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air distribution…

Cold Air Makes Good $ense

from the editor…
Since 1995, the Engineers Newsletter has examined various aspects of chilled-water-plant design. With technological advancements in refrigeration and controls providing the context, these articles have challenged flow rates for chilled water (ENEWS/28–3, 26–2), temperature control for cooling towers (ENEWS/24–1), and plant partitioning (ENEWS/28–4, 25–5).

But value-added, cost-saving designs are by no means confined to the waterside of an HVAC system. Rules-of-thumb for air distribution demand equal scrutiny. As evidence, this EN:

- *Identifies the effects of lowering the supply-air temperature by several degrees*
- *Outlines the application considerations that must be addressed to successfully design, install, and operate an HVAC system that supplies colder-than-55°F air*

Many choices in the design of comfort systems are predetermined by experience. Designers repeatedly choose to supply 42°F chilled water or 55°F supply air because they know that it works. The temperature rises that result from these choices directly impact the cost of pipes and ducts. Fan and pump horsepower are affected, too, extending the cost impact to the electrical distribution system. A successful HVAC design is as much a matter of cost savings as it is comfort.

Undoubtedly, lack of experience is what makes many engineers and contractors reluctant to consider HVAC designs that distribute cold air, despite the obvious benefits of doing so. "No one in our area uses cold air—I don't want to be the first … Don't you have to worry about condensation and moisture carryover? … How can I keep people comfortable if I'm 'dumping' cold air into the space? … Won't the system use more energy?"

Shedding a bit of light on the subject should help dispel these fears. Let's discuss what "cold air distribution" is, why it's worth your time (and is in your clients' best interest), and how it affects other aspects of system design.

How Cold Is "Cold"?

Cold-air distribution describes comfort-cooling applications that deliver supply air of 48°F or less to the occupied spaces in a building. Supply-air temperatures of 45°F to 48°F are most common, but applications with temperatures as cold as 42°F also exist. Successful installations throughout this spectrum demonstrate how readily cold-air distribution can be adapted to individual job requirements.

The appeal of cold-air distribution lies in the dramatic impact on the amount of air required for sensible cooling. As Table 1 suggests, simply lowering the supply-air temperature from the 55°F dictated by current practice to 45°F reduces the supply-air volume by 30 to 40 percent!

Table 1–Comparison of Airside Designs

	Conventional	Cold Air[a]
Supply air	55°F	45°F
Room set point	75°F	78°F
Room humidity	55–60% RH	40–45% RH
Cooling-coil ΔT	20°F	33°F
Airflow rate for space sensible cooling load	553 cfm/ton	335 cfm/ton

[a] This phenomenon is described by the equation, $Q = 1.085 \times cfm \times \Delta T$.

Benefits of Cold Air

Reducing the amount of supply air needed in an application triggers a series of related benefits, ranging from lower construction costs for both the building and the mechanical system to improved comfort, acoustics, and indoor air quality. Lower energy costs are possible, too, depending on the application.

Let's take a closer look at the opportunities extended by a 40-percent reduction in supply air volume:

- **Smaller air-handling equipment** lessens capital expense. The extra space made available by the smaller air-handler footprint can be used for attenuation in sound-sensitive applications or recovered—along with the space savings of **smaller vertical air shafts**—as usable/rentable floor space.

  How much space can cold-air distribution save? A typical air handler that requires 34,485 cfm for conventional, 55°F supply air needs only 20,900 cfm for cold, 45°F air. At 500 fpm, coil face area is reduced from more than 60 square feet to 40 square feet.

- **Smaller VAV terminals** ease tight installations, are less expensive, and quieter.
- **Smaller ductwork** means less sheet metal, easier installation, and more space above the ceiling for cable trays. Round duct also becomes a viable option, permitting slightly higher air velocities and further reducing duct size.
- **Shorter floor-to-floor height**, attributable to smaller ductwork, may significantly reduce the cost of glass and steel in a multistory building … perhaps even add a floor of rentable space.
- **Less fan horsepower** reduces the cost of the electrical installation and lowers operating costs for the life of the building, as shown in Table 2. It may also reduce fan-generated sound.

Table 2–Comparison of Supply Fans

Air temperature	55°F	45°F
Airflow volume	34,485 cfm	20,900 cfm
Total static pressure	4.0 in. wg	4.0 in. wg
Power consumption	37.9 bhp	23.0 bhp
Motor size	40 hp	25 hp

"Cold-air distribution is appealing because it dramatically reduces the air volume needed for sensible cooling."

What about electrical distribution costs? To maintain proper air movement in cold-air distribution systems, some designers add fan-powered VAV boxes to the interior zones. This addition also increases the number of electrical drops in the building. But don't forget that the **overall** ampacity of the electrical distribution system is reduced … and not just on the airside of the HVAC system. Cold-air distribution requires colder chilled water and deeper coils, which in turn, require lower chilled-water flow rates. Table 3 hints at the potential savings by recounting the actual reduction of installed power in a six-story, Denver office building.

Note: The reduction in fan and pump horsepower is attractive to electric utilities. Rebates offered for thermal-storage systems often apply to cold-air distribution, too.

Building environment. Other benefits that are directly attributable to cold supply air include better indoor air quality and improved comfort. Both improvements result from a lower relative humidity in the occupied space—55 to 65 percent for 55°F air versus 40 to 45 percent for 45°F air.

Obviously, lower relative humidity deters the growth of mold and mildew. Carpets, furniture, and other building materials last longer and are less likely to develop moisture-related odors.

The positive effect of cold-air designs on occupant comfort results from the common practice of raising the room set point by several degrees since there is less heat in the surrounding air. People who are more heavily dressed benefit from the lower relative humidity, while those who are more lightly attired appreciate the warmer space temperature.

Table 3–Comparison of Installed Power

System Component	Supply-Air Temperature	
	55°F	45°F
Air handler	171.3 kW	120.1 kW
VAV terminals	— kW	17.0 kW[a]
Pump	11.3 kW	6.5 kW
Chiller	135.0 kW	144.0 kW
Total	317.6 kW	287.6 kW, saving 10%

[a] Installed power is only included for the additional fan-powered VAV terminals used in this cold-air application when compared to a conventional system based on 55°F supply air.

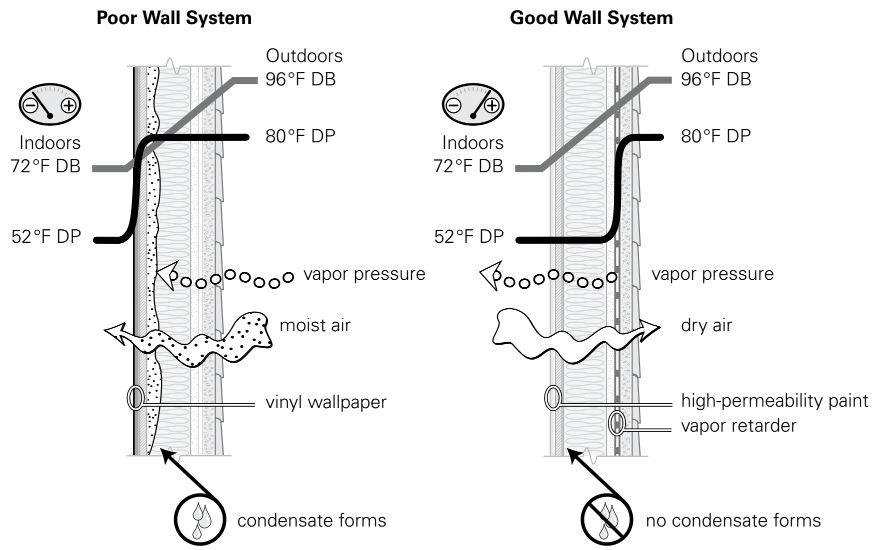
Impact on System Design

The checklist on page 6 provides a number of guidelines that address various aspects of cold-air distribution. Designers unfamiliar with this practice typically cite three concerns:

- condensation
- delivering cold air to comfort zones
- system energy consumption

Sound engineering practice is the critical ingredient in any successful cold-air comfort system. Rules-of-thumb simply do not apply when the objective is lower installed-costs and/or lower operating-costs.

Figure 1–Operation of Exterior Walls



Condensation. Uncontrolled condensation can be a problem in any building. Unwanted condensation in walls, plenums, fan rooms, or other areas of the building can lead to mold and mildew growth and the attendant problems with indoor air quality.

Cold air is an excellent tool to help solve unwanted moisture problems. Supplying comfort zones with 45°F-to-48°F air significantly lowers the room dew-point temperature. Buildings that employ cold-air systems enjoy the benefits of 40-to-45-percent relative humidity. One of these benefits is the potential reduction of unwanted condensation attributable to spaces with lower relative humidity. To realize this benefit, however, cold surfaces must be **inside** the humidity-controlled envelope. Any cold surfaces located **outside** this envelope must be completely insulated.

The humidity-controlled envelope ends where infiltration begins. Maintaining a slightly positive pressure in the space extends the humidity-controlled envelope to include exterior walls, and allows dry air to bathe cracks and wall cavities. Figure 1 illustrates the operation of an effective wall system. Buildings with cold-air distribution and positive space pressurization are typically healthy buildings.

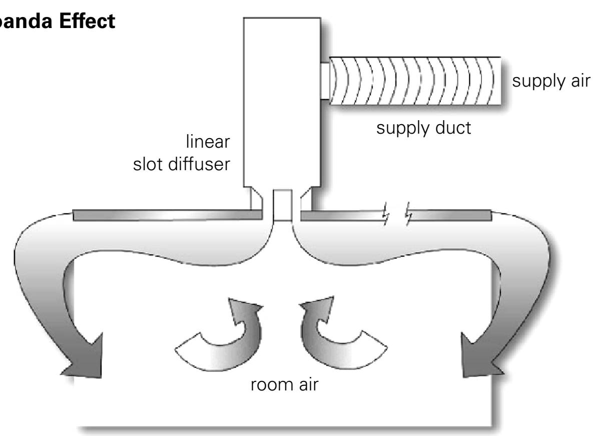
"Dumping" cold air. Designers typically use one of two approaches to address occupant comfort in cold-air designs:

- High-aspiration diffusers
- Fan-powered VAV terminals in all zones, with continuous fan operation during occupied hours

A diffuser design with a high aspiration ratio induces room air toward the supply diffuser. It also enhances cold-air systems by increasing both room-air motion and diffuser throw. For example, a linear slot diffuser recirculates 1 cfm of room air for each 1 cfm of supply air that it delivers.

The higher momentum (mass flow rate × velocity) of cold air increases the throw, or **coanda effect**, of aspirating-type diffusers; see Figure 2. Improved

Figure 2–Coanda Effect

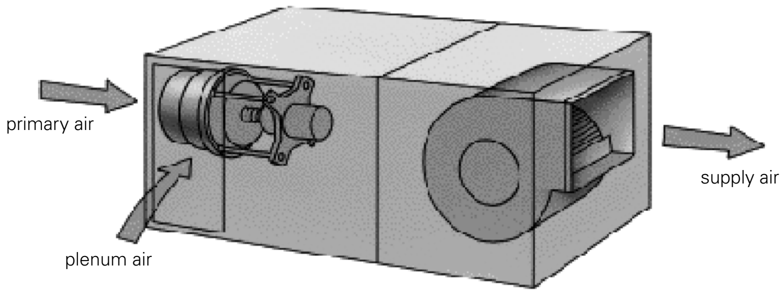


performance provides greater latitude for diffuser selection.

Note: Non-aspirating diffusers such as perforated plates or concentric grilles may not perform well in cold-air applications. If this type of device is used, couple it with fan-powered VAV terminals to blend the room air with cold supply air above the ceiling.

Fan-powered VAV terminals also avoid "dumping" cold air into the space. Series fan-powered VAV terminals (Figure 3) may be preferred for large conference rooms or other applications where constant airflow is desirable. Parallel fan-powered VAV terminals (Figure 4), on the other hand, are well suited for comfort zones where less air motion during off-peak conditions is preferred. With either terminal configuration, the air-blending fans run continuously during occupied hours.

Figure 3–Series Fan-Powered VAV Terminal



System energy consumption.

Cold-air systems require much less "transport" energy (energy used to pump water and move air) than conventional designs. The balance struck between first cost and operating cost determines the extent of the savings achieved; Table 4 summarizes representative savings. Annual savings in pump energy resulting from the use of colder chilled water typically range from 1¢ to 3¢ per square foot. Moving colder air usually cuts 6¢ to 8¢ per square foot from supply-fan energy costs, but savings in excess of 10¢ per square foot are not uncommon.

Not every component of a cold-air system contributes energy savings, however. Despite small individual wattages, the continuous operation of fan-powered VAV terminals, as air blenders, during occupied hours adds up. Energy costs for parallel terminals increase by as much as 1¢ to 1.5¢ per square foot … series terminals add more.

The costs of producing colder water and colder air require attention, too. Lowering the leaving chilled water temperature from 42°F to 38°F can increase chiller horsepower-per-ton by 6 to 10 percent, depending on compressor type. Maintaining a lower relative humidity in the building increases the amount of cooling required.

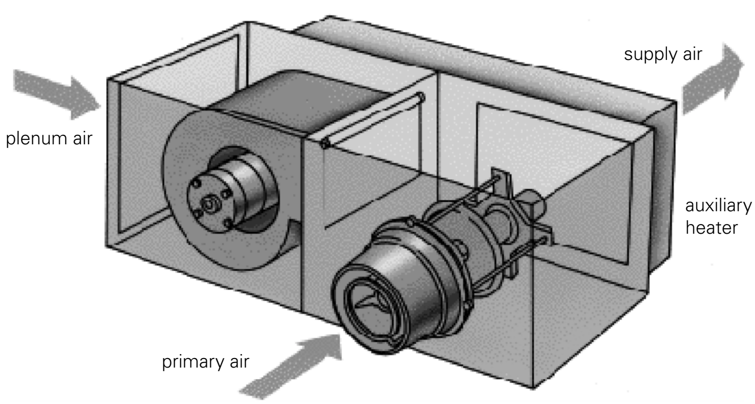
Offsetting these increases in chiller energy is the substantial and continuous reduction of heat generated by supply and return fans. (Reducing system airflow may be a reason to reconsider the need for return fans.)

The varying impact of cold-air designs on chiller energy reflects the complex relationship between building utilization, climate, airside design, and the intelligent reset of supply-air temperature. It is also a function of chilled-water-plant

Table 4–Typical Energy Savings

	Annual Savings Per Square Foot
Supply fans	6.0¢ to 8.5¢
Fan-powered VAV terminals	(1.2¢)
Chiller	(3.0¢) to 2.0¢
Pump	0.7¢ to 2.6¢
Reheat	(6.0¢) to (2.0¢)
Total	(3.5¢) to 9.9¢

Figure 4–Parallel Fan-Powered VAV Terminal



strategy, sometimes trading chiller energy for condenser-water-pump energy and cooling-tower energy. A low-flow, low-temperature, cold-air system that increases chiller energy by 2¢ to 3¢ per square foot in one application may reduce chiller energy by as much as 1¢ to 2¢ per square foot in another.

Zone reheat also deserves careful consideration. Cold air reduces the quantity of primary air sent to each zone, while the ventilation requirements of ASHRAE Standard 62 increase the minimum airflow required. A ventilation-reset control strategy can minimize the operating-cost impact of providing proper ventilation at part load by matching outdoor airflow to the actual ventilation required. Implementing such a strategy requires communicating controls throughout the system:

- VAV terminals with pressure-independent (DDC/VAV) controllers
- A building automation system with simple, equation-solving capability
- An air handler with a DDC controller and the means for sensing outdoor airflow and maintaining it at a set point

When properly controlled, the energy cost of providing additional reheat for a cold-air design need not exceed 2¢ to 5¢ per square foot.

(For more information, refer to ENEWS/27–1, "The Threefold Challenge of Ventilating Single-Duct VAV Systems." You can find it among the archived newsletters in the Commercial section of the Trane Web site.)

In aggregate, the net energy savings provided by cold-air distribution can be impressive. The reduction in fan horsepower is primarily responsible for reducing the energy cost of comfort cooling, but chilled-water-plant energy and reheat energy

Paradigms of Design

How did the design conventions of 42°F chilled water and 55°F supply air come to be? A 10°F ΔT for chilled water and a 20°F ΔT for the airside of the system are not advantageous for efficiency or material cost. Nor do these temperatures impart any preordained, thermodynamic advantage to the design or operation of air-conditioning systems. In fact, supply air several degrees colder than 55°F was not uncommon several decades ago. So how did 55°F become the norm in today's HVAC designs? There is no one right answer, but it may be useful to review a bit of HVAC history.

Before 1970, typical comfort systems included fan-coil, multizone, double-duct, and induction. The size of the building and the magnitude of its perimeter (or "skin") loads determined the choice of system.

Induction systems were often preferred for high-rise applications. They provided sufficient heating and cooling capacity to both offset a –30°F winter night and neutralize the solar load from large expanses of glass. Many of these applications used 40°F-to-45°F supply air; doing so permitted limited quantities of primary air to offset the high gains conducted through floor-to-ceiling glass … substantially reducing duct sizes.

The influence of VAV. Energy-conserving construction practices and materials—including less glass, better U-factors, and better shading coefficients—eliminated the radical load variations that induction systems addressed. It also paved the way for the "new kid on the block": variable-air-volume (VAV) designs that simply changed the volume of air introduced to the space to match the load. The VAV approach reduced system control costs; it required no more than a damper or air valve to vary zone airflow.

Like any new concept, VAV created a new set of design challenges. With a varying volume of supply air to the space, how would air motion be maintained? Would the difference in diffuser noise be noticeable? Heating with VAV also posed difficulties.

The rise of DX unitary. The 20 years that followed the energy crisis of the 1970s witnessed the explosive growth of commercial and light-commercial unitary air conditioners. These direct-expansion (DX) products adapted well to VAV because temperature control was centralized at the air handler. Controllers were developed to minimize compressor starts while maintaining the supply air temperature within an acceptable range, often by employing some type of supply-air reset strategy.

The "acceptable range" was defined by the performance characteristics of DX equipment. When the supply-air temperature exceeded 60°F, fan horsepower and fan noise increased noticeably, and loss of humidity control became more likely. At the same time, temperatures colder than 55°F more often led to frost formation on direct-expansion cooling coils, especially when coil air volume was reduced. How much did this phenomenon contribute to our present design conventions?

Perhaps it is less important to know how the paradigm of 55°F supply air came to be than to know whether that paradigm yields the best possible design on a case-by-case basis. ■

cannot be ignored. Intelligent system control is crucial to realize the potential energy savings.

A Final Thought

Distributing cold air (48°F or less) presents an excellent opportunity *and motive* to refine VAV-system designs. After all, air conditioning must compete with the dollars spent on glass, office furniture, communication systems, and other building technology. A comfort system that reduces building cost, reduces HVAC cost, and lowers energy cost while improving comfort and indoor air quality makes sense in today's competitive marketplace. ■

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To comment on this article, send a note to The Trane Company, Engineers Newsletter Editor, 3600 Pammel Creek Road, La Crosse, Wisconsin 54601; or e-mail us from www.trane.com. You can also find back issues of recent Engineers Newsletters on the Trane Web site.

Application Checklist for Cold-Air Distribution

Sound engineering and construction practices can yield significant cost savings for any comfort-system design, but the economic potential associated with cold supply air is particularly compelling.

This checklist identifies the basic attributes of an HVAC system designed for effective cold-air distribution.

For more information on this subject, be sure to consult ASHRAE publication 90396, *Cold Air Distribution System Guide*, and Trane publication SYS-AM-15, *Managing Building Moisture*. ■

Building construction

- Use a vapor retarder on the **warm** side of perimeter walls to minimize vapor-pressure diffusion
- Place all HVAC equipment within the vapor barrier
- Do not install the cold-air system in unconditioned spaces such as attics or plenums

Equipment room

- Duct outdoor air directly to the air handler—do not use the equipment room as a plenum
- Consider pressurizing and dehumidifying the equipment room with a small volume of supply air
- If the return air is unducted, use a return fan to pressurize the equipment room.
- Insulate and vapor-seal cold surfaces such as condensate drain pipes and chilled water pipes (leaving and entering)

Air handlers and ductwork

- Size the cooling coil to limit carryover
- Use dual-slope drain pans to prevent standing water
- Plan for proper trapping: the air handler must be mounted high enough above the floor to accommodate total trap height and depth
- Gasket all access panels, door openings, and inspection windows in positive-pressure sections
- Insulate all air-handler and supply-duct walls
- Specify sealing at all air-handler penetrations, including connections for coil piping and electrical service

Terminal devices

- Select linear slot diffusers with a high-aspiration ratio to provide proper air movement
- When using regular diffusers, install fan-powered VAV terminals in all zones
- Specify VAV terminals with gasketed panels and insulated surfaces

Intelligent control strategies

- Maintain positive building pressure in cooling climates to prevent infiltration
- Use set-point reset for supply-air temperature to minimize reheat
- Provide set-point reset for supply-air static pressure to minimize fan energy and improve zone control
- Operate fan-powered VAV terminals continuously during occupied periods, and when primary airflow is less than 20 to 30 percent of design

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