# AVOIDING SICK BUILDINGS WHILE ASSURING OCCUPANT PRODUCTIVITY AND BUILDING OPTIMIZATION

**Dan Int-Hout**
***Chief Engineer***

## ABSTRACT

In order to meet the increased expectations of building occupants, designers must be aware of the conflicts between first cost economics, occupant productivity and life cycle costs. Cost savings measures have resulted in severe building problems, including deaths due to Legionnaire's disease, and often energy consumption increases. Buildings that do not meet the needs of the occupants often result in expensive redesign or worse, result in lawsuits against all parties involved. Recent court cases make it imperative that designers understand the changing 'rules' of the road. Building owners need to understand how their building systems work, to avoid creating problems in a tightening economy.

## AIR DISTRIBUTION: SELECTING COMPONENTS & SYSTEM PARAMETERS FOR EFFECTIVE AIR MIXING

Modern environmentally controlled spaces consume significant amounts of energy in maintaining a stable environment within the structure. The demands for heating and cooling vary greatly over time, and different strategies are employed to respond to these varying loads. Ventilation requirements must also be met by the HVAC system. This conditioned and ventilated air must be effectively delivered to the building occupants.

There are two principal design conditions in HVAC systems. Perimeter zone loads vary over a broad range from heating to cooling, and are affected by both exterior and interior factors. In interior zones the heat generated by lights, occupants, and office machinery provides a continuous cooling demand.

The proper selection of diffusers is necessary to ensure that both occupant comfort and adequate ventilation mixing are provided. Both the engineer and the architect need to have input into the final selection as the choice of diffusers is based on both engineering and architectural concerns,. In addition, the ideal selection is dependent on the type and operation of the air supply to the diffuser. While there are many ways of supplying conditioned air to an office space including displacement ventilation, underfloor vertical air distribution and task cooling, this article will deal with the predominant method, the ceiling supply air diffuser.

ASHRAE continues to sponsor research into the performance of air distribution elements, and much has been written on the subject in both technical papers and magazine articles. At the same time, ASHRAE Standards for Indoor Air Quality (62.1-2004) and Comfort (55-2004) state requirements for the resulting ventilation mixing, air temperatures and airspeeds, and even turbulence intensity (the rate of change of the local air speed) in the zone. Awareness of Indoor Air Quality issues and concern over occupant complaints has increased the visibility of proper diffuser selection, location, and design. At the same time, complaints of 'stuffiness' are finally being diagnosed as temperature, not Indoor Air Quality complaints.

The ASHRAE handbook provides recommendations for diffuser selection, which have been included in at least one lawsuit as being considered to be the 'acceptable standard of care'. All available research indicates that when air distribution is provided from the ceiling, a thoroughly mixed condition, throughout the space, is the desired result. Furthermore, the research has proven that with properly selected ceiling diffusers, excellent air distribution and ventilation mixing can be achieved with many types of diffusers, and in many types of spaces, including the open landscape office.

The key element here is the term 'properly selected'. Diffuser selection ranges from the selection of a 'hole cover' (or 'architectural duct termination') to a detailed analysis of the diffuser air supply pattern in each zone. A zone-by-zone analysis is almost always prohibitive in terms of design time, and probably not necessary for most spaces. Interior open office spaces, which are typically in cooling mode year round, can usually be characterized in a general way, and excellent diffuser selections made for these locations. Perimeter zones are more complicated, with both thermal and aerodynamic concerns to be considered, but again, these can often be characterized in a general way for the building design, and excellent air distribution can be achieved. When there are problems, it is our experience that there was little (or no) actual air distribution design, loads are considerably different than planned, or products were installed which did not meet the designer's specifications.

The 2005 ASHRAE Handbook of Fundamentals, Chapter 33, provides two basic rules for overhead heating and cooling:

- In cooling mode, diffuser selection should be based the ratio of the diffuser's throw to the length of the zone/area being supplied, at all design air flow rates, to achieve an acceptable Air Diffusion Performance Index (ADPI).

- In heating mode, the diffuser to room temperature difference (delta-t) should not exceed 15$^{o}$ F, to avoid excessive temperature stratification. ASHRAE Standard 55-2004 defines the level of acceptable vertical temperature gradation at not to exceed 5$^{o}$F.

ASHRAE has recently defined a term for describing the mixing of supply and room air, replacing the somewhat ambiguous terms Ventilation Effectiveness and Ventilation Efficiency with a new term, Air Change Effectiveness, or ACE. This term is used in both ASHRAE Standard 129, the Method of Test for Air Change Effectiveness, and in the public review draft (recently withdrawn) to ASHRAE Indoor Air Quality Standard 62-89R. It was also the basis for a LEED point in the 2.1 version (the V2.2 of LEED has ASHRAE 62.1 as a prerequisite, and a point is no longer gained for meeting this requirement)

The relationship between ADPI and ACE has not been fully evaluated at this time, but the data available at present indicate that if a high ADPI is attained, the ACE will also be high. While there have been no reported tests where the ACE was significantly below 100% when cooling from the ceiling, it has been demonstrated that in heating mode the ACE may decrease significantly, as low as 20%.

The ASHRAE IAQ Standard, 62-1989, assumed ventilation mixing of 100% in setting minimum ventilation rates. If it can be shown that the ACE is less than 100%, then the amount of outside air must be increased above the required minimums. With presently available information, when a high ADPI is measured, the ACE is high as well. The current version of ASHRAE 62.1 (2004) now includes Air Change Effectiveness in a table. When cooling form the ceiling ACE is always assumed to be 100%. When heating from the ceiling with ceiling returns, the discharge to room delta-t must not exceed 15$^{o}$F, and the 150fpm throw must reach to within 4.5 ft from the floor, or the outside air ventilation rate must be increased by 25%.

While it is possible to have a high ACE and a low ADPI, especially if the HVAC system air is supplied directly into the occupied zone, uniform comfort will not be likely.

## COOLING SELECTION BASED ON ADPI

ADPI is intended as a measure of performance in cooling mode. When in heating mode, the ADPI criteria often become overly sensitive to temperature differences (due to the very low air speeds present in heating mode), and as a result ADPI is not a good means of performing heating evaluations. Heating is best analyzed as a function of vertical temperature gradients as compared to ASHRAE 55's requirements. Interior spaces, however, are predominantly in a cooling mode of operation so this limitation is seldom a problem in interior zone evaluations.

Utilizing the ASHRAE Handbook table for predicting ADPI is cumbersome, and is seldom accomplished. It is possible, however, to simplify this analysis by combining a diffuser's throw performance with a cfm/sqft analysis and diffuser spacing, to produce an ADPI "Performance Envelope" graph.

An example is shown, in several Figures below. In these graphs, the x axis is flow rate/unit area, and the y axis is half the separation distance, or L, the characteristic room length. The horizontal curved lines are simply calculations of flow vs. area served, while the vertical boundaries are computed from the ASHRAE maximum and minimum $T_{50}/L$ ratios for this type of diffuser. Performance within the area bounded by the lines should achieve an ADPI of 80% or greater.

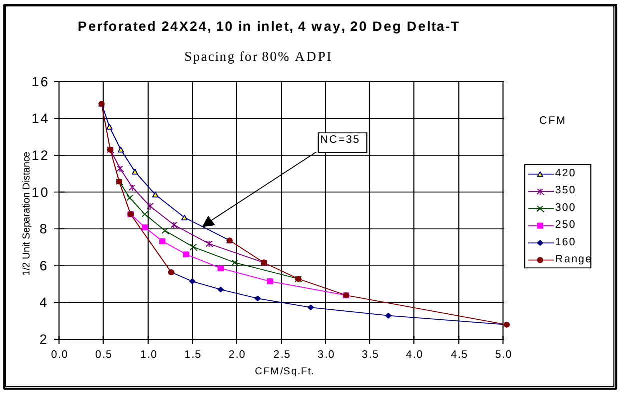



Figure 1 shows the performance envelope of a low cost perforated diffuser with a load resulting from a room/supply differential of $20^{o}\Delta T$. (Different delta-t's will result in different load rates, and will change the location of the vertical boundaries somewhat.)

One can see that this diffuser will not have acceptable performance at reduced flows, and in a VAV application will have limited "turn down".

*__Figure 1: Low Cost Perforated Diffuser__*

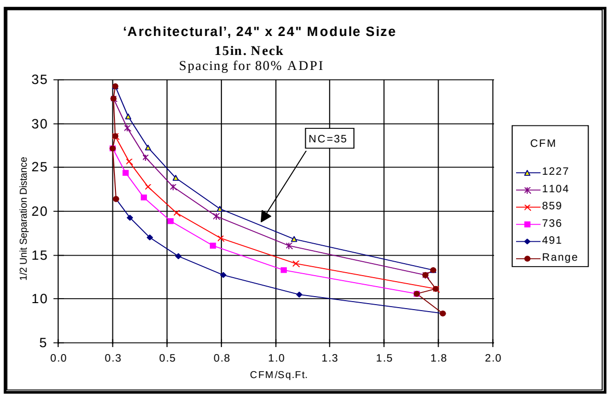
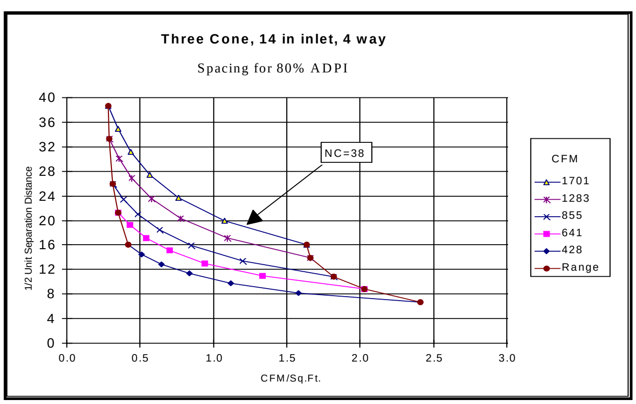



Figure 2 shows the performance of an Architectural (plaque) diffuser with a similar 4 jet pattern. Here one can see that this diffuser will both have better performance at low flows, and will have greater turn down in a VAV situation.

*__Figure 2: Architectural Diffuser__*




In figure 3, we show the performance for a circular pattern, three cone diffuser. This diffuser performs well in all applications.

*__Figure 3: Cone Diffuser__*

Analyzing these charts, it can be seen that they have quite different ‘turn down’ limits. The perforated diffuser shown has quite a high flow capability, but can’t be used below 0.7 cfm/sq.ft. On the other hand, the others shown will operate down as low as 0.2 cfm/sq.ft but are limited above 1 cfm/sq.ft. at this separation. If different neck areas are analyzed, a different separation distance results, but it will be seen that the flow/unit area limits don’t change appreciably.

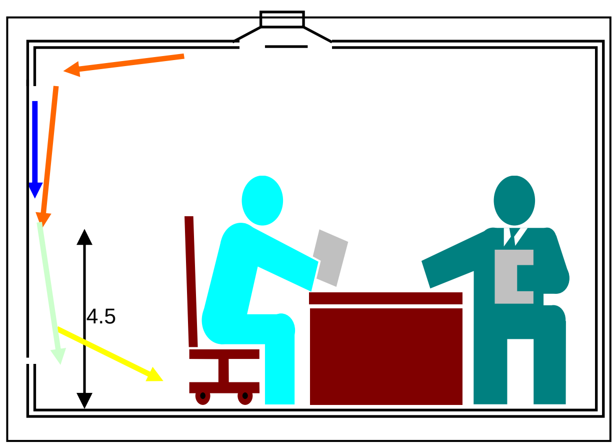
Selecting diffusers using the above charts will result in selections which meet the requirements of the ASHRAE Handbook of Fundamentals, and will also meet ASHRAE Standard 55’s limitation of 5°F Degrees vertical temperature stratification. In fact, using ADPI analysis is the only proven method of demonstrating compliance to the Standard 55 requirement, at the design stage. Experience has shown that when these guidelines are followed, excellent air distribution, uniform temperatures and no objectionable drafts should be expected in the space. Providing that acceptable temperatures are established as a function of the occupant’s clothing and activity levels, occupant comfort should be assured as well. Indoor Air Quality, which is many times a perception issue, will be assured as well.

## OVERHEAD HEATING

Heating perimeter zones from the ceiling became possible when perimeter glass became better, and in response to needs for better space utilization along the glass. A number of technical papers presented in the late 70’s defined the parameters of this design, and established a repeatable method of test for evaluation of these spaces (ASHRAE113). The ASHRAE Fundamentals Handbook (now 2005, Chapter 33) incorporated these results in the early 80’s, and overhead heating became a ‘standard’ method of heating perimeter zones.

Today we see a surprising number of designs which are obviously established in the absence of an understanding of these studies. The cfm and kW settings often specified on VAV terminals as well as discharge temperature requirements for small package units evidence this. Discharging low velocity, highly heated air at the ceiling may work in residential applications with low returns, but it will ensure highly stratified, poorly ventilated spaces with uncomfortable occupants in commercial applications with overhead returns. One of the authors has recently polled over 2000 consulting engineers regarding awareness of the overhead heating ‘rules’. Almost none were aware of the ASHRAE design limitations.

The ASHRAE Handbook of Fundamentals has since 1979 provided specific guidance on the maximum room discharge temperature difference (not to exceed 15°F) for effective control of the perimeter environment. In fact, the author and others have conducted several hundred tests of perimeter designs in full-scale mock-ups, all confirming the ASHRAE guidelines. ASHRAE Standard 62.1 2004 now requires that ventilation rates be increased by 25% when discharging air that exceeds this limitation, and when the diffuser discharge velocity is insufficient to get the heated air to within 4.5 feet of the floor.

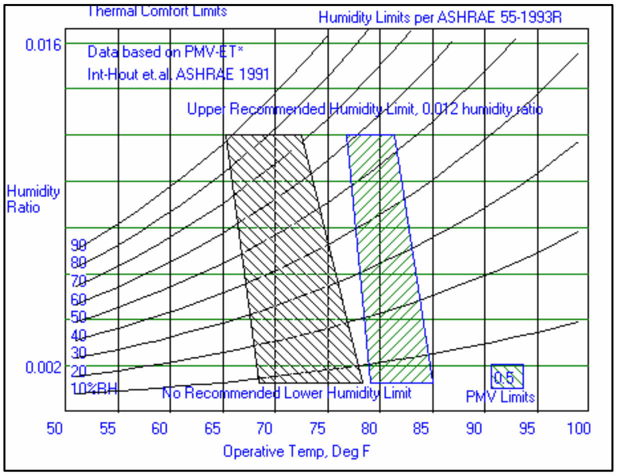



## THERMAL COMFORT: DETERMINING OPTIMUM OCCUPANT COMFORT STRATEGIES

Many IAQ complaints, especially those characterized by the expressions ‘STUFFY’ and “DRAFTY” are probably misdiagnosed thermal comfort problems. The ASHRAE Standard on comfort (Std 55-2004) can be misleading in terms of occupant comfort. The older versions of this standard assumed both a ‘Winter’ and a ‘Summer’ condition, shown as comfort envelopes on a psychometric chart. These envelopes are shown with some overlap, suggesting that there is a single temperature, which may satisfy both conditions.

*Figure 5: Thermal Comfort Envelopes*

When comfort is analyzed using the equations now incorporated in current version of the standard, very realistic conditions can be developed which do not, in fact, overlap. Shown below are two envelopes, one for an active individual wearing 1 Clo (on the left) and another for a very sedentary occupant wearing 0.5 Clo. (on the right).




It can be seen that a single setpoint, such as 75°F 50% RH, will satisfy neither individual above. Another problem results from occupant's misdiagnosis of what is troubling them. The individual on the left, (1.1 Met, 1.0 Clo), at 75°F, will probably not complain of being too warm, as indicated, but will more likely complain of being 'Stuffy'. Building operators may be inclined to ignore these complaints, especially when the building is in economizer mode, as they know that 100% outside air is being supplied. A slight drop in temperature, however, will result in comments of how much 'fresher' the air seems now.

The individual on the right (0.9 Met, 0.5 Clo), at 75°F, will probably complain of drafts, not of being too cold. The women in the office will be the most sensitive to this phenomenon due to the fact that they are less likely than men to wear socks, and it is coldest at the floor. Cures often involve cardboard and/or tape on the diffusers, interfering with building air balance. The solution, of course, is to modify either the temperature or the occupant's clothing.

Even though occupant productivity is a difficult thing to measure, the economics of thermal comfort are easily calculated. Several factors illustrate why it is important to maintain occupant comfort.

- Occupants who occupy 150 sq.ft. and are paid $30K / year, cost $200/Sq.Ft., annual salary.
- Buildings seldom cost more than $2.00/Sq.Ft.-year to heat and cool.
- Adjusting thermostats to save energy is unlikely to save more than 5% (interior and perimeter zone) HVAC energy use, or 1/2000 of the salary cost.
- We seldom spend more than $20/Sq.Ft. on HVAC, First Cost, on a building. (In Europe, they spend as much as $70 / sq.ft.)
- Any heat produced in a commercial building will be paid for twice (once to make the heat, once to pass it through the cooling coils)
- Occupants will do whatever they can to maintain their own comfort levels.

Obviously, we cannot afford to have people uncomfortable in buildings.

## EMERGENCY BUILDING TEMPERATURE REGULATIONS

Imposed by President Carter in the late 70's, the EBTR established 68°F heating and 78°F cooling set point in federal buildings, in response to the Arab oil embargo. These regulations found their way, in various forms, into other state, local, and corporate codes, regulations and guidelines.

Several studies and many observations have all confirmed that in fact, energy consumption often increases when these arbitrary set points are enforced, and while the negative effect on productivity cannot be measured, it is obvious.  It is rumored that the GSA actually had a study confirming this increase in energy use (one of the authors saw a draft of the study), but it was never made public. Nonetheless, in response to the recent energy crisis in California, the EBTR was again imposed on federal buildings there.

Calculations show that with a few climatic exceptions, the maximum savings is on the order of 1% / degree set point modification for the HVAC system. The discomfort created by this causes occupants to add their own measures, such as fans or heaters, and when system conflicts result, the actual effect is to actually increase the energy consumption of the building. One should remember that a cooling fan is in fact a 100% energy to heat converter, which adds to the interior load to be sent to the cooling system. In one building investigated in the early 80's, the occupants had installed 1.5 w/SqFt. of fans to offset the 80F-space temperature that resulted from the 78F set point. Buildings with constant volume reheat systems such as the Library at the University of Richmond used more gas for heating in July than in January. Humidity levels in schools can be significantly increased resulting in better breeding grounds for mold and mildew.

## ACOUSTICS: ACCURATELY PREDICTING END USE

All manufacturers provide two sets of acoustical data: performance and application. Performance data are intended to be determined in accordance with a test code. Application data typically include assumptions about end reflection loss, sound power division through multiple outlets, lined discharge duct, ceiling transmission loss, etc. It is assumed that application data is based on the rated Sound Power submitted to ARI. There may be a significant difference, however, between ARI rating sound power and calculated application data, even given the same application assumptions.

The obsolete ADC Test Code permitted the application of a 10-dB reduction in sound power levels when calculating performance NC values for VAV boxes. Ceiling transmission loss, however, has been a matter of some dispute. The combination of ceiling material and the plenum in which the unit is located is difficult to quantify in a traditional laboratory. A number of tests, however, with full-scale mock-ups at several manufacturers have resulted in a consensus "ceiling/plenum" effect, included in the ARI 885-90 Standard. This table was again revised in the 1998 version of the ARI standard, reflecting data from an ASHRAE research project. At present, room and ceiling plenum effects have been combined, in the 1998 version of ARI 885.

### Unit Selection

Proper selection of a unit on the basis of acoustics requires an understanding of all the above issues by both the specifying engineer and the provider of the product. In general, the specification provided by the engineer is typically vague. Almost always, an NC is required (typically an NC 35), without any supporting documentation on the assumptions required to achieve that space sound level. In addition, in order to accurately predict sound power levels, the unit inlet static must be. If both are given, the selection is straight forward, and a selection program can easily perform the analysis and generate a submittal.

Lacking that, however, the supplier has to make some assumptions. ARI 885-98 provides a list of typical assumptions, and these are a good starting place.

### Duct Lining Issues

Parallel fan boxes and single duct boxes are not nearly as sound critical, when lined duct and flexible duct are involved in the design. If duct lining is not allowed, add several NC to the predicted space sound level. Some Single duct units with Foil lining, even if no lining is provided in the discharge ductwork, will still probably result in an NC<35 at all rated flows. With others, it may be significant. The following are the available linings for many VAV boxes (not all linings available with all units – check the price pages and catalogs). Some selection software accounts for lining options in preparing sound performance data, others do not.

- 1/2” 1.5lb Density (Std): Dual density fiberglass insulation with a tough outer layer.
- 1” 3lb Density: A thicker version of the above.
- Duct Board (Foil inside) w/Steel flanges (adds 7-10 NC to the 45K): High density duct board with a heavy foil facing on the exposed surface. Steel strips seal all edges.
- Double Wall, solid (adds 12 NC) and perforated (adds 8 NC to the 45K), 5 insulation options. The perforated is a 23% free area perforated sheet.
- Foil Encapsulated Glass - becoming obsolete in the industry, and no longer available on the 35E (0.5 and 1”) (Adds 7-10 NC to the 45K): This is a ‘pillow’ of light density fiberglass with a foil/scrim/kraft facing (duct liner) installed with the foil on the side exposed to the air stream. Edges are taped with foil tape and tucked under.
- Cellular (Adds 3-5 NC to the 45K): A closed cell plastic-type foam, meeting NFPA-90a and UL-181 characteristics. Note; this is not an ‘armaflex’ insulation.
- No liner (Adds 12 NC to the 45K): Unit must be externally wrapped to avoid condensation.

The effect of linings is different in each octave band, and must be carefully applied, by band, to a units reported sound power / pressure. The effect in the resultant NC is shown above for a quiet series flow fan terminal. The effect is quite different with a lower cost series flow fan terminal, due to differing base sound level spectra.

Because of the combined effects of diffuser, terminal, and ductwork noise generation, it may be difficult to pinpoint the source of noise in a space. The two sources of box noise (fan and VAV) are also different. The catalog data for airborne sound performance assumes the sound is transmitted to a diffuser outlet. This sound is reduced for "room effect", to calculate airborne NC.

Catalog radiated performance data is a measurement of the radiated energy of the box. Application NC values for both airborne and radiated sound includes further reduction factors. Not included in the assumptions are sound radiated by flexible and rigid duct sections. These radiated sounds are, however, included in the measured values when verifying sound levels in an installed installation. When comparing measured and predicted sound levels, these duct radiated sound levels seldom add significantly to the critical 125 Hz and 250 Hz octave bands measured underneath the terminal unit. The effect of varying the ceiling components has also been seen to be minimal.

## FIBERGLASS DUCT LINING

Many building owners are eliminating all exposed Fiberglass insulation in their buildings. While this will reduce the potential for moisture to become trapped, and will make systems easier to clean, fiberglass, or other similar materials, is the only thing we have to provide fire-safe sound attenuation of airborne HVAC noises. We know of several cases where excessive noise has resulted in a space from this practice.

Vinyl core, lined Flexible duct (used with discretion) can have a significant attenuation in a building. Installing this when the building is under construction is easy, but is complicated and expensive to add after complaints come in. An acoustician should be utilized in evaluating any design before eliminating duct linings.

Contrary to a commonly understood ‘fact’, there is no evidence that the Fiberglass insulation used in today’s buildings offers any long-term health hazard. It is not going to be, as we have heard often, ‘the next Asbestos’ threat. In fact, it was recently removed from the ISO list of “Potential Cancer Causing” substances. What is important, however, is to keep duct linings a clean as possible (change filters regularly) and above all, keep them dry. ASHRAE Standard 62.1 2004 prohibits exposed lining for a short distance downstream of any cooling coil. It will not prohibit exposed linings. There are millions of square feet of buildings operating just fine with fiber-lined ducts, and with no complaints.

Another problem coming to the surface is low frequency noise. This often results from poor duct configurations immediately downstream from an air handler. These low frequency noises can result in subtle stress on building occupants. Low frequency noises are suspected of causing a number of poorly understood complaints, and can result in an overall feeling of 'oppression', and is often overlooked is solving occupant complaints

## ACOUSTICAL SPECIFICATIONS

The specification of acoustical parameters is an important issue in the selection of VAV terminals. Certified octave band sound power data has been available since the 1980's through ARI Standard 880, originally released in 1981 (Updated in 1991, 1994 and 1998). Following ARI 880, an application standard, ARI 885, was developed, and released in 1990 (updated in 1998). This standard provides accepted methods of determining the path attenuation factors for estimating and specifying sound levels, both in the room (typically as NC), and at the source (as octave band sound power).

In the consulting engineer poll mentioned above, the engineers were questioned regarding awareness of the ARI 885 standard. Of those polled, only 5 actually had copies of the standard, which is available at no cost from the ARI website (www.ARI.org). Even today, specifications are frequently received requiring tests to the ADC test code (obsoleted in 1984 in favor of the ARI standard) or worse, per ASHRAE 36B, which was obsoleted in 1972. Many times, these specifications require a set room NC level based on one of the above obsolete test codes with no guidance on the acoustical assumptions to be employed in the analysis. These specifications often omit significant variables such as design inlet static pressure, which is critical to any analysis. Other specifications simply contain a favored manufacturer's published sound power, or worse, published estimated NC levels, which may be based on nothing that is specific to the design.

When no guidance is given, the supplier has the option of selecting whatever application factors favor his selection. As the engineer has probably been burned in the past with this approach, products are often specified at much lower sound levels than necessary. This results in oversizing of units and/or the addition of unnecessary silencers, which in turn results in poor operation, poor ventilation, excessive energy use and shortened motor life. Sound power should be used to compare products, and each octave band should be reviewed within the design parameters to insure the desired outcome.

## SUMMARY

In summary, a number of issues need to be understood in providing an acceptable indoor environment. These include a number of non-IAQ items that have a strong influence on perceived air quality. Issues include a need for occupant education and awareness of their own response to slight hot and cool environments ('Stuffy' & 'Drafty' are key terms). Don't fix the wrong thing. Occupant control of their environment is a major step forward. Don't worry about the energy spent in providing comfort; it is insignificant compared to salary costs (or the costs of what occupants will do to maintain comfort).

Codes need to be written and understood in a way that doesn't cause moisture and IAQ problems in buildings. Owners who eliminate insulation from ducts often get very noisy buildings. Fiberglass is seldom the problem; dirt and moisture often are. Moisture is the real enemy. Keep it dry!

Overall, we need to understand how our buildings operate. We need to train operators on what is happening, how occupants respond, and we need to design systems that can be understood. We are finding that cutting costs and saving energy can be very expensive.

As an industry, we have conducted significant research into the proper way to apply systems to buildings to maintain energy efficiency, first costs, comfort and productivity. These lessons have apparently been lost on many in the design community, as well as the agencies and politicians affecting the operation of buildings. The information is available, often in the ASHRAE Handbooks, and certainly in the body of ASHRAE sponsored research.

Manufacturers are being asked to provide products that we know will not perform when installed. Due to competitive pressures, we often have no choice but to meet the flawed specifications with products that will cause discomfort when applied as specified.

The mandatory setting of uncomfortable temperatures has repeatedly been shown to create reduced productivity, and to often actually increase energy consumption as occupants do what they can to maintain their comfort levels, thereby causing the systems to be operated in ways contrary to both their design and good sense.

How the situation looks, however, depends on the observer. Architects and Engineers have their own concerns on the design, many times not to the benefit of occupants. Rules of thumb are used to avoid costly analysis. Developers and contractors are concerned with first costs, and getting on to the next project. Occupants and owners, however, want safe comfortable spaces. And when there are problems, 'call in the lawyers'.