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(54) HIGH-EFFICIENCY AIR HANDLER

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(57) ABSTRACT

A high-efficiency air handler that is suitable for moving large volumes of air. The air handler comprises a coil and a fan. The total air volume moving through the unit is preferably over about 2000 CFM, with a total fan static pressure rise of less than about 0.5 inches and preferably less than 0.2 inches of water. The fan is preferably a modified ceiling fan with a rotational speed of less than about 450 rpm. The air handler is preferably part of a system where dehumidification and sensible cooling functions are separate. The separate dehumidification allows for much higher supply air temperatures, preferably within about 10° F. of the space temperature. Low-velocity air distribution through a ceiling plenum or a vent into the space allows for very low fan static pressures, which greatly reduces fan energy use compared to conventional ducted systems. The low static pressures and high supply-air temperatures allow the use of existing drop ceiling construction with little modification. The system can also include low-cost thermal storage. Latent thermal storage is in the form of a concentrated liquid desiccant solution. Chilled water storage is another option. The result is a major improvement in energy efficiency and comfort while reducing installed cost and peak electrical demand of the system.

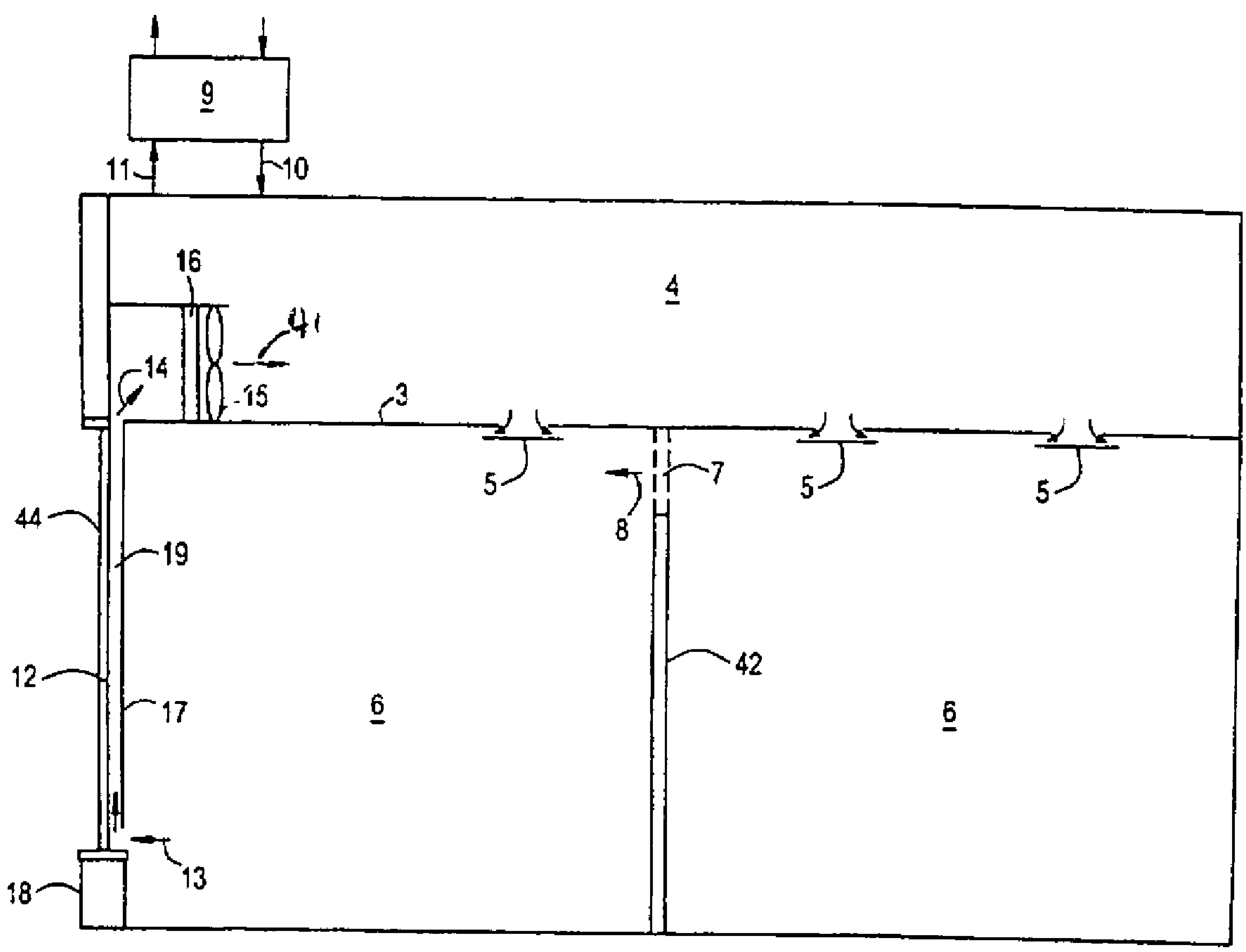


FIG. 1

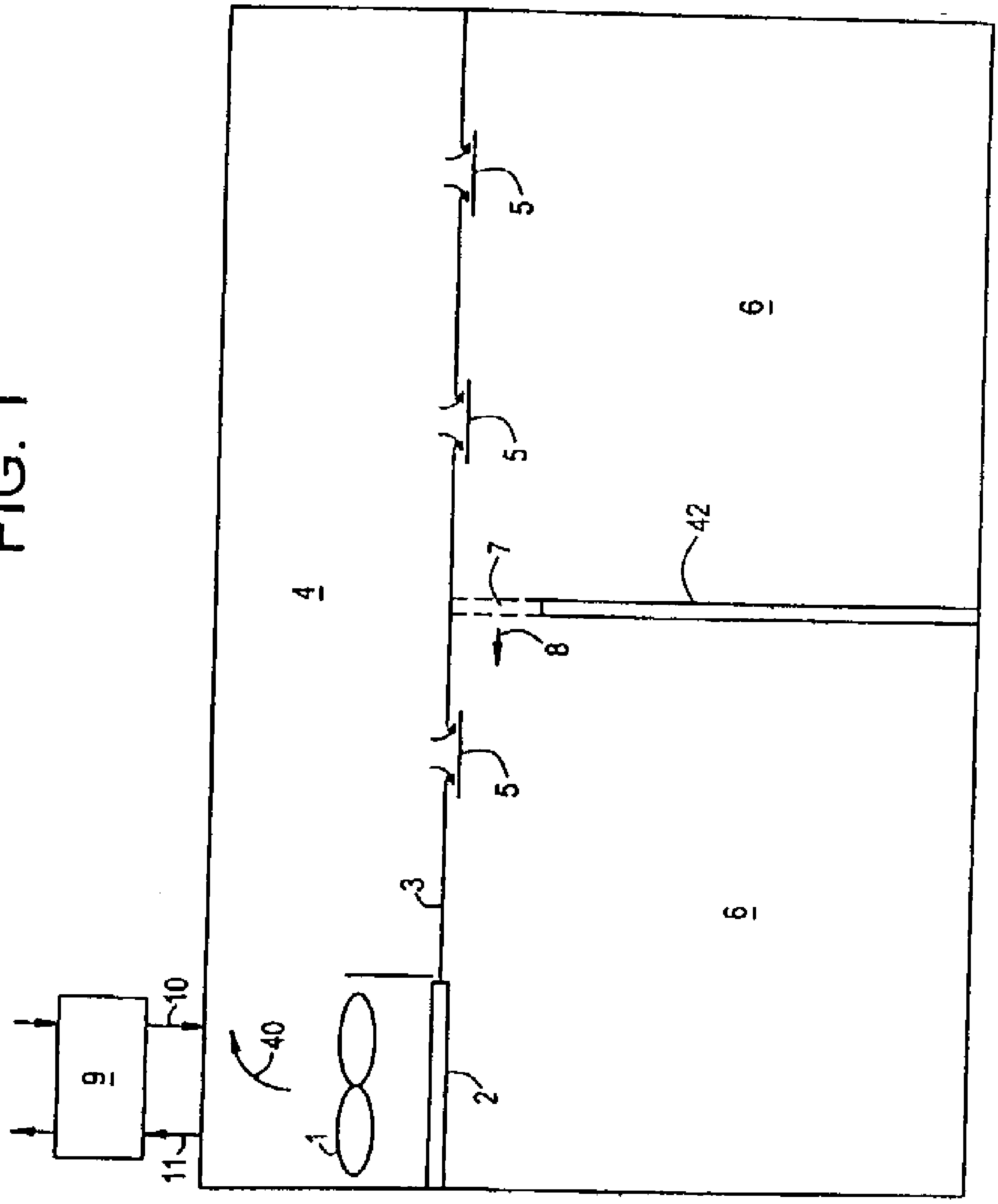
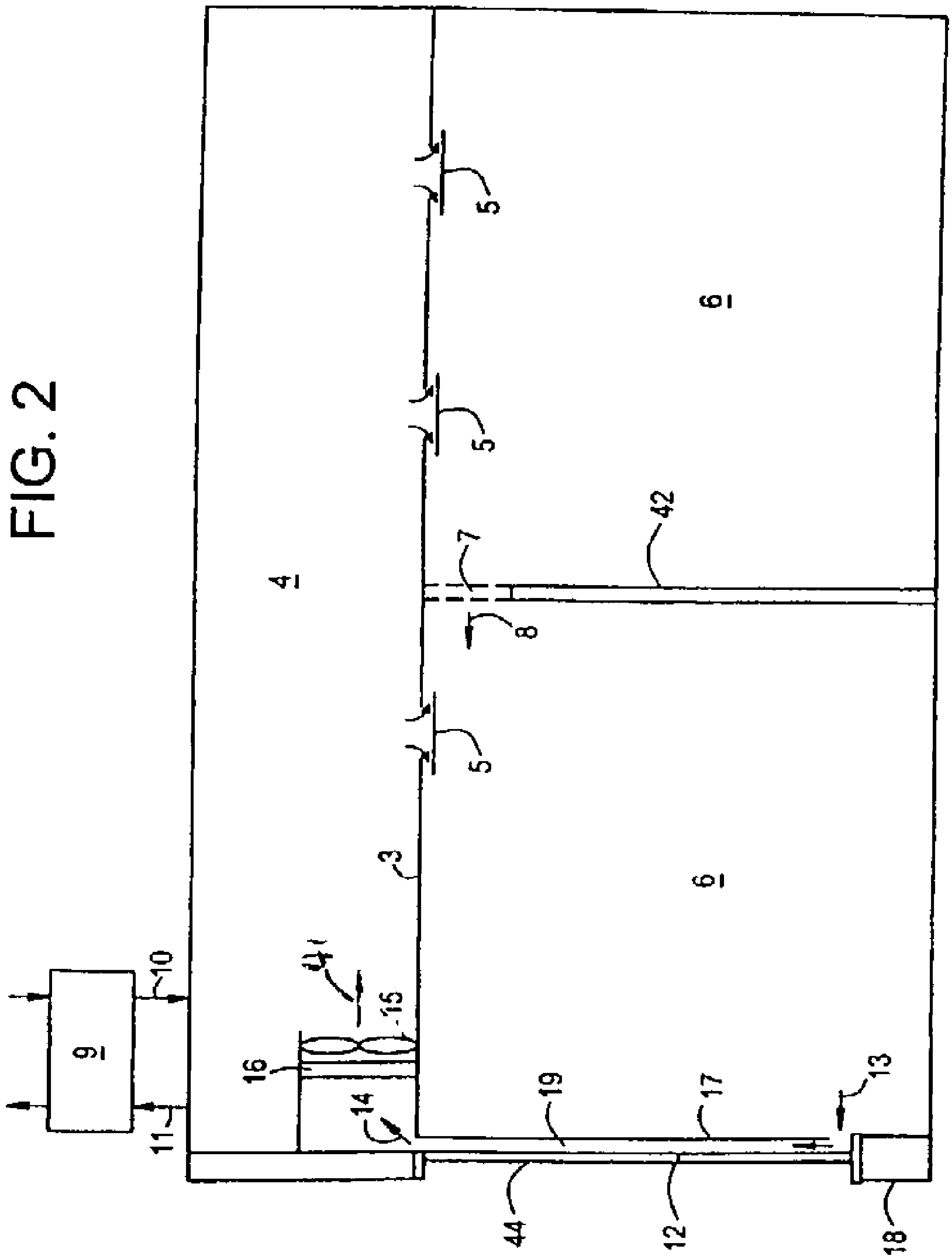


FIG. 2



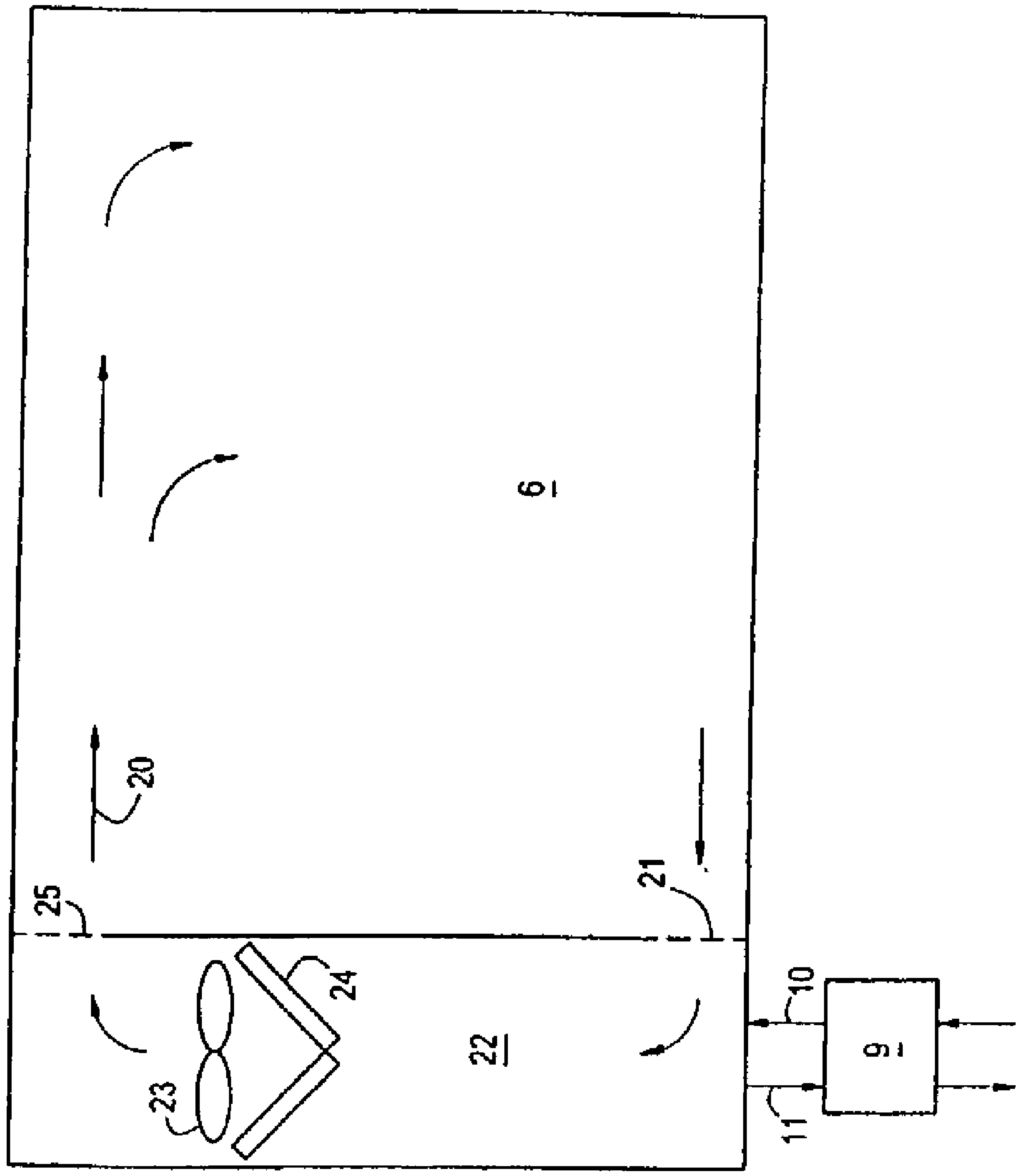



FIG. 3

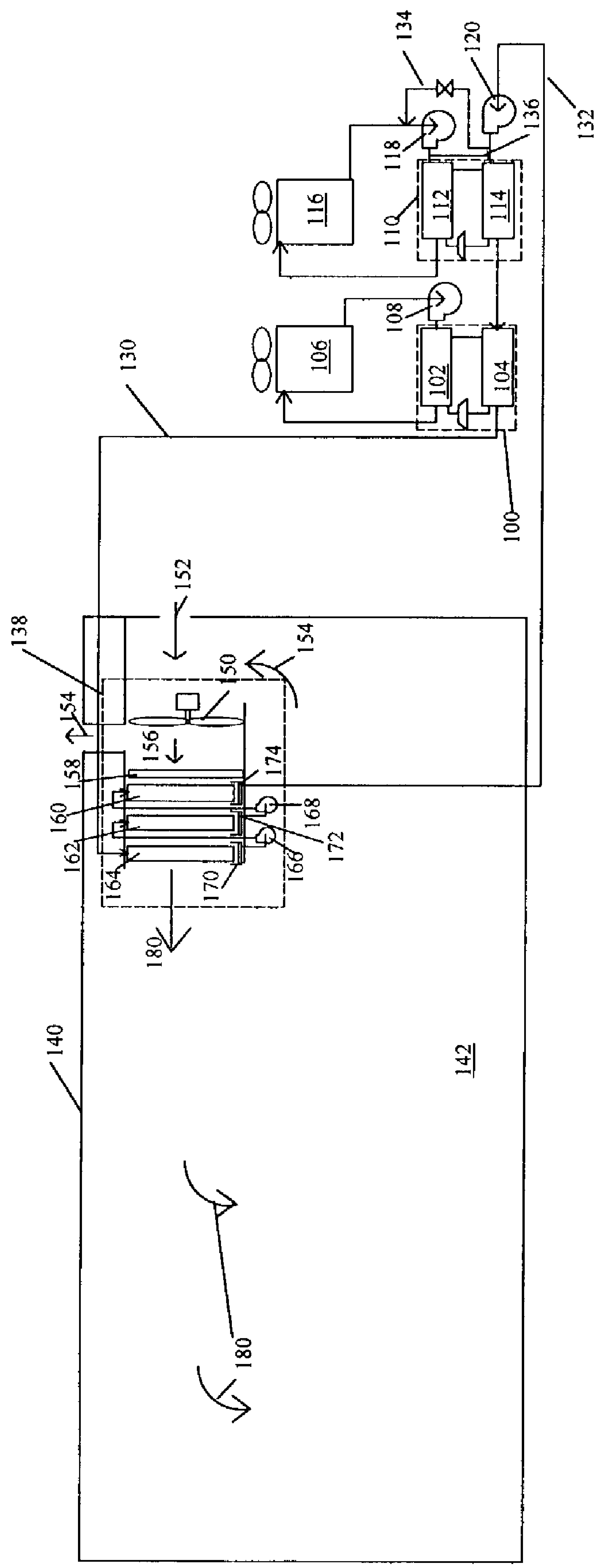


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Figure 4

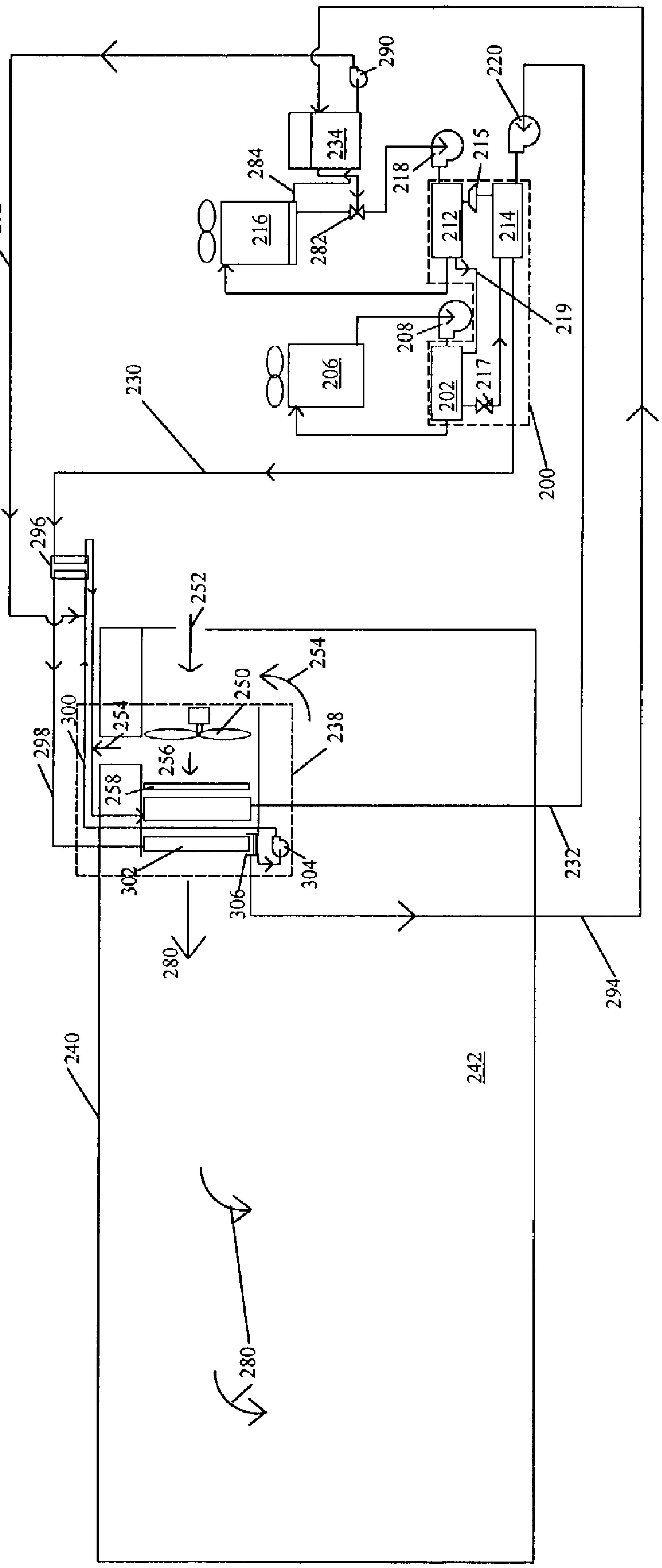
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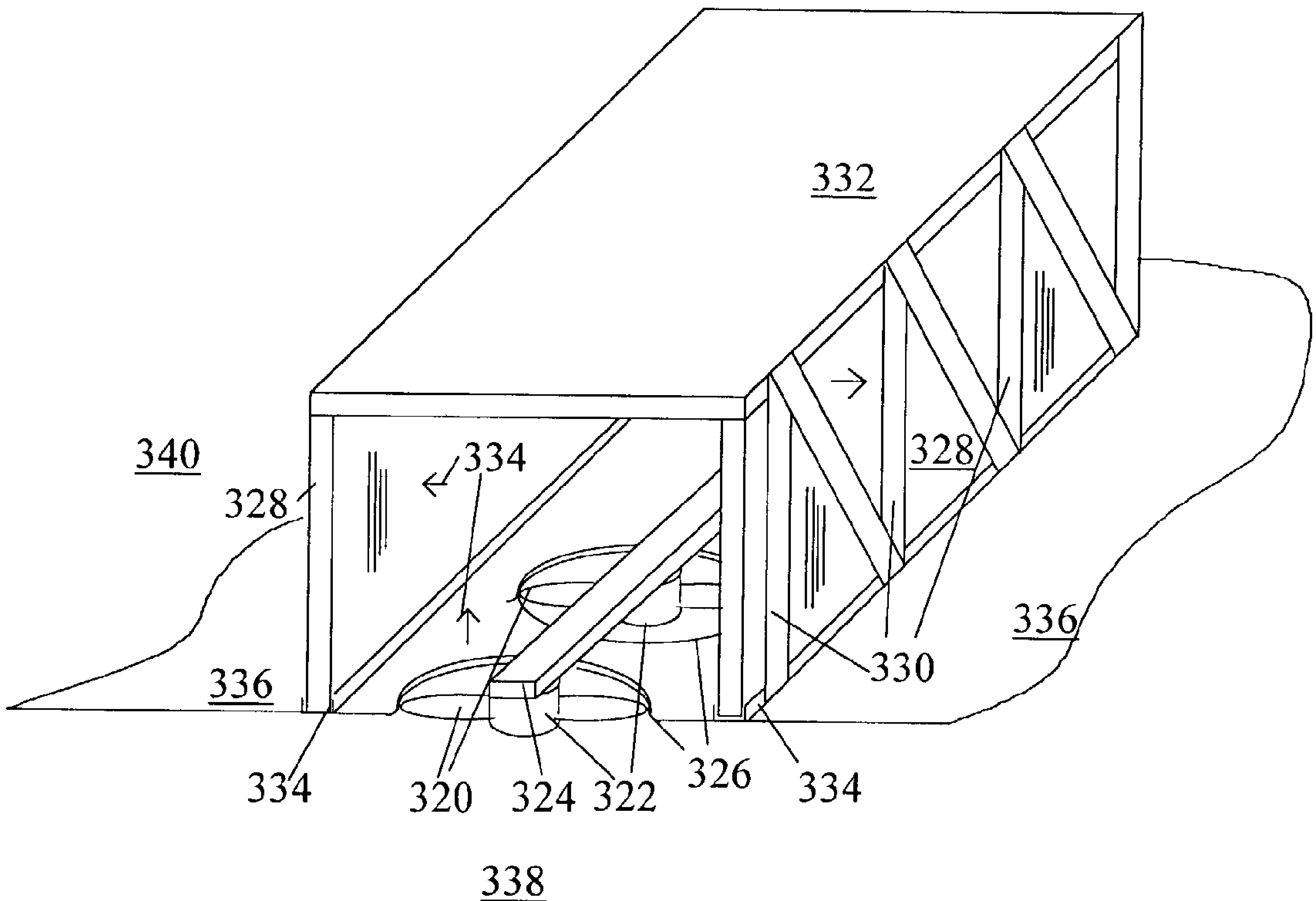
Figure 5

Figure 6



# HIGH-EFFICIENCY AIR HANDLER

## CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application is a continuation-in-part of application Ser. No. 09/772,306 filed on Jan. 29, 2001. Ser. No. 09/772,306 is scheduled to issue as U.S. Pat. No. 6,405,543 on Jun. 18, 2002, and is a continuation-in-part of application Ser. No. 09/331,758 filed on Jun. 25, 1999, which is now patent U.S. Pat. No. 6,185,943. Applicant claims benefit of a co-pending provisional US application entitled “High-Efficiency Air Handler” filed on Jun. 7, 2002.

## BACKGROUND

[0002] 1. Field of Invention

[0003] This invention relates to air handlers that are suitable for use in air conditioning systems. Specifically it relates to an improved air handler that is capable of cooling large volumes of air efficiently and quietly that is suitable for use in a high-efficiency cooling system.

[0004] 2. Description of Prior Art

[0005] Air handlers are commonly used in residential and commercial air conditioning. Air handlers typically comprise a centrifugal blower that moves air over an evaporator or water coil and pressurizes the air for distribution through a duct. For residential systems the typical design static pressure rise across the fan is on the order of one-inch of water. These systems normally use direct-drive blowers with a speed of 800 to 1200 rpm.

[0006] For commercial systems, the pressure rise can be much larger, as much as 8 or 10 inches of water. The fans are normally belt-driven centrifugal fans. Direct-drive variable-pitch fans are also occasionally used with speed of 1150 rpm or greater.

[0007] For residential minisplit systems or room air conditioners, the pressure rise can be smaller because of the absence of ductwork. The fans are direct-drive centrifugal or crossflow. The total air volumes are small, typically less than about 1000 to 2000 CFM.

[0008] There are many problems with large conventional air handlers. The high fan static pressures found in larger systems require greatly increase fan energy input. The high pressures also create high fan sound levels, which require the use of expensive and bulky silencers or sound-absorbing ductwork. The high fan power and high pressures greatly increase the cost of the fans and housings because of higher strength requirements.

[0009] While small minisplit or room systems generally do not have these problems, they are not suitable for conditioning large areas. They generally have a high cost per unit of output. They also lack any means for distributing air for large distances because they supply air directly to the conditioned space.

[0010] One important problem with conventional air handlers is that they are not suitable a new high-efficiency air-conditioning system of the type described in U.S. Pat. No. 6,185,943. This system uses a high volume of air that is supplied at a relatively high temperature, roughly 68 F to provide sensible cooling. A separate dehumidification system handles the latent load. This approach can provide a large reduction energy input compared to conventional systems, but it requires a special air handler that is capable of moving large cooling large volumes of air efficiently.

[0011] These limitations with conventional air handlers are an outgrowth of a fundamental design philosophy found in conventional air conditioning systems. Air-conditioning manufacturers, architects, and professional design engineers have expended huge efforts in optimizing the design of building air-conditioning systems. Annual sales of equipment amount to tens of billions of dollars and annual energy use for heating and cooling have similar values. In addition the costs associated with reduced productivity of workers because of uncomfortable environmental conditions may be several times these figures, although difficult to quantify. Yet despite this effort the fundamental process for air conditioning buildings has remained essentially the same since the introduction of the first air conditioners in the 1920’s. Conventional approaches to air conditioning have inherent problems that severely limit their efficiency, raise installed cost, and frequently produce poor comfort conditions in the building space. Solving these problems requires major changes in the basic configuration of air-conditioning systems.

[0012] Conventional air-conditioning systems use a relatively small volume of air for cooling. The typical arrangement uses a vapor-compression refrigeration system to cool a mixture of return air and outside air to approximately 55° F. and then distribute the cooled air through ducts to the building space. The low supply air temperatures are a result of the need to cool air below its dew point to remove moisture. The low air temperatures are also necessary to meet the sensible cooling needs of the space without excessively large ducts.

[0013] There are several important problems with this approach. The first is related to fan energy use. Since air flow is through relatively restrictive ductwork, fan static pressures are quite high. Typical pressures range from less than 0.5 inches of water for residential systems to as much as 5 to 10 inches of water for large commercial cooling systems. These high static pressures result in large energy use from the fan, which also adds to the cooling load for the rest of the system. In many commercial systems, the fan heat accounts for as much as 20 to 30 percent of the total cooling load for the building. The net result is a very inefficient cooling system.

[0014] A second problem is with high compressor energy required. The low supply air temperatures mean even lower evaporating temperatures, typically 40 to 50 F for the compressor system. The low evaporating temperatures create more work for the compressor, which further reduces the efficiency of the system.

[0015] A third problem is poor indoor air quality associated with high duct humidity. Conditions over 70% relative humidity allow the growth of mold and fungus in ductwork. The relative humidity in the supply ducts for conventional systems is frequently over 90%. In addition water from wet coils wets drain pans and can also wet nearby ductwork. These wet conditions create a potential breeding grounds for many types of microbes that can cause health and odor problems.

[0016] A fourth issue is high noise levels with conventional systems. The high static pressure creates a need for a powerful fan that usually is quite noisy. In addition, metal ducts transmit the noise quite well. Common fixes for the noise problem include the use of fiberglass duct liners. Unfortunately these liners increase cost and pressure drop and also can contribute to problems with molds given the high relative humidity in most ducts.

[0017] A fifth problem is the potential for drafts with conventional cooling systems. The low supply air temperatures and high velocities create the possibility of extremely uncomfortable conditions near the vents. Designers must take special care to ensure adequate mixing of room air and supply air to reduce drafts to acceptable levels.

[0018] A sixth problem is the need for simultaneous heating and cooling. Most office buildings have a single air handling system for the interior and exterior zones. In cold weather the interior zones still need cooling because of heat from people, lights, equipment, etc., while the exterior needs heat. The usual solution is to supply cool air to the entire building in order to satisfy the cooling needs of the interior. Perimeter heaters or heaters in the ducts servicing the exterior zones then provide the heat necessary to satisfy the heating load and overcome the cooling from the supply air. The objective of the present invention is to improve energy efficiency and to reduce or eliminate the problems associated with existing air conditioning systems.

## SUMMARY

[0019] The invention is a new design of air handler. The air handler comprises a coil and a fan. The total air volume moving through the unit is preferably over about 2000 CFM, with a total fan static pressure rise of less than about 0.2 inches of water. The fan is preferably a modified ceiling fan with a rotational speed of less than about 450 rpm.

[0020] The invention preferably is included a new type of air-conditioning system that uses a fundamentally different approach to air conditioning. The approach involves the use of a large volumetric flow rate of air with a temperature that is close to that of the building space for space heating and cooling. A separate dehumidification system is used in humid climates. In one embodiment, a ceiling plenum is used as for the supply air and air returns through the space. In another embodiment, supply air enters the space through a vent near the ceiling along one wall and returns near the floor along the same wall. Pressure drops are kept very low because of the low air velocities. The low pressure and small temperature difference between the supply air and the room air allow for very low energy use and improved comfort.

[0021] Objects and Advantages:

[0022] 1) Minimize fan power

[0023] 2) Minimize fan noise

[0024] 3) Move large volumes of air efficiently

[0025] 4) Minimize first cost

## DESCRIPTION OF THE FIGURES

[0026] **FIG. 1** shows a preferred embodiment that uses a ceiling plenum to distribute supply air.

[0027] **FIG. 2** is a preferred embodiment that returns air through a channel in a window.

[0028] **FIG. 3** is a preferred embodiment for buildings without ceiling plenum.

[0029] **FIG. 4** is an alternate embodiment that uses a cooled liquid desiccant for both cooling and dehumidification.

[0030] **FIG. 5** is another alternate embodiment that uses liquid desiccant for dehumidification.

[0031] **FIG. 6** shows a preferred air-handler embodiment.

## DESCRIPTION OF THE INVENTION

[0032] Preferred system embodiment: **FIG. 1** shows a preferred embodiment of the invention. Fan, **1**, draws air across coil, **2**, where it is cooled or heated to create a supply air stream **40**. Ceiling, **3**, defines the bottom of a ceiling plenum, **4**, that serves as a flow path for air leaving the fan. Vents, **5**, provide openings into that allow supply air to mix with air in an occupied portion of the building space, **6**. Vent, **7**, provides an opening to allow air, **8**, to return through a partition **42** in the space. A separate ventilation system, **9**, provides dehumidified outside air, **10**, to the space and recovers energy from exhaust air, **11**.

[0033] The fan may be a propeller or centrifugal fan. It would have to provide only a small static pressure, typically less than 0.2 inches of water. The low static pressures favor the uses of low-speed fans, which should help to reduce fan sound levels and should reduce fan energy use.

[0034] The coil can contain water or brine or a refrigerant. The supply air temperature for cooling would normally be greater than about 63° F. and preferably about 68 to 70° F. The high temperatures prevent unwanted heat transfer through the ceiling and help to keep the relative humidity in the plenum below 70%. The coil temperature should be a least a few degrees above the dewpoint of the return air and preferably as close as practical to that of the supply air temperature. The high coil temperatures minimize the compressor energy required for cooling and eliminate problems associated with wet coils.

[0035] The ceiling would normally be a suspended ceiling. The tiles should sufficiently rigid to withstand the pressure of the plenum, which would normally be less the 0.1 inches of water. The low static pressures in the plenum reduce the loads on the tiles and reduce the problems associated with leaks around the edge of the tiles. The tiles should provide sufficient resistance to leakage and conduction to prevent undesirable heat transfer between the plenum and the space. In many cases, existing suspended ceilings would meet these requirements without any significant modification.

[0036] This configuration preferably uses very low velocities for the supply air compared to conventional duct systems. According to the *ASHRAE Handbook* 1985 *Fundamentals* for a conventional "low-velocity" duct with at least 10,000 CFM flow rate, the duct would have a velocity of 1300 to 2600 feet per minute. For the present invention at similar volumetric flow rates, the maximum supply air velocity would be less than about 1000 feet per minute and preferably about 100 to 400 feet per minute. This lower velocity is readily achievable because of the huge flow area available in a ceiling plenum compared to conventional ductwork. The low velocities assure low flow noise. They also provide very low pressure drops, which helps to assure proper air distribution to the entire building.

[0037] The vents, **5**, are designed to handle a large volume of air with a minimal pressure drop, typically only a few hundredths of an inch of water. Adjustment may be manual or automatic. The vents should introduce sufficient mixing so as to prevent undesirable drafts.

[0038] Vents, **7**, that allow air to move between zones should be able to handle the required airflow with pressure drops that are smaller than the pressure drop across the ceiling vents. In buildings with raised floors, another option is return air though the space under the floor.

[0039] Ideally the vents would have a control mechanism that is responsive to space temperature without need of a source of outside power. For example wax actuators and shape-memory actuators are capable of producing significant motion in response relatively small changes in space temperature and could be used to control air flow through the vents. Co-pending provisional U.S. application Ser. No. 60/077,008 describes a roller damper mechanism that can work with these types of actuators.

[0040] While this drawing shows the ventilation air entering the ceiling plenum, the exact location where the air is added to the building is somewhat arbitrary, so long as the air temperature is close to that of the space. Likewise the exhaust air can be drawn from anywhere in the building and normally at least a portion would come from toilet exhaust. The ventilation/dehumidification system should incorporate an enthalpy wheel or other heat recovery device, and would preferably be a desiccant-based system capable of providing low dewpoints. The temperature of the air should be close to that of the building space, although this is not required if the air is mixed into the supply air. The ventilation system should also provide a small positive pressure for the building space to reduce possible of infiltration of outside air.

[0041] While the preferred dehumidification system is combined with a heat recovery ventilation system, many other configurations are possible. For example, the dehumidification system can simply further cool a portion of the air leaving the cooling coil so that its temperature drops below the dewpoint. A heat pipe or other device for exchanging heat between the air on the coil and the air leaving the coil can increase the amount of moisture remove compared to sensible cooling, which can reduce energy use. This arrangement is acceptable in cases where adequate outside air is available to the space from infiltration or other sources. Numerous other dehumidification systems that appear in the prior art could be used in the new system. The ASHRAE Handbooks describe many of these dehumidification options.

[0042] In dry climates the dehumidification system can be eliminated, although sensible heat recovery may still be a valuable option. There is also potential for eliminating the need for a compressor, with sensible cooling provided with an indirect evaporative cooler or cooling tower.

[0043] The table below shows the massive energy advantages of the invention when compared to a conventional air-conditioning system in handling the sensible cooling load:

Comparison of Energy Use for a Conventional Cooling System and New Invention

| | conventional | new high-flow | units |
|---|---|---|---|
| zone sensible load | 20 | 20 | btu/hr/ft2 |
| supply air temperature | 55 | 70 | deg F |
| room temperature | 75 | 77 | deg F |
| cfm/ton of total sensible load | 556 | 1587 | cfm/ton |
| fan static pressure | 6 | 0.2 | inches H2O |
| fan static efficiency | 70% | 50% | |
| motor efficiency | 90% | 80% | |
| fan power (hp/1000 CFM) | 1.349 | 0.063 | hp/1000 cfm |
| fan power(w/CFM) | 1.12 | 0.06 | w/cfm |
| fan heating | 3.53 | 0.19 | deg F |
| fan heat (% of sensible load) | 18% | 3% | |
| coil load | 23.5 | 20.5 | btu/hr/ft2 |
| chilled water temperature | 45 | 65 | deg F |
| chiller energy use | 0.6 | 0.3 | kw/coil ton |
| chiller energy use | 0.706 | 0.308 | kw/building ton |
| fan energy use | 0.528 | 0.091 | kw/building ton |
| total energy use | 1.234 | 0.399 | kw/building ton |
| percent energy saved | | 67.7% | |

[0044] This analysis shows that the new system can save over two thirds of the energy used for sensible cooling at design conditions. At off-design conditions the savings can be even larger because of the increased availability of free cooling because of the much high chilled water and supply air temperatures. This free cooling option means that the chiller may be shut down for a large portion of what is normally the cooling season.

[0045] The system should also have a major advantage in handling latent load. The use of an enthalpy wheel or other suitable heat exchanger can reduce loads associated with bringing in outside air by 80%. Heat recovery also greatly reduces heating requirements. For most office and retail buildings, the outside air is the main source of moisture. Use of a gas-driven desiccant system also gives the opportunity to greatly reduce electric demand charges while efficiently handling the ventilation load. Electrically driven systems are also an option.

[0046] Use of a separate dehumidification system also greatly reduces the need to run the whole system when the building is unoccupied. Current systems frequently require continuous operation during conditions of high humidity in order to prevent excessive accumulation of moisture in building materials during off periods. The present invention allows the operation of the dehumidification system alone, which greatly reduces operating costs while providing good moisture control.

[0047] Embodiment with Alternate Return-Air Configuration: **FIG. 2** shows a variation of the first embodiment that is designed to greatly reduce the need for heating. The basic idea is to move a large volume of air from the interior toward the exterior of the building. The system also draws return air from the building envelope. Return air, **13**, is drawn upward through channel, **19**, that is formed between exterior glazing, **12**, and interior glazing, **17** of a window **44**. This arrangement effectively removes any cold air associated with heat loss through glazing, **12** and an exterior wall, **18**. The return air then moves into channel, **14**. Fan, **15**, draws air from the channel through coil, **16**, and then discharges the conditioned air into the ceiling plenum **4** as a supply air stream **41**.

[**0048**] This configuration several advantages that greatly reduce winter heating requirements. The first is that it removes cold from the building envelop before it enters the conditioned space. The second is that it then moves this air toward the interior so as to provide necessary cooling. Third it then uses the air returning from the interior to provide as source of warm air for the exterior zones. This system should not require any significant amount of heat so long as the interior heat generation exceeds the exterior heating load. Proper insulation of windows and walls can effectively eliminate the need for heat in most larger buildings even in the most severe climates. The only time that heat would normally be required, would be if the building were unoccupied for a long period of time with limited sunlight. Under these circumstances, the coils provide heat to warm the entire building.

[**0049**] **FIG. 3** shows a third preferred embodiment of the invention. This configuration is suitable in retail space or similar applications with large open areas and few obstructions near the ceiling. Fan, **23**, moves supply air, **20**, from coil, **24**, through vent, **25**, to mix with air in building space, **6**. The air returns through register, **21**, and return duct, **22**, back to coil, **24**. As with the other embodiments, a separate dehumidification system supplies outside air and recovers heat from exhaust air.

[**0050**] A key feature of this embodiment is the combination of high air volume, high temperature, low velocity, and low relative humidity of the supply air compared to conventional systems. The preferred velocity of the air flowing through the vent is low, less than 1000 feet per minute and preferably about 100 to 500 feet per minute. The air volume flow requirements are large, typically over twice that of convention systems per unit of cooling capacity, which corresponds to at least 10,000 CFM for a small commercial building (5 to 10-ton load). For a typical retail building (50,000 to 100,000 square feet) the volumetric flow rate amounts to over 100,000 CFM. A preferred supply air temperature is high, at least about 63° F. and preferably 68 to 70° F. The relative humidity of the supply air is low compared to conventional systems, less than about 90 percent and preferably about 75 percent or lower. The combination of low velocity, high air volumetric flow rate, and high supply air temperature allows for a very long throw of 100 feet or more without risk of cold, high-velocity drafts. The low relative humidity of the supply air assures proper humidity control in the space. These supply-air conditions provide comfort in the building space in addition to providing great opportunities for energy savings.

[**0051**] The large volumetric flow rates and relatively warm temperatures of the supply air allow for very long throws that may be necessary to supply air to a large space. The higher supply temperatures also greatly reduce the risk of uncomfortable drafts in the space. As with the other embodiments, this system has a large advantage in efficiency because of the high coil temperatures and low fan static pressures. It should have a major first cost advantage since it virtually eliminates the need for ductwork. One disadvantage is that it does not provide local temperature control within the building space, which may limit its application.

[**0052**] Cooled-Desiccant Embodiment: **FIG. 4** shows an embodiment that uses a cooled liquid desiccant for both cooling and dehumidification for comfort air conditioning in a building **140**. This embodiment uses two chillers. A water-cooled chiller **100** includes a water-cooled condenser **102** and a desiccant cooler **104**. A condenser water pump circulates cooling water through the condenser **102** to a cooling tower **106**.

[**0053**] The second chiller is a desiccant-cooled chiller, **110**. It comprises a desiccant-cooled condenser **112** and a desiccant cooler **114**. A condenser pump **118** circulates a liquid desiccant through the condenser to a cooling tower **116**. The waste heat from the condenser cools heats the desiccant fluid, which cause water to evaporate out of the desiccant and creates concentrated desiccant. The desiccant cooling tower should be of special design to ensure material compatibility and prevent excessive loss of desiccant material.

[**0054**] A desiccant loop provides sensible and latent cooling to a building **140**. The desiccant loop comprises a cooled-desiccant pump **120** the pumps desiccant through the desiccant coolers **114** and **104**. A supply desiccant line **130** supplies the cooled desiccant to an air handler **138**. A return desiccant line **132** returns the desiccant from the air handler to the cooled-desiccant pump **120** to complete the loop.

[**0055**] The air handler **138** uses cooled desiccant to cooled and dehumidify a mixed air stream **156**. The mixed-air stream **156** is a mixture of outside air **152** and return air **154** that is moved by a fan **150**. A portion of the mixed-air stream leaves the building as exhaust air **154**. The remaining mixed-air stream enters a filter **158** and then goes through a first, second, and third direct-contact heat exchangers **160**, **162**, and **164** respectively. The direct-contact heat exchangers allow simultaneous heat and mass transfer between the air and the cooled desiccant and are preferably arranged in a counter-crossflow configuration.

[**0056**] The first direct-contact heat exchanger receives desiccant from a first sump pump **168**. A first sump **174** collects desiccant that drains off of the first direct-contact heat exchanger. Likewise the second direct-contact heat exchanger receives desiccant from a second sump pump **166** and has a second sump **172**. The desiccant collected in the second sump **172** supplies the first sump pump **168**. A third direct-contact heat exchanger **164** receives desiccant from the supply desiccant line **130**. Air flows through the first then the second and the third direct-contact heat exchangers so as to approximate a couterflow configuration. This setup allows for a close approach temperature. While three passes of cooling are shown in **FIG. 4**, other numbers a possible and may be desirable depending on the details of the design of the air handler. The preferred number is between 1 and 5 passes.

[**0057**] A supply air stream **180** exits in an approximately horizontal direction from the air handler **138**. As with the previous embodiments as for example **FIG. 3**, the supply air has a relatively high temperature, low speed, and low relative humidity compared to conventional designs. The preferred values for these conditions are similar to those for the earlier embodiments. The supply air stream gradually slows as it moves away from the air handler and mixes into the air in an occupied portion **142** of the building **140**.

[**0058**] The preferred desiccant material is calcium chloride, although other materials such as various glycols, lithium chloride, or lithium bromide are possible. The advantages of calcium chloride include its low-cost, availability, and very low toxicity. Its long history of use as a brine for refrigeration applications means that compatibility with materials of construction is well known. Because the required relative humidity is relatively high (about 70%) compared to the low values required in most other desiccant applications (typically about 30% or less), the relatively high equilibrium vapor pressure of calcium chloride solutions is not a problem.

[**0059**] Alternate liquid desiccant system embodiment: **FIG. 5** shows an embodiment with liquid-desiccant dehumidification with a single chiller. The chiller 200 comprises a liquid cooler 214 that evaporates refrigerant and supplies refrigerant vapor to a compressor 215. While the chiller is shown outside, it can also be located inside a building to be cooled. The discharge of the compressor goes into an auxiliary condenser **212** the heats desiccant. The refrigerant leaving the auxiliary condenser then goes into a main condenser **202** that exchanges heat with condenser water. Liquid refrigerant leaves the main condenser through a liquid line **217** that includes valve, orifice, or other pressure drop.

[**0060**] The main condenser is part of a cooling tower loop. A condenser-water pump **208** moves water through the condenser to a cooling tower **206**, which normally cools the water by evaporation into the atmosphere. Dry cooling towers are also an option.

[**0061**] The auxiliary condenser is used to heat liquid desiccant. A desiccant pump **218** move desiccant through the auxiliary condenser to a direct-contact heat exchanger **216** that is a type of cooling tower. The direct-contact heat exchanger evaporates water from the warm desiccant. The majority of desiccant leaving the heat exchanger returns through a control valve **282** to the desiccant pump. The rest goes through line **284** to a desiccant storage tank **234**. The desiccant storage tank keeps a supply of desiccant for dehumidification when the auxiliary condenser is not operating.

[**0062**] The operation of the condenser pumps allows for efficient production of chilled water and concentrated desiccant. To produce concentrated desiccant, the chiller is run with the desiccant pump **218** is on, and the condenser water pump **208** turned off. The chiller then runs with a relatively high condensing temperature (about 100 to 130° F.) to regenerate the desiccant. When additional concentrated desiccant is not required, the desiccant pump is turned off; and the condenser water pump **208** is turned on, which allows the chiller to run with a lower condensing temperature (typically less than 105° F.). When no chilled water is required, the chiller is turned off. While **FIG. 5** shows a series refrigerant flow configuration for the condensers, the condensers can also share a common shell (shellside refrigerant) with separate tube bundles and liquid connections.

[**0063**] The storage tank is preferably sized to provide desiccant for at least an hour or two of operation. Storage capacity with at 8 to 12 hours of storage allows for significant demand shifting. The volume of storage required is small since because the desiccant uses the heat of vaporization of water as the storage mechanism. Latent energy storage of several hundred Btu/lbm is possible. The storage capacity of the desiccant related primarily to concentration of solution, not temperature, so insulation of the tank is not normally required.

[**0064**] A desiccant pump **290** draws desiccant from the bottom of the storage tank, which normally contains the most-concentrated desiccant, and move the desiccant through a desiccant supply line **292**. The desiccant supply line adds concentrated desiccant to a warm desiccant line **300** that is part of fluid circuit includes a direct-contact heat exchanger **302** in an air handler **238**. The warm desiccant flows into a heat exchanger **296** that cools the desiccant using chilled water. The desiccant then flows through a cooled-desiccant line **298** to the direct-contact heat exchanger **302**. A sump **306** collects warm desiccant from the direct-contact heat changer **302**. Most of the desiccant then enters a desiccant pump **304** that pumps it through the warm-desiccant line **300** to complete the loop. A portion of the desiccant drains from the sump **306** through a diluted-desiccant line **294**, which returns it to the desiccant storage tank **234**, preferably at a location near the top liquid in the tank.

[**0065**] The air handler **238** also comprises fan **250** and a coil **260**. The fan **250** draws return air **254** and outside air **252** into the air handler. A portion of a mixed air stream **256** leaving the fan exits as an exhaust air stream **254**. The rest of the air stream goes through a filter **258** to a coil **260** and then through the direct-contact heat exchanger **302**, which dries the air with the desiccant. The air leave the air handler as a supply air stream **280**, which flows in a roughly horizontal direction at low speed as in the earlier embodiments. The supply air **280** mixes with air in an occupied portion of a building space **242**.

[**0066**] Cooling water from the coil come from the chiller. A chilled water supply line **230** leaves the water cooler **214** and enters the heat exchanger **296** that cools the desiccant. The chilled water then enters the coil **260** and returns to a chilled-water pump **220** through return line **232**. The chilled-water pump **220** pumps the water through the cooler **214** to complete the chilled-water circuit.

[**0067**] Some features can be changed while keeping the basic function of the system. For example, the heat exchanger **296** for cooling desiccant can be eliminated. Another possibility is to place the desiccant upstream of the cooling coil. These changes would increase the desiccant concentration and lower chilled water temperature necessary to achieve a given supply air temperature and relative humidity. Another possibility is to uses the auxiliary condenser as a desuperheater that operates at the same time as the main condenser. This change is possible with refrigerants that produce a high discharge temperature, such as R-22, but it is not normally an option with R-123, which has little superheat. While the chiller and associated equipment is shown outside, it can also be located inside the building to be cooled or in a separate structure.

[**0068**] Yet another option is to incorporate a heat-recovery heat exchanger or enthalpy recovery wheel to reduce ventilation energy requirements. This approach reduces energy use, but may be feasible in every case depending on the ability to economically recover energy from the exhaust air. These changes or similar changes or combinations of changes do not affect the basic function of the system

[0069] Other possible system configurations: There are many possible variations of these embodiments. For example, through not preferred, a conventional heat-pipe reheat system with air cooled below the dewpoint can provide similar supply-air conditions. Mixing return air and supply air to achieve a high supply air temperature is also an option, though not preferred.

[0070] Other systems for regenerating desiccant using heat from combustion or heat from solar energy is another option. The solar option is explore more fully in a co-pending application entitled, "Solar air conditioner." Other configurations of chillers or heat pumps are also possible for supplying heat to regenerate a desiccant.

[0071] Thermal storage using chilled water is another possibility. This option is discussed in a co-pending application entitled, "Air conditioner with thermal storage."

[0072] **FIG. 6**—Preferred Air-Handler Embodiment:

[0073] **FIG. 6** shows cross-sectional view of a preferred air-handler embodiment that uses modified ceiling fans. The basic configuration is a box that is formed by a housing **332**, coils **328**, and a bottom housing **327**. The housing approximates an airtight enclosure for the top and ends of the box. Fans **318** comprise fan impellers **320** that are each driven by fan motors **322**. Orifices **326** in the bottom housing **327** encircle each impeller. The fan motors are attached to a support **324**.

[0074] Coils **328** provide a heat-transfer surface between air and another fluid, normally water or a refrigerant such as R-22 or R-134a. A truss **330** and a coil support **334** provide a structural support. While preferred applications would use a high supply air temperature and a dry coil, the coil support **334** can also serve as a channel for catching condensate that may collect on the coil during transient or unusual operating conditions.

[0075] The coils are preferably one or two rows deep with ⅜-inch copper tubes and aluminum fins. The coils preferably include enhanced heat-transfer surfaces such as louvered or slit fins and rifled tubes to minimize material and size requirements. In contrast to conventional air handlers, which have a coils face velocity of about 500 to 600 feet per minute, the preferred face velocity for the present invention is about 100 to 300 feet per minute so as to minimize airside pressure drop and reduce sound levels. For larger units, each coil face has multiple sections to simplify assembly.

[0076] While **FIG. 1** shows the coils in a vertical arrangement, it may be desirable to arrange the coils in an inverted "V". An inverted "V" configuration has the advantage of eliminating the need for a top cover for the air handler, but it may reduce the strength of the coils to resist vertical shipping loads, which may require additional structural re-enforcement to the coils.

[0077] The air handler preferably sits on top of a hole in a drop ceiling **36** and moves air from an occupied space **38** through the coils to a ceiling plenum **40**. Air from the ceiling plenum is then distributed through vents to the occupied space. This flow configuration is generally preferred for space cooling.

[0078] The fans may also operate to move air in the opposite direction so as to draw from the ceiling plenum and discharge air to the occupied space. An advantage of this configuration is that the velocity of air from the fans can overcome buoyancy effects to ensure good air circulation in heating mode. For propeller fans, simply reversing the rotation reverses the direction of airflow.

[0079] The fan impellers are preferably propeller fans with an efficient twisted, airfoil cross-section, although a range of designs is possible. A modification of a fan made sold under the Gossamer Wind brand name is an example design. The modification can be as simple as cutting down the fan blade from 52 inches to about 36 inches to allow the fan to rotate at a higher speed without overloading the fan motor. The smaller fan diameter allows the fan rotational speed to increase from about 200 rpm for the conventional ceiling fan to about 250 to 350 rpm for the modified fan.

[0080] While an airfoil shape is preferred for maximum efficiency, flat, metal or wooden blades such as those found in conventional ceiling fans may also be used. In addition to wood or metal, the fans can be made of plastic, foamed plastic, cardboard, or paper.

[0081] Radial-flow plenum fans or centrifugal fans are another option. For a given static pressure difference, radial-flow fans allow a significant reduction in tip speed (on the order of 20 to 70% lower) compared to axial fans. A preferred radial fan would be a plenum fan with backwardly curved airfoil blades. In order to match the capability if conventional ceiling fan motors, the diameter of a radial-flow fan may be substantially smaller than that of an axial-flow fan. Diameters of about 12 to 30 inches may be suitable.

[0082] Radial-flow fans may have an advantage in lower sound levels and reduced strength requirements, and may be preferred for some applications. The reduced strength requirements may allow the use of low-cost materials such as cardboard, paper, or plastic in the construction of fan impellers.

[0083] In addition to axial and radial-flow fans, other fans, such as mixed-flow or transverse-flow fans are also an option. A key feature is the need to maintain a low rotational speed and tip speed to prevent excessive noise and to reduce power requirements. This design also implies the use of a very low static pressure compared to conventional air handlers.

[0084] For minimum cost and maximum use of readily available equipment, the motors are preferably ceiling fan motors. These motors are normally 18-pole or 16-pole, single-phase induction motors. They may be either shade-pole or PSC (permanent split capacitor) motors, but PSC design is preferred because of better efficiency characteristics and the ability to reverse direction.

[0085] Other motor options include DC motors, and permanent-magnet motors, or three-phase induction motors with associated electronic drives. These motors may have a large efficiency advantage over conventional fan motors. They also have the advantage of providing efficient variablespeed operation. For some of these options, such as for induction motors, a single electronic drive may generate a low-frequency output for multiple motors. The chief disadvantages of these approaches are higher cost and increase engineering required compared to conventional ceiling-fan motors.

[0086] Belt drives or gear drives are another option. Belt drives or gears allow the motor to run at a higher rotational speed than the fan impeller, which reduces motor size and cost. Mechanical drives can also allow a single motor to drive multiple fans, which can reduce motor cost. Using a single larger, high-speed motor can also improve motor efficiency compared to multiple low-speed motors. The chief problems with mechanical drives are maintenance, reliability, and complexity compared to direct drives. Mechanical drives can also produce significant noise. For these reasons, belts and gears are generally not preferred.

[0087] The preferred fan rotational speed ranges from about 100 to 450 rpm for propeller fans, which is consistent with the capabilities of conventional ceiling fan motors. The corresponding fan tip speed is approximately 1000 to 4000 feet per minute, which is much lower than that for conventional axial-flow fans. The lower tip speed greatly reduces fan sound levels and reduces fan input power. It also reduces the pressure capability of the fan. In contrast to conventional fans with a design static pressure rise excess of 0.5 inches of water, fan static pressure rise in the present invention is normally less than 0.2 inches of water, and is preferably about 0.01 to 0.05 inches of water so as to minimize fan noise and fan power requirements.

[0088] The airflow capability of the air handler is quite large. Each fan would normally move roughly 1000 CFM or more, which an air handler having a total capability of 1,000 to 40,000 CFM.

[0089] When multiple fans are used with a single air handler, a check valve may be included in series with each fan to prevent backflow of air when it is not operating. The check valve should be of lightweight construction to prevent excessive pressure drop and to ensure that it opens in response to the operation of the fan. The check valve can be constructed of lightweight plastic sheets that open in response to a small pressure difference. In the case where reversing operation is desired, this check valve may be replaced with an actuator-driven damper. Another option is to include a divider (such as a sheet metal plate) between the fans to separate air handler into compartments. While these options may improve reliability and allow more variation in operation, they are not required for the basic function of the unit and may introduce unnecessary cost in many installations.

[0090] For improved motor efficiency, modification of this motor design may be preferred. A relatively simple change is to reduce rotor resistance. This may be accomplished by selecting a lower-resistance material for forming the rotor conductors or by increasing the cross-sectional area of the conductors. In contrast to conventional ceiling fan motors, varying fan speed by changing the supplied voltage to the motor is not required for operation of the air handler. This allows the selection of a lower rotor resistance, since high-slip operation at reduced voltage is not required. Proper capacitors for starting and running the motor may be selected in accordance with design approaches found the prior art. While conventional ceiling fans have as large as 60% slip at design conditions, the preferred slip for the present invention is as low as possible with 5 to 30% being reasonable design target.

[0091] An alternative approach is to supply two-phase power to the fan motors. Two-phase power may be supplied from a three-phase to two-phase transformer or an electronic inverter, which are found in the prior art. Two-phase power allows PSC fan motors to run with without capacitors and can improve starting torque characteristics. This approach may allow for a further improvement in motor efficiency through the use of a lower rotor resistance and optimization for two-phase operation.

[0092] Air filtration is another issue. Filtration generally needs to address to issues. First there is the need to prevent clogging or fouling of the coils. The high face velocity in conventional systems especially sensitive to this problem. In addition, the large number of rows deep (generally at least 3 to 12 rows) makes it difficult to remove lint or dust from the interior of a coil.

[0093] A second issue is to remove pollen, dust, and similar materials for the comfort and health of the building occupants. The particle size of these materials is generally much smaller than those that can clog heat exchangers, so a greater level of filtration is required. This high-level of filtration requires a higher pressure drop for the same filter velocity, which can greatly increase the corresponding fan energy.

[0094] There are several options for air filtration. One approach, though not preferred, is to include a conventional filter in the air handler. The filter should be designed with as large of a face area as possible to minimize pressure drop. The filter is preferably placed upstream of the coil and may include pleats or "V" configuration to maximize face area.

[0095] A better option is to separate filtration from the air handler. A higher-pressure fan can provide a high level of filtration to a smaller volume of air, so as to remove pollen, smoke, or other materials from the air stream. This approach eliminates the need for filtering the air these purposes in the air handler.

[0096] As for the issue of clogging the coil, the present invention should be able to handle this problem without filtration. The low-face velocity, few rows deep, and dry coil surface make it relatively resistant to clogging. No filter upstream of the coil is an option, especially in relatively lint-free environments. A simple vacuum cleaner should be sufficient to remove dust from the coil if necessary. Washable screens or low-efficiency filters are also alternatives.

[0097] 8. better indoor air quality,

[0098] 9. low noise,

[0099] 10. no cold drafts,

[0100] 11. increased economizer use possible

[0101] 12. ability to use thermal storage for demand shifting, and

[0102] 13. efficient dehumidification using liquid desiccant.

**1)** An air handler for air conditioning buildings comprising:

a) A cooling coil that transfers thermal energy from air to a second fluid and

b) A fan with a total design airflow greater than about 2000 CFM and a design static pressure less than about 0.5 inches of water.

c) A flow path between said fan and said cooling coil so that said fan moves air over said cooling coil, which cools the air and thereby produces a cooled air stream.

**2)** The air handler of claim 1 wherein said fan has a design static pressure rise of less than about 0.2 inches of water.

**3)** The air handler of claim 2 wherein said fan has a design pressure rise of less than 0.05 inches of water.

**4)** The air handler of claim 2 wherein the temperature of said cooled air stream is above the dewpoint temperature of air entering the air handler.

**5)** The air handler of claim 2 wherein said fan is an axial-flow fan.

**6)** The air handler of claim 2 wherein said fan is a radial flow fan.

**7)** The air handler of claim 6 wherein said radial fan is a plenum fan with backwardly curved blades.

**8)** The air handler of claim 2 wherein said fan has blade with an airfoil cross-section.

**9)** The air handler of claim 2 wherein air leaving said air handler is discharged into a ceiling plenum.

**10)** The air handler of claim 2 wherein said flow channel comprises a ceiling plenum.

**11)** A heat-exchange device comprising:

a) A heat exchanger for transferring heat between air and a second fluid,

b) An electric motor with a design rotational speed of less than about 500 rpm, and

c) An impeller that is directly driven by said motor that moves air across said heat exchanger.

**12)** The air handler of claim 11 wherein said electric motor comprises an induction motor.

**13)** The air handler of claim 12 wherein said electric motor has a synchronous speed of less than about 600 rpm.

**14)** The air handler of claim 12 wherein said induction motor is a single-phase motor.

**15)** The air handler of claim 12 wherein said induction motor is supplied with two-phase power.

**16)** The air handler of claim 12 wherein the rotational speed of said induction motor is greater than about 70% of the synchronous speed.

**17)** An air handler comprising:

a) A fan,

b) A cooling coil with a design face velocity of less than 300 feet per minute, and

c) A flow path between said cooling coil and said fan so that said fan mover air through said cooling coil.

**18)** The air handler of claim 17 wherein said cooling coil is less than 3 rows deep.

**19)** The air handler of claim 18 wherein said cooling coil has a supply air temperature of above the dewpoint temperature of air entering said air handler.

**20)** The air handler of claim 19 wherein said supply air temperature is above about 62 F.

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