A New Ventilation System Integrates Total Energy Recovery, Conventional Cooling and a Novel "Passive" Dehumidification Wheel to Mitigate the Energy, Humidity Control and First Cost Concerns Often Raised when Designing for ASHRAE Standard 62-1999 Compliance

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Abstract:

This paper introduces a novel, "passive" desiccant based outdoor air preconditioning system (PDH) that is shown to be significantly more energy-efficient than all known alternatives, and has the unique ability to dehumidify outdoor air streams to very low dewpoints unattainable with conventional cooling approaches. The system allows for precise control of the indoor space humidity while delivering high quantities of outdoor air, at both peak and part load conditions, and during both occupied and unoccupied modes. Low operating cost, reasonable first cost and a significant reduction in cooling plant capacity requirements provide a life cycle cost that is substantially less than that of more conventional system approaches.

Introduction

The quality of indoor air has been directly linked to the frequency of illness and has been shown to have a direct impact on worker productivity. Research suggests that indoor humidity levels may have a greater impact on the health and comfort of building occupants than previously suspected (Arundel 1986). Microbial activity (e.g., mold and fungus), which increases at higher indoor humidity levels, has been shown to emit harmful organic compounds and toxins (Bayer 1995). Childhood asthma has been linked to mold, microbial activity and high humidity (Nevalainen etal 1997).

In addition to direct health effects, the odors associated with inadequate ventilation and/or microbial activity are often cited as a primary reason why indoor air quality is considered unacceptable to occupants. When odors are encountered in a building, building operators often respond by increasing outdoor air quantities in an attempt to eliminate odors. This often exacerbates the problem because increasing outdoor air quantities to conventional HVAC systems often results in higher indoor air humidity levels, which, in turn, fosters continued microbial activity (Downing, 1993).

ASHRAE provides guidance to the design community through its IAQ Standard 62, entitled, "Ventilation for Acceptable Indoor Air Quality." It emphasizes the importance of continuous outdoor air ventilation as well as the need for controlling indoor humidity levels. The current version, Standard 62-1999 has been written in code language and is undergoing modification through addenda. One of the proposed modifications to the standard would place further emphasis on humidity control, especially in hot and humid climates, highlighting the need for maintaining proper humidity levels during both peak and partial loading conditions, and during both occupied conditions (where ASHRAE recommends limiting the relative humidity to 60%) and unoccupied conditions (where ASHRAE recommends limiting the relative humidity to 70%).

As engineers have worked to identify effective ways to accommodate ASHRAE Standard 62 requirements, many have realized the need for a fundamental change to the way they have designed HVAC systems in the past. Through experience, many have recognized first hand the limitations of conventional, especially packaged cooling equipment, when challenged with supplying an increased amount of outdoor air on a continuous basis. These limitations are well presented by the references listed for Henderson (1996), Khattar (1985), Fischer (1996) and others.

One of the more effective solutions for accommodating the ASHRAE standard involves preconditioning the outdoor air stream, to "decouple" the latent load from a downsized conventional HVAC system sized to handle primarily the space sensible load. This design approach is described by references Fischer (1996), Smith (1996) and Coad (1995). The conventional approach being used to precondition or "decouple" outdoor air loads involves increased cooling capacity to over-cool the outdoor air to the required humidity content followed by reheat. More advanced systems, such as total energy recovery and active desiccant systems provide far more energy efficient solutions yet may involve a higher project first cost. Each of these systems are described later, including both design benefits and limitations.

This paper introduces a novel, "passive" desiccant based outdoor air preconditioning system (PDH) that is shown to be significantly more energy-efficient than all known alternatives, and has the unique ability to dehumidify outdoor air streams to very low dewpoints, unattainable with conventional cooling approaches. The PDH system allows for precise control of the indoor space humidity while processing high quantities of outdoor air, and does so at both peak and part load conditions, during both occupied and unoccupied modes. The combination of low operating cost, reasonable first cost and the significant reduction in cooling capacity requirements provide for a life cycle operating cost that is substantially less than that recognized with conventional system approaches.

Review of Outdoor Air Preconditioning Systems

To date, most projects designed to "decouple" the outdoor load from the space load utilize traditional over-cooling and reheat systems (Figure 1). With the use of this conventional over-cooling/reheat approach comes a very high energy consumption, and often requires a more complex chilled water system to perform as desired (as opposed to DX) due to the wide variation in outdoor air conditions over time. Energy consumption is high since the outdoor air needs to be cooled to a very low outlet temperature (typically 50 to 55 degrees) in order to reach the desired supply air dewpoint. Since this air is often too cold to provide directly to the occupied space, especially at part load conditions, it is frequently reheated before being delivered to the occupied space.

Over time an increasing number of desiccant based system solutions have been successfully applied to accommodate the requirements of ASHRAE Standard 62. These approaches are far more energy efficient than the conventional over-cooling/reheat method described previously. The most fundamental desiccant based system utilizes a total energy wheel to recovery up to 80% of the sensible (temperature) and latent (moisture) energy from the air exhausted from a facility (Figure 2). This significantly reduces the energy required to cool and dehumidify the outdoor air, while providing most of the heat and humidification required during the heating season. This approach does not reduce the amount of reheat energy required when compared with Figure 1.

The total/sensible recovery dual wheel system (DWS) combines a total energy and sensible only recovery wheel, along with a conventional chilled water or D.X. coil, to cool and dehumidify or heat and humidify the outdoor air stream (Figure 3). The outdoor air stream passes through the total energy wheel where it is precooled and dehumidified. The outdoor air is then passed through the cooling coil where it is further cooled and dehumidified until reaching the absolute humidity level (moisture content) required by the occupied space. The cold, dehumidified outdoor air stream is finally passed through a sensible only wheel where it is reheated to the desired space neutral temperature, using only the energy contained within the return air stream being exhausted from the space. Until now this DWS has been the most energy efficient outdoor air preconditioner available.

The "active" desiccant based cooling system (DBC) combines a traditional active desiccant dehumidification wheel, a sensible only recovery wheel, a regeneration heat source and in most cases, an evaporative cooler. With this system the outdoor air is passed through the thermally regenerated desiccant wheel, wherein the air is dehumidified and warmed. The hot, dehumidified outdoor air exiting the desiccant wheel is post cooled by passing it through the sensible only wheel. On humid days, a significant amount of mechanical cooling is required to reduce the supply air temperature before it can be delivered to the occupied space.

Each of these preconditioning systems will be discussed later when they are compared with a novel Total Recovery/Passive Dehumidification System (PDH).

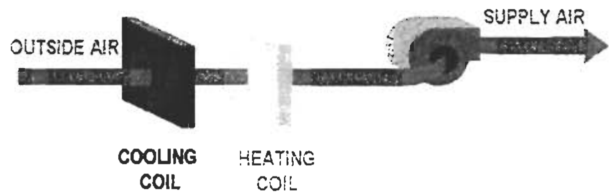


Figure 1: Conventional Cooling and Reheat Approach

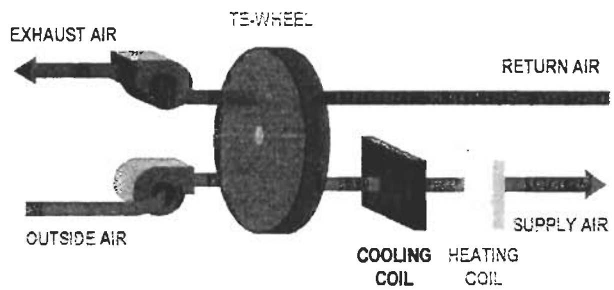


Figure 2: Total Energy Recovery, Cooling and Reheat

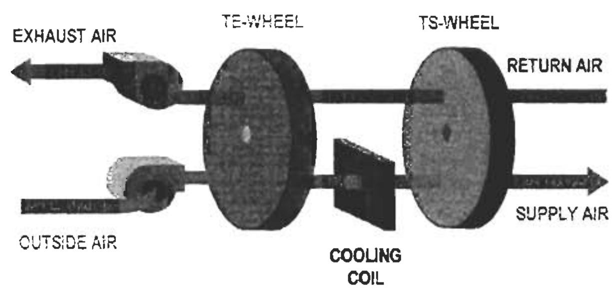


Figure 3: Total/Sensible Recovery Dual Wheel System

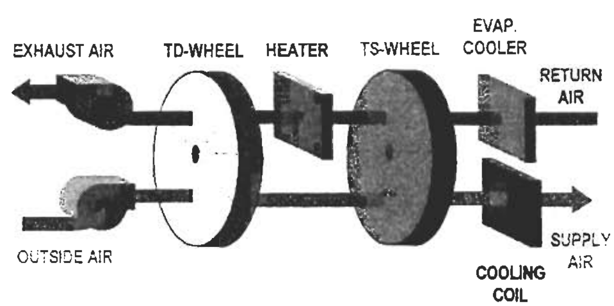


Figure 4: Active Desiccant Based Cooling System

PDH Enhances Performance and Energy Efficiency while Reducing Project First Cost

The novel Total Recovery/Passive Dehumidification System (PDH) described by this paper, allows outdoor air streams to be dehumidified to very low dewpoints, to levels unattainable with either the conventional cooling or the desiccant based systems described previously. This enhanced dehumidification capacity is achieved without the heated regeneration source required by "active" desiccant based dehumidification systems. The PDH approach minimizes cooling requirements while delivering a continuous supply of outdoor air to buildings while simultaneously controlling indoor humidity levels as recommended by ASHRAE, even at part load conditions. Data presented confirms the PDH system to be by far the most energy efficient way to handle the increased outdoor air latent loads placed on building HVAC systems.

How It Works:

A typical PDH system comprises a supply air fan, an exhaust air fan, a total energy recovery wheel, a "passive" dehumidification wheel and a cooling coil positioned in the supply air stream between the total energy recovery wheel and the passive dehumidification wheel.

In the cooling mode, (Figure 5) the supply air stream is cooled and dehumidified by passing it through a dry and cool zone of the total energy recovery wheel which has been rotated through and reached near equilibrium with the relatively cool, dry exhaust air stream leaving the "passive" dehumidification wheel. The air stream is then further cooled and dehumidified by passing it through the cooling coil. Before it is supplied to the controlled space, the supply air is moderately reheated and further dehumidified by passing it through a warm and dry zone of the passive dehumidification wheel, which has been rotated through and reached near equilibrium with the warm, dry exhaust air stream leaving the conditioned space.

The "Passive" Dehumidification Wheel:

The key to providing the exceptional dehumidification capability provided by the PDH system was the development of a new class of product, the passive dehumidification wheel. This unique product utilizes a desiccant composite optimized to remove moisture from a saturated air stream (condition leaving a cooling coil) using only the driving force provided by return air stream relative humidity. Unlike the "active" DBC system, neither a high temperature regeneration source nor an evaporative cooling section is necessary.

This system can therefore provide very dry outdoor air in an extremely energy efficient manner since it uses the conventional cooling technology in its most efficient performance envelope, uses the total energy recovery wheel in its most efficient envelope and combines the two with the "passive" desiccant dehumidification wheel operating in its most efficient envelope. The result is minimal cooling energy input and maximum latent cooling output.

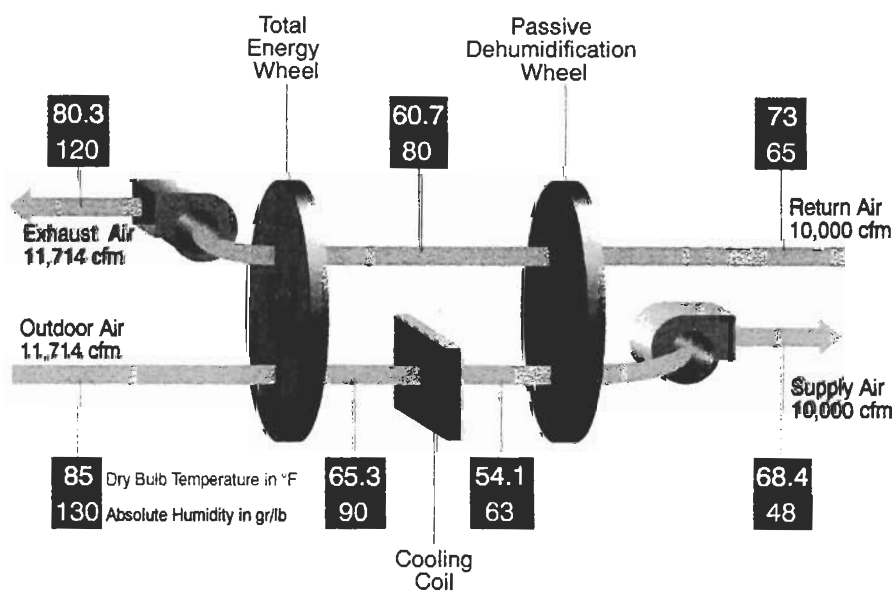


Figure 5: Typical Dehumidification Mode Operation for the Total Recovery/Passive Dehumidification System (PDH)

Figure 6: PDH system controlled to optimize sensible cooling output by maximizing the cooling input and minimizing the reheat provided by the passive dehumidification wheel

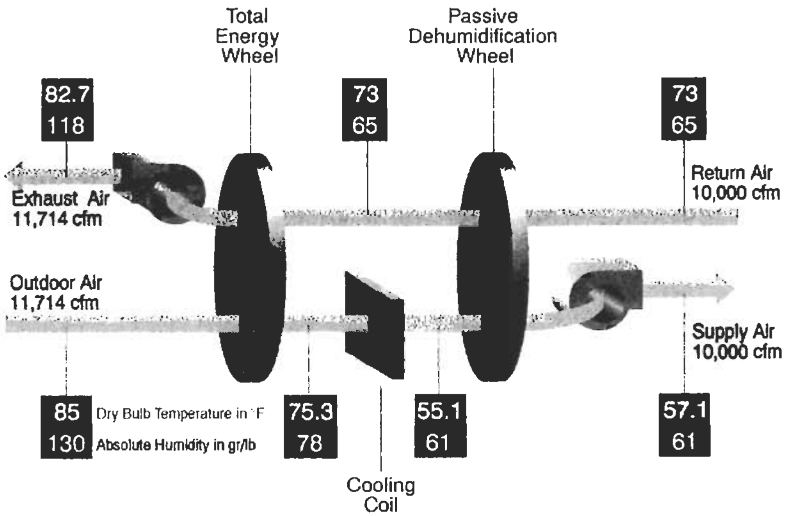


Figure 7: PDH system controlled to optimize both the dehumidification and sensible cooling provided by maximizing the cooling output and optimizing the speed of the passive dehumidification wheel

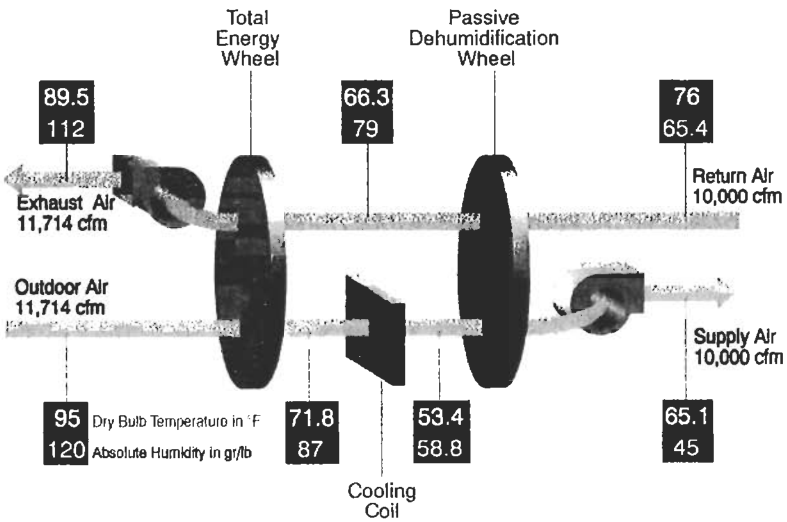


Figure 8: PDH system controlled to provide both the dehumidification and reheat required at a typical part load condition by minimizing the cooling input and optimizing the speed of the passive dehumidification wheel

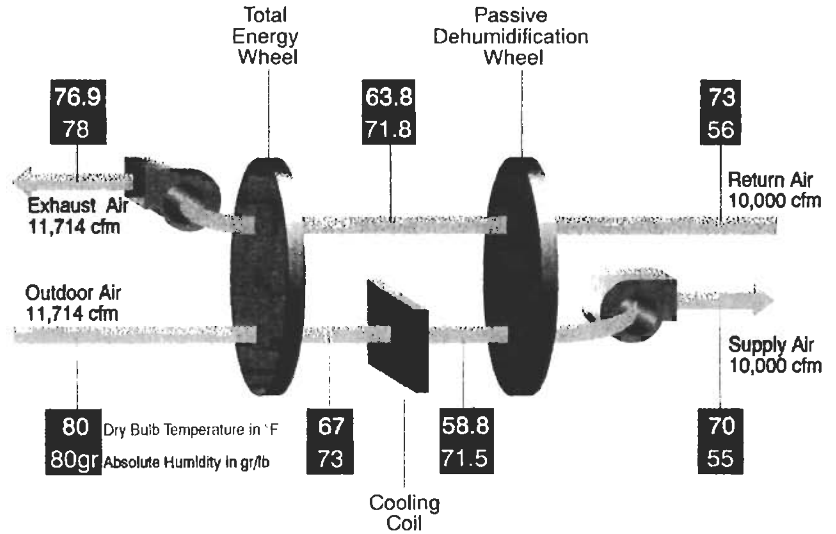
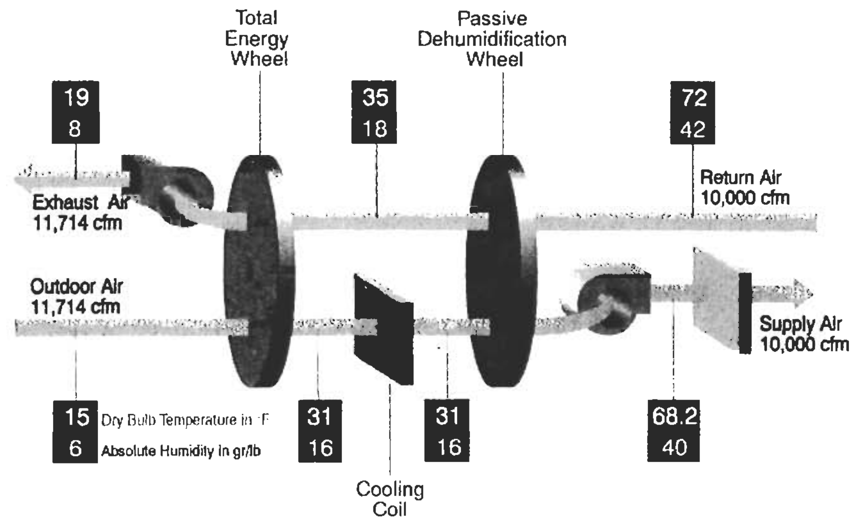


Figure 9: PDH system controlled to optimize both temperature and humidity recovery during the heating season by maximizing the speed of the passive dehumidification wheel



Unique Capacity Control Matches PDH Output with Space Load Requirements

A significant performance advantage results from the unique control logic used by the PDH system. By controlling the rotational speed of the passive dehumidification wheel, the amount of dehumidification capacity and the amount of reheat energy available can both be optimized to match the load associated with the varying temperature and humidity conditions that exist within the occupied space. Figure 10 shows an example of the performance relationship between wheel speed, dehumidification capacity and reheat capacity for an effective passive dehumidification wheel.

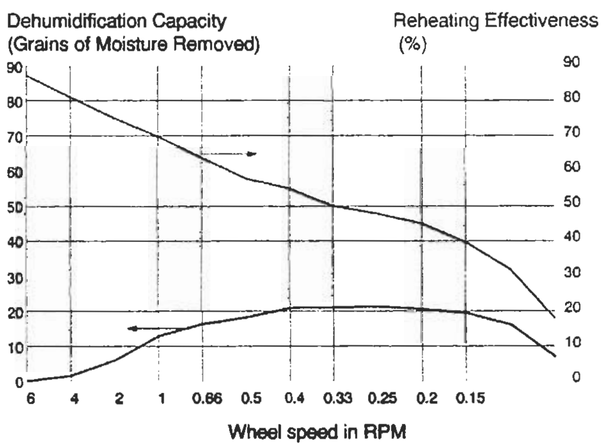


Figure 10: Unique performance of passive DH wheel

Figures 5 through 8 respectively show how the PDH system would be controlled during the cooling season to respond to each of the following conditions:

1) occupied space is too cool and too humid.

2) occupied space is too hot, humidity acceptable

3) occupied space is too hot and too humid

4) occupied space is too cool, humidity acceptable

Another control advantage offered by the PDH system, and shown by Figures 5 through 8, is that the input to the cooling coil is modulated to minimize the energy required to deliver the desired dewpoint. In all cases, the dewpoint leaving the cooling coil is significantly higher than would have been required by a conventional preconditioning approach. In many cases, the dewpoint delivered from the PDH system using a moderate leaving coil temperature would not be obtainable with a conventional cooling approach.

Figure 9 shows the advantage offered by the PDH system during the heating season. By optimizing the temperature and latent recovery of both desiccant wheels, total recovery in excess of 90% is common allowing many spaces to be self-heating despite the high percentage of outdoor air being provided.

Summary of Benefits Provided by the PDH System

The three most significant advantages offered by the PDH system when compared with all other options, are that (1) the dehumidification or latent capacity (e.g., dryness of the air provided to the controlled space) is significantly increased, (2) the energy efficiency is greatly improved and (3) the life cycle cost is greatly reduced. A detailed comparison of life cycle cost, energy consumption, dehumidification capacity and other performance criteria are summarized in Figures 12, 13 and 14, highlighting benefits of the PDH system.

The total energy recovery, DWS and other conventional over-cooling reheat systems discussed previously are limited by the dew point of the air leaving the cooling coil (e.g. how much the air can be cooled). Since even the best commercial cooling systems have a practical limit of approximately 50°F to 52°F for a cooling coil leaving air temperature, the absolute humidity level obtainable from such systems is limited to 52 to 55 grains (.0074 to .0079 lb. moisture/lb. of dry air).

The absolute humidity content of the supply air provided to the controlled space by an active DBC system processing all outdoor air on a latent design day (85°F and 130 grains) is limited to approximately 60 grains with the best technology currently available. To reach this condition requires the equipment to be operated at very low face velocities (resulting in very large system space requirements) and regenerated at very high regeneration temperatures.

As a result, the only commercially available way to dehumidify outdoor air below approximately 52 grains of moisture involves cooling the outdoor air below approximately 50°F, and requires expensive, non-standard cooling equipment with very deep cooling coils, complex controls with defrost cycles and significantly elevated KW/ton energy consumption (i.e. poor energy efficiency).

The PDH system can easily provide outdoor air at a humidity content of 40 grains using standard cooling equipment. This results in a 90 grain reduction at the typical latent design condition of 130 grains, and can be designed and operated to provide air as dry as 35 grains in many cases.

Providing very dry air offers many advantages including reduced energy consumption and allowing lower airflow quantities to be used to satisfy a given latent load. Numerous other system wide benefits are recognized if the total latent load can be handled by the PDH system.

For example, an office building could reduce energy consumption by operating its VAV air handling systems serving the space with dry cooling coils allowing the supply air leaving temperature to be set by the controlled space sensible loads and not that required to dehumidify the air. Operating the main system VAV air handling units with dry coils significantly reduces fan horsepower due to the reduction in static pressure loss across the cooling coils. First cost savings are recognized by the application of smaller chiller water piping, fittings and insulation. In some cases, the higher coil face velocities possible with dry coils will reduce the cost of the main air handling units. As importantly, the improved space humidity control often allows for the space temperature setpoint to be raised while maintaining comfort conditions.

These savings are possible if the outdoor air volume provided to the VAV air handling system is dehumidified enough to handle both the outdoor air and space latent loads. Because the percentage of outdoor air compared to the total supply air volume of a typical office designed to comply with ASHRAE Standard 62 may only be 15-20%, the outdoor air would need to be very dry if all the internal latent load is to be handled by over-drying the outdoor air volume (dry cooling coils).

The PDH system is also particularly well suited for school classrooms, especially where engineers attempt to reduce project first cost by designing for only 7.5 cfm/student in lieu of the recommended 15 cfm/student or when over-crowding or poor filter maintenance reduce the quantity of outdoor air per student.

For example, a typical school classroom contains approximately 30 children. The sensible load associated with the lights, occupants, etc is approximately 2 tons. The latent load associated with the occupants and infiltration is approximately 4.3 pounds per hour. Assuming the space is to be controlled at 75 degrees and 50% RH, a humidity content of 65 grains is desired. If the latent load is to be handled with an outdoor air volume of 450 cfm (based on 15 cfm/student) then the outdoor air must be dehumidified to 50 grains.

Now, if only 7.5 cfm/student is applied in lieu of the 15 cfm considered previously, then the required moisture differential doubles from the 15 grains previously calculated to 30 grains. As a result, to handle the latent load with 225 cfm of outdoor (7.5 cfm/student) the outdoor air must now be delivered at 35 grains (65 grains - 30 grain moisture differential). Because this level of dehumidification is generally not obtainable with conventional HVAC equipment, this advantage of the PDH system is apparent.

The ability of the PDH system to provide very dry outdoor air is also particularly advantageous where extreme indoor air humidity loads are encountered. For example, at least two times a day the doors to a school facility may be kept open for extended periods of time. The reserve capacity offered by the PDH system is particularly important in the morning hours because the outdoor air infiltrating the building is typically cool and humid. As a result, little dehumidification is provided by the conventional HVAC systems controlled off of space temperature during these part load conditions.

Benefits Recognized During Unoccupied Periods:

Another advantage offered by the PDH system is that it has the ability to control humidity levels in unoccupied facilities, not effectively accomplished with other preconditioning or conventional HVAC systems.

During unoccupied times, research has shown that the building materials (e.g., carpeting, furnishings, etc.) act as a humidity storage site as the humidity level rises. This rise in humidity is typical because many building operators cycle off the HVAC system in an attempt to conserve energy. Because the internal sensible load in an unoccupied building is minimal, controlling humidity often requires reheating the air leaving the cooling coil. Since this reheat capability is seldom integrated into the building design, the humidity level in most unoccupied buildings is not controlled.

The PDH system can be operated to either "dry out" the humidity stored in the building materials or to provide the necessary humidity control during unoccupied periods. Since the necessary reheat and much of the dehumidification capacity is provided by the passive dehumidification wheel, and since the system can be cycled to operate only when dehumidification is needed, controlling the humidity in unoccupied facilities is both practical and energy efficient. The "unoccupied mode" of the PDH system is shown in figure 11.

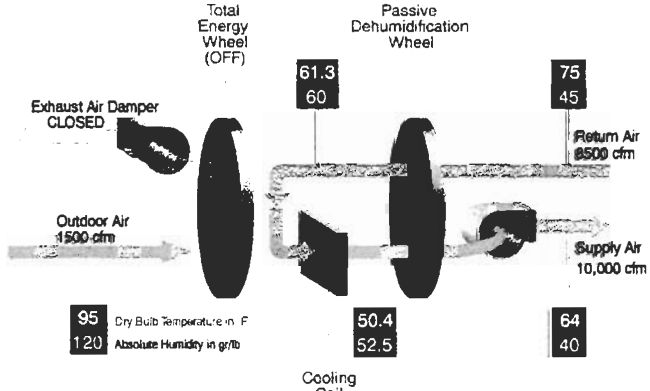


Figure 11: PDH unoccupied mod

Comparative Example 1: PDH System vs. Other Outdoor Air Preconditioning Systems

To quantify the benefits offered by the PDH system, an analysis compares key performance parameters including cooling capacity input requirement, energy consumption, equipment first cost, life cycle cost, and dehumidification capacity for a typical application.

Figures 12 through 14 show the results of a comparison made for a project involving 10,000 cfm of outdoor air provided to a facility located in Atlanta, Georgia. Assumptions made for the energy estimates include 24 hour per day operation, an electric cost of \$.06/KWH and a gas cost of \$4.50/million BTU. It is assumed that outdoor air will be conditioned to 68°F and 55 grains (51°F dewpoint) during the cooling season and 70°F and 38 grains during the heating season. The space is maintained at 75°F and 50% RH and 72°F and 35% RH during the cooling and heating seasons.

Energy Consumption:

As shown in Figure 12, the PDH system has by far the lowest estimated operating cost, using only 27% of the energy required by the packaged rooftop gas reheat baseline system.

Conventional Cooling Input Required:

Figure 12 also shows the significant reduction in required cooling input associated with the PDH system, which needed only 24 ton (vs. 76 tons for the baseline system) to produce 42.5 tons of latent cooling, with no reheat energy needed.

Equipment Cost and Life Cycle Cost:

When comparing total equipment cost; (i.e. air handling system, chiller and boiler cost) the PDH system was within 10% of the lowest cost alternative. When comparing the life cycle cost, the PDH is the clear winner, costing an estimated \$490,970 less than the lowest first cost alternative over a 20 period (see Figure 13).

Dehumidification Capacity Delivered:

Figure 14 shows the benefit of providing dehumidified outdoor air at a much lower dewpoint that possible with current preconditioning options. Note that the PDH operated at 10,000 cfm and delivering air at 40 grains produces 51 tons of overall latent capacity and 14 tons of room latent capacity, 120% and 230% of that provided by the baseline system respectively.

Equipment Options	Dewpoint Delivered	Latent Capacity Delivered	Conventional Cooling Capacity Required	Reheat Required	Energy % Baseline
Packaged Rooftop Gas Reheat	51 °F	42.5 Tons	75.8 Tons	166,100 btu/hr	100%
Chilled Water/Hot Water Reheat	51 °F	42.5 Tons	75.8 Tons	166,100 btu/hr	100%
Customized Package DX with Condensor Reheat	51 °F	42.5 Tons	75.8 Tons	0 btu/hr	85%
Active Desiccant Based Cooling	55 °F	36.8 Tons	18.9 Tons (1)	489,000 (2)	70%
Total Energy Recovery, Chilled Water with Hot Water Reheat	51 °F	42.5 Tons	37.5 Tons	166,100 btu/hr	57%
Dual Wheel Total Energy Recovery	51 °F	42.5 Tons	29.5 Tons	0 btu/hr	33%
PDH Passive Dehumidificaiton System	51 °F	42.5 Tons	24 Tons	0 btu/hr	27%

Figure 12: Comparing cooling input and energy consumption at a common delivered dewpoint (not obtained by DBC), (1) DBC post cooling tons (2) DBC regeneration energy

Equipment Options	Latent Capacity Delivered	Equipment (1) Cost	Annual Cost (2) of Operation	Life Cycle Cost	Simple Payback
Packaged Rooftop Gas Reheat	42.5 Tons	\$56,500	\$24,045	\$610,020	-
Chilled Water/Hot Water Reheat	42.5 Tons	\$63,000	\$20,130	\$514,960	1.1 yrs.
Customized Package DX with Condensor Reheat	42.5 Tons	\$66,000	\$21,920	\$556,370	2.6 yrs
Active Desiccant Based Cooling	36.8 Tons	\$87,000	\$12,240	\$332,250	2.4 yrs
Total Energy Recovery, Chilled Water with Hot Water Reheat	42.5 Tons	\$60,500	\$11,290	\$300,490	.30 yrs
Dual Wheel Total Energy Recovery	42.5 Tons	\$61,500	\$4,465	\$146,330	.20 yrs
PDH Passive Dehumidificaiton System	42.5 Tons	\$65,300	\$3,420	\$119,050	.40 yrs

Figure 13: (1) 1999 market price (2) SEMCO analysis software using DOE TMY2 weather data

Equipment Options	Lowest Practicle Dewpoint Delivered	Humidity Level Delivered	Latent Capacity (1) (Outdoor Load)	Latent Capacity (2) (Room Load)
Packaged Rooftop Gas Reheat	51 °F	55 grains	43 tons	6 tons
Chilled Water/Hot Water Reheat	49 °F	52 grains	44 Tons	7 tons
Customized Package DX with Condensor Reheat	50 °F	54 grains	43 tons	6 tons
Active Desiccant Based Cooling	55 °F	65 grains	37 tons	0 tons
Total Energy Recovery, Chilled Water with Hot Water Reheat	49 °F	52 grains	44 tons	7 tons
Dual Wheel Total Energy Recovery	49 °F	52 grains	44 tons	7 tons
PDH Passive Dehumidificaiton System	42 °F	40 grains	51 tons	14 tons

Figure 14: (1) Latent capacity based on outdoor air (2) Capacity available for room latent load

Case History: University Library in Augusta, Ga.

The PDH system approach was chosen to resolve numerous indoor air quality and humidity control problems experienced by the occupants of a multi-story library facility located in Augusta, Georgia.

The library facility was initially designed with a conventional variable air volume (VAV) system served by a 200 ton chiller. Outdoor air was supplied to the facility in accordance with ASHRAE Standard 62. After adjustment with the appropriate "Z" factor to account for the "fresh air" distribution efficiency of the VAV system, the outdoor air quantity amounted to approximately 20,000 cfm.

The initial HVAC system design called for the outdoor air to be mixed with the return air stream, then cooled to approximately 56°F by a bank of 10 row cooling coils, then delivered to the occupied space in the amount necessary to maintain the setpoint temperature of 75°F. At peak design conditions, this resulted in a total system airflow of approximately 80,000 cfm.

Performance Problems with the Initial HVAC Design:

Numerous comfort and performance problems resulted from the initial project design. The first problem was that the design conditions used for calculating the outdoor air load were based on inappropriate dry bulb with corresponding wet bulb data (90° db/100 grains) in lieu of the more appropriate dewpoint design data now available (87° db/130 grains). This resulted in a leaving coil condition of only 58°F, and a dewpoint too high to satisfy the space humidity requirements (see Figure 17).

Enough 58°F air could be delivered to handle the space sensible load, but the humidity level within the space remained high which effected comfort conditions and resulted in damage to books and other documents housed at the facility.

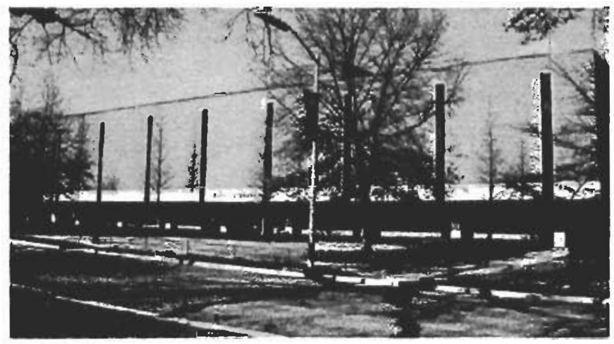


Photo of Augusta Library

At more typical part load conditions, the problems were compounded since the percentage of outdoor air increased as the reduced space sensible load allowed the supply air quantity to decrease. The available chiller capacity was able to drive the supply air temperature down to 55°F at the reduced airflow quantity, but the space latent load still was not satisfied despite the effort to dry out the space by controlling off of space humidity. The net result was that the space was often both too cool and too humid (see Figures 15 and 17).

Given the repeated complaints regarding high humidity, cool temperatures and damage to the books and other important documents, a decision was made to identify a modification to the original HVAC system that would eliminate problems experienced at the facility.

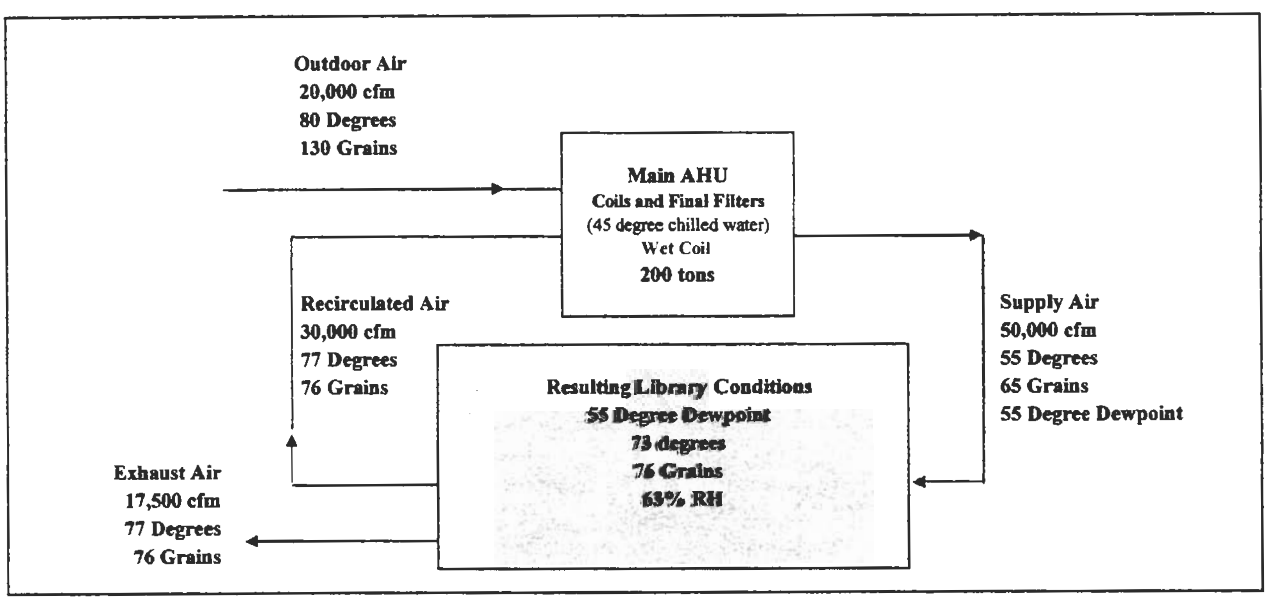


Figure 15: Typical part load performance for the original VAV system installed at the library facility in Augusta Georgia

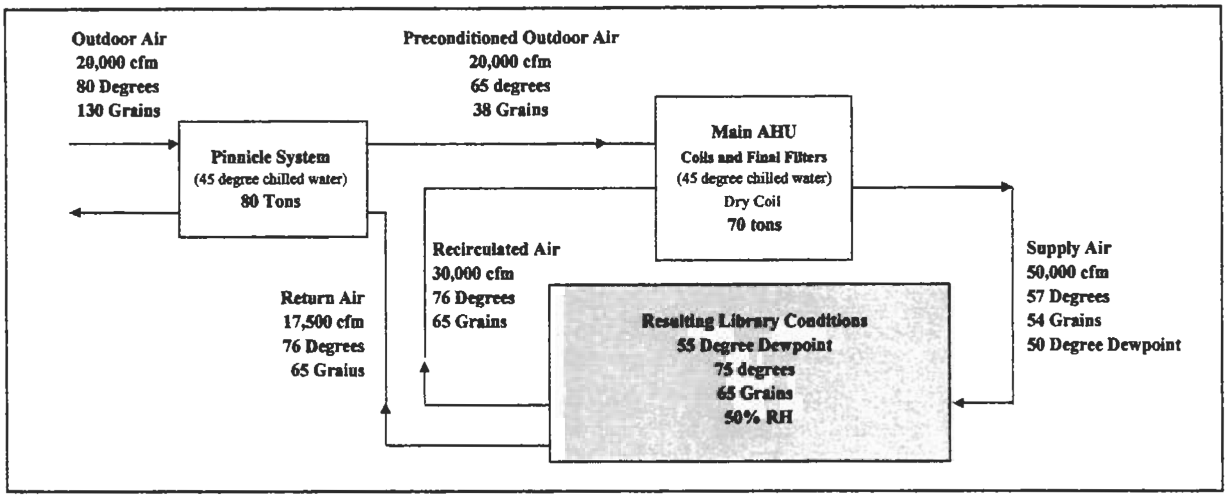


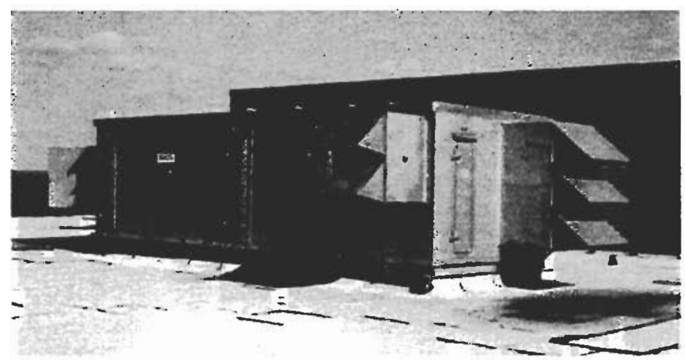
Figure 16: Typical part load performance for the original VAV system following the retrofit of the PDH system

A PDH System Retrofit Proves and Effective Solution:

A careful evaluation of the problems that existed at the Augusta library facility identified the need for a system to efficiently dehumidify and pre-cool the outdoor air being supplied to the existing VAV system. Following a review of the available options, it was determined that a PDH approach would provided numerous operating advantages in addition to resolving the identified problems.

As shown by Figure 16, the ability to operate the PDH system to provide the very dry, 38 grain air allowed all of the latent load to be handled by over-drying the outdoor air stream. This enabled the existing chiller capacity to maintain the desired space conditions at both peak and part load conditions (see Figures 16 and 17). This approach also allows the existing cooling coils to operate dry under most conditions, which significantly reduces the required fan horsepower.

As shown in Figure 16, only 150 tons of chiller capacity is now used at part load conditions where previously 200 tons was not adequate. Another benefit is that the leaving coil temperature can be optimized to control space temperature since all dehumidification is now accomplished with the outdoor air volume.



In addition to providing the necessary temperature and humidity control, the PDH system proved to be a very energy efficient and cost effective solution. Figure 18 summarizes the chiller and boiler capacity that would have been added to this facility in order to approximate the desired space conditions. It was determined that the cost of adding of the additional 100 tons of chiller capacity, cooling tower and 40 boiler horsepower required would have exceeded the cost of the PDH retrofit, so there was no first cost premium.

As importantly, the estimated $28,000 in annual energy savings offered by the PDH approach resulted in a very attractive return on investment (ROI), which helped to support the decision to address the IAQ problems.

Peak Sensible Conditions	System Airflow	Outdoor Air %	Cooling Tons	Delivered Supply Air Conditions			Resulting Space Conditions			Comments
				Temp.	Humidity	Dewpoint	Temp.	Humidity	Dewpoint	
Initial VAV	80000	25%	200	58°	70 grains	57°	75°	77 grains	60°	Too Humid
Initial VAV + PDH	80000	25%	200	56°	58 grains	52°	75°	65 grains	55°	As Desired

Part Load Sensible	System Airflow	Outdoor Air %	Cooling Tons	Delivered Supply Air Conditions			Resulting Space Conditions			Comments
				Temp.	Humidity	Dewpoint	Temp.	Humidity	Dewpoint	
Initial VAV	50000	40%	200	55°	65 grains	55°	73°	76 grains	60°	Too Cool & Humid
Initial VAV + PDH	50000	40%	150	57°	54 grains	50°	75°	65 grains	55°	As Desired

Figure 17: Comparing system performance of the initial VAV design and the same VAV system after the PDH retrofit

Figure 18: Comparison of chiller capacity, boiler capacity and estimated operating cost of both the Augusta Library PDH system and the baseline approach, initially considered, which would have increased the size of the physical plant in an attempt to maintain the desired space conditions.

PDH vs Conventional Alternative	Required Chiller Capacity	Required Boiler Capacity	Cost of Conditioning Outdoor Air
Modified Conventional VAV	300 Tons	85 BHP	$37,414/yr.
Existing VAV + PDH System	200 Tons	45 BHP	$9,188/yr
Savings Due to PDH Approach	100 Tons	40 BHP	$28,226/yr.

Conclusion

A new class of equipment, referred to as a "passive" desiccant based system (PDH), is now available to precondition the outdoor air volume delivered to a wide variety of facility types. The performance advantages offered by the novel technology mitigates the energy consumption, humidity control and life cycle cost concerns often associated with accommodating the requirements of ASHRAE Standard 62.

A detailed analysis has shown the equipment to be significantly more energy-efficient than all known alternatives, and that it has the unique ability to dehumidify outdoor air streams to very low dewpoints, unattainable with conventional cooling approaches. The system allows for precise control of the indoor space humidity while providing high quantities of outdoor air, at both peak and part load conditions, and during both occupied and unoccupied modes. Low operating cost, reasonable first cost and the significant reduction in cooling capacity requirements provide a life cycle cost that is substantially less than that of conventional system approaches.

The PDH system offers engineers the flexibility to consider whole new design schemes as a result of its ability to deliver outdoor air as dry as 35 grains (39°F dewpoint), even in hot and humid climates. Systems can be designed to handle the total latent load with the dehumidified outdoor air volume, allowing for dry coils in the conventional HVAC units, down-sized to handle only the space sensible load. First cost savings can also be recognized due to reduced chiller capacity, cooling tower, boiler capacity, pipe size and fittings. The system integrates equally well with HVAC systems using packaged equipment or chilled water systems.

A detailed analysis of the available preconditioning options concluded that the PDH system has the best life cycle cost, lowest energy consumption, requires the least amount of cooling input, provides the most latent capacity, provides the driest air and has a first cost similar to that of the least expensive alternative.

Based on the 10,000 cfm comparison completed, the PDH system equipment first cost was only $8,800 more than the lowest cost option, but the life cycle cost over a 20 year period was estimated to be $490,970 less. This investigation highlights both the many benefits offered by the technology and the importance of making design decisions based on life cycle cost, not first cost, especially when evaluating outdoor air systems.

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