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Ceiling Panel Cooling Systems

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Ceiling panel radiant cooling has been refined and used successfully in Europe for more than 15 years. Global consulting firms, and their European engineers, have been frustrated when presenting the option of panel cooling to U.S. engineers, architects, contractors and owners. The three most common reasons in the United States for dismissing panel cooling are condensation, capacity, and first cost.

This article analyzes the reasons given for not using panel cooling to see if they are still warranted when panel cooling is used with dedicated outdoor air systems (DOAS).

Overview of Ceiling Panel Cooling

Ceiling cooling panels are built as an architectural finish product with necessary acoustical qualities, color, and pattern. They are compatible with the traditional drop ceiling "Tee grid" system, or as a free hanging element. Widths are generally 2 ft (0.6 m). Lengths vary from 2 ft to 12 ft (0.6 to 3.7 m) or more.

For cooling applications, the heat flux to the panel surface is in the range of 30 Btu/h-ft² (95 W/m²) for drop ceiling applications, and about twice that for free hanging designs. As a result, the aluminum absorber surface is only about 22 gage (0.76 mm), and the thermally bonded copper cooling water piping is generally ½ in. (12 mm) in diameter or less, and on about 6 in. (150 mm) centers. Panel piping arrangements are generally in a serpentine pattern. However, parallel header arrangements are available on request.

Typical panel construction is illustrated in *Figure 1*. As installed, the "drop-

in" ceiling cooling panels weigh 0.96 lb/ft² (4.7 kg/m²) while the conventional 0.875 in. (22 mm) thick mineral fiber acoustical tile that they replace weigh 1.15 lb/ft² (5.6 kg/m²). The lightweight construction results in a transient response "time constant" of only about three to five minutes. That means they respond rapidly to changing space sensible load conditions.

Hydraulically, the ceiling panels are frequently connected with flexible push-on coupling hoses for fast and safe installation. The couplings also permit quick removal for easy office-scape revisions.

Panel Heat Transfer. Sensible heat is removed from the space by a combination of convection and radiation. In most applications, the heat removed by each of the two mechanisms is roughly equal, and governed by the differential between the panel mean temperature and the enclosure mean temperature.¹

Under favorable conditions, the "drop-in" panel with back insulation can remove 32 Btu/h-ft² (101 W/m²). The free hanging design can remove 62 Btu/h-ft² (196 W/m²).

Principal advantages of a chilled ceiling approach. The 16 radiant cooling advantages discussed in the general

evaluation section of the *2000 ASHRAE Handbook—HVAC Systems and Equipment*² strongly support the application of this technology.

Moisture Condensation Issues

Because of the potential for condensation, it is *critical* to understand panel cooling cannot be considered unless a parallel system is in place to decouple the space sensible and latent loads. The author recommends^{3, 4, 5} using a DOAS parallel system capable of removing all of the space latent loads, thus achieving the required load decoupling.

ANSI/ASHRAE/IESNA Standard 90-1999, *Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings*,⁶ Section 6.3.6.1 Exhaust Air Energy Recovery, addresses the requirements for total energy recovery in the DOAS. The required heat recovery can reduce the OA load on the cooling coil consistently by 75–80%. By comparison, an ordinary all-air system using demand-controlled ventilation will only provide a comparable reduction in OA load when the occupancy has dropped below 25%.

The resulting integrated panel cooling/DOAS mechanical system (some might refer to it as a hybrid⁷ system) also offers the potential to generate Green Building Leadership in Energy and Envi-

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ronmental Design (LEED)⁸ rating points. Points are accumulated in five of the major categories (i.e., water efficiency, energy and atmosphere, materials and resources, indoor environmental quality, and LEED innovation credits). The panel cooling/DOAS approach can generate up to 23 LEED points, or up to 88% of the minimum points needed for certification.

Steady-State Condensation Formation. Using the DOAS to decouple the space loads at design does not guarantee that some spaces will not have transient occupancies exceeding design. The extra occupant latent load generation has the potential to create a condensation problem. To understand the nature of this potential problem, let us explore the steady-state rate of moisture condensation in a typical office situation. Let us make the following assumptions:

- Ventilation airflow rate and thermodynamic state point remains constant at the design conditions;
- Occupant latent load, 205 Btu/h-person (60 W/person);
- Infiltration negligible;
- Occupancy exceeds design by 100% (i.e., two people, when one was used in design);
- Enclosure and contents are completely non-hygroscopic (i.e., they do not participate in the moisture transients [an extremely conservative assumption]);
- Typical panel area per person at design occupancy is 70 ft² (6.5 m²). This assumes seven people per 1000 ft² (93 m²) and a 50% chilled ceiling fill factor; and
- Chilled ceiling panel temperature is uniform overall.

Under these assumed conditions, the occupant generates less than 0.2 lbm/h (0.025 g/s) of water vapor. When uniformly distributed over 70 ft² (6.5 m²) of panel per person, the water thickness after one hour is 5/10,000 of an inch (0.013 mm). For reference, a human hair ranges in diameter from 0.0007 to 0.007 in. (0.017 mm to 0.177 mm) in diameter. Under these conservative steady-state moisture condensation assumptions, it would take one person's latent generation 90 minutes to 14 hours for the condensation thickness to equal the diameter of a human hair.

If space permitted, it could be shown that with a 3°F (1.7°C) difference between the panel inlet temperature and the space dew-point temperature (DPT), it takes nearly an hour for condensation to commence when the design occupancy is doubled.

Condensation also could result from envelope integrity problems. However, if the structure is confirmed to be in compliance with Standard 90.1-1999, Section 5.2.3, Envelope Air Leakage, that should not be a problem.

An extreme case was investigated under steady boundary conditions. In the experiment, a portion of a panel surface temperature was chilled 14°F (8°C) below the conditioned space DPT. The balance of the panel was not chilled. Moving away from the chilled portion of the panel, the surface temperature increased to and above the space DPT. After 8.5 hours, the condensation beads under the chilled section, illustrated in *Figure 2* to the right of the 14°F (8°C) line, grew full. Under these extreme conditions of temperature differential and time, the beads of water did not grow large enough to release one drop of water.

The portion of *Figure 2* between 0°F and 14°F (0°C and 8°C) represents that portion of the panel that was not chilled. There the panel temperature was decreasing from the space

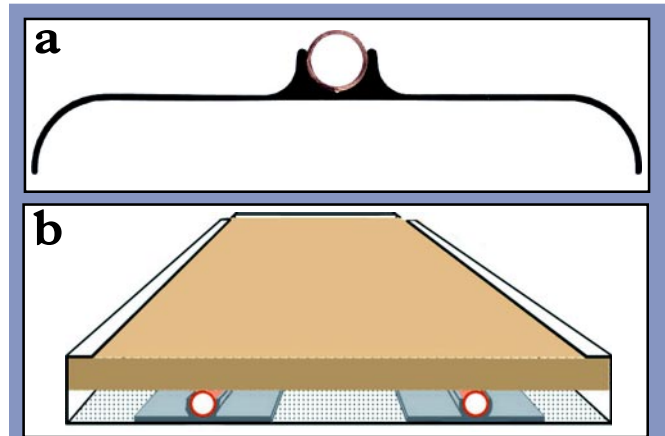


Figure 1: Typical ceiling panel construction: a) free hanging panel design for enhanced upper surface HT; b) "drop-in" panel with back insulation and acoustical perforation.

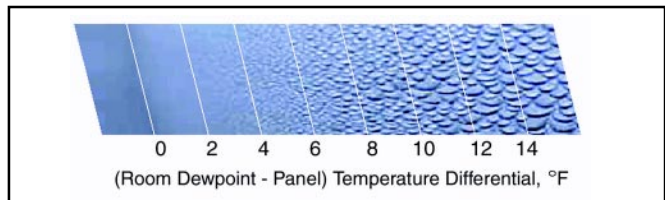


Figure 2: Condensation formation on a panel after 8.5 hours.

dew point temperature at the 0°F (0°C) line over to the 14°F (8°C) line. Not much condensation is observed on the panel, even after 8.5 hours, for panel temperatures between 0°F and 8°F (0°C and 4°C) colder than the space DPT.

The onset of condensation occurs slowly, and can be easily avoided when the DOAS and panel loop temperature controls are operating correctly. In the event the controls fail, there are simple control based safety remedies, which are:

- Monitor the space dew point temperature and reset the panel coolant low limit temperature above the space DPT. This may have a negative impact on the panel capacity and require attention—a good thing.
- Place a water sensor, consisting of a normally closed switch and an element that swells when dampened, under the vertical inlet piping (so the condensate can fall directly onto the water sensor) to the first panel for each group controlled by a normally closed spring return modulating control valve. The switch in the sensor is wired in series with the control signal to the modulating control valve. When water droplets fall on the sensor element, it swells, pushing the normally closed switch open, thereby breaking the control signal to the control valve. The spring return (normally closed control valve) closes, thus isolating the panels from the source of cold water. Of course, panel cooling in that space will cease, and the occupants will demand corrective action—also a good thing.

Cooling Capacity

Experienced design engineers are exceptionally skilled at bounding problems with rules developed from experience. This

skill has avoided many problems in the past, and works particularly well when applied to an area where significant experience exists. However, when experience based rules are applied to an unfamiliar technology, such as chilled ceilings working in parallel with DOAS's, caution is necessary.

A universal rule is: 300–400 ft²/ton (7.9–10.6 m²/kW) of cooling. Clearly, this rule requires that 40–30 Btu/h·ft² (126–95 W/m²), of total heat energy be removed. Since “drop-in” panel cooling has a sensible heat removal capacity of around 30 Btu/h·ft² (95 W/m²), rule based thinkers conclude that panel cooling simply could not meet the loads, even with the ceiling 100% filled with cooling panels.

However, using another universal industry rule, i.e., 1 cfm/ft² (5.08 L/s·m²) for all-air VAV systems operating with 55°F (12.8°C) supply air, is a certain tip off that the space loads must be less than 40 Btu/h·ft² (126 W/m²). Clearly, at design under the 1 cfm/ft² (5.08 L/s·m²) rule, an all-air VAV system could only remove about 20 Btu/h·ft² (63 W/m²) of space sensible heat (panels can only be allowed to remove sensible heat) when the design space temperature is 75°F (23.9°C).

Dividing the building floor area by the design chiller load yields the 300–400 ft²/ton (7.9–10.6 m²/kW) rule. The design chiller load consists of the following three load components:

- The total OA load which is: $\dot{m}_{oa} \times (h_{oa} - h_{ret.air})$
Note: it is not $\dot{m}_{oa} \times (h_{oa} - h_{supplyair})$.
- The space sensible load (illumination, equipment, building envelope, and the sensible portion of the occupant loads).
- The space latent load (primarily the occupants, infiltration, and perhaps coffeepots and live plants).

The only portion of the design chiller load contributed by the ceiling panel cooling system is the space sensible load (all or a portion of it). The balance of the design chiller load comes from the DOAS, used in parallel with the panel cooling, which must be designed to remove the total OA load and the entire space latent load. If the OA is supplied with a dry bulb temperature equal to the required supply air DPT, it can remove 5–6 Btu/h·ft² (16–19 W/m²) of the space sensible load. As a result, the sensible load actually remaining for the panel-cooling system is, on average, only 14–15 Btu/h·ft² (44–47 W/m²) (or about 50% ceiling fill factor). From this perspective, it should be clear that there is no capacity problem with panel cooling when properly applied with a parallel DOAS.

Economic Issues

The integrated panel-cooling/DOAS approach provides superior indoor air quality and thermal comfort,⁹ and that alone should be sufficient incentive for the industry to use the concept. This is especially true since the Lawrence Berkeley National Laboratory¹⁰ estimates that U.S. companies could save as much as \$58 billion annually by preventing sick-building illnesses and could benefit from up to \$200 billion in productivity increases each year. However, it is well recognized that these issues are not always sufficiently compelling to motivate prospective building investors.

Investors generally expect to realize at least a first-cost benefit. Operating cost savings are an added benefit, but they have rarely been a major factor in the decision making pro-

cess. Therefore, a careful first-cost analysis is necessary to justify this design approach.

First Cost of the Panel Cooling/DOAS System. With the integrated panel cooling/DOAS approach compared to a conventional all-air VAV system, mechanical as well as building cost issues arise. The major items are identified below:

1. The building design chiller size is reduced due to the need for less OA and the use of total energy (sensible and latent) recovery in the DOAS;
2. Pump size is smaller due to chiller size reduction;
3. The first cost of the ductwork and associated terminal units is greatly reduced since the DOAS airflow rate is only about 15 to 20% that of an all-air VAV system;
4. Plenum depth can be reduced in new construction due to the smaller ductwork and elimination of terminal VAV boxes. The first cost savings potential can be significant due to reduced materials used for the building enclosure, i.e., interior partitions, structural, plumbing, fire protection, and vertical transportation;
5. Air-handling unit size reduction;
6. Electrical service reduction for the mechanical equipment due to smaller chiller, fans and pumps;
7. Piping material reduction due to functional integration of the thermal and fire suppression transport systems;¹¹
8. Acoustical ceiling panels reduction where replaced with the cooling panels; and,
9. Less lost rentable space due to mechanical shaft reduction as a result of much lower air volumes.

On the other hand, the panels contribute to the first cost of the panel-cooling/DOAS system.

A first cost analysis¹² for a six-story, 31,000 ft² (2883 m²) per floor, office building (in compliance with Standard 90.1-1999) in Philadelphia is presented in *Table 1*. This cost analysis shows that there is a net first cost savings of \$2/ft² (\$22/m²) with the panel cooling/DOAS over a conventional all-air VAV system. The brick facade building has a footprint of 125 by 250 ft (38 by 76 m) with the long axis oriented in an E-W direction. The wall U value is 0.044 Btu/h·ft²·F (0.25 W/m²·C), the roof U value is 0.03 Btu/h·ft²·F (0.17 W/m²·C), and the glazing U value and shading coefficients are 0.48 Btu/h·ft²·F (2.73 W/m²·C) and 0.365 respectively. The glazing constitutes 27% of the building wall area. As for internal generation, the occupancy density is seven people per 1,000 ft² (93 m²), the overhead lighting is 1.3 W/ft² (14 W/m²), task lighting is 0.7 W/ft² (7.5 W/m²), and equipment plug loads are 2 W/ft² (21.5 W/m²).

Minimizing First Cost of the Chilled Ceiling Panel Array. Currently, cooling panels manufactured and shipped to the United States from abroad, are relatively expensive (approximately \$13/ft² [\$140/m²] of panel installed) as perceived by the building industry. For panels to be a financially attractive investment, six specific steps are recommended to either minimize the sensible load they bear, or to enhance their thermal performance.

1. Hold the space dew point temperature as low as is practical and healthy, i.e., around 52°F (11°C);
2. Use a supply air temperature equal to the required supply air DPT, i.e., approximately 45°F (7°C);
3. Supply the DOAS air to the space via high aspiration diffusers.¹³ This air delivery method increases the convective

heat transfer to the panels by about 15% more than panels operating in still air;¹⁴

4. Use long panels, greater than 2 ft (0.6 m), (it's better to use 8–10 ft [2.43–3.04 m] long or more to minimize installation handling and the number of connections);

5. Long panels require parallel header rather than serpentine piping, resulting in more uniform panel temperature, and lower pumping costs;

6. Functional integration of the thermal transport and fire suppression piping.¹¹

Operating Cost Issues

An hourly energy analysis, for 12 hours a day, 5 days per week, was performed for the 186,000 ft² (17 300 m²) building located in Philadelphia. An all-air VAV system and the panel cooling/DOAS system were analyzed. This analysis was performed using existing load and energy analysis software.¹⁵ When the DOAS uses an enthalpy wheel and a cooling coil, as is the case for this example, existing software can be used. In the air system menu of the program, the single wheel problem is set up with a common ventilation system using total energy recovery and supplying the air at 44°F (7°C).

A parallel fan coil system with a supply air temperature above the resulting space DPT, to make sure only sensible cooling is done, is also specified. The fan coil fan pressure drop is set to zero, and the panel pumping head is included in the hydronic specifications. This approach serves as a good model for summer operation. However, it may not work well to model winter humidification, if it is to be accomplished by modulating the speed of the enthalpy wheel, since that is not a feature of current software. Needless to say, more work on design tool development is a critical need. The utility rates for this analysis are:

Energy Charges:

- Demand Block 1: 200 kWh/kW, \$0.065/kWh
 - Demand Block 2: 200 kWh/kW, \$0.052/kWh
 - Demand Block 3: remaining kWh, \$0.05/kWh
- Demand Charge: \$6.94/kW

The results of the simulations are presented in Table 2. Like the first cost analysis, the operating cost data also favor the panel cooling/DOAS system. The mechanical system annual operating cost savings is \$17,620 or about \$0.10/ft²-yr (\$1.07/m²-yr). Due to the smaller mechanical plant for the panel cooling/DOAS, the building demand charges were smaller, resulting in an annual building operating cost savings of \$26,349, or about \$0.15/ft²-yr (\$1.60/m²-yr). It cost about 29% more to operate a conventional VAV system each year than the panel cooling/DOAS system.

Conclusions and Recommendations

This article explored the primary concerns expressed by the building industry about panel cooling; condensation, capac-

Cost Item	Unit Cost	Units VAV	Units Panel-Cooling/DOAS	Cost Savings \$
Chiller	\$1,000/ton (\$284/kW)	506 ton (1780 kW)	306 ton (1076 kW)	200,000
Chilled Water Pump	\$25/gpm (\$400/L/s)	1215 gpm (76.5 L/s)	737 gpm (46.4 L/s)	11,950
Ductwork	\$1/ft ² (\$11/m ²) DOAS \$4/ft ² (\$43/m ²) VAV	186,000 ft ² (17 300 m ²)	186,000 ft ² (17 300 m ²)	558,000
AHU	\$2/cfm (\$4.25/L/s) VAV \$4/cfm (\$8.50/L/s) DOAS	135,000 cfm (73 720 L/s)	25,000 cfm (11 800 L/s) 100% Ventilation Air	170,000
Electrical Service	\$50/kW	630 kW	372 kW	12,400
Facade/Partitions	\$35/ft ² (\$376/m ²) of Facade	No Depth Reduction	1 ft (0.3 m) Plenum Depth/Floor or 4308 ft ² (400 m ²)	150,780
Integrated Thermal and Fire Suppression Piping	\$0.65/ft ² (\$7/m ²) Savings	N/A	186,000 ft ² (17 300 m ²)	120,900
Drop Ceiling	\$1.50/ft ² (\$16/m ²)	N/A	79,200 ft ² (7365 m ²)	118,800
Mechanical Shaft Impact on Lost Rental Space	\$125/ft ² (\$1,344/m ²)	N/A	500 ft ² (47 m ²) Saved	62,500
Savings Subtotal				1,405,300
Panel	\$13/ft ² (\$140/m ²) of Panel	N/A	79,200 ft ² (7365 m ²)	-1,029,600
Net Savings				375,700 or 2/ft² (22/m²)

Table 1: First cost comparison of the panel-cooling/DOAS vs. a conventional all-air VAV system serving a six-story 186,000 ft² (17 300 m²) building in Philadelphia.

System	Annual Mechanical Operating Cost	Annual Total Mechanical, Illumination, and Equipment Operating Cost
VAV	\$77,350	\$299,914
Panel-Cooling/DOAS	\$59,730	\$273,565
Annual Savings	\$17,620	\$26,349
Annual Savings: \$/ft ² (\$/m ²)	\$0.10 (\$1.10)	\$0.15 (\$1.60)
Annual Cost Ratio, VAV/(Panel-Cooling/DOAS)	1.29	1.10

Table 2: Operating cost comparison, panel-cooling/DOAS system vs. a standard VAV system.

ity, and cost. It has been demonstrated that when a DOAS is used to decouple the space sensible and latent loads, the panels are left with only a portion of the space sensible loads. And if the occupancy exceeds design by a factor of 2 or 3, it could take hours for the condensation thickness to equal the diameter of a human hair. Control measures necessary to prevent condensate dripping were also discussed. It may safely be concluded that condensation can easily be avoided, and it must be for aesthetic as well as IAQ reasons.

The capacity concern was addressed by considering the rules of thumb engineers have used to question the panel cooling capacity. The article illustrates that a large percentage of the design

chiller load is a result of the OA load and the space latent loads. Further it was demonstrated that with low ventilation air supply temperatures, only a portion of the space sensible loads fall onto the panels. In conclusion, panel cooling can meet its capacity duty, and only use about 50% of the ceiling in most cases.

As for economic concerns, through a careful first and operating cost analysis of both conventional all-air VAV systems and the panel cooling/DOAS systems, that concern has been dismissed. The root of the concern came from the notion that there

were capacity problems, and to meet it the building ceiling and walls required panels. This is simply not the case.

The three concerns addressed in this article cannot be used as an excuse to reject panel cooling when properly designed. This approach holds great potential not only for first and operating cost advantages, but also improved IAQ and thermal comfort. The environmental improvements will enhance the productivity of the folks working in the buildings and if universally applied potentially save U.S. companies up to \$258 billion/yr.

Advertisement in the print edition formerly in this space.

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