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Low temperature air, thermal comfort and indoor air quality

Cold air distribution can reduce installation and operating costs while allowing more comfort and increased energy savings

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The use of low temperature air in building systems has several economic advantages. These include both lower fan horsepower and reduced installed ductwork costs resulting from reduced ductwork size. However, there are several technical issues that have an effect on the final system cost. These include thermal comfort and indoor air quality issues.

Thermal comfort

The potential effects on occupant thermal comfort with cold air distribution systems are three-fold. First, there is an opportunity to reduce the relative humidity in the space, both because of the lower potential moisture carrying capacity of colder air, and because of the lower coil surface temperatures.

Second, with lower temperature supply air, diffuser selection will be quite different. The result will probably be more diffusers supplying less air per diffuser to achieve the same air distribution performance as a conventional system. As more diffusers are employed, the probability of improved temperature uniformity and comfort in a VAV system increases.

On the other hand, if standard diffusers are employed, the opportunity for non-uniform air distribution increases as a result of the lower discharge velocities and reduced induction rates.

Third, there is evidence that mean room air motion, at low delivery rates, is highly affected by room-discharge ΔT.

Humidity effects. The potential effect of humidity on comfort when using cold air systems is discussed by Berglund,[1] and can be calculated by a computer program.[2] This program shows that a 1°F (0.5°C) increase in dry-bulb temperature has the same effect on comfort as a 10°F (5.5°C) increase in dewpoint.

About the author

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Caution should be used in adjusting space setpoints to save energy, as occupant productivity costs are typically 100 times the potential energy savings. Lowered humidity has beneficial effects other than allowing a slightly higher temperature setting, including reduction in mold and/or mildew, reduced sensitivity to odors, and a greater feeling of "air freshness."[1]

From a computer program on thermal comfort, a graph has been prepared that shows the relationship between air temperature, relative humidity and 5% dissatisfied occupant comfort (see *Figure 1*). The calculations have input values for clothing, room air motion level and occupant activity level, as well as local temperatures.

Figure 1 shows that as humidity increases, the range of acceptable temperatures decreases and the absolute setpoint goes down as well. Notice that in this situation (as in most), the setpoint is nowhere near the 78°F (26°C) setting suggested as a means of energy savings. (Remember the EBTR rules in the late 1970s?) 1.0 Clo is a typical light suit; the activity level of 68 represents moderate office activity.

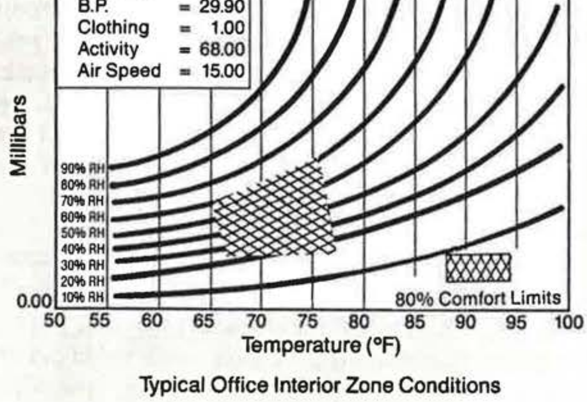

Figure 1. Thermal comfort limits.

Diffuser sizing. To provide good mixing of very cold air, the discharge velocity from the diffuser must be maximized. At the same time, the quantity of supply air is much lower than with conventional designs. These parameters, when combined with acoustical and pressure limitations, will often result in the application of shorter diffuser throws if standard rules-of-thumb are used for diffuser sizing and spacing. A proper selection will likely call for more smaller diffusers operating with a greater range of throws than typically seen with conventional systems.

The *ASHRAE Handbook—Fundamentals*[3] recommends that diffusers be selected on the basis of the ratio of throw (to 100 fpm, 0.5 m/s, for slot diffusers) to room length (or the distance to the next diffuser). This value should be between 0.3 and 1.5 for high induction slots. This means that the throw must go at least 1/3 of the way to the wall, or the next diffuser, at the lowest VAV design flow rate, to a 100 fpm (0.5 m/s) terminal velocity.

For example:

Throw at minimum flow = 6 ft (to 100 fpm)
room length = 18 ft
T(100) / L = 6/18 = 0.35

At maximum flow, sound is often the determining factor. Selecting a noise criteria slightly above the design sound level (such as an NC=37 versus a typical NC=35) may not be a problem. For example, when loads exceed design, the space will be warm and a noisy diffuser may reduce complaints ("At least the system is trying to maintain comfort.") In addition, throws should not exceed 1.5 times the distance to the wall (or next diffuser).

For example:

Throw at maximum flow = 22 ft (to 100 fpm)
room length = 18 ft
T(100) / L = 22/18 = 1.24

With cold air system designs, the delivery rates on a square foot basis are lower than a conventional system's range of 0.7 to 1.2 cfm (35.6 to 60.97 L/s per 3.6 to 6.1 mm/s) for interior zones. Research conducted on linear diffusers at low ΔT has shown that a well-designed and high-induction linear diffuser at less than 0.20 cfm (1 mm/s) is capable of achieving an Air Diffusion Performance Index (ADPI) of 100% in an office equipped with space dividers and typical interior zone office loads.

An ADPI greater than 80% is considered the minimum acceptable condition for most office environments.[4] ADPI is defined as the percentage of points within the occupied zone of a space with the comfort criteria of an airspeed less than 70 fpm (0.36 m/s) and a calculated draft temperature between −3° to 2°F (−19° to −17°C).

Draft temperature is calculated from an equation using temperature difference between each point and the room mean temperature, and a cooling factor based on that point's air speed. *ASHRAE Standard 113-1990* defines the measurement of these points.[5]

Single office selection is fairly straightforward, with one diffuser. Open offices are more complicated, with partitions, variable loads and multiple diffusers interacting on each other. In any case, at some point there is a danger of a mass of cold air falling into the space and causing discomfort (dumping). Research is underway to better determine when this is likely, based on diffuser performance, the type of heating load and the room to diffuser discharge ΔT.

High-induction diffusers (as yet undefined) rapidly mix room and supply air. They may avoid the problem by mixing enough room air to increase the temperature sufficiently to minimize the problem. The point of separation where a jet of air falls off the ceiling may be determined from a number of factors.[6] These include factors for air jet expansion, acceleration, mass and gravity.

Effect of ΔT on room air motion. Several detailed air motion studies were conducted in the late 1970s and early 1980s with two-dimensional diffuser arrays. These studies show that, at low air flow rates (less than 0.6 cfm/ft^2; 3 mm/s, the steady state mean room air speed is strongly affected by the room-discharge ΔT, all other things (except load, of course) being equal.

While not verified at very low flow rates, it seems that supplying air at colder temperatures should result in average room air motion levels similar to those found with higher conventional supply air temperature and higher volumes.

However, it should be noted that *ASHRAE Standard 55-1981*[7] states, "There is no minimum air motion for comfort." This assumes, of course, that all other parameters of the comfort equation are satisfied. This means that low air motion by itself is not an indicator or predictor for discomfort.

However, in practice, it is difficult to maintain temperature uniformity in the presence of heat loads with too low an air motion level. What seems to be desired is some feeling of air motion, but at a low level. As the mean space temperature rises, the desirability of air motion increases (see *Figure 2*).

Again, some assumptions are made about clothing, humidity and activity in calculating *Figure 2*, with winter clothing and slightly greater than sedentary conditions selected in this example. Also, the ideal temperature is well below the 78°F (26°C) setting that was mandatory in cooling zones in the late 1970s as an energy saving means. Energy savings cannot come at the expense of occupant comfort, especially if there is a competition between spaces for occupancy in the over-built office environment of today.

Air exchange efficiency and indoor air quality

The relationship between air diffusion performance and ventilation effectiveness is still not fully understood. One side of

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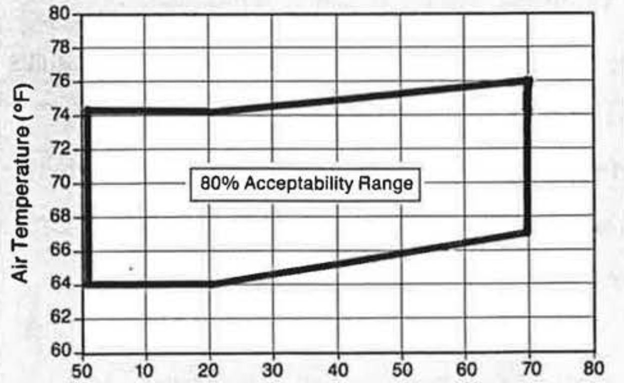

Figure 2. Air speed for comfort (1.1 Met, 35% RH, 1.2 Clo).

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the argument says that if the quantity of supply air can handle the load such that uniform temperatures are achieved, then ventilation mixing is also achieved. (It is easy to measure temperature uniformity, but difficult to verify ventilation mixing.)

The other side says that there is no necessary correlation between load and ventilation mixing, and that significant portions of ventilation air can short-circuit to system returns without mixing with room air.

Significant research is being conducted to evaluate this problem. ASHRAE-funded and other research programs are underway to predict room air movement using Computational Fluid Dynamic (CFD) techniques. One goal of CFD analysis is to attempt to predict room air mixing and thereby to predict ventilation effectiveness.

An ASHRAE standards project committee (SPC 129P) has been formed to develop a standard test method to evaluate ventilation effectiveness. When combined with the recently published *ASHRAE Standard 113-1990*,[5] much of the mystery will be solved. However, the limited data available so far has shown no cases of good ADPI and simultaneous poor room ventilation.

The predominant problem is not one of ventilation mixing, but of ventilation rate. Even with 100% mixing, when a VAV system senses a low load and reduces the total air supply rate to the room, the absolute quantity of outside (fresh) air reduces proportionally, depending on the system controls and setpoints.

With cold air systems, the total quantity of air is as little as 50% of that with conventional systems, for a given load. As lighting levels are reduced, the total quantity of air required to handle the load is lowered further, and the required proportion of fresh ventilation air increases.

ASHRAE Standard 62-1989[8] requires 20 cfm/person of ventilation air, in most circumstances. This rate (if 100% outside air and with 34° ΔT) provides:

$$\Delta T \times 1.08 \times \text{cfm} = \text{Btu/h, or}$$
$$34 \times 1.08 \times 20 = 734 \text{ Btu/h (cooling)}$$

As a person generates approximately 350 Btu/h of sensible heat, this leaves an additional capacity equivalent to 105 watts of load. (Loads are expressed here in watts, rather than Btu/h, so the loads can be analyzed in terms of watts/ft^2):

$$\begin{array}{l} 734 \text{ Btu/h} \\ \underline{-350} \\ 384 \text{ Btu/h excess } /(3.41 \times 1.08) \\ = 105 \text{ watts of extra capacity} \end{array}$$

With one person per 120 ft^2 (11 m^2; typical occupancy load) and 1.5 W/ft^2 of lighting, this requires an additional quantity of supply air:

$$\begin{array}{ll} 1.5 \times 120 & = 180 \text{ watts} \\ \times\ 3.41 \times 1.08 & = 663 \text{ Btu/h} \\ -\ 384 & = 278 \text{ Btu/h of additional cooling, or} \\ 278 / 34 / 1.08 & = 7.5 \text{ cfm} \end{array}$$

This additional 7.5 cfm (3.5 L/s) of supply air at 41°F (5°C) results in a total of 27 cfm per 120 ft^2 (13 L/s per 11 m^2), or 0.22 cfm/ft^2 (1.1 mm/s). This seems to be an absolute minimum for a design flow rate at this lighting load level. Should lighting loads get down to 1.0 W/ft^2 (10.8 W/m^2), the minimum becomes 0.17 cfm/ft^2 (0.86 mm/s).

This assumes no internal equipment load. As personal computers often draw 150 watts,[9] the VAV system will over-ventilate by the proportion of outside air in the additional cfm of cooling required for that device's load.

In theory, VAV systems should not cause ventilation shortages because the reduction in airflow corresponds to a reduction in load, which is typically due to variable occupancy. Assuming that ventilation rates are based on 100% design occupancy and if, in fact, occupancy is the variable load, and assuming that the thermostat senses the actual load, VAV systems provide ventilation proportional to occupancy, without re-heat requirements. This is a simple problem for interior spaces.

The problem comes with perimeter zones where the exterior building load will often offset interior loads, reducing cooling demand (and ventilation). Fan-powered terminals (which can provide constant airflow to the space) do not provide constant ventilation. The ventilation delivered is still proportional to the sensed cooling load, with the remainder coming from the ceiling plenum.

One solution to this problem is to ensure that the interior is over-ventilated, with the excess ventilation being drawn to the perimeter by the plenum-located fan terminals. This requires some assumptions about plenum circulation that may be difficult to verify.

If the minimum flow rate is set to worst-case ventilation minimums in perimeter zones, some re-heating will be required to maintain comfort (which is not allowed with many newer building codes). This is the quandary that must be solved by a combination of mechanical layout, plenum configuration, testing and control design.

Several methods can be employed to guarantee that minimum outside air quantities are maintained. These include air monitoring stations, interconnected supply and return dampers, pressure tracking systems, carbon dioxide monitoring, and other methods. (If the air is cold enough and if the loads permit, a system with 100% outside air has no measurement problems.)

Other factors in cold air systems

Balancing. The measurement method selected for determining the volume of air delivered by a diffuser can cause problems with building start-up. The flowrate can be determined by several methods, of which some compensate for density and some do not. Hot wire anemometers are mass flow sensors, while vane or pressure-type velometers are not.

When conducting evaluations, it is important that consistent flow rate measures and units be employed. If a DDC system is used to determine field system flow rates, the measurement means must be evaluated to determine the effect of density.

For systems delivering a designed 55°F (13°C) air (typically 59° to 60°F, or 15° to 16°C, at the diffuser), the flow rate difference between standard air and measured flow rate are not significant. However, with air at 42°F (6°C), the density induced flow rate error can approach 10%. These factors must be considered when optimizing flows to spaces.

Condensation and building start-up. The potential for condensation in terminal elements is a major concern with cold air delivery systems. There is seldom a problem when operating, as building humidities are lowered by the cold (and dry) air. However, on start-up, especially after a long weekend, there is a potential to have short-term, but significant condensation.

The expense of additional insulation and special construction must be weighed against the additional controls required to provide a soft-start of the building to bring humidity levels down before dropping distribution temperatures below normal (55 °F; 13 °C) levels.

Summary

Cold air distribution has the potential for reducing both installation and operating costs. If properly applied, it may allow more comfort at a lower energy cost. However, there is some new technology to be learned by engineers, operators and owners if comfort and productivity are to be maintained.

Some installed cost savings potentials may be offset by problems providing minimum quantities of fresh air. Only a careful understanding of all design issues can result in a successful and economical design. ■

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