Ventilation Air Taken through Cooling Unit Which Becomes a Part of the Space Load (see Equations 10 and 11):

q_{si} = 1275 × 1.08 (95–80) (0.15) = 3,100 Btuh, sensible.

q_{ei} = 1275 × 4840 (0.0168–0.0098) (0.15) = 6,480 Btuh, latent.

Ventilation Air Taken through Cooling Unit Which Does Not Become a Part of the Space Load (see Equations 12 and 13):

q_{sx} = 1275 × 1.08 (95–80) (1–0.15) = 17,600 Btuh, sensible.

q_{ex} = 1275 × 4840 (0.0168–0.0098) (1–0.15) = 36,715 Btuh, latent.

$q_t = q_{si} + q_{sx} + q_{ei} + q_{ex}$ = 3100 + 17,600 + 6480 + 36,715 = 63,895 Btuh total.

Heat Gain from Sources within the Conditioned Space:

For the occupants, use the data of Table 27 for moderately active office work.

Sensible heat gain = 85 × 200 = 17,000 Btu per hr.
Latent heat gain = 85 × 250 = 21,250 Btu per hr.
Total = 38,250 Btu per hr.

For the gain from *lighting*, use Equation 15 with a use factor of unity, and a special allowance factor of 1.20 for the fluorescents and of unity for the tungsten globes.

q_{sl} = (12,000 × 1.20 + 4000) × 3.41 = 62,700 Btu per hr.

For the *fan motor*, use Equation 16 with a load factor of unity, and omit term *Motor Efficiency* because the motor is not within the space.

q_{sm} = 7.5 × 2544 = 19,100 Btu per hr.

Moisture Permeation, Miscellaneous Allowance, and the Load-Lag Estimate:

Moisture permeation will be negligible, since this is a comfort job with a good building construction.

There would be some heat gain in the ductwork, but this would not be great because of the short run involved. Practical judgment for this job would suggest that no adjustment for load lag need be made to the load as computed. (Refer to Fig. 4). While it is true that inside radiation forms an important part of the total heat gain, it is advisable to be conservative in recognizing the effect of the large, flat, hot roof on the *comfort sensations of the occupants*. Radiation from the relatively low ceiling, augmented by heat absorption from the lighting fixtures, would produce a sensation of warmth in excess of the nominal effective temperature (see Chapter 6) established by the wet-bulb and dry-bulb temperatures. Hence, it is not desirable to take advantage of every small decrease possible in the peak design load, especially since the peak occurs in mid-afternoon when everything would be rather well warmed.

Total Loads and Required Air Quantity through Conditioning Equipment:

The total loads are summarized in the following table:

SUMMARY OF TOTAL LOADS—EXAMPLE 11

LOAD COMPONENT	SENSIBLE BTU/HR	LATENT BTU/HR
All walls, roof and doors	78,500	....
Glass areas	5,970	....
Infiltration 67 cfm	1,085	2,270
Ventilation air (1275 cfm × 0.15)	3,100	6,480
Occupants	17,000	21,250
Lighting	62,700	....
Motor, fan	19,100	....
Space load	187,455	30,000
Ventilation 1275 cfm × (1–0.15)	17,600	36,715
Totals	205,055	66,715
Grand total sensible and latent		271,770

Compute the enthalpy difference ratio from Equation 18.

$$\frac{h_i - h_o}{W_i - W_o} = \frac{(187{,}455 + 30{,}120)}{30{,}120} \times 1076 = 7770.$$

From the ASHAE PSYCHROMETRIC CHART, Chapter 3, determine that the apparatus dew point is 53.9 F.

Compute the effective air quantity (Equation 19). Then,

$$Q_{re} = \frac{187{,}455}{1.08(80 - 53.9) \times 0.85} = 7830 \text{ cfm.}$$

(Refer to Chapter 23 for coil selection.)

From note under Equation 19 the dry-bulb range will be (80 − 53.9) × 0.85 = 22.2 deg, and the dry-bulb temperature of air leaving the coil will be 80 − 22.2 = 57.8 F. The dry-bulb temperature leaving the fan (including the heat supplied by the fan motor) or, delivered into the room, will be (from Equation 21):

$$t_s = 80 - \frac{187{,}455 - 19{,}100}{1.08 \times 7830} = 80 - 19.9 = 60.1 \text{ F.}$$

With good distribution and diffusion, this temperature should not produce objectionable drafts.

The various calculations for the sensible, latent, and total heat loads for Example 11 may be summarized as follows:

EXAMPLE 11: SUMMARY

OUTDOOR CONDITIONS	95 DB	78 WB	0.0168 HUMIDITY RATIO
SPACE CONDITIONS	80 DB	65 WB	0.0098 HUMIDITY RATIO
DIFFERENCE	15		0.0070

SENSIBLE LOAD		
Transmission	Btu/Hr.	
Roof 4000 sq ft × 53° × 0.34 =	72,000	
S. wall 405 sq ft × 6° × 0.41 =	995	
E. wall 765 sq ft × 11° × 0.52 =	4,380	
N. wall ex. 170 sq ft × 3° × 0.52 =	265	
N. & W. party wall 1065 sq ft × 2° × 0.26 =	550	
Floor none		
Door 35 sq ft × 15 × 0.59 =	310	
All glass and rest of doors =	3,020	
Solar Radiation		
S. glass 60 sq ft × 13 =	780	
S. glass (doors) 35 sq ft × 42 =	1,470	
E. glass (doors) 18 sq ft × 14 =	250	
N. glass 30 sq ft × 15 =	450	
Internal Load		
Infiltration 67 cfm × 1.08 × 15° =	1,085	
Ventilation 1275 cfm × 1.08 × 15° × 0.15 =	3,100	
Lights (12,000 × 1.20 + 4000) 3.41 =	62,700	
People 85 × 200 =	17,000	
Motor, fan 7.5 hp × 2544 =	19,100	
TOTAL SENSIBLE SPACE LOAD		187,455
LATENT LOAD		
Infiltration 67 cfm × 4840 × 0.0071 =	2,270	
Ventilation 1275 cfm × 4840 × 0.0071 × 0.15	6,480	
People 85 × 250 =	21,250	
TOTAL LATENT SPACE LOAD		30,000
VENTILATION AIR WHICH DOES NOT BECOME PART OF SPACE LOAD		
Sensible 1275 cfm × 1.08 × 15° × (1–0.15) =		17,600
Latent 1275 cfm × 4840 × 0.0071 × (1–0.15) =		36,715
GRAND TOTAL LOAD		271,770

LETTER SYMBOLS USED IN CHAPTER 13

α = fraction of incident solar radiation absorbed, dimensionless; subscripts D, d, and t refer to direct, diffuse, and total, respectively.

β = solar altitude, degrees.

γ = wall solar azimuth, degrees.

ϵ = emissivity, dimensionless.

θ = incident angle, degrees.

τ = fraction of incident solar radiation transmitted, dimensionless.
Subscripts D, d, and t refer to direct, diffuse, and total, respectively.

ϕ = solar azimuth, degrees.

ψ = wall azimuth, degrees.

A = area across which heat is being transferred, square feet.

b = fraction of air passing through coil which does not contact surfaces, coil bypass factor.

f = unit surface conductance, Btu per (hour) (square foot) (Fahrenheit degree).
Subscripts c, r, o, and i refer to convection, radiation, outdoor, and indoor, respectively.

G_f = fraction of total window area receiving direct solar radiation when shaded by window reveal, dimensionless.

h = enthalpy of air per pound of dry air, Btu per pound. Subscripts i, o, and s refer to indoor, outdoor, and supply air, respectively.

I = incident solar radiation, Btu per (hour) (square foot). Subscripts D, d, Dn, and t refer to direct, diffuse, direct normal, and total solar radiation, respectively.

K = cosine of angle of incidence for direct solar radiation striking a surface, dimensionless.

k = thermal conductivity of building material, Btu per (square foot) (hour) (Fahrenheit degree per inch).

l = height of window, feet.

M = the permeance of the specimen in *perms* or grains per (square foot) (hour) (inch of mercury vapor pressure difference).

Q = rate of entry of outdoor air, cubic feet per minute.

Q_{re} = required air quantity through conditioning equipment, cubic feet per minute.

q = instantaneous rate of heat transfer, Btu per hour.

q_e = instantaneous latent heat load, Btu per hour.

q_{ei} = instantaneous space latent ventilation load, Btu per hour.

q_{ex} = instantaneous latent ventilation load which does not become a part of space load, Btu.

q_m = latent heat load due to moisture transmission through materials, Btu per (hour) (square foot).

q_s = instantaneous sensible heat load, Btu per hour.

q_{si} = instantaneous space sensible ventilation load, Btu per hour.

q_{sx} = instantaneous sensible ventilation load which does not become a part of space load, Btu per hour.

$q_t = q_s + q_e$, also $q_{si} + q_{sx} + q_{ei} + q_{ex}$, Btu per hour.

R_o = low temperature radiant energy received from outdoor surroundings (does not include solar radiation), Btu per (hour) (square foot of receiving surface).

R = radiant energy emitted by a black body, Btu per (hour) (square foot). Subscripts go and L refer to outdoor surfaces of glass and building, respectively.

S = rate of heat storage within a glass section, Btu per (hour) (square foot).

t_e = sol-air temperature, Fahrenheit.

t_{gi} = temperature of indoor glass surface, Fahrenheit.

t_{go} = temperature of outdoor glass surface, Fahrenheit.

t_i = indoor air temperature, Fahrenheit.

t_m = 24-hr cyclic average sol-air temperature, Fahrenheit.

t_o = outdoor air temperature, Fahrenheit.

t_s = room supply air dry-bulb temperature, Fahrenheit.

U = overall coefficient of heat transfer of a structural section, Btu per (square foot) (hour) (Fahrenheit degree).

v_o = volume of outdoor air per pound of dry air, cubic feet.

w = width of window, feet.

W = humidity ratio, pounds moisture per pound of dry air. Subscripts i, o, and s refer to indoor, outdoor, and supply air, respectively.

REFERENCES

[1] *Application Engineering Standards for Air Conditioning for Comfort* (Air Conditioning and Refrigerating Machinery Association, Inc., 1947, p. 4).

[2] W. V. Consolazio and L. J. Pecora: Minimal replenishment air required for living spaces (ASHVE TRANSACTIONS, Vol. 53, 1947, p. 127).

[3] *Recommended Safe Practice of the NBFU for Hospital Operating Rooms* (*National Board of Fire Underwriters Pamphlet* No. 56).

[4] P. Moon: Proposed standard solar radiation curves for engineering use (*Journal of the Franklin Institute*, Vol. 230, November 1940, p. 583).

[5] C. O. Mackey: ASHVE RESEARCH REPORT No. 1268—Summer weather data and sol-air temperature—Study of data for Lincoln, Nebr. (ASHVE TRANSACTIONS, Vol. 51, 1945, p. 93).

[6] C. O. Mackey and E. B. Watson: Summer weather data and sol-air temperature—Study of data for New York City (ASHVE TRANSACTIONS, Vol. 51, 1945, p. 75).

[7] G. A. Hendrikson and J. H. Walker: Summer cooling for comfort as affected by solar radiation (*Heating and Ventilating*, Vol. 29, November 1932, p. 14).

[8] *Tables of Computed Altitude and Azimuth* (U. S. Navy Dept. *Hydrographic Office Bulletin* No. 214, Vol. 1–9, 1940).

[9] *The American Nautical Almanac* (U. S. Naval Observatory, annual).

[10] F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott: ASHVE RESEARCH REPORT No. 923—Heat transmission as influenced by heat capacity and solar radiation (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 231). J. S. Alford, J. E. Ryan, and F. O. Urban: Effect of heat storage and variation in outdoor temperature and solar intensity on heat transfer through walls (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 369). Victor Paschkis: Periodic heat flow in building walls determined by electrical analogy method (ASHVE TRANSACTIONS, Vol. 48, 1942, p. 75). C. O. Mackey and L. T. Wright, Jr.: Periodic heat flow—Homogeneous walls or roofs (ASHVE TRANSACTIONS, Vol. 50, 1944, p. 293). C. O. Mackey and L. T. Wright, Jr.: Periodic heat flow—Composite walls or roofs (ASHVE TRANSACTIONS, Vol. 52, 1946, p. 283). H. A. Johnson: Periodic heat transfer at the inner surface of a homogeneous wall (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 143).

[11] F. C. Houghten, Carl Gutberlet, and A. A. Rosenburg: The effect of solar radiation on the heat transmission through walls (*American Society of Testing Materials Symposium on Thermal Insulating Materials*, 1939).

[12] F. C. Houghten, H. T. Olson, and Carl Gutberlet: ASHVE RESEARCH REPORT No. 1157—Summer cooling load as affected