

NONRESIDENTIAL COOLING AND HEATING LOAD CALCULATION PROCEDURES

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THIS CHAPTER presents two load calculation methods that represent a significant departure from those in common use. 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 calculation procedures and scientific principles are explained in equation format. These equations are coded in a generic computer program named Hbfort, released with *Cooling and Heating Load Calculation Principles* (Pedersen et al. 1998), and linked to a user interface program to allow input and output in either inch-pound or SI units. The source code for these programs has been refined and enhanced under ASHRAE Research Project 987 and can be found in the *ASHRAE Load Calculation Toolkit*.

The second method is called the **radiant time series (RTS) method**, which is a simplified method directly related to and derived from the HB calculation procedure. This chapter presents the principles of both procedures rather than a working tool for the cooling load practitioner. The load prediction of an actual, multiple-room building requires a complex computer program.

While the procedures described in this chapter are the most reliable means for estimating cooling load for a defined building space, the methods described in earlier editions of the Handbook series are valid for many applications. Each of these earlier procedures is, however, a simplification of the heat balance principles, and their use requires experience to deal with atypical situations or unusual circumstances. In fact, any cooling or heating load estimate is no better than the assumptions used to define conditions and parameters such as physical makeup of the various envelope surfaces, conditions of occupancy and use, and ambient weather conditions outside the building. The experience of the practitioner can never be ignored.

The procedures described in this chapter are concerned with a given space or zone in a building. When estimating the loads for a group of spaces such as might be handled by a single air-handling system, the assembled zones must be analyzed to consider (1) the

simultaneous effects taking place; (2) any diversification of heat gains for occupants, lighting, or other internal load sources; (3) ventilation; and/or (4) any other unique circumstances. With large buildings that involve more than a single HVAC system, simultaneous loads and any additional diversities also must be considered. The 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.

In comparing the HB and RTS methods against the methods described in earlier versions of this chapter, the primary difference is the directness of approach as opposed to the simplifying techniques necessitated by the limited computer capability available in earlier days. The **transfer function method (TFM)**, for example, required many calculation steps. Also, this method was originally designed for energy analysis with emphasis on daily, monthly, and annual energy use and, thus, was more oriented to average hourly cooling loads than peak design loads.

The **total equivalent temperature differential method with time averaging (TETD/TA)** has been a highly reliable—if subjective—method of load estimating since its initial presentation in the 1967 *ASHRAE Handbook—Fundamentals*. Originally conceived as a manual method of calculation, it proved suitable only as a computer application because of the need to calculate an extended profile of hourly heat gain values from which the radiant components had to be averaged over a time perceived to represent the general mass of the building involved. Because this perception of thermal storage characteristics of a given building was almost entirely subjective, with little specific information for the user to judge variations, the TETD/TA method's primary usefulness has always been to the experienced engineer.

The **cooling load temperature differential method with solar cooling load factors (CLTD/CLF)** was an attempt to simplify the two-step TFM and TETD/TA methods into a single-step technique that allowed proceeding directly from raw data to cooling load without the intermediate conversion of radiant heat gain to cooling load. A series of factors were taken from cooling load calculation results (produced by more sophisticated methods) as “equivalent temperature differences” for use in traditional conduction ($q = UA\Delta t$) equa-

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

tions. The results, however, are approximate cooling load values rather than simple heat gain values. The simplifications required for this process limit the applicability of this method. In fact, past versions of this chapter listed several areas of potential use that should be avoided as beyond the range of applicability.

COOLING LOAD PRINCIPLES

The variables affecting cooling load calculations are numerous, often difficult to define precisely, and always intricately interrelated. Many cooling load components vary in magnitude over a wide range during a 24 h period. Because these cyclic changes in load components often are not in phase with each other, each must be analyzed to establish the maximum cooling load for a building or zone. A zoned system (a system of conditioning equipment serving several independent areas, each with its own temperature control) need provide no greater total cooling load capacity than the largest hourly summary of simultaneous zone loads throughout a design day; however, it must handle the peak cooling load for each zone at its individual peak hour. At certain times of the day during the heating or intermediate seasons, some zones may require heating while others require cooling.

Calculation Accuracy

A realistic cooling load calculation gives values adequate for acceptable system performance. Variation in the heat transmission coefficients of typical building materials and composite assemblies, the differing motivations and skills of those who construct the building, and the manner in which the building is actually operated are some of the variables that make a 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 cooling load.

Heat Flow Rates

In air-conditioning design, four related heat flow rates, each of which varies with time, must be differentiated: (1) space heat gain, (2) space cooling load, (3) space heat extraction rate, and (4) cooling coil load.

Space Heat Gain. This instantaneous rate of heat gain is the rate at which heat enters into and/or is generated within a space. Heat gain is classified by (1) the mode in which it enters the space and (2) whether it is a sensible or latent gain.

Mode of entry. The mode of entry includes (1) solar radiation through transparent surfaces; (2) heat conduction through exterior walls and roofs; (3) heat conduction through ceilings, floors, and interior partitions; (4) heat generated in the space by occupants, lights, and appliances; (5) energy transfer as a result of ventilation and infiltration of outdoor air; and (6) miscellaneous heat gains.

Sensible or latent heat. Sensible heat is added directly to the conditioned space by conduction, convection, and/or radiation. Latent heat gain occurs when moisture is added to the space (e.g., from vapor emitted by occupants and equipment). To maintain a constant humidity ratio, water vapor must condense on the cooling apparatus and be removed at a rate equal to the rate it is added to the space. The amount of energy required to offset the latent heat gain essentially equals the product of the rate of condensation and the latent heat of condensation. In selecting cooling apparatus, it is necessary to distinguish between sensible and latent heat gain. Every cooling apparatus has a maximum sensible heat removal capacity and a maximum latent heat removal capacity for particular operating conditions.

Space Cooling Load. This is the rate at which heat must be removed from the space to maintain a constant space air temperature. The sum of all space instantaneous heat gains at any given time does not necessarily (or even frequently) equal the cooling load for the space at that same time.

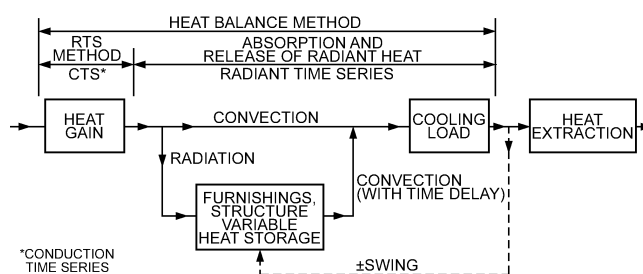


Fig. 1 Origin of Difference Between Magnitude of Instantaneous Heat Gain and Instantaneous Cooling Load

Radiant heat gain. Radiant energy must first be absorbed by the surfaces that enclose the space (walls, floor, and ceiling) and the objects in the space (furniture, etc.). When these surfaces and objects become warmer than the surrounding air, some of their heat is transferred to the air by convection. The composite heat storage capacity of these surfaces and objects determines the rate at which their respective surface temperatures increase for a given radiant input and thus governs the relationship between the radiant portion of heat gain and its corresponding part of the space cooling load (Figure 1). The thermal storage effect is critically important in differentiating between instantaneous heat gain for a given space and its cooling load at that moment. Predicting the nature and magnitude of this phenomenon in order to estimate a realistic cooling load for a particular combination of circumstances has long been a subject of interest to design engineers. The section on Bibliography lists some of the early work on the subject.

Space Heat Extraction Rate. The rate at which heat is removed from the conditioned space equals the space cooling load only if the room air temperature is held constant. Along with the intermittent operation of the cooling equipment, the control system characteristics usually permit a minor cyclic variation or swing in room temperature. Therefore, a proper simulation of the control system gives a more realistic value of energy removal over a fixed period than using the values of the space cooling load. However, this concept is primarily important for estimating energy use over time; it is not needed to calculate design peak cooling load for equipment selection.

Cooling Coil Load. The rate at which energy is removed at the cooling coil that serves one or more conditioned spaces equals the sum of the instantaneous space cooling loads (or space heat extraction rate if it is assumed that the space temperature does not vary) for all the spaces served by the coil, plus any external loads. Such external loads include fan heat gain, duct heat gain, and outdoor air heat and moisture brought into the cooling equipment to satisfy the ventilation requirement.

Cooling Load Estimation in Practice

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 where detailed partition requirements, furnishings, lighting selection, and layout cannot be predefined. Potential tenant modifications once the building is occupied also must be considered. The load estimating process requires proper engineering judgment that includes a thorough understanding of heat balance fundamentals.

INITIAL DESIGN CONSIDERATIONS

To calculate a space cooling load, detailed building design information and weather data at selected design conditions are required. Generally, the following steps should be followed.

Data Assembly

Building Characteristics. Obtain characteristics of the building. 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 by a site plan or by visiting the proposed site but should be carefully evaluated as to its probable permanence before it is included in the calculation. The possibility of abnormally high ground-reflected solar radiation (i.e., from adjacent water, sand, or parking lots) or solar load from adjacent reflective buildings should not be overlooked.

Outdoor Design Conditions. Obtain appropriate weather data, and select outdoor design conditions. For outdoor design conditions for a large number of weather stations, see [Chapter 27](#). Note, however, that these values for the 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.

In recent years, several research projects have greatly expanded the amount of available weather data (Colliver et al. 1995, 1998, 2000). In addition to the conventional dry-bulb with mean coincident wet-bulb, data are now available for wet-bulb and dew-point with mean coincident dry-bulb. The peak load for a space that requires both large quantities of outside air and close control of moisture may occur at peak wet-bulb or peak dew-point conditions when the corresponding dry-bulb temperature is significantly lower than normal design conditions.

Indoor Design Conditions. Select indoor design conditions, such as indoor dry-bulb temperature, indoor wet-bulb temperature, and ventilation rate. Include permissible variations and control limits.

Operating Schedules. Obtain a proposed schedule of lighting, occupancy, internal equipment, appliances, and processes that contribute to the internal thermal load. Determine the probability that the cooling equipment will be operated continuously or shut off during unoccupied periods (e.g., nights and/or weekends).

Date and Time. Select the time of day and month to do the cooling load calculation. Frequently, several different times of day and several different months must be analyzed to determine the peak load time. The particular day and month are often dictated by peak solar conditions. For southern exposures in north latitudes above 32° having large fenestration areas, the peak space cooling load usually occurs in December or January. To calculate a space cooling load under these conditions, the warmest temperature for the winter months must be known. These data can be found for the United States in [Chapter 27, Table 4B](#).

Additional Considerations

The proper design and sizing of all-air or air-and-water central air-conditioning systems require more than calculation of the cooling load in the space to be conditioned. The type of air-conditioning system, fan energy, fan location, duct heat loss and gain, duct leakage, heat extraction lighting systems, and type of return air system all affect system load and component sizing. Adequate system design and component sizing require that system performance be analyzed as a series of psychrometric processes.

HEAT SOURCES AND HEAT GAIN CALCULATION CONCEPTS

TIME DELAY EFFECT

The energy absorbed by walls, floor, furniture, etc., contributes to space cooling load only after a time lag, with some part of this

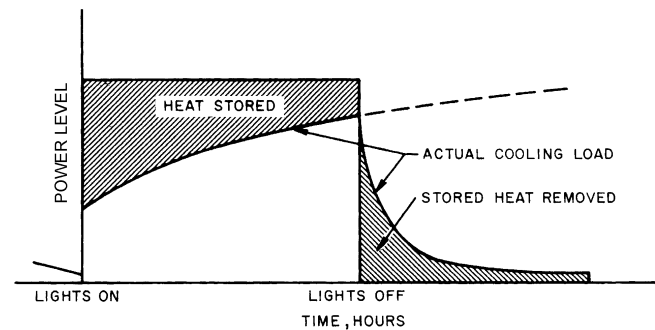


Fig. 2 Thermal Storage Effect in Cooling Load from Lights

energy still present and reradiating after the heat sources have been switched off or are no longer present ([Figure 2](#)).

There is always significant delay between the time of switching on or otherwise activating a heat source and the point when reradiated energy equals that being instantaneously stored. This time lag must be considered when calculating cooling load because the load felt by the space can be much lower than the instantaneous heat gain being generated, and the peak load for the space may be affected significantly.

PEOPLE

[Table 1](#) gives representative rates at which heat and moisture are given off by human beings in different states of activity. Often these sensible and latent heat gains constitute a large fraction of the total load. Even for short-term occupancy, the extra heat and moisture brought in by people may be significant. [Chapter 8](#) should be consulted for detailed information; however, [Table 1](#) summarizes design data representing conditions commonly encountered.

The conversion of sensible heat gain from people to space cooling load is affected by the thermal storage characteristics of that space, since some percentage of the sensible load is radiant energy. Latent heat gains are considered instantaneous.

LIGHTING

Because lighting is often the major space load cooling component, an accurate estimate of the space heat gain it imposes is needed. Calculation of this load component is not straightforward; the rate of cooling load due to lighting at any given moment can be quite different from the heat equivalent of power supplied instantaneously to those lights.

Instantaneous Heat Gain from Lighting

The primary source of heat from lighting comes from light-emitting elements, or lamps, although significant additional heat may be generated from associated appurtenances in the light fixtures that house such lamps. Generally, the instantaneous rate of heat gain from electric lighting may be calculated from

$$q_{el} = WF_{ul}F_{sa} \quad (1)$$

where

q_{el} = heat gain, W

W = total light wattage

F_{ul} = lighting use factor

F_{sa} = lighting special allowance factor

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

The **lighting use factor** is the ratio of the wattage in use, for the conditions under which the load estimate is being made, to the total

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

		Total Heat, W		Sensible Heat, W	Latent Heat, W	% Sensible Heat that is Radiant ^b	
Degree of Activity		Adult Male	Adjusted, M/F ^a			Low <i>V</i>	High <i>V</i>
Seated at theater	Theater, matinee	115	95	65	30		
Seated at theater, night	Theater, night	115	105	70	35	60	27
Seated, very light work	Offices, hotels, apartments	130	115	70	45		
Moderately active office work	Offices, hotels, apartments	140	130	75	55		
Standing, light work; walking	Department store; retail store	160	130	75	55	58	38
Walking, standing	Drug store, bank	160	145	75	70		
Sedentary work	Restaurant ^c	145	160	80	80		
Light bench work	Factory	235	220	80	140		
Moderate dancing	Dance hall	265	250	90	160	49	35
Walking 4.8 km/h; light machine work	Factory	295	295	110	185		
Bowling ^d	Bowling alley	440	425	170	255		
Heavy work	Factory	440	425	170	255	54	19
Heavy machine work; lifting	Factory	470	470	185	285		
Athletics	Gymnasium	585	525	210	315		

Notes:

1. Tabulated values are based on 24°C room dry-bulb temperature. For 27°C room dry bulb, the total heat remains the same, but the sensible heat values should be decreased by approximately 20%, and the latent heat values increased accordingly.

2. Also refer to [Table 4, Chapter 8](#), for additional rates of metabolic heat generation.

3. All values are rounded to nearest 5 W.

^aAdjusted heat gain is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is

85% of that for an adult male, and that the gain from a child is 75% of that for an adult male.

^bValues approximated from data in [Table 6, Chapter 8](#), where *V* is air velocity with limits shown in that table.

^cAdjusted heat gain includes 18 W for food per individual (9 W sensible and 9 W latent).

^dFigure one person per alley actually bowling, and all others as sitting (117 W) or standing or walking slowly (231 W).

installed wattage. For commercial applications such as stores, the use factor would generally be unity.

The **special allowance factor** is for fluorescent fixtures and/or fixtures that are either ventilated or installed so that only part of their heat goes to the conditioned space. For fluorescent or high-intensity discharge fixtures, the special allowance factor accounts primarily for ballast losses. [Table 2](#) shows that the special allowance factor for a two-lamp fluorescent fixture ranges from 0.94 for T8 lamps with an electronic ballast to 1.21 for energy-saver T12 lamps with a standard electromagnetic ballast. High-intensity discharge fixtures, such as metal halide, may have special allowance factors varying from 1.07 to 1.44, depending on the lamp wattage and quantity of lamps per fixture, and should be dealt with individually. A wide variety of lamp and ballast combinations is available, and ballast catalog data provide the overall fixture wattage.

For ventilated or recessed fixtures, manufacturers' or other data must be sought to establish the fraction of the total wattage that may be expected to enter the conditioned space directly (and subject to time lag effect) versus that which must be picked up by return air or in some other appropriate manner.

Light Heat Components

Cooling load caused by lights recessed into ceiling cavities is made up of two components: one part (known as the **heat-to-space** load) comes from the light heat directly contributing to the space heat gain, and the other is the light heat released into the above-ceiling cavity, which (if used as a return air plenum) is mostly picked up by the return air that passes over or through the light fixtures. In such a ceiling return air plenum, this second part of the load (sometimes referred to as **heat-to-return**) never enters the conditioned space. It does, however, add to the overall load and significantly influences the load calculation.

Even though the total cooling load imposed on the cooling coil from these two components remains the same, the larger the fraction of heat output picked up by the return air, the more the space cooling

load is reduced. The minimum required airflow rate for the conditioned space is decreased as the space cooling load decreases. Supply fan power decreases accordingly, which ultimately results in reduced energy consumption for the system and possibly reduced equipment size as well.

For ordinary design load estimation, the heat gain for each component may be calculated simply as a fraction of the total lighting load by using judgment to estimate heat-to-space and heat-to-return percentages (Mitalas and Kimura 1971).

Return Air Light Fixtures

Two generic types of return air light fixture are available—those that allow and those that do not allow return air to flow through the lamp chamber. The first type is sometimes called a heat-of-light fixture. The percentage of light heat released through the plenum side of various ventilated fixtures can be obtained from lighting fixture manufacturers. For representative data, see Nevins et al. (1971). Even unventilated fixtures lose some heat to plenum spaces; however, most of the heat ultimately enters the conditioned space from a dead-air plenum or is picked up by return air via ceiling return air openings. The percentage of heat to return air ranges from 40 to 60% for heat-to-return ventilated fixtures or 15 to 25% for unventilated fixtures.

Plenum Temperatures

As heat from lighting is picked up by the return air, the temperature differential between the ceiling cavity and the conditioned space causes part of that heat to flow from the ceiling back to the conditioned space. Return air from the conditioned space can be ducted to capture light heat without passing through a ceiling plenum as such, or the ceiling space can be used as a return air plenum, causing the distribution of light heat to be handled in distinctly different ways. Most plenum temperatures do not rise more than 0.5 to 1.5 K above space temperature, thus generating only a relatively small thermal gradient for heat transfer through plenum surfaces but

Table 2 Typical Nonincandescent Light Fixtures

Description	Ballast	Watts/Lamp	Lamps/Fixture	Lamp Watts	Fixture Watts	Special Allowance Factor	Description	Ballast	Watts/Lamp	Lamps/Fixture	Lamp Watts	Fixture Watts	Special Allowance Factor
Compact Fluorescent Fixtures													
Twin, (1) 5 W lamp	Mag-Std	5	1	5	9	1.80	Twin, (2) 40 W lamp	Mag-Std	40	2	80	85	1.06
Twin, (1) 7 W lamp	Mag-Std	7	1	7	10	1.43	Quad, (1) 13 W lamp	Electronic	13	1	13	15	1.15
Twin, (1) 9 W lamp	Mag-Std	9	1	9	11	1.22	Quad, (1) 26 W lamp	Electronic	26	1	26	27	1.04
Quad, (1) 13 W lamp	Mag-Std	13	1	13	17	1.31	Quad, (2) 18 W lamp	Electronic	18	2	36	38	1.06
Quad, (2) 18 W lamp	Mag-Std	18	2	36	45	1.25	Quad, (2) 26 W lamp	Electronic	26	2	52	50	0.96
Quad, (2) 22 W lamp	Mag-Std	22	2	44	48	1.09	Twin or multi, (2) 32 W lamp	Electronic	32	2	64	62	0.97
Quad, (2) 26 W lamp	Mag-Std	26	2	52	66	1.27							
Fluorescent Fixtures													
(1) 450 mm, T8 lamp	Mag-Std	15	1	15	19	1.27	(4) 1200 mm, T8 lamp	Electronic	32	4	128	120	0.94
(1) 450 mm, T12 lamp	Mag-Std	15	1	15	19	1.27	(1) 1500 mm, T12 lamp	Mag-Std	50	1	50	63	1.26
(2) 450 mm, T8 lamp	Mag-Std	15	2	30	36	1.20	(2) 1500 mm, T12 lamp	Mag-Std	50	2	100	128	1.28
(2) 450 mm, T12 lamp	Mag-Std	15	2	30	36	1.20	(1) 1500 mm, T12 HO lamp	Mag-Std	75	1	75	92	1.23
(1) 600 mm, T8 lamp	Mag-Std	17	1	17	24	1.41	(2) 1500 mm, T12 HO lamp	Mag-Std	75	2	150	168	1.12
(1) 600 mm, T12 lamp	Mag-Std	20	1	20	28	1.40	(1) 1500 mm, T12 ES VHO lamp	Mag-Std	135	1	135	165	1.22
(2) 600 mm, T12 lamp	Mag-Std	20	2	40	56	1.40	(2) 1500 mm, T12 ES VHO lamp	Mag-Std	135	2	270	310	1.15
(1) 600 mm, T12 HO lamp	Mag-Std	35	1	35	62	1.77	(1) 1500 mm, T12 HO lamp	Mag-ES	75	1	75	88	1.17
(2) 600 mm, T12 HO lamp	Mag-Std	35	2	70	90	1.29	(2) 1500 mm, T12 HO lamp	Mag-ES	75	2	150	176	1.17
(1) 600 mm, T8 lamp	Electronic	17	1	17	16	0.94	(1) 1500 mm, T12 lamp	Electronic	50	1	50	44	0.88
(2) 600 mm, T8 lamp	Electronic	17	2	34	31	0.91	(2) 1500 mm, T12 lamp	Electronic	50	2	100	88	0.88
(1) 900 mm, T12 lamp	Mag-Std	30	1	30	46	1.53	(1) 1500 mm, T12 HO lamp	Electronic	75	1	75	69	0.92
(2) 900 mm, T12 lamp	Mag-Std	30	2	60	81	1.35	(2) 1500 mm, T12 HO lamp	Electronic	75	2	150	138	0.92
(1) 900 mm, T12 ES lamp	Mag-Std	25	1	25	42	1.68	(1) 1500 mm, T8 lamp	Electronic	40	1	40	36	0.90
(2) 900 mm, T12 ES lamp	Mag-Std	25	2	50	73	1.46	(2) 1500 mm, T8 lamp	Electronic	40	2	80	72	0.90
(1) 900 mm, T12 HO lamp	Mag-Std	50	1	50	70	1.40	(3) 1500 mm, T8 lamp	Electronic	40	3	120	106	0.88
(2) 900 mm, T12 HO lamp	Mag-Std	50	2	100	114	1.14	(4) 1500 mm, T8 lamp	Electronic	40	4	160	134	0.84
(2) 900 mm, T12 lamp	Mag-ES	30	2	60	74	1.23	(1) 1800 mm, T12 lamp	Mag-Std	55	1	55	76	1.38
(2) 900 mm, T12 ES lamp	Mag-ES	25	2	50	66	1.32	(2) 1800 mm, T12 lamp	Mag-Std	55	2	110	122	1.11
(1) 900 mm, T12 lamp	Electronic	30	1	30	31	1.03	(3) 1800 mm, T12 lamp	Mag-Std	55	3	165	202	1.22
(1) 900 mm, T12 ES lamp	Electronic	25	1	25	26	1.04	(4) 1800 mm, T12 lamp	Mag-Std	55	4	220	244	1.11
(1) 900 mm, T8 lamp	Electronic	25	1	25	24	0.96	(1) 1800 mm, T12 HO lamp	Mag-Std	85	1	85	120	1.41
(2) 900 mm, T12 lamp	Electronic	30	2	60	58	0.97	(2) 1800 mm, T12 HO lamp	Mag-Std	85	2	170	220	1.29
(2) 900 mm, T12 ES lamp	Electronic	25	2	50	50	1.00	(1) 1800 mm, T12 VHO lamp	Mag-Std	160	1	160	180	1.13
(2) 900 mm, T8 lamp	Electronic	25	2	50	46	0.92	(2) 1800 mm, T12 VHO lamp	Mag-Std	160	2	320	330	1.03
(2) 900 mm, T8 HO lamp	Electronic	25	2	50	50	1.00	(2) 1800 mm, T12 lamp	Mag-ES	55	2	110	122	1.11
(2) 900 mm, T8 VHO lamp	Electronic	25	2	50	70	1.40	(4) 1800 mm, T12 lamp	Mag-ES	55	4	220	244	1.11
(1) 1200 mm, T12 lamp	Mag-Std	40	1	40	55	1.38	(2) 1800 mm, T12 HO lamp	Mag-ES	85	2	170	194	1.14
(2) 1200 mm, T12 lamp	Mag-Std	40	2	80	92	1.15	(4) 1800 mm, T12 HO lamp	Mag-ES	85	4	340	388	1.14
(3) 1200 mm, T12 lamp	Mag-Std	40	3	120	140	1.17	(1) 1800 mm, T12 lamp	Electronic	55	1	55	68	1.24
(4) 1200 mm, T12 lamp	Mag-Std	40	4	160	184	1.15	(2) 1800 mm, T12 lamp	Electronic	55	2	110	108	0.98
(1) 1200 mm, T12 ES lamp	Mag-Std	34	1	34	48	1.41	(3) 1800 mm, T12 lamp	Electronic	55	3	165	176	1.07
(2) 1200 mm, T12 ES lamp	Mag-Std	34	2	68	82	1.21	(4) 1800 mm, T12 lamp	Electronic	55	4	220	216	0.98
(3) 1200 mm, T12 ES lamp	Mag-Std	34	3	102	100	0.98	(1) 2400 mm, T12 ES lamp	Mag-Std	60	1	60	75	1.25
(4) 1200 mm, T12 ES lamp	Mag-Std	34	4	136	164	1.21	(2) 2400 mm, T12 ES lamp	Mag-Std	60	2	120	128	1.07
(1) 1200 mm, T12 ES lamp	Mag-ES	34	1	34	43	1.26	(3) 2400 mm, T12 ES lamp	Mag-Std	60	3	180	203	1.13
(2) 1200 mm, T12 ES lamp	Mag-ES	34	2	68	72	1.06	(4) 2400 mm, T12 ES lamp	Mag-Std	60	4	240	256	1.07
(3) 1200 mm, T12 ES lamp	Mag-ES	34	3	102	115	1.13	(1) 2400 mm, T12 ES HO lamp	Mag-Std	95	1	95	112	1.18
(4) 1200 mm, T12 ES lamp	Mag-ES	34	4	136	144	1.06	(2) 2400 mm, T12 ES HO lamp	Mag-Std	95	2	190	227	1.19
(1) 1200 mm, T8 lamp	Mag-ES	32	1	32	35	1.09	(3) 2400 mm, T12 ES HO lamp	Mag-Std	95	3	285	380	1.33
(2) 1200 mm, T8 lamp	Mag-ES	32	2	64	71	1.11	(4) 2400 mm, T12 ES HO lamp	Mag-Std	95	4	380	454	1.19
(3) 1200 mm, T8 lamp	Mag-ES	32	3	96	110	1.15	(1) 2400 mm, T12 ES VHO lamp	Mag-Std	185	1	185	205	1.11
(4) 1200 mm, T8 lamp	Mag-ES	32	4	128	142	1.11	(2) 2400 mm, T12 ES VHO lamp	Mag-Std	185	2	370	380	1.03
(1) 1200 mm, T12 ES lamp	Electronic	34	1	34	32	0.94	(3) 2400 mm, T12 ES VHO lamp	Mag-Std	185	3	555	585	1.05
(2) 1200 mm, T12 ES lamp	Electronic	34	2	68	60	0.88	(4) 2400 mm, T12 ES VHO lamp	Mag-Std	185	4	740	760	1.03
(3) 1200 mm, T12 ES lamp	Electronic	34	3	102	92	0.90	(2) 2400 mm, T12 ES lamp	Mag-ES	60	2	120	123	1.03
(4) 1200 mm, T12 ES lamp	Electronic	34	4	136	120	0.88	(3) 2400 mm, T12 ES lamp	Mag-ES	60	3	180	210	1.17
(1) 1200 mm, T8 lamp	Electronic	32	1	32	32	1.00	(4) 2400 mm, T12 ES lamp	Mag-ES	60	4	240	246	1.03
(2) 1200 mm, T8 lamp	Electronic	32	2	64	60	0.94	(2) 2400 mm, T12 ES HO lamp	Mag-ES	95	2	190	207	1.09
(3) 1200 mm, T8 lamp	Electronic	32	3	96	93	0.97	(4) 2400 mm, T12 ES HO lamp	Mag-ES	95	4	380	414	1.09

Table 2 Typical Nonincandescent Light Fixtures (Concluded)

Description	Ballast	Watts/Lamp	Lamps/Fixture	Lamp Watts	Fixture Watts	Special Allowance Factor	Description	Ballast	Watts/Lamp	Lamps/Fixture	Lamp Watts	Fixture Watts	Special Allowance Factor
(1) 2400 mm, T12 ES lamp	Electronic	60	1	60	69	1.15	(1) 2400 mm, T8 HO lamp	Electronic	59	1	59	68	1.15
(2) 2400 mm, T12 ES lamp	Electronic	60	2	120	110	0.92	(1) 2400 mm, T8 VHO lamp	Electronic	59	1	59	71	1.20
(3) 2400 mm, T12 ES lamp	Electronic	60	3	180	179	0.99	(2) 2400 mm, T8 lamp	Electronic	59	2	118	109	0.92
(4) 2400 mm, T12 ES lamp	Electronic	60	4	240	220	0.92	(3) 2400 mm, T8 lamp	Electronic	59	3	177	167	0.94
(1) 2400 mm, T12 ES HO lamp	Electronic	95	1	95	80	0.84	(4) 2400 mm, T8 lamp	Electronic	59	4	236	219	0.93
(2) 2400 mm, T12 ES HO lamp	Electronic	95	2	190	173	0.91	(2) 2400 mm, T8 HO lamp	Electronic	86	2	172	160	0.93
(4) 2400 mm, T12 ES HO lamp	Electronic	95	4	380	346	0.91	(4) 2400 mm, T8 HO lamp	Electronic	86	4	344	320	0.93
(1) 2400 mm, T8 lamp	Electronic	59	1	59	58	0.98							
Circular Fluorescent Fixtures													
Circlite, (1) 20 W lamp	Mag-PH	20	1	20	20	1.00	(2) 200 mm circular lamp	Mag-RS	22	2	44	52	1.18
Circlite, (1) 22 W lamp	Mag-PH	22	1	22	20	0.91	(1) 300 mm circular lamp	Mag-RS	32	1	32	31	0.97
Circline, (1) 32 W lamp	Mag-PH	32	1	32	40	1.25	(2) 300 mm circular lamp	Mag-RS	32	2	64	62	0.97
(1) 150 mm circular lamp	Mag-RS	20	1	20	25	1.25	(1) 400 mm circular lamp	Mag-Std	40	1	40	35	0.88
(1) 200 mm circular lamp	Mag-RS	22	1	22	26	1.18							
High-Pressure Sodium Fixtures													
(1) 35 W lamp	HID	35	1	35	46	1.31	(1) 250 W lamp	HID	250	1	250	295	1.18
(1) 50 W lamp	HID	50	1	50	66	1.32	(1) 310 W lamp	HID	310	1	310	365	1.18
(1) 70 W lamp	HID	70	1	70	95	1.36	(1) 360 W lamp	HID	360	1	360	414	1.15
(1) 100 W lamp	HID	100	1	100	138	1.38	(1) 400 W lamp	HID	400	1	400	465	1.16
(1) 150 W lamp	HID	150	1	150	188	1.25	(1) 1000 W lamp	HID	1000	1	1000	1100	1.10
(1) 200 W lamp	HID	200	1	200	250	1.25							
Metal Halide Fixtures													
(1) 32 W lamp	HID	32	1	32	43	1.34	(1) 250 W lamp	HID	250	1	250	295	1.18
(1) 50 W lamp	HID	50	1	50	72	1.44	(1) 400 W lamp	HID	400	1	400	458	1.15
(1) 70 W lamp	HID	70	1	70	95	1.36	(2) 400 W lamp	HID	400	2	800	916	1.15
(1) 100 W lamp	HID	100	1	100	128	1.28	(1) 750 W lamp	HID	750	1	750	850	1.13
(1) 150 W lamp	HID	150	1	150	190	1.27	(1) 1000 W lamp	HID	1000	1	1000	1080	1.08
(1) 175 W lamp	HID	175	1	175	215	1.23	(1) 1500 W lamp	HID	1500	1	1500	1610	1.07
Mercury Vapor Fixtures													
(1) 40 W lamp	HID	40	1	40	50	1.25	(1) 250 W lamp	HID	250	1	250	290	1.16
(1) 50 W lamp	HID	50	1	50	74	1.48	(1) 400 W lamp	HID	400	1	400	455	1.14
(1) 75 W lamp	HID	75	1	75	93	1.24	(2) 400 W lamp	HID	400	2	800	910	1.14
(1) 100 W lamp	HID	100	1	100	125	1.25	(1) 700 W lamp	HID	700	1	700	780	1.11
(1) 175 W lamp	HID	175	1	175	205	1.17	(1) 1000 W lamp	HID	1000	1	1000	1075	1.08

Abbreviations: Mag = electromagnetic; ES = energy saver; Std = standard; HID = high-intensity discharge; HO = high output; VHO = very high output; PH = preheat; RS = rapid start

a relatively large percentage reduction in space cooling load. (Many engineers believe that a major reason for plenum temperatures not becoming more elevated is due to leakage into the plenum from supply air ducts normally concealed there.)

Energy Balance

Where the ceiling space is used as a return air plenum, an energy balance requires that the heat picked up from the lights into the return air (1) become a part of the cooling load to the return air (represented by a temperature rise of the return air as it passes through the ceiling space), (2) be partially transferred back into the conditioned space through the ceiling material below, and/or (3) may be partially "lost" (from the space) through the floor surfaces above the plenum. In a multistory building, the conditioned space frequently gains heat through its floor from a similar plenum below, offsetting the loss just mentioned. The radiant component of heat leaving the ceiling or floor surface of a plenum is normally so small that all such heat transfer is considered convective for calculation purposes.

Figure 3 shows a schematic diagram of a typical return air plenum. The following equations, using the heat flow directions shown

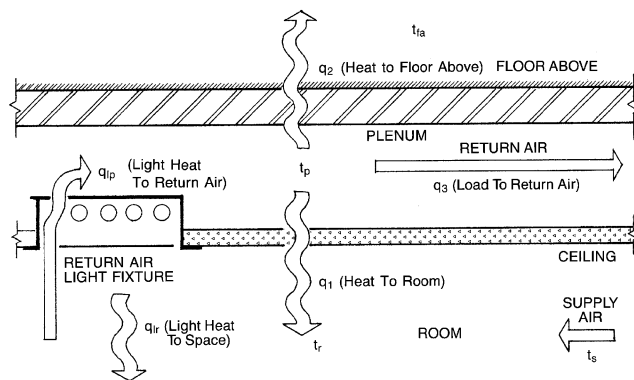


Fig. 3 Schematic Diagram of Typical Return Air Plenum

in Figure 3, represent the heat balance of a return air plenum design for a typical interior room in a multifloor building:

$$q_1 = U_c A_c (t_p - t_r) \quad (2)$$

$$q_2 = U_f A_f (t_p - t_{fa}) \quad (3)$$

$$q_3 = 1.23 Q (t_p - t_r) \quad (4)$$

$$q_{lp} - q_2 - q_1 - q_3 = 0 \quad (5)$$

$$Q = \frac{q_r + q_1}{1.23(t_r - t_s)} \quad (6)$$

where

- q_1 = heat gain to space from plenum through ceiling, W
- q_2 = heat loss from plenum through floor above, W
- q_3 = heat gain "pickup" by return air, W
- Q = return airflow, L/s
- q_{lp} = light heat gain to plenum via return air, W
- q_{lr} = light heat gain to space, W
- q_f = heat gain from plenum below, through floor, W
- q_w = heat gain from exterior wall, W
- q_r = space cooling load, including appropriate treatment of q_{lr} , q_f , and/or q_w , W
- t_p = plenum temperature, °C
- t_r = space temperature, °C
- t_{fa} = space temperature of floor above, °C
- t_s = supply air temperature, °C

From heat balance Equation (5), t_p can be found as the resultant return air temperature or plenum temperature. The results, although rigorous and best solved by computer, are important in determining the cooling load, which affects equipment size selection, future energy consumption, and other factors.

Equations (2) through (6) are simplified to illustrate the heat balance relationship. Heat gain into a return air plenum is not limited to the heat of lights alone. Exterior walls directly exposed to the ceiling space will transfer heat directly to or from the return air. For single-story buildings or the top floor of a multistory building, the roof heat gain or loss enters or leaves the ceiling plenum rather than entering or leaving the conditioned space directly. The supply air quantity calculated by Equation (6) is only for the conditioned space under consideration and is assumed equal to the return air quantity.

The amount of airflow through a return plenum above a conditioned space may not be limited to that supplied into the space under consideration; it will, however, have no noticeable effect on plenum temperature if the surplus comes from an adjacent plenum operating under similar conditions. Where special conditions exist, heat balance Equations (2) through (6) must be modified appropriately. Finally, even though the building's thermal storage has some effect, the amount of heat entering the return air is small and may be considered as convective for calculation purposes.

ELECTRIC MOTORS

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

$$q_{em} = (P/E_M) F_{UM} F_{LM} \quad (7)$$

where

- q_{em} = heat equivalent of equipment operation, W
- P = motor power rating, W
- 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

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 would be 1.0.

The motor load factor is the fraction of the rated load being delivered under the conditions of the cooling load estimate. In Equation (7), it is assumed that both the motor and the driven

equipment are in the conditioned space. If the motor is outside the space or airstream,

$$q_{em} = P F_{UM} F_{LM} \quad (8)$$

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

$$q_{em} = P \left(\frac{1.0 - E_M}{E_M} \right) F_{UM} F_{LM} \quad (9)$$

Equation (9) also applies to a fan or pump in the conditioned space that exhausts air or pumps fluid outside that space.

Tables 3A and 3B give average efficiencies and related data representative of typical electric motors, generally derived from the lower efficiencies reported by several manufacturers of open, drip-proof motors. These reports indicate that totally enclosed

Table 3 Average Efficiencies and Related Data Representative of Typical Electric Motors

Motor Name-plate or Rated Horse-power	(kW)	Motor Type	Nominal rpm	Full Load Motor Efficiency, %	Location of Motor and Driven Equipment with Respect to Conditioned Space or Airstream		
					A	B	C
					Motor in, Driven Equipment in, W	Motor out, Driven Equipment in, W	Motor in, Driven Equipment out, W
0.05	(0.04)	Shaded pole	1500	35	105	35	70
0.08	(0.06)	Shaded pole	1500	35	170	59	110
0.125	(0.09)	Shaded pole	1500	35	264	94	173
0.16	(0.12)	Shaded pole	1500	35	340	117	223
0.25	(0.19)	Split phase	1750	54	346	188	158
0.33	(0.25)	Split phase	1750	56	439	246	194
0.50	(0.37)	Split phase	1750	60	621	372	249
0.75	(0.56)	3-phase	1750	72	776	557	217
1	(0.75)	3-phase	1750	75	993	747	249
1.5	(1.1)	3-phase	1750	77	1453	1119	334
2	(1.5)	3-phase	1750	79	1887	1491	396
3	(2.2)	3-phase	1750	81	2763	2238	525
5	(3.7)	3-phase	1750	82	4541	3721	817
7.5	(5.6)	3-phase	1750	84	6651	5596	1066
10	(7.5)	3-phase	1750	85	8760	7178	1315
15	(11.2)	3-phase	1750	86	13 009	11 192	1820
20	(14.9)	3-phase	1750	87	17 140	14 913	2230
25	(18.6)	3-phase	1750	88	21 184	18 635	2545
30	(22.4)	3-phase	1750	89	25 110	22 370	2765
40	(30)	3-phase	1750	89	33 401	29 885	3690
50	(37)	3-phase	1750	89	41 900	37 210	4600
60	(45)	3-phase	1750	89	50 395	44 829	5538
75	(56)	3-phase	1750	90	62 115	55 962	6210
100	(75)	3-phase	1750	90	82 918	74 719	8290
125	(93)	3-phase	1750	90	103 430	93 172	10 342
150	(110)	3-phase	1750	91	123 060	111 925	11 075
200	(150)	3-phase	1750	91	163 785	149 135	14 738
250	(190)	3-phase	1750	91	204 805	186 346	18 430

Table 3B Typical Overload Limits with Standard Motors

Motor Type	Watts			
	40 to 190	120 to 250	500 to 560	750 and up
AC open	1.4	1.35	1.25	1.15
AC TEFC ^a and DC	—	1.0	1.0	1.0

Note: Some shaded pole, capacitor start, and special purpose motors have a service factor varying from 1.0 up to 1.75.

^aSome totally enclosed fan-cooled (TEFC) motors have a service factor above 1.0.

fan-cooled (TEFC) motors are slightly more efficient. For speeds lower or higher than those listed, efficiencies may be 1 to 3% lower or higher, depending on the manufacturer. Should actual voltages at motors be appreciably higher or lower than rated nameplate voltage, efficiencies in either case will be lower. If electric motor load is an appreciable portion of cooling load, the motor efficiency should be obtained from the manufacturer. Also, depending on design, the maximum efficiency might occur anywhere between 75 to 110% of full load; if underloaded or overloaded, the efficiency could vary from the manufacturer's listing.

Overloading or Underloading

Heat output of a motor is generally proportional to the motor load, within the 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, can be accurately established, and the reduced load efficiency data can be obtained from the motor manufacturer.

Radiation and Convection

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

APPLIANCES

In a cooling load estimate, heat gain from all appliances—electrical, gas, or steam—should be taken into account. 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.

Cooking Appliances

These appliances include common heat-producing cooking equipment found in conditioned commercial kitchens. Marn (1962) concluded that appliance surfaces contributed most of the heat to commercial kitchens and that when appliances were installed under an effective hood, the cooling load was independent of the fuel or energy used for similar equipment performing the same operations.

Gordon et al. (1994) and Smith et al. (1995) found that gas appliances may exhibit slightly higher heat gains than their electric counterparts under wall-canopy hoods operated at typical ventilation rates. This is due to the fact that the heat contained in the combustion products exhausted from a gas appliance may increase the temperatures of the appliance and surrounding surfaces, as well as the hood above the appliance, more than the heat produced by its electric counterpart. These higher temperature surfaces radiate heat to the kitchen, adding moderately to the radiant gain directly associated with the appliance cooking surface.

Marn (1962) confirmed that where the appliances are installed under an effective hood, only radiant gain adds to the cooling load; convected and latent heat from the cooking process and combustion products are exhausted and do not enter the kitchen. Gordon et al. (1994) and Smith et al. (1995) substantiated these findings.

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.) or as an alternative approach, recommended heat gains listed in Table 5 for a wide variety of commonly encountered equipment items may be used. In estimating the appliance load, probabilities of simultaneous use and operation for different appliances located in the same space must be considered.

The radiant heat gain from hooded cooking equipment can range from 15 to 45% of the actual appliance energy consumption (Talbert et al. 1973, Gordon et al. 1994, Smith et al. 1995). This ratio of heat gain to appliance energy consumption may be expressed as a radiation factor. It is a function of both appliance type and fuel source. The radiation factor F_R is applied to the average rate of appliance energy consumption, determined by applying usage factor F_U to the nameplate or rated energy input. Marn (1962) found that radiant heat temperature rise can be substantially reduced by shielding the fronts of cooking appliances. Although this approach may not always be practical in a commercial kitchen, radiant gains can also be reduced by adding side panels or partial enclosures that are integrated with the exhaust hood.

Heat Gain from Meals. For each meal served, the heat transferred to the dining space is approximately 15 W, of which 75% is sensible and 25% is latent.

Heat Gain for Electric and Steam Appliances. The average rate of appliance energy consumption can be estimated from the nameplate or rated energy input q_{input} by applying a duty cycle or usage factor F_U . Thus, the sensible heat gain $q_{sensible}$ for generic types of electric, steam, and gas appliances installed under a hood can be estimated using one of the following equations:

$$q_{sensible} = q_{input} F_U F_R \quad (10)$$

or

$$q_{sensible} = q_{input} F_L \quad (11)$$

where F_L is defined as the ratio of sensible heat gain to the manufacturer's rated energy input.

Table 4 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.

Unhooded Equipment. For all cooking appliances not installed under an exhaust hood or directly vent-connected and located in the conditioned area, the heat gain may be estimated as 50% ($F_U = 0.50$)

Table 4A Hooded Electric Appliance Usage Factors, Radiation Factors, and Load Factors

Appliance	Usage Factor F_U	Radiation Factor F_R	Load Factor $F_L = F_U F_R$ Elec/Steam
Griddle	0.16	0.45	0.07
Fryer	0.06	0.43	0.03
Convection oven	0.42	0.17	0.07
Charbroiler	0.83	0.29	0.24
Open-top range without oven	0.34	0.46	0.16
Hot-top range without oven	0.79	0.47	0.37
with oven	0.59	0.48	0.28
Steam cooker	0.13	0.30	0.04

Sources: Alereza and Breen (1984), Fisher (1998).

Table 4B Hooded Gas Appliance Usage Factors, Radiation Factors, and Load Factors

Appliance	Usage Factor F_U	Radiation Factor F_R	Load Factor $F_L = F_U F_R$ Gas
Griddle	0.25	0.25	0.06
Fryer	0.07	0.35	0.02
Convection oven	0.42	0.20	0.08
Charbroiler	0.62	0.18	0.11
Open-top range without oven	0.34	0.17	0.06

Sources: Alereza and Breen (1984), Fisher (1998).

or the rated hourly input, regardless of the type of energy or fuel used. On average, 34% of the heat may be assumed to be latent and the remaining 66% sensible. Note that cooking appliances ventilated by “ductless” hoods should be treated as unhooded appliances from the perspective of estimating heat gain. In other words, all energy consumed by the appliance and all moisture produced by the cooking process is introduced to the kitchen as a sensible or latent cooling load.

Recommended Heat Gain Values. As an alternative procedure, [Table 5](#) lists recommended rates of heat gain from typical commercial cooking appliances. The data in the “with hood” columns assume installation under a properly designed exhaust hood connected to a mechanical fan exhaust system.

Hospital and Laboratory Equipment

Hospital and laboratory equipment items are major sources of heat gain in conditioned spaces. Care must be taken in evaluating the probability and duration of simultaneous usage when many components are concentrated in one area, such as a laboratory, an operating room, etc. Commonly, heat gain from equipment in a laboratory ranges from 50 to 220 W/m² or, in laboratories with outdoor exposure, as much as four times the heat gain from all other sources combined.

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 and can be presented, but the equipment included represents only a small sample of the type of equipment that may be encountered.

The data presented for medical equipment in [Table 6](#) are relevant for portable and bench-top equipment. Medical equipment is very specific and can vary greatly from application to application. The data are presented to provide guidance in only the most general sense. For large equipment, such as MRI, engineers must obtain heat gain from the manufacturer.

Laboratory Equipment. Equipment in laboratories is similar to medical equipment in that it will vary significantly from space to space. [Chapter 14 of the ASHRAE Handbook—Applications](#) discusses heat gain from equipment, stating that it may range from 50 to 270 W/m² in highly automated laboratories. [Table 7](#) lists some values for laboratory equipment, but, as is the case for medical equipment, it is for general guidance only. Wilkins and Cook (1999) also examined laboratory equipment heat gains.

Office Equipment

Computers, printers, copiers, calculators, checkwriters, posting machines, etc., can generate 9 to 13 W/m² for general offices or 18 to 22 W/m² for purchasing and accounting departments. ASHRAE *Research Project 822* developed a method to measure the actual heat gain from equipment in buildings and the radiant/convective percentages (Hosni et al. 1998; Jones et al. 1998). This methodology was then incorporated into ASHRAE *Research Project 1055* and applied to a wide range of equipment (Hosni et al. 1999) as a follow-up to independent research by Wilkins et al. (1991) and Wilkins and McGaffin (1994). Komor (1997) found similar results. Analysis of measured data showed that results for office equipment could be generalized, but results from laboratory and hospital equipment proved too diverse. The following general guidelines for office equipment are a result of these studies.

Nameplate Versus Measured Energy Use. Nameplate data rarely reflect the actual power consumption of office equipment. Actual power consumption of such equipment is assumed equal to the total (radiant plus convective) heat gain, but the ratio of such energy to the nameplate value varies widely. ASHRAE *Research Project 1055* (Hosni et al. 1999) found that for general office equipment with nameplate power consumption of less than 1000 W, the

actual ratio of total heat gain to nameplate ranged from 25% to 50%, but when all tested equipment is considered, the range is broader. Generally, if the nameplate value is the only information known and no actual heat gain data are available for similar equipment, it would be conservative to use 50% of nameplate as heat gain and more nearly correct if 25% of nameplate were used. Much better results can be obtained, however, by considering the heat gain as being predictable based on the type of equipment.

Office equipment is grouped into categories such as computers, monitors, printers, facsimile machines, and copiers, with heat gain results within each group analyzed to establish patterns.

Computers. Based on tests by Hosni et al. (1999) and Wilkins and McGaffin (1994), nameplate values on computers should be ignored when performing cooling load calculations. [Table 8](#) presents typical heat gain values for computers with varying degrees of safety factor.

Monitors. Based on monitors tested by Hosni et al. (1999), heat gain correlates approximately with screen size as

$$q_{mon} = 0.2S - 20 \quad (12)$$

where

q_{mon} = heat gain from monitor, W
 S = nominal screen size, mm

Wilkins and McGaffin tested ten monitors (330 to 480 mm), finding the average heat gain value to be 60 W. This testing was done in 1992 when DOS was prevalent and the Windows™ operating system was just being introduced. Monitors displaying Windows consumed more power than those displaying DOS. [Table 8](#) tabulates typical values.

Laser Printers. Hosni et al. (1999) found that the power consumed by laser printers, and therefore the heat gain, depended largely on the level of throughput for which the printer was designed. It was observed that smaller printers are used more intermittently and that larger printers may run continuously for longer periods. [Table 9](#) presents data on laser printers.

These data can be applied by taking the value for continuous operation and then applying an appropriate diversity factor. This would likely be most appropriate for larger open office areas. Another approach could be to take the value that most closely matches the expected operation of the printer with no diversity. This may be appropriate when considering a single room or small area.

Copiers. Hosni et al. (1999) also tested five copy machines considered to be of two types, desktop and office (freestanding high-volume copiers). Larger machines used in production environments were not addressed. [Table 9](#) summarizes of the results. It was observed that desktop copiers rarely operated continuously but that office copiers frequently operated continuously for periods of an hour or more.

Miscellaneous Office Equipment. [Table 10](#) presents data on miscellaneous office equipment such as vending machines and mailing equipment.

Diversity. The ratio of the measured peak electrical load at the 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](#) through [10](#) can be expected to contribute heat gain to the space at the sum of the appropriate listed values. Progressively larger areas with many equipment items will always experience some degree of usage diversity resulting from whatever percentage of such equipment is not in operation at any given time.

Wilkins and McGaffin (1994) measured diversity in 23 areas within five different buildings totaling over 25 600 m². Diversity was found to range between 37 and 78%, with the average (normalized based on area) being 46%. [Figure 4](#) illustrates the relationship between nameplate, the sum of the peaks, and the actual electrical load with diversity accounted for, based on the average of the total

Table 5 Recommended Rates of Heat Gain From Typical Commercial Cooking Appliances

Appliance	Size	Energy Rate, W		Recommended Rate of Heat Gain, ^a W			
		Rated	Standby	Without Hood			With Hood
				Sensible	Latent	Total	Sensible
Electric, No Hood Required							
Barbeque (pit), per kilogram of food capacity	36 to 136 kg	88	—	57	31	88	27
Barbeque (pressurized) per kilogram of food capacity	20 kg	210	—	71	35	106	33
Blender, per litre of capacity	1.0 to 3.8 L	480	—	310	160	470	150
Braising pan, per litre of capacity	102 to 133 L	110	—	55	29	84	40
Cabinet (large hot holding)	0.46 to 0.49 m ³	2080	—	180	100	280	85
Cabinet (large hot serving)	1.06 to 1.15 m ³	2000	—	180	90	270	82
Cabinet (large proofing)	0.45 to 0.48 m ³	2030	—	180	90	270	82
Cabinet (small hot holding)	0.09 to 0.18 m ³	900	—	80	40	120	37
Cabinet (very hot holding)	0.49 m ³	6150	—	550	280	830	250
Can opener		170	—	170	—	170	0
Coffee brewer	12 cup/2 brnrs	1660	—	1100	560	1660	530
Coffee heater, per boiling burner	1 to 2 brnrs	670	—	440	230	670	210
Coffee heater, per warming burner	1 to 2 brnrs	100	—	66	34	100	32
Coffee/hot water boiling urn, per litre of capacity	11 L	120	—	79	41	120	38
Coffee brewing urn (large), per litre of capacity	22 to 38 L	660	—	440	220	660	210
Coffee brewing urn (small), per litre of capacity	10 L	420	—	280	140	420	130
Cutter (large)	460 mm bowl	750	—	750	—	750	0
Cutter (small)	360 mm bowl	370	—	370	—	370	0
Cutter and mixer (large)	28 to 45 L	3730	—	3730	—	3730	0
Dishwasher (hood type, chemical sanitizing), per 100 dishes/h	950 to 2000 dishes/h	380	—	50	110	160	50
Dishwasher (hood type, water sanitizing), per 100 dishes/h	950 to 2000 dishes/h	380	—	56	123	179	56
Dishwasher (conveyor type, chemical sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	340	—	41	97	138	44
Dishwasher (conveyor type, water sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	340	—	44	108	152	50
Display case (refrigerated), per cubic metre of interior	0.17 to 1.9 m ³	1590	—	640	0	640	0
Dough roller (large)	2 rollers	1610	—	1610	—	1610	0
Dough roller (small)	1 roller	460	—	460	—	460	0
Egg cooker	12 eggs	1800	—	850	570	1420	460
Food processor	2.3 L	520	—	520	—	520	0
Food warmer (infrared bulb), per lamp	1 to 6 bulbs	250	—	250	—	250	250
Food warmer (shelf type), per square metre of surface	0.28 to 0.84 m ³	2930	—	2330	600	2930	820
Food warmer (infrared tube), per metre of length	1.0 to 2.1 m	950	—	950	—	950	950
Food warmer (well type), per cubic metre of well	20 to 70 L	37400	—	12400	6360	18760	6000
Freezer (large)	2.07 m ³	1340	—	540	—	540	0
Freezer (small)	0.51 m ³	810	—	320	—	320	0
Griddle/grill (large), per square metre of cooking surface	0.43 to 1.1 m ²	29000	—	1940	1080	3020	1080
Griddle/grill (small), per square metre of cooking surface	0.20 to 0.42 m ²	26200	—	1720	970	2690	940
Hot dog broiler	48 to 56 hot dogs	1160	—	100	50	150	48
Hot plate (double burner, high speed)		4900	—	2290	1590	3880	1830
Hot plate (double burner stockpot)		4000	—	1870	1300	3170	1490
Hot plate (single burner, high speed)		2800	—	1310	910	2220	1040
Hot water urn (large), per litre of capacity	53 L	130	—	50	16	66	21
Hot water urn (small), per litre of capacity	7.6 L	230	—	87	30	117	37
Ice maker (large)	100 kg/day	1090	—	2730	—	2730	0
Ice maker (small)	50 kg/day	750	—	1880	—	1880	0
Microwave oven (heavy duty, commercial)	20 L	2630	—	2630	—	2630	0
Microwave oven (residential type)	30 L	600 to 1400	—	600 to 1400	—	600 to 1400	0
Mixer (large), per litre of capacity	77 L	29	—	29	—	29	0
Mixer (small), per litre of capacity	11 to 72 L	15	—	15	—	15	0
Press cooker (hamburger)	300 patties/h	2200	—	1450	750	2200	700
Refrigerator (large), per cubic metre of interior space	0.71 to 2.1 m ³	780	—	310	—	310	0
Refrigerator (small) per cubic metre of interior space	0.17 to 0.71 m ³	1730	—	690	—	690	0
Rotisserie	300 hamburgers/h	3200	—	2110	1090	3200	1020
Serving cart (hot), per cubic metre of well	50 to 90 L	21200	—	7060	3530	10590	3390
Serving drawer (large)	252 to 336 dinner rolls	1100	—	140	10	150	45
Serving drawer (small)	84 to 168 dinner rolls	800	—	100	10	110	33
Skillet (tilting), per litre of capacity	45 to 125 L	180	—	90	50	140	66
Slicer, per square metre of slicing carriage	0.06 to 0.09 m ²	2150	—	2150	—	2150	680
Soup cooker, per litre of well	7 to 11 L	130	—	45	24	69	21
Steam cooker, per cubic metre of compartment	30 to 60 L	214000	—	17000	10900	27900	8120
Steam kettle (large), per litre of capacity	76 to 300 L	95	—	7	5	12	4
Steam kettle (small), per litre of capacity	23 to 45 L	260	—	21	14	35	10
Syrup warmer, per litre of capacity	11 L	87	—	29	16	45	14

Table 5 Recommended Rates of Heat Gain From Typical Commercial Cooking Appliances (Concluded)

Appliance	Size	Energy Rate, W		Recommended Rate of Heat Gain, ^a W			
		Rated	Standby	Without Hood			With Hood
				Sensible	Latent	Total	
Toaster (bun toasts on one side only)	1400 buns/h	1500	—	800	710	1510	480
Toaster (large conveyor)	720 slices/h	3200	—	850	750	1600	510
Toaster (small conveyor)	360 slices/h	2100	—	560	490	1050	340
Toaster (large pop-up)	10 slice	5300	—	2810	2490	5300	1700
Toaster (small pop-up)	4 slice	2470	—	1310	1160	2470	790
Waffle iron	0.05 m ²	1640	—	700	940	1640	520
Electric, Exhaust Hood Required							
Broiler (conveyor infrared), per square metre of cooking area	0.19 to 9.5 m ²	60800	—	—	—	—	12100
Broiler (single deck infrared), per square metre of broiling area	0.24 to 0.91 m ²	34200	—	—	—	—	6780
Charbroiler, per linear metre of cooking surface	0.6 to 2.4 m	10600	8900	—	—	—	2700
Fryer (deep fat)	15 to 23 kg oil	14000	850	—	—	—	350
Fryer (pressurized), per kilogram of fat capacity	6 to 15 kg	1010	—	—	—	—	38
Griddle, per linear metre of cooking surface	0.6 to 2.4 m	18800	3000	—	—	—	1350
Oven (full-size convection)		12000	5000	—	—	—	850
Oven (large deck baking with 15.2 m ³ decks), per cubic metre of oven spacer	0.43 to 1.3 m ³	17300	—	—	—	—	710
Oven (roasting), per cubic metre of oven space	0.22 to 0.66 m ³	28300	—	—	—	—	1170
Oven (small convection), per cubic metre of oven space	0.04 to 0.15 m ³	107000	—	—	—	—	1520
Oven (small deck baking with 7.7 m ³ decks), per cubic metre of oven space	0.22 to 0.66 m ³	28700	—	—	—	—	1170
Open range (top), per 2 element section	2 to 10 elements	4100	1350	—	—	—	620
Range (hot top/fry top), per square metre of cooking surface	0.36 to 0.74 m ²	22900	—	—	—	—	8500
Range (oven section), per cubic metre of oven space	0.12 to 0.32 m ³	40600	—	—	—	—	1660
Gas, No Hood Required							
Broiler, per square metre of broiling area	0.25	46600	190 ^b	16800	9030	25830	3840
Cheese melter, per square metre of cooking surface	0.23 to 0.47	32500	190 ^b	11600	3400	15000	2680
Dishwasher (hood type, chemical sanitizing), per 100 dishes/h	950 to 2000 dishes/h	510	190 ^b	150	59	209	67
Dishwasher (hood type, water sanitizing), per 100 dishes/h	950 to 2000 dishes/h	510	190 ^b	170	64	234	73
Dishwasher (conveyor type, chemical sanitizing), per 100 dishes/h							
	5000 to 9000 dishes/h	400	190 ^b	97	21	118	38
Dishwasher (conveyor type, water sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	400	190 ^b	110	23	133	41
Griddle/grill (large), per square metre of cooking surface	0.43 to 1.1 m ²	53600	1040	3600	1930	5530	1450
Griddle/grill (small), per square metre of cooking surface	0.23 to 0.42 m ²	45400	1040	3050	1610	4660	1260
Hot plate	2 burners	5630	390 ^b	3430	1020	4450	1000
Oven (pizza), per square metre of hearth	0.59 to 1.2 m ²	14900	190 ^b	1970	690	2660	270
Gas, Exhaust Hood Required							
Braising pan, per litre of capacity	102 to 133 L	3050	190 ^b	—	—	—	750
Broiler, per square metre of broiling area	0.34 to 0.36 m ³	68900	1660	—	—	—	5690
Broiler (large conveyor, infrared), per square metre of cooking area/minute							
	0.19 to 9.5 m ²	162000	6270	—	—	—	16900
Broiler (standard infrared), per square metre of broiling area	0.22 to 0.87 m ²	61300	1660	—	—	—	5040
Charbroiler (large), per linear metre of cooking area	0.6 to 2.4 m	34600	21000	—	—	—	3650
Fryer (deep fat)	15 to 23 kg	23500	1640	—	—	—	560
Oven (bake deck), per cubic metre of oven space	0.15 to 0.46 m ³	79400	190 ^b	—	—	—	1450
Griddle, per linear metre of cooling surface	0.6 to 2.4 m	24000	6060	—	—	—	1540
Oven (full-size convection)		20500	8600	—	—	—	1670
Oven (pizza), per square metre of oven hearth	0.86 to 2.4 m ²	22800	190 ^b	—	—	—	410
Oven (roasting), per cubic metre of oven space	0.26 to 0.79 m ³	44500	190 ^b	—	—	—	800
Oven (twin bake deck), per cubic metre of oven space	0.31 to 0.61 m ³	45400	190 ^b	—	—	—	810
Range (burners), per 2 burner section	2 to 10 burners	9840	390	—	—	—	1930
Range (hot top or fry top), per square metre of cooking surface	0.26 to 0.74 m ³	37200	1040	—	—	—	10700
Range (large stock pot)	3 burners	29300	580	—	—	—	5740
Range (small stock pot)	2 burners	11700	390	—	—	—	2290
Range top, open burner (per 2 element section)	2 to 6 elements	11700	4000	—	—	—	640
Steam							
Compartment steamer, per kilogram of food capacity/h	21 to 204 kg	180	—	14	9	23	7
Dishwasher (hood type, chemical sanitizing), per 100 dishes/h	950 to 2000 dishes/h	920	—	260	110	370	120
Dishwasher (hood type, water sanitizing), per 100 dishes/h	950 to 2000 dishes/h	920	—	290	120	410	130
Dishwasher (conveyor, chemical sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	350	—	41	97	138	44
Dishwasher (conveyor, water sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	350	—	44	108	152	50
Steam kettle, per litre of capacity	12 to 30 L	160	—	12	8	20	6

Sources: Alreza and Breen (1984), Fisher (1998).

^aIn some cases, heat gain data are given per unit of capacity. In those cases, the heat gain is calculated by: $q = (\text{recommended heat gain per unit of capacity}) \times (\text{capacity})$

^bStandby input rating is given for entire appliance regardless of size.

Table 6 Recommended Heat Gain from Typical Medical Equipment

Equipment	Nameplate, W	Peak, W	Average, W
Anesthesia system	250	177	166
Blanket warmer	500	504	221
Blood pressure meter	180	33	29
Blood warmer	360	204	114
ECG/RESP	1440	54	50
Electrosurgery	1000	147	109
Endoscope	1688	605	596
Harmonical scalpel	230	60	59
Hysteroscopic pump	180	35	34
Laser sonics	1200	256	229
Optical microscope	330	65	63
Pulse oximeter	72	21	20
Stress treadmill	N/A	198	173
Ultrasound system	1800	1063	1050
Vacuum suction	621	337	302
X-ray system	968		82
X-ray system	1725	534	480
X-ray system	2070		18

Source: Hosni et al. (1999)

Table 7 Recommended Heat Gain from Typical Laboratory Equipment

Equipment	Nameplate, W	Peak, W	Average, W
Analytical balance	7	7	7
Centrifuge	138	89	87
Centrifuge	288	136	132
Centrifuge	5500	1176	730
Electrochemical analyzer	50	45	44
Electrochemical analyzer	100	85	84
Flame photometer	180	107	105
Fluorescent microscope	150	144	143
Fluorescent microscope	200	205	178
Function generator	58	29	29
Incubator	515	461	451
Incubator	600	479	264
Incubator	3125	1335	1222
Orbital shaker	100	16	16
Oscilloscope	72	38	38
Oscilloscope	345	99	97
Rotary evaporator	75	74	73
Rotary evaporator	94	29	28
Spectronics	36	31	31
Spectrophotometer	575	106	104
Spectrophotometer	200	122	121
Spectrophotometer	N/A	127	125
Spectro fluorometer	340	405	395
Thermocycler	1840	965	641
Thermocycler	N/A	233	198
Tissue culture	475	132	46
Tissue culture	2346	1178	1146

Source: Hosni et al. (1999)

area tested. Data on actual diversity can be used as a guide, but diversity varies significantly with occupancy. The proper diversity factor for an office of mail order catalog telephone operators is different from that for an office of sales representatives who travel regularly.

Heat Gain per Unit Area. Wilkins (1998) and Wilkins and Hosni (2000) summarized recent research on a heat gain per unit area basis. The diversity testing showed that the actual heat gain per unit area, or load factor, ranged from 4.7 to 11.6 W/m², with an average (normalized based on area) of 8.7 W/m². Spaces tested were

Table 8 Recommended Heat Gain from Typical Computer Equipment

	Continuous, W	Energy Saver Mode, W
Computers^a		
Average value	55	20
Conservative value	65	25
Highly conservative value	75	30
Monitors^b		
Small monitor (330 to 380 mm)	55	0
Medium monitor (400 to 460 mm)	70	0
Large monitor (480 to 510 mm)	80	0

Sources: Hosni et al. (1999), Wilkins and McGaffin (1994).

^aBased on 386, 486, and Pentium grade.

^bTypical values for monitors displaying Windows environment.

Table 9 Recommended Heat Gain from Typical Laser Printers and Copiers

	Continuous, W	1 page per min., W	Idle, W
Laser Printers			
Small desktop	130	75	10
Desktop	215	100	35
Small office	320	160	70
Large office	550	275	125
Copiers			
Desktop copier	400	85	20
Office copier	1,100	400	300

Source: Hosni et al. (1999).

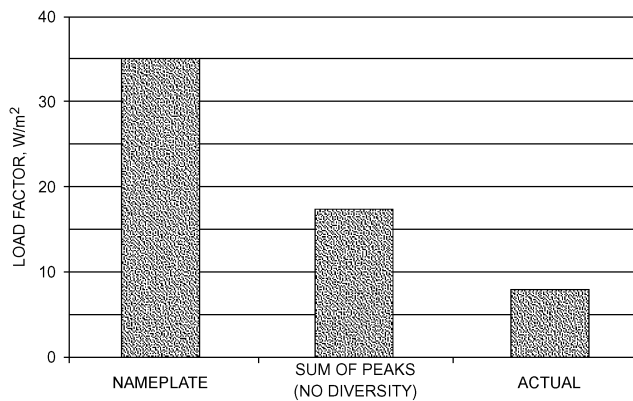
Table 10 Recommended Heat Gain from Miscellaneous Office Equipment

Appliance	Maximum Input Rating, W	Recommended Rate of Heat Gain, W
Mail-processing equipment		
Folding machine	125	80
Inserting machine, 3,600 to 6,800 pieces/h	600 to 3300	390 to 2150
Labeling machine, 1,500 to 30,000 pieces/h	600 to 6600	390 to 4300
Postage meter	230	150
Vending machines		
Cigarette	72	72
Cold food/beverage	1150 to 1920	575 to 960
Hot beverage	1725	862
Snack	240 to 275	240 to 275
Other		
Bar code printer	440	370
Cash registers	60	48
Check processing workstation, 12 pockets	4800	2470
Coffee maker, 10 cups	1500	1050 sens., 450 latent
Microfiche reader	85	85
Microfilm reader	520	520
Microfilm reader/printer	1150	1150
Microwave oven, 28 L	600	400
Paper shredder	250 to 3000	200 to 2420
Water cooler, 30 L/h	700	350

Table 11 Recommended Load Factors for Various Types of Offices

Load Density of Office	Load Factor, W/m ²	Description
Light	5.4	Assumes 15.5 m ² /workstation (6.5 workstations per 100 m ²) with computer and monitor at each plus printer and fax. Computer, monitor, and fax diversity 0.67, printer diversity 0.33.
Medium	10.8	Assumes 11.6 m ² /workstation (8.5 workstations per 100 m ²) with computer and monitor at each plus printer and fax. Computer, monitor, and fax diversity 0.75, printer diversity 0.50.
Medium/Heavy	16.1	Assumes 9.3 m ² /workstation (11 workstations per 100 m ²) with computer and monitor at each plus printer and fax. Computer and monitor diversity 0.75, printer and fax diversity 0.50.
Heavy	21.5	Assumes 7.8 m ² /workstation (13 workstations per 100 m ²) with computer and monitor at each plus printer and fax. Computer and monitor diversity 1.0, printer and fax diversity 0.50.

Source: Wilkins and McGaffin (1994).

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

fully occupied and highly automated, comprising 21 unique areas in five buildings, with a computer and monitor at every workstation. Table 11 presents a range of load factors with a subjective description of the type of space to which they would apply. Table 12 presents more specific data that can be used to better quantify the amount of equipment in a space and the expected load factor. The medium load density is likely to be appropriate for most standard office spaces. Medium/heavy or heavy load densities may be encountered but can be considered extremely conservative estimates even for densely populated and highly automated spaces.

Radiant Convective Split. Hosni et al. (1999) found that the radiant-convective split for equipment was fairly uniform, the most important differentiating feature being whether or not the equipment had a cooling fan. Table 13 is a summary of those results.

HEAT GAIN THROUGH FENESTRATION AREAS

The primary weather-related variable influencing the cooling load for a building is solar radiation. The effect of solar radiation is more pronounced and immediate in its impact on exposed non-opaque surfaces. The calculation of solar heat gain and conductive heat transfer through various glazing materials and associated mounting frames, with or without interior and/or exterior shading devices, is discussed in Chapter 30. This chapter covers the

Table 12 Cooling Load Estimates for Various Office Load Densities

	Number	Each, W	Total, W	Diversity	Load, W
Light Load Density^a					
Computers	6	55	330	0.67	220
Monitors	6	55	330	0.67	220
Laser printer—small desk top	1	130	130	0.33	43
Fax machine	1	15	15	0.67	10
Total Area Load					494
Recommended equipment load factor = 5.4 W/m ²					
Medium Load Density^a					
Computers	8	65	520	0.75	390
Monitors	8	70	560	0.75	420
Laser printer—desk	1	215	215	0.5	108
Fax machine	1	15	15	0.75	11
Total Area Load					929
Recommended equipment load factor = 10.8 W/m ²					
Medium/Heavy Load Density^a					
Computers	10	65	650	1	650
Monitors	10	70	700	1	700
Laser printer—small office	1	320	320	0.5	160
Facsimile machine	1	30	30	0.5	15
Total Area Load					1525
Recommended equipment load factor = 16.1 W/m ²					
Heavy Load Density^a					
Computers	12	75	900	1	900
Monitors	12	80	960	1	960
Laser printer—small office	1	320	320	0.5	160
Facsimile machine	1	30	30	0.5	15
Total Area Load					2035
Recommended equipment load factor = 21.5 W/m ²					

Source: Wilkins and McGaffin (1994).

^a See Table 11 for descriptions of load densities.

Table 13 Summary of Radiant-Convective Split for Office Equipment

Device	Fan	Radiant	Convective
Computer	Yes	10 to 15%	85 to 90%
Monitor	No	35 to 40%	60 to 65%
Computer and monitor	—	20 to 30%	70 to 80%
Laser printer	Yes	10 to 20%	80 to 90%
Copier	Yes	20 to 25%	75 to 80%
Fax machine	No	30 to 35%	65 to 70%

Source: Hosni et al. (1999).

application of such data to the overall heat gain evaluation and the conversion of the calculated heat gain into a composite cooling load for the conditioned space. Table 14 includes some useful solar equations.

Fenestration Direct Solar, Diffuse Solar, and Conductive Heat Gains

For fenestration heat gain, use the following equations:

Direct beam solar heat gain q_b :

$$q_b = AE_D \text{ SHGC}(\theta) \text{ IAC} \quad (13)$$

Diffuse solar heat gain q_d :

$$q_d = A(E_d + E_r)(\text{SHGC})_D \text{ IAC} \quad (14)$$

Conductive heat gain q_c :

$$q_c = UA(T_{out} - T_{in}) \quad (15)$$

Total fenestration heat gain Q :

$$Q = q_b + q_d + q_c \quad (16)$$

where

A = window area, m^2

E_D , E_d , and E_r = direct, diffuse, and ground-reflected irradiance, calculated using the equations in [Table 14](#)

$\text{SHGC}(\theta)$ = direct solar heat gain coefficient as a function of incident angle θ ; may be interpolated between values in [Table 13 of Chapter 30](#)

$(\text{SHGC})_D$ = diffuse solar heat gain coefficient (also referred to as hemispherical SHGC); from [Table 13 of Chapter 30](#)

T_{in} = inside temperature, $^{\circ}\text{C}$

T_{out} = outside temperature, $^{\circ}\text{C}$

U = overall U-factor, including frame and mounting orientation from [Table 4 of Chapter 30](#), $\text{W}/(\text{m}^2 \cdot \text{K})$

IAC = inside shading attenuation coefficient, = 1.0 if no inside shading device

If specific window manufacturer's SHGC and U-factor data are available, those should be used. For fenestration equipped with inside shading (blinds, drapes or shades), IAC is listed in [Tables 19](#),

[20](#), and [22 of Chapter 30](#). The inside shading attenuation coefficients given are used to calculate both direct and diffuse solar heat gains.

Exterior Shading

Nonuniform exterior shading, caused by roof overhangs, side fins, or building projections, requires separate hourly calculations for the externally shaded and unshaded areas of the window in question, with the inside shading SHGC still used to account for any internal shading devices. 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 fundamental algorithms for analysis of shade patterns. McQuiston and Spitler (1992) provide graphical data to facilitate shadow line calculation.

An alternative, more accurate method described by Todorovic et al. (1993) first calculates cooling loads as if the external shading were absent, then adjusts (reduces) the result to account for the shading effect. This correction applies a "negative cooling load factor," calculated in much the same way as a conventional cooling load but using the time-varying area of the shaded portion of the glass as the heat gain element. Todorovic (1987) describes the solution of the moving shade line problem in the context of consequent cooling load.

The equations for calculating shade angles [[Chapter 30, Equations \(107\) to \(110\)](#)] can be used to determine the shape and area of moving shadow falling across a given window from external shading elements during the course of a design day. Thus, a subprofile of heat gain for that window can be created by separating its sunlit and shaded areas for each hour.

Table 14 Solar Equations

Solar Angles

All angles are in degrees. The solar azimuth ϕ and the surface azimuth ψ are measured in degrees from south; angles to the east of south are negative, and angles to the west of south are positive. Calculate solar altitude, azimuth, and surface incident angles as follows:

Apparent solar time AST, in decimal hours:

$$\text{AST} = \text{LST} + \text{ET}/60 + (\text{LSM} - \text{LON})/15$$

Hour angle H , degrees:

$$H = 15(\text{hours of time from local solar noon}) = 15(\text{AST} - 12)$$

Solar altitude β :

$$\sin \beta = \cos L \cos \delta \cos H + \sin L \sin \delta$$

Solar azimuth ϕ :

$$\cos \phi = (\sin \beta \sin L - \sin \delta) / (\cos \beta \cos L)$$

Surface-solar azimuth γ :

$$\gamma = \phi - \psi$$

Incident angle θ :

$$\cos \theta = \cos \beta \cos \gamma \sin \Sigma + \sin \beta \cos \Sigma$$

where

ET = equation of time, decimal minutes

L = latitude

LON = local longitude, decimal degrees of arc

LSM = local standard time meridian, decimal degrees of arc

= 60° for Atlantic Standard Time

= 75° for Eastern Standard Time

= 90° for Central Standard Time

= 105° for Mountain Standard Time

= 120° for Pacific Standard Time

= 135° for Alaska Standard Time

= 150° for Hawaii-Aleutian Standard Time

LST = local standard time, decimal hours

δ = solar declination, $^{\circ}$

ψ = surface azimuth, $^{\circ}$

Σ = surface tilt from horizontal, horizontal = 0°

Values of ET and δ are given in [Table 7 of Chapter 30](#) for the 21st day of each month.

Direct, Diffuse, and Total Solar Irradiance

Direct normal irradiance E_{DN}

$$\text{If } \beta > 0 \quad E_{DN} = \left[\frac{A}{\exp(B/\sin \beta)} \right] \text{CN}$$

Otherwise, $E_{DN} = 0$

Surface direct irradiance E_D

If $\cos \theta > 0$ $E_D = E_{DN} \cos \theta$

Otherwise, $E_D = 0$

Ratio Y of sky diffuse on vertical surface to sky diffuse on horizontal surface

If $\cos \theta > -0.2$ $Y = 0.55 + 0.437 \cos \theta + 0.313 \cos^2 \theta$

Otherwise, $Y = 0.45$

Diffuse irradiance E_d

Vertical surfaces $E_d = CYE_{DN}$

Surfaces other than vertical $E_d = CE_{DN}(1 + \cos \Sigma)/2$

Ground-reflected irradiance $E_r = E_{DN}(C + \sin \beta)\rho_g(1 - \cos \Sigma)/2$

Total surface irradiance $E_t = E_D + E_d + E_r$

where

A = apparent solar constant

B = atmospheric extinction coefficient

C = sky diffuse factor

CN = clearness number multiplier for clear/dry or hazy/humid locations. See [Figure 5 in Chapter 33 of the ASHRAE Handbook—Applications](#) for CN values.

E_d = diffuse sky irradiance

E_r = diffuse ground-reflected irradiance

ρ_g = ground reflectivity

Values of A , B , and C are given in [Table 7 of Chapter 30](#) for the 21st day of each month. Values of ground reflectivity ρ_g are given in [Table 10 of Chapter 30](#).

Temperature Considerations

To estimate the conductive heat gain through fenestration at any time, applicable values of the outdoor and indoor dry-bulb temperatures must be used. [Chapter 27](#) gives monthly cooling load design values of outdoor dry-bulb temperatures for many locations. These are generally midafternoon temperatures; for other times, the method described below in the section on Outdoor Air Temperatures can be used to estimate temperatures. Where local microclimatic conditions prevail or data are not included in [Chapter 27](#), local weather stations or the National Oceanic and Atmospheric Administration can supply temperature data. Winter design temperatures should not be used because such data are for heating design rather than coincident conductive heat gain with sunlit glass during the winter months.

HEAT GAIN THROUGH EXTERIOR SURFACES

Heat gain through exterior opaque surfaces is derived from the same elements of solar radiation and thermal gradient as that for fenestration areas. It differs primarily as a function of the mass and nature of the wall or roof construction, since those elements affect the rate of conductive heat transfer through the composite assembly to the interior surface.

SOL-AIR TEMPERATURE

Sol-air temperature is the temperature of the outdoor air that in the absence of all radiation changes gives the same rate of heat entry into the surface as would the combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air.

Heat Flux into Exterior Sunlit Surfaces

The heat balance at a sunlit surface gives the heat flux into the surface q/A as

$$\frac{q}{A} = \alpha E_t + h_o(t_o - t_s) - \varepsilon \Delta R \quad (17)$$

where

- α = absorptance of surface for solar radiation
- E_t = total solar radiation incident on surface, $\text{W}/(\text{m}^2 \cdot \text{K})$
- h_o = coefficient of heat transfer by long-wave radiation and convection at outer surface, $\text{W}/(\text{m}^2 \cdot \text{K})$
- t_o = outdoor air temperature, $^{\circ}\text{C}$
- t_s = surface temperature, $^{\circ}\text{C}$
- ε = hemispherical emittance of surface
- ΔR = difference between long-wave radiation incident on surface from sky and surroundings and radiation emitted by blackbody at outdoor air temperature, W/m^2

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

$$\frac{q}{A} = h_o(t_e - t_s) \quad (18)$$

and from Equations (17) and (18),

$$t_e = t_o + \frac{\alpha E_t}{h_o} - \frac{\varepsilon \Delta R}{h_o} \quad (19)$$

Horizontal Surfaces. For horizontal surfaces that receive long-wave radiation from the sky only, an appropriate value of ΔR is about $63 \text{ W}/\text{m}^2$, so that if $\varepsilon = 1$ and $h_o = 17 \text{ W}/(\text{m}^2 \cdot \text{K})$, the long-wave correction term is about 4 K (Bliss 1961) and the correction itself thus -4 K .

Vertical Surfaces. 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 $\varepsilon \Delta R = 0$ for vertical surfaces.

Tabulated Temperature Values

The sol-air temperatures in [Table 15](#) have been calculated based on $\varepsilon \Delta R/h_o$ values of 4 K for horizontal surfaces and 0 K for vertical surfaces; total solar intensity values used for the calculations were calculated using the equations included in [Table 14](#).

Surface Colors

Sol-air temperature values are given for two values of the parameter α/h_o ([Table 15](#)); the value of 0.026 is appropriate for a light-colored surface, while 0.052 represents the usual maximum value for this parameter (i.e., for a dark-colored surface or any surface for which the permanent lightness can not reliably be anticipated).

Example 1. Calculating sol-air temperature using solar equations. Calculate the sol-air temperature at 3 P.M., Central Daylight Time on July 21 for a vertical light-colored wall surface, facing southwest, located at 40° North latitude and 90° West longitude, with an outdoor temperature of 34.4°C . The clearness number CN is assumed to be 1.0 and ground reflectivity $\rho_g = 0.2$.

Solution: Sol-air temperature is calculated using Equation (19). For a light-colored wall, $\alpha/h_o = 0.026$, and for vertical surfaces, $\varepsilon \Delta R/h_o = 0$. The solar irradiance E_t must be determined using the equations in [Table 14](#).

Solar Angles:

ψ = southwest orientation = $+45^{\circ}$. Σ = surface tilt from horizontal (where horizontal = 0°) = 90° for vertical wall surface. Local solar time (LST) is one hour earlier than Daylight Time, so 3 P.M. DST = 2 P.M. LST = hour 14. Calculate solar altitude, solar azimuth, surface solar azimuth, and incident angle as follows.

From [Table 7 in Chapter 30](#), solar position data and constants for July 21 are

$$\begin{aligned} ET &= -6.2 \text{ min} \\ \delta &= 20.6^{\circ} \\ A &= 1085 \text{ W}/\text{m}^2 \\ B &= 0.207 \\ C &= 0.136 \end{aligned}$$

Local standard meridian (LSM) for Central Time Zone = 90° .
Apparent solar time AST

$$\begin{aligned} \text{AST} &= \text{LST} + ET/60 + (\text{LSM} - \text{LON})/15 \\ &= 14 + (-6.2/60) + [(90 - 90)/15] \\ &= 13.897 \end{aligned}$$

Hour angle H , degrees:

$$\begin{aligned} H &= 15(\text{AST} - 12) \\ &= 15(13.897 - 12) \\ &= 28.45^{\circ} \end{aligned}$$

Solar altitude β :

$$\begin{aligned} \sin \beta &= \cos L \cos \delta \cos H + \sin L \sin \delta \\ &= \cos(40) \cos(20.6) \cos(28.45) + \sin(40) \sin(20.6) \\ &= 0.8566 \\ \beta &= \sin^{-1}(0.8566) = 58.9^{\circ} \end{aligned}$$

Solar azimuth ϕ :

$$\begin{aligned} \cos \phi &= (\sin \beta \sin L - \sin \delta) / (\cos \beta \cos L) \\ &= [(\sin(58.9) \sin(40) - \sin(20.6)) / [\cos(58.9) \cos(40)]] \\ &= 0.502 \\ \phi &= \cos^{-1}(0.502) = 59.9^{\circ} \end{aligned}$$

Surface-solar azimuth γ :

$$\begin{aligned} \gamma &= \phi - \psi \\ &= 59.9 - 45 \\ &= 14.9^{\circ} \end{aligned}$$

Table 15 Sol-Air Temperatures for July 21, 40° North Latitude

Local Standard Hour	Air Temp. $t_o, ^\circ\text{C}$	Light-Colored Surface, $\alpha/h_o = 0.026$										Local Standard Hour	Air Temp. $t_o, ^\circ\text{C}$	Dark-Colored Surface, $\alpha/h_o = 0.052$									
		N	NE	E	SE	S	SW	W	NW	Horiz.	N			NE	E	SE	S	SW	W	NW	Horiz.		
1	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	20.6	1	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	20.6		
2	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	20.6	2	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	20.6		
3	23.9	23.9	23.9	23.9	23.9	23.9	23.9	23.9	23.9	20.0	3	23.9	23.9	23.9	23.9	23.9	23.9	23.9	23.9	23.9	20.0		
4	23.3	23.3	23.3	23.3	23.3	23.3	23.3	23.3	23.3	19.4	4	23.3	23.3	23.3	23.3	23.3	23.3	23.3	23.3	23.3	19.4		
5	23.3	23.3	23.3	23.3	23.3	23.3	23.3	23.3	23.3	19.4	5	23.3	23.3	23.3	23.3	23.3	23.3	23.3	23.3	23.3	19.4		
6	23.3	27.7	34.5	35.3	29.7	24.3	24.3	24.3	24.3	23.1	6	23.3	32.1	45.6	47.3	36.1	25.3	25.3	25.3	25.3	23.1		
7	23.9	28.4	39.6	43.0	36.2	25.9	25.8	25.8	25.8	29.0	7	23.9	32.9	55.4	62.1	48.5	28.0	27.8	27.8	27.8	29.0		
8	25.0	27.9	40.2	45.8	40.7	28.3	27.6	27.6	27.6	35.1	8	25.0	30.7	55.4	66.6	56.4	31.5	30.2	30.2	30.2	35.1		
9	26.7	29.9	38.9	46.0	43.7	33.4	29.8	29.8	29.8	41.2	9	26.7	33.2	51.1	65.4	60.7	40.2	33.0	33.0	33.0	41.2		
10	28.3	31.9	36.4	44.0	44.7	38.0	31.9	31.9	31.9	46.2	10	28.3	35.4	44.4	59.7	61.1	47.7	35.6	35.4	35.4	46.2		
11	30.6	34.4	34.6	41.2	44.6	42.2	35.3	34.4	34.4	50.7	11	30.6	38.2	38.7	51.8	58.7	53.8	40.2	38.2	38.2	50.7		
12	32.2	36.1	36.1	36.4	41.8	44.6	42.5	37.0	36.1	53.2	12	32.2	40.0	40.0	40.6	51.3	57.0	53.1	41.9	40.0	53.2		
13	33.9	37.7	37.7	37.7	39.7	45.8	47.1	43.4	37.9	54.3	13	33.9	41.5	41.5	41.5	45.5	57.7	60.8	52.8	42.0	54.3		
14	34.4	38.1	38.1	38.1	38.2	44.6	50.2	49.2	41.5	52.9	14	34.4	41.7	41.7	41.7	42.0	54.8	66.6	63.9	48.6	52.9		
15	35.0	38.4	38.2	38.2	38.2	42.4	51.8	53.7	46.4	50.3	15	35.0	41.7	41.5	41.5	41.5	49.8	69.1	72.5	57.8	50.3		
16	34.4	37.4	37.2	37.2	37.2	38.4	50.4	55.2	49.2	45.5	16	34.4	40.4	39.9	39.9	39.9	42.4	66.8	75.9	63.9	45.5		
17	33.9	38.1	36.0	36.0	36.0	36.1	46.9	53.7	49.8	40.0	17	33.9	42.2	38.1	38.1	38.1	38.4	60.2	73.5	65.7	40.0		
18	32.8	37.5	34.0	34.0	34.0	34.0	40.5	46.9	45.5	33.6	18	32.8	42.3	35.2	35.2	35.2	35.2	48.4	60.9	58.3	33.6		
19	30.6	31.0	30.6	30.6	30.6	30.6	31.0	31.5	31.5	26.8	19	30.6	31.5	30.7	30.7	30.7	30.7	31.4	32.5	32.5	26.8		
20	29.4	29.4	29.4	29.4	29.4	29.4	29.4	29.4	29.4	25.6	20	29.4	29.4	29.4	29.4	29.4	29.4	29.4	29.4	29.4	25.6		
21	28.3	28.3	28.3	28.3	28.3	28.3	28.3	28.3	28.3	24.4	21	28.3	28.3	28.3	28.3	28.3	28.3	28.3	28.3	28.3	24.4		
22	27.2	27.2	27.2	27.2	27.2	27.2	27.2	27.2	27.2	23.3	22	27.2	27.2	27.2	27.2	27.2	27.2	27.2	27.2	27.2	23.3		
23	26.1	26.1	26.1	26.1	26.1	26.1	26.1	26.1	26.1	22.2	23	26.1	26.1	26.1	26.1	26.1	26.1	26.1	26.1	26.1	22.2		
24	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	21.1	24	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	21.1		

Incident angle θ :

$$\begin{aligned}\cos\theta &= \cos\beta\cos\gamma\sin\Sigma + \sin\beta\cos\Sigma \\ &= \cos(58.9)\cos(14.9)\sin(90) + \sin(58.9)\cos(90) \\ &= 0.499 \\ \theta &= \cos^{-1}(0.499) = 60.1^\circ\end{aligned}$$

Direct, Diffuse, and Total Solar Irradiance:Direct normal irradiance E_{DN}

$$\begin{aligned}E_{DN} &= \left[\frac{A}{\exp(B/\sin\beta)} \right] \text{CN} \\ &= \frac{1085}{\exp[0.207/\sin(58.9)]} (1.0) \\ &= 852 \text{ W/m}^2\end{aligned}$$

Surface direct irradiance E_D

$$\begin{aligned}E_D &= E_{DN}\cos\theta \\ &= (852)\cos(60.1) \\ &= 425 \text{ W/m}^2\end{aligned}$$

Ratio Y of sky diffuse on vertical surface to sky diffuse on horizontal surface

$$\begin{aligned}Y &= 0.55 + 0.437\cos\theta + 0.313\cos^2\theta \\ &= 0.55 + 0.437\cos(60.1) + 0.313\cos^2(60.1) \\ &= 0.846\end{aligned}$$

Diffuse irradiance E_d

Vertical surfaces

$$\begin{aligned}E_d &= CYE_{DN} \\ &= (0.136)(0.846)(852) \\ &= 98 \text{ W/m}^2\end{aligned}$$

Ground-reflected irradiance E_r

$$\begin{aligned}E_r &= E_{DN}(C + \sin\beta)\rho_g(1 - \cos\Sigma)/2 \\ &= (852)[0.136 + \sin(58.9)](0.2)[1 - \cos(90)]/2 \\ &= 85 \text{ W/m}^2\end{aligned}$$

Total surface irradiance E_t

$$\begin{aligned}E_t &= E_D + E_d + E_r \\ &= 425 + 98 + 85 \\ &= 608 \text{ W/m}^2\end{aligned}$$

Sol-air temperature [from Equation (19)]:

$$\begin{aligned}t_e &= t_o + \alpha E_t/h_o - \varepsilon\Delta R/h_o \\ &= 34.4 + (0.026)(608) - 0 \\ &= 50.2^\circ\text{C}\end{aligned}$$

This procedure was used to calculate the sol-air temperatures included in [Table 15](#). Due to the tedious solar angle and intensity calculations, use of a simple computer spreadsheet or other computer software implementing these calculations can reduce the effort involved. A spreadsheet is illustrated in [Table 16](#), calculating a 24 h sol-air temperature profile for the data of this example.

OUTDOOR AIR TEMPERATURES

The hourly air temperatures in Column 2, [Table 16](#), are for a location with a design temperature of 35°C and a range of 11.7 K. To compute corresponding temperatures for other locations, select a suitable design temperature from [Table 4B of Chapter 27](#) for the month being calculated and note the outdoor daily range from [Table 1B of Chapter 27](#). For each hour, take the percentage of the daily range indicated in [Table 17](#) of this chapter and subtract it from the design temperature.

Example 2. Air temperature calculation. Calculate the July dry-bulb temperature at 1200 h for Reno, Nevada.

Solution: From [Table 1B, Chapter 27](#), the daily range is 20.7 K, and from [Table 4B, Chapter 27](#), the 1% design dry-bulb temperature for July is 36°C. From [Table 17](#), the percentage of the daily range at 1200 h is 23%. Thus, the dry-bulb temperature at 1200 is $36 - (0.23 \times 20.7) = 31.2^\circ\text{C}$.

Table 16 Solar Calculations—Sol-Air Temperature for Example 1

GENERAL INPUT:		SURFACE INPUT:		SOLAR CONSTANTS ^a						TIME ZONE MERIDIANS			
Time Zone	= 3	Azimuth = 45		SOL-AIR TEMP. INPUT: Absorptance = 0.45 h (outside) = 17.3 Emittance = 1 ΔR = 0	Month	Equation of Time, min.	Declination, degrees	A	B	C	Time Zone Standard Meridian 1 Atlantic60 2 Eastern75 3 Central90 4 Mountain105 5 Pacific120 6 Alaska135 7 Hawaii150		
Month	= 7	Tilt = 90											
Longitude	= 90												
Latitude	= 40												
Clearness	= 1												
Ground Refl.	= 0.2												

HEAT GAIN THROUGH INTERIOR SURFACES

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

$$q = UA(t_b - t_i) \quad (20)$$

where

q = heat transfer rate, W

U = coefficient of overall heat transfer between adjacent and conditioned space, $W/(m^2 \cdot K)$

A = area of separating section concerned, m^2

t_b = average air temperature in adjacent space, $^{\circ}C$

t_i = air temperature in conditioned space, $^{\circ}C$

Values of U can be obtained from [Chapter 25](#). Temperature t_b may differ greatly from t_i . The temperature in a kitchen or boiler room, for example, may be as much as 8 to 28 K above the outdoor air temperature. Actual temperatures in adjoining spaces should be measured when possible. Where nothing is known except that the adjacent space is of conventional construction, contains no heat sources, and itself receives no significant solar heat gain, $t_b - t_i$ may be considered to be the difference between the outdoor air and conditioned space design dry-bulb temperatures minus 3 K. In some cases, the air temperature in the adjacent space will correspond to the outdoor air temperature or higher.

Floors

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

INFILTRATION AND VENTILATION HEAT GAIN

Ventilation

Outdoor air must be introduced to ventilate conditioned spaces. [Chapter 26](#) suggests minimum outdoor air requirements for representative applications, but the minimum levels are not necessarily adequate for all psychological attitudes and physiological responses. Where maximum economy in space and load is essential, as in submarines or other restricted spaces, as little as 0.5 L/s of outdoor air per person can be sufficient, provided that recirculated air is adequately decontaminated (Consolazio and Pecora 1947).

Local codes and ordinances frequently specify ventilation requirements for public places and for industrial installations. For example, minimum requirements for safe practice in hospital operating rooms are given in NFPA *Standard 99*. Although 100% outdoor air is sometimes used in operating rooms, this standard does not require it, and limiting the outdoor air to 6 to 8 changes per hour is finding increasing acceptance.

ASHRAE *Standard 62* recommends minimum ventilation rates for most common applications. For general applications, such as offices, 10 L/s per person is suggested.

Ventilation air is normally introduced at the air-conditioning apparatus rather than directly into the conditioned space and thus becomes a cooling coil load component instead of a space load component. Calculations for estimating this heat gain are discussed later in the section on Heat Gain Calculations Using Standard Air Values.

Reducing heat gain from outdoor air by using filtered recirculated air in combination with outdoor air should be considered. Recirculated air can also be treated to control odor (see [Chapter 13](#) of this volume and [Chapter 45 of the ASHRAE Handbook—Applications](#)).

Infiltration

The principles of estimating infiltration in buildings, with emphasis on the heating season, are discussed in [Chapter 26](#). For

the cooling season, infiltration calculations are usually limited to doors and windows. Air leakage through doors can be estimated using the information in [Chapter 26, Table 3 in Chapter 26](#), adjusted for the average wind velocity in the locality, may be used to compute infiltration for windows. In calculating window infiltration for an entire structure, the total window area on all sides of the building is not involved since wind does not act on all sides simultaneously. In any case, infiltration from all windows in any two adjacent wall exposures should be included. A knowledge of the prevailing wind direction and velocity is helpful in selecting exposures.

When economically feasible, sufficient outdoor air should be introduced as ventilation air through the air-conditioning equipment to maintain a constant outward escape of air and thus eliminate the infiltration portion of the gain. The pressure maintained must overcome wind pressure through cracks and door openings. When the quantity of outside air introduced through the cooling equipment is not sufficient to maintain the required pressure to eliminate infiltration, the entire infiltration load should be included in the space heat gain calculations.

Standard Air Volumes

Because the specific volume of air varies appreciably, calculations are more accurate when made on the basis of air mass instead of volume. However, volume values are often required for selection of coils, fans, ducts, etc., in which cases volume values based on measurement at standard conditions may be used for accurate results. One standard value is 1.2 kg (dry air)/ m^3 (0.833 m^3/kg). This density corresponds to about 16°C at saturation and 21°C dry air (at 101.325 kPa). Because air usually passes through the coils, fans, ducts, etc., at a density close to standard, the accuracy desired normally requires no correction. When airflow is to be measured at a particular condition or point, such as at a coil entrance or exit, the corresponding specific volume can be read from the psychrometric chart.

Example 3. Standard air calculation. Assume outdoor air at standard conditions is flowing at 10 m^3/s . What is the flow rate when the outdoor air is at 35°C dry-bulb and 24°C wet-bulb (0.893 m^3/kg)?

Solution: The measured rate at that condition should be 10 (0.893/0.833) = 10.7 m^3/s .

Heat Gain Calculations Using Standard Air Values

Air-conditioning design often requires calculation of the following.

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

$$q_t = 1.2 Q_s \Delta h \quad (21)$$

where air density = 1.2 kg/m^3 .

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

$$q_s = 1.2(1.006 + 1.84W)Q_s \Delta t \quad (22)$$

where

1.006 = specific heat of dry air, $kJ/(kg \cdot K)$

W = humidity ratio, $kg (water)/kg (air)$

1.84 = specific heat of water vapor, $kJ/(kg \cdot K)$

The specific heats are for a range from about -75 to $90^{\circ}C$. When $W = 0$, the value of $1.20(1.006 + 1.84W) = 1.21$; when $W = 0.01$, the value is 1.23; when $W = 0.02$, the value is 1.25; and when $W = 0.03$,

the value is 1.27. Because a value of $W = 0.01$ approximates conditions found in many air-conditioning problems, the sensible heat change (in W) can normally be found as

$$q_s = 1.23 Q_s \Delta t \quad (23)$$

3. Latent heat

Latent heat gain q_l corresponding to the change of humidity ratio ΔW for given airflow (standard conditions) Q_s is

$$q_l = 1.20 \times 2500 Q_s \Delta W = 3010 Q_s \Delta W \quad (24)$$

where 2500 is the approximate heat content of 50% rh vapor at 24°C less the heat content of water at 10°C. 50% rh at 24°C is a common design condition for the space, and 10°C is normal condensate temperature from cooling and dehumidifying coils.

The constants 1.20, 1.23, and 3010 are useful in air-conditioning calculations at sea level (101.325 kPa) and for normal temperatures and moisture ratios. For other conditions, more precise values should be used. For an altitude of 1500 m (84.556 kPa), appropriate values are 1.00, 1.03, and 2500.

LATENT HEAT GAIN FROM MOISTURE THROUGH PERMEABLE BUILDING MATERIALS

The diffusion of moisture through all common building materials is a natural phenomenon that is always present. [Chapters 23 and 24](#) cover the principles and specific methods used to control moisture. Moisture transfer through walls is often neglected in the usual comfort air-conditioning application because the actual rate is quite small and the corresponding latent heat gain is insignificant. The permeability and permeance values for various building materials are given in [Table 9, Chapter 25](#). Vapor retarders are frequently installed to keep moisture transfer to a minimum.

Certain industrial applications call for a low moisture content to be maintained in a conditioned space. In such cases, the latent heat gain accompanying moisture transfer through walls may be greater than any other latent heat gain. This gain is computed by

$$q_m = MA \Delta p_v (h_g - h_f) \quad (25)$$

where

- q_m = latent heat gain, W
- M = permeance of wall assembly, $\text{ng}/(\text{s} \cdot \text{m}^2 \cdot \text{Pa})$
- A = area of wall surface, m^2
- Δp_v = vapor pressure difference, Pa
- h_g = enthalpy at room conditions, kJ/kg
- h_f = enthalpy of water condensed at cooling coil, kJ/kg
= 2500 kJ/kg when room temperature is 24°C and condensate off coil is 10°C

HEAT GAIN FROM MISCELLANEOUS SOURCES

The calculation of the cooling load is affected by such factors as (1) type of HVAC system, (2) effectiveness of heat exchange surfaces, (3) fan location, (4) duct heat gain or loss, (5) duct leakage, (6) heat-extraction lighting systems, (7) type of return air system, and (8) sequence of controls. System performance needs to be analyzed as a sequence of individual psychrometric processes. The most straightforward method first defines all known (or desired) state points on a psychrometric chart. Next, the actual entering and leaving dry- and wet-bulb conditions are calculated for such components as the cooling and/or heating coils (based on zone or space load), the amount of outside air introduced into the system through the equipment, and the amount of heat gain or loss at various points.

This overall process must verify that the space conditions originally sought can actually be met by the designed system by considering all sensible and latent heat changes to the air as it travels from

the space conditions through the return air system and equipment back to the conditioned space. If the design is successful (i.e., within the degree of correctness of the various design assumptions), appropriate equipment components can safely be selected. If not, the designer must judge if the results will be “close enough” to satisfy the needs of the project, or if one or more assumptions and/or design criteria must first be modified and the calculations rerun.

Heat Gain from Fans

Fans that circulate air through HVAC systems add energy to the system by one or all of the following processes:

- Temperature rise in the airstream from fan inefficiency. Depending on the equipment, fan efficiencies generally range between 50 and 70%, with an average value of 65%. Thus, some 35% of the energy required by the fan appears as instantaneous heat gain to the air being transported.
- Temperature rise in the airstream as a consequence of air static and velocity pressure. The “useful” 65% of the total fan energy that creates pressure to move air spreads out throughout the entire air transport system in the process of conversion to sensible heat. Designers commonly assume that the temperature change equivalent of this heat occurs at a single point in the system, depending on fan location as noted below.
- Temperature rise from heat generated by motor and drive inefficiencies. The relatively small gains from fan motors and drives are normally disregarded unless the motor and/or drive are physically located within the conditioned airstream. Equations (7), (8), and (9) may be used to estimate heat gains from typical motors. Belt drive losses are often estimated as 3% of the motor power rating. Conversion to temperature rise is calculated by Equation (26).

The location of each fan relative to other elements (primarily the cooling coil) and the type of system (e.g., single zone, multizone, double-duct, terminal reheat, VAV), along with the concept of equipment control (space temperature alone, space temperature and relative humidity, etc.), must be known before the analysis can be completed. A fan located upstream of the cooling coil (blowthrough supply fan, return air fan, outside air fan) adds the heat equivalent of its inefficiency to the airstream at that point; thus, a slightly elevated entering dry-bulb temperature to the cooling coil results. A fan located downstream of the cooling coil raises the dry-bulb temperature of air leaving the cooling coil. This rise can be offset by reducing the cooling coil temperature or, alternatively, by increasing airflow across the cooling coil as long as its impact on space conditions is considered.

Duct Heat Gain and Leakage

Unless return air duct systems are extensive or subjected to rigorous conditions, only the heat gained or lost by supply duct systems is significant; it is normally estimated as a percentage of space sensible cooling load (usually about 1%) and applied to the dry-bulb temperature of the air leaving the coil in the form of an equivalent temperature reduction.

Air leakage out of (or into) ductwork can have much greater impact than conventional duct heat gain or loss, but it is normally about the same or less. Outward leakage from supply ducts is a direct loss of cooling and/or dehumidifying capacity and must be offset by increased airflow (sometimes reduced supply air temperatures) unless it enters the conditioned space directly. Inward leakage to return ducts causes temperature and/or humidity variations, but these are often ignored under ordinary circumstances due to the low temperature and pressure differentials involved. [Chapter 34](#) has further details on duct sealing and leakage.

A well-designed and installed duct system should not leak more than 1 to 3% of the total system airflow. All HVAC equipment and

volume control units connected into a duct system are usually delivered from manufacturers with allowable leakage not exceeding 1 or 2% of maximum airflow rating. Where duct systems are specified to be sealed and leak tested, both low- and medium-pressure types can be constructed and required to fall within this range. Designers normally assume this loss to approximate 1% of the space load, handled in a similar manner to that for duct heat gain. Latent heat considerations are frequently ignored.

Poorly designed or installed duct systems can have leakage rates of 10 to 30%. Leakage from low-pressure lighting troffer connections lacking proper taping and sealing can be 35% or more of the terminal air supply. Improperly sealed high-pressure systems can leak as much as 10% or more from the high-pressure side alone. Such extremes destroy the validity of any load calculation. Although leaks do not always affect overall system loads enough to cause problems, they will always adversely impact required supply air quantities. Also, uninsulated supply ductwork running through return air plenums results in high thermal leakage, which reduces the space-cooling capability of the supply air and may cause condensation during a warm startup.

HEAT BALANCE METHOD OF COOLING LOAD CALCULATION

The estimation of cooling load for a space 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. Sometimes called the **exact solution**, these principles form the foundation for all methods described in this chapter.

Some of the computations required by this rigorous approach to calculating space cooling load make the use of modern digital computers essential. 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 NBSLD (Kusuda 1967). The heat balance procedure is also implemented in both the BLAST and TARP energy analysis programs (Walton 1983). Prior to the implementation of ASHRAE *Research Project 875*, the method had never been described completely or in a form applicable to cooling load calculations. The papers resulting from RP-875 describe the heat balance procedure in detail (Pedersen et al. 1997, Liesen and Pedersen 1997, McClellan and Pedersen 1997).

The HB method is codified in the software called Hbfort that accompanies *Cooling and Heating Load Calculation Principles* (Pedersen et al. 1998).

HEAT BALANCE MODEL ASSUMPTIONS

All calculation procedures involve some kind of model. All models require simplifying assumptions and therefore are approximate. The most fundamental assumption is that the air in the thermal zone can be modeled as **well mixed**, meaning it has a uniform temperature throughout the zone. ASHRAE *Research Project 664* (Fisher et al. 1997) established that this assumption is valid over a wide range of conditions.

The next major assumption is that the surfaces of the room (walls, windows, floor, etc.) can be treated as having

- Uniform surface temperatures
- Uniform long-wave (LW) and short-wave (SW) irradiation
- Diffuse radiating surfaces
- One-dimensional heat conduction within

The resulting formulation is called the **heat balance model**. It is important to note that the foregoing assumptions, although common, are quite restrictive and set certain limits on the information that can be obtained from the model.

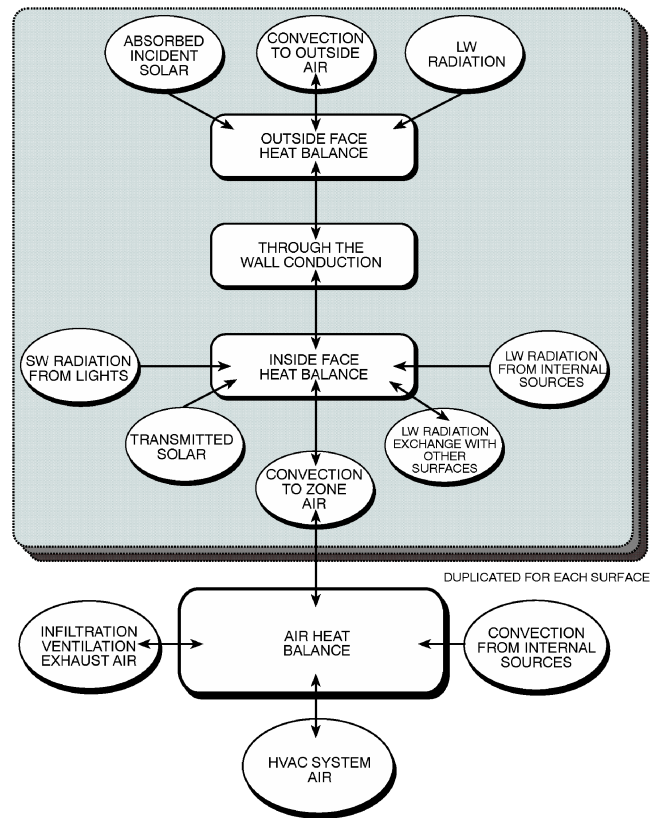


Fig. 5 Schematic of Heat Balance Processes in a Zone

ELEMENTS OF HEAT BALANCE MODEL

Within the framework of the foregoing assumptions, the heat balance model can be viewed as four distinct processes:

1. Outside face heat balance
2. Wall conduction process
3. Inside face heat balance
4. Air heat balance

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 of the surfaces enclosing the zone. The process for transparent surfaces would be similar but would have the absorbed solar component appear in the conduction process block instead of at the outside face. Also, the absorbed component would split into an inward-flowing fraction and an outward-flowing fraction. These components would participate in the surface heat balances.

Outside Face Heat Balance

The heat balance on the outside face of each surface is

$$q''_{asol} + q''_{LWR} + q''_{conv} - q''_{ko} = 0 \quad (26)$$

where

q''_{asol} = absorbed direct and diffuse solar radiation flux (q/A), W/m^2

q''_{LWR} = net long-wave radiation flux exchange with air and surroundings, W/m^2

q''_{conv} = convective exchange flux with outside air, W/m^2

q''_{ko} = conductive flux (q/A) into wall, W/m^2

All terms are positive for net flux to the face except the conduction term, which is traditionally taken to be positive in the direction from outside to inside the wall.

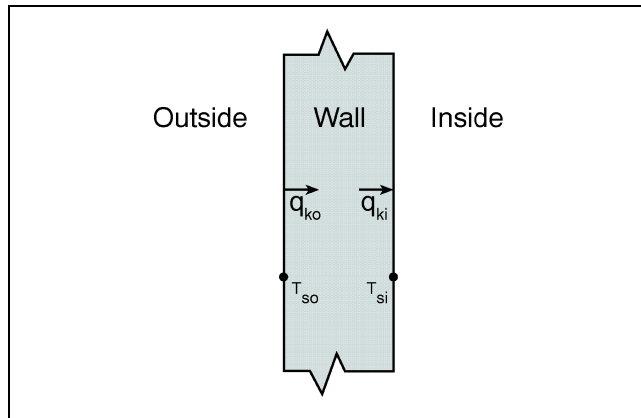


Fig. 6 Schematic of Wall Conduction Process

Each of the heat flux terms in Equation (26) has been modeled in several ways, and in some formulations the first three terms are combined by using the sol-air temperature.

Wall Conduction Process

The wall conduction process has been formulated in more ways than any of the other processes. Among the possible ways to model this process are

1. Numerical finite difference
2. Numerical finite element
3. Transform methods
4. Time series methods

This process introduces part of the time dependence inherent in the load calculation process. Figure 6 shows schematically the surface temperatures on the inside and outside faces of the wall element and corresponding conductive heat fluxes away from the outside face and toward the inside face. All four quantities are functions of time. The direct formulation of the process has the two temperature functions as input or known quantities and the two heat fluxes as outputs or resultant quantities.

In some models, the surface heat transfer coefficients are included as part of the wall element. Then the temperatures in question are the inside and outside air temperatures. This is not an acceptable formulation because it hides the heat transfer coefficients and prohibits changing them as airflow conditions change. Also, it prohibits treating the internal long-wave radiation exchange appropriately.

Since the heat balances on both sides of the element induce both the temperature and heat flux, the solution technique must deal with this simultaneous condition. From a computational standpoint, two methods that have been used widely are a finite difference procedure and a method using conduction transfer functions. Because of the computational time advantage, the conduction transfer function formulation has been selected for the heat balance procedure presented here.

Inside Face Heat Balance

The heart of the heat balance method is the internal heat balance involving the inside faces of the zone surfaces. This heat balance has many heat transfer components, and they are all coupled. Both long-wave (LW) and short-wave (SW) radiation are important, as well as wall conduction and convection to the air. The inside face heat balance for each surface can be written as follows:

$$q''_{LWX} + q''_{SW} + q''_{LWS} + q''_{ki} + q''_{sol} + q''_{conv} = 0 \quad (27)$$

where

q''_{LWX} = net long-wave radiant flux exchange between zone surfaces, W/m²

q''_{SW} = net short-wave radiation flux to surface from lights, W/m²

q''_{LWS} = long-wave radiation flux from equipment in zone, W/m²

q''_{ki} = conductive flux through the wall, W/m²

q''_{sol} = transmitted solar radiative flux absorbed at surface, W/m²

q''_{conv} = convective heat flux to zone air, W/m²

These terms are explained in the following paragraphs.

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

1. Zone air is completely transparent to LW radiation
2. Zone air completely absorbs LW radiation from surfaces in the zone

Most heat balance models treat air as completely transparent, and then it does not participate in the LW radiation exchange among the surfaces in the zone. The second model is attractive because it can be formulated simply using a combined radiative and convective heat transfer coefficient from each surface to the zone air and in that way decouples the radiant exchange among surfaces in the zone. However, because the transparent air model allows the radiant exchange and is more realistic, the second model is inferior.

Furniture in a zone increases the amount of surface area that can participate in the radiative and convective heat exchanges. It also adds thermal mass to the zone. These two changes can affect the time response of the zone cooling load.

SW Radiation from Lights. The short-wavelength radiation from lights is usually assumed to be distributed over the surfaces in the zone in some manner. The HB procedure retains this approach but allows the distribution function to be changed.

LW Radiation from Internal Sources. The traditional model for this source defines a radiative/convective split for the heat introduced into a zone from equipment. The radiative part is then distributed over the surfaces in the zone in some manner. This is not a completely realistic model, and it departs from the heat balance principles. If it were handled in a true heat balance model, the equipment surfaces would be treated just as other LW radiant sources within the zone. However, since information about the surface temperature of equipment is rarely known, it is reasonable to keep the radiative/convective split concept even though it ignores the true nature of the radiant exchange. ASHRAE Research Project 1055 (Hosni et al. 1999) determined radiative/convective splits for many additional equipment types, as listed in Table 13.

Transmitted Solar Heat Gain. The calculation procedure for determining transmitted solar energy through fenestration as described in Chapter 30 uses the solar heat gain coefficient (SHGC) directly rather than relating it to double-strength glass as is done when using a shading coefficient (SC). The difficulty with this plan is that the SHGC includes both the transmitted solar and the inward-flowing fraction of the solar radiation absorbed in the window. With the heat balance method, this latter part should be added to the conduction component so that it can be included in the inside face heat balance.

Transmitted solar radiation is also distributed over surfaces in the zone in a prescribed manner. It is possible to calculate the actual position of beam solar radiation, but this involves partial surface irradiation, which is inconsistent with the rest of the zone model that assumes uniform conditions over an entire surface.

Using SHGC to Calculate Solar Heat Gain

The total solar heat gain through fenestration consists of the directly transmitted solar radiation plus the inward-flowing fraction of the solar radiation that is absorbed in the glazing system.

Both parts contain beam and diffuse contributions. The transmitted radiation goes directly onto surfaces within the zone and is accounted for in the surface inside heat balance. The zone heat balance model accommodates the resulting heat fluxes without difficulty. The second part, the inward-flowing fraction of the absorbed solar radiation, gets involved in an interaction with the other surfaces of the enclosure through long-wave radiant exchange and with the zone air through convective heat transfer. As such, it is dependent both on the geometric and radiative properties of the zone enclosure and the convection characteristics inside and outside the zone. The solar heat gain coefficient (SHGC) combines the transmitted solar radiation and the inward-flowing fraction of the absorbed radiation. The SHGC is defined as

$$\text{SHGC} = \tau + \sum_{k=1}^n N_k \alpha_k \quad (28)$$

where

- τ = solar transmittance of glazing
- α_k = solar absorptance of the k th layer of the glazing system
- n = number of layers
- N_k = inward-flowing fraction of absorbed radiation in the k th layer

Note that Equation (28) is written in a generic way. It can be written for a specific incidence angle and/or radiation wavelength and integrated over the wavelength and/or angle, but the principle is the same in each case. Refer to [Chapter 30](#) for the specific expressions.

Unfortunately, the inward-flowing fraction N interacts with the zone in many ways. This interaction can be expressed as

$$N = f(\text{inside convection coefficient, outside convection coefficient, glazing system overall heat transfer coefficient, zone geometry, zone radiation properties})$$

The only way to model these interactions correctly is to combine the window model with the zone heat balance model and solve the two simultaneously. This combination has been done recently in some energy analysis programs but is not generally available in load calculation procedures. In addition, the SHGC used for rating glazing systems is based on specific values of the inside, outside, and overall heat transfer coefficients and does not include any zonal long-wavelength radiation considerations. So, the challenge is to devise a way to use SHGC values within the framework of a heat balance calculation in the most accurate way possible. This will be done 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 4.1 program (LBL 1994). 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 18](#).

[Table 18](#) shows the parameters as a function of incident solar angle and also the diffuse values. The specific parameters shown are

V_{tc} = transmittance in the visible spectrum

R_f and R_b = front and back surface visible reflectances

T_{sol} = solar transmittance [symbol τ in Equations (28), (29), and (30)]

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 [symbol α in Equations (28), (29), and (30)]

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 the Window 4.1 program, the inward-flowing fraction of the absorbed solar can be obtained by rearranging Equation (28) to give:

$$N_k \alpha_k = \text{SHGC} - \tau \quad (29)$$

This quantity, when multiplied by the appropriate incident solar intensity, will provide the amount of absorbed solar radiation that flows inward. In the heat balance formulation for zone loads, this heat flux is combined with that caused by conduction through the glazing and included in the surface heat balance.

The outward-flowing fraction of the absorbed solar radiation is used in the heat balance on the outside face of the glazing and is determined from

$$(1 - N_k) \alpha_k = \alpha_k - N_k \alpha_k = \alpha_k - (\text{SHGC} - \tau) \quad (30)$$

If there is more than one layer, the appropriate summation of absorptances must be done.

There is some potential inaccuracy in using the Window 4.1 SHGC values because the inward-flowing fraction part was determined under specific conditions for the inside and outside heat transfer coefficients. However, it is possible to rerun the program with inside and outside coefficients of one's own choosing. Normally, however, this is not a large effect, and only in highly absorptive glazing systems might it cause significant error.

Table 18 Single Layer Glazing Data Produced by Window 4.1

Parameter	Incident Angle										Diffuse (Hemis.)
	0	10	20	30	40	50	60	70	80	90	
V_{tc}	0.901	0.901	0.9	0.897	0.89	0.871	0.824	0.706	0.441	0	0.823
R_f	0.081	0.081	0.082	0.083	0.09	0.108	0.155	0.271	0.536	1	0.146
R_b	0.081	0.081	0.082	0.083	0.09	0.108	0.155	0.271	0.536	1	0.146
T_{sol}	0.85	0.85	0.848	0.844	0.835	0.814	0.766	0.652	0.399	0	0.77
R_f	0.075	0.074	0.075	0.076	0.082	0.099	0.144	0.255	0.509	1	0.136
R_b	0.075	0.074	0.075	0.076	0.082	0.099	0.144	0.255	0.509	1	0.136
A_{bs1}	0.075	0.076	0.077	0.08	0.083	0.087	0.091	0.093	0.092	0	0.084
SHGC	0.87	0.87	0.868	0.865	0.857	0.837	0.79	0.677	0.423	0	0.792

Source: LBL (1994).

For solar heat gain calculations, then, it seems reasonable to use the generic window property data that comes from Window 4.1. Considering Table 18, the procedure is as follows:

1. Determine the angle of incidence for the glazing.
2. Determine the corresponding SHGC.
3. Evaluate $N_k \alpha_k$ using Equation (28).
4. Multiply T_{sol} by the incident beam radiation intensity to get the transmitted beam solar radiation.
5. Multiply $N_k \alpha_k$ by the incident beam radiation intensity to get inward-flowing absorbed heat.
6. Repeat steps 2 through 5 with the diffuse parameters and diffuse radiation.
7. Add the beam and diffuse components of transmitted and inward-flowing absorbed heat.

Table 13 in Chapter 30 contains SHGC information for many additional glazing systems. That table is similar to Table 18 but is slightly abbreviated. Again, the information needed for heat gain calculations is T_{sol} , SHGC, and A_{bs} .

The same caution regarding the inside and outside heat transfer coefficients applies to the information in Table 13 in Chapter 30. Those values were also obtained with specific inside and outside heat transfer coefficients, and the inward-flowing fraction N is dependent upon those values.

Convection to Zone Air. The inside 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 the 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 664 (Fisher 1998) can be incorporated into the procedures. It also permits determining the sensitivity of the load calculation to these parameters.

Air Heat Balance

In heat balance formulations aimed at determining cooling loads, the capacitance of the air in the zone is neglected and the air heat balance is done as a quasi-steady balance in each time period. Four factors contribute to the air heat balance:

$$q_{conv} + q_{CE} + q_{IV} + q_{sys} = 0 \quad (31)$$

where

- q_{conv} = convective heat transfer from surfaces, W
- q_{CE} = convective parts of the internal loads, W
- q_{IV} = sensible load due to infiltration and ventilation air, W
- q_{sys} = heat transfer to/from the HVAC system, W

The **convection from zone surfaces** q_{conv} is the sum of all the convective heat transfer quantities from the inside surface heat balance. This comes to the air via the convective heat transfer coefficient on the surfaces.

The **convective parts of the internal loads** q_{CE} is the companion to q'_{LWS} , the radiant contribution from internal loads described previously [Equation (27)]. It is added directly to the air heat balance. Such a treatment also violates the tenets of the heat balance approach because surfaces producing the internal loads exchange heat with the zone air through normal convective processes. However, once again, this level of detail is generally not included in the heat balance, so it is included directly into the air heat balance instead.

In keeping with the well-mixed model for the zone air, any air that enters by way of **infiltration or ventilation** (q_{IV}) is immediately mixed with the zone air. The amount of infiltration air is uncertain. Sometimes it is related to the indoor-outdoor temperature

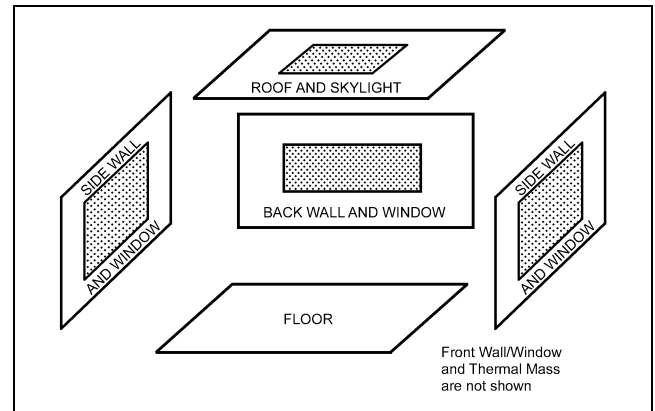


Fig. 7 Schematic View of General Heat Balance Zone

difference and wind speed; however it is determined, it is added directly to the air heat balance.

The conditioned air that enters the zone from the HVAC system and provides q_{sys} is also mixed directly with the zone air.

GENERAL ZONE FOR LOAD CALCULATION

The heat balance procedure is tailored to a single thermal zone, shown in Figure 7. The definition of a thermal zone depends on how the fixed temperature is going to be controlled. If air is circulated through an entire building or an entire floor in such a way that it is uniformly well stirred, the entire building or floor could be considered a thermal zone. On the other hand, if each room has a different control scheme, each room may need to be considered as a separate thermal zone. The framework needs to be flexible enough to accommodate any zone arrangement, but the heat balance aspect of the procedure also requires that a complete zone be described. This zone consists of four walls, a roof or ceiling, a floor, and a "thermal mass surface" (described in the section on Input Required for Heat Balance Procedure). Each wall and the roof can include a window (or skylight in the case of the roof). This makes a total of 12 surfaces, any of which may have zero area if it is not present in the zone to be modeled.

The heat balance processes for this general zone are formulated for a 24 h steady-periodic condition. The variables of the problem are the inside and outside temperatures of the 12 surfaces plus either the HVAC system energy required to maintain a specified air temperature or the air temperature, if the system capacity is specified. This makes a total of 25×24 or 600 variables. While it is possible to set up the problem for a simultaneous solution of these variables, the relatively weak coupling of the problem from one hour to the next permits a double iterative approach. One iteration is through all the surfaces in each hour, and the other iteration is through the 24 h of a day. This procedure automatically reconciles the nonlinear aspects of the surface radiative exchange and the other heat flux terms.

MATHEMATICAL DESCRIPTION OF HEAT BALANCE PROCEDURE

Conduction Process

Because it links the outside and inside heat balances, the wall conduction process regulates the time dependence of the cooling load. For the heat balance procedure presented here, the wall conduction process is formulated using **conduction transfer functions** (CTFs), which relate conductive heat fluxes to the current and past surface temperatures and the past heat fluxes. The general form for the inside heat flux is

$$q''_{ki}(t) = -Z_o T_{si, \theta} - \sum_{j=1}^{nz} Z_j T_{si, \theta-j\delta} + Y_o T_{so, \theta} + \sum_{j=1}^{nz} Y_j T_{so, \theta-j\delta} + \sum_{j=1}^{nq} \Phi_j q''_{ki, \theta-j\delta} \quad (32)$$

For the outside heat flux, the form is

$$q''_{ko}(t) = -Y_o T_{si, \theta} - \sum_{j=1}^{nz} Y_j T_{si, \theta-j\delta} + X_o T_{so, \theta} + \sum_{j=1}^{nz} X_j T_{so, \theta-j\delta} + \sum_{j=1}^{nq} \Phi_j q''_{ko, \theta-j\delta} \quad (33)$$

where

X_j = outside CTF, $j = 0, 1, \dots, nz$
 Y_j = cross CTF, $j = 0, 1, \dots, nz$
 Z_j = inside CTF, $j = 0, 1, \dots, nz$
 Φ_j = flux CTF, $j = 1, 2, \dots, nq$
 θ = time
 δ = time step
 T_{si} = inside face temperature, °C
 T_{so} = outside face temperature, °C
 q''_{ki} = conductive heat flux on inside face, W/m²
 q''_{ko} = conductive heat flux on outside face, W/m²

The subscript following the comma indicates the time period for the quantity in terms of the time step δ . Also, the first terms in the series have been separated from the rest in order to facilitate solving for the current temperature in the solution scheme.

The two summation limits nz and nq depend on wall construction and depend somewhat on the scheme used for calculating the CTFs. If $nq = 0$, the CTFs are generally referred to as **response factors**, but then theoretically nz is infinite. The values for nz and nq are generally set to minimize the amount of computation. A development of CTFs can be found in Hittle (1981).

Heat Balance Equations

The primary variables in the heat balance for the general zone are the 12 inside face temperatures and the 12 outside face temperatures at each of the 24 h, assigning i as the surface index and j as the hour index, or, in the case of CTFs, the sequence index. Then, the primary variables are:

T_{soij} = outside face temperature, $i = 1, 2, \dots, 12; j = 1, 2, \dots, 24$

$T_{si,j}$ = inside face temperature, $i = 1, 2, \dots, 12; j = 1, 2, \dots, 24$

In addition, q_{sysj} = cooling load, $j = 1, 2, \dots, 24$

Equations (26) and (33) are combined and solved for T_{so} to produce 12 equations applicable in each time step:

$$T_{soij} = \left(\sum_{k=1}^{nz} T_{si,j-k} Y_{i,k} - \sum_{k=1}^{nz} T_{soij-k} Z_{i,k} - \sum_{k=1}^{nq} \Phi_{i,k} q''_{koij-k} + q''_{asolij} + q''_{LWRij} + T_{si,j} Y_{i,0} + T_{oj} h_{coij} \right) / (Z_{i,0} + h_{coij}) \quad (34)$$

where

T_o = outside air temperature

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

Equation (34) shows the need for separating $Z_{i,0}$ because the contribution of the current surface temperature to the conductive flux can be collected with the other terms involving that temperature.

Equations (27) and (32) are combined and solved for T_{si} to produce the next 12 equations:

$$T_{si,j} = \left(T_{si,j} Y_{i,0} + \sum_{k=1}^{nz} T_{soij-k} Y_{i,k} - \sum_{k=1}^{nz} T_{si,j-k} Z_{i,k} + \sum_{k=1}^{nq} \Phi_{i,k} q''_{ki,j-k} + T_{aj} h_{ci,j} + q''_{LWS} + q''_{LWX} + q''_{SW} + q''_{sol} \right) / (Z_{i,0} + h_{ci,j}) \quad (35)$$

where

T_a = zone air temperature

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

Note that in Equations (34) and (35), the opposite surface temperature at the current time appears on the right-hand side. The two equations could be solved simultaneously to eliminate those variables. Depending on the order of updating the other terms in the equations, this can have a beneficial effect on solution stability.

The remaining equation comes from the air heat balance, Equation (31). This provides the cooling load q_{sys} at each time step:

$$q_{sysj} = \sum_{i=1}^{12} A_i h_{ci} (T_{si,j} - T_{aj}) + q_{CE} + q_{IV} \quad (36)$$

In Equation (36), the convective heat transfer term is expanded to show the interconnection between the surface temperatures and the cooling load.

Overall HB Iterative Solution Procedure

The iterative HB procedure consists of a series of initial calculations that proceed sequentially, followed by a double iteration loop as shown in the following steps:

1. Initialize areas, properties, and face temperatures for all surfaces, 24 h.
2. Calculate incident and transmitted solar flux for all surfaces and hours.
3. Distribute transmitted solar energy to all inside faces, 24 h.
4. Calculate internal load quantities for all 24 h.
5. Distribute LW, SW, and convective energy from internal loads to all surfaces for all hours.
6. Calculate infiltration and ventilation loads for all hours.
7. Iterate the heat balance according to the following scheme:

```

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

```

8. Display results.

Generally, four or six surface iterations are sufficient to provide convergence. The convergence check on the day iteration should be based on the difference between the inside and the outside conductive heat flux terms q_k . A limit such as requiring the difference between all inside and outside flux terms to be less than 1% of either flux works well.

INPUT REQUIRED FOR HEAT BALANCE PROCEDURE

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. In the case of the heat balance procedure, no precalculations are made, so the procedure requires a fairly complete description of the zone.

Global Information. Because the procedure incorporates a solar calculation, some global information is required, including latitude, longitude, time zone, month, day of month, north axis of the zone, and the zone height (floor to floor). Additionally, to take full advantage of the flexibility of the method to incorporate, for example, variable outside heat transfer coefficients, such things as wind speed, wind direction, and terrain roughness may be specified. Normally, these variables and others default to some reasonable set of values, but the flexibility remains.

Wall Information (Each Wall). Because the walls are involved in three of the fundamental processes (external and internal heat balance and wall conduction), each wall of the zone requires a fairly large set of variables. They include

- Facing angle with respect to building north
- Tilt (degrees from horizontal)
- Area
- Solar absorptivity outside
- Long-wave emissivity outside
- Short-wave absorptivity inside
- Long-wave emissivity inside
- Exterior boundary temperature condition (solar vs. nonsolar)
- External roughness
- Layer-by-layer construction information

Again, some of these parameters can be defaulted, but they are changeable, and they indicate the more fundamental character of the heat balance method since they are related to true heat transfer processes.

Window Information (Each Window). The situation for windows is similar to that for walls, but the windows require some additional information because of their role in the solar load. The necessary parameters include:

- Area
- Normal solar transmissivity
- Normal SHGC
- Normal total absorptivity
- Long-wave emissivity outside
- Long-wave emissivity inside
- Surface-to-surface thermal conductance
- Reveal (for solar shading)
- Overhang width (for solar shading)
- Distance from overhang to window (for solar shading)

Roof and Floor Details. The roof and floor surfaces are specified similarly to walls. The main difference is that the ground outside boundary condition will probably be specified more often for a floor.

Thermal Mass Surface Details. An “extra” surface, called a thermal mass surface, can serve several functions. It is included in the radiant heat exchange with the other surfaces in the space but is

only exposed to the inside air convective boundary condition. As an example, this surface would be used to account for the movable partitions in a space. The construction of the partitions is specified layer by layer, similar to specification for walls, and those layers store and release heat via the same conduction mechanism as walls. As a general definition, the extra thermal mass surface should be sized to represent all of the surfaces in the space that are exposed to the air mass, except the walls, roof, floor, and windows. In the formulation, both sides of the thermal mass participate in the exchange.

Internal Heat Gain Details. The space can be subjected to several internal heat sources: people, lights, electrical equipment, and infiltration. In the case of infiltration, the energy is assumed to go immediately into the air heat balance, so it is the least complicated of the heat gains. For the others, several parameters must be specified. These include the following fractions:

- Fraction of heat gain that is sensible energy
- Fraction of heat gain that is latent energy
- Fraction of energy that enters as short-wave radiation
- Fraction of energy that enters as long-wave radiation
- Fraction of energy that enters the air immediately as convection
- Activity level of people
- Fraction of energy of lighting heat gain that goes directly to the return air

Radiant Distribution Functions. As mentioned previously, the generally accepted assumptions for the heat balance method include specifying the distribution of radiant energy from several sources to the surfaces that enclose the space. This requires a distribution function that specifies the fraction of the total radiant input that is absorbed by each surface. The types of radiation that require distribution functions are

- Long-wave radiation from equipment and lights
- Short-wave radiation from lights
- Transmitted solar radiation

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

- Heat transfer coefficients/convection models
- Solar coefficients
- Sky models

The amount of input information required may seem extensive, but many of the parameters can be set to default values in most routine applications. However, all of the 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 new simplified method for performing design cooling load calculations that is derived from the heat balance (HB) method described above. It effectively replaces all other simplified (non-heat-balance) methods, 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.

The casual observer might well ask why yet another load calculation method is necessary. This method was developed in response to the desire to offer a method that is rigorous, yet does not require iterative calculations, and that quantifies each component contribution to the total cooling load. In addition, it is desirable for the user to be able to inspect and compare the coefficients for different construction and zone types in a form illustrating their relative impact

on the result. These characteristics of the RTS method make it easier to apply engineering judgment during the cooling load calculation process.

The RTS method is suitable for peak design load calculations, but it should not be used for annual energy simulations due to its inherent limiting assumptions. The RTS method, while simple in concept, involves too many calculations to be used practically as a manual method, although it can easily be implemented in a simple computerized spreadsheet, as illustrated in the examples. For a manual cooling load calculation method, refer to the CLTD/CLF method included in the 1997 *ASHRAE Handbook—Fundamentals*.

RTS COOLING LOAD ASSUMPTIONS AND PRINCIPLES

Design cooling loads are based on the assumption of **steady-periodic conditions** (i.e., the design day's weather, occupancy, and heat gain conditions are identical to those for preceding days such that the loads repeat on an identical 24 h cyclical basis). Thus, the heat gain for a particular component at a particular hour is the same as 24 h prior, which is the same as 48 h prior, etc. This assumption is the basis for the RTS derivation from the HB method.

Cooling load calculations must address two time-delay effects inherent in building heat transfer processes: (1) delay of conductive heat gain through opaque massive exterior surfaces (walls, roofs, or floors) and (2) delay of radiative heat gain conversion to cooling loads.

Exterior walls and roofs conduct heat due to temperature differences between outdoor and indoor air. In addition, solar energy on exterior surfaces is absorbed, then transferred by conduction to the building interior. Due to 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 earlier in the section on Cooling Load Principles, most heat sources transfer energy to a room by a combination of convection and radiation. The convection part of heat gain immediately becomes cooling load. The radiation part must first be absorbed by the finishes and mass of the interior room surfaces and becomes cooling load only when it is later transferred by convection from those surfaces to the room air. Thus, radiant heat gains become cooling loads over a delayed period of time.

OVERVIEW OF THE RADIANT TIME SERIES METHOD

Figure 8 gives an overview of the radiant time series method. In the calculation of solar radiation, transmitted solar heat gain through windows, sol-air temperature, and infiltration, the RTS method is exactly the same as previous simplified methods (TFM and TETD/TA). Important areas that are different include the computation of conductive heat gain, the splitting of all heat gains into radiant and convective portions, and the conversion of radiant heat gains into cooling loads.

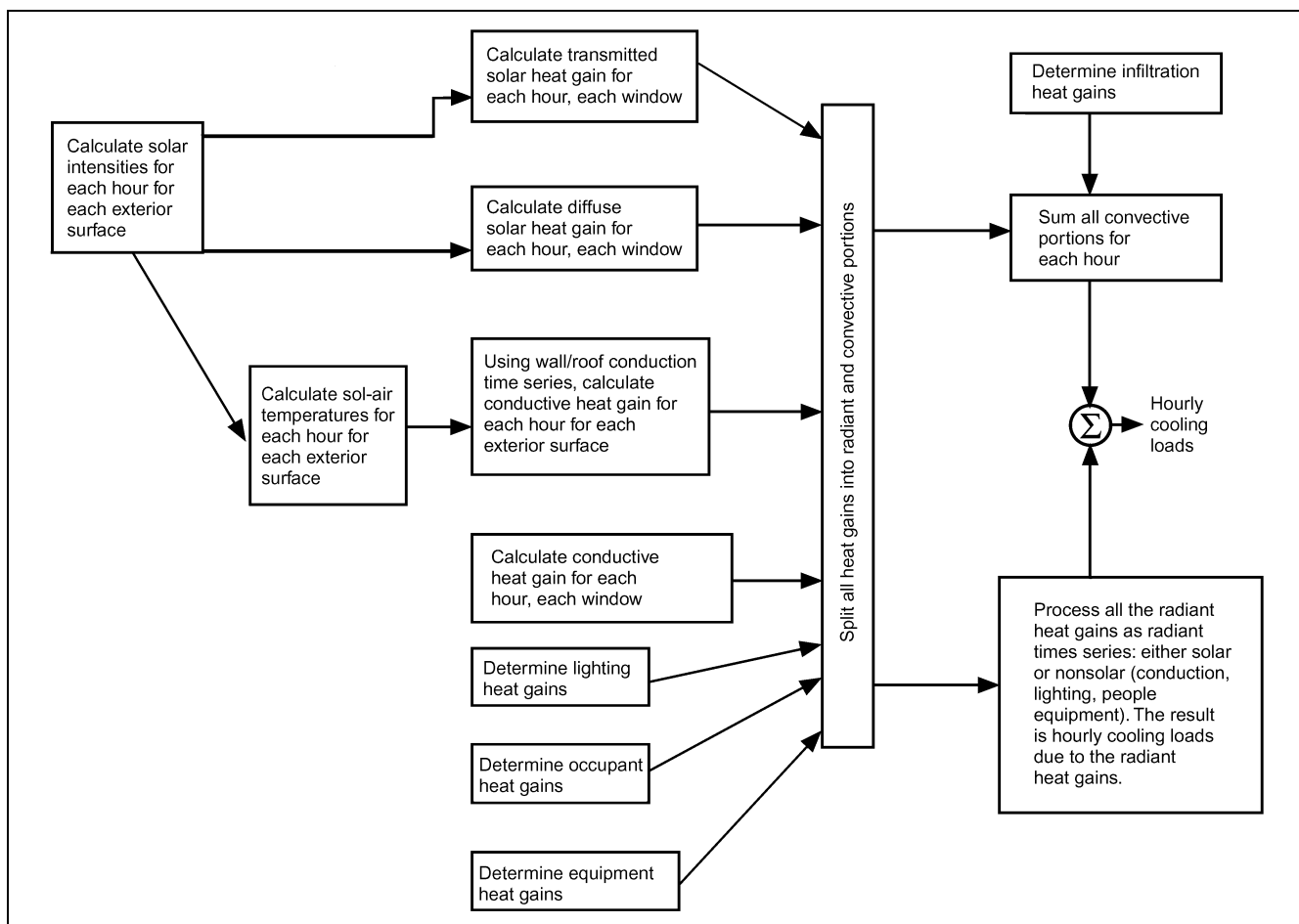


Fig. 8 Overview of Radiant Time Series Method

The RTS method accounts for both conduction time delay and radiant time delay effects by multiplying hourly heat gains by 24 h time series. The time series multiplication, in effect, distributes heat gains over time. Series coefficients, which are called **radiant time factors** and **conduction time factors**, are derived using the heat balance method. Radiant time factors reflect the percentage of an earlier radiant heat gain that becomes cooling load during the current hour. Likewise, conduction time factors reflect the percentage of an earlier heat gain at the exterior of a wall or roof that becomes

heat gain at the inside during the current hour. By definition, each radiant or conduction time series must total 100%.

These series can be used to easily compare the time-delay impact of one construction versus another. This ability to compare choices is of particular benefit in the design process, when all construction details may not have been decided. Comparison can illustrate the magnitude of difference between the choices, allowing the engineer to apply judgment and make more informed assumptions in estimating the load.

Figure 9 illustrates 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 U-factors. Figure 11 illustrates RTS values for zones varying from light to heavy construction.

RADIANT TIME SERIES PROCEDURE

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

1. Calculate 24 h profile of component heat gain for design day (for conduction, first account for conduction time delay by applying conduction time series).
2. Split heat gains into radiant and convective parts (see Table 19 for radiant and convective fractions).
3. Apply appropriate radiant time series to radiant part of heat gains to account for time delay in conversion to cooling load.
4. Sum convective part of heat gain and delayed radiant part of heat gain to determine cooling load for each hour for each cooling load component.

After calculating cooling loads for each component for each hour, sum those to determine the total cooling load for each hour and select the hour with the peak load for design of the air-conditioning system. This process should be repeated for multiple design months to determine the month when the peak load occurs, especially with windows on southern exposures (northern exposure in southern latitudes), which can result in higher peak room cooling loads in winter months than in summer.

Conductive Heat Gain Using Conduction Time Series

In the RTS method, conduction through exterior walls and roofs is calculated using conduction time series (CTS). Wall and roof conductive heat input at the exterior is defined by the familiar conduction equation as

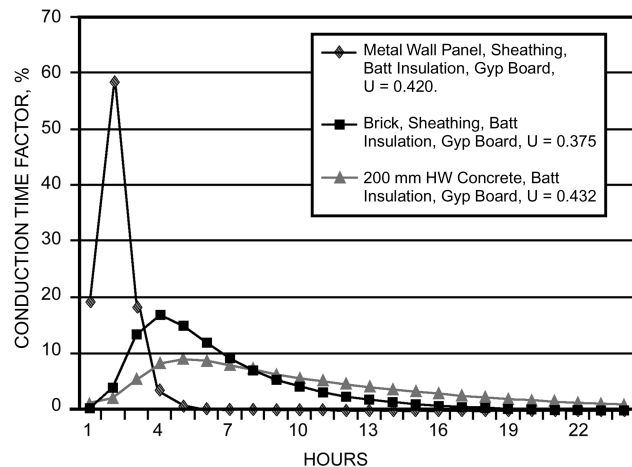


Fig. 9 CTS for Light to Heavy Walls

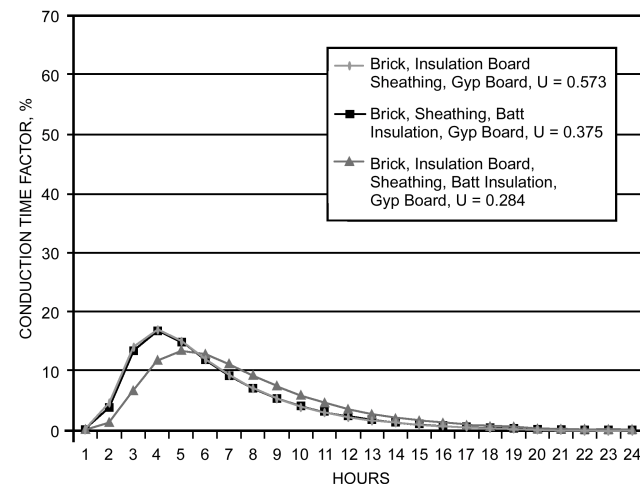


Fig. 10 CTS for Walls with Similar Mass and Increasing Insulation

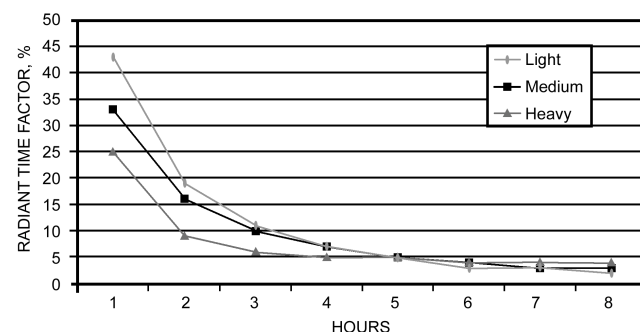


Fig. 11 RTS for Light to Heavy Construction

Table 19 Convective and Radiant Percentages of Total Sensible Heat Gain

Heat Gain Source	Radiant Heat, %	Convective Heat, %
Transmitted solar, no inside shade	100	0
Window solar, with inside shade	63	37
Absorbed (by fenestration) solar	63	37
Fluorescent lights, suspended, unvented	67	33
Fluorescent lights, recessed, vented to return air	59	41
Fluorescent lights, recessed, vented to return air and supply air	19	81
Incandescent lights	80	20
People	See Table 1	
Conduction, exterior walls	63	37
Conduction, exterior roofs	84	16
Infiltration and ventilation	0	100
Machinery and appliances (see Table 13)	20 to 80	80 to 20

Sources: Pedersen et al. (1998), Hosni et al. (1999).

Table 20 Wall Conduction Time Series (CTS)

Wall Number =	CURTAIN WALLS			STUD WALLS				EIFS			BRICK WALLS									
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
U-factor, W/(m ² ·K)	0.428	0.429	0.428	0.419	0.417	0.406	0.413	0.668	0.305	0.524	0.571	0.377	0.283	0.581	0.348	0.628	0.702	0.514	0.581	0.389
Total R	2.3	2.3	2.3	2.4	2.4	2.5	2.4	1.5	3.3	1.9	1.7	2.7	3.5	1.7	2.9	1.6	1.4	1.9	1.7	2.6
Mass, kg/m ²	31.0	20.9	80.0	25.5	84.6	25.6	66.7	36.6	38.3	130.9	214.1	214.7	215.8	290.6	304.0	371.7	391.5	469.3	892.2	665.1
Thermal Capacity, W/(m ² ·K)	8.5	5.4	19.0	7.0	20.5	9.3	17.1	10.2	10.6	33.2	49.2	49.3	49.7	66.6	70.6	89.1	86.7	108.0	218.4	161.5
Hour	Conduction Time Factors, %																			
0	18	25	8	19	6	7	5	11	2	1	0	0	0	1	2	2	1	3	4	3
1	58	57	45	59	42	44	41	50	25	2	5	4	1	1	2	2	1	3	4	3
2	20	15	32	18	33	32	34	26	31	6	14	13	7	2	2	2	3	3	4	3
3	4	3	11	3	13	12	13	9	20	9	17	17	12	5	3	4	6	3	4	4
4	0	0	3	1	4	4	4	3	11	9	15	15	13	8	5	5	7	3	4	4
5	0	0	1	0	1	1	2	1	5	9	12	12	13	9	6	6	8	4	4	4
6	0	0	0	0	1	0	1	0	3	8	9	9	11	9	7	6	8	4	4	5
7	0	0	0	0	0	0	0	0	2	7	7	7	9	9	7	7	8	5	4	5
8	0	0	0	0	0	0	0	0	1	6	5	5	7	8	7	7	8	5	4	5
9	0	0	0	0	0	0	0	0	0	6	4	4	6	7	7	6	7	5	4	5
10	0	0	0	0	0	0	0	0	0	5	3	3	5	7	6	6	6	5	4	5
11	0	0	0	0	0	0	0	0	0	5	2	2	4	6	6	6	6	5	5	5
12	0	0	0	0	0	0	0	0	0	4	2	2	3	5	5	5	5	5	5	5
13	0	0	0	0	0	0	0	0	0	4	1	2	2	4	5	5	4	5	5	5
14	0	0	0	0	0	0	0	0	0	3	1	2	2	4	5	5	4	5	5	5
15	0	0	0	0	0	0	0	0	0	3	1	1	1	3	4	4	3	5	4	4
16	0	0	0	0	0	0	0	0	0	3	1	1	1	3	4	4	3	5	4	4
17	0	0	0	0	0	0	0	0	0	2	1	1	1	2	3	4	3	4	4	4
18	0	0	0	0	0	0	0	0	0	2	0	0	1	2	3	3	2	4	4	4
19	0	0	0	0	0	0	0	0	0	2	0	0	1	2	3	3	2	4	4	4
20	0	0	0	0	0	0	0	0	0	2	0	0	0	1	3	3	2	4	4	4
21	0	0	0	0	0	0	0	0	0	1	0	0	0	1	2	2	1	4	4	4
22	0	0	0	0	0	0	0	0	0	1	0	0	0	1	2	2	1	4	4	3
23	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	3	4	3
Total Percentage	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Layer ID from outside to inside (see Table 22)	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01
	F09	F08	F10	F08	F10	F11	F07	F06	F06	F06	M01	M01	M01	M01	M01	M01	M01	M01	M01	M01
	F04	F04	F04	G03	G03	G02	G03	I01	I01	I01	F04	F04	F04	F04	F04	F04	F04	F04	F04	F04
	I02	I02	I02	I04	I04	I04	I04	G03	G03	G03	I01	G03	I01	I01	M03	I01	I01	I01	I01	M15
	F04	F04	F04	G01	G01	G04	G01	F04	I04	M03	G03	I04	G03	M03	I04	M05	M01	M13	M16	I04
	G01	G01	G01	F02	F02	F02	F02	G01	G01	F04	F04	G01	I04	F02	G01	G01	F02	F04	F04	G01
	F02	F02	F02	0	0	0	0	F02	F02	G01	G01	F02	G01	0	F02	F02	0	G01	G01	F02
	0	0	0	0	0	0	0	0	0	F02	F02	0	F02	0	0	0	0	F02	F02	0

Wall Number Descriptions

1. Spandrel glass, insulation board, gyp board
2. Metal wall panel, insulation board, gyp board
3. 25 mm stone, insulation board, gyp board
4. Metal wall panel, sheathing, batt insulation, gyp board
5. 25 mm stone, sheathing, batt insulation, gyp board
6. Wood siding, sheathing, batt insulation, 13 mm wood
7. 25 mm stucco, sheathing, batt insulation, gyp board
8. EIFS finish, insulation board, sheathing, gyp board
9. EIFS finish, insulation board, sheathing, batt insulation, gyp board
10. EIFS finish, insulation board, sheathing, 200 mm LW CMU, gyp board
11. Brick, insulation board, sheathing, gyp board
12. Brick, sheathing, batt insulation, gyp board
13. Brick, insulation board, sheathing, batt insulation, gyp board
14. Brick, insulation board, 200 mm LW CMU
15. Brick, 200 mm LW CMU, batt insulation, gyp board
16. Brick, insulation board, 200 mm HW CMU, gyp board
17. Brick, insulation board, brick
18. Brick, insulation board, 200 mm LW concrete, gyp board
19. Brick, insulation board, 300 mm HW concrete, gyp board
20. Brick, 200 mm HW concrete, batt insulation, gyp board

$$q_{i,0-n} = UA(t_{e,0-n} - t_{rc}) \quad (37)$$

where

- $q_{i,0-n}$ = conductive heat input for the surface n hours ago, W
 U = overall heat transfer coefficient for the surface, W/(m²·K)
 A = surface area, m²
 $t_{e,0-n}$ = sol-air temperature n hours ago, °C
 t_{rc} = presumed constant room air temperature, °C

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:

where

- q_0 = hourly conductive heat gain for the surface, W
 $q_{i,0}$ = heat input for the current hour, W
 $q_{i,0-n}$ = heat input n hours ago, W
 c_0, c_1 , etc. = conduction time factors

Conduction time factors for representative wall and roof types are included in Tables 20 and 21. Those values were derived by first calculating conduction transfer functions for each example wall and roof construction. The assumption of steady-periodic heat input conditions for design load calculations allowed the conduction transfer functions to be reformulated into periodic response factors

Table 20 Wall Conduction Time Series (CTS) (Concluded)

Wall Number =	CONCRETE BLOCK WALL						PRECAST AND CAST-IN-PLACE CONCRETE WALLS								
	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35
U-factor, W/(m ² ·K)	0.383	0.335	0.414	1.056	0.834	0.689	0.673	0.418	0.434	0.650	0.387	0.467	0.434	0.266	3.122
Total R	2.6	3.0	2.4	0.9	1.2	1.5	1.5	2.4	2.3	1.5	2.6	2.1	2.3	3.8	0.3
Mass, kg/m ²	108.8	108.8	224.3	94.3	107.1	168.9	143.9	144.6	262.5	291.8	274.7	488.1	469.9	698.9	683.2
Thermal Capacity, W/(m ² ·K)	27.4	27.4	57.0	23.1	26.9	42.0	34.5	34.6	61.3	68.9	64.9	122.7	118.3	175.7	171.0
Hour	Conduction Time Factors, %														
0	0	1	0	1	1	0	1	2	1	3	1	2	1	0	1
1	2	11	3	1	10	8	1	2	2	3	2	2	2	2	11
2	8	21	12	2	20	18	3	3	3	4	5	3	4	8	21
3	12	20	16	5	18	18	6	5	6	5	8	3	7	12	20
4	12	15	15	7	14	14	8	6	7	6	9	5	8	12	15
5	11	10	12	9	10	11	9	6	8	6	9	5	8	11	10
6	9	7	10	9	7	8	9	6	8	6	8	6	8	9	7
7	8	5	8	8	5	6	9	6	7	5	7	6	8	8	5
8	7	3	6	8	4	4	8	6	7	5	6	6	7	7	3
9	6	2	4	7	3	3	7	6	6	5	6	6	6	6	2
10	5	2	3	6	2	2	7	5	6	5	5	6	6	5	2
11	4	1	3	6	2	2	6	5	5	5	5	5	5	4	1
12	3	1	2	5	1	2	5	5	5	4	4	5	4	3	1
13	2	1	2	4	1	1	4	5	4	4	4	5	4	2	1
14	2	0	1	4	1	1	4	4	4	4	3	4	4	2	0
15	2	0	1	3	1	1	3	4	3	4	3	4	3	2	0
16	1	0	1	3	0	1	2	4	3	4	3	4	3	1	0
17	1	0	1	2	0	0	2	3	3	4	2	4	3	1	0
18	1	0	0	2	0	0	1	3	2	4	2	4	2	1	0
19	1	0	0	2	0	0	1	3	2	3	2	3	2	1	0
20	1	0	0	2	0	0	1	3	2	3	2	3	2	1	0
21	1	0	0	2	0	0	1	3	2	3	2	3	1	1	0
22	1	0	0	1	0	0	1	3	2	3	1	3	1	1	0
23	0	0	0	1	0	0	1	2	2	2	1	3	1	0	0
Total Percentage	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Layer ID from outside to inside (see Table 22)	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01
	F07	M08	M08	M09	M11	M11	M11	F06	M13	F06	M15	M16	M16	F07	M08
	M05	F02	F04	F04	I01	I04	I02	I01	I04	I02	I04	I05	F02	M05	F02
	I04	—	G01	G01	F04	G01	M11	M13	G01	M15	G01	G01	—	I04	—
	G01	—	F02	F02	G01	F02	F02	G01	F02	G01	F02	F02	—	G01	—
	F02	—	—	—	F02	—	—	F02	—	F02	—	—	—	F02	—
Wall Number Descriptions															
21. 200 mm LW CMU, batt insulation, gyp board							29. 100 mm LW concrete, board insulation, 100 mm LW concrete								
22. 200 mm LW CMU with fill insulation, batt insulation, gyp board							30. EIFS finish, insulation board, 200 mm LW concrete, gyp board								
23. 25 mm stucco, 200 mm HW CMU, batt insulation, gyp board							31. 200 mm LW concrete, batt insulation, gyp board								
24. 200 mm LW CMU with fill insulation							32. EIFS finish, insulation board, 200 mm HW concrete, gyp board								
25. 200 mm LW CMU with fill insulation, gyp board							33. 200 mm HW concrete, batt insulation, gyp board								
26. 300 mm LW CMU with fill insulation, gyp board							34. 300 mm HW concrete, batt insulation, gyp board								
27. 100 mm LW concrete, board insulation, gyp board							35. 300 mm HW concrete								
28. 100 mm LW concrete, batt insulation, gyp board															

as demonstrated by Spitler and Fisher (1999a). The periodic response factors were further simplified by dividing the 24 periodic response factors by the respective overall wall or roof U-factor to form the conduction time series (CTS). The conduction time factors can then be used in Equation (38) and provide a means for comparison of time delay characteristics between different wall and roof constructions. Construction material data used in the calculations for walls and roofs included in Tables 20 and 21 are listed in Table 22.

Heat gains calculated for walls or roofs using periodic response factors (and thus CTS) are identical to those calculated using conduction transfer functions for the steady periodic conditions assumed in design cooling load calculations. The methodology for calculating periodic response factors from conduction transfer functions was originally developed as part of ASHRAE Research Project 875 (Spitler et al. 1997, Spitler and Fisher 1999b).

Example 4. Wall heat gain using conduction time series. Using the data from Example 1 and Table 15, calculate the heat gain at 3 P.M. Central Daylight Time on July 21 through 9.3 m² of a wall composed of light-colored 100 mm brick, 50 mm of insulation ($R = 1.76 \text{ m}^2\cdot\text{K}/\text{W}$) and 200 mm lightweight concrete block. An air space is included between the brick and insulation. Overall U-factor of the wall is $0.386 \text{ W}/(\text{m}^2\cdot\text{K})$. Inside room temperature is 23.9°C .

Solution: Conductive heat gain is calculated using Equations (37) and (38). First, calculate the 24 h heat input profile using Equation (37) and the sol-air temperatures from Table 15 for a southwest-facing wall with light exterior color:

$$\begin{aligned}
 q_{i,1} &= (0.386)(9.3)(24.4 - 23.9) = 2.0 \text{ W} \\
 q_{i,2} &= (0.386)(9.3)(24.4 - 23.9) = 2.0 \\
 q_{i,3} &= (0.386)(9.3)(23.9 - 23.9) = 0 \\
 q_{i,4} &= (0.386)(9.3)(23.3 - 23.9) = -2.0 \\
 q_{i,5} &= (0.386)(9.3)(23.3 - 23.9) = -2.0 \\
 q_{i,6} &= (0.386)(9.3)(24.3 - 23.9) = 1.6
 \end{aligned}$$

Table 21 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
U-factor, W/(m ² ·K)	0.249	0.227	0.255	0.235	0.239	0.231	0.393	0.329	0.452	0.370	0.323	0.206	0.297	0.304	0.296	0.288	0.315	0.313	0.239
Total R	4.0	4.4	3.9	4.2	4.2	4.3	2.5	3.0	2.2	2.7	3.1	4.9	3.4	3.3	3.4	3.5	3.2	3.2	4.2
Mass, kg/m ²	26.7	21.0	14.0	34.7	55.5	34.9	48.9	55.9	23.9	30.9	25.0	27.2	57.6	149.2	214.3	279.3	360.7	474.5	362.3
Thermal Capacity, W/(m ² ·K)	7.3	4.6	3.5	12.9	20.2	13.0	21.0	22.1	7.8	8.9	8.1	8.9	15.7	37.6	52.8	67.9	92.8	121.3	91.8
Hour	Conduction Time Factors, %																		
0	6	10	27	1	1	1	0	1	18	4	8	1	0	1	2	2	2	3	1
1	45	57	62	17	17	12	7	3	61	41	53	23	10	2	2	2	2	3	2
2	33	27	10	31	34	25	18	8	18	35	30	38	22	8	3	3	5	3	6
3	11	5	1	24	25	22	18	10	3	14	7	22	20	11	6	4	6	5	8
4	3	1	0	14	13	15	15	10	0	4	2	10	14	11	7	5	7	6	8
5	1	0	0	7	6	10	11	9	0	1	0	4	10	10	8	6	7	6	8
6	1	0	0	4	3	6	8	8	0	1	0	2	7	9	8	6	6	6	7
7	0	0	0	2	1	4	6	7	0	0	0	0	5	7	7	6	6	6	7
8	0	0	0	0	0	2	5	6	0	0	0	0	4	6	7	6	6	6	6
9	0	0	0	0	0	1	3	5	0	0	0	0	3	5	6	6	5	5	5
10	0	0	0	0	0	1	3	5	0	0	0	0	2	5	5	6	5	5	5
11	0	0	0	0	0	1	2	4	0	0	0	0	1	4	5	5	5	5	5
12	0	0	0	0	0	0	1	4	0	0	0	0	1	3	5	5	4	5	4
13	0	0	0	0	0	0	1	3	0	0	0	0	1	3	4	5	4	4	4
14	0	0	0	0	0	0	1	3	0	0	0	0	0	3	4	4	4	4	3
15	0	0	0	0	0	0	1	3	0	0	0	0	0	2	3	4	4	4	3
16	0	0	0	0	0	0	0	2	0	0	0	0	0	2	3	4	3	4	3
17	0	0	0	0	0	0	0	2	0	0	0	0	0	2	3	4	3	4	3
18	0	0	0	0	0	0	0	2	0	0	0	0	0	1	3	3	3	3	2
19	0	0	0	0	0	0	0	2	0	0	0	0	0	1	2	3	3	3	2
20	0	0	0	0	0	0	0	1	0	0	0	0	0	1	2	3	3	3	2
21	0	0	0	0	0	0	0	1	0	0	0	0	0	1	2	3	3	3	2
22	0	0	0	0	0	0	0	1	0	0	0	0	0	1	2	3	2	2	2
23	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	2	2	2	2
	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Layer ID from outside to inside (see Table 22)	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01
	F08	F08	F08	F12	F14	F15	F13	F13	F13	F13	F13	F13	M17	F13	F13	F13	F13	F13	F13
	G03	G03	G03	G05	G05	G05	G03	G03	G03	G03	G03	G03	F13	G03	G03	G03	G03	G03	M14
	F05	F05	F05	F05	F05	F05	I02	I02	I02	I02	I03	I02	G03	I03	I03	I03	I03	I03	F05
	I05	I05	I05	I05	I05	I05	G06	G06	F08	F08	F08	I03	I03	M11	M12	M13	M14	M15	I05
	G01	F05	F03	F05	F05	F05	F03	F05	F03	F05	F03	F08	F08	F03	F03	F03	F03	F03	F16
	F03	F16	—	G01	G01	G01	—	F16	—	F16	—	—	F03	—	—	—	—	—	F03
	—	F03	—	F03	F03	F03	—	F03	—	F03	—	—	—	—	—	—	—	—	—

Roof Number Descriptions

- | | |
|---|--|
| 1. Metal roof, batt insulation, gyp board | 11. Membrane, sheathing, insulation board, metal deck |
| 2. Metal roof, batt insulation, suspended acoustical ceiling | 12. Membrane, sheathing, plus insulation boards, metal deck |
| 3. Metal roof, batt insulation | 13. 50 mm concrete roof ballast, membrane, sheathing, insulation board, metal deck |
| 4. Asphalt shingles, wood sheathing, batt insulation, gyp board | 14. Membrane, sheathing, insulation board, 100 mm LW concrete |
| 5. Slate or tile, wood sheathing, batt insulation, gyp board | 15. Membrane, sheathing, insulation board, 150 mm LW concrete |
| 6. Wood shingles, wood sheathing, batt insulation, gyp board | 16. Membrane, sheathing, insulation board, 200 mm LW concrete |
| 7. Membrane, sheathing, insulation board, wood deck | 17. Membrane, sheathing, insulation board, 150 mm HW concrete |
| 8. Membrane, sheathing, insulation board, wood deck, suspended acoustical ceiling | 18. Membrane, sheathing, insulation board, 200 mm HW concrete |
| 9. Membrane, sheathing, insulation board, metal deck | 19. Membrane, 150 mm HW concrete, batt insulation, suspended acoustical ceiling |
| 10. Membrane, sheathing, insulation board, metal deck, suspended acoustical ceiling | |

$$\begin{aligned}
 q_{i,7} &= (0.386)(9.3)(25.8 - 23.9) = 6.8 \\
 q_{i,8} &= (0.386)(9.3)(27.6 - 23.9) = 13.2 \\
 q_{i,9} &= (0.386)(9.3)(29.8 - 23.9) = 21.1 \\
 q_{i,10} &= (0.386)(9.3)(31.9 - 23.9) = 28.8 \\
 q_{i,11} &= (0.386)(9.3)(35.3 - 23.9) = 40.9 \\
 q_{i,12} &= (0.386)(9.3)(42.5 - 23.9) = 66.7 \\
 q_{i,13} &= (0.386)(9.3)(47.1 - 23.9) = 83.3 \\
 q_{i,14} &= (0.386)(9.3)(50.2 - 23.9) = 94.6 \\
 q_{i,15} &= (0.386)(9.3)(51.8 - 23.9) = 100.1 \\
 q_{i,16} &= (0.386)(9.3)(50.4 - 23.9) = 94.9 \\
 q_{i,17} &= (0.386)(9.3)(46.9 - 23.9) = 82.4 \\
 q_{i,18} &= (0.386)(9.3)(40.5 - 23.9) = 59.5 \\
 q_{i,19} &= (0.386)(9.3)(31.0 - 23.9) = 25.4 \\
 q_{i,20} &= (0.386)(9.3)(29.4 - 23.9) = 19.9
 \end{aligned}$$

$$\begin{aligned}
 q_{i,21} &= (0.386)(9.3)(28.3 - 23.9) = 15.9 \\
 q_{i,22} &= (0.386)(9.3)(27.2 - 23.9) = 12.0 \\
 q_{i,23} &= (0.386)(9.3)(26.1 - 23.9) = 8.0 \\
 q_{i,24} &= (0.386)(9.3)(25.0 - 23.9) = 4.0 \\
 \text{Total 24 h heat input} &= 779.1 \text{ W}
 \end{aligned}$$

These data are used with conduction time series to calculate the wall heat gain. From Table 20, the most similar wall construction is wall number 14. This is a brick and block wall that has similar mass and thermal capacity. Using Equation (38), the conduction time factors for wall 14 can be used in conjunction with the 24 h heat input profile calculated above to determine the wall heat gain at 3 P.M. Central Daylight Time (which is actually 2 P.M. or hour 14 local standard time):

Table 22 Thermal Properties and Code Numbers of Layers Used in Wall and Roof Descriptions for [Tables 20](#) and [21](#)

Layer ID	Description	Thickness, mm	Conductivity, W/(m·K)	Density, kg/m ³	Specific Heat, kJ/(kg·K)	Resistance, m ² ·K/W	R	Mass, kg/m ²	Thermal Capacity, W·h/(m ² ·K)	Notes
F01	Outside surface resistance	—	—	—	—	0.04	0.04	—	—	1
F02	Inside vertical surface resistance	—	—	—	—	0.12	0.12	—	—	2
F03	Inside horizontal surface resistance	—	—	—	—	0.16	0.16	—	—	3
F04	Wall air space resistance	—	—	—	—	0.15	0.15	—	—	4
F05	Ceiling air space resistance	—	—	—	—	0.18	0.18	—	—	5
F06	EIFS finish	9.5	0.72	1856	0.84	—	0.01	17.7	4.12	6
F07	25 mm stucco	25.4	0.72	1856	0.84	—	0.04	47.2	10.98	6
F08	Metal surface	0.8	45.28	7824	0.50	—	0.00	6.0	0.83	7
F09	Opaque spandrel glass	6.4	0.99	2528	0.88	—	0.01	16.1	3.39	8
F10	25 mm stone	25.4	3.17	2560	0.79	—	0.01	65.1	14.39	9
F11	Wood siding	12.7	0.09	592	1.17	—	0.14	7.5	2.45	10
F12	Asphalt shingles	3.2	0.04	1120	1.26	—	0.08	3.6	1.24	
F13	Built-up roofing	9.5	0.16	1120	1.46	—	0.06	10.7	4.35	
F14	Slate or tile	12.7	1.59	1920	1.26	—	0.01	24.4	8.52	
F15	Wood shingles	6.4	0.04	592	1.30	—	0.17	3.8	1.36	
F16	Acoustic tile	19.1	0.06	368	0.59	—	0.31	7.0	1.14	11
F17	Carpet	12.7	0.06	288	1.38	—	0.22	3.7	1.41	12
F18	Terrazzo	25.4	1.80	2560	0.79	—	0.01	65.1	14.39	13
G01	16 mm gyp board	15.9	0.16	800	1.09	—	0.10	12.7	3.85	
G02	16 mm plywood	15.9	0.12	544	1.21	—	0.14	8.6	2.92	
G03	13 mm fiberboard sheathing	12.7	0.07	400	1.30	—	0.19	5.1	1.83	14
G04	13 mm wood	12.7	0.15	608	1.63	—	0.08	7.7	3.51	15
G05	25 mm wood	25.4	0.15	608	1.63	—	0.17	15.5	7.01	15
G06	50 mm wood	50.8	0.15	608	1.63	—	0.33	30.9	14.03	15
G07	100 mm wood	101.6	0.15	608	1.63	—	0.66	61.8	28.05	15
I01	25 mm insulation board	25.4	0.03	43	1.21	—	0.88	1.1	0.37	16
I02	50 mm insulation board	50.8	0.03	43	1.21	—	1.76	2.2	0.74	16
I03	75 mm insulation board	76.2	0.03	43	1.21	—	2.64	3.3	1.11	16
I04	89 mm batt insulation	89.4	0.05	19	0.96	—	1.94	1.7	0.46	17
I05	154 mm batt insulation	154.4	0.05	19	0.96	—	3.34	3.0	0.79	17
I06	244 mm batt insulation	243.8	0.05	19	0.96	—	5.28	4.7	1.25	17
M01	100 mm brick	101.6	0.89	1920	0.79	—	0.11	195.2	43.16	18
M02	150 mm LW concrete block	152.4	0.49	512	0.88	—	0.31	78.1	19.08	19
M03	200 mm LW concrete block	203.2	0.50	464	0.88	—	0.41	94.3	23.06	20
M04	300 mm LW concrete block	304.8	0.71	512	0.88	—	0.43	156.2	38.16	21
M05	200 mm concrete block	203.2	1.11	800	0.92	—	0.18	162.7	41.65	22
M06	300 mm concrete block	304.8	1.40	800	0.92	—	0.22	244.0	62.47	23
M07	150 mm LW concrete block (filled)	152.4	0.29	512	0.88	—	0.53	78.1	19.08	24
M08	200 mm LW concrete block (filled)	203.2	0.26	464	0.88	—	0.78	94.3	23.06	25
M09	300 mm LW concrete block (filled)	304.8	0.29	512	0.88	—	1.04	156.2	38.16	26
M10	200 mm concrete block (filled)	203.2	0.72	800	0.92	—	0.28	162.7	41.65	27
M11	100 mm lightweight concrete	101.6	0.53	1280	0.84	—	0.19	130.1	30.29	
M12	150 mm lightweight concrete	152.4	0.53	1280	0.84	—	0.29	195.2	45.43	
M13	200 mm lightweight concrete	203.2	0.53	1280	0.84	—	0.38	260.3	60.58	
M14	150 mm heavyweight concrete	152.4	1.95	2240	0.90	—	0.08	341.6	85.47	
M15	200 mm heavyweight concrete	203.2	1.95	2240	0.90	—	0.10	455.5	113.96	
M16	300 mm heavyweight concrete	304.8	1.95	2240	0.90	—	0.16	683.2	170.94	
M17	50 mm LW concrete roof ballast	50.8	0.19	640	0.84	—	0.27	32.5	7.57	28

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

1. [Chapter 25, Table 1](#) for 3.4 m/s wind
2. [Chapter 25, Table 1](#) for still air, horizontal heat flow
3. [Chapter 25, Table 1](#) for still air, downward heat flow
4. [Chapter 25, Table 3](#) for 40 mm space, 32.2°C, horizontal heat flow, 0.82 emittance
5. [Chapter 25, Table 3](#) for 90 mm space, 32.2°C, downward heat flow, 0.82 emittance
6. EIFS finish layers approximated by [Chapter 25, Table 4](#) for 10 mm cement plaster, sand aggregate
7. [Chapter 38, Table 3](#) for steel (mild), 22 gage
8. [Chapter 25, Table 4](#) for architectural glass
9. [Chapter 25, Table 4](#) for marble and granite
10. [Chapter 25, Table 4](#), density assumed same as Southern pine
11. [Chapter 25, Table 4](#) for mineral fiberboard, wet molded, acoustical tile
12. [Chapter 25, Table 4](#) for carpet and rubber pad, density assumed same as fiberboard
13. [Chapter 25, Table 4](#), density assumed same as stone

14. [Chapter 25, Table 4](#) for nail-base sheathing
15. [Chapter 25, Table 4](#) for Southern pine
16. [Chapter 25, Table 4](#) for expanded polystyrene
17. [Chapter 25, Table 4](#) for glass fiber batt, specific heat per glass fiber board
18. [Chapter 25, Table 4](#) for clay fired brick
19. [Chapter 25, Table 4](#), 7.3 kg block, 200 mm × 400 mm face
20. [Chapter 25, Table 4](#), 8.6 kg block, 200 mm × 400 mm face
21. [Chapter 25, Table 4](#), 14.5 kg block, 200 mm × 400 mm face
22. [Chapter 25, Table 4](#), 15 kg normal weight block, 200 mm × 400 mm face
23. [Chapter 25, Table 4](#), 22.7 kg normal weight block, 200 mm × 400 mm face
24. [Chapter 25, Table 4](#), 7.3 kg block, vermiculite fill
25. [Chapter 25, Table 4](#), 8.6 kg block, 200 mm × 400 mm face, vermiculite fill
26. [Chapter 25, Table 4](#), 14.5 kg block, 200 mm × 400 mm face, vermiculite fill
27. [Chapter 25, Table 4](#), 15 kg normal weight block, 200 mm × 400 mm face, vermiculite fill
28. [Chapter 25, Table 4](#) for 640 kg/m³ LW concrete

Table 23 Wall Heat Gain for Example 4

Wall Area = 9.3 m ²					
Inside Temperature = 23.9°C					
Wall U-Factor = 0.386					
CTS Table 20					
Wall Number: 14					
Hour	Sol-Air Temp., °C	Inside Temp., °C	Heat Input, W	CTS	Heat Gain, W
1	24.4	23.9	2.0	1%	47.8
2	24.4	23.9	2.0	1%	43.8
3	23.9	23.9	—	2%	39.5
4	23.3	23.9	2.0	5%	35.2
5	23.3	23.9	2.0	8%	30.8
6	24.3	23.9	1.6	9%	26.7
7	25.8	23.9	6.8	9%	23.0
8	27.6	23.9	13.2	9%	19.6
9	29.8	23.9	21.1	8%	16.7
10	31.9	23.9	28.8	7%	14.5
11	35.3	23.9	40.9	7%	13.0
12	42.5	23.9	66.7	6%	12.8
13	47.1	23.9	83.3	5%	13.5
14	50.2	23.9	94.6	4%	15.5
15	51.8	23.9	100.1	4%	19.4
16	50.4	23.9	94.9	3%	25.0
17	46.9	23.9	82.4	3%	31.7
18	40.5	23.9	59.5	2%	38.9
19	31.0	23.9	25.4	2%	45.6
20	29.4	23.9	19.9	2%	51.0
21	28.3	23.9	15.9	1%	54.4
22	27.2	23.9	12.0	1%	55.3
23	26.1	23.9	8.0	1%	54.0
24	25.0	23.9	4.0	0%	51.3
			779.1	100%	779.1

$$\begin{aligned}
 q_{14} &= c_0 q_{i,14} + c_1 q_{i,13} + c_2 q_{i,12} + c_3 q_{i,11} + \dots + c_{23} q_{i,15} \\
 &= (0.01)(94.6) + (0.01)(83.3) + (0.02)(66.7) + (0.05)(40.9) + \\
 &\quad (0.08)(28.8) + (0.09)(21.1) + (0.09)(13.2) + (0.09)(6.8) + \\
 &\quad (0.08)(1.6) + (0.07)(-2) + (0.07)(-2) + (0.06)(0) + (0.05)(2) + \\
 &\quad (0.04)(2) + (0.04)(4) + (0.03)(8) + (0.03)(12) + (0.02)(15.9) + \\
 &\quad (0.02)(19.9) + (0.02)(25.4) + (0.01)(59.4) + (0.01)(82.4) + \\
 &\quad (0.01)(94.9) + (0.0)(93.1) \\
 &= 15.5 \text{ W}
 \end{aligned}$$

Due to the tedious calculations involved, use of a simple computer spreadsheet or other computer software implementing these calculations can reduce the effort involved. A spreadsheet is illustrated in [Table 23](#), calculating a 24 h heat gain profile for the data of this example.

Calculating Cooling Load Using RTS

The instantaneous cooling load is defined as the rate at which heat energy is convected to the zone air at a given point in time. The computation of cooling load is complicated by the radiant exchange between surfaces, furniture, partitions, and other mass in the zone. Most heat gain sources transfer energy by both convection and radiation. Radiative heat transfer introduces to the process a time dependency that is not easily quantified. Radiation is absorbed by the thermal masses in the zone and then later transferred by convection into the space. This process creates a time lag and dampening effect. The convection portion of heat gains, on the other hand, is assumed to immediately become cooling load in the hour in which that heat gain occurs.

Heat balance procedures calculate the radiant exchange between surfaces based on their surface temperatures and emissivities, but they typically rely on estimated “radiative-convective splits” to determine the contribution of internal loads, including people, light-

ing, appliances, and equipment, to the radiant exchange. The radiant time series procedure further simplifies the heat balance procedure by also relying on an estimated radiative-convective split of wall and roof conductive heat gain instead of simultaneously solving for the instantaneous convective and radiative heat transfer from each surface, as is done in the heat balance procedure.

Thus, the cooling load for each load component (lights, people, walls, roofs, windows, appliances, etc.) for a particular hour is the sum of the convective portion of the heat gain for that hour plus the time-delayed portion of radiant heat gains for that hour and the previous 23 h. [Table 19](#) contains recommendations for splitting each of the heat gain components into convective and radiant portions.

The radiant time series method converts the radiant portion of hourly heat gains to hourly cooling loads using radiant time factors, the coefficients of the radiant time series. Radiant time factors are used to calculate the cooling load for the current hour on the basis of current and past heat gains. The radiant time series for a particular zone gives the time-dependent response of the zone to a single pulse of radiant energy. The series shows the portion of the radiant pulse that is convected to the zone air for each hour. Thus, r_0 represents the fraction of the radiant pulse convected to the zone air in the current hour r_1 in the previous hour, and so on. The radiant time series thus generated is used to convert the radiant portion of hourly heat gains to hourly cooling loads according to the following equation:

$$Q_{r,\theta} = r_0 q_{r,\theta} + r_1 q_{r,\theta-1} + r_2 q_{r,\theta-2} + r_3 q_{r,\theta-3} + \dots + r_{23} q_{r,\theta-23} \quad (39)$$

where

$Q_{r,\theta}$ = radiant cooling load (Q_r) for the current hour (θ), W

$q_{r,\theta}$ = radiant heat gain for the current hour, W

$q_{r,\theta-n}$ = radiant heat gain n hours ago, W

r_0, r_1 , etc. = radiant time factors

The radiant cooling load for the current hour, which is calculated using RTS and Equation (39), is added to the convective portion to determine the total cooling load for that component for that hour.

Radiant time factors are generated by a heat balance based procedure. A separate series of radiant time factors is theoretically required for each unique zone and for each unique radiant energy distribution function assumption. For most common design applications, RTS variation depends primarily on the overall massiveness of the construction and the thermal responsiveness of the surfaces the radiant heat gains strike.

One of the goals in developing RTS was to provide a simplified method that was based directly on the heat balance method; thus, it was deemed desirable to generate the RTS coefficients directly from a heat balance. To this end, a heat balance computer program was developed. This program, called Hbfort, which may be used to generate RTS coefficients, is included as part of *Cooling and Heating Load Calculation Principles* (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 analogous to the custom weighting factor generation procedure used by DOE 2.1 (Kerrisk et al. 1981; Sowell 1988a, 1988b). In both cases, a zone model is pulsed with a heat gain. In the case of DOE 2.1, the resulting loads are used to estimate the best values of the transfer function method weighting factors to most closely match the load profile. In the case of the procedure described here, a unit periodic heat gain pulse is used to generate loads for a 24 h period. As long as the heat gain pulse is a unit pulse, the resulting loads are equivalent to the RTS coefficients.

Two different series of radiant time factors are utilized—one for direct transmitted solar heat gain (radiant energy assumed to be distributed to the floor and furnishings only) and one for all other types of heat gains (radiant energy assumed to be uniformly distributed on all internal surfaces). Nonsolar RTS apply to radiant heat gains from people, lights, appliances, walls, roofs, and floors. Also, for diffuse

Table 24 Representative Nonsolar RTS Values for Light to Heavy Construction

% Glass																			Interior Zones					
	Light						Medium						Heavy						Light		Medium		Heavy	
	With Carpet			No Carpet			With Carpet			No Carpet			With Carpet			No Carpet			With Carpet	No Carpet	With Carpet	No Carpet	With Carpet	No Carpet
	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%
Hour	Radiant Time Factor, %																							
0	47	50	53	41	43	46	46	49	52	31	33	35	34	38	42	22	25	28	46	40	46	31	33	21
1	19	18	17	20	19	19	18	17	16	17	16	15	9	9	9	10	9	9	19	20	18	17	9	9
2	11	10	9	12	11	11	10	9	8	11	10	10	6	6	5	6	6	6	11	12	10	11	6	6
3	6	6	5	8	7	7	6	5	5	8	7	7	4	4	4	5	5	5	6	8	6	8	5	5
4	4	4	3	5	5	5	4	3	3	6	5	5	4	4	4	5	5	4	4	5	3	6	4	5
5	3	3	2	4	3	3	2	2	2	4	4	4	4	3	3	4	4	4	3	4	2	4	4	4
6	2	2	2	3	3	2	2	2	2	4	3	3	3	3	3	4	4	4	2	3	2	4	3	4
7	2	1	1	2	2	2	1	1	1	3	3	3	3	3	3	4	4	4	2	2	1	3	3	4
8	1	1	1	1	1	1	1	1	1	3	2	2	3	3	3	4	3	3	1	1	1	3	3	4
9	1	1	1	1	1	1	1	1	1	2	2	2	3	3	2	3	3	3	1	1	1	2	3	3
10	1	1	1	1	1	1	1	1	1	2	2	2	3	2	2	3	3	3	1	1	1	2	3	3
11	1	1	1	1	1	1	1	1	1	2	2	2	2	2	2	3	3	3	1	1	1	2	2	3
12	1	1	1	1	1	1	1	1	1	1	1	1	2	2	2	3	3	3	1	1	1	1	2	3
13	1	1	1	0	1	0	1	1	1	1	1	1	2	2	2	3	3	2	1	1	1	1	2	3
14	0	0	1	0	1	0	1	1	1	1	1	1	2	2	2	3	2	2	1	0	1	1	2	3
15	0	0	1	0	0	0	1	1	1	1	1	1	2	2	2	2	2	2	0	0	1	1	2	3
16	0	0	0	0	0	0	1	1	1	1	1	1	2	2	2	2	2	2	0	0	1	1	2	3
17	0	0	0	0	0	0	1	1	1	1	1	1	2	2	2	2	2	2	0	0	1	1	2	2
18	0	0	0	0	0	0	1	1	1	1	1	1	2	2	1	2	2	2	0	0	1	1	2	2
19	0	0	0	0	0	0	0	1	0	0	1	1	2	2	1	2	2	2	0	0	1	0	2	2
20	0	0	0	0	0	0	0	0	0	0	1	1	2	1	1	2	2	2	0	0	0	0	2	2
21	0	0	0	0	0	0	0	0	0	0	1	1	2	1	1	2	2	2	0	0	0	0	2	2
22	0	0	0	0	0	0	0	0	0	0	1	0	1	1	1	2	2	2	0	0	0	0	1	2
23	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	2	2	1	0	0	0	0	1	2
	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 25 Representative Solar RTS Values for Light to Heavy Construction

% Glass	Light						Medium						Heavy											
	With Carpet			No Carpet			With Carpet			No Carpet			With Carpet			No Carpet								
	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%
Hour	Radiant Time Factor, %																							
0	53	55	56	44	45	46	52	54	55	28	29	29	47	49	51	26	27	28						
1	17	17	17	19	20	20	16	16	15	15	15	15	11	12	12	12	13	13						
2	9	9	9	11	11	11	8	8	8	10	10	10	6	6	6	7	7	7						
3	5	5	5	7	7	7	5	4	4	7	7	7	4	4	3	5	5	5						
4	3	3	3	5	5	5	3	3	3	6	6	6	3	3	3	4	4	4						
5	2	2	2	3	3	3	2	2	2	5	5	5	2	2	2	4	4	4						
6	2	2	2	3	2	2	2	1	1	4	4	4	2	2	2	3	3	3						
7	1	1	1	2	2	2	1	1	1	4	3	3	2	2	2	3	3	3						
8	1	1	1	1	1	1	1	1	1	3	3	3	2	2	2	3	3	3						
9	1	1	1	1	1	1	1	1	1	3	3	3	2	2	2	3	3	3						
10	1	1	1	1	1	1	1	1	1	2	2	2	2	2	2	3	3	3						
11	1	1	1	1	1	1	1	1	1	2	2	2	2	2	1	3	3	2						
12	1	1	1	1	1	0	1	1	1	2	2	2	2	1	1	2	2	2						
13	1	1	0	1	0	0	1	1	1	2	2	2	2	1	1	2	2	2						
14	1	0	0	0	0	0	1	1	1	1	1	1	2	1	1	2	2	2						
15	1	0	0	0	0	0	1	1	1	1	1	1	1	1	1	2	2	2						
16	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	2	2	2						
17	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	2	2	2						
18	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	2	2	2						
19	0	0	0	0	0	0	0	0	0	1	1	1	1	1	1	2	2	2						
20	0	0	0	0	0	0	0	0	0	1	1	1	1	1	1	2	2	2						
21	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	2	2	2						
22	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	2	1	1						
23	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	2	1	1						
	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100						

Table 26 RTS Representative Zone Construction for [Tables 24](#) and [25](#)

Construction Class	Exterior Wall	Roof/Ceiling	Partitions	Floor	Furnishings
Light	steel siding, 50 mm insulation, air space, 19 mm gyp	100 mm LW concrete, ceiling air space, acoustic tile	19 mm gyp, air space, 19 mm gyp	acoustic tile, ceiling air space, 100 mm LW concrete	25 mm wood @ 50% of floor area
Medium	100 mm face brick, 50 mm insulation, air space, 19 mm gyp	100 mm HW concrete, ceiling air space, acoustic tile	19 mm gyp, air space, 19 mm gyp	acoustic tile, ceiling air space, 100 mm HW concrete	25 mm wood @ 50% of floor area
Heavy	100 mm face brick, 200 mm HW concrete air space, 50 mm insulation, 19 mm gyp	200 mm HW concrete, ceiling air space, acoustic tile	19 mm gyp, 200 mm HW concrete block, 19 mm gyp	acoustic tile, ceiling air space, 200 mm HW concrete	25 mm wood @ 50% of floor area

Table 27 Wall Heat Gain and Cooling Load for Example 5

Wall Area = 9.3 m ² Inside Temp. = 23.9°C Wall U-factor = 0.386			CTS Table 20 Wall Number: 14			RTS medium-weight construction, 50% glass, with carpet				
Hour	Sol-Air Temp	Inside Temp	Heat Input	CTS	Heat Gain	Split Heat Gain		Nonsolar RTS	Radiant Cooling Load	Total Cooling Load
						Convective %	Radiant %			
						37%	63%			
1	24.4	23.9	2.0	1%	47.8	17.7	30.1	49%	29.0	46.7
2	24.4	23.9	2.0	1%	43.8	16.2	27.6	17%	27.4	43.6
3	23.9	23.9	—	2%	39.5	14.6	24.9	9%	25.6	40.2
4	23.3	23.9	2.0	5%	35.2	13.0	22.2	5%	23.7	36.7
5	23.3	23.9	2.0	8%	30.8	11.4	19.4	3%	21.6	33.0
6	24.3	23.9	1.6	9%	26.7	9.9	16.8	2%	19.6	29.5
7	25.8	23.9	6.8	9%	23.0	8.5	14.5	2%	17.7	26.2
8	27.6	23.9	13.2	9%	19.6	7.3	12.4	1%	15.9	23.2
9	29.8	23.9	21.1	8%	16.7	6.2	10.5	1%	14.3	20.5
10	31.9	23.9	28.8	7%	14.5	5.4	9.1	1%	13.0	18.3
11	35.3	23.9	40.9	7%	13.0	4.8	8.2	1%	11.9	16.7
12	42.5	23.9	66.7	6%	12.8	4.7	8.0	1%	11.3	16.0
13	47.1	23.9	83.3	5%	13.5	5.0	8.5	1%	11.2	16.2
14	50.2	23.9	94.6	4%	15.5	5.8	9.8	1%	11.6	17.4
15	51.8	23.9	100.1	4%	19.4	7.2	12.3	1%	12.8	20.0
16	50.4	23.9	94.9	3%	25.0	9.2	15.7	1%	14.8	24.1
17	46.9	23.9	82.4	3%	31.7	11.7	19.9	1%	17.5	29.2
18	40.5	23.9	59.5	2%	38.9	14.4	24.5	1%	20.7	35.1
19	31.0	23.9	25.4	2%	45.6	16.9	28.7	1%	23.9	40.8
20	29.4	23.9	19.9	2%	51.0	18.9	32.2	1%	26.9	45.8
21	28.3	23.9	15.9	1%	54.4	20.1	34.3	0%	29.2	49.3
22	27.2	23.9	12.0	1%	55.3	20.4	34.8	0%	30.4	50.9
23	26.1	23.9	8.0	1%	54.0	20.0	34.0	0%	30.7	50.7
24	25.0	23.9	4.0	0%	51.3	19.0	32.3	0%	30.1	49.1
			779.1	100%	779.1	288.3	490.8	100%	490.8	779.1

solar heat gain and direct solar heat gain from fenestration with inside shading (blinds, drapes, etc.), the nonsolar RTS should be used. Radiation from those sources is assumed to be more uniformly distributed onto all room surfaces. Effect of beam solar radiation distribution assumptions is addressed by Hittle (1999).

Representative RTS data for light, medium, and heavyweight constructions are provided in [Tables 24](#) and [25](#). Those were calculated using the Hbfort computer program (Pedersen et al. 1998) with zone characteristics listed in [Table 26](#). Customized RTS values may be calculated using the HB method where the zone is not reasonably similar to these typical zones or where more precision is desired.

ASHRAE *Research Project 942* compared HB and RTS results over a wide range of zone types and input variables (Spitler et al. 1998, Rees et al. 2000). In general, total cooling loads calculated using the RTS method closely agreed with or were slightly higher than those of the HB method with the same inputs. The project examined more than 5000 test cases of varying zone parameters. The dominating variable was overall thermal mass, and results were grouped into lightweight, U.S. medium-weight, U.K. medium-weight, and heavyweight construction. Best agreement between RTS and HB results was obtained for light- and medium-weight

construction. Greater differences occurred in heavyweight cases, with RTS generally predicting slightly higher peak cooling loads than HB. Greater differences also were observed in zones with extremely high internal radiant loads and large glazing areas or with a very lightweight exterior envelope. In this case, heat balance calculations predict that some of the internal radiant load will be transmitted to the outdoor environment and never becomes cooling load within the space. The RTS methodology does not account for energy being transferred out of the space to the environment and thus predicted higher cooling loads.

Example 5. Wall cooling load using radiant time series. Using the data from Example 4, calculate the cooling load at 3 P.M. Central Daylight Time on July 21 through 9.3 m² of a wall composed of 100 mm brick, 50 mm of insulation ($R = 1.76 \text{ m}^2 \cdot \text{K/W}$), and 200 mm lightweight concrete block in an office building of average commercial construction.

Solution: 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, from Example 4, times the convective fraction for walls from [Table 19](#):

$$Q_c = (15.8)(37\%) = 5.85 \text{ W}$$

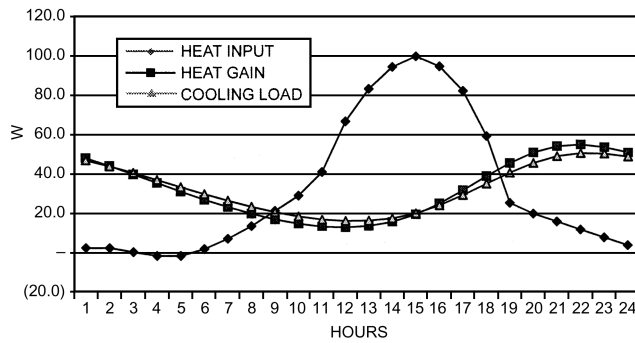


Fig. 12 Wall Heat Input, Heat Gain for Example 5 and Cooling Load Using CTS and RTS

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 19](#) (63%), and radiant time series from [Table 24](#), in accordance with Equation (39). From [Table 24](#), select the RTS for medium-weight construction, assuming 50% glass and carpeted floors as is often found in modern office construction. Take the wall heat gains from [Table 23](#). Thus, the radiant cooling load for the wall is

$$\begin{aligned} Q_{r,14} &= r_0(0.63)q_{i,14} + r_1(0.63)q_{i,13} + r_2(0.63)q_{i,12} + r_3(0.63)q_{i,11} + \\ &\quad \dots + r_{23}(0.63)q_{i,15} \\ &= (0.49)(0.63)(15.5) + (0.17)(0.63)(13.5) + (0.09)(0.63)(12.8) \\ &\quad + (0.05)(0.63)(13) + (0.03)(0.63)(14.5) + (0.02)(0.63)(16.7) \\ &\quad + (0.02)(0.63)(19.6) + (0.01)(0.63)(23.0) + (0.01)(0.63)(26.7) \\ &\quad + (0.01)(0.63)(30.8) + (0.01)(0.63)(35.2) + (0.01)(0.63)(39.5) \\ &\quad + (0.01)(0.63)(43.8) + (0.01)(0.63)(47.8) + (0.01)(0.63)(51.3) \\ &\quad + (0.01)(0.63)(54.0) + (0.01)(0.63)(55.3) + (0.01)(0.63)(54.4) \\ &\quad + (0.01)(0.63)(51.0) + (0.01)(0.63)(45.6) + (0.00)(0.63)(38.9) \\ &\quad + (0.00)(0.63)(31.7) + (0.00)(0.63)(25.0) + (0.00)(0.63)(19.4) \\ &= 11.6 \text{ W} \end{aligned}$$

The total wall cooling load at the designated hour is thus

$$Q_{\text{wall}} = Q_c + Q_{r,14} = 5.8 + 11.6 = 17.4 \text{ W}$$

Again, due to the tedious calculations involved, use of a simple computer spreadsheet or other computer software implementing these calculations can reduce the effort involved. The spreadsheet illustrated in [Table 23](#) is expanded in [Table 27](#) 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 hour cooling load profile for the data of this example. [Figure 12](#) shows wall heat input, heat gain, and total cooling load versus time for the 24 h calculated.

Example 6. Window cooling load using radiant time series. Calculate the cooling load for a 1.858 m² double-glazed bronze low-e window at 3 P.M. Central Daylight Time on July 21, at the location and for the conditions defined in Example 1, in an office building of average commercial construction. Use SHGC data for glass type 17f from [Table 13 in Chapter 30](#). Use window U-factor = 3.18.

Solution: To determine the window cooling load, 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 at the designated time. The window heat gain components are calculated using Equations (13) through (15).

From Example 1, at hour 14 standard time (3 P.M. Central Daylight Time):

$$E_D = 425 \text{ W/m}^2$$

$$E_d = 98 \text{ W/m}^2$$

$$E_r = 85 \text{ W/m}^2$$

$$\theta = 60.1^\circ$$

From [Chapter 30, Table 13](#), for glass type 17f,

$$\text{SHGC}(\theta) = \text{SHGC}(60.1) = 0.349 \text{ (interpolated)}$$

$$\langle \text{SHGC} \rangle_D = 0.38$$

Therefore, window heat gain components for hour 14 are

$$q_{b,14} = AE_D \text{SHGC}(\theta) = (1.858)(425)(0.349) = 276 \text{ W}$$

$$q_{d,14} = A(E_d + E_r) \langle \text{SHGC} \rangle_D = (1.858)(98 + 85)(0.38) = 129 \text{ W}$$

$$q_{c,14} = UA(t_{\text{out}} - t_{\text{in}}) = (3.18)(1.858)(34.4 - 23.9) = 62.4$$

This procedure is repeated to determine these values for a 24 h heat gain profile, as illustrated in [Table 28](#).

Total cooling load for the window is calculated by summing the convective and radiant portions. For windows with inside 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 inside 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 inside shading, is treated as 100% radiant, and solar RTS factors from [Table 25](#) 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 convection portions according to [Table 19](#) percentages, and nonsolar RTS factors from [Table 24](#) 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 25](#), in accordance with Equation (39). From [Table 25](#), select the solar RTS for medium-weight construction, assuming 50% glass and carpeted floors as is often found in modern office construction. Using [Table 28](#) values for direct solar heat gain, the radiant cooling load for the window direct beam solar component is

$$\begin{aligned} Q_{b,14} &= r_0q_{b,14} + r_1q_{b,13} + r_2q_{b,12} + r_3q_{b,11} + \dots + r_{23}q_{b,15} \\ &= (0.54)(276) + (0.16)(175) + (0.08)(82) + (0.04)(1) \\ &\quad + (0.03)(0) + (0.02)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) \\ &\quad + (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) \\ &\quad + (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) \\ &\quad + (0.00)(6) + (0.00)(143) + (0.00)(262) + (0.00)(326) \\ &\quad + (0.00)(329) \\ &= 184 \text{ W} \end{aligned}$$

This process is repeated for other hours; results are listed in [Table 29](#).

For diffuse and conductive heat gains, the radiant fraction according to [Table 19](#) is 63%. The radiant portion is processed using nonsolar RTS coefficients from [Table 24](#). The results are listed in [Table 29](#).

The total window cooling load at the designated hour is thus

$$Q_{\text{window}} = Q_b + Q_{\text{diff} + \text{cond}} = 184 + 170 = 354 \text{ W}$$

Again, due to the tedious calculations involved, use of a simple computer spreadsheet or other computer software implementing these calculations can reduce the effort involved. The spreadsheet illustrated in [Table 28](#) is expanded in [Table 29](#) 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 the data of this example.

Example 7. Internal cooling load using radiant time series. Calculate the cooling load at 10 A.M. for a 100 m² office space due to suspended fluorescent lights with a total load of 2000 W, in an office building of average commercial construction. The lights are 50% on at 7 A.M., 100% on from 8 A.M. to 6 P.M., then off the remainder of the time.

Solution: To determine the lighting cooling load, first calculate the 24 h heat gain profile for the lighting load, 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. The lighting heat gain profile, based on the schedule indicated, is

$$\begin{aligned} q_1 &= (2000 \text{ W})(0\%) &= 0 \\ q_2 &= (2000 \text{ W})(0\%) &= 0 \\ q_3 &= (2000 \text{ W})(0\%) &= 0 \\ q_4 &= (2000 \text{ W})(0\%) &= 0 \\ q_5 &= (2000 \text{ W})(0\%) &= 0 \\ q_6 &= (2000 \text{ W})(0\%) &= 0 \end{aligned}$$

Table 28 Solar Calculations—Solar Heat Gain Using SHGC—for Example 6

									Equa- tion of Time,		Declina- tion		A		B		C		Time Zone		Std. Merid- ian	
					Surface:										(Dimensionless Ratios)							
					Azimuth 45		Month		min.		degrees		W/m ²						1 Atlantic		60	
					Tilt 90		1		-11.2		-20		1230		0.142		0.058		2 Eastern		75	
							2		-13.9		-10.8		1215		0.144		0.060		3 Central		90	
					Fenestration		3		-7.5		0		1186		0.156		0.071		4 Mountain		105	
					Area1.858 m ²		4		1.1		11.6		1136		0.180		0.097		5 Pacific		120	
					U 0.56		5		3.3		20		1104		0.196		0.121		6 Alaska		135	
					IAC 1.0														7 Hawaii		150	
					Angle SHGC:		6		-1.4		23.45		1088		0.205		0.134					
					0 0.450		7		-6.2		20.6		1085		0.207		0.136					
For Month = 7					40 0.420		8		-2.4		12.3		1107		0.201		0.122					
Equation of Time = -6.2					50 0.400		9		7.5		0		1151		0.177		0.092					
Declination = 20.6					60 0.350		10		15.4		-10.5		1192		0.160		0.073					
A = 344					70 0.270		11		13.8		-19.8		1221		0.149		0.063					
B = 0.207					80 0.140		12		1.6		-23.45		1233		0.142		0.057					
C = 0.136					90 —																	
Local Standard Time Meridian = 90					Hemis: 0.380																	
					Direct Beam Solar Heat Gain					Diffuse Solar Heat Gain								Conduction		Total		
Local Standard Hour	Apparent Solar Time, hours	Hour Angle	Solar Altitude	Solar Azimuth	Direct Normal Irradiance	Surface Incident Angle	Surface Direct, W/m ²	Direct SHGC	Direct Solar Heat Gain, W	Ground Diffuse	Y Ratio	Sky Diffuse	Total Diffuse, W/m ²	Hemis. SHGC	Diffuse Solar Heat Gain, W	Outside Temp.	Conductive Heat Gain, W	Window Heat Gain, W				
1	0.90	166.55	-28.1	-165.7	—	139.3	0.0	—	0.0	0.0	0.45	0.00	0.0	0.380	0.0	24.4	3.3	3				
2	1.90	151.55	-23.8	-150.8	—	151.6	0.0	—	0.0	0.0	0.45	0.00	0.0	0.380	0.0	24.4	3.3	3				
3	2.90	136.55	-17.1	-137.7	—	162.7	0.0	—	0.0	0.0	0.45	0.00	0.0	0.380	0.0	23.9	0.0	0				
4	3.90	121.55	-8.6	-126.2	—	167.8	0.0	—	0.0	0.0	0.45	0.00	0.0	0.380	0.0	23.3	3.3	3				
5	4.90	106.55	1.3	-116.2	0	161.1	0.0	—	0.0	0.0	0.45	0.01	0.0	0.380	0.0	23.3	3.3	3				
6	5.90	91.55	11.9	-107.0	399	149.7	0.0	—	0.0	13.7	0.45	24.40	38.1	0.380	26.9	23.3	3.3	3				
7	6.90	76.55	23.1	-98.1	641	137.3	0.0	—	0.0	33.9	0.45	39.21	73.1	0.380	51.6	23.9	0.0	52				
8	7.90	61.55	34.6	-88.8	754	124.7	0.0	—	0.0	53.0	0.45	46.11	99.1	0.380	70.0	25.0	6.6	77				
9	8.90	46.55	46.0	-78.0	814	112.2	0.0	—	0.0	69.6	0.45	49.80	119.4	0.380	84.3	26.7	16.4	101				
10	9.90	31.55	56.8	-63.6	847	100.0	0.0	—	0.0	82.5	0.48	55.70	138.2	0.380	97.5	28.3	26.3	124				
11	10.90	16.55	66.0	-41.0	865	88.4	24.8	0.023	1.1	90.8	0.56	66.21	157.0	0.380	110.8	30.6	39.4	151				
12	11.90	1.55	70.6	4.4	871	75.4	220.1	0.200	81.9	94.0	0.68	80.61	174.6	0.380	123.3	32.2	49.2	254				
13	12.90	13.45	67.5	34.6	867	67.8	327.0	0.287	174.6	91.9	0.76	89.55	181.4	0.380	128.1	33.9	59.1	362				
14	13.90	28.45	58.9	59.8	852	60.1	425.0	0.349	275.9	84.6	0.85	98.02	182.6	0.380	128.9	34.4	62.4	467				
15	14.90	43.45	48.3	75.4	822	55.0	471.6	0.375	328.6	72.6	0.90	101.05	173.6	0.380	122.6	35.0	65.6	517				
16	15.90	58.45	37.0	86.7	769	53.4	458.4	0.383	326.2	56.7	0.92	96.40	153.1	0.380	108.1	34.4	62.4	497				
17	16.90	73.45	25.5	96.2	671	55.6	379.0	0.372	262.0	38.0	0.90	81.82	119.8	0.380	84.6	33.9	59.1	406				
18	17.90	88.45	14.2	105.1	467	61.1	225.4	0.341	142.7	17.8	0.83	52.96	70.8	0.380	50.0	32.8	52.5	245				
19	18.90	103.45	3.4	114.2	33	69.3	11.8	0.276	6.0	0.6	0.74	3.36	4.0	0.380	2.8	30.6	39.4	48				
20	19.90	118.45	-6.6	124.0	—	79.1	0.0	0.151	0.0	0.0	0.64	0.00	0.0	0.380	0.0	29.4	32.8	33				
21	20.90	133.45	-15.5	135.2	—	90.2	0.0	—	0.0	0.0	0.55	0.00	0.0	0.380	0.0	28.3	26.3	26				
22	21.90	148.45	-22.6	147.9	—	101.9	0.0	—	0.0	0.0	0.45	0.00	0.0	0.380	0.0	27.2	19.7	20				
23	22.90	163.45	-27.5	162.5	—	114.2	0.0	—	0.0	0.0	0.45	0.00	0.0	0.380	0.0	26.1	13.1	13				
24	23.90	178.45	-29.4	178.3	—	126.7	0.0	—	0.0	0.0	0.45	0.00	0.0	0.380	0.0	25.0	6.6	7				

$$\begin{aligned}
 q_7 &= (2000 \text{ W})(50\%) &= 1000 \\
 q_8 &= (2000 \text{ W})(100\%) &= 2000 \\
 q_9 &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{10} &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{11} &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{12} &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{13} &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{14} &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{15} &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{16} &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{17} &= (2000 \text{ W})(100\%) &= 2000 \\
 q_{18} &= (2000 \text{ W})(100\%) &= 2000
 \end{aligned}$$

$$\begin{aligned}
 q_{19} &= (2000 \text{ W})(0\%) &= 0 \\
 q_{20} &= (2000 \text{ W})(0\%) &= 0 \\
 q_{21} &= (2000 \text{ W})(0\%) &= 0 \\
 q_{22} &= (2000 \text{ W})(0\%) &= 0 \\
 q_{23} &= (2000 \text{ W})(0\%) &= 0 \\
 q_{24} &= (2000 \text{ W})(0\%) &= 0
 \end{aligned}$$

The convective portion is simply the lighting heat gain for the hour being calculated times the convective fraction for unvented fluorescent lighting from [Table 19](#):

$$Q_{c,10} = (2000)(33\%) = 660 \text{ W}$$

Table 29 Fenestration Cooling Load for Example 6

	Direct Beam Solar Cooling Load						Diffuse Solar and Conduction Cooling Load								Total
	Direct Solar Heat Gain, W	Convect., %	Rad., %	Solar RTS Coefficients from Table 25	Direct Solar Radiant Cooling Load	Direct Solar Total Cooling Load	Diffuse Solar Heat Gain	Cond. Heat Gain, W	Tot Diff. & Cond., W	Convect., %	Rad., %	Nonsolar RTS Coefficients from Table 24	Diff. & Cond. Radiant Cooling Load	Diff. & Cond. Total Cooling Load	Total Window Cooling Load
Hour		0%	100%							37%	63%				
1	0	—	—	54%	16	16	—	3.28299	3	1.21	2.07	49%	15	16	32
2	0	—	—	16%	16	16	—	3	3	1.21	2.07	17%	14	15	31
3	0	—	—	8%	16	16	—	0	0	0.00	0.00	9%	12	12	28
4	0	—	—	4%	16	16	—	−3	3	1.21	2.07	5%	10	9	25
5	0	—	—	3%	16	16	0	−3	3	1.21	2.06	3%	9	8	24
6	0	—	—	2%	16	16	27	−3	24	8.73	14.86	2%	16	25	41
7	0	—	—	1%	15	15	52	0	52	19.10	32.52	2%	26	46	61
8	0	—	—	1%	13	13	70	7	77	28.33	48.24	1%	26	66	79
9	0	—	—	1%	11	11	84	16	101	37.26	63.45	1%	49	86	97
10	0	—	—	1%	7	7	98	26	124	45.81	78.00	1%	60	106	113
11	1	—	4	1%	5	5	111	39	150	55.59	94.65	1%	73	128	133
12	82	—	1	1%	46	46	123	49	173	63.83	108.68	1%	84	148	194
13	175	—	82	1%	108	108	128	59	187	69.26	117.92	1%	94	163	271
14	276	—	175	1%	184	184	129	62	191	70.78	120.51	1%	100	170	354
15	329	—	276	1%	239	239	123	66	188	69.65	118.59	1%	102	172	410
16	326	—	329	1%	260	260	108	62	170	63.07	107.39	1%	98	161	422
17	262	—	326	1%	238	238	85	59	144	53.16	90.52	1%	90	143	381
18	143	—	262	1%	171	171	50	53	102	37.92	64.57	1%	74	112	283
19	6	—	143	1%	78	78	3	39	42	15.62	26.60	1%	50	66	144
20	0	—	—	0%	45	45	—	33	33	12.15	20.68	1%	39	51	95
21	0	—	—	0%	29	29	—	26	26	9.72	16.54	0%	31	41	70
22	0	—	—	0%	22	22	—	20	20	7.29	12.41	0%	26	33	55
23	0	—	—	0%	18	18	—	1	13	4.86	8.27	0%	22	26	44
24	0	—	—	0%	16	16	—	7	7	2.43	4.14	0%	18	20	36
	1599	—	1599	100%	1599	1599	1,190	634	1823	675	1149	100%	1149	1823	3422

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 19 (67%), and radiant time series from Table 24, in accordance with Equation (39). From Table 24, select the RTS for medium-weight construction, assuming 50% glass and carpeted floors as is often found in modern office construction. Thus, the radiant cooling load for lighting is

$$\begin{aligned}
 Q_{r,10} &= r_0(0.67)q_{10} + r_1(0.67)q_9 + r_2(0.67)q_8 + r_3(0.67)q_7 + \dots + r_{23}(0.67)q_{11} \\
 &= (0.49)(0.67)(2000) + (0.17)(0.67)(2000) \\
 &\quad + (0.09)(0.67)(2000) + (0.05)(0.67)(1000) + (0.03)(0.67)(0) \\
 &\quad + (0.02)(0.67)(0) + (0.02)(0.67)(0) + (0.01)(0.67)(0) \\
 &\quad + (0.01)(0.67)(0) + (0.01)(0.67)(0) + (0.01)(0.67)(0) \\
 &\quad + (0.01)(0.67)(0) + (0.01)(0.67)(0) + (0.01)(0.67)(0) \\
 &\quad + (0.01)(0.67)(0) + (0.01)(0.67)(0) + (0.01)(0.67)(2000) \\
 &\quad + (0.01)(0.67)(2000) + (0.01)(0.67)(2000) \\
 &\quad + (0.01)(0.67)(2000) + (0.00)(0.67)(2000) \\
 &\quad + (0.00)(0.67)(2000) + (0.00)(0.67)(2000) \\
 &\quad + (0.00)(0.67)(2000) \\
 &= 1092 \text{ W}
 \end{aligned}$$

The total lighting cooling load at the designated hour is thus

$$Q_{wall} = Q_{c,10} + Q_{r,10} = 660 + 1092 = 1752 \text{ W}$$

COMPARISON WITH PREVIOUS METHODS

Some users of this chapter during the past eight editions may have been disappointed that it did not contain (nor was it accompanied by) fully developed multiple-room, multiple-system calculation computer programs for immediate use by HVAC

designers. Nor have the several attempts to create and present hand-calculation procedures been acceptable in practice, as they were simply incapable of dealing realistically with contemporary demands. As ASHRAE evolves in the 21st century, the issue of practical load calculation procedures clearly demands computer operation.

Who should provide the computer applications for this purpose? While it could be argued that ASHRAE itself is the best source of experience and talent to generate such tools, the nature of ASHRAE as a technical society prevents this type of activity. Instead, ASHRAE has provided technically reliable mechanisms for both the heat balance and the radiant time series procedures to be accurately implemented, as the most scientifically correct methodologies available, anticipating that various qualified private enterprises will take these frameworks and complete their development into practical applications for HVAC designers.

The user may question what benefits may be expected now that the TFM, TETD/TA, and CLTD/CLF procedures presented in earlier chapter versions have been superseded (*not* invalidated or discredited). The primary benefit will be improved accuracy, with reduced dependency upon purely subjective input (such as determining a proper time-averaging period for TETD/TA, or ascertaining appropriate safety factors to add to the “rounded off” TFM results). As a generic example, the space sensible cooling load for the traditional little ASHRAE store building (used for example purposes since the 1940s) was calculated by means of the heat balance procedure and independently calculated by application of the radiant time series procedure, with each set of results plotted as one of the load profile curves of Figure 13. Also plotted on this chart are the corresponding curves produced by

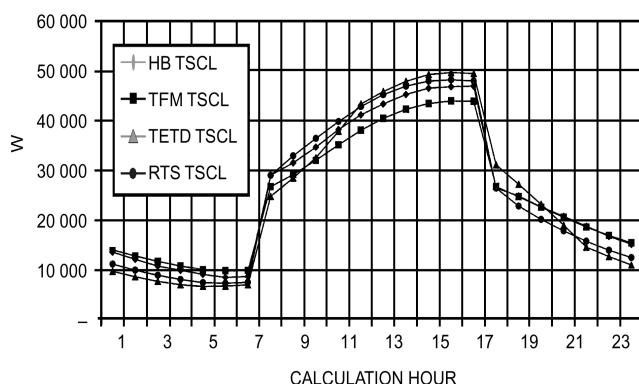


Fig. 13 Cooling Load Profile Comparison

the TFM and TETD/TA methodologies in the 1997 edition of this chapter. The user may draw his or her own conclusions from this chart.

HEATING LOAD PRINCIPLES

Techniques for estimating design heating load for commercial, institutional, and industrial applications are essentially the same as for those estimating design cooling loads for such uses, except that (1) temperatures outside the conditioned spaces are generally lower than the space temperatures maintained; (2) credit for solar heat gains or for internal heat gains is not included; and (3) the thermal storage effect of building structure or content is ignored. Heat losses (negative heat gains) are thus considered to be instantaneous, heat transfer essentially conductive, and latent heat treated only as a function of replacing space humidity lost to the exterior environment.

This simplified approach is justified because it evaluates “worst case” conditions that can reasonably occur during a heating season. The worst case is the load that must be met under design interior and exterior conditions, including infiltration and/or ventilation, but with no solar effect (at night or on cloudy winter days) and before the periodic presence of people, lights, and appliances has an offsetting effect.

Safety Factors and Load Allowances. Before mechanical cooling became a usual procedure, 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 occasional other safety factors reflecting the experience and/or concern of the individual designer. Such measures are rarely used today, with the uncompensated net heating load normally considered as having an adequate margin for error. Armstrong et al. (1992a, 1992b) provide a design method to deal with warm-up and cool-down load.

Cooling Needs During Noncooling Months. Perimeter spaces exposed to high solar heat gain often justify mechanical cooling during sunlit portions of traditional heating months, as do completely interior spaces with significant internal heat gain. Such spaces can also have significant heating loads during nonsunlit hours or after periods of nonoccupancy when adjacent spaces have cooled below interior design temperatures. The loads involved can be estimated conventionally to offset or to compensate for them and prevent overheating, but they have no direct relationship to design heating loads for the spaces in question.

Other Considerations. Calculation of design heating load estimates for this general category of applications has essentially

become a subset of the more involved and complex estimation of cooling loads for such spaces. Chapter 31 discusses using the heating load estimate to predict or analyze energy consumption over time. Special provisions to deal with special problems are covered in the *ASHRAE Handbook—Applications* and the *ASHRAE Handbook—Systems and Equipment*.

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