COMPUTER ROOM FRESH AIR COOLING

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ABSTRACT

This paper discusses the concept of a computer room fresh air cooling system with evaporative humidification. The system offers significantly lower energy consumption than conventional cooling units, with 24% reduction for Dallas and 56% reduction for Denver. A control scheme is suggested that should satisfy the strict temperature and humidity specifications of computer rooms. The controls also allow flexibility to meet the wide range of dehumidification loads that can occur in a room with an imperfect vapor barrier. The project is presently in the conceptual stage, but is being considered for installation in a Texas Instruments' building if economic feasibility can positively be demonstrated.

INTRODUCTION

Because of strict temperature and humidity control specifications and constant load profiles, computer rooms are often ignored as potential sites for building energy use reduction. These load characteristics, along with high load densities in comparison to office space, can make computer rooms prime energy users that should not be ignored. The strict temperature and humidity specifications are often met through the use of expensive steam humidification and reheat. In addition, the constant nature of the cooling load may dictate part load operation of a large, central building chiller system during periods of the year when the equipment could otherwise be turned off. Alternatively, computer rooms may be conditioned by expensive dedicated small chillers or direct expansion systems.

In an excellent overview of computer room conditioning alternatives, H.P. Becker suggests that because of the cycling nature of their controls, direct expansion (DX) systems may not be able to meet the strict humidity specifications for underfloor delivery at all times (1). Becker also points out that water vapor migration due to the inadequate design or construction of vapor barriers can often lead to very high summer dehumidification and winter humidification loads. These loads can be much greater than packaged computer room conditioning units are designed to meet.

There is a need to consider alternative computer room conditioning schemes to better meet the requirements of low energy consumption and tight temperature and humidity control over varying dehumidification loads. The system must also have an acceptable payback and must not compromise reliability.

POSSIBILITIES FOR REDUCING ENERGY CONSUMPTION

To offer reduced energy consumption during periods of low ambient temperature, computer room air handlers can be fitted with economizer coils to precool the air using fluid from a sensible, or for lower fluid temperatures, an evaporative heat rejector (Figure 1). As the ambient temperature rises close to the required supply air temperature, operation of the economizer coil is limited by the evaporative heat rejector wet bulb approach temperature and the economizer coil effectiveness. Similar schemes can use cooling tower water through strainers or heat exchangers directly into the chilled water coil, but these systems cannot be used for precooling.

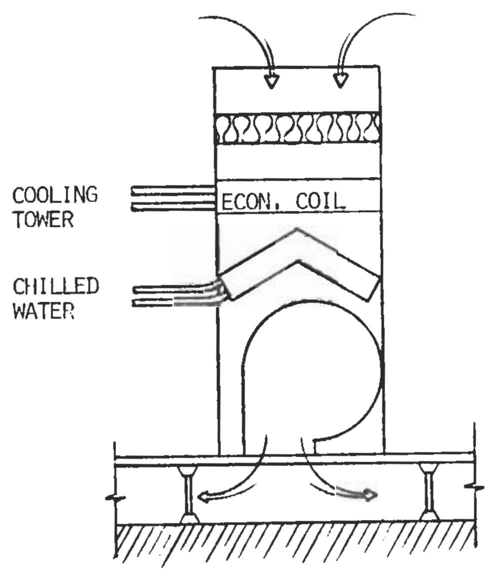

Fig. 1 Economizer Coil Cooling Scheme

The equipment required to implement an economizer coil precooling scheme will include a heat rejector, coil, piping, valves, pump and controls. The additional pump and fan use a small amount of energy. The economizer coil may increase the requirement for humidification and reheat, but some reheat schemes can be set up to use waste heat. Questions of control accuracy for DX systems are still relevant.

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A direct fresh air cooling scheme with evaporative humidification (Figure 2) is not limited by the same heat transfer considerations as the economizer coil and hence can meet the total cooling load for many more hours per year. The carefully engineered control system that is required to control the evaporative humidification process may be able to provide enough flexibility to eliminate the need for reheat and steam or electric powered humidification. A fresh air cooling system will require ductwork, dampers, increased fan capacity, humidification equipment, air filters and controls. The equipment and installation costs for any economizer system must be evaluated for each specific site.

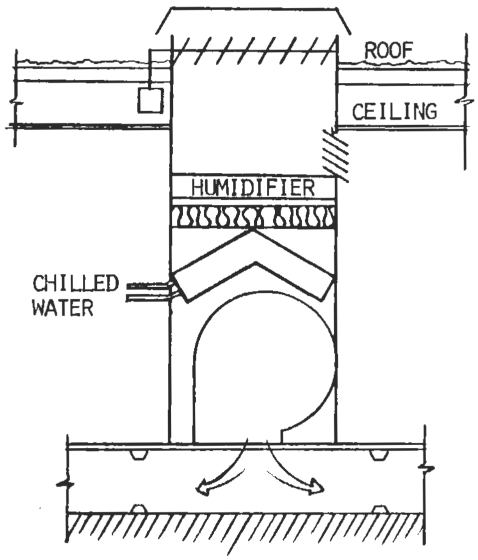

Fig. 2 Fresh Air Cooling Scheme

Table 1 lists energy consumption for 3 computer room conditioning systems in 4 cities that have climates that might be described as hot and humid, hot and moderately humid, cool and dry, and cool and moderately humid. The conditioning systems being compared are 1) standard chilled water coil, 2) chilled water coil with economizer coil, and 3) chilled water coil with fresh air cooling. The economizer coil is cooled by an evaporative heat rejector with an assumed wet bulb approach temperature of 10 degrees F. The computer room is 1500 ft^2 with a room cooling load (not including air conditioning equipment) of 20 tons. The room setpoint is assumed to be 72°F, 50% RH. Other specific assumptions are stated below Table 1. Operation without chilled water can take place for ambient wet bulb temperatures of less than 55°F for the fresh air scheme and less than 35°F for the economizer coil scheme. The analysis uses bin mean coincident wet bulb hours as summarized in Table 2. Calculation methods are presented in Table 3. The analysis neglects energy consumption for humidification and reheat for the standard coil and the economizer coil. Inclusion of this quantity could raise the total energy consumption by 10% or more for these systems, bearing in mind that humidification and reheat use are site specific and affected by climate, vapor barrier condition, equipment condition and the general nature of the load. The fan and pump power required to move system fluids (air or water) for free cooling is system specific, although these effects are secondary energy consumption considerations. The energy consumption quantities should be considered as reasonable estimates for comparative purposes and more detailed analyses of the fan and pump power should be performed for calculating actual system payback.

Table 1

Energy Use Summary

SYSTEM	HOUSTON	DALLAS	DENVER	NEW YORK
CHILLED WATER				
TOTAL KWh	176,000	176,000	176,000	176,000
KWh/ft^2	117.3	117.3	117.3	117.3
ECONOMIZER COIL				
TOTAL KWh	161,700	150,200	104,000	123,500
KWh/ft^2	107.8	100.1	69.9	82.3
% SAVINGS	8	15	40	30
FRESH AIR				
TOTAL KWh	157,300	134,200	76,900	108.400
KWh/ft^2	104.9	89.5	51.3	72.2
% SAVINGS	11	24	56	38

ASSUMPTIONS:

1) COMPUTER ROOM AREA = 1500 ft^2
2) COOLING LOAD (WITHOUT AIR-CONDITIONING EQUIPMENT) = 20 TONS
3) FAN POWER = 6KW STANDARD, 8KW FRESH AIR, 8/6 KW ECONOMIZER
4) SETPOINT = 72°F, 50% RH
5) HEAT REJECTOR APPROACH TO WET BULB = 10°F
6) SUPPLY AIR TEMPERATURE = 60°F
7) ECONOMIZER COIL EFFECTIVENESS = 50%
8) CHILLER CONSUMPTION = 0.65 KW/TON
9) HUMIDIFICATION AND REHEAT ENERGY NEGLECTED
10) SMALL FRACTION OUTSIDE AIR FOR CLOSED SYSTEMS

The standard chilled water coil is estimated to require 117 KWH/ ft^2 per year to operate. In Houston, the economizer coil reduces this consumption by 8% and the fresh air system by 11%. In Dallas, the reduction for the economizer and fresh air systems are 15% and 24% respectively. The similar reductions for Denver are 40% and 56% and those for New York are 30% and 38% respectively. It can easily be seen the the cooler and drier the climate, the better either free cooling scheme is. The fresh air system is superior to the economizer coil in energy savings in every case, and again it should be noted that inclusion of humidification and reheat will increase that advantage. The consumption figures

Table 2

Wet Bulb Temperatures, Hours & KWh

RANGE °F WB	HOUSTON (°F) HRS	HOUSTON KWh	DALLAS (°F) HRS	DALLAS KWh	DENVER (°F) HRS	DENVER KWh	NEW YORK (°F) HRS	NEW YORK KWh
ECONOMIZER								
≤ 35	307	2,456	802	6,416	3,691	29,528	2,567	20,536
35-36	(39)321		(38)516		(39)729		(39)829	
	(43)460		(42)582		(43)739		(43)771	
	(48)580		(46)651		(47)776		(48)751	
	(53)668		(50)645		(51)794		(53)780	
	(58)782		(55)727		(53)618		(58)827	
	(63)964	65,244	(60)805		(55)494		(62)865	75,443
			(65)966	82,159	(56)384			
					(58)291			
					(59)174			
					(60) 71	75,144		
> 65	4,678	93,990	3,056	61,596	7	141	1,370	27,523
		166,690		150,171		104,813		123,502
FRESH AIR								
≤ 55	2,336	18,688	3,923	31,384	7,841	62,728	5,698	45,584
55-60	782	11,908	805	12,259	920	14,010	827	12,594
> 60	5,642	126,697	4,032	90,543	7	157	2,235	108,367
		157,293		134,186		76,895		108,367

Table 3

Energy Consumption Calculations

Chilled Water Coil

$$\left[KW_{fan}+(KW_{fan}*0.28\frac{ton}{KW}+20ton)*0.65\frac{KW}{ton}\right]*hrs=KWh$$

Fresh Air Cooling With Evaporative Humidification

Fresh Air Mode ($T_{WB}\leq 55^{o}F$)

$$(KW_{fan})*hrs=KWh$$

Mixed Mode ($55^{o}F<T_{WB}\leq 60^{o}F$)

$$\left[KW_{fan}+0.5*(KW_{fan}*0.28\frac{ton}{KW}+20ton)*0.65\frac{KW}{ton}\right]*hrs=KWh$$

Economizer Coil

Economizer Mode ($T_{WB}\leq 35^{o}F$)

$$(KW_{fan,pump})*hrs=KWh$$

Precooler Mode ($35^{o}F<T_{WB}\leq 65^{o}F$)

$$Tp_i=T_R-\varepsilon*\left[T_R-(T_{WBi}+10)\right] \qquad f_i=1-\frac{T_R-Tp_i}{T_R-T_S}$$

$$\left[KW_{fan,pump}*\sum_{i=1}^{n} hrs_i+\sum_{i=1}^{n} f_i*hrs_i*(KW_{fan}*0.28\frac{ton}{KW}+20ton)*0.65\frac{KW}{ton}\right]=KWh$$

T_{WB}=wet bulb temperature
T_R =room temperature
T_S =supply air temperature
Tp_i=precooler exit temperature for T_{WB} bin_i
ε =economizer coil effectiveness

presented here do not imply that the fresh air scheme is the best system for all computer room cooling applications or that it will even provide adequate payback, but the magnitude of possible energy savings certainly suggests that fresh air cooling should be seriously considered.

SYSTEM DESIGN CRITERIA

Typical computer room temperature and humidity setpoints are 72 ± 2°F and 50 ± 5% RH. Computer equipment manufacturers usually specify maximum allowable supply air relative humidity of 80%. These specifications are set to provide equipment protection and enhance reliability by controlling low humidity static electricity and high humidity corrosion, electrical shorting and silver migration. Strict standards of air quality must also be maintained to prevent damage to delicate equipment by dust particles.

A great deal of expense is associated with computer room downtime for conditioning equipment or computer equipment failures. Any changes to the computer room conditioning status quo in the pursuit of energy conservation will not be tolerated if the environment is not controlled at least as well and if the present conditioning system reliability is not equalled or improved. In addition, the energy conserving conditioning system must provide an acceptable return on investment. To help meet these criteria, the improved computer room conditioning system should be reasonably simple, easy to retrofit or design

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in, easy to maintain and provide fail safe operation. Improvements can be made over some standard packaged conditioning systems by providing smoother control to positively meet temperature and humidity specifications 100 percent of the time. Increased humidification and dehumidification capacity would improve the control of many existing computer rooms with imperfect vapor barriers.

FRESH AIR COOLING SYSTEM

The suggested computer room fresh air cooling system with evaporative humidification is shown schematically in Figure 3 as it would interface with a typical packaged conditioning system or as a stand alone unit including a cooling coil. Return and outside air ducts with dampers must be provided, as well as ductwork to the humidification system and the cooling coil. Because of the large amounts of outside air that will be drawn into the room, dampers are also required to relieve room pressure to the outside or to the rest of the building. In a retrofit application, greater fan pressure is required to overcome the pressure drop through the added ductwork and humidification equipment. The evaporative humidifier can be either a fogging nozzle that is accurately modulated, or a continuously wetted humidification pad with face and bypass dampers for control, as suggested by Becker. The fogger would be chosen if an adequate supply of distilled or hyperfiltered water is available and the wetted pad would be chosen if tap water is to be used. This discussion will concentrate on wetted pad humidification since tap water use would be most common. Fogger control will be essentially similar, with fogger capacity modulated rather than the humidifier face damper. The blowdown that should be used with the pad is dependent on the mineral content of the available water. Care must be exercised in maintaining the pad to prevent bacterial growth. Temperature and humidity sensors are required at the locations shown in the figure. Outside air temperature and humidity, space temperature and humidity, mixed air temperature, duct temperature and supply temperature sensors are all required.

MODES OF OPERATION AND CONTROL

Control of temperature and humidity in an evaporative process requires careful attention to the process fundamentals to assure stable and accurate control. The fresh air cooling cycle has 3 modes of operation: fresh air, mixed and refrigeration. The fresh air mode is depicted on the psychrometric chart in Figure 4. Outside and return airstreams are mixed along path OR to point M. A fraction of that air is humidified along the wet bulb line to point H and then mixed with air from state M to reach state D. The air at state D passes through the inactive cooling coil without changing temperature or humidity to state S where it is delivered to the underfloor plenum. If the room humidity is off of setpoint, the face and bypass dampers at the humidifier (H dampers) are modulated to adjust it by shifting the mixing point up or down line HM. As the air temperature at S (T_S) changes because of the increase or decrease in evaporative cooling, the O & R dampers are modulated to quickly adjust T_S by shifting point M. As the room temperature T_R moves off of setpoint, the setpoint for T_S is adjusted to shift T_R back by modulating the O and R dampers.

If the O damper is fully open and T_R is too high, the cooling load cannot be completely met by fresh air and evaporative cooling, and the system must change over to the mixed mode as shown on the psychrometric chart of Figure 5. The mixed mode is controlled almost identically to the fresh air mode except that the O damper remains fully open and T_S is now controlled by modulating the chilled water valve. Using the respective

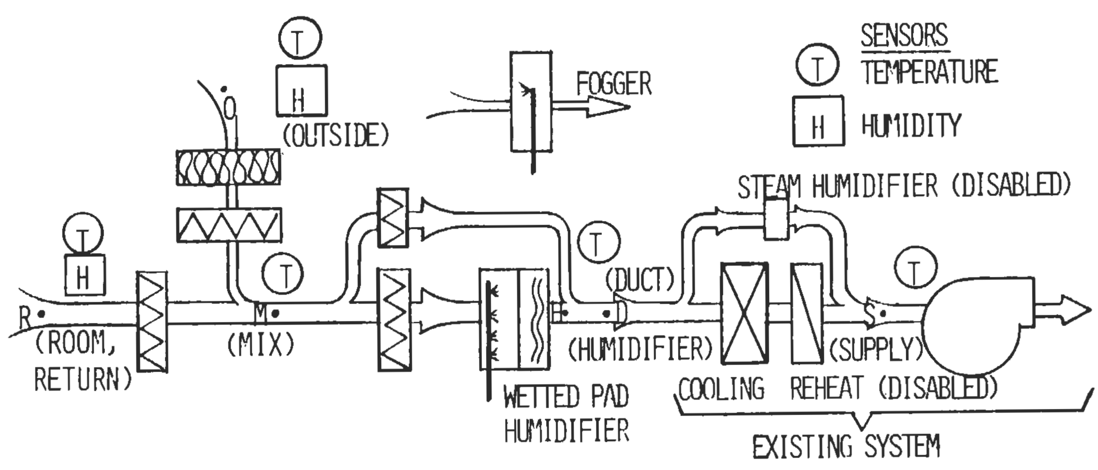

Fig. 3 Fresh Air Cooling System

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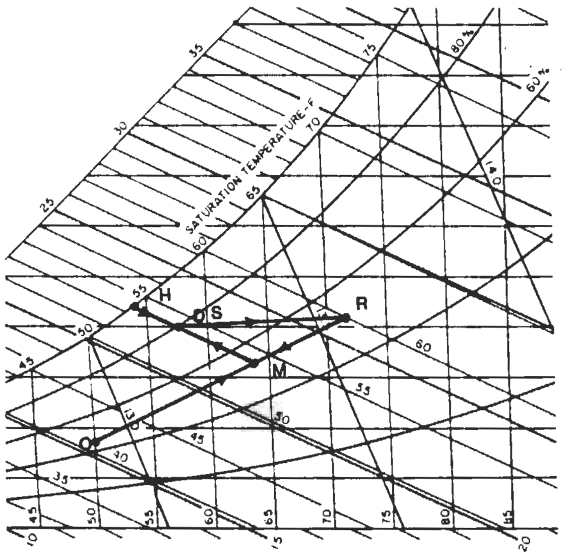

Fig. 4 Cooling Cycle - Fresh Air Mode

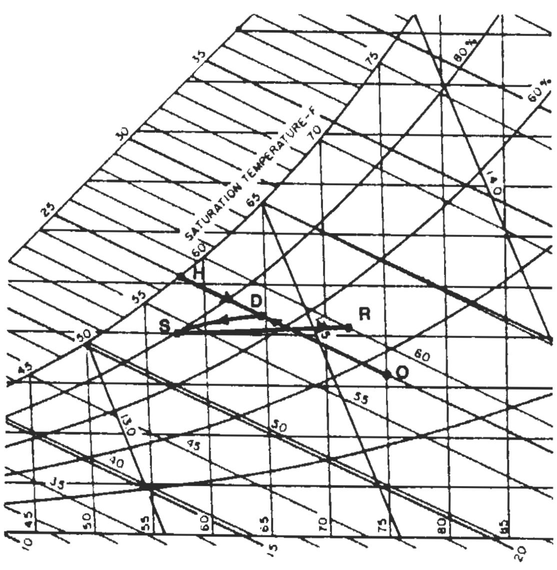

Fig. 5 Cooling Cycle - Mixed Mode

temperature and humidity readings, room air enthalpy and outside air enthalpy can be calculated and then compared to control changeover from the mixed mode to the refrigeration mode. When the outside enthalpy is greater than the room enthalpy, the outside damper is closed to the minimum position and the return damper is opened so that return air is conditioned by the cycle shown for the refrigeration mode in Figure 6. Humidification is again controlled by the face and bypass dampers and T_S is again controlled by modulating the cooling coil.

In his studies of the effect of air velocity on the cooling and dehumidification capacity of cooling coils, A. Shaw has identified the general characteristic that, as face velocity decreases, the relative capacity of dehumidification to sensible cooling increases (2). This result suggests that a cooling coil bypass scheme may be useful in reducing or eliminating overcooling and the associated reheating of conditioned air in certain circumstances. The solid line on the psychrometric chart in Figure 7 depicts the hypothetical cooling and dehumidification process of a coil. If the target supply condition is a humidity ratio of 0.008 lbm_w/lbm_{da} at 60^{o}F, the air would have to be cooled to 55^{o}F and then reheated to 60. The dashed line represents the process through the same coil with 25% bypass airflow. The bypassed air is mixed with the air that goes through the coil, to provide a humidity ratio of 0.008 at a temperature of 60^{o}F with no reheat. Even the 75% of the airstream that passes through the coil is cooled only to 56^{o}F rather than 55. In this example, the bypass scheme uses about 5 BTU/lbm_{da} for cooling while the reheat scheme uses 7 BTU/lbm_{da} for cooling and 2 BTU/ lbm_{da} for reheating. Of course, coil geometry and chilled water temperature will determine the limits of dehumidification by this scheme, but careful coil specification should

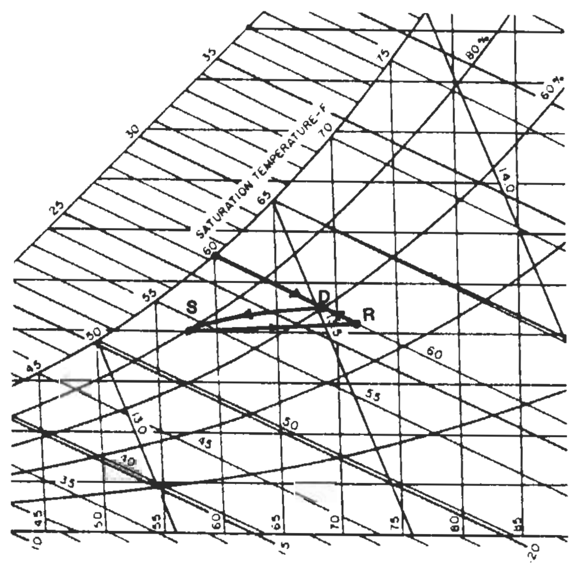

Fig. 6 Cooling Cycle - Refrigeration Mode

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allow the designer to eliminate the need for reheat in many designs.

The amount of dehumidification provided to condition a computer room with an imperfect vapor barrier could be varied by modulating face and bypass dampers at the coil. The dynamics of this control method have not been investigated in detail and may present stability problems. Alternatively, the bypass could be set manually to meet the worst case dehumidification load, and evaporative humidification could be used to compensate during periods of lower dehumidification load. A combination of these two strategies could also be employed. For the purposes of this paper, it will be assumed that a fixed bypass is maintained sufficient to meet peak dehumidification requirements in the refrigeration mode.

In the fresh air or mixed modes, if the humidity at state M or O is so high that space humidity remains too high even with H dampers closed, the O damper will close as the R damper opens and the chilled water coil will be used to maintain T_S as before.

In the fresh air mode, the minimum allowable temperature T_S, for the 80% RH limit, can be calculated from T_O, T_R, T_M, and outside and return humidities. For the mixed and refrigeration modes, the humidity at D can be calculated from T_D and the temperature and humidity of outside or return air respectively. To link the humidity at S to the humidity at D, an approximation of the coil characteristic between T_S and T_D can be used. Alternatively, a conservative estimate of the minimum temperature at S could be made by assuming the humidity at S equal to the humidity at D. A humidity sensor could also be used to directly measure the supply condition, but these sensors are typically more costly and slower in response than temperature sensors.

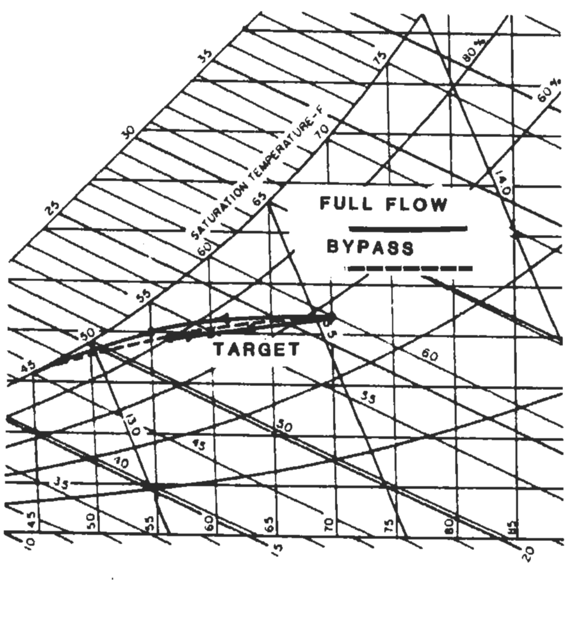

Fig. 7 Cooling Coil Bypass

Ideally, a variable speed fan would be used in the system to compensate for changes in system resistance as the control dampers modulate. As long as the resistance of the outside air ductwork is not significantly different than that of the return air ductwork, carefully designed face and bypass dampers should allow only minor variations in the volume of supplied air. A variable volume system would also allow increased protection against supply air relative humidity greater than 80%, by increasing air volume when large amounts of humidification are required. Becker suggests a system for precise process cooling by modulating a VAV damper in the underfloor opening to each piece of equipment. Many modifications can be made to tailor the basic system to best fit the needs of each application.

The use of modulated control rather than ON/OFF control should produce smooth operation and transitions between operational modes. The stability of the control has yet to be analyzed, but the continuous operation of the facility and the almost constant nature of the load would tend not to perturb the system. The control and controlled devices are standard HVAC equipment that have demonstrated stable performance in numerous installations. The use of temperature sensors for high humidity protection rather than a humidity sensor provides very fast detection of that condition. A certain amount of redundancy can be built into the sensors for failure detection.

The obvious choice for controlling this system would be a programmable controller, preferably with proportional and integral capability. The controller would require a small amount of computational power for enthalpy control and high humidity protection based on temperature. Because of the importance of system reliability, the control for the dampers and valves in the fresh air cooling system should be designed to return operation to that of a conventional computer room cooling system in the event of an excessive deviation from setpoint. The system relies on 2 humidity sensors and 5 temperature sensors, and their accuracy and reliability are of prime importance. The duct and supply temperature sensors must also have quick responses.

CONCLUSIONS

The fresh air cooling scheme with evaporative humidification presented in this paper offers impressive energy savings over conventional systems of greater than 24%, 56% and 38% for Dallas, Denver and New York respectively. The more humid climate in Houston reduces the savings to 11%. The savings in all cases are significantly greater than the savings obtained from an evaporatively cooled economizer coil.

The system requires control equipment that is not overly sophisticated to be handled by a reasonably straightforward control strategy. The controls appear to be capable of providing accurate maintenance of temperature and humidity setpoints as well as positive sensing of conditions required for computer equipment protection. The system can be designed to provide much greater humidification and dehumidification capacity than conventional conditioning systems to maintain accurate setpoints, without using powered humidification or reheat, in spite of vapor barrier inadequancies.

Installation of a fresh air cooling system will be very site specific. In addition to being better suited to cooler and drier climates, the system requires easy access to a fresh air supply. The system appears well suited for design into new buildings, but may also be retrofit in many cases. The system may have applications in any space where strict setpoint specifications have previously limited conditioning systems. Costs of installation remain to be considered, but will vary significantly from site to site.

The fresh air cooling system will not be universally applicable, but for certain installations, the energy savings potential and the control flexibility warrant further consideration and development.

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