BTU output to the load; the system does not compute the cycle rate based on a developed software program. The load response of control system 10 can be characterized by automatic initialization of the stages for an optimum cycle rate.

Control system 10 can adapt to the operating characteristics of the various modes, heating, economizer and cooling, whether staging or proportional. The control system can match the BTU output of the unit to the load in the space without cycling from one mode to the other or short cycle between stages. The control system does not require time delays between stages. Control system 10 can adapt automatically to a change in the latent load in the space of a change in the temperature of the outside ventilation air, and vary the cycle rate of DX cooling for optimum latent heat removal and improved IAQ.

Control system 10 will not heat and cool simultaneously, nor will it cycle between heating and cooling. Control system 10 does not require a heating or cooling mode switch. Rather, the system can measure the load and responds accordingly.

Control system 10 can recognize changes in the load, either internal (space) or external (entering the unit) that will affect the relationship of matching the BTU output to the measured sensible thermal load, and can respond immediately.

Control system 10 can identify a stage failure, heating or cooling, and can activate the next stage if available and activate an alarm. Control system 10 can monitor the HVAC package performance continuously. Any malfunction can be alarmed, if desired.

It will be appreciated that the present invention has been described herein with reference to certain preferred or exemplary embodiments. The preferred or exemplary embodiments described herein may be modified, changed, added to or deviated from without departing from the intent, spirit and scope of the present invention, and it is intended that all such additions, modifications, amendment and/or deviations be included within the scope of the following claims.

What is claimed is:

1. A method of controlling room temperature within a zone of a temperature control system utilizing supply air having a temperature T, comprising the steps of:

defining a thermal demand set point temperature curve for said temperature control system;

measuring a sensible thermal load within said zone;

calculating a thermal demand set point temperature based upon said sensible thermal load;

defining at least one load band for said temperature control system corresponding to an equilibrium condition; and

operating said temperature control system to maintain individual components of said system in a constant operating condition for as long as said system operates within said load band.

2. The method according to claim 1, wherein said first defining step includes the steps of establishing a heating curve extending between a minimum heating thermal demand set point corresponding to a condition of minimum heating output and a maximum heating thermal demand set point corresponding to a condition of maximum heating output and establishing a cooling curve extending between a minimum cooling thermal demand set point corresponding to a condition of minimum cooling output and a maximum cooling thermal demand set point corresponding to a condition of maximum cooling output.

3. The method according to claim 2, wherein said sensible thermal load is equal to the amount of deviation between a set point temperature for said zone and an actual room temperature for said zone, and wherein said measuring step includes the step of calculating the difference between said set point temperature and said actual room temperature.

4. The method according to claim 3, wherein said calculating step includes the further steps of:

establishing a point on said thermal demand set point temperature curve corresponding to said sensible thermal load;

determining a delta temperature T from said set point temperature; and

calculating said thermal demand set point temperature based upon said room set point temperature and said delta temperature T.

5. The method according to claim 4, wherein said second defining step includes the steps of establishing an operating load band having a preselected width corresponding generally to the operating characteristics of a unit temperature control stage.

6. The method according to claim 5, further comprising the step of defining an upper integrating region located above said operating load band and a lower integrating region located below said operating load band; and

providing integrating action for increasing the responsiveness of said system when a signal enters one of said upper and lower integrating regions.

7. The method according to claim 6, wherein said operating step includes the step of energizing a temperature control unit stage to move said temperature T of said supply air into said load band, and maintaining said unit in an energized state as long as said temperature T remains within said load band.

8. A thermal balance temperature control system for controlling room temperature within a predefined zone, comprising:

at least one air handling unit for providing supply air at a preselected temperature, said air handling unit including at least one unit stage;

a supply duct for transporting said supply air from said air handling unit to said predefined zone;

at least one controller for controlling room temperature within said predefined zone, said controller comprising at least one processor circuit for measuring a sensible thermal load within said zone and for calculating a thermal demand set point temperature based upon said sensible thermal load in accordance with a predefined thermal demand set point temperature curve, and wherein said processor circuit operates said temperature control system to maintain said unit stage in an energized condition for as long as said system operates within a predefined load band corresponding to an equilibrium condition.

9. The system according to claim 8, wherein said sensible thermal load is equal to the deviation between a set point temperature for said zone and an actual room temperature for said zone.

10. The system according to claim 9, wherein said predefined thermal demand set point temperature curve includes a heating curve extending between a minimum thermal demand set point corresponding to minimum heating output and a maximum thermal demand set point corresponding to maximum heating output and also includes a cooling curve extending between a minimum thermal demand set point corresponding to minimum cooling output and a maximum thermal demand set point corresponding to maximum cooling output.

11. The system according to claim 11, wherein said predefined load band includes an upper integrating region located above said operating load band and a lower integrating region located below said operating load band, and wherein said integrating regions provide integrating action for increased responsiveness to signals entering one of said upper and lower integrating regions.

12. A controller for controlling room temperature within a zone of a temperature control system utilizing supply air A, said supply air having a temperature T, comprising:

at least one processor circuit for measuring a sensible thermal load within said zone and for calculating a thermal demand set point temperature based upon said sensible thermal load in accordance with a predefined thermal demand set point temperature curve, and wherein said processor circuit operates said temperature control system to maintain individual components in a constant operating condition for as long as said system operates within a predefined load band corresponding to an equilibrium condition.

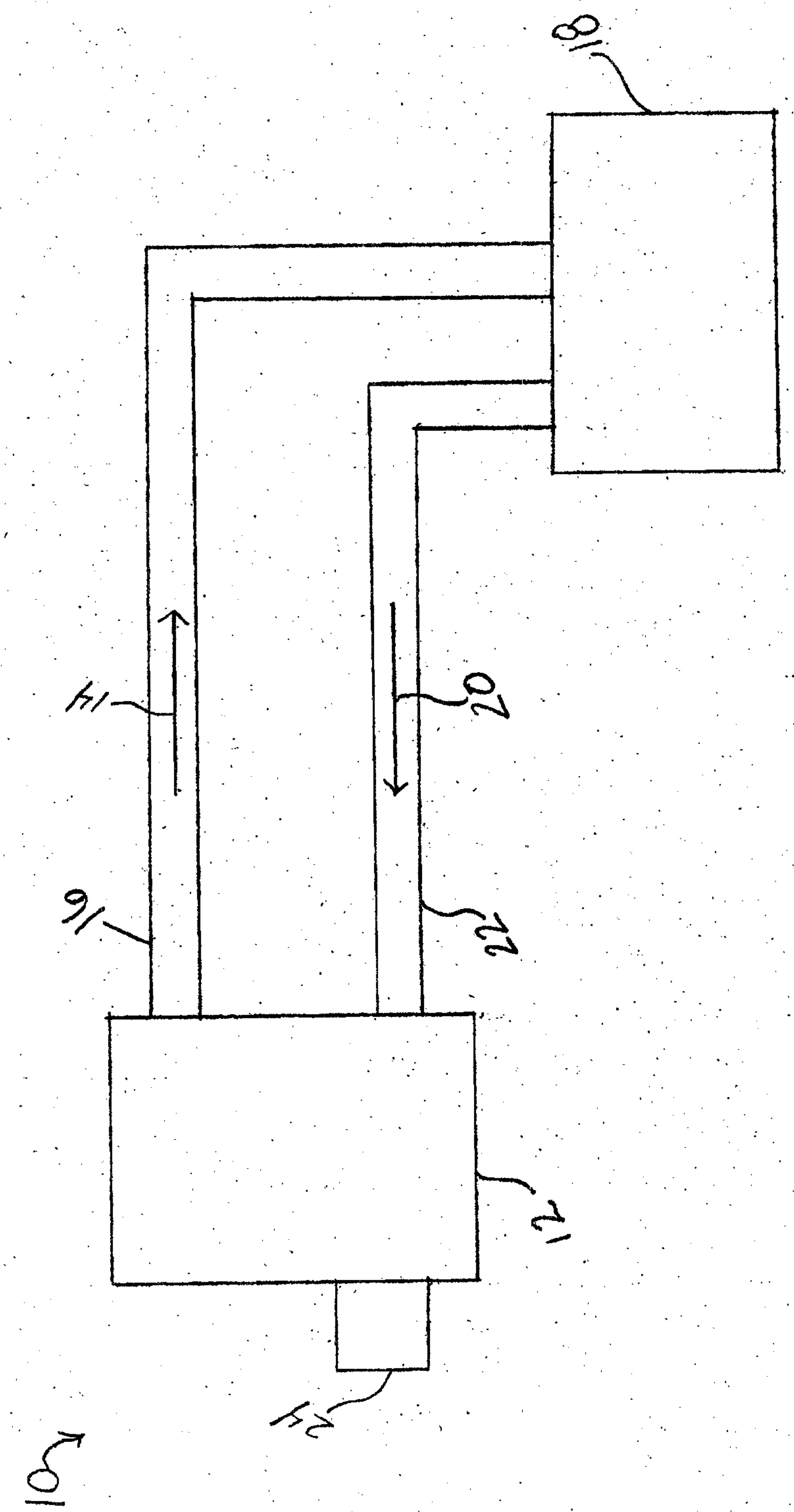


Fig. 1

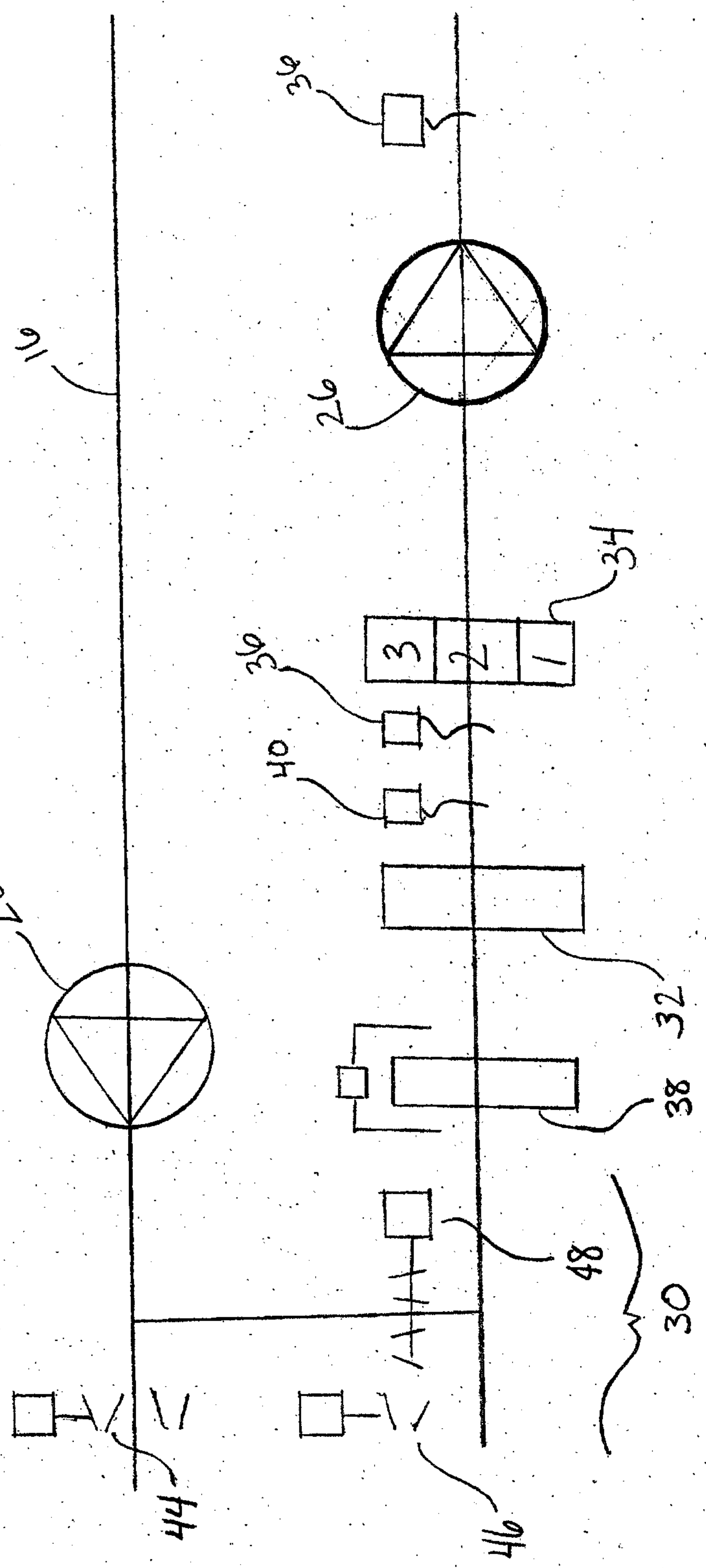
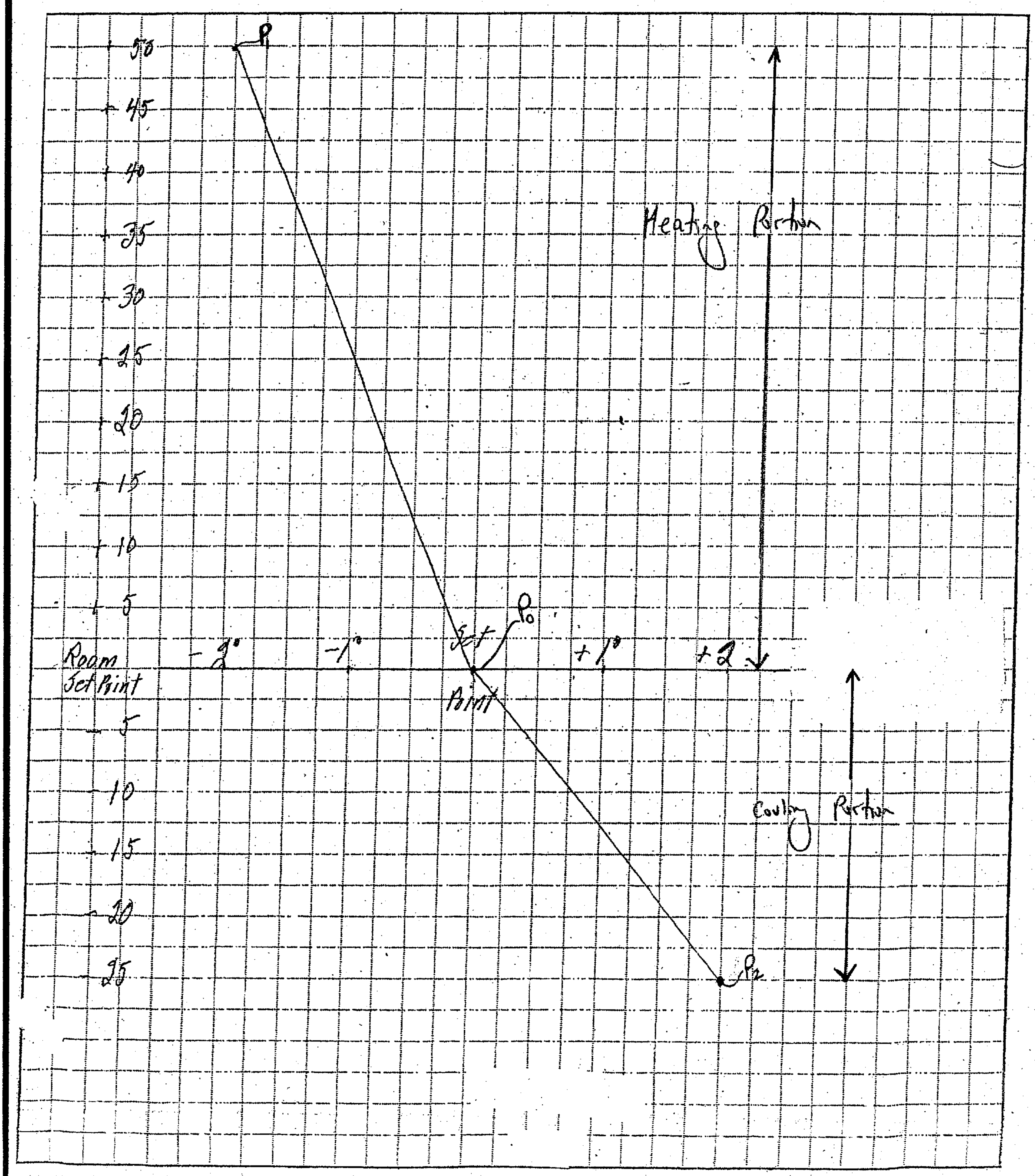


Fig. 2

Fig. 3



P1
+50
+45
+40
+35
+30
+25
+20
+15
+10
+5
Room Set Point
-2°
-1°
Set Point
P0
+1°
+2
-5
-10
-15
-20
-25
P2
Heating Portion
Cooling Portion

Fig. 4

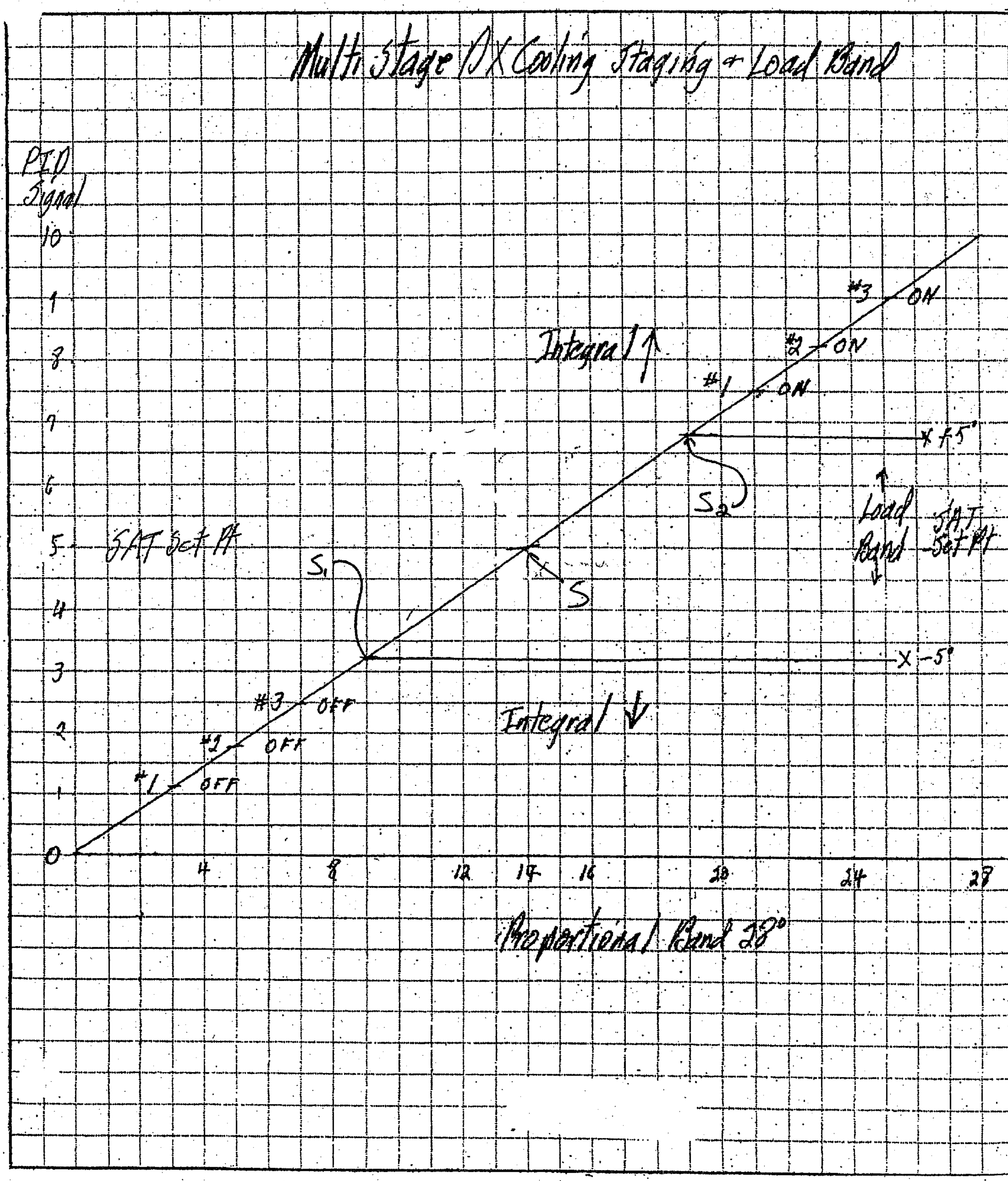

Multi Stage DX Cooling Staging + Load Band
PID Signal
10
9
8
7
6
5
4
3
2
1
0
4
8
12
14
16
20
24
28
Proportional Band 28°
#3 - ON
#2 - ON
#1 - ON
Integral ↑
X +5°
S2
Load Band
SAT Set Pt
SAT Set Pt
S1
S
X -5°
#3 - OFF
#2 - OFF
#1 - OFF
Integral ↓

Fig. 5

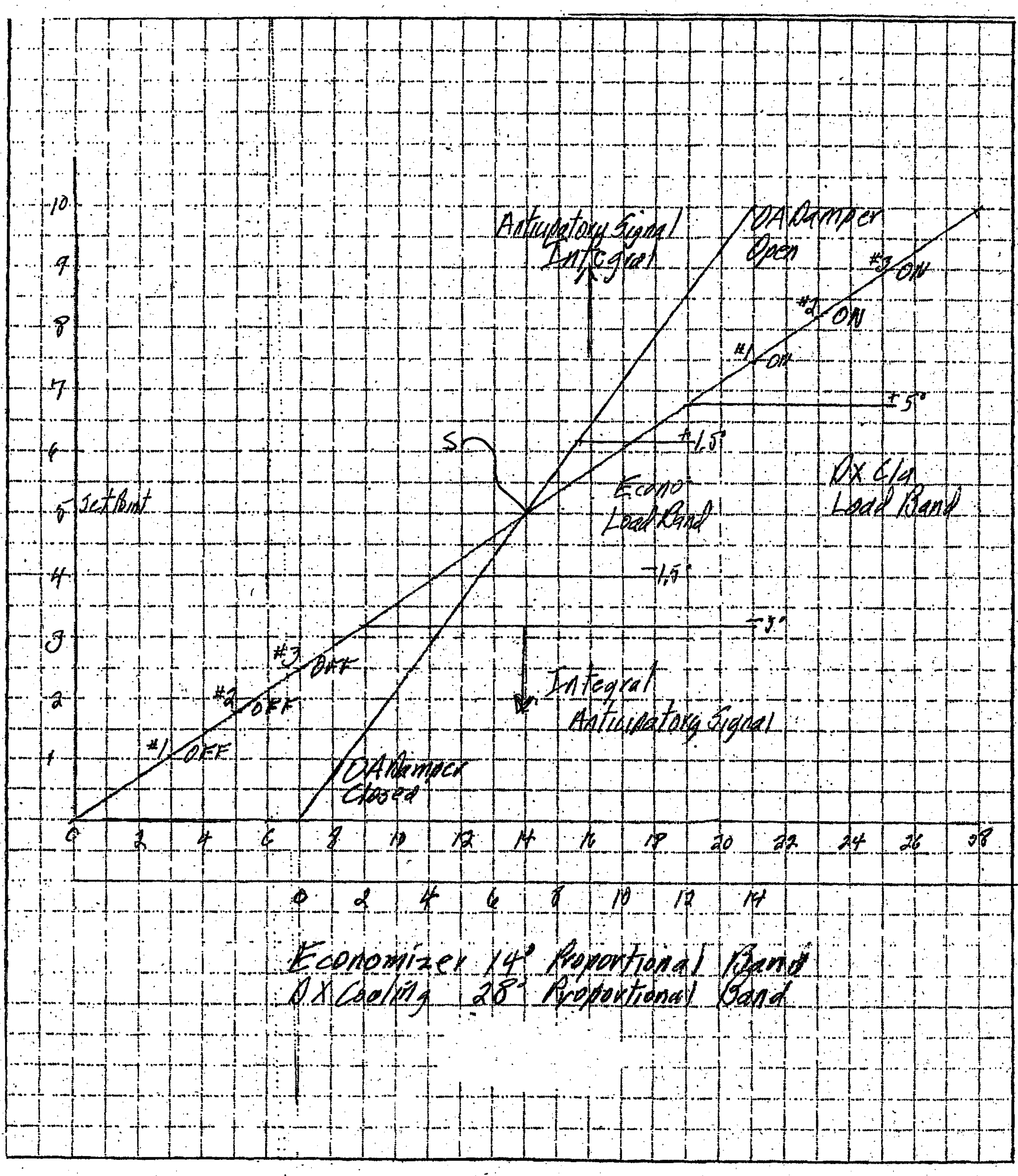
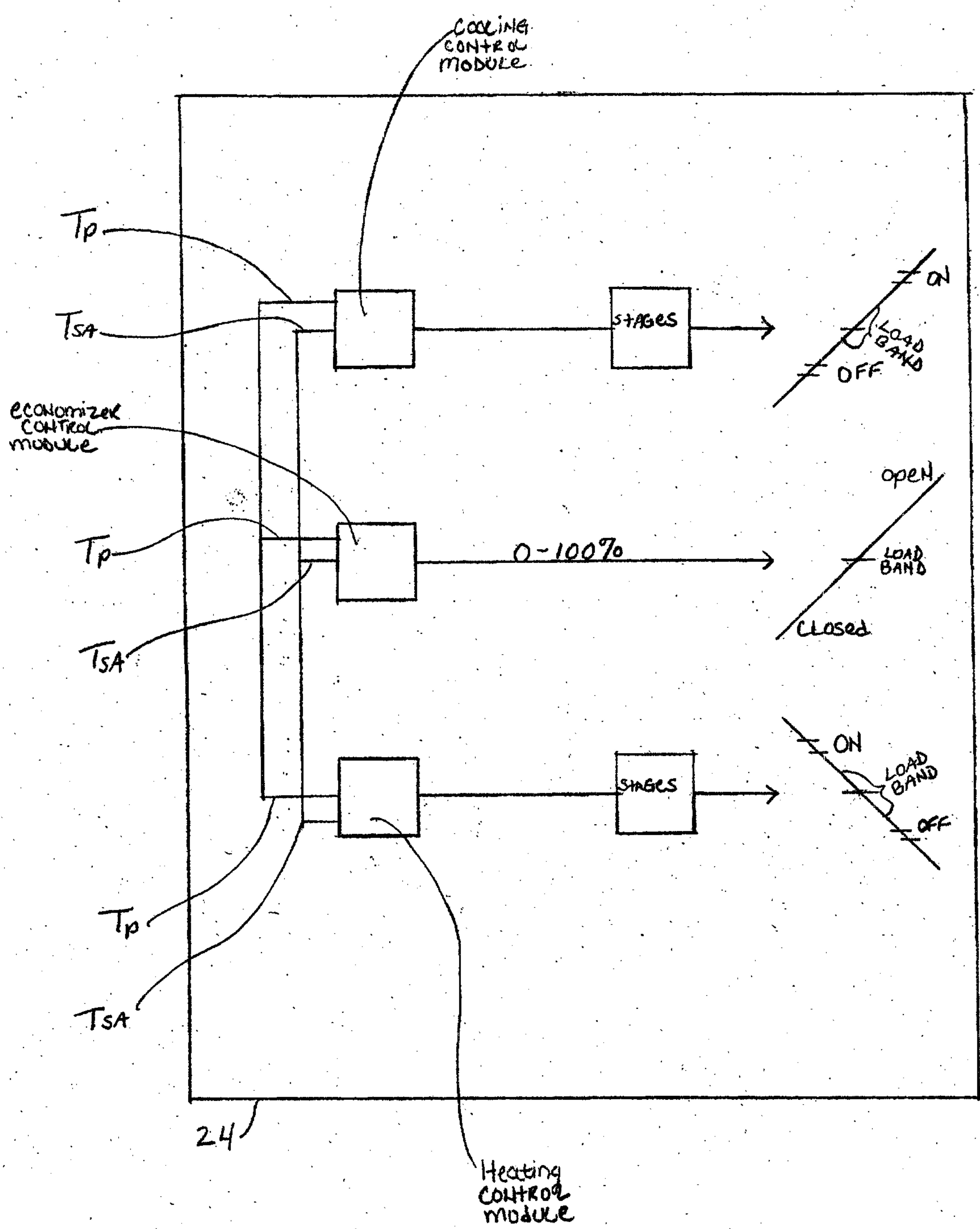

Anticipatory Signal
Integral
OA Damper
Open
#3 ON
#2 ON
#1 ON
±5°
+1.5°
S
Econo-
Load Band
DX Clg
Load Band
Set Point
-1.5°
-3°
#3 OFF
#2 OFF
#1 OFF
Integral
Anticipatory Signal
OA Damper
Closed
Economizer 14° Proportional Band
DX Cooling 28° Proportional Band

Fig. 6


Cooling control module
Tp
TSA
stages
ON
LOAD BAND
OFF
economizer control module
open
Tp
0-100%
LOAD BAND
TSA
Closed
ON
LOAD BAND
stages
OFF
Tp
TSA
24
Heating control module


10
16
14
24
12
22
20
18