

The radiation absorbed by the interior furnishings and structure reaches the conditioning equipment after a considerable delay in time.

Fig. 4 .... Origin of the Difference Between the Magnitudes of the Instantaneous Heat Gain and Instantaneous Cooling Load

handling equipment serving a certain space must be sufficient to handle the load created by the throngs of people attending sales in that space. Under such a condition the number of people in other spaces is usually normal or below normal. While this means that the air-handling equipment for certain departments must be large enough to cope with the situation, the refrigeration equipment need be only large enough to handle the average maximum. If a system employing zone recirculating fans and a single central fan and dehumidifier were used, the saving would be reflected in the capacity of the central fan and dehumidifier. Another example of this diversity is found in an office building having restaurants and stores of certain types in the first story and basement. At noon, when the restaurants and stores are crowded, the offices are below normal occupancy.

Heat lag should be carefully considered in the cooling load calculations. In certain types of buildings the effect of solar radiation is still apparent several hours after the sun has shifted from that exposure. In other types having a much lighter construction, the heat gain due to solar radiation decreases markedly with the passing of the sun. Some walls, warmed by the sun, may radiate heat long after the passing of the sun, thus requiring lower indoor temperatures to offset the radiant energy.

Buildings have considerable heat storage capacity which can often be utilized to great advantage, and which has provided an unexpected safety factor in many installations. If a space is kept below the design indoor temperature for some time, the interior walls, floors, furniture, and fixtures begin to assume the temperature of the space. Where the time is sufficient the entire mass, rather than merely its surface, may reach the room temperature. Thus, when a space has been precooled below the design maximum temperature for a period of time prior to the advent of the peak load, and the heat gain begins to increase to peak conditions, some of the increase is used in raising the temperature of the furniture, fixtures, etc., to the design conditions and the cooling load can be reduced accordingly. However, unless very accurate data with regard to the mass, surface, specific heat, etc., of the items within the space are available, due caution must be used in discounting the cooling load for this storage effect. In the absence of reliable data this allowance is often a matter of experience rather than calculation.

No comprehensive data are presently available for use in design load estimates to evaluate the interior load-lag effect, but several investigators[1-11,21,22,C-4] have made a study of the problem and have presented many useful data. Tables 8 and 9 include a lag factor for the wall and roof. The data should be classified as instantaneous rates of heat transfer, however, because they do not consider the convective versus the radiation heat transfer as shown in Fig. 4. Hence, practical judgment and experience offer the only basis of procedure. Until the needed data become available, it is recommended that the noncontinuous load be averaged over two or three hours during the time of maximum load, when determining the total instantaneous cooling load where a large portion of the heat gain is radiant. This suggestion applies only to conditions near the time of maximum heat gain, as the heat stored within the structure would necessarily appear in the cooling load eventually; but if it appears at a time when the gain from outdoors is relatively low, the equipment will be able to maintain satisfactory conditions within the range of maximum capacity.

LOAD FROM INTERIOR PARTITIONS, CEILINGS, AND FLOORS

Whenever a conditioned space is adjacent to another space in which a different temperature prevails, the transfer of heat through the separating structural section must be considered. Calculations are made according to the relation:

$$q = U_i A_i (t_i - t_i) \text{ Btu per hour.} \quad (5)$$

where

U_i = coefficient of overall heat transfer between the adjacent and the conditioned space, Btu per (hour) (square foot) (Fahrenheit degree).
A_i = area of separating section concerned, square feet.
t_i = air temperature in adjacent space, Fahrenheit.
t_i = air temperature in conditioned space, Fahrenheit.

Magnitudes of U_i may be obtained from Chapter 24. The temperature t_i 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 $(t_i - t_i)$ 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 north wall should be used from Table 9.

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.

Where air-conditioning supply and return ducts pass through unconditioned spaces, there will be a transfer of heat from these spaces to the air in the ducts, even though these ducts are well insulated. An allowance should be made for this heat gain and included in the heat estimate so that air can be supplied at a temperature low enough to offset the rise caused by this heat gain (see Chapter 31). There will also be some heat gain to the air in ducts passing through conditioned spaces, but since a cooling effect is produced in the space through which the duct passes, this is not a loss and usually can be compensated for by adjustment of air quantities between the various spaces.

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. Consideration should be given to reducing the cooling load from outdoor air by the use of filtered recirculated air in combination with outdoor air. Filtered recirculated air may be supplied to the occupied zone to reduce the quantity of outdoor air, depending upon the filter efficiency for the particular contaminants. Treatment of the recirculated air to provide some form of odor control may also be used. See Chapter 12 in this volume and Chapter 81 in the 1964 Guide And Data Book. The outdoor air quantity should never be less than 4 cfm per person, in order to provide for physiological needs.

Infiltration. The principles of infiltration calculations have been discussed in Chapter 25, with emphasis on the heating season. For the cooling season, infiltration calculations are usually limited to doors and windows. However, in multistory commercial buildings reversed chimney effect may exist. The exfiltration through cracks due to reversed chimney effect and exfiltration through doors due to entrance traffic may be estimated from Chapter 25.

To compute cooling-load infiltration for windows by the crack method, use the data of Table 2, Chapter 25, for a wind velocity of 10 mph or adjusted for the average wind velocity in the locality. 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 doors[41] may be obtained from Chapter 25.

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.

Sensible Load

$$q_s = Q \times 60 \times 0.244 \times 0.075 \left(1 - \frac{0.00923}{0.62}\right)(t_o - t_i)$$
$$= Q \times 1.08\,(t_o - t_i), \text{ Btu per hour} \quad (6)$$

Latent Load

$$q_l = Q \times 60 \times 0.075 \times 1076(W_o - W_i)$$
$$= Q \times 4840(W_o - W_i), \text{ Btu per hour} \quad (7)$$

Total Load

$$q_t = q_s + q_l \quad (8)$$

where

Q = rate of entry of outdoor air, cubic feet per minute.
t_o = outdoor dry-bulb temperature, Fahrenheit.
t_i = indoor dry-bulb temperature, Fahrenheit.
W_o = outdoor humidity ratio, pounds moisture per pound of dry air.
W_i = 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).
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,[C] is believed to be very easy and accurate to use. Similar equations, therefore, can be written:

Space Sensible Ventilation Load,

$$q_{os} = Q \times 1.08\,(t_o - t_i) \times b \quad (9)$$

Space Latent Ventilation Load,

$$q_{ol} = Q \times 4840(W_o - W_i) \times b \quad (10)$$

Remaining Sensible Ventilation Load,

$$q_{os} = Q \times 1.08(t_o - t_i)(1 - b) \quad (11)$$

Remaining Latent Ventilation Load,

$$q_{ol} = Q \times 4840(W_o - W_i)(1 - b) \quad (12)$$

$$q_t = q_{os} + q_{ol} + q_{os} + q_{ol} \quad (13)$$

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.