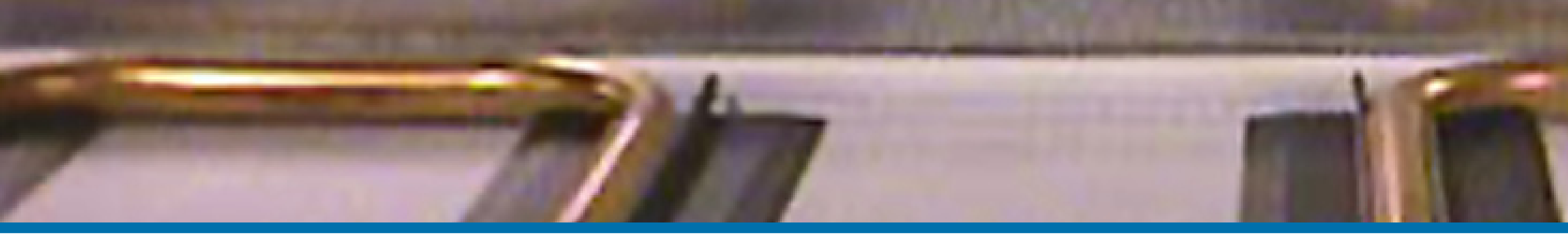

Designing a Dedicated Outdoor Air System with Ceiling Radiant Cooling Panels

Ceiling radiant panels.

By **Jae-Weon Jeong, Ph.D.,** Associate Member ASHRAE; and **Stan Mumma, Ph.D., P.E.,** Fellow ASHRAE

The concept of a dedicated outdoor air system (DOAS) with parallel sensible cooling was born from the decoupled system concept, which can be summarized as decoupling of ventilation and air-conditioning functions, or decoupling of sensible and latent load functions. First, remove the latent loads from the outside air (OA) intake and generated in spaces using a 100% OA ventilation system (i.e., DOAS). Second, remove the space sensible loads using a parallel mechanical cooling system, such as fan coil units, conventional variable air volume, and ceiling radiant cooling panel (CRCP) independent of the ventilation system.

Among several candidates for a parallel sensible cooling system, Mumma[1] shows CRCPs may be the best choice for the DOAS in most aspects: first cost, energy consumption, thermal comfort, and indoor air quality. *Figure 1* shows the typical configuration of the DOAS/CRCP combined system.

For the last few decades, core technologies required for the DOAS and the CRCP system have been developed separately. Now, a few equipment manufacturers for each core technology exist in the U.S., Canada, and Europe. However, as recently as five years ago, there was no evidence the U.S. HVAC market would consider the integrated DOAS/CRCP system. It has been only in the last few years that serious attention has been devoted to the integration of those technologies with focus on their improved thermal and economical advantages.

Today, engineers are forced to design the DOAS/CRCP system on the basis of qualitative merits, limited experience, and conservative estimates. The unfamiliarity of HVAC designers and contractors with the new system concept is a significant barrier to the wider application of DOAS/CRCP systems. To overcome

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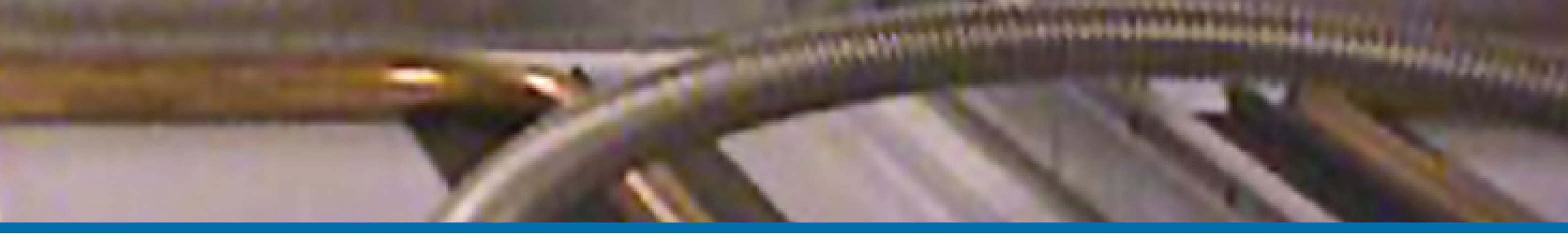


this situation, eight simple steps for designing a DOAS/CRCP system are provided in this article.

Designing DOAS/CRCP Systems

By designing a DOAS/CRCP system serving four classrooms located at Williamsport, Pa., the general design procedure is presented clearly. For simplicity, it is assumed each classroom has identical basic design conditions as described in *Table 1* except the number of occupants.

Step 1: Determine design outdoor air conditions.

In many HVAC system design practices, the climatic design conditions listed in the *2005 ASHRAE Handbook—Fundamentals*[2] are used as a design OA condition. As for cooling and/or dehumidification designs, ASHRAE provides three design OA data sets: (1) peak dry bulb with mean coincident wet-bulb temperature (WBT); (2) peak dew point with mean coincident dry-bulb temperature; and (3) peak wet bulb with mean coincident dry-bulb temperature. This data allow the designer to consider various operational peak conditions. The first data set is used to determine peak sensible loads, the second data set is selected for calculating the peak latent loads, and the third data set is used for estimating peak total cooling loads. Among these three, one data set appropriate for designing the DOAS should be chosen by considering the unique characteristics of the DOAS.

Ceiling radiant cooling panels.

In the DOAS, the OA enthalpy is reduced by the enthalpy wheel during the summer. The enthalpy wheel transfers excess moisture and sensible heat contained in the OA stream to the relatively dry and cool exhaust airstream (i.e., pre-cooling and dehumidification). The cooling coil is sized based on the OA enthalpy after the enthalpy wheel preconditioning. Therefore, the climatic design data set providing the highest design OA enthalpy should be considered (i.e., peak wet bulb with mean coincident dry-bulb temperature) for selecting a cooling coil that has adequate cooling and dehumidification capacity. In *Table 2*, ASHRAE's three OA condition data sets (annual percentile of 0.4) for Williamsport, Pa., are presented. Among these three, the third data set (75.6°F [24.2°C]) WBT and 84.7°F [29.3°C] DBT) showing the highest OA enthalpy is chosen for this DOAS/CRCP design.

Step 2: Determine target space conditions.

Before determining a target space condition (i.e., room dry-bulb temperature and percent relative humidity [RH]) maintained by the DOAS/CRCP system, a design mean panel surface temperature should be chosen first. The design panel surface temperature should be higher than the room dew-point temperature (DPT) to avoid condensation on the cooling panel surfaces. Sixty-two degrees Fahrenheit (16°C) is commonly used as a design panel surface temperature. Therefore, the room DPT under the target space condition should be less than or equal to 62°F (16°C) mean panel surface temperature.

In conventional cooling system design, many engineers use 75°F (24°C) DBT and 50% RH as a target space condition. This condition corresponds to 65.4 gr/lb (9.34 g/kg) humidity ratio (HR) and 55°F (12.9°C) DPT. This design condition can also be used for DOAS/CRCP system design. However, according to the literature,[3,4] if ceiling radiant panels are used for space cooling, 2°F to 4°F (1°C to 2°C) higher design space temperature (i.e., 77°F to 79°F [25°C to 26°C] DBT) can be used without significant negative impact on thermal comfort rated by the operative temperature (OT).

OT can be approximated by a simple average of the space DBT and mean radiant temperature (MRT). In general, space MRT is reduced about 2°F to 4°F (1°C to 2°C) by ceiling radiant cooling. Consequently, the room thermostat can be set to 2°F to 4°F (1°C to 2°C) higher temperature without any change in OT. More space sensible loads are met by supply air, and the required CRCP area can be reduced. Based on above findings, 79°F (26°C) DBT/50% RH is selected as the target space condition for this DOAS/CRCP design. It corresponds to 73.8 gr/lb (10.54 g/kg) HR and 58.6°F (14.8°C) DPT. This room DPT is lower than the design mean panel surface temperature (62°F [16°C]); therefore, the target space condition of 79°F (26°C) DBT/50% RH is acceptable.

Step 3: Determine design cooling load and required ventilation rate for each space.

Based on *Table 1* and predetermined OA and target space conditions, the design sensible and latent cooling load for each space are calculated. Required ventilation for each space is estimated using the minimum ventilation rates recommended by ANSI/ASHRAE Standard 62.1-2004, *Ventilation for Acceptable Indoor Air Quality*. According to Standard 62.1, a classroom (for students ages 9 and older) requires 10 cfm (5 L/s) of OA per person and 0.12 cfm (0.6 L/s) of OA per unit floor area. *Table 3* shows the estimated design cooling loads and the ventilation rate required for each space. In the DOAS/CRCP system, the total supply air is the sum of the required minimum ventilation rates (i.e., 1,649 cfm [778.6 L/s]), and the ventilation air distributed to each conditioned space at constant volume. No contaminated return air is recirculated to the conditioned space.

Step 4: Determine supply air conditions.

In DOAS/CRCP, the supply air must be dehumidified enough by the DOAS to maintain the target space humidity level in each conditioned space. However, the dryness of the supply air (SA) required for each space may be different because each space experiences different latent load and needs different SA quantity although they may be served by one DOAS unit. Consequently, the critical space in the DOAS approach is the space that requires the driest (lowest HR) supply air. The required SA humidity ratio for each space can be calculated using Equation 1, and the lowest SA humidity ratio among them should be selected as a design SA humidity level.

$$W_{sa} = W_{sp} - \frac{Q_L}{0.68 \times \dot{V}_{sa}} \qquad (1)$$

where

W_{sa} = SA humidity ratio, gr/lb
W_{sp} = target space humidity ratio, gr/lb
Q_L = space latent load, Btu/h
$\dot{V}_{sa}$ = space SA flow rate, cfm

In *Table 4,* the required SA HR for each classroom is calculated using Equation 1. In this design, the critical space is Classroom 2 where the driest supply air (50.4 gr/lb [7.26 g/kg] HR) should be supplied. The cooling coil (*Figure 1*) cools and dehumidifies the SA preconditioned by the enthalpy wheel to meet this SA dryness. Assuming the supply air leaves the cooling coil at the saturation condition, the supply air DBT is 48.6°F (9.2°C) and the humidity

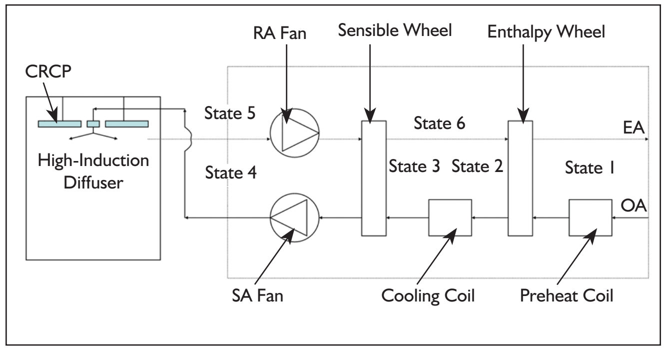


Figure 1: Typical DOAS/CRCP system configuration.

Location	Williamsport, Pa.
System Description	DOAS/CRCP System Serving Four Classrooms
Room Size	26.2 ft × 26.2 ft × 11.5 ft
Number of Occupants	30 people (Classroom 1), 35 people (Classroom 2), 28 people (Classroom 3), 32 people (Classroom 4)
Occupant Heat	Sensible: 256 Btu/h per Person Latent: 205 Btu/h per Person
Envelope Wall UA value	123.1 Btu/h·°F
Roof UA value	18.9 Btu/h·°F
Lighting Heat	75 Btu/h per Unit Floor Area
Solar	9.9 kBtu/h per Room
Other Assumptions	No Infiltration, No Moisture Generation Source Except Occupants

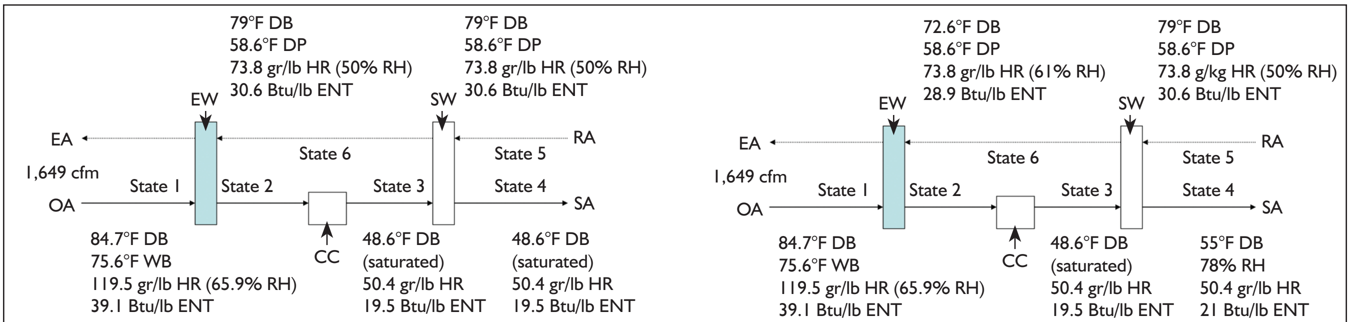
Table 1: Basic design data for conditioned spaces.

	Design Condition	Enthalpy
Peak DB, Mean Coincident WB	89.4°F DB 72.5°F WB	36.1 Btu/lb
Peak DP, Mean Coincident DB	72.9°F DP 79.9°F DB	38.3 Btu/lb
Peak WB, Mean Coincident DB	75.6°F WB 84.7°F DB	39.1 Btu/lb (selected)

Table 2: Design outdoor air conditions (0.4%, Williamsport, Pa.) (From the* 2005 ASHRAE Handbook—Fundamentals.*)

	Classroom 1	Classroom 2	Classroom 3	Classroom 4
Sensible Load (Q_s)	23.2 kBtu/h	24.5 kBtu/h	22.7 kBtu/h	23.7 kBtu/h
Latent Load (Q_L)	6.1 kBtu/h	7.2 kBtu/h	5.7 kBtu/h	6.5 kBtu/h
Number of Occupants	30	35	28	32
Required Ventilation (By Components)	318 cfm (People) 81 cfm (Floor)	371 cfm (People) 81 cfm (Floor)	297 cfm (People) 81 cfm (Floor)	339 cfm (People) 81 cfm (Floor)
Required Ventilation (Each Room) ($\dot{V}_{sa}$)	399 cfm	452 cfm	378 cfm	420 cfm
Total SA Quantity ($\dot{V}_{sa,tot}$) = 1,649 cfm				

Table 3: Design cooling loads and ventilation rate for each space.



(A) Case 1: 48.6°F SA Temperature **(B) Case 2: 55°F SA Temperature**

***Figure 2:** Design outdoor air, supply air and return air conditions.*

	Classroom 1	Classroom 2*	Classroom 3	Classroom 4
Latent Load (Q_L)	6.1 kBtu/h	7.2 kBtu/h	5.7 kBtu/h	6.5 kBtu/h
SA quantity ($\dot{V}_{sa}$)	399 cfm	452 cfm	378 cfm	420 cfm
Target HR (W_{sp})	73.8 gr/lb	73.8 gr/lb	73.8 gr/lb	73.8 gr/lb
Required SA HR (W_{sa})	51.3 gr/lb	50.4 gr/lb	51.6 gr/lb	51.0 gr/lb
* Classroom 2 is the critical space that requires the driest supply air.				

***Table 4:** Required supply air humidity ratio for each space.*

		Sensible Effectiveness, ε_s	Latent Effectiveness, ε_L	Total Effectiveness, ε_T
Silica Gel	Case 1*	85.6%	83.5%	83.9%
EW	Case 2**	85.6%	83.9%	84.5%
Molecular	Case 1	84.8%	68.5%	71.4%
Sieves EW	Case 2	84.8%	69.2%	74.0%
* 48.6°F SA temperature case. ** 55°F SA temperature case.				

***Table 5:** Design enthalpy wheel effectiveness values.*

ratio is 50.4 gr/lb (7.26 g/kg). It is a lower temperature than the design SA DBT used in common practice (55°F [12.8°C]). However, this relatively low-temperature air can be supplied directly to the conditioned spaces through the high induction diffusers without reheating. According to the research,[1] the SA temperature for the DOAS/CRCP system can be lowered to 45°F (7.2°C) without penalty in thermal comfort as long as the air is supplied through the high induction diffusers.

On the other hand, if an engineer wants to use higher design SA temperature (e.g., from 55°F [12.8°C] to neutral temperature) and conventional ceiling diffusers in the DOAS approach, the low-temperature air leaving the cooling coil can be reheated by the sensible wheel shown in *Figure 1*. The sensible wheel can maintain the design SA DBT via rotating speed modulation. However, the required CRCP area in each space is inevitably increased because of the reduced sensible cooling capacity of the supply air.

In this design, two different SA temperatures are considered: (1) 48.6°F (9.2°C) SA DBT (no reheating by the sensible wheel) with high induction diffusers; and (2) 55°F (12.8°C) SA DBT (reheating by the sensible wheel) with conventional ceiling diffusers. In both cases, the SA humidity ratio is not changed (50.4 gr/lb [7.26 g/kg]) because the sensible wheel does not recover moisture from the return air (RA) stream. Similarly, RA humidity ratio (73.8 gr/lb [10.54 g/kg]) is not affected by sensible wheel operation. RA DBT is decreased from 79°F to 72.6°F (26°C to 22.4°C) in Case (2), increasing the SA temperature from 48.6°F to 55°F (9.2°C to 12.8°C).

The design conditions determined so far are summarized in *Figure 2*. For simplicity, the effect of the fan-generated heat is not considered in this design example.

Step 5: Determine enthalpy wheel effectiveness and design cooling coil load.

To estimate the design cooling coil load (required cooling coil capacity), the SA conditions after the enthalpy wheel (State 2 in *Figure 1*) should be known. The enthalpy wheel leaving SA DBT and HR are easily determined using Equations 2 and 3 with knowledge of wheel entering SA and exhaust air (EA) conditions (DBTs and HRs at State 1 and State 6) and the sensible and latent effectiveness of the enthalpy wheel.

$$T_2 = T_1 - \varepsilon_s \frac{\left(\dot{m}C_p\right)_{\min}}{\left(\dot{m}C_p\right)_1}\left(T_1 - T_6\right) \tag{2}$$

$$W_2 = W_2 - \varepsilon_L \frac{\dot{m}_{\min}}{\dot{m}_1}\left(W_1 - W_6\right) \tag{3}$$

where

T_1, T_2, and T_6 = DBTs at States 1, 2, and 6, °F
W_1, W_2, and W_6 = HRs at States 1, 2, and 6, gr/lb
$\left(\dot{m}C_p\right)_{\min}$ = the minimum capacitance flow rate between SA and EA, Btu/h·°F
$\dot{m}_{\min}$ = the minimum pass flow rate between SA and EA, lb/min
ε_s = sensible effectiveness of the enthalpy wheel
ε_L = latent effectiveness of the enthalpy wheel

		Classroom 1	Classroom 2	Classroom 3	Classroom 4
Case 1 (T_{sa}= 48.6°F)	Space Sensible Load (Q_S)	23.2 kBtu/h	24.5 kBtu/h	22.7 kBtu/h	23.7 kBtu/h
	SA Quantity ($\dot{V}_{sa}$)	399 cfm	452 cfm	378 cfm	420 cfm
	SA Cooling Capacity ($Q_{sen,sa}$)	13.1 kBtu/h	14.8 kBtu/h	12.4 kBtu/h	13.8 kBtu/h
	CRCP Cooling Load ($Q_{sen,p}$)	10.1 kBtu/h	9.7 kBtu/h	10.3 kBtu/h	10.1 kBtu/h
Case 2 (T_{sa}= 55°F)	Space Sensible Load (Q_S)	23.2 kBtu/h	24.5 kBtu/h	22.7 kBtu/h	23.7 kBtu/h
	SA Quantity ($\dot{V}_{sa}$)	399 cfm	452 cfm	378 cfm	420 cfm
	SA Cooling Capacity ($Q_{sen,sa}$)	10.3 kBtu/h	11.6 kBtu/h	9.7 kBtu/h	10.8 kBtu/h
	CRCP Cooling Load ($Q_{sen,p}$)	12.9 kBtu/h	12.9 kBtu/h	13 kBtu/h	12.9 kBtu/h

Table 6: Sensible cooling load for the CRCP system.

Because the enthalpy wheel entering SA and EA conditions are determined in the previous steps, the remaining unknowns in Equations 2 and 3 are the design sensible and latent effectiveness of the enthalpy wheel at normal rotating speed (more than 20 rpm).

The design enthalpy wheel effectiveness can be found from the manufacturer wheel performance data or selection software using known parameters: desiccant material, wheel entering air conditions, face velocity, and airflow ratio (ratio of SA flow to RA flow). In general, the catalog data reflect ANSI/ARI Standard 1060-2001, *Performance Rating of Air-to-Air Heat Exchangers for Energy Recovery Ventilation Heat Equipment,*[5] rating conditions, and the manufacturers do not guarantee that their rated effectiveness values are reproduced under the non-standard design conditions. However, they provide some correction factors for engineers to consider the impact of non-standard design condition on the enthalpy wheel performance. The determination of design enthalpy wheel effectiveness is critical because it affects major equipment sizing, such as the cooling coil and chiller.

The design enthalpy wheel effectiveness values also can be estimated by the practical enthalpy wheel effectiveness correlations developed by Jeong and Mumma.[6] They proposed simple linear equations returning the sensible, latent, and total effectiveness for the silica gel or molecular sieve coated enthalpy wheel at normal rotating speed using six predetermined design perimeters including face velocity, entering SA DBT and RH, entering EA DBT and RH, and airflow ratio.

In this example, the remaining unknowns (design sensible and latent effectiveness) are estimated using the correlations for the silica gel and the molecular sieve coated enthalpy wheels found in the literature[6] (*Table 5*). It is assumed that the OA and EA flows are balanced (the airflow ratio is 1.0), and the design face velocity at the wheel inlet is 590 fpm (3 m/s). The wheel entering OA and EA temperatures and relative humidity values for the Cases 1 and 2 required for estimating the design wheel effectiveness are already determined in the previous step (*Figure 2*).

As shown in *Table 5*, the silica gel coated enthalpy wheel provides higher sensible, latent, and total effectiveness values than the molecular sieve enthalpy wheel in both cases, and is selected for this design. Consequently, the SA DBT and HR after the enthalpy wheel (State 2) are determined by Equations 2 and 3 with known enthalpy wheel sensible and latent effectiveness values for each design case (*Figure 3*).

Once the thermodynamic properties of the air before and after the cooling coil (States 2 and 3 in *Figure 3*) are known in both design cases, the design cooling coil load (Q_{cc}) can be calculated using Equation 4. The average density (ρ) of the SA conditioned by the cooling coil can be approximated to 0.075 lb/ft³ (1.2 kg/m³) in both cases. Consequently, the required cooling coil capacities for the Cases 1 and 2 are 91.1 kBtu/h or 7.6 ton (26.7 kW) and 80.9 kBtu/h or 6.7 ton (23.7 kW), respectively.

$$Q_{cc} = 0.06 \cdot \rho \dot{V}_{sa,tot} \left(h_2 - h_3\right) \tag{4}$$

where

Q_{cc} = cooling coil capacity required, kBtu/h
ρ = average supply air density, lb/ft³
$\dot{V}_{sa,tot}$ = total air supply quantity, cfm
h_2 and h_3 = SA enthalpy at States 2 and 3, Btu/lb

As discussed in the previous step, the EA stream is cooled by the sensible wheel in Case 2, while the SA stream is reheated by the recovered sensible heat to maintain the design SA DBT setpoint (55°F [12.8°C]). Finally, in Case 2, the incoming OA is precooled more by the enthalpy with lower temperature wheel entering EA compared with Case 1, and the cooling coil size is reduced about 1 ton (35.17 kW). However, this cooling coil load saving will be offset by the increased CRCP area.

Step 6: Determine sensible cooling load for the CRCP system.

In the DOAS/CRCP system, the ceiling radiant panels installed in each space should accommodate the remaining sensible load not met by the supply air from the DOAS. In this design example, the design sensible load (Q_S) for each class-

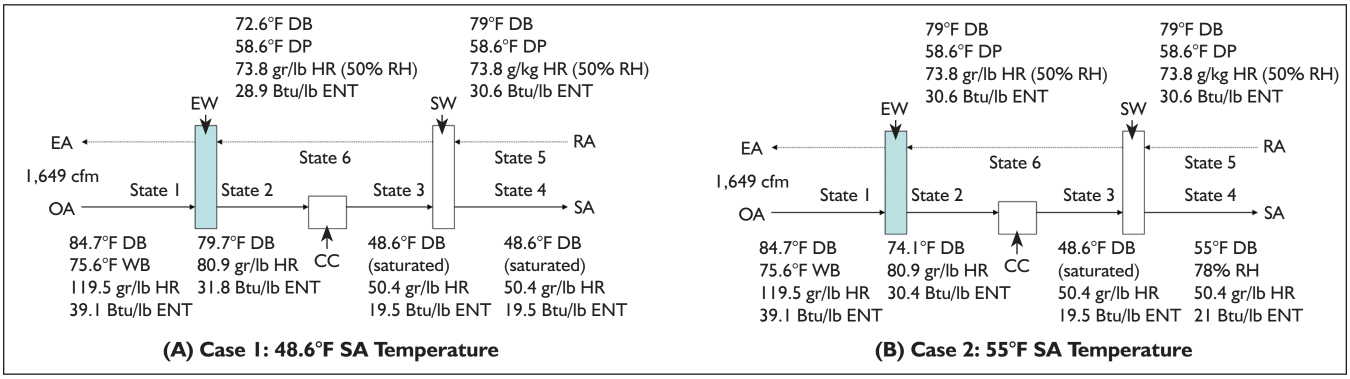


Figure 3: Supply air properties at State 2.

room was estimated in Step 3 (*Table 3*). The sensible cooling provided by the SA ($Q_{sen,sa}$) can be calculated using Equation 5. Consequently, the difference between the space sensible load and the SA cooling capacity is the sensible cooling load allocated to the CRCPs installed in each classroom (Equation 6).

$$Q_{sen,sa} = 1.08 \cdot \dot{V}_{sa}\left(T_{sp} - T_{sa}\right) \tag{5}$$

$$Q_{sen,p} = Q_s - Q_{sen,sa} \tag{6}$$

where

$Q_{sen,sa}$ = SA cooling capacity, Btu/h
$Q_{sen,sp}$ = panel sensible cooling load, Btu/h
Q_s = space sensible cooling load, Btu/h
$\dot{V}_{sa}$ = SA flow rate in each space, cfm
T_{sp} = space dry-bulb temperature, °F
T_{sa} = SA dry-bulb temperature, °F

In *Table 6,* the sensible load that should be met by the CRCP system in each space is presented. As expected, the CRCPs must accommodate more sensible load in Case 2 because of the reduced cooling capacity of the supply air (higher SA temperature).

Step 7: Determine design panel cooling capacity.

In practice, the design cooling capacity per unit panel area (Btu/h·ft² [W/m²]) is determined from the panel manufacturer's catalog data rated for the test standard, such as DIN 4715[7] and ANSI/ASHRAE Standard 138-2005, *Method of Testing for Rating Ceiling Panels for Sensible Heating and Cooling.*[8] The unit panel cooling capacity can be selected from the design capacity tables provided by the panel manufacturer based on the difference between the room temperature and the mean panel surface temperature (or mean fluid temperature). However, the manufacturer's data rated in their standard test chamber is usually 5% to 30% less than the actual capacities measured in the real space after installation[9] because the tests are performed in a test chamber under ideal conditions (mechanical ventilation and adiabatic walls). Consequently, the required panel area may be overestimated, and the initial and operating costs for the CRCP system also may be increased.

This over-design problem in the CRCP system can be avoided by estimating the design panel cooling capacity for the real operating conditions. Recently, the unit cooling capacity of top insulated panels lying on conventional false ceiling T-bar grid in a mechanically ventilated space was proposed[10] as a function of eight variables including tube spacing (w), panel thickness (δ), panel thermal conductivity (k), panel inlet chilled water temperature (T_{fi}), room temperature (T_{sp}), room position (interior or perimeter space with or without fenestration), diffuser discharge air velocity (V), and diffuser characteristic width (W).

In this example, it is assumed the top insulated aluminum panels (δ = 0.04 in. [0.001 m], k = 137 Btu/h·ft·°F [237 W/m·°C]) with 5.9 in. (150 mm) tube spacing (w) are installed in each classroom. The chilled water temperature supplied to the panel is close to the target room DPT (T_{fi} = 59°F [15°C]), and the design space temperature (T_{sp}) is 79°F (26°C) as determined previously. Assume each classroom has one exterior wall with fenestration greater than 5% of the total room surface area.

	Case 1*	Case 2**
Manufacturer's Data	31 Btu/h·ft²	31 Btu/h·ft²
Jeong and Mumma's (2004) Correlation	42 Btu/h·ft²	32 Btu/h·ft²
* 48.6°F SA DB with high induction diffusers. ** 55°F SA DB with conventional ceiling diffusers.		

Table 7: Design panel cooling capacities.

Based on the general diffuser selection procedure, 2 two-way high induction diffusers (W = 24 in. (0.6 m), V = 984 fpm [5.0 m/s]) are selected for each classroom in Case 1, and two conventional square ceiling diffusers (W = 7.9 in. [0.2 m], V = 590 fpm [3.0 m/s]) are chosen in Case 2. The design air diffusion performance index (ADPI) for high induction diffusers and conventional ceiling diffusers are 95% and 86%, respectively.

Based on the previous design conditions, the unit panel cooling capacities for each design case are determined using the manufacturer's data and the panel capacity correlation proposed in the literature[10] (*Table 7*). As expected, the correlation that considers real operating conditions (enhanced air motion around the panel caused by mechanical ventilation) gives higher design panel cooling capacity in both cases than the manufacturer's data rated under the conservative test conditions.

Step 8: Determine required CRCP area.

The required CRCP area (A_p) for each classroom is easily calculated (Equation 7) by dividing the panel sensible cooling

Design Capacity	Design Case	Item	Classroom 1	Classroom 2	Classroom 3	Classroom 4
Manufacturer's Data	Case 1*	$Q_{sen,p}$	10.1 kBtu/h	9.7 kBtu/h	10.3 kBtu/h	10.1 kBtu/h
		q_p	31 Btu/h·ft^2	31 Btu/h·ft^2	31 Btu/h·ft^2	31 Btu/h·ft^2
		A_p	329 ft^2	319 ft^2	340 ft^2	329 ft^2
		CCR	48%	46%	49%	48%
	Case 2**	$Q_{sen,p}$	12.9 kBtu/h	12.9 kBtu/h	13.0 kBtu/h	12.9 kBtu/h
		q_p	31 Btu/h·ft^2	31 Btu/h·ft^2	31 Btu/h·ft^2	31 Btu/h·ft^2
		A_p	418 ft^2	418 ft^2	428 ft^2	418 ft^2
		CCR	61%	61%	62%	61%
Jeong and Mumma's (2004) Correlation	Case 1	$Q_{sen,p}$	10.1 kBtu/h	9.7 kBtu/h	10.3 kBtu/h	10.1 kBtu/h
		q_p	42 Btu/h·ft^2	42 Btu/h·ft^2	42 Btu/h·ft^2	42 Btu/h·ft^2
		A_p	244 ft^2	237 ft^2	253 ft^2	244 ft^2
		CCR	35%	34%	37%	35%
	Case 2	$Q_{sen,p}$	12.9 kBtu/h	12.9 kBtu/h	13.0 kBtu/h	12.9 kBtu/h
		q_p	32 Btu/h·ft^2	32 Btu/h·ft^2	32 Btu/h·ft^2	32 Btu/h·ft^2
		A_p	409 ft^2	409 ft^2	420 ft^2	409 ft^2
		CCR	59%	59%	61%	59%

* 48.6°F SA DB with high induction diffusers. ** 55°F SA DB with conventional ceiling diffusers.

Table 8: Ceiling radiant cooling panel areas required.

load ($Q_{sen,p}$) estimated in Step 6 by the unit design panel capacity (q_p) determined in Step 7.

$$A_p = \frac{Q_{sen,p}}{q_p} \tag{7}$$

where

A_p = the CRCP area required, ft^2
$Q_{sen,p}$ = space sensible cooling load, Btu/h
q_p = cooling capacity of the panel, Btu/h·ft^2

As shown in *Table 8,* Case 2 requires more radiant panels in every classroom compared to Case 1 because of increased panel load caused by reduced SA cooling capacity as discussed in Step 6. Consequently, the ceiling coverage ratio (CCR), the ratio of ceiling covered by the radiant panels is higher in Case 2. In Case 1, 12% to 13% of the panel area can be saved by considering the panel capacity enhancement caused by the mechanical ventilation instead of manufacturer's conservative design capacity. Similarly, 1% to 2% of panel area savings is possible even in conventional design conditions (Case 2). It seems to be very small savings, but its impact on the total initial cost of the DOAS/CRCP system may be significant because of the relatively high price of the CRCPs.

Conclusion

In this article, a general DOAS/CRCP system design procedure was presented using a simple design example. Although a DOAS/CRCP system design tool consisting of reliable design data, verified simulation models, and more systematic design guide is still missing, the eight simple steps for DOAS/CRCP system design presented in this work would be useful to engineers considering a DOAS/CRCP system and being forced to design the DOAS/CRCP system on the basis of qualitative merits, limited experience, and conservative estimates.

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