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Chilled Ceilings in Parallel with Dedicated Outdoor Air Systems: Addressing the Concerns of Condensation, Capacity, and Cost

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ABSTRACT

Emphasis is placed upon the integration of chilled ceiling/ceiling radiant cooling panel technology with other building mechanical systems in this paper. Applicable radiative and convective heat transfer equations are applied to illustrate the rates of heat removal that are representative of this technology. Also explored are the comfort advantages of radiant cooling and space design dry-bulb temperature criteria. The issue of potential steady state and transient condensation on the exposed 55-60°F (12.8-15.6°C) panel and supply piping surfaces is analyzed. Also explored, in light of ASHRAE standards 62 [IAQ] and 90.1 [energy], are dedicated outdoor air preconditioning requirements that enable decoupling of the space sensible and latent loads, thus eliminating both condensation and capacity concerns. Finally, the economic issues related to the first and operating cost of the integrated system compared to conventional all-air VAV systems are explored. It is a recognized fact that building investment decisions are based almost entirely on first cost in the United States, and if this technology is to blossom in that marketplace, victory over first costs must be achieved. The paper concludes that technical and economic barriers do not currently exist to inhibit the widespread application of ceiling radiant cooling panels when integrated with dedicated outdoor air systems.

INTRODUCTION

Ceiling radiant cooling has failed to appeal to the majority of those practicing in the United States consulting engineering community to date. While this technology has been refined and successfully utilized in Europe for over 15 years, global consulting firms, and their European engineers (and ASHRAE colleagues), have almost universally been met with frustration when the concept of radiant cooling has been presented to U.S. engineers, architects, contractors, and owners. The three most common major concerns given in the U.S. for dismissing radiant cooling out of hand (referred to here as the dreaded 3-**Cs**) are:

- **C**ondensation concerns,
- cooling **C**apacity doubts and concerns, and
- first **C**ost penalty concerns compared to conventional systems.

The central thrust of this paper will be to explore the reasons given for not utilizing radiant cooling systems, i.e., condensation, capacity, and cost, in sufficient detail to see if they are warranted when integrated with dedicated outdoor air systems (DOAS).

Overview of Ceiling Radiant Cooling/Chilled Ceilings

Ceiling radiant cooling panels (CRCP) are generally built as an architectural finish product (with necessary acoustical qualities, color, and pattern), compatible with the traditional drop ceiling “tee grid” system or as a free hanging element. Widths are generally 2 ft (0.608 m), and lengths can vary from 2 ft to 12 ft (0.61 to 3.65 m) or more. For cooling applications, the heat flux to the panel surface is in the 30 Btu/h·ft^2 (95 W/m^2) range for drop ceiling applications and about twice that for the free hanging designs (this heat flux rate will be discussed in more detail later in the paper). As a result, the aluminum absorber surface is only about 22 ga. (0.76 mm), and the thermally bonded copper cooling water piping is generally 1/2 in. (1.27 cm) in diameter or less and on about 6 in. (15 cm) centers. Panel piping arrangements are generally in a serpen-

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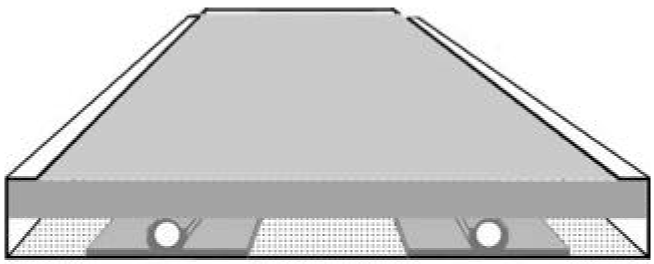
(a) Free hanging panel design for enhanced upper surface HT



(b) "Drop in" panel with back insulation

Figure 1 *Typical ceiling radiant panel construction.*

tine pattern; however, parallel header arrangements are also available on request. Typical panel construction is illustrated in Figure 1. As installed, the "drop in" radiant panels weigh 1.61 lb/ft^2 (7.8 kg/m^2) while the conventional 7/8-in. (2.2 cm) thick mineral fiber acoustical tile that they replace weigh 1.15 lb/ft^2 (5.6 kg/m^2) (Manufacturer's catalogs 1 and 2, 2001). The lightweight construction results in a transient response "time constant" of only about 3-5 minutes. That means they respond rapidly to changing space sensible load conditions.

Hydraulically, the ceiling panels are most frequently connected with flexible-push on coupling hoses for fast and safe installations, as illustrated in Figure 2. Panels can be moved aside without disconnecting the hoses, for easy access above. They can also be easily removed and reconnected for either extensive maintenance or evolving space use requirements without breaking normal threaded or sweat solder plumbing connections.

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, governed by the differential between the panel mean temperature and the enclosure mean temperature.

The radiant heat transfer is governed by the Stefan-Boltzmann equation. For most building enclosure cases encountered in practice, the enclosure emittances are about 0.9, and the view factor between the ceiling and the balance of the enclosure is at least 0.87. When these common values are placed into the Stefan-Boltzmann equation, the following equation emerges (ASHRAE 2000, page 6.2, equation 5):

$$q_r = 0.15 \times 10^{-8}[(t_p)^4 - (AUST)^4] \qquad (1)$$

Figure 2 *Hose with quick connect push-on coupling.*

where

q_r = radiant cooling, Btu/h·ft^2 (W/m^2);

t_p = mean panel surface temperature, °R (K);

AUST = area weighted average temperature of the non-radiant panel surfaces of the room, °R (K).

The rate of heat transfer by convection is a combination of natural and forced convection. Natural convection results from the cooled air in the boundary layer just below the panels being displaced by warmer air in the room. This natural process can be altered or even changed to forced convection by infiltration, human activity, and the mechanical ventilation systems. Research suggests that for practical panel cooling applications without forced convection, the cooling natural convection heat transfer is given by the following equation (ASHRAE 2000, page 6.4 equation 10):

$$q_c = 0.31|t_p - t_a|^{0.31}(t_p - t_a) \qquad (2)$$

The equations necessary to determine the panel mean temperature are presented in an ASHRAE paper, "Ceiling Radiant Cooling Panels as a Viable Distributed Parallel Sensible Cooling Technology Integrated with Dedicated Outdoor-Air Systems" (Conroy and Mumma 2001). In addition, many of the European ceiling radiant panel manufacturers offer spreadsheet design tools to aid in estimating the radiant panel thermal performance. An example is presented in Table 1.

The example presented in Table 1 assumes that the design space air temperature (not the mean radiant temperature) is 78°F (25.6°C) (justification for this slightly elevated thermostat setting will be addressed later). It also assumes that the space dew-point temperature is maintained below the water inlet temperature of 53°F (11.7°C) (a topic of discussion later in the paper). Finally it was assumed that the panel flow rate would be adjusted to maintain no more than a 5°F (2.8°C) water temperature rise through the panel at design load conditions. It is also possible to specify asymmetrical loads, which contribute to an increase in the panel heat removal capacity.

TABLE 1
Typical Results from Manufacturer Software

Variable	Input value	Output value Drop-in w/Back Insulation	Output value Free Hanging w/o Back Insulation
Room air temperature, F (C)	78 (25.6)		
Water inlet temperature, F (C)	53 (11.7)		
Water outlet temperature, F (C)	58 (14.4)		
Net cooling cap, Btu/hr-ft^2 (W/m^2)		34.47 (108.75)	66.91 (211.1)
Asymmetric load	yes [no]		
Hot window surface	yes [no]		
Panel height above floor, ft. (m)	9 (2.74)		
Asymmetric load received, Btu/hr-ft^2 (W/m^2)		1.32 (4.16) [0]	2.55 (8.04) [0]
Hot window surface load received, Btu/hr-ft^2 (W/m^2)		2.2 (6.9) [0]	4.25 (13.4) [0]
Total capacity, Btu/hr-ft^2 (W/m^2)		35.2 (111) [31.9 (100.6)]	68.3 (215.5) [62.0 (195.6)]
Water flow, gpm/ft^2 (l/s-m^2)		0.015 (0.013) [0.010 (0.009)]	0.027 (0.025) [0.018 (0.017)]
Key: Items in [] *are for internal spaces with no radiant asymmetry or hot windows*			

The asymmetrical heat removal calculations are only approximate, since the spatial geometry required to accurately compute the respective radiant view factors is not requested or specified. In any event the “drop in” panel with back insulation and radiant asymmetry is capable of removing 35.2 Btu/h·ft^2 (111 W/m^2), while the free hanging design can remove 68.3 Btu/h·ft^2 (215.5 W/m^2). For most internal zones, where minimal radiant asymmetry would occur, the “drop in” panel with back insulation has a heat removal capacity of 31.9 Btu/h·ft^2 (100.6 W/m^2), while the free hanging design can remove 62.0 Btu/h·ft^2 (195.6 W/m^2).

Principal Advantages of a Chilled Ceiling Approach

All of the 16 advantages discussed in the general evaluation section (ASHRAE 2000, page 6.1) strongly support the application of this technology, but five relate strongly to this paper. They are:

- Comfort levels can be better than those of other conditioning systems because radiant loads are treated directly and air motion in the space is at normal ventilation levels.
- Supply air quantities usually do not exceed those required for *ventilation and dehumidification* (emphasis added).
- A 100% outdoor air system may be installed with smaller penalties, in terms of refrigeration load, because of reduced outdoor air quantities (multiple spaces Equation 6.1 of *ANSI/ASHRAE Standard 62-1999* [ASHRAE 1999a] does not apply to this situation).
- Wet surface cooling coils are eliminated from the occupied space, reducing the potential for septic contamination.
- The panel system can use the automatic sprinkler system piping (see NFPA Standard 13, Chapter 3, Section 3.6; Janus 2001).

Other CRCP advantages worth noting (Simmonds 1996, 1997; Dedicated Outdoor Air Web Site 2001) but not discussed at length above include the following.

- *Compact design*. The compact design is an advantage for either retrofit design or new construction. In existing buildings, where ceiling heights and plenum space are important issues, the cooling panels can be used to save on plenum space and allow ceiling heights to be raised to an architecturally pleasing level. When used in new construction, the CRCPs achieve all of the advantages that a retrofit project achieves, plus the owner can save money in construction by decreasing the overall height of the building or adding about one floor for every five to ten floors when compared to conventional construction.
- *Quick accommodation of dynamics*, since the panels have a time constant of about three minutes.
- *Spaces may be zoned* by the use of a control valve for each zone.
- *LEED Green Building rating Std.*, the proposed radiant/DOAS mechanical system has the potential to generate rating points in five of the major categories, i.e., Water Efficiency, Energy and Atmosphere, Materials and Resources, Indoor Environmental Quality, and LEED Innovation Credits. Each specific building project will require its own analysis; however, in general the following rating points should be realized with the DOAS approach. The radiant/DOAS approach has the potential to generate up to 23 Green Building Rating points, or up to 88% of the minimum points needed for certification.

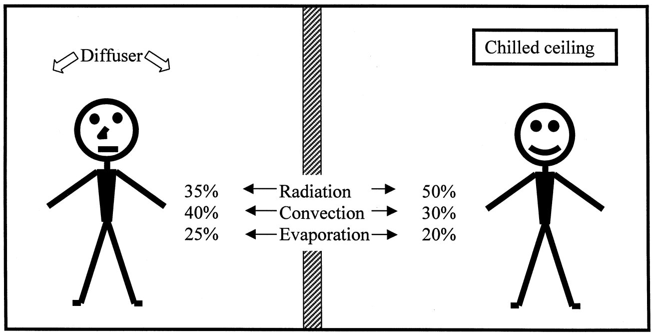


Figure 3 *Energy balance on a human without and with radiant cooling.*

Ceiling Radiant Cooling Thermal Comfort, Radiant Asymmetry, and Space Design DBT issues

Thermal comfort must always be an important design consideration and is strongly governed by variables that influence the energy balance on the human occupants. The primary variables include clothing, activity level, mixed air temperature, mean radiant temperature, vertical air temperature gradients, radiant asymmetry, air motion, and air moisture content. Secondary variables that influence the perception of thermal comfort include age, sex, biorhythms, and physical health. Since these secondary variables are generally outside the control of the mechanical system, they will not be addressed here. The impact of the radiant/DOAS system on thermal comfort will now be addressed.

With ceiling radiant cooling, the energy balance on the human body is different than without the cooled ceiling in two ways. First (Kulpmann 1993), the heat rejection, as illustrated in Figure 3, from the human body by radiation is increased from about 35% without radiant cooling to 50% with radiant cooling. Likewise, the heat loss due to convection decreases from about 40% without the chilled ceiling to about 30% with radiant cooling. The net effect is that less heat is rejected by perspiration in the presence of the radiant cooling field. Secondarily (Jones et. al. 1998), the human head, which emits much of the body's heat, can more effectively emit energy with the cooled ceiling above. The result of the cooled ceiling is a cool face and warm feet for increased comfort and alertness.

As a result of these two occurrences, it is possible to maintain the space dry-bulb temperature higher with chilled ceilings; in fact a space at 78°F (25.6°C) with radiant cooling gives the perception of a space at about 75°F (23.9°C) without radiant cooling (Hittinger 1986). This results in a reduction in the building skin and ventilation air cooling loads. It also means that the ventilation air can remove more sensible heat since there is about a 3°F (1.7°C) larger temperature rise as the air passes through the space. With the growing energy supply problems in the western U.S., there is growing pressure on the government to require that thermostats be set up to 78°F (25.6°C). This move would cause no comfort problems for radiantly cooled buildings, but it would for conventionally cooled spaces.

Another issue worth addressing is the potential discomfort that might result from radiant asymmetry experienced by the human occupant. With most of the enclosure at 78°F (25.6°C) and the chilled ceiling panels at approximately 60°F (15.6°C), an 18°F (10°C) radiant asymmetry temperature differential exists. The archival literature (Olesen 2000; Fanger 1986) indicates that the predicted percent dissatisfied is less than 6% as a result of an 14°C (25°F) or less radiant asymmetry. And in fact, for most cases, as will be discussed more later, only about 50% of the ceiling is chilled, so the effective mean radiant ceiling temperature of the two nearly equal areas is close to 69°F (20.6°C), resulting in a radiant asymmetry of approximately 9°F (5°C)—much too small for radiant asymmetry discomfort.

The remaining three major factors influencing thermal comfort—air motion, vertical air temperature gradients, and air moisture content—are influenced by the design of the DOAS. Air is introduced at constant volume (no possibility of cold air dumping out of the diffusers as may occur in VAV systems under turn down conditions) into the space at about 45°F via high aspiration ceiling diffusers capable of creating a secondary flow to primary airflow ratio of approximately 20:1. This high mixing ratio causes the cold primary air to be warmed to room temperature at about 12 to 15 in. (0.3 to 0.4 m), eliminating the possibility of cold drafts. It also creates sufficient air motion in the space to achieve satisfactory air diffuser performance index (ADPI) values, thus effectively eliminating vertical air temperature gradients.

Displacement ventilation and underfloor air distribution systems (from here on referred to as *floor delivery systems*) have received attention from engineers in Europe for some time, and there is growing interest in them here in the United States. Consequently, it is logical to consider how floor delivery systems might impact the environmental performance of a radiant/DOAS system. Keep in mind that with the radiant/DOAS system, the only air introduced into the space is ventilation air. So the comments that follow are from that perspective. Reasons that floor delivery systems are undesirable with the radiant/DOAS approach will be discussed next.

The low DOAS supply air temperature means that floor level air delivery systems cannot even be considered, for comfort reasons, with radiant cooling. If the supply air temperature were increased to accommodate floor delivery, the added sensible load that the radiant system must bear would increase the required panel area and the first cost by about 50%. Published European literature (Brunk 1993) makes a case for using displacement ventilation, introduced at floor level at about 65°F to 70°F (18.3°C to 21°C), with radiant panels. Supplying the ventilation air 20°F to 25°F (11°C to 14°C) warmer than available with the DOAS requires the radiant panels to absorb more sensible load than necessary, introducing a significant first cost penalty. Another reason for not employing floor delivery systems is that they do little to

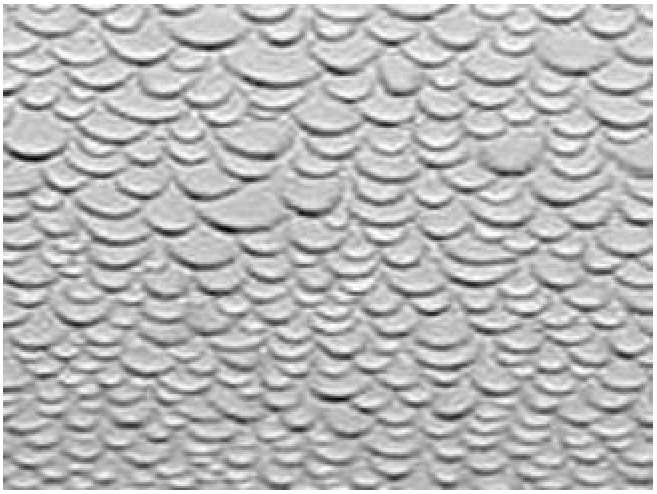


Figure 4 *Condensation after 8.5 hours on a chilled surface intentionally held 14°F (7.8°C) colder than the space DPT. (Note: not one droplet fell under these conditions.)*

enhance the convective heat transfer to the radiant panels. However, by supplying the DOAS air to the space via high aspiration ceiling diffusers, the convective heat transfer to the radiant panels can be increased. The overall increase in heat transfer is about 15% greater than when the panels are operating in still air and 10% greater than when the panels are operating with displacement ventilation (Min et al. 1956). The enhanced convective heat transfer performance further reduces the ceiling area devoted to radiant panels and, hence, first cost. Therefore, for the reasons enumerated above, floor delivery systems are strongly discouraged with radiant panels. Rather, the 45°F (7°C) air should be supplied to the space via high aspiration diffusers located in the ceiling. The diffuser throw needs to parallel the longitudinal pattern of the radiant panels.

Some other reasons, in the author's opinion, that floor delivery systems are a poor choice for all-air systems (consisting of ventilation air and return air) are:

- The elevated supply air temperatures require very high flow rates. As a result a great quantity of air must be recirculated, eliminating the advertised IAQ benefits, which suggests that there is minimal mixing of the air with room contaminants. In addition the very high flow rates also increase the fan energy use and demand. Finally, the high flow rates will whip up the irritants tracked into the space on the shoes of the occupants and put them in the occupant's breathing zone.
- The OA is mixed with recirculated room air, and hence the multiple spaces Equation of Standard 62 (ASHRAE 1999a) apply, and verifiable fresh air distribution is very difficult just like other all-air systems.
- The high supply air temperatures mean that the potential for serious high humidity problems exists including condensation on the chilled floor carpet and associated biological incubation. Or the air must first be deep cooled for dehumidification, then reheated, an energy use and demand penalty.

The vertical air temperature distribution of approximately 20°F (11°C) far exceeds the less than 7°F (4°C) recommended upper limit for thermal comfort (Olesen 2000). It is said that the bulk of this temperature gradient occurs above the breathing zone of a seated person, but this is not likely to be true. First, with the required high flow rates of underfloor air distribution systems, a well-mixed room would occur, preventing a nonlinear gradient. Even with much lower flow displacement systems, it is the author's experience that the air above the breathing zone transitions to turbulent flow and is well mixed with a very small temperature gradient from the top of laminar breathing zone to the ceiling.

MOISTURE CONDENSATION ISSUES

Because of the potential for condensation, radiant cooling cannot even be considered unless there is another parallel system in place to decouple the space sensible and latent loads, or the situation illustrated in Figure 4 may occur. The author strongly recommends (Mumma 2001a) that a DOAS be used to remove all of the space latent loads, thus achieving the required load decoupling. The DOAS is also required, in the author's opinion (Mumma 2001b), to ensure compliance with ASHRAE Standard 62, something nearly impossible to verify with an all-air system. ASHRAE Standard 90.1-1999 (ASHRAE 1999b), section 6.3.6.1, "Exhaust Air Energy Recovery," addresses the requirements for total energy recovery in the DOAS. The required heat recovery reduces the OA load on the cooling coil by 75% to 80%. This results in reduced energy demand and consumption (generally exceeding savings that could be realized with demand-controlled ventilation in conventional all-air VAV systems), as well as reduced chiller size. In general, the supply air conditions from the DOAS (Mumma and Shank 2001) required to decouple the space sensible and latent loads and minimize the sensible load on the parallel radiant cooling system are about 45°F (7.2°C) and saturated.

Steady-State and Transient Condensation Formation Considerations

Utilization of the DOAS to decouple the space loads at design does not guarantee that some spaces will not from time to time have transient occupancies exceeding design. The extra occupant latent load generation has the potential to create a condensation problem in time. To better understand the nature of this potential problem, let us explore two conservative boundary conditions, first, the steady-state (SS) rate of moisture condensation in a typical office situation. Let us make the following assumptions:

- Ventilation airflow rate and thermodynamic state point remain constant at the design conditions.
- Occupant latent load is 205 Btu/hour per person (60 W/ person).
- Infiltration is negligible.
- Occupancy exceeds design by 100%, i.e., two people where one was used in design.
- Enclosure and contents are completely nonhygroscopic, i.e., they do not participate in the moisture transients (an extremely conservative assumption).
- Typical radiant panel area per person at design occupancy, 70 ft^2 (6.5 m^2). This assumes 7 people per 1000 ft^2 (93 m^2) and a 50% chilled ceiling fill factor.
- Chilled ceiling radiant panel temperature is uniform overall [not completely true since as much as a 5°F (2.8°C) differential can exist from one position to another].

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 ft^2 (6.5 m^2) of panel per person, the water thickness after one hour is 5/10,000 of an in. (13 μm). For reference, a human hair ranges in diameter from 0.0007 to 0.007 in. (17 μm -181 μm) in diameter. Needless to say, under these conservative steady-state moisture condensation assumptions, it would take one person's latent generation from 90 minutes to 14 hours for the condensation thickness to equal the diameter of a human hair. Now consider the transient rate of moisture condensation in a typical office situation.

As in the steady-state case, it is assumed that at or below design occupancy, condensation will not occur when the DOAS is working properly. However if the design occupancy is exceeded, the space dew-point temperature will rise leading to potential condensation when the space dew-point temperature exceeds the radiant panel cooling water temperature. Conceptually, this is similar to a bucket of water partially full when filling begins. It does not overflow at first as water is added. In the case of the transient moisture situation, the following additional design condition assumptions were made beyond those used for the SS case:

- 12 ft (3.65 m) high ceiling, in addition to 7 people per 1000 ft^2 (93 m^2).
- Ventilation rate per person, 20 scfm (0.011kg/s).
- Supply air condition, 44°F (6.7°C) and saturated; or a humidity ratio of 42.6 gr./lbm (0.00608 kg/kg).
- Resulting space dew point temperature, 52°F (11°C), or a humidity ratio of 57.73 gr./lbm (0.00825 kg/kg).
- Cooling water temperature to the panels, 55°F (12.8°C) [3°F (1.7°C) greater than the design space DPT]. The humidity ratio for 55°F (12.8°C) and saturated is 64.63 gr./lbm (0.00923 kg/kg)
- The space air is assumed to be well mixed.

The governing differential equation for this problem, based upon water vapor, is

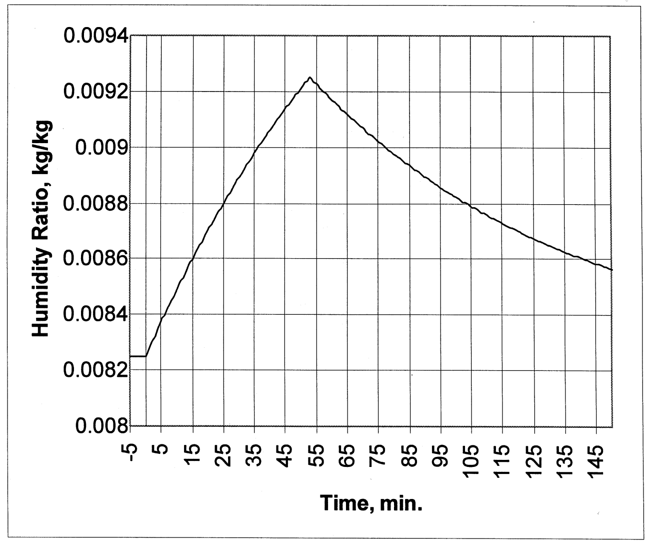


Figure 5 *Transient humidity ratio response to a sudden doubling of the design occupancy, followed by a sudden return to design occupancy.*

$$dm_{room}/d\tau = \dot{m}_{in} + \dot{m}_{IG} - \dot{m}_{out} \quad (3)$$

where

$dm_{room/dt}$ = rate of change of the mass of water vapor in the space at any instant in time;

$\dot{m}_{in}$ = mass flow rate of water vapor into the space with the ventilation air at an instant in time;

$\dot{m}_{IG}$ = rate of moisture released by the occupants in the space at an instant in time;

$\dot{m}_{out}$ = mass flow rate of water vapor leaving the space at an instant in time.

Under the assumed conditions, if the occupancy suddenly doubled, it would take nearly an hour for the space dew-point temperature to rise to that of the chilled ceiling feed water temperature [55°F (12.8°C)]. If the occupancy suddenly tripled, (resulting in an SS space humidity ratio of 87.9 gr./lbm [0.01256 kg/kg] if no condensation formed on the radiant panels), it would take nearly half an hour for condensation to begin forming. If the occupancy were to suddenly double (resulting in an SS space humidity ratio of 72.66 gr./lbm [0.01038 kg/kg] if no condensation formed on the radiant panels), then after nearly an hour, the occupancy returned to design, the transient response appears as illustrated in Figure 5.

As expected, the space humidity ratio begins an exponential rise toward steady state, and then, when the space DPT equals the temperature of the cooling fluid, the occupancy suddenly returns to design. The fall in humidity following the rise responds more slowly than it did on the way up. Under the assumed conditions this is not a problem. However, if the hygroscopic nature of the enclosure and space contents had been considered, the rise in humidity ratio would have been

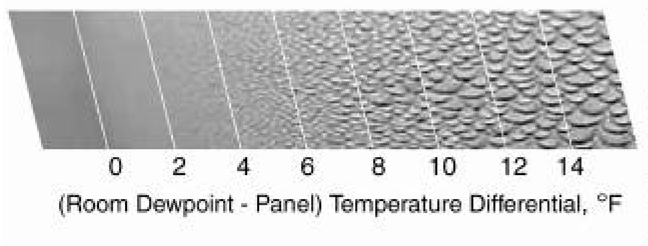


Figure 6 *Condensation formation on a panel after 8.5 hours.*

much slower and the return to design conditions even slower. Real response characteristics will be important experimentally determined values for each installation. The data will be useful for determining the required DOAS preconditioning dehumidification run time prior to activation of the chilled ceiling and occupancy after a weekend/holiday shutdown.

Only deviations from design resulting from occupancy changes have been addressed to this point. It is possible that condensation could also result from envelope integrity problems. However, if the structure is confirmed to truly be in compliance with ASHRAE Standard 90.1, Section 5.2.3, "Envelope Air Leakage," that should not be a problem. Particularly when care is taken to avoid negative pressures within the building, such as return air plenums.

An extreme condensation case was investigated experimentally under steady boundary conditions. In the experiment, a portion of a panel surface temperature was chilled 14°F (8°C) below the conditioned space dew-point temperature. 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 6 to the right of the 14°F 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 6 between 0°F and 14°F represents that portion of the panel that was not chilled. There the panel temperature was decreasing from the space dew point temperature at the 0°F line over to the 14°F line. Not much condensation is observed on the panel, even after 8.5 hours, for panel temperatures between 0°F and 8°F colder than the space dew-point temperature.

It can safely be stated that the onset of condensation occurs slowly and can easily be avoided so long as the DOAS and panel loop temperature controls are operating correctly. In the event the controls fail, there are several simple control-based *safety* remedies. They 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. And 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 be taken—also a good thing.

COOLING CAPACITY ISSUE

Experienced design engineers are exceptionally skilled at bounding problems with rules developed from experience. This skill has avoided many a problem 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 DOASs, caution is advised/required.

One universal rule that all in this industry use is: 300-400 ft^2/ton (7.9–10.6 m^2/kW) of cooling. Clearly this rule appears to require that 40-30 Btu/h·ft^2 (126-95 W/m^2) of total heat energy be removed. Since "drop in" radiant cooling has an upper sensible heat removal limit of around 30 Btu/h·ft^2 (95 W/m^2), rule based thinkers conclude that radiant cooling simply could not meet the loads, even with the ceiling 100% filled with cooling panels. However, application of another universal industry rule, i.e., 1 cfm/ft^2 (5.08 L/s·m^2) for all-air VAV systems operating with 55°F (12.8°C) supply air, is a certain tipoff that the space loads must be less than 40 Btu/h·ft^2 (126 W/m^2). Clearly, at design under the 1-cfm/ft^2 (5.08 L/s·m^2) rule, an all-air VAV system could only remove about 20 Btu/h·ft^2 (63 W/m^2) of space sensible heat (radiant panels can only be allowed to remove sensible heat) when the design space temperature is 75°F (23.9°C). While it is probably clear why there is an apparent conflict between the conclusions drawn from the two universal rules, it will be briefly discussed next.

Dividing the building floor area by the design chiller load yields the 300-400 ft^2/ton (7.9–10.6 m^2/kW) rule. The design chiller load consists of the following three load components:

- The total OA load, which is: $\dot{m}_{OA}*(h_{OA}-h_{ret.air})$. Note: It is not $\dot{m}_{OA}*(h_{OA}-h_{supply.air})$.
- 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 borne by the ceiling radiant 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 radiant cooling, which must be designed to remove the total OA load and all of the space latent load. And if the OA is supplied with a dry-bulb temperature equal to the required supply air DPT, it can also remove 5-6 Btu/h·ft^2 (16-19 W/m^2) of the space sensible load. As a result, the sensible load actually remaining for the radiant cooling system is, on average, only 14 to 15 Btu/h·ft^2 (44 to 47 W/m^2) (or about 50% ceiling fill factor). From this perspective, it should be clear that there is no capacity problem with radiant cooling when properly applied with a parallel DOAS. In some buildings, with a high percentage of glazed walls, corner offices will require more than 15 Btu/h·ft^2 (47 W/m^2). In limited cases, increasing the design flow rate of OA to such spaces, thus shifting more of the space sensible load to the DOAS, can nicely accommodate this capacity issue. The implications of such a strategy will be discussed in the economic issues section of the paper.

ECONOMIC ISSUES

The integrated radiant cooling/DOAS approach provides superior indoor air quality and thermal comfort, and that alone should be sufficient incentive for the industry to use the concept. This is especially true since the Lawrence Berkeley National Laboratory (Vaughn 2001) estimates that U.S. companies could save as much as \$58 billion annually by preventing sick-building illnesses and could benefit from \$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 process (the current energy woes in the western U.S. may change this mindset). Therefore, a careful first cost analysis is necessary to justify this design approach.

The Major Factors That Impact the First Cost of the Radiant/DOAS System Compared to a Conventional All-Air VAV System

With the integrated radiant/DOAS approach, mechanical as well as building cost issues arise. The major items are identified below:

1. Building design chiller size reduced due to the need for less OA and the use of total energy (sensible and latent) recovery in the DOAS.
2. Pump size smaller due to chiller size reduction.
3. First cost of the ductwork and associated terminal units greatly reduced since the DOAS air flow 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/interior partition/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.
8. Acoustical ceiling panels reduction where replaced with the CRCP system.
9. Less lost rentable space due to mechanical shaft reduction as a result of much lower air volumes.

On the other hand, the radiant panels contribute significantly to the first cost of the radiant/DOAS system.

A first cost analysis (Mean's 2000) for a six-story, 31,000 ft^2 (2,883 m^2) per floor, office building (in compliance with the ASHRAE energy conservation Standard 90.1-1999) located in Philadelphia, Pa., is presented in Table 2. The brick facade building has a footprint of 125 by 250 ft (38 by 76 m) with the long axis pointing in the E-W direction. The wall U-factor is 0.044 Btu/h·ft^2°F (0.25 W/m^2-C), the roof U-factor is 0.03 Btu/h·ft^2-F (0.17 W/m^2-C), and the glazing U-factor and shading coefficients are 0.48 Btu/h·ft^2°F (2.73 W/m^2·°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 1000 ft^2 (93 m^2), the overhead lighting is 1.3 W/ft^2 (14 W/m^2), task lighting is 0.7 W/ft^2 (7.5 W/m^2), and equipment plug loads are 2 W/ft^2 (21.5 W/m^2).

Controlling the First Cost of the Chilled Ceiling Panel Array

At this point in time, radiant cooling panels produced and shipped from abroad are relatively expensive [approximately \$13/ft^2 (\$140/m^2) of panel installed] as perceived by the building industry. Therefore, in order for them to show a financially attractive investment, six specific steps are recommended to either minimize the sensible load they bear or to enhance their thermal performance.

- Hold the space dew-point temperature as low as is practical and healthy, i.e., around 52°F (11°C). The lower the space DPT, the lower the panel cooling water temperature can be without condensation formation, thus increasing the panel heat removal capability per square foot.
- Use a supply air temperature equal to the required supply air dew-point temperature, i.e., approximately 45°F (7°C). Requirements for terminal reheat are almost universally less with the DOAS system than with a conventional all-air VAV system (Mumma 2001b).
- Use long panels, certainly greater than 2 ft (0.61 m), i.e., more like 8 to 10 ft (2.43 to 3.04 m) long, or more to minimize installation handling and the number of connections (cost and potential leak points).

TABLE 2
First Cost Comparison of the Radiant/DOAS System vs. Conventional All-Air VAV System Serving a six-Story 186,000 ft^2 (17,300 m^2) Building in Philadelphia, PA

Cost item	Unit cost	Units VAV	Units Radiant/DOAS	Cost savings
Chiller	\$1,000/ton (\$284/kW)	506 ton (1,780 kW)	306 ton (1,076 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 Serv.	\$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	NA	186,000 ft^2 (17,300 m^2)	\$120,900
Drop ceiling	\$1.50/ ft^2 (\$16/m^2)	NA	79,200 ft^2 (7,365 m^2)	\$118,800
Mechanical shaft impact on lost rentable space	\$125/ ft^2 (\$1,344/m^2)	NA	500 ft^2 (47 m^2) saved	\$62,500
Savings Subtotal				**\$1,405,300**
Radiant Panel	\$13/ ft^2 (\$140/m^2) of panel	NA	79,200 ft^2 (7,365 m^2)	–\$1,029,600
Net savings				**\$375,700, or \$2/ ft^2 (\$22/m^2)**

TABLE 3
Operating Cost Comparison, Radiant/DOAS System vs. Standard VAV System

System	Annual Mechanical Operating Cost	Annual Total Mech., Illum., & Equip. Operating Cost
VAV	\$77,350	\$299,914
Radiant/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/(Radiant/DOAS)	1.29	1.10

TABLE 4
Impact of Increasing the Ventilation Air with the Radiant/DOAS System on First and Operating Costs

OA flow ➡	DOAS, 25,000 scfm (13.85 kg/s)	DOAS, 30,000 scfm (16.6 kg/s)	DOAS, 35,000 scfm (19.4 kg/s)	VAV, 34,000 scfm (18.8 kg/s)
Category	**Savings, $**	**Savings, $**	**Savings, $**	**Savings, $**
Chiller	200,000	194,000	188,000	0
Pump	11,950	11,620	11,250	0
Ductwork	558,000	520,800	483,600	0
AHU	170,000	150,000	130,000	0
Elec. Serv.	12,400	12,030	11,660	0
Facade	150,780	150,780	150,780	0
Piping	120,900	120,900	120,900	0
Drop ceiling	118,800	109,800	100,200	0
Mech. Shaft	62,500	60,600	58,750	0
Savings subtotal	**1,405,300**	**1,330,530**	**1,255,140**	**0**
Radiant Panel	(1,029,600)	(951,600)	(868,400)	0
Net Savings	**375,700**	**378,930**	**386,740**	**0**
First cost improvement ref. 25,000 scfm (13.85 kg/s)	0	3,230	11,040	NA
Annual mech. op. cost	59,730	64,840	65,480	77,350
Mech. op. cost penalty, Ref. 25,000 scfm (13.85 kg/s)	0	5,110	5,750	17,620
Annual total Op. cost	273,560	281,270	282,470	299,910
Total. op. cost penalty, Ref. 25,000 scfm (13.85 kg/s)	0	7,700	8,900	26,350

- Long panels require parallel header rather than serpentine piping, resulting in more uniform panel temperature and lower pumping costs. Also makes joining panels together on a single control valve easier.
- Functional integration of the thermal transport and fire suppression piping.

Operating Cost Issues

An hourly energy analysis, for 12 hours a day, 5 days per week, was performed for the 186,000 ft^2 (17,300 m^2) building located in Philadelphia. Both an all-air VAV system and the radiant/DOAS system were analyzed. This analysis was easily performed using existing load and energy analysis software. When the DOAS uses a single enthalpy wheel and a cooling coil, as was the case for this example, existing software can be used. Dual-wheel DOAS system analysis is another matter with existing readily available software. In the air system menu portion 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 radiant panel pumping head is included in the hydronic specifications. The summer operation is thought to be well modeled by this approach. 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 used for this analysis follow.

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 3. Like the first cost analysis, the operating cost data also favor the radiant/DOAS system. The mechanical system annual operating cost savings is $17,620 or about $0.10/ft^2-yr. ($1.07/m^2-yr.). Due to the smaller mechanical plant for the radiant/DOAS, the building demand charges were smaller, resulting in an annual building operating cost savings of $26,349, or about $0.15/ft^2-yr. ($1.60/m^2-yr.) It cost about 29% more to operate a conventional VAV system each year than the radiant/DOAS system.

Cost Summary

The first and operating cost results clearly both favor the radiant/DOAS approach. However, the structure of professional fees may discourage crediting items outside one's own discipline (i.e., mechanical in this case). For example, reductions in the first cost of the building enclosure, mechanical shaft floor area reduction, and dual use of fire suppression piping could be perceived as having an adverse impact on the architect's fees. If actual practice supports these first and operating cost estimates, there can be no impediment to rapid implementation, short of defending the status quo out of fear of change or failure!

Trade-Offs, More OA and Sensible Cooling with the DOAS and Less Radiant Panel

The first cost of the radiant panels could be reduced by increasing the ventilation air supplied to the space and hence the sensible cooling done by the ventilation air. Modest increases in the OA flow rate will not impact the chiller plant size measurably because of the energy recovery incorporated into the DOAS unit. However, it will increase the size of the DOAS and associated ductwork, as well as the operating cost of the fans in the unit. Using the same building and cost data as presented above, the impact of increasing the OA from 25,000 scfm (13.85 kg/s) to 30,000 and 35,000 scfm (16.6 and 19.4 kg/s) were explored. The results of that investigation are presented in Table 4.

As expected, the first cost of the radiant panels dropped by about $80,000 for each 5,000 scfm (2.8 kg/s) increase in the OA supplied. This savings was offset by a reduction in the savings elsewhere in the project of about $75,000 for each 5,000 scfm (2.8 kg/s) increase in the OA supplied. So the reduction in first cost of the system was roughly $1/scfm ($1,800/kg/s) of increase in the OA flow rate. As expected, the operating cost rose nonlinearly with increasing OA flow. And even when the OA flow rate exceeded the required OA flow rate for the VAV system, the annual mechanical operating and total building operating costs were about $12,000 and $17,500 per year, respectively, lower in favor of the radiant/DOAS system. However since the first cost savings were nearly or completely eliminated by the increase in operating cost, it is not recommended that the OA flow rate be increased as a means of reducing the first cost further.

CONCLUSIONS AND RECOMMENDATIONS

The central thrust of this paper was to explore the primary concerns expressed by the building industry about radiant cooling: condensation, capacity, and cost. It has been demonstrated that when a DOAS is used to decouple the space sensible and latent loads, the radiant panels are only left with a portion of the space sensible loads. And if the occupancy exceeds design by a factor of 2 or 3, it takes quite a bit of time for the space humidity ratio to raise to the point where condensation can form. Then, once it does start to form, 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 in light of the rules of thumb that engineers have used to come to the conclusion that capacity is an area of concern. The paper 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 radiant panels. In conclusion, radiant 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 radiant/DOAS systems, that concern has been dismissed. The root of the concern to begin with came from the notion that there were capacity problems and to meet them, the building ceiling and walls required radiant panels. This is simply not the case as illustrated in the paper.

The three concerns addressed in this paper cannot be used as an excuse to reject radiant cooling when properly designed. This approach holds great potential not only for first and operating cost advantages but also for improved IAQ and thermal comfort. Both of these improvements will enhance the productivity of the people working in the buildings.

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