APPENDIX B
DESIGN CALCULATIONS FOR A TYPICAL SINGLE-ZONE HVAC SYSTEM

B.1 INTRODUCTION

The sample calculations in this appendix and in Appendix C illustrate the application of the procedures presented in Chapter 6, All-Air Systems. *Any conclusions drawn from this system analysis apply only to the specific application presented; they 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.* A step-by-step procedure is given for the load components of each air system accompanied by a corresponding psychrometric chart plot. Several off-season operating conditions are also analyzed to illustrate the ability of each system to maintain design conditions under a wide range of loads.

The building analyzed here and in Appendix C is a multi-exposure, multistory building. Treatment of the roof as to loads or control zones is ignored, and all vertical separations are assumed to be ceiling-to-floor assemblies without a roof. All decks are considered floors separating conditioned spaces; therefore, deck losses are a heat loss from, not a heat gain to, the ceiling plenum.

Supply and return ducts are assumed to be placed outside the conditioned spaces, so that they do not affect ceiling temperatures. This assumption is made for simplicity but, if the supply ducts in the return air plenum are insulated and there are only small runs of return air stub-ducting in the ceiling, the resulting error is small. It is also assumed that the peak airflow for all rooms is governed by room sensible heat loads.

All fan heat gains are assumed to occur as a temperature rise at the fan discharge even though the velocity pressure component of fan energy occurs along the length of the ductwork. This introduces a small psychrometric error when draw-through and blow-through systems are compared. The velocity at the fan discharge is assumed to be 2,500 fpm [12.7 m/s] or 0.39 in. of water [97 Pa] velocity pressure (VP). In a system with 6 in. of water [1.5 kPa] total pressure (TP), this VP represents only 6.5% of the total pressure (or energy) of the fan. Such an error is accepted for simplicity in this analysis.

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.

B.2 BUILDING LOADS

B.2.1 Physical Data

Total conditioned area			160,000 ft^2 (10 stories, 126.5 by 126.5 ft) [14,870 m^2; 10 stories, 38.6 by 38.6 m]
Perimeter area			70,700 ft^2 [6,570 m^2] based on 16-ft [4.9-m] depth
Interior area			89,300 ft^2 [8,300 m^2]
U-factors for ceiling assembly			
Ceiling		is	0.5-in. [12.7-mm] acoustical lay-in tiles; heat flow down
	U	=	1/(0.23 + 1.19 + 0.92) = 0.43 Btu/h ft^2 °F [1/(0.041 + 0.209 + 0.162) = 2.44 W/m^2 K]
Floor		is	2-in. [50-mm] concrete slab, tile covered; heat flow up
	U	=	1/(0.23 + 0.40 + 0.05 + 0.61) = 0.77 Btu/h ft^2 °F [1/(0.041 + 0.070 + 0.009 + 0.107) = 4.37 W/m^2 K]

Return-air suspended ceiling plenum with return air grilles in ceiling on 150-ft^2 [13.9-m^2] module locations, resulting in the plenum temperature being midway between room temperature and the temperature of the air entering the return air duct.

B.2.2 Design Conditions for Full Load

Room design summer condition	76°F [24.4°C] db, 45% rh
Room design winter condition	76°F [24.4°C] db, 20% rh
Outdoor design summer condition	95°F [35.0°C] db/75°F [23.9°C] wb
Outdoor design winter condition	0°F db [-17.8°C], 20% rh
Minimum outdoor air	0.2 cfm/ft^2 [1.0 L/s m^2] = 32,000 cfm [15,100 L/s]

B.2.3 Summer Design Transmission Loads (Solar and Conduction)

Assume no roof (for simplicity, see Section B-1); therefore, loads are from walls and glass only, distributed over the 16-ft [4.9-m] perimeter area that is partitioned from the interior area. The following figures are in Btu/h ft^2 [W/m^2] of perimeter floor area.

Exposure	N	E	S	W
Block load	5.88 [18.6]	6.90 [21.8]	7.92 [25.0]	39.9 [125.9]
Maximum in each exposure	5.88 [18.6]	32.2 [101.6]	18.2 [57.4]	39.9 [125.9]

Each exposure (of 10 stories) has 17,675 ft^2 [1,643 m^2] of floor area. Total loads for the perimeter area are, therefore:

Block load	104 + 122 + 140 + 705 = 1,071,000 Btu/h [314 kW]
∑max load	104 + 569 + 322 + 705 = 1,700,000 Btu/h [498 kW]

Conduction load per square foot of perimeter floor area is 5.45 Btu/h ft^2 [17.2 W/m^2]. The remainder is solar load:

Loads (1000 Btu/h [kW])	Total	Conduction	Solar
Block	1071 [314]	385 [113]	685 [201]
∑max	1700 [498]	385 [113]	1315 [385]

B.2.4 Winter Design Block Transmission Load (Conduction Only)

Load is 1,540,000 Btu/h [451 kW] or 21.8 Btu/h per ft^2 [68.8 W/m^2] of perimeter floor area. Loads do not include transmission from ceiling and floor cavities to room area. These loads are noted for completeness only. Winter conditions are not analyzed in this appendix.

B.2.5 Lighting and Miscellaneous Electric Loads

The following tabulation includes ballasts, office equipment, and other plug loads shown as design values with and without a 10% diversity allowance. Only systems with true VAV characteristics in multiple zone applications are allowed diversity reductions, since constant-volume systems must provide capacities for ∑max (the sum of the individual peaks). When the ceiling is a return air plenum, only part of the lighting heat gain to the ceiling is retransmitted to the room as a space heat gain. With a stagnant ceiling in a ducted return air system, however, all of it is retransmitted.

Loads per unit area[a] and total[b]	∑max full load		with 10% diversity	
65% of heat of lights	3.04	1660	2.74	1496
(emitted directly to ceiling plenum)	[32.7]	[487]	[29.5]	[439]
35% of heat of lights	1.63	890	1.47	803
(emitted directly to room)	[17.5]	[261]	[15.8]	[235]
Miscellaneous room electric loads	0.56	306	0.50	273
	[6.0]	[90]	[5.4]	[80]
Total electric load	5.23	2856	4.71	2572
	[56.3]	[837]	[50.7]	[754]
Ceiling lighting alone	4.67	2550	4.21	2299
	[50.3]	[747]	[45.3]	[674]

[a] W/ft^2 [w/m^2] [b] 1,000 Btu/h [kW]

B.2.6 Occupancy Loads (Assumed as a High-Occupancy Building)

The full occupant load is 2,378 occupants; with 67 ft^2 [6.2 m^2] per occupant. If the design permits diversity, use 2,140 occupants with 75 ft^2 [7.0 m^2] per occupant. Use 240 Btu/h [70 W] sensible and 210 Btu/h [62 W] latent heat per occupant.

B.2.7 Design Data Full Load

Supply fan total pressure	6 in. [1490 Pa]
Return fan total pressure	1 in. [250 Pa]

Cooling coil leaving air dry-bulb temperature (summer)	50°F [10.0°C]
Minimum room supply air temperature (winter)	55°F [12.8°C]

B.2.8 Intermediate Season Loads

Outdoor temperature	=	65°F db/65°F wb [18.3°C / 18.3°C]
Indoor minus outdoor temperature	=	11°F [6.1°C]
Cooling coil temperature	=	48°F [8.9°C]minimum; control droop 2°F [1.1°C] lower
Transmission load	=	-223,000 Btu/h (heat loss) at -11°F Δt [-65.4 kW @ 6.1°C] without solar load
Heat gain from lights		
at 75% full load	=	2,143,000 Btu/h [628 kW]
with 10% diversity	=	1,927,000 Btu/h [565 kW]
People load		
at full load	=	570,700 Btu/h [167 kW]
with 10% diversity	=	513,600 Btu/h [151 kW]

B.2.9 Special Room Loads

Table B-1 lists standard load components for typical rooms under various operating conditions. The first set of calculations in this appendix will help in understanding the numbers in the table.

B.2.10 System Calculations

For simplicity, all overall system calculations are based upon either the block loads or ∑max loads, assuming that summer sensible heat conditions govern all room air volumes. In unusual cases, certain room air volumes and room peak calculations may indicate that ventilation, humidity, or winter sensible heat concerns may govern, but these conditions rarely occur and are ignored in the present analysis. In all cases, particular stress is given to system calculations as they affect room conditions, particularly humidity.

B.3 LOAD CALCULATION FOR CONSTANT-VOLUME SINGLE-ZONE SYSTEM

Although the nature and the magnitude of the stated design criteria were chosen to permit analysis and comparison of the results for fairly sizable buildings with a requirement for zoning flexibility, they are used here as the loads for a simple, single-zone system to illustrate basic procedures. Even though the type of system analyzed here is not recommended as a design solution for this application, the results serve to illustrate that the use of more sophisticated air systems might result in appreciable increases in air, refrigeration, and heating capacities to achieve the desired effects. Figure B-1 represents the system cycle and shows the state of air at all locations noted in Figure 6-l(A). The corresponding points on the psychrometric chart and the component heat quantities absorbed or liberated between points are illustrated in Figures 6-1(B) and 6-1(C). Assume that the simple, single-zone load is identical to the simultaneous peak load of the entire building, as if there were just one large area with diversity.

B.3.1 Cooling Cycle Calculations at System Peak

Step 1. Find trial room sensible heat gain; it cannot be finalized until heat gains from the ceiling and floor cavities are determined after finding the average plenum temperature. Assume that 40% of heat from ceiling-mounted lighting will be transmitted to the room, both down through the ceiling and up through the floor below (see Figure B-1).

In Btu/h [kW]:

Heat gain from ceiling and floor	=	0.4 x 1,496,000 = 598,400 [0.4 x 439 = 175]
Lighting and miscellaneous heat gain	=	803,000 + 273,000 = 1,076,000 [235 + 80 = 315]
Simultaneous maximum transmission heat gain	=	1,071,000 [314]
People load	=	2,140 occupants @ 240 = 513,600 [2,140 @ 70 W = 150 kW]
Sensible heat load (trial)	=	3,259,000 [955]

TABLE B-1a Special Room Loads (Basic Form) I-P Units

	1 Lecture and Projection	2 Interior Clerical	3 Perimeter Executive Office	4 Perimeter Conference Room	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 Peak Cooling, 95/75 day, 100% 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]					
Occupancy sensible load	60,000	720	1,440	8,650	960
Room sensible load					
Occupancy latent load	52,500	630	1,260	7550	840
Room internal load					
SHR = Room sensible heat ratio					
Cooling at Part Load[5], 65/65 day	@ 0% Lighting	@ 65% Lighting	@ 80% Lighting	@ 25% Lighting	@80% Lighting
Conduction load[6] (solar = 0)	0	0	-1,260[5]	-5,690[5]	-1,010[5]
Plenum load					
Occupancy sensible load	60,000	720	1,440	8,650	960
Room sensible load					
Occupancy latent load	52,500	630	1,260	7550	840
Room internal load					
SHR = Room sensible heat ratio					

All loads are in Btu/h (unless otherwise noted).

TABLE B-1b Special Room Loads (Basic Form) SI Units

	1 Lecture and Projection	2 Interior Clerical	3 Perimeter Executive Office	4 Perimeter Conference Room	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 Peak Cooling, 35.0/23.9 day, 100% 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]					
Occupancy sensible load	17.59	0.21	0.42	2.54	0.28
Room sensible load					
Occupancy latent load	15.39	0.18	0.37	2.21	0.25
Room internal load					
SHR = Room sensible heat ratio					
Cooling at Part Load[5], 18.3/18.3 day	@ 0% Lighting	@ 65% Lighting	@ 80% Lighting	@ 25% Lighting	@80% Lighting
Conduction load[6] (solar = 0)	0	0	-0.37[5]	-1.67[5]	-0.30[5]
Plenum load					
Occupancy sensible load	17.59	0.21	0.42	2.54	0.28
Room sensible load					
Occupancy latent load	15.39	0.18	0.37	2.21	0.25
Room internal load					
SHR = Room sensible heat ratio					

All loads are in kW (unless otherwise noted).

Notes for Table B-1 (a and b)

Only the basic criteria that apply to any system design are given in this table. *Blanks are left where data will vary from one system to another.* No diversity in individual rooms is allowed at peak.

(1) Electrical loads in specific areas vary from the average of 5.23 W/ft^2 [56.3 W/m^2], depending upon plug loads and/or nonstandard ceiling lighting. Loads are intended to be 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 Btu/ft^2 [kW/m^2] maximum loads 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 emissions from lighting to any room are considered to be the same for all systems, but indirect transmission through floor and ceiling 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 is indicated for each room. Systems lighting for all 65/65°F [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 business machines and task lamps, 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 system 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. B-1 Psychrometric Analysis for Single-Duct, Single-Zone System.

Step 2. Find the supply air temperature and the trial supply and return airflow. With a draw-through coil and the fan and drive completely enclosed in the fan plenum, all the heat of the supply fan, motor, and drive, as well as the supply duct transmission loss, becomes a temperature rise in the supply air after the cooling coil.

$$\text{Fan temperature rise} = \frac{(0.363)\ (\text{total pressure in inches})}{(\text{fan efficiency})\ (\text{motor efficiency})\ (\text{drive efficiency})} \qquad \text{(B-1a)}$$

$$\frac{(0.829)\ (\text{total pressure in kPa})}{(\text{fan efficiency})\ (\text{motor efficiency})\ (\text{drive efficiency})} \qquad \text{(B-1b)}$$

Fan temperature rise = (0.363) (6) / (0.6) (0.9) (0.95) = 4.25°F, independent of air flow.
= [(0.829) (1.49) / (0.6) (0.9) (0.95) = 2.4°C]

Assuming a 1°F [0.6°C] temperature gain from that portion of the supply duct that passes through 90°F [32°C] unconditioned surroundings, the supply air temperature becomes 50 + 4.25 + 1 = 55.25°F [10.0 + 2.4 + 0.6 = 13.0°C].

Using Equation (1) in Chapter 6, the trial supply air quantity for sensible cooling is obtained as 3,259,000 / (1.1) (76 - 55.25) = 142,800 cfm [955,213 / (1.2) (24.4 – 12.9) = 67,387 L/s]

Recirculated air flow equals supply air flow minus outdoor air flow.
Recirculated air flow = 142,800 - 32,000 = 110,800 cfm [67,387 – 15,100 = 52,286 L/s].

Now check the assumed heat gain from the ceiling and floor. Using the appropriate equation from the Duct Design chapter of the *ASHRAE Handbook—Fundamentals*, the difference between room and return air plenum temperatures is found from:

(lighting heat in plenum) / ((0.5) (floor and ceiling conduction) + (1.1) (return air flow)

(1,496,000) / (0.5) ((160,000) (0.77 + 0.43)) + ((1.1) (110,800)) = 6.87°F

[(438,478) / (0.5) ((14,870) (4.37 + 2.44)) + ((1.2) (52,286)) = 3.86°C]

where,

1,496,000 [438,478] = lighting heat into plenum (from Step 1)
160,000 [14,870] = floor area (from B-2.1)
0.77 [4.37] = floor U-factor (from B-2.1)
0.43 [2.44] = ceiling U-factor (from B-2.1)
110,800 [52,286] = return (recirculated) air flow (see above)

The difference between room temperature and average return air plenum temperature is one-half of that value, and the heat gain from the ceiling and floor plenums then becomes:

(160,000) (0.77 + 0.43) (6.87) / 2 = 659,520 Btu/h

[(14,870) (4.37 + 2.44) (3.86) / 2 = 195,441 W

Step 3. A revised try for airflow, repeating steps 1 and 2 with new values gives the following:

Heat gain from ceiling and floor = 659,520 Btu/h [193.3 kW]
Lighting and miscellaneous heat gain = 803,000 + 273,000 = 1,076,000 Btu/h
= [235.4 + 80.0 = 315.4 kW]
Simultaneous maximum transmission gain = 1,071,000 Btu/h [313.9 kW]
People load = (2,140 occupants) (240) = 513,600 Btu/h
= [(2,140) (0.07 kW) = 150.5 kW]
Sensible heat load (revised) = 3,320,120 Btu/h [973.1 kW]
Revised supply air flow = (3,320,120) / (1.1) (76 – 55.25) = 145,460 cfm
= (973,127) / (1.2) (24.4 – 12.92) = 70, 639 L/s]
Revised recirculated air flow = (145,460 – 32,000) = 113,460 cfm
= [70,639 – 15,101 = 55,538 L/s]
Temperature difference = 1,496,000 / (96,000 + (1.1) (113,460)) = 6.77°F
= [438,478 / (50,632 + (1.2) (55,538)) = 3.74°C]
Return air plenum temperature = (76 + 6.8) = 82.8°F [24.4 + 3.76 = 28.2°C]
Heat gain from ceiling and floor = (160,000) (0.77 + 0.43) (6.77/2) = 649,920 Btu/h
= [(14,870) (4.37 + 2.44) (3.74/2) = 189.4 kW]
Return plenum heat gain = (1.1) (113,460) (6.77°F) = 844,937 Btu/h
= (1.2) (55,538) (3.74) = 249.3 kW]

Total direct lighting heat emission to plenum = (649,920 + 844,937) = 1,494,857 Btu/h
= (189.4 + 249.3) = 438.7 kW]

This is within 0.08% of the direct heat emission stated in the breakdown criteria. Note that this is 44% instead of the 40% lighting load fraction assumed. The total heat gain contribution from direct and indirect heat of light is 1,076,000 + 649,920 = 1,725,920 Btu/h [505,867 kW] or 67% of the total electric load.

Step 4. Find the room sensible heat ratio and check to see that the design room temperature is obtainable. Use Figure B-1(A). Plot the room design air state of 76°F [24.4°C] db, 45% rh.

Room sensible heat load	3,321,100 Btu/h [973.4 kW]
Room latent heat load	2,140 occupants x 210 [61.6 W]
	= 449,400 Btu/h [131.7 kW]
Room total internal load	3,770,500 Btu/h [1,105 kW]
Sensible heat ratio	(3,321,100) / (3,770,500) = 0.88

Draw a line with this slope through the design state and note that the required supply air temperature of 55.25°F [12.92°C], after a 5.25°F [2.92°C] rise from the 50°F [10°C] coil temperature, could not possibly fall anywhere along this process line—see dotted line in Figure B-l(A). A reasonable coil temperature must be assumed before the actual room humidity ratio and total system load can be determined, since the latter is a function of actual, not design, humidity.

A final adjustment to the calculated loads must always be made based upon a given room design condition, when the state of air leaving the actual selected coil is low enough in dew point to depress the room humidity below the selected design condition. If reconciliation is not made between the calculated load and the air load from the psychrometric chart conditions, the calculated load may be too low. Also, since the final leaving coil air condition cannot be determined until the total system load is known, a generalization will suffice to zero in on a reasonable coil condition. It is based upon the average coil surface temperature, which may be approximated closely as the intersection of the cooling line with the saturation curve (assumed straight), in this case, line *m-cc* extended in Figure B-1(A). The generalization may be expressed as:

$$(tccs - tEchw) / (wbtcc - tEchw) = Kcc \qquad \text{(B-2)}$$

where,

tccs = coil surface temperature as defined previously
tEchw = entering chilled-water temperature
wbtcc = wet-bulb temperature of air leaving coil
Kcc = chilled water coil constant

Kcc is a function of the ratio of extended surface to prime surface of a coil. It varies from about 0.5 for circular fin coils with 8 fins-per-in. [0.32 per mm] spacing to 0.7 for those with 14 fins per in. [0.55 per mm], and from about 0.6 to 0.75 for continuous plate fin coils. The overall effect for a given *tEchw* is to raise the leaving coil dew-point temperature as the extended surface ratio increases. The designer should always check the assumed cooling coil temperature against that obtainable with final coil selection.

Step 5. Find the dry-bulb temperatures and humidities of return air and mixed air. These humidities can only be found by trial and error using Equations (B-1) and (B-2). From Equation (B-1), the return fan temperature rise equals (0.36) (1 in.) / (0.6) (0.9) (0.95) = 0.71°F [(0.829) (0.248) / (0.6) (0.9) (0.95) = 0.40°C]. Assume a return duct transmission gain (in the unconditioned space) of 0.29°F [0.16°C]. Then the return air temperature becomes 76 + 6.77 + 0.29 + 0.71 = 83.8°F [24.4 + 3.76 + 0.16 + 0.39 = 28.7°C] (point *r* in Figure B-la). Also plot point *o*, the state of outdoor air. From Equation (5a) in Chapter 6, and the developed 22% outdoor air ratio, the mixed-air temperature is found to be 83.8 + 0.22 (95 - 83.8) = 86.26°F [28.7 + 0.22 (35 – 28.7) = 30.09°C].

Assume a coil with *Kcc* = 0.5. Graphically, start with a lower room humidity than the design condition and plot in sequence the temperatures of Equation (B-2) such that *Kcc* = 0.5. This occurs when *wbtcc* = 48.4°F [9.1°C], *tccs* = 46.4°F [8.0°C], and *tEchw* = 44.4°F [6.89°C]. Figure B-la shows this at a room condition of 76°F db/60°F wb [24.4/15.6°C] (rh = 38.7%), which is considerably below the 45% design relative humidity. At this point, coil selection should be checked with actual manufacturer's data to verify that conditions are attainable with an optimum selection of coil area, rows, water flow rate, *tEchw,* and water temperature rise.

Step 6. Find the final total system cooling load from load component summation and psychrometric chart values.

Room sensible heat	3,321,100 Btu/h [973.4 kW]
Room latent heat	449,400 [131.7]
Room total heat	3,770,500 Btu/h [1,105 kW]
Light heat in return air	844,937 [247.7]
Return duct and fan	
(1.1) (113,460) (1°F)	124,806 [36.6]
Supply duct and fan, sensible	
(1.1) (145,460) (5.25)	840,032 [246.2]
Outside air, sensible	
(1.1) (32,000) (95 - 76)	669,000 [196.1]
Outside air, latent	
(4840) (32,000) (0.01410-0.00752)	1,019,000 [298.7]
Total calculated load	7,268,275 Btu/h [2,130 kW]

Step 7. Find the total coil load from the enthalpy difference between mixed air and coil leaving air.

Coil load (4.5) (145,460) (30.56 - 19.43) = 7,287,400 Btu/h

[(1.2) (68,643) (71.08 - 45.19) = 2,132 kW]

Step 8. Find the total load from temperature and humidity ratio differences across the coil.

Sensible coil load (1.1) (145,460) (86.26 - 50.0) = 5,801,818 Btu/h

[(1.2) (68,643) (30.15 - 10.0) = 1,660 kW]

Latent coil load 4840 x 145,460 (0.00885 - 0.00683) = 1,422,520 Btu/h

[(3) (68,643) (8.85 – 6.83) = 416 kW]

Total coil load = 7,224,338 Btu/h [2,076 kW]

B.3 COMMENTS

(a) Steps 6, 7, and 8 illustrate good reconciliation between three calculation methods, including the psychrometric chart analysis, to synthesize the total load.

(b) Note that the entire ceiling plenum load becomes a system load only when all the air from the room is returned to the supply fan. In the extreme case (if all the return air were relieved after the return air fan) the heat from the ceiling plenum that is added to the return air never becomes a system cooling load.

(c) The full load with a 5.23 W/ft^2 [56.3 W/m^2] electric load requires only (145,460 cfm) / (160,000 ft^2) = 0.91 cfm/ft^2 [(68,643 L/s) / (14,870 m^2 = 4.62 L/s m^2] because of the heat of lighting in the return air, even with a moderate supply air temperature difference of 20.75°F [11.5°C]. If the ceiling were not used as a return air plenum, this load of 845,230 Btu/h [248 kW] would become a room sensible load requiring approximately 37,000 more cfm [17,460 L/s], or a total supply of 1.14 cfm/ft^2 [5.79 L/s m^2].

(d) Figure B-l(B) shows the psychrometrics for winter conditions. For an explanation of winter conditions, see Appendix C.

Fig. B-1 >>

VALUES:

m = 62.44 [16.91]	thc = 75.53 [24.18]	temperature in °F [°C]
ts = 79.85 [26.58]	tsf = 80.38 [26.88]	enthalpy in Btu/lb [kJ/kg]
Wm = 0.00307	WR = 0.0038	humidity ratio in lb/lb [kg/kg]
tr = 79.44 [26.36]	TR = 76.0 [24.44]	
cc = 50.0 [10]	ts = 55.25 [12.92]	
SHRR = 0.885	tm = 86.44 [30.24]	
hm = 30.68 [71.36]	ho = 38.61 [89.81]	

DESIGN (wbtR = 62.0 [16.67]   hR = 27.85 [64.78])

wbtcc = 48.4 [9.11]	tEchw = 44.4 [6.89]

ACTUAL (wbtR = 60.1 [15.61])