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Project Summary

Energy Costs of IAQ Control Through Increased Ventilation in a Small Office in a Warm, Humid Climate: Parametric Analysis Using the DOE-2 Computer Model

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Abstract

A series of computer runs has been completed using the DOE-2.1E building energy model, simulating a small (4,000 ft²)* strip mall office cooled by two packaged single-zone systems, in a hot, humid climate (Miami). These simulations assessed the energy penalty, and the impact on indoor relative humidity (RH), when the outdoor air (OA) ventilation rate of the office is increased from 5 to 20 cfm/person in this challenging climate to improve indoor air quality (IAQ). One objective was to systematically assess how each parameter associated with the building and with the mechanical system impacts the energy penalty resulting from increased OA. Another objective was to assess the cost and effectiveness of off-hour thermostat setup (vs. system shutdown), and of humidity control (using overcooling with reheat), as means for reducing the number of hours that the office space is at an RH above 60% at the 20 cfm/person ventilation rate.

With the baseline set of variables selected for this analysis, an OA increase from 5 to 20 cfm/person is predicted to increase the annual cost of energy consumed by the heating, ventilating, and air conditioning (HVAC) system by 12.9%. The analysis showed that the parameters offering the greatest practical potential for energy savings are conversion to very efficient lighting and equipment (1.5 W/ft²) and conversion to very efficient cooling coils (electric input ratio = 0.284). If the increase to 20 cfm/person were accompanied by either of these conversions, the 12.9% HVAC energy penalty for the increased OA rate would be eliminated; the modified system at 20 cfm/person would have a *lower* annual HVAC energy cost than the baseline system at 5 cfm/person. Other parameters offering significant practical potential for energy savings are: conversion from packaged single-zone units to a variable air volume system; conversion to cold-air distribution (minimum supply air temperature = 42°F); or improvements in the glazing or in the roof resistance to heat transfer. If the OA increase were accompanied by any one of these modifications, the 12.9% penalty would be reduced to between 2 and 7% (the modified system at 20 compared against the baseline at 5 cfm/person).

According to the DOE-2.1E model, the increase in ventilation rate could be achieved with an 85% reduction in the number of occupied hours above 60% RH, compared to the baseline system at 5 cfm/person — with only a $19/year increase in energy cost — if the economizer were eliminated. That is, most of the elevated-RH hours in the baseline case were predicted to be the result of economizer operation. If the control system were modified so that it controlled the humidity as well as the temperature in the office space, *all* of the elevated-RH occupied hours would be eliminated, at an energy cost of $90/year.

*Readers more familiar with metric units may use the factors provided at the end of this Summary to convert to that system.

Neither economizer elimination nor humidity control would address *un*occupied periods, when most of the elevated-RH hours occur. Building operators concerned about biological growth at elevated RH should consider operation of the cooling system during unoccupied hours, perhaps with the thermostat set up, rather than system shut-down off-hours. Off-hour setup from 75 to 81°F would add only $10/year to energy costs, and would provide some modest reduction in unoccupied elevated-RH hours. Setup to 79°F would provide a greater reduction, at an energy cost of $38/year.

DOE-2.1E underestimates the number of elevated-RH hours because it does not address the moisture capacitance of building materials and furnishings, or re-evaporation off the cooling coils when they cycle off with the air handler operating. As a result, the performance of the RH reduction steps above may be overestimated, or the costs of the steps underestimated.

This Project Summary was developed by the National Risk Management Research Laboratory's Air Pollution Prevention and Control Division, Research Triangle Park, NC, to announce key findings of the research project that is fully documented in a separate report of the same title (see Project Report ordering information at back).

Introduction

Among the three basic techniques for improving IAQ — improved ventilation, air cleaning, and source management — improved ventilation is perhaps the most commonly utilized. In ANSI/ASHRAE Standard 62-1989, ASHRAE recommended that OA ventilation rates in office space be increased from 5 to 20 cfm/person to maintain acceptable IAQ. There will be an energy penalty associated with an OA increase, which will usually be most pronounced in hot, humid climates. Also of particular concern in humid climates, an OA increase can result in increased indoor RH levels, which can be of concern both from the standpoint of occupant comfort, and from the standpoint of fungal growth.

Objectives and Approach

To assess these energy and RH penalties associated with increased ventilation, a systematic series of computer simulations have been run using the DOE-2.1E software to model a small (4,000 ft^2) office in a hot, humid climate (Miami). These simulation runs comprised a parametric analysis to systematically quantify how each of the building and HVAC system variables impacts energy consumption and cost, and HVAC performance (in particular, indoor RH levels), at ventilation rates of both 5 and 20 cfm/person.

By defining the building and HVAC parameters having the greatest impact on HVAC energy consumption and cost, this assessment was intended to suggest those parameters which — if modified in conjunction with the increase in OA — could at least partially offset the energy and cost penalties associated with the increased ventilation rate. Likewise, by defining the parameters having the greatest impact on indoor RH, the assessment was intended to suggest parametric modifications which could reduce the RH impacts of the OA increase.

As part of this analysis, the DOE-2.1E model was used to further assess the energy penalty and the effectiveness of two specific approaches for reducing the number of hours at RH levels greater than 60%. These approaches are: 1) turning the thermostat up (rather than shutting the HVAC system down) during unoccupied cooling hours; and 2) use of a humidity controller on the HVAC system, employing overcooling and reheat as necessary to maintain the RH below 60% during occupied hours.

This analysis did *not* address the equipment/installation costs associated with the parametric variations, or any impact of the variables on maintenance costs.

The Baseline Building and HVAC System

The building type selected for this analysis was a small, one-story office in a strip mall, with adjoining space (occupied by other tenants) on either side. The office had a frontage of 40 ft and a depth of 100 ft for a total floor area of 4,000 ft^2, and was subdivided into two 2,000 ft^2 zones (of 40 by 50 ft). A small office was selected because the U. S. population spends a substantial number of hours inside offices, and Government statistics indicate that approximately half of the office buildings in the U. S. are 5,000 ft^2 and smaller.

The floor plan for this office is illustrated in Figure 1.

Full occupancy is 27 persons (150 ft^2/person). The occupancy varies throughout the day on weekdays, between 6 am to 7 pm. The building is unoccupied overnight (7 pm to 6 am), and all day on weekends and holidays.

The baseline HVAC system consists of two rooftop, constant-volume, packaged single-zone (PSZ) units, one dedicated to each of the 2,000 ft^2 zones. The units included electric resistance heating; annual heating requirements are minimal in the Miami climate. Ventilation rates of both 5 and 20 cfm/person were considered. The cooling setpoint was 75°F during occupied hours; the cooling was shut down overnight and on weekends. The heating setpoint was 70°F, set back to 55°F during off-hours. The cooling electric input ratio (EIR) was 0.341 Btu/hr of electric input per Btu/hr of cooling output, considered to be representative of modern PSZ units.

The Impact of Building and HVAC Parameters on the Penalties Associated with Increased Ventilation

Table 1 summarizes how each of the building and HVAC system parameters impacts the computed cooling coil capacity, the annual HVAC energy cost, and the percentage of occupied hours having an RH above 60%.

For ease in comparison, the impact of each parameter is presented in Table 1 as the percentage change from the baseline building and baseline system operating at a ventilation rate of 5 cfm/person. Under baseline conditions at 5 cfm/person, the cooling capacity computed by the software is 103.6 kBtu/hr (8.6 tons of refrigeration), the annual HVAC energy cost is $2,510, and the number of occupied hours above 60% RH is 40 hours per year (1.2% of 3,276 occupied hours), as shown by the first entry in the table.

All of the other entries are for operation at 20 cfm/person.

The second entry in the table shows the predicted impacts when the baseline building and system are simply operated at 20 cfm/person, without any other variations in the building and HVAC variables. For example, this entry shows that operation at the increased ventilation rate increases HVAC energy cost by 12.9% (an increase of $325, to $2,835 per year). Use of the HVAC energy costs in this table is intended to emphasize the impact on the HVAC system. If one instead used the *total building* energy costs — which are $4,273 per year, including lighting and equipment, at 5 cfm/person — the $325 increment caused by the OA increase would correspond to only a 5.4% increase.

The second entry in the table also shows that the OA increase in the baseline system is computed to *decrease* the percentage of elevated-RH occupied hours by 25%, from 40 to 29 hours per year.

The remainder of the entries show the predicted impacts (at 20 cfm/person) as each of the building and HVAC param-

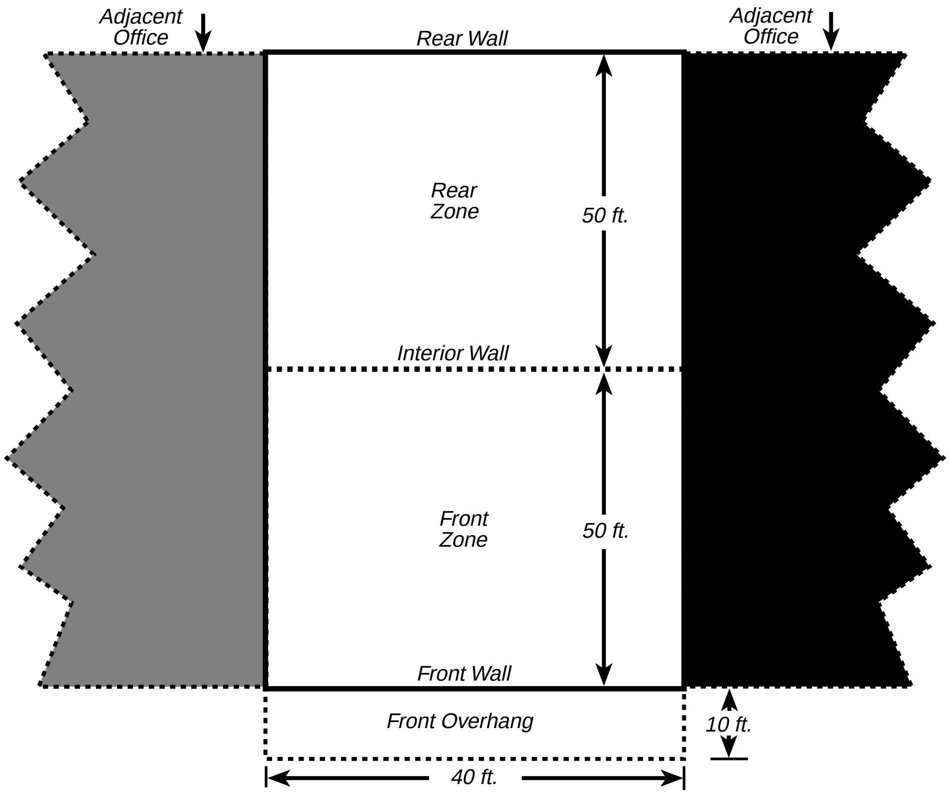

Figure 1. Floor plan for the baseline 4,000 ft^2 office in a Miami strip mall.

Table 1. Effects of Building and HVAC Variables on HVAC Capacity and Energy Cost, and on Occupied Hours Above 60% RH

	Cooling Coil Capacity	Annual HVAC Energy Cost[a]	Occupied Hours with RH >60%
OA Rate = 5 cfm/person			
Baseline system with OA rate of 5 cfm/person	103.6 kBtu/hr	$2,510	40 hr/yr
OA Rate = 20 cfm/person	Results below are expressed as the percentage change from the baseline numbers at 5 cfm/person, above		
Baseline system with OA rate of 20 cfm/person	15.1 (increase to 119.2 kBtu/hr)	12.9 (increase to $2,835)	-25 (decrease to 29 hr/yr)
Effect of (baseline):			
<u>Building (LOADS) Variables</u>			
Building orientation (building faces north)			
- *Building faces south*	15.5	10.2	-25
Building shading (door, window overhangs)			
- *Delete all overhangs*	21.0	16.	-25
Occupant density (150 ft^2/person			
- *Reduce density to 300 ft^2/person*	-0.2	-1.5	-31
- *Increase 60 4.0 W/ft^2*	29.6	26.9	-25
Lighting/equipment power use (2.55 W/ft^2)			
- *Reduce to 1.5 W/ft^2*	0.4	-5.1	+7
Infiltration rate (0.1 ACH[b])			
- *Decrease to 0 ACH*	13.1	11.4	-25

(continued)

eters are systematically varied from its baseline values. The percentage changes with each parameter should be compared with the percentage changes with the baseline system at 20 cfm/person, discussed in the preceding two paragraphs. If, for example, the percentage change in annual HVAC energy cost becomes less than 12.9% when a given parameter is varied, this parametric variation is predicted to consume less HVAC energy at 20 cfm/person than would the baseline at 20 cfm/person. In concept, the HVAC energy penalty associated with increasing the baseline from 5 to 20 cfm/person could be correspondingly reduced if the OA increase could practically be accompanied by this variation in this parameter.

In some cases, the percentages become negative. This means that a building or HVAC system incorporating that parametric variation could operate at 20 cfm/person *at a savings* compared to the baseline at 5 cfm/person.

Table 2 — presented in the same format as Table 1 — lists those entries from Table 1 that are predicted to offer the greatest potential reductions in HVAC energy cost at 20 cfm/person, compared to the baseline at 5 cfm/person. These entries are listed in descending order, with the parametric variation offering the greatest reduction listed first.

Similarly, Table 3 lists those entries from Table 1 that are predicted to offer the greatest potential reductions in hours at elevated RH at 20 cfm/person, compared to the baseline at 5 cfm/person. Again, the entries are listed in descending order.

Parameters Creating the Greatest Reductions in HVAC Energy

Six of the ten parameters listed in Table 2 are associated with the building: elimination of all exterior surfaces (a hypothetical consideration); reduced lighting/equipment wattage; reduced occupant density; decreased glazing; improved glass type; and increased roof insulation.

That each of these parameters would significantly reduce annual HVAC energy cost, of course, is not surprising. However, it is instructive to explore why these parameters fall in the order they do in Table 2.

As shown, lighting and equipment are the largest individual contributors to the HVAC load, contributing about half of the total load from all sources. Thus, it is not surprising that a 40% reduction in lighting plus equipment wattage (from 2.55 to 1.5 W/ft^2) would provide the greatest reduction in HVAC energy costs among the practical alternatives in Table 2. (Only the

Table 1. (continued)

	Percentage Increase Over Baseline at 5 cfm/person		
	Cooling Coil Capacity	Annual HVAC Energy Cost[a]	Occupied Hours with RH >60%
Exterior wall resistance (U_o = 0.16 Btu/hr ft^2 F^0)			
- Decrease to 0 Btu/hr ft^2 F^0	14.2	11.4	-25
- Decrease to 0.6 Btu/hr ft^2 F^0	14.8	12.3	-25
Amount of glazing (33% of exterior walls)			
- Decrease to 0%	10.4	3.5	-43
OA Rate = 20 cfm/person			
Glass type (U_o = 0.94 Btu/hr ft^2 F^0, S-C = 0.55[c])			
- Improve to U_o = 0.32, S-C = 0.16	13.3	6.1	-40
Roof resistance (U_o = 0.066 Btu/hr ft^2 F^0)			
- Reduce to U_o = 0	15.1	6.7	-34
Total office insulation (exterior walls, roof)			
- Eliminate all exterior surfaces (hypothetical)	7.6	-8.4	-60
HVAC (SYSTEMS) Variables			
Thermostat setup off-hours (shuts down off-hours)			
- Cooling setpoint 81^{0}F off-hours	15.1	13.3	-25
Alternative HVAC systems (2 PSZ units/2 zones)			
- 1 PSZ unit/1 zone	12.1	11.4	-30
- 1 PSZ unit/1 zone + 1 subzone	15.1	10.1	-33
- 1 PVAVS[d] unit/2 zones	16.8	4.9	+135
- 2 PTAC[e] units/2 zones	7.9	9.1	-100
Ducted return air (plenum return)			
- Air return via ducts	15.1	12.4	-33
Cold-air distribution (PSZ/55^{0}F min. supply T)			
- PSZ/42^{0}F minimum supply T	22.2	6.2	-72
- PVAVS/42^{0}F minimum supply T	23.3	1.6	-52
Economizer modiciations (T-controlled econo.)			
- No economizer	15.1	13.7	-85
- Enthalpy-controlled economizer	15.1	13.0	-55
Cooling electric input ratio (EIR = 0.341)			
- Cooling EIR = 0.284	15.1	-1.8	-25
- Cooling EIR = 0.427	15.1	105.5	-25
Cooling capacity and SHR[f] (8.6 tons/SHR = 0.75)			
- 10 tons/SHR = 0.78	15.8	13.8	-25
- 10 tons/SHR = 0.73	15.8	13.7	-25
- 11 tons/SHR = 0.78	27.4	15.4	-25
- 11 tons/SHR = 0.73	27.3	15.6	-25
Weather File Variables			
Alternative weather files (typical meteorological year)			
- Weather year for energy calcs.	*30.8*	*12.2*	*-8*

[a]Energy costs include electricity for: the air-conditioning compressor and condenser fan; the electric resistance heating coils; the motor for the central air handling fan; and auxiliaries (compressor crankcase heaters). Cost of electricity is $0.0473/kWh plus a demand charge of $9.96/kW above 10 kW.
[b]ACH = Air changes/h.
[c]S-C = Shading coefficient.
[d]PVAVS = Packaged variable-air-volume system.
[e]PTAC = Packaged terminal air conditioner.
[f]SHR = Sensible heat ratio.

hypothetical scenario of eliminating all exterior surfaces provided a greater reduction.) This reduction in lighting plus equipment wattage could be achieved by converting from the prescriptive or average wattages in ASHRAE 90.1-1989 to very efficient lighting (e.g., including daylighting) and more efficient (or more limited) equipment usage.

As shown in Table 2, the HVAC energy cost savings from more efficient lighting/equipment would more than offset the increase in HVAC energy costs resulting from an increase in OA from 5 to 20 cfm/person. The building with efficient lighting/equipment could operate at 20 cfm/person with a HVAC cost savings of 5.1% compared to the baseline 5 cfm/person case. Of course, efficient lighting/equipment would provide even greater savings in total building energy costs, by reducing the energy costs for lighting and equipment as well as for the HVAC system.

As shown in Table 4, occupants are tied with glazing and (at 20 cfm/person) with OA as the second largest contributor to HVAC energy consumption. Accordingly, it is not surprising that cutting occupancy in half (from 150 to 300 ft^2/person) would provide the next greatest reduction among the 6 building parameters in Table 2. Of course, reducing occupant density will not generally be a viable option for reducing energy costs.

Table 4 shows that — among the exterior surfaces — conduction and radiation through the glazing are the most important contributors to HVAC energy consumption in this office. As a result, it is not surprising that adjustments to the glazing — eliminating it altogether, or increasing its resistance to conduction and radiation — should show up on Table 2 as the next most effective building parameters for reducing HVAC energy costs. Eliminating (or substantially reducing) the glazing might not often be a viable option. However, improving the glazing resistance is a viable option, if the building owner is prepared to accept the increased construction costs.

Finally, Table 4 shows that — among the exterior surfaces — roof conduction is the second most important contributor, about half as important as glass conduction and radiation. The roof is important because it represents such a large exterior surface area for this building (4,000 ft^2, compared to only 700 ft^2 for the unglazed portion of the exterior walls), and it has the most consistent direct exposure to solar radiation. Consequently, it is not surprising that hypothetically increasing the roof resistance to infinity (i.e., reducing the roof U_o — heat transfer coefficient —

Table 2. Building and HVAC Variables Creating the Greatest Reductions in Annual HVAC Energy Cost at 20 cfm/person (from Table 1)

	Cooling Coil Capacity	Annual HVAC Energy Cost	Occupied Hours with RH > 60%
OA Rate = 5 cfm/person			
Baseline system with OA rate of 5 cfm/person	103.6	$2,510	40 hr/yr
OA Rate = 20 cfm/person	Results below are expressed as the percentage change from the baseline numbers at 5 cfm/person, above.		
Baseline system with OA rate of 20 cfm/person	15.1	12.9	-25
Variables giving the greatest reduction in HVAC energy cost at 20 cfm/person, in descending order.			
Eliminate all exterior surfaces (ideal)	7.6	-8.4	-60
Reduce lighting/equipment to 1.5 W/ft^2	0.4	-5.1	7
Reduce cooling electric input ratio to 0.284	15.1	-1.8	-25
Reduce occupant density to 300 ft^2/person	-0.2	-1.5	-31
Convert to PVAVS with cold-air distribution (minimum supply air T = 42^{0}F)	23.3	1.6	-52
Decrease glazing to 0% of wall area	10.4	3.5	-43
Convert from 2 PSZ units to 1 PVAVS unit - standard minimum supply air T (55^{0}F)	16.8	4.9	135
Improve glass type to U_o = 0.32, S-C = 0.16	13.3	6.1	-40
Convert PSZ to cold-air distribution (42^{0}F)	22.2	6.2	-72
Increase roof resistance to U_o = 0	15.1	6.7	-34

Table 3. Building and HVAC Variables Creating the Greatest Reductions in Hours above 60% RH at 20 cfm/person (from Table 1.)

	Cooling Coil Capacity	Annual HVAC Energy Cost[a]	Occupied Hours with RH > 60%
OA Rate = 5 cfm/person			
Baseline system with OA rate of 5 cfm/person	103.6	$2,510	40 hr/yr
OA Rate = 20 cfm/person	Results below are expressed as the percentage change from the baseline numbers at 5 cfm/person, above.		
Baseline system with OA rate of 20 cfm/person	15.1	12.9	-25
Variables giving the greatest reduction in hours at elevated RH at 20 cfm/person, in descending order.			
Eliminate economizer	15.1	13.7	-85
Convert PSZ unit to cold-air distribution (minimum supply air T = 42^{0}F)	22.2	6.2	-72
Eliminate all exterior surfaces (ideal)	7.6	-8.4	-60
Convert to enthalpy-controlled economizer	15.1	13.0	-55
Convert to PVAVS with cold-air distribution	23.3	1.6	-52
Decrease glazing to 0% of wall area	10.4	3.5	-43
Improve glass type to U_o = 0.32, S-C = 0.6	13.3	6.1	-40
Increase roof resistance to U_o = 0	15.1	6.7	-34

[a]The DOE-2.1E model used here does not account for moisture capacitance of the building materials/furnishings, or for re-evaporation of moisture from the cooling coils when the coils cycle off with the air handler operating. As a result, the number of hours computed to have RH>60% at any given set of conditions will usually be low.

from 0.066 Btu/hr ft^2 F^o to zero) is the exterior surface parameter that provides the next greatest reduction in HVAC energy cost in Table 2 (cutting the cost penalty from the OA increase about in half, from 12.9% to 6.7%).

Of course, reducing the roof U_o all the way to zero is not practical. However, these results show that — if additional resources are going to be expended to better insulate the shell of this particular office configuration — one is best served directing those resources towards improved glazing and increased roof resistance, rather than towards increased wall or slab resistance.

The other 4 of the 10 parameters listed in Table 2 are associated with the HVAC system: improving the cooling system efficiency; converting from a constant-volume PSZ system to a packaged variable-volume system (PVAVS); and conversion to cold-air distribution (i.e., a minimum supply air temperature of 42°F rather than 55°F), with either the PSZ or the PVAVS.

Of these four, the parameter providing the greatest reduction in HVAC energy cost is improved efficiency of the PSZ cooling coils. In this calculation, the EIR was decreased from the baseline value of 0.341 — corresponding to an energy efficiency ratio (EER) of 10 Btu/hr per W), representing a typical efficiency — to an EIR of 0.284 (EER = 12 Btu/hr per W), representing a high-efficiency unit. If the building owner were prepared to invest in high-efficiency cooling units, this office could operate at 20 cfm/person while simultaneously *saving* 1.8% of the HVAC energy cost compared to operation at 5 cfm/person with the baseline, moderate-efficiency system. This 1.8% savings corresponds to a modest $46/year.

As shown in Table 2, conversion of the pair of PSZ units to a single two-zone PVAVS (operating at the standard minimum supply air temperature of 55°F) would reduce by 60% the HVAC energy cost penalty associated with the OA increase. That is, the penalty would drop from 12.9% to 4.9%. PVAVS can be slightly more complicated and more expensive than the PSZ units, and hence do not appear to be as widely used in strip mall space of the type being modeled here. However, PVAVS of this capacity are commercially available, and can reasonably be considered as a means to reduce the energy penalty in this application.

PVAVS reduce energy consumption and cost by reducing the volume of supply air being delivered. Most of the savings result from reduced power consumption by the central air handling fan, since power

Table 4. Approximate Contribution of the Various Heat Sources toothe Annual HVAC Energy Consumption in the Baseline Building

Heat Source	Percentage Contribution to Annual HVAC Energy Consumption[a] OA = 5 cfm/person	OA = 20 cfm/person
Conduction and Radiation Through Exterior Surfaces (sensible)		
Exterior walls - conduction	2	2
Glazing - conduction and radiation	14	12
Door - conduction	0.5	0.4
Roof - conduction	8	6
Slab - conduction	-0.5	-0.4
Infiltration		
(sensible and latent)	2	2
Mechanically Introduced Outdoor Air		
(sensible and latent)	2	15
Internal Sources		
Occupants (sensible and latent)	15	13
Lighting (sensible)	40	35
Equipment (sensible)	17	15
Domestic hot water heater (sensible)	~0	~0
TOTAL	100	100

[a]Annual HVAC energy consumption for the baseline building is 26,145 kWh/year for a ventilation rate of 5 cfm/person, and 29,390 kWh/year for 20 cfm/person.

consumption varies with the cube of the volumetric flow rate. A small portion of the savings results from reduced cooling coil consumption, since reduced central fan operation results in less heat being added to the circulating air stream by the fan motor.

Finally, Table 2 shows that conversion to cold-air distribution (with either the PVAVS or the PSZ system) will provide a significant reduction in the HVAC energy penalty associated with the OA increase. Operation at a minimum supply air temperature of 42°F instead of 55°F reduces volumetric flow rates, thus reducing fan power consumption as well as the amount of heat added to the air stream by the fan motor. Superimposing cold-air distribution and a PVAVS — for which volumetric flows are already significantly reduced — provides the greater reduction in HVAC energy costs, among the two HVAC types.

The use of cold-air distribution creates a number of design and operating complications that could make such an approach impractical for small offices such as the one modeled here, where simplicity in maintenance is important. Among these complications is the need for: a) increased care to reduce the risk of moisture condensation on the ductwork and the diffusers; and b) possible powered terminals to provide adequate throw of the reduced volume of air out through the diffusers (a step which would offset part of the energy savings achieved through the reduction in volumetric flow).

Parameters Creating the Greatest Reductions in Hours at Elevated RH

According to the DOE-2 model, occupied hours having RH levels greater than 60% occur on cool mornings in Miami. During the first hours after system startup on cool mornings, the outdoor RH can be high (over 90%), but the indoor and outdoor temperatures can be sufficiently low that the cooling coils operate at greatly reduced capacity (or remain off altogether). As soon as the cooling coils begin operating at a significant fraction of their capacity — usually within 2 or 3 hours after startup — the indoor RH drops below 60%. (On warm summer mornings, the coils begin operating near full capacity immediately upon startup; thus, elevated-RH indoor hours never occur during warm weather, despite the high outdoor RH levels that exist.)

On some cool morning hours, when the economizer is able to provide all of the sensible cooling required by the space, the economizer will activate in lieu of coil operation. (The economizer and the cooling coils cannot operate simultaneously in the packaged PSZ units.) During economizer operation — when a large amount of untreated, potentially high-moisture-content outdoor air can be introduced into the building — there is an increased potential for indoor RH levels to exceed 60%. In practice, the economizer on the HVAC systems being modeled here does not operate often.

As shown in Table 3, simply increasing the OA rate from 5 to 20 cfm/person using the baseline system (with no changes in any other variables) is predicted by DOE-2 to reduce the number of elevated-RH occupied hours by 25%. Although this percentage may seem significant, the actual number of hours involved is small, corresponding to a reduction from 40 hours per year at 5 cfm/person (1.2% of all occupied hours) to 29 hours at 20 cfm/person (0.9%).

The number of elevated-RH hours occurs because, on average, the increased OA rate increases the sensible load. In attempting to address this increased load, the PSZ coils operate at a lower temperature during the cool morning periods when elevated-RH hours occur. This increases the latent cooling provided by the system. This increase in latent cooling at 20 cfm/person is predicted by DOE-2 to more than offset the increase in latent load caused by the increased OA rate.

Other researchers have made similar calculations using a model that includes factors not addressed by DOE-2, namely, moisture capacitance and re-evaporation off the cooling coils. These researchers predict that — in contrast to the DOE-2 predictions — an increase in OA rate in Miami would significantly *increase*, not *decrease*, hours at elevated RH. Also, when capacitance and re-evaporation are considered, it is predicted that some of the elevated-RH hours will occur during warm weather, not just on cool mornings.

Table 3 lists the eight parametric variations predicted by DOE-2 to provide the greatest reductions in the number of elevated-RH occupied hours.

Two of the most effective of these eight variations involve adjustments to the economizer. This is not surprising, since — in the Miami climate, as discussed above — the economizer is likely to cause elevated indoor RH during those hours when it operates.

When the economizer is eliminated altogether (and the system is operating at 20 cfm/person), as shown in the table, occupied hours above 60% RH are reduced by 85% compared to the baseline 5 cfm/person case (from 40 to 6 hours/year). This result confirms that, in this humid climate, the bulk of the elevated-RH hours are caused by the economizer.

If the economizer were converted to enthalpy control, rather than standard temperature control, hours above 60% RH are reduced by 55% (from 40 to 18 hours/year). Economizer enthalpy control prevents the economizer from operating if the outdoor enthalpy is greater than the indoor enthalpy (even if the outdoor temperature is lower). But this controller does not make any effort to control the indoor humidity. Thus, if the outdoor enthalpy were lower, the controller would allow the economizer to operate — and hence allow the cooling coils to shut down — even if this meant that indoor RH values would exceed 60%. Thus, enthalpy control would eliminate only *some* of the economizer-induced elevated-RH hours.

Elimination of the economizer altogether is predicted by DOE-2 to almost eliminate occupied hours above 60% RH in this climate, and it does so with only a modest energy cost penalty. The annual HVAC energy cost for the no-economizer case at 20 cfm/person is only $23/year greater than that for the baseline temperature-controlled-economizer case at 20 cfm/person. Thus, this is a viable option to consider for reducing indoor RH. By comparison, the option of economizer enthalpy control is less attractive, since the cost and maintenance requirements make such controllers less desirable for small office applications, and since enthalpy control is less effective in reducing elevated-RH hours.

Two of the other parametric variations in Table 3, offering significant reductions in the number of elevated-RH hours, involve conversion to cold-air distribution. These include conversion of the baseline PSZ units to cold-air distribution (providing a 72% reduction, from 40 to 11 hours), and conversion to a PVAVS with cold-air distribution (providing a 52% reduction, from 40 to 19 hours). This occurs largely because — at the very low coil temperatures in cold-air systems — the amount of latent cooling increases significantly relative to the standard (55 °F supply air temperature) case. Thus — after the coils activate on cool mornings, when the elevated-RH hours occur — RH levels in the office space drop more rapidly with the cold-air system.

However, due to the operating complications and likely increased maintenance of cold-air systems, it is not likely that this approach would often be considered for use in a small strip mall office such as the one modeled here.

The remaining four parameters listed in Table 3 involve efforts to make the building shell more heat resistant: hypothetical total isolation of the space; elimination of the glazing; improving the glazing; and increasing the roof resistance. These four parameters appear in Table 3 in the same order that they appeared in Table 2.

These parameters have this effect on the number of elevated-RH hours because the better insulated the building, the less it cools off over cool winter nights and weekends. Consequently, the cooling coils see a greater cooling load more quickly after startup on the cool mornings, when the elevated-RH occupied hours occur. The temperature-activated coils come on earlier after startup, and provide greater total (and hence latent) cooling during these morning hours, thus reducing the number of elevated-RH hours. The more effective the shell insulating step, the better the building retains its heat overnight, and the greater the resulting latent heat removal in the morning. For this reason, the insulation steps that provide the greatest reduction in total HVAC energy cost (Table 2) also provide the largest reduction in elevated-RH hours (Table 3).

These results show that resources devoted toward improved glazing and increased roof resistance will have the greatest impact, not only on reducing HVAC energy cost, but also in reducing (modestly) the number of hours at elevated RH.

It is interesting to note that Table 3 does not include any of the parameters that involve latent heat entry into, or generation inside, the building. Reducing occupant density to 300 ft^2/person reduces the number of elevated-RH hours by 31%, just below the cut-off used in preparing the table. Reducing outdoor air infiltration from 0.1 to 0 ACH has essentially no impact on the number of elevated-RH hours.

The Impact of Steps to Reduce Indoor Humidity

As indicated previously, the DOE-2 model incorporates neither the moisture capacitance of building materials and furnishings, nor moisture re-evaporation off the cooling coils when the coils cycle off with the air handler operating. As a result — unlike a model that *does* include these phenomena — DOE-2 does not predict an increase in elevated-RH occupied hours when the OA rate is increased. On this basis, DOE-2 might not be expected to precisely simulate the actual energy and performance impacts that would result when steps are taken to reduce the number of hours at elevated RH.

Despite this shortcoming, it is still felt that a DOE-2 analysis can provide useful perspective regarding the possible magnitude of the effects of steps to reduce RH. For example, the conclusion in the preceding section — that elimination of the economizer would substantially reduce the number of occupied hours above 60% RH — is felt to be valid, despite the fact that the absolute number of computed elevated-RH hours might be low.

A variety of steps can be taken to reduce the number of hours at elevated indoor RH in warm, humid climates. These steps fall into two categories:

a) Utilize an HVAC control system that relies solely on temperature control, as is typical for office space. But design and operate the HVAC system such that — as the system operates to control temperature in the space — there will be as few hours as possible having RH levels above 60%.

b) Incorporate humidity control as well as temperature control into the HVAC control system, which is not common for an office of this type. The humidity control could be achieved, using overcooling with reheat or using desiccants.

The RH results presented in Tables 1 and 3 can be viewed as an assessment of a wide range of building and HVAC parameters that might serve as steps that would fall into Category a) above. The most practical conclusion apparent from Table 3 is that occupied hours at elevated RH can be substantially reduced at 20 cfm/person if the economizer is deleted in warm, humid climates.

Two additional RH reduction steps are considered in further detail here. One — which falls into Category a) above — involves setting the thermostat temperature up to 81°F during off-hours (overnight, weekends, and holidays) rather than turning the system off altogether during cooling periods. The second — which falls into Category b) — involves using a humidity controller on the system, overcooling and reheating the supply air as necessary. The humidity control approach was considered in order to assess the energy penalty associated with this procedure, recognizing that humidity control is not commonly used in small offices, and that reheat is generally prohibited by Florida code.

Thermostat Setup vs. System Shutdown

According to the DOE-2 simulation, setting the thermostat up to 81 °F during off-hours, rather than shutting off the system, will have no impact on the number of occupied hours above 60% RH. This result occurs because elevated-RH occupied hours are predicted by DOE-2 to occur during the first hours after startup on cool mornings. During such cool weather, the overnight temperatures will not have been sufficiently high to cause the overnight office temperature to exceed 81°F. Thus, even if the thermostat is set up rather than the system being turned off, the cooling coils will not activate overnight. No latent cooling will be provided overnight, and, as a result, the latent load encountered by the system upon startup in the morning will remain unchanged. Accordingly, the number of elevated-RH *occupied* hours will remain unchanged.

This predicted result could be different if the DOE-2 model had addressed moisture capacitance and coil re-evaporation. In that case, some elevated-RH occupied hours occur during hot weather, when temperatures sometimes *can* be sufficient to activate the coils overnight. This would reduce the latent load seen by the system upon startup, by reducing the re-evaporated moisture that remains in the air overnight, and by reducing the amount of sorbed moisture. It could thus reduce the number of elevated-RH *occupied* hours occurring during warm weather.

With or without consideration of capacitance and re-evaporation, switching to thermostat set-up rather than system shutdown *will* reduce the number of elevated-RH *un*occupied hours during warm weather. On some hot nights and weekends in Miami, with the system off, the indoor RH can be above 60% much of the time, due to infiltration alone (even in the absence of re-evaporation effects). With thermostat setup, when the off-hour office temperature exceeds 81°F and the coils come on, the RH drops below 60%, at least for the hours when the coils are activated. On some warm days, this can represent 15 to 20% of the unoccupied hours that would otherwise be at elevated RH. The same reduction in elevated unoccupied hours will be achieved regardless of the OA rate during occupied hours, since OA ventilation is not provided during unoccupied hours.

Thermostat setup will not impact the elevated-RH unoccupied hours that occur during cool weather in Miami, since the office temperatures will generally not exceed the 81°F level that would trigger coil operation.

Of course, reducing the number of elevated-RH *un*occupied hours will not improve occupant comfort, since no-one will be in the building. But it *will* reduce the risk of biological growth. Since most of the elevated-RH hours in this building occur during unoccupied hours — regardless of whether the OA rate during occupied hours is 5 or 20 cfm/person — switching from system shutdown to thermostat setup would appear to be an important step for any building operator concerned about microbiologicals.

The operating cost associated with set-up vs. shutdown is low, according to the DOE-2 predictions. As shown in Table 1, switching to 81 °F thermostat setup at 20 cfm/person would increase the annual HVAC energy cost by only 0.4% (amounting to only $10 per year) compared to the baseline shutdown case at 20 cfm/person. Selecting an even lower set-up temperature of 79°F — which would reduce the number of unoccupied elevated-RH hours by an even greater amount — would increase annual HVAC energy costs by only $38.

Humidity Control by Overcooling and Reheat

Overcooling the supply air to condense moisture, then reheating to achieve the proper supply air temperature, has historically been a method for controlling humidity. Although humidity control is not commonly utilized in small offices (except in special cases), and although Florida codes now generally prohibit reheat due to the energy penalty involved, it is of interest to assess the costs and effectiveness of this approach, in comparison with the other approaches considered here.

The results of this analysis of a reheat-based humidity control system are summarized in Table 5, presented in the same format as Table 1.

As shown in the table, the humidity control system operating at 20 cfm/person is predicted by DOE-2 to increase the HVAC energy cost by 16.5% compared to the baseline (temperature-control) system at 5 cfm/person, and by 3.6% (i.e., 16.5 vs. 12.9%) compared to the baseline system at 20 cfm/person. But the humidity control system does achieve its objective, of eliminating all occupied hours above 60% RH.

Comparing Tables 3 and 5, it is apparent that — among the parametric variations predicted to provide the greatest reductions in elevated-RH hours — conversion to a humidity controller involves the largest increase in HVAC energy cost (16.5%), but provides the greatest reduction in hours above 60% RH (100%). Second to the humidity controller in both these categories — at least as estimated by DOE-2 — is elimination of the economizer (13.7% increase in HVAC energy cost, 85% reduction in elevated-RH hours).

This comparison suggests that elimination of the economizer would eliminate 85% of the elevated-RH occupied hours at 20 cfm/person, at an energy cost increase of $19/year (compared to the baseline case at 20 cfm/person). To eliminate the remaining 15% of the elevated-RH occupied hours, one could convert to a humidity controller, at an energy cost increase of $90/year (compared to the baseline at 20 cfm/person). Conversion to a humidity controller automatically prevents economizer operation during those hours when the economizer is responsible for the elevated RH, and provides the additional cooling/reheat required to address the remaining elevated-RH hours.

This DOE-2 comparison would change if one included moisture capacitance and coil re-evaporation in the model. In that more rigorous case, the effectiveness of economizer elimination at 20 cfm/person would decrease — i.e., the percentage reduction in elevated-RH occupied hours would be much less than 85% — since the new model would show a much greater number of elevated-RH hours being caused by factors other than economizer operation. Conversion to a humidity controller would remain 100% effective, but the energy cost would increase, since, again, the new model would show many more elevated-RH occupied hours.

Most of the energy penalty incurred by the humidity-controlled system results from additional sensible and latent cooling on cool, humid days, when the elevated-RH occupied hours occur, according to the DOE-2 model. As would be expected, the penalty is relatively small during mild and hot weather. And the contribution of reheat to the total penalty is small, on the order of 10% of the total; the increased sensible and latent cooling is responsible for the remainder.

It is emphasized that humidity control will maintain the RH in the offices only during *occupied* hours, when the HVAC system is operating. Regardless of which simulation model is used, a large fraction of the total elevated-RH hours in the space occur during *un*occupied hours. Thus, if biological growth is a concern, some off-hour operation would be required even if a humidity control system were used to eliminate all elevated-RH hours during occupied periods. This is true regardless of the OA rate during occupied hours.

Table 5. Effect of Humidity Control by Overcooling and Reheat

	Cooling Coil Capacity	Annual HVAC Energy Cost[a]	Occupied Hours with RH > 60%
OA Rate = 5 cfm/person			
Baseline system with OA rate of 5 cfm/person	103.6	$2,510	40 hr/yr
OA Rate = 20 cfm/person	Results below are expressed as the percentage change from the baseline numbers at 5 cfm/person, above.		
Baseline system (temperature control only) with OA rate of 20 cfm/person	15.1	12.9	-25
Humidity control system (temperature plus RH control) with OA rate of 20 cfm/person	15.1	16.5	-100

[a]Energy costs include electricity for: the air conditioning compressor and condenser fan; the electric resistance heating coils; the motor for the central air handling fan; and auxiliaries (compressor crankcase heaters). Cost of electricity is $0.0473/kWh plus a demand charge of $9.96/kW above 10 kW.

METRIC EQUIVALENTS

Readers more familiar with the metric system may use the following factors to convert to that system:

Nonmetric	*multiplied by*	*yields Metric*
Btu/hr	*0.293*	*W*
cfm	*37*	*L/s*
°F	*5/9 (°F-32)*	*°C*
ft	*0.305*	*m*
ft²	*0.0929*	*m²*
ton (of refrigeration) (12,000Btu/hr)	*3.520*	*kW (of cooling capacity*

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The complete report, entitled "Energy Costs of IAQ Control Through Increased Ventilation in a Small Office in a Warm, Humid Climate: Parametric Analysis Using the DOE-2 Computer Model," (Order No. PB98-113368; Cost: $38.00, subject to change) will be available only from:

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Springfield, VA 22161
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