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Preface

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Research Promotion Committee

CHAPTER 18

NONRESIDENTIAL COOLING AND HEATING LOAD CALCULATIONS

HEATING and cooling load calculations are the [redacted] basis for [redacted] These calculations affect the [redacted] Cooling and heating load calculations can [redacted]

Simply put, heating and cooling loads are the [redacted] Heating and air conditioning systems are [redacted]

Peak design heating and cooling load calculations, which are this chapter's focus, seek to determine the [redacted] Similar principles, [redacted] as described in Chapter 19.

This chapter discusses [redacted] of cooling load calculation (e.g., [redacted] and [redacted] methods of [redacted]

COOLING LOAD CALCULATION PRINCIPLES

Cooling loads result from many [redacted] Building components or contents that [redacted] include the following:

- **External:** [redacted]
- **Internal:** [redacted]
- **Infiltration:** [redacted]
- **System:** [redacted]

TERMINOLOGY

The variables affecting cooling load calculations are [redacted] often [redacted] and [redacted]

Many cooling load components [redacted] and [redacted] Because these cyclic changes in [redacted] for a building or zone. A **zoned system** (i.e., one serving several independent areas, each with its own temperature control) needs to [redacted] however, it [redacted] At some times of [redacted]

Heat Flow Rates

In air-conditioning design, the following four related heat flow rates, each of which varies with time, must be [redacted]

Space Heat Gain. [redacted] **Entry modes** include (1) [redacted] (2) [redacted] (3) [redacted] (4) [redacted] (5) [redacted] and (6) [redacted] **Sensible heat** [redacted] **Latent heat** [redacted]

Radiant Heat Gain. [redacted] When these surfaces and objects become warmer than the surrounding air, some of their heat transfers to the air by convection. The composite heat storage capacity of these surfaces and objects [redacted] (Figure 1). The thermal storage effect is [redacted] Predicting

The preparation of this chapter is assigned to TC 4.1, Load Calculation Data and Procedures.

[redacted]

Fig. 1 Origin of Difference Between Magnitude of Instantaneous Heat Gain and Instantaneous Cooling Load

the nature and magnitude of this phenomenon to estimate a realistic cooling load for a particular set of circumstances has long been of interest to design engineers; the Bibliography lists some early work on the subject.

Space Cooling Load. This is the rate at which [redacted]

Space Heat Extraction Rate. The rates at which [redacted] Along with the [redacted] Therefore, [redacted] However, [redacted]

Cooling Coil Load. The rate at which [redacted] System loads include [redacted]

Time Delay Effect

Energy absorbed by walls, floor, furniture, etc., contributes to space cooling load [redacted] as shown in Figure 2.

There is [redacted]

COOLING LOAD CALCULATION METHODS

This chapter presents [redacted] methods that [redacted] from previous methods. The technology involved, however (the principle of calculating a heat balance for a given space) is not new. The first of the two methods is the **heat balance (HB) method**; the second is **radiant time series (RTS)**, which is [redacted] of the HB procedure. Both methods are explained in their respective sections.

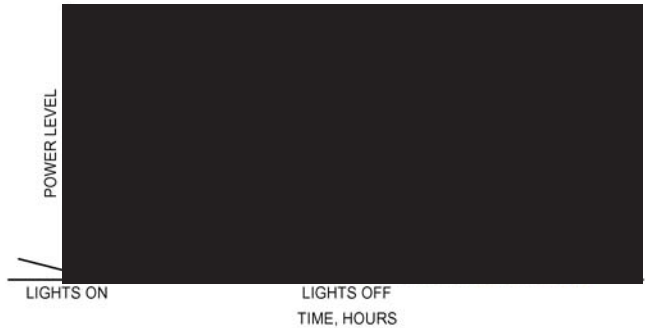

Fig. 2 Thermal Storage Effect in Cooling Load from Lights

Cooling load calculation of an actual, multiple-room building requires a complex computer program implementing the principles of either method.

Cooling Load Calculations in Practice

Load calculations should [redacted] are some of the variables that make precise calculation impossible. Even if the designer uses reasonable procedures to account for these factors, the calculation can never be more than a good estimate of the actual load. Frequently, a cooling load must be calculated before every parameter in the conditioned space can be properly or completely defined. An example is a cooling load estimate for a new building with many floors of unleased spaces for which detailed partition requirements, furnishings, lighting, and layout cannot be predefined. Potential tenant modifications once the building is occupied also must be considered. Load estimating requires proper engineering judgment that includes a thorough understanding of heat balance fundamentals.

Perimeter spaces exposed to [redacted]

Correct design and sizing of air-conditioning systems require more than calculation of the cooling load in the space to be conditioned. The [redacted] require that system performance be analyzed as a series of psychrometric processes.

System design could be driven by either sensible or latent load, and both need to be checked. In a sensible-load-driven [redacted] For a space driven by latent load [redacted]

This chapter is primarily concerned with a given space or zone in a building. When estimating loads for a group of spaces (e.g., for an air-handling system that serves multiple zones), the assembled zones must be analyzed to consider (1) [redacted] (2) [redacted] (3) [redacted] (4) [redacted] Methods presented in this chapter are expressed as hourly load summaries, reflecting 24 h input schedules and profiles of the individual load variables. Specific systems and applications may require different profiles.

DATA ASSEMBLY

Calculating space cooling loads [redacted] Generally, the following information should be compiled.

Building Characteristics. Building materials, component size, external surface colors, and shape are usually determined from building plans and specifications.

Configuration. Determine building location, orientation, and external shading from building plans and specifications. Shading from adjacent buildings can be determined from a site plan or by visiting the proposed site, but its probable permanence should be carefully evaluated before it is included in the calculation. [redacted] should not be overlooked.

Outdoor Design Conditions. Obtain appropriate weather data, and select outdoor design conditions. Chapter 14 provides information for many weather stations; note, however, that these design dry-bulb and mean coincident wet-bulb temperatures may vary considerably from data traditionally used in various areas. Use judgment to ensure that results are consistent with expectations. Also, consider prevailing wind velocity and the relationship of a project site to the selected weather station.

Recent research projects have [redacted] (e.g., ASHRAE 2012). [redacted]

To estimate conductive heat gain through exterior surfaces and infiltration and outdoor air loads at any time, applicable [redacted] must be used. Chapter [redacted] gives monthly [redacted] Chapter [redacted]

Indoor Design Conditions. Select indoor dry-bulb temperature, indoor relative humidity, and ventilation rate. Include permissible variations and control limits. Consult ASHRAE *Standard* [redacted] for [redacted] and *Standard* [redacted] for [redacted]

Internal Heat Gains and Operating Schedules. Obtain planned density and a proposed schedule of lighting, occupancy, internal equipment, appliances, and processes that contribute to the internal thermal load.

Areas. Use consistent methods for calculation of building areas. For fenestration, [redacted]

Gross surface area. It is efficient and conservative to derive gross surface areas from outer building dimensions, ignoring wall and floor thicknesses and avoiding separate accounting of floor edge and wall corner conditions. Measure floor areas to the outside of adjacent exterior walls or to the center line of adjacent partitions. When apportioning to rooms, façade area should be divided at partition center lines. Wall height should be taken as floor-to-floor height.

The outer-dimension procedure is expedient for load calculations, but [redacted] The resulting differences [redacted]

Fenestration area. As discussed in Chapter [redacted]

Net surface area. Net surface area is the gross surface area less any enclosed fenestration area.

INTERNAL HEAT GAINS

Internal heat gains from people, lights, motors, appliances, and equipment can [redacted] in a modern building. As building envelopes have improved in response to more restrictive energy codes, internal loads have increased because of factors such as increased use of computers and the advent of dense-occupancy spaces (e.g., call centers). Internal heat gain calculation techniques are [redacted]

PEOPLE

Table 1 gives representative rates at which sensible heat and moisture are emitted by humans in different states of activity. In high-density spaces, such as auditoriums, these sensible and latent heat gains comprise [redacted] Even for short-term occupancy, the extra sensible heat and moisture introduced by people may be [redacted] See Chapter [redacted] for detailed information; however, Table 1 summarizes design data for common conditions.

The conversion of sensible heat gain from people to space cooling load is affected by [redacted] energy. Latent heat gains are usually considered instantaneous, but research is yielding [redacted]

LIGHTING

Because lighting is often a major space cooling load component, an accurate estimate of the space heat gain it imposes is needed. Calculation of this load component is [redacted]

Instantaneous Heat Gain from Lighting

The primary source of heat from lighting comes from light-emitting elements, or lamps, although significant additional heat

Table 1 Representative Rates at Which Heat and Moisture Are Given Off by Human Beings in Different States of Activity

Degree of Activity	Location	Total Heat, Btu/h: Adult Male	Total Heat, Btu/h: Adjusted, M/F[a]	Sensible Heat, Btu/h	Latent Heat, Btu/h	% Sensible Heat that is Radiant[b]: Low V	% Sensible Heat that is Radiant[b]: High V
Seated at theater	Theater, matinee	[redacted]	[redacted]	[redacted]	[redacted]		
Seated at theater, night	Theater, night	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Seated, very light work	Offices, hotels, apartments	[redacted]	[redacted]	[redacted]	[redacted]		
Moderately active office work	Offices, hotels, apartments	[redacted]	[redacted]	[redacted]	[redacted]		
Standing, light work; walking	Department store; retail store	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Walking, standing	Drug store, bank	[redacted]	[redacted]	[redacted]	[redacted]		
Sedentary work	Restaurant[c]	[redacted]	[redacted]	[redacted]	[redacted]		
Light bench work	Factory	[redacted]	[redacted]	[redacted]	[redacted]		
Moderate dancing	Dance hall	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Walking 3 mph; light machine work	Factory	[redacted]	[redacted]	[redacted]	[redacted]		
Bowling[d]	Bowling alley	[redacted]	[redacted]	[redacted]	[redacted]		
Heavy work	Factory	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Heavy machine work; lifting	Factory	[redacted]	[redacted]	[redacted]	[redacted]		
Athletics	Gymnasium	[redacted]	[redacted]	[redacted]	[redacted]		

Notes:
1. Tabulated values are based on [redacted]
2. Also see Table 4, Chapter [redacted] for additional rates of metabolic heat generation.
3. All values are rounded to nearest 5 Btu/h.

[a] Adjusted heat gain is based on [redacted] of that for an adult male, and gain from a child is [redacted] of that for an adult male.

[b] Values approximated from data in Table 6, Chapter [redacted] where V is [redacted]

[c] [redacted]

[d] Figure [redacted]

may be generated from ballasts and other appurtenances in the luminaires. Generally, the instantaneous rate of sensible heat gain from electric lighting may be calculated from

[redacted] (1)

[redacted]

The **total light wattage** is obtained from the ratings of all lamps installed, both for general illumination and for display use. Ballasts are not included, but are addressed by a separate factor. Wattages of magnetic ballasts are [redacted]

The **lighting use factor** is the [redacted]

The **special allowance factor** [redacted] Use manufacturers' values for system (lamps + ballast) power, when available.

For high-intensity-discharge lamps (e.g. metal halide, mercury vapor, high- and low-pressure sodium vapor lamps), the actual lighting system power consumption should be available from the manufacturer of the fixture or ballast. Ballasts available for metal halide and high pressure sodium vapor lamps may have special allowance factors from about [redacted]

An alternative procedure is to estimate the lighting heat gain on a per square foot basis. Such an approach may be required when final lighting plans are not available. Table 2 shows the maximum lighting power density (LPD) (lighting heat gain per square foot) allowed by ASHRAE *Standard* [redacted] for a range of space types.

In addition to determining the lighting heat gain, the fraction of lighting heat gain that enters the conditioned space may need to be distinguished from the fraction that enters an unconditioned space; of the former category, the distribution between radiative and convective heat gain must be established.

Fisher and Chantrasrisalai (2006) experimentally studied 12 luminaire types and recommended [redacted] different categories of luminaires, as shown in Table 3. The table provides a range of design data for the conditioned space fraction, short-wave radiative fraction, and long-wave radiative fraction under typical operating conditions: airflow rate of 1 cfm/ft², supply air temperature between 59 and 62°F, and room air temperature between 72 and 75°F. The recommended fractions in Table 3 are based on lighting heat input rates range of 0.9 to 2.6 W/ft². For higher design power input, the [redacted] for design power input below this range, [redacted] The **space fraction** in the table is the fraction of lighting heat gain that goes to the room; the fraction going to the plenum can be computed as 1 – the space fraction. The **radiative fraction** is the radiative part of the lighting heat gain that goes to the room. The convective fraction of the lighting heat gain that goes to the room is 1 – the radiative fraction. Using values in the middle of the range yields [redacted] However, [redacted]

Table 3's data apply to [redacted]

If the [redacted] Figure 3 can be used to estimate the lighting heat gain parameters. [redacted]

Table 2 Lighting Power Densities Using Space-by-Space Method

Common Space Types*	LPD, W/ft²	Building-Specific Space Types*	LPD, W/ft²	Building-Specific Space Types*	LPD, W/ft²
Atrium		Automotive		Library	
First 40 ft in height	0.03 per ft (height)	Service/repair	[redacted]	Card file and cataloging	[redacted]
		Bank/office		Reading area	[redacted]
Height above 40 ft	0.02 per ft (height)	Banking activity area	[redacted]	Stacks	[redacted]
		Convention center		Manufacturing	
Audience/seating area—permanent		Audience seating	[redacted]	Corridor/transition	[redacted]
For auditorium	[redacted]	Exhibit space	[redacted]	Detailed manufacturing	[redacted]
For performing arts theater	[redacted]	Courthouse/police station/penitentiary		Equipment room	[redacted]
For motion picture theater	[redacted]	Courtroom	[redacted]	Extra high bay (>50 ft floor-to-ceiling height)	[redacted]
		Confinement cells	[redacted]		
Classroom/lecture/training	[redacted]	Judges' chambers	[redacted]	High bay (25 to 50 ft floor-to-ceiling height)	[redacted]
Conference/meeting/multipurpose	[redacted]	Penitentiary audience seating	[redacted]		
Corridor/transition	[redacted]	Penitentiary classroom	[redacted]	Low bay (<25 ft floor-to-ceiling height)	[redacted]
		Penitentiary dining	[redacted]		
Dining area	[redacted]	Dormitory		Museum	
For bar lounge/leisure dining	[redacted]	Living quarters	[redacted]	General exhibition	[redacted]
For family dining	[redacted]	Fire stations		Restoration	[redacted]
Dressing/fitting room for performing arts theater	[redacted]	Engine room	[redacted]	Parking garage	
		Sleeping quarters	[redacted]	Garage area	[redacted]
		Gymnasium/fitness center		Post office	
Electrical/mechanical	[redacted]	Fitness area	[redacted]	Sorting area	[redacted]
Food preparation	[redacted]	Gymnasium audience seating	[redacted]	Religious buildings	
		Playing area	[redacted]	Audience seating	[redacted]
Laboratory		Hospital		Fellowship hall	[redacted]
For classrooms	[redacted]	Corridor/transition	[redacted]	Worship pulpit, choir	[redacted]
For medical/industrial/research	[redacted]	Emergency	[redacted]	Retail	
		Exam/treatment	[redacted]	Dressing/fitting room	[redacted]
Lobby	[redacted]	Laundry/washing	[redacted]	Mall concourse	[redacted]
For elevator	[redacted]	Lounge/recreation	[redacted]	Sales area	[redacted]
For performing arts theater	[redacted]	Medical supply	[redacted]	Sports arena	
For motion picture theater	[redacted]	Nursery	[redacted]	Audience seating	[redacted]
		Nurses' station	[redacted]	Court sports arena—class 4	[redacted]
Locker room	[redacted]	Operating room	[redacted]	Court sports arena—class 3	[redacted]
Lounge/recreation	[redacted]	Patient room	[redacted]	Court sports arena—class 2	[redacted]
		Pharmacy	[redacted]	Court sports arena—class 1	[redacted]
Office		Physical therapy	[redacted]	Ring sports arena	[redacted]
Enclosed	[redacted]	Radiology/imaging	[redacted]	Transportation	
Open plan	[redacted]	Recovery	[redacted]	Air/train/bus—baggage area	[redacted]
		Hotel/highway lodging		Airport—concourse	[redacted]
Restrooms	[redacted]	Hotel dining	[redacted]	Waiting area	[redacted]
Sales area	[redacted]	Hotel guest rooms	[redacted]	Terminal—ticket counter	[redacted]
Stairway	[redacted]	Hotel lobby	[redacted]	Warehouse	
Storage	[redacted]	Highway lodging dining	[redacted]	Fine material storage	[redacted]
Workshop	[redacted]	Highway lodging guest rooms	[redacted]	Medium/bulky material storage	[redacted]

Source: ASHRAE *Standard* [redacted]

*In cases where both a common space type and a building-specific type are listed, the building-specific space type applies.

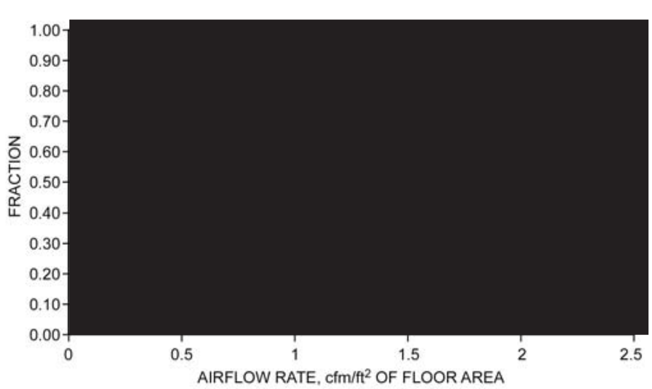

Fig. 3 Lighting Heat Gain Parameters for Recessed Fluorescent Luminaire Without Lens
(Fisher and Chantrasrisalai 2006)

Although design data presented in Table 3 and Figure 3 [redacted]

For other luminaire types, [redacted]

Because of the directional nature of downlight luminaires, a large portion of the short-wave radiation typically falls on the floor. When converting heat gains to cooling loads in the RTS method, [redacted]

Table 3 Lighting Heat Gain Parameters for Typical Operating Conditions

Luminaire Category	Space Fraction	Radiative Fraction	Notes
[redacted]	[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]	[redacted]

Source: Fisher and Chantrasrisalai (2006).

is intercepted by the floor; nonsolar RTFs assume uniform distribution by area over all interior surfaces.) This effect may be significant for rooms where lighting heat gain is high and for which solar RTFs are significantly different from nonsolar RTFs.

ELECTRIC MOTORS

Instantaneous sensible heat gain from equipment operated by electric motors in a conditioned space is calculated as

$$q_{em} = 2545(P/E_M)F_{UM}F_{LM} \tag{2}$$

where

q_{em} = heat equivalent of equipment operation, Btu/h
P = motor power rating, hp
E_M = motor efficiency, decimal fraction <1.0
F_{UM} = motor use factor, 1.0 or decimal fraction <1.0
F_{LM} = motor load factor, 1.0 or decimal fraction <1.0
2545 = conversion factor, Btu/h·hp

The motor use factor may be applied when motor use is known to be intermittent, with significant nonuse during all hours of operation (e.g., overhead door operator). For conventional applications, its value is 1.0.

The motor load factor is the fraction of the rated load delivered under the conditions of the cooling load estimate. Equation (2) assumes that both the motor and driven equipment are in the conditioned space. If the motor is outside the space or airstream,

$$q_{em} = 2545PF_{UM}F_{LM} \tag{3}$$

When the motor is inside the conditioned space or airstream but the driven machine is outside,

$$q_{em} = 2545P\left(\frac{1.0 - E_M}{E_M}\right)F_{UM}F_{LM} \tag{4}$$

Equation (4) also applies to a fan or pump in the conditioned space that exhausts air or pumps fluid outside that space.

Table 4 gives minimum efficiencies and related data representative of typical electric motors from ASHRAE *Standard* [redacted] If electric motor load is an appreciable portion of cooling load, the motor efficiency should be obtained from the manufacturer. Also, depending on design, maximum efficiency might occur anywhere between 75 to 110% of full load; if under- or overloaded, efficiency could vary from the manufacturer's listing.

Overloading or Underloading

Heat output of a motor is generally proportional to motor load, within rated overload limits. Because of typically high no-load motor current, fixed losses, and other reasons, F_{LM} is generally assumed to be unity, and no adjustment should be made for underloading or overloading unless the situation is fixed and can be accurately established, and reduced-load efficiency data can be obtained from the motor manufacturer.

Table 4 Minimum Nominal Full-Load Efficiency for 60 HZ NEMA General Purpose Electric Motors (Subtype I) Rated 600 Volts or Less (Random Wound)*

Minimum Nominal Full Load Efficiency (%) for Motors Manufactured on or after December 19, 2010						
	Open Drip-Proof Motors			Totally Enclosed Fan-Cooled Motors		
Number of Poles ⇒	2	4	6	2	4	6
Synchronous Speed (RPM) ⇒	3600	1800	1200	3600	1800	1200
Motor Horsepower						
1	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
1.5	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
2	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
3	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
5	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
7.5	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
10	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
15	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
20	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
25	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
30	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
40	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
50	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
60	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
75	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
100	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
125	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
150	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
200	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
250	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
300	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
350	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
400	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
450	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
500	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Source: ASHRAE *Standard* 90.1-2010

*Nominal efficiencies established in accordance with NEMA *Standard* MG1. Design A and Design B are National Electric Manufacturers Association (NEMA) design class designations for fixed-frequency small and medium AC squirrel-cage induction motors.

Radiation and Convection

Unless the manufacturer's technical literature indicates otherwise, motor heat gain normally should be equally divided between radiant and convective components for the subsequent cooling load calculations.

APPLIANCES

A cooling load estimate should take into account heat gain from all appliances (electrical, gas, or steam). Because of the variety of appliances, applications, schedules, use, and installations, estimates can be very subjective. Often, the only information available about heat gain from equipment is that on its nameplate, which can overestimate actual heat gain for many types of appliances, as discussed in the section on Office Equipment.

Cooking Appliances

These appliances include common heat-producing cooking equipment found in conditioned commercial kitchens. Marn (1962) concluded that [redacted]

Gordon et al. (1994) and Smith et al. (1995) found that [redacted]

Marn (1962) confirmed that, [redacted] Gordon et al. (1994) and Smith et al. (1995) substantiated these findings. Chapter [redacted] of the 2011 ASHRAE *Handbook—HVAC Applications* has more information on [redacted]

Sensible Heat Gain for Hooded Cooking Appliances. To establish a heat gain value, nameplate energy input ratings may be used with appropriate usage and radiation factors. Where specific rating data are not available (nameplate missing, equipment not yet purchased, etc.), representative heat gains listed in Tables 5A to E (Swierczyna et al. 2008, 2009) for a wide variety of commonly encountered equipment items. In estimating appliance load, [redacted]

[redacted] (Gordon et al. 1994; Smith et al. 1995; Swierczyna et al. 2008; Talbert et al. 1973). This ratio of heat gain to appliance energy consumption may be expressed as a [redacted]

Heat Gain from Meals. [redacted]

Heat Gain for Generic Appliances. The average rate of appliance energy consumption can be estimated from the [redacted] using one of the following equations:

$$[redacted] \quad (5)$$

or

$$[redacted] \quad (6)$$

where [redacted] However, recent ASHRAE research (Swierczyna et al. 2008, 2009) showed the [redacted] Because large errors could occur in the heat load calculation for specific appliance lines by using a general radiation factor, heat gain values in Table 5 should be applied in the HVAC design.

Table 5 lists usage factors, radiation factors, and load factors based on appliance energy consumption rate for typical electrical, steam, and gas appliances under standby or idle conditions, hooded and unhooded.

Recirculating Systems. Cooking appliances ventilated by recirculating systems or "ductless" hoods should be [redacted] In other words, [redacted]

Recommended Heat Gain Values. Table 5 lists recommended rates of heat gain from typical commercial cooking appliances. Data in the "hooded" columns assume installation under a properly designed exhaust hood connected to a mechanical fan exhaust system operating at an exhaust rate for complete capture and containment of the thermal and effluent plume. Improperly operating hood systems load the space with a significant convective component of the heat gain.

Hospital and Laboratory Equipment

Hospital and laboratory equipment items are [redacted] Care is needed in [redacted]

Medical Equipment. It is more difficult to provide generalized heat gain recommendations for medical equipment than for general office equipment because medical equipment is much more varied in type and in application. Some heat gain testing has been done, but the equipment included represents only a small sample of the type of equipment that may be encountered.

Data presented for medical equipment in Table 6 are relevant for [redacted] The data are presented to provide guidance in only the most general sense. For large equipment, such as MRI, heat gain must be obtained from the manufacturer.

Laboratory Equipment. Equipment in laboratories is similar to medical equipment in that [redacted]

Table 5A Recommended Rates of Radiant and Convective Heat Gain from Unhooded Electric Appliances During Idle (Ready-to-Cook) Conditions

Appliance	Energy Rate, Btu/h: Rated	Energy Rate, Btu/h: Standby	Rate of Heat Gain, Btu/h: Sensible Radiant	Sensible Convective	Latent	Total	Usage Factor F_U	Radiation Factor F_R
Cabinet: hot serving (large), insulated*	6,800	1,200	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
hot serving (large), uninsulated	6,800	3,500	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
proofing (large)*	17,400	1,400	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
proofing (small 15-shelf)	14,300	3,900	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Coffee brewing urn	13,000	1,200	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Drawer warmers, 2-drawer (moist holding)*	4,100	500	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Egg cooker	10,900	700	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Espresso machine*	8,200	1,200	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Food warmer: steam table (2-well-type)	5,100	3,500	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Freezer (small)	2,700	1,100	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Hot dog roller*	3,400	2,400	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Hot plate: single burner, high speed	3,800	3,000	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Hot-food case (dry holding)*	31,100	2,500	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Hot-food case (moist holding)*	31,100	3,300	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Microwave oven: commercial (heavy duty)	10,900	0	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Oven: countertop conveyorized bake/finishing*	20,500	12,600	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Panini*	5,800	3,200	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Popcorn popper*	2,000	200	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Rapid-cook oven (quartz-halogen)*	41,000	0	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Rapid-cook oven (microwave/convection)*	24,900	4,100	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Reach-in refrigerator*	4,800	1,200	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Refrigerated prep table*	2,000	900	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Steamer (bun)	5,100	700	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Toaster: 4-slice pop up (large): cooking	6,100	3,000	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
contact (vertical)	11,300	5,300	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
conveyor (large)	32,800	10,300	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
small conveyor	5,800	3,700	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Waffle iron	3,100	1,200	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 5B Recommended Rates of Radiant Heat Gain from Hooded Electric Appliances During Idle (Ready-to-Cook) Conditions

Appliance	Energy Rate, Btu/h: Rated	Energy Rate, Btu/h: Standby	Rate of Heat Gain, Btu/h: Sensible Radiant	Usage Factor F_U	Radiation Factor F_R
Broiler: underfired 3 ft	36,900	30,900	[redacted]	[redacted]	[redacted]
Cheesemelter*	12,300	11,900	[redacted]	[redacted]	[redacted]
Fryer: kettle	99,000	1,800	[redacted]	[redacted]	[redacted]
Fryer: open deep-fat, 1-vat	47,800	2,800	[redacted]	[redacted]	[redacted]
Fryer: pressure	46,100	2,700	[redacted]	[redacted]	[redacted]
Griddle: double sided 3 ft (clamshell down)*	72,400	6,900	[redacted]	[redacted]	[redacted]
Griddle: double sided 3 ft (clamshell up)*	72,400	11,500	[redacted]	[redacted]	[redacted]
Griddle: flat 3 ft	58,400	11,500	[redacted]	[redacted]	[redacted]
Griddle-small 3 ft*	30,700	6,100	[redacted]	[redacted]	[redacted]
Induction cooktop*	71,700	0	[redacted]	[redacted]	[redacted]
Induction wok*	11,900	0	[redacted]	[redacted]	[redacted]
Oven: combi: combi-mode*	56,000	5,500	[redacted]	[redacted]	[redacted]
Oven: combi: convection mode	56,000	5,500	[redacted]	[redacted]	[redacted]
Oven: convection full-size	41,300	6,700	[redacted]	[redacted]	[redacted]
Oven: convection half-size*	18,800	3,700	[redacted]	[redacted]	[redacted]
Pasta cooker*	75,100	8,500	[redacted]	[redacted]	[redacted]
Range top: top off/oven on*	16,600	4,000	[redacted]	[redacted]	[redacted]
Range top: 3 elements on/oven off	51,200	15,400	[redacted]	[redacted]	[redacted]
Range top: 6 elements on/oven off	51,200	33,200	[redacted]	[redacted]	[redacted]
Range top: 6 elements on/oven on	67,800	36,400	[redacted]	[redacted]	[redacted]
Range: hot-top	54,000	51,300	[redacted]	[redacted]	[redacted]
Rotisserie*	37,900	13,800	[redacted]	[redacted]	[redacted]
Salamander*	23,900	23,300	[redacted]	[redacted]	[redacted]
Steam kettle: large (60 gal) simmer lid down*	110,600	2,600	[redacted]	[redacted]	[redacted]
Steam kettle: small (40 gal) simmer lid down*	73,700	1,800	[redacted]	[redacted]	[redacted]
Steamer: compartment: atmospheric*	33,400	15,300	[redacted]	[redacted]	[redacted]
Tilting skillet/braising pan	32,900	5,300	[redacted]	[redacted]	[redacted]

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 5C Recommended Rates of Radiant Heat Gain from Hooded Gas Appliances During Idle (Ready-to-Cook) Conditions

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h		
Appliance	Rated	Standby	Sensible Radiant	Usage Factor F_U	Radiation Factor F_R
Broiler: batch*	95,000	69,200	[redacted]		
Broiler: chain (conveyor)	132,000	96,700			
Broiler: overfired (upright)*	100,000	87,900			
Broiler: underfired 3 ft	96,000	73,900			
Fryer: doughnut	44,000	12,400			
Fryer: open deep-fat, 1 vat	80,000	4,700			
Fryer: pressure	80,000	9,000			
Griddle: double sided 3 ft (clamshell down)*	108,200	8,000			
Griddle: double sided 3 ft (clamshell up)*	108,200	14,700			
Griddle: flat 3 ft	90,000	20,400			
Oven: combi: combi-mode*	75,700	6,000			
Oven: combi: convection mode	75,700	5,800			
Oven: convection full-size	44,000	11,900			
Oven: conveyor (pizza)	170,000	68,300			
Oven: deck	105,000	20,500			
Oven: rack mini-rotating*	56,300	4,500			
Pasta cooker*	80,000	23,700			
Range top: top off/oven on*	25,000	7,400			
Range top: 3 burners on/oven off	120,000	60,100			
Range top: 6 burners on/oven off	120,000	120,800			
Range top: 6 burners on/oven on	145,000	122,900			
Range: wok*	99,000	87,400			
Rethermalizer*	90,000	23,300			
Rice cooker*	35,000	500			
Salamander*	35,000	33,300			
Steam kettle: large (60 gal) simmer lid down*	145,000	5,400			
Steam kettle: small (10 gal) simmer lid down*	52,000	3,300			
Steam kettle: small (40 gal) simmer lid down	100,000	4,300			
Steamer: compartment: atmospheric*	26,000	8,300			
Tilting skillet/braising pan	104,000	10,400			

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 5D Recommended Rates of Radiant Heat Gain from Hooded Solid Fuel Appliances During Idle (Ready-to-Cook) Conditions

	Energy Rate, Btu/h	Rate of Heat Gain, Btu/h			
Appliance	Rated	Standby	Sensible	Usage Factor F_U	Radiation Factor F_R
Broiler: solid fuel: charcoal	40 lb	42,000	[redacted]	[redacted]	[redacted]
Broiler: solid fuel: wood (mesquite)*	40 lb	49,600	[redacted]	[redacted]	[redacted]

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 5E Recommended Rates of Radiant and Convective Heat Gain from Warewashing Equipment During Idle (Standby) or Washing Conditions

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h: Unhooded				Hooded		
Appliance	Rated	Standby/ Washing	Sensible Radiant	Sensible Convective	Latent	Total	Sensible Radiant	Usage Factor F_U	Radiation Factor F_R
Dishwasher (conveyor type, chemical sanitizing)	46,800	5700/43,600	[redacted]						
Dishwasher (conveyor type, hot-water sanitizing) standby	46,800	5700/N/A							
Dishwasher (door-type, chemical sanitizing) washing	18,400	1200/13,300							
Dishwasher (door-type, hot-water sanitizing) washing	18,400	1200/13,300							
Dishwasher* (under-counter type, chemical sanitizing) standby	26,600	1200/18,700							
Dishwasher* (under-counter type, hot-water sanitizing) standby	26,600	1700/19,700							
Booster heater*	130,000	0							

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Note: Heat load values are prorated for 30% washing and 70% standby.

space. Chapter 16 of the 2011 *ASHRAE Handbook—HVAC Applications* discusses heat gain from equipment, which may range from [redacted] Table 7 lists some values for laboratory equipment, but, as with medical equipment, it is for general guidance only. Wilkins and Cook (1999) also examined laboratory equipment heat gains.

Office Equipment

Computers, printers, copiers, etc., can generate [redacted] ASHRAE research project [redacted] developed a [redacted] (Hosni et al. 1998; Jones et al. 1998). This

Table 6 Recommended Heat Gain from Typical Medical Equipment

Equipment	Nameplate, W	Peak, W	Average, W
Anesthesia system	[redacted]	[redacted]	[redacted]
Blanket warmer	[redacted]	[redacted]	[redacted]
Blood pressure meter	[redacted]	[redacted]	[redacted]
Blood warmer	[redacted]	[redacted]	[redacted]
ECG/RESP	[redacted]	[redacted]	[redacted]
Electrosurgery	[redacted]	[redacted]	[redacted]
Endoscope	[redacted]	[redacted]	[redacted]
Harmonical scalpel	[redacted]	[redacted]	[redacted]
Hysteroscopic pump	[redacted]	[redacted]	[redacted]
Laser sonics	[redacted]	[redacted]	[redacted]
Optical microscope	[redacted]	[redacted]	[redacted]
Pulse oximeter	[redacted]	[redacted]	[redacted]
Stress treadmill	[redacted]	[redacted]	[redacted]
Ultrasound system	[redacted]	[redacted]	[redacted]
Vacuum suction	[redacted]	[redacted]	[redacted]
X-ray system	[redacted]	[redacted]	[redacted]

Source: Hosni et al. (1999).

Table 7 Recommended Heat Gain from Typical Laboratory Equipment

Equipment	Nameplate, W	Peak, W	Average, W
Analytical balance	[redacted]	[redacted]	[redacted]
Centrifuge	[redacted]	[redacted]	[redacted]
Electrochemical analyzer	[redacted]	[redacted]	[redacted]
Flame photometer	[redacted]	[redacted]	[redacted]
Fluorescent microscope	[redacted]	[redacted]	[redacted]
Function generator	[redacted]	[redacted]	[redacted]
Incubator	[redacted]	[redacted]	[redacted]
Orbital shaker	[redacted]	[redacted]	[redacted]
Oscilloscope	[redacted]	[redacted]	[redacted]
Rotary evaporator	[redacted]	[redacted]	[redacted]
Spectronics	[redacted]	[redacted]	[redacted]
Spectrophotometer	[redacted]	[redacted]	[redacted]
Spectro fluorometer	[redacted]	[redacted]	[redacted]
Thermocycler	[redacted]	[redacted]	[redacted]
Tissue culture	[redacted]	[redacted]	[redacted]

Source: Hosni et al. (1999).

[redacted] was then incorporated into ASHRAE research project [redacted] and [redacted] The following general guidelines for office equipment are a result of these studies.

Nameplate Versus Measured Energy Use. [redacted]

Computers. Based on tests by Hosni et al. (1999) and Wilkins and McGaffin (1994), [redacted]

Monitors. Based on monitors tested by Hosni et al. (1999), [redacted]

[redacted]

[redacted]

[redacted]

[redacted]

Table 8 shows typical values.

Flat-panel monitors have replaced CRT monitors in many workplaces. Power consumption, and thus heat gain, for flat-panel displays are [redacted]

Laser Printers. Hosni et al. (1999) found [redacted]

Table 9 presents data on laser printers. These data can be [redacted]

Copiers. Hosni et al. (1999) also tested five photocopy machines, including [redacted]

Miscellaneous Office Equipment. Table 10 presents data on miscellaneous office equipment such as [redacted]

Diversity. The ratio of measured peak electrical load at equipment panels to the sum of the maximum electrical load of each individual item of equipment is the usage diversity. A small, one- or two-person office containing equipment listed in Tables 8 to 10

Table 8 Recommended Heat Gain from Typical Computer Equipment

Equipment	Description	Nameplate Power, W	Average Power, W	Radiant Fraction
Desktop computer[a]	Manufacturer A (model A); 2.8 GHz processor, 1 GB RAM	[redacted]		
	Manufacturer A (model B); 2.6 GHz processor, 2 GB RAM			
	Manufacturer B (model A); 3.0 GHz processor, 2 GB RAM			
	Manufacturer B (model B); 3.0 GHz processor, 2 GB RAM			
	Manufacturer A (model C); 2.3 GHz processor, 3 GB RAM			
Laptop computer[b]	Manufacturer 1; 2.0 GHz processor, 2 GB RAM, 17 in. screen			
	Manufacturer 1; 1.8 GHz processor, 1 GB RAM, 17 in. screen			
	Manufacturer 1; 2.0 GHz processor, 2 GB RAM, 14 in. screen			
	Manufacturer 2; 2.13 GHz processor, 1 GB RAM, 14 in. screen, tablet PC			
	Manufacturer 2; 366 MHz processor, 130 MB RAM (4 in. screen)			
	Manufacturer 3; 900 MHz processor, 256 MB RAM (10.5 in. screen)			
Flat-panel monitor[c]	Manufacturer X (model A); 30 in. screen			
	Manufacturer X (model B); 22 in. screen			
	Manufacturer Y (model A); 19 in. screen			
	Manufacturer Y (model B); 17 in. screen			
	Manufacturer Z (model A); 17 in. screen			
	Manufacturer Z (model C); 15 in. screen			

Source: Hosni and Beck (2008).

[a]Power consumption for [redacted]

[b]Power consumption of [redacted]

[redacted]

Table 9 Recommended Heat Gain from Typical Laser Printers and Copiers

Equipment	Description	Nameplate Power, W	Average Power, W	Radiant Fraction
Laser printer, typical desktop, small-office type[a]	Printing speed up to 10 pages per minute	[redacted]		
	Printing speed up to 35 pages per minute			
	Printing speed up to 19 pages per minute			
	Printing speed up to 17 pages per minute			
	Printing speed up to 19 pages per minute			
	Printing speed up to 24 page per minute			
Multifunction (copy, print, scan)[b]	Small, desktop type			
	Medium, desktop type			
Scanner[b]	Small, desktop type			
Copy machine[c]	Large, multiuser, office type			
Fax machine	Medium			
	Small			
Plotter	Manufacturer A			
	Manufacturer B			

Source: Hosni and Beck (2008).

[a]Various [redacted]

[b]Small [redacted]

[c]Power consumption for [redacted]

usually [redacted]

Wilkins and McGaffin (1994) measured diversity in 23 areas within five different buildings totaling over 275,000 ft^2. Diversity was found to range between 37 and 78%, with the average (normalized based on area) being 46%. Figure 4 illustrates the relationship between [redacted]

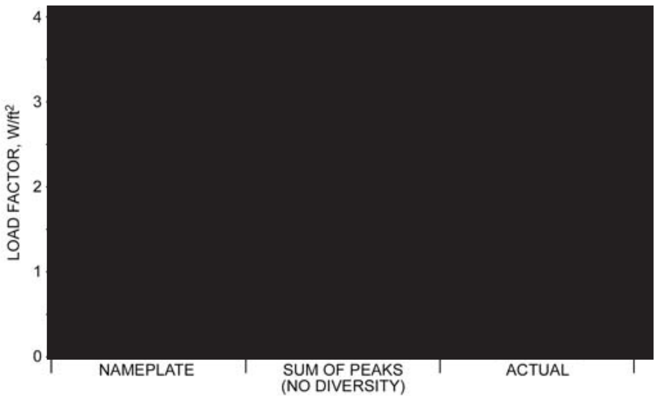

Fig. 4 Office Equipment Load Factor Comparison
(Wilkins and McGaffin 1994)

Table 10 Recommended Heat Gain from Miscellaneous Office Equipment

Equipment	Maximum Input Rating, W	Recommended Rate of Heat Gain, W
Mail-processing equipment	[redacted]	
Folding machine		
Inserting machine, 3600 to 6800 pieces/h		
Labeling machine, 1500 to 30,000 pieces/h		
Postage meter		
Vending machines		
Cigarette		
Cold food/beverage		
Hot beverage		
Snack		
Other		
Bar code printer		
Cash registers		
Check processing workstation, 12 pockets		
Coffee maker, 10 cups		
Microfiche reader		
Microfilm reader		
Microfilm reader/printer		
Microwave oven, 1 ft³		
Paper shredder		
Water cooler, 32 qt/h		

ASHRAE research project [redacted] (Abushakra et al. 2004; Claridge et al. 2004). [redacted]

Heat Gain per Unit Area. Wilkins and Hosni (2000, 2011) and Wilkins and McGaffin (1994) [redacted] Table 11 presents a range of load factors with a subjective description of the type of space to which they would apply. [redacted]

Table 11 Recommended Load Factors for Various Types of Offices

Type of Use	Load Factor, W/ft²	Description
100% Laptop, light	[redacted]	
medium		
50% Laptop, light		
medium		
100% Desktop, light		
medium		
100% Desktop, two monitors		
100% Desktop, heavy		
100% Desktop, full on		

Source: Wilkins and Hosni (2011).

Table 12 Recommended Diversity Factors for Office Equipment

Device	Recommended Diversity Factor
Desktop computer	[redacted]
LCD monitor	[redacted]
Notebook computer	[redacted]

[redacted] Table 12 [redacted]

Radiant/Convective Split. ASHRAE research project [redacted] (Hosni and Beck 2008) [redacted]

INFILTRATION AND MOISTURE MIGRATION HEAT GAINS

Two other load components contribute to space cooling load directly without time delay from building mass: (1) infiltration, and (2) moisture migration through the building envelope.

INFILTRATION

Principles of estimating infiltration in buildings, with emphasis on the heating season, are discussed in Chapter [redacted] When economically feasible, [redacted] However, there is concern, especially in some climates, that [redacted]

When positive air pressure is assumed, most designers do not include infiltration in cooling load calculations for commercial buildings. However, including some infiltration for spaces [redacted]

[redacted]

Infiltration also depends [redacted] Designers usually predict overall rates of infiltration using the number of **air changes per hour (ACH)**. A common guideline for climates and buildings [redacted]

Standard Air Volumes

Because the specific volume of air varies appreciably, calculations are [redacted] One standard value is [redacted] When airflow is to be measured at a particular condition or point, such as at a coil entrance or exit, the corresponding specific volume [redacted]

Heat Gain Calculations Using Standard Air Values

Air-conditioning design often requires the following information:

1. Total heat

   Total heat gain q_t corresponding to the change of a given standard flow rate Q_s through an enthalpy difference Δh is

[redacted] (8)

where [redacted]

This total heat equation can also be expressed as

[redacted]

where [redacted] is the air total heat factor, in Btu/h·cfm per Btu/lb enthalpy h.

2. Sensible heat

   Sensible heat gain q_s corresponding to the change of dry-bulb temperature Δt for given airflow (standard conditions) Q_s is

[redacted] (10)

This sensible heat equation can also be expressed as

[redacted]

3. Latent heat

   Latent heat gain q_l corresponding to the change of humidity ratio [redacted]

where [redacted]

This latent heat equation can also be expressed as

[redacted]

4. Elevation correction for total, sensible, and latent heat equations

   The constants [redacted]

Elevation Correction Examples

To correct the C values for El Paso, Texas, the elevation listed in the appendix of Chapter 14 is 3917 ft. C values for Equations (8) to (11) can be corrected using Equation (3) in Chapter 1 as follows:

$$C_{t,3917} = 4.5 \times [1 - (3917 \times 6.8754 \times 10^{-6})]^{5.2559} = 3.90$$

$$C_{s,3917} = 1.10 \times [1 - (3917 \times 6.8754 \times 10^{-6})]^{5.2559} = 0.95$$

$$C_{l,3917} = 4840 \times [1 - (3917 \times 6.8754 \times 10^{-6})]^{5.2559} = 4192$$

To correct the C values for Albuquerque, New Mexico, the elevation listed in the appendix of Chapter 14 is 5315 ft. C values for Equations (8) to (11) can be corrected as follows:

$$C_{t,5315} = 4.5 \times [1 - (5315 \times 6.8754 \times 10^{-6})]^{5.2559}\ C_{t,531} = 3.70$$

$$C_{s,5315} = 1.10 \times [1 - (5315 \times 6.8754 \times 10^{-6})]^{5.2559}\ C_{s,5315} = 0.90$$

$$C_{l,5315} = 4840 \times [1 - (5315 \times 6.8754 \times 10^{-6})]^{5.2559}\ C_{l,5315} = 3979$$

LATENT HEAT GAIN FROM MOISTURE DIFFUSION

Diffusion of moisture through building materials is a natural phenomenon that is always present. Chapters [redacted] cover principles, materials, and specific methods used to control moisture. Moisture transfer through walls and roofs is often neglected in comfort air conditioning because the actual rate is quite small and the corresponding latent heat gain is insignificant. Permeability and permeance values for various building materials are given in Chapter [redacted] Vapor retarders should be [redacted] Moisture migration [redacted]

[redacted] Under-slab continuous moisture retarders and drainage [redacted]

Some industrial applications require low moisture to be maintained in a conditioned space. In these cases, the latent heat gain accompanying moisture transfer through walls and roofs may be greater than any other latent heat gain. This gain is computed by

[redacted]

[redacted]

q_l [redacted]

OTHER LATENT LOADS

Moisture sources within a building (e.g., shower areas, swimming pools or natatoriums, arboretums) [redacted] Unlike sensible loads, which [redacted] However, system loads associated with [redacted]

For natatoriums, occupant comfort and humidity control are [redacted] Chapter [redacted] of the [redacted]

FENESTRATION HEAT GAIN

For spaces with neutral or positive air pressurization, the primary weather-related variable affecting cooling load is solar radiation. The effect of solar radiation is [redacted] Chapter [redacted] includes [redacted] discussed in Chapter [redacted] This chapter covers [redacted]

FENESTRATION DIRECT SOLAR, DIFFUSE SOLAR, AND CONDUCTIVE HEAT GAINS

For fenestration heat gain, use the following equations:

[redacted]

If specific window manufacturer's SHGC and U-factor data are available, those should be used. For fenestration equipped with indoor shading (blinds, drapes, or shades), the indoor solar attenuation coefficients [redacted] are listed in Tables 13A to 13G of Chapter [redacted]

Note that, as discussed in Chapter [redacted] fenestration ratings [redacted] are based on the entire product area, including frames. Thus, for load calculations, fenestration area is the area [redacted]

EXTERIOR SHADING

Nonuniform exterior shading, caused by roof overhangs, side fins, or building projections, [redacted] The areas, shaded and unshaded, depend on the location of the shadow line on a surface in the plane of the glass. Sun (1968) developed [redacted] McQuiston and Spitler (1992) [redacted]

Equations for calculating shade angles [Chapter 15, Equations (34) to (37)] can be used to [redacted]

HEAT BALANCE METHOD

Cooling load estimation involves calculating a surface-by-surface conductive, convective, and radiative heat balance for each room surface and a convective heat balance for the room air. These principles form the foundation for all methods described in this chapter. The heat balance (HB) method solves the problem directly instead of introducing transformation-based procedures. The advantages are [redacted]

Some computations required by this rigorous approach require the use of computers. The heat balance procedure is not new. Many energy calculation programs have used it in some form for many years. The first implementation that incorporated all the elements to form a complete method was [redacted] The heat balance procedure is also [redacted] Before ASHRAE research project [redacted]

The HB method is codified in the software [redacted]

ASHRAE research project [redacted] HB calculations [redacted]

ASSUMPTIONS

All calculation procedures involve some kind of model; all models require simplifying assumptions and, therefore, are approximate. The most fundamental assumption is that air in the thermal zone can be modeled as [redacted] ASHRAE research project [redacted] established that this assumption is valid over a wide range of conditions.

[redacted]

[redacted]

The resulting formulation is called [redacted] Note that the assumptions, although common, are quite restrictive and set certain limits on the information that can be obtained from the model.

ELEMENTS

Within the framework of the assumptions, the HB can be viewed as [redacted] processes:

[redacted]

Figure 5 shows the relationship between these processes for a single opaque surface. The top part of the figure, inside the shaded box, is repeated for each surface enclosing the zone. The process for transparent surfaces is [redacted]

Outdoor-Face Heat Balance

The heat balance on the outdoor face of each surface is

[redacted]

Fig. 5 Schematic of Heat Balance Processes in Zone

[redacted]

All terms are [redacted]

Each term in Equation (17) has been modeled in several ways, and in [redacted]

Wall Conduction Process

The wall conduction process has been formulated in more ways than any of the other processes. Techniques include

[redacted]

This process introduces [redacted] Figure 6 [redacted] Direct formulation of the process uses [redacted]

In some models, [redacted]

[redacted]

Fig. 6 Schematic of Wall Conduction Process

[redacted]

Because heat balances on both sides of the element [redacted]

Indoor-Face Heat Balance

The heart of the HB method is the [redacted] The indoor-face heat balance for each surface can be written as follows:

[redacted]

These terms are explained in the following paragraphs.

LW Radiation Exchange Among Zone Surfaces. The limiting cases for modeling internal LW radiation exchange are

[redacted] However, because the [redacted]

SW Radiation from Lights. [redacted]

LW Radiation from Internal Sources. [redacted] ASHRAE research project [redacted]

Transmitted Solar Heat Gain. Chapter [redacted] calculation procedure for determining transmitted solar energy through fenestration uses the [redacted]

Transmitted solar radiation is [redacted]

Using SHGC to Calculate Solar Heat Gain

The total solar heat gain through fenestration consists of [redacted] As such, it [redacted]

$$\mathrm{SHGC} = \tau + \sum_{k=1}^{n} N_k \alpha_k \tag{19}$$

[redacted]

Note that Equation (19) is written [redacted]

Table 13 Single-Layer Glazing Data Produced by WINDOW 5.2

Parameter	Incident Angle 0	10	20	30	40	50	60	70	80	90	Diffuse (Hemis.)
V_{tc}	0.899	0.899	0.898	0.896	0.889	0.870	0.822	0.705	0.441	0	0.822
R_{fv}	0.083	0.083	0.083	0.085	0.091	0.109	0.156	0.272	0.536	1	0.148
R_{bv}	0.083	0.083	0.083	0.085	0.091	0.109	0.156	0.272	0.536	1	0.148
T_{sol}	0.834	0.833	0.831	0.827	0.818	0.797	0.749	0.637	0.389	0	0.753
R_f	0.075	0.075	0.075	0.077	0.082	0.099	0.143	0.253	0.506	1	0.136
R_b	0.075	0.075	0.075	0.077	0.082	0.099	0.143	0.253	0.506	1	0.136
A_{bs1}	0.091	0.092	0.094	0.096	0.100	0.104	0.108	0.110	0.105	0	0.101
SHGC	0.859	0.859	0.857	0.854	0.845	0.825	0.779	0.667	0.418	0	0.781

Source: LBL (2003).

[redacted]

The only way to model these interactions correctly is to [redacted] as discussed in the following paragraphs.

Using SHGC Data. The normal incidence SHGC used to rate and characterize glazing systems is not sufficient for determining solar heat gain for load calculations. These calculations require solar heat gain as a function of the incident solar angle in order to determine the hour-by-hour gain profile. Thus, it is necessary to use angular SHGC values and also diffuse SHGC values. These can be obtained from the WINDOW 5.2 program (LBL 2003). This program does a detailed optical and thermal simulation of a glazing system and, when applied to a single clear layer, produces the information shown in Table 13.

Table 13 shows the parameters as a function of incident solar angle and also the diffuse values. The specific parameters shown are

V_{tc} = transmittance in visible spectrum
R_{fv} and R_{bv} = front and back surface visible reflectances
T_{sol} = solar transmittance [τ in Equations (19), (20), and (21)]
R_f and R_b = front and back surface solar reflectances
A_{bs1} = solar absorptance for layer 1, which is the only layer in this case [α in Equations (19), (20), and (21)]
SHGC = solar heat gain coefficient at the center of the glazing

The parameters used for heat gain calculations are T_{sol}, A_{bs}, and SHGC. For the specific convective conditions assumed in WINDOW 5.2 program, the inward-flowing fraction of the absorbed solar can be obtained by rearranging Equation (19) to give

$$N_k \alpha_k = \text{SHGC} - \tau \tag{20}$$

This quantity, when multiplied by the appropriate incident solar intensity, provides the amount of absorbed solar radiation that flows [redacted]

[redacted]

There is some potential [redacted]

[redacted]

This procedure is incorporated into the HB method so the solar gain is calculated accurately for each hour.

Table 10 in Chapter [redacted] contains SHGC information for many additional glazing systems. That table is similar to Table 13 but is [redacted]

Convection to Zone Air. Indoor convection coefficients presented in past editions of this chapter and used in most load calculation procedures and energy programs are based on very old, natural convection experiments and do not accurately describe heat transfer coefficients in a mechanically ventilated zone. In previous load calculation procedures, these coefficients were buried in the procedures and could not be changed. A heat balance formulation keeps them as working parameters. In this way, research results such as those from ASHRAE research project [redacted]

Air Heat Balance

In HB formulations aimed at determining cooling loads, the capacitance of air in the zone is [redacted]

[redacted]

Convection from zone surfaces[redacted]

The **convective parts of the internal loads**[redacted]

In keeping with the well-mixed model for zone air, any air that [redacted] The amount of infiltration or natural ventilation air is [redacted]n. Sometimes it is [redacted]

Conditioned air that enters the zone from the HVAC system and provides [redacted] as discussed in Chapter [redacted]

GENERAL ZONE FOR LOAD CALCULATION

The HB procedure is tailored to a single thermal zone, shown in Figure 7. The definition of a thermal zone depends on [redacted]

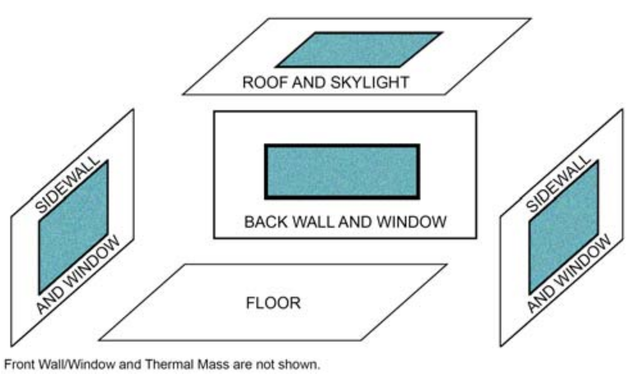

Fig. 7 Schematic View of General Heat Balance Zone

[redacted]

The heat balance processes for this general zone are formulated for [redacted]

MATHEMATICAL DESCRIPTION

Conduction Process

Because it links the outdoor and indoor heat balances, the wall conduction process regulates the cooling load's time dependence. For the HB procedure presented here, wall conduction is formulated [redacted]

For outdoor heat flux, the form is

[redacted]

where

[redacted]

The subscript following the comma indicates the time period for the quantity in terms of time step δ. Also, the first terms in the series have been [redacted]

Heat Balance Equations

The primary variables in the heat balance for the general zone are the [redacted]

[redacted]

Equations (17) and (24) are [redacted] (25)

[redacted]

where

T_o = outdoor air temperature

h_{co} = outdoor convection coefficient, introduced by using $q''_{conv} = h_{co}(T_o - T_{so})$

Equation (25) shows the need to [redacted]

Equations (18) and (23) are [redacted] (26)

[redacted]

where

T_a = zone air temperature

h_{ci} = convective heat transfer coefficient indoors, obtained from $q''_{conv} = h_{ci}(T_a - T_{si})$

Note that in Equations (25) and (26), the [redacted] (27)

In Equation (27), the [redacted]

Overall HB Iterative Solution

The iterative HB procedure consists of a [redacted]

```
For Day = 1 to Maxdays
    For j = 1 to 24                         {hours in the day}
        For SurfaceIter = 1 to MaxIter
            For i = 1 to 12                 {The twelve zone surfaces}
                Evaluate Equations (34) and (35)
            Next i
        Next SurfaceIter
        Evaluate Equation (36)
    Next j
If not converged, Next Day
```

8. Display results.

Generally, [redacted] are [redacted]

INPUT REQUIRED

Previous methods for calculating cooling loads attempted to simplify the procedure by precalculating representative cases and grouping the results with various correlating parameters. This generally tended to reduce the amount of information required to apply the procedure. With heat balance, no precalculations are made, so the procedure requires a fairly complete description of the zone.

Global Information. Because the procedure incorporates [redacted] Additionally, to take full advantage of the flexibility of the method to incorporate, for example, variable outdoor heat transfer coefficients, things such as wind speed, wind direction, and terrain roughness may be specified. [redacted]

Wall Information (Each Wall). Because the walls are involved in [redacted] of the fundamental processes [redacted] They include

[redacted]

Again, some of these parameters can be defaulted, but they are changeable, and they indicate the more fundamental character of the HB method because [redacted]

Window Information (Each Window). The situation for windows is similar to that for walls, but the windows require [redacted]

[redacted]

Roof and Floor Details. The roof and floor surfaces are specified similarly to walls. The main difference is that the ground [redacted]

Thermal Mass Surface Details. An "extra" surface, called a thermal mass surface, can serve several functions. It is included in radiant heat exchange with the other surfaces in the space but is only exposed to the indoor air convective boundary condition. As an example, this surface would be used to account for movable partitions in a space. Partition construction is specified layer by layer, similar to specification for walls, and those layers store and release heat by the same conduction mechanism as walls. As a general definition, the extra thermal mass surface should be [redacted] In the formulation, [redacted]

Internal Heat Gain Details. The space can be subjected to several internal heat sources: people, lights, electrical equipment, and infiltration. Infiltration energy is assumed to [redacted] These include the following fractions:

[redacted]

Radiant Distribution Functions. As mentioned previously, the generally accepted assumptions for the HB method include [redacted]

Other Required Information. Additional flexibility is included in the model so that results of research can be incorporated easily. This includes the capability to specify such things as

[redacted]

The amount of input information required may seem extensive, but many parameters can be set to default values in most routine applications. However, all parameters listed can be changed when necessary to fit unusual circumstances or when additional information is obtained.

RADIANT TIME SERIES (RTS) METHOD

The radiant time series (RTS) method is a [redacted] that is derived from the heat balance (HB) method. It [redacted] such as the transfer function method (TFM), the cooling load temperature difference/cooling load factor (CLTD/CLF) method, and the total equivalent temperature difference/time averaging (TETD/TA) method.

[redacted]

The RTS method is suitable for [redacted] Although simple in concept, RTS involves too many calculations for practical use as a manual method, [redacted] For a manual cooling load calculation method, refer to the CLTD/CLF method in Chapter [redacted] of the 1997 *ASHRAE Handbook—Fundamentals*.

ASSUMPTIONS AND PRINCIPLES

Design cooling loads are based on the assumption of **steady-periodic conditions** (i.e., the design day's weather, [redacted] This assumption is the basis for the RTS derivation from the HB method.

Cooling load calculations must address [redacted]

Exterior walls and roofs conduct heat because of temperature differences between outdoor and indoor air. In addition, solar energy on exterior surfaces is absorbed, then transferred by conduction to the building interior. Because of the mass and thermal capacity of the wall or roof construction materials, there is a substantial time delay in heat input at the exterior surface becoming heat gain at the interior surface.

As described in the section on Cooling Load Principles, most heat sources transfer energy to a room by a combination of convection and radiation. The convective part of heat gain immediately [redacted] The radiative part must [redacted] Thus, [redacted]

OVERVIEW

Figure 8 gives an overview of the RTS method. When calculating solar radiation, transmitted solar heat gain through windows, sol-air temperature, and infiltration, RTS is [redacted] as previous simplified methods (TFM and TETD/TA). Important areas that differ from previous simplified methods include

[redacted]

[redacted]

Fig. 8 Overview of Radiant Time Series Method

[redacted]

The RTS method accounts for both [redacted]

These series can be used to [redacted]

Figure 9 illustrates conduction time series (CTS) values for three walls with similar U-factors but with light to heavy construction. Figure 10 illustrates CTS for three walls with similar construction but with different amounts of insulation, thus with significantly different [redacted] Figure 11 illustrates RTS values for zones varying from light to heavy construction.

RTS PROCEDURE

The general procedure for calculating cooling load for each load component (lights, people, walls, roofs, windows, appliances, etc.) with RTS is as follows:

[redacted]

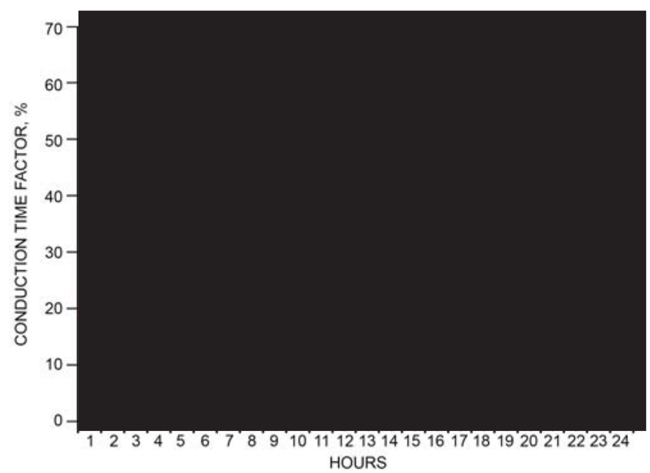

Fig. 9 CTS for Light to Heavy Walls

[redacted]

After calculating cooling loads for [redacted]

Table 14 Recommended Radiative/Convective Splits for Internal Heat Gains

Heat Gain Type	Recommended Radiative Fraction	Recommended Convective Fraction	Comments
Occupants, typical office conditions	[redacted]		
Equipment			
Office, with fan			
Without fan			
Lighting			
Conduction heat gain			
Through walls and floors			
Through roof			
Through windows			
Solar heat gain through fenestration			
Without interior shading			
With interior shading			
Infiltration			

Source: Nigusse (2007).

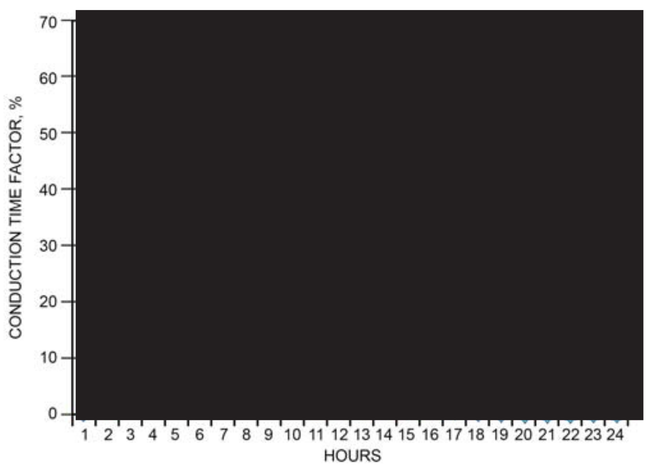

Fig. 10 CTS for Walls with Similar Mass and Increasing Insulation

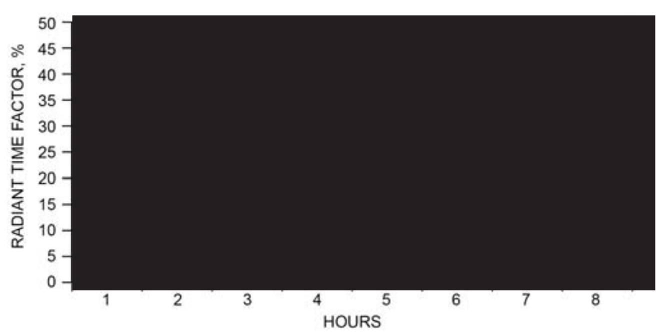

Fig. 11 RTS for Light to Heavy Construction

HEAT GAIN THROUGH EXTERIOR SURFACES

Heat gain through exterior opaque surfaces is [redacted]

Sol-Air Temperature

Sol-air temperature is the outdoor air temperature that, in the absence of all radiation changes [redacted]

Heat Flux into Exterior Sunlit Surfaces. The heat balance at a sunlit surface gives the heat flux into the surface q/A as

[redacted] (28)

[redacted]

Assuming the rate of heat transfer can be expressed in terms of the sol-air temperature t_e,

[redacted] (29)

and from Equations (28) and (29),

[redacted] (30)

For **horizontal surfaces** that receive long-wave radiation from the sky only, [redacted]

Because **vertical surfaces** receive long-wave radiation from the ground and surrounding buildings as well as from the sky, accurate ΔR values are difficult to determine. When solar radiation intensity is high, surfaces of terrestrial objects usually have a higher temperature than the outdoor air; thus, their long-wave radiation compensates to some extent for the sky's low emittance. Therefore, it is common practice to assume [redacted]

Tabulated Temperature Values. The sol-air temperatures in Example Cooling and Heating Load Calculations section have been calculated based on [redacted] calculations were calculated using equations in Chapter 1[redacted]

Table 15 Solar Absorptance Values of Various Surfaces

Surface	Absorptance
Brick, red (Purdue) [a]	[redacted]
Paint	
Red[b]	
Black, matte[b]	
Sandstone[b]	
White acrylic[a]	
Sheet metal, galvanized	
New[a]	
Weathered[a]	
Shingles	
Gray[b]	
Brown[b]	
Black[b]	
White[b]	
Concrete[a,c]	

[a]Incropera and DeWitt (1990).
[b]Parker et al. (2000).
[c]Miller (1971).

Surface Colors. Sol-air temperature values are given in the Example Cooling and Heating Load Calculations section for two values of the parameter [redacted] Solar absorptance values of various surfaces are included in Table 15.

This procedure was used to calculate the sol-air temperatures included in the Examples section. Because of the tedious solar angle and intensity calculations, using a simple computer spreadsheet or other software for these calculations can reduce the effort involved.

Calculating Conductive Heat Gain Using Conduction Time Series

In the RTS method, conduction through exterior walls and roofs is calculated using [redacted] Wall and roof conductive heat input at the exterior is defined by the familiar conduction equation as

[redacted] (31)

[redacted]

Conductive heat gain through walls or roofs can be calculated using conductive heat inputs for the current hours and past 23 h and conduction time series:

[redacted] (32)

[redacted]

Conduction time factors for representative wall and roof types are included in Tables 16 and 17. Those values were derived by first [redacted] as demonstrated by Spitler and Fisher (1999a). The periodic response factors were further simplified by [redacted] in Equation (32) and [redacted] Tables 16 and 17 are listed in Table 18.

Heat gains calculated for walls or roofs using periodic response factors (and thus CTS) are [redacted] The methodology for calculating periodic response factors from conduction transfer functions was originally developed as part of ASHRAE research project [redacted] (Spitler and Fisher 1999b; Spitler et al. 1997). For walls and roofs that are [redacted] described by Iu and Fisher (2004). For walls and roofs with thermal bridges, the procedure described by Karambakkam et al. (2005) [redacted]

The tedious calculations involved make a simple computer spreadsheet or other computer software a useful labor saver.

HEAT GAIN THROUGH INTERIOR SURFACES

Whenever a conditioned space is adjacent to a space with a different temperature, heat transfer through the separating physical section must be considered. The heat transfer rate is given by

[redacted] (33)

where

q = heat transfer rate, Btu/h
U = coefficient of overall heat transfer between adjacent and conditioned space, Btu/h·ft²·°F
A = area of separating section concerned, ft²
t_b = average air temperature in adjacent space, °F
t_i = air temperature in conditioned space, °F

U-values can be obtained from Chapter [redacted] Temperature [redacted]

Floors

For floors directly in contact with the ground or over an underground basement that [redacted]

Table 16 Wall Conduction Time Series (CTS)

	Curtain Walls			Stud Walls				EIFS			Brick Walls									
Wall Number =	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Hour	**Conduction Time Factors, %**																			
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Total Percentage	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Layer ID from outdoors to indoors (see Table 18)	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Wall Number Descriptions

1. Spandrel glass, R-10 insulation board, gyp board
2. Metal wall panel, R-10 insulation board, gyp board
3. 1 in. stone, R-10 insulation board, gyp board
4. Metal wall panel, sheathing, R-11 batt insulation, gyp board
5. 1 in. stone, sheathing, R-11 batt insulation, gyp board
6. Wood siding, sheathing, R-11 batt insulation, 1/2 in. wood
7. 1 in. stucco, sheathing, R-11 batt insulation, gyp board
8. EIFS finish, R-5 insulation board, sheathing, gyp board
9. EIFS finish, R-5 insulation board, sheathing, R-11 batt insulation, gyp board
10. EIFS finish, R-5 insulation board, sheathing, 8 in. LW CMU, gyp board
11. Brick, R-5 insulation board, sheathing, gyp board
12. Brick, sheathing, R-11 batt insulation, gyp board
13. Brick, R-5 insulation board, sheathing, R-11 batt insulation, gyp board
14. Brick, R-5 insulation board, 8 in. LW CMU
15. Brick, 8 in. LW CMU, R-11 batt insulation, gyp board
16. Brick, R-5 insulation board, 8 in. HW CMU, gyp board
17. Brick, R-5 insulation board, brick
18. Brick, R-5 insulation board, 8 in. LW concrete, gyp board
19. Brick, R-5 insulation board, 12 in. HW concrete, gyp board
20. Brick, 8 in. HW concrete, R-11 batt insulation, gyp board

CALCULATING COOLING LOAD

The **instantaneous cooling load** is the [redacted] Computation of cooling load is [redacted]

[redacted] RTS further [redacted]

Thus, the cooling load for each load component (lights, people, walls, roofs, windows, appliances, etc.) for a particular hour is [redacted]

Table 16 Wall Conduction Time Series (CTS) (*Concluded*)

	Concrete Block Wall						Precast and Cast-in-Place Concrete Walls								
Wall Number =	**21**	**22**	**23**	**24**	**25**	**26**	**27**	**28**	**29**	**30**	**31**	**32**	**33**	**34**	**35**
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Hour	**Conduction Time Factors, %**														
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Total Percentage	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Layer ID from outdoors to indoors (see Table 18)	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Wall Number Descriptions

21. 8 in. LW CMU, R-11 batt insulation, gyp board
22. 8 in. LW CMU with fill insulation, R-11 batt insulation, gyp board
23. 1 in. stucco, 8 in. HW CMU, R-11 batt insulation, gyp board
24. 8 in. LW CMU with fill insulation
25. 8 in. LW CMU with fill insulation, gyp board
26. 12 in. LW CMU with fill insulation, gyp board
27. 4 in. LW concrete, R-5 board insulation, gyp board
28. 4 in. LW concrete, R-11 batt insulation, gyp board
29. 4 in. LW concrete, R-10 board insulation, 4 in. LW concrete
30. EIFS finish, R-5 insulation board, 8 in. LW concrete, gyp board
31. 8 in. LW concrete, R-11 batt insulation, gyp board
32. EIFS finish, R-10 insulation board, 8 in. HW concrete, gyp board
33. 8 in. HW concrete, R-11 batt insulation, gyp board
34. 12 in. HW concrete, R-19 batt insulation, gyp board
35. 12 in. HW concrete

[redacted]

RTS converts the [redacted]

[redacted]

The radiant cooling load for the current hour, which is calculated using RTS and Equation (34), is [redacted]

Radiant time factors are generated by a heat-balance-based procedure. A separate series of radiant time factors is [redacted]

Table 17 Roof Conduction Time Series (CTS)

	Sloped Frame Roofs						Wood Deck		Metal Deck Roofs					Concrete Roofs					
Roof Number	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**
[redacted]	[redacted]						[redacted]	[redacted]											
	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Hour	**Conduction Time Factors, %**																		
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Total Percentage	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
Layer ID from outdoors to indoors (see Table 18)	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Roof Number Descriptions

1. Metal roof, R-19 batt insulation, gyp board
2. Metal roof, R-19 batt insulation, suspended acoustical ceiling
3. Metal roof, R-19 batt insulation
4. Asphalt shingles, wood sheathing, R-19 batt insulation, gyp board
5. Slate or tile, wood sheathing, R-19 batt insulation, gyp board
6. Wood shingles, wood sheathing, R-19 batt insulation, gyp board
7. Membrane, sheathing, R-10 insulation board, wood deck
8. Membrane, sheathing, R-10 insulation board, wood deck, suspended acoustical ceiling
9. Membrane, sheathing, R-10 insulation board, metal deck
10. Membrane, sheathing, R-10 insulation board, metal deck, suspended acoustical ceiling
11. Membrane, sheathing, R-15 insulation board, metal deck
12. Membrane, sheathing, R-10 plus R-15 insulation boards, metal deck
13. 2 in. concrete roof ballast, membrane, sheathing, R-15 insulation board, metal deck
14. Membrane, sheathing, R-15 insulation board, 4 in. LW concrete
15. Membrane, sheathing, R-15 insulation board, 6 in. LW concrete
16. Membrane, sheathing, R-15 insulation board, 8 in. LW concrete
17. Membrane, sheathing, R-15 insulation board, 6 in. HW concrete
18. Membrane, sheathing, R-15 insulation board, 8 in. HW concrete
19. Membrane, 6-in HW concrete, R-19 batt insulation, suspended acoustical ceiling

[redacted]

One goal in developing RTS was to provide a simplified method based directly on the HB method; thus, it was deemed desirable to [redacted] A heat balance computer program was developed to do this: Hbfort, which is included as part of [redacted] (Pedersen et al. 1998). The RTS procedure is described by Spitler et al. (1997). The procedure for generating RTS coefficients may be thought of as [redacted]

[redacted]

Two different radiant time series are used: **solar**, for direct transmitted solar heat gain [redacted] and **nonsolar**, for all other types of heat gains [redacted] Nonsolar RTS apply to [redacted]

Table 18 Thermal Properties and Code Numbers of Layers Used in Wall and Roof Descriptions for Tables 16 and 17

Layer ID	Description	Thickness, in.	Conductivity, Btu·in/h·ft^2·°F	Density, lb/ft^3	Specific Heat, Btu/lb·°F	Resistance, ft^2·°F·h/Btu	*R*	Mass, lb/ft^2	Thermal Capacity, Btu/ft^2·°F	Notes
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Notes: The following notes give sources for the data in this table.

1. Chapter 26, Table 1 for 7.5 mph wind
2. Chapter 26, Table 1 for still air, horizontal heat flow
3. Chapter 26, Table 1 for still air, downward heat flow
4. Chapter 26, Table 3 for 1.5 in. space, 90°F, horizontal heat flow, 0.82 emittance
5. Chapter 26, Table 3 for 3.5 in. space, 90°F, downward heat flow, 0.82 emittance
6. EIFS finish layers approximated by Chapter 26, Table 4 for 3/8 in. cement plaster, sand aggregate
7. Chapter 33, Table 3 for steel (mild)
8. Chapter 26, Table 4 for architectural glass
9. Chapter 26, Table 4 for marble and granite
10. Chapter 26, Table 4, density assumed same as Southern pine
11. Chapter 26, Table 4 for mineral fiberboard, wet molded, acoustical tile
12. Chapter 26, Table 4 for carpet and rubber pad, density assumed same as fiberboard
13. Chapter 26, Table 4, density assumed same as stone
14. Chapter 26, Table 4 for nail-base sheathing
15. Chapter 26, Table 4 for Southern pine
16. Chapter 26, Table 4 for expanded polystyrene
17. Chapter 26, Table 4 for glass fiber batt, specific heat per glass fiber board
18. Chapter 26, Table 4 for clay fired brick
19. Chapter 26, Table 4, 16 lb block, 8 × 16 in. face
20. Chapter 26, Table 4, 19 lb block, 8 × 16 in. face
21. Chapter 26, Table 4, 32 lb block, 8 × 16 in. face
22. Chapter 26, Table 4, 33 lb normal weight block, 8 × 16 in. face
23. Chapter 26, Table 4, 50 lb normal weight block, 8 × 16 in. face
24. Chapter 26, Table 4, 16 lb block, vermiculite fill
25. Chapter 26, Table 4, 19 lb block, 8 × 16 in. face, vermiculite fill
26. Chapter 26, Table 4, 32 lb block, 8 × 16 in. face, vermiculite fill
27. Chapter 26, Table 4, 33 lb normal weight block, 8 × 16 in. face, vermiculite fill
28. Chapter 26, Table 4 for 40 lb/ft^3 LW concrete

[redacted] Hittle (1999).

Representative solar and nonsolar RTS data for light, medium, and heavyweight constructions are provided in Tables 19 and 20. Those were calculated using the [redacted] (Pedersen et al. 1998) with zone characteristics listed in Table 21. Customized RTS values may be calculated using [redacted]

ASHRAE research project [redacted] (Rees et al. 2000; Spitler et al. 1998). In general, total cooling loads calculated using RTS [redacted] The project examined more than 5000 test cases of varying zone parameters. The domi-[redacted]

ASHRAE research project [redacted] The results [redacted] (Chantrasrisalai et al. 2003; Eldridge et al. 2003; Iu et al. 2003). HB calculations [redacted]

HEATING LOAD CALCULATIONS

Techniques for estimating design heating load for commercial, institutional, and industrial applications are [redacted]

Heat losses (negative heat gains) are thus [redacted]

Table 19 Representative Nonsolar RTS Values for Light to Heavy Construction

	Light						Medium						Heavy						Interior Zones					
	With Carpet			No Carpet			With Carpet			No Carpet			With Carpet			No Carpet			Light		Medium		Heavy	
% Glass	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	With Carpet	No Carpet	With Carpet	No Carpet	With Carpet	No Carpet
Hour	Radiant Time Factor, %																							
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100

Table 20 Representative Solar RTS Values for Light to Heavy Construction

	Light						Medium						Heavy					
	With Carpet			No Carpet			With Carpet			No Carpet			With Carpet			No Carpet		
% Glass	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%
Hour	Radiant Time Factor, %																	
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100

Table 21 RTS Representative Zone Construction for Tables 19 and 20

Construction Class	Exterior Wall	Roof/Ceiling	Partitions	Floor	Furnishings
Light	Steel siding, 2 in. insulation, air space, 3/4 in. gyp	4 in. LW concrete, ceiling air space, acoustic tile	3/4 in. gyp, air space, 3/4 in. gyp	Acoustic tile, ceiling air space, 4 in. LW concrete	1 in. wood @ 50% of floor area
Medium	4 in. face brick, 2 in. insulation, air space, 3/4 in. gyp	4 in. HW concrete, ceiling air space, acoustic tile	3/4 in. gyp, air space, 3/4 in. gyp	Acoustic tile, ceiling air space, 4 in. HW concrete	1 in. wood @ 50% of floor area
Heavy	4 in. face brick, 8 in. HW concrete air space, 2 in. insulation, 3/4 in. gyp	8 in. HW concrete, ceiling air space, acoustic tile	3/4 in. gyp, 8 in. HW concrete block, 3/4 in. gyp	Acoustic tile, ceiling air space, 8 in. HW concrete	1 in. wood @ 50% of floor area

[redacted]

Typical commercial and retail spaces have nighttime unoccupied periods at a setback temperature where little to no ventilation is required, building lights and equipment are off, and heat loss is primarily through conduction and infiltration. Before being occupied, buildings are warmed to the occupied temperature (see the following discussion). During occupied time, building lights, equipment, and people cooling loads can [redacted] Ventilation heat load may be [redacted]

HEAT LOSS CALCULATIONS

The general procedure for calculation of design heat losses of a structure is as follows:

[redacted]

Outdoor Design Conditions

The ideal heating system provides enough heat to match the structure's heat loss. However, weather conditions vary considerably from year to year, and heating systems designed for the worst weather conditions on record would have a great excess of capacity most of the time. A system's failure to maintain design conditions during brief periods of severe weather usually is not critical. However, close regulation of indoor temperature may be critical for some occupancies or industrial processes. Design temperature data and discussion of their application are given in Chapter 14. Generally, the 99% temperature values given in the tabulated weather data are used. However, caution is needed, and local conditions should always be investigated. In some locations, outdoor temperatures are commonly much lower and wind velocities higher than those given in the tabulated weather data.

Indoor Design Conditions

The main purpose of the heating system is to maintain indoor conditions that make most of the occupants comfortable. It should be kept in mind, however, that the purpose of heating load calculations is to obtain data for sizing the heating system components. In many cases, the system will rarely be called upon to operate at the design conditions. Therefore, the use and occupancy of the space are general considerations from the design temperature point of view. Later, when the building's energy requirements are computed, the actual conditions in the space and outdoor environment, including internal heat gains, must be considered.

The indoor design temperature should be selected at the lower end of the acceptable temperature range, so that the heating equipment will not be oversized. Even properly sized equipment operates under partial load, at reduced efficiency, most of the time; therefore, any oversizing aggravates this condition and lowers overall system efficiency. A maximum design dry-bulb temperature of [redacted] is recommended for most occupancies. The indoor design value of relative humidity should be compatible with a healthful environment and the thermal and moisture integrity of the building envelope. A minimum relative humidity of [redacted] is recommended for most situations.

Calculation of Transmission Heat Losses

Exterior Surface Above Grade. All above-grade surfaces exposed to outdoor conditions (walls, doors, ceilings, fenestration, and raised floors) are treated identically, as follows:

[redacted] (35)

[redacted] (36)

where HF is the heating load factor i[redacted]

Below-Grade Surfaces. An approximate method for estimating below-grade heat loss [based on the work of Latta and Boileau (1969)] assumes that the heat flow paths shown in Figure 12 can be used to find the steady-state heat loss to the ground surface, as follows:

[redacted] (37)

[redacted]

The effect of soil heat capacity means that [redacted] which varies with geographic location and surface cover. The minimum ground surface temperature, suitable for heat loss estimates, is therefore

[redacted] $\bar{t}_{gr}$ [redacted] (38)

[redacted]

$\bar{t}_{gr}$ [redacted]

Figure 14 shows depth parameters used in determining U_{avg}. For walls, the region defined [redacted]

The value of soil thermal conductivity [redacted]

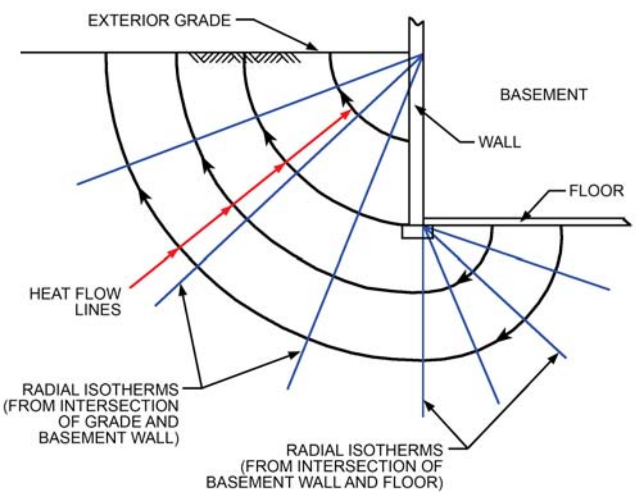

Fig. 12 Heat Flow from Below-Grade Surface

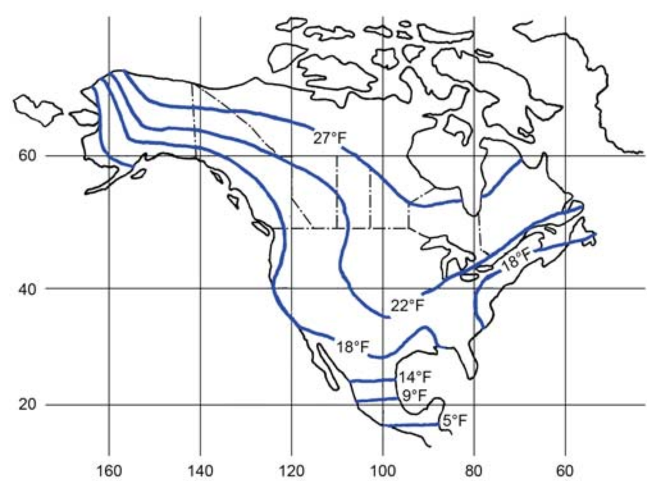

Fig. 13 Ground Temperature Amplitude

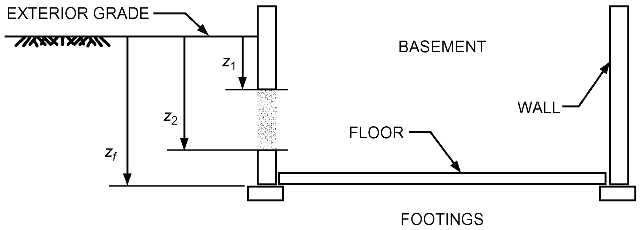

Fig. 14 Below-Grade Parameters

Table 22 Average U-Factor for Basement Walls with Uniform Insulation

Depth, ft	[redacted]			
	Uninsulated	R-5	R-10	R-15
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Soil conductivity = 0.8 Btu/h·ft·°F; insulation is over entire depth. For other soil conductivities and partial insulation, use Equation (39).

Table 23 Average U-Factor for Basement Floors

z_f (Depth of Floor Below Grade), ft	[redacted]			
	w_b (Shortest Width of Basement), ft			
	20	24	28	32
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Soil conductivity is 0.8 Btu/h·ft·°F; floor is uninsulated. For other soil conductivities and insulation, use Equation (39).

[redacted] are shown in Table 22.

The average below-grade floor U-factor (where the entire basement floor is uninsulated or has uniform insulation) is given by

[redacted] (40)

[redacted]

Representative values of $U_{avg,bf}$ for uninsulated basement floors are shown in Table 23.

At-Grade Surfaces. Concrete slab floors may be (1) unheated, relying for warmth on heat delivered above floor level by the heating system, or (2) heated, containing heated pipes or ducts that constitute a radiant slab or portion of it for complete or partial heating of the house.

The simplified approach that treats heat loss as proportional to slab perimeter allows slab heat loss to be estimated for both unheated and heated slab floors:

Table 24 Heat Loss Coefficient F_p of Slab Floor Construction

Construction	Insulation	F_p, Btu/h·ft·°F
[redacted]	[redacted]	[redacted]

*Weighted average temperature of heating duct was assumed at 110°F during heating season (outdoor air temperature less than 65°F).

[redacted] (41)

[redacted] (42)

[redacted]

Surfaces Adjacent to Buffer Space. Heat loss to adjacent unconditioned or semiconditioned spaces can be calculated using a heating factor based on the partition temperature difference:

[redacted] (43)

Infiltration

Infiltration of outdoor air through openings into a structure is caused by thermal forces, wind pressure, and negative pressure (planned or unplanned) with respect to the outdoors created by mechanical systems. Typically, in building design, if the mechanical systems are designed to maintain positive building pressure, infiltration need not be considered except in ancillary spaces such as entryways and loading areas.

Infiltration is treated as a room load and has both sensible and latent components. During winter, this means heat and humidity loss because cold, dry air must be heated to design temperature and moisture must be added to increase the humidity to design condition. Typically, during winter, controlling indoor humidity is not a factor and infiltration is reduced to a simple sensible component. Under cooling conditions, both sensible and latent components are added to the space load to be treated by the air conditioning system.

Procedures for estimating the infiltration rate are discussed in Chapter [redacted] The infiltration rate is reduced to a volumetric flow rate at a known dry bulb/wet bulb condition. Along with indoor air condition, the following equations define the infiltration sensible and latent loads.

[redacted] (44)

[redacted]

Assuming standard air conditions [redacted] may be written as

[redacted] (45)

The infiltrating air also introduces a latent heating load given by

[redacted] (46)

[redacted]

[redacted]

For standard air and nominal indoor comfort conditions, the latent load may be expressed as

[redacted] (47)

The coefficients [redacted] in Equation (45) and [redacted] in Equation (47) are given for standard conditions. They depend on [redacted]

HEATING SAFETY FACTORS AND LOAD ALLOWANCES

Before mechanical cooling became common in the second half of the 1900s, and when energy was less expensive, buildings included much less insulation; large, operable windows; and generally more infiltration-prone assemblies than the energy-efficient and much tighter buildings typical of today. Allowances of 10 to 20% of the net calculated heating load for piping losses to unheated spaces, and 10 to 20% more for a warm-up load, were common practice, along with other occasional safety factors reflecting the experience and/or concern of the individual designer. Such measures are less conservatively applied today with newer construction. A combined warm-up/safety allowance of [redacted] is fairly common but [redacted] Engineering judgment must be applied for the particular project. Armstrong et al. (1992a, 1992b) provide a design method to deal with warm-up and cooldown load.

OTHER HEATING CONSIDERATIONS

Calculation of design heating load estimates has essentially become a subset of the more involved and complex estimation of cooling loads for such spaces. Chapter 19 discusses using the heating load estimate to predict or analyze energy consumption over time. Special provisions to deal with particular applications are covered in the 2011 *ASHRAE Handbook—HVAC Applications* and the 2012 *ASHRAE Handbook—HVAC Systems and Equipment*.

The 1989 *ASHRAE Handbook—Fundamentals* was the last edition to contain a chapter dedicated only to heating load. Its contents were incorporated into this volume's Chapter 17, which describes steady-state conduction and convection heat transfer and provides, among other data, information on losses through basement floors and slabs.

SYSTEM HEATING AND COOLING LOAD EFFECTS

The heat balance (HB) or radiant time series (RTS) methods are used to [redacted] Principal factors to consider in determining the plant size are [redacted]

Detailed analysis of system components and methods calculating their contribution to equipment sizing are beyond the scope of this chapter, which is general in nature. Table 25 lists the most frequently used calculations in other chapters and volumes.

Table 25 Common Sizing Calculations in Other Chapters

Subject	Volume/Chapter	Equation(s)
[redacted]	[redacted]	
[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]

ZONING

The organization of building rooms as defined for load calculations into zones and air-handling units has no effect on room cooling loads. However, specific grouping and ungrouping of rooms into zones may cause peak system loads to occur at different times during the day or year and may significantly affected heat removal equipment sizes.

For example, if each room is cooled by a separate heat removal system, the total capacity of the heat transport systems equals the sum of peak room loads. Conditioning all rooms by a single heat transport system (e.g., a variable-volume air handler) requires less capacity (equal to the simultaneous peak of the combined rooms load, which includes some rooms at off-peak loads). This may significantly reduce equipment capacity, depending on the configuration of the building.

VENTILATION

Consult ASHRAE *Standard* [redacted] and building codes to determine the required quantity of ventilation air for an application, and the various methods of achieving acceptable indoor air quality. The following discussion is confined to the effect of mechanical ventilation on sizing heat removal equipment. Where natural ventilation is used, through operable windows or other means, [redacted] Where ventilation air is conditioned and supplied through the mechanical system, its sensible and latent loads [redacted] If the mechanical ventilation rate sufficiently exceeds exhaust airflows, [redacted] Chapter 16 includes more information on ventilating commercial buildings.

AIR HEAT TRANSPORT SYSTEMS

Heat transport equipment is usually selected to provide adequate heating or cooling for the peak load condition. However, selection must also consider maintaining desired indoor conditions during all occupied hours, which requires [redacted] Automatic control systems normally vary the heating and cooling system capacity during these off-peak hours of operation.

On/Off Control Systems

On/off control systems, common in residential and light commercial applications, cycle equipment on and off to match room load. They are adaptable to heating or cooling because they can cycle both heating and cooling equipment. In their purest form, [redacted]

[redacted] Systems

[redacted]

[redacted]

[redacted] Systems

[redacted]

[redacted] Systems

[redacted]

1. Determine the ratio of cold-deck flow to hot-deck flow from

[redacted]

2. From Equation (10), the hot-deck contribution to room load during off-peak cooling is

[redacted]

Heat Gain from Fans

Fans that circulate air through HVAC systems add energy to the system through the following processes:

[redacted]

The power required to provide airflow and static pressure can be determined from the first law of thermodynamics with the following equation:

[redacted]

The power necessary at the input to the fan motor must account for fan motor inefficiencies and drive losses. Fan motor efficiencies generally vary from 80 to 95%, and drive losses for a belt drive are 3% of the fan power. This may be determined from

$$P_M = (1 + \mathrm{DL})\, P_F / E_M E_D$$

where

P_M = power required at input to motor, hp
E_D = belt drive efficiency, dimensionless
E_M = fan motor efficiency, dimensionless
P_F = power required at fan shaft, hp
DL = drive loss, dimensionless

Almost all the energy required to generate airflow and static pressure is ultimately dissipated as heat within the building and HVAC system; a small portion is discharged with any exhaust air. Generally, it is assumed that all the heat is released at the fan rather than dispersed to the remainder of the system. The portion of fan heat released to the airstream depends on the location of the fan motor and drive: if they are within the airstream, all the energy input to the fan motor is released to the airstream. If the fan motor and drive are outdoor the airstream, the energy is split between the airstream and the room housing the motor and drive. Therefore, the following equations may be used to calculate heat generated by fans and motors:

If motor and drive are **outside** the airstream,

$$q_{fs} = 2545 P_F$$

$$q_{fr} = 2545(P_M - P_F)$$

If motor and drive are **inside** the airstream,

$$q_{fs} = 2545 P_M$$

$$q_{fr} = 0.0$$

where

P_F = power required at fan shaft, hp
P_M = power required at input to motor, hp
q_{fs} = heat release to airstream, Btu/h
q_{fr} = heat release to room housing motor and drive, Btu/h
2545 = conversion factor, Btu/h·hp

Supply airstream temperature rise may be determined from psychrometric formulas or Equation (9).

Variable- or adjustable-frequency drives (VFDs or AFDs) often drive fan motors in VAV air-handling units. These devices release heat to the surrounding space. Refer to manufacturers' data for heat released or efficiencies. The disposition of heat released is determined by the drive's location: in the conditioned space, in the return air path, or in a nonconditioned equipment room. These drives, and other electronic equipment such as building control, data processing, and communications devices, are temperature sensitive, so the rooms in which they are housed require cooling, frequently year-round.

Duct Surface Heat Transfer

Heat transfer across the duct surface is one mechanism for energy transfer to or from air inside a duct. It involves conduction through the duct wall and insulation, convection at inner and outer surfaces, and radiation between the duct and its surroundings. Chapter 4 presents a rigorous analysis of duct heat loss and gain, and Chapter 23 addresses application of analysis to insulated duct systems.

The effect of duct heat loss or gain [redacted]

[redacted]

Duct Leakage

Air leakage from supply ducts can [redacted]

Determining leakage from a duct system is complex because of the variables in paths, fabrication, and installation methods. Refer to Chapter 21 and publications from the Sheet Metal and Air Conditioning Contractors' National Association (SMACNA) for methods of determining leakage. In general, good-quality ducts and post-installation duct sealing provide highly cost-effective energy savings, with improved thermal comfort and delivery of ventilation air.

Ceiling Return Air Plenum Temperatures

The space above a ceiling, when used as a return air path, is a ceiling return air plenum, or simply a **return plenum**. Unlike a traditional ducted return, the plenum may have multiple heat sources in the air path. These heat sources may be radiant and convective loads from lighting and transformers; conduction loads from adjacent walls, roofs, or glazing; or duct and piping systems within the plenum.

As heat from these sources is picked up by the unducted return air, the temperature differential between the ceiling cavity and conditioned space is small. Most return plenum temperatures do not rise more [redacted]

Where the ceiling space is used as a return air plenum, energy balance requires that heat picked up from the lights into the return air [redacted] (Rock and Wolfe 1997).

Figure 15 shows a schematic of a typical return air plenum. The following equations, using the heat flow directions shown in Figure 15, represent the heat balance of a return air plenum design for a typical interior room in a multifloor building:

$$q_1 = U_c A_c (t_p - t_r) \tag{48}$$

$$q_2 = U_f A_f (t_p - t_{fa}) \tag{49}$$

$$q_3 = 1.1 Q (t_p - t_r) \tag{50}$$

$$q_{lp} - q_2 - q_1 - q_3 = 0 \tag{51}$$

$$Q = \frac{q_r + q_1}{1.1(t_r - t_s)} \tag{52}$$

[redacted]

Fig. 15 Schematic Diagram of Typical Return Air Plenum

By substituting Equations (48), (49), (50), and (52) into heat balance Equation (51), [redacted] The results, although rigorous and best solved by computer, are important in determining the cooling load, which affects [redacted]

Equations (48) to (52) are simplified to illustrate the heat balance relationship. Heat gain into a return air plenum is not [redacted] The supply air quantity calculated by Equation (52) is only for the conditioned space under consideration, and is assumed to equal the return air quantity.

The amount of airflow through a return plenum above a conditioned space may [redacted] Where [redacted] exist, Equations (48) to (52) must be modified appropriately. Finally, although the building's thermal storage has some effect, the amount of heat entering the return air is [redacted]

Ceiling Plenums with Ducted Returns

Compared to those in unducted plenum returns, temperatures in ceiling plenums that have well-sealed return or exhaust air ducts float considerably. In cooling mode, [redacted]

Underfloor Air Distribution Systems

Room cooling loads determined by methods in this chapter cannot model two distinguishing aspects of the thermal performance of underfloor air distribution (UFAD) systems under cooling operation:

- Room air stratification: UFAD systems supply cool air at the floor and extract warmer air at the ceiling, thus creating vertical thermal stratification. Cooling load models assume a well-mixed uniform space temperature.
- Underfloor air supply plenums: cool supply air flowing through the underfloor plenum is exposed to heat gain from both the concrete slab (conducted from the warm return air on the adjacent floor below in a multistory building) and the raised floor panels (conducted from the warmer room above).

Extensive simulation and experimental research led to the development of a whole-building energy simulation program capable of modeling energy performance and load calculations for UFAD systems (Bauman et al. 2007; Webster et al. 2008). Previously, it was thought that cooling loads for UFAD and overhead (OH) mixing systems were nearly identical. However, energy modeling studies show that the UFAD cooling load is generally higher than that calculated in the same building for a well-mixed system (Schiavon et al. 2010a). The difference is primarily caused by the thermal storage effect of the lighter-weight raised-floor panels compared to the greater mass of a structural floor slab. Schiavon et al. (2010b) showed that the presence of the raised floor reduces the slab's ability to store heat, thereby producing higher peak cooling loads for a raised-floor system than for one without a raised floor. A second contributing factor is that the raised-floor surface above the underfloor plenum tends to be cooler (except when illuminated by the sun) than most other room surfaces, producing a room surface temperature distribution resembling a chilled radiant floor system, which has a different peak cooling load than an all-air system (Feng et al. 2012). The precise magnitude of difference in design cooling loads between OH and UFAD systems is still under investigation, but mainly depends on zone orientation and floor level, and possibly the effects of furniture. Methods for determining UFAD cooling loads will be updated as additional research results become available. For more information about simplified approaches to UFAD cooling load calculations, see Bauman et al. (2010), Schiavon et al. (2010c), and the updated ASHRAE *Underfloor Air Distribution (UFAD) Design Guide* (Bauman and Daly 2013).

Plenums in Load Calculations

Currently, most designers include ceiling and floor plenums within neighboring occupied spaces when thermally zoning a building. However, temperatures in these plenums, and the way that they behave, are significantly different from those of occupied spaces. Thus, they should be defined as a separate thermal zone. However, most hand and computer-based load calculation routines currently do not allow floating air temperatures or humidities; assuming a constant air temperature in plenums, attics, and other unconditioned spaces is a poor, but often necessary, assumption. The heat balance method does allow floating space conditions, and when fully implemented in design load software, should allow more accurate modeling of plenums and other complex spaces.

CENTRAL PLANT

Piping

Losses must be considered for piping systems that transport heat. For water or hydronic piping systems, heat is [redacted] see Chapter [redacted] for ways to [redacted] However, distribution of this transferred heat depends [redacted]

Consider a heating hot-water pipe. If the pipe serves a room heater and is routed through the heated space, any heat loss from the pipe adds heat to the room. Heat transfer to the heated space and heat loss from the piping system is null. If the piping is exposed to ambient conditions en route to the heater, the loss must [redacted]

In summary, the designer must [redacted]

Pumps

Calculating heat gain from pumps is addressed in the section on Electric Motors. For pumps serving hydronic systems, disposition of heat from the pumps depends on the service. For chilled-water systems, energy applied to the fluid to generate flow and pressure becomes a chiller load. For condenser water pumps, pumping energy must be rejected through the cooling tower. The magnitude of pumping energy relative to cooling load is generally small.

EXAMPLE COOLING AND HEATING LOAD CALCULATIONS

To illustrate the cooling and heating load calculation procedures discussed in this chapter, an example problem has been developed

Table 26 Summary of RTS Load Calculation Procedures

Equation	Equation No. in Chapter	Equation	Equation No. in Chapter
[redacted]	[redacted]	[redacted]	[redacted]

Table 26 Summary of RTS Load Calculation Procedures (*Concluded*)

Equation	Equation No. in Chapter	Equation	Equation No. in Chapter
[redacted]	[redacted]	[redacted]	[redacted]

based on the ASHRAE headquarters building located in Atlanta, Georgia. This example is a two-story office building of approximately 35,000 ft^2, including a variety of common office functions and occupancies. In addition to demonstrating calculation procedures, a hypothetical design/construction process is discussed to illustrate (1) application of load calculations and (2) the need to develop reasonable assumptions when specific data is not yet available, as often occurs in everyday design processes.

Table 26 provides a summary of RTS load calculation procedures.

SINGLE-ROOM EXAMPLE

Calculate the peak heating and cooling loads for the office room shown in Figure 16, for Atlanta, Georgia. The room is on the second floor of a two-story building and has two vertical exterior exposures, with a flat roof above.

Room Characteristics

Area: 130 ft^2.

Floor: Carpeted 5 in. concrete slab on metal deck above a conditioned space.

Roof: Flat metal deck topped with rigid closed-cell polyisocyanurate foam core insulation (R = 30), and light-colored membrane roofing. Space above 9 ft suspended acoustical tile ceiling is used as a return air plenum. Assume 30% of the cooling load from the roof is directly absorbed in the return airstream without becoming room load. Use roof U = 0.032 Btu/h·ft^2·°F.

Spandrel wall: Spandrel bronze-tinted glass, opaque, backed with air space, rigid mineral fiber insulation (R = 5.0), mineral fiber batt insulation (R = 13), and 5/8 in. gypsum wall board. Use spandrel wall U = 0.077 Btu/h·ft^2·°F.

Brick wall: Light-brown-colored face brick (4 in.), lightweight concrete block (6 in.), rigid continuous insulation (R = 5), mineral fiber batt insulation (R = 13), and gypsum wall board (5/8 in.). Use brick wall U = 0.08 Btu/h·ft^2·°F.

Windows: Double glazed, 1/4 in. bronze-tinted outdoor pane, 1/2 in. air space and 1/4 in. clear indoor pane with light-colored interior miniblinds. Window normal solar heat gain coefficient (SHGC) = 0.49. Windows are nonoperable and mounted in aluminum frames with thermal breaks having overall combined U = 0.56 Btu/h·ft^2·°F (based on Type 5d from Tables 4 and 10 of Chapter 15). Indoor attenuation coefficients (IACs) for indoor miniblinds are based on light venetian blinds (assumed louver reflectance = 0.8 and louvers positioned at 45° angle) with heat-absorbing double glazing (Type 5d from Table 13B of Chapter 15), IAC(0) = 0.74, IAC(60) = 0.65, IAD(diff) = 0.79, and radiant fraction = 0.54. Each window is 6.25 ft1.91 m wide by 6.4 ft tall for an area per window = 40 ft^2.

South exposure:	Orientation	= 30° east of true south
	Window area	= 40 ft^2
	Spandrel wall area	= 60 ft^2
	Brick wall area	= 60 ft^2
West exposure:	Orientation	= 60° west of south
	Window area	= 40 ft^2
	Spandrel wall area	= 60 ft^2
	Brick wall area	= 40 ft^2

Occupancy: 1 person from 8:00 AM to 5:00 PM.

Lighting: One 4-lamp pendant fluorescent 8 ft type. The fixture has four 32 W T-8 lamps plus electronic ballasts (special allowance factor 0.85 per manufacturer's data), for a total of 110 W for the room. Operation is from 7:00 AM to 7:00 PM. Assume 0% of the cooling load from lighting is directly absorbed in the return airstream without becoming room load, per Table 3.

Equipment: One computer and a personal printer are used, for which an allowance of 1 W/ft^2 is to be accommodated by the cooling system, for a total of 130 W for the room. Operation is from 8:00 AM to 5:00 PM.

Infiltration: For purposes of this example, assume the building is maintained under positive pressure during peak cooling conditions and therefore has no infiltration. Assume that infiltration during peak heating conditions is equivalent to one air change per hour.

Weather data: Per Chapter 14, for Atlanta, Georgia, latitude = 33.64, longitude = 84.43, elevation = 1027 ft above sea level, 99.6% heating design dry-bulb temperature = 21.5°F. For cooling load calculations, use 5% dry-bulb/coincident wet-bulb monthly design day profile calculated per Chapter 14. See Table 27 for temperature profiles used in these examples.

Indoor design conditions: 72°F for heating; 75°F with 50% rh for cooling.

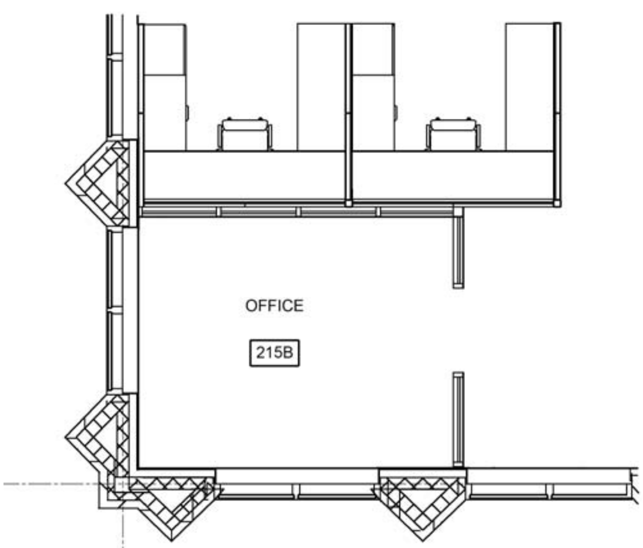

Fig. 16 Single-Room Example Office

Cooling Loads Using RTS Method

Traditionally, simplified cooling load calculation methods have estimated the total cooling load at a particular design condition by independently calculating and then summing the load from each component (walls, windows, people, lights, etc). Although the actual heat transfer processes for each component do affect each other, this simplification is appropriate for design load calculations and useful to the designer in understanding the relative contribution of each component to the total cooling load.

Cooling loads are calculated with the RTS method on a component basis similar to previous methods. The following example parts illustrate cooling load calculations for individual components of this single room for a particular hour and month. Equations used are summarized in Table 26.

Part 1. Internal cooling load using radiant time series. Calculate the cooling load from lighting at 3:00 PM for the previously described office.

Solution: First calculate the 24 h heat gain profile for lighting, then split those heat gains into radiant and convective portions, apply the appropriate RTS to the radiant portion, and sum the convective and radiant cooling load components to determine total cooling load at the designated time. Using Equation (1), the lighting heat gain profile, based on the occupancy schedule indicated is

[redacted]

The convective portion is simply the lighting heat gain for the hour being calculated times the convective fraction for non-in-ceiling fluorescent luminaire (pendant), from Table 3:

[redacted]

The radiant portion of the cooling load is calculated using lighting heat gains for the current hour and past 23 h, the radiant fraction from Table 3 [redacted] and radiant time series from Table 19, in accordance with Equation (34). From Table 19, select the RTS for medium-weight construction, assuming 50% glass and carpeted floors as representative of the described construction. Thus, the radiant cooling load for lighting is

[redacted]

Table 27 Monthly/Hourly Design Temperatures (5% Conditions) for Atlanta, GA, °F

	January		February		March		April		May		June		July		August		September		October		November		December	
Hour	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb
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Table 28 Cooling Load Component: Lighting, Btu/h

Hour	Usage Profile, %	Heat Gain, Btu/h Total	Convective 43%	Radiant 57%	Nonsolar RTS Zone Type 8, %	Radiant Cooling Load	Total Sensible Cooling Load	% Lighting to Return 26%	Room Sensible Cooling Load
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The total lighting cooling load at the designated hour is thus

$$Q_{light} = Q_{c,15} + Q_{r,15} = 161.3 + 190.3 = 351.6 \text{ Btu/h}$$

See Table 28 for the office's lighting usage, heat gain, and cooling load profiles.

Part 2. Wall cooling load using sol-air temperature, conduction time series and radiant time series. Calculate the cooling load contribution from the spandrel wall section facing 60° west of south at 3:00 PM local standard time in July for the previously described office.

Solution: Determine the wall cooling load by calculating [redacted]

This procedure is used to calculate the sol-air temperatures for each hour on each surface. Because of the tedious solar angle and intensity calculations, using a simple computer spreadsheet or other computer software can reduce the effort involved. A spreadsheet was used to calculate a 24 h sol-air temperature profile for the data of this example. See Table 29A for the solar angle and intensity calculations and Table 29B for the sol-air temperatures for this wall surface and orientation.

Conductive heat gain is calculated using Equations (31) and (32). First, calculate the 24 h heat input profile using Equation (31) and the sol-air temperatures for a southwest-facing wall with dark exterior color:

[redacted]

[redacted]

Next, calculate wall heat gain using conduction time series. The preceding heat input profile is used with conduction time series to calculate the wall heat gain. From Table 16, the most similar wall construction is wall number 1. This is a spandrel glass wall that has similar mass and thermal capacity. Using Equation (32), the conduction time factors for wall 1 can be used in conjunction with the 24 h heat input profile to determine the wall heat gain at 3:00 PM LST:

[redacted]

Because of the tedious calculations involved, a spreadsheet is used to calculate the remainder of a 24 h heat gain profile indicated in Table 29B for the data of this example.

Finally, calculate wall cooling load using radiant time series. Total cooling load for the wall is calculated by summing the convective and radiant portions. The convective portion is simply the wall heat gain for the hour being calculated times the convective fraction for walls from Table 14 [redacted]

[redacted]

The radiant portion of the cooling load is calculated using conductive heat gains for the current and past 23 h, the radiant fraction for walls from Table 14 [redacted] and radiant time series from Table 19, in accordance with Equation (34). From Table 19, select the RTS for medium-weight construction, assuming 50% glass and carpeted floors as representative for the described construction. Use the wall heat gains from Table 29B for 24 h design conditions in July. Thus, the radiant cooling load for the wall at 3:00 PM is

[redacted]

The total wall cooling load at the designated hour is thus

[redacted]

Again, a simple computer spreadsheet or other software is necessary to reduce the effort involved. A spreadsheet was used with the heat gain profile to split the heat gain into convective and radiant portions, apply RTS to the radiant portion, and total the convective and radiant loads to determine a 24 h cooling load profile for this example, with results in Table 29B.

Part 3. Window cooling load using radiant time series. Calculate the cooling load contribution, with and without indoor shading (venetian blinds) for the window area facing 60° west of south at 3:00 PM in July for the conference room example.

Solution: First, calculate the 24 h heat gain profile for the window, then split those heat gains into radiant and convective portions, apply the appropriate RTS to the radiant portion, then sum the convective and radiant cooling load components to determine total window cooling load for the time. The window heat gain components are calculated using Equations (13) to (15). From Part 2, at hour 15 LST (3:00 PM):

[redacted]

From Chapter 15, Table 10, for glass type 5d,

[redacted]

From Chapter 15, Table 13B, for light-colored blinds (assumed louver reflectance = 0.8 and louvers positioned at 45° angle) on double-glazed, heat-absorbing windows (Type 5d from Table 13B of Chapter 15), IAC(0) = 0.74, IAC(60) = 0.65, IAC(diff) = 0.79, and radiant fraction = 0.54. Without blinds, IAC = 1.0. Therefore, window heat gain components for hour 15, without blinds, are

[redacted]

This procedure is repeated to determine these values for a 24 h heat gain profile, shown in Table 30.

Total cooling load for the window is calculated by summing the convective and radiant portions. For windows with indoor shading (blinds, drapes, etc.), the direct beam, diffuse, and conductive heat gains may be summed and treated together in calculating cooling loads. However, in this example, the window does not have indoor shading, and the direct beam solar heat gain should be treated separately from the diffuse and conductive heat gains. The direct beam heat gain, without indoor shading, is treated as 100% radiant, and solar RTS factors from Table 20 are used to convert the beam heat gains to cooling loads. The diffuse and conductive heat gains can be totaled and split into radiant and convective portions according to Table 14, and nonsolar RTS factors from Table 19 are used to convert the radiant portion to cooling load.

The solar beam cooling load is calculated using heat gains for the current hour and past 23 h and radiant time series from Table 20, in accordance with Equation (39). From Table 20, select the solar RTS for medium-weight construction, assuming 50% glass and carpeted floors for this example. Using Table 30 values for direct solar heat gain, the radiant cooling load for the window direct beam solar component is

[redacted]

This process is repeated for other hours; results are listed in Table 31.

For diffuse and conductive heat gains, the radiant fraction according to Table 14 is 46%. The radiant portion is processed using nonsolar RTS coefficients from Table 19. The results are listed in Tables 30 and 31. For 3:00 PM, the diffuse and conductive cooling load is 1297 Btu/h.

The total window cooling load at the designated hour is thus

[redacted]

Again, a computer spreadsheet or other software is commonly used to reduce the effort involved in calculations. The spreadsheet illustrated in Table 30 is expanded in Table 31 to include splitting the heat gain into convective and radiant portions, applying RTS to the radiant portion, and totaling the convective and radiant loads to determine a 24 h cooling load profile for a window without indoor shading.

Table 29A Wall Component of Solar Irradiance

Local Standard Hour	Apparent Solar Time	Hour Angle H	Solar Altitude β	Solar Azimuth φ	Air Solar Mass m	Direct Beam Solar: E_b, Direct Normal Btu/h·ft²	Direct Beam Solar: Surface Incident Angle θ	Direct Beam Solar: Surface Direct Btu/h·ft²	Diffuse Solar Heat Gain: E_d, Diffuse Horizontal, Btu/h·ft²	Diffuse Solar Heat Gain: Ground Diffuse Btu/h·ft²	Diffuse Solar Heat Gain: Y Ratio	Diffuse Solar Heat Gain: Sky Diffuse Btu/h·ft²	Diffuse Solar Heat Gain: Subtotal Diffuse Btu/h·ft²	Total Surface Irradiance Btu/h·ft²
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
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Table 29B Wall Component of Sol-Air Temperatures, Heat Input, Heat Gain, Cooling Load

Local Standard Hour	Total Surface Irradiance Btu/h·ft²	Outdoor Temp., °F	Sol-Air Temp., °F	Indoor Temp., °F	Heat Input, Btu/h	CTS Type 1, %	Heat Gain, Btu/h: Total	Heat Gain, Btu/h: Convective 54%	Heat Gain, Btu/h: Radiant 46%	Nonsolar RTS Zone Type 8, %	Radiant Cooling Load, Btu/h	Total Cooling Load, Btu/h
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
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Table 30 Window Component of Heat Gain (No Blinds or Overhang)

	Beam Solar Heat Gain						Diffuse Solar Heat Gain							Conduction		
Local Std. Hour	Beam Normal, Btu/h·ft²	Surface Incident Angle	Surface Beam, Btu/h·ft²	Beam SHGC	Adjusted Beam IAC	Beam Solar Heat Gain, Btu/h	Diffuse Horiz. E_d, Btu/h·ft²	Ground Diffuse, Btu/h·ft²	*Y* Ratio	Sky Diffuse, Btu/h·ft²	Subtotal Diffuse, Btu/h·ft²	Hemis. SHGC	Diff. Solar Heat Gain, Btu/h	Outside Temp., °F	Conduction Heat Gain, Btu/h	Total Window Heat Gain, Btu/h
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Table 31 Window Component of Cooling Load (No Blinds or Overhang)

	Unshaded Direct Beam Solar (if AC = 1)						Shaded Direct Beam (AC < 1.0) + Diffuse + Conduction									
Local Standard Hour	Beam Heat Gain, Btu/h	Convective 0%, Btu/h	Radiant 100%, Btu/h	Solar RTS, Zone Type 8, %	Radiant Btu/h	Cooling Load, Btu/h	Beam Heat Gain, Btu/h	Diffuse Heat Gain, Btu/h	Conduction Heat Gain, Btu/h	Total Heat Gain, Btu/h	Convective 54%, Btu/h	Radiant 46%, Btu/h	Nonsolar RTS, Zone Type 8	Radiant Btu/h	Cooling Load, Btu/h	Window Cooling Load, Btu/h
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Table 32 Window Component of Cooling Load (With Blinds, No Overhang)

	Unshaded Direct Beam Solar (if AC = 1)						Shaded Direct Beam (AC < 1.0) + Diffuse + Conduction									
Local Standard Hour	Beam Heat Gain, Btu/h	Con-vective 0%, Btu/h	Radiant 100%, Btu/h	Solar RTS, Zone Type 8, %	Radiant Btu/h	Cooling Load, Btu/h	Beam Heat Gain, Btu/h	Diffuse Heat Gain, Btu/h	Con-duction Heat Gain, Btu/h	Total Heat Gain, Btu/h	Con-vective 54%, Btu/h	Radiant 46%, Btu/h	Non-solar RTS, Zone Type 8	Radiant, Btu/h	Cooling Load, Btu/h	Window Cooling Load, Btu/h
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

Table 33 Window Component of Cooling Load (With Blinds and Overhang)

	Overhang and Fins Shading					Shaded Direct Beam (AC < 1.0) + Diffuse + Conduction									
Local Standard Hour	Surface Solar Azimuth	Profile Angle	Shadow Width, ft	Shadow Height, ft	Direct Sunlit Area, ft^2	Beam Heat Gain, Btu/h	Diffuse Heat Gain, Btu/h	Con-duction Heat Gain, Btu/h	Total Heat Gain, Btu/h	Con-vective 54%, Btu/h	Radiant 46%, Btu/h	Non-solar RTS, Zone Type 8	Radiant, Btu/h	Cooling Load, Btu/h	Window Cooling Load, Btu/h
[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]	[redacted]

If the window has an indoor shading device, it is accounted for with the indoor attenuation coefficients (IAC), the radiant fraction, and the RTS type used. If a window has no indoor shading, 100% of the direct beam energy is assumed to be radiant and solar RTS factors are used. However, if an indoor shading device is present, the direct beam is assumed to be interrupted by the shading device, and a portion immediately becomes cooling load by convection. Also, the energy is assumed to be radiated to all surfaces of the room, therefore nonsolar RTS values are used to convert the radiant load into cooling load.

IAC values depend on several factors: (1) type of shading device, (2) position of shading device relative to window, (3) reflectivity of shading device, (4) angular adjustment of shading device, as well as (5) solar position relative to the shading device. These factors are discussed in detail in Chapter 15. For this example with venetian blinds, the IAC for beam radiation is treated separately from the diffuse solar gain. The direct beam IAC must be adjusted based on the profile angle of the sun. At 3:00 PM in July, the profile angle of the sun relative to the window surface is 58°. Calculated using Equation (45) from Chapter 15, the beam IAC = 0.653. The diffuse IAC is 0.79. Thus, the window heat gains, with light-colored blinds, at 3:00 PM are

[redacted]

Because the same radiant fraction and nonsolar RTS are applied to all parts of the window heat gain when indoor shading is present, those loads can be totaled and the cooling load calculated after splitting the radiant portion for processing with nonsolar RTS. This is illustrated by the spreadsheet results in Table 32. The total window cooling load with venetian blinds at 3:00 PM = 2171 Btu/h.

Part 4. Window cooling load using radiant time series for window with overhang shading. Calculate the cooling load contribution for the previous example with the addition of a 10 ft overhang shading the window.

Solution: In Chapter 15, methods are described and examples provided for calculating the area of a window shaded by attached vertical or horizontal projections. For 3:00 PM LST IN July, the solar position calculated in previous examples is

[redacted]

From Chapter 15, Equation (40), shadow height S_H is

[redacted]

Because the window is 6.4 ft tall, at 3:00 PM the window is completely shaded by the 10 ft deep overhang. Thus, the shaded window heat gain includes only diffuse solar and conduction gains. This is converted to cooling load by separating the radiant portion, applying RTS, and adding the resulting radiant cooling load to the convective portion to determine total cooling load. Those results are in Table 33. The total window cooling load = 1098 Btu/h.

Part 5. Room cooling load total. Calculate the sensible cooling loads for the previously described office at 3:00 PM in July.

Solution: The steps in the previous example parts are repeated for each of the internal and external loads components, including the southeast facing window, spandrel and brick walls, the southwest facing brick wall, the roof, people, and equipment loads. The results are tabulated in Table 34. The total room sensible cooling load for the office is 3674 Btu/h at 3:00 PM in July. When this calculation process is repeated for a 24 h design day for each month, it is found that the peak room sensible cooling load actually occurs in July at hour 14 (2:00 PM solar time) at 3675 Btu/h as indicated in Table 35.

Table 34 Single-Room Example Cooling Load (July 3:00 PM) for ASHRAE Example Office Building, Atlanta, GA

[redacted]

Although simple in concept, these steps involved in calculating cooling loads are tedious and repetitive, even using the “simplified” RTS method; practically, they should be performed using a computer spreadsheet or other program. The calculations should be repeated for multiple design conditions (i.e., times of day, other months) to determine the maximum cooling load for mechanical equipment sizing. Example spreadsheets for computing each cooling load component using conduction and radiant time series have been compiled and are available from ASHRAE. To illustrate the full building example discussed previously, those individual component spreadsheets have been compiled to allow calculation of cooling and heating loads on a room by room basis as well as for a “block” calculation for analysis of overall areas or buildings where detailed room-by-room data are not available.

SINGLE-ROOM EXAMPLE PEAK HEATING LOAD

Although the physics of heat transfer that creates a heating load is identical to that for cooling loads, a number of traditionally used simplifying assumptions facilitate a much simpler calculation procedure. As described in the Heating Load Calculations section,

Table 35 Single-Room Example Peak Cooling Load (Sept. 5:00 PM) for ASHRAE Example Office Building, Atlanta, GA

[redacted]

design heating load calculations typically assume a single outdoor temperature, with no heat gain from solar or internal sources, under steady-state conditions. Thus, space heating load is determined by computing the heat transfer rate through building envelope elements ($UA\Delta T$) plus heat required because of outdoor air infiltration.

Part 6. Room heating load. Calculate the room heating load for the previous described office, including infiltration airflow at one air change per hour.

Solution: Because solar heat gain is not considered in calculating design heating loads, orientation of similar envelope elements may be ignored and total areas of each wall or window type combined. Thus, the total spandrel wall area = 60 + 60 = 120 ft^2, total brick wall area = 60 + 40 = 100 ft^2, and total window area = 40 + 40 = 80 ft^2. For this example, use the U-factors that were used for cooling load conditions. In some climates, higher prevalent winds in winter should be considered in calculating U-factors (see Chapter 25 for information on calculating U-factors and surface heat transfer coefficients appropriate for local wind conditions). The 99.6% heating design dry-bulb temperature for Atlanta is [redacted] and the indoor design temperature is 72°F. The room volume with a 9 ft ceiling = 9 × 130 = 1170 ft^3. At one air change per hour, the infiltration airflow = 1 × 1170/60 = 19.5 cfm. Thus, the heating load is

[redacted]

WHOLE-BUILDING EXAMPLE

Because a single-room example does not illustrate the full application of load calculations, a multistory, multiple-room example building has been developed to show a more realistic case. A hypothetical project development process is described to illustrate its effect on the application of load calculations.

Design Process and Shell Building Definition

A development company has acquired a piece of property in Atlanta, GA, to construct an office building. Although no tenant or end user has yet been identified, the owner/developer has decided to proceed with the project on a speculative basis. They select an architectural design firm, who retains an engineering firm for the mechanical and electrical design.

At the first meeting, the developer indicates the project is to proceed on a fast-track basis to take advantage of market conditions; he is negotiating with several potential tenants who will need to occupy the new building within a year. This requires preparing **shell-and-core** construction documents to obtain a building permit, order equipment, and begin construction to meet the schedule.

The shell-and-core design documents will include finished design of the building exterior (the **shell**), as well as permanent interior elements such as stairs, restrooms, elevator, electrical rooms and mechanical spaces (the **core**). The primary mechanical equipment must be sized and installed as part of the shell-and-core package in order for the project to meet the schedule, even though the building occupant is not yet known.

The architect selects a two-story design with an exterior skin of tinted, double-glazed vision glass; opaque, insulated spandrel glass, and brick pilasters. The roof area extends beyond the building edge to form a substantial overhang, shading the second floor windows. Architectural drawings for the shell-and-core package (see Figures 17 to 22) include plans, elevations, and skin construction details, and are furnished to the engineer for use in “block” heating and cooling load calculations. Mechanical systems and equipment must be specified and installed based on those calculations. (*Note*: Full-size, scalable electronic versions of the drawings in Figures 17 to 22, as well as detailed lighting plans, are available from ASHRAE at www.ashrae.org.)

The HVAC design engineer meets with the developer’s operations staff to agree on the basic HVAC systems for the project. Based on their experience operating other buildings and the lack of specific information on the tenant(s), the team decides on two variable-volume air-handling units (AHUs), one per floor, to provide operating flexibility if one floor is leased to one tenant and the other floor to someone else. Cooling will be provided by an air-cooled chiller located on grade across the parking lot. Heating will be provided by electric resistance heaters in parallel-type fan-powered variable-air-volume (VAV) terminal units. The AHUs must be sized quickly to confirm the size of the mechanical rooms on the architectural plans. The AHUs and chiller must be ordered by the mechanical subcontractor within 10 days to meet the construction schedule. Likewise, the electric heating loads must be provided to the electrical engineers to size the electrical service and for the utility company to extend services to the site.

The mechanical engineer must determine the (1) peak airflow and cooling coil capacity for each AHU, (2) peak cooling capacity required for the chiller, and (3) total heating capacity for sizing the electrical service.

Table 36 Block Load Example: Envelope Area Summary, ft^2

	Floor Area	Brick Areas				Spandrel/Soffit Areas				Window Areas			
		North	South	East	West	North	South	East	West	North	South	East	West
First Floor	19,000	680	560	400	400	1400	1350	1040	360	600	1000	120	360
Second Floor	15,700	510	390	300	300	1040	920	540	540	560	840	360	360
Building Total	34,700	1190	950	700	700	2440	2270	1580	900	1160	1840	480	720

Solution: First, calculate "block" heating and cooling loads for each floor to size the AHUs, then calculate a block load for the whole building determine chiller and electric heating capacity.

Based on the architectural drawings, the HVAC engineer assembles basic data on the building as follows:

Location: Atlanta, GA. Per Chapter 14, latitude = 33.64, longitude = 84.43, elevation = 1027 ft above sea level, 99.6% heating design dry-bulb temperature = 21.5°F. For cooling load calculations, use 5% dry-bulb/coincident wet-bulb monthly design day profile from Chapter 14 (on CD-ROM). See Table 27 for temperature profiles used in these examples.

Indoor design conditions: 72°F for heating; 75°F with 50% rh for cooling.

Building orientation: Plan north is 30° west of true north.

Gross area per floor: 19,000 ft^2 first floor and 15,700 ft^2 second floor

Total building gross area: 34,700 ft^2

Windows: Bronze-tinted, double-glazed. Solar heat gain coefficients, U-factors are as in the single-room example.

Walls: Part insulated spandrel glass and part brick-and-block clad columns. The insulation barrier in the soffit at the second floor is similar to that of the spandrel glass and is of lightweight construction; for simplicity, that surface is assumed to have similar thermal heat gain/loss to the spandrel glass. Construction and insulation values are as in single-room example.

Roof: Metal deck, topped with board insulation and membrane roofing. Construction and insulation values are as in the single-room example.

Floor: 5 in. lightweight concrete slab on grade for first floor and 5 in. lightweight concrete on metal deck for second floor

Total areas of building exterior skin, as measured from the architectural plans, are listed in Table 36.

The engineer needs additional data to estimate the building loads. Thus far, no tenant has yet been signed, so no interior layouts for population counts, lighting layouts or equipment loads are available. To meet the schedule, assumptions must be made on these load components. The owner requires that the system design must be flexible enough to provide for a variety of tenants over the life of the building. Based on similar office buildings, the team agrees to base the block load calculations on the following assumptions:

Occupancy: 7 people per 1000 ft^2 = 143 ft^2/person
Lighting: 1.1 W/ft^2
Tenant's office equipment: 1 W/ft^2

Normal use schedule is assumed at 100% from 7:00 AM to 7:00 PM and unoccupied/off during other hours.

With interior finishes not finalized, the owner commits to using light-colored interior blinds on all windows. The tenant interior design could include carpeted flooring or acoustical tile ceilings in all areas, but the more conservative assumption, from a peak load standpoint, is chosen: carpeted flooring and no acoustical tile ceilings (no ceiling return plenum).

For block loads, the engineer assumes that the building is maintained under positive pressure during peak cooling conditions and that infiltration during peak heating conditions is equivalent to one air change per hour in a 12 ft deep perimeter zone around the building.

Table 37 Block Load Example—First Floor Loads for ASHRAE Example Office Building, Atlanta, GA

[redacted]

To maintain indoor air quality, [redacted] ASHRAE *Standard* [redacted] is the design basis for ventilation rates; however, no interior tenant layout is available for application of *Standard* [redacted]

Block load calculations were performed using the RTS method, and results for the first and second floors and the entire building are summarized in Tables 37, 38, and 39. Based on these results, the engineer performs psychrometric coil analysis, checks capacities versus vendor catalog data, and prepares specifications and schedules for the equipment. This information is released to the contractor with the shell-and-core design documents. The air-handling units and chiller are purchased, and construction proceeds.

Table 38 Block Load Example—Second Floor Loads for ASHRAE Example Office Building, Atlanta, GA

[redacted]

Table 39 Block Load Example—Overall Building Loads for ASHRAE Example Office Building, Atlanta, GA

[redacted]

Tenant Fit Design Process and Definition

About halfway through construction, a tenant agrees to lease the entire building. The tenant will require a combination of open and enclosed office space with a few common areas, such as conference/training rooms, and a small computer room that will operate on a 24 h basis. Based on the tenant's space program, the architect prepares interior floor plans and furniture layout plans (Figures 23 and 24), and the electrical engineer prepares lighting design plans. Those drawings are furnished to the HVAC engineer to prepare detailed design documents. The first step in this process is to prepare room-by-room peak heating and cooling load calculations, which will then be used for design of the air distribution systems from each of the VAV air handlers already installed.

The HVAC engineer must perform a room-by-room "takeoff" of the architect's drawings. For each room, this effort identifies the floor area, room function, exterior envelope elements and areas, number of occupants, and lighting and equipment loads.

The tenant layout calls for a dropped acoustical tile ceiling throughout, which will be used as a return air plenum. Typical 2 by 4 ft fluorescent, recessed, return-air-type lighting fixtures are selected. Based on this, the engineer assumes that 20% of the heat gain from lighting will be to the return air plenum and not enter rooms directly. Likewise, some portion of the heat gain from the roof will be extracted via the ceiling return air plenum. From experience, the engineer understands that return air plenum paths are not always predictable, and decides to credit only 30% of the roof heat gain to the return air, with the balance included in the room cooling load.

For the open office areas, some areas along the building perimeter will have different load characteristics from purely interior spaces because of heat gains and losses through the building skin. Although those perimeter areas are not separated from other open office spaces by walls, the engineer knows from experience that they must be served by separate control zones to maintain comfort conditions.

Room-by-Room Cooling and Heating Loads

The room-by-room results of RTS method calculations, including the month and time of day of each room's peak cooling load, as well as peak heating loads for each room and all input data, are available at www.ashrae.org in spreadsheet format similar to Table 39. These results are used by the HVAC engineer to select and design room air distribution devices and to schedule airflow rates for each space. That information is incorporated into the tenant fit drawings and specifications issued to the contractor.

Conclusions

The example results illustrate issues which should be understood and accounted for in calculating heating and cooling loads:

[redacted]

PREVIOUS COOLING LOAD CALCULATION METHODS

Procedures described in this chapter are the most current and scientifically derived means for estimating cooling load for a defined building space, but methods in earlier editions of the ASHRAE Handbook are valid for many applications. These earlier procedures are [redacted]

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