the room load, $\frac{h_i - h_s}{W_i - W_s}$. Use the equation

$$\frac{h_i - h_s}{W_i - W_s} = \frac{\text{(Space sensible load + space latent load)}}{\text{Space latent load/1076}} \quad (18)$$

where

h_s = enthalpy of moist air supplied to the space, Btu per pound of dry air.

h_i = enthalpy of moist air at room design conditions, Btu per pound of dry air.

W_s = humidity ratio of moist air supplied to the space, pounds of vapor per pound of dry air.

W_i = humidity ratio of moist air at room design conditions, pounds of vapor per pount of dry air.

Note that the ratio (space latent load/1076) is the equivalent of the required rate of water vapor removal in pounds per hour. If the rate of water removed is known, it may be used directly in Equation 18.

3. Draw a line through the reference point on the ASHAE Psychrometric Chart and the value of $(h_i - h_s)/(W_i - W_s)$ determined above. Draw a second line through the state point of the room air (design wet-bulb and dry-bulb temperatures) parallel to this line. This is the *condition line* for the process.

4. Read the temperature where the condition line from step 3 intersects the saturation line. This is called the *apparatus dew point*.

See Fig. 5. Note that instead of using this graphical method the left hand side of Equation 18 may be solved by trial and error by substituting values of h_s and W_s corresponding to assumed apparatus dew-point temperatures.

5. Compute the *required air quantity* from the relation

$$Q_{rs} = \begin{matrix}\text{Space}\\\text{sensible}\\\text{load}\end{matrix} \Bigg/ \left\{1.08\left[\begin{pmatrix}\text{Space}\\\text{dry-bulb}\end{pmatrix} - \begin{pmatrix}\text{Apparatus}\\\text{dew-point}\end{pmatrix}\right] \times \left(1 - \begin{matrix}\text{Coil}\\\text{bypass}\\\text{factor}\end{matrix}\right)\right\}. \quad (19)$$

The magnitude of Q_{rs} is substantially the quantity, cfm, of cooled and dehumidified air for which the distribution system must be designed.

The numerical factor 1.08 is derived from the product 1 cfm $\times$ 60 min $\times$ 0.244 $\times$ $0.075\left(1 - \frac{0.00923}{0.62}\right) = 1.08$, assuming an average supply air dew point of 55 F. Since standard air density (0.075) includes the weight of the water vapor, it is desirable to reduce it to the basis of dry air by the last factor where 0.00923 = humidity ratio of air at 55 F dew point, and 0.62 = ratio of density of water vapor to dry air at same temperature and pressure. Refer to Chapter 23 for coil selection.

Note that the product [(space dry-bulb) − (apparatus dew point)] $\times$ (1 − coil bypass factor) is equal to the dry-bulb range through which the conditioned air is cooled. Hence, in rare instances when the *condition line* of the process may not intersect the saturation line, any other convenient reference temperature on the *condition line* may be used instead, provided that the coil bypass factor is specified accordingly on the proper basis.

MINIMUM ENTERING AIR TEMPERATURE

Due consideration must be given to the temperature of the air entering the conditioned space in order to prevent objectionable drafts. With ceiling-type diffusers or wall grilles with a high aspect ratio (see Chapter 20), many engineers consider 20 deg as the maximum difference for good design under average conditions. This difference can only be exceeded with extremely high ceiling outlets or wall grilles. Thus, if 80 F dry-bulb is to be maintained in a space with average ceiling height, the minimum delivered air temperature would be limited to about 60 F dry-bulb temperature. If the latent heat load is relatively high, it is often necessary to circulate more air with a higher delivered dry-bulb temperature in order to produce a thermodynamic balance. If the dry-bulb temperature of the air supplied to the space is known, the required air quantity can be calculated from the formula,

$$Q_{rs} = \frac{q_s}{1.08\,(t_i - t_s)} \quad (20)$$

or the supply temperature t_s can be determined as follows,

$$t_s = t_i - \frac{q_s}{1.08 \times Q_{rs}} \quad (21)$$

EXAMPLE—COOLING-LOAD CALCULATION

Example 11: A one-story office building Fig. 6 is located in an eastern state near 40 deg latitude. The adjoining buildings on the north and west are not conditioned, and the air temperature within them is known to be substantially equal to the outdoor air temperature at any time of the day.

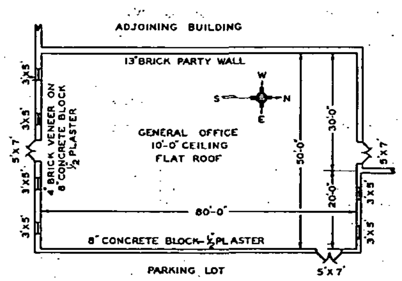

Fig. 6 .... Plan of One-Story Office Building

South wall construction: 8-in. concrete block, 4-in. brick veneer. (Table 7, Chapter 9, $U = 0.41$.)

East wall and outside north wall construction: 8-in. concrete block, no plaster on walls. (Table 6, Chapter 9, $U = 0.52$.)

West wall and adjoining north party wall construction: 13-in. solid brick, no plaster:

$$\frac{1}{U} = \frac{1}{1.65} + \frac{13}{5} + \frac{1}{1.65} \quad \text{or,} \quad U = 0.263. \quad \text{Use } U = 0.26.$$

Roof construction: 2½-in. flat roof deck of 2-in. gypsum slab on ¼-in. asbestos cement board surfaced with built-up roofing. (Table 13B, $U = 0.34$ *for summer*.)

Floor construction: 4-in. concrete on ground.

Window: 3 ft x 5 ft, nonopening type, with medium colored Venetian blinds for windows on south wall. Approximately 4-in. reveal on all windows.

Front doors: Two 2 ft-6 in. x 7 ft (glass panels).

Side doors: Two 2 ft-6 in. x 7 ft (½ glass panels).

Rear doors: Two 2 ft-6 in. x 7 ft (wood panels).

Outdoor design conditions: Maximum dry-bulb 95 F, wet-bulb 78 F; $W_o = 0.0168$ lbs vapor per lb dry air; $h_o = 41.38$ Btu per lb dry air.

Indoor design conditions: Dry-bulb 80 F, wet-bulb 65 F; $W_i = 0.0098$ lb vapor per lb dry air; $h_i = 29.95$ Btu per lb dry air.

Occupancy: 85 office workers.

Lights: 12,000 watts, fluorescent; 4000 watts, tungsten.

Fan motor: 7½ hp.

Assume that cooling coil has a bypass factor of 0.15, i.e., that 15 percent of the air passes through the coil without contacting the coil surface.

Conditioning equipment to be located in adjoining structure to north.

Find: Total, sensible, and latent maximum cooling loads and required air quantity through conditioning equipment.

Solution: From Table 3, the recommended ventilation rate is 15 cfm per person. Total necessary = 85 $\times$ 15 = 1275 cfm or 76,500 cu ft per hr.

As the room volume is 40,000 cu ft, the air changes per hour will be 76,500/40,000 = 1.91 which is more than one air change.

Estimated Time of Maximum Cooling Load:

For this job, judgment indicates that the roof will make the greatest single contribution to the cooling load. Hence, the time of maximum cooling load probably will be the time of maximum heat gain through the roof. From Table 9 the maximum temperature differential for a 2-in. gypsum roof of medium weight construction is 54 deg at 4:00 p.m., and 53 deg at 3:00 p.m. Examination of Table 12 (40 deg N Latitude) shows that solar heat gain through glass on the south wall is 18 Btu per (hr) (sq ft) at 4:00 p.m., and 42 Btu at 3:00 p.m. This indicates that the maximum cooling load occurs at approximately 3:00 p.m. Therefore make load calculations at 3:00 p.m. sun time. (This may be slightly different from 3:00 p.m. local time.) In some cases, there would be no clear-cut evidence of this nature, and consequently, it would be necessary to estimate the load for several successive times, and then to select the maximum.

Heat Gain Through Outer Wall and Roof Areas:

From Table 10 the temperature differential for the south wall (8-in. concrete block with 4-in. brick veneer) may be about the same as a 12-in. brick which is 6 deg at 3:00 p.m. for a dark colored wall. From the same table, the temperature differential for the east wall (8-in. concrete block with plaster) will be 11 deg at 3:00 p.m. for a *light colored wall* (interpolating between 2:00 and 4:00 p.m.). Likewise, the temperature differential for the north exposed wall (8-in. concrete block plus plaster) will be 3 deg at 3:00 p.m. (by interpolation) for a light wall.

The party wall of 13-in. brick on the west side and part of the north side may be treated as if it were an outside wall in the shade which has a temperature differential (from Table 10) of 2 deg.

For the door in north wall, estimate $U = 0.59$ from Chapter 9. The outdoor temperature at 3:00 p.m. is 95 F. Neglect time lag and any decrement factor. The temperature differential is $(t_o - t_i) = 95 - 80 = 15$ deg. The tabulation of the preceding values at 3:00 p.m. is given in the following table:

Section	Net Area Sq Ft	Temperature Differential F Deg	Heat Transmission Coefficient U	Heat Flow Rate per Hour Btu
Roof	4000	53	0.34	72,000
South wall	405[a]	6	0.41	995
East wall	765[a]	11	0.52	4,380
North exposed wall	170[a]	3	0.52	265
West & north party wall	1065[a]	2	0.26	550
Door in north wall	35	15	0.59	310
Total				78,500

[a] Calculated from gross wall area, less windows and doors.

Heat Gain Through Glass Areas:

In computing the load for 3:00 p.m., only the south windows and doors will be exposed to direct sunlight. Tables 12 and 13 will give the total heat gain from the glass areas. The window reveals will shade the south windows; the fraction of the window area receiving direct radiation is obtained from Equation 5 by substituting values as follows:

$r_1 = s/l = 4/60$; $r_2 = 4/36$; $\beta = 45.5$ deg, $\tan\beta = 1.02$

$\gamma = 74$ deg, $\tan\gamma = 3.487$, $\cos\gamma = 0.276$

$$G_f = 1 - \frac{4}{60}\left(\frac{1.02}{0.276}\right) - \frac{4}{36}(3.487) + \left(\frac{4}{60}\right)\left(\frac{4}{36}\right)\frac{(1.02)(3.487)}{0.276} = 0.462.$$

The south doors will be considered entirely sunlit. The outdoor air temperature is 95 F at 3:00 p.m. From Table 25 the inside Venetian blind factor is 0.65. The instantaneous heat gains due to transmitted direct and diffuse solar radiation, and from convection and radiation gain, are found in Tables 12 and 13 as listed below for the south-facing doors and windows, the north-facing windows and the ½ glass doors in the east wall. The gain through the solid portion of the east doors may be approximated by use of Fig. 3, since the wood panels have little heat capacity. From Table 4 the diffuse radiation value is taken as 18 Btu per (hr) (sq ft) from which $t_o + \alpha_s I_t/f_{ov}$ is found to be 98.2 for $\alpha = 0.7$ and $f_{ov} = 4.0$. From Fig. 3, $q = 22.0$ Btu per (hr) (sq ft). These heat gains are itemized in the following table.

Location	Area Sq Ft	Fraction Sunlit	Trans Solar Gain Btu/(hr) (sq ft)	Conv and Rad Gain Btu/(hr) (sq ft)	Total Gain Btu/(hr) (sq ft)	Total Gain Btu/hr
South windows*	60	0.462	13	19	32	1920
South doors	35	1.00	42	19	61	2135
East doors						
Glass	18	—	14	17	31	560
Wood	18	—	—	22	22	395
North windows	30	—	15	17	32	960
Total						5970

* Shade factor = 0.65.

In some jobs it would be desirable to increase (or decrease) the instantaneous radiation heat gain by a load-lag factor. The reason for not doing so in this case is that the solar gain is of a low magnitude, and reference to the table indicates that 0.8 of the previous hour would not affect the results materially.

Heat Gain from Ventilation and Infiltration:

Since the desired outdoor air rate 1275 cfm is greater than one air change per hour, it will be satisfactory for determining the ventilation component of the heat gain.

Window infiltration can be taken as negligible since the windows do not open.

Door infiltration requires some judgment. Assume that for each person passing through the double doors, the infiltration will be 100 cu ft of outdoor air, see Chapter 10, Table 3. Assume that the outside doors will be used at the rate of 10 persons per hour and the inside doors at the rate of 30 persons per hour. Total infiltration will then be 40 $\times$ 100 = 4000 cfh or 67 cfm.

The design rate of entry of outdoor air is then:

$$Q = 1275 + 67 = 1342 \text{ cfm}.$$

The sensible, latent, and total loads are determined from Equations 7, 8, and 9, respectively, at 3:00 p.m. ($t_o = 95$, $t_i = 80$, $W_o = 0.0168$, $W_i = 0.0098$). All the air entering the room as infiltration becomes a part of the space load.

Infiltration (*see Equations* 7, 8, *and* 9):

$q_s = 67 \times 1.08\,(95 - 80) = 1085$ Btuh, sensible.

$q_e = 67 \times 4840\,(0.0168 - 0.0098) = 2270$ Btuh, latent.

$q_t = q_s + q_e = 1085 + 2300 = 3385$ Btuh, total.