APPENDIX C
DESIGN CALCULATIONS FOR A SINGLE-DUCT VAV SYSTEM WITH PERIMETER RADIATION

C.1 INTRODUCTION

This analysis is for the same building considered in Appendix B. Data not shown here are taken from that appendix. The following description defines the building systems in more detail. The focus of this appendix is upon the interrelationship of loads, indoor air quality concerns, and controls for a VAV HVAC system with perimeter radiation as applied in this particular building context. *Any conclusions drawn from analysis of systems apply only to the specific application presented and do not necessarily apply universally. Specific values for various design variables on a particular project would need to comply with local codes and match the design intent of the owner and design team.* Although the values presented herein are specific to this example, the issues discussed are typical for applications of this type of system.

1. A continuous perimeter hot-water radiation system is assumed with scheduling of the supply water temperature from the outdoor air temperature. It is sized throughout for adequate auxiliary heat to maintain a 76°F [24.4°C] room temperature by replacing the heat loss via conduction plus any infiltration. Scheduling options are illustrated.

2. Full-closing VAV units are assumed in this example, but means for maintaining a minimum acceptable air motion and ventilation rate without minimum positioning are illustrated. Note that, in general, it is not good practice to use full-closing VAV boxes in occupied spaces.

3. Multiple-zone flexibility is illustrated by examining interior and perimeter exposed areas of different uses.

4. The benefits of full diversity are taken.

5. Table C-1 (part “a” in I-P units; part “b” in SI units) completes the basic information from Table B-1 for this particular system.

As noted in Chapter 1, this appendix presents an example from a real project. This project was selected for the first edition of this Manual and has been retained for the second edition. Some design values may, as a result, appear dated. This is, however, not too critical in the context of a process example. The purpose of this appendix is not to show expected results or recommended inputs for any given project, but rather to outline design procedures and considerations. Economic analyses and conclusions are particularly vulnerable to local conditions and assumptions. The conclusions presented herein regarding economic decisions should not be viewed as establishing general patterns or directions for design decision making.

The analyses in this appendix were conducted in I-P units. SI units have also been provided, but I-P and SI numerical values may not correspond exactly due to rounding and approximations during conversions.

C.2 CALCULATIONS AT SUMMER PEAK

The calculations and psychrometric chart for the simple, single-zone cooling application with a draw-through fan may be considered identical to this VAV application for the analysis of average system conditions at summer design. Figure C-1(A) therefore shows, using solid lines for system averages, the identical summer plot as in Figure B-1, while the dotted processes are for special rooms defined in Table C-1. The room state subscript numerals correspond to the column numbers in the table.

TABLE C-1a. Special Room Loads (I-P Units)

	1 Lecture and Projection	2 Interior General Offices	3 Perimeter Executive	4 Perimeter Conference	5 Perimeter Office
Area, ft^2	5,000	320	400	1,800	320
Full load electric[1] (W/ft^2)	4.0	6.0	5.75	5.0	4.5
Full load occupancy	250	3	6	36	4
Location	Interior	Interior	South	North	West
Room cooling peak on 95°F db day, 100% (full load) lighting and occupancy					
Conduction load[2]	0	0	2,180	9,800	1,740
Solar load[3]	0	0	5,100	774	11,024
Plenum load[4]	55,300	4,657	5,260	22,056	3,375
Occupancy sensible load	60,000	720	1,440	8,650	960
Room sensible load	115,300	5,377	13,980	41,280	17,099
Occupancy latent load	52,500	630	1,260	7,550	840
Room internal load	167,840	6,007	15,240	48,830	17,939
SHF = Room sensible heat factor	0.69	0.9	0.92	0.85	0.95
cfm/ft^2 at t_s = 55.25°F	1.01	0.74	1.85	1.0	2.34
Room relative humidity, %	46.6	38.6	40.0	40.0	37.6
Cooling at part load[5] **(65/65 day)**	@ 0% FL Lighting	@ 65% FL Lighting	@ 80% FL Lighting	@ 25% FL Lighting	@80% FL Lighting
Conduction load offset by radiation[6] (solar = 0)	0	0	0	0	0
Plenum load[4]	19,200	3,409	4,441	10,595	2,887
Occupancy sensible load	60,000	720	1,440	8,650	960
Excess radiation	0	0	0	0	400
Room sensible load	79,200	4,129	5,881	19,245	4,247
Occupancy latent load	52,500	630	1,260	7,550	840
Room internal load	131,700	4,759	7,141	26,794	5,087
SHF = Room sensible heat factor	0.60	0.87	0.82	0.72	0.83
cfm/ft^2 at t_s = 58.1°F	0.80	0.66	0.75	(exh) 0.65	0.65
Room relative humidity (%)	55.0	47.7	45.5	48.5	44.5

TABLE C-1a. Special Room Loads (continued)

Cooling at Heating Peak on 0°F Day					
Conduction load[6] (reference only)	0	0	-8,730	-39,300	-6,990
Peak solar load	0	0	8,800	300	5,250
Maximum plenum load	55,300	4,657	5,260	22,056	3,375
Maximum occupancy sensible load	60,000	720	1,440	8,650	960
Excess radiation	0	0	4,150	0	4,100
Maximum room cooling load	115,300	5,377	19,650	31,006	13,685
cfm/ft^2 at t_s = 55.25°F	1.17	0.74	2.15	0.76	2.17
Minimum room cooling load					
at ts = 61.5°F	15,000	2,620	4,150	10,595	4,100
cfm/ft^2 at t_s = 61.5°F	1.01	0.05	0.65	(exh) 0.65	0.65

All loads are in Btu/h.

Notes for Table C-1a

(1) Electrical loads in specific areas vary from the average of 5.23 W/ft^2, depending upon wall and floor outlet usage or nonstandard ceiling lighting. Loads are intended to represent actual demands, not connected loads.

(2) Loads are pure wall and glass conduction with no allowance or prorating for roof.

(3) These are solar loads over and above conduction loads. Items (2) and (3) for each area correspond to the maximum loads (Btu/ft^2) from Section B-2.3, Summer Design Transmission Loads. Only Room 3, south, is given at its noon peak; all other rooms peak individually but simultaneously in late afternoon.

(4) Direct heat emission from lights to any room is considered to be the same for all systems, but indirect transmissions through floor and ceilings from the return air plenum are a function of the average ceiling temperature, which varies with the return air volume from the room and the temperature above the deck or below the floor adjacent to unconditioned spaces.

(5) The percent of full load lighting indicated applies only to the indicated rooms. General building lighting for all analyses using 65/65°F outdoor conditions is 75%. For interior spaces, 65% of full load lighting is assumed a realistic minimum in a preplanned, modular lighting system for an area without extensive plug loads and task lighting, since local occupancy switches are rarely provided. For a perimeter office with a light switch, 80% lighting load is assumed when occupied and 0% when unoccupied; for conference rooms, 25% when occupied and 0% when unoccupied.

(6) The conduction values shown constitute calculated values at the Δt for the indicated outdoor temperature. They are applied as internal loads only in specific systems tabulations, for air volume calculations, to the extent that they are not balanced by auxiliary heating systems (e.g., perimeter radiation or terminal heating coils).

TABLE C-1b. Special Room Loads (SI Units)

	1 Lecture and Projection	2 Interior General Offices	3 Perimeter Executive	4 Perimeter Conference	5 Perimeter Office
Area, m^2	465	30	37	167	30
Full load electric[1] (W/m^2)	43	65	62	54	48
Full load occupancy	250	3	6	36	4
Location	Interior	Interior	South	North	West
Room cooling peak on 35°C db day, 100% (full load) lighting and occupancy					
Conduction load[2]	0	0	0.64	2.87	0.51
Solar load[3]	0	0	1.49	0.23	3.23
Plenum load[4]	16.21	1.36	1.54	6.46	0.99
Occupancy sensible load	17.59	0.21	0.42	2.54	0.28
Room sensible load	33.79	1.58	4.10	12.10	5.01
Occupancy latent load	15.39	0.18	0.37	2.21	0.25
Room internal load	49.19	1.76	4.47	14.31	5.26
SHF = Room sensible heat factor	0.69	0.9	0.92	0.85	0.95
L/s m^2 at t_s = 12.9°C	5.13	3.76	9.40	5.08	11.89
Room relative humidity, %	46.6	38.6	40.0	40.0	37.6
Cooling at part load[5] (18.3/18.3°C day)	@ 0% FL Lighting	@ 65% FL Lighting	@ 80% FL Lighting	@ 25% FL Lighting	@80% FL Lighting
Conduction load offset by radiation[6] (solar = 0)	0	0	0	0	0
Plenum load[4]	5.63	1.00	1.30	3.11	0.85
Occupancy sensible load	17.59	0.21	0.42	2.54	0.28
Excess radiation	0	0	0	0	0.12
Room sensible load	23.21	1.21	1.72	5.64	1.24
Occupancy latent load	15.39	0.18	0.37	2.21	0.25
Room internal load	38.60	1.39	2.09	7.85	1.49
SHF = Room sensible heat factor	0.60	0.87	0.82	0.72	0.83
L/s m^2 at t_s = 14.5°C	4.06	3.35	3.81	(exh) 3.30	3.30
Room relative humidity (%)	55.0	47.7	45.5	48.5	44.5

TABLE C-1b. Special Room Loads (continued)

Cooling at Heating Peak on -17.8°C Day					
Conduction load[6] (reference only)	0	0	-2.56	-11.52	-2.05
Peak solar load	0	0	2.58	0.09	1.54
Maximum plenum load	16.21	1.36	1.54	6.46	0.99
Maximum occupancy sensible load	17.59	0.21	0.42	2.54	0.28
Excess radiation	0	0	1.22	0	1.20
Maximum room cooling load	33.79	1.58	5.76	9.09	4.01
L/s m^2 at t_s = 12.0°C	5.94	3.76	10.92	3.86	11.02
Minimum room cooling load at t_s = 16.4°C	4.40	0.77	1.22	3.11	1.20
L/s m^2 at t_s = 16.4°C	5.13	0.25	3.30	(exh) 3.30	3.30

All loads are in kW.

Notes for Table C-1b

(1) Electrical loads in specific areas vary from the average of 56.3 W/m^2, depending upon wall and floor outlet usage or nonstandard ceiling lighting. Loads are intended to represent actual demands, not connected loads.

(2) Loads are pure wall and glass conduction with no allowance or prorating for roof.

(3) These are solar loads over and above conduction loads. Items (2) and (3) for each area correspond to the maximum loads (kW/m^2) from Section B-2.3, Summer Design Transmission Loads. Only Room 3, south, is given at its noon peak; all other rooms peak individually but simultaneously in late afternoon.

(4) Direct heat emission from lights to any room is considered to be the same for all systems, but indirect transmissions through floor and ceilings from the return air plenum are a function of the average ceiling temperature, which varies with the return air volume from the room and the temperature above the deck or below the floor adjacent to unconditioned spaces.

(5) The percent of full load lighting indicated applies only to the indicated rooms. General building lighting for all analyses using 18.3/18.3°C outdoor conditions is 75%. For interior spaces, 65% of full load lighting is assumed a realistic minimum in a preplanned, modular lighting system for an area without extensive plug loads and task lighting, since local occupancy switches are rarely provided. For a perimeter office with a light switch, 80% lighting load is assumed when occupied and 0% when unoccupied; for conference rooms, 25% when occupied and 0% when unoccupied.

(6) The conduction values shown constitute calculated values at the Δt for the indicated outdoor temperature. They are applied as internal loads only in specific systems tabulations, for air volume calculations, to the extent that they are not balanced by \ auxiliary heating systems (e.g., perimeter radiation or terminal heating coils).

Fig. C-1. Psychrometric Analysis for Single-Duct VAV System with Separate Perimeter Radiation at Design Loads.

Note 4 in Table C-1 shows the direct lighting heat emission to each room from ceiling fixtures and miscellaneous room electric loads plus the indirect heat as a function of the building's average ceiling temperature rather than as a percentage of the individual maximum direct lighting heat emission to the ceiling. The plenum temperature is therefore a function of the percent of the system's entire full-load ceiling lighting, independent of the specific ceiling lighting intensity for any specific room.

Room 1: Lecture and Projection, Peak Load (see column 1, Table C-1)

Take ceiling lighting alone as 3 W/ft^2 [32.2 W/m^2], miscellaneous electrical loads as 1 W/ft^2 [10.8 W/m^2], and the temperature difference between plenum air and room air as 3.39°F [1.88°C] (from Appendix B). Again, as in Appendix B, assume 35% of the lighting load enters the room. Room loads are then as follows (in Btu/h [kW]):

Electrical load	(1 + (0.35) (3)) (5,000) (3.41) [(10.76 + (3.76) (3)) (465)]	=	35,000 [10.26]
Load from plenums	(5,000) (1.2) (3.39) [(465) (6.8) (1.88)]	=	20,340 [5.96]
Occupant sensible	(250) (240) [(250) (0.070)]	=	60,000 [17.59]
Room sensible			115,340 [33.81]
Occupant latent	(250) (210) [(250) (0.0620)]	=	52,500 [15.39]
Room total internal			167,840 [49.19]

Sensible heat ratio SHR = 115,300/167,840 = 0.69
[33.81/49.19 = 0.69]

Room supply air rate (115,300) / ((1.1) (76 – 55.25)) = 5,051 cfm = 1.01 cfm/ft^2
[(33,810) / ((1.2) (24.4 – 12.9)) = 2,384 L/s = 5.13 L/s m^2]

Draw line *s-R1* through point *s* in Figure C-1(A) at a slope of SHR = 0.69 for graphical solution of room state at 76°F [24.4°C] db, 46.6% relative humidity (*RH1*). This is not as low as the design condition but is considered acceptable; therefore, no design reheat (or its equivalent) is required for this mode.

Columns 2 through 5 in Table C-1 are addressed in a similar manner. Only results that require discussion are explained below.

Room 2: Interior Clerical, Peak Load (see column 2, Table C-1) Btu/h ft^2 [kW]

Electrical	(1.5 W/ft^2 + (0.35) (4.5 W/ft^2 ceiling)) (320 ft^2) (3.41) =	3,355 [0.98]
	[(16.1 + (.35) (48.4)) (29.7)]	
Plenums	(320 ft^2) (1.2) (3.39) =	1,302 [0.38]
	[(29.7) (6.8) (1.88)]	
Sum		4,657 [1.36]

Supply air rate = (5,377) / (1.1) (20.75) = 236 cfm at full load = 0.74 cfm/ft^2
[(1576) / (1.2) (11.5)] = 111 L/s = 3.76 L/s m^2
Draw line *s-R2* at SHR = 0.9 for room relative humidity (*RH2*) of 38.6%

Room 3: Perimeter Executive Office, South-Facing, Noon Peak (column 3, Table C-l)

From assumed loads, Appendix B, section B-2.3 (Btu/h [kW]):

Conduction	(400) (5.45) =	2,180 [0.64]
	[(37.2) (17.2)]	
Solar	(400) (18.2 - 5.45) =	5,100 [1.50]
	[(37.2) (57.4 - 17.2)]	
Lighting	(1 + (0.35) (4.75)) (400) (3.41) =	3,632 [1.07]
	[(10.8 + (.35) (51.1)) (37.2)]	
Plenums	(400) (1.2) (3.39) Btu/h ft^2 =	1,628 [0.48]
	[(37.2) (6.8) (1.88)]	
Sum		12,540 [3.67]

Room supply air rate (13,980 peak) / (1.1) (20.75) = 612 cfm = 1.53 cfm/ft^2
[(4098) / (1.2) (11.53)] = 297 L/s = 7.98 L/s m^2

Draw line *s-R3* at SHR = 0.92 for room relative humidity (*RH3*) of 40% (this line is not shown in Figure C-la)

Room 4: Conference Room, North-Facing, Peak Load (column 4, Table C-l) Btu/h [kW]

Conduction	(1,800) (5.45) =	9,800 [2.87]
	[(167) (17.2)]	
Solar	(1,800) (5.88 - 5.45) =	774 [0.23]
	[(167) (18.6 - 17.2)]	
Electrical	(1 + (0.35) (4)) (1,800) (3.41)=	14,730 [4.32]
	[(10.8 + (.35) (43.0)) (167)]	
Plenums	(1,800) (1.2) (3.39) =	7,326 [2.15]
	[(167) (6.8) (1.88)]	
Sum		~~22,056~~ [6.47]

Room supply air rate (41,280 peak) / (1.1) (20.75) = 1,808 cfm = 1.0 cfm/ft^2
[(12,099) / (1.2) (11.53)] = 875 L/s = 5.24 L/s m^2
Draw line *s-R4* at SHR = 0.85 for room relative humidity (*RH4*) of 40%

Room 5: Perimeter Office, West, Late Afternoon Peak (column 5, Table C-l) Btu/h [kW]

Solar	(320) (39.9 - 5.45) = [(29.7) (125.9 - 17.2)]	11,024 [3.23]
Electrical	(0.5 + (0.35) (4)) (320) (3.41) = [(5.4 + (.35) (43.0)) (29.7)]	2,073 [0.61]
Plenums	(320) (1.2) (3.39) = [(29.7) (6.8) (1.88)]	1,302 [0.38]
Sum		14,399 [4.22]

Room supply air rate (17,099 peak) / (1.1) (20.75) = 749 cfm = 2.34 cfm/ft^2
[(5,012) / (1.2) (11.53)] = 362 L/s = 1.13 L/s m^2
Draw line *s-R5* at SHR = 0.95 for room relative humidity (*RH5*) of 37.6%

C.3 CALCULATIONS FOR PART-LOAD COOLING

C.3.1 System Analysis for Outdoor Conditions 65/65°F [18.3/18.3 °C], No Sun, 75% Electric and 100% Occupancy Loads

With true VAV, only the following changes on the psychrometric chart have an effect on room conditions (see Figure C-2):

1. The cooling coil temperature will suffer a natural control droop from the discharge thermostat that controls it as well as from the controller of the refrigerated medium. For part-load humidity control, it is desirable to permit this to occur, and some designs even have a desired droop programmed into their control cycle. Note, however, that this tends to increase the throttling ratio for VAV-controlled zones.

2. The temperature rise from supply fan heat and duct gains may change as the result of a drop in system air volume and different ceiling plenum balances and temperatures. Although Equation (B-1) shows the fan heat temperature rise to be independent of airflow rate, it is an inverse function of fan efficiency at any given operating condition. This may vary in any given fan, as the constant static pressure control operates the various devices and may rise or fall, depending upon the position of the full-load operating point in the efficiency curve, the type of control, and the part-load value. For consistency in comparisons within these appendices, it is assumed that variable-inlet vanes are used and, within the range of 100 to 50% of system volume, the bhp is taken to be proportional to the percent of full-load volume. Therefore, the temperature change is 48 + 5.25 = 53.25°F [8.9 + 2.9 = 11.8°C]. This tends to keep the fan rise constant within the stated range regardless of volume. Note, however, that the fan efficiency for variable-speed drives remains constant in the range typical for VAV systems. Variable-speed drives would be the norm for most current projects.

3. For any given room latent load, the room SHR decreases as the room supply air volume is decreased to meet reduced room sensible loads.

4. At constant room air flow rate, the temperature rise from the ceiling heat of lighting is proportional to the percent of full lighting load. This percentage, however, varies inversely with room air supply volume. Repeat the trial-and-error solution for this quantity, illustrated in steps 1 through 3 of the cooling load calculations in Appendix B, for 75% lighting and miscellaneous electric loads, full occupancy, and a greater ratio of lighting loads into the room than the 44% found previously. With a substantially lower room air volume, assume a ratio of 57% because of the inverse relationship.

Fig. C-2. Psychrometric Analysis for Single-Duct VAV System with Separate Perimeter Radiation at Outdoor Conditions of 65/65°F [18.3/18.3°C].

		Btu/h [kW]
Direct lighting to room and miscellaneous electrical loads	(0.75) (1,076,000) = [(0.75) (315,376)]	807,000 [237]
Plenum loads	(0.57) (0.75) (1,496,100) = [(0.57) (0.75) (438,507]	639,580 [187]
Sensible occupant load at full load with diversity =		513,600 [151]
Transmission load (balanced by radiation) =		0 [0]
Total room sensible load		1,960,180 [575]

Supply air of 78,300 cfm [36,950 L/s] is at a temperature of 53.25°F [11.8°C], allowing for a rise to room air temperature of 22.75°F [12.64°C]. This represents 54% of the full-load air supply and an average of 0.49 cfm/ft^2 [2.49 L/s m^2]. Assume, however, that 0.65 cfm/ft^2 [3.30 L/s m^2] is the minimum acceptable value (see Section C-3.2.5) for a total of 104,000 cfm [49,078 L/s]. In actual operation, when a controller senses a system reduction to this desired minimum point, the supply air temperature can be scheduled to rise to some tolerable point that still permits adequate cooling at this minimum volume. In this case, a Δt of 17.9°F [9.9°C], i.e., a supply air temperature of 58.1°F [14.5°C], would be necessary with a cooling coil temperature of 52.85°F [11.6°C] as shown in the upper cycles of Figure C-2. If this coil temperature were too high or if the load were lower, the basic outdoor air radiation schedule could be elevated as a second step, after the maximum acceptable supply air temperature has been reached—all from the same flow-volume sensing device. It is reasonable to assume that 58.1°F [14.5°C] is satisfactory until its use is checked for desired performance in the special rooms, and the 104,000 cfm [49,078 L/s] may be used for the first try in the modified ceiling temperature rise equation, as before:

$$\frac{(0.75)\ (1{,}496{,}110)}{((0.5)\ (160{,}000)\ (1.2)) + ((1.1)\ (104{,}000 - 32{,}000))} = 6.4^{\circ}\mathrm{F}$$

$$\frac{(0.75)\,(438{,}510)}{((0.5)\,(14{,}870)\,(6.8)) + ((1.2)\,(49{,}078 - 15{,}101))} = 3.6^{o}C$$

There is no need to check the heat load from the plenums, since it was predicated on an arbitrary return air rate of 104,000 - 32,000 = 72,000 cfm [49,078 - 15,101 = 33,977 L/s]. However, the trial-and-error cycle must be closed by checking the amount of heat from lighting directly emitted to the plenum against the assumed value of (0.75) (1,496,000 Btu/h) = 1,122,000 Btu/h [(0.75) (438.5 kW) = 328.9 kW], which checks very closely as shown in the following calculation (Btu/h [kW]).

Load from plenums to room	(160,000) (1.2) (3.2) = [(14,870) (6.8) (1.8)]	614,400 [180.1]
Load from plenums to return air	(1.1) (72,000) (6.4) = [(1.2) (33,977) (3.56)]	506,900 [148.6]
Total =		1,121,300 [328.7]

5. Were it not for the need to raise the supply air temperature above the design point to control air volume, the system percent-of-full-load-operation would have no effect on the room relative humidity, since the latter is strictly a function of that temperature, not the system load. For this particular system load, and a required supply air temperature of 58.1°F [14.5°C], simple chilled-water temperature or flow control may be considered first if most or all room conditions can be satisfied with the higher dew-point temperature at the cooling coil. Terminal reheat, ceiling induction, or local fan recirculation with ceiling plenum mix may be a more economical solution (in both dollars and energy use) than main air system reheat to take care of especially difficult rooms (see lower left of Figure C-2). All room conditions can now be checked from Figure C-2 without the need to plot average system conditions. Only cooling coil and supply air temperatures are relevant.

C.3.2 Room Analyses for Outdoor Conditions 65/65°F [18.3/18.3°C], No Sun, with Electric and Occupancy Loads as Noted in Table C-1

The calculations follow the previous pattern. Special conditions are highlighted below.

1. Room 1 at zero lighting and full occupancy (Btu/h [kW]).

Room electrical load during projection from isolation booth =		0 [0]
Plenum loads	($5{,}000\ ft^2$) (1.2) (3.2) = [($465\ m^2$) (6.8) (1.77)]	19,200 [5.63]

Completing the summation in Table C-1 as before with SHR = 0.60 requires a coil temperature to find the supply air temperature. With a rise of 5.25°F [2.92°C] (assuming that Δt will stay substantially the same with decreased air volume and a smaller temperature difference between plenum and supply air), then the coil temperature is 58.1 - 5.25 = 52.85°F [14.5 – 2.92 = 11.58°C]. With a lower air velocity across the coil, the leaving air will be more saturated than at full load; so, in Figure C-2 the upper cycle is plotted at 52.85/51.6°F [11.58/10.89°C], and the supply air will be at 58.1/53.0°F [14.5/11.67°C]. For SHR = 0.60, relative humidity (*RH2*) = 55%.

This result warrants an interesting observation that is seldom considered by designers when examining part-load and no-load conditions. Unless a particular room is under a roof (or over an unconditioned space, or its ceiling cavity is completely isolated from the remaining ceiling cavities on the floor, or unless all building lighting is suspended within the room) the room is never under a true no-load condition, even with no lighting or occupancy. Thus, there is always a notable plenum load when a building is occupied and, if the perimeter conduction loss is adequately treated, the floor and ceiling heat gains constitute a year-round heat gain and also act as radiant heating panels. Without this effect, this crowded lecture room without operating lighting would have an SHR of 0.535 and a relative humidity of 59.5% (see Figure C-2, line *s-R1*). This effect is even more pronounced in lowering room humidity when recessed light fixtures are used in a *dead* ceiling with a higher plenum temperature.

2. Room 2 at 65% electric and 100% occupancy loads.

The table is self-explanatory. In order to avoid overloading Figure C-2, the psychrometric conditions for this room are not shown in the figure.

3. Room 3 at 80% electric and 100% occupancy loads.

The conduction load is neutralized by radiation and is therefore zero.

4. Room 4 at 25% lighting and 100% occupancy loads.

Similarly, the conduction load is zero. The room air flow rate is 977 cfm [461 L/s], which equals 0.54 cfm/ft^2 [2.74 L/s m^2]. Since this is less than the 0.65 cfm/ft^2 [3.30 L/s m^2] criterion (see Section C-3.2.5), even though it meets cooling requirements and the room has an infrequent use pattern, an expedient that takes care of such a space without penalizing the entire system is worth considering. One of the simplest ways to provide minimum air motion in a conference room, if temperature and humidity control are not a problem, is to employ an exhaust fan at the minimum desired unit air flow value with make-up from neighboring spaces through relief grilles. This permits the target criteria to be satisfied with minimal additional cost. From a psychrometric perspective, 977 cfm [461 L/s] mixes with an extra 198 cfm [93 L/s] of relief air at room conditions when the room is occupied. This yields the same room relative humidity with a satisfactory volume of total air movement.

5. Room 5 at 80% lighting and 100% occupancy loads.

This room needs 188 cfm [89 L/s] or 0.59 cfm/ft^2 [3.0 L/s m^2]. The intense solar effect at design condition places too much of a throttling demand on a west perimeter zone during sunless periods, with a similar but less intense situation for the east exposure. During weather below 76°F [24.4°C], this can be taken care of with separate zoning of east and west radiation and selective reset of the radiation schedule for excess radiation, when appropriate, to maintain the minimum airflow in the shaded rooms. Thus, Room 5 would need only 400 Btu/h [117 W] excess radiation to maintain 0.65 cfm/ft^2 [3.30 L/s m^2]. If moving shadows result in simultaneous sunlit and shaded rooms on the same exposure, with the shaded room set for radiation to maintain 0.65 cfm/ft^2 [3.30 L/s m^2], the sunlit room will have excess cooling available to neutralize the solar radiation effect in each module (16 ft deep, 1 ft wide [4.9 m by 0.3 m]) in a west exposure of (1.1) (2.34 - 0.65 cfm/ft^2) (76 - 58.1) = 33.3 Btu/h ft^2 [(1.2) (11.89 - 3.30 L/s m^2) (24.4 - 14.5) = 102 kW/m^2], which is more than required to neutralize the excess radiation and the sun effect in a sunlit room. During weather above 76°F [24.4°C], the problem is nonexistent because the additional conduction load brings the sensible load and the room air rate within the criteria limits.

Since room types 2, 3, and 5 constitute practically the entire building and can be maintained at 45.5% relative humidity or less with a supply temperature of 58.1°F [14.5°C], the designer should explore raising this temperature and even lowering the room temperature somewhat below 76°F [24.4°C] in order to raise the room air flow rate. In the illustration given, there is no doubt that the west- and east-side exposures could be brought up to 0.65 cfm/ft^2 [3.30 L/s m^2] with supply air scheduling alone, leaving radiation sequence reset only for more stringent requirements. The tabulation is shown with excess radiation.

With effective distribution, 0.65 cfm/ft^2 [3.30 L/s m^2] can produce in excess of 10 fpm [0.05 m/s] air motion in the occupied zone. Simple VAV systems under certain part-load conditions can operate satisfactorily with considerably less than that value, although occupant complaints have been encountered. The designer must use judgment in appraising these results, and the effect of air quality control and code restrictions during part-load operation, upon the design. Diffusers with a high induction ratio may be helpful in avoiding potential low-circulation problems.

C.4 COOLING CALCULATIONS FOR WINTER PEAK LOAD

This analysis assumes outdoor conditions of 0°F [-17.8°C] and relates to Figure C-1(B).

C.4.1 System Load Calculations at Peak Cooling, 100% Lighting, Occupancy, and Solar Loads

When the entire perimeter conduction and infiltration heat loss is balanced by auxiliary radiation, there is no transmission heat ~~gain~~ from the wall, only a solar gain.

The entire air system is on a year-round room cooling cycle except for an interior warm-up period or for interior spaces with exposed roof or floor. Therefore, no main system coil heat is required unless the mixed air temperature at minimum outdoor air volume is lower than needed to maintain the desired supply air temperature after fan and supply duct heat gains have been added to the mixed-air temperature.

Since a high-limit humidity control problem is nonexistent below 58°F [14.4°C] outdoor air temperature, the supply air temperature may be determined on the basis of providing enough cooling for all spaces—while not being so low as to create air movement or cold draft problems.

For maximum cooling conditions, take full lighting and occupancy loads and an assumed block solar gain of 530 MBh [155.3 kW]. This load has not been calculated, but can be shown to be approximately 77% of the 685 MBh [200.8 kW] summer block peak.

	Btu/h [kW]
Solar load	530,000 [155.3]
Plenum load (assumed same as on 95/75 [35.0/23.9]day)	650,900 [190.8]
Direct lighting and electrical loads	1,076 000 [315.4]
Occupant sensible load	513,600 [150.5]
Total room sensible load	2,770,000 [811.9]

This requires 140,680 cfm [66,387 L/s] at 58.1°F [14.5°C], the same supply temperature as that used for the 65/65°F [18.3/18.3°C] condition. Hence, for practical purposes (since the full-load volume was 145,500 cfm [68,662 L/s]), take all plenum, lighting, electrical, and occupancy values the same as for full load, and the above loads for the 95°F [35°C] day can be used for the 0°F [-17.8°C] day in the Table C-1 summary. Room conditions may be examined for the extreme conditions of maximum and minimum cooling in each room without regard to room humidity.

C.4.2 Room Calculations at 0°F [-17.8°C] for Maximum and Minimum Room Cooling under Maximum System Cooling Conditions

1. Rooms 1 and 2, Interior

At maximum cooling, Table C-1 indicates that any interior space would require the same air quantities and temperatures as for peak summer conditions. As a practical matter, wide experience with interior constant-volume cooling systems has shown that the supply air temperature must be several degrees higher in the winter than in the summer to avoid complaints. Therefore, assume that ordinary interior rooms, such as Room 2, can be satisfied with 58.1°F [14.5°C] supply air and the same air volumes as for the peak summer condition. Consequently, the cfm/ft^2 [L/s m^2] values for such interior offices are left at peak summer values, even though the higher supply temperature calculates out to 1.17 cfm/ft^2 [5.94 L/s m^2]. The lecture room is left at the calculated higher 1.17 cfm/ft^2 [5.94 L/s m^2] peak volume because of the high occupancy density and its effect on indoor air quality.

For minimum cooling, the worst type of situation that could occur in general office areas would be for local air distribution to be designed for greater modular flexibility with a peak capability of handling 6 W/ft^2 [64.6 W/m^2] of lighting for full load in the Table C-1 criteria, including a 1 W/ft^2 [10.8 W/m^2] allowance for miscellaneous electrical loads and a 60 ft^2 [5.6 m^2] per occupant concentration. If a particular tenant had no miscellaneous electrical loads and only 4 W/ft^2 [43.0 W/m^2] ceiling lighting with no occupancy, the only cooling load is the direct lighting load plus a reduced plenum load if the situation were to exist over the entire floor without air volume reduction. The designer is left with many choices, such as (a) accepting lower minimum air volumes, (b) raising system volume with higher coil and supply temperatures to preserve cooling capability in perimeter areas, (c) terminal reheat or induction reheat for low part-load areas, or (d) local recirculation. To carry through the simple VAV approach, it is assumed here that a combination of higher supply air temperature (controlled from system volume) and lower room temperature will be programmed to maintain a minimum 0.65 cfm/ft^2 [3.30 L/s m^2] of supply air at 61.5°F [16.4°C]. These results are shown in Table C- 1 as the two extremes with 58.1°F [14.5°C] supply air for maximum cooling and 61.5°F [16.4°C] for minimum cooling.

The lecture room with an assumed minimum load of no lighting, 25% occupancy, and a substantial load from a common ceiling plenum would have a load of only 15,000 + 20,340 = 35,340 Btu/h [4.4 + 5.9 = 10.3 kW]. In the worst case, however, such a room would be isolated for fire rating purposes and remain with no heat load except the 15,000 Btu/h [4.4 kW] from occupancy, shown for minimum cooling in Room 1. A local fan unit, which mixes VAV system supply air and recirculated air to produce a constant 1.01 cfm/ft^2 [5.13 L/s m^2], can handle all situations (in lieu of over-airing). The processes for such a unit are shown as a dotted line in the upper cycle of Figure C-2. With occupancy as the only load, process line *s-R1'* is at an SHR of 240/450 = 0.53. The supply air state *sl*" at 64.3°F [17.9°C] is the mixture required for full occupancy, and *sl*' at 73.1°F [22.8°C] for 25% occupancy, both yielding room air of 76°F db [24.4°C], 59% rh. If this type of room suffered any heat losses (i.e., exposure), then reheat should be considered to keep relative humidity below 60%.

2. Room 3

At minimum cooling with 61.5°F [16.4°C] supply air, a minimum volume of 0.65 cfm/ft^2 [3.30 L/s m^2] requires a sensible load of 4,150 Btu/h [1.2 kW] to maintain a room temperature of 76°F [24.4°C], or 3,300 Btu/h [0.97 kW] to maintain 73°F [22.8°C]. The former will occur with 80% lighting and one occupant, while the latter will occur with about 70% lighting and one occupant, both without any excess radiation or reheat. When a building facade has moving shadows, it is usually impractical to provide individual room control of radiation output or facade zoning to avoid excess radiation in the sunlit areas. It is easier to design on the basis of all areas receiving enough radiation to neutralize the minimum VAV at some supply temperature, such as the 61.5°F [16.4°C] of this example, when any area is sunlit without an internal load. Thus, the shaded areas will receive enough excess radiation to keep all south-facing VAV controls at the minimum air volume, while rooms in the sun require some additional air to neutralize the radiation and maintain control on a year-round cooling cycle. Therefore, 4,150 Btu/h [1.2 kW] is used in Table C-1 and is assumed to have come from some combination of lighting and excess radiation. For example, a shaded south-facing room could conceivably be unoccupied, unlit, and without solar gain, while the remainder of the building is generally occupied. Assuming individual room VAV control, such a room would have to be heated by radiation only enough to handle the conduction loss plus the minimum air volume of 0.65 cfm/ft^2 [3.30 L/s m^2] tempered from 61.5 to 76°F [16.4 to 24.4°C]. To design for this worst condition requires 4,150 Btu/h [1.2 kW] of excess radiation, which becomes a cooling burden in a sunlit, fully loaded identical room on the same radiation riser without individualized room radiation control. This is the reason for the 4,150 Btu/h [1.2 kW] of radiation load (excess) for Room 3 and 4,100 Btu/h [1.2 kW] for Room 5.

At maximum noon cooling, the winter design condition in the south governs the peak room air flow but, with true VAV, only the south-facing branches and diffusers need be sized to handle the peak—not the system fan, since air that is not required in other areas is probably available. With supply air at 58.1°F [14.5°C] and the 4,150 Btu/h [1.2 kW] excess radiation for the worst-case scenario, this room needs 2.15 cfm/ft^2 [10.9 L/s m^2] to maintain 76°F [24.4°C] instead of the summer peak of 1.85 cfm/ft^2 [9.4 L/s m^2], an increase of 0.3 cfm/ft^2 [1.5 L/s m^2]. If all south-facing offices were the same, an additional volume of (17,675 ft^2 of south-facing perimeter) (0.30 cfm/ft^2) = 5,302 cfm [(1,643 m^2) (1.5 L/s m^2) = 2,465 L/s] would be required at the period of south peak. Other calculations for full-load cooling on a 0°F [–17.8°C] day (not shown) indicate a diversity reserve of (145,000 - 121,400) = 24,100 cfm [(68,426 - 57,289 = 11,137 L/s] several hours later, which translates into a much greater reserve during the south peak, especially when occupancy and lighting diversity is allowed for the entire southern facade, because most of the offices have a lighting load of 4.5 W/ft^2 [48.8 W/m^2] rather than 5.75 W/ft^2 [61.9 W/m^2].

3. Room 4, North Conference Room

The north, winter, maximum cooling load without conduction gain, being considerably lower than the summer load, is still enough to permit 0.76 cfm/ft^2 [3.86 L/s m^2]. No excess radiation is required, since all north-facing rooms would receive only enough radiation to balance the conduction losses.

The minimum cooling load with 25% lighting and no solar and no occupancy needs only 0.37 cfm/ft^2 [1.88 L/s m^2] from the VAV system at 61.5°F [16.4°C] supply air, but air movement can be kept to the minimum 0.65 cfm/ft^2 [3.30 L/s m^2] with the exhaust fan and relief from an adjacent area.

4. Room 5

The cooling load can vary from zero to 13,685 Btu/h [4.0 kW] with full solar, occupancy, and lighting. A maximum 0°F [-17.8°C] day cooling load requires 2.87 cfm/ft^2 [14.6 L/s m^2] of 55.25°F [12.92°C] air, while a no-load room requires 4,100 Btu/h [1.2 kW] excess radiation to provide the minimum 0.65 cfm/ft^2 [3.30 L/s m^2] with 61.5°F [16.4°C] air. This excess radiation may be treated as described for Room 3.

5. Condition at Zero Lighting

Although there might be an occasional occupied perimeter space without lighting or plenum gains, a practical solution is excess radiation for typical rooms, such as 3 and 5, and exhausters for special rooms. It is assumed that general office interior areas are not occupied unless the lighting is on, and, if the entire interior were unlit and unoccupied, a full-throttling VAV terminal could go to complete shut-off. Occasional interior spaces without operating lighting would have a plenum gain of 3.4 Btu/h ft^2 [10.7 W/m^2] if surrounded by spaces with an average of only 65% full-load lighting. This would result in a reduced room temperature of 66.3°F [19.1°C] with 0.65 cfm/ft^2 [3.30 L/s m^2] of 61.5°F [16.4°C] supply air. This is more tolerable, however, than the lower temperatures encountered with a constant-air-volume system and less energy-consuming than adding reheat to the system.

Fig. C-1 >>

VALUES:
m = 55.74 [13.19]
hc = 57.25 [14.03]
sB = 61.5 [16.39], AF = 104,000 cfm, [49,078 L/s]
tr = 80.44 [26.91]
sA = 55.25 [12.92]

temperature in °F [°C]
air flow (AF) in cfm [L/s]

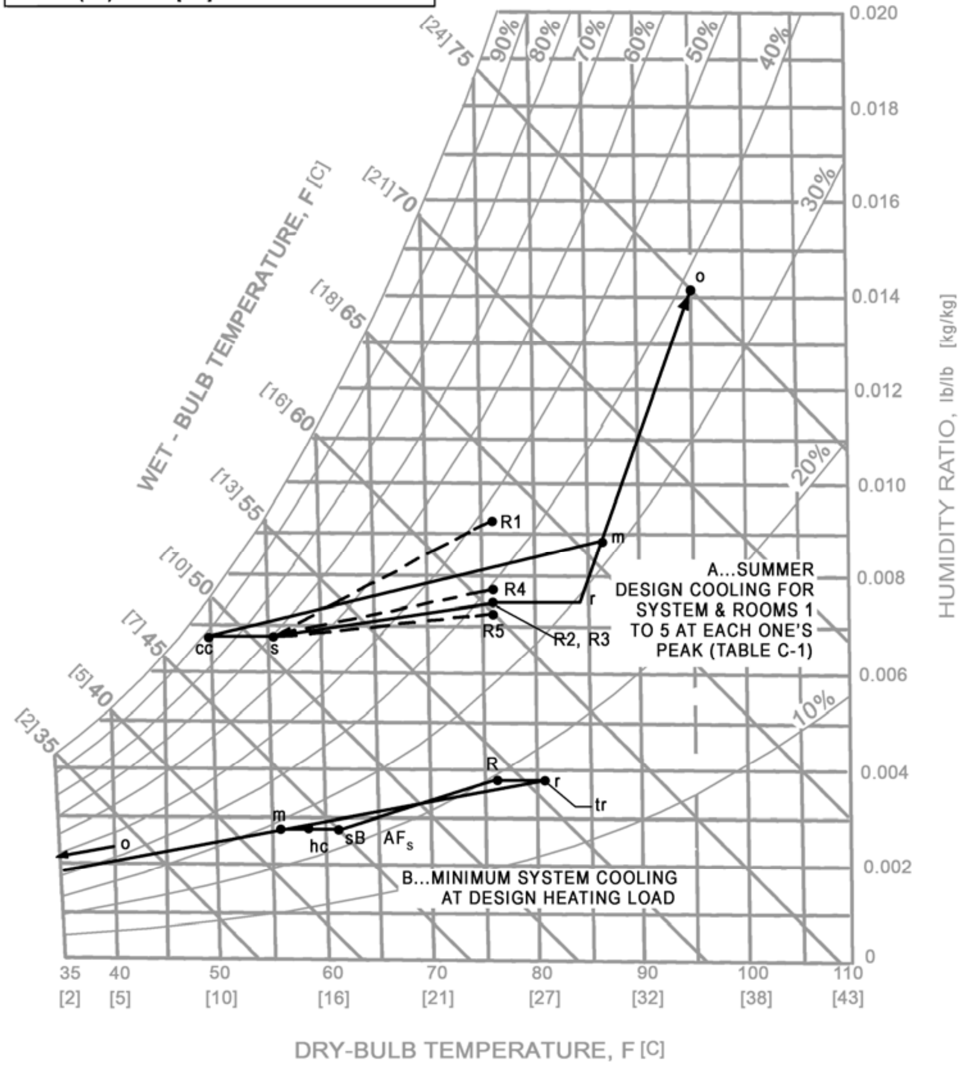
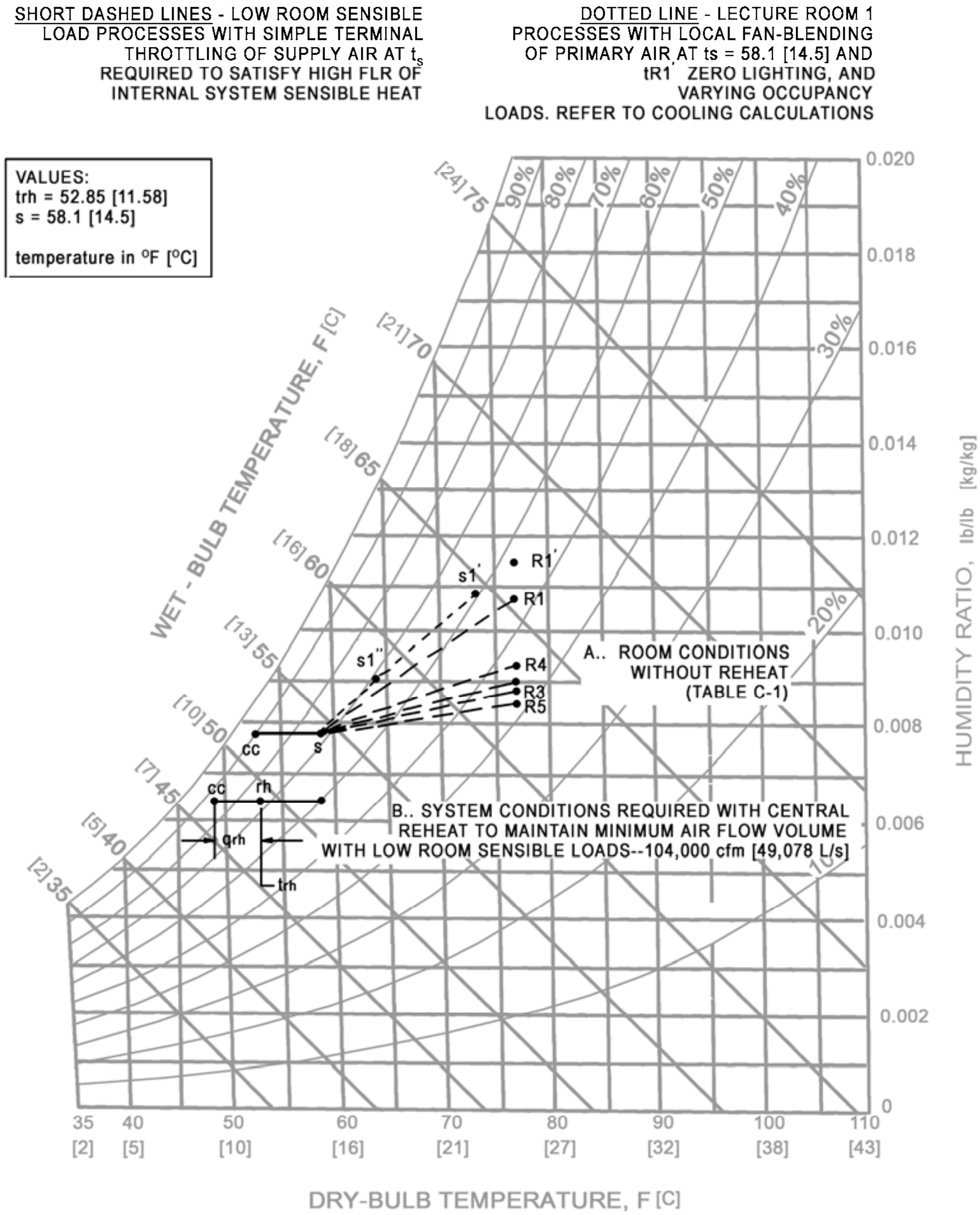


Fig. C-2 >>



SHORT DASHED LINES - LOW ROOM SENSIBLE LOAD PROCESSES WITH SIMPLE TERMINAL THROTTLING OF SUPPLY AIR AT t_s REQUIRED TO SATISFY HIGH FLR OF INTERNAL SYSTEM SENSIBLE HEAT
DOTTED LINE - LECTURE ROOM 1 PROCESSES WITH LOCAL FAN-BLENDING OF PRIMARY AIR AT ts = 58.1 [14.5] AND tR1' ZERO LIGHTING, AND VARYING OCCUPANCY LOADS. REFER TO COOLING CALCULATIONS
VALUES:
trh = 52.85 [11.58]
s = 58.1 [14.5]
temperature in °F [°C]
WET - BULB TEMPERATURE, F [C]
HUMIDITY RATIO, lb/lb [kg/kg]
DRY-BULB TEMPERATURE, F [C]
A.. ROOM CONDITIONS WITHOUT REHEAT (TABLE C-1)
B.. SYSTEM CONDITIONS REQUIRED WITH CENTRAL REHEAT TO MAINTAIN MINIMUM AIR FLOW VOLUME WITH LOW ROOM SENSIBLE LOADS--104,000 cfm [49,078 L/s]
R1'
s1'
R1
s1''
R4
R3
R5
cc
s
cc
rh
qrh
trh
90%
80%
70%
60%
50%
40%
30%
20%
10%
[24] 75
[21] 70
[18] 65
[16] 60
[13] 55
[10] 50
[7] 45
[5] 40
[2] 35
0.020
0.018
0.016
0.014
0.012
0.010
0.008
0.006
0.004
0.002
0
35 [2]
40 [5]
50 [10]
60 [16]
70 [21]
80 [27]
90 [32]
100 [38]
110 [43]