Heat Gain Through Outer Wall and Roof Areas:

From Table 9 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 2:00 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 12 deg at 2:00 p.m. for a *light colored wall*. Likewise, the temperature differential for the north exposed wall (8-in. concrete block plus plaster) will be 2 deg at 2:00 p.m.

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 9) of 2 deg.

For the doors in north and east walls, estimate $U_f = 0.59$ from Chapter 24. The outdoor temperature at 2:00 p.m. is 94 F. Neglect time lag and any decrement factor. The temperature differential is $(t_o - t_i) = 94 - 75 = 19$ deg. From Table 9, the temperature differential for the south door may be considered the same as dark frame construction and estimated at 30 F. The tabulation of the preceding values is given in the following table:

Section	Net Area Sq Ft	Temperature Differential F Deg[b]	Heat Transmission Coefficient U	Heat Flow Rate per Hour Btu
Roof	4000	52+4=56	0.34	76,200
South wall	405[a]	6+4=10	0.39	1,580
East wall	765[a]	12+4=16	0.48	5,880
North exposed wall	170[a]	2+4=6	0.48	490
West & north party wall	1065[a]	2+4=6	0.26	1,660
Doors in north and east walls	70	19	0.59	780
Door in south wall	35	34	0.59	700
Total	......	......	......	87,290

[a] Calculated from gross wall area, less fenestration and doors.
[b] Values given are corrected for $t_o - t_i = 19$ deg.

Heat Gain Through Fenestration Areas:

Examination of Tables 11 through 17 shows that for south windows at 40° N. latitude, the Solar Heat Gain Factors at 2:00 p.m. suntime are highest in December and lowest in June. Some judgment is therefore required to select the heat gain through windows which, combined with the other loads, will result in the maximum total heat gain. The equivalent temperature differential method for calculating roof loads is shown in the notes for Table 8 to apply for the months of April through August without reduction in heat flow. Therefore, August is selected for the calculation of heat gains through windows.

Using Tables 13, 18, and 20A, the total heat gains through the south and north windows were calculated as shown in the following tabulation. The solar heat gains for the south windows were taken as the product of the Shading Coefficient and the average of the Heat Gain Factors for the three hours, 12:00 noon, 1:00 p.m., and 2:00 p.m. Averaging of the solar heat gains for the three hours, which is a method based on experience and judgment, is intended to adjust the instantaneous heat gains to air conditioning loads. The solar heat gain for the north windows were calculated for 2:00 p.m. The reasons for not averaging these Heat Gain Factors is that the magnitude and variations are low, and previous hours would not affect the results materially. The air-to-air heat gain must be added to the solar beat gain for the total load through the windows.

Location	Area Sq ft	Total Solar Heat Gain Btu/(hr)(sq ft)	Air to Air Heat Gain Btu/(hr)(sq ft)	Total Gain Btu/(hr)(sq ft)	Total Gain Btu/(hr)
South windows	60	67	11	78	4,680
North windows	30	11	11	22	660
Total					5,340

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 zero since the windows are sealed.

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 24. 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 × 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 6, 7, and 8, respectively, at 2:00 p.m. ($t_o = 94$, $t_i = 75$, $W_o = 0.0161$, $W_i = 0.0093$). All the air entering the room as infiltration becomes a part of the space load.

Infiltration (see Equations 6, 7, and 8):

$$q_s = 67 \times 1.08\ (94 - 75) = 1375 \text{ Btuh, sensible.}$$

$$q_e = 67 \times 4840\ (0.0161 - 0.0093) = 2210 \text{ Btuh, latent.}$$

$$q_t = q_s + q_e = 1375 + 2210 = 3585 \text{ Btuh, total.}$$

Ventilation Air Taken through Cooling Unit Which Becomes a Part of the Space Load (see Equations 9 and 10):

$$q_{si} = 1275 \times 1.08\ (94\text{–}75)\ (0.15) = 3920 \text{ Btuh, sensible.}$$

$$q_{ei} = 1275 \times 4840\ (0.0161\text{–}0.0093)(0.15) = 6300 \text{ Btuh, latent.}$$

Ventilation Air Taken Through Cooling Unit Which Does Not Become a Part of the Space Load (see Equations 11 and 12):

$$q_{sx} = 1275 \times 1.08\ (94\text{–}75)\ (1\text{–}0.15) = 22{,}200 \text{ Btuh, sensible.}$$

$$q_{ex} = 1275 \times 4840\ (0.0161\text{–}0.0093)\ (1\text{–}0.15)$$
$$= 35{,}700 \text{ Btuh, latent.}$$

$$q_t = q_{si} + q_{sx} + q_{ei} + q_{ex} = 3920 + 22{,}200 + 6300 + 35{,}700$$
$$= 68{,}120 \text{ Btuh total.}$$

Heat Gain from Sources within the Conditioned Space:

For the occupants, use the data of Table 26 for moderately active office work, corrected for an indoor temperature of 75 F.

Sensible heat gain = 85 × 250 = 21,250 Btu per hr.

Latent heat gain = 85 × 200 = 17,000 Btu per hr.

Total = 38,250 Btu per hr.

For the gain from *lighting*, use Equation 14 with a use factor of unity, and a special allowance factor of 1.20 for the fluorescents and of unity for the tungsten globes.

$$q_{el} = (12{,}000 \times 1.20 + 4000) \times 3.41 = 62{,}700 \text{ Btu per hr.}$$

For the *fan motor*, use Equation 15 with a load factor of unity, and omit term *Motor Efficiency* because the motor is not within the space.

$$q_{em} = 7.5 \times 2544 = 19{,}100 \text{ Btu per hr.}$$

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

Moisture permeation will be negligible, since this 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. Because of the relatively high heat gain from the roof and the relatively high thermal capacity for the building, practical judgment for this job would suggest that an adjustment for the load lag should be made to the load as computed for a design condition of 75 F. Because of insufficient information on this subject at the present time, no factor can be given.

Total Loads and Required Air Quantity Through Conditioning Equipment:

The total loads are summarized in table 29.

Compute the enthalpy difference ratio from Equation 21.

$$\frac{h_i - h_s}{W_i - W_s} = \frac{(200{,}975 + 25{,}510)}{25{,}510} \times 1076 = 9545.$$

Table 29....Summary of Total Loads for *Example 10*

Load Component	*Sensible Btu/hr*	*Latent Btu/hr*
All walls, roof and doors	87,290	....
Windows	5,340	....
Infiltration 67 cfm	1,375	2,210
Ventilation air (1275 cfm × 0.15)	3,920	6,300
Occupants	21,250	17,000
Lighting	62,700	....
Motor, fan	19,100	....
Space load	200,975	25,510
Ventilation 1275 cfm × (1–0.15)	22,200	35,700
Totals	223,175	61,210
Grand total sensible and latent		284,385

From the ASHRAE Psychrometric Chart, (Chart I) Chapter 3, determine that the apparatus dew point is 53.0 F.

Compute the effective air quantity (Equation 22). Then,

$$Q_{re} = \frac{200{,}975}{1.08(75 - 53) \times 0.85} = 9950 \text{ cfm.}$$

(Refer to Chapter 34 for coil selection.)

From note under Equation 22 the dry-bulb range will be (75 – 53) × 0.85 = 18.7 deg, and the dry-bulb temperature of air leaving the coil will be 75 – 18.7 = 56.3 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 24):

$$t_s = 75 - \frac{200{,}975 - 19{,}100}{1.08 \times 9950} = 58 \text{ F.}$$

Table 30.... Summary of Calculations for *Example 10*

Outdoor Conditions	94 DB	78 WB	0.0161 Humidity Ratio
Space Conditions	75 DB	62.5 WB	0.0093 Humidity Ratio
Difference	19		0.0068

Sensible Load		
Transmission	Btu/Hr	
Roof 4000 sq ft × 56° × 0.34 =	76,200	
S. wall 405 sq ft × 10° × 0.34 =	1,580	
E. wall 765 sq ft × 16° × 0.48 =	5,880	
N. wall ex. 170 sq ft × 6° × 0.48 =	490	
N. & W. party wall 1065 sq ft × 6° × 0.26 =	1,660	
Floor none		
Doors N. & E. 70 sq ft × 19 × 0.59 =	780	
Door S. 35 sq ft × 34 × 0.59 =	700	
Transmitted Solar Radiation and Transmission		
S. glass 60 sq ft × 78 =	4,680	
N. glass 30 sq ft × 22 =	660	
Internal Load		
Infiltration 67 cfm × 1.08 × 19° =	1,375	
Ventilation 1275 cfm × 1.08 × 19° × 0.15 =	3,920	
Lights (12,000 × 1.20 + 4000) 3.41 =	62,700	
People 85 × 250 =	21,250	
Motor, fan 7.5 hp × 2544 =	19,100	
Total Sensible Space Load		200,975
Latent Load		
Infiltration 67 cfm × 4840 × 0.0068 =	2,210	
Ventilation 1275 cfm × 4840 × 0.0068 × 0.15	6,300	
People 85 × 200 =	17,000	
Total Latent Space Load		25,510
Ventilation Air Which Does Not Become Part of Space Load		
Sensible 1275 cfm × 1.08 × 19° × (1–0.15) =		22,200
Latent 1275 cfm × 4840 = 0.0068 × (1–0.15)		35,700
Grand Total Load		284,385

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 10* are shown in Table 30.

PART II: RESIDENTIAL COOLING LOADS

The acknowledged differences between *heat gains* and *cooling loads* are of essential importance in calculating residential cooling load. This is largely due to the following factors:

1. The loads on residential cooling systems are primarily those imposed by heat flow through structural components and by air leakage or ventilation. Internal loads, particularly those imposed by occupants and lights, are small in comparison to those experienced in commercial or industrial installations.
2. Most residences are cooled as a single zone, there being no means to redistribute cooling unit capacity from one area to another as loads change from one hour of the day to another.
3. Most residential systems employ units of relatively small capacity (from about 20,000 to 60,000 Btuh) which have no means for controlling capacity except by cycling the condensing unit. When it is remembered that cooling load is largely affected by conditions outside the house and that only a few days each season are *design* days, it becomes apparent that a partial load situation exists during most hours of the season. An oversized unit which has no capacity control is detrimental to good system performance under these circumstances.
4. Dehumidification is achieved only during periods of cooling unit operation, there being only very limited use of reheat or by-pass systems to achieve direct control of humidity under conditions of relatively light sensible load. Space condition control is usually limited to the use of room thermostats, essentially sensible heat actuated devices.
5. The systems in most residences are operated twenty-four hours per day, thus permitting full advantage to be taken of thermal *flywheel* effects of structural members, and furnishings within the structure. (This accentuates the partial load effects mentioned in item 3 above.)
6. Equipment should be of the smallest possible capacity commensurate with good comfort performance, in order to minimize initial equipment and distribution system costs.

From the foregoing, it is apparent that accurate cooling load calculation is, if anything, more important for residential installations than for non-residential installations.

Interpretation of Data

In attempts to calculate cooling loads for residences, the equivalent temperatures of Tables 8 and 9 have been found unsuccessful when applied directly. By making use of results of studies conducted in 5 research residences at the University of Illinois,[64] these equivalent temperature differences have been successfully applied to residential cooling load calculations and a comprehensive study of these data has resulted in the development of an all-industry residential heat gain calculation procedure.[65] Although the procedure was developed primarily for use by installers of residential equipment and systems, its engineering basis and examples of its use in engineering design will be given here.

Design Temperatures

Residences, unlike many other structures and types of occupancy, can be assumed to be occupied, and usually conditioned, for 24 hours per day, every day of the cooling season. The indoor design temperature is therefore a constant value, usually 75 F. The outdoor design temperature is that shown in Table 1.

An assumption of a single indoor temperature is not a sufficient condition for equipment selection and system design if the design goal is occupant comfort without requiring uneconomic equipment capacity. The procedure is therefore based upon an assumed indoor temperature swing of not more than 3 deg on a design day, when the residence is conditioned 24 hr per day, and the thermostat setting is 75 F. An