

A_s = area of separating section concerned, square feet.

t_s = air temperature in adjacent space, Fahrenheit.

t_c = air temperature in conditioned space, Fahrenheit.

Magnitudes of U , may be obtained from Chapter 9. The temperature t_c may have any value over a considerable range, according to conditions in the adjacent space. The temperature in a kitchen or boiler room may be as much as 15 to 50 deg above the outdoor air temperature. It is recommended that actual temperatures be measured in adjoining spaces wherever practicable. Where nothing is known, except that the adjacent space is of conventional construction and contains no heat sources, it is recommended that the difference ($t_s - t_c$) be taken as the difference between the outdoor air and conditioned-space design dry-bulb temperatures minus 5 deg. In some cases it may be that the air temperature in the adjacent space will correspond closely to the outdoor air temperature at all times. Under these latter conditions, the heat gain through the partition will be periodic in nature, and the value of a shaded wall should be used from Table 10.

For floors directly in contact with the ground, or over an underground basement that is neither ventilated nor warmed, the heat transfer may be neglected for cooling-load estimates.

LOAD FROM OUTDOOR AIR, VENTILATION, AND INFILTRATION

Ventilation. Data for determining the necessary ventilation rate have been presented previously in this chapter. Ventilation required is primarily dependent upon the number of occupants and upon the materials and apparatus within the space which may give off odors. For spaces having ceiling heights 10 ft or less, the total requirement should be checked against the volume, and in no case should the ventilation air rate be less than one air change per hour. In spaces having ceilings higher than 10 ft where the occupant load is low, a check calculation can be made against the volume of the space below an assumed 10-ft ceiling.

Infiltration must never be counted upon to provide ventilation, because on still days there will be little or no infiltration.

Infiltration. The principles of infiltration calculations have been discussed in Chapters 11 and 12, with emphasis on the heating season. For the cooling season, infiltration calculations are usually limited to doors and windows.

To compute cooling-load infiltration for windows by the crack method, use the data of Table 2, Chapter 11, for a wind velocity of 10 mph. Note that for double-hung windows the length of crack is three times the width plus twice the height; while for metal-sash windows the crack length is the total perimeter of the movable or ventilating sections. In calculating window infiltration for an entire structure, it is not necessary to consider the total crack length on all sides of the building, for the wind would not act simultaneously on all sides at once. In no case, however, should less than half of the total crack length be figured. A knowledge of the prevailing wind direction will aid judgment in this consideration.

Cooling-load infiltration for doors²³ may be obtained from Table 3, Chapter 11. For conditions other than those covered, the notes appended to the table will provide a basis for estimates. The tabulated data may also be used as the basis of estimates for interior doors between an air-conditioned and a non-air-conditioned space.

Infiltration load must be included whenever the new air introduced through the system is not sufficient to maintain

sufficient pressure within the enclosure to prevent the infiltration. Whenever economically feasible, it is desirable to introduce sufficient outdoor air through the air-conditioning equipment to maintain a constant outward escape of air, and thus eliminate the infiltration portion of the load. The pressure maintained must be sufficient to overcome wind pressure through cracks and door openings. When this condition prevails it is not necessary to include any infiltration load. When the quantity of new air introduced through the cooling equipment is not sufficient to build up the required pressure to offset infiltration, the entire infiltration load should be included in the cooling load calculations.

Total Outdoor Air Load. Many cooling coil manufacturers publish tables giving psychrometric data based on the average conditions of the leaving air for various coil temperatures, air velocities, and entering dry-bulb and wet-bulb conditions. When these tables are used, it is necessary to calculate the mixed-air condition entering the coil, and determine from the tables what coil and air velocity will produce the desired leaving-air conditions as required for the space to be conditioned. When cooling coils are listed as 80 to 95 percent efficient, the manufacturer indicates that 20 to 5 percent of the air passes through the coil without being cooled. If data of this nature are used, the uncooled portion of the air must be added to the space load before determining the effective air quantity. See later section on Apparatus Dew Point.

To determine the design cooling load caused by the introduction of outdoor air, the maximum rate of outdoor-air entry is first established. In some applications the use of special exhausters from the conditioned space may add to the outdoor-air requirements in determining the maximum rate. When this design quantity is established, and the design indoor- and outdoor-air states are known, the cooling load may be computed. There are several methods in use, the more accurate of which require rather detailed calculations. Refer to Chapter 3, and also section on Apparatus Dew Point in this chapter. The following equations are considered to be of sufficient precision for use at usual design conditions, as their accuracy is within 1 percent.

$$\text{Sensible Load } q_s = Q \times 60 \times 0.244$$

$$\times 0.075 \left(\frac{1 - 0.00923}{0.62} \right) (t_s - t_c)$$

$$= Q \times 1.08 (t_s - t_c), \text{ Btu per hour} \quad (7)$$

$$\text{Latent Load } q_L = Q \times 60 \times 0.075 \times 1076 (W_s - W_c)$$

$$= Q \times 4840 (W_s - W_c), \text{ Btu per hour} \quad (8)$$

$$\text{Total Load } q_t = q_s + q_L \quad (9)$$

where

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

t_s = outdoor dry-bulb temperature, Fahrenheit.

t_c = indoor dry-bulb temperature, Fahrenheit.

W_s = outdoor humidity ratio, pounds moisture per pound of dry air.

W_c = indoor humidity ratio, pounds moisture per pound of dry air.

0.075 = standard air density, pounds per cubic foot.

0.244 = a constant approximating the specific heat of dry air corrected for moisture, Btu per (pound) (Fahrenheit degree).

Cooling Load

1076 = a factor approximating the average Btu released in condensing one pound of water vapor from air.

As explained later in the section on Apparatus Dew Point, some methods of load calculations break down the ventilation air into two parts: one portion which does not contact the coil surfaces (i.e., bypasses the coil) in passing through the coil and thus becomes a part of the room load; and a second portion, the remainder of the air which contacts the coil surfaces and is cooled down to the apparatus dew point. This detailed method explained in the literature by Ashley,²⁴ is believed to be very easy and accurate to use. Similar equations, therefore, can be written:

Space Sensible Ventilation Load,

$$q_{ss} = Q \times 1.08 (t_s - t_c) \times b \quad (10)$$

Space Latent Ventilation Load,

$$q_{sl} = Q \times 4840 (W_s - W_c) \times b \quad (11)$$

Remaining Sensible Ventilation Load,

$$q_{rs} = Q \times 1.08 (t_s - t_c) (1 - b) \quad (12)$$

Remaining Latent Ventilation Load,

$$q_{rl} = Q \times 4840 (W_s - W_c) (1 - b) \quad (13)$$

$$q_t = q_{ss} + q_{sl} + q_{rs} + q_{rl} \quad (14)$$

where

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

Standard air weight (0.075 lb per cu ft) is recommended for use in all calculations, as this is the basis for rating fans and its consistent use keeps all parts of the calculations in conformity.

HOW OUTDOOR AIR LOAD AFFECTS ROOM LOAD

Actually, the outdoor air used for ventilation would pass through the conditioning equipment, and be cooled and dehumidified to a lower temperature and humidity ratio than room conditions before entering the room; but for heat-balance purposes the cooling load chargeable to the outdoor air is that corresponding to the difference between the outdoor and indoor air conditions.

One important purpose of the cooling-load estimate is to determine the conditions and quantity of air supplied to the space. All the various sensible and latent heat loads within the space must be included. Infiltration must be included in the space load since this air enters the doors and windows, and its heat and moisture load must be offset by the introduction of cooler, dryer air to the space. However, since outdoor air is taken through the conditioning equipment and cooled, this portion does not become a part of the space load, except a small portion which passes through the coil untreated. To determine the total load on the refrigeration machine, the remaining outdoor-air load must be included in the grand total load.

Example 10: For outdoor design conditions of 95 F dry-bulb and 75 F wet-bulb, and indoor design conditions of 80 F dry-bulb and 67 F wet-bulb, and for the supply of outdoor air at the rate of 1000 cfm and the exhaust of room air at the corresponding rate, calculate the total, sensible, and latent heat gains.

Solution: Substituting in Equation 7:

$$q_s = 1000 \times 1.08 (95 - 80) = 16,200 \text{ Btu per hr.}$$

From psychrometric data $W_s = 0.01413$, $W_c = 0.01122$.

Substituting in Equations 8 and 9:

$$q_L = 1000 \times 4840 (0.01413 - 0.01122) = 14,100 \text{ Btu per hr.}$$

$$q_t = q_s + q_L = 30,300 \text{ Btu per hr.}$$

HEAT SOURCES WITHIN THE CONDITIONED SPACE

People. The rates at which heat and moisture are given off by human beings under different states of activity are given in Table 27. In many applications these sensible and latent heat gains become a large fraction of the total load. Appreciable variations in heat-emission rates must be recognized according to the age and sex of the individual, state of activity, environmental influences, and duration of occupancy (since for short occupancy the extra heat and moisture brought in by people may be a significant factor).

While Chapter 6 should be referred to for detailed information, Table 27 in this chapter summarizes practical data representing conditions commonly encountered.

Lighting. In general, the instantaneous rate of heat gain from electric lighting²⁵ may be calculated from the following relation:

$$q_u = \left(\frac{\text{total light}}{\text{wattage}} \right) \times \left(\frac{\text{use}}{\text{factor}} \right) \times \left(\frac{\text{special allowance factor}}{\text{factor}} \right) \times 3.41, \text{ Btu per hr.} \quad (15)$$

The total light wattage is obtained from the ratings of all fixtures installed, both for general illumination and for display use.

The use factor is the ratio of the wattage in use, for the conditions under which the load estimate is being made, to the total installed wattage. For commercial applications such as stores, the use factor would be unity.

The special allowance factor is introduced to care for fluorescent fixtures, and for fixtures which are either ventilated or installed so that only part of their heat goes to the conditioned space. For fluorescent fixtures, the special allowance factor is recommended to be taken as 1.20 in order to allow for power consumed in the ballast. For ventilated fixtures, recessed fixtures, and the like, manufacturers' or other data²⁶ must be sought to establish the fraction of the total wattage which may be expected to enter the conditioned space.

Power. When equipment of any sort is operated within the conditioned space by electric motors, the heat equivalent of this operation must be considered in the cooling load. The general equation for calculating this load is:

$$q_m = \left(\frac{\text{Horsepower Rating}}{\text{Motor Efficiency}} \right) \times \left(\frac{\text{Load}}{\text{Factor}} \right) \times 2544, \text{ Btu/h} \quad (16)$$

It is assumed that both the motor and the driven equipment are within the conditioned space. If the motor is outside the space, then do not divide by the motor efficiency in Equation 16. The load factor is merely the fraction of the rated load which is being delivered under the conditions of the cooling-load estimate. Motor efficiencies may be approximated as follows: about 50 to 60 percent at $\frac{1}{2}$ hp rating, increasing to 80 percent at 1 hp, and to 88 percent at 10 hp and above.

Appliances. Care must be taken in a cooling-load estimate to take into account the heat gain from all appliances, electrical, gas, or steam. Table 28 presents recommended data.²⁷ Note that the maintaining rate in Table 28 is the heat input required to maintain the appliance at the normal operating temperature even though it is not being used, i.e., no coffee is being made, no toast is being made, no food is being cooked in the fry kettle, etc. The maintaining rate is useful in setting