CAN ASHRAE STANDARD 62-1989 REQUIREMENTS BE SATISFIED WHILE MAINTAINING MOISTURE CONTROL USING STOCK HVAC EQUIPMENT IN HOT, HUMID CLIMATES?

STEPHEN C. TURNER, PE, CIAQP SENIOR ENGINEER CHELSEA GROUP LTD. DELRAY BEACH, FLORIDA

ABSTRACT

Outdoor air intake rates are studied to determine their impacts on moisture control in buildings, especially in hot, humid climates. Key impacts of outdoor air intake rates can be readily modeled and studied using computer simulations of building energy costs. Increased ventilation rates create real capital and operating costs for building owners and operators, with implications beyond energy costs relating to increased ventilation requirements. In hot, humid climates, increased ventilation rates increase latent loads more than sensible loads, requiring lower sensible heat ratios. Stock HVAC package units and split systems are not available with the requisite sensible heat ratios, and cannot maintain moisture control in small commercial buildings without costly modifications.

PURPOSE OF STUDY

Outdoor air intake rates are studied to determine their impacts on moisture control in buildings, especially in hot, humid climates. Key impacts of outdoor air intake rates can be readily modeled and studied using computer simulations of building energy costs. These studies permit an objective analysis of the costs and benefits of such an approach. Through computer simulation studies, impacts of various scenarios for managing outdoor air intake rates in hot humid climates are quantified.

PRESCRIPTIVE COMPLIANCE WITH CODES BASED ON ASHRAE 62-1989

The provisions of Standard 62-1989 mandate that many buildings operate with a high rate of outside air intake for their heating, ventilation, and air-conditioning (HVAC) systems. This provision is stated as follows:

> "**6.1.3 Ventilation Requirements.** Indoor air quality shall be considered acceptable if the required rates of acceptable outdoor air in Table 2 are provided for the occupied space."

Since most local mechanical codes have substantially adopted ASHRAE Standard 62-1989, this wording requires new or substantially renovated buildings to comply with the outside air intake rate provisions of this standard. Providing ventilation air flow rates with outside air, as prescribed in Table 2 of ASHRAE Standard 62-1989, is virtually the only method in actual use for implementing this standard.[1]

The prescriptive approach centered on Table Two of ASHRAE Standard 62-1989 "Ventilation for Acceptable Indoor Air Quality" has many impacts on building owners, operators, and tenants in hot, humid climates. The outside air intake flow rates are shown to have substantial and quantifiable energy penalties. Furthermore, the capital costs associated with providing the capacity or equipment to handle increased mechanical heating, ventilation,

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[1] Godish, T. 1994. Sick Buildings: Definition, Diagnosis, and Mitigation. (Boca Raton, Florida: CRC Press, 1994) p. 377

and air-conditioning system loads cannot be ignored. The most important consequence of this prescriptive approach for practitioners in hot, humid climates is this: increased ventilation rates in hot, humid climates may cause loss of moisture control. *Uninterrupted* elevated equilibrium relative humidity may result. Equilibrium relative humidity levels constantly above 65% have been associated with heightened potential for microbiological growth.[2]

STUDY METHODOLOGY

Any study of the impacts of outdoor air intake rates must consider a variety of building types, regional climates, and occupancies. The need to represent these variations must be balanced with the need to present an understandable study based on acceptable assumptions.

DOE 2.1E BUILDING ENERGY ANALYSIS PROGRAM

There are a variety of building simulation software packages available to model building performance. Department of Energy, version 2.1E (DOE 2.1E) software was selected as a widely used, validated, and accepted package. The DOE 2.1E building energy analysis program was designed by the Simulation Research Group at Lawrence Berkeley Laboratory at the University of California, Berkeley, with the support of the United States Department of Energy. The program was developed to assist engineers and architects in design studies of whole-building energy use under actual weather conditions. It facilitates the study of energy effects which result from changing various parameters with respect to location, weather, and building mechanical system design, construction, and operation. It is reasonably accurate in predicting energy costs, allowing the user to input actual weather files, geographical, and utility rate information for any region of the country. It is extremely accurate in producing costs for comparison from parametric runs in which one or more building system factors are varied. Such parametric runs permit accurate study of the key impacts of indoor air quality standards.

Other simulation programs are available which incorporate routines designed to account for two mechanisms for moisture transport of particular concern in hot, humid climates. DOE-2..1E was selected due to wider validation studies and greater acceptance nation-wide. These two mechanisms are re-evaporation of condensate from evaporator coils and drain pans when fans continue to operate after cooling coils are cycled off upon satisfaction of thermal load, and moisture absorption and desorption in building materials, fixtures, and furnishings.[3] Because of these effects, DOE-2.1E may overestimate dehumidification capabilities, making any conclusions drawn about the inadequacy of stock equipment to handle latent loads even more conservative.

BUILDING TYPES

In order to evaluate the effect of indoor air quality regulation on commercial buildings, variations on building types must be represented. However, to avoid creating

[2] Morey, Philip R. "Suggested Guidance on Prevention of Microbial Contamination for the Next Revision of ASHRAE Standard 62," IAQ '94:Engineering Indoor Environments. (Atlanta, Georgia: ASHRAE, 1995) p.141

[3] Rengarajan, K., D.B. Shirey, & R.A. Raustad. Cost-Effective HVAC Technologies to Meet ASHRAE Standard 62-1989 in Hot and Humid Climates. (Atlanta, Georgia: ASHRAE Preprint, 1996) p. 2

model buildings unfamiliar to experienced DOE 2.1E users, three buildings were chosen from the sample set provided by Lawrence Berkeley Laboratories with the DOE 2 software. These buildings, though they are not based on built structures, are extremely detailed and realistic. They are described in detail in *DOE-2 Sample Run Book Version 2.1E*, by F.C. Winkelmann et. al., November 1993. Samples number 3, 7, and 8 were selected:

- 31 story office building
- 2 story office building with atrium
- 1 story small bar/lounge

Because the systems modeled with these sample buildings were meant to demonstrate the sophisticated capabilities of the software, they do not reflect actual mechanical systems. These systems included features atypical of current building stock, such as desiccant wheel dehumidification, thermal energy storage, and gas engine drive chillers. To accurately reflect real buildings, standard systems provided with the software were used instead of the more complex systems originally modeled with the buildings.[4] These systems more closely reflect what is actually installed in many buildings.

CLIMATE ZONES

To model the effects of regulation on the United States building stock, these buildings were modeled in five cities, one in each climate zone represented in the *Commercial Buildings Energy Characteristics 1992*, Energy Information Administration, April 1994, page 317. Weather data as required by DOE 2.1E software is only available for certain cities (*DOE-2 Reference Manual Part 2 Version 2.1*, US DOE, May 1980). Of these cities, ten have been studied by Eto and Meyer in 1989[5] in a study of energy costs associated with increased ventilation air. Representative cities for each climate zone were selected from Eto and Meyer's list.

Particular attention is paid to the results from the two story office building and the bar/lounge in Dallas, to compare the requirements of total loads with the characteristics of available stock unitary HVAC equipment.

UTILITY RATES

Utility rates used were taken directly from the tariff sheets in effect in August 1995, as obtained from the utilities serving the selected cities. Taxes were not included, but demand, energy, and fuel adjustment charges were applied as appropriate to each building.

RESULTS

Buildings operating with various control configurations and different outside air intake rates are compared, as detailed below. These results are expressed in terms of total energy dollars per gross square foot, and HVAC dollars per gross square foot.

ENERGY COST IMPACT OF OUTDOOR AIR INTAKE RATES

To understand the effects of the outside air intake on existing buildings, a baseline case representing the existing mechanical systems in buildings was established. To establish this baseline case, each sample office building

[4] Birdsall, B.E., W.F. Buhl, K.L. Ellington, A.E. Erdem, F.C. Winkelmann, J.J. Hirsch, & S. Gates. DOE-2 Basics, Version 2.1E. Berkeley, California: University of California. May 1994.

[5] Eto, J.H. &C. Meyer. "The HVAC Costs of Increased Fresh Air Ventilation Rates in Office Buildings." ASHRAE Journal September 1988: 30-39

was studied operating with outside air being provided as ventilation air at the rate of 10 cfm per person. Both ASHRAE 62-1973 and 62-1981 required 5 cfm for office buildings. The bar/lounge was studied operating with outside air being provided as ventilation air at the rate of 20 cfm per person, left unchanged from the sample case furnished by Lawrence Berkeley Laboratories.

Buildings were modeled with control systems incorporated into the DOE 2.1E samples by Lawrence Berkeley Laboratories. For both office buildings, this includes night setback and other energy efficient strategies. For the bar/lounge, there are no such strategies incorporated.

To represent these same buildings after being brought into compliance with ASHRAE Standard 62-1989, each sample office building was then studied operating with outside air being provided as ventilation air at the rate of 20 cfm per person. The bar/lounge was studied operating with outside air being provided as ventilation air at the rate of 30 cfm per person. The difference between the two cases demonstrates the increase in costs of energy required to heat and cool the higher outside air intake rate.

The costs for increased outside air are demonstrated to be substantial. The average increase in HVAC energy costs across all building types and locations studied is 11%. Offsetting these increased energy costs with heat exchange or demand control equipment requires substantial capital investment. In many buildings, existing equipment capacities no longer meet the load imposed by increased outside air intake rates. These buildings must then invest in supplementary equipment, or replace existing equipment with larger equipment. There is no way to escape the cost penalty attached to outside air intake rate increase. Operating costs increase, and capital costs are also incurred.

The results for Dallas reflect the increase in internal and other loads relative to ventilation loads as building size increases. Ironically, larger buildings equipped with Air-Water systems and built-up air handlers are more easily designed or adapted to lower Sensible Heat Ratios associated with higher latent loads, but are less likely to require such adaptation. Significant increases in latent loads associated with higher ventilation rates are more likely to occur in smaller buildings where internal and other sensible loads are less dominant, and where stock HVAC equipment is more likely to be used. It is imperative to consider these changing load profiles when seeking to improve IAQ in smaller buildings.

HOT AND HUMID CLIMATE CONSIDERATIONS

LOADS DETAIL

Units manufactured for applications across a wide geographic range must meet many criteria. A widely used criteria in the marketplace for many years has been efficiency at standard ARI conditions of 95°ambient db, 80°db & 67°wb to evaporator. Particularly in hot and humid climates, sensible heat ratio under a wide range of load conditions is also important in proper equipment selection.

The traditional design and selection process can exacerbate problems relating to inadequate sensible heat ratios. If a designer performs room by room heat load calculations, figuring sensible and latent for each room, and then sums these loads separately for the entire zone or building, then sensible, latent, and grand total loads result which are often used for equipment selection. In many designs, latent load dominates. If equipment is selected based on the smallest unit which will satisfy the latent load, the sensible and total capacities will be excessive. Such equipment is almost always controlled by a conventional thermostat which responds only to dry bulb temperature – responding only to the sensible load.

The resulting oversized system, at all but peak conditions, pulls down very quickly to design dry bulb temperature, cycling off while leaving humidity unsatisfactorily high. Since return air dry-bulb temperatures are rarely as high as the ARI standard rating point, the difference between actual coil temperatures and dewpoint of air traveling over these coils is reduced, resulting in actual sensible heat ratios under partial load conditions that are often lower than those derived from performance at the standard ARI rating point. Conventional systems are not equipped with adequate capacity stepping or modulation to keep compressors on-line during part-load conditions. Since cooling coil temperatures are not maintained below dewpoint, intermittent and inadequate dehumidification takes place. Using ASHRAE 1% design conditions results in equipment selection for loads which occur less than 90 hours each year.

ASHRAE's Standing Standards Project Committee has proposed revisions for Standard 62 which require that ventilation air be delivered continuously to occupied spaces. Under this provision, fan switches on thermostat subbases would be required to operate only in the "On" position, with the "Auto" cycling mode disabled with jumper wires. The resulting continuous operation of the evaporator fan even when the compressor has cycled off causes subsequent re-evaporation of condensate. In hot and humid climates, the amount of condensate re-entrained has been shown to exceed 15%.[6]

Many other factors can contribute to moisture control problems with unitary equipment. With drier coils, more interstitial area between fins opens up. Bypass factors increase, resulting in even lower dehumidification performance. In some package units, poor mixing of outdoor air with return air may result in uneven coil loading and decreased dehumidification of outdoor air. Failure to deliver outdoor air directly to air handlers often results in loss of moisture control.

Of course, uncontrolled airflows associated with inappropriate pressure relationships, leaky envelopes, or concentrated internal moisture sources, will also contribute to moisture control problems associated with inadequate dehumidification.

DALLAS RESULTS

By requesting and retaining detail from the loads reports generated by DOE-2.1E, important insight can be gained into the way that load profiles change when ventilation rates are increased.

BAR/LOUNGE

For the bar/lounge modeled in Dallas, tables 4 and 5 show a 6.4% decrease in sensible heat ratio when ventilation rates increase. Cooling load increases by 19.1 % or two tons.

The original sensible heat ratio value, when compared with Table 6, shows that stock

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[6] Khattar, M.K., M.V. Swami, and N. Ramanan. "Another Aspect of Duty Cycling: Effects on Indoor Humidity," ASHRAE Transactions 93(1):1678-1687

equipment is already deficient in this application. The increased gap between required and available sensible heat ratio will cause higher humidity levels. Increased chronic equilibrium relative humidity will cause greater moisture activity in building materials, fixtures, and furnishings. Especially in hydrophilic materials such as the paper surfaces of wallboard, this increase in moisture activity is likely to reach the range which has been shown to support microbiological activity.

TWO-STORY OFFICE BUILDING

For the two-story office building modeled in Dallas, tables 7 and 8 show a 4.1 % decrease in sensible heat ratio when ventilation rates increase.

As internal and sensible envelope gains outweigh the ventilation loads, the latent load increase due to increased ventilation has less effect upon sensible heat ratio. The resulting sensible heat ratio value is still within the reach of stock equipment. Cooling load increases by 7.1 % or five tons.Thirty-one story Office Building.

For the 31-story office building modeled in Dallas, tables 8 and 9 show a 3.3 % decrease in sensible heat ratio when ventilation rates increase. Cooling load increases by 3.5 % or 67 tons. Internal and sensible envelope loads far outweigh the ventilation loads, so the increased ventilation has even less effect upon sensible heat ratio and capacities than in the smaller buildings.

MODIFICATIONS TO ACCOMMODATE INCREASED VENTILATION RATES

Several after-market modifications can be made to improve dehumidification capabilities of HVAC systems. The cost to implement these modifications cannot be ignored. Nor can the cost of increased capacity associated with increased ventilation requirements.

Enthalpy wheels are effective where an exhaust stream is available adjacent to the outside air intake. To avoid contamination in the event of wheel or seal failure, care must be taken that the outdoor air intake airstream be forced draft and the regenerative airstream be induced draft. Adequate pre-filtration of regeneration airstream is also required. These approaches, which are recommended by manufacturers as part of due care in installation, require two additional dedicated blowers and a filter rack.

These measures are often overlooked in estimating the costs of this approach. High first costs, cleaning and maintenance requirements, and the energy costs associated with pressure drops are valid considerations when evaluating the applicability of enthalpy wheels.

Heat pipes are also effective, where excess sensible capacity is available to be traded off for increased latent capacity. High first costs, the addition of Freon-bearing components requiring monitoring for leaks, the need to maintain cleanliness for airflow and effective heat exchange, and the energy costs associated with higher pressure losses are all considerations in evaluating heat pipe applications.

Demand controlled ventilation has been applied to reduce ventilation air quantities in accordance with actual occupancies. The only sector-wide application of this approach has been in new public assembly buildings such as sports arenas, where energy costs associated with the ventilation rate procedure in ANSI/ASHRAE Standard 62-

1989 are so high that builders are willing to accept the costs of demand control ventilation. For the most part, designers are able to obtain fees adequate to perform the indoor air quality procedure, and have become willing to accept the inherent liabilities, although only in this limited market sector. In some studies, and even in some applications of demand control ventilation, Standard 62-1989 requirements to implement filtration, including gas-phase filtration, to address constituents of concern other than CO_2 have been ignored. Sensor drift for controllers, requirements for low velocity over gas phase filtration beds, and high pressure losses are also concerns with demand control ventilation applications.

Run-around coils can be effective in heating climates, but rarely develop adequate enthalpy transfer between air streams for hot and humid climate applications. They have been used to supplement other methods of transfer.

Dedicated 100% outside air units, though costly to install, have demonstrated exceptional ability to control latent loads while freeing the conventional HVAC system to handle fluctuating sensible loads. If part load dehumidification concerns are adequately addressed, this approach can provide excellent control, albeit at substantial first cost.

Refrigerant desuperheating coils are increasingly being used to reheat air streams downstream of evaporator coils. This approach has proven effective in factory engineered applications, but may be difficult to implement and control properly on after-market or retrofit applications. One way to facilitate this approach is to apply widely available water heating desuperheaters, with which there is wider field experience, and use pumped water coil reheat to improve sensible heat ratio. Of course, fan and pump energy considerations apply.

Proprietary hot-gas bypass controls are available for after-market or retrofit application. Concerns which must be met when applying these devices include achieving part load operation while maintaining dehumidification capability, ensuring suction gas flow adequate to cool compressor motor windings, and maintaining refrigerant oil return.

Electric reheat, applied in accordance with ASHRAE Standard 90.1, may have the lowest first cost of after-market or retrofit dehumidification measures, but carries with it substantial operating energy expense. It is currently illegal in most applications in Florida under that state's energy efficiency code.

The single, most cost-effective way to control humidity in smaller buildings in hot and humid climates is this: *manufacturers should improve stock units to have lower sensible heat ratios*. This approach would minimize first costs, maintenance requirements, liability, and energy costs. Significant improvements could be gained by optimizing manufactured systems for sensible heat ratio on an integrated part load value basis, rather than maximizing total capacity ratings at standard ARI conditions, energy efficiency ratio, and seasonal energy efficiency ratings. Currently, several manufacturers are offering costly units equipped with one or more of the above approaches. Conventional systems with equipment matches to render lower sensible heat ratios may offer an optimal combination of first and operating costs.

CONCLUSIONS

In larger buildings, built-up equipment can be specified to handle low sensible heat ratios. In such buildings, ventilation loads do not dominate total loads, so increased ventilation does not require these lower sensible heat ratios. In smaller buildings, ventilation loads assume a larger role in determining required equipment capacities and sensible heat ratios.

In the smallest buildings, designers are more likely to use stock packaged and split direct expansion equipment, which is not available in a wide enough range of sensible heat ratios to handle higher latent loads associated with buildings in hot and humid climates. Chronic loss of moisture control may lead to unusual microbiological activity. In such situations, increased ventilation rates without costly humidity control measures may detract from indoor air quality.

Increased ventilation rates create real capital and operating costs for building owners and operators, with implications beyond energy costs relating to increased ventilation requirements. In hot, humid climates, increased ventilation rates increase latent loads more than sensible loads, requiring lower sensible heat ratios. Stock HVAC package units and split systems are not available with the requisite sensible heat ratios, and cannot maintain moisture control in small commercial buildings without costly modifications.

Table 1. Modeled Buildings and Systems

Building	System
31 story office building	Standard chiller with air handlers
2 story office building with atrium	Rooftop package units (two)
1 story small bar/lounge	Packaged single zone air conditioner

Table 2. City Selection for Five Climate Zones

Region	Cooling Degree Days	Heating Degree Days	City
Climate Zone 1	under 2,000	over 7,000	Minneapolis
Climate Zone 2	under 2,000	5,500 to 7,000	Chicago
Climate Zone 3	under 2,000	4,000 to 5,499	Washington DC
Climate Zone 4	under 2,000	under 4,000	San Francisco
Climate Zone 5	2,000 or more	under 4,000	Dallas

Table 3. Increased Costs from Increased Outside Air Intake Rates

City	Building	Total $/sq.ft.		HVAC $/sq.ft.		Increase in HVAC costs
		before	after	before	after	
Minneapolis	Bar/Lounge	$2.32	$2.57	$1.60	$1.85	16%
Minneapolis	2-story Office	$0.97	$1.01	$0.50	$0.54	8%
Minneapolis	31-story Office	$0.83	$0.87	$0.40	$0.44	10%
Chicago	Bar/Lounge	$2.62	$2.94	$1.76	$2.05	16%
Chicago	2-story Office	$1.46	$1.50	$0.71	$0.75	6%
Chicago	31-story Office	$1.46	$1.49	$0.70	$0.73	4%
Washington DC	Bar/Lounge	$2.67	$2.90	$1.74	$1.97	13%
Washington DC	2-story Office	$1.65	$1.69	$0.78	$0.81	4%
Washington DC	31-story Office	$1.44	$1.48	$0.68	$0.72	6%
San Francisco	Bar/Lounge	$2.69	$2.93	$1.44	$1.69	17%
San Francisco	2-story Office	$1.30	$1.30	$0.50	$0.50	0%
San Francisco	31-story Office	$1.50	$1.51	$0.60	$0.61	2%
Dallas	Bar/Lounge	$1.74	$1.90	$1.18	$1.35	14%
Dallas	2-story Office	$1.14	$1.17	$0.62	$0.64	3%
Dallas	31-story Office	$0.97	$0.98	$0.51	$0.52	2%
Average		$ 1.65	$ 1.75	$ 0.91	$ 1.01	11%

Table 4. Bar/Lounge Cooling Load and Sensible Heat Ratio at 20 cfm per Person

Maximum Cooling Load	121.536 kBTU/hr	10 tons
SHR @ Maximum Cooling Load	0.623	

Table 5. Bar/Lounge Cooling Load and Sensible Heat Ratio at 30 cfm per Person

Maximum Cooling Load	144.763 kBTU/hr	12 tons
SHR @ Maximum Cooling Load	0.583	

Table 6. Ten Ton Stock Rooftop Package Units - Standard Ratings (4000 cfm, 95°ambient db, 80°db & 67°wb to evaporator)

	Cooling Capacity (kBTU/hr)	SHR
Manufacturer #1 High Efficiency	127.8	0.746
Manufacturer #1 Standard Efficiency	126.1	0.722
Manufacturer #2 High Efficiency	121	0.704
Manufacturer #2 Standard Efficiency	123	0.724
Manufacturer #3 High Efficiency	125.0	0.746
Manufacturer #3 Standard Efficiency	126.0	0.733

Table 7. 2. Story Office Building Cooling Load and Sensible Heat Ratio at 10 cfm per Person

Maximum Cooling Load	933.791 kBTU/hr	78 tons
SHR @ Maximum Cooling Load	0.807	

Table 8. 2. Story Office Building Cooling Load and Sensible Heat Ratio at 20 cfm per Person

Maximum Cooling Load	1000.277 kBTU/hr	83 tons
SHR @ Maximum Cooling Load	0.774	

Table 9. 31. Story Office Building Loads and Sensible Heat Ratio at 10 cfm per Person

Maximum Cooling Load	23,083.918 kBTU/hr	1924 tons
SHR @ Maximum Cooling Load	0.961	

Table 10. 31. Story Office Building Loads and Sensible Heat Ratio at 20 cfm per Person

Maximum Cooling Load	23,889.375 kBTU/hr	1991 tons
SHR @ Maximum Cooling Load	0.929	

Table 11. Changes in Cooling Loads Due to 10 cfm Per Person Increase in Outside Air Intake Rate

Building	Decrease in Sensible Heat Ratio	Increase in Total Cooling Load
Bar/lounge	6.4%	19.1%
2-Story Office Building	4.1%	7.1%
31-Story Office Building	3.3%	3.5%