Selecting the Supply Air Conditions for a Dedicated Outdoor Air System Working in Parallel with Distributed Sensible Cooling Terminal Equipment

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ABSTRACT

The central thrust of this paper is to place into question the current practice of supplying air from dedicated outdoor air systems at or near room temperature (a neutral temperature) and to develop a methodology for selecting the supply air conditions in an energy- and cost-effective manner. Hypotheses are advanced concerning the supply air dry-bulb temperature, dew-point temperature, and terminal reheat. The three hypotheses are then tested and found to be correct. In general, it is recommended that the supply air temperature from the dedicated outdoor air system be no higher than 55°F (13°C). The recommended supply air dew-point temperature is whatever it takes to provide all of the latent cooling while maintaining the space relative humidity at no more than 40%, or a supply air dew-point temperature of approximately 44°F (7°C). Finally, it was demonstrated that terminal reheat is generally not required to prevent over-cooling with 44-55°F (7-13°C) supply air dry-bulb temperatures for spaces with a combined lighting and equipment load of 3-5 W/ft^2 (32-54 W/m^2). This conclusion applies to spaces with design occupancy densities from 7 to more than 90 people per 1000 ft^2 (93 m^2). Greater space design occupancy densities have more ventilation air than required to remove the 3-5 W/ft^2 (32-54 W/m^2) of internal generation. Supplying the air at a neutral temperature shifts virtually all of the space sensible loads out onto the distributed parallel cooling system at a huge first and operating cost penalty, a practice that normally cannot be justified. Automatic controls, a subject beyond the scope of this paper, are envisioned to offer the potential to further improve the economic benefits of the lower supply air conditions.

INTRODUCTION

The engineering design community appears to be on the verge of a major shift in paradigm concerning the delivery of ventilation air because of the significant economic, comfort, and health benefits brought about by new ways of integrating the equipment with the building. A complete overview of this entire integrated concept is presented in a companion paper by Mumma (2001a). The ventilation air is delivered by a separate dedicated outdoor air system (DOAS) designed to efficiently remove the ventilation air latent load as well as 100% of the space latent loads (Mumma and Shank 2001). The terminal equipment, operating in parallel with the DOAS equipment, is required to remove only the sensible loads that remain after the dry ventilation air has been introduced into the conditioned space. The parallel equipment envisioned includes fan coil units, ceiling radiant cooling panels, water-source heat pumps, and a parallel variable air volume (VAV) system operating on 100% recirculated air. Only fan coil units and ceiling radiant cooling panel equipment will be further investigated in this paper.

Articles on this subject (Scofield and Des Champs 1993; Brady 1997) appear in the trade literature, manufacturers' literature, and web sites. In all cases that the authors are aware of, the ventilation air is supplied to the conditioned space with dew-point temperatures sufficiently low to remove some or all of the space latent loads and at a neutral dry-bulb temperature. The archival literature is silent on the technical issues surrounding the selection of the supply air thermal conditions. To provide the dehumidified air at a neutral temperature, the DOAS equipment (Figure 1) must provide considerable reheat. By using both sensible and total energy heat recovery

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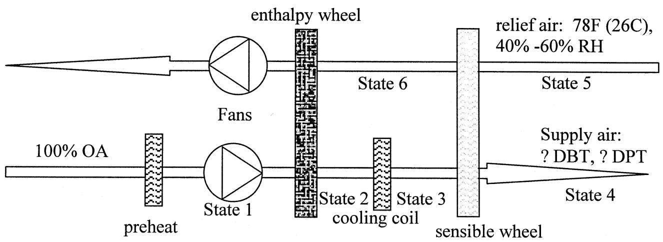

Figure 1 *DOAS equipment configuration.*

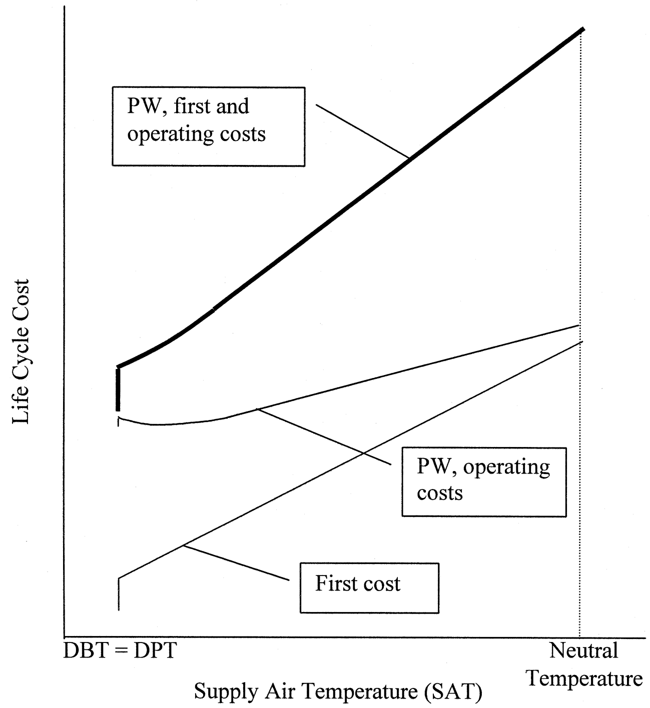

Figure 2 *Hypothetical life-cycle cost, as a function of the supply air temperature.*

equipment in the DOAS configuration, the outside air (OA) load on the cooling coil is greatly diminished and no heating energy is expended to accomplish the reheat. In effect, a neutral supply air temperature shifts virtually all of the space sensible load to the parallel terminal equipment. Under this typical operating strategy, only the OA and space latent loads are borne by the DOAS.

This paper will introduce three hypotheses and attempt to support them.

Hypothesis One

The first hypothesis concerns the selection of the supply air dry-bulb temperature. It is hypothesized that supply air temperatures much lower than the customary neutral, even as low as the supply air dew-point temperature, will result in a major reduction in the first cost of the entire heating, ventilating, and air-conditioning (HVAC) system.

Part (a) of the first hypothesis predicts that the first cost, as a function of supply air temperature, for a given supply air dew-point temperature, will drop as the DOAS supply air temperature (SAT) drops, as depicted in Figure 2. The prediction that the first cost will drop as the SAT drops is based solely upon the reduced size of the parallel terminal equipment. Shifting the sensible load to or from the DOAS is hypothesized to have minimal, if any, impact on the size of the cooling plant or the DOAS equipment costs. However, when the SAT and the dew-point temperature (DPT) are equal (no heat recovery reheat), the cost of the DOAS unit can be reduced by the cost of the sensible heat recovery device, and this is reflected by the vertical first cost drop at that point in Figure 2.

Part (b) of the first hypothesis predicts that a drop in SAT will be accompanied by a drop in the operating costs, at least initially, as depicted in Figure 2. Since the enthalpy wheel is less than 100% effective, only a portion of the energy removed from the return air by the sensible wheel during reheat can be recovered at the enthalpy wheel. This will be discussed in more detail later in the paper. The change in the operating cost curve slope at low SATs reflects the possibility that, in some applications and situations, terminal reheat may be required. (*Note*: terminal reheat done at the spaces to avoid local overcooling is not to be confused with the heat recovery reheat performed by the sensible wheel in the DOAS unit). The sharp vertical dip at the point where the dry-bulb temperature (DBT) and the DPT are equal reflects a drop in the fan power when the sensible wheel is removed. The consequence of the first hypothesis is a prediction that the life-cycle cost will follow a trend, as depicted in Figure 2.

Hypothesis Two

The second hypothesis concerns the impact that lowering the supply air DPT has on the performance of both the DOAS equipment and the parallel sensible cooling equipment. Good practice (Sterling et al. 1985) suggests that the space relative humidity be maintained between 40% and 60%. With a summer space design temperature of 78ºF (26ºC) (ASHRAE 1999), the range of room DPTs is only 52-63ºF (11-17ºC). Since the ventilation air supplied by the DOAS unit is designed to remove the entire space latent load, its supply air DPT must be lower than the desired room DPT. The supply ventilation air DPT can be easily determined, given a knowledge of the space latent loads and the ventilation air flow rates.

Part (a) of the second hypothesis predicts that lowering the space design DPT will increase the first cost of the chiller and may increase its operating cost. These predictions are based upon a knowledge of chiller performance. Lowering the chilled water temperature below the standard 45ºF (7ºC) derates the chiller and may increase the kW/ton to operate it (pumping costs not included).

Part (b) of the second hypothesis predicts that even an 11ºF (6ºC) reduction in the space design DPT, and, hence, the temperature of the chilled water serving the terminal equipment, will substantially reduce the size of the terminal equipment and the associated first and transport energy costs. The rate of heat transfer for the two parallel terminal sensible cooling equipment types considered in this paper, fan coil units and ceiling radiant cooling panels, is nearly proportional to the temperature difference between the inlet fluid temperature (dictated by the space DPT) and the space design temperature.

Hypothesis Three

The third and final hypothesis concerns the need for, and the energy use of, terminal reheat. It is hypothesized that for many of the most common design occupancy densities currently found in buildings, terminal reheat will be needed sparingly, if at all. The relationship between the envelope loads and the sensible internal generation from lighting and equipment necessary to just balance the sensible cooling that is performed by the required design occupancy-based ventilation air flow rates will be explored later in the paper. In addition, this paper will attempt to make a case for not supplying the DOAS ventilation air at temperatures above 55ºF (13ºC), currently a common practice in all-air VAV systems.

The authors wish to emphasize that this paper is not intended to prescribe a specific set of supply air conditions for DOAS-parallel sensible cooling applications. Rather, it is intended to offer guidance to the design engineering community in selecting the supply air DBT and DPT. Later in the paper, where the three hypotheses are analyzed, a general methodology for selecting these variables will emerge. The analysis portion of this paper will only address some of the major interrelated issues in the decision-making process. The role of optimal control will not be addressed in the paper. Neither are winter operating conditions addressed, except as they impact terminal reheat and water side free cooling. These topics have been omitted primarily because the DOAS and terminal sensible cooling equipment selection is most profoundly impacted by summer operating conditions.

DEDICATED OUTDOOR AIR SYSTEM OPERATION OVERVIEW

A detailed presentation of the DOAS operation is documented in two other ASHRAE papers (Mumma and Shank 2001; Mumma 2001b). Therefore, only an overview will be provided in this paper, with the aid of the psychrometric chart. The DOAS components operate in a manner appropriate for the OA conditions. The possible OA conditions are illustrated

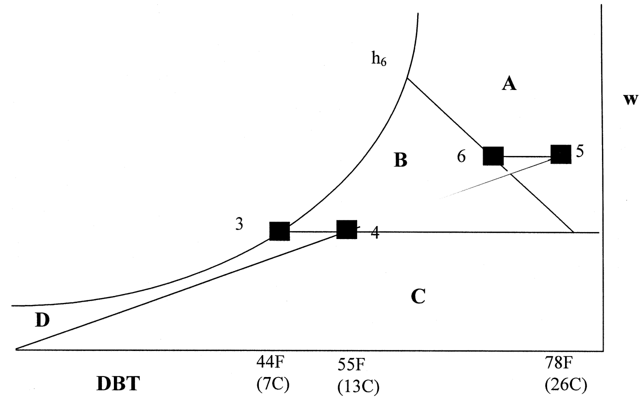

Figure 3 *Four regions on the psychrometric chart.*

on the psychrometric chart in Figure 3 and assigned four regions, A, B, C, and D. Table 1 identifies how the DOAS components are to operate when the OA is in any one of the four regions. Since this paper is focused upon the summer design conditions and operation, discussion will be limited to regions A and B.

When the OA is in region A, the enthalpy wheel does a very effective job of precooling and dehumidifying the OA to a condition close to that leaving the sensible wheel. The preconditioning substantially reduces the cooling load seen by the chiller and the deep cooling coil. In region B, only free reheat is supplied by the sensible wheel when the desired SAT is greater than the desired supply air DPT.

SELECTION OF THE DOAS SUPPLY AIR DRY-BULB TEMPERATURE

This section will provide support for hypothesis 1, which predicted first and operating cost benefits for SATs well below neutral. In order to provide a concrete example, a 10,000 scfm (5.5 kg/s) DOAS unit is investigated, based upon Atlanta, Ga., hourly typical meteorological year (TMY) data. The illustration is based on a six-day week, excluding Sundays, and 12-hour days starting at 7 a.m. and ending at 7 p.m. Further, this illustration is based upon a constant supply air DPT of 44ºF (7ºC). A 44ºF (7ºC) supply air DPT results in a 52ºF (11ºC) space DPT, assuming 20 scfm (0.01 kg/s) of OA per person

TABLE 1
Control Status of the DOAS Equipment

Region	Enthalpy wheel CTL	Cooling coil CTL	Sensible wheel CTL
A	100% speed for max. effectiveness	Modulate to hold 44ºF (7ºC) LAT	Modulate to hold55ºF (13ºC) SAT
B	Off! Must not modulate	Modulate to hold 44ºF (7ºC) LAT	Modulate to hold 55ºF (13ºC) SAT
C	Modulate to required DPT	Modulate to hold 55ºF (13ºC) LAT	Will modulate off
D	Modulate to required DPT	Will modulate off	Modulate to hold 55ºF (13ºC) SAT

and that all of the latent loads are from human occupants. Finally, for this illustration, it is assumed that the return air conditions are 78ºF (26ºC) DBT and 40% RH.

A model simulating the performance of the DOAS unit (Mumma and Shank 2001) was used to compute the peak design and annual results presented in Table 2. The simulation was performed assuming identical supply air DPTs for three different supply air dry-bulb temperatures. Column 1 of Table 2 identifies the three SATs as 70ºF (21ºC) (26ºF [14ºC] of reheat with the sensible wheel), 55ºF (13ºC) (11ºF [6ºC] of reheat with the sensible wheel), and 44ºF (7ºC) (no sensible reheat or wheel). As expected, the total ton hours of cooling needed at the cooling coil is greatly reduced as the supply air temperature is elevated. Column two of Table 2 contains those data. The cooling coil ton hours (TH) for the 70ºF (21ºC) SAT are only 43% as much as when no reheat is added with the sensible wheel (i.e., 44ºF [7ºC] SAT). The 55ºF (13ºC) SAT requires only 75% as many TH of cooling as the 44ºF (7ºC) SAT case. Column 3 illustrates the sensible cooling that is available from 10,000 scfm (5.5 kg/s) of air at the various SATs when the space temperature is maintained at 78ºF (26ºC). As expected, when the temperature difference between the supply air and the room air increases, the sensible cooling capability of the supply air increases. With the SATs chosen, the supply air temperature to room air temperature difference is as low as 8ºF (4ºC) and as high as 34ºF (18ºC), making the sensible cooling capability vary by a factor of 425%. For now, it is assumed that no terminal reheat is required for any of the three SATs. To provide an equivalent basis for comparison, a 70ºF (21ºC) SAT will require that the parallel terminal cooling equipment (column 4 of Table 2) remove 87,600 TH (308,000 kWh) more sensible heat from the space than is required with the 44ºF (7ºC) SAT. In the case of 55ºF (13ºC) SAT, the parallel terminal cooling equipment must remove 37,100 TH (130,400 kWh) more sensible heat from the space than would be required with the 44ºF (7ºC) SAT. Column 5 of Table 2 summarizes the TH of cooling that must be provided. It is the summation of the DOAS annual TH and the parallel system TH for each SAT to work against the same space sensible cooling loads. Contrary to what one might initially think, decreasing the SAT reduces the total annual TH of cooling required. And the difference is not trivial. A 70ºF (21ºC) SAT requires 13% more TH than a 55ºF (13ºC) SAT and 23% more than 44ºF (7ºC) SAT. And, the 55ºF (13ºC) SAT requires 9% more than a 44ºF (7ºC) SAT requires. These numbers are based upon an assumed enthalpy wheel effectiveness (ASHRAE 1996) of 0.85. The percent differences would be even greater if the enthalpy wheel effectiveness were less. The peak load on the DOAS cooling coil, with an OA condition of 84ºF (29ºC) and 147gr/lb (0.021 kg/kg) humidity ratio (conditions with the highest enthalpy during the occupied hours in Atlanta), is presented in column 6 of Table 2. As expected, the peak cooling coil (CC) load decreased with increasing SAT.

This is a good place to demonstrate why the energy consumption increases (column 5 of Table 2) as the reheat by the sensible wheel increases. Notice that the peak load on the CC with the 44ºF (7ºC) SAT (no sensible wheel) is 47 ton (165 kW). In the case of 70ºF (21ºC) SAT, the 44ºF (7ºC) air leaving the cooling coil is sensibly elevated to 70ºF (21ºC) by the sensible wheel. At the same time, the 78ºF (26ºC) return air is sensibly cooled by the same amount or by 23.4 ton (82 kW). Since the enthalpy wheel effectiveness is only 0.85, the cooling coil load could only

TABLE 2
Annual Cooling and Peak Design Loads for Three DOAS SAT, Supply DPT = 44ºF (7ºC)

Col. 1	2	3	4	5	6	7	8
SAT ºF (ºC)	Annual cooling required at the DOAS TH (kWh)	Annual space sensible cooling available from the DOAS supply air TH (kWh)	Difference between the sensible cooling available from the DOAS at the given SAT and a 44ºF (7ºC) SAT TH (kWh)	Combined cooling at the DOAS coil and the cooling that must be done at the terminal to make up the difference between the given SAT temp. and 44ºF (7ºC). TH (kWh)	Peak CC load on the DOAS, ton (kW)	Peak load on the terminal equipment to meet the same peak load as the 44 SAT case could meet. tons (kW)	Total peak load ton (kW)
70 (21), Neutral	48,150 (169,000)	27,000 (95,000)	87,600 (308,000)	135,750 (477,300)	27 (95)	22.5 (79)	49.5 (174)
55 (13)	83,100 (292,000)	77,500 (272,500)	37,100 (130,400)	120,200 (422,600)	39 (137)	9 (32)	48 (169)
44 (7)	110,700 (389,000)	114,600 (403,000)	0	110,700 (389,000)	47 (165)	0	47 (165)

be reduced by 20 ton (70 kW). The equivalent of about 3.4 ton (12 kW) of available cooling has been lost in this reheat process.

Ten thousand standard cubic feet per minute (5.5 kg/s) of ventilation air, supplied to a 78ºF (26ºC) space from the DOAS unit at 70ºF (21ºC), 55ºF (13ºC), or 44ºF (7ºC) will provide 7.2 ton (25 kW), 20.7 ton (73 kW), or 29.7 ton (104 kW) of sensible cooling, respectively. Column 7 of Table 2 shows the deficiency in sensible cooling capacity of the ventilation air at various supply air temperatures compared to 44ºF (7ºC). A comparison of the sum of the peak DOAS CC loads and the deficient sensible cooling load that must be supplied by the parallel system for equivalency (column 8 of Table 2), illustrates that SAT has little impact on the chiller size. Although lowering the SAT is slightly favored.

This illustration supports hypothesis 1, which predicted that lowering the SAT would reduce both first cost and energy consumption. These findings are based on the assumption that terminal reheat was not necessary, a subject to be addressed later under the section entitled "The Role of Terminal Reheat on Selecting the DOAS Supply Air Conditions."

PARALLEL SENSIBLE COOLING EQUIPMENT FIRST COST DIFFERENCE ESTIMATES

As presented above, 10,000 scfm (5.5 kg/s) of air at 55ºF (13ºC) can remove 13.5 ton (47 kW) more sensible heat than air at 70ºF (21ºC). And 44ºF (7ºC) air can remove 22.5 ton (79 kW) more sensible heat than air at 70ºF (21ºC). Assume that fan coil units, designed to remove 19 Btu/h per cfm (11.7 W per m^3/s) and cost approximately \$6/cfm (\$12,600/m^3/s) are used as the parallel terminal units. With a 55ºF (13ºC) SAT from the DOAS unit, the cost of the fan coil installation could be reduced by \$51,200. With a 44ºF (7ºC) SAT from the DOAS unit, the cost of the fan coil installation could be reduced by \$85,300. Next, assume ceiling radiant cooling panels, costing approximately \$8/ft^2 (\$86/m^2) and capable of removing approximately 30 Btu/h·ft^2 (95 W/m^2) of heat (Conroy and Mumma 2001), are used as the parallel terminal units. With a 55ºF (13ºC) SAT from the DOAS unit, the cost of the ceiling radiant cooling panel installation could be reduced by \$43,200. With a 44ºF (7ºC) SAT from the DOAS unit, the cost of the fan coil installation could be reduced by \$72,000. These appear to be significant savings in what might be about a 60,000 ft^2 (5600 m^2) building (the approximate size of an office building requiring 10,000 scfm [5.5 kg/s] of ventilation air).

Based upon the very small difference in peak loads for the different supply air temperatures, no difference in the chiller first cost is realized.

Finally, if the sensible heat recovery wheel could be eliminated from the DOAS unit, a first cost savings of approximately \$2/scfm (\$3600/kg/s) might be expected. This translates to a first cost savings of about \$20,000 in a 10,000 scfm (5.5 kg/s) system.

Based upon the potential first cost savings and associated operating cost savings, there is a strong incentive to abandon the practice of supplying air from dedicated outdoor air systems at a neutral temperature.

SELECTION OF THE DOAS SUPPLY AIR DEW-POINT TEMPERATURE

Hypothesis 2, introduced above, predicted that lowering the supply air DPT would have a negative impact on the size and operating cost of the chiller serving the building and a positive impact on the first cost and operating expenses of the parallel terminal cooling equipment. In the section above, dealing with the selection of the supply air DBT, the building loads were not a factor so long as it was assumed that the sensible cooling loads were sufficient to avoid terminal reheat. However, selection of the DPT must consider the entire building peak design load and annual THs. By way of illustration, consider that the 10,000 scfm (5.5 kg/s) of ventilation air, used as an example throughout this paper, is serving a building with a peak design sensible cooling load (does not include the OA loads) of 75 ton (264 kW). The impact of selecting equipment within the rather narrow range of room dew-point temperatures of 52-63ºF (11-17ºC) (corresponds to a 78ºF [26ºC] DBT and 40-60% RH) will be illustrated next, following a characterization of a chiller, fan coil units, and ceiling radiant cooling panels.

Typical Water-Cooled Chiller Performance

The capacity and energy use rate for a commercially available water-cooled chiller is as depicted in Figure 4

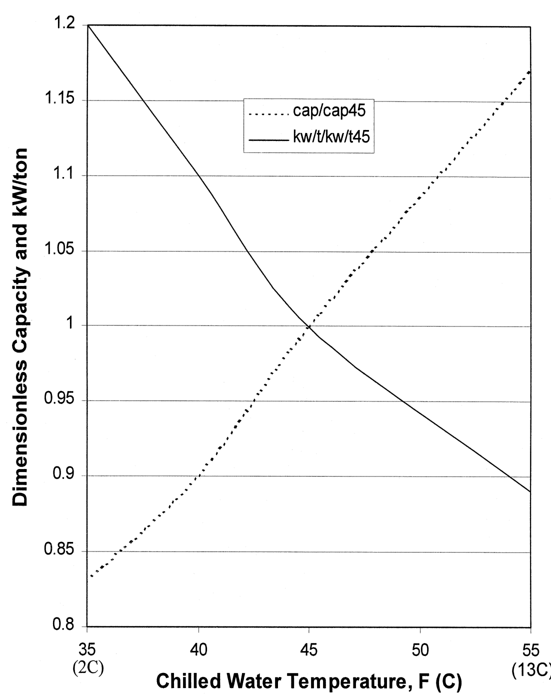

Figure 4 *Typical water-cooled chiller performance.*

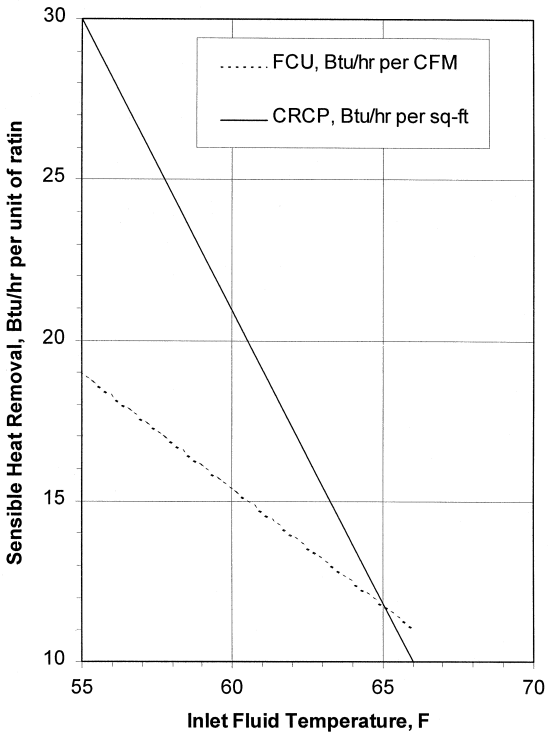

Figure 5 *Typical fan coil and ceiling radiant cooling panel (CRCP) performance in a 78ºF (26ºC) room.*

(Carrier 1998). If it is assumed that the 52ºF (11ºC) room design DPT is achieved with 40ºF (4ºC) chilled water at the DOAS, it may be noted from Figure 4 that the chiller is derated 10% when compared to a 45ºF (7ºC) chilled water temperature. It may also be noted that for this example, the kW/ton (dashed line) has increased by 10% (from 0.72 to 0.79 kW/ton) when the chilled water temperature is 40ºF (4ºC) rather than 45ºF (7ºC). It will be assumed that the 63ºF (17ºC) room design DPT is achieved with 45ºF (7ºC) chilled water. Care is needed in selecting the DOAS cooling coil to produce 44ºF (7ºC) air with 40ºF (4ºC) water; however, it is possible.

Typical Fan Coil Performance

For the upper and lower bounds of the space DPT, it is assumed that the fan coil can be fed with chilled water at 66ºF (19ºC) and 55ºF (13ºC), respectively, without condensation on the piping or the cooling coils. Typical cooling capacity as a function of inlet fluid temperature for a fan coil unit performing sensible-only cooling in a space at 78ºF (26ºC) is depicted in Figure 5 (dashed line). The capacity of a given size fan coil unit drops precipitously as the chilled water inlet temperature is elevated. In this case, it drops from 19 Btu/h per cfm (11.7 W per m^3/s) of airflow at a 55ºF (13ºC) inlet fluid temperature to 11 Btu/h per cfm (6.8 W per m^3/s) of airflow at a 66ºF (19ºC) inlet fluid temperature. That represents a 42% derating of the equipment.

Typical Ceiling Radiant Cooling Panel Performance

For the upper and lower bounds of the space DPT, it is assumed that the ceiling radiant cooling panel can be fed with chilled water at 66ºF (19ºC) and 55ºF (13ºC), respectively. Typical cooling capacity (Conroy and Mumma 2001), as a function of inlet fluid temperature, for a ceiling radiant cooling panel performing sensible-only cooling in a space at 78ºF (26ºC) is depicted in Figure 5 (solid line). The capacity of a unit area of ceiling radiant cooling panel drops precipitously as the chilled water inlet temperature is elevated, in this case from 30 Btu/h per ft^2 (95 W/m^2) at a 55ºF (13ºC) inlet fluid temperature to 10 Btu/h per ft^2 (32 W/m^2) at a 66ºF (19ºC) inlet fluid temperature. That represents a 67% derating of the equipment. Clearly, ceiling radiant cooling panels are more sensitive to increases in inlet water temperature than fan coil units.

Estimating the First and Operating Cost Differences for the Chiller and Parallel Sensible Cooling Equipment with a Constant 55ºF (13ºC) SAT and Varying Supply DPT

To facilitate this analysis, Table 3 was developed. Some of the data in columns 2 and 3 are repeated for convenience from Table 2. However, a new row has been added in Table 3 to reflect the performance of a DOAS unit supplying saturated 55ºF (13ºC) air without the sensible wheel and represents a new supply air DPT (55ºF [13ºC]). The data in column 4 are based upon an annual energy simulation for the hypothetical building requiring 75 ton (264 kW) of sensible cooling at design. That hypothetical building required a total of 217,000 TH (763,000 kWh) of sensible cooling for the year in Atlanta. Therefore, the values in column 4 of Table 3 are the difference between the annual sensible cooling THs provided by the DOAS unit (presented in column 3) and the building total sensible cooling requirements. That difference represents the annual sensible cooling that must be borne by the parallel sensible cooling equipment. The values in column 5 are the summation of the TH of cooling required at the DOAS unit, column 2, and that borne by the parallel equipment, column 4. Therefore, column 5 is the total TH of cooling that must be provided by the chiller and will be used later to compute the operating cost estimates in combination with the data in Figure 4. The values in columns 6-8 are peak load values that will be used later to compute the first cost estimates in combination with the data in Figures 4 and 5.

The information presented in Table 3 and Figures 4 and 5 is used to produce Table 4. The first three rows of Table 4 apply to the chiller. Column 3 of Table 4 presents the first cost of the chiller, obtained by first dividing the design chiller load in column 2 by its rating, relative to a 45ºF (7ºC) chilled water temperature, from Figure 4 (solid line). That value is then multiplied by the assumed first cost per rated ton of \$1000 (\$3516/kW). For example, row 3, column 4, of Table 4 is (91 ton/0.9) · \$1000/ton = \$101,000.

TABLE 3

Annual Cooling and Peak Design Loads for Two Supply Air DPT, SAT 55ºF (13C)

Col. 1	2	3	4	5	6	7	8
Supply DPT/DBT, ºF (ºC)	**Annual cooling required at the DOAS, TH (kWh)**	**Annual space sensible cooling available from the DOAS supply air, TH (kWh)**	**Difference between the sensible cooling available from the DOAS at the given SAT & the building sensible load, 217,000 TH, (763,000 kWh) TH (kWh)**	**Combined cooling at the DOAS coil and the cooling that must be done at the terminal to make up the difference between the given SAT temp. and 44ºF (7ºC), TH (kWh)**	**Peak CC load on the DOAS, ton (kW)**	**Peak load on the terminal equipment to meet the peak load, ton (kW)**	**Total peak load, ton (kW)**
55/55 (13/13)	72,000 (253,000)	77,500 (272,500)	139,500 (490,500)	211,500 (743,600)	39 (137)	54 (190)	93 (327)
55/44 (13/7)	83,100 (293,000)	77,500 (272,500)	139,500 (490,500)	222,600 (782,700)	39 (137)	54 (190)	93 (327)
Note: New DBT below							
44/44 (7/7)	110,700 (389,000)	114,600 (403,000)	102,400 (360,000)	213,100 (749,300)	47 (165)	44 (155)	91 (320)

TABLE 4

First and Operating Cost Estimates for the Chiller and Terminal Cooling Equipment

Col #	1	2	3	4	5
Row #	**DBT/DPT Supply, ºF (ºC)**	**Design chiller load, Ton (kW)**	**Chiller first cost at $1000/ton-rated capacity**	**Chiller operating energy consumption, based upon TH from Table 3, kWh**	**Annual operating cost, based upon $0.09/kWh**
1	55/55 (13/13)	93 (327)	$93,000	148,000	$13,300
2	55/44 (13/7)	93 (327)	$103,000	176,000	$15,800
3	44/44 (7/7)	91 (320)	$101,000	168,000	$15,000
	DBT/DPT Supply, ºF (ºC)	**Fan coil size based upon design load and capacity at the design space DPT, cfm (m^3/s)**	**Fan coil cost at $6/cfm ($12,600/m^3/s)**	**Fan coil operating kWh assuming 2 in.w.g. (500 Pa) pressure drop and 3744 operating hours**	**Annual operating cost, based upon $0.09/kWh**
4	55/55 (13/13)	59,000 (28)	$354,000	70,200	$6300
5	55/44 (13/7)	34,100 (16)	$204,600	40,600	$3600
6	44/44 (7/7)	28,000 (13)	$168,000	33,300	$3000
	DBT/DPT Supply, ºF (ºC)	**Radiant panel size based upon design load and capacity at the design space DPT, ft^2 (m^2)**	**Radiant panel cost at $8/$ft^2$ ($86/$m^2$)**	**Radiant panel pumping energy, kWh, assuming a 30-ft head (90 kPa) and a 5ºF (3ºC) delta T for 3744 operating hours**	**Annual operating cost, based upon $0.09/kWh**
7	55/55 (13/13)	64,800 (6000)	$518,000	7000	$630
8	55/44 (13/7)	21,600 (2000)	$173,000	7000	$630
9	44/44 (7/7)	17,600 (1640)	$141,000	5600	$504

The annual energy consumption, column 4 of Table 4, is the product of the appropriate column 5 row from Table 3 and the kW/ton for the appropriate chilled water temperature. For example, the value in row 2, column 4, of Table 4 is 222,600 ton-h · 0.79 kW/ton = 176,000 kWh.

The values in column 5 are based upon an assumed simple energy rate of $0.09/kWh.

The next three rows, 4-6, focus on the first and operating costs of the fan coil units serving as the parallel sensible-only cooling units. The size of the fan coil units (i.e., air flow rates) is obtained by dividing the peak load that the terminal equipment must handle selected from the appropriate row of column 7 of Table 3 by the capacity for the appropriate entering chilled water temperature (either 55ºF [13ºC] or 66ºF [19ºC]) from Figure 5. For example, the value in row 4, column 2, is (54 ton · 12,000 Btu/h/ton)/11 Btu/h per cfm = 59,000 cfm (28 m^3/s).

The fan coil unit's fan motor annual operating kWh calculation is based on an assumed fan efficiency of 74%, a fan pressure differential of 2 in.w.g. (500 Pa), and continuous operation for the entire 3744 hours of occupancy. The pumping energy associated with the fan coil units, while not computed here, is expected to be about the same as for the ceiling radiant cooling panels to be discussed next.

The next three rows, 7-9, focus on the first and operating costs of the ceiling radiant cooling panel units serving as the parallel sensible-only cooling units. The total area of ceiling radiant cooling panels is obtained by dividing the peak load that the terminal equipment must handle from the appropriate row of column 7 of Table 3 by the capacity for the appropriate entering chilled water temperature (either 55ºF [13ºC] or 66ºF [19ºC]) from Figure 5. For example, the value in row 8, column 2, Table 4, is (54 ton · 12,000 Btu/h/ton)/30 Btu/h/ft^2 = 21,600 ft^2 (2000 m^2).

The ceiling radiant cooling panel annual pump operating kWh calculation is based on an assumed pump efficiency of 80%, a pump pressure differential of 30 in. w.g. (90 kPa), a 5ºF (3ºC) temperature rise, and continuous operation for the entire 3744 hours of occupancy.

The first and operating cost estimate data presented in columns 3 and 5 of Table 4 support the second hypothesis, which predicted that the chiller's first and operating costs would increase with decreasing DOAS supply air DPTs. It also supports the prediction that the size and annual operating costs of the parallel sensible cooling equipment both decline with decreasing DOAS supply air DPTs. The magnitude of savings in first and operating costs for the parallel terminal sensible cooling equipment greatly overcome the negative impact of lowering the DOAS supply air DPTs on the chiller. Therefore, the design goal should be to maintain the space DPT as low as would be considered good practice or about 52ºF (11ºC).

THE ROLE OF TERMINAL REHEAT ON SELECTING THE DOAS SUPPLY AIR CONDITIONS

ANSI/ASHRAE Standard 90.1-1999 (ASHRAE 1999) does not permit terminal reheat except where required to meet *ANSI/ASHRAE Standard 62-1999* (ASHRAE 1999). This provision of Standard 90.1 explains why nearly all VAV systems use terminal reheat, at least for the perimeter zones, to avoid overcooling at off design, low cooling load conditions. When VAV box minimums are set to ensure ventilation (i.e., the VAV boxes are not allowed to shut off), terminal reheat is infrequently required at off design operating conditions to avoid overcooling. This practice is common for VAV systems designed to supply 55ºF (13ºC) air, as well as for low temperature systems designed to supply air from 40ºF to 50ºF (4ºC to 10ºC). Few VAV systems are designed with supply air temperature reset to avoid terminal reheat because, as currently practiced (Ke and Mumma 1997), there are essentially no energy-saving benefits.

Given the current situation, the authors see that VAV systems supplying air at 55ºF (13ºC) or lower, employing terminal reheat with fixed minimum VAV box settings to ensure ventilation would always be supplying more air than a DOAS unit employing terminal reheat. The required box minimum in the case of VAV systems would always be set to deliver a greater volumetric flow rate at the fixed minimum box settings (due to the multiple spaces equation and the critical Z of Standard 62) and a constant volume DOAS unit. Because of the severe first and operating cost penalties associated with supplying the DOAS air at a neutral temperature and the current practice of using terminal reheat in VAV systems, it is very difficult to justify designing a DOAS-parallel system with a supply air temperature above 55ºF (13ºC). Controls, a subject that is beyond the scope of this paper, could easily be used to further improve the economic benefits of the lower supply air conditions.

Since VAV systems frequently employ terminal reheat, let us explore design occupancy densities below which terminal reheat is not required. Focus is placed on systems where the minimum airflow to the space is dictated by the ventilation requirements. Currently, *ANSI/ASHRAE Standard 62-1999* requires some activities to receive 20 scfm (0.01 kg/s) per person and others 15 scfm (0.008 kg/s) per person of ventilation air. Standard 62 also lists the maximum design occupancy per 1000 ft^2 (93 m^2) for various activities. When the airflow to a space is at the minimum to satisfy the design ventilation requirements, it is possible to determine the envelope and sensible internal generation loads that to just balance the cooling capability of that air. Internally dominated buildings that comply with *ANSI/ASHRAE Standard 90.1-1999* generally have a UA value of approximately 0.09 Btu/h·ºF (0.05 W/ºC). If this were applied in an energy balance with assumed summer and winter OA design temperatures of 90ºF (32ºC) and 20ºF (–7ºC), respectively, the internal generation from lights and equipment required to make terminal reheat of the DOAS supplied air unnecessary can be computed. That has been done for the various design occupancy figures that appear in *ANSI/ASHRAE Standard 62-1999* and is presented in Figure 6. The figure illustrates a summer (dashed lines) and winter (solid lines) design condition, with 44ºF (7ºC) and 55ºF (13ºC) supply air temperatures, with design ventilation flow rates of 15 and 20 scfm (0.01 kg/s) per person design occupancy, and with the spaces occupied and unoccupied (16 combinations). The worst possible case is line “L” (the one on top). It is a winter condition, so the envelope is providing sensible cooling, the supply airflow rate is based on 20 scfm (0.01 kg/s) per person, the SAT is 44ºF (7ºC), and there is nobody in the space. Under these worst conditions, it may be noted that the balance point design occupancy density (and, hence, the supply air flow rate per ft^2 of building) at 3 W/ft^2 (32 W/m^2) (internal generation from lighting and equipment) is about 7 people/1000 ft^2 (93 m^2) and at 5 W/ft^2 (54 W/m^2) is about 17 people/1000 ft^2 (93 m^2). Standard 62 notes that 7 people/1000 ft^2 (93 m^2) is the maximum expected in office buildings. If the spaces were under the same conditions as just discussed, but with the design occupancy present (line “K”), the balance point design occupancy density at 3 W/ft^2 (32 W/m^2) is about 10 people/1000 ft^2 (93 m^2) and at 5 W/ft^2 (54 W/m^2) is about 25 people/1000 ft^2 (93 m^2). Elevating the SAT to 55ºF (13ºC), and otherwise maintaining the same conditions as discussed above, the unoccupied and occupied balance point design occupancy density at 3 W/ft^2 (32 W/m^2) are about 10 and 20 people/1000 ft^2 (93 m^2) and at 5 W/ft^2 (54 W/m^2) are about 25 and 47 people/1000 ft^2 (93 m^2). The balance point design occupancy is much higher in the summer when the envelope contributes to the cooling load. The balance point design occupancy is also higher with the lower 15 scfm per person ventilation rate. Clearly, there are many situations where terminal reheat will not be a significant energy user if required at all.

Consequently, for many building applications encountered, terminal reheat will seldom, if ever, be necessary with the DOAS-parallel system approach at supply air temperatures of 55ºF (13ºC) or even as low as 44ºF (7ºC). Consequently, the old paradigm of supplying the OA at a neutral temperature needs to be reconsidered.

CONCLUSIONS AND RECOMMENDATIONS

The DOAS supply air conditions have been addressed in this paper. Currently, nearly all systems employing dedicated OA systems supply air at or near neutral thermal conditions. The intent of this paper was to challenge that practice. At the outset of the paper, three hypotheses were advanced. The first hypothesis predicted that for a given supply air dew-point temperature, supply air dry-bulb temperatures below neutral would result in both a lower first and operating energy costs. That hypothesis was confirmed. Based upon the results presented in Table 4, combining a low SAT and a low DPT provides the best first and operating cost option. The second hypothesis predicted that for a given supply air dry-bulb condition, supply air DPTs that would provide space relative humidities around 40% RH would result in lower first and operating expenditures than higher supply air dew-point temperatures. That hypothesis was also confirmed. The third hypothesis predicted that with supply air dry-bulb temperatures lower than neutral, i.e., 55ºF (13ºC) or less, minimal or no terminal reheat would be required for many building applications. That hypothesis was also confirmed, and the reasons explored.

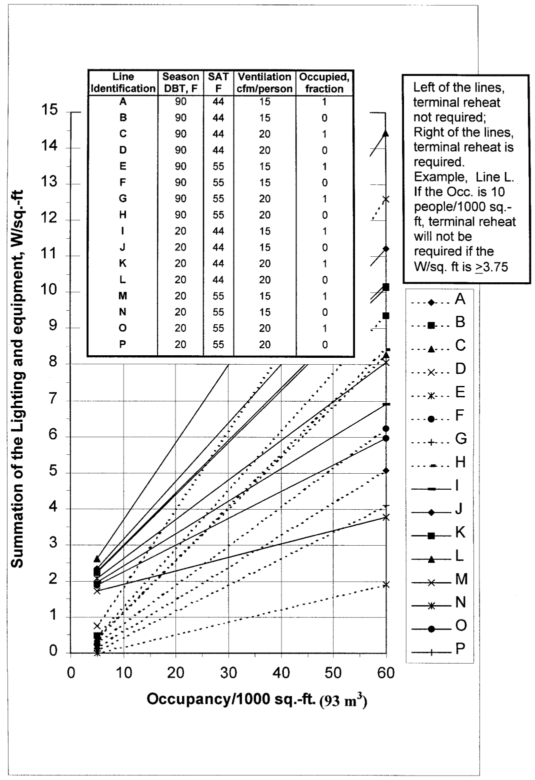

Figure 6 *Internal generation balance point for terminal reheat.*

In conclusion, it is recommended that for DOAS applications coupled with distributed parallel sensible cooling equipment, the supply air dew-point temperature be low enough to maintain a summer space RH no greater than 40%. That means a supply air DPT around 44ºF (7ºC). Likewise, the supply air DBT should be at or below 55ºF (13ºC). In cases where the design occupancy density is very high and terminal reheat is required, the terminal reheat energy should come from a recoverable source such as a chiller, engine, or other if possible.

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