Predictions from ceiling jet theory

The usual practice to avoid draught in cooling situations is to dimension the separation distance of the jet to be a fraction of the throw of the isothermal jet, see references (4) and (5). This amounts to saying that the maximum velocity in the occupied zone (u_{occz}) is proportional to the velocity of the jet (u_a) at the separation point (x_a). Thus assuming similarity equations to be valid, the following proportionality was found (3) for radial and three dimensional ceiling jets

$$u_{occz} \sim u_a \sim Kq_0^{1/2}u_0^{1/2}/x_a \sim \Delta t^{1/2}K^n(q_0/u_0)^{1/4} = \Delta t^{1/2}K^n a_0^{1/4} \quad (1)$$

where the effects of the temperature difference (Δt), air flow rate (q_0), supply air velocity (u_0), opening area (a_0) and velocity constant (K) are shown. At a constant supply temperature the equation predicts constant thermal comfort when the opening area is fixed. This important finding is supported by measurements of the radial case except at very low air flow rates and velocities, when the flow structure changes and also temperature has more effect on comfort. The effect of the supply air area is quite small: a 50 % reduction in the area will decrease the velocity by only 16 %. It is surprising that the mixing properties of the jet (small K or large effective origin) have negative effect in the radial case (n = -1/2) and no effect (n = 0) in two and three dimensional cases. A negative n-value is supported by the fact that the simple radial diffuser was better than the multinozzle radial diffuser in the experiments. The corresponding equation for the two dimensional jet (3) shows that only the air flow rate per width has importance: this is confirmed by the results of the slot diffuser in Fig. 6 and may be the reason for the air flow limit for other diffusers as well. The above reasoning is valid up to the point where the throw of the jet comes close to the room dimensions.

DISCUSSION

Air flow rates up to 8 l/s are possible at temperatures of 0 °C and -20 °C if the supply air velocity is higher than 2 - 3 m/s and the draught criterion is based on a 20 % level of dissatisfied occupants. This is an improvement compared with the products found on the market. Better comfort can be achieved by using the convective heating power which is greater than is needed solely for ventilation. The heater should be of the convective type. The properties and locations of the heater, outdoor air inlet and flow obstructions should be well planned. Useful predictions can be found using jet theory.

ACKNOWLEDGEMENTS

The study was supported by the Technology Development Centre (TEKES).

REFERENCES

1. Fanger PO, Melikow AK, Hanzawa H, Ring J. Air turbulence and sensation of draught. Energy and Buildings, 1988;12;21-39.
2. National Building Code of Finland. D2, Indoor climate and ventilation in buildings. Regulation and guidelines. Ministry of the Environment. 1987.
3. Heikkinen J, Kovanen K, Ojanen T, Pallari M-L, Piira K, Siitonen V. Principles of draughtless outdoor air inlets (in Finnish). VTT Research Notes (in print).
4. Skåret, E. Luftbevegelse i ventilerte rom. Inst. for VVS, NTH, Tapir, Trondheim, 1976.
5. Nielsen, PV. Models for the prediction of room air distribution. 12th AIVC Conference on Air Movement and Ventilation Control within Buildings, Ottawa, 1991.

INCREASING OUTDOOR AIR FLOW RATES IN EXISTING BUILDINGS

David H. Mudarri[1] and John D. Hall[2]

[1]Indoor Air Division, Environmental Protection Agency, Washington, DC, USA
[2]ICF Incorporated, Fairfax, VA, USA

ABSTRACT

Many existing buildings were designed for relatively low rates of outdoor air ventilation (2.5 l/s/occupant or 5 cfm/occupant). Nevertheless, building owners and managers face requests by occupants and consultants to raise outdoor ventilation rates in order to satisfy demands for improved indoor air quality. This paper examines the feasibility, and potential energy and IAQ impact of attempts to raise outdoor ventilation rates from 2.5 to 10 l/s/occupant (5 to 20 cfm/occupant) in existing buildings. A parametric analysis using DOE-2 simulations was used to generate both energy and indoor air quality data. Our analysis suggests that the energy impacts are minimal, but that capacity problems are sometimes encountered when ventilation rates are raised in existing buildings.

INTRODUCTION

ASHRAE's latest ventilation standard (Standard 62-1989) increases outdoor air ventilation rates from 2.5 to 10 liters per second per occupant (5 to 20 cubic feet per minute per occupant or cfm/occ) for office environments. Many owners and managers whose buildings were designed under the old standard, face requests to meet the new outdoor air standard. This constitutes a four-fold increase in the amount of outdoor ventilation air required. This recommendation is a contradiction to commonly accepted energy conservation practices focused on keeping the outdoors out, and it raises the question of whether or not such a change will overload the ventilation equipment.

Several perspectives of the relationship between energy use and outdoor air flow rates are available in the current literature [2,3,4,5,6,7]. Some reports suggest that increasing outdoor air ventilation rates causes only a marginal increase in energy use [2,3,4]. However, these studies do not evaluate the relative effects of various building designs, HVAC system types, or outdoor air flow control strategies, and they do not address the issues faced in raising outdoor ventilation rates in existing buildings.

In order to develop an improved understanding of the relationship between outdoor air ventilation rates, energy use, and IAQ in office buildings, a study was initiated at the Indoor Air Division of the Environmental Protection Agency. The two main purposes of this study are: (1) to determine the impact of increased outdoor air ventilation rates in commercial buildings; and (2) to determine the energy and IAQ performance of various ventilation system configurations, IAQ control measures (ICMs), and energy conservation measures (ECMs).

This paper presents preliminary findings from a portion of this study dealing with the

#6966

energy use impacts of changing outdoor air and VAV box damper settings to increase outdoor air flow rates. Data are presented for a few selected cases only. More complete results on IAQ control measures, energy conservation measures and IAQ performance will be presented in subsequent papers.

GENERAL APPROACH

Results have been prepared for five building configurations only, designated as buildings "A" through "E". These are all multi-story office buildings with central HVAC systems. Building A (the base building) has an occupancy of 13 square meters per person (143 square feet per person), a perimeter to core space ratio of 0.84, typical shell efficiencies (RSI-value = 1.8, infiltration = 0.5 ACH), and typical efficiencies for the boiler (Eff = 70%) and chiller (EER = 10.0). Variations around the base building (Buildings B through E) are shown in Table 1.

Table 1: Design parameters for selected base case buildings

Building	Primary design parameters	
A. Base case	(see text)	
B. Core-dominated	perimeter/core ratio:	0.33
C. High occupant density	occupant density:	6.2 m^2/occ
D. Efficient shell	wall and roof RSI-value:	3.52 $W/m^2/°C$
	infiltration:	0.5 ACH
E. Efficient HVAC equipment	boiler efficiency:	80%
	chiller efficiency:	12.0 EER

The data for this project were generated using the DOE-2.1d building energy computer simulation model [8] for a temperate climate (Washington, D.C.). Five different HVAC systems including constant volume (CV) and variable volume (VAV) supply systems with different outdoor air control strategies were modelled in each of the five buildings: (1) CV with constant outdoor air flow; (2) CV with air economizer; (3) VAV with fixed damper; (4) VAV with air economizer; and (5) VAV with constant outdoor air flow.

All energy data were sorted into three bins. The bins were defined by a few significant **outdoor air temperature** conditions. The first bin was -18 to 13°C (0 to 55°F). Thirteen degrees celsius (fifty-five degrees fahrenheit) is the transition outdoor air temperature at which the building's heating energy requirements begin (i.e. the building's balance temperature). The second bin is from 14 to 26°C (56 to 79°F). Twenty-six degrees celsius (seventy-nine degrees fahrenheit) is the approximate room air temperature - the point at which warmer outdoor air becomes a sensible cooling burden to the building. (When outdoor air at less than 26°C (79°F) is drawn into the supply air stream at the main cooling coil, instead of warmer recirculated air, the temperature of the air at the cooling coil is marginally reduced, thereby resulting in a reduction in cooling energy use - or an energy saving bonus). The third bin is from 27 to 38°C (80 to 100°F). In this temperature range, the whole building is expected to be in a cooling mode, and all outdoor air drawn into the supply air stream should result in a sensible cooling energy increase.

CAPACITY LIMITATIONS IN EXISTING BUILDINGS

The five HVAC configurations in each building were designed to operate at outdoor air flow rates of 2.5 l/s/occ (5 cfm/occ). When control settings were increased to 10 l/s/occ (20 cfm/occ), most systems experienced capacity problems which resulted in the loss of control of space temperature and humidity conditions. However, increasing the design cooling capacity by approximately 10 to 20 percent eliminated the temperature control problem in most buildings. This suggests that unless existing buildings designed under the old ASHRAE standard have some excess capacity, meeting ASHRAE Standard 62-1989 may not be feasible until the capacity issue is properly addressed. A more detailed analysis of the relationship between heating and cooling capacities, space temperatures, and humidity levels is currently underway.

BUILDING ENERGY USE PATTERNS

DOE-2 simulations were used to generate energy data for HVAC equipment which was sized to meet the higher 10 l/s/occ (20 cfm/occ) requirement. (Heating and cooling coil loads (kWh or MBtu) were used as surrogates for analyzing heating and cooling energy use respectively, since the detailed hourly coil load data were available in DOE-2.) Analysis for the base building (Building A) with a VAV system and a fixed outdoor air damper is shown in Figure 1. The cooling coil load has been subdivided into its two component loads: the sensible cooling load and the latent cooling load. The heating coil load increases in the winter (Bin 1) with the higher outdoor air flow rate. The sensible cooling coil load increases in the summer (Bin 3) with a higher outdoor air flow rate. However, the sensible cooling coil load drops in the winter (Bin 1) and intermediate spring and fall seasons (Bin 2). Latent cooling loads increased most in the warmer summer weather (Bin 3), with a moderate decline in winter (Bin 1). Overall, the building experiences a moderate net annual **increase** in both heating and latent cooling energy use, and a decrease in sensible cooling with increased outdoor air flow. This net annual result is a function of the energy pattern in each bin, and the proportion of the year spent in each bin.

Binned energy use results for the core dominated building (Building B) with a CV system and a constant outdoor air flow control strategy are shown in Figure 2. The heating coil load for this building increased with increased outdoor air flow in the winter (Bin 1). However, the drop in sensible cooling load in the winter (Bin 1) and spring and fall seasons (Bin 2) was considerably greater than in the previous case. As expected, the latent cooling load experienced its greatest increase in the summer (Bin 3).

OTHER BUILDING CONFIGURATIONS

The overall effect of an increase in outdoor air flow rate from 2.5 to 10 l/s/occ (5 to 20 cfm/occ) on total annual HVAC energy costs for the temperate Washington, D.C. climate is shown in Table 2. This confirms other authors' conclusions that the energy impact is typically quite minimal. Our analysis shows that it may result in annual energy cost savings for some buildings.

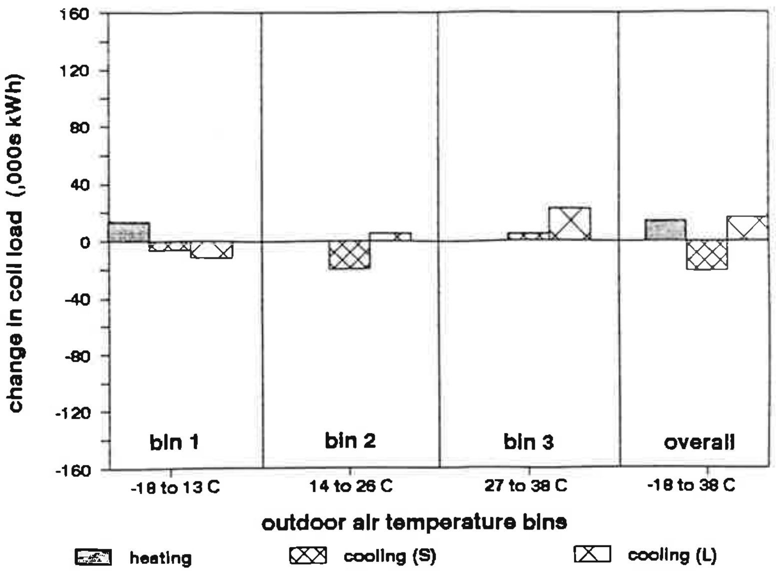

Figure 1: Change in coil oads with increased outdoor air flow rate for Building A with VAV system and fixed outdoor air damper.

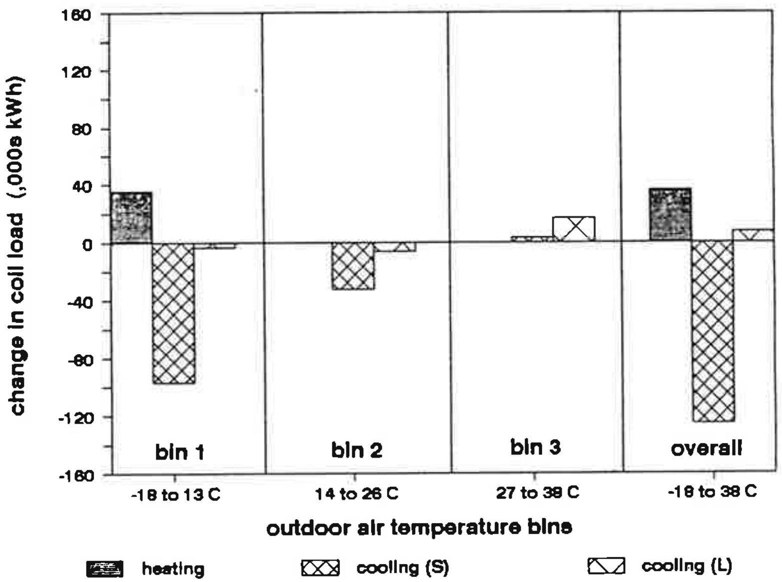

Figure 2: Change in coil loads with increased outdoor air flow rate for Building B with CV system and constant outdoor air flow.

Table 2: Percent change in building and HVAC energy cost and use for various building configurations

Case		#1 CV/ECON	#2 CV/COA	#3 VAV/FIX	#4 VAV/ECON	#5 VAV/COA
Change in HVAC energy use						
A Base case	heating	1.5%	25.9%	16.9%	19.6%	21.7%
	cooling	2.3%	2.3%	2.3%	2.3%	2.3%
	fan	0.5%	0.1%	-0.8%	-0.7%	-0.7%
Change in HVAC energy costs*						
A Base case		2.9%	2.2%	3.0%	4.0%	2.1%
B Core-dominated		1.6%	0.1%	1.8%	4.2%	1.8%
C High occupant density		3.1%	1.5%	5.4%	7.1%	5.7%
D Efficient shell		3.0%	0.0%	-0.4%	2.9%	-0.8%
E Efficient HVAC equipment		2.9%	2.2%	2.8%	3.7%	2.1%
Change in building energy costs*						
A Base case		1.2%	1.0%	1.2%	1.5%	0.8%
B Core-dominated		0.6%	0.0%	0.6%	1.4%	0.6%
C High occupant density		1.4%	0.7%	2.4%	2.9%	2.4%
D Efficient shell		1.2%	0.0%	-0.2%	1.0%	-0.3%
E Efficient HVAC equipment		1.1%	0.9%	1.0%	1.2%	0.8%

* Rate Structure: $2.00/kW; $0.07/kWh; $0.16/m³

Although this paper presents only the impacts of increased outdoor air flows for buildings located in Washington DC, the results show that increased outdoor air flow during hot weather (Bin 3) will result in increased cooling energy use. This cooling energy increase is a function of two factors: (1) the number of hours of occurrence of hot weather, and (2) the **actual** increase in the volume of outdoor air flow achieved by the HVAC system configuration of interest. Thus, for any location with a climate that has more hours of hot weather than Washington DC, greater increases in cooling energy costs should be expected.

The impacts of increased outdoor air flow in the winter is not as straight forward. Because of the need for concurrent heating and cooling in commercial buildings in the winter, the introduction of cold outdoor air into a building both helps and hurts. The cold air reduces cooling energy use, but increases heating energy use. This problem is further exacerbated by reheat systems. Thus, depending on the HVAC system type, and the relative magnitude of heating and cooling loads (i.e. other building design parameters), the net effect of increased outdoor air flow in the winter season (Bin 1) is highly variable. For the special case of buildings with high internal gains (e.g. Building C), the results shown that energy costs increase significantly in all weather conditions.

CONCLUSIONS

Buildings designed for 2.5 l/s/occ (5 cfm/occ) of outdoor air per occupant may experience capacity problems if outdoor air settings are increased to 10 l/s/occ (20 cfm/occ) per occupant. Buildings designed for 2.5 l/s/occ (5 cfm/occ), but with slight excess capacity (about 10 to 20 percent excess cooling capacity for temperate climates), appear to be capable of meeting the 10 l/s/occ (20 cfm/occ) requirement of ASHRAE 62-1989. When the extra capacity is not available, problems in maintaining thermal comfort may be significant unless the capacity issue is properly addressed. Caution needs to be exercised when advising building owners to increase outdoor air flow rates in existing buildings.

The energy cost impacts of increased outdoor air flows are dependent on the relative changes in heating and cooling energy occurring in each of the three seasonal temperature bins introduced in this paper. The increased cooling energy demand in summer and heating energy demand in winter is counterbalanced by decreases in cooling energy demand in winter and the transitional seasons. The effect on annual energy cost depends on the relative magnitude of these changes, the relative importance of the temperature bin for that climate zone, and the relative prices of gas and electricity. The net effect typically results in a marginal increase or a slight decrease in energy costs. Warmer climates will experience a greater energy cost increase due to the larger increase in both sensible and latent cooling loads. Colder climates will have mixed results, but buildngs with high internal gains will likely experience a higher energy cost increase in colder climates.

REFERENCES

1. ASHRAE , 1989, ASHRAE Standard 62-1989: Ventilation for Acceptable Indoor Air Quality, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta.
2. Eto, J., and Meyer, C., 1988, "The HVAC Costs of Fresh Air Ventilation in Office Buildings", ASHRAE Transactions, V. 94, Pt.2.
3. Eto, J., 1990, "The HVAC Costs of Increased Fresh Air Ventilation Rates in Office Buildings, Part 2", from Proc. of Indoor Air 90: The Fifth International Conference on Indoor Air Quality and Climate, Toronto, Canada.
4. Sauer, H., and Howell, R., 1992, "Estimating the Indoor Air Quality and Energy Performance of VAV Systems", ASHRAE Journal, Atlanta, July.
5. Steele, T, and Brown, M., 1990, "Energy and Cost Implications of ASHRAE Standard 62-1989", Bonneville Power Administration.
6. Mutammara, A., and Hittle, D., 1990, "Energy Effects of Various Control Strategies for Variable Air Volume Systems", ASHRAE Transactions, V. 96, Pt. 1. Atlanta.
7. Levenhagen, J., 1992, "Control Systems to Comply with ASHRAE Standard 62-1989", ASHRAE Journal, Atlanta, September.
8. Curtis, R., Birdsall, B., Buhl, W., Erdem, E., Eto, J., Hirsch, J., Olson, K., and Winkelmann, F., 1984, DOE-2 Building Energy Use Analysis Program, Lawrence Berkeley Laboratory, LBL-18046.

A GAS SENSOR ARRAY FOR MEASUREMENT OF INDOOR AIR POLLUTION - PRELIMINARY RESULTS

J.D. Wenger, R.C. Miller, and D. Quistgaard

Controls Group Research, Johnson Controls Inc, Milwaukee USA

ABSTRACT

An array of gas sensors was used to evaluate air contaminants in 36 spaces, concurrent with judgements by a trained sensory evaluation panel. Data were processed by advanced pattern recognition techniques to discern patterns relating sensor readings to voted indoor air pollution (IAP) level in decipol. The mean vote of the panel could be predicted with a mean error of 2.9 decipol and a standard deviation of error of 2.4 decipol, comparing favorably with panel member votes across all spaces with 2.1, and 2.9 decipol respectively. While the results of this study are promising, the small number and variety of sample spaces limit the generality of the recognized patterns. Additional work to extend the capabilities of this method is ongoing.

INTRODUCTION

A technique for direct measurement of indoor air pollution (IAP) level is desirable for building diagnostics and for continuous monitoring of indoor environments. An IAP sensor would allow control actions such as modulation of outdoor air and zone dampers for ventilation, operation of air cleaning devices, or alarming of unhealthy conditions. Various analytical techniques exist for measurement of low concentrations of gases and vapors in air, and even a few reasonably priced sensors can monitor fairly low concentrations of some compounds. These measurements are not representative indicators of IAP, because they do not reflect human response.

One of the most promising proposed indicators of IAP is the sensory evaluation-based decipol scale, because it focuses on human response (Fanger, 1988). The decipol is primarily a measure of unpleasant odor concentration, but may also be affected by other sensory inputs. While the human olfactory organ is a very sensitive detector of many chemicals that are of concern in indoor air, the receptors quickly become saturated, and the perceived response diminishes. To avoid this adaptation to odorous environments, panelists must enter the environment to be judged from a clean environment, usually outdoors, and quickly make a judgement on their initial perception. To maintain consistency in judging the decipol level, panelists must undergo a daily calibration.

Practical problems in using the decipol to judge the pollution level in indoor environments includes the inability to perform continuous measurements, daily calibration requirements, the lack of portability of a panel of 10 people or more, and the overall expense of the technique. As a measure of IAP, the decipol is essentially limited to those contaminants that have an odor. Odor can not be a complete indicator of IAP however, even when consideration is limited to gaseous contaminants. Carbon monoxide is a common and potentially harmful air contaminant that does not have a detectable odor. In general, irritation experienced by occupants after extended exposure will not necessarily correlate with odor response.

An ideal sensing system for measuring IAP would have the following features:
1. human-equivalent response for odors, unhealthful non-odorous compounds, and irritants
2. capability to perform real-time measurements without sensors becoming saturated
3. stable and accurate operation without frequent calibration
4. portability for diagnostic activities
5. small size and low cost for permanent installation