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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

by

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16. ABSTRACT The report gives results of a series of computer runs using the DOE-2.1E building energy model, simulating a small office in a hot, humid climate (Miami). These simulations assessed the energy and relative humidity (RH) penalties when the outdoor air (OA) ventilation rate is increased from 5 to 20 cfm/person to improve indoor air quality. The effect was systematically assessed of each building and mechanical system parameter on the energy penalty resulting from increased OA. The cost and effectiveness were also assessed of methods for reducing elevated-RH hours. The parameters offering the greatest practical potential for energy savings are conversion to: very efficient lighting and equipment; very efficient cooling coils; to a variable air volume (from a constant volume) system; cold-air distribution; or improved glazing or roof resistance to heat transfer. If the OA increase were accompanied by any one of these modifications, the energy penalty would be significantly reduced (comparing the modified system at 20 against the baseline at 5 cfm/person). The number of occupied hours above 60% RH could be dramatically reduced (with a minimal energy cost impact) if the economizer were eliminated. Conversion to a system that controlled office humidity would eliminate all of the elevated-RH occupied hours, at an energy cost of $90/year.

17. KEY WORDS AND DOCUMENT ANALYSIS

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Energy	Stationary Sources	14G
Expenses	Indoor Air Quality	05C, 14A
Ventilation		13A
Humidity		04B
Computerized Simulation		

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FOREWORD

The U.S. Environmental Protection Agency is charged by Congress with protecting the Nation's land, air, and water resources. Under a mandate of national environmental laws, the Agency strives to formulate and implement actions leading to a compatible balance between human activities and the ability of natural systems to support and nurture life. To meet this mandate, EPA's research program is providing data and technical support for solving environmental problems today and building a science knowledge base necessary to manage our ecological resources wisely, understand how pollutants affect our health, and prevent or reduce environmental risks in the future.

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E. Timothy Oppelt, Director
National Risk Management Research Laboratory

ABSTRACT

A series of computer runs has been completed using the DOE-2.1E building energy model, simulating a small (4,000 ft^2) 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. 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 set-up (vs. system shut-down), 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/ft^2) 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.

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 set-up from 75 to 81 °F would add only $10/year to energy costs, and would provide some modest reduction in unoccupied elevated-RH hours. Set-up 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.

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METRIC CONVERSION FACTORS

Although it is EPA's policy to use metric units in its documents, non-metric units have been used in this report consistent with common practice in the heating, ventilating, and air-conditioning industry. Readers more accustomed to the metric system may use the following factors to convert to that system.

Non-Metric	Times	Yields Metric
foot (ft)	0.305	meter (m)
square foot (ft^2)	0.0929	square meter (m^2)
cubic foot per minute (cfm)	37	liters per second (L/s)
pound (lb)	0.454	kilogram (kg)
degrees Fahrenheit (°F)	5/9 (°F - 32)	degrees Celsius (°C)
British thermal unit (Btu)	0.293	watt-hour (W-hr)
British thermal unit per hour (Btu/h)	0.293	watt (W)
ton (of refrigeration) (12,000 Btu/h)	3,520	watts (of cooling capacity)

SECTION 1

OBJECTIVES AND APPROACH

1.1 BACKGROUND

In 1989, the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) issued ANSI/ASHRAE Standard 62-1989 (ASHRAE, 1989a) recommending increased outdoor air (OA) ventilation rates to maintain acceptable indoor air quality (IAQ) inside the full array of commercial, institutional, and residential structures. This standard recommends, for example, an OA ventilation rate of 20 cfm per person in the office space within office buildings, and 15 cfm/person in the classroom space within schools. These ventilation rates are greater than those which had been recommended in the earlier version of Standard 62 (ASHRAE, 1981), which had recommended 5 cfm/person in non-smoking offices and classrooms.

Increased OA ventilation rate, and improved distribution of the ventilation air, are perhaps the most commonly utilized techniques for improving IAQ. Accordingly, IAQ researchers are concerned with the costs associated with increased ventilation rates, and with possible approaches for reducing these costs, as a means for increasing the acceptance and effective utilization of this IAQ control technique.

Since the time that increased outdoor air ventilation rates were being considered during the development of Standard 62-1989, a number of computer simulation studies have been conducted to estimate the energy consumption and energy cost impacts of this increase in various building types (Eto and Meyer, 1988; Eto, 1990; Steele and Brown, 1990; Ventresca, 1990; Mudarri and Hall, 1993; Rojeski et al., 1995; Shirey and Rengarajan, 1996). Most, although not all, of the published simulation studies have involved the use of DOE-2, the building energy simulation software developed for the U. S. Department of Energy (York et al., 1981; U. S. Department of Energy, 1994a). These studies estimate that increasing OA from 5 to 20 cfm/person in office buildings would increase total annual building energy consumption and cost by 0 to 5% in large office buildings, and by 2 to 15% in medium to small office buildings, depending upon climate.

Some of the simulation studies cited above addressed a single, specific building with a specific heating, ventilating, and air-conditioning (HVAC) system. Other studies considered multiple buildings. Those that addressed multiple buildings generally addressed multiple specific buildings with specific HVAC systems; for any one building, the building parameters and the HVAC design and operating parameters were generally fixed for the analysis (except for the variation in OA rates).

Therefore -- while these several studies on a variety of realistic buildings consistently show the energy cost of increased OA to be limited -- the studies do not quantify the extent to which the selection of the many specific building and HVAC parameters impacts the cost of increasing OA from 5 cfm/person to the higher ASHRAE 62-1989 value. Perhaps the cost of increasing OA might be at least partially offset by judicious selection of building and HVAC design and operating conditions.

The Mudarri and Hall study did vary a few of the building parameters (internal configuration, shell heat transfer resistance, and occupant density) and HVAC parameters (HVAC system type, presence or absence of an economizer, and boiler and chiller efficiencies) in a large office building. But this study was conducted for the moderate climate of Washington, D. C., where the cost of increasing OA in a large office is relatively small. Thus, the incremental effects of these building/HVAC parameters on the costs of increased OA were difficult to distinguish. Steele and Brown also addressed the effect of one building parameter, namely, occupant density, in ten different building types in the Seattle and Richland, WA, climates. It would be of interest to systematically determine the effects of varying a broader array of building and HVAC design and operating parameters, in a climate where any effects of these variations would be as pronounced as possible.

Most of the energy modeling to date has focussed on cold (or temperate) climates. However, the greatest energy cost increases resulting from increased OA tend to occur in hot, humid climates, due to the high sensible and latent cooling loads. In addition to its impact on energy costs, increased OA in humid climates can necessitate added efforts to control the indoor relative humidity (RH) below the upper limit of 60% recommended by ASHRAE, for purposes both of thermal comfort (ASHRAE, 1992a) and of reduced allergenic/pathogenic organism growth (ASHRAE, 1989a). For these reasons, some of the greatest concerns about the increased OA rates recommended by ASHRAE 62-1989 have been expressed in regions having hot, humid climates. Eto and Meyer (1988) and Eto (1990) did include the humid Miami, FL, climate among the 13 cities that they considered; these analyses predicted that, among the cities studied, Miami would experience the greatest increase in energy consumption and costs resulting from the OA increase, in medium-sized and large offices. However, these two studies did not address the impact of increased OA on indoor RH levels, or any energy penalty resulting from an effort to control RH increases created by the increase in ventilation rate.

Among the published simulation studies, only Shirey and Rengarajan have attempted to rigorously address the cost impacts of controlling indoor humidity when increasing OA in humid climates. This latter study showed that an increase from 5 to 20 cfm/person in a small office in Miami would indeed increase annual total building energy costs by roughly 6%, consistent with Eto (1990), if one addresses only the control of temperature in the office. But if one wishes to simultaneously control RH to less than or equal to 60%, special steps would be required which could cause the energy cost increase to rise to 10 to 15%, depending upon the humidity control approach that is taken. Taking into consideration the increased *installation* costs (as well as the energy costs), Shirey and Rengarajan estimate that an increase from 5 to 20 cfm OA/person could increase the life cycle costs of the HVAC system (including installed hardware plus energy) by 7 to 32%, if indoor RH is to be controlled to 60% or less. Because standard over-cooling and reheat of the supply air to control enthalpy is generally prohibited by Florida code (FDCA, 1993), Shirey and Rengarajan did not address this approach.

1.2 OVERALL APPROACH

This study involved a systematic series of DOE-2.1E computer simulations to estimate indoor conditions (i.e., the number of hours at elevated temperature and RH), and the energy consumption and cost, in a small (4,000 ft^2) office in Miami conditioned by packaged, direct expansion HVAC equipment. This office is similar, although not identical, to the office modeled by Shirey and Rengarajan.

These simulation runs comprised a parametric analysis to systematically quantify how each of the potentially important building design and operating variables (in DOE-2 terminology, the LOADS variables), and each of the HVAC design and operating variables (the SYSTEMS variables), impact the computed indoor conditions and the computed energy consumption and cost in that office. For each building and HVAC parameter, simulations were run to determine the incremental effect of that parameter on the increase in energy use, and on the change in indoor conditions, resulting from an increase in OA ventilation rate from 5 to 20 cfm/person.

Particular attention was devoted to assessing the effectiveness and costs of possible means for controlling RH $\leq$60% using the conventional direct expansion HVAC systems typical of small offices. This analysis was not as rigorous as that of Shirey and Rengarajan, but did address the over-cool/reheat approach that was not analyzed in the other study.

1.3 OBJECTIVES

This effort had three objectives.

1) To assess the extent to which the selection of the baseline building conditions impacts the computed indoor conditions and energy usage/cost, and the computed penalty caused by an OA ventilation rate increase from 5 to 20 cfm/person.

2) To assess whether variations in specific building and HVAC design and operating parameters might at least partially offset the increase in energy costs resulting from the increase in ventilation rate, under the challenging conditions of a hot, humid climate.

3) To develop a better understanding of the practical issues and costs involved in controlling indoor RH when the ventilation rate is increased in small offices in hot, humid climates.

If the costs and the indoor temperature/RH impacts associated with an increase in the OA rate can be reduced through judicious selection of building and HVAC design and operating conditions, increased ventilation might be more widely accepted and more effectively utilized for IAQ control in humid climates.

Another, secondary objective of the study was to develop familiarity with the DOE-2 software.

1.4 LIMITATIONS

The cost analysis conducted in this report addressed only the energy cost impacts of the various building and HVAC system modifications. No attempt was made here to address the impacts on the installation cost of the building or of the HVAC system, or to address any impacts on maintenance costs.

The standard DOE-2.1E software does not address the moisture capacitance of the building materials and furnishings; nor does it address condensed moisture that is re-evaporated off of the cooling coils when the compressor cycles off during air handler operation (Birdsall, 1995). Shirey and Rengarajan (1996) utilize a customized version of DOE-2 to conclude that these two moisture-related issues that can have significant impacts on the computed indoor RH values in humid climates. On worst-case summer days in Miami, the RH values predicted by DOE-2.1E during HVAC operating hours might be as much as 10 to 20 percentage points lower than those predicted by a model that *does* incorporate these moisture considerations, according to Shirey and Rengarajan.

If this is true, the DOE-2.1E calculations presented in this report will under-estimate the humidity levels in the small office, including the number of hours having RH $\geq$ 60% (one of the parameters reported here as a measure of indoor conditions). This potential problem would also impact the assessment here of the equipment performances and the energy costs that would be required in order to control indoor RH levels below 60% as OA is increased from 5 to 20 cfm/person, since the assessment may be being made using artificially low RH values.

Despite this potential problem, this analysis still meets its basic objectives, utilizing what is arguably the most widely used software in the U. S. for modeling building energy consumption and costs. Specifically, this analysis quantifies how individual building and HVAC parameters impact energy consumption and costs, and whether the selection of these parameters might significantly affect the estimated costs of increased ventilation, even though there might be some question regarding the accuracy of the predicted RH values. The analysis also provides useful perspective regarding the effectiveness and relative energy costs of various options for controlling RH, again despite the uncertainty in the RH values.

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SECTION 2

EXECUTIVE SUMMARY

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 outdoor air 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.

2.1 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 over-cooling 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.

2.2 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 front and rear exterior walls are concrete block with exterior stucco and interior insulation, with 46% glazing in the front and 20% in the rear. The side walls are considered to be thermally neutral interior walls, adjoining neighboring offices. The roof is assumed to be built-up roofing over insulated decking, and the floor is a carpeted concrete slab.

Full occupancy is 27 persons (150 ft^2/person). The occupancy varies throughout the day on weekdays, between 6 am and 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/h of electric input per Btu/h of cooling output, considered to be representative of modern PSZ units.

Further details regarding the baseline building and HVAC system are presented in Section 3 and in Appendix B.

2.3 THE IMPACT OF BUILDING AND HVAC PARAMETERS ON THE PENALTIES ASSOCIATED WITH INCREASED VENTILATION

Table S-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 S-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/h (8.6 tons of refrigera-

TABLE S-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[1]	Occupied Hours with RH > 60%
OA Rate = 5 cfm/person			
Baseline system with OA rate of 5 cfm/person	103.6 kBtu/h	$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 (incr. to 119.2 kBtu/h)	12.9 (incr. to $2,835)	-25 (decr. to 29 hr/yr)
Building (LOADS) Variables			
Effect of building orientation (baseline building faces north)			
- Building faces south	15.5	10.2	-25
Effect of building shading (baseline has door, window overhangs)			
- Delete all overhangs	21.0	16.0	-25
Effect of occupant density (baseline is 150 ft^2/person)			
- Reduce density to 300 ft^2/person	-0.2	-1.5	-31
- Increase to 100 ft^2/person	29.6	26.9	-25
Effect of lighting/equipment power use (baseline is 2.55 W/ft^2)			
- Reduce to 1.5 W/ft^2	0.4	-5.1	+7
- Increase to 4.0 W/ft^2	36.1	38.7	-48

(continued)

TABLE S-1 (continued)

	Percentage Increase Over Baseline at 5 cfm/person		
	Cooling Coil Capacity	Annual HVAC Energy Cost[1]	Occupied Hours with RH > 60%
OA Rate = 20 cfm/person (continued)			
Effect of infiltration rate (baseline is 0.1 ACH)			
- Decrease to 0 ACH	13.1	11.4	-25
Effect of exterior wall resistance (baseline U_o=0.16 Btu/h ft^2 F°)			
- Decrease to 0 Btu/h ft^2 F°	14.2	11.4	-25
- Decrease to 0.6 Btu/h ft^2 F°	14.8	12.3	-25
Effect of amount of glazing (baseline is 33% of exterior walls)			
- Decrease to 0%	10.4	3.5	-43
Effect of glass type (baseline U_o=0.94 Btu/h ft^2 F°, shading coefficient = 0.55)			
- Improve to U_o=0.32, S-C=0.16	13.3	6.1	-40
Effect of roof resistance (baseline U_o=0.066 Btu/h ft^2 F°)			
- Reduce to U_o = 0	15.1	6.7	-34
Effect of total insulation of office (baseline has exterior walls, roof)			
- Eliminate all exterior surfaces (hypothetical)	7.6	-8.4	-60
HVAC (SYSTEMS) Variables			
Effect of thermostat set-up off-hours (baseline shuts down off-hours)			
- Cooling setpoint 81°F off-hours	15.1	13.3	-25

(continued)

TABLE S-1 (continued)

	Percentage Increase Over Baseline at 5 cfm/person		
	Cooling Coil Capacity	Annual HVAC Energy Cost[1]	Occupied Hours with RH > 60%
OA Rate = 20 cfm/person (continued)			
Effect of alternative HVAC systems (baseline is 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 unit/2 zones	16.8	4.9	+135
- 2 PTAC units/2 zones	7.9	9.1	-100
Effect of ducted return air (baseline is plenum return)			
- Air return via ducts	15.1	12.4	-33
Effect of cold-air distribution (baseline is PSZ/55°F min. supply T)			
- PSZ/42°F minimum supply T	22.2	6.2	-72
- PVAVS/42°F minimum supply T	23.3	1.6	-52
Effect of economizer modifications (baseline is T-controlled econo.)			
- No economizer	15.1	13.7	-85
- Enthalpy-controlled economizer	15.1	13.0	-55
Effect of cooling electric input ratio (baseline is EIR = 0.341)			
- Cooling EIR = 0.284	15.1	-1.8	-25
- Cooling EIR = 0.427	15.1	105.5	-25
Effect of cooling capacity and SHR (baseline is 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	27.4	15.6	-25

(continued)

TABLE S-1 (concluded)

	Percentage Increase Over Baseline at 5 cfm/person		
	Cooling Coil Capacity	Annual HVAC Energy Cost[1]	Occupied Hours with RH > 60%
OA Rate = 20 cfm/person (continued)			
Weather File Variables			
Effect of alternative weather files (baseline-Typical Meteorological Year)			
- Weather Year for Energy Calcs.	30.8	12.2	-8

Note:

[1] 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.

tion), the annual HVAC energy cost is $2,510, and the percentage 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 explanation for this effect is discussed later.)

The remainder of the entries show the predicted impacts (at 20 cfm/person) as each of the building and HVAC parameters is systematically varied from its baseline value. The percentage change with each parameter should be compared with the corresponding percentage increase experienced by the baseline system at 20 cfm/person, discussed in the preceding two paragraphs. If, for example, the percentage increase 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.

For some parameters, 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.

Section 5 of this report presents a detailed analysis of why each of these parameters is predicted to have the result that it does.

Table S-2 -- presented in the same format as Table S-1 -- lists those entries from Table S-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.

TABLE S-2

Building and HVAC Variables Creating the Greatest Reductions in Annual HVAC Energy Cost at 20 cfm/person (from Table S-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 kBtu/h	$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 °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 °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°F)	22.2	6.2	-72
Increase roof resistance to U_o = 0	15.1	6.7	-34

Similarly, Table S-3 lists those entries from Table S-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.

2.3.1 Parameters Creating the Greatest Reductions in HVAC Energy

Six of the ten parameters listed in Table S-2 are parameters 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 S-2.

Table S-4 summarizes the contribution of each of the individual heat sources to the annual HVAC energy consumption as predicted by the DOE-2.1E model, at ventilation rates of both 5 and 20 cfm/person.

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 S-2. (Only the 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 (ASHRAE, 1989b) to very efficient lighting (e.g., including daylighting) and more efficient (or more limited) equipment usage.

As shown in Table S-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 S-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 S-2. Of course, reducing occupant density will not generally be a viable option for reducing energy costs.

TABLE S-3

Building and HVAC Variables Creating the Greatest Reductions in Hours Above 60% RH at 20 cfm/person (from Table S-1)

	Cooling Coil Capacity	Annual HVAC Energy Cost[1]	Occupied Hours with RH > 60%
OA Rate = 5 cfm/person			
Baseline system with OA rate of 5 cfm/person	103.6 kBtu/h	$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
<u>Variables giving the greatest reduction in hours at elevated RH at 20 cfm/person, in descending order</u>			
Eliminate economizer	15.1	13.7	-85
Convert PSZ unit to cold-air distribution (minimum supply air T = 42 °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=0.16	13.3	6.1	-40
Increase roof resistance to U_o = 0	15.1	6.7	-34

Note: 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.

TABLE S-4

Approximate Contribution of the Various Heat Sources to the Annual HVAC Energy Consumption in the Baseline Building

Heat Source	Percentage Contribution to Annual HVAC Energy Consumption[1] 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

Note:

1 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.

Table S-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 S-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. The results shown in Table S-2 were computed when the baseline case -- single-pane glass with a moderate combination of tinting, coatings, or shading, having an overall heat transfer coefficient (U_o) of 0.94 Btu/h ft^2 F°, and a shading coefficient (S-C) of 0.55 -- was converted to double-pane glass with tinting and a highly reflective coating, having U_o = 0.32 Btu/h ft^2 F° and S-C = 0.16.

Finally, Table S-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 from 0.066 Btu/h ft^2 F° to zero) is the exterior surface parameter that provides the next greatest reduction in HVAC energy cost in Table S-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 better served directing those resources towards improved glazing and increased roof resistance, rather than towards increased wall or slab resistance.

The other four of the ten parameters listed in Table S-2 are parameters associated with the HVAC system: improving the cooling system efficiency; converting from a constant-volume PSZ system to a packaged variable-air-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 system 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/h per W), representing a typical efficiency -- to an EIR of 0.284 (EER = 12 Btu/h 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 S-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 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 S-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 are 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).

2.3.2 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 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 S-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 decrease in the number of elevated-RH hours occurs because, on average, the increased OA rate increases the sensible load. In addressing this increased load, the PSZ coils to 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 (Shirey and Rengarajan, 1996). These analysts 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 S-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 S-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 S-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 S-3 in the same order that they appeared in Table S-2.

The reason why these parameters have this effect on the number of elevated-RH hours is that -- 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 S-2) also provide the largest reduction in elevated-RH hours (Table S-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 S-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 air change per hour to zero has essentially no impact on the number of elevated-RH hours.

2.4 THE IMPACT OF STEPS TO REDUCE INDOOR HUMIDITY

As indicated previously, the DOE-2 model does not incorporate 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 (Shirey and Rengarajan, 1996) -- 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, e.g., using over-cooling with reheat, or using desiccants.

The RH results presented in Tables S-1 and S-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. As discussed in Section 2.3.2, the most practical conclusion apparent from Table S-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, over-cooling 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 (FDCA, 1993).

2.4.1 Thermostat Set-Up vs. System Shut-Down

According to the DOE-2 simulation, setting the thermostat up to 81 °F during off-hours, rather than shutting the system off, 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 shut-down *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 set-up, 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 set-up 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 shut-down to thermostat set-up would appear to be an important step for any building operator concerned about micro-biologicals.

The operating cost associated with set-up vs. shut-down is low, according to the DOE-2 predictions. As shown in Table S-1, switching to 81 °F thermostat set-up 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 shut-down 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.

The detailed analysis of thermostat set-up is presented in Section 6.2.2.

2.4.2 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.

It is re-emphasized that -- since DOE-2 does not include the moisture capacitance and re-evaporation phenomena -- DOE-2 underestimates the number of occupied hours where the RH exceeds 60%. Correspondingly, the computations here will necessarily underestimate the energy consumption and costs for a system that is designed to control these elevated-RH hours.

The results of this analysis of a reheat-based humidity control system are summarized in Table S-5, presented in the same format as Table S-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 S-3 and S-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

TABLE S-5

Effect of Humidity Control by Overcooling and Reheat

	Cooling Coil Capacity	Annual HVAC Energy Cost	Occupied Hours with RH > 60%
OA Rate = 5 cfm/person			
Baseline system (temperature control only) with OA rate of 5 cfm/person	103.6 kBtu/h	$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

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.

As indicated above, preventing economizer operation during elevated-RH hours reduces the number of elevated hours by 85% at an energy cost of $19/year. Considering that the humidity controller does prevent economizer operation under these conditions, it seems surprising that conversion to humidity control raises the energy cost penalty to $90/year simply to address the remaining 15% (only 6 hours/year). The reason is that, during many cool-weather hours, the humidity-controlled system -- as modeled by DOE-2 -- cools the office space down toward the heating set-point (70 °F) instead of the cooling set-point (75 °F), at a significant energy penalty. This seems to occur because, during cool weather, the moisture content of the office air (lb moisture per lb dry air) can sometimes be so close to 60% RH that the office temperature can determine whether the office is above or below the 60% set-point. The humidity controller tends to operate the offices at a lower temperature -- where a given moisture content would result in a higher RH -- perhaps as the result of occasional supply air over-cooling required to prevent elevated-RH hours. This seems to establish a cycle, whereby the simulated system has to continue to over-cool the supply air in order to maintain 60% RH in the cooler offices.

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.

A detailed discussion of the humidity controller analysis is presented in Sec. 6.3.

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SECTION 3

DESCRIPTION OF THE STUDY

3.1 THE SOFTWARE

The computer program used for this analysis was DOE-2.1E, the current version of the DOE-2 building energy simulation software, which evolved from efforts begun by the predecessor of the U. S. Department of Energy, and by the State of California, in the mid-1970's (York et al., 1981; U. S. Department of Energy, 1994a; Ayres and Stamper, 1995). The DOE-2 program is well documented and supported, and is among the most widely used software in the country for simulating building energy consumption and energy costs.

In the DOE-2 program, the building and HVAC system to be modeled are described by the user in an input file written in the Fortran-based "Building Description Language". A weather file describing the key climatic conditions of the location of the building is selected from one of three sources.

Appendix A presents an example of a DOE-2.1E input file used for the analysis presented in this report. This particular input file is for the "baseline" set of office and HVAC conditions, as discussed later.

The DOE-2 energy simulation involves four primary steps.

1) The LOADS calculation, a meticulous, hourly heat transfer and heat balance calculation which determines the loads in each space within the defined building. This calculation considers external loads (created by ambient temperature, solar angle, etc.), internal loads (resulting from occupants, lighting, equipment, etc.), and infiltration. The heat transfer resistance and heat capacitance of the building shell, and the characteristics of the glazing, for example, have significant impacts on the external loads.

2) The SYSTEMS calculation, which computes the hourly performance of the "secondary" HVAC equipment (the coils, air handler, and air distribution system) in response to the calculated loads, and the conditions (temperature, RH) which this equipment is able to maintain in the defined zones within the building. For an all-air HVAC system, for example, a central element of the SYSTEMS calculation is the computation of what mass of supply air that must be supplied at what temperature to each zone in order to maintain the user-specified temperature setpoint in that zone -- a computation which will be

impacted by the control method used by the particular HVAC system being modeled (modulation of supply air volume, modulation of supply temperature, or both). DOE-2 allows the user to select from among about two dozen different HVAC system types for modeling; for a given HVAC system type, users may allow the program to use default system design values (e.g., for coil efficiencies and for air handler performance curves), or may specify their own design values.

3) The PLANT calculation, which computes the hourly performance of the "primary" HVAC equipment (e.g., boilers and chillers) required to provide the coil temperatures, heat extraction rates, etc., determined under SYSTEMS. The PLANT is responsible for providing the total actual energy requirements of the building. This includes not only the fuel and electricity requirements for running the boilers and chillers, but also, e.g., any purchased electricity computed under SYSTEMS for operating the air handler, and calculated under LOADS for powering the building lighting and internal equipment.

4) The ECONOMICS calculation, where the total fuel and electricity requirements tallied under PLANT are converted to costs, based upon specified energy cost rates.

3.2 THE BASELINE BUILDING

A small office was the building type selected for this analysis.

Office space was selected because the U. S. population spends a substantial number of person-hours inside offices. Only residential structures would seem to have a clearly greater number of person-hours of occupancy.

Government statistics (U. S. Department of Commerce, 1994; U. S. Department of Energy, 1994b) indicate that approximately half of the office buildings in the U. S. are 5,000 ft^2 and smaller; almost 90% of the office buildings are 25,000 ft^2 and smaller. Expressed in terms of square footage of office space, something less than 10% of the total square footage in the U. S. is in offices of 5,000 ft^2 and less, and about one-third in offices of 25,000 ft^2 and less. On this basis, it was decided to focus this study on *small* offices, less than 25,000 ft^2. In addition to the population of small offices, another reason for selecting small offices is that they typically utilize packaged, direct expansion HVAC systems which offer the challenge of somewhat reduced flexibility for humidity control, compared to the built-up systems common in large office buildings. The specific office size selected for this study -- 4,000 ft^2, toward the lower end of the size range -- was selected for simplicity, and to be consistent with Shirey and Rengarajan.

The baseline office defined for this simulation is depicted in Figures 1 and 2, and further described in Table 1. The parametric analysis was conducted by systematically varying these baseline conditions, as discussed later. To model a hot, humid climate, this office is located in Miami, Florida. The office has 4,000 ft^2 of total space, and is subdivided into two 2,000 ft^2 zones (one zone in the front, and one in the rear). This office is located in the middle of a strip shopping mall, such as an office occupied by an insurance or travel agency. As a result, only the front and rear walls are exterior walls; the two side walls adjoin conditioned space occupied by neighboring tenants, and are assumed to be thermally neutral.

The baseline values summarized in Table 1 for the building design and operating parameters are presented more completely in the LOADS section of Appendix B. Appendix B also presents the rationale for the selection of these baseline values, and, as discussed later, indicates the alternative values that are considered for each of these parameters during this parametric analysis. The baseline values are incorporated into the LOADS portion of the baseline input file presented in Appendix A.

The baseline values for the building variables are believed to represent typical values for space in strip malls, based upon inspections of a number of malls. This office is similar to the one considered by Shirey and Rengarajan, although some of the baseline values have been changed from their values based upon the rationales presented in Appendix B, and in an effort to more precisely describe the building.

3.3 THE BASELINE HVAC SYSTEM

The baseline HVAC system is summarized in Table 2, with further definition in the SYSTEMS sections of Appendices A and B. The baseline system consists of two packaged single-zone (PSZ) rooftop units, one for each of the two zones in the office. These constant-volume units provide direct expansion cooling, and have electric resistance heating.

PSZ units -- which can be either unitary (rooftop or outside-the-wall) systems, as in this case, or split systems -- are a common choice for small offices. They can be among the simplest and least expensive of the central systems (Birdsall, 1995). Although other types of systems (such as packaged VAV systems) can also be considered, it is not uncommon for small offices, much larger than the one being considered here, to be conditioned by multiple PSZ units, one for each of the building's control zones. The use of two PSZ units in the baseline building here -- which results in each unit having approximately 5 tons of refrigeration capacity -- is consistent with the configuration used by Shirey and Rengarajan, and represents a commonly utilized capacity for this type of unit. From a practical standpoint, the use of two units will also improve the comfort in the offices; the north side of the building will have a much lower solar load than the south side, with the result that some occupants on one side could be too cool and/or some on the other side too warm if

TABLE 1

Baseline Building Design and Operating Conditions:
Small Office in Miami Strip Mall

Parameter	Baseline Value
Total size	40 ft frontage by 100 ft depth (4,000 ft^2).
Zones	Subdivided into 2,000 ft^2 front and rear zones (each 40 by 50 ft).
Height	9 ft ceiling height, with 4-ft-high plenum overhead for utilities and return air.
Orientation	Front faces north.
Building shade	A 10-ft overhang along the front at ceiling height (9 ft) to protect mall customers; a 3-ft high parapet along the front roof to visually shield rooftop equipment, protect service personnel.
Maximum occupancy	27 persons (150 ft^2/person).
Occupancy pattern	Full occupancy at 8-11am and 2-5pm weekdays; 80% at 11am-noon and 1-2pm; 40% at noon-1pm; 30% at 6-7am and 5-6pm; 10% at 6-8pm; zero at 8pm-6am weekdays, and at all times on weekends and holidays.
Lighting	1.8 W/ft^2.
Equipment (elec. outlets)	0.75 W/ft^2.
Lighting/equipment schedule	Full power at 7am-5pm weekdays; 30% at 6-7am and 5-6pm; 10% at 6-8pm; 5% at all other times on weekdays, weekends, and holidays.
Infiltration rate	0.1 air changes per hour.
Exterior walls	Concrete block with exterior stucco and interior insulation; overall heat transfer coefficient (including interior and exterior film resistances) U_o = 0.16 Btu/h ft^2 F°.
Window area	46% of front wall area, 20% of rear wall area.
Window glass type	Single pane with tinting, coating, and/or interior shading [shading coefficient = 0.55, U_o (incl. aluminum frame) = 0.94 Btu/h ft^2 F°].
Roof	Built-up roofing over mineral board insulation and metal deck; U_o (incl. film resistances) = 0.066 Btu/h ft^2 F°.
Floor slab	4-in. thick concrete slab with carpeting.

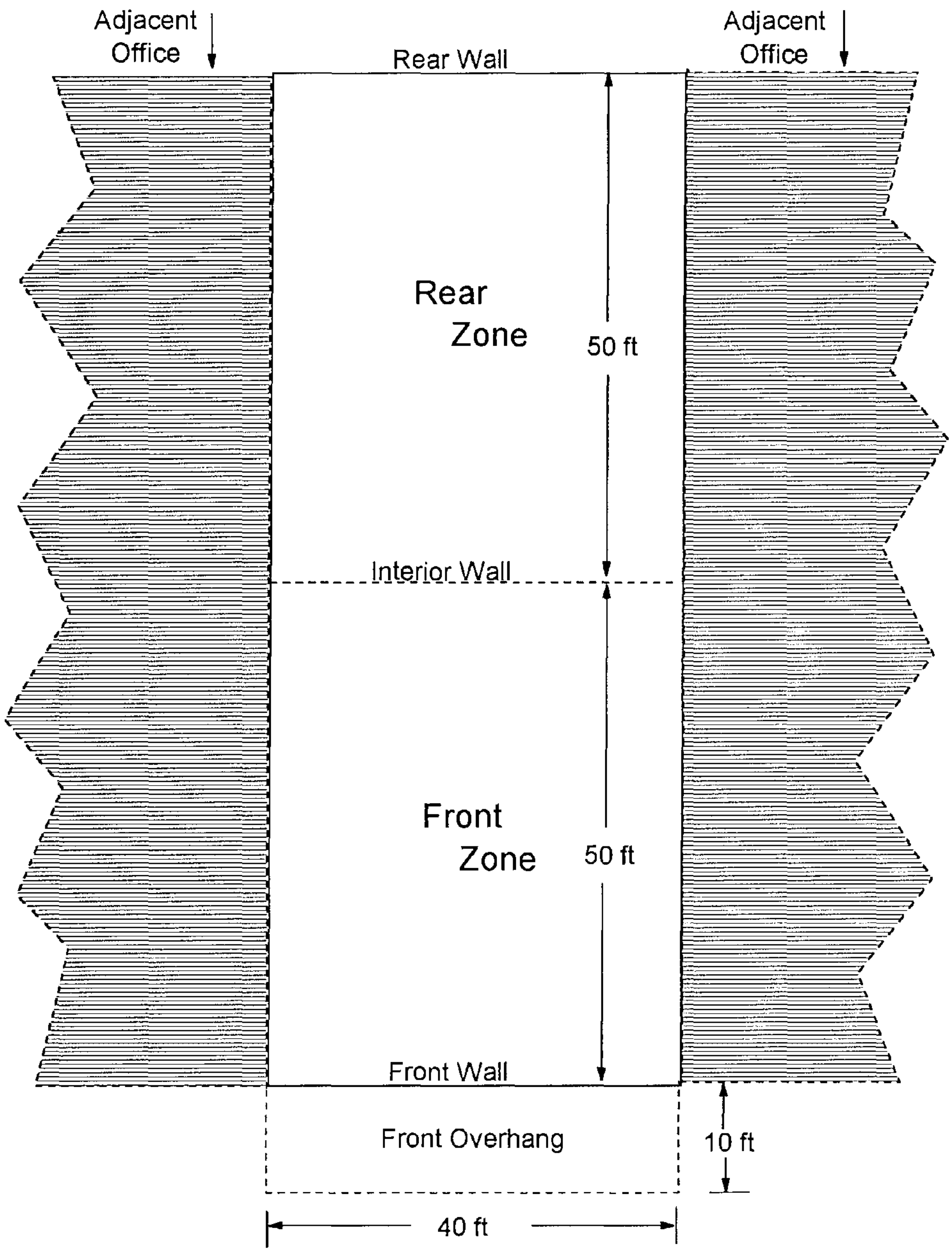

Figure 1. Floor plan for the baseline 4,000 ft^2 office in a Miami strip mall.

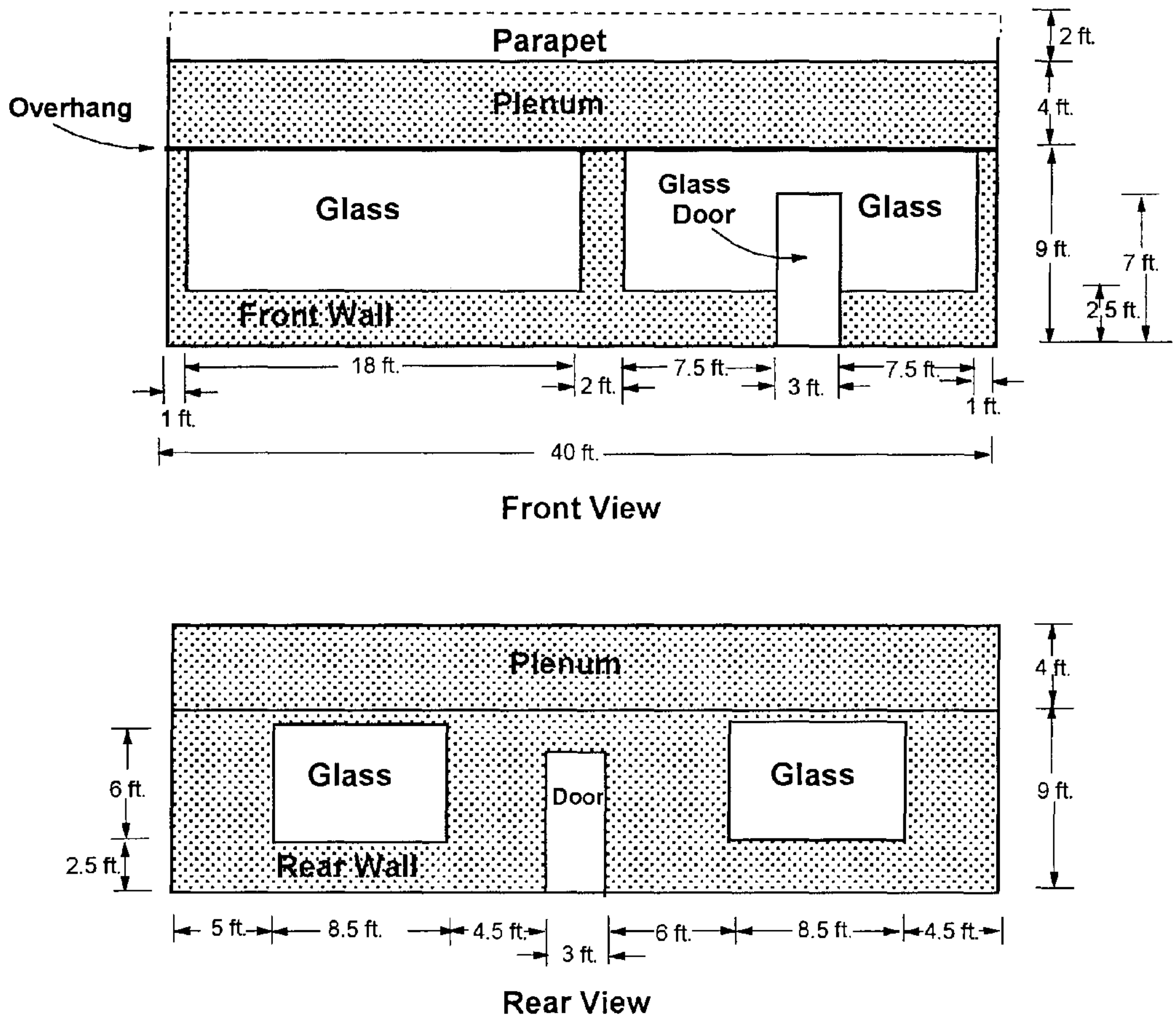

Figure 2. Front and rear views of the baseline office.

TABLE 2

Baseline HVAC Design and Operating Conditions:
Small Office in Miami Strip Mall

Parameter	Baseline Value
HVAC system type	Two rooftop direct expansion, constant volume, packaged single-zone (PSZ) units, one dedicated to each of the 2,000 ft^2 zones.
Office cooling setpoint	75°F during occupied hours (6am-7pm weekdays); cooling is shut off at all other times.
Office heating setpoint	70°F during occupied hours; heating is set back to 55°F at all other times.
OA ventilation rate	5 (and, as warranted, 20) cfm/person
Minimum supply air temperature for cooling	55°F
Maximum supply air temperature for heating	100°F
Maximum humidity in offices	Not set.
Economizer	Economizer present, controlled by indoor vs. outdoor temperatures.
Cooling capacity and sensible heat ratio	Calculated by program.
Cooling efficiency	Electric input ratio (EIR) = 0.341 Btu/h electric input per Btu/h cooling output [corresponding to energy efficiency ratio (EER) = 10 Btu/h cooling output per W electric input].
Heating efficiency	EIR = 1.0 Btu/h electric input per Btu/h cooling output (electric resistance heating).
Return air method	Return via overhead plenum (unducted).

one attempted to condition the entire 4,000 ft^2 with a single, 10-ton PSZ system. Technically, though, individual PSZ units are commercially available in capacities greater than 10 tons.

3.4 THE "PLANT"

Packaged systems, such as PSZ systems, incorporate their "primary" and "secondary" HVAC equipment into a single unit. The classical "primary" equipment required for built-up systems, addressed in the PLANT portion of DOE-2 -- e.g., boilers, chillers, and cooling towers -- are absent. These components are replaced by, e.g., electric resistance heating, an electric-driven compressor, and an air-cooling condenser fan within the packaged unit, defined under SYSTEMS (Section 3.3 above).

As a result, the PLANT portion of the DOE-2 computation in this modeling of packaged systems may be viewed simply as the supplier of purchased electricity for: the lighting and office equipment in the building; the compressor, outdoor condenser fan, electric resistance heaters, and the air handling unit associated with the packaged HVAC system; and the electric domestic hot water (DHW) heater assumed for this office.

3.5 ENERGY COSTS

The electric rate structure used for this analysis was the GSD-1 schedule for Florida Power and Light Company, which serves the Miami area. This is the rate structure used by Shirey and Rengarajan.

This structure involves a basic energy charge of $0.0473 per kWh, plus a peak demand charge of $9.96/kW for each kilowatt above 10 kW. There is no demand charge for the first 10 kW.

3.6 WEATHER DATA

The Typical Meteorological Year (TMY) weather file for Miami was used as the baseline file, for essentially all of these calculations. For comparison, one run was made with the Weather Year for Energy Calculations (WYEC) file for Miami.

The DOE-2 user has three choices of weather files for Miami (each of which provides one year of hourly data): the Test Reference Year (TRY) files and the TMY files, available from the National Climatic Data Center, and the WYEC files, available from ASHRAE. TRY files provide data from a single, selected year; as such, these files might result in a somewhat greater variation between predicted and observed energy consumption, since no single year is likely to be "typical" during all 12 months.

TMY files improve on the TRY files by assembling 12 "typical" months from different years (e.g., a typical January from one year, a typical February from another). WYEC files attempt to provide even further improvement by assembling "weather events" (of a duration shorter than a month) from different years (e.g., a typical first two weeks of July from one year, a typical second two weeks from another).

3.7 PARAMETRIC VARIATIONS FROM THE BASELINE VALUES

The baseline values presented in Tables 1 and 2 (and in Appendix B) for the building and HVAC design and operating parameters were selected because they were felt to be reasonably typical for an office of the type being considered.

For this sensitivity analysis, alternative values were selected for most of these parameters, varying from the baseline values. These alternative values are presented in Appendix B. These alternative values commonly include a lower value and a higher value, bracketing the baseline value. The range defined by these alternative values usually define the broadest reasonable range that might be expected in practice; for example, the EER of the cooling coils is varied through a range extending from 8 to 12 Btu/h cooling output per W electric input. In some cases, extreme values are selected in order to demonstrate maximum effects (for example, perfect total insulation of the walls and roof, or the total absence of glazing).

In the semantics used in Appendix B, the "low" alternative value shown in the appendix is the value expected to result in reduced energy consumption, and the "high" value is that expected to result in increased consumption.

3.8 STRATEGY FOR THE CALCULATIONS

As the first step, runs were made with the baseline building and HVAC system to define baseline HVAC performance, HVAC capacity requirements, energy consumption, and energy costs. Baseline runs were made at OA ventilation rates of both 5 and 20 cfm/person, to estimate the impact of increased ventilation. The baseline results are presented in Section 4 of this report.

As the next steps, each of the individual building and HVAC parameters was varied in turn, through the values discussed in Section 3.7, at a ventilation rate of 5 cfm/person. For each parametric variation, the predicted incremental impacts on performance, capacity requirements, and energy use were calculated. Where a given parameter appeared to have a significant effect, or where otherwise of interest, the calculation for a given parametric variation was repeated at a ventilation rate of 20 cfm/person. The results of this sensitivity analysis are reported in Section 5.

As the final step, calculations were made to assess the practical ability of packaged HVAC systems to control RH in this office, and the approximate energy costs involved in doing so as ventilation rate is increased from 5 to 20 cfm/person. These calculations were made recognizing the potential uncertainties in using DOE-2 to estimate indoor RH in humid climates, as discussed in Section 1.4. The results of this assessment of RH control are discussed in Section 6.

In the sensitivity analysis calculations reported in Sections 4 and 5, the HVAC capacities reported in the tables are those calculated by the computer program as being necessary to meet the load. [Only in Section 5.17, where the effects of capacity are explored, were HVAC capacities and sensible heat ratios (SHRs) specified for the runs, simulating commercially available equipment.] The computed capacities never correspond exactly to the capacities available in commercial cooling units, which are commonly marketed in increments of one-half to one ton of refrigeration (6 to 12 kBtu/h). Thus, a variation in a building or HVAC parameter that increased the computed cooling capacity requirement by some fraction of a ton of refrigeration could, in practice, necessitate the installation of a unit at the next highest tonnage increment. Installation of a higher-capacity unit would impact the system operation in a number of ways, impacting system performance and energy use.

This issue could have been addressed by specifying, in the DOE-2 input file for each run, the capacity and SHR of the commerically available unit that would be needed for that parametric variation, rather than accepting the program's computed (fractional tonnage) default value. It was decided to use the program-calculated default capacities because that approach was felt to provide a clearer measure of how the parametric variations impact capacity requirements.

HVAC performance is measured in terms of the predicted percentage of total hours throughout the year that the office space is undercooled, and in terms of the percentage of the occupied hours when the RH exceeds 60%. With the zone design temperatures specified in Appendices A and B, and with the decision discussed above to allow the program to calculate the cooling capacities, the percentage of hours undercooled is generally less than 0.5%, and the percentage of occupied hours calculated as having RH > 60% is generally about 1%.

SECTION 4

BASELINE RESULTS

4.1 SUMMARY TABLES

The results of the computations for the baseline building and HVAC system (as defined in Tables 1 and 2) are shown in Table 3, for OA ventilation rates of both 5 and 20 cfm/person.

As discussed in the previous section, system *performance* is summarized in terms of the percentage of hours undercooled and at RH values >60% (averaged over the two zones to represent the entire office space). System *design* requirements are summarized in terms of the program-calculated HVAC cooling capacity (summed for the two PSZ units combined). Building and system *operating* requirements are summarized in terms of the energy consumption and energy costs for the HVAC system by itself, and for the total building (including the HVAC system, the lighting and office equipment, and the DHW heater).

To show how the total electric energy consumption in the office is distributed, Table 4 breaks down the total annual building energy consumption figures in Table 3, according to the various end uses.

4.2 DISCUSSION

<u>End uses of energy</u>. Two of the significant conclusions from Table 4 are:

1) The HVAC system consumes something less than half of the total power required by the office, a fraction that remains essentially unchanged as the OA ventilation rate is increased from 5 to 20 cfm/person. The largest single power consumer -- requiring half of the office's consumption -- is the lighting and office equipment (computers, copiers, fax machines, etc.).

2) In the Miami climate, power consumption for heating is almost negligible in this office. The bulk of the HVAC power consumption is for sensible and latent cooling. For this reason, the HVAC capacity addressed in Table 3 (and in subsequent tables) is the cooling capacity.

<u>HVAC capacity requirements</u>. As shown in Table 3, the combined cooling capacity of the two PSZ units would have to increase by 15% as the OA ventilation

TABLE 3

Results from DOE-2.1E Modeling of
the Baseline Small Office in Miami

Output Variable	Value of Output Variable when: OA = 5 cfm/person	OA = 20 cfm/person	Increase caused by increase in OA from 5 to 20 cfm/person Variable Units	%
Total required cooling capacity[1] (kBtu/h)	103.558	119.193	15.635	15.1
Annual HVAC energy consumption[1] (kWh)	26,145	29,390	3,245	12.4
Annual HVAC energy costs[1] ($)	2,510	2,835	325	12.9
Annual building energy consumption[1] (kWh)	60,161	63,406	3,245	5.4
Annual building energy costs[1] ($)	4,273	4,598	325	7.6
% of all hours (8760 hr/year) undercooled[1]	0.4	0.4	~0	~0
% of occupied hours (3276 hr/year) when RH>60%[1]	1.2	0.9	-0.3	~0

1 HVAC capacity, energy consumption, and energy cost numbers represent the sum for the two PSZ units combined; building energy figures include both zones. Zone undercooling and humidity performance numbers represent the total 4,000 ft^2 of office space, i.e., the average for the two zones.

TABLE 4

Annual Electric Energy Consumption in the Baseline Small Office, Broken Down According to End Use

End Use	Annual Electric Energy Consumption for that End Use (kWh)		Percentage of Total Annual Consumption in the Office	
	For OA=5 cfm/person	For OA=20 cfm/person	For OA=5 cfm/person	For OA=20 cfm/person
HVAC System				
Cooling (compressor and condenser fan)	18,778	22,001	31	34
Heating (electric resistance)	30	52	1	1
Air handling fan	7,328	7,328	12	11
Auxiliaries (comp. crankcase heater)	9	9	~0	~0
Subtotal - HVAC	26,145	29,390	44	46
Other than HVAC				
Lighting and office equipment	30,429	30,429	50	48
Domestic hot water heater	3,587	3,587	6	6
Subtotal - Non-HVAC	34,016	34,016	56	54
TOTAL FOR OFFICE	60,161	63,406	100	100

rate is increased, if similar temperature and RH performances are to be maintained in the zones.

Given that one ton of refrigeration corresponds to 12 kBtu/h, and that off-the-shelf packaged units are typically marketed in ton or half-ton increments, one would likely choose a pair of units adding up to at least 9 tons (108 kBtu/h) for this office if the ventilation rate were to be 5 cfm/person, and at least 10 tons (120 kBtu/h) if the rate were to be 20 cfm/person. As shown later in Section 5, specification of those higher capacities in the DOE-2 calculation (rather than using the program-calculated values of 103.558 and 119.193 kBtu/h) would have notably improved the temperature control performance shown in Table 3, would have had almost no effect on the RH performance, and would have slightly increased energy consumption and costs.

<u>Energy consumption and costs</u>. Table 3 indicates that increasing the ventilation rate increases electric consumption by 3,245 kWh per year. (As shown in Table 4, this increase results essentially entirely from an increase in HVAC cooling costs, as expected.) Expressed as a percentage of HVAC energy consumption, this 3,245 kWh increase represents an increase of over 12%; expressed as a percentage of total office consumption, it represents an increase of over 5% [consistent with the results of Eto (1990) in modeling small offices in Miami]. Since HVAC energy consumption is something less than half the total office consumption (in this baseline case and in the parametric variations covered in Section 5), the *percentage increase* in HVAC consumption will always be something greater than twice the percentage increase in total office consumption.

The corresponding energy cost increase resulting from increased ventilation is $325 per year. This amount corresponds to a 12.9% increase in the HVAC energy costs, and a 7.6% increase in total office energy costs. The percentage increase in energy costs (for the baseline and in Section 5) is consistently greater than the percentage increase in consumption, due to the effect of the electric demand charge.

<u>Performance</u>. As shown in Table 3, there is no significant change in the percentage of hours undercooled as the ventilation rate increases, remaining at 0.4% of the 8,760 hours in the year. This is not surprising. The computer program designs the capacity of each PSZ unit to maintain the temperature in the zone that it is conditioning. Accordingly, at 20 cfm/person, the program has increased the capacity of each unit as necessary to meet the additional load created by the increased OA flow.

The percentage of occupied hours >60% RH also remains almost unchanged, at about 1%, as the ventilation rate increases. As a matter of fact, according to these DOE-2 computations, there is a small *reduction* in the number of elevated-RH hours with the increase in OA ventilation rate -- from 1.2% of the 3,276 occupied hours (40 hours per year) to 0.9% (29 hours per year). While this decrease in elevated-RH hours is tiny, it is useful to note conceptually why the program would predict a decrease.

Elevated-RH hours tend to occur during the first several hours after startup on cool, humid mornings. This occurs because the cooling coils are operating at a small fraction of capacity (or are off altogether) due to the low sensible load during those cool hours, so that there is little or no latent cooling. Increasing the OA rate to 20 cfm/person causes, on average, an increase in the sensible load during these morning hours. Since constant-volume PSZ systems handle load changes by modulating supply air temperature, this increased sensible load triggers a reduction in the PSZ cooling coil temperature. The reduction in coil temperature increases the latent cooling provided by the system at 20 cfm/person. The increased latent cooling can more than offset the increased latent load caused by the increased influx of humid OA. Table 3 indicates that the increase in latent cooling exceeds the increase in latent load during cool morning hours sufficiently to eliminate 11 elevated-RH hours when the OA is increased. (This effect is discussed in greater detail in Section 5.12.2.)

As discussed in Section 6, Shirey et al. (1995), and Shirey and Rengarajan (1996), suggest that, in fact, RH levels could increase at increased OA flows, rather than decreasing (or remaining about the same) as predicted here. According to their calculations, RH increases resulting from moisture capacitance and condensate re-evaporation effects at 20 cfm/person would more than compensate for the RH decreases due to the two effects discussed above. There is no way to independently verify the Shirey and Rengarajan results here using the DOE-2 model.

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SECTION 5

THE EFFECTS OF PARAMETRIC VARIATIONS

5.1 BUILDING ORIENTATION

In the baseline configuration, the front of the office was facing north. The predicted effects of orientation in the other directions are summarized in Table 5.

For this comparison, the effects of building orientation are shown as incremental changes from the baseline configuration with an OA ventilation rate of 5 cfm/person. Runs at 20 cfm/person are shown only for the cases where the building is facing north (since that is the baseline direction), and where it is facing south (since that is the direction which, at 5 cfm/person, resulted in the greatest variation in energy consumption and cost relative to the baseline).

5.1.1. The Effect of Orientation at 5 cfm/person

The first three rows in Table 5 (for the ventilation rate of 5 cfm/person) show that building orientation has only a minor impact on energy consumption in this building.

Among the four compass directions, a *south* orientation results in the lowest required cooling capacity and energy consumption/cost. When the front office faces south, the program predicts a 3% reduction in HVAC energy consumption and costs, relative to the north-facing baseline. Review of the program reports confirms that the reason for this result is that both the window solar load and the wall conduction load for the building are distinctly lower for the south orientation, relative to any other orientation. With the south orientation, the 10-foot front overhang is on the south side, providing exterior shading in the direction from which the maximum solar gain would be received. The unshaded rear glass and wall is on the north side, and hence will be shaded by the building itself. As a result, annual cooling (compressor and condenser) power consumption is the lowest for any of the four orientations. Also, with the shading, peak loads are lower with the result that system flows are lower; hence, power consumption by the air handling fan is reduced.

The greatest capacity and energy requirements result when the front of the building is facing *east* (with HVAC energy consumption and costs increasing about 1% relative to the baseline). East and west are the two orientations providing the greatest window solar load, since the rising and setting suns, respectively, have a low incident angle which allows the sun to hit the highly glazed front of the building

TABLE 5

Effect of Building Orientation:
Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values: Bldg faces **north**, OA=5 cfm/p (absolute values)[2]	103.558[2]	--	26,145[2] (HVAC)	--	--	2,510[2] (HVAC)	--	--	0.4	1.2
			60,161[2] (building)	--	--	4,273[2] (building)	--	--		
OA ventilation rate = 5 cfm/person										
Bldg faces **south**	-0.196	-0.2	-820	-3.1	-1.4	-73	-2.9	-1.7	0.4	1.3
Bldg faces **east**	5.050	4.9	186	0.7	0.3	31	1.2	0.7	0.5	1.2
Bldg faces **west**	1.276	1.2	24	0.1	~0	4	0.2	0.1	0.7	1.2
OA ventilation rate = 20 cfm/person										
Bldg faces **north**	15.635	15.1	3,245	12.4	5.4	325	12.9	7.6	0.4	0.9
Bldg faces **south**	16.009	15.5	2,451	9.4	4.1	256	10.2	6.0	0.4	0.9

1 Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

2 These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

underneath the overhang; and, during latter or initial part of the day, respectively, the sun will be hitting the unshaded rear windows. In fact, for this building, the west orientation results in a slightly higher window solar load and wall conduction load than does the east orientation. However, the east orientation has the somewhat higher peak load -- probably because it receives the early morning solar gain on the front glazing on summer Monday mornings when peak loads will occur. As a result, the east orientation has somewhat higher air flows, and hence higher power consumption by the air handling fan, causing this orientation to have an annual power consumption (and a cooling capacity requirement) slightly greater than that for the west orientation.

For all four orientations, the percentage of hours undercooled remains at about 0.5%, and the percentage of occupied hours >60% RH remains at about 1.2%.

5.1.2 The Effect of Orientation on Increased Ventilation Rates

The last two rows in Table 5 show the effects of north vs. south orientation at the increased OA ventilation rate of 20 cfm/person.

At 5 cfm/person, the south orientation results in a reduction of about 800 kWh/year and $70/year, relative to the north orientation. This same differential exists between the two orientations at 20 cfm/person. For example, increasing the ventilation rate in the north orientation results in an increased energy consumption of 3,245 kWh/year relative to the north-facing/5 cfm baseline; increased ventilation in the south orientation results in an increase of only 2,451 kWh/year from the north-facing/5 cfm baseline, 794 kWh less.

Stating the above point in another way, the difference in annual energy consumption between the south-facing/20 cfm case and the south-facing/5 cfm case is (2,451) - (-820) = 3,271 kWh. This is essentially identical to the 3,245 kWh difference between the north-facing/20 cfm case and the north-facing/5 cfm case. For a given climate and HVAC system, the incremental cost of treating an additional 15 cfm/person of outdoor air will not be heavily dependent on the orientation of the building. Therefore, if one had happened to choose a south-facing office rather than a north-facing office to assess the effects of increased ventilation, this choice would not have significantly affected the conclusions.

For either the north or south orientation, an increase to 20 cfm/person would require approximately a 16 kBtu/h (1.3 ton) increase in cooling capacity. In either case, from a practical standpoint (with PSZ units assumed to be available only in 0.5-ton increments), this would translate into the need to install units totalling 10 tons of cooling capacity rather than 9 tons.

5.2 BUILDING SHADE

In the baseline configuration, the front of the office was shaded with a 10-foot overhang at ceiling height along its entire frontage, whereas the rear was not shaded at all. Table 6 shows the effects for the cases where: a) the rear is also shaded, with overhangs that extend 6 ft outward at ceiling height over the rear windows and door; and b) there are no shading overhangs at all, either on the front or the rear.

5.2.1 The Effect of Building Shade at 5 cfm/person

From Table 6, adding shading on the rear decreases HVAC energy consumption and costs by 2½ to 3%. The small reduction in cooling capacity requirements (0.713 kBtu/h) would result in no change in the capacity of the commercial units that would have to be purchased (9 tons total), since the units will likely have to be purchased in 0.5-ton increments. This rear shading is having its maximum effect in this baseline orientation, which has the rear facing south. Computations confirm that the energy consumption and cost savings resulting from the addition of the rear shading would have been slightly less had the rear been facing one of the other compass directions, where the solar gain through the *un*shaded rear windows would be less.

Removing the front shading from the baseline case increases HVAC energy consumption and costs by 2½ to 3%. The 5.213 kBtu/h increase in cooling capacity requirements could result in the need to increase the installed capacity by 0.5 ton, to 9.5 tons. Deletion of the front shading has its minimum effect in the baseline orientation, since the front is facing north. The large amount of front glazing is thus receiving substantial shading from the building itself even in the absence of the front overhang. If the front office were facing south, the penalty for deleting the front overhang would jump from 667 kWh and $69 per year, shown in Table 6 for the north orientation, to 1,641 kWh and $159 per year, an increase of about 6% over the baseline.

The changes in building shading have essentially no impact on system performance, in terms of hours undercooled or at elevated RH. Again, major changes in these output parameters would not be expected, especially for the hours undercooled, since the program should be sizing the system capacity in each case to achieve consistent thermal performance.

5.2.2 The Effect of Shade on Increased Ventilation Rates

From Table 6, in a building where all overhangs are deleted, the added energy consumption resulting from increasing the ventilation rate from 5 to 20 cfm/person in the north-facing building is (3,950 - 667) = 3,283 kWh/year, and the added cost is ($401 - $69) = $332/year. In the worst-case situation where the overhangs were deleted in a south-facing building, these differentials from increased ventilation would

TABLE 6

Effect of Building Shading: Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values: 10-ft overhang in front, none in rear, OA=5 cfm/p (absolute values)[2]	103.558[2]	--	26,145[2] (HVAC) 60,161[2] (building)	-- --	-- --	2,510[2] (HVAC) 4,273[2] (building)	-- --	-- --	0.4	1.2
OA ventilation rate = 5 cfm/person										
Add 6-ft overhang over **rear** door and windows	-0.713	-0.7	-734	-2.8	-1.2	-65	-2.6	-1.5	0.3	1.2
Delete all overhangs (front and rear)	5.213	5.0	667	2.6	1.1	69	2.7	1.6	0.4	1.2
OA ventilation rate = 20 cfm/person										
Delete all overhgs	21.718	21.0	3,950	15.1	6.6	401	16.0	9.4	0.4	0.9

1 Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

2 These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

be 3,175 kWh and $311. These impacts of increased ventilation are similar to those shown for the baseline building in Table 3, and for different orientations in Section 5.1.2. Again, for a given climate and HVAC system, the incremental energy to treat an additional 15 cfm/person of OA does not vary significantly as a function of the building overhang configuration. Even if the modeler had happened to choose the worst-case situation of a south-facing office with no overhang, this choice would not have significantly affected the calculated effects of increased ventilation.

However, the calculations do show that -- especially when the building is facing south -- such shading can have a meaningful impact in reducing energy consumption (in addition to protecting clientele from the weather). In a south-facing office *without* an overhang, increasing the ventilation rate from 5 to 20 cfm/person would increase annual energy consumption by 3,175 kWh at a cost of $311, as indicated above. But if a front overhang were installed at the same time that the ventilation rate were increased, the net increased annual consumption would be only 810 kWh at a cost of $97. Thus, adding an overhang in that case could nominally recover more than two-thirds of the energy penalty resulting from the increase in OA rate.

In terms of practical cooling capacity, the addition of rear overhangs results in no effective change in the required increase in capacity when ventilation rate is increased from 5 to 20 cfm/person; units totalling 9 tons must be increased by 1 ton, to 10 tons, with or without the rear overhang. If the front overhang is deleted, increased ventilation would again result in a capacity increase of 1 ton, except in this case, the increase would be from 9.5 tons to 10.5 tons (whether the building were facing north or south). For the hypothetical case suggested in the previous paragraph -- where a south-facing building with no front overhang and a ventilation rate of 5 cfm/person is increased to 20 cfm/person and an overhang added -- the required practical capacity increase would be only 0.5 ton, from 9.5 to 10 tons.

5.3 OCCUPANT DENSITY

In the baseline configuration, the occupant density was that specified by ASHRAE 62-1989 (7 persons per 1,000 ft^2, or approximately 150 ft^2/person), corresponding to about 27 persons in the 4,000 ft^2 office. Occupant density is particularly important in the calculations, because it not only determines the sensible and latent load from people, but it also significantly impacts the actual cfm of outdoor air that is brought into the building in response to the specified cfm per person (and hence the sensible and latent load for conditioning the OA).

Table 7 shows the calculated effects of: a) decreasing occupancy by 50%, to 13 persons (300 ft^2/person); and b) increasing occupancy by 50%, to 40 persons (100 ft^2/person).

TABLE 7

Effect of Building Occupancy:
Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values:	103.558[2]	--	26,145[2]			2,510[2]			0.4	1.2
150 ft^2/person,			(HVAC)	--	--	(HVAC)	--	--		
OA = 5 cfm/p			60,161[2]			4,273[2]				
(absolute values)[2]			(building)	--	--	(building)	--	--		
OA ventilation rate = 5 cfm/person										
300 ft^2/person	-9.171	-8.9	-2,151	-8.2	-3.6	-207	-8.2	-4.8	0.3	1.0
100 ft^2/person	8.387	8.1	2,152	8.2	3.6	204	8.1	4.8	0.3	1.2
OA ventilation rate = 20 cfm/person										
300 ft^2/person	-0.252	-0.2	-436	-1.7	-0.7	-37	-1.5	-0.9	0.4	0.8
100 ft^2/person	30.678	29.6	6,703	25.6	11.1	674	26.9	15.8	0.4	0.9

[1] Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

[2] These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

5.3.1 The Effect of Occupancy at 5 cfm/person

The first two rows in Table 7 show that, at 5 cfm/person, decreasing and increasing the number of occupants by 50% have meaningful, equal effects in opposite directions, as expected. HVAC energy consumption, HVAC energy cost, and cooling capacity are each decreased by about 8% by decreasing the number of occupants, and increased by 8% by increasing occupancy.

At 300 ft^2/person, the predicted percentage of hours above 60% RH decreases slightly relative to the baseline, due to the reduced latent load from people and the reduced flow of moisture-containing OA.

In the extreme, if the number of occupants were decreased to zero, HVAC energy consumption would be reduced by approximately 15%. Stated another way, roughly 15% of the HVAC energy consumption in the baseline building at 5 cfm/person is due to the sensible and latent load contributed by the occupants.

5.3.2 The Effect of Occupancy on Increased Ventilation Rates

The last two rows in Table 7 show that increasing the ventilation rate to 20 cfm/person at 300 ft^2/person results in HVAC energy consumption, costs, and cooling capacity requirements that are slightly less than the 150 ft^2/person baseline at 5 cfm/person. At 100 ft^2/person, increasing the ventilation rate results in more than a 25% increase in HVAC energy and capacity over the baseline.

At 300 ft^2/person, increasing the OA ventilation from 5 to 20 cfm/person increases energy consumption by (-436) - (-2,151) = 1,715 kWh, and increases energy costs by (-37) - (-207) = $170. By comparison, at 100 ft^2/person, the increase from 5 to 20 cfm/person results in increases of 4,551 kWh and $470. Occupancy is one of the few parameters that causes these incremental increases from increased ventilation to vary dramatically from the baseline increases shown in Table 3 (3,245 kWh and $325).

In terms of required cooling capacity, at 300 ft^2/person, the increase in ventilation rate would necessitate an increase of 1 ton in the practical installed capacity, from 8 to 9 tons. This is the same 1-ton increment as required by the baseline, except in that case, the increase is from 9 to 10 tons. At 100 ft^2/person, the increase is 1.5 tons, from 10 to 11.5 tons.

This result underscores the obvious conclusion: The fewer the number of people in the space, the lower the cost of increasing the flow of OA per person. Clearly, the assumed occupant density can have a significant impact on the energy penalty that a modeler would predict from increasing ventilation from 5 to 20 cfm/person.

On this same basis, the use of demand-controlled ventilation (DCV) could conceptually reduce the incremental penalty of increased ventilation. DCV would effectively reduce the number of occupants used by the HVAC system in the OA cfm per person calculation during periods of reduced occupancy.

5.4 LIGHTING AND OFFICE EQUIPMENT POWER CONSUMPTION

In the baseline building, overhead lighting was assumed to consume 1.8 W/ft^2 and office equipment 0.75 W/ft^2 (for a total of 2.55 W/ft^2), consistent with ASHRAE 90.1-1989. Increased lighting and equipment power consumption in the building would have two effects: a) it would increase the sensible heat load that would have to be addressed by the HVAC system; and b) it would increase the non-HVAC component of building power consumption, causing the incremental energy penalty of increased ventilation to seem to be less when expressed as a percentage of total building energy.

Table 8 shows the estimated effects of: a) decreasing the lighting + equipment power consumption to 1.0 + 0.5 = 1.5 W/ft^2; and b) increasing this consumption to 2.25 + 1.75 = 4.0 W/ft^2. The rationales for these selections are summarized in Appendix B. The lower value for power consumption would correspond to the use of efficient modern lighting, daylighting, and Energy Star appliances.

An increase in the lighting/equipment power consumption causes an increase in the HVAC energy consumption and cost estimates (due to increased sensible heat load), plus a much larger increase in total building energy consumption and costs (due to the increased HVAC energy plus increased consumption by the lights and equipment). In Table 8, the increases/decreases shown for energy consumption and costs include only the *HVAC* energy variations, not those for the total building.

5.4.1 The Effect of Lighting/Equipment Power at 5 cfm/person

The results in Table 8 for the two runs at 5 cfm/person show dramatic decreases and increases in HVAC energy and capacity requirements as lighting/equipment power requirements are decreased/increased. These large decreases/increases demonstrate the importance of this load on HVAC operation. No other building or HVAC parameter studied here has had as great an impact on HVAC energy consumption, HVAC energy costs, and required cooling capacity ($\pm$ 14-27%) at the low ventilation rate of 5 cfm/person.

The range of lighting/equipment power consumptions considered here (1.5 to 4.0 W/ft^2) admittedly covers the extremes, causing the size of the observed variations; modelers generally do not use these extreme values. However, the selected power consumptions do vary between published modeling studies from about 2.5 to 3.5 W/ft^2, a differential that could still create a significant difference between estimates. This would appear to be a variable which needs to be selected with some care.

In the extreme, if the lighting wattage were reduced to zero, HVAC energy consumption would be reduced by approximately 39%. If equipment wattage were reduced to zero, HVAC energy would be reduced by about 17%. Thus, sensible heat from the lighting and equipment is responsible for approximately 39% and 17%, respectively, of the total HVAC energy consumption in the baseline building at 5 cfm/person -- the largest contributions from any individual source.

TABLE 8

Effect of Lighting and Office Equipment Power Consumption: Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values:	103.558[2]	--	26,145[2]			2,510[2]			0.4	1.2
2.55 W/ft^2,			(HVAC)	--	--	(HVAC)	--	--		
OA=5 cfm/p			60,161[2]			4,273[2]				
(absolute values)[2]			(building)	--	--	(building)	--	--		
OA ventilation rate = 5 cfm/person										
1.5 W/ft^2	-14.695	-14.2	-5,020[3]	-19.2	-8.3	-437[3]	-17.4	-10.2	0.3	2.0
4.0 W/ft^2	20.961	20.2	7,038[3]	26.9	11.7	632[3]	25.2	14.8	0.4	0.8
OA ventilation rate = 20 cfm/person										
1.5 W/ft^2	0.404	0.4	-1,958[3]	-7.5	-3.3	-127[3]	-5.1	-3.0	0.3	1.3
4.0 W/ft^2	37.350	36.1	10,410[3]	39.8	17.3	971[3]	38.7	22.7	0.5	0.6

1 Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

2 These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

3 Increase (or decrease) in *HVAC* energy and costs; the increase (or decrease) in *total building* energy and costs is much greater.

The $\pm$ 17-27% variation in energy consumption and costs from baseline values shown in Table 8 is for *HVAC* energy. If one considered the impacts on *total building* energy consumption and costs -- considering the impact of lighting/equipment power on non-HVAC, as well as HVAC, energy requirements -- the results are even more dramatic. At a ventilation rate of 5 cfm/person, varying lighting/equipment power between 1.5 and 4.0 W/ft^2 causes total building energy consumption and cost to vary by an impressive **$\pm$ 29-50%** from the baseline (2.55 W/ft^2) total building energy values.

From the standpoint of performance, it is noted that the percentage of occupied hours above 60% RH is slightly higher than the baseline when lighting/equipment power drops to 1.5 W/ft^2, and slightly lower than the baseline when this value rises to 4.0 W/ft^2. This effect occurs because the higher lighting/equipment power consumption results in a consistently higher sensible heat load on the HVAC system. As discussed in Section 4.2, increased sensible load tends to increase the latent removal of this system -- including during cool mornings after startup, when elevated-RH occupied hours occur. Thus, higher lighting/equipment power consumption results in fewer elevated-RH hours. The lower lighting/ equipment power case has reduced sensible loads, and hence *increases* the number of elevated-RH hours.

5.4.2 The Effect of Lighting/Equipment Power on Increased Ventilation Rates

At 1.5 W/ft^2, increasing the OA ventilation rate from 5 to 20 cfm/person increases HVAC energy consumption by (-1,958) - (-5,020) = 3,062 kWh/year, and HVAC energy costs by (-127) - (-437) = $310/year. The comparable values for increasing ventilation at 4.0 W/ft^2 are 3,372 kWh and $339. Thus, the absolute values for the incremental costs of increased ventilation are approximately the same for both of these cases, and for the baseline case (Table 3), as would be expected.

Expressed as percentages of the total building energy consumption at 5 cfm/ person, these impacts of increased ventilation are: 7.2% increase at 1.5 W/ft^2 (total building energy at 5 cfm/person = 42,611 kWh); 5.4% increase at the baseline value of 2.55 W/ft^2 (total building energy at 5 cfm/person = 60,161 kWh, from Table 3); and 4.0% increase at 4.0 W/ft^2 (total building energy at 5 cfm/person = 84,502 kWh). So the selected value of lighting/equipment power requirements could make a couple percentage point difference in the impact of increased ventilation, when expressed in this manner. Except for occupant density (Section 5.3), no other single building or HVAC parameter makes this percentage vary so much from the 5.4% baseline value.

From Table 8, it is noted that HVAC energy consumption and cost at 1.5 W/ft^2 and a ventilation rate of 20 cfm/person are 1,958 kWh and $127 lower (5 to 8% lower) than they are for the baseline case (at 2.55 W/ft^2 and 5 cfm/person). This effect is even more pronounced when total building energy consumption is considered: total building energy consumption is 45,674 kWh/year (at a cost of $3,040/year) at 1.5 W/ft^2 and 20 cfm/person, which is about 25% less than the 60,161 kWh and

\$4,273/year at 2.55 W/ft^2 and 5 cfm/person. There would be some increase in installed cost associated with the efficient lighting systems, which cannot be addressed here. However, in concept, if one designed a new building with highly efficient lighting and conserving appliances, one could more than offset the HVAC energy cost penalty of increasing the ventilation rate from 5 to 20 cfm/person.

A similar point can be made regarding HVAC cooling capacity. The required capacity at 1.5 W/ft^2 and 20 cfm/person is 103.962 kBtu/h, which would translate into a practical installed capacity of 108 kBtu/h (9 tons). For the 2.55 W/ft^2 and 5 cfm/person case, the required capacity is 103.558 kBtu/h, which would also translate to 9 tons. Thus, an improvement in lighting and equipment efficiency could nominally offset any increase in cooling capacity that would otherwise be necessitated by an increase in ventilation rate.

5.5 INFILTRATION RATE

The baseline building is assumed to have an infiltration rate (i.e., an uncontrolled entry rate for outdoor air through leaks around windows, doors, etc.) of 0.1 air changes per hour (ACH). This was felt to represent a reasonably typical rate for a well-performing building (see Appendix B). Table 9 shows the effects when the assumed infiltration rate is changed to: a) 0 ACH (i.e., a building that is pressurized everywhere, preventing infiltration); and b) 0.3 ACH, one of the higher infiltration rates commonly assumed for modern offices.

5.5.1 The Effect of Infiltration at 5 cfm/person

As shown in the 5 cfm/person rows of Table 8, in the narrow ACH range considered here, HVAC energy consumption and cost appear to vary by 1 to 2% (either upward or downward, depending on the direction of the ACH change) for each 0.1 ACH change in the assumed infiltration rate. Required cooling capacity appears to vary by perhaps 2 to 4% per 0.1 ACH change.

Stating the results at 0 ACH in another manner, sensible and latent heat from infiltration are responsible for approximately 2% of the total HVAC energy consumption in the baseline building.

5.5.2 The Effect of Infiltration on Increased Ventilation Rates

At 0 ACH, increasing ventilation rate from 5 to 20 cfm/person would increase cooling capacity requirements by 16.129 kBtu/h, energy consumption by 3,288 kWh, and energy cost by \$329. These increments are essentially identical to the increments experienced at the baseline infiltration rate of 0.1 ACH, as expected.

One question of interest is: If an increased ventilation rate pressurizes a building and hence reduces infiltration, to what extent might the savings from reduced infiltration compensate for the energy penalty resulting from increased ventilation? As shown in Table 3, relative to the baseline (0.1 ACH, 5 cfm/person), the energy

TABLE 9

Effect of Infiltration Rate:
Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values:	103.558[2]	--	26,145[2]			2,510[2]			0.4	1.2
0.1 ACH,			(HVAC)	--	--	(HVAC)	--	--		
OA=5 cfm/p			60,161[2]			4,273[2]				
(absolute values)[2]			(building)	--	--	(building)	--	--		
OA ventilation rate = 5 cfm/person										
0 ACH	-2.567	-2.5	-385	-1.5	-0.6	-43	-1.7	-1.0	0.4	1.2
0.3 ACH	8.577	8.3	895	3.4	1.5	106	4.2	2.5	0.2	1.1
OA ventilation rate = 20 cfm/person										
0 ACH	13.562	13.1	2,903	11.1	4.8	286	11.4	6.7	0.4	0.9

[1] Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

[2] These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

penalty of increasing the ventilation rate to 20 cfm/person (0.1 ACH/20 cfm/person) is 3,245 kWh, and the cost penalty is $325. By comparison, from Table 9, the energy penalty of increased ventilation at 0 ACH (0 ACH/20 cfm/person) relative to the baseline (0.1 ACH/5 cfm/person) is 2,903 kWh (342 kWh, or 10%, less than the 3,245 kWh at 0.1 ACH), and the cost penalty is $286 ($39, or 12%, less than the $325). Thus, if a building with an infiltration rate of 0.1 ACH and a ventilation rate of 5 cfm/person had its infiltration rate ideally reduced to zero when its ventilation rate was increased to 20 cfm/person, the reduction in infiltration would offset only 10 to 12% of the energy penalty associated with the increased ventilation.

If the initial building had had an infiltration rate of *0.3 ACH* at 5 cfm/person (rather than the baseline 0.1 ACH), and if this 0.3 ACH had been reduced to zero by the increase in ventilation rate to 20 cfm/person, then the reduction in infiltration would offset approximately 40% of the ventilation energy penalty. This is probably about the maximum benefit that might ideally be anticipated.

In terms of commercial cooling capacity, the baseline case with 20 cfm/person (0.1 ACH/20 cfm/person) would require a pair of PSZ units totalling 10 tons, rounded upward to the nearest half ton, as indicated previously. Even if the increase in ventilation rate caused the infiltration to drop to zero (0 ACH/20 cfm/person), the required commercial capacity would still round upward to 10 tons.

5.6 EXTERIOR WALL RESISTANCE TO HEAT TRANSFER

The baseline building has exterior walls constructed of heavy-weight hollow concrete block with stucco on the exterior and with insulation and gypsum board on the interior. This wall construction is typical, and represents an overall heat transfer coefficient for the wall (including the inside and outside film coefficients) of U_o = 0.16 Btu/h ft^2 F°.

Table 10 shows the estimated effects of the following.

a) Reducing the exterior wall U_o to zero (i.e., providing infinite wall resistance), representing the extreme case. The glazing and doors associated with the exterior (front and rear) walls remain in place, so that there is still conduction and solar gain through those sources.

b) Replacing the block walls with 4-in. stud walls insulated with R-11 batts (U_o = 0.064 Btu/h ft^2 F°), a realistic wall construction having better resistance to heat transfer than does the baseline block wall, but not the ideal infinite resistance assumed in a) above.

c) Deleting the inside insulation on the exterior walls, increasing U_o to 0.34 Btu/h ft^2 F°, representing a case of poor heat transfer resistance.

TABLE 10

Effect of Exterior Wall Resistance: Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values:	103.558[2]	--	26,145[2]			2,510[2]			0.4	1.2
U_o = 0.16[3],			(HVAC)	--	--	(HVAC)	--	--		
OA=5 cfm/p			60,161[2]			4,273[2]				
(absolute values)[2]			(building)	--	--	(building)	--	--		
OA ventilation rate = 5 cfm/person										
U_o = 0	-0.755	-0.7	-401	-1.5	-0.7	-38	-1.5	-0.9	0.3	1.1
U_o = 0.06[3]	-0.307	-0.3	-185	-0.7	-0.3	-16	-0.6	-0.4	0.3	1.2
U_o = 0.34[3]	2.079	2.0	527	2.0	0.9	54	2.2	1.3	0.4	1.3
OA ventilation rate = 20 cfm/person										
U_o = 0	14.731	14.2	2,834	10.8	4.7	287	11.4	6.7	0.3	0.9
U_o = 0.06[3]	15.317	14.8	3,046	11.7	5.1	308	12.3	7.2	0.3	0.9

1 Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

2 These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

3 Units of U_o are Btu/h ft^2 F°. Wall U_o includes interior and exterior film resistances.

5.6.1 The Effect of Wall Resistance at 5 cfm/person

As shown in Table 10, varying the wall resistance at 5 cfm/person had only a limited effect. Decreasing the wall heat transfer coefficient by a factor of more than 2.5 from the baseline, to 0.064 Btu/h ft^2 F°, reduces HVAC energy consumption and costs by only 0.6 to 0.7%. In the extreme, reducing this coefficient all the way to zero would reduce HVAC energy consumption/costs only by 1.5%. On the other hand, *doubling* U_o to 0.34 Btu/h ft^2 F° (an extreme achieved by deleting all insulation from the baseline wall) has a slightly greater impact in the opposite direction, increasing HVAC energy consumption/costs by about 2%.

The results at a wall U_o of zero indicate that sensible heat conduction through the exterior walls is responsible for approximately 2% of the total HVAC energy consumption in the baseline building at a ventilation rate of 5 cfm/person.

On the basis of the results shown in Table 10, one would have to look closely at the cost-effectiveness before incurring any significant additional construction costs in an effort to improve wall resistance to heat transfer beyond the baseline value. This limited benefit from improved exterior wall insulation is consistent with results reported by others (Parker, 1996). As shown in later sections, improvements to the glass and the roof offer somewhat greater potential.

5.6.2 The Effect of Wall Resistance on Increased Ventilation Rates

As shown in Table 3, increasing OA ventilation rate from 5 to 20 cfm/person in the baseline building -- without any change in the wall resistance -- resulted in an increase of 3,245 kWh/year in HVAC energy consumption. As shown in Table 10, increasing ventilation rate from 5 cfm/person in the baseline building with the insulated block walls (5 cfm/0.16 Btu/h ft^2 F°) to 20 cfm/person in a building with frame walls (20 cfm/0.06 Btu/h ft^2 F°) increases consumption by 3,046 kWh. That is, if one accompanied the increase in ventilation rate with a re-design of the walls in this manner to improve the wall resistance, one might expect to reduce the energy penalty resulting from the increased ventilation by only 199 kWh/year, or 6%.

In the extreme, if one accompanied the increase in ventilation rate with a wall re-design that ideally raised wall resistance to infinity, one could reduce the energy penalty to 2,834 kWh/year -- a modest reduction of 411 kWh/year, or 13%.

Thus, improving exterior wall resistance to heat transfer does not appear to be the most promising avenue for achieving major reductions in energy consumption.

5.7 AMOUNT OF GLAZING

The baseline building has a front wall area 46% of which consists of glazing (including the glass front door), and a rear wall area which is 20% glazed. Table 11 shows the estimated effects of:

a) deleting all of the glazing, front and rear; and

b) increasing the glazing in the rear such that it is identical to the glazing on the front.

5.7.1 The Effect of Glazing Amount at 5 cfm/person

As shown in Table 11 for the ventilation rate of 5 cfm/person, deleting all glazing decreases energy consumption and costs, and decreases design cooling capacity, by 8 to 10%. Increasing the glazing on the rear from 20% to 46% increases these parameters by approximately the same amount.

The energy savings resulting from deleting the glazing are reduced by the fact that the front windows are shaded. Likewise, the energy penalty associated with increasing the rear glazing is increased by the fact that there is no rear shading. For example, deleting the shading overhang on the front would have increased the energy savings due to glazing elimination from the 2,535 kWh/year shown in Table 11 to 3,135 kWh/year, or about 12%.

The 9.7% reduction in HVAC energy consumption shown in Table 11 when the windows are deleted assumes that, realistically, the glazing is replaced by exterior wall area that conducts a lesser amount of heat. If one instead assumes, as an extreme, that the glazing is replaced by a surface that is completely thermally neutral, the results (with other adjustments) indicate that heat conduction and solar radiation through the glass are responsible for roughly 14% of the total HVAC energy consumption in the baseline building.

Complete elimination of the glazing is an extreme measure that would often not be a practical option from an aesthetics standpoint. However, these figures indicate the maximum savings that might be achieved by reducing the amount of glazing. The savings appear meaningful but modest.

Reasons for observed effects on hours undercooled and at elevated RH. Variations in the amount of glazing have only minor effects on the performance of the system (the percentage of hours undercooled or at elevated RH). Although these effects are small, it is of interest to understand conceptually why the observed variations occur. The hours undercooled (typically 20 to 30 hours per year) and the hours above 60% RH (typically 20 to 50 hours per year) commonly occur during the first occupied hours on warm weekday mornings, especially on summer Monday mornings. The HVAC system has been off overnight or over the weekend; thus,

TABLE 11

Effect of Amount of Glazing: Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values: 46% glazing in front, 20% in rear OA = 5 cfm/p (absolute values)[2]	103.558[2]	--	26,145[2] (HVAC)	--	--	2,510[2] (HVAC)	--	--	0.4	1.2
			60,161[2] (building)	--	--	4,273[2] (building)	--	--		
OA ventilation rate = 5 cfm/person										
0% in front, 0% in rear	-8.574	-8.3	-2,535	-9.7	-4.2	-258	-10.3	-6.0	0.2	0.8
46% in front, 46% in rear	9.922	9.6	2,069	7.9	3.4	215	8.6	5.0	0.3	1.3
OA ventilation rate = 20 cfm/person										
0% in front, 0% in rear	10.805	10.4	779	3.0	1.3	88	3.5	2.1	0.2	0.7

[1] Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

[2] These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

temperatures and RH levels are elevated, and heat has gotten stored in the building materials and furnishings during this warm unconditioned period, depending upon their heat capacities. During the first few hours on a warm Monday morning in Miami, the air entering the cooling coils has the highest temperature and moisture content that the system will see during that day. As a result, even though the cooling compressor is operating continuously during those hours, it might be unable to reduce the warm entering air down to the supply air temperature desired by the zone, depending upon the amount of over-design in the system. The warm entering air will also cause the coil surface temperature to be higher, with the result that less of the moisture in the air will be condensed.

When the glazing is deleted, the calculations predict that the interior of the building can be 1 F° or more cooler just before system startup on a summer Monday morning, relative to the baseline case with glazing. Thus, the air initially entering the cooling coils on startup is cooler, with the result that the supply air temperature leaving the coils is cooler by up to 1 F°, helping to avoid undercooling of the space. Also, the coil surface temperature is lower by up to 1 F°, condensing more moisture and thus helping to reduce the RH of the space. The lower space temperature prior to startup also results in less heat buildup in the building materials and furnishings, with the result that the space cools faster; on a Monday where the baseline building with glazing remains undercooled for the first four hours, the building with no glazing might remain undercooled for only the first two hours. Contributing to the faster cooling is the fact that, with the glazing absent, there is no glass conduction and glass solar radiation load on the space during these first hours.

As a result of the lower supply air temperatures achievable by the PSZ systems in the building with no glazing, Table 11 shows that, at the ventilation rate of 5 cfm/person, the percentage of hours undercooled drops from the 0.4% with the baseline building, to 0.2% for the no-glazing case. Similarly, due to the lower coil surface temperatures in the no-glazing case, the percentage of occupied hours above 60% RH drops to 0.8% from the baseline 1.2% at 5 cfm/person. These reductions in supply air and coil temperatures occur despite the reduction in cooling capacity shown in Table 11 for the no-glazing case; the reduced load more than compensates for the reduced capacity.

When the amount of glazing is *increased* from the baseline at 5 cfm/person, Table 11 shows that hours undercooled decrease slightly relative to the baseline values (to 0.3% from 0.4%), and hours at elevated RH increase slightly (from 1.2% to 1.3%). The reason for these effects is that the increased cooling capacity designed for the increased-glazing case more than compensates for the increased entering air temperatures on warm weekday mornings, but does not quite compensate for the increased humidity. With the glazing increased, the temperature in the rear zone is perhaps 0.7 F° warmer on a summer Monday morning just before system startup, but -- whereas the baseline rear zone might remain undercooled for two hours on that morning -- the greater capacity in the increased-glazing case is able to adequately reduce the zone temperature within only one hour, despite the elevated initial coil inlet temperature.

This ability of the higher-capacity system to reduce the number of hours undercooled in the increased-glazing case does not translate into a similar ability to reduce hours at elevated RH. The higher-capacity system achieves its increased cooling in this case to a large extent by providing a greater flow rate of supply air, and to a lesser extent by reducing the temperature of that supply air relative to baseline values. Thus, coil surface temperatures in the higher-capacity system are only a fraction of a degree lower than in the lower-capacity baseline case; sometimes the coil temperatures are the same in both cases. Thus, the higher-capacity system is not always able to condense enough additional moisture to compensate for the greater initial moisture content in the increased-glazing case.

Similar small variations in hours undercooled and at elevated RH will be observed in the other tables in Section 5. While exact scenarios will vary from case to case, the explanations will always involve the same factors discussed above: the entering air temperature (and moisture content) at startup on warm weekday mornings; the cooling capacity; and the constant PSZ design flows. These factors will determine: the temperature (and flow) of the supply air exiting the coils, and hence the ability to reduce hours undercooled; and the coil surface temperature, hence the ability to reduce hour at elevated RH.

5.7.2 The Effect of Glazing Amount on Increased Ventilation Rates

Hypothetically, if one deleted the glazing in the design of a new building and designed the HVAC system to provide 20 rather than 5 cfm OA/person, the deleted glazing would reduce the net energy penalty from the 3,245 kWh/year shown in Table 3 for the ventilation increase alone, to the 779 kWh/year shown in Table 11. Thus, deleted glazing would nominally recover about 75% of the energy penalty associated with the increased ventilation. Reducing rather than eliminating the glazing would, of course, recover a smaller percentage of the energy penalty.

This observation is not intended as a recommendation that glazing be eliminated or significantly reduced. The decision regarding the glazing configuration will be determined by aesthetics and cost-effectiveness considerations on a site-specific basis. The observation here is intended only to illustrate the relative energy impacts of increased ventilation versus substantially reduced amounts of glazing.

In terms of performance, Table 11 shows that the hours undercooled and at elevated RH in the no-glazing case at 20 cfm/person are about the same (or are slightly less than) as in the no-glazing case at 5 cfm/person (and, correspondingly, are lower than in the baseline building at 5 cfm/person). This decrease occurs despite the increase in both sensible and latent heat resulting from the increased OA flow rate. The reason is that the increase in cooling capacity in the 20 cfm/person case compensates for the increase in incoming sensible and latent heat. Thus, after startup on a summer Monday morning, zone temperature is reduced more rapidly, despite the elevated temperature of the air entering the coils. Where the zone was undercooled during the first two hours in the no-glazing/5 cfm case, it might be undercooled for only one hour in the no-glazing/20 cfm case.

The design air flow rate in the 20 cfm/person case is the same as in the corresponding 5 cfm/person case, due to the manner in which the systems are designed. Thus, the increase in capacity at 20 cfm/person must be achieved by operating at lower coil temperatures rather than by increasing flow. The coil temperature in the no-glazing/20 cfm case is up to 1.5 F° lower than in the no-glazing/5 cfm case (and up to 2.5 F° lower than in the baseline glazing/5 cfm case). As a result, the elevated initial humidities on Monday mornings are more rapidly reduced in the 20 cfm/person case (despite the increased latent load added by the incoming OA), and the number of hours at elevated RH decreases.

This improved performance for 20 cfm/person systems, relative to the corresponding 5 cfm/person systems, is seen consistently in the tables throughout Section 5, for the same reasons discussed above.

5.8 GLASS TYPE

The baseline building has single-pane glass in a thermally broken aluminum frame, providing an overall heat transfer coefficient (including the frame) of $U_o = 0.94$ Btu/h ft^2 F°, determining heat conduction through the glass. It has a shading coefficient (S-C) of 0.55, determining the transmittance, reflectance, and absorptance of solar radiation. A S-C of 0.55 can be achieved with single-pane glass through various combinations of glass tinting, reflective coatings, and interior shading.

Table 12 shows the estimated effects of:

a) Decreasing the fenestration U_o to 0.32 Btu/h ft^2 F° by switching to double-pane glass with a coating, and decreasing S-C to 0.16, about the lowest value that can reasonably be achieved through the use of tinting, highly reflective coatings, and interior shading.

b) Increasing the S-C to 0.94 (with a U_o of 0.94 Btu/h ft^2 F°), by switching to simple single-pane glass with no tinting, coating, or interior shading (representing glass having minimum resistance).

An even more extreme case than a) above would be to delete the glazing altogether, a case considered in Section 5.7.

TABLE 12

Effect of Glass Type: Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values: U_o=0.94 Btu/h-ft^{2}F°, S-C=0.55, OA=5 cfm/p (absolute values)[2]	103.558[2]	--	26,145[2] (HVAC)	--	--	2,510[2] (HVAC)	--	--	0.4	1.2
			60,161[2] (building)	--	--	4,273[2] (building)	--	--		
OA ventilation rate = 5 cfm/person										
U_o = 0.32, S-C = 0.16	-5.755	-5.6	-1,936	-7.4	-3.2	-194	-7.7	-4.5	0.2	0.9
U_o = 0.94, S-C = 0.94	7.366	7.1	1,930	7.4	3.2	188	7.5	4.4	0.5	1.1
OA ventilation rate = 20 cfm/person										
U_o = 0.32, S-C = 0.16	13.725	13.3	1,405	5.4	2.3	153	6.1	3.6	0.2	0.7

[1] Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

[2] These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

5.8.1 The Effect of Glass Type at 5 cfm/person

As shown in Table 12, increasing the resistance of the glass to conduction and solar radiation as described in a) above reduces HVAC energy consumption and costs by 7 to 8%. Decreasing resistance as described in b) increases consumption and costs by a similar amount. The computed required cooling capacity also varies by similar amounts.

As discussed in Section 5.7.1, increasing glass resistance in the extreme -- by deleting the glass altogether -- reduces energy consumption and costs by about 10%.

As with changes in the *amount* of glazing, the incremental effects observed due to changes in the characteristics of the glazing are impacted by the configuration of the exterior shading around this office. The greater the amount of exterior shading, the less the incremental benefits of improving glazing (or the less the incremental penalty of using less resistant glazing). For example, if the baseline building had had no front overhang, the reduction in energy consumption achieved by using the glazing in a) above would have been 2,374 kWh (rather than 1,936 kWh), a 9% reduction (rather than 7%).

The 7 to 10% energy savings shown in Tables 11 and 12 for improvement in (or elimination of) the glazing compare with the maximum savings of less than 1 to 2% shown in Table 10 for improvements to exterior wall resistance, and the maximum savings of about 6% that will be discussed in Table 13 for improvements to roof resistance. It is not surprising that glazing improvements are more effective than wall improvements, since, as indicated previously, the walls account for only 5% of the baseline building's annual sensible heat load, whereas the glazing accounts for 14%.

But it is less obvious why glazing improvements should be more effective than roof improvements, since the baseline roof accounts for an even larger portion of the sensible load, 21%. The reason is probably that the heat conducted through the roof into the plenum area goes largely to increasing plenum temperature, with only a relatively small portion being conducted across the ceiling into office space. This heat in the plenum is channeled directly into the cooling coils with the return air, where the increased entering air temperature creates a larger ΔT driving force between the air and the coils, giving more effective heat removal. By comparison, the heat introduced by conduction and radiation through the glazing enters the office space, where it is mixed with the larger volume of office air before being routed to the cooling coils. Thus, it has less effect in increasing the temperature of the air entering the coils, makes less contribution to the ΔT driving force, and thus is not removed as efficiently.

Hours undercooled and at elevated RH vary slightly with the changes in glass type, for reasons similar to those discussed in Section 5.7.1.

5.8.2 The Effect of Glass Type on Increased Ventilation Rates

If one replaced the baseline glass with the high-performance glass considered here at the same time that ventilation rate were increased from 5 to 20 cfm/person, the improved glazing would decrease the net energy penalty from the 3,245 kWh/year shown in Table 3 for the ventilation increase alone, to the 1,405 kWh/year shown in Table 12 for the 20 cfm/person case. Thus, improving the glazing in this manner would nominally recover 57% of the energy penalty associated with the increased ventilation.

Of course, if the baseline glazing were replaced with improved glazing that was not as resistant as the high-performance glazing assumed here, the energy benefits from improving the glazing would be less.

No effort is made here to assess the installation costs, and hence the overall cost-effectiveness, of improving the glazing. The goal here is simply to indicate the variables having the greatest impact on energy consumption.

The percentage of hours undercooled for the 20 cfm/person case in Table 12 are about the same as that for the corresponding 5 cfm/person case; the percentage of hours at elevated RH are slightly less. These are consistent with the results discussed in Section 5.7.2, and have the same explanation.

5.9 ROOF RESISTANCE TO HEAT TRANSFER

The baseline building has a roof construction consisting of built-up roofing over R-12 mineral board and a metal deck. This roof represents an overall heat transfer coefficient (including the inside and outside film coefficients) of U_o = 0.066 Btu/ h ft^2 F°.

Table 13 shows the estimated effects of the following.

a) Reducing the roof U_o to zero (i.e., providing infinite roof resistance), representing the extreme case.

b) Replacing the R-12 mineral board with lesser insulation, approximately doubling U_o to 0.12 Btu/h ft^2 F°).

5.9.1 The Effect of Roof Resistance at 5 cfm/person

As shown in Table 13, eliminating all heat transfer through the roof (U_o = 0) at 5 cfm/person reduces HVAC energy consumption and costs by about 6 to 7%. Approximately doubling the heat transfer *increases* energy consumption by about a similar amount.

TABLE 13

Effect of Roof Resistance:
Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values:	103.558[2]	--	26,145[2]			2,510[2]			0.4	1.2
U_o = 0.066[3],			(HVAC)	--	--	(HVAC)	--	--		
OA=5 cfm/p			60,161[2]			4,273[2]				
(absolute values)[2]			(building)	--	--	(building)	--	--		
OA ventilation rate = 5 cfm/person										
U_o = 0	0	0	-1,609	-6.2	-2.7	-165	-6.6	-3.9	~0	0.8
U_o = 0.12[3]	9.725	9.4	1,619	6.2	2.7	206	8.2	4.8	0.4	1.7
OA ventilation rate = 20 cfm/person										
U_o = 0	15.635	15.1	1,673	6.4	2.8	168	6.7	3.9	~0	0.8

1 Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

2 These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

3 Units of U_o are Btu/h ft^2 F°. Roof U_o includes interior and exterior film resistances.

The results for the case where roof U_o is zero, when adjusted, indicate that sensible heat conduction through the roof is responsible for approximately 8% of the total HVAC energy consumption in the baseline building. This modest contribution from the roof is greater than the 2% from the exterior walls (Section 5.6.1), but is less than the 14% estimated from the glazing (Section 5.7.1).

Table 13 shows no change in required cooling coil capacity when U_o is decreased to zero, a result that might seem curious. But when U_o is increased to 0.12 Btu/h ft^2 F°, capacity requirements increase by 9%, consistent with intuition. The explanation for these results is as follows.

DOE-2 computes the cooling coil capacity based upon *design* temperatures for the office space and the plenum (specified in the input file as DESIGN-COOL-T). These *design* temperatures are generally selected to provide some selected degree of over-design. As discussed in Appendix B, for the calculations here, the design cooling temperature for the office space is 74°F, a little below the actual cooling setpoint of 75°F, so that the coils are somewhat over-designed. Likewise, the design temperature for the plenum is 90°F, whereas, for the baseline building with the baseline roof U_o of 0.066 Btu/h ft^2 F°, the peak hourly temperature actually encountered in the plenum is above 89°F. Accordingly, the cooling coils are slightly over-designed, assuming that these coils will have to cool the entering air from 90°F to the desired supply air temperature, when in fact, the entering air will have to be cooled only from about 89°F or less.

Since the plenum never quite reaches the design temperature at the baseline roof U_o, it does not, either, when U_o is reduced to zero and the actual plenum temperature becomes even lower. Accordingly, in both cases, the program designs the coil capacity assuming a plenum temperature of 90°F, and there is no change in capacity when the roof U_o is decreased below the baseline.

But when the roof U_o is increased to 0.12 Btu/h ft^2 F°, more heat can enter the plenum, and actual plenum temperatures *do* reach and exceed 90°F. When this occurs, the program adjusts the plenum design cooling temperature upward, so that the coils will not be under-designed. As a result, when the roof U_o is increased to this extent over the baseline value, the computed capacity does increase as intuitively anticipated.

When U_o is reduced to zero, the percentage of hours undercooled drops to about zero. At $U_o = 0$, the actual plenum temperatures are further below the design temperatures than they were in the baseline case, and the cooling coils are thus more over-designed. Accordingly, the coils now have sufficient excess capacity to handle the spikes in entering air temperature that they encounter at startup on warm week-day mornings, as discussed in Section 5.7.1, and these early-morning undercooled hours are essentially eliminated.

But when U_o is raised to 0.12 Btu/h ft^2 F°, the percentage of undercooled hours remains about the same as in the baseline case. This is the result of two competing phenomena.

a) The program has adjusted the plenum design cooling temperature upward, so that the cooling capacity is no longer over-designed as it was in the case where $U_o = 0$. However, because of the heat now conducting through the roof in the afternoon, the design cooling capacity is 9% greater than in the baseline case, with the result that the capacity at $U_o = 0.12$ is better able than the baseline to handle the first morning hours on warm weekday startups. As a result, a summer Monday morning that might have been undercooled for the first four hours in the baseline case and the first one hour in the $U_o = 0$ case, is predicted to be undercooled for the first *two* hours in the $U_o = 0.12$ case.

b) But because so much more heat conducts through the roof in the $U_o = 0.12$ case, this case experiences some undercooled hours around 4 pm in the afternoon, a situation not encountered in the baseline case. On the particular Monday morning in a) above, the $U_o = 0.12$ case experiences two undercooled hours at 4 and 5 pm in the afternoon, with the result that this case, like the baseline, experiences a total of four undercooled hours on that day. The hours are just spread between the morning and the afternoon, rather than all being in the morning.

The causes for the reduced percentage of occupied hours at RH>60% at reduced U_o are also similar to those discussed in Section 5.7.1. At $U_o = 0$, the plenum and office space remain cooler over nights and weekends; the lower entering air temperatures during startup on warm weekday mornings result in lower coil temperatures, and the number of early-morning elevated RH hours is thus reduced compared to the baseline.

When U_o is increased to 0.12 Btu/h ft^2 F°, the percentage of occupied hours at RH>60% increases to 1.7%, among the highest predicted for any of the cases considered in this report. At first, this result seems counter-intuitive. The cooling coils in the $U_o = 0.12$ case commonly operate at temperatures 1 to 2 F° colder than those in the baseline case during warm weather, so that they condense more moisture. (The increase in capacity shown in Table 13 for this case is achieved by lowering supply air temperature while maintaining the same constant PSZ flow as that in the baseline case.) Correspondingly, hourly reports for the particular summer Monday considered above show that the latent cooling is greater and the moisture levels in the offices (expressed in lb moisture/lb dry air) are lower for the $U_o = 0.12$ case than for the baseline case, as expected.

The reason for the apparent anomaly is that this *improved* RH control during *warm* weather in the $U_o = 0.12$ case is apparently more than offset by *poorer* RH control during *cooler* weather. On a cool winter Monday, increased heat loss through the roof in the U_o = 0.12 case makes it unnecessary for the cooling coils to come on (to handle internal heat gains) until 11 am, whereas the baseline coils have to come on at 9 am. Then, for the first two hours after the coils switch on in the U_o = 0.12 case, the coils are operating at a smaller part load ratio than are the baseline coils (due to the increased free roof cooling), with the result that the Btu/h of *latent* cooling is lower. As a result, RH levels in the offices (from internal generation) during the first four hours of this winter Monday morning are greater for the U_o = 0.12 case than for the baseline case.

A similar effect occurs on mild spring and fall Mondays. In the baseline case, the economizer opens to a greater extent in the hour or two before the cooling coils switch on; and, once the coils do switch on, they are operating closer to full load and are achieving more Btu/h of latent cooling. So again, the baseline case achieves lower RH in the offices than does the U_o = 0.12 case.

The net effect of these phenomena is that the U_o = 0.12 case has a much greater percentage of occupied hours at RH>60% than does the baseline case, with fewer of these hours occurring during the summer, and more occurring during cool and mild seasons.

5.9.2 The Effect of Roof Resistance on Increased Ventilation Rates

Using the same rationale as that in Section 5.6.1, Table 13 shows that -- if an increase in ventilation rate in the baseline building were accompanied by steps that reduced the roof heat transfer to zero -- the increase in energy consumption would be 1,673 kWh/year, rather than the baseline increase of 3,245 kWh/year. The improvement in roof resistance would thus conceptually recover 48% of the energy penalty associated with the increased ventilation.

Of course, reducing roof heat transfer to zero is an extreme case that would not be achievable in practice. In a more "practical" extreme case, where R-19 batts are added beneath the roof deck (decreasing U_o to 0.028 Btu/h ft^2 F°), the improved roof insulation would compensate for 28% (rather than 48%) of the energy penalty of the increased ventilation.

Again, no effort is made here to assess the installation costs, and hence the overall cost-effectiveness, of improving roof insulation.

At U_o = 0, the hours undercooled and at elevated RH remain essentially the same with a ventilation rate of 20 cfm/person as in the 5 cfm/person case. The reasons are essentially the same as those discussed in Section 5.7.2.

5.10 ELIMINATION OF ALL EXTERIOR SURFACES

The previous four sections have addressed the individual effects of varying the heat transfer resistances of the three types of above-grade exterior surface: the walls, the windows, and the roof. In each case, the extreme value was to assume that the surface had infinite resistance.

In Table 14, for illustration, the extreme case is considered where all three types of exterior surface have infinite resistance simultaneously. In practice, such a case could be approached only if the space being considered were completely enclosed within a larger structure, i.e., if the space had no exterior walls, and all boundaries of the space were interior walls adjoining other conditioned space.

5.10.1 The Effect of Infinite Exterior Resistance at 5 cfm/person

In the extreme, with all external loads except for infiltration and slab conduction deleted, Table 14 shows that energy consumption and cost would decrease by about 20%, and design cooling capacity would decrease by about 10%.

Thus, of the energy consumed by the baseline HVAC system, about 20% results from loads on the walls, windows, and roof. (The sum of the individual percentages from Sections 5.6.1, 5.7.1, and 5.9.1, which have been adjusted, actually add up to 24%.) The remainder of the HVAC power consumption results from: internally generated loads (occupants, lighting, and equipment); mechanically introduced outdoor ventilation air (5 cfm/person for the percentage shown here); infiltration; and very small contributions from conduction through the slab and door.

This represents the hypothetical maximum energy savings that could be achieved with improved insulation.

5.10.2 The Effect of Infinite Exterior Resistance on Increased Ventilation Rates

The results for the 5 cfm/person case in Table 14 show that the combined contribution of conduction and radiation through the walls, windows, and roof to HVAC energy consumption in the baseline building is about 5,135 kWh/year. Table 3 showed that the incremental energy penalty of increasing the baseline ventilation rate from 5 to 20 cfm/person is 3,245 kWh/year (which gets adjusted slightly, to 3,257 kWh/year when modeling the case of infinite exterior resistance).

Thus, the nominal energy savings of deleting the walls, windows, and roof is greater than the energy penalty of increasing the ventilation rate, by a difference of 1,878 kWh/year. One could hypothetically increase the ventilation rate and save 1,878 kWh/year at the same time, if one could make the walls, windows, and roof thermally neutral in the process.

TABLE 14

Effect of Infinite Resistance for All Exterior Surfaces: Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values: 2 ext. walls, U_o= 0.16[3], with glass; roof U_o=0.066[3]; OA=5 cfm/p (absolute values)[2]	103.558[2]	--	26,145[2] (HVAC) 60,161[2] (building)	-- --	-- --	2,510[2] (HVAC) 4,273[2] (building)	-- --	-- --	0.4	1.2
OA ventilation rate = 5 cfm/person										
No exterior surfaces	-11.204	-10.8	-5,135	-19.6	-8.5	-561	-22.4	-13.1	0	0.4
OA ventilation rate = 20 cfm/person										
No exterior surfaces	7.922	7.6	-1,878	-7.2	-3.1	-211	-8.4	-4.9	0	0.5

1 Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

2 These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

3 Units of U_o are Btu/h ft^2 F°. Values of U_o include interior and exterior film resistances.

Of course, completely eliminating heat gain through walls, windows, and the roof is an extreme which is not achievable in practice (except by enclosing the space within another building). A more "practical" extreme case would be to combine the most resistive alternatives considered in Sections 5.6, 5.8, and 5.9: insulated frame walls replacing the hollow-block walls, switching to highly reflective double-pane glass, and adding R-19 insulation underneath the baseline roof. These steps together would recover roughly 90% of the energy penalty associated with the increase in ventilation rate. But there would still be some energy penalty, not a savings.

Table 14 shows that, with infinite exterior resistance, the number of hours undercooled drops to zero with either ventilation rate. This result is not surprising. With infinite resistance, the temperature buildup in the offices over warm nights and weekends will be greatly reduced, limited to that resulting from infiltration, from the 5% of the lighting that remains on during unoccupied periods, and from the small amount of conduction through the slab and door. As a result, the HVAC systems are not overwhelmed by a high entering air temperatures upon startup on warm weekday mornings. They always have sufficient capacity to reduce the supply air to the temperature required by the zones.

5.11 THERMOSTAT SET-UP RATHER THAN SYSTEM SHUT-DOWN DURING UNOCCUPIED HOURS

The baseline HVAC system operating procedure involves complete shut-down of the cooling coils overnight, and over weekends and holidays. This saves energy, but allows temperatures and humidity levels in the offices and plenums to build up during warm weather off-hours. As a result, the first operating hours on warm weekday mornings (and especially on Monday mornings) can experience elevated temperatures and RH levels.

Table 15 shows the results when -- instead of shutting the cooling coils off on nights and weekends -- the office thermostats are set up to a cooling setpoint temperature of 81°F, 6 F° above the 75° setpoint used during occupied hours. For these runs, the outside air dampers were kept closed during unoccupied hours, so that the temperature control was achieved by circulating building air with no OA ventilation.

As shown in the table, operation at 81°F during unoccupied hours has the desired effect of eliminating the percentage of hours undercooled to zero, for both ventilation rates. On summer Sunday afternoons in Miami, the temperatures in the baseline office space (with the cooling coils completely off) can reach 85°F, and those in the plenums can approach 90°F. The temperatures entering the cooling coils upon startup on Monday morning can be as high as 84°F in the baseline case. Keeping these temperatures near 81°F by leaving the coils on at the higher setpoint has two

TABLE 15

Effect of Thermostat Set-Up Rather than System Shut-Down: Increase Compared to Baseline Case with 5 cfm/person[1]

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values:	103.558[2]	--	26,145[2]			2,510[2]			0.4	1.2
Cooling coils			(HVAC)	--	--	(HVAC)	--	--		
shut down during			60,161[2]			4,273[2]				
off hours;			(building)	--	--	(building)	--	--		
OA = 5 cfm/p										
(absolute values)[2]										
OA ventilation rate = 5 cfm/person										
Cooling setpoint 81°F off hours	0	0	622	2.4	1.0	6	0.2	0.1	0	1.2
OA ventilation rate = 20 cfm/person										
Cooling setpoint 81°F off hours	15.635	15.1	3,892	14.9	6.5	335	13.3	7.8	0	0.9

1 Capacity, energy consumption, and cost numbers represent the two PSZ units combined. Undercooling and RH performance numbers represent the average for the two zones.

2 These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

effects which have been successful in eliminating undercooled hours. First, by reducing the temperature of the air entering the cooling coils by several degrees at weekday morning startups, this approach has made it possible for the systems of the given capacities to reduce the supply air temperature to the levels demanded by their zones during these crucial first hours of the day. Second, by keeping the temperatures of the zones down over the night or weekend, this approach has reduced the zones' demand at startup, again making it easier for the systems of the given capacities to meet the demand.

No reductions are achieved in the percentage of *occupied* hours at elevated RH; the 1.2% and 0.9% shown in Table 15 for the 81° set-up case are identical to the values seen at 5 and 20 cfm/person, respectively, in the baseline (shut-down) case. As discussed later, most of the occupied hours with elevated RH occur after startup during cool weather, when the humidity can be high but when temperatures are so low that the cooling coils operate at greatly reduced capacity (or are off altogether). In such cool weather, off-hour ambient temperatures are often below 81 °F. With an 81 °F set-up temperature, the coils would not come on when the building is unoccupied. As a result, the behavior of the building and the system during those elevated-RH hours after startup on cool mornings is identical, regardless of whether the cooling coils have been shut off during the unoccupied hours, or whether they have been set up. Accordingly, there is no net effect on RH levels in the offices *during occupied hours*.

However, there is a significant impact on RH levels during *unoccupied* hours in warm weather. When the coils are shut off entirely during unoccupied hours, as in the baseline case, the RH levels commonly reach 60 to 75% over warm summer weekends, and also overnight in mid-week, due to the latent content of infiltrating outdoor air. When the coils are instead left on at the 81°F setpoint, the coils tend to cycle on during warm weekend afternoons, and sometimes on weekday mornings; with this cycling, peak RH levels during unoccupied summer hours are reduced to perhaps 40-50%. Since indoor biocontaminant growth can be significantly increased at elevated RH levels, with 60% RH being the upper level recommended by ASHRAE (ASHRAE, 1989a), the elevated RH levels experienced in humid climates when the cooling coils are turned off over a weekend should be of concern. The relatively high indoor temperatures over the weekends (up to almost 90°F in the plenum), combined with the high RH, would help expedite biocontaminant growth.

Shirey and Rengarajan (1996) suggest that actual indoor RH levels during occupied hours in humid climates might be higher than those predicted by DOE-2, because DOE-2 does not take into consideration either the moisture capacitance of furnishings and building materials, or the re-evaporation of moisture off the cooling coils when the coils switch off with the air handler operating. At least in some circumstances, moisture capacitance is calculated by those authors to be the more significant of these two causes of the discrepancy between the DOE-2 RH predictions and their own. That is, moisture in the infiltrating air is absorbed into the furnishings and building materials over nights and weekends, and then takes some time to desorb when the HVAC system is started up again, contributing to the occupied hours at elevated RH in the Shirey and Rengarajan computations. To the extent that this is happening, it would add further importance to the need to keep RH levels down on nights and weekends by leaving the coils on at a set up setpoint.

As would be expected, since DOE-2 calculates system capacities based upon the design temperatures, the cooling coil capacity in the 81° set-up cases are unchanged from their baseline (shut-down) counterparts. The 15.635 kBtu/h increase shown in Table 15 for the 81° set-up/20 cfm case, relative to the shut-down/5 cfm baseline, is identical to the increase for the shut-down/20 cfm case.

As would be expected, Table 15 shows an increase in energy consumption in the 81° set-up cases, relative to their baseline counterparts. At 5 cfm/person, the increase is 622 kWh/year; at 20 cfm/person, the increase is 647 kWh/year relative to the shut-down/20 cfm case.

But interestingly, the increases in *cost* are much smaller than the increases in consumption would initially suggest -- only \$6/year at 5 cfm/person, and only \$10/year at 20 cfm/person (relative to the shut-down/20 cfm case). This is smaller than the cost that would be calculated by simply applying the \$0.0473/kWh electricity charge to the increased kilowatt-hours of usage. The increase in usage is being largely offset by reduced peak demands during warm months, and hence reduced demand charges. Keeping the plenums (and offices) at more moderate temperatures over warm nights and weekends shaves the peak load that is experienced during startup hours on weekday mornings.

Based on these calculations, the operator of an office such as this would be well advised to consider operation of the HVAC at a set-up cooling setpoint during unoccupied periods, rather than turning the cooling system off. Such an approach could significantly reduce the risk of biocontaminant growth over nights and weekends (and would be of added importance if elevated RH levels during unoccupied hours leads to elevated RH levels during occupied hours due to the moisture capacitance of materials). It would also improve occupant comfort on warm mornings, from the standpoints both of temperature and humidity. And it would achieve these benefits at minimal cost.

5.12 ALTERNATIVE HVAC SYSTEMS

The baseline configuration assumes that the office is subdivided into two 2,000 ft^2 zones, each conditioned by a dedicated packaged single-zone (PSZ) unit. Under this assumption, each of the two PSZ units requires approximately 5 tons of cooling capacity.

Table 16 shows the results when several possible alternative HVAC system configurations are considered instead. The alternative HVAC system configurations

TABLE 16

Effect of Assuming Alternative HVAC Systems: Increase Compared to Baseline Case with 5 cfm/person

Case	Increase in total required cooling capacity: In kBtu/h	As %	Increase in annual energy consumption: In kWh	As % of HVAC energy	As % of building energy	Increase in annual energy cost: In $	As % of HVAC energy cost	As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values:	103.558[1]	--	26,145[1]			2,510[1]			0.4	1.2
Two PSZ units,			(HVAC)	--	--	(HVAC)	--	--		
OA=5 cfm/p			60,161[1]			4,273[1]				
(absolute values)[1]			(building)	--	--	(building)	--	--		
OA ventilation rate = 5 cfm/person										
One PSZ unit	-3.060	-3.0	-377	-1.4	-0.6	-38	-1.5	-0.9	0.5	1.2
One PSZ, subzone[3]	0	0	-761	-2.9	-1.3	-43	-1.7	-1.0	0.3	1.2
One PSZ, subzone[4]	0	0	-1,105	-4.2	-1.8	-67	-2.7	-1.6	0.3	1.0
One PVAVS	1.479	1.4	-3,149	-12.0	-5.2	-186	-7.4	-4.4	~0	1.5
Two PTAC units	-6.718	6.5	-2,719	-10.4	-4.5	-232	-9.2	-5.4	0.2	~0
OA ventilation rate = 20 cfm/person										
Two PSZ units[2]	15.635	15.1	3,245	12.4	5.4	325	12.9	7.6	0.4	0.9
One PSZ unit	12.512	12.1	2,858	10.9	4.8	285	11.4	6.7	0.5	0.8
One PSZ, subzone[3]	15.639	15.1	2,461	9.4	4.1	279	11.1	6.5	0.3	0.8
One PSZ, subzone[4]	15.639	15.1	2,067	7.9	3.4	254	10.1	5.9	0.3	0.8
One PVAVS	17.413	16.8	-78	-0.3	-0.1	124	4.9	2.9	~0	2.9
Two PTAC units	8.152	7.9	1,944	7.4	3.2	228	9.1	5.3	0.2	0

1 These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.
2 The baseline case at 20 cfm/person (from Table 3), repeated here for comparison.
3 The front zone is the control zone; the rear zone is the subzone.
4 The rear zone is the control zone; the front zone is the subzone.

considered are:

1) a single PSZ unit conditioning the entire 4,000 ft^2 office, with the entire space being considered as a single zone. The required size of this single PSZ unit -- approximately 10 tons of cooling capacity -- is well within the range of commercially available units.

2) a single PSZ unit conditioning the entire space, but with the front (more shaded) 2,000 ft^2 zone being used as the control zone (determining compressor operation and hence supply air temperature), and with the rear (more exposed) zone being a variable air volume (VAV) subzone. Because the rear subzone is no longer able to control the temperature of the supply air entering the subzone, subzone space temperature is now controlled instead by modulating the flow rate of the air entering the subzone.

3) a single PSZ unit conditioning the entire space, as in 2) above, except with the rear 2,000 ft^2 zone being used as the control zone, and the front zone being a VAV subzone.

4) a single packaged variable air volume system (PVAVS) serving the two 2,000 ft^2 zones. PVAVS would be a commercially reasonable choice for this application, although perhaps less common than PSZ. Consistent with typical design of *packaged* VAV systems, it is assumed that the PVAVS supplies an essentially constant supply air temperature (which is as close to the minimum supply temperature, 55°F, as the on/off compressor can deliver); and that zone temperatures are controlled solely by modulating supply air flow rate to each zone.

5) two through-the-wall packaged terminal air conditioner (PTAC) units, one conditioning each zone. The roughly 5-ton units required for this application are somewhat larger than PTAC units commonly available commercially; PTAC units would not usually be considered for an office such as that being studied here. PTAC units are being considered in this study solely for illustration.

Overall, Table 16 shows that the greatest reductions in energy consumption occur with the PVAVS. In fact, the table indicates that a PVAVS could provide 20 cfm OA/person while consuming 78 kWh/year *less* than the baseline dual PSZ units providing only 5 cfm/person. One could increase the ventilation rate and simultaneously save energy if one installed the PVAVS instead of the two PSZ units. As discussed further later, this effect results because the PVAVS moves much less air and thus consumes less energy in operating the central air handling fan.

Table 16 also suggests that a pair of PTAC units would require less capacity, and reduce energy consumption and costs, relative to two PSZ units, despite the fact that the PTAC units are assumed to have a poorer cooling efficiency. In part, this result occurs because the PTAC units are not conditioning the overhead plenum space to the extent that the PSZ units are; the return air to the PTAC units flows directly from the offices, rather than through the plenum. Also, the air handler fan power for the PTAC fans (and the air temperature rise through the fans) are only 12% of that for the PSZ units according to the DOE-2 default values, because the PTAC fans do not have to move the air through either supply or return ductwork. However, these very factors that are causing PTACs to appear advantageous in these calculations would also cause PTACs to inadequately distribute cooling and ventilating air throughout the two 2,000 ft^2 zones. Thus, one would not likely select individual PTAC units for this application, despite the computed energy savings.

5.12.1 The Effect of Alternative HVAC Systems at 5 cfm/person

Single PSZ system conditioning entire 4,000 ft^2 as one zone

Capacity. As shown in Table 16, the cooling capacity of the single PSZ unit at 5 cfm/person is calculated to be 3.060 kBtu/h less than the combined capacities of the two individual units in the baseline case at that ventilation rate. This result occurs because the single unit is being sized for the peak load of the total space (which occurs at 5 pm on October 11), rather than for the peak load of the front office plus the peak load of the rear (which occur at 5 pm on June 14 and at 4 pm on January 25, respectively).

DOE-2 computes cooling capacities by first calculating the maximum supply air flows required to a given space (at the minimum supply temperature) in order to handle the peak load for that space, determined in the LOADS part of the program. (With PSZ systems, of course, this maximum flow becomes the constant volume that is provided whenever the air handler is operating.) The required system cooling capacity is then computed by multiplying this maximum flow times the required reduction in the enthalpy of the air entering the cooling coils during the hour that this peak load occurs, in order to reduce the air leaving the coils to the minimum supply temperature. Thus, changes either in the flow rate, or in the entering air enthalpy at peak load, create the differences in the calculated capacities.

Because the peak loads for the two PSZ systems in the baseline case occur at different times than the peak for the single-PSZ case, the flows and the entering enthalpies are both different in the two cases. In the baseline case, the combined maximum air handler design flow for the front plus rear units is 3,810 cfm; by comparison, the design flow for the air handler in the single-PSZ case is 3,689 cfm, about 3% lower. Presumably, the entering enthalpies during the peak-load hours also vary somewhat between the two cases. If one specifies a design flow of 3,810 cfm for the single-PSZ case, so that the two cases now have the same flow rates (but different peak-hour entering enthalpies), the computed cooling capacities are similar.

Thus, the difference in design air flow -- and not the difference in peak-hour entering enthalpies -- is primarily responsible for the difference in the computed capacities between the two cases. This conclusion is also apparent from the fact that the percentage reduction in cooling capacity shown in Table 16 for the single-PSZ case (3%) is essentially the same as the percentage reduction in total flow for this case.

Not surprisingly, this reduction in total capacity in the single-PSZ case results in a tiny increase in the percentage of hours undercooled (0.5% vs. 0.4%).

Energy consumption and cost. The 377 kWh reduction in annual energy consumption in the single-PSZ case, and the $38 reduction in annual energy cost, also result almost entirely from the reduction in air handler design flow. As discussed previously, electrical power to the air handling fan is a significant contributor to system electric consumption. If one specifies that fan for the single-PSZ unit have the same design flow (3,810 cfm) as the combined flows of the baseline fans, the single-PSZ case would consume only 37 kWh/yr less than the baseline, and the annual HVAC energy costs of the two cases would be exactly the same ($2,510).

Hours at elevated temperature or RH. Although treating the entire 4,000 ft^2 as a single zone in this manner appears on paper to provide about the same performance at a slightly lower cost, there would be a comfort penalty in practice. Occupants in the rear (south) office would likely be too warm, and/or occupants in the front (north) office would likely be too cool.

Single PSZ system with 2,000 ft^2 control zone and 2,000 ft^2 subzone

Capacity. As shown in Table 16, the calculated cooling capacities for the cases of a single PSZ unit with a control zone and a subzone are exactly the same as that for the baseline case with two PSZ units. This holds true regardless of which of the two zones is the control zone. This result would be expected. As in the baseline case, these control zone/subzone cases compute design capacity based on the peak load in the front zone plus the peak load in the rear (unlike the case of the single 4,000 ft^2 zone discussed above). As a result, the total design flows in the control zone/subzone cases are identical to that for the baseline (3,810 cfm); the dates and hours for which the peak loads are computed are the same as for the baseline, so that the enthalpies of the air entering the cooling coils during the peak hours are the same; and, consequently, the design capacities are identical.

Energy consumption and cost. Energy consumptions and costs are moderately lower for the two control zone/ subzone cases, relative to the baseline. Diagnosis shows that 65 to 75% of this reduction is caused by reduced power consumption by the fan, and that almost all of the remainder results from reduced power consumption by the cooling coils.

The reduced fan power consumption occurs because one of the two zones (the subzone) is now being operated in a VAV mode. Thus, during most of the operating hours, flows to that subzone will be reduced relative to what they would be in the baseline, constant-volume case. Because the fan is moving less air, the power required to operate the fan decreases by about 500 to 800 kWh/year, depending upon which of the two zones is used as the control zone.

The reduced cooling coil power consumption results from a variety of reasons. One reason is that -- since fan heat dissipation is assumed to increase the temperature of the circulating air by 1.8 F° -- the reduced flow through the fan results in less heat being introduced into the air stream from this source. Another reason is that the average temperature in one or both zones will sometimes be a couple tenths of a degree higher in the control zone/subzone cases, relative to the baseline, so that sometimes slightly less cooling is being provided. In addition, the efficiency of the cooling coils varies slightly when the coils are operated at different fractions of full load -- and the fraction of full load operation during a given hour will vary somewhat between the baseline case and the control zone/subzone cases.

Table 16 indicates that energy consumption and costs are slightly lower when the rear zone is the control zone, compared to when the front zone is the control zone. For example, consumption is 1,105 kWh/year less than the baseline when the rear zone controls, and only 761 kWh/year less when the front zone controls, a difference of 344 kWh/year. Although this difference is small, it is of interest to understand why it occurs.

The control zone determines what the supply air temperature will be. Regardless of which zone controls, the supply air temperature determined by the control zone will sometimes be *lower* than that required for the other zone (the subzone). When the supply temperature is too low, flows to the subzone are reduced, thus reducing fan power consumption. On the other hand, regardless of which zone controls, sometimes the supply air temperature selected by the control zone will be *higher* than that required for the subzone. When supply temperature is too high, flows to the subzone rise to their design maximum, but the subzone is not cooled to the extent that it would have been had it been able to control its own supply temperature. When this occurs, energy consumption by the cooling coils is reduced relative to the baseline, because the subzone is no longer receiving the amount of cooling that it did with the baseline two-PSZ system.

Thus, when either one of the zones is treated as a subzone, there will sometimes be an energy savings due to reduced fan power consumption, and sometimes a savings due to reduced cooling coil consumption. Which of the two zones results in the greatest savings when considered as the subzone depends upon the mix and magnitude of the savings from these two sources. As it turns out, in this case, treating the front (north) zone as the subzone -- and the rear (south) zone as the control zone -- results in somewhat greater savings. Of the 344 kWh difference in energy consumption between these two cases, almost the entire savings results because the rear-control case requires less fan power. That is, the south-facing rear zone tends more often to demand cooler supply air than is required by the north-facing front zone, so that front zone flows are reduced to a greater degree. This result is intuitively reasonable.

Of the 344 kWh/year additional savings between the rear-control-zone and front-control-zone cases, only 10 kWh/year results due to reduced cooling coil (and heating coil) energy consumption. Both the rear-control-zone and the front-control-zone cases result in a cooling coil energy savings of about 250 kWh/year relative to the baseline. For some hours on some days, the front-control-zone case requires less cooling energy than the rear-control-zone case; for other hours, the situation is reversed. These hours apparently offset each other, so that the net effect is that both control-zone cases reduce cooling energy consumption to the same extent relative to the baseline.

Hours at elevated temperature or RH. For both of the control-zone/subzone cases, the hours undercooled (and some-times the hours at elevated RH) are lower, by a tiny amount, compared to the baseline case. This occurs despite the fact that, during occasional hours when the supply air temperature is too high, the subzone is receiving less cooling than it does in the baseline case, and is hence a couple tenths of a degree warmer. The reason for the slight net reduction in hours undercooled must be that the one large PSZ system in these cases, with about 10 tons of cooling capacity, sometimes has the reserve capacity necessary to handle peak loads in one of the zones -- peak loads that could not have been handled if that zone were conditioned by a dedicated 5-ton unit.

Taking one summer Monday morning (July 8) to provide an example, in the baseline case with two 5-ton PSZ units, the front zone is undercooled for the first four hours after the system begins operating. But when there is a single PSZ system with the front system serving as the control zone, the front zone is undercooled during only *one* hour. During the first two hours on that Monday morning, the supply air temperature to the front zone averages 1.8 F° cooler in the front-control-zone/rear-subzone case than in the baseline case, because the 10-ton coils have sufficient capacity to provide the additional cooling. On this same summer Monday morning, the number of undercooled hours in the *rear* zone is also reduced, from two hours (in the baseline case) to one hour (in the front-control-zone case).

Single PVAVS system conditioning both zones

With the two constant-volume PSZ units assumed in the baseline case, the flow to each zone remains constant; zone temperature is controlled by modulating the average supply air temperature. With the PVAVS, it is the supply air temperature that remains (relatively) constant (at about the minimum supply temperature of 55°F); in this case, zone temperature is controlled primarily by modulating the flow to each zone. In larger and more complex VAV systems, it is possible to modulate supply air temperature as well as flow rate; but that ability is not commonly included (or needed) in simple PVAVS.

Capacity. Referring to Table 16, the design cooling capacity for the two-zone packaged VAV system is 1.479 kBtu/h (1.4%) greater than the sum of the capacities for the two PSZ units in the baseline case. The sole reason for the higher PVAVS design capacity is that the default fan that DOE-2 assumes for PVAVS increases supply air temperature by 2.1 F° (due to dissipation of fan heat), whereas the temperature rise assumed for the PSZ fan is only 1.8 F°. The added cooling capacity computed for PVAVS is solely to remove this incremental additional fan heat during peak load. If one specifies a PVAVS fan that raises the supply air temperature by 1.8 F°, identical to the PSZ fan, the computed design cooling capacity becomes identical for PVAVS and PSZ.

This result would be expected. All of the parameters used by DOE-2 in calculating design cooling capacity are the same for both HVAC systems; i.e., system flows at peak load, and the date and hour of peak loads in each zone (and hence the enthalpies of the air entering the coils, as determined by weather, internal loads, and ventilation rate). The heat added by the air handling fan is the only variable used in the calculation that varies.

Energy consumption and cost. As shown in Table 16, the PVAVS results in a 12% reduction in HVAC energy consumption compared to the baseline two-PSZ system, the greatest reduction of any of the HVAC alternatives considered here. PVAVS also results in a 7% reduction in HVAC energy cost, again the greatest reduction, with the exception of the anomalous PTAC result. The only other parameters which enable an energy savings this great at 5 cfm/person are: dramatically reducing lighting and equipment power consumption (19% reduction in energy consumption, see Section 5.4); elimination of (or achieving infinite thermal resistance in) the windows, walls, and roof (20% reduction, Section 5.10); and the use of cold-air distribution (11% reduction, Section 5.14). Of these parameters offering the greatest potential, switching from a PSZ to a PVAVS is the most practical and easily achievable.

Diagnosis of these results indicates that, of the 3,149 kWh/year energy savings shown in Table 16 for PVAVS compared to the baseline two-PSZ system, 87% of this savings results from reduction in power consumption by the air handling fan. Almost all of the remaining 13% results from reduced energy consumption by the cooling coils.

The substantial reduction in fan power consumption results because, with the VAV system, the fan is almost always moving less air than in the baseline (constant-volume) case. Only at peak loads do variable flows reach the maximum value (3,810 cfm) that the constant-volume baseline fan is always moving whenever it operates. The power consumption of a given fan decreases with the cube of the volumetric flow rate through the fan (ASHRAE, 1992c). As shown in Table 4, the air handler represents 28% of the HVAC energy requirements in the baseline case. Hence, a major reduction in fan power consumption through reductions in flows would be expected to have an important impact on total system consumption. Switching from the baseline PSZ configuration to the PVAVS reduces fan power consumption by 2,742 kWh/year, or 37% (from 7,328 to 4,586 kWh/year).

The remaining 13% of the 3,149 kWh/year total reduction in HVAC energy consumption -- i.e., 407 kWh/year -- results primarily from a modest reduction in cooling coil energy requirements for the PVAVS relative to the two PSZ units. This reduction in cooling coil energy occurs because the reduced PVAVS flows result in less heat being introduced into the air through dissipation of heat from the air handler, causing a net reduction in the load on the coils.

As indicated previously, heat dissipation from the PVAVS supply fan is assumed in the computations to increase air temperature by 2.1 F°, compared to a 1.8 F° increase for the PSZ fans. The total heat added to the air stream by the fan is proportional to the volume of air flow multiplied by this increase in temperature across the fan. During hours when PVAVS flows are between about 85% and 100% of the combined PSZ flows in the baseline system, the total heat added by the PVAVS fan -- and hence the contribution of the fan to cooling coil energy consumption -- will be *greater* than the contribution from the PSZ fans. But when the PVAVS flows are less than 85% of the PSZ flows, the contribution of the PVAVS fan will be *less* than that of the PSZ fans. The net effect over the course of a year is for the PVAVS fan to contribute *less* heat to the air stream, creating the net reduction in cooling coil energy consumption observed here.

Hours at elevated temperature or RH. Table 16 shows that, with the PVAVS, the total number of hours undercooled each year is reduced to essentially zero. By comparison, with the baseline two-PSZ configuration, 0.4% of all hours are undercooled (about 30 hours per year in each zone).

This difference results because the DOE-2 simulation models the PSZ and PVAVS control systems in different ways. Upon startup on a warm summer morning, the PVAVS immediately begins operating at peak capacity, providing supply air near the minimum supply temperature (55 °F); the supply air temperature remains near this MIN-SUPPLY-T all day, with supply flows varying as necessary to accommodate the varying load. For example, on Monday morning, July 8, the PVAVS operates at maximum capacity for the first 3 hours after startup, and the space temperatures in each zone -- which were above 84 °F before startup -- average below 76 °F during the first hour after startup. This is within the 2 F° throttling range of the cooling setpoint (75 °F), and there are thus no undercooled hours on that day with the PVAVS.

By comparison, the two PSZ systems start up on that same Monday morning (July 8) operating at only 83 to 91% of maximum capacity over the first two hours, even though the zones are telling the system that they are undercooled. As a result, the 84 °F pre-startup zone temperatures are reduced only to an average of 77.7 °F during the first two hours of July 8, more than 2 F° above the 75 °F setpoint. Each zone is thus considered to be undercooled during those hours with the baseline PSZ configuration.

Table 16 also indicates that the percentage of occupied hours with relative humidities above 60% are about the same with the PVAVS and the baseline dual PSZ configuration. Although these percentages of hours at elevated RH are almost identical (1.5% vs. 1.2%), it is of interest to understand how these similar percentages result from two different control systems.

The hours at elevated indoor RH tend to occur during winter mornings in Miami, when the outdoor RH can be high (over 90%), but when the indoor temperature is low enough (about 70 °F) such that the cooling coils are operating at greatly reduced capacity. Sometimes in the winter, the cooling coils do not operate at all during the first few hours. (On *summer* mornings, the coils are operating close to maximum capacity beginning at startup, so that indoor RH levels are consistently below 60%.)

Take Monday morning, January 14, as an example. The office spaces are at 70 to 71 °F just before startup, well below the cooling setpoint of 75 °F that should trigger cooling. But anticipating that occupation will cause the temperature to rise, the PVAVS control equipment starts the system operating immediately in the cooling mode, an approach that maintains the offices well below 75 °F until early afternoon. Typical of PVAVS operation, the system provides supply air near to the minimum supply temperature beginning at startup; but because of the reduced cooling load (technically, the offices are over-cooled during the morning), the PVAVS supply air is provided at the minimum variable-volume flow throughout the morning hours.

The low PVAVS cooling coil temperature condenses a meaningful fraction of the moisture, causing a decrease in the indoor RH from its pre-startup value of over 80%. However, because the flows are at their minimum, this moisture removal is not sufficient to reduce the office RH below 60%. The RH ranges between 70 and 72% for the first four operating hours, and does not drop below the 60% target level until mid-afternoon on this particular Monday. The first nine occupied hours are above 60% RH on January 14 with the PVAVS, and the RH range during this 9-hour period is 60 to 72%.

By comparison, due to a different control approach, the baseline PSZ systems have their cooling coils entirely off during the first two to three hours on this Monday, allowing office temperatures to rise to about 74 °F before cooling begins. During those first occupied hours with the cooling coils off, the office RH rises significantly as a result of: a) latent heat release by the occupants; and b) mechanical supply of outdoor air by the activated HVAC system, containing moisture equivalent to $\geq$80% RH at office temperature. During these initial hours with the cooling coils inactive, the office RH rises to 90 to 100%, substantially higher than the maximum values with the PVAVS.

But once the cooling coils for the two PSZ units do come on, their combined hourly latent cooling rate soon surpasses that of the PVAVS. Even though the PSZ coil surface temperature is not as low as in the PVAVS during the initial hours, the fact that the PSZ units are treating a much greater quantity of air results in significantly greater total moisture removal. As a result, with the baseline PSZ system, the office RH drops below 60% within 6 hours in the front office, and within 4 hours in the rear office, compared to 9 hours with the PVAVS.

In summary, the PSZ units result in elevated office RH for fewer hours, as indicated in Table 16. However, during those fewer hours, the PSZ RH levels can reach much greater values than those seen with the PVAVS, since the PSZ coils can be off completely.

One PTAC unit serving each zone

As shown in Table 16 -- in comparison with the baseline dual-PSZ configuration at 5 cfm/person -- the use of PTAC units would appear to decrease the cooling capacity requirement by 6.5% (i.e., by 6.718 kBtu/h), the annual HVAC energy use by 10% (by 2,719 kWh), and the HVAC energy cost by 9% (by $232). But as indicated previously, this result is an artifact. An attempt in practice to cool a 2,000 ft^2 zone with an unducted, through-the-wall PTAC unit would result in the occupants near the unit being uncomfortably cool, and those remote from the unit being uncomfortably warm.

Most of the apparent savings with the PTAC units results because the default PTAC air handler is a low-power, low-static-pressure fan, since this fan does not have to move air through either supply or return ducting. This default PTAC fan provides a static pressure rise of only 0.3 in. WG (compared to 3.0 in. WG for the default PSZ air handler), consumes only 0.000070 kW per cfm of air moved (12% of the value for PSZ, 0.000587 kW/cfm), and raises the air temperature by only 0.2 F° (compared to 1.8 F° for the PSZ fan).

The PTAC computations were repeated with the power consumption and the temperature rise for the PTAC fan set equal to the values for the PSZ fan (0.000587 kW/cfm, 1.8 F°), rather than being allowed to default to the low PTAC values. With these fan values:

a) the required PTAC cooling capacity rose almost to the value of the baseline PSZ system. That is, the 6.718 kBtu/h decrease shown in Table 16 dropped to almost zero.

b) the annual HVAC energy consumption by the PTAC system rose dramatically. The 2,719 kWh *decrease* shown in Table 16 (relative to the PSZ baseline) became a 6,150 kWh *increase* over baseline consumption, since the PTAC units (having a default electric input ratio of 0.438) are inherently less efficient than the PSZ units (with an EIR set equal to 0.341).

c) correspondingly, the annual HVAC energy cost for the PTAC system rose substantially, from a \$232 *savings* compared to PSZ costs, to a \$515 *increase* over PSZ.

These figures indicate that the very low default performance (and hence very low power consumption) of the PTAC fans is largely responsible for the apparent PTAC savings shown in Table 16. It is this low fan power consumption (and the resulting lack of static pressure to provide effective air distribution) that makes the PTAC system generally not a good choice for this application.

In addition to low PTAC fan power consumption, another, secondary contributor to the reduced cooling load and power requirements of PTAC systems might be that the PTAC system is not conditioning the overhead plenum to the same extent.

With the baseline PSZ configuration, the 4-ft-high space above the offices is used as the return air plenum. As such, it is being swept with conditioned office air whenever the HVAC fan is operating. Consequently, it is always within a couple degrees of the office temperature -- and sometimes within a couple tenths of a degree -- during fan operation. By comparison, with the PTAC units -- which draw the return air directly from the offices -- this overhead space is simply dead space which exchanges heat with the offices only via conduction through the uninsulated ceiling, but with no air exchange. As a result, with the PTAC units, the unused plenum can average perhaps 3 F° warmer on a summer day than it does when it serves as the PSZ plenum. This increased PTAC plenum temperature represents heat that does not have to be removed by the PTAC units, and hence, a potential savings in PTAC cooling capacity and power requirements. On the other hand, the heat that *does* conduct through the ceiling into the offices is removed less efficiently by the PTAC units; plenum heat swept directly into the PSZ cooling coils by return air is removed more efficiently.

In Section 5.13, results are reported for the case where return air to the PSZ system is accomplished using (insulated) ducts, rather than using the overhead plenum as in the baseline PSZ case. Those results indicate that net effect of avoiding the partial conditioning of the plenum -- but of then having to remove some of that plenum heat after it has conducted into the office space -- seems to be a small savings in energy consumption. Extrapolating that PSZ result to the PTAC case, it is likely that the fact that the PTAC units are avoiding partial conditioning of the overhead space is only a minor part of the reason why the PTAC units require less power than the baseline PSZ case. The default low power requirements of the PTAC fans are the predominant explanation.

<u>Hours at elevated temperature or RH</u>. Table 16 indicates that the PTAC systems result in slightly fewer hours undercooled than do the PSZ systems (0.2%

compared to 0.4%). This effect results because the PTAC units are calculated to provide cooler supply air during the first couple hours after startup on warm summer mornings. For example, during the first two hours after startup on Monday, July 8, the computed temperature of the air leaving the PTAC cooling coils is 1 to 5 F° cooler than the air leaving the PSZ coils. The PTAC units are operating at a greater percentage of full capacity during those first two hours, compared to the PSZ units. Consequently, while the PSZ systems result in the zones being undercooled for the first four hours on July 8, the PTAC units result in undercooling for only one or two of those first four hours (depending on the zone). This effect might result from differences in the manner by which the PTAC vs. PSZ control systems are simulated by the program, and/or from the fact that the less powerful PTAC fans add less heat to the air stream.

Table 16 also indicates that, according to the calculations, the PTAC systems reduce the number of occupied hours above 60% RH to about zero. The PTAC coils do tend to operate at a lower temperature than the PSZ coils under some conditions, consistent with this prediction that they might remove more moisture. However, diagnosis indicates -- if PTAC units *do* result in fewer elevated RH hours than do PSZ units -- the difference is not as great as that suggested in Table 16, and PTAC units do *not* in fact reduce elevated-RH hours to zero. The apparent PTAC vs. PSZ RH effects in Table 16 are believed to result from errors in the DOE-2 simulation and reporting.

To illustrate, the morning of Monday, January 14, is taken as an example. During the first three to four hours after startup on that day, the cooling coils are not operating in the front zone with either PTAC or PSZ; the system is either in the heating mode, or is just ventilating with neither the heating nor cooling coils operating. Thus, neither system is removing moisture from the air during those hours. Yet, the RH in the front zone rises to 96 to 100% with the PSZ system (reaching a high of 0.0210 lb moisture per lb dry air) -- whereas, with the PTAC system, RH is calculated to hold at 77 to 84% (its maximum being 0.0144 lb/lb, lower than the moisture content of the outdoor air, despite indoor latent sources). Since the DOE-2 program cannot address the moisture capacitance of indoor materials, moisture capacitance is not contributing to these apparently anomalous results. Accordingly, it seems clear that the program is underestimating the moisture levels with the PTAC system, and/or is overestimating moisture levels with the PSZ system.

It is also noted that there must be an error in DOE-2's relative humidity scatter plot report for the PTAC system (the "SS-N" report), on which Table 16 is based. The SS-N report suggests that, with the PTAC system, the RH will be above 60% for only one occupied hour (in the front zone) during the entire year. But hourly reports for January 14 with the PTAC system show RH's above 60% (actually, above 70%) for four occupied hours in the front zone, and for one hour in the rear zone, on that day alone. [By comparison, the PSZ system is computed to be above 60% (and above

70%) for five hours in the front zone on that day, and for three hours in the rear zone.]

5.12.2 The Effect of Alternative HVAC Systems on Increased Ventilation Rates

As discussed in Section 4, and as repeated in Table 16, increasing the OA ventilation rate from 5 to 20 cfm/person with the baseline two-PSZ/two-zone system increases the required cooling capacity by 15.635 kBtu/h, the energy consumption by 3,245 kWh/yr, and the energy cost by $325/yr.

Alternative PSZ configurations

For the three alternative HVAC systems in Table 16 that are PSZ-based -- the one-PSZ/one-zone configuration and the two one-PSZ/zone/sub-zone configurations -- increasing the ventilation rate to 20 cfm/person increases capacity requirements, energy consumption, and costs by essentially identical increments above each configuration's respective values at 5 cfm/person. This is consistent with the expectation that such similar HVAC configurations will respond to the increased OA sensible and latent loads in the same manner.

For example, at 20 cfm/person, the one-PSZ/one-zone configuration requires an increase in capacity of 15.572 kBtu/h relative to this same configuration at 5 cfm/person, essentially identical to the 15.635 kBtu/h OA-induced increase for the baseline system. But the capacity of this one-PSZ/one-zone configuration at 5 cfm/person was 3.060 kBtu/h less than that required for the baseline two-PSZ/two-zone system at 5 cfm/person, for the reasons discussed in Section 5.12.1. As a result, the capacity for the one-PSZ/one-zone configuration at 20 cfm/person is increased by only (15.572 - 3.060 =) 12.512 kBtu/h relative to the baseline two-PSZ/two-zone system at 5 cfm/person, the figure shown in Table 16. Similarly, the increase in energy consumption for the one-PSZ/one-zone configuration at 20 vs. 5 cfm/person is 3,235 kWh/yr (essentially identical to the baseline's 3,245 kWh/yr), and the increase in energy cost is $323/yr (identical to the baseline's $325/yr).

Single PVAVS conditioning both zones

Capacity. For the PVAVS, the required increase in capacity for the increase from 5 to 20 cfm/person is 15.934 kBtu/h, slightly greater than the OA-induced increase for the baseline two-PSZ system. (As a result, the required PVAVS capacity at 20 cfm/person is 15.934 + 1.479 = 17.413 kBtu/h greater than that for the PSZ system at 5 cfm/person, as shown in Table 16.) This minor difference results from the greater temperature rise in the air stream assumed across the default PVAVS air handler (2.1 vs. 1.8 F° for PSZ), the same issue discussed for PVAVS in Section 5.12.1. Reducing the assumed PVAVS fan temperature rise to 1.8 F° results in a computed capacity increase for the PVAVS at 20 vs. 5 cfm/person of 15.614 kBtu/h, essentially identical to the baseline PSZ system.

Energy consumption and cost. The OA-induced increase in energy consumption for the PVAVS is only 3,070 kWh/yr, somewhat lower than the 3,245 kWh/yr increase for the baseline case. This 175 kWh/yr difference results almost entirely because cooling coil energy consumption (i.e., consumption by the compressor and the condenser fan) increases to a lesser degree with increasing OA in the case of the PVAVS, compared to the PSZ system. Changes in consumption by the air handler and the heating element are *not* significant contributors to this difference.

PVAVS cooling coil energy consumption is impacted less by an increase from 5 to 20 cfm/person because the hourly EIRs for the PVAVS are impacted slightly differently by the OA increase than the EIRs for the PSZ system are impacted. The EIR during any given hour for a given system is determined, in part, by the part-load ratio (i.e., the fraction of full capacity) at which the system is operating. The default equation relating EIR to part-load ratio is the same for the PVAVS and the PSZ system. However, due to the inherent characteristics of PVAVS vs. PSZ, an increase in OA from 5 to 20 cfm/person impacts the hourly part-load ratios (and hence the EIRs) slightly differently. Therefore -- even though an increase from 5 to 20 cfm/person should increase the load to the same extent for both systems -- the differences in EIRs results in a different number of kilowatt-hours being required to address this load. This is the cause of the net 175 kWh/yr difference indicated in the preceding paragraph.

Due to the lesser increase in PVAVS energy consumption resulting from the increased OA -- and since PVAVS at 5 cfm/person was consuming 3,149 kWh/yr less than PSZ at 5 cfm/person -- Table 16 estimates that a PVAVS could operate at 20 cfm/person in this building, while consuming 78 *fewer* kWh/yr than the baseline dual PSZ system at 5 cfm/person. Thus, switching from a dual PSZ system to a PVAVS in designing a new building could theoretically enable an increase in OA ventilation rate without an increase in energy consumption.

Increasing the PVAVS from 5 to 20 cfm/person increases energy *cost* by $310 per year. This is slightly less than the OA-induced increase in PSZ energy costs ($325), consistent with the fact that, with PVAVS, the OA-induced increase in energy *consumption* is 175 kWh/yr less, as discussed above. Thus, the PVAVS is estimated to be able to deliver 20 cfm/person at an annual cost only $124 greater than that of the baseline PSZ system delivering only 5 cfm/person, as indicated in Table 16.

Hours at elevated temperature or RH. Table 16 shows that, at 20 cfm/person, the PVAVS is estimated to result in essentially no occupied hours being undercooled, just as at 5 cfm/person. The reasons why PVAVS gives no undercooled hours, whereas the PSZ-based systems do result in some undercooling, are the same as those given previously (Section 5.12.1). Due to its control system, the PVAVS operates closer to full capacity than the PSZ units during startup on summer weekday mornings, when the PSZ undercooled hours occur; correspondingly, PVAVS delivers more power to its cooling coils (i.e., to the compressor and the condenser fan) during those hours.

Table 16 indicates that, at 20 cfm/person, the occupied hours at RH levels above 60% approximately doubles with the PVAVS (to 2.9% of all occupied hours, or 94 hours per year), compared to the 5 cfm/person case. By comparison, the PSZ-based systems all show a tiny *decrease* in elevated-RH hours with the increase from 5 to 20 cfm/person. On winter weekday mornings, when the elevated-RH hours occur, the increase in the PVAVS latent cooling resulting from the increased OA does not compensate for the increase in latent load with the increased OA. By comparison, with the PSZ systems, the increased latent cooling *does* compensate for the increase in load. This effect results from the differences between the PVAVS and PSZ control systems.

Consider Monday, January 14 -- the same day addressed previously for the 5 cfm/person case. In terms of flows and temperatures, the PVAVS performs the same at 20 cfm/person as at 5 cfm/person on that day. The hourly flows to each zone -- and the hourly temperatures of each zone, the supply air, and the cooling coil surface -- are essentially identical, regardless of the OA rate. This similarity results because the externally induced and occupant-induced thermal loads on the zones are independent of OA rate, and because of the characteristics of the PVAVS control system (which modulates supply air flow at a fairly steady temperature near the minimum supply temperature). Because the mixed air entering the coils has a higher latent load with the increased intake of humid outdoor air at 20 cfm/person, the identical total flows and coil temperatures result in an increase in the amount of moisture removed relative to the 5 cfm/person case. Increasing the PVAVS OA rate from 5 to 20 cfm/person increases latent cooling by over 50% (from 82,600 to 125,000 Btu/day) on January 14. Latent cooling is increased from 19% of total cooling at 5 cfm/person, to 25% of total cooling at 20 cfm/person.

But this increase in PVAVS latent cooling is insufficient to compensate for the increased latent load. At 20 cfm/person, the hourly moisture content of the supply air leaving the PVAVS coils (lb moisture per lb dry air) is 5 to 15% higher than the content at 5 cfm/person. The RH in the office space during occupied hours reaches higher maximum levels during the initial hours after startup (78% at 20 cfm/person, compared to 72% at 5 cfm/person). And the office RH remains above 60% during all 13 occupied hours on January 14, whereas at 5 cfm/person, the RH was above 60% only during the first 9 hours with the PVAVS.

The PSZ-based systems react differently to the increase in OA. As with the PSZ system at 5 cfm/person, the PSZ cooling coils remain off in each zone for the first two to four hours after startup at 20 cfm/person, allowing the humidity to increase as the result of occupant latent heat. But due to the increased in-flow of outdoor air at 20 cfm/person -- outdoor air whose moisture content corresponds to about 80% RH at office temperature -- the maximum humidity levels reach only about 90% RH at 20 cfm/person, compared to 90 to 100% at 5 cfm/person.

Like the PVAVS, the total PSZ flows remain unchanged as OA is increased, as do the office temperatures and the supply air temperatures. But since the PSZ control system is designed to modulate supply air temperature at constant flow, the PSZ system -- unlike the PVAVS -- reduces coil temperature (sometimes by several degrees or more) in response to the greater sensible and latent loads in the mixed air entering the coils when OA is increased.

Thus -- once the PSZ coils do come on, after remaining off during the first few hours -- the lower coil temperatures at 20 cfm/person result in greater moisture removal than occurred at 5 cfm/person. The moisture content (lb of moisture per lb of dry air) of the supply air leaving the PSZ cooling coils during the first one to two hours of operation is 5 to 15% *lower* at 20 cfm/person than at 5 cfm/person. (By comparison, with PVAVS, the moisture content was 5 to 15% *higher* at 20 than at 5 cfm/person.) As a result, the RH in the office space drops slightly more rapidly at 20 than at 5 cfm/person during the first hour after the coils come on. However -- on January 14, at least -- the offices remain above 60% RH for exactly the same number of hours at 20 cfm/person with the PSZ system as they did at 5 cfm/person (6 hours in the front, 4 hours in the rear). This is consistent with the results shown in Table 16: The percentage of occupied hours above 60% RH with the PSZ system at 20 cfm/person is not significantly different than the percentage with the PSZ system at 5 cfm/person, and the slight change that *does* occur is in the direction of reduced elevated-RH hours at the higher OA rate.

Further comparing the effect of increased OA on the PVAVS vs. the PSZ system, with the baseline two-PSZ configuration, increasing the OA rate from 5 to 20 cfm/person increases the latent cooling by 87% on January 14, from a combined total of 78,000 to 146,000 Btu/day. By comparison, as indicated previously, the increased OA increased PVAVS latent cooling by only 50%. At 20 cfm/person, the dual PSZ configuration provides greater latent cooling than the PVAVS (146,000 vs. 125,000 Btu/day). In the PSZ system, latent cooling is increased from 20% of total cooling at 5 cfm/person, to 32% of total cooling at 20 cfm/person (compared to 25% with PVAVS). As a result, the increase in latent cooling by the PSZ system at 20 cfm/person compensates for the increase in latent load caused by the higher OA rate.

In summary, when the OA rate is increased from 5 to 20 cfm/person on January 14, the PVAVS experiences an increase in the number of occupied hours at RH > 60% (13 rather than 9), and an increase in the maximum RH levels (78% rather than 72%). By comparison, the PSZ system experiences no significant change in the number of elevated-RH hours, and a *de*crease in the average RH during those elevated hours (about 78% rather than 85%).

One PTAC unit serving each zone

Capacity. For an increase from 5 to 20 cfm/person, the required increase in the total PTAC capacity is 14.870 kBtu/h, somewhat less than the 15.635 kBtu/h increase for the baseline PSZ configuration. Diagnosis indicates that this somewhat reduced impact of OA in PTAC capacity is due to the different approach used by the DOE-2 software in computing capacities for zonal systems such as PTAC vs. central systems such as PSZ (Winkelmann et al., 1993). It is *not* the result of differences between the PSZ and PTAC default values for the key variables that influence the capacity calculation. Specifically, the reduced impact of increased OA on estimated PTAC capacity is *not* due to the reduced temperature gradient across the PTAC supply fan (as discussed in Section 5.12.1), the higher PTAC coil bypass factor, or the different standard equations expressing bypass factor, total capacity, and sensible capacity as functions of wet- and dry-bulb temperatures. Even if the values for all of these parameters are set the same for PTAC as for PSZ, the impact of 5 vs. 20 cfm/person on the computed PTAC system capacity remains at about 14.7 kBtu/h. Correspondingly, even with these "corrections", the computed PTAC capacity at 20 cfm/person remains about 1.5 kBtu/h lower than PSZ capacity at 20 cfm/person. (By comparison, in Table 16, PTAC is 15.635-8.152 ≈ 7.5 Btu/h lower).

The difference in computed PTAC capacities is also not due to additional heat picked up in the plenum by the return air in the PSZ case. Except for heat released into the plenum by the office lights (of which there is none in this case), plenum heat does not factor into the capacity calculation.

Energy consumption and cost. As shown in Table 16, two PTAC units operating at 20 cfm/person use less power than the baseline PSZ system at 20 cfm/person (by 3,245 - 1,944 = 1,301 kWh/yr), and have a lower energy cost (by $325 - $228 = $97). But the *increases* in power consumption and cost for PTAC at 20 cfm/person vs. PTAC at 5 cfm/person are significantly *greater* than the increase for PSZ with that increase in OA. Specifically, increased OA causes PTAC power consumption to rise by 2,719 + 1,944 = 4,663 kWh/yr (compared to 3,245 kWh/yr for PSZ), and energy cost to rise by $232 + $228 = $460/year (compared to $325 for PSZ). This result is no surprise. The PTAC unit cools less efficiently than does PSZ (cooling EIR = 0.438 for PTAC, vs. 0.341 for PSZ), so that the incremental increase in power usage resulting from the OA load increase is correspondingly greater.

As discussed in Section 5.12.1, the primary reasons why PTAC at 20 cfm/person is consumes less power than PSZ at 20 cfm/person is that the default PTAC supply fan is assumed to consume far less power and to add far less heat to the supply air. If the PTAC calculations are repeated assuming PSZ fan characteristics, the inefficient PTAC system at 20 cfm/person consumes about 25% *more* power than the PSZ system at 20 cfm/person, at about the same percentage increase in energy cost.

5.13 DUCTED AIR RETURNS

The baseline configuration assumes that the office air returns to the HVAC system by way of the overhead plenum spaces, without return ducting. This common approach results in some effective conditioning of the plenum space by the return air, such that the temperatures in the plenums are generally within a degree or two of the temperatures in the conditioned offices.

It is also possible to model the case where ducting is provided for the return air. In a small office such as this, it would be common for such return ducting to be routed through the overhead spaces, although, for the purposes of the DOE-2 modeling, it is not necessary to specify exactly where the ducting is located. The model assumes that the ducts are perfectly insulated, i.e., that there is no heat transfer between the return air (inside the ducting) and the space outside the ducting. The one exception is that any energy from the office lighting that is not released into the conditioned space is assumed to be released into the return ducting. But in this study, all energy consumed by the lights has been assumed to be released into the office space (LIGHT-TO-SPACE = 1); consequently, no heat would be added to the air in the return ducting.

Table 17 shows the results when the office air is assumed to return to the baseline PSZ units via return ducting rather than via the overhead plenum.

5.13.1 The Effect of Ducted Returns at 5 cfm/person

Capacity. As shown in Table 17, ducting the returns has no effect on the computed cooling capacity of the PSZ system at 5 cfm/person. Ducting has no effect on capacity because -- even when the return air flows through the overhead plenum -- the capacity calculations do not take into consideration any of the heat picked up in the plenum (Winkelmann et al., 1993). The one exception is that the lighting energy that is not released into the office space is added to the return air stream in the capacity calculation -- and that exception applies regardless of whether or not there is return ducting. Thus, the presence of perfectly insulated return air ducting does not influence the cooling capacity calculation.

Energy consumption and cost. But as shown in Table 17, there is some small reduction in HVAC energy consumption and costs as a result of the return ducting (298 kWh and $35 per year). This modest savings is the net result of two offsetting phenomena.

The first of these phenomena is that ducting the returns avoids the partial conditioning of the plenum space by the return air. The overhead space now serves as a 4-ft-thick layer of air insulation between the outdoors and the conditioned space, thus reducing the load on the cooling coils. When that space is not serving as a return plenum, and is thus not being swept by return air from the offices, the tempera-

TABLE 17

Effect of Ducted Air Returns: Increase Compared to Baseline Case with 5 cfm/person

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values: Air return via plenum, OA=5 cfm/p (absolute values)[1]	103.558[1]	--	26,145[1] (HVAC)	--	--	2,510[1] (HVAC)	--	--	0.4	1.2
			60,161[1] (building)	--	--	4,273[1] (building)	--	--		
OA ventilation rate = 5 cfm/person										
Air return via duct	0	0	-298	-1.1	-0.5	-35	-1.4	-0.8	0.5	1.2
OA ventilation rate = 20 cfm/person										
Air return via duct	15.635	15.1	3,089	11.8	5.1	311	12.4	7.3	0.6	0.8

[1] These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

ture of the overhead space can average perhaps 3 F° warmer on a summer day than in cases where this space *does* serve as a return plenum. That 3 F° represents a sensible cooling load from which the coils are being spared.

The second phenomenon is that some portion of this sensible heat in the overhead space flows into the offices via conduction through the ceiling, which is not insulated in the baseline building. Once the conducted heat is in the office space, it must be removed by circulating the bulk office air through the cooling coils. The removal of this heat from the larger mass of office air will be less efficient than it would have been had the warm plenum air been swept directly into the cooling coils, as it would have been had the overhead space been being used as a return plenum.

So the offsetting phenomena are: a) return air ducting reduces the amount of overhead heat that has to be removed by the system; but b) the portion that *does* have to be removed is removed less efficiently. The relatively small difference in energy consumption and cost between the plenum- and ducted-return cases shows that these two phenomena just about cancel each other.

The modest 298 kWh energy savings shown in Table 17 for the ducted-return case would increase significantly if the office ceiling were insulated, so that less of the heat in the overhead space could conduct down into the conditioned space. For example, if it were assumed that the uninsulated office ceiling were instead insulated with R-30 batt insulation (in the extreme), the savings from switching from plenum return to (perfectly insulated) ducted returns at 5 cfm/person would increase to 1,388 kWh and $178 per year.

<u>Hours at elevated temperature or RH</u>. Table 17 shows that ducting the return air results is a tiny increase in the percentage of hours undercooled (0.5% vs. 0.4%). During startups on summer weekday mornings, the return air in the baseline unducted case is warmer, due to the heat in the plenum. This causes the PSZ units to have a slightly higher actual capacity, and to operate at a greater fraction of total capacity (a higher part-load ratio), than is the case with the identical PSZ units in the ducted case. Under these circumstances, without ducting, the plenum space cools rapidly, reducing heat conduction through the ceiling from the plenum into the office space. The net result of these phenomena is that the office temperatures are consistently computed to be a couple tenths of a degree cooler in the unducted case, compared to the ducted case.

This small temperature difference explains why the baseline unducted plenum case has a tiny percentage fewer undercooled hours. For example, on Monday, July 8, the rear zone in the ducted case is undercooled during the first three hours after startup (with the third hour being a tenth of a degree above the 77 °F limit that defines the zone as being undercooled). But in the baseline unducted case, the rear zone is undercooled only during the first two hours (with the third hour being a couple tenths of a degree below the 77 °F maximum, and thus not undercooled).

Table 17 indicates that, at 5 cfm/person, the percentage of occupied hours above 60% RH remains unchanged at 1.2% as the air return is changed from unducted to ducted. Actually, ducting the returns *does* have a small impact on the office RH values, but not enough to change the percentage of hours that are above 60%.

Elevated-RH hours tend to occur after startup on cool, humid winter mornings. When the plenum is used for the unducted return under these circumstances, it serves to cool the return air during the initial hours, until the outdoor temperature warms up later in the morning. But with ducted returns -- insulated from the cool plenum -- the return air does not receive this plenum cooling. Accordingly, with the ducted returns, slightly more power is put into the cooling coils during these initial hours, compared to the baseline unducted case, and the humidity levels in the supply air (and in the offices) are somewhat lower during these hours.

Take, for example, the morning of Monday, January 14. The front office is at an RH level above 60% for the first six hours after startup, regardless of whether the return air is ducted or unducted. This is consistent with the indication in Table 17 that the percentage of occupied hours above 60% RH remain unchanged when the returns are ducted. However, the *average* RH in the front office during those first six hours decreases from 88% in the baseline unducted case, to 82% in the ducted-return case.

Of course, on warm summer mornings, the reverse occurs. The baseline unducted return air now *picks up* (rather than *loses*) heat in the plenum. So under these circumstances, it is now the unducted case that consumes more cooling coil power, and provides slightly lower-humidity supply air, during the initial hours. However, during the summer, the RH levels in the offices are generally well below 60% with or without return air ducting, so that this effect has no impact on the number of occupied hours above 60%.

5.13.2 The Effect of Ducted Returns on Increased Ventilation Rates

The relationship of the ducted-return results in Table 17 to the unducted results is the same at 20 cfm/person as it is at 5 cfm/person, and for the same reasons discussed in Section 5.13.1.

<u>Capacity</u>. With ducted returns at 20 cfm/person, the required cooling capacity increases by 15.635 kBtu/h above the unducted 5 cfm/person baseline. This is exactly the same increase as for the *un*ducted 20 cfm/person case, for the same reasons discussed previously.

Energy consumption and cost. With ducted returns at 20 cfm/person, annual HVAC energy consumption and costs increase by 3,089 kWh and $311, respectively, relative to the unducted 5 cfm/person baseline. As expected, these figures are slightly less than the 3,245 kWh and $325 annual increases predicted for the *un*ducted 20 cfm/person case.

Hours at elevated temperature or RH. The percentage of hours undercooled for the ducted 20 cfm/person case (0.6% in Table 17) is slightly higher than the 0.4% for the unducted 20 cfm/person case, for the same reasons discussed in the preceding section.

The percentage of occupied hours above 60% RH for the ducted case at 20 cfm/person (0.8% in Table 17) is slightly lower than the 0.9% for the *un*ducted case at 20 cfm/person. The reason for this is that -- in the ducted case -- slightly more power is consumed by the cooling coils during the first few hours after startup on winter weekday mornings. The ducted return is not cooled by the cool plenum during these early hours, as occurs in the unducted case. Thus, in the ducted case, there is a slightly increased cooling coil load during the early hours, and hence slightly increased latent cooling.

For example, on the morning of Monday, January 14, operating at 20 cfm/person, the cooling coils provide over 1,000 Btu of additional latent cooling during the first 4 to 5 hours after startup in the ducted case, relative to the unducted case. The front and rear plenum spaces range between 69 and 72 °F during these early hours in the ducted case, indicating that the relatively cool plenums could have contributed to cooling the office space below the cooling setpoint of 75 °F had the ducting not kept the plenum air isolated.

The reason why the percentage of occupied hours at elevated RH is slightly reduced at 20 vs. 5 cfm/person with the ducted PSZ system is the same as that for the *un*ducted PSZ system, as discussed in Section 4.2 (see *Performance*) and Section 5.12.2 (see *Single PVAVS conditioning both zones*).

5.14 USE OF COLD-AIR DISTRIBUTION

The previous sections have assumed that the minimum temperature of the supply air being distributed to the offices during cooling is 55 °F. This is a typical value for the minimum supply temperature, selected in an effort to remain above the dew point of the supply air and thus to reduce the risk of moisture condensation on the ductwork, and at the diffusers.

In some applications, consideration is sometimes given to designing the HVAC system to operate at cooling supply air temperatures well below 55 °F (i.e., to the use of "cold-air distribution"). Operation at lower supply air temperatures would have the advantage of reducing the volume of cooling air that must be distributed. Since power consumption by a given fan is proportional to the cube of the volumetric flow rate through the fan, a decrease in fan flow rate could result in a significant decrease in fan power consumption. The reduced flow rates to the zones would also offer the advantage of smaller duct size.

The possible disadvantages of cold-air diffusers include: a) the need for more sophisticated designs and controls to reduce the risk of moisture condensation on the ductwork and at the diffusers; and b) the possible need for fan-assisted ("powered") terminals in order to achieve adequate "throw" of the reduced volume of air out through the diffusers into the office space.

For this assessment, the use of cold-air diffusers was considered, with a minimum supply air temperature of 42 °F, for both the baseline dual-PSZ configuration and for the PVAVS (which substantially reduces fan power consumption even at 55 °F). The temperature of 42 °F is a typical value for cold-air diffuser designs, high enough to keep cooling coil surface temperatures above 34 to 35 °F, thus avoiding frosting on the outside of the coils.

It must be emphasized that this analysis of cold-air diffusers in this application is academic, simply to determine what the effect *could* be. It is unlikely that cold-air distribution would be used in an application such as this in practice. Strip mall offices of the type assumed here tend to have simple mechanical systems to reduce installation cost and to simplify maintenance. The complications that would be added by a cold-air diffusion system would be inconsistent with the usual desire for a simple system in this application. Moreover, in such a small office, reduction in duct installation cost (one of the benefits of cold-air distribution) might not be fully realized.

The results of the computations regarding cold-air distribution are presented in Table 18.

5.14.1 The Effect of Cold-Air Distribution at 5 cfm/person

Capacity - PSZ system. Table 18 indicates that the use of cold-air distribution results in a very small (0.514 kBtu/h) reduction in the cooling capacity of the two-PSZ configuration, relative to the baseline case with 55 °F supply air. This occurs because the increase in latent cooling capacity required for the cold-air system (since more moisture is condensed at the lower supply temperature) is slightly more than offset by the decrease in required sensible cooling capacity resulting from the reduction in the amount of heat added to the air stream by the supply fan (since the fan is now having to move less air).

Because the supply air is cooled by an additional 13 F° in the cold-air case, the required flow drops from the baseline 3,810 cfm value to 2,263 cfm, a 40% reduction, based on a straightforward thermal balance. At 5 cfm/person, the colder coils

TABLE 18

Effect of Cold-Air Distribution:
Increase Compared to Baseline Case with 5 cfm/person

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH > 60%, %
Baseline values: Two PSZ units, MIN-SUPPLY-T = 55 °F, OA = 5 cfm/p (absolute values)[1]	103.558[1]	--	26,145[1] (HVAC)	--	--	2,510[1] (HVAC)	--	--	0.4	1.2
			60,161[1] (building)	--	--	4,273[1] (building)	--	--		
OA ventilation rate = 5 cfm/person										
Two PSZ units, MIN-SUP-T = 42°F	-0.514	-0.5	-2,930	-11.2	-4.9	-268	-10.7	-6.3	0.1	0.7
One PVAVS unit, MIN-SUP-T = 42°F	0.339	0.3	-4,941	-18.9	-8.2	-392	-15.6	-9.2	~0	0.3
OA ventilation rate = 20 cfm/person										
Two PSZ units, MIN-SUP-T = 42°F	23.051	22.2	1,160	4.4	1.9	157	6.2	3.7	0.7	0.3
One PVAVS unit, MIN-SUP-T = 42°F	24.119	23.3	-823	-3.1	-1.4	39	1.6	0.9	0.2	0.6

[1] These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

in the cold-air case can remove perhaps 2.6 additional pounds of water per hour from that reduced air stream on a typical summer day, relative to the baseline case. At a latent heat of vaporization of approximately 1,050 Btu/lb at coil temperatures, the additional 2.6 lb of water removal would correspond to a required increase in latent capacity of about 2.7 kBtu/h in the cold-air case. But the reduction in flow rate would decrease the sensible heat added by the fan (and hence the required sensible capacity) by about 3.3 kBtu/h. This simple analysis shows why switching the dual-PSZ configuration to cold-air operation would be expected to reduce the required total cooling capacity by < 1 kBtu/h for the building being considered here.

Changing to cold-air operation also impacts some of the other parameters used by the DOE-2 program to compute cooling capacity, including: a) the coil bypass factor; and b) the wet-bulb temperature in the offices (used to convert capacities to values at the conditions specified by the Air-Conditioning and Refrigeration Institute). These factors lead to the precise value of 0.514 kBtu/h as the computed reduction in cooling capacity for the cold-air case at 5 cfm/person.

Capacity - PVAVS. A PVAVS operating with cold-air distribution requires a cooling capacity of 103.897 kBtu/h. As shown in Table 18, this is very slightly (0.339 kBtu/h) larger than the 103.558 kBtu/h computed for the baseline PSZ system with the standard 55 °F supply air temperature. It is 0.853 kBtu/h greater than the 103.044 kBtu/h computed for the PSZ system with the 42 °F supply, and 1.140 kBtu/h *less* than the 105.037 kBtu/h for the PVAVS with 55 °F supply.

The PVAVS at 42 °F requires less capacity than the PVAVS at 55 °F for the same reasons that the PSZ at 42 °F requires a lower capacity than the PSZ at 55 °F. The PVAVS at 42 °F condenses more moisture than its 55 °F PVAVS counterpart, and thus requires a greater latent capacity. But this latent increase is more than offset by a reduction in the sensible capacity requirement, resulting from reduced supply fan heat due to the reduced air flows.

The reduction in supply air temperature causes a slightly greater reduction in the computed capacity for a PVAVS (1.140 kBtu/h) than for a PSZ system (0.514 kBtu/h). This occurs because the default PVAVS supply fan is assumed to add more sensible heat to the air stream (2.1 F°) than the PSZ fan (1.8 F°). Hence, a given reduction in peak air flow rate results in a somewhat greater reduction in fan-imparted sensible heat for the PVAVS. If the PVAVS computer runs were made specifying a supply fan temperature rise of 1.8 F°, identical to the PSZ system, then the PVAVS capacity at either supply temperature would become essentially equal to the PSZ capacity at the same temperature. The reductions in capacity resulting from a reduction in supply temperature would then necessarily become the same for both systems.

The PVAVS at 42 °F requires a slightly greater cooling capacity (by 0.853 kBtu/h) than the PSZ system at that same temperature, as is the case when the two systems are compared at 55 °F (see Section 5.12.1). Again, this relationship results because of the increased amount of sensible heat added by the PVAVS supply fan, compared to the PSZ fan. But at the reduced flows in the 42 °F case, the capacity difference between the two systems has shrunk -- from 1.479 kBtu/h in the 55 °F case (see Table 16) to 0.853 kBtu/h in the 42 °F case.

Energy consumption and cost - PSZ system. With cold-air distribution, Table 18 shows that the energy consumption by the PSZ system decreases by 2,930 kWh/yr compared to the baseline PSZ system with 55 °F supply. Annual energy costs decrease by $268.

This 2,930 kWh reduction in energy consumption approximately equals the reduction in power consumption by the supply fan (a little less than 3,000 kWh, resulting from the reduced flows at 42 °F). The reduction in fan consumption is partially offset by an increase in cooling coil energy consumption in the 42 °F case. However, at 5 cfm/person, the increase in cooling coil consumption is small (about 60 kWh/yr). That is, at 5 cfm/person, the increase in annual cooling coil power consumption for latent cooling at 42 °F (due to increased moisture condensation) is just slightly greater than the reduction in annual coil power consumption for sensible cooling (due to reduced sensible heat added by the supply fans at the reduced flows).

Thus, the overall 2,930 kWh reduction is explained by the roughly 3,000 kWh reduction in supply fan consumption, slightly offset by the roughly 60 kWh increase in cooling coil consumption.

The 2,930 kWh reduction with the cold-air PSZ configuration corresponds to 11% of the baseline energy consumption by the HVAC system, and 11% of the HVAC energy cost. If cold-air distribution were practical in this application, it would represent one of the single most effective steps for reducing energy consumption in this building, following: achievement of infinite thermal resistance in the windows, walls, and roof (20% reduction); dramatic reduction of power consumption by lighting and equipment (19%); and conversion from PSZ to PVAVS, with or without cold-air distribution (12 to 19%).

Energy consumption and cost - PVAVS. As shown in Table 18, with cold-air distribution, the PVAVS requires 4,941 kWh/yr less than the baseline PSZ system at 55 °F, at a cost savings of $392. The cold-air PVAVS requires 1,792 kWh/yr (and $206/yr) less than the 55 °F PVAVS, and 2,011 kWh/yr (and $124/yr) less than the 42 °F PSZ system.

Of the 4,941 kWh/yr reduction in energy consumption relative to the 55 °F PSZ baseline, over 90% results from a reduction in supply fan power consumption (which decreases by 62%, from 7,328 to 2,758 kWh/yr). The remaining percentage results from a reduction in cooling coil power consumption. With the 42 °F PVAVS, the reduction in annual coil power consumption for sensible cooling (due to reduced sensible heat added by the supply fan) more than offsets the increase in annual coil consumption for latent cooling, relative to the baseline.

The 1,792 kWh/yr reduction for the 42 °F vs. the 55 °F PVAVS is due entirely to reduction in fan power consumption between the two cases. Annual cooling coil power consumption is almost identical for PVAVS at the two supply temperatures, and does not contribute to the difference. Apparently, over the course of the year, the increased energy consumption for latent cooling at the lower supply temperature is just about exactly offset by the reduced consumption for sensible cooling due to reduced fan heat at the lower flows.

Of the 2,011 kWh/yr reduction for the 42 °F PVAVS compared to the PSZ system at that supply temperature, 80% is due to a reduction in supply fan power consumption due to the lower flows in PVAVS vs. PSZ. The remaining 20% results from reduced cooling coil power consumption, due to net reduced fan heat generation at the reduced PVAVS flows.

In practice, the net energy savings with cold-air distribution -- and especially with the cold-air PVAVS -- would not be as great as the figures shown here. The low flows resulting at this low supply temperature might well necessitate the use of powered terminals -- i.e., auxiliary fans at the terminal boxes to ensure adequate throw of the supply air out of the diffusers, for proper mixing in the offices. Such auxiliary fans would consume some portion of the power savings computed above.

Hours at elevated temperature and RH - PSZ system. As shown in Table 18, the percentage of hours undercooled with the PSZ system at 5 cfm/person drops slightly -- from 0.4% to 0.1% (a reduction of 20 hours/yr) -- when the minimum supply air temperature is reduced to 42 °F. This reduction occurs because, relative to the 55 °F case, the control system for the PSZ system in the 42 °F case (as modelled by DOE-2) causes it to operate at closer to full power -- providing greater sensible and latent cooling -- immediately after startup on warm summer mornings, when undercooled hours tend to occur.

Taking the morning of Monday, July 8, as an example, the baseline 55 °F PSZ system operates at between 83 and 91% of maximum capacity during the first two hours after startup, providing an average total cooling of 96,000 Btu/h (both PSZ units combined). The two offices average 77.7 °F during those two hours, more than 2 F° above the 75 °F setpoint. Hence, the offices are both considered to be undercooled during these two hours.

By comparison, with the 42 °F supply air case, the PSZ system (as modelled by DOE-2) operates at between 93 and 100% of full capacity during the first two hours after startup, providing an average of 102,000 Btu/h. The office temperatures now average 76.9 °F during these two hours -- i.e., within the 2 F° throttling range of 75 °F -- and hence no undercooled hours are recorded.

Table 18 shows that the percentage of occupied hours above 60% RH decreases slightly -- from 1.2% to 0.7% (a decline of about 15 hours/yr) -- when the supply temperature is reduced to 42 °F. Elevated-RH hours occur during the first few hours after startup on relatively cool winter mornings with high outdoor humidity. With both the 55 and the 42 °F supply temperatures, the cooling coils remain off during the initial 2 or 3 hours after startup, leading to RH values above 60% in both offices during those hours. But after the coils *do* come on, the 42 °F supply case provides more latent cooling than does the 55 °F case (while providing less sensible cooling and somewhat less total cooling). Thus, in the 42 °F case, the humidity levels drop more rapidly, and there are thus fewer total hours above 60% RH, as reflected in Table 18.

Take the morning of Monday, January 14, as an example. In the baseline 55 °F case, the front office is above 60% RH during the first 6 hours after startup (with the coils off totally during the first 3 hours); the latent cooling provided by the front PSZ unit during those 6 hours is 6,566 Btu. By comparison, in the 42 °F case, the front office is above 60% RH only during the first 5 hours, and the latent cooling by the front unit during the first 6 hours is somewhat higher, 6,717 Btu. The rear office is above 60% RH during the first 4 hours with the 55 °F supply, with latent cooling of 6,211 Btu by the rear unit during those hours. But with the 42 °F supply, the rear office is above 60% RH only during the first 3 hours, with latent cooling of 7,828 Btu during the first 4 hours.

<u>Hours at elevated temperature and RH - PVAVS</u>. Like the PVAVS with a 55 °F supply temperature (Table 16), Table 18 shows that the PVAVS with the 42 °F supply temperature at 5 cfm/person results in no hours undercooled during the year. The reasons are the same as those given in Section 5.12.1 (see *<u>Single PVAVS conditioning both zones</u>*). As with the 55 °F PVAVS, the 42 °F PVAVS begins operating at full capacity during the first hours after startup on warm summer mornings, when undercooled hours tend to occur. As a result, there are no undercooled hours.

As shown in Table 18, there are *fewer* occupied hours above 60% RH with the 42 °F PVAVS at 5 cfm/person (0.3%) than there are with the 42 °F PSZ system (0.7%). This is a reversal of the situation with the 55 °F PVAVS, where the percentage of elevated-RH hours (1.5%) were slightly *greater* than the percentage with the 55 °F PSZ system (1.2%).

The first issue that will be discussed is why there are so many fewer elevated-RH hours with the 42 °F PVAVS compared to the 55 °F PVAVS (0.3 vs. 1.5%, corresponding to a difference of 40 occupied hours per year). During the first hour after startup on cool, humid summer mornings, the 42 °F system -- operating at a low coil surface temperature of 40 °F and experiencing relatively high-humidity inlet air, but operating at minimum flow -- consistently provides more Btu's of latent cooling than does its 55 °F PVAVS counterpart. This drops office humidity rapidly, although not always to RH $\leq$ 60% during that first hour. During subsequent hours, the *55 °F* PVAVS is commonly providing more Btu's of latent cooling; although its coil temperature is higher, its flows are greater and the moisture content of the entering air is higher. But because of the high level of moisture removal during that first hour with the 42 °F system, the RH levels in the office spaces drop below 60% sooner with that system.

On Monday, January 14, for example, the 55 °F PVAVS reduces the office RH (which is above 80% prior to startup) to 70% during the first hour, and to below 60% after 9 hours. The average RH during those 9 hours is about 67%. By comparison, with the 42 °F PVAVS, the office RH drops to 62% during the first hour, drops below 60% after only 5 hours (instead of 9), and averages about 64% during those 5 hours. The trend is similar on Tuesday, January 15: the offices are at 61 to 62% RH during the first two hours with the 55 °F system, but are never above 60% (on an hourly average) with the 42 °F system.

The reason that the PVAVS at 42 °F results in fewer elevated-RH hours than the PSZ system at 42 °F (0.3% vs. 0.7%, corresponding to about 15 hours per year) is more difficult to explain. As discussed in the previous subsection, on January 14, the 42 °F PSZ system results in 5 hours above 60% RH in the front zone (ranging between 72 and 100% RH), and 3 hours in the rear zone (ranging between 65 and 100% RH). By comparison, on that same day, the 42 °F PVAVS results in 6 hours above 60% RH in both zones (ranging between 62 and 65%, approximately). On the next day (Tuesday, January 15), neither system has any hours above 60% RH. Thus, the 42 °F PVAVS would seem to be having *more* elevated-RH hours than the cold-air PSZ system, and it is unclear why PVAVS is being reported as having *fewer* hours. Possibly, the hours above 60% RH with the 42 °F PVAVS are so close to the 60% RH target, that the program is not counting them as being above 60% RH.

5.14.2 The Effect of Cold-Air Distribution on Increased Ventilation Rates

Capacity - PSZ system. As shown in Table 18, increasing the OA rate to 20 cfm/person with the 42 °F PSZ system increases the computed cooling capacity by 23.051 kBtu/h, relative to the baseline 55 °F PSZ system at 5 cfm/person. This increase is larger than the 15.635 kBtu/h increase required when OA rate is raised to 20 cfm/person with the 55 °F PSZ system (see Table 3).

This larger OA-induced increase in required capacity with the 42 °F system results from increased latent cooling. The increase in OA to 20 cfm/person in this humid climate significantly increases the latent load on the system, and the colder cooling coils in the 42 °F case will remove a greater amount of this additional latent heat.

Intuitively, one might expect that -- in this warm climate -- increased OA flows might also increase the *sensible* cooling requirements for the 42 °F system, since the increased amount of (generally warm) OA will have to be cooled to a lower temperature than in the 55 °F case. However, analysis indicates that, in fact, the impact of increased OA on sensible cooling is small. Of the increase in cooling capacity resulting from the increase in OA to 20 cfm/person in the 42 °F case, no more than a few percent appears to be required for additional *sensible* cooling of the OA; essentially all of the added capacity is required for *latent* cooling of the OA.

As indicated in the earlier discussion of the cold-air system at 5 cfm/person, the increased latent load at 42 °F is partially offset by a decrease in the fan-induced sensible load. At the reduced flows in the cold-air system, the amount of sensible heat added to the air by the supply fan is reduced. As discussed in Section 5.14.1, at 5 cfm/person, the increase in peak latent capacity is entirely offset by the decrease in peak sensible capacity. But with the cold-air system at 20 cfm/person, the significant increase in peak latent capacity (resulting from the increased OA inflow) is much greater than the decrease in peak sensible capacity (which remains almost unchanged from the 5 cfm/person cold-air case, since the total 42 °F airflow remains unchanged). Therefore, the total required cooling capacity experiences a net increase for the 42 °F PSZ system at 20 cfm/person, relative to the 55 °F PSZ system at 20 cfm/person, by an amount equal to 23.051 - 15.635 = 7.416 kBtu/h. This 7.4 kBtu/h differential results from an increase of roughly 9.5 kBtu/h in latent capacity, partially offset by a decrease of roughly 2 kBtu/h in sensible capacity, in the 42 °F case.

<u>Capacity - PVAVS</u>. At 20 cfm/person, the PVAVS with 42 °F supply air has a computed capacity requirement of 127.677 kBtu/h. This capacity is 24.199 kBtu/h greater than that of the baseline 55 °F PSZ system at 5 cfm/person, as indicated in Table 18.

This computed capacity of the 42 °F PVAVS at 20 cfm/person is 23.780 kBtu/h greater than that of the 42 °F PVAVS at 5 cfm/person. This is almost exactly the same differential as exists between the 42 °F PSZ systems at 20 vs. 5 cfm/person (23.565 kBtu/h), and occurs for the same reason. This 23.780 kBtu/h difference equals the increase in latent cooling load created by the increased inflow of humid outdoor air.

This capacity of the 42 °F PVAVS at 20 cfm/person is 6.706 kBtu/h greater than that required for the *55 °F* PVAVS at 20 cfm/person. Diagnosis indicates this 6.7 kBtu/h increase is the net result of an increase in latent capacity by over 7 kBtu/h, partially offset by a decrease in sensible capacity by about 1 kBtu/h.

Energy consumption and cost - PSZ system. As shown in Table 18, the 42 °F PSZ system at 20 cfm/person consumes 1,160 kWh/yr more power (at an increased cost of $157/yr), compared to the baseline 55 °F PSZ system at 5 cfm/person. Thus, if an increase from 5 to 20 cfm/person were accompanied by a switch from 55 °F to cold-air distribution, the energy penalty of the OA ventilation rate increase would theoretically decline from 3,245 kWh and $325/yr (see Table 3) to 1,160 kWh and $157/yr, decreases of 50% or more.

Compared with the 55 °F PSZ system at *20* cfm/person, the 42 °F PSZ system at 20 cfm/person consumes 2,085 kWh/yr *less* power, at a cost savings of $168/yr. This 2,085 kWh/yr reduction for the 42 ° vs. the 55 °F system at 20 cfm/person is the net effect of roughly a 3,000 kWh reduction in fan power consumption (due to reduced air flows), partially offset by roughly a 900 kWh increase in cooling power consumption (due to the increased latent load at the lower supply temperature).

As discussed in Section 5.14.1 (Energy consumption and cost - PSZ system), the increase in PSZ cooling coil consumption at 42 ° vs. 55 °F was only about 60 kWh/yr at 5 cfm/person. At that lower OA rate, the increased latent cooling at 42 °F was almost entirely offset by a decrease in sensible cooling due to reduced sensible heat addition by the reduced-flow 42 °F supply fan. But at 20 cfm/person, the required latent cooling energy increases, such that the latent increase at 42 °F becomes 900 kWh/yr (rather than only 60 kWh/yr) greater than the sensible decrease due to the reduced fan power. Thus, the 3,000 kWh reduction in fan power consumption at 42 °F vs. 55 °F -- a reduction which remains the same, regardless of the OA rate -- is partially offset by a 900 kWh increase in 42 °F cooling coil power consumption at 20 cfm/person, while it had been offset only by a 60 kWh increase in cooling coil power at 5 cfm/person.

Compared with the *42 °F* PSZ system at *5* cfm/person, the 42 °F PSZ system at 20 cfm/person consumes 4,090 kWh/yr more power at an increased cost of $425/yr. These increases caused by increased OA with the 42 °F PSZ system are greater than those predicted for the OA increase with the 55 °F PSZ system (3,245 kWh and $325/yr, from Table 3). An analysis of the discussion in the preceding paragraph shows that the incremental effect of 20 vs. 5 cfm/person is greater in the 42 °F case because the OA increase causes an added (900 - 60 =) 840 kWh/yr increase in cooling coil power consumption in the 42 °F PSZ system. Thus, 4,090 kWh = 3,245 kWh + ~840 kWh.

<u>Energy consumption and cost - PVAVS</u>. From Table 18, the 42 °F PVAVS at 20 cfm/person consumes 823 kWh/yr *less* power (but at an energy cost *increase* of $39/yr), compared to the baseline 55 °F PSZ system at 5 cfm/person. (The fact that cost goes *up* while consumption goes *down* indicates that the 42 °F PVAVS results in an increase in kW demand charges that more than offsets the decrease in kWh usage charges.)

Thus -- hypothetically -- an increase in OA ventilation from 5 to 20 cfm/person would nominally result in an energy *savings* of 823 kWh/yr, *if* this increase were accompanied by a switch from a 55 °F PSZ system to a cold-air PVAVS. The energy cost penalty associated with the increase in ventilation would be negligible ($39/yr).

Compared with the 55 °F PVAVS at 20 cfm/person (see Table 16), the 42 °F PVAVS at 20 cfm/person consumes 745 kWh/yr less power, at a cost savings of $85/yr. This 745 kWh savings with the 42 °F system is the net effect of a roughly 1,800 kWh reduction in fan power consumption, partially offset by a 1,050 kWh increase in cooling coil power consumption.

This 745 kWh and $85/yr savings from operating a 20 cfm/person PVAVS at 42 ° rather than 55 °F is fairly modest. By comparison, the 55 °F, 20 cfm/person PVAVS had offered a 3,323 kWh and $201/yr savings compared to the 55 °F, 20 cfm/person PSZ system. Thus, if one is prepared to convert from a 55 °F PSZ system, the greatest incremental energy savings is to achieved by switching to a standard 55 °F PVAVS. Converting further, from a 55 °F PVAVS to a 42 °F PVAVS, would seem to buy only a modest additional incremental energy savings, considering the added complexities that the cold-air distribution system would introduce.

As discussed in the preceding subsection (<u>Energy consumption and cost - PSZ system</u>), reducing the supply air temperature of a 20 cfm/person PSZ system from 55 ° to 42 °F results in a savings of 2,085 kWh and $168/yr. By comparison, the predicted savings are smaller (745 kWh and $85/yr) for reducing the temperature of the 20 cfm/person PVAVS. The relatively modest reductions achieved by converting a PVAVS to 42 °F occur because -- even at 55 °F -- the PVAVS has already substantially reduced flow rates (and hence fan power consumption). The ability to further reduce PVAVS flows by reducing supply temperatures is thus more limited. This situation is illustrated by the fact that -- with the 20 cfm/person PSZ system -- reducing the supply temperature reduces fan power consumption by about 3,000 kWh/yr, as discussed previously. However -- with the 20 cfm/person PVAVS -- reducing the temperature reduces fan power consumption by only about 1,800 kWh/yr.

With either the PSZ system or the PVAVS at 20 cfm/person, the total annual savings in power consumption (745 to 2,085 kWh) and cost ($85 to $168) achieved by using cold-air distribution would be at least partially consumed by the auxiliary fans that would likely be required to power the terminals for proper throw of the supply air out of the diffusers. This is especially true with the PVAVS.

Hours at elevated temperature and RH - PSZ system. Table 3 shows that, as OA is increased from 5 to 20 cfm/person with the 55 °F PSZ system, the percentage of hours undercooled remains essentially unchanged, at 0.4% of all hours. By contrast, Table 18 shows that as OA is increased with the 42 °F system, the percentage of undercooled hours increases, from 0.1% to 0.7% of all hours.

The reason for this effect is that -- with the 42 °F system -- an increase in OA actually results in a decrease of the amount of sensible cooling provided by the system during summer morning startups. Although the increase in OA increases the *total* cooling provided by the 42 °F system's coils, it decreases the *sensible* cooling by the coils, in part because the increased flow of OA (which is slightly cooler than the return air at startup on summer mornings) provides an economizer sensible cooling effect. But this increase in "economizer" sensible cooling does not compensate for the reduction in sensible cooling by the coils, as modeled. As a result -- with the 42 °F system at 20 vs. 5 cfm/person -- the office space operates at higher temperatures during the first few hours after startup on summer mornings, and there are more undercooled hours.

Take the morning of Monday, July 8, as an example. At an OA rate of 5 cfm/person, the 42 °F PSZ system provides total cooling of 101,358 Btu to the front zone during the first 2 hours after operation begins. Of this total, 89,289 Btu are sensible cooling and 12,069 Btu are latent. At 5 cfm/person, the average temperature in the front zone is 77.0 °F during those 2 hours, and neither of those hours is recorded as undercooled (i.e., all are within 2 F° of the 75 °F setpoint).

At an OA rate of 20 cfm/person, the total cooling in the front zone during the first 2 hours on July 8 with the 42 °F PSZ system is 110,724 Btu, 9,000 Btu greater than the total cooling at 5 cfm/person. However, of this total cooling at 20 cfm/person, only 77,684 Btu is sensible cooling -- almost 12,000 Btu *less* than the sensible cooling at 5 cfm/person. Latent cooling at 20 cfm/person jumps to 33,040 Btu -- almost 21,000 Btu greater than at 5 cfm/person. The reduction in sensible cooling by the coils results in part because the increased flow of outdoor air -- which averages 76.5 °F during those 2 hours -- provides some "economizer" cooling of the return air, which averages 82.4 °F prior to OA mixing during those hours. But the reduction in sensible cooling by the coils at 20 cfm/person over-compensates for the increase in "economizer" sensible cooling, because, at 20 cfm/person, the front office temperature averages 78.2 °F -- well above the 77.0 °F average at 5 cfm/person. As a result, while the 42 °F PSZ/5 cfm case had no undercooled hours in the front office on July 8, the 42 °F PSZ/20 cfm case is undercooled for the first 5 hours, according to the DOE-2 model.

This effect does not occur with the OA increase in the 55 °F PSZ system. With the 55 °F system, the sensible cooling in the front office during the first 2 hours on July 8 is 86,507 Btu at 5 cfm/person and 86,910 Btu at 20 cfm/person -- essentially unchanged with the OA increase. As a result, the front office temperatures in the 55 °F PSZ/20 cfm case are within 0.1 to 0.2 F° of those in the 55 °F PSZ/5 cfm case. With the 55 °F system, the front office is undercooled for the first 2 hours after startup on July 8 at both OA rates. Consequently, Table 3 shows the percentage of undercooled hours for the 55 °F PSZ to be the same at both 5 and 20 cfm/person.

It would appear that the substantial increase in latent cooling in the 42 °F system with the increase in OA is confounding either the PSZ unit's control system, or the DOE-2 model, resulting in insufficient sensible cooling. The 42 °F system at 20 cfm/person is not operating at full capacity, according to the model; during the first 2 hours on July 8, the front PSZ unit is operating only at 77 to 90% of capacity, an even lower percentage than the 55 °F/20 cfm unit (which operated at 82 to 93%). And the supply air is at a temperature above the 42 °F minimum supply temperature during the first 2 hours. Thus, the 42 °F/20 cfm system had the capacity to cool the office more effectively than it did.

Table 18 shows that operating the 42 °F PSZ system at increased OA slightly reduces the number of occupied hours above 60% RH, from 0.7% at 5 cfm/person to 0.3% at 20 cfm/person. The difference between 0.7% and 0.3% corresponds to 12 hours per year per zone. The apparent explanation is that -- in the first few occupied hours on cool winter mornings, before the cooling coils have activated but when the indoor latent sources have caused indoor RH to exceed outdoor RH -- the increased OA flow reduces indoor RH.

To illustrate this impact on RH, take the morning of Monday, January 14. During the first 4 hours after startup on this particular morning, the outdoor humidity ratio in Miami is about 0.014 lb moisture per lb dry air, corresponding to RH at office temperature ranging between 80 and 90%. The cooling coils in the PSZ units do not activate for the first 4 hours in the front zone, or the first 2 hours in the rear zone. As a result, during those hours, there is no moisture removal by the HVAC system. When the PSZ units are operating at 5 cfm/person, latent heat released by the occupants causes the RH in the office space to average 99% during the first 4 hours in the front zone, and 94% during the first 2 hours in the rear zone. But at 20 cfm/person, these averages drop by 10 percentage points, to 89% in the front zone and 84% in the rear zone. Note that these decreases are independent of the design cooling supply air temperature for the system, since the coils are off.

When the cooling coils *do* come on, RH levels drop faster for the 5 cfm/person system. Even though the 20 cfm/person systems achieve many more Btu's of latent cooling, as discussed previously, this increase is insufficient to compensate for the increased latent heat in the incoming air at the higher OA flow. As a result, during the several hours *after* the coils have activated, the 42 °F PSZ at 5 cfm/person reduces zone RH to levels 0 to 10 percentage points lower than the RH levels predicted for the 42 °F PSZ at 20 cfm/person.

From the figures above, it is clear that January 14 is not one of the mornings when the increased OA causes RH to drop below 60% during the initial hours before the cooling coils come on. Presumably, the 12 fewer elevated-RH hours per year occur on other days during cool-weather months.

As noted above, the impact of increased OA in reducing indoor RH before the coils come on is independent of the minimum supply air temperature that the system is designed to deliver once the coils are activated. Accordingly, this apparent explanation for why increased OA causes the decrease in elevated-RH hours shown in Table 18 for the 42 °F PSZ system, must also be the explanation for the reduction shown in Table 3 for the 55 °F PSZ system. The decrease in elevated-RH hours for the 55 °F system with the increase in OA also corresponds to 12 hours per zone per year (a reduction from 1.2% to 0.9% of occupied hours, as shown in Table 3).

<u>Hours at elevated temperature and RH - PVAVS</u>. Table 18 shows that, as the 42 °F PVAVS is increased from 5 to 20 cfm OA/person, the percentage of hours undercooled increases from about zero to 0.2% of all hours. This slight increase -- corresponding to about 11 hours per zone per year -- occurs for the same reason indicated previously for the PSZ system. That is -- perhaps confounded by the high latent load at 20 cfm/person, and/or the anticipated "economizer" cooling by the additional OA -- the 42 °F PVAVS provides insufficient sensible cooling at the higher OA rate after startup on summer mornings, even though it has sufficient capacity to provide the needed cooling.

Take the morning of Monday, July 29, as an example. During the first 4 hours after startup on that morning, the 42 °F PVAVS operating at 5 cfm/person provides 371,500 Btu of sensible cooling; the front zone is undercooled during 1 of those 4 hours at 5 cfm/person, and the rear zone is not recorded as undercooled during any of the hours. But when OA is increased to 20 cfm/person, the system provides only 351,500 Btu during those first 4 hours, 20,000 Btu less than in the 5 cfm/person case; the front zone is undercooled for 2 of the 4 hours, and the rear zone for 1 of the 4 hours. This occurs even though -- for 3 of the 4 hours (including all of the hours when one or both zones are undercooled) -- the 20 cfm/person system is operating below its available capacity (i.e., at part-load ratios of 0.93 to 0.97), and thus had the capability of providing more sensible cooling.

Regarding hours at elevated RH, Table 18 shows that -- with the 42 °F PVAVS, the percentage of hours at RH > 60% increase slightly (from 0.3 to 0.6%) as OA is increased. This result is consistent with that for the 55 °F PVAVS, where the OA increase also resulted in an approximate doubling of elevated-RH hours (from 1.5% to 2.9%, in Table 16).

And the explanation in the 42 °F case is the same as that for the 55 °F PVAVS (see Section 5.12.2, *Single PVAVS conditioning both zones*, Hours at elevated temperature or RH). Although the 20 cfm/person system at a given supply temperature removes substantially more latent heat than does the 5 cfm/person system at the same temperature, this increased removal does not compensate for the increased latent inflow due to the increased OA rate.

Take Monday, January 14, as an example. Because the externally induced and occupant-induced loads on the office spaces are independent of OA rate, the flows to each zone from the 42 °F PVAVS are unchanged with increasing OA (remaining at or near the minimum flow rate throughout that day). Because PVAVS characteristically supply air near the minimum temperature, the supply air temperatures and the coil surface temperatures are essentially unchanged with increasing OA; the coil surface temperature averages 37.8 °F with the 5 cfm/person 42 °F PVAVS, and 37.9 °F with the 20 cfm/person system on January 14. The coil bypass factor decreases from an average of 0.67 at 5 cfm to 0.61 at 20 cfm, a decrease of under 10%. That is, at 20 cfm, more of the air nominally comes into contact with the coil surface, reflecting, e.g., a deeper tube bank or reduced velocity, as necessary to cool the warmer entering air down to the same coil outlet temperature.

And, correspondingly, the 20 cfm/person system removes 16% more latent heat on January 14 (501,000 Btu compared to 431,000 Btu at 5 cfm/person). But this increase is insufficient to compensate for the increased moisture present in the increased OA flow. The moisture content of the air leaving the cooling coils increases 15%, from an average of 0.0087 lb of moisture per lb of dry air at 5 cfm to an average of 0.0100 lb/lb at 20 cfm. As a result, the office space -- which is above 60% RH only for the first 5 hours on January 14 at 5 cfm/person -- is above 60% RH for 10 of the 13 hours at 20 cfm/person.

5.15 MODIFICATIONS TO THE ECONOMIZER

The systems discussed to date have all included an economizer (with the exception of the PTAC unit in Section 5.12, which is not designed to accommodate an economizer). Economizer operation has been controlled by the outdoor air temperature, with an economizer limit temperature of 68 °F. That is, the economizer ceases to function when the outdoor air temperature rises above 68 °F, and the OA flow rate drops back to its minimum value (of 5 or 20 cfm/person).

To assess the effect of the economizer on energy consumption and costs in the study building, two modifications to this baseline economizer were considered:

a) the economizer is eliminated altogether; or

b) economizer operation is controlled by a controller which senses the enthalpy rather than the temperature of the outdoor air.

It would not be anticipated that the baseline economizer would be in operation very many hours for the building being considered here. For one thing, in the warm climate of southern Florida, the outdoor temperature is not often below the economizer limit temperature of 68 °F. Even during winter months in Miami, the outdoor temperature will generally be at 68 °F or below only during the first hour or two after startup in the morning.

A second factor limiting economizer operation is that PSZ systems commonly operate with economizer lockout. That is, the economizer will operate only when it is able to provide *all* of the required cooling, so that the air conditioning compressor remains off entirely. The hermetic motors in packaged compressors are cooled by the circulating refrigerant flow. If the compressors operated at very low loads -- as might occur if the economizer were providing much but not all of the required cooling -- the motors could overheat, thus reducing motor lifetime. Economizer lockout is intended to avoid this problem by ensuring that the compressor and the economizer are never operating at the same time.

With these two limits on economizer performance, the economizer will typically operate, at most, only during the first hour after startup, even during the coolest weather. With economizer lockout, economizer usage remains low even when the economizer limit temperature is raised to values well above typical values (e.g., to 80 °F).

If the economizer operation were controlled by the *enthalpy* rather than the temperature of the outdoor air, economizer operation would be limited even further. With enthalpy-based control, it would no longer be sufficient for outdoor temperature to be below 68 °F (and the mixed air entering the cooling coils to be above 68 °F). Now, in addition, the enthalpy of the outdoor air must be less than the enthalpy of the mixed air. The high outdoor humidity levels during those morning hours immediately after startup would be expected to result in elevated outdoor enthalpies during those few hours when the economizer might otherwise be expected to operate, thus further limiting economizer operation.

Enthalpy controllers are more expensive than dry-bulb temperature controllers, and require more maintenance. Thus, enthalpy controllers would not be typical of small HVAC units in the capacity range being considered here. Enthalpy control calculations will be made here despite this fact, in order to demonstrate what the effect would be on energy and performance.

5.15.1 The Effect of Eliminating the Economizer

Table 19 shows the effect of eliminating the economizer, at both 5 and 20 cfm OA/person. As would be expected, the effect is small.

As would be expected, elimination of the economizer has no impact on the required peak cooling capacity, at either 5 or 20 cfm/person. The economizer will not be operating when the cooling coils are operating at their peak, and hence does not enter into the capacity calculation.

HVAC energy consumption increases by 1.9% when the baseline temperature-controlled economizer is deleted at 5 cfm/person, and by 1.6 percentage points (14.0% minus 12.4%) at 20 cfm/person. HVAC energy cost increases by 0.9% (by $23) at 5 cfm/person, and by 0.8 percentage points (by 344 - 325 = $19) at 20 cfm/person. These small increases from deletion of the economizer are consistent with the expectation that the economizer will be operating only a small portion of the time.

The percentage of total hours undercooled remain unchanged when the economizer is eliminated, as would be expected.

The percentage of occupied hours at RH > 60% decline when the economizer is deleted (from 1.2 to 0.6% at 5 cfm/person, and from 0.9 to 0.2% at 20 cfm/person). The economizer substantially increases the inflow of humid outdoor air during the first hour of operation on cool mornings -- exactly the times when hours of elevated RH occur. Elimination of the economizer thus would be expected to reduce the hours at elevated RH.

5.15.2 The Effect of an Enthalpy-Controlled Economizer

As shown in Table 19, the results with the enthalpy-controlled economizer are in between those with the temperature-controlled economizer and those with no economizer. They are almost exactly equal to the results with the baseline temperature-controlled economizer, indicating that the use of enthalpy control instead of temperature control eliminates only a few hours of economizer operation.

The PSZ cooling capacity is unaffected by the switch to an enthalpy-controlled economizer, as expected.

Use of an enthalpy-controlled economizer increases HVAC energy consumption by 33 kWh/yr (from 3,245 to 3,278 kWh/yr), resulting in essentially no energy cost

TABLE 19

Effect of Modifying Economizer Operation:
Increase Compared to Baseline Case with 5 cfm/person

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH > 60%, %
Baseline values: T-controlled economizer, OA = 5 cfm/p (absolute values)[1]	103.558[1]	--	26,145[1] (HVAC) 60,161[1] (building)	-- --	-- --	2,510[1] (HVAC) 4,273[1] (building)	-- --	-- --	0.4	1.2
OA ventilation rate = 5 cfm/person										
No economizer	0	0	505	1.9	0.8	23	0.9	0.5	0.4	0.6
Enthalpy-controlled economizer	0	0	45	0.2	0.1	2	0.1	~0	0.4	0.8
OA ventilation rate = 20 cfm/person										
T-controlled economizer	15.635	15.1	3,245	12.4	5.4	325	12.9	7.6	0.4	0.9
No economizer	15.635	15.1	3,654	14.0	6.1	344	13.7	8.1	0.4	0.2
Enthalpy-controlled economizer	15.635	15.1	3,278	12.5	5.4	326	13.0	7.6	0.4	0.6

[1] These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

increase ($326 - $325 = $1/yr) relative to the temperature-controlled economizer. This result indicates that the conversion to enthalpy control has only a minimal impact on the already-limited hours of economizer operation.

As expected, the enthalpy-controlled economizer has no impact at all on the number of undercooled hours, relative to either the temperature-controlled economizer or the case with no economizer.

Relative to the temperature-controlled economizer, the enthalpy-controlled economizer decreases the percentage of occupied hours at RH > 60% from 1.2% to 0.8% at 5 cfm/person, and from 0.9% to 0.6% at 20 cfm/person. These reductions correspond to 10 to 15 hours per year per zone. This is probably an approximate indication of the number of hours by which the conversion to enthalpy control reduces economizer operation. (Of course, eliminating the economizer altogether reduces the number of elevated-RH hours even further.)

5.16 ALTERNATIVE COOLING ELECTRIC INPUT RATIOS

All of the preceding energy consumption and cost calculations assume that the cooling system has an electric input ratio (EIR) of 0.341 Btu/h of electric input per Btu/h of cooling output (except for the PTAC unit in Section 5.12, for which the EIR is higher). This is a reasonably representative value, corresponding to an energy efficiency ratio (EER) of 10 Btu/h cooling output per W of electric input.

Table 20 shows the effect on the baseline PSZ system of varying the cooling EIR to: 0.284 Btu/h per Btu/h (EER = 12 Btu/h per W), representing a very efficient cooling system; and 0.427 Btu/h per Btu/h (EER = 8 Btu/h per W), representing a relatively inefficient system.

5.16.1 The Effect of EIR at 5 cfm/person

Capacity. As expected, Table 20 shows that variations in EIR have no impact on computed maximum cooling capacity. The efficiency with which the cooling system converts electric input into cooling output has no impact on the parameters that determine the amount of cooling output that is necessary. The weather- and occupant-induced loads on the space remain unchanged, as do the coil bypass factor and the heat added to the air stream by the air handler.

Energy consumption and cost. Improvement of cooling efficiency to an EIR of 0.284 results in a significant reduction in HVAC energy consumption (12.0%) and cost (12.6%), compared to the baseline case (with EIR = 0.341). These reductions at 5 cfm/person are greater than those from any of the other parameters considered in this report, with the exception of:

TABLE 20

Effect of Cooling EIR: Increase Compared to Baseline Case with 5 cfm/person

Case	Increase in total required cooling capacity: In kBtu/h	Increase in total required cooling capacity: As %	Increase in annual energy consumption: In kWh	Increase in annual energy consumption: As % of HVAC energy	Increase in annual energy consumption: As % of building energy	Increase in annual energy cost: In $	Increase in annual energy cost: As % of HVAC energy cost	Increase in annual energy cost: As % of building energy cost	Hours/yr under-cooled, %	Occupied hours/yr RH>60%, %
Baseline values:	103.558[1]	--	26,145[1]			2,510[1]			0.4	1.2
EIR = 0.341,			(HVAC)	--	--	(HVAC)	--	--		
OA = 5 cfm/p			60,161[1]			4,273[1]				
(absolute values)[1]			(building)	--	--	(building)	--	--		
OA ventilation rate = 5 cfm/person										
EIR = 0.284	0	0	-3,139	-12.0	-5.2	-317	-12.6	-7.4	0.4	1.2
EIR = 0.427	0	0	4,736	18.1	7.9	477	19.0	11.2	0.4	1.2
OA ventilation rate = 20 cfm/person										
EIR = 0.341	15.635	15.1	3,245	12.4	5.4	325	12.9	7.6	0.4	0.9
EIR = 0.284	15.635	15.1	-433	-1.7	-0.7	-46	-1.8	-1.1	0.4	0.9
EIR = 0.427	15.635	15.1	8,794	33.6	14.6	2,647	105.5	61.9	0.4	0.9

[1] These baseline values for capacity, consumption, and cost are the absolute values from Table 3, *not* the *increase* from the baseline.

- conversion to PVAVS, which also gives a 12% reduction in energy consumption, although only a 7% reduction in cost (Section 5.12.1);

- conversion to highly efficient lighting and equipment, giving 19% reduction in consumption and 17% reduction in cost (Section 5.4.1); and

- achieving infinite thermal resistance in the walls, windows, and roof (a hypothetical situation), giving 20% reduction in consumption and 22% reduction in cost (Section 5.10).

Conversion of the PSZ system to cold-air distribution (Section 5.14.1) gives similar but slightly smaller reductions (11% in energy consumption and 11% in cost).

Thus, accepting an increase in installed costs for the more efficient cooling equipment would result in a 12% to 13% reduction in energy consumption and cost each year at 5 cfm/person. Of the parameters listed above that provide the greatest energy savings, the conversion to a more efficient cooling system (and/or conversion to PVAVS) may be the most readily and practically achieved. At the indicated energy cost savings of $317 per year for the study building with the two PSZ units, it would take perhaps 7 years to recover the increased equipment costs for the more efficient units.

Just as a reduction in EIR results in one of the most significant *decreases* in energy consumption and cost, an increase in EIR to 0.427 causes one of the most significant *increases* in energy consumption and cost. As shown in Table 20, the higher EIR results in an 18% increase in annual energy consumption and a 19% increase in cost. No other single parameter causes such an increase in energy use at 5 cfm/person, with the exception of the use of inefficient lighting and equipment (Section 5.4.1). With an energy cost penalty of $477 per year for using the less efficient units in this building, the initial savings in equipment costs that would be achieved by using the lower efficiency units would be consumed within perhaps 2 years. The use of lower-efficiency units would not appear cost-effective.

Hours at elevated temperature and RH. Table 20 shows that the percentages of undercooled hours and of hours above 60% RH do not change at all as EIR is varied. Variation of the EIR changes only the computed amount of electric power required to achieve the computed amount of cooling. There are no other changes to the space or the system: hourly zone temperatures and humidities remain the same, as do the hourly values of, e.g., total and latent cooling provided by the system, cooling coil and supply air temperatures, bypass factor, and supply air humidity. Therefore, EIR is not calculated to create any change in hours undercooled or at elevated RH.

5.16.2 The Effect of EIR on Increased Ventilation Rates

Capacity. Table 20 shows that, at 20 cfm/person -- as at 5 cfm/person -- the cooling EIR has no impact on the computed maximum cooling capacity, since none of the parameters that impact the capacity calculation are influenced by the EIR.

Energy consumption and cost. As indicated in Table 20, the PSZ system with EIR = 0.284 could provide 20 cfm OA/person while consuming 433 kWh/yr less power and costing $46/yr less than the baseline PSZ system (EIR = 0.341) at 5 cfm/person. That is, OA ventilation rate could nominally be *increased* while achieving a 1.7% to 1.8% *reduction* in HVAC energy consumption and cost, if the ventilation rate increase were accompanied by conversion of the system to a higher-efficiency system.

The only other parameters which provided a reduction in energy consumption or cost (relative to the baseline) when the ventilation rate is increased are:

- conversion to PVAVS, providing a 0.3% reduction in consumption, although an increase in cost (Section 5.12);

- conversion to highly efficient lighting and equipment, providing a 5% to 7.5% reduction (Section 5.4);

- reduction in the number of occupants by one-half, providing a 1.5% to 1.7% reduction (Section 5.3); and

- the hypothetical case of achieving infinite thermal resistance in the walls, windows, and roof, providing a 7% to 8% reduction (Section 5.10).

It is of interest to compare the consumptions and costs involved with providing 20 cfm/person using the more efficient PSZ (EIR = 0.284), with the values involved with providing 20 cfm/person using the system having the baseline EIR (0.341). Referring to Table 20, at 20 cfm/person, the energy consumption with the more efficient system is 3,245 - (-433) = 3,678 kWh/yr less than the consumption with the system having the baseline EIR (a 12.5% reduction in annual consumption). The annual energy cost is $325 - (-$46) = $371 less for the study building (a 13% reduction). With this annual energy cost savings at 20 cfm/person, it would take perhaps about 6 years to recover the increased equipment costs for buying the more efficient units. As would be expected, this recovery period is somewhat less than the roughly 7 years at 5 cfm/person (Section 5.16.1).

With the less effective EIR of 0.427, Table 20 shows that, at 20 cfm/person, energy consumption increases by one-third (by 8,794 kWh/yr) and energy cost approximately doubles (increasing by $2,647/yr), compared to the baseline (EIR =

0.341) system at 5 cfm/person. No other parameter causes the 20 cfm/person system to increase in cost so dramatically relative to the baseline, with the exception of the use of inefficient lighting and equipment (Section 5.4.2).

The use of the less efficient cooling equipment in this building would be very cost-ineffective at the 20 cfm/person OA ventilation rate. Compared to the PSZ units with the baseline EIR (0.341) at 20 cfm/person, the units with the poorer EIR (0.427) at 20 cfm/person consume 8,794 - 3,245 = 5,549 kWh/yr more power at an increased cost of $2,647 - $325 = $2,322/yr. With this increase in annual power cost, the savings in equipment cost that would be achieved by buying the less efficient equipment would be consumed in perhaps half a year.

Hours at elevated temperature and RH. Table 20 shows that -- at 20 cfm/person, as at 5 cfm/person -- the percentages of undercooled hours and of hours above 60% RH do not change at all as EIR is varied at constant OA flow rate.

5.17 ALTERNATIVE COOLING CAPACITIES AND SENSIBLE HEAT RATIOS

During the variation of the parameters discussed previously, the system cooling capacity has been allowed to default to the value computed by the DOE-2 program; i.e., to the capacity computed as being required to meet the peak cooling load. As indicated in the preceding tables, these default capacities have varied somewhat, depending upon the building or HVAC system parameters being varied.

For the baseline configuration with 5 cfm OA/person, as shown in Table 3, the total capacity (both zones combined) at Air-Conditioning and Refrigeration Institute (ARI) conditions is 103.558 kBtu/h, or 8.6 tons of refrigeration. This total capacity is divided into 4.3 tons for the PSZ unit treating the front zone, and 4.3 tons for the unit treating the rear zone. The default sensible heat ratio (SHR) at 5 cfm/person (i.e., the fraction of the total capacity that is sensible capacity) computes to be 0.75 in each zone. For the baseline configuration with 20 cfm/person, the total capacity is 119.193 kBtu/h (9.9 tons), with 4.9 tons in the front zone and 5.0 tons in the rear; the SHR is 0.70.

To illustrate the effect of cooling capacity and SHR on energy costs and system performance, calculations were made in which the capacity and SHR of the unit serving each zone was varied through several values representative of commercially available units. Units are assumed to be available in half-ton increments of ARI-rated capacity. Representative SHRs are assumed to be 0.78 (for three rows of coils) and 0.73 (for four rows of coils).

5.17.1 The Effect of Cooling Capacity and SHR at 5 cfm/person

Table 21 summarizes the computed effect of varying cooling capacity and SHR at an OA ventilation rate of 5 cfm/person.

TABLE 21

Effect of Varying Cooling Capacity and SHR at 5 cfm/person

Total Cooling Capacity[1] (tons)	HVAC Energy Cost ($/yr)			% of Total Hours Undercooled			% of Occupied Hours at RH > 60%		
	SHR=0.78	SHR=0.75[2]	SHR=0.73	SHR= 0.78	SHR= 0.75[2]	SHR= 0.73	SHR= 0.78	SHR= 0.75[2]	SHR= 0.73
8.5	2,502	--	2,473	0.4	--	1.2	1.2	--	1.2
8.6[2]	--	2,510	--	--	0.4	--	--	1.2	--
9.0	2,541	--	2,527	0.1	--	0.3	1.2	--	1.2
9.5	2,562	--	2,551	~0	--	0.1	1.2	--	1.2
10.0	2,584	--	2,581	~0	--	~0	1.2	--	1.2

[1] The combined cooling capacities of the front and rear PSZ units, expressed as tons of refrigeration. With total capacities of 9.0 and 10.0 tons, the front and rear units are of equal capacity; with total capacities of 8.5 and 9.5 tons, the rear unit is 0.5 tons larger than the front unit. The rated capacities and SHRs are at the conditions specified by the Air-Conditioning and Refrigeration Institute.

[2] Default values of cooling capacity and SHR computed by DOE-2 when these values are not set in the input file. These are the "baseline" values used elsewhere in this report. With the baseline system, the front and rear zones are each served by a nominal 4.3-ton unit.

HVAC energy cost. As shown in Table 21, the HVAC energy cost increases slightly with increasing capacity. There are two reasons that this occurs.

First, the units having greater capacity are better able to address the peak cooling loads that are encountered on summer Monday mornings, and they consume slightly more power in doing so. But the baseline (8.6-ton) system already handles most of the peak-load hours, with only 0.4% of hours being undercooled (corresponding to about 30 hours per year per zone). As a result, the increase in energy costs involved in going to greater-capacity systems (and reducing the undercooled hours to even smaller percentages) are small.

The second reason that power consumption is greater with the larger-capacity units is that, on a given summer day, the larger units tend to keep the office space a couple tenths of a degree cooler.

These effects are small. As shown in Table 21, increasing the total capacity to 9.5 or 10.0 tons -- by which point the number of undercooled hours is reduced almost to zero -- increases energy costs by only \$41 to \$74/yr above the \$2,510 baseline value, an increase of only 1.5% to 3%.

Of the increase in cost due to capacity increase, 70% to 80% results from increases in the monthly power demand (per-kW) charges, and the remainder due to an increase in energy (per-kWh) charges. The increase in demand charges consists of a \$1 to \$2 per month increase during the winter months, and a \$2 to \$5 per month increase during the summer months. The increase in energy charges consists of a \$0 to \$1 per month increase during the winter months, and a \$1 to \$2 per month increase during the summer months.

As shown in Table 21, at a given total capacity, the system having the SHR of 0.73 consistently consumes less power annually than the one having the SHR of 0.78. This occurs because the system having the lower SHR has less sensible cooling capacity. It thus performs somewhat less sensible cooling throughout the year, resulting in more hours undercooled. Because system operation is controlled by sensible demand, the lower-SHR system winds up performing less latent cooling as well, on many days. Correspondingly, at the 5 cfm/person OA rate, the lower-SHR system consumes slightly less power on an annual basis.

To illustrate this impact of SHR on power costs, consider the 9.5-ton system during the mid-summer week of July 8 through 12. During this week, the entire system (front and back units combined) would provide 15,700 Btu less total cooling (and 8,700 Btu less latent cooling) if the SHR were 0.73 instead of 0.78. Of course, the effect is small; given the EER assumed for the cooling unit (10 Btu/h of cooling output per W of electric input), 15,700 Btu of cooling corresponds to only 4.6 kWh of power consumption over an entire week. But it illustrates why, over the course of a year, the SHR value would create the small energy cost differences seen in Table 21 at 5 cfm OA/person.

This effect of SHR is smaller during milder weather. For example, during the week of March 18 through 22, the 9.5-ton system operating at 5 cfm/person would provide only 1,000 Btu less total cooling if the SHR were 0.73 instead of 0.78. Latent cooling would be essentially identical with either SHR.

This impact of capacity and SHR on power consumption is further addressed below in the discussion of hours undercooled.

Hours at elevated temperature and RH. As shown in Table 21, at constant SHR, increasing the total capacity systematically reduces the percentage of total hours undercooled. This results, of course, because -- as sensible capacity increases -- the system is better able to respond to the high sensible loads immediately after startup on warm mornings, when undercooled hours occur.

At constant capacity, decreasing the SHR from 0.78 to 0.73 consistently increases the percentage of undercooled hours. As indicated previously, decreasing the SHR decreases the sensible cooling capacity, rendering the system less able to handle the high sensible loads on summer mornings.

To illustrate, consider the first four hours in the front office after startup on Monday, July 8. In the baseline case at 5 cfm/person, the front office was undercooled during those four hours (averaging 77.5 °F). The 4.3-ton PSZ unit (SHR = 0.75) conditioning the front office in the baseline case -- which is rated at 3.2 tons sensible capacity and 1.1 tons latent capacity -- provided a 194,400 Btu of total cooling during that four-hour period, of which 21,400 Btu were latent cooling.

In comparison with the baseline, cooling increases significantly during these four hours when total building cooling capacity is increased to 9.0 tons at SHR = 0.78. In this case, the unit conditioning the front office is increased from 4.3 to 4.5 tons; the rated sensible capacity of this larger unit is 3.5 tons, and the latent capacity 1.0 ton. With the larger unit, the front office is not undercooled during any of the first four hours on July 8 (the average temperature being 76.4 °F), since the total cooling provided by the system during those hours increases to 213,300 Btu (about a 10% increase over the cooling provided by the baseline unit).

The latent cooling with the larger (4.5-ton/0.78 SHR) unit increases to 22,800 Btu, a 6.5% increase over the baseline (4.3-ton/0.75 SHR) case. This increase in latent cooling occurs despite the fact that -- with the lower SHR -- the smaller, baseline front unit actually has a slightly greater rated latent capacity (1.1 tons vs. 1.0 ton). The operation of the cooling coils is dictated by the sensible cooling demands, and the ability of the larger unit to respond to the high sensible load during those first four hours results in increased latent cooling, despite the reduced latent capacity of the larger unit.

When the SHR of the 9.0 ton system is reduced from 0.78 to 0.73 -- increasing the latent capacity but reducing the sensible capacity -- the larger system still has a greater sensible capacity than does the smaller, baseline system at SHR = 0.75. With SHR = 0.73, the rated sensible capacity of the 4.5-ton unit conditioning the front office is 3.3 tons, and the latent capacity is 1.2 tons. With this slightly greater sensible capacity relative to the baseline (3.3 vs. 3.2 tons), the 4.5-ton/0.73 SHR unit conditioning the front zone is better able to reduce office temperature during the first four hours on July 8; only two of the four hours are undercooled (average temperature 77.2 °F). This is somewhat better than the 4.3-ton/0.75 SHR baseline unit (four hours undercooled, 77.5 °F average), but not as effective as the 4.5-ton/0.78 SHR unit (zero hours undercooled, 76.4 °F average). Power consumption and cost naturally follow this same relationship. The 4.5-ton/0.73 SHR unit provides total cooling of 200,900 Btu to the front zone during the first four hours -- 3% more than the baseline 4.3-ton/0.75 SHR unit, but 6% less than the 4.5-ton/0.78 SHR unit.

Again, the latent cooling provided by the units tracks the sensible cooling that is provided, rather than the rated latent capacity of the units. The latent cooling provided to the front zone during the first four hours by the 4.5-ton/0.73 SHR unit (latent capacity 1.2 tons) is 21,900 Btu -- 2.5% more than the 21,400 Btu provided by the 4.3-ton/0.75 SHR unit (latent capacity 1.1 tons), but 4% less than the 22,800 Btu provided by the 4.5-ton/0.78 SHR unit (latent capacity 1.0 ton). The greater the sensible capacity of the unit, the more vigorously it responds to the high sensible loads during startup on warm mornings, and, as a result, the greater the latent cooling it provides. In this example, the unit with the least latent capacity winds up providing the greatest amount of latent cooling. The unit with the greatest latent capacity is only in the middle, in terms of latent removal during these hours. The 4.5-ton/0.73 SHR unit would appear to be over-designed for latent removal, and -- at least in comparison with the 4.5-ton/0.78 SHR unit -- under-designed for sensible removal during peak hours.

Table 21 shows no impact of capacity or SHR on the percentage of occupied hours at RH > 60%. This effect results because the hours at elevated RH occur after startup on cool, humid mornings, during the initial hours before the cooling coils have come on (or when the cooling coils are operating well below capacity). Therefore, increasing total cooling capacity (or increasing the latent capacity via a decrease in SHR) would not be expected to impact the percentage of elevated-RH hours. Hours above 60% RH are not being caused by inadequate total or latent cooling capacity.

Hours at elevated RH do not occur during hot months in the Miami climate, when an increase in cooling capacity might otherwise have an impact. In the earlier discussion, it was indicated that the latent cooling by the unit serving the front office during the first four hours on July 8 could vary between 21,400 and 22,800 Btu, depending upon the capacity and SHR of the systems considered. But even the lesser amount of cooling is sufficient to keep the front office below 50% RH during the first hours after startup.

5.17.2 The Effect of Cooling Capacity on Increased Ventilation Rates

Table 22 shows the estimated effect of varying cooling capacity and SHR at a ventilation rate of 20 cfm/person.

HVAC energy cost. Table 22 shows that -- at 20 cfm/person as at 5 cfm/person -- energy costs increase slightly as cooling capacity increases. As at 5 cfm/person, this occurs for two reasons: a) additional cooling by the larger units in reducing the percentage of undercooled hours below 0.4%; and b) additional cooling because the larger units tend to keep the office a couple tenths of a degree cooler on some summer days.

As at 5 cfm/person, the effects of increased capacity on power consumption and cost at 20 cfm/person are small. Increasing the total cooling capacity of the two PSZ units from the 9.9 tons to 10.5 or 11.0 tons increases annual energy costs by only $40 to $66, or about 1.5% to 2.5% of the power cost for the 9.9-ton system. This is about the same cost increase that was computed for a comparable capacity increase in the 8.6-ton system at 5 cfm/person.

Of the increase in cost due to capacity increase at 20 cfm/person, 60% to 75% results from increases in the monthly power demand (per-kW) charges, with the remainder due to an increase in energy (per-kWh) charges. By comparison, at 5 cfm/person, 70% to 80% of the increase was due to demand charges. At 20 cfm/person, the increase in demand charges consists of a $0 to $2 per month increase during the winter months, and a $3 to $4 per month increase during the summer months. The increase in energy charges consists of no monthly increase during the winter months, but a $2 to $3 per month increase during the summer months.

One difference between the results at 5 cfm/person (Table 21) and those at 20 cfm/person (Table 22) is that -- at 20 cfm/person -- the annual power costs for the 0.73 SHR unit are slightly *greater* (by $2 to $5) than those for the 0.78 SHR unit at the two higher capacities. At 5 cfm/person, power costs with the 0.73 SHR unit were always slightly *less*.

The reason for this small effect is that -- when the inflow of humid outdoor air is increased from 5 to 20 cfm/person -- the latent load on the system throughout the year is increased disproportionately relative to the increase in sensible load. This large increase in the latent load makes more effective use of the increased latent capacity of the 0.73 SHR system. In doing so, it causes the 0.73 SHR system at 20 cfm/per-

TABLE 22

Effect of Varying Cooling Capacity and SHR at 20 cfm/person

Total Cooling Capacity[1] (tons)	HVAC Energy Cost ($/yr) SHR=0.78	HVAC Energy Cost ($/yr) SHR=0.73	HVAC Energy Cost ($/yr) SHR=0.70[2]	% of Total Hours Undercooled SHR= 0.78	% of Total Hours Undercooled SHR= 0.73	% of Total Hours Undercooled SHR= 0.70[2]	% of Occupied Hours at RH > 60% SHR= 0.78	% of Occupied Hours at RH > 60% SHR= 0.73	% of Occupied Hours at RH > 60% SHR= 0.70[2]
9.9[2]	--	--	2,835	--	--	0.4	--	--	0.9
10.0	2,856	2,853	--	0.1	0.2	--	0.9	0.9	--
10.5	2,875	2,877	--	~0	0.1	--	0.9	0.9	--
11.0	2,896	2,901	--	0	~0	--	0.9	0.9	--

1 The combined cooling capacities of the front and rear PSZ units, expressed as tons of refrigeration. With total capacities of 10.0 and 11.0 tons, the front and rear units are of equal capacity; with a total capacity of 10.5 tons, the rear unit is 0.5 tons larger than the front unit. The rated capacities and SHRs are at the conditions specified by the Air-Conditioning and Refrigeration Institute.

2 Default values of cooling capacity and SHR computed by DOE-2 for the baseline system operating at 20 cfm/person, when these values are not set in the input file. With the baseline system operating at 20 cfm/person, a nominal 4.9-ton unit serves the front zone and a 5.0-ton unit serves the rear.

son to consume proportionally more power, relative to the 0.78 SHR system, than it did at 5 cfm/person.

To illustrate this, consider the 10.5-ton system operating at 20 cfm/person during the week of July 8 through 12. In previous discussion, it was indicated that -- with the 9.5-ton, 5 cfm/person system -- decreasing the SHR from 0.78 to 0.73 reduced the total cooling provided by the system by 15,700 Btu during that summer week (and reduced the latent cooling by 8,700 Btu). By comparison -- with the 10.5-ton, 20 cfm/person system -- decreasing SHR from 0.78 to 0.73 decreases the total cooling by only 3,500 Btu during that same week (and actually *increases* the latent cooling by 1,600 Btu).

During the week of March 18 through 22, decreasing the SHR of the 9.5-ton, 5 cfm/person system from 0.78 to 0.73 reduced the total cooling by only 1,000 Btu (and did not change the latent cooling at all). With the 10.5-ton, 20 cfm/person system, reducing the SHR results in an *increase* of 5,900 Btu in total cooling, and an increase of 6,500 Btu in latent cooling, over the course of that March week.

Clearly, then, the explanation for why Table 22 shows the 0.73 SHR system with slightly higher annual power costs than the 0.78 SHR system at 20 cfm/person is that the decreases in total cooling with reduced SHR during high-cooling weeks (e.g., July 8-12) are not as large as they were at 5 cfm/person. And, these smaller summer decreases are offset -- or slightly more than offset -- by increases in cooling by the 0.73 SHR system during milder weeks (e.g., March 18-22). The increases in latent cooling by the 0.73 SHR system at 20 cfm/person are largely responsible for why the 0.73 vs. 0.78 SHR power consumption relationship is different at 20 cfm/person than it was at 5 cfm/person.

But as shown in Table 22, the power cost differences between the 0.73 and 0.78 SHR systems are tiny. At an EER of 10, the 3,500 Btu reduction in total cooling provided by the 0.73 SHR system at 20 cfm/person during July 8-12 corresponds to a reduction in power consumption of only 0.35 kWh over the course of the week. The 5,900 Btu increase in total cooling provided by the lower-SHR system at 20 cfm/person during March 18-22 corresponds to an increased power consumption of only 0.59 kWh for the week. Over the course of a year, these tiny differences add up to only a few dollars in power costs.

<u>Hours at elevated temperature and RH</u>. The trends shown in Table 22 for the 20 cfm/person cases are the same as those in Table 21 for 5 cfm/person.

At a given SHR, an increase in total capacity (and hence in sensible capacity) always results in a reduction in the number of hours undercooled. A unit having higher sensible capacity is better able to handle the peak-load hours immediately after startup on warm Monday mornings, when the undercooled hours occur. And, at a given total capacity, the percentage of undercooled hours is lower for the higher-SHR system, since the higher-SHR system has greater sensible capacity.

Also, increases in the total capacity or adjustment of the SHR has no impact on the percentage of occupied hours at RH > 60%. These elevated-humidity hours occur on relatively cool winter days, when the system is operating well below capacity. Thus, increases in total or latent capacity will not significantly reduce the number of elevated-RH hours.

5.18 ALTERNATIVE WEATHER INPUT FILES

For computation of the atmospherically induced loads on the building being modeled, the DOE-2 program requires weather on an hourly basis for the entire year at the building's location. The weather parameters of concern include, for example, dry- and wet-bulb temperatures, wind speed, cloud cover, and total horizontal solar radiation).

Over the years, hourly weather data have been compiled via several different procedures, for use in various applications (such as DOE-2 modeling). These alternative procedures vary according to how they attempt to represent typical weather conditions at the location.

Typical Reference Year (TRY) - The data in TRY weather files describe a single year that has been selected from all of the years for which data are available. This one year has been selected as being the most typical overall for the particular location. For example, 1964 is the selected reference year for Miami. Since any one year is almost certain to have some weeks or months that vary significantly from the multi-year average, TRY weather data are viewed as being potentially useful for comparing one building or HVAC design against another, but as not being suitable for estimating average energy requirements over several years.

Typical Meteorological Year (TMY) - The TMY file contains a typical year that has been compiled by selecting the 12 most typical months from the various years covered by the available weather data. For example, the TMY file for Miami incorporates the January weather data from 1962, the February data from 1974, the March data from 1967, etc.

Weather Year for Energy Calculations (WYEC) - WYEC files take the TMY concept a step further. The most typical months are still selected from different years, but -- when a given individual day within a typical month is considered not to be representative -- the weather data for that day is replaced by the data for that same day from another year (ASHRAE, 1993). WYEC files thus best reflect the long-term mean weather conditions.

All of the previous results in this report were computed using the TMY weather file for Miami. To assess the impact of the weather file, the baseline conditions were re-run using the WYEC file for Miami.

The results using the two weather files are compared in Table 23.

Capacity. As shown in Table 23, the WYEC weather file results in higher computed cooling capacities for both OA flow rates (by 5.6% to 12.7%). This seems to occur in part because the WYEC file incorporates a higher level of solar radiation on the days that are used by the program in computing the capacity.

To illustrate this effect, consider the front office. The cooling capacity of the unit serving this office is computed for the hour when -- according to the LOADS portion of the DOE-2 program -- the peak cooling load is imposed on this space by the weather, by internal sources (occupants, lights, equipment), and by infiltration.

For the WYEC weather file, this hour for the front office is 5 pm on August 2 (with both OA flow rates). During that hour, the WYEC file predicts that the outdoor dry-bulb temperature will be one degree warmer than the TMY file predicts (88 vs. 87 °F). As a result, conduction through the walls, windows, and roof is a few percent greater with WYEC (in terms of kBtu conducted into the office and overhead plenum during that hour). But the increase in the solar component with WYEC (radiation through the window glass, and absorption by the walls and roof) is even greater; solar gain through the front windows increases 20% relative to the TMY conditions (from 2.13 to 2.57 kBtu during that hour). With the WYEC file, the total heat entering the front office and plenum by conduction and radiation from outdoors is 16.26 kBtu during the hour. The increased temperature of the outdoor air also increases the sensible heat added by infiltration, and by the OA mechanically supplied by the system.

Under these conditions at 5 pm on August 2, based on the WYEC file, the program computes a required ARI-rated cooling capacity for the front system of 53.61 kBtu/h at 5 cfm/person, or 66.11 kBtu/h with the higher OA load at 20 cfm/person.

By comparison, with the baseline TMY weather file, the hour with the peak load in the front office (from LOADS) is 5 pm on June 14. This hour is selected primarily because the TMY file records the outdoor dry-bulb temperature as being 91 °F during that hour (one of the hottest hours during the year). The WYEC file records an ambient temperature of only 82 °F during that hour. The high TMY temperature increases heat conduction through the walls, windows, and roof, relative to the WYEC case. With the TMY file, the total heat entering the front office and plenum by conduction and radiation from outdoors is 15.67 kBtu during the 5 pm hour on June 14 (slightly less than the 16.26 kBtu indicated above for the WYEC case at 5 pm on August 2). The increased temperature of the outdoor air in the TMY case on June 14

TABLE 23

The Effect of the WYEC vs. the TMY Weather File on the Results
for the Baseline Small Office in Miami

Output Variable	Value of Output Variable at 5 cfm/person			Value of Output Variable at 20 cfm/person		
	TMY File	WYEC File	% Difference	TMY File	WYEC File	% Difference
Total required cooling capacity (kBtu/h)	103.558	109.336	+5.6	119.193	134.308	+12.7
Annual HVAC energy consumption (kWh)	26,145	25,338	-3.1	29,390	28,531	-2.9
Annual HVAC energy costs ($)	2,510	2,477	-1.3	2,835	2,817	-0.6
Annual building energy consumption (kWh)	60,161	59,406	-1.3	63,406	62,599	-1.3
Annual building energy costs ($)	4,273	4,244	-0.7	4,598	4,584	-0.3
% of all hours (8760 hr/year) undercooled	0.4	0.3	-0.1	0.4	0.2	-0.2
% of occupied hours (3276 hr/year) when RH > 60%	1.2	1.4	+0.2	0.9	1.1	+0.2

also increases the sensible heat added by infiltration, and by the OA mechanically supplied by the system.

Under these conditions at 5 pm on June 14, based on the TMY file, the program computes a required ARI-rated cooling capacity for the front system of 51.42 kBtu/h at 5 cfm/person, or 59.18 kBtu/h with the higher OA load at 20 cfm/person. Consistent with the results in Table 23, these TMY capacities for the front unit are 4% to 12% smaller than the WYEC capacities indicated earlier.

It is of interest to note that the increased conduction of heat into the space in the TMY case at 5 pm on June 14 (resulting from the high TMY outdoor air temperature) is partially offset by increased solar radiation in the WYEC case. Solar gain through the front windows during that hour with the WYEC weather file is 6% greater than in the TMY case (2.62 vs. 2.48 kBtu). And even though there is a much greater nominal temperature differential between the outdoor air and the air inside the plenum in the TMY case at 5 pm on June 14 (19 vs. 10 F°), there is nevertheless 18% more heat conduction across the roof during that hour in the WYEC case (10.08 vs. 8.57 kBtu) due to increased solar absorption on the roof.

<u>Energy consumption and cost</u>. Table 23 indicates that -- despite the greater capacities of the PSZ units in the WYEC case -- the computed HVAC power consumptions are lower by about 3% at both OA flow rates, and HVAC power costs are lower by about 1%, when the WYEC weather file is used.

An analysis of hourly data from a variety of days over the course of the year confirms that the reduced power consumption with the WYEC file results because, on balance over the year, the WYEC weather data impose a slightly lesser load on the system during the hours of system operation. Thus, while the load during the single peak hour in Miami is greater for WYEC -- resulting in a greater computed capacity for the system under WYEC -- the load over the entire year is slightly less with the WYEC file.

To illustrate this effect, consider the week of July 8 through 12. On Monday, July 8, the TMY data predict higher outdoor dry-bulb temperatures in Miami than do the WYEC data (averaging 1.6 F° higher over the 13-hour period while the system is operating). As a result, the TMY file results in greater total cooling by the system on July 8. At 5 cfm/person, the total cooling with the TMY file is 1,225 kBtu (front and rear units combined), about 5% greater than the 1,170 kBtu predicted by the WYEC file. At 20 cfm/person, the TMY file also estimates a 5% larger number (1,470 kBtu vs. 1,400 kBtu).

The following Friday, July 12, the situation is reversed. The WYEC file now predicts the higher outdoor dry-bulb temperature (by 4.3 F° on average), and the WYEC file now results in 6% to 8% greater total cooling on that day (950 vs. 900 kBtu at 5 cfm/person, 1,210 vs. 1,110 kBtu at 20 cfm/person).

For the entire week of July 8 through 12, it turns out that the TMY file results in 2% greater total cooling provided at 5 cfm/person (5,010 vs. 4,930 kBtu), while the WYEC file results in 0.2% greater cooling at 20 cfm/person (6,120 vs. 6,110 kBtu).

A more dramatic effect is seen during the winter week of January 14 through 18. During that week, the TMY file assumes mid-day outdoor dry-bulb temperatures above 75 °F. The cooling coils are typically off -- and, as necessary, the economizer operating -- for perhaps the first couple hours after startup. But after the first couple hours, the coils come on and the economizer is locked out. With the TMY file, the total cooling provided by the system during that week (front and rear units combined) is 2,960 kBtu at the 5 cfm/person OA rate.

By comparison, with the WYEC weather file, outdoor dry-bulb temperatures during the week of January 14 through 18 are much lower, hitting highs of 46 °F on Monday and 64 °F on Friday. For three of the days that week, the cooling coils never come on at all in either office, with the economizer operating all day. With the WYEC file, the total cooling provided during that week at 5 cfm/person is only 1,190 kBtu, only 40% of the value estimated with the TMY file.

The WYEC file generally results in a greater amount latent cooling during any given warm day or week. This occurs even on days when the TMY file assumes a higher moisture content in the outdoor air. The greater latent cooling occurs because, during warm weather, the WYEC system tends to operate at coil surface temperatures averaging 1 to 2 F° colder than the TMY system, thus condensing more moisture, even when the computed average supply air temperature leaving the coils is similar for the two weather files.

<u>Hours at elevated temperature and RH</u>. As shown in Table 23, the WYEC file results in a small reduction in the percentage of hours undercooled at both OA flow rates (from 0.4% of all hours, to 0.2-0.3%). This results because the PSZ units have a somewhat greater capacity in the WYEC case, and are thus better able to handle the elevated coil inlet temperatures during summer Monday morning startups, when the undercooled hours occur.

For example, consider the morning of Monday, July 8. With the 103.558 kBtu/h system at 5 cfm/person computed using the baseline TMY weather file, the front office remains undercooled -- i.e., more than 2 F° above the 75 °F setpoint temperature -- for the first 4 hours after startup, and the rear office remains undercooled for the first 2 hours. But with the 109.336 kBtu/h system at 5 cfm/person computed using the WYEC file, the front office is undercooled for only 1 hour that morning, and the rear office is not undercooled during *any* hour.

However, with the exception of these initial hours after startups on warm Monday mornings, the PSZ units maintain the office space at almost exactly the same temperatures during the other hours, regardless of which weather file is used. Office temperatures maintained by the system during any given hour remain within 0.3 F° of each other as the weather files are changed, with the higher office temperature usually being predicted by the weather file that assumes the higher outdoor temperature.

As shown in Table 23, the WYEC file results in a slight increase in the percentage of occupied hours when the RH is greater than 60% (e.g., 1.4% vs. 1.2% of occupied hours at 5 cfm/person). This small effect appears to result because the WYEC file often assumes a lower outdoor temperature on cool mornings than does the TMY file, with the result that the economizer is operating more often in the WYEC case (bringing in additional outdoor moisture with the cooling coils off). However, this effect is partially offset by the fact that the WYEC file also often assumes a lower moisture content in the outdoor air, so that this additional unconditioned outdoor air does not contribute to indoor RH to the same extent that it would under TMY conditions.

As discussed in earlier sections, the hours of elevated indoor RH tend to occur during the first few hours after startup on cool mornings, when the outdoor temperature in Miami is sufficiently low such that increased OA flow can provide all of the cooling required by the space. The economizer lockout feature on these packaged units dictates that the economizer will operate only during hours when the cooling coils can be off entirely, so that there is an increased influx of potentially humid outdoor air during these hours without any removal of the moisture content. Whether this economizer-induced OA flow will result in indoor RH values above 60% will depend upon, among other things, the amount of OA flow and the moisture content of that OA.

As one example of why Table 23 would be showing additional elevated-RH hours with the WYEC file, consider the morning of Monday, January 28. During the first three hours after startup, the WYEC file assumes an average outdoor temperature of 62.7 °F, and an average moisture content of 0.0120 lb moisture/lb dry air. With the WYEC file, the economizer operates for those first three hours (i.e., the cooling coils do not come on until the fourth hour) for the PSZ system serving the front office. And, during those hours, the moisture content of the indoor air in the front office ranges between 0.0119 and 0.0125 lb/lb -- corresponding to indoor RH values of 63% to 67%.

But with the TMY file on January 28, the average outdoor temperature during those first three operating hours is 69.7 °F, 7 F° warmer than in the WYEC case. (The outdoor moisture content also happens to be higher, 0.0143 lb/lb.) Because of this higher outdoor temperature, the program computes that the economizer cannot come on at all on January 28 in the TMY case. Thus, while the WYEC file results in the coils being off entirely during those first three hours, the TMY file results in the coils providing an average of about 26,700 Btu/h of total cooling, including a reasonable amount of latent cooling. Thus, despite the higher moisture content of the outdoor air in the TMY case, the maximum moisture level in the indoor air of the front office during those three hours is 0.0089 lb/lb -- corresponding to an RH of 48%.

Of course, this situation is not universally true on all days. In a few cases, the TMY file predicts the lower outdoor temperature, and the situation is reversed.

The WYEC file often assumes a lower moisture content in the outdoor air, consistent with the lower outdoor temperature, which can sometimes offset the RH impact of the increased WYEC economizer operation. An illustration of this effect can be seen in the front zone on January 14.

On January 14, the WYEC file assumes that the outdoor temperature remains between 43 and 46 °F during the entire day while the PSZ system is operating, and the outdoor moisture content remains between 0.0034 and 0.0047 lb moisture/lb dry air. With this cool outdoor temperature, the economizer is operating all day, and, consequently, the cooling coils never come on. But because of the low moisture content in the outdoor air, the indoor moisture content in the front office never exceeds 0.0102 lb/lb (corresponding to an indoor RH of 57%). The indoor RH never reaches as high as 60% on that day with the WYEC file.

But with the TMY file, the outdoor temperature at startup on January 14 is 68 °F, increasing to 78 °F at mid-day. The outdoor moisture content is generally at values above 0.0135 lb/lb. Under these conditions, the economizer never comes on during that day. But the TMY outdoor temperature *is* low enough such that the cooling coils *also* remain off for the first three hours after startup, and are on at only a reduced level during the fourth and fifth hours. As a result of the latent heat released by the building occupants, the high moisture content in the OA, and the absence of any latent removal by cooling coils, the RH in the front office is well above 60% during the first five hours on January 14 with the TMY file. In fact, during the first four hours, it is computed to be 95% to 100%.

Thus, despite the fact that the WYEC file often results in a greater degree of economizer usage -- and despite the fact that, in humid climates, increased economizer usage might intuitively be expected to lead to increased indoor RH -- the above example for January 14 shows that, under some circumstances, the WYEC file can nevertheless result in *fewer* hours at elevated RH.

The above examples for January 14 and 28 illustrate that various offsetting phenomena occur with the switch from the TMY to the WYEC weather files, and that the impact on indoor RH is not straightforward. However, as shown in Table 23, the net effect over the course of an entire year is for the WYEC file to result in a slightly greater number of occupied hours above 60% RH.

SECTION 6

THE IMPACT OF HUMIDITY CONTROL

6.1 INTRODUCTION

A primary concern with increasing the OA flow rate in humid climates is the impact that such an increase might have on the indoor RH levels. Outdoor RH levels in Miami can often be well above 60% [the indoor maximum recommended by ASHRAE to reduce the growth of fungi, such as molds and mildew (ASHRAE, 1989a), and to maintain occupant comfort (ASHRAE, 1992a)]. An increased influx of such humid OA -- combined with the latent energy being generated inside the building -- could result in a high latent load on the HVAC system, possibly resulting in indoor RH levels exceeding 60% during system operation.

Thus, two primary questions arise when considering increased OA in humid climates.

a) What will be the impact of such an OA increase on the RH levels in the space, if no steps are taken to reduce RH?

b) What will the costs be of trying to maintain the space below 60% RH when the OA is increased to 20 cfm/person?

6.1.1 Impacts of Increased OA on RH When No RH Reduction Steps are Taken

Table 3 in Section 4 predicts that -- when OA is increased from 5 to 20 cfm/person with the baseline PSZ units -- the percentage of occupied hours above 60% RH will actually decrease, from 1.2% to 0.9%. The DOE-2.1E modeling predicts that elevated-RH hours will remain essentially unchanged -- actually, *decrease* slightly -- when OA is increased.

As discussed in Section 5.12.2, this tiny predicted decrease in elevated-RH hours with the PSZ units results because -- at increased OA -- the cooling coils operate at lower temperatures, and provide increased latent cooling that more than offsets the increase in latent load created by the increased OA flow. Moreover, as discussed in Section 5.17, the number of elevated-RH hours (at either 5 or 20 cfm/person) is not impacted by increases or decreases in either the HVAC system capacity or SHR, because the elevated-RH hours occur immediately after startup on cool mornings when the coils are operating far below capacity (or perhaps are not operating at all).

Some investigators indicate that this apparent non-effect on increased OA results because the DOE-2.1E model does not properly simulate certain parameters that will impact the indoor humidity (Shirey et al., 1995; Shirey and Rengarajan, 1996). Specifically, DOE-2.1E does not account for two phenomena.

a) The moisture capacitance of the of the building materials and furnishings. DOE-2 does not account for moisture adsorbed due to the infiltration of humid air when the system is off overnight and over weekends and holidays. This moisture would be released when the system comes back on, providing an incremental additional latent load throughout the day.

b) The condensed moisture on the surfaces of the cooling coils, which can re-evaporate into the circulating air stream when the coils cycle off (but when the air handler continues to operate) during operating hours.

Shirey et al. utilize a model -- a broad building flow and energy model incorporating a variation of the DOE-2 model -- which addresses these moisture sources. Their model predicts that, when OA in this small Miami office is increased to 20 cfm/person, the percentage of occupied hours above 60% RH -- rather than remaining essentially unchanged -- would increase to 11-23%. And these elevated-RH hours would occur not only on cool days, but on hot summer days as well.

Because the elevated-RH hours are predicted to sometimes occur during hot days, when the system will be operating near capacity, the calculations by Shirey et al. predict that changes in system capacity and SHR *will* impact the percentage of occupied hours at elevated RH.

Because the DOE-2.1E model used in this study does not address moisture capacitance and re-evaporation, it is not possible to independently assess the significance of these phenomena under different conditions.

6.1.2 Approaches for Reducing the Impact of Increased OA on RH

In concept, two general approaches might be considered for reducing the impact of the increased OA on space RH.

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

b) Alternatively, incorporate RH control as well as temperature control into the HVAC control system. The RH control could be achieved using, e.g., desiccants, or overcooling with reheat.

HVAC systems in general -- and especially systems serving small spaces, as considered in this study -- generally function only with temperature control, and not with RH control. Humidity control involves somewhat greater system installed cost, greater energy costs, and higher maintenance requirements (which the owners of a strip mall or small office building might not have the personnel to conveniently provide). Thus, in practice, RH control is considered only in cases where it is critical to the occupants of the space.

6.1.3 Limitations of the DOE-2 Model for RH Analysis

In this section, the DOE-2.1E model is used to assess some alternatives for reducing the RH impacts of increased OA, based on the two general approaches listed in Section 6.1.2. Primarily, the assessment here addresses the incorporation of an enthalpy controller into the system, with over-cooling and reheating of the air to control moisture (an option which falls under the second of the two approaches above).

As indicated in Section 6.1.1, it is recognized that the DOE-2 model might not fully model RH effects, since it does not simulate building moisture capacitance or re-evaporation off the cooling coils. Thus, in the analysis here using that model, it must be understood that the RH levels during a given hour could be higher than those predicted here -- i.e., the RH performance of the control option being evaluated could be poorer, and the energy consumption/costs higher -- if the capacitance and re-evaporation phenomena were considered. Despite these limitations, the DOE-2 results here are felt to show, at least qualitatively, the relative magnitude of the effects of RH control, and the possible potential of enthalpy control.

6.2 REDUCING RH THROUGH ADJUSTMENTS TO HVAC DESIGN USING TEMPERATURE CONTROL ALONE

6.2.1 Results from Other Investigators

In the study conducted by the Florida Solar Energy Center, or FSEC (Shirey et al., 1995; Shirey and Rengarajan, 1996), six HVAC design modifications were considered to assess their potential for reducing the impact of increased OA on indoor RH, on energy consumption and costs, and on system installed cost. This study addressed a small office in Miami having a PSZ system, essentially identical to the office and the system considered in this report.

The six design modifications considered in that study were as follows.

a) Reduction of the PSZ sensible heat ratio (SHR) to 0.73.

b) An increase in the system cooling efficiency, from an EER of 10 to an EER of 12 (combined with reduction in the SHR to 0.73).

c) Use of demand-controlled ventilation (DCV), so that OA flow rate is reduced accordingly to maintain the 20 cfm/person ratio when the number of occupants decreases. This calculation assumed an accurate sensor of actual building occupancy, based on CO_2 monitoring.

d) Use of an enthalpy recovery wheel, exchanging sensible and latent heat between the incoming OA and the outgoing building exhaust, in an effort to reduce the load on the system.

e) Use of a heat pipe, whereby a very efficient recuperative heat exchanger (based on heat pipe technology) is used to reduce the temperature of the warm return air entering the coils, by exchanging it against the cool supply air leaving the coils. The principle is that -- with the cooler entering air -- the coils provide a colder outlet air, thus condensing more moisture. The cooling coils provide about the same amount of sensible cooling as they would without the heat pipe, but operate at a lower temperature.

f) Addition of a 100% OA cooling system. In this approach, the two PSZ units treating the zones of the office space are supplemented by a third high-efficiency, very low-SHR unit designed specifically to cool and dehumidify the inlet OA stream. This cooled/dehumidified OA is then introduced into the supply air downstream of the two main PSZ units.

In all cases, the system operation was controlled by space temperature alone, not by enthalpy.

Hours at elevated RH. In summary, the computations showed that -- at 20 cfm/person -- DCV, the enthalpy wheel, the heat-pipe system, and the 100% OA unit were able to keep elevated-RH occupied hours down to about the same number as that experienced by the baseline system (conventional PSZ, SHR = 0.78, EER = 10) at 5 cfm/person (i.e., about zero). That is, if the increase from 5 to 20 cfm/person were accompanied by one of these design modifications, the number of occupied hours above 60% RH would hardly increase (or would not increase at all), even with a temperature-based (rather than an enthalpy-based) control system.

By comparison, simply increasing the OA flow in the baseline system without any modifications (other than to increase capacity) increased the percentage of occupied hours above 60% RH from about zero to about 16%. This is dramatically different from the DOE-2 results presented in Table 3 (where the OA increase has essentially no effect on elevated-RH hours), due to the capacitance and re-evaporation issues discussed previously.

Reducing the system SHR to 0.73 would reduce the impact of the increased OA somewhat, according to FSEC, so that -- instead of 16% of occupied hours being above 60% RH -- the value falls to 11%. But there is still a substantial increase in elevated-RH hours at 20 cfm/person relative to the baseline case at 5 cfm/person (zero hours). By comparison, the DOE-2 modeling in the current report predicts that a decrease in SHR would have no effect on elevated-RH hours at all (see Section 5.17), with the percentage of elevated-RH occupied hours remaining at 0.9-1.2%. This results because, according to DOE-2, all of the elevated-RH occupied hours occur after startup on cool Monday mornings, when the coils are operating at greatly reduced capacity (if at all), and their SHR accordingly does not fully come into play. This differs from the model used by FSEC, where -- due to moisture capacitance and re-evaporation -- many of the elevated-RH occupied hours occur during warm weather, when the SHR of the equipment plays a more important role.

In the calculations by FSEC, an increase in the EER in combination with a decrease in the SHR provides no additional impact on the percentage of elevated-RH hours. The high-EER case (with SHR = 0.73) results in the same percentage of elevated-RH occupied hours (11%) as the standard-EER case with SHR = 0.73. This is consistent with the predictions of DOE-2, that EER (or EIR) should not have a significant impact on elevated-RH hours (see Section 5.16).

<u>Energy consumption and cost</u>. According to the FSEC predictions, an increase in OA from 5 to 20 cfm/person would increase annual HVAC energy consumption and cost by approximately 9% in this Miami building, if no modifications were made to the baseline system (other than to increase capacity). This is somewhat less than the 12.4% and 12.9% increase predicted by DOE-2 for HVAC energy consumption and cost (respectively), shown in Table 3.

If the increase in OA were accompanied by installation of an enthalpy wheel, HVAC energy consumption and costs would actually *decline* compared to the 5 cfm/person baseline case, according to the FSEC estimates. Thus, the enthalpy wheel would not only avoid any increase in elevated-RH hours resulting from the OA increase, as discussed above, but would save energy in the process.

If the increase in OA were accompanied by a switch to the higher-EER coils, FSEC's HVAC energy consumption and costs would again *decline* relative to the 5 cfm/person baseline. The DOE-2 results provided a similar prediction for the case of the improved-efficiency coils (see Section 5.16.2). Although the improved-efficiency coils have no impact on RH performance, they do, of course, reduce energy consumption and cost.

If the increase in OA were accompanied by a switch to DCV, the HVAC energy penalty associated with the OA increase would decline from the 9% (with no modifications), to 6-8%. Thus, DCV, if it could be implemented rigorously, would reduce the energy penalty while, at the same time, substantially reducing the increase in elevated-RH hours accompanying the increased OA.

If the OA increase were accompanied by a switch to lower-SHR coils, the HVAC energy penalty would increase above that for the unmodified system, to 13-14%. Thus, the modest reduction in the increase in elevated-RH hours with increased OA is offset by an increase in the energy penalty of the reduced-SHR system.

The increased energy penalty predicted by FSEC with the lower-SHR coils differs from the DOE-2 results presented in Section 5.17.2. DOE-2 predicted almost no change in power consumption and costs at 20 cfm/person as SHR was decreased from 0.78 to 0.73. Actually, DOE-2 did predict a *tiny* increase in energy cost at lower SHR, due to increased latent cooling energy consumption during mild weather. But this increase was insignificant compared to the effect predicted by FSEC.

If the increase in OA were accompanied by the addition of a heat pipe system, or of a 100% OA unit, the HVAC energy penalty would be substantially greater than the 9% with the unmodified baseline system, according to the FSEC estimates. The heat pipe system would increase the penalty to about 27%, the 100% OA unit to 21-25%. Thus, while both of these options are very effective at reducing or eliminating hours at elevated RH, they have a significant energy penalty.

Installed cost. The installed cost estimates prepared in conjunction with the FSEC modeling predicted that an increase from 5 to 20 cfm/person for the baseline system (with no modifications other than a capacity increase) would increase the installed cost of the system by less than 2%, resulting from the capacity increase.

All six of the system modifications considered by FSEC involved an increase in the installed cost at 20 cfm/person significantly greater than the baseline 2%.

The enthalpy wheel had the least increase in installed cost (11%). Thus, the only one of the six options predicted to simultaneously eliminate elevated-RH hours and save energy when OA is increased, also turned out to be the least expensive to install, under FSEC's assumptions.

Reduction of the SHR to 0.73 had the next lowest installed cost increase, 14%. But, as discussed earlier, this option gave only a modest reduction in the increase in elevated-RH hours, and had a larger energy penalty than did the 20 cfm/person baseline.

Simultaneously increasing the EER, along with the SHR decrease, resulted in an installed cost increase of 21%. The high-EER/low-SHR case resulted in an energy cost savings at 20 cfm/person, due to the improved coil efficiency, but it still provided only the modest reduction in the increase in elevated-RH hours. And the installation cost penalty is high.

Conversion to DCV resulted in an installed cost increase of just over 14%, about the same as the reduced-SHR option. So DCV -- if it could be implemented rigorously -- would greatly reduce the elevated-RH hours resulting from the OA increase, would slightly reduce the associated energy penalty, and would result in a moderate installed cost penalty.

The heat pipe system and the 100% OA unit (at 20 cfm/person) both resulted in significant increases in installed cost (relative to the baseline at 5 cfm/person): 21% for the 100% OA unit, 43% for the heat pipe system. Thus -- although both of these approaches would be effective at reducing or eliminating the elevated-RH hours resulting from the OA increase -- they would significantly increase both the energy penalty and the installed cost penalty.

6.2.2 Cooling Coil Set-Up vs. Shut-Down During Unoccupied Hours

The analysis in Section 6.2.1 assumes that the coils are shut off altogether overnight and during weekends and holidays, when the office is unoccupied. This allows the indoor RH to increase during off-hours, resulting from the infiltration of humid outdoor air. Alternatively, rather than shutting the system down altogether, the thermostat might instead be set up to, e.g., 81 °F, as discussed in Section 5.11. If the thermostat were set up, off-hour RH levels would be reduced, at least during hot weather when the temperature became sufficiently high to cause the cooling coils to activate during unoccupied hours.

Hours at elevated RH - DOE-2 model predictions. Switching from shut-down to 81 °F set-up will not have any impact on RH levels -- during either occupied or unoccupied hours -- when the weather is cool, and when the set-up temperature is thus too high to activate the coils when the office is unoccupied.

As discussed in Section 5.11, DOE-2 predicts that all elevated-RH *occupied* hours will occur on cool mornings. Thus, switching to 81 °F set-up has no impact on the number of occupied hours annually above 60% RH, as shown in Table 15. In addition, the indoor RH will sometimes exceed 70% during *un*occupied hours during cool weather (because the outdoor RH will exceed 90%). Switching to 81 °F set-up will not reduce the number of *un*occupied elevated-RH hours during cool weather, either.

But during warm weather, there is a significant impact. As discussed previously, DOE-2 predicts that no elevated-RH *occupied* hours occur during warm weather. The cooling coils are predicted to operate at sufficient power after startup on warm Monday mornings to remove sufficient moisture from the air as soon as occupancy begins. But the RH levels commonly reach 60 to 75% during *un*occupied hours overnight and on weekends in the baseline case when the system is shut down. *Switching to 81 °F set-up substantially reduces these warm-weather elevated-RH unoccupied hours.* It eliminates them altogether on the warmer days when the temperature is sufficiently high such that the coils are activated for even a modest fraction of the unoccupied hours. Although the controller is controlling office temperature rather than humidity, the activity by the cooling coils to maintain the 81 °F office temperature is predicted by DOE-2 to remove sufficient moisture to maintain RH below 60% during many warm-weather unoccupied hours.

Thus -- according to the DOE-2 predictions -- switching from shut-down to 81 °F set-up would not impact RH during occupied hours, and hence would not contribute to the goal of maintaining occupant comfort (ASHRAE, 1992a). But, by greatly reducing that portion of the elevated-RH hours that occur when the office is unoccupied during warm weather, it would make a major contribution toward reducing fungal growth (ASHRAE, 1989a).

This statement is true regardless of whether the system is operating at 5 or at 20 cfm OA/person. As modeled here, there is no outside air flow when the cooling coils cycle on during unoccupied hours in response to the 81 °F set-up temperature; the system is simply recirculating office air. Hence, the OA rate during occupied hours has no impact on the ability of the system to eliminate elevated-RH hours during unoccupied periods.

As discussed in Section 6.1.1, DOE-2 predicts that increased OA will not increase elevated RH hours during occupied periods. Thus, the conclusion above could be stated in another manner: *If an increase in OA from 5 to 20 cfm/person were accompanied by a switch from off-hour shut-down to 81 °F set-up, there would be a significant reduction in the total number of elevated-RH hours* (with this reduction occurring during warm-weather unoccupied hours).

<u>Hours at elevated RH - the FSEC model</u>. As indicated in Section 6.1.1, FSEC utilizes a model which addresses the moisture capacitance of building materials and furnishings, and moisture re-evaporation from the coils when they cycle off. These features -- not incorporated in the DOE-2.1E model used here -- can result in significantly more hours above 60% RH, when the system is operating at 20 cfm/

person. And many of the additional elevated-RH hours occur during occupied periods in warm weather, when DOE-2 predicts that elevated RH will not occur.

As with the baseline configuration used in the current study, the FSEC study assumed that the cooling coils were shut down completely during occupied hours. With the DOE-2 model available here, it is not possible to reproduce the FSEC results, or to quantify what the impact would have been had the FSEC study considered off-hour set-up rather than shut-down. However, based upon the underlying principles, it is possible to consider qualitatively how thermostat set-up vs. system shut-down would have impacted the FSEC results.

As with the DOE-2 model, it would not be expected that the FSEC model would predict any significant effect of set-up vs. shut-down on elevated-RH hours (occupied or unoccupied) during cool weather. The impact will occur only during warm weather, when the set-up temperature would trigger coil operation during unoccupied hours.

To assess the effects of thermostat set-up on the FSEC modeling during hot weather, it is necessary to understand the effects that the moisture capacitance and re-evaporation phenomena are having on the FSEC model predictions. These effects are presented in the FSEC publications (Shirey et al., 1995; Shirey and Rengarajan, 1996), and are summarized below.

On a typical hot summer weekday, the DOE-2 model -- which *excludes* the effects of capacitance and re-evaporation -- predicts that office RH steadily increases overnight when the cooling coils are shut down, as humid outdoor air infiltrates. The RH will commonly exceed 60% for some portion of the night, and be above 60% when the coils come on in the morning. When the system starts up in the morning, the RH drops steeply, and remains well below 60% during the main occupied portion of the day, when the sensible cooling loads are high and the coils are operating near capacity (so that latent removal is also high). During the last three hours of coil operation during the day -- between 6 and 9 p.m. -- RH increases (though commonly remaining below 60%), because the reduced sensible cooling load during those hours result in the coils operating at a lower part-load ratio, with resulting lower latent removal. When the coils go off, the RH begins its steady rise that will continue throughout the night due to infiltration.

FSEC predicts that the moisture capacitance phenomenon will impact this pattern in two ways. During the off-hours, when DOE-2 predicts a steady increase in RH, moisture capacitance tends to hold RH levels steady, preventing them from rising above 60%. The reason is that the infiltrating moisture from outdoors is being adsorbed by certain building materials and furnishings. Then, throughout the main portion of the day after startup the next morning -- when DOE-2 predicts a significant decrease in RH -- capacitance tends again to keep the RH levels steady, at levels well above what DOE-2 would predict, because this sorbed moisture is being desorbed.

Thus, moisture capacitance has the effect of decreasing the number of unoccupied hours at elevated RH, relative to the DOE-2 predictions, and of increasing the RH during occupied hours (although not necessarily to levels above 60%).

The coil re-evaporation phenomenon also impacts the DOE-2 pattern in two ways. First, during the off-hours, there appears to be a continuing source of moisture from the drain pans associated with the cooling coils. At least for the one summer day (August 22) analyzed by Shirey et al., this off-hour evaporation added about 5 percentage points to the RH at 20 cfm/person, relative to what the RH would otherwise have been. Second, when the coils are operating at sufficiently reduced capacity during the day -- i.e., when the sensible cooling load is sufficiently reduced such that the coils cycle off for significant periods -- re-evaporation can contribute significant spikes in the RH. These spikes are much more severe at 20 than at 5 cfm/person. Comparing DOE-2 results for August 22 against the curve presented by Shirey et al., major RH contributions from re-evaporation seem to occur when the coil part-load ratio drops below about 0.7. Depending upon the ambient conditions, this ratio can occur during the final hours of operation (6 to 9 p.m.), and during the initial hours after startup at 6 a.m., on certain days during warm weather.

In summary, DOE-2 -- *without* capacitance and re-evaporation included -- can commonly show a certain number of elevated-RH hours overnight on warm weekdays, but no elevated-RH hours after system startup. This is true at either 5 or 20 cfm/person. FSEC -- *with* capacitance and re-evaporation -- commonly shows RH levels below 60% almost all the time at 5 cfm/person. But, at 20 cfm/person -- depending on ambient conditions on a given day -- the FSEC model can show not only a significant number of unoccupied hours above 60% RH, but also some portion of the occupied hours, in the morning and early evening.

Switching to 81 °F thermostat set-up during unoccupied hours, rather than system shut-down, would likely have the following impacts on the FSEC predictions (on those days when the ambient temperature is sufficiently high to trigger coil operation during at least a modest fraction of the unoccupied hours).

Moisture capacitance effects would be reduced. Moisture removal during occasional operation of the coils overnight and on weekends would reduce moisture adsorption by the building materials and furnishings during those off hours. As a result, less sorbed moisture would be available for re-release into the office space after the coils start up in the morning.

Likewise, coil and drain pan re-evaporation effects would be reduced over nights and weekends. As indicated, the FSEC model predicts a substantial re-evaporation effect during off-hours when the coils are shut down, presumably from water standing in drain pans. This off-hour re-evaporation can be sufficient to raise the RH above 60%, even while the capacitance phenomenon is adsorbing all of the moisture contained in the off-hour infiltrating air. Operation of the coils for a couple of the overnight hours would be sufficient to remove a significant amount of this evaporated moisture from the office air, reducing average RH. This would be true even if it were assumed that all of the re-evaporated moisture thus condensed remained in the drain pans, where it would potentially have the opportunity to re-evaporate a second time.

Thus, even with the FSEC model (incorporating capacitance and re-evaporation), off-hour set-up rather than shut-down would significantly reduced the number of elevated-RH hours during warm weather. Most of the elevated-RH *un*occupied hours during warm weather seem to be due to re-evaporation from the drain pans; much of this re-evaporated moisture would be removed from the air, reducing elevated-RH unoccupied hours.

The elevated-RH *occupied* hours that can occur soon after startup result because moisture desorption, along with re-evaporation from cycling, reduced-load coils, create a moisture spike which is superimposed on top of the already-elevated off-hour RH that resulted from off-hour drain pan re-evaporation. If the coils had come on for a couple hours overnight triggered by the 81 °F set-up temperature, the off-hour RH level that existed at startup would be reduced, and the portion of the moisture spike contributed by desorption would also be reduced. Thus, set-up (rather than shut-down) would also be expected to reduce the number of elevated-RH occupied hours predicted by the FSEC model at 20 cfm/person.

With either the DOE-2 or the FSEC models, thermostat set-up would *not* eliminate *all* warm-weather elevated-RH hours. It would reduce or eliminate only those hours occurring on days sufficiently warm such that the set-up temperature triggered off-hour coil operation. The example day shown in FSEC's publications -- Thursday, August 22 -- is one warm day where set-up vs. shut-down would *not* have an impact on RH (at least not with an 81 °F set-up temperature). On that particular day, the highest overnight temperature in the office space is 79 °F (with the TMY weather file), so the coils are never activated during the unoccupied period.

Energy consumption and cost. As discussed in Section 5.11, and shown in Table 15, the reduction in elevated hours achievable with thermostat set-up can be achieved with a minimal energy consumption and cost penalty.

As shown in Table 3, increasing the OA flow from 5 to 20 cfm/person without changing the temperature control strategy -- i.e., with total system shut-down during unoccupied hours -- increases annual HVAC energy costs by \$325/yr. As shown in Table 15, if the increase in OA were accompanied by a switch to 81 °F set-up, this would increase energy cost only by an additional \$10/yr (a total cost increase of \$335/yr). Thus, a significant reduction in total elevated-RH hours could be achieved at a minimal additional cost.

6.3 REDUCING RH USING ENTHALPY CONTROL

In applications where humidity control is critical, enthalpy controllers are installed on the system. One common approach for controlling humidity is to overcool the gas stream to condense moisture, followed by reheat to return the supply air to the temperature needed for proper thermal control. Another approach is to install desiccant units in the system. These steps increase energy consumption and cost, and, especially for the desiccant units, can increase installed cost and maintenance requirements.

Enthalpy control is most commonly used in selected zones where some special need -- e.g., rare document storage, special manufacturing processes, etc. -- is felt to warrant the costs involved. Enthalpy control does not appear to be commonly utilized solely for the purposes of occupant comfort and prevention of fungal growth in offices. Recognizing this reality, the assessment here of the use of enthalpy control was undertaken to define what the costs and benefits of using this approach would be, relative to the other approaches presented in Section 6.2, if it were decided that occupant comfort and indoor microbial control were sufficiently important to warrant this step.

In this assessment, the humidity control approach considered is the traditional procedure of overcooling followed by reheat. Reheat has been generally discouraged for years due to the energy inefficiencies involved, and is generally prohibited by the Florida code (FDCA, 1993). It is being considered in this assessment despite these concerns, on the basis that -- if indoor microbial control is considered to be important -- it is necessary to weigh the effectiveness and the energy penalties of the reheat approach for comparison against the other approaches.

It was recognized at the outset that -- since the DOE-2.1E model does not incorporate the moisture capacitance and re-evaporation phenomena -- the estimates here will not be addressing the full latent load, and hence underestimating the full energy impacts of enthalpy control. But it is anticipated that the DOE-2 analysis here will suggest at least the relative impacts of the variables influencing enthalpy control. Moreover -- when steps are taken to maintain RH at reduced levels -- the impacts of capacitance and re-evaporation will become less, and the difference between the FSEC and DOE-2 RH predictions will shrink.

Unfortunately, the DOE-2 model is able to model enthalpy control only during occupied hours. Setting a maximum allowable relative humidity -- e.g., MAX-HUMIDITY = 60 -- will not cause the coils to cycle on during unoccupied hours when the humidity exceeds that level. Humidity would be controlled during unoccupied hours only if there were simultaneous off-hour temperature control -- e.g. thermostat set-up to 81 °F during unoccupied hours -- which caused the coils to cycle on during certain unoccupied hours. And, even in that case, the humidity control would occur only during those particular unoccupied hours when the temperature controller cycled the coils on. Thus, it is not possible with the standard DOE-2 model to simulate around-the-clock RH control.

Recognizing the limitations of the model, the results of the DOE-2 calculations -- controlling RH at 60% or less during occupied hours -- are presented in Table 24.

6.3.1 The Effect of Enthalpy Control at 5 cfm/person

Energy consumption and cost. For three different coil capacity/SHR combinations, Table 24 shows the predicted effect at 5 cfm/person of switching from standard temperature-based system control ("No Humidity Control") to enthalpy control during occupied hours ("Maximum Humidity = 60%").

In all three capacity/SHR cases, conversion to enthalpy control at a constant OA rate of 5 cfm/person is estimated to increase HVAC energy cost by about $145 per year, an increase of less than 6%.

As shown in Table 24, adjustments to the total refrigeration capacity and the sensible heat ratio have only a minor impact on energy cost, with or without enthalpy control. This is consistent with the results discussed in Section 5.17. The explanation for this minor effect of capacity and SHR -- presented in that earlier section for the temperature-controlled case -- also applies for the enthalpy-controlled case.

The manner in which enthalpy control impacts energy consumption and costs at a constant OA rate of 5 cfm/person can be illustrated by considering hourly performance on selected winter and summer days.

First, consider the winter days of Thursday, January 10, and Friday, January 11.

At the end of the occupied period on Thursday, January 10, the temperatures of both zones are 74-75 °F in the temperature-controlled case, and a little below 71 °F in the enthalpy-controlled case. This difference results because the enthalpy-controlled case has provided substantial additional cooling of the supply air during the day in order to remove moisture, and the reheat coils have supplied sufficient heat only to maintain the zone temperatures at the heating set-point (70 °F) throughout the day. By comparison, in the temperature-controlled case, the system has provided no heating during the day; the zones were at 75-77 °F just before startup, so that only cooling was ever called for. As a result, with temperature control, the zones are maintained at the cooling set-point (75 °F) all day. But in both cases, the moisture content of the zone air at the end of the day is the same (0.008 lb moisture/lb dry air), corresponding to an RH of about 43% in the 75 °F temperature-controlled case, and about 49% in the 70 °F enthalpy-controlled case.

TABLE 24

Effect of Humidity Control During Occupied Hours, Under Various Conditions

System Conditions	HVAC Energy Cost ($/yr) No Humidity Control	HVAC Energy Cost ($/yr) Maximum Humidity = 60%	% of Total Hours Undercooled No Humidity Control	% of Total Hours Undercooled Maximum Humidity = 60%	% of Occupied Hours at RH > 60% No Humidity Control	% of Occupied Hours at RH > 60% Maximum Humidity = 60%
Program-Calculated Capacities/SHRs						
OA = 5 cfm/person - 8.6 tons refrigeration capacity - SHR = 0.75	2,510	2,656	0.4	0	1.2	0
OA = 20 cfm/person - 9.9 tons refrigeration capacity - SHR = 0.70	2,835	2,925	0.4	0	0.9	0
Increased Capacity/SHR = 0.78						
OA = 5 cfm/person - 9.0 tons capacity, SHR = 0.78	2,541	2,683	0.1	0	1.2	0
OA = 20 cfm/person - 10.0 tons capacity, SHR = 0.78	2,856	2,945	0.1	0	0.9	0
Increased Capacity/SHR = 0.73						
OA = 5 cfm/person - 9.0 tons capacity, SHR = 0.73	2,527	2,669	0.3	0	1.2	0
OA = 20 cfm/person - 10.0 tons capacity, SHR = 0.73	2,853	2,940	0.2	0	0.9	0

Overnight between Thursday and Friday, the outdoor temperature drops from 72 °F at shut-down (according to the TMY weather file) to a low of 62 °F at startup the next morning. The ambient humidity ranges between 0.011 and 0.013 lb/lb (with ambient RH values varying between 77% and 94%). Between shut-down on Thursday and startup on Friday -- due to the combined effects of infiltration, shell heat transfer, and release of heat stored in the building materials and furnishings -- the zone temperatures rise by 1 to 1.5 F° in the temperature-controlled case (so that the temperature just before Friday startup is 75-76.5 °F). In the enthalpy-controlled case, the temperature rise is slightly greater -- about 2 F° -- so that the temperature before startup is 72.5-73 °F. The moisture in the infiltrating air causes overnight office humidity levels to reach a peak of 0.013 lb/lb, corresponding to an RH of roughly 70%, varying slightly depending upon office temperature.

According to the TMY weather file, Friday, January 11, is a fairly mild day, with ambient temperatures holding in the low to mid-60's throughout the morning, and reaching a high of 78 °F in mid-afternoon. The ambient RHs are in the range of 85% to 95% in the morning, and 65% to 80% in the afternoon.

When the air handler starts up on Friday morning in the *temperature-controlled* case, the cooling coils do not cycle on because the office temperature is below the cooling set-point of 75 °F. In fact, in the temperature-controlled case at 5 cfm/person, the coils do not cycle on until noon. As a result, no moisture is removed, and the relative humidity remains above 60% all morning (a total of 5 occupied hours), but drops below 60% immediately when the coils finally cycle on. The total cooling coil energy consumption during that day in the 5 cfm/person temperature-controlled case is just over 428,000 Btu (of which 57,000 Btu is latent cooling). Of course, there is no heating energy consumption. As on Thursday, the office space is just below 75 °F at the end of the occupied period on Friday.

When the system starts up on Friday morning in the *enthalpy-controlled* case, the coils cycle on immediately in response to the elevated RH that exists. In the first hour after startup, the average RH in the office space is reduced to about 52% (and the office temperature is reduced below 71 °F). None of the occupied hours on Friday are above 60% RH in the enthalpy-controlled case, as would be expected. Of course, this added moisture removal is reflected in the energy consumption. On Friday, the enthalpy-controlled system at 5 cfm/person has a total cooling coil energy consumption of 864,000 Btu (of which 89,000 is latent), and a total reheat energy consumption of 132,500 Btu. The total cooling and heating energy consumption that day -- 996,500 Btu -- is 2.3 times greater than that for the temperature-controlled case.

This relationship between energy consumptions in the temperature- and enthalpy-controlled cases is representative of what is generally experienced on such cool winter days when the elevated-RH occupied hours tend to occur (in the absence of enthalpy control). The increase in consumption for the enthalpy-controlled case is not so great on certain winter days when the TMY weather file predicts less humid conditions.

This energy consumption relationship is very different during hot weather. Since the temperature-controlled case controls occupied-hour RH levels effectively during hot weather, the difference in consumption between the two cases is small.

To illustrate this, consider the week beginning on Monday, July 8. Overnight, prior to startup on Monday, the outdoor temperature has dropped from 82 to 76 °F. The overnight ambient humidity has ranged from 0.018 to 0.019 lb/lb, corresponding to an RH of about 90%.

Under these conditions, the indoor temperature is about 85 °F just before Monday morning startup in the temperature-controlled case, and 83 to 83.5 °F in the enthalpy-controlled case. Interestingly, even after the coils have been off over the summer weekend, a couple-degree office temperature differential still exists between the two cases, resulting from the office space having been 4 F° cooler in the enthalpy-controlled case -- 71 vs. 75 °F -- when the system shut down on Friday evening. The indoor humidity has also ranged between 0.018 and 0.019 lb/lb overnight between Sunday and Monday, corresponding to an RH greater than 70%.

When the system starts up on Monday morning, July 8, in the *temperature-controlled* case, the cooling coils cycle on at almost full capacity, since the space is well above the 75 °F cooling set-point at startup. As a result, the office RH averages below 60% starting with the first occupied hour, and there are no elevated-RH occupied hours during the day. During this day -- when the outdoor temperature holds in the mid to upper 80's throughout most of the morning and afternoon -- the 5 cfm/person temperature-controlled system consumes 1,234,000 Btu of cooling energy, 134,000 Btu of which is latent cooling.

When the *enthalpy-controlled* system starts up on Monday morning, it actually consumes slightly *less* cooling energy than does the temperature-controlled case. This presumably occurs because the enthalpy-controlled office space is at a slightly lower temperature at startup, as indicated above. Of course, the RH is below 60% throughout the day with this system, as it is with the temperature-controlled system. During the course of this day, the cooling energy consumption by the enthalpy-controlled system is 1,225,000 Btu (of which 135,000 Btu is latent cooling) at 5 cfm/person. The reheat coils never have to activate on this day; the system is able to maintain the RH below 60% without having to over-cool the supply air. Thus, on this day, the enthalpy-controlled system actually consumes slightly less cooling/heating energy than does the temperature-controlled system.

Because the enthalpy-controlled system is able to provide the needed moisture removal without reheat on July 8, the system is controlled by the cooling set-point.

Hence, the office space is at about 75 °F during most of the day, just as is the temperature-controlled system. Later in the day, as the sensible heat load decreases, the enthalpy-controlled system does perform a small amount of supply air over-cooling in order to remove moisture, and this reduces office temperature to 73 to 74 °F. But, since this over-cooling is not sufficient to drop the office temperature below 70 °F, the reheat coils do not come on.

The high sensible load created by the weekend shutdown is largely addressed on Monday. Peak loads are less severe during the remainder of the week, and, as a result, energy consumption by the enthalpy-controlled system again becomes greater than that for the temperature-controlled system. For example, on Friday, July 12, the enthalpy-controlled system is having to over-cool the supply air in order to remove sufficient moisture, and the office temperature ranges between 71 and 72 °F, close to the heating set-point (compared to a steady 75 °F with the temperature-controlled system). The enthalpy-controller reheat coils actually have to come on during the last operating hour on Friday, to prevent office over-cooling. As a result, on Friday, total cooling and heating energy consumption by the enthalpy-controlled system is 1,019,000 Btu, 13% greater than the 900,000 Btu in cooling energy required by the temperature-controlled system.

The preceding examples for selected days in January and July illustrate why -- over the course of the year -- the enthalpy-controlled case increases HVAC energy consumption (i.e., cooling, heating, and air handler consumption) by about 11% at a constant OA rate of 5 cfm/person, and HVAC energy cost by about 6%.

Over 85% of the increase in HVAC energy consumption with the enthalpy-controlled system over the course of the year results from increased cooling coil (compressor and condenser fan) energy. Of the 2,820 kWh increase in annual HVAC consumption caused by enthalpy control at 5 cfm/person, only 380 kWh (13%) is the result of increased heating energy caused by the reheat coils. Increased cooling coil energy is responsible for the remainder. Of course, air handler energy requirements are unchanged by the switch to enthalpy control. Also, the small heating energy requirements to maintain the heating set-point during normal heating hours (*excluding* reheat) remain essentially unchanged.

<u>Hours at elevated temperature and RH</u>. As shown in Table 24, switching to enthalpy control (with MAX-HUMIDITY = 60) eliminates all undercooled hours at 5 cfm/person. This occurs because on warm Monday mornings -- when the undercooled hours occur with the temperature-controlled system -- the enthalpy-controlled system encounters a cooler zone temperature upon startup, since the enthalpy system had left the zones several degrees cooler on Friday evening. Thus, less cooling is demanded from the enthalpy-controlled system upon startup, and undercooled hours are avoided.

For illustration, again consider the morning of Monday, July 8. Just before startup on this morning, the office space temperature is a little below 85 °F in the temperature-controlled case, and about 83 to 83.5 °F in the enthalpy-controlled case, because the enthalpy-controlled zones had been about 4 F° cooler at shut-down on Friday evening, as indicated earlier. Starting at 83 °F, the 5 cfm/person enthalpy-controlled case is able to reduce the temperature of both zones sufficiently close to the 75 °F cooling set-point -- i.e., to 77 °F or less -- beginning with the first hour after startup on Monday. Thus, the enthalpy system is not charged with any undercooled hours on that day. But starting at 85 °F, the temperature-controlled case is slightly above 77 °F for the first 4 hours in the front zone (with temperatures of 77.1 to 78.3 °F), and the first 2 hours in the rear zone (77.2 to 77.9 °F). The 1.5 to 2 F° warmer zone encountered at startup by the temperature-controlled system would appear to explain why this system resulting in undercooled hours while the enthalpy-controlled system did not.

Due to the lower startup temperature, the enthalpy-controlled system is able to avoid undercooled hours while consuming slightly less cooling energy than the temperature-controlled system. During the first 4 hours on July 8 -- when the temperature-controlled system is encountering undercooled hours in one or both zones -- the enthalpy-controlled cooling coils are consuming a total of 387,000 Btu (front and rear units combined), while the temperature-controlled cooling coils are consuming 391,000 Btu, about 1% more energy.

Table 24 also confirms that -- as commanded -- the enthalpy-controlled case eliminates occupied hours above 60% RH at the 5 cfm/person OA rate. This occurs, of course, because the enthalpy-controlled system provides more cooling (and, as necessary, reheat) on cool mornings when elevated-RH hours occur.

Consider the morning of Monday, January 14. Just before startup on this day, the zones in the enthalpy-controlled case are at 68.5 to 69.5 °F, 1 to 2.5 F° cooler than the 69.5 to 72 °F in the temperature-controlled case. Again, this differential at startup exists because the enthalpy-controlled zones were 3.5 to 4 F° cooler at shut-down on Friday evening. In both cases, the humidity before startup is 0.013 lb/lb, corresponding to an RH of about 87% in the cooler, enthalpy-controlled zones, and about 82% in the warmer, temperature-controlled zones.

When the temperature-controlled system starts up on January 14, the air handler comes on, but neither cooling nor heating are provided. The zone temperature is at or above the heating set-point, and below the cooling set-point, so that neither of the coils are activated. As latent heat is generated inside the offices, and as the outdoor humidity increases (in terms of lb/lb), the indoor humidity increases to a peak of 0.02 lb/lb during the morning. At 5 cfm/person, the cooling coils do not come on until the third hour after startup in the rear zone, and the fourth hour in the front zone -- and then at greatly reduced capacity. As a result, the RH is above 60% during the first 6 hours in the front zone, and the first 4 hours in the rear.

By comparison, when the enthalpy-controlled system starts up on January 14, the cooling coils cycle on immediately -- and the reheat activates as well. As a result, there are no elevated-RH occupied hours during the course of the day. The reheat coils maintain the office space just below 71 °F throughout the day, in response to the heating set-point.

Of course, as emphasized previously, there is an energy penalty associated with this dehumidification. During the first 6 hours, when one or both of the zones are at elevated RH in the temperature-controlled case, the enthalpy-controlled system consumes a total of 570,000 Btu for cooling and reheat in the two zones combined (with about one-third of this total being energy for reheat), at 5 cfm/person. By comparison, the temperature-controlled system consumes only 79,000 Btu during this same 6-hour period -- only 14% of the enthalpy-controlled total. For the entire day, the temperature-controlled system consumes 395,000 Btu, 40% of the 997,000 Btu consumed in the enthalpy-controlled case.

6.3.2 The Effect of Enthalpy Control on Increased Ventilation Rates

Energy consumption and cost. As shown in Table 24, at a constant OA rate of 20 cfm/person, the increase in energy cost resulting from a switch from temperature to enthalpy control is slightly less than it is when this switch is made at a constant OA rate of 5 cfm/person. At 5 cfm/person, the energy cost increase of switching to enthalpy control is predicted to be about $145 per year for each of the capacity/SHR combinations shown in the table (e.g., $2,656/yr with enthalpy control vs. $2,510/yr with temperature control). At 20 cfm/person, this increase is reduced to about $90 ($2,925/yr vs. $2,835/yr).

Another way of looking at this difference is that -- with temperature control -- increasing the OA rate from 5 to 20 cfm/person results in a predicted energy cost increase of about $325/yr (from $2,510 to $2,835 for the first case in Table 24). By comparison, if this switch to 20 cfm/person were accompanied by a switch from temperature control to enthalpy control, the energy cost increase would be $415/yr (from $2,510 to $2,925), $90 greater than if the accompanying switch to enthalpy control were not made.

As with the increase in energy *cost*, the increase in HVAC energy *consumption* caused by the switch to enthalpy control is less when the OA is 20 cfm/person. At a constant OA rate of 5 cfm/person, the switch from temperature control to enthalpy control results in added energy consumption of about 2,800 kWh/year, for all three capacity/SHR combinations considered in Table 24. More than 85% of this increase at 5 cfm/person is due to increased compressor and condenser fan consumption, and less than 15% by reheat coil operation. By comparison, at a constant OA rate of 20 cfm/person, this penalty is only about 1,800 kWh/year. And only about 3% of this 1,800 kWh increase at 20 cfm/person is due to reheat coil operation.

This modest 1,000 kWh/yr reduction in the energy penalty (1,800 instead of 2,800 kWh/yr), achieved by switching to enthalpy control at 20 instead of 5 cfm/person, results from two phenomena.

a) About one-third of the 1,000 kWh/yr reduction results because the enthalpy-controlled system sometimes consumes *less* reheat energy at 20 cfm/person than it does at 5 cfm/person, in cool and in mild weather. The enthalpy-controlled system tends to provide less sensible cooling (and more latent cooling) at the higher OA rate during cool and mild weather, with the result that less reheat is required. Reheat energy is also reduced at 20 cfm/person for the reason described in b) below.

b) The remainder of the 1,000 kWh reduction results because the enthalpy-controlled system sometimes consumes *less* total cooling energy at 20 than at 5 cfm/person, in mild weather. Typically -- when over-cooling and then reheating -- the enthalpy-controlled system maintains the offices at the heating set-point of 70 °F. But at mild conditions when the RH in the office is just slightly above 60%, the enthalpy-controlled system (as modelled by DOE-2) sometimes reduces RH instead by simply letting the office temperature rise to the cooling set-point of 75 °F, at a substantial savings in cooling (and reheat) energy. This occurs during a greater number of hours when operating at the increased OA rate.

To illustrate these effects, it is useful to consider three separate weeks throughout the year -- a cool week (ending January 18), a mild week (ending March 22), and a hot week (ending July 12).

During each of these weeks, the temperature-controlled system at 20 cfm/person consumes more power for total cooling -- and more power for latent cooling -- than does the temperature-controlled system at 5 cfm/person, as would be expected.

Also, during each of these weeks, the enthalpy-controlled system at either OA rate consumes more power for total cooling, more power for latent cooling, and (of course) more power for reheat, than does the temperature-controlled system at the same OA rate. Again, this would be expected.

But during the week ending January 18, the enthalpy-controlled system at 20 cfm/person consumes *less* energy for reheat than does the enthalpy-controlled system at 5 cfm/person. And during the week ending March 22, the enthalpy-controlled system at 20 cfm/person consumes *less* energy for total cooling and *less* energy for reheat than does the enthalpy-controlled system at 5 cfm/person. But during the week ending July 12 -- when the system tends to be operating near full capacity, and essentially no reheat is needed -- the enthalpy-controlled system at 20 cfm/person consumes more power for total (and latent) cooling than does the enthalpy-controlled system at 5 cfm/person.

Thus, the largest energy penalty for converting from temperature to enthalpy control occurs when the switch is made at the lower OA rate. When the switch to enthalpy control is instead implemented at a higher OA rate, the energy penalty associated with the conversion is less during cool and mild weeks, and is about the same during hot weeks. As a result, the net effect over the course of the entire year is a modest reduction in penalty for conversion to enthalpy control as the OA rate increases.

Consider the week ending January 18. During this week, the outdoor temperature is generally mild, and outdoor humidity is high. As a result, when the OA rate is increased from 5 to 20 cfm/person with the enthalpy-controlled system, the latent cooling for the week more than doubles, from 484 kBtu to 1,010 kBtu for the two zones combined. But at these outdoor temperature and RH conditions with these PSZ units, the *sensible* cooling with the enthalpy-controlled system actually *decreases* with the increase in OA (from 3,982 to 3,757 kBtu), partially off-setting the increase in latent cooling. And because there is less sensible cooling at 20 cfm/person, less reheat is required in order to maintain office temperature at the heating set-point (or above); the increase in OA decreases reheat requirements for the week from 648 to 367 kBtu.

In summary, for the week ending January 18, an increase from 5 to 20 cfm/person with the enthalpy-controlled system increases *total* cooling requirements by just over 300 kBtu, and decreases reheat requirements by just over 280 kBtu. Thus, the OA increase results in only a minor 20 kBtu increase in total cooling plus reheat energy for the week.

Consider now the week ending March 22. During this week, the outdoor temperature is mild; it averages less than 1 F° warmer than during the January week considered above, according to the TMY file for Miami. But the outdoor humidity is much lower during this March week, averaging only 0.010 lb moisture per lb dry air (compared to 0.012 lb/lb during the January week). Under these conditions, the moisture content of the office space (in terms of lb/lb) often corresponds to an RH just above, or just below, 60%. As a result, a small change in office temperature can sometimes determine whether the office is above or below the RH set-point for the enthalpy controller.

The effect of this "borderline" situation is that the enthalpy control system will over-cool and reheat the supply air during some hours of the day, in order to maintain the RH below 60% at the heating set-point of 70 °F. But then, during other hours, the system -- as modelled by DOE-2 -- will allow office temperature to rise to the cooling set-point of 75 °F, where the warmer temperature will cause the RH to drop below 60% without the substantial energy penalty associated with the over-cooling and reheating. Increasing the OA rate from 5 to 20 cfm/person with the enthalpy-controlled system increases the number of hours during this March week when the RH control is accomplished by letting the office temperature rise. Hence, the increase in OA rate reduces both the total cooling energy and the reheat energy during this week.

First, consider the 5-hour period between 7 am and noon on Wednesday, March 20. During this period, the system is maintaining the office space at about 71 °F, near the heating set-point, with either OA rate. The outdoor temperature during this period ranges between 70 and 74 °F, and the outdoor humidity ranges between 0.0089 and 0.0096 lb/lb (corresponding to RHs of about 55% to 60% at office temperatures).

Under these conditions, for this 5-hour period in the morning, an increase in the enthalpy-controlled system's OA rate from 5 to 20 cfm/person increases sensible cooling by 2.5% (from 298 to 306 kBtu), and latent cooling by one third (from 28 to 39 kBtu). But, for reasons that are not clear, the model calculates that this increase in OA *decreases* reheat requirements substantially (from 40 kBtu to zero). As a result, the total cooling plus reheat energy requirements for the enthalpy-controlled system during these 5 hours are somewhat reduced (from 366 kBtu to 345 kBtu) by virtue of increasing the OA from 5 to 20 cfm/person.

This reduction during this 5-hour period on the morning of March 20 might thus be attributed to a phenomenon similar to that responsible for the effects during the January week, discussed above. That is, at 20 cfm/person, more of the total cooling is latent cooling, with the result that the temperature of the supply air is reduced to a lesser extent and the need for reheat (to maintain the 71 °F office temperature) is thus reduced. However, it is not apparent why the model is predicting the elimination of reheat requirements at 20 cfm/person during this period. The mixed-air temperatures entering the coils are about the same at both OA rates, essentially the same amount of sensible cooling is provided by the coils at both rates, and, correspondingly, exactly the same reduction in air temperature occurs across the coils at both rates. It is curious that the 5 cfm/person case should be computed as providing 40 kBtu of reheat under these conditions, while the 20 cfm/person case is providing none.

Now consider the afternoon hours on March 20, when a different phenomenon occurs. The outdoor air is somewhat drier in the afternoon compared to the morning -- 0.0081 to 0.0089 lb/lb, corresponding to RHs of 45% to 55% at office temperatures. Outdoor temperatures are relatively mild during the afternoon, ranging from 72 to 75 °F, according to the TMY weather file.

Presumably because of the increased influx of this relatively dry outdoor air at 20 cfm/person, the enthalpy-controlled system at this higher OA flow rate shifts from the heating set-point of 70 °F to the cooling set-point of 75 °F during the entire afternoon on March 20. That is, RH is reduced by raising the office temperature. By comparison, at 5 cfm/person, the enthalpy controller keeps the system operating at the heating set-point (i.e., reducing RH by over-cooling and reheating) during the entire afternoon, until the last hour before shut-down in the evening.

The energy impact of this action by the controller is substantial. For the 7-hour period from noon through 7 pm, the total cooling energy consumed by the enthalpy-controlled system drops from 422 kBtu to 284 kBtu as OA is increased from 5 to 20 cfm/person (although the latent cooling increases modestly, from 25 to 36 kBtu). Correspondingly, the OA increase decreases reheat energy consumption during this 7-hour period, from 29 kBtu to zero.

These results for Wednesday, March 20, explain the results observed for the entire week ending March 22. For the entire week, the increase in OA from 5 to 20 cfm/person in the enthalpy-controlled system results in a 12% *decrease* in sensible cooling requirements (from 3,849 to 3,371 kBtu), a 57% *increase* in latent cooling (from 312 to 490 kBtu), and almost complete *elimination* of reheat requirements (decreasing reheat consumption from 238 to 10 kBtu). The total cooling plus reheat energy consumption is reduced by a little more than 500 kBtu (from 4,400 to 3,872 kBtu) by the increase in OA.

Finally, consider the week ending July 12. During this hot week, essentially no reheat is needed in the enthalpy-controlled system; even the temperature-controlled system, which operates near full capacity without reheat capability, maintains the RH below 60%. Also, the outdoor conditions are so hot and humid that the enthalpy-controlled system will always be controlled by the cooling set-point. Thus, neither of the phenomena discussed above, that caused increased OA to reduce energy consumption during the January and March weeks, would be expected to come into play during July.

And indeed, this is the case. Increasing OA from 5 to 20 cfm/person with the enthalpy-controlled system increases total cooling during this July week by 1,000 kBtu (975 kBtu of which is increased latent cooling). This is almost identical to the 1,100 kBtu increase in total cooling (940 kBtu increase in latent cooling) experienced when OA is increased in the *temperature*-controlled system. And there is essentially no change in reheat energy consumption caused by the OA increase in the enthalpy-controlled system; there is essentially no reheat energy consumed during July at either OA rate.

In summary, with the enthalpy-controlled system, an increase in OA from 5 to 20 cfm/person causes almost no change in energy consumption during the January week, a 500 kBtu decrease in consumption during the March week, and a 1,000 kBtu increase in consumption during the July week. By comparison, with the temperature-controlled system, that same increase in OA causes an increase in energy consumption during each of those three weeks. The net results are that:

a) with the enthalpy-controlled system, increasing OA from 5 to 20 cfm/person increases annual energy consumption; but

b) the resulting increase in annual energy consumption with the enthalpy-controlled system is less than the increase that would be caused by that same OA increase with the temperature-controlled system.

Hours at elevated temperature and RH. As shown in Table 24, the enthalpy-controlled system operating at 20 cfm OA/person eliminates all undercooled hours and, as commanded, all occupied hours at elevated RH. This occurs for the same reasons discussed in connection with the enthalpy-controlled system at 5 cfm/person (see Section 6.3.1).

SECTION 7.

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APPENDIX A

Baseline Input DOE-2.1E File:
4,000 ft^2 Office in Miami Strip Mall

INPUT LOADS ..

```
TITLE        LINE-1   * SMALL OFFICE STUDY #2 *
             LINE-2   *4000 FT2 OFFICE IN STRIP MALL (MIAMI)*
             LINE-3   *CASE 01A:  -- BASE CASE --*
             LINE-4   *HVAC:  2 PSZ UNITS WITH ECONOMIZERS*
             LINE-5   *OA = 5 CFM/PERSON* ..

$  DOE-2.1E FILE BY BRUCE HENSCHEL -- REVISED SEPTEMBER 6, 1995

$  THIS SMALL OFFICE IS MODIFIED FROM A SIMILAR ONE MODELED BY THE
$  FLORIDA SOLAR ENERGY CENTER (FSEC).  IT IS A 4000 FT2 "SLICE" OUT
$  OF A LARGER, SINGLE-STORY STRIP MALL.  THIS OFFICE IS IN TURN SUB-
$  DIVIDED INTO TWO 2000 FT2 ZONES, EACH COOLED WITH A DEDICATED
$  PACKAGED SINGLE-ZONE DIRECT EXPANSION AIR-CONDITIONER (PSZ), HEATED
$  WITH ELECTRIC RESISTANCE HEATING.

$  BECAUSE THIS OFFICE IS ADJOINED ON EACH SIDE BY OTHER OFFICES,
$  LEASED BY OTHER TENANTS WHO ARE CONDITIONING THEIR ADJOINING
$  SPACE USING THEIR OWN DEDICATED UNITS, THIS MODELING EFFORT
$  ADDRESSES ONLY THE FRONT AND REAR (EXTERIOR) WALLS.  IT IS
$  ASSUMED THAT THE TWO SIDE WALLS ARE THERMALLY NEUTRAL, AND THE
$  MODEL THUS PRETENDS THAT THEY DO NOT EXIST.

        DIAGNOSTIC                CAUTIONS ..
        ABORT                     ERRORS ..
        RUN-PERIOD                JAN 1 1974 THRU DEC 31 1974 ..

        BUILDING-LOCATION         LATITUDE = 25.8   LONGITUDE = 80.3
                                  ALTITUDE = 7      TIME-ZONE = 5
                                  AZIMUTH = 0       HOLIDAY = YES ..

$  LATITUDE, LONGITUDE, ETC., CORRESPOND TO MIAMI, FLORIDA.

                $-----BUILDING SHADING-----$

        FRONT-OVERHANG = BUILDING-SHADE
                                  X=40    Y=100    Z=9    AZ=0
                                  H=10    W= 40    TILT=180 ..

        FRONT-PARAPET  = BUILDING-SHADE
                                  X=40    Y=100    Z=13   AZ=0
                                  H=2     W= 40    TILT=90 ..

                $-----LOADS SCHEDULES-----$

OCCUPY-1       = SCHEDULE      THRU DEC 31
                               (MON,FRI)    (1,6)(0)     (7)(.3)
                                            (8,11)(1)    (12,14)(.8,.4,.8)
                                            (15,17)(1)   (18,20)(.3,.1,.1)
                                            (21,24)(0)
                               (SAT,HOL)    (1,24)(0) ..
```

```
LIGHTS-1       = SCHEDULE     THRU DEC 31
                              (MON,FRI)    (1,6)(.05) (7)(.3)
                                           (8,17)(1)  (18,20)(.3,.1,.1)
                                           (21,24)(.05)
                              (SAT,HOL)    (1,24)(.05) ..

INFILT-1       = SCHEDULE     THRU DEC 31
                              (ALL)        (1,24)(1) ..


                  $-----SPACE CONDITIONS-----$

OFFICE-ENVIRON= SPACE-CONDITIONS
                  TEMPERATURE      = (72)
                  PEOPLE-SCHEDULE  = OCCUPY-1
                  AREA/PERSON      = 150
                  PEOPLE-HEAT-GAIN = 450
                  LIGHTING-SCHEDULE= LIGHTS-1
                  LIGHTING-TYPE    = REC-FLUOR-NV
                  LIGHTING-W/SQFT  = 1.8
                  LIGHT-TO-SPACE   = 1
                  EQUIP-SCHEDULE   = LIGHTS-1
                  EQUIPMENT-W/SQFT = 0.75
                  INF-METHOD       = AIR-CHANGE
                  INF-SCHEDULE     = INFILT-1
                  AIR-CHANGES/HR   = 0.1
                  FLOOR-WEIGHT     = 70
                  ZONE-TYPE        = CONDITIONED ..


                  $-----CONSTRUCTIONS AND GLASS TYPE-----$

EX-WALL-LAYER = LAYERS        MATERIAL = (SC01,CB11,IN32,GP02)
                              INSIDE-FILM-RES = 0.68 ..

$  THIS LAYER OF MATERIALS IN THE EXTERIOR WALL INCLUDES 1-IN.
$  STUCCO, 8-IN. HEAVY-WEIGHT HOLLOW BLOCK, 3/4-IN. POLYSTYRENE
$  INSULATION, AND 5/8-IN. GYPSUM BOARD, PLUS THE INSIDE AND
$  OUTSIDE FILM RESISTANCES.

IN-WALL-LAYER = LAYERS        MATERIAL = (GP02,AL21,GP02)
                              INSIDE-FILM-RES = 0.68 ..

CEILING-LAYER = LAYERS        MATERIAL = (AC02)
                              INSIDE-FILM-RES = 0.61 ..

  R-12-INSUL  = MATERIAL      TH=0.333   COND=0.024   DENS=15   S-H=0.17 ..
  METAL-DECK  = MATERIAL      TH=0.021   COND=26      DENS=480  S-H=0.10 ..

ROOF-LAYER    = LAYERS        MATERIAL = (BR01,R-12-INSUL,METAL-DECK)
                              INSIDE-FILM-RES = 0.61 ..

SLAB-LAYER    = LAYERS        MATERIAL = (CC03,CP02)
                              INSIDE-FILM-RES = 0.92 ..


EXT-WALL      = CONSTRUCTION    LAYERS = EX-WALL-LAYER
                                ABSORPTANCE = 0.7   ROUGHNESS = 3 ..

INT-WALL      = CONSTRUCTION    LAYERS = IN-WALL-LAYER ..

CEILING       = CONSTRUCTION    LAYERS = CEILING-LAYER ..
```

```
MALL-ROOF       = CONSTRUCTION     LAYERS = ROOF-LAYER
                                   ABSORPTANCE = 0.8 ..

SLAB            = CONSTRUCTION     LAYERS = SLAB-LAYER ..

BACK-DOOR       = CONSTRUCTION     U-VALUE = 0.59
                                   ABSORPTANCE = 0.84 ..

FRNT-DOOR       = GLASS-TYPE       PANES = 1
                                   SHADING-COEF = 0.55
                                   GLASS-CONDUCTANCE = 1.47
                                        $ DEFAULT GLASS-CONDUCTANCE WHEN PANES=1
                                   FRAME-CONDUCTANCE = 1.245 ..
                                        $ FRAME-COND. FOR THERMALLY BROKEN ALUMINUM

ALL-WINDOWS     = GLASS-TYPE       PANES = 1
                                   SHADING-COEF = 0.55
                                   GLASS-CONDUCTANCE = 1.47
                                   FRAME-CONDUCTANCE = 1.245 ..


                $-----SPACE DEFINITIONS-----$

$  THE 13-FT-TALL BUILDING INCLUDES 9-FT CEILINGS IN THE OFFICE
$  AREA, WITH A 4-FT PLENUM OVERHEAD.

FRONT-PLENUM  = SPACE
                 X=0   Y=0   Z=0   AZ=0
                 AREA = 2000
                 VOLUME = 8000
                 TEMPERATURE = (72)
                      $ PLENUM TEMPERATURE FOR LOADS CALCULATION IS SAME AS
                      $ THAT FOR OFFICE SO THAT CEILING IS THERMALLY NEUTRAL.
                 FLOOR-WEIGHT = 5
                 ZONE-TYPE = PLENUM ..

   FRONT-P-WALL  = EXTERIOR-WALL
                    X=40   Y=100   Z=9   AZ=0
                    H=4    W=40    TILT=90
                    CONSTRUCTION = EXT-WALL ..

   FRONT-ROOF    = ROOF
                    X=40   Y=100   Z=13   AZ=0
                    H=50   W=40    TILT=0
                    CONSTRUCTION = MALL-ROOF
                    GND-REFLECTANCE = 0 ..

REAR-PLENUM   = SPACE
                 LIKE FRONT-PLENUM ..

   REAR-P-WALL   = EXTERIOR-WALL
                    X=0   Y=0   Z=9   AZ=180
                    H=4   W=40   TILT=90
                    CONSTRUCTION = EXT-WALL ..

   REAR-ROOF     = ROOF
                    X=0   Y=0   Z=13   AZ=180
                    H=50  W=40   TILT=0
                    CONSTRUCTION = MALL-ROOF
                    GND-REFLECTANCE = 0 ..

$  AGAIN, NO SIDE WALLS ARE INDICATED FOR THE PLENUMS, ASSUMING
$  NO HEAT TRANSFER WITH THE PLENUMS OF ADJOINING OFFICES (OR
$  BETWEEN THE FRONT AND REAR PLENUMS).
```

```
FRONT-OFFICE   = SPACE
                 X=0   Y=0   Z=0   AZ=0
                 AREA = 2000
                 VOLUME = 18000
                 SPACE-CONDITIONS = OFFICE-ENVIRON ..

   FRONT-O-WALL =  EXTERIOR-WALL
                   X=40   Y=100   Z=0   AZ=0
                   H=9    W=40    TILT=90
                   CONSTRUCTION = EXT-WALL ..

       FRONT-WIN-1 = WINDOW     X=1.1   Y=2.6   H=6.3   W=17.8   SETBACK=0.33
                     GLASS-TYPE = ALL-WINDOWS      FRAME-WIDTH = 0.1  ..

       FRONT-WIN-2 = WINDOW    LIKE FRONT-WIN-1
                     EXCEPT    X=21.1   W=7.3 ..

       FRONT-WIN-3 = WINDOW    LIKE FRONT-WIN-1
                     EXCEPT    X=28.6   Y=7.1   H=1.8   W=2.8 ..

       FRONT-WIN-4 = WINDOW    LIKE FRONT-WIN-1
                     EXCEPT    X=31.6   W=7.3 ..

       FRONT-DOOR  = WINDOW    X=28.67   Y=0.17   H=6.66   W=2.66   SETBACK=0.33
                     GLASS-TYPE = FRNT-DOOR        FRAME-WIDTH = 0.17 ..

   FRONT-CEILING=  INTERIOR-WALL
                   AREA = 2000
                   X=40   Y=100   Z=9   AZ=0   TILT=0
                   CONSTRUCTION = CEILING
                   NEXT-TO  FRONT-PLENUM ..

   FRONT-SLAB    =  UNDERGROUND-FLOOR
                   AREA = 2000
                   X=40   Y=50   Z=0   AZ=0   TILT=180
                   CONSTRUCTION = SLAB
                   U-EFFECTIVE = 0.035 ..

$  THE CALCULATED OVERALL U-VALUE FOR THE SLAB + CARPET + INSIDE-FILM-RESISTANCE
$  IS 0.39 BTU/HR-FT2-Fo.  THE REDUCED U-EFFECTIVE -- USING DOE'S CONVENTION FOR
$  AVOIDING OVER-ESTIMATION OF HEAT TRANSFER THROUGH THE SLAB -- IS CALCULATED
$  BY MULTIPLYING THIS U BY THE RATIO OF THE SLAB PERIMETER (180 FT) TO THE SLAB
$  AREA (2000 FT2).

REAR-OFFICE    = SPACE
                 LIKE FRONT-OFFICE ..

   REAR-O-WALL  =  EXTERIOR-WALL
                   X=0   Y=0   Z=0   AZ=180
                   H=9   W=40  TILT=90
                   CONSTRUCTION = EXT-WALL ..

       REAR-WIN-1 = WINDOW   X=5.1   Y=2.6   H=5.8   W=8.3   SETBACK=0.33
                    GLASS-TYPE = ALL-WINDOWS      FRAME-WIDTH = 0.1 ..

       REAR-WIN-2 = WINDOW   LIKE REAR-WIN-1
                    EXCEPT  X=27.1 ..

       REAR-DOOR  = DOOR
                    X=18   Y=0   H=7   W=3
                    CONSTRUCTION = BACK-DOOR ..
```

```
   MID-O-WALL    =  INTERIOR-WALL
                     X=0    Y=50    Z=0    AZ=180
                     H=9    W=40
                     CONSTRUCTION = INT-WALL
                     NEXT-TO  FRONT-OFFICE ..

   REAR-CEILING =  INTERIOR-WALL
                     AREA = 2000
                     X=0    Y=0    Z=9    AZ=180    TILT=0
                     CONSTRUCTION = CEILING
                     NEXT-TO  REAR-PLENUM ..

   REAR-SLAB     =  UNDERGROUND-FLOOR
                     AREA = 2000
                     X=0    Y=50    Z=0    AZ=180    TILT=180
                     CONSTRUCTION = SLAB
                     U-EFFECTIVE = 0.035 ..

                 $-----REPORTS-----$

LOADS-REPORT     VERIFICATION=(LV-D)
                 SUMMARY=(LS-C) ..

END ..
```

COMPUTE LOADS ..

INPUT SYSTEMS ..

```
                 $-----SYSTEMS SCHEDULES-----$

OFFICE-COOL-T = SCHEDULE     THRU DEC 31
                             (MON,FRI)    (1,6)(99)   (7,19)(75)
                                          (20,24)(99)
                             (SAT,HOL)    (1,24)(99) ..

OFFICE-HEAT-T = SCHEDULE     THRU DEC 31
                             (MON,FRI)    (1,6)(55)   (7,19)(70)
                                          (20,24)(55)
                             (SAT,HOL)    (1,24)(55) ..

FAN-ON = SCHEDULE            THRU DEC 31
                             (MON,FRI)    (1,6)(0)   (7,19)(1)
                                          (20,24)(0)
                             (SAT,HOL)    (1,24)(0) ..

OCCUPY-1 = SCHEDULE          THRU DEC 31
                             (MON,FRI)    (1,6)(0)    (7)(.3)
                                          (8,11)(1)   (12,14)(.8,.4,.8)
                                          (15,17)(1)  (18,20)(.3,.1,.1)
                                          (21,24)(0)
                             (SAT,HOL)    (1,24)(0) ..

                 $-----DEFINITION OF ZONES-----$

OFFICE-ZAIR    = ZONE-AIR

   OA-CFM/PER   = 5 ..
```

```
OFFICE-ZCONT   = ZONE-CONTROL

   DESIGN-COOL-T = 74
   COOL-TEMP-SCH = OFFICE-COOL-T
   DESIGN-HEAT-T = 71
   HEAT-TEMP-SCH = OFFICE-HEAT-T
   THERMOSTAT-TYPE = PROPORTIONAL
   THROTTLING-RANGE= 2  ..
        $ DEFAULT THROTTLING-RANGE VALUE WHEN THERMOSTAT-TYPE = PROPORTIONAL

FRONT-OFFICE = ZONE

   ZONE-TYPE      = CONDITIONED
   ZONE-AIR       = OFFICE-ZAIR
   ZONE-CONTROL   = OFFICE-ZCONT
   SIZING-OPTION = ADJUST-LOADS ..
        $  SIZING-OPTION MUST BE SET AT ADJUST-LOADS WHEN AN OVERHEAD PLENUM IS
        $  PRESENT, SO THAT THE SYSTEMS CALCULATION WILL NOT IGNORE ROOF LOAD
        $  WHEN SIZING EQUIPMENT.

REAR-OFFICE  = ZONE         LIKE FRONT-OFFICE ..

FRONT-PLENUM = ZONE

   ZONE-TYPE      = PLENUM
   DESIGN-COOL-T = 90
   DESIGN-HEAT-T = 50
   SIZING-OPTION = ADJUST-LOADS ..

REAR-PLENUM  = ZONE         LIKE FRONT-PLENUM ..


                $-----DEFINITION OF SYSTEMS-----$

$  THE FRONT AND REAR OFFICES EACH HAVE A PACKAGED SINGLE-ZONE, CONSTANT-VOLUME,
$  DIRECT EXPANSION COOLING SYSTEM WITH CENTRAL ELECTRIC HEATING.

PKG-AC-CONTROL = SYSTEM-CONTROL

   MAX-SUPPLY-T  = 100
   MIN-SUPPLY-T  = 55
   ECONO-LIMIT-T = 68
   ECONO-LOCKOUT = YES ..
        $ DEFAULT ECONO-LOCKOUT VALUE FOR PSZ SYSTEMS

PKG-AC-AIR = SYSTEM-AIR

   OA-CONTROL = TEMP
   MAX-OA-FRACTION = 1  ..

$  THESE SETTINGS INDICATE THAT THE SYSTEM HAS AN ECONOMIZER (CONTROLLED BY
$  SUPPLY AIR T), AND CAN OPERATE AT UP TO 100% OA.

PKG-AC-FAN = SYSTEM-FANS

  FAN-CONTROL = CONSTANT-VOLUME
  FAN-SCHEDULE = FAN-ON
  NIGHT-CYCLE-CTRL = CYCLE-ON-FIRST
  FAN-PLACEMENT = DRAW-THROUGH ..

$  THESE SETTINGS INDICATE THAT THE FANS ARE TURNED OFF OVERNIGHT AND ON
$  WEEKENDS/HOLIDAYS WHEN NO-ONE IS IN THE BUILDING, BUT THAT THEY WILL CYCLE ON
$  IF NEEDED TO MAINTAIN TEMPERATURE.
```

```
PKG-AC-TERM = SYSTEM-TERMINAL

   MIN-CFM-RATIO = 1.0
   REHEAT-DELTA-T = 0 ..

$  TRADITIONAL SETTINGS FOR CONSTANT VOLUME SYSTEMS.

PKG-EQUIP-FRONT = SYSTEM-EQUIPMENT

   COOLING-EIR       = 0.341
        $ THIS EIR CORRESPONDS TO EER = 10.0 BTU/H OUTPUT/WATT OF ELECTRIC INPUT
   HEATING-EIR       = 1.0  ..
        $ IF HEATING-EIR WERE ALLOWED TO DEFAULT, PROGRAM WOULD MODEL HEAT PUMP.

PKG-EQUIP-REAR  = SYSTEM-EQUIPMENT

   LIKE PKG-EQUIP-FRONT ..


FRONT-SYSTEM = SYSTEM

   SYSTEM-TYPE       = PSZ
   SYSTEM-CONTROL    = PKG-AC-CONTROL
   SYSTEM-AIR        = PKG-AC-AIR
   SYSTEM-FANS       = PKG-AC-FAN
   SYSTEM-TERMINAL   = PKG-AC-TERM
   SYSTEM-EQUIPMENT= PKG-EQUIP-FRONT
   HEAT-SOURCE       = ELECTRIC
   RETURN-AIR-PATH = PLENUM-ZONES
   PLENUM-NAMES      = (FRONT-PLENUM)
   ZONE-NAMES        = (FRONT-OFFICE, FRONT-PLENUM)

   HEAT-ELEC-METER = M2
   COOL-ELEC-METER = M3
   HTREJ-ELEC-METER= M4
   AUX-ELEC-METER  = M4
   VENT-ELEC-METER = M5 ..
        $  THE VARIOUS HVAC SYSTEM COMPONENTS ARE PLACED ON SEPARATE METERS,
        $  TO TRACK HOW HVAC kW AND kWh ARE DISTRIBUTED.


REAR-SYSTEM  = SYSTEM          LIKE FRONT-SYSTEM  EXCEPT

   SYSTEM-EQUIPMENT= PKG-EQUIP-REAR
   PLENUM-NAMES      = (REAR-PLENUM)
   ZONE-NAMES        = (REAR-OFFICE, REAR-PLENUM) ..


                 $-----PLANT ASSIGNMENTS-----$

TOTAL-OFFICE = PLANT-ASSIGNMENT

   SYSTEM-NAMES    = (FRONT-SYSTEM, REAR-SYSTEM)
   DHW-GAL/MIN     = 0.1
   DHW-SIZE        = 30
        $  DEFAULT DHW TANK SIZE FOR 0.1 GAL/MIN USAGE IS 30 GAL
   DHW-SUPPLY-T    = 140
        $  DEFAULT SUPPLY T.  ALSO, HIGH T PROTECTS AGAINST LEGIONELLA.
   DHW-TYPE        = ELECTRIC
   DHW-SCH         = OCCUPY-1
   DHW-ELEC-METER = M1 ..

$  THIS ASSUMES THAT THERE IS A DEDICATED DOMESTIC HOT WATER (DHW) HEATER, AND
$  THAT MAXIMUM DHW CONSUMPTION IS 2 GAL/DAY PER PERSON DURING OCCUPIED HOURS.
```

```
                $-----SYSTEM REPORTS-----$

SYSTEMS-REPORT      VERIFICATION=(SV-A)
                    SUMMARY=(SS-A,SS-F,SS-N) ..


END ..
```

COMPUTE SYSTEMS ..

INPUT PLANT ..

```
TOTAL-OFFICE = PLANT-ASSIGNMENT ..

                $-----DEFINE PLANT EQUIPMENT-----$

HT-WTR-HTR = PLANT-EQUIPMENT

   TYPE = ELEC-DHW-HEATER
   SIZE = -999
   INSTALLED-NUMBER = 1 ..


PLANT-PARAMETERS

   ELEC-DHW-LOSS = 0.03 ..

                $-----PLANT REPORTS-----$

PLANT-REPORT          SUMMARY = (PS-B,PS-E,BEPS) ..

END ..
```

COMPUTE PLANT ..

INPUT ECONOMICS ..

```
                $-----COST OF ELECTRICITY-----$

$  THE FOLLOWING CHARGES REPRESENT THE FLORIDA POWER & LIGHT GSD-1 RATE SCHEDULE.

ELEC-COST = UTILITY-RATE

   RESOURCE         = ELECTRICITY
   ENERGY-CHG       = 0.0473
   BLOCK-CHARGES    = (DEMAND-MIN) ..

   DEMAND-MIN       = BLOCK-CHARGE
      BLOCK1-TYPE   = DEMAND
      BLOCK1-DATA   = (10,  0,
                         1, 9.96) ..

$  THIS RATE INCLUDES A CONSTANT ENERGY CHARGE OF 4.73 CENTS PER kWh; THERE IS
$  NO DEMAND CHARGE FOR THE FIRST 10 kW, THEN A CHARGE OF 9.96 DOLLARS/kW FOR
$  EACH ADDITIONAL kW.
```

```
                $-----ECONOMICS REPORTS-----$

ECONOMICS-REPORT            SUMMARY = (ES-E)  ..

END ..
COMPUTE ECONOMICS ..

STOP ..
```

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APPENDIX B

Rationale for Values Selected for Variables in DOE-2.1E Input File: 4,000 ft^2 Office in Strip Mall

Variable	Value	Rationale for Selection
LOADS Variables		
General Building Description	Baseline: One-story office in slab-on-grade strip mall. Office dimensions are 40 ft frontage by 100 ft depth; subdivided into two 40 x 50-ft offices. Side walls are interior walls adjoining conditioned space occupied by other tenants; the front and rear walls are exterior. Conditioned space has 9-ft ceilings, with 4-ft unconditioned space overhead which can serve as a return air plenum if desired. Parametric variations: None	Represents one typical type of small office (e.g., insurance or travel agencies). The length of frontage and the depth are typical of strip mall space. Subdivision of total floor area into two spaces is representative, and allows separate cooling of the north and south zones. The 9-ft ceilings are common for office space, although higher ceilings are also commonly encountered in strip malls. A plenum height of 4 ft is reasonable, although shorter (3-ft) plenums are commonly used by modelers, and plenums taller than 4 ft are commonly encountered.
BUILDING-LOCATION	Baseline: Miami, Florida Parametric variations: None	Desire to test hot, humid climate.
Building AZIMUTH (AZIMUTH = 0 for front wall)	Baseline: 0 (building faces north) Alternative values: 90, 180, 270 (building faces east, south, and west)	Arbitrary assumption.
BUILDING-SHADE	Baseline: 10-ft exterior overhang on front, at ceiling height. No overhang on rear. 2-ft parapet extending above front of roof (but not rear). Low value: 6-ft awning added over windows and door on rear, at ceiling height.	Typical of strip malls containing offices.
	High value: No overhang on either front or rear.	Models extreme case.

(continued)

APPENDIX B (continued)

Variable	Value	Rationale for Selection
AREA/PERSON	Baseline: 150 ft²/person	Approximately the maximum occupancy for offices (seven persons per 1000 ft²) recommended for office space in ASHRAE 62-1989 (ASHRAE, 1989a). The ASHRAE 62-1989 value for malls (20 persons/1000 ft², or 50 ft²/person) is not used since this space is deemed to represent office space rather than typical mall retail space.
	Low value: 300 ft²/person	50% decrease in number of occupants, compared to ASHRAE 62-1989 recommendation.
	High value: 100 ft²/person	50% increase in number of occupants.
PEOPLE-HEAT-GAIN	Baseline: 450 Btu/h/person	Adjusted sensible plus latent heat gain per person indicated in ASHRAE Fundamentals (ASHRAE, 1993) for moderately active office work.
	Parametric variations: None	
Occupancy Pattern (PEOPLE-SCHEDULE)	Baseline: Full occupancy at 8-11am and 2-5pm; 80% at 11am-noon and 1-2pm; 40% at noon-1pm; 30% at 6-7am and 5-6pm; 10% at 6-8pm; zero at other times.	A reasonable occupancy schedule for a small strip-mall office. The schedule used in prior modeling by FSEC (Shirey and Rengarajan, 1996).
	Parametric variations: None	
LIGHTING-TYPE	Baseline: Recessed fluorescent non-vented.	Representative of modern offices.
	Parametric variations: None	
LIGHTING-W/SQFT	Baseline: 1.8 W/ft²	The prescriptive lighting power allowance for office buildings of this size in ASHRAE 90.1-1989 (ASHRAE, 1989b).
	Low value: 1.0 W/ft²	Representing efficient lighting in modern offices, e.g., utilizing daylighting (Todesco, 1996; Birdsall, 1995).
	High value: 2.25 W/ft²	A 25% increase in the prescriptive allowance from ASHRAE 90.1-1989. Representing less efficient modern lighting (or better-lit space).
EQUIPMENT-W/SQFT	Baseline: 0.75 W/ft²	The average receptacle power density for offices specified in ASHRAE 90.1-1989.
	Low value: 0.5 W/ft²	More limited usage of computers, copiers, etc., relative to the average receptacle power density in ASHRAE 90.1-1989.
	High value: 1.75 W/ft²	Heavier usage of computers, copiers, etc.

(continued)

Variable	Value	Rationale for Selection
LIGHTING- and EQUIP-SCHEDULE	Baseline value: Full power at 7am-5pm; 30% at 6-7am and 5-6pm; 10% at 6-8pm; 5% at all other times.	A reasonable schedule for a small office. The schedule used in prior modeling by FSEC (Shirey and Rengarajan, 1996).
	Parametric variations: None	
AIR-CHANGES/HR	Baseline: 0.1 ACH	Representing average air infiltration rate for offices which are reasonably tight and/or which operate with much of the interior near neutral pressure, or slightly pressurized.
	Low value: 0 ACH	Representing offices which are in fact pressurized at all locations.
	High value: 0.3 ACH	One of the higher infiltration rates assumed by DOE-2 modelers for modern offices.
EXTERIOR WALL Construction (front and rear walls)	Baseline: 8-in. heavy weight hollow concrete block wall; 1-in. stucco layer on exterior, 3/4-in. expanded polystyrene insulation and 5/8-in. gypsum board layers on interior. U-value (*including* h_i and h_o)=0.16 Btu/h ft^2 F$^\circ$ (h_o averages 2.5 Btu/h ft^2 F$^\circ$ at Miami weather conditions).	Construction used by Florida Solar Energy Center (FSEC) in modeling a similar office (Shirey and Rengarajan, 1996). Representative of a construction encountered in Florida (Odom and DuBose, 1994). The U-value of the wall is lower than the maximum (1 Btu/h ft^2 F$^\circ$) determined using either the prescriptive or the performance criteria for Miami in ASHRAE 90.1-1989 (ASHRAE, 1989b).
	Intermediate value: Frame wall with 3.5-in. R-11 batt insulation. U-value (incl. h_i and h_o)=0.06 Btu/h ft^2 F$^\circ$.	Representative of a construction encountered in Florida (Odom and DuBose, 1994).
	Low value: No heat transfer across exterior wall; infinite wall resistance.	Models the extreme case.
	High value: Concrete block wall, like baseline, except delete 3/4-in. insulation. U-value (incl. h_i and h_o) = 0.34 Btu/h ft^2 F$^\circ$.	Models extreme case.

(continued)

Variable	Value	Rationale for Selection
WINDOW Area (incl. frame)	Baseline: Windows comprise 33% of gross exterior wall area. - 46% of front wall area. - 20% of rear wall area.	 Common for front walls in strip malls. Occasionally observed in rear walls in strip malls, depending on use of space adjoining rear wall. (More commonly, rear walls have no window area.)
	Low value: No windows on either front or rear wall.	Models extreme case.
	High value: Windows comprise 46% of gross exterior wall area.	Assumes rear wall has same amount of fenestration as front wall (extreme case). Towards upper end of range considered by other modelers (Eto, 1990).
Window Frame Type	Baseline: Thermally broken aluminum, 1.25 in. wide; glass set back 4 in. from exterior face of wall. Parametric variations: None	Representative of office space in strip malls.
GLASS-TYPE	Baseline: One pane, shading coefficient = 0.55 (PANES = 1, SHADING-COEF = 0.55). Overall U for fenestration (incl. frame) = 0.94 Btu/h ft^2 F° at Miami weather conditions (average h_o = 2.5 Btu/h ft^2 F°).	SHADING-COEF selected based on ASHRAE 90.1-1989 for Miami climate for: front wall (48% glazing, significant exterior shading); and rear wall (20% glazing, no exterior shading). This SHADING-COEF can be achieved with single-pane glass through various combinations of glass tinting, glass reflective coatings, and interior shading (ASHRAE, 1993; ASHRAE, 1992b).
	Low value: PANES = 2, SHADING-COEF = 0.16. Fenestration U = 0.32 Btu/h ft^2 F° (Miami weather).	Double-pane glass with tinting and highly-reflective coating (and/or interior shading), representing glass having good resistance to conduction and superior resistance to solar transmission (Todesco, 1996; ASHRAE, 1993; ASHRAE, 1992b).
	High value: PANES = 1, SHADING-COEF = 0.94. U = 0.94 Btu/h ft^2 F°.	A typical single-pane clear glass, 1/4-in. thick, representing glass having minimum resistance.
DOOR Constructions	Baseline: Front door is glass with 2-in. aluminum frame (treated as a window, with glazing as above). Rear door is brown steel door with fiber core (U-VALUE = 0.59 Btu/h ft^2 F°, ABSORPTANCE = 0.84). Parametric variations: None	Reasonably typical.

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Variable	Value	Rationale for Selection
ROOF Construction	Baseline: Built-up roofing over 4-in., R-12 mineral board and metal deck. U-Overall U-value of roof (incl. h_i and h_o) = 0.066 Btu/h ft^2 F°.	The roof construction used by FSEC in modeling a similar office (Shirey and Rengarajan, 1996).
	Low value: No heat transfer across roof; infinite roof resistance.	Models extreme case.
	High value: Replace R-12 mineral board in baseline with 2-in., R-6.9 board. Overall U = 0.12 Btu/h ft^2 F°.	Approximately doubles rate of heat transfer through roof compared to baseline.
INTERIOR-WALL Constructions	Baseline: Walls adjoining neighbors and subdividing office are 5/8-in. gypsum board on each side of 2- by 4-in. studs. Ceiling is acoustic tile. Parametric variations: None	Typical.
UNDERGROUND-FLOOR Construction	Baseline: 4-in. thick floor slab with carpet. Overall U-EFFECTIVE (incl. h_i) = 0.035 Btu/h ft^2 F°. Parametric variations: None	Typical construction. The effective U-value was estimated using the DOE-2 convention for avoiding over-estimation of slab heat transfer (multiplying the actual calculated U-value by the ratio slab perimeter:slab area).
SYSTEMS Variables		
Cooling SCHEDULE	Baseline: Cooling set point in conditioned space is 75°F during occupied hours (6am-7pm weekdays); off at all other times.	Set point is a typical value, and is consistent with ASHRAE 55-1992 (ASHRAE, 1992a). Turning cooling off after hours, and on weekends and holidays, is a common assumption used by modelers, even in hot, humid climates (Shirey and Rengarajan, 1996). In this case, office temperatures reach a high of 85°F on summer Sunday afternoons.
	Low value: Cooling set point 75°F during occupied hours; set up to 81°F at all other times during cooling season. High value: None	Assess impact of night set-up in hot weather to protect computers from high temperatures.
Heating SCHEDULE	Baseline: Heating set point 70°F during occupied hours; set back to 55°F at all other times (heating season). Parametric variations: None	Typical values. Set point during occupied hours is consistent with ASHRAE 55-1992.

(continued)

Variable	Value	Rationale for Selection
Design Temperatures for Zones	Baseline: Conditioned space: DESIGN-COOL-T = 74°F, DESIGN-HEAT-T = 71°F.	These temperatures, which control the sizing of the cooling and heating coils by DOE-2, are reasonable selections for the conditioned space, in view of the setpoints selected above under the cooling and heating SCHEDULES.
	Plenum: DESIGN-COOL-T = 90°F, DESIGN-HEAT-T = 50°F.	Values sometimes selected by other modelers. Since the plenum will in fact rarely get as hot as 90°F, even when the system is off over weekends, selection of this plenum DESIGN-COOL-T will result in moderate oversizing of the cooling coils, since the conditioned space will be estimated by LOADS as receiving artificially high heat gain from the plenum. At 90°F, this oversizing will be an amount which will limit the hours undercooled and at RH $\geq$ 60% to about 1%, considered to be a reasonable percentage for these computations. [The selection of DESIGN-HEAT-T is of little importance, since, in Miami, heating requirements are so small that the cooling load predominates in equipment sizing.]
	Parametric variations: None	
HVAC SYSTEM-TYPE	Baseline: Two rooftop direct expansion, constant-volume, packaged single-zone (PSZ) units; one unit dedicated to each of the two 2,000 ft^2 zones in the office.	Typical HVAC systems for offices of this size. Allows reasonably effective control of temperatures on the sunny vs. shaded sides of the office. Most computations made with this HVAC configuration.
	Alternative #1: A single PSZ unit serving the entire 4,000 ft^2 space.	Also representative. Provides less effective temperature control in the space, now treated as one large zone. PSZ units of this capacity commonly available.
	Alternative #2: A single PSZ unit serving the entire space, with one of the two sub-offices treated as a VAV subzone. South office is the control zone.	Included to illustrate the effect of this approach.
	Alternative #3: Same as Alternative #2, except that north office is the control.	
	Alternative #4: A single two-zone packaged VAV system (PVAVS).	Also a reasonable choice; might be somewhat less common than PSZ for this application.
	Alternative #5: Two packaged terminal air conditioner (PTAC) units, one in each 2,000 ft^2 office.	Generally less representative of new construction.

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Variable	Value	Rationale for Selection
RETURN-AIR-PATH	Baseline: PLENUM-ZONES (all systems except PTAC; DIRECT for PTAC). Alternative value: DUCT	HVAC system air return via the overhead plenum is a common choice. Through-the-wall units such as PTAC necessarily draw the return air directly from the room. Ducted returns are also a reasonable choice.
Supply Temperatures for System Air	Baseline: MIN-SUPPLY-T = 55°F; MAX-SUPPLY-T = 100°F (for PVAVS, REHEAT-DELTA-T= 45 F°).	Supply temperature for cooling air (55°F) is the commonly selected value, being above the typical dew point and thus avoiding condensation on the supply ducts. Supply temperature for heating air (100°F) is lower than other modelers commonly use (105-120°F), in view of Miami's limited heating needs; MAX-SUPPLY-T has minimal impact on these computations.
	Low value: MIN-SUPPLY-T = 42°F.	Models case of cold-air diffusers.
	High value: None	
Economizer	Baseline: Economizer with lockout; ECONO-LIMIT-T = 68°F. (Exception: PTAC units have no economizer.)	Economizers are increasingly common in packaged HVAC units. Economizer lockout (preventing economizer from functioning when cooling system must also operate) is common in these small systems, to prevent the hermetic compressor motors from overheating at times when economizer operation would create refrigerant flows too low to adequately cool the motor. PTAC units are not configured to accommodate an economizer. 68°F is one typical value for the outdoor air temperature at which the economizer returns to minimum outdoor air operation.
	Alternative value: No economizer.	
COOLING-EIR	Baseline: 0.341 (EER = 10). (Exception: PTAC units have EIR = 0.438, or EER = 7.8.)	An electric input ratio of 0.341 Btu/h electric input per Btu/h of cooling output (corresponding to an energy efficiency ratio of 10 Btu/h of cooling output per W of electric input), is representative of reasonably efficient modern PSZ units. (The DOE-2 default value for the COOLING-EIR of PSZ units is 0.360.) The EIR for the less efficient PTAC units is the DOE-2 default value.
	Low value: 0.284 (EER = 12).	Represents about the most efficient commercial units.
	High value: 0.427 (EER = 8).	Represents about the least efficient commercial units.
Heating	Baseline: Electric resistance (HEATING-EIR = 1). Central heating coils for all systems except PVAVS; terminal reheat for PVAVS.	One reasonable selection.
	Parametric variations: None	Air-source heat pumps (HEATING-EIR < 1) not considered in view of low heating load in Miami.

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APPENDIX B (continued)

Variable	Value	Rationale for Selection
COOLING-CAPACITY	Baseline: 103.6 kBtu/h at 5 cfm/person, 119.2 kBtu/h at 20 cfm/person.	The default values computed by the DOE-2 program. Slightly more than half of these capacities resides in the unit conditioning the rear zone.
	Low value: 102 kBtu/h (5 cfm/person)	The commercially available capacity (corresponding to 8.5 tons of refrigeration) closest to the default capacity computed by the program at 5 cfm/person.
	High values: 108, 114, and 120 kBtu/h (5 cfm/person); 120, 126, and 132 kBtu/h (20 cfm/person).	Commercially available capacities providing some excess capacity above the computed default values.
Sensible heat ratio (SHR)	Baseline: 0.75 at 5 cfm/person, 0.70 at 20 cfm/person	The default values computed by the program.
	Alternative #1: SHR = 0.78	A value typical of commercial units containing three rows of cooling coils.
	Alternative #2: SHR = 0.73	A value typical of commercial units containing four rows of cooling coils.

PLANT Variables

Variable	Value	Rationale for Selection
Domestic Hot Water (DHW)	Baseline: Electric resistance 30-gal hot water heater, delivering up to 0.1 gal/min (depending on occupancy) at 140°F.	DHW delivery rate based on maximum rate of 2.0 gal/person/day estimated for offices in the ASHRAE Applications Handbook (ASHRAE, 1995), based on an 8-hour work day. Storage tank size is DOE-2 default minimum value. Supply temperature, higher than the 105-115°F recommended by ASHRAE for office lavatories, is to protect against growth of *Legionella* (ASHRAE, 1995). [Note: For simplicity, no lavatory is included in the modeled office; a lavatory would be modeled as part of one of the 2,000 ft^2 sub-offices, and would have minimal impact on the calculations. To enable these calculations, it is assumed that the exhaust rate from such a lavatory would never be greater than the rate at which outdoor air is supplied by the HVAC system, even at 5 cfm/person (67 cfm per sub-office), even though 67 cfm could be insufficient bathroom exhaust by some codes.]
	Parametric variations: None	

APPENDIX B (concluded)

Variable	Value	Rationale for Selection
ECONOMICS Variables		
Cost of Electricity	Baseline: Energy charge of 4.73 ¢/kWh; demand charge of zero up to 10 kW usage, then $9.96/kW for each kW above 10. Parametric variations: None	The GSD-1 rate structure for Florida Power and Light Co., which serves the Miami area (Shirey and Rengarajan, 1996).
Weather File Variables		
Weather File	Baseline: Typical Meteorological Year (TMY) file for Miami. Alternative: Weather Year for Energy Calculations (WYEC) file for Miami.	