Energy and Buildings, 1 (1977) 69 - 76

Energy Savings Due to Changes in Design of Ventilation and Air Flow Systems

F. W. HUTCHINSON
University of California, Berkeley, Calif. 94720 (U.S.A.)

Procedures for the design of ventilation and air flow systems are shown to be energy wasteful. The cause lies both in the methodologies and their technical bases, and in the influence of other factors that enter into a final systems choice for a building (particularly economics).

The issue of general outside air requirements is discussed. It is found that any system in which outside air supply is sufficient to control body odor necessarily meets oxygen requirements many times over, and that the ventilation rate can be reduced by about 45% to 60% if higher humidities are used. The long-standing belief that required ventilation flow rate for odor control must increase as air space per occupant decreases has previously been proven untrue, yet a survey of the ASHRAE recommendations and various building codes shows that present standards are based on this inaccurate and energy-wasteful principle. New standards, including a rate of 7.5 cmf/person of outdoor air (not conditioned) during the heating season, are proposed based on the result that the ventilation rate is independent of air space per person.

Air distribution systems and sub-systems are analyzed in terms of minimum energy requirements. Energy saving by velocity reduction is discussed. Fan power requirements for a high velocity air distribution system are increased by 2500% over that for a low velocity system in commercial buildings.

A comparison of the "old" ASHRAE definition of Effective Temperature, which serves as a thermal comfort criterion in AC systems design, is made with the "new" definition, and the consequent energy savings that should result when the "new" definition is accepted in practice is demonstrated.

INTRODUCTION

In many respects present design procedures (and particularly various code requirements) are energy wasteful for two main reasons:

1. Failure of the art of heating and air conditioning to rapidly incorporate into codes and design procedures the results of latest research. Two examples of this (to be discussed in subsequent sections) are: (a) the energy saving possibilities related to acceptance of the "new" ASHRAE definition of Effective Temperature, ET*, and the consequent revised psychrometric location of the socalled comfort zone. (b) Research results that disprove the long standing belief that ventilation flow rate for odor control increases inversely with available air space per occupant. (c) Specification of outside air ventilation rates for maintenance of metabolic equilibrium (oxygen supply and carbon dioxide removal) and for body odor control in terms of required air flow rate per occupant rather than in terms of the largely meaningless values of cfm (ft^3/min) per unit floor area or air changes per hour.

2. Emphasis by the designer on minimization of first cost — or *total* cost — at the expense of excess energy requirements. Examples of such designs are:

(a) Use of high velocity systems with resultant greater power input to the fans.

(b) Use of *constant* air supply volume systems requiring that the air flow rate through the system be sufficient to carry a load equal to the sum of the maximum loads in the various zones.

(c) Use of dumping methods (as by fan bypass) with variable air volume (VAV), systems that reduce the constant fan flow rate to a value sufficient to carry the block load (the temperature and humidity; for this reason it is very difficult to establish a reproducible scale of odor response. Analysis of odor sources is usually divided between such primary sources as occupants or cigarettes and such secondary sources as odor contaminated furnishings, filters, or cooling coils. Secondary sources can be largely controlled by good HVAC system hygiene: avoidance of prolonged entrapment of dust, viable organisms, and other contaminants in system components.

With respect to energy conservation in commercial buildings, the primary odor sources of concern are those due to the metabolic processes of the occupants or to smoking by the occupants. The problem is one of minimizing the required supply of outside air in order to reduce the energy requirements for heating, cooling and propelling the air through the system. The sources of body odor are exhalations, insensible perspiration and the products of organic decay. Butyl acetate is an important and offensive constituent of body odor. The rate at which odoriferous material is released from the human body depends on personal hygiene and the degree of physical exertion. However, some of the organic materials contributing to body odor are self oxidizing, and thus would disappear in an unventilated space in time.

ODOR CONTROL

Reduction of odor to an acceptable intensity can be achieved by the following methods or by combinations of these methods. An analysis will establish the particular design that will provide greatest energy conservation.

1. Isolation of the source to prevent migration of the odoriferous gases and vapors to the entire occupied region. Typical examples of this method are the use of negative pressure (exhaust induced) to localize kitchen odors, or separate zoning for conference rooms or waiting rooms where there may be heavy smoking.

2. Decontamination of room air before recirculation. Recent developments make this method economically competitive in total cost with dilution by outside air; in terms of minimum energy usage decontamination is in many cases more advantageous than dilution.

3. Modification of room odor by addition of other odoriferous materials to mask or counteract body odor, as with a pleasant additive. (This method is not recommended and many codes prohibit its use.)

4. Interference with the olfactory sensor (as by addition of ozone) so that it becomes irritated and loses sensitivity (not recommended and often prohibited).

5. Dilution by ventilation with outside or cleaned outside air. This is the most common method of body odor control.

DECONTAMINATION OF ROOM AIR

Many procedures are available for purifying return air before recirculation. Among these are various methods of chemical control for converting the odor causing gases and vapors to inert or less odoriferous species. Activated carbon is widely used as an adsorbent for entrapment of condensed objectionable vapors in the capillary spaces, but the carbons must be regenerated periodically by heating. Absorption systems are also used, as are washing and scrubbing devices with water or with a chemical solution (e.g., caustic soda or lithium bromide). Ultra violet light is used to kill airborne organisms.

When it is economical to use a decontamination process, the outdoor air requirement can be substantially reduced with resultant energy saving. For example, if odor control dictates that supply air contain 30% outside air and 70% recirculated room air, the outside air fraction could be eliminated (except for that amount necessary to meet physiological needs) if 30% of return room air were decontaminated with 100% effectiveness. An equivalent situation would exist if all of the return room air were decontaminated with 30% effectiveness. Thus by reconstructing supply air from partially decontaminated return air mixed with a reduced fraction of outside air, the energy requirement for ventilation would be reduced.

Further, when the reduction in outdoor air can be reflected in a corresponding reduction in total supply air there will be a further energy saving due to reduction in the power input to the fan; the possibility of achieving such a saving in fan air will depend on meeting the requirement for an adequate approximately 4 cfm/person. In rooms where people are smoking the minimum ventilation rate goes up to 25 cfm/person with a recommended value of 40 cfm/person. Honma ("The Ventilation of Dwellings and Its Disturbances", Royal Institute of Technology, Stockholm, 1975, p. 14) states that "... the minimum fresh air requirement is 0.003 m^3/sec per person (6.35 cfm/person) in order that a person entering from relatively clean air may get the impression of allowable odor intensity in the room while it is 0.002 m^3/sec per person (4.25 cfm/person) in order that the occupants of the room may maintain an air quality of fair to good".

Thus, when occupants are the sole source of odor within a conditioned space, 7.5 cfm/person of odor-free outside air should suffice.

ENERGY SAVING BY VELOCITY REDUCTION

With increasing costs of multi-story buildings there has been a growing trend towards use of high velocity air distribution systems. The smaller ducts used with such systems sometimes permit reduction in the size of ceiling crawl spaces and thereby serve to reduce the height between floors. The smaller ducts require less material, are less expensive to fabricate (except where anti-leakage construction requires more careful installation), and are less expensive to insulate. However, the cost of attenuating sound is increased. For large systems (e.g., 50,000 cfm at fan discharge) maximum velocity in the main ducts may be as high as 6000 fpm (ft/min) with up to 3000 fmp in the branches; for smaller systems, (e.g., 2000 cfm at fan discharge) maximum main duct velocity approaches 2500 fpm with up to 1000 fpm in the branches.

Low velocity systems operate with main duct speeds varying from an average of 800 fpm for residences to 1200 fpm for schools and public buildings.

In energy terms, the advantages of high velocity are paid for by a substantial increase in the power input to the fan. Fan horsepower increases as the square of the velocity. Thus use of a 6000 fpm high velocity system in a commercial building would increase the fan power requirement over that for a 1200 fpm low velocity system by $(6000/1200)^2$, or 2500%.

An additional factor to consider in estimating the cost of a high *vs.* low velocity system is that all the energy input to the fan (except motor and drive losses when this equipment is outside the conditioned space) enters the discharge air from the fan either as a thermal load or as an increase in kinetic energy, and thus increased fan power also represents an increased cooling load. In a blow-through cooling cycle the added thermal load will be picked up in the cooling coil so will increase the required refrigeration capacity, but will not become part of the space sensible heat load. In a draw through cycle, the fan load will be part of the space sensible load and will therefore require an increase in the flow rate of supply air to the conditioned space.

To minimize energy requirements, a blow-through cooling system is preferable, with the lowest acceptable distribution velocities to minimize fan power requirements.

MINIMIZING FAN ENERGY BY USE OF TRUE VARIABLE AIR VOLUME SYSTEM

When an all-air zoned system is used for heating and/or air conditioning a commercial building, one of the following three types is likely to be selected:

Dual-duct. This system, particularly when used with constant volume, is inherently wasteful of energy as it is based on the use of irreversible mixing, with consequent loss of availability.

Re-heat System for Cooling. This system is thermodynamically equivalent to making up for a reduction in space load by supplying additional energy (reheating) to cancel the otherwise excess cooling capacity of the constant volume supply air. The system is wasteful of energy.

Variable Air Volume (VAV). According to ASHRAE Systems, 1973, p. 3 - 15: "True variable volume systems have inherent potential to provide flexible heating and cooling *with minimum annual energy consumption compared with other air systems.* ... true VAV implies a system volume reduction that coincides exactly with the sum of zone volume reductions. It permits substantial savings in

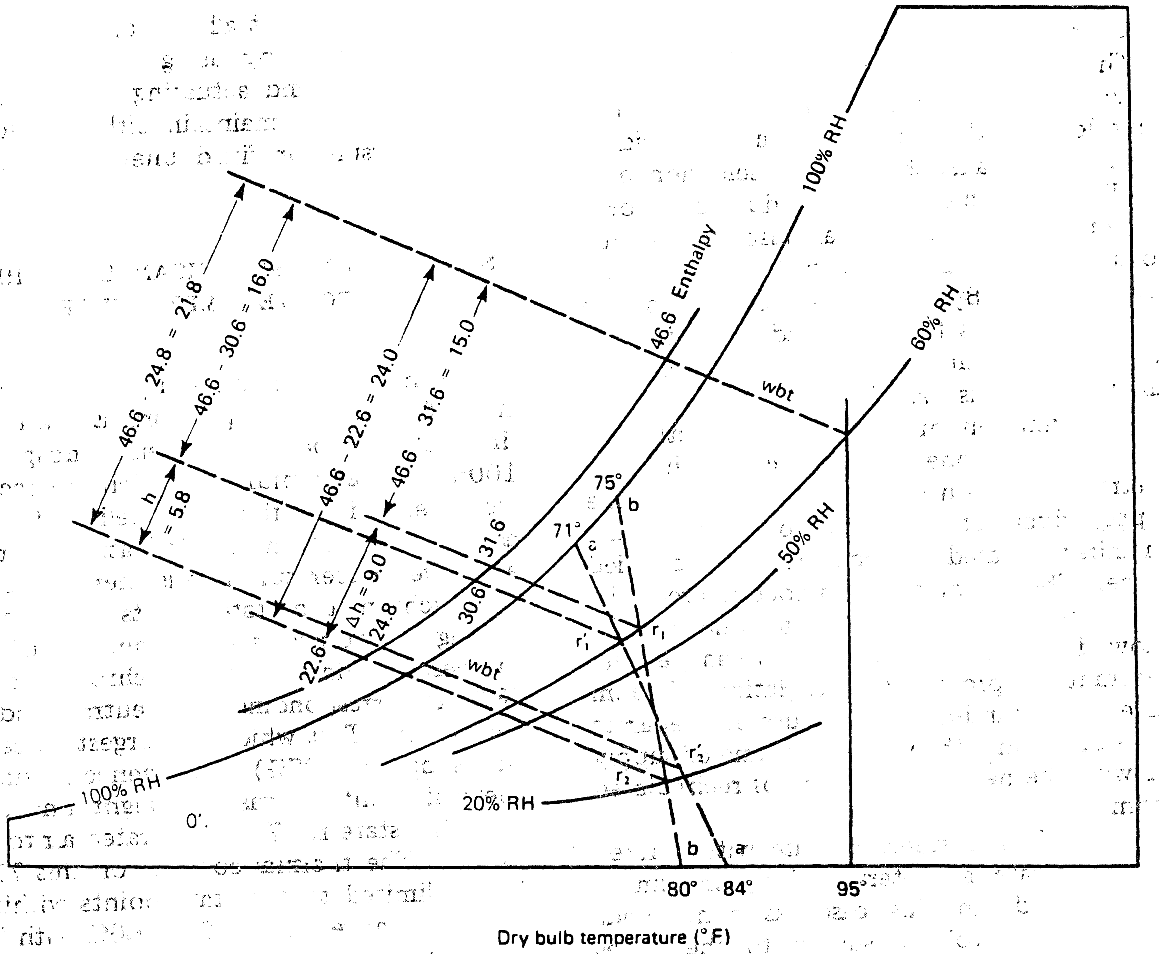

Fig. 1. Psychrometric chart and diagram for numerical example demonstrating the energy saving significance of the new ASHRAE Effective Temperature.

Example for old ET

Referring to Fig. 1, assume that the state of outside air, 0, is at 95 °F dry bulb temperature and 60% relative humidity with corresponding enthalpy of 46.6 Btu/lb. If the selected inside design state is taken at r_2 on old ET line aa, 71 °ET, (the lowest permissible relative humidity for thermal comfort) the corresponding enthalpy at r_2 is 24.8 Btu/lb. The energy extraction required for cooling and dehumidifying the ventilation air will therefore be $h_0 - h_{r_2'} = 46.6 - 24.8 = 21.8$ Btu/lb.

If, in contrast, a state r_1' is selected on the same 71 °ET line at maximum permissible relative humidity of 60%, the corresponding enthalpy would be 30.6 Btu/lb and energy extraction from ventilation air would be reduced to 46.6 — 30.6 = 16.0 Btu/lb. Thus the saving in energy required to process outside air to state r_1', as compared with state r_2', is 21.8 — 16.0 = 5.8 Btu/lb, or a reduction of, (5.8/21.8)100 = 36.6%. This saving has been achieved with no change in the indoor feeling of warmth since r_1' and r_2' are on the same ET line. It should be noted, however, that different air conditioning processes (and possibly equipments) would be needed to condition the air: from stable 0 to state r_1' there is required a greater extraction of sensible heat and a lesser extraction of latent heat than would be needed to condition outside air to state r_2'.

*Example for new ET**

Consider now that room state is selected at 20% relative humidity, r_2, on the new ET* summer neutral line bb with corresponding enthalpy of 22.6 Btu/lb. The energy extraction to condition ventilation air is then $h_0 - h_{r_2} = 46.6 - 22.6 = 24$ Btu/lb.

If, in contrast, the state r_1 (at 60% relative humidity, but at the same ET* as r_2) is selected as the inside state, the enthalpy of