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CHAPTER 18

NONRESIDENTIAL COOLING AND HEATING LOAD CALCULATIONS

HEATING and cooling load calculations are the primary design basis for most heating and air-conditioning systems and components. [redacted]

Simply put, heating and cooling loads are the rates of energy input (heating) or removal (cooling) required to maintain an indoor environment at a desired temperature and humidity condition. Heating and air conditioning systems are designed, sized, and controlled to accomplish that energy transfer [redacted]

Peak design heating and cooling load calculations, which are this chapter's focus, seek to determine the maximum rate of heating and cooling energy transfer needed to maintain the space conditions at the desired level (set point). Similar principles, but with different assumptions, data, and application, can be used to estimate building energy consumption, as described in Chapter 19.

This chapter discusses common elements of cooling and heating load calculation [redacted]

1. COOLING LOAD CALCULATION PRINCIPLES

Cooling loads result from many conduction, convection, and radiation heat transfer processes [redacted] Building components or contents that may affect cooling loads include the following:

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

1.1 TERMINOLOGY

The variables affecting cooling load calculations are numerous, often difficult to define precisely, and always intricately interrelated. Many cooling load components vary widely in magnitude, and possibly direction, during a 24 h period. Because these cyclic changes in load components often are not in phase with each other, each component must be analyzed to establish the maximum cooling load for a building or zone. A **zoned system** (i.e., one serving several independent areas, each with its own temperature control) [redacted] At some times of day during heating or intermediate seasons, some zones may require heating while others require cooling. The zones' ventilation, humidification, or dehumidification needs must also be considered.

The current terminology presented here has been developed over time based on th [redacted] ASHRAE research project RP-1729 (Moftakhari et al. 2020) [redacted] It is always important to distinguish between room load and HVAC system load. This is true for both convective-based and radiant cooling systems. This chapter deals with calculation of the room cooling load. For radiant cooling systems, those room loads should be applied to calculation of HVAC system loads using principles described in Chapter 6 of the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*. This current chapter is largely based on convective or air-system-based cooling, but some key differences that would exist with radiant systems will be pointed out.

Heat Flow Rates

In air-conditioning design, the following four related heat flow rates [redacted], must be differentiated.

Space Heat Gain. [redacted] Heat gain is classified by its mode of entry into the space and whether it is sensible or latent. **Entry modes** include [redacted] **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 [redacted]. To maintain a constant humidity ratio, water vapor must condense on the cooling apparatus and be removed at the same rate it is added to the space. [redacted] In selecting cooling equipment, distinguish between sensible and latent heat gain: every cooling

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

[redacted]

Radiant Heat Gain. Radiant heat gain will first be absorbed by surfaces that enclose the space (walls, floor, and ceiling) and objects in the space (furniture, etc.). With convective or air-based cooling systems, these surfaces and objects [redacted]. The thermal storage effect is critical in differentiating between instantaneous heat gain for a given space and its cooling load at that moment [redacted]

With radiant-based cooling systems, radiant gains are still absorbed by surfaces in the space but the process of conversion from radiant heat gains to cooling load can be very different than would occur with convective or air-based cooling systems [redacted] In these cases, though, the heat gain [redacted] ASHRAE research project RP-1729 (Moftakhari et al. 2020) [redacted]

The heat transfer mechanics [redacted] RP-1729 [redacted]

[redacted]

Space Cooling Load. This is the rate at which sensible and latent heat must be removed from the space to maintain a constant space air temperature and humidity [redacted] as described previously.

Space Heat Extraction Rate. The rates at which sensible and latent heat are removed from the conditioned space equal the space cooling load when the room air temperature and humidity are constant [redacted]

Cooling Coil Load. The rate at which energy is removed at a cooling coil serving one or more conditioned spaces equals the sum of instantaneous space cooling loads [redacted]

Time Delay Effect

Heat gain absorbed by walls, floor, furniture, etc., contributes to space cooling load only after it has been converted to cooling load, as described previously [redacted]

For other than purely convective heat gains, there is some delay between the time a heat source is activated and the point when the rate of conversion to cooling load equals the rate of heat gain (steady-state) [redacted]

Accounting for the time delay effect is the major challenge in cooling load calculations. Several methods, including the two presented in this chapter, have been developed to take the time delay effect into consideration.

1.2 COOLING LOAD CALCULATION METHODS

This chapter presents two load calculation methods that vary significantly from previous methods. The technology involved, however [redacted] is not new. [redacted] Both methods are explained in their respective sections.

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

Cooling Load Calculations in Practice

Load calculations should accurately describe the building. All load calculation inputs should be as accurate as reasonable, without using safety factors. Introducing compounding safety factors at multiple levels in the load calculation results in an unrealistic and oversized load.

Variation in heat transmission coefficients of typical building materials and composite assemblies, differing motivations and skills of those who construct the building, unknown infiltration rates, and the manner in which the building is actually operated are some of the variables that make precise calculation impossible. Even if the designer uses reasonable procedures to account for these factors, the calculation can never be more than a good estimate of the actual load [redacted] An example is a cooling load estimate for a new building with many floors of unleased spaces for which detailed partition requirements, furnishings, lighting, and layout cannot be predefined. Potential tenant modifications once the building is occupied also must be considered. Load estimating requires proper engineering judgment that includes a thorough understanding of heat balance fundamentals.

Perimeter spaces exposed to high solar heat gain often need cooling during sunlit portions of traditional heating months, as do completely interior spaces with significant internal heat gain. [redacted]

Correct design and sizing of air-conditioning systems require more than calculation of the cooling load in the space to be conditioned [redacted]

System design could be driven by either sensible or latent load, and both need to be checked. In a sensible-load-driven space (the most common case), the cooling supply air has surplus capacity to dehumidify, but this is usually permissible. For a space driven by latent load (e.g., an auditorium), supply airflow based on sensible load is likely not to have enough dehumidifying capability, so subcooling and reheating or some other dehumidification process is needed.

This chapter is primarily concerned with a given space or zone in a building. When estimating loads for a group of spaces (e.g., for an air-handling system that serves multiple zones) [redacted] With large buildings that involve more than a single HVAC system, simultaneous loads and any additional diversity also must be considered when designing the central equipment that serves the systems. 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.

1.3 DATA ASSEMBLY

Calculating space cooling loads requires detailed building design information and weather data at design conditions. Generally, the following information should be compiled.

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

Configuration. Determine building location, orientation, and external shading from building plans and specifications. Shading from adjacent buildings can be determined from a site plan or by visiting the proposed site [redacted]

Outdoor Design Conditions. Obtain appropriate weather data, and select outdoor design conditions [redacted] Use judgment to ensure that results are consistent with expectations [redacted] The relationship between space and system loads is discussed further in following sections of the chapter.

To estimate conductive heat gain through exterior surfaces and infiltration and outdoor air loads at any time, applicable outdoor dry- and wet-bulb temperatures must be used. Chapter 14 gives monthly cooling load design values of outdoor conditions for many locations [redacted]

Indoor Design Conditions. Select indoor dry-bulb temperature, indoor relative humidity, and ventilation rate [redacted]

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

Areas. Use consistent methods for calculation of building areas. For fenestration, the definition of a component's area must be consistent with associated ratings.

Gross surface area [redacted]

[redacted]

The outer-dimension procedure is expedient for load calculations, but it is not consistent with rigorous definitions used in building-related standards. The resulting differences do not introduce significant errors in this chapter's procedures.

Fenestration area. [redacted]

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

2. INTERNAL HEAT GAINS

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

2.1 PEOPLE

Table 1 gives representative rates at which sensible heat and moisture are emitted by humans in different states of activity. In high-density spaces, such as auditoriums, these sensible and latent heat gains comprise a large fraction of the total load[redacted]

[redacted]

2.2 LIGHTING

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

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

[redacted]

The **total light wattage** is obtained from the ratings of all lamps installed[redacted]

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

[redacted]

The **special allowance factor** is the ratio of the lighting fixtures' power consumption, including lamps and ballast, to the nominal power consumption of the lamps [redacted]

2.3 ELECTRIC MOTORS

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

[redacted]

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) [redacted]

The motor load factor is the fraction of the rated load delivered under the conditions of the cooling load estimate [redacted]

[redacted]

[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]

[redacted]

[redacted]

Radiation and Convection

[redacted]

2.4 APPLIANCES

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

Overloading or Underloading

Heat output of a motor is generally proportional to motor load, within rated overload limits[redacted]

[redacted]

Cooking Appliances

These appliances include common heat-producing cooking equipment found in conditioned commercial kitchens.

Fundamental Principles. In commercial kitchens, appliances are typically turned on at the beginning of each operating period and are not turned off until closing time [redacted]

[redacted]

or

[redacted]

Heat Gain from Unhooded Cooking Appliances[redacted]

Heat Gain from Hooded Cooking Appliances[redacted]

Warewashing Applications. Typically, hot-water sanitizing and conveyor-type dish machines have either a dishwasher/condensing hood or direct-connected ductwork. If the ventilation is not operating properly, there are significant sensible and latent gains to the space[redacted]

Recirculating Systems[redacted]

Recommended Heat Gain Values[redacted]

Hospital and Laboratory Equipment

Hospital and laboratory equipment items are major sources of sensible and latent heat gains in conditioned spaces[redacted]

Medical Equipment. It is more difficult to provide generalized heat gain recommendations for medical equipment than for general office equipment because medical equipment is much more varied

[redacted]

in type and in application. Some heat gain testing has been done, but the equipment included represents only a small sample of the type of equipment that may be encountered.

[redacted]

Laboratory Equipment. Equipment in laboratories is similar to medical equipment in that it varies significantly from space to space. [redacted]

Office Equipment

Computers, printers, copiers, etc., can generate very significant heat gains, sometimes greater than all other gains combined [redacted]

Nameplate Versus Measured Energy Use. Nameplate data rarely reflect the actual power consumption of office equipment. [redacted]

[redacted]

[redacted]

[redacted]

[redacted]

Computers[redacted]

Monitors[redacted]

Laser Printers[redacted]

[redacted]

Copiers. [redacted]

Miscellaneous Office Equipment[redacted]

[redacted]

Diversity. [redacted]

Heat Gain per Unit Area [redacted]

[redacted]

Radiant/Convective Split. [redacted]

[redacted]

[redacted]

3. INFILTRATION AND MOISTURE MIGRATION HEAT GAINS

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

[redacted]

[redacted]

[redacted]

3.1 INFILTRATION

Principles of estimating infiltration in buildings, with emphasis on the heating season, are discussed in Chapter 16. When economically feasible, somewhat more outdoor air may be introduced to a building than the total of that exhausted, to create a slight overall positive pressure in the building relative to the outdoors. [redacted] When positive air pressure is assumed, most designers do not include infiltration in cooling load calculations for commercial buildings. However, including some infiltration for spaces such entry areas or loading docks may be appropriate, especially when those spaces are on the windward side of buildings[redacted]

[redacted] As such, it is impossible to accurately predict infiltration rates. Designers usually predict overall rates of infiltration using the number of **air changes per hour (ACH)**. A common guideline for climates and buildings typ-

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

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, volumetric flow rates are often required for selecting coils, fans, ducts, etc.; basing volumes on measurement at standard conditions may be used for accurate results [redacted] Because air usually passes through the equipment at a density close to standard for locations below about 1000 ft, 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 sea-level psychrometric chart. For higher elevations, the mass flow rates of air must be adjusted and higher-elevation psychrometric charts or algorithms must be used.

Heat Gain Calculations Using Standard Air Values

Air-conditioning design often requires the following information:

1. Total heat

[redacted]

2. Sensible heat

[redacted]

3. Latent heat

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

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

[redacted]

[redacted]

[redacted]

Elevation Correction Examples

To correct the *C* values for El Paso, Texas, the elevation listed in the appendix of Chapter 14 is 3918 ft. *C* values for Equations (7) to (10) can be corrected using Equation (3) in Chapter 1 as follows:

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

3.2 LATENT HEAT GAIN FROM MOISTURE DIFFUSION

Diffusion of moisture through building materials is a natural phenomenon that is always present. Chapters 25 to 27 cover principles, materials, and specific methods used to control moisture. Moisture transfer through walls and roofs is often neglected in comfort air conditioning because the actual rate is quite small and the corresponding latent heat gain is insignificant. Permeability and permeance values for various building materials are given in Chapter 26. Vapor retarders should be specified and installed in the proper location to keep moisture transfer to a minimum, and to minimize condensation within the envelope. Moisture migration up through slabs-on-grade and basement floors has been found to be significant, but has historically not been addressed in cooling load calculations. Under-slab continuous moisture retarders and drainage can reduce upward moisture flow.

[redacted]

q_{l_m} [redacted]

[redacted]

q_{l_m} [redacted]

[redacted]

3.3 OTHER LATENT LOADS

Moisture sources within a building (e.g., shower areas, swimming pools or natatoriums, arboretums) can also contribute to latent load. Unlike sensible loads, which correlate to supply air quantities required in a space, latent loads usually only affect cooling coils sizing or refrigeration load. Because air from showers and some other moisture-generating areas is exhausted completely, those airborne latent loads do not reach the cooling coil and thus do not contribute to cooling load. However, system loads associated with ventilation air required to make up exhaust air must be recognized, and any recirculated air's moisture must be considered when sizing the dehumidification equipment.

For natatoriums, occupant comfort and humidity control are critical. In many instances, size, location, and environmental requirements make complete exhaust systems expensive and ineffective. Where recirculating mechanical cooling systems are used, evaporation (latent) loads are significant. Chapter 6 of the 2019 *ASHRAE Handbook—HVAC Applications* provides guidance on natatorium load calculations.

4. FENESTRATION HEAT GAIN

For spaces with neutral or positive air pressurization, the primary weather-related variable affecting cooling load is solar radiation. The effect of solar radiation is more pronounced and immediate on exposed, nonopaque surfaces. Chapter 14 includes procedures for calculating clear-sky solar radiation intensity and incidence angles for weather conditions encountered at specific locations. That chapter also includes some useful solar equations. 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 15. This chapter covers application of such data to overall heat gain evaluation, and conversion of calculated heat gain into a composite cooling load for the conditioned space.

4.1 FENESTRATION DIRECT SOLAR, DIFFUSE SOLAR, AND CONDUCTIVE HEAT GAINS

For fenestration heat gain, use the following equations:

[redacted]

[redacted]

If specific window manufacturer's SHGC and U-factor data are available, those should be used. [redacted]

Note that, as discussed in Chapter 15, fenestration ratings (U-factor and SHGC) are based on the entire product area, including frames. Thus, for load calculations, fenestration area is the area of the entire opening in the wall or roof.

4.2 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 indoor 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[redacted]

5. HEAT BALANCE METHOD

Cooling load estimation involves calculating a surface-by-surface conductive, convective, and radiative heat balance for each room surface and a convective heat balance for the room air. These [redacted] The heat balance (HB) method solves the problem using the most fundamental principles and the fewest simplifications. The advantages are that it contains the fewest parameters and general assumptions.

Some computations required by this rigorous approach require the use of computers. The heat balance procedure is not new. Many energy calculation programs have used it in some form for many years[redacted]

5.1 ASSUMPTIONS

All calculation procedures involve some kind of model; all models require simplifying assumptions and, therefore, are approximate. The most fundamental assumption inherent in the traditional heat balance solution is that air in the thermal zone can be modeled as **well mixed**, meaning its temperature is uniform throughout the zone. [redacted]

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

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

The resulting formulation is called the **heat balance (HB) model**. Note that the assumptions, although common, set certain limits on the information that can be obtained from the model. [redacted]The discussion that follows is primarily focused on the use of heat balance with convective air system based cooling systems.

5.2 ELEMENTS

Within the framework of the assumptions, the HB can be viewed as four distinct processes:

1. Outdoor-face heat balance
2. Wall conduction process
3. Indoor-face heat balance
4. Air heat balance

[redacted]

Outdoor-Face Heat Balance

The heat balance on the outdoor face of each surface is

[redacted]

[redacted]

Wall Conduction Process

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

- Numerical finite difference
- Numerical finite element
- Transform methods
- Time series methods

This process introduces part of the time dependence inherent in load calculation [redacted]

In some models, surface heat transfer coefficients are included as part of the wall element, making the temperatures in question the indoor and outdoor air temperatures. This is not a desirable formulation, because it hides the heat transfer coefficients and prohibits changing them as airflow conditions change. It also prohibits treating the internal long-wave radiation exchange appropriately.

[redacted]

Indoor-Face Heat Balance

The heart of the HB method is the internal heat balance involving the inner 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 indoor-face heat balance for each surface can be written as follows:

[redacted]

[redacted]

These terms are explained in the following paragraphs.

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

• [redacted]

Most HB models treat air as completely transparent and not participating in LW radiation exchange among 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 thus decouples radiant exchange among surfaces in the zone. However, because the transparent air model allows 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 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 heat introduced into a zone from equipment. The radiative part is then distributed over the zone's surfaces in some manner. This model is not completely realistic, and it departs from HB principles. In a true HB model, equipment surfaces are treated just as other LW radiant sources in the zone. [redacted]

Transmitted Solar Heat Gain [redacted]

Using SHGC to Calculate Solar Heat Gain

The total solar heat gain through fenestration consists of directly transmitted solar radiation plus the inward-flowing fraction of solar radiation that is absorbed in the glazing system. Both parts contain beam and diffuse contributions. Transmitted radiation goes directly onto surfaces in the zone and is accounted for in the surface indoor heat balance. [redacted]

The only way to model these interactions correctly is to combine the window model with the zone heat balance model and solve both simultaneously. This 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 indoor, outdoor, 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 heat balance calculation in the most accurate way possible, as discussed in the following paragraphs. [redacted]

[redacted]

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

[redacted]

The same caution about the indoor and outdoor heat transfer coefficients applies to the information in Table 10 in Chapter 15.

[redacted]

Convection to Zone Air. Indoor convection coefficients presented in past editions of this chapter and used in most load calculation procedures and energy programs are based on very old, natural convection experiments and do not accurately describe heat transfer coefficients in a mechanically ventilated zone. In previous load calculation procedures, these coefficients were buried in the procedures and could not be changed. [redacted]

Air Heat Balance

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

$$q_{conv} + q_{CE} + q_{IV} + q_{sys} = 0 \tag{21}$$

where

q_{conv} = convective heat transfer from surfaces, Btu/h
q_{CE} = convective parts of internal loads, Btu/h
q_{IV} = sensible load caused by infiltration and ventilation air, Btu/h
q_{sys} = heat transfer to/from HVAC system, Btu/h

Convection from zone surfaces q_{conv} is the sum of all the convective heat transfer quantities from the indoor-surface heat balance. This comes to the air through the convective heat transfer coefficient on the surfaces.

[redacted]

Conditioned air that enters the zone from the HVAC system and provides q_{sys} is also mixed directly with the zone air. For commercial HVAC systems, ventilation air is most often provided using outdoor air as part of this mixed-in conditioned air; ventilation air is thus normally a system load rather than a direct-to-space load. An exception is where infiltration or natural ventilation is used to provide all or part of the ventilation air, as discussed in Chapter 16.

5.3 GENERAL ZONE FOR LOAD CALCULATION

The HB procedure is tailored to a single thermal zone, shown in Figure 7. The definition of a thermal zone depends on how the fixed temperature is controlled. If air circulated through an entire building or an entire floor 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). 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 are the indoor and outdoor temperatures of the 12 surfaces plus either the HVAC system energy required to maintain a specified air temperature or the air temperature, if system capacity is specified. This makes a total of $25 \times 24 = 600$ variables. Although 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 allows a double iterative approach. One iteration is through all the surfaces in each hour, and the other is through the 24 h of a day. This procedure automatically reconciles nonlinear aspects of surface radiative exchange and other heat flux terms.

5.4 MATHEMATICAL DESCRIPTION

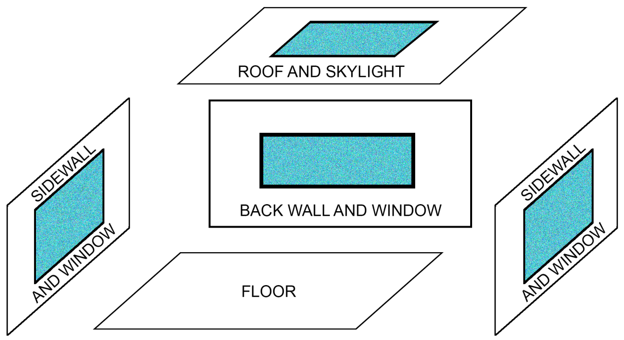
Conduction Process

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

[redacted] [redacted]

For outdoor heat flux, the form is

[redacted] [redacted]



Front Wall/Window and Thermal Mass are not shown.

Fig. 7 Schematic View of General Heat Balance Zone

[redacted]

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

[redacted]

Heat Balance Equations

The primary variables in the heat balance for the general zone are the 12 indoor face temperatures and the 12 outdoor 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. Thus, the primary variables are

$T_{soi,j}$ = outdoor face temperature, $i = 1,2,\ldots,12$; $j = 1,2,\ldots, 24$

$T_{sii,j}$ = indoor face temperature, $i = 1,2,\ldots,12$; $j = 1,2,\ldots, 24$

In addition, q_{sys_j} = cooling load, $j = 1,2,\ldots, 24$.

Equations (16) and (23) are combined and solved for T_{so} to produce 12 equations applicable in each time step:

[redacted]

where

T_o = outdoor air temperature

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

Equation (24) shows the need to separate $X_{i,0}$, because the contribution of current surface temperature to conductive flux cannot be collected with the other historical terms involving that temperature.

Equations (17) and (22) are combined and solved for T_{si} to produce the next 12 equations:

[redacted]

[redacted]

Overall HB Iterative Solution

[redacted]

5.5 INPUT REQUIRED

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

Global Information. Because the procedure incorporates a solar calculation, some global information is required, including latitude, longitude, time zone, month, day of month, directional orientation of the zone, and zone height (floor to floor). Additionally, to take full advantage of the flexibility of the method to incorporate, for example, variable outdoor heat transfer coefficients, things such as wind speed, wind direction, and terrain roughness may be specified. 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. [redacted]

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

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

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

[redacted]

Radiant Distribution Functions. As mentioned previously, the generally accepted assumptions for the HB method include specifying the distribution of radiant energy from several sources to surfaces that enclose the space. [redacted]

Other Required Information. Additional flexibility is included in the model so that results of research can be incorporated easily. [redacted]

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

6. RADIANT TIME SERIES (RTS) METHOD

The radiant time series (RTS) method is a simplified method for performing design cooling load calculations that is derived from the heat balance (HB) method. It is intended as a more up-to-date alternative to 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.

This method was developed to offer an approach that is rigorous, yet does not require iterative calculations, and that quantifies each component's 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 showing their relative effect on the result. These characteristics of the RTS method make it easier to apply engineering judgment during cooling load calculation.

The RTS method is suitable for peak design load calculations, but it should not be used for annual energy simulations because of its inherent limiting assumptions. Although simple in concept, RTS involves too many calculations for practical use as a manual method, although it can easily be implemented in a simple computerized spreadsheet, as shown in the examples. For a manual cooling load calculation method, refer to the CLTD/CLF method in Chapter 28 of the 1997 *ASHRAE Handbook—Fundamentals*.

6.1 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:

[redacted]

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

As described in the section on Cooling Load Principles, most heat sources transfer energy to a room by a combination of convection and radiation. The convective part of heat gain immediately becomes cooling load. The radiative 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.

6.2 OVERVIEW

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

[redacted]

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 HB 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 indoors 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 effect of one construction versus another. This ability to compare choices is of particular benefit during design, when all construction details may not have been decided. Comparison can show the magnitude of difference between the choices, allowing the engineer to apply judgment and make more informed assumptions in estimating the load.

[redacted]

[redacted]

[redacted]

6.3 RTS PROCEDURE

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

[redacted]

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

[redacted]

system. Repeat this process 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.

6.4 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, because 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 outdoor air temperature 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 outdoor air.

Heat Flux into Exterior Sunlit Surfaces. [redacted]

Because **vertical surfaces** receive long-wave radiation from the ground and surrounding buildings as well as from the sky, accurate ΔR values are difficult to determine. When solar radiation intensity is high, surfaces of terrestrial objects usually have a higher temperature than the outdoor air; thus, their long-wave radiation compensates to some extent for the sky's low emittance. Therefore, it is common practice to assume $\varepsilon\Delta R = 0$ for vertical surfaces.

Tabulated Temperature Values. [redacted]

Surface Colors. [redacted]

Calculating Conductive Heat Gain Using Conduction Time Series

In the RTS method, conduction through exterior walls and roofs is calculated using CTS values. [redacted]

[redacted]

[redacted]

[redacted]

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. [redacted]

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

6.5 HEAT GAIN THROUGH INTERIOR SURFACES

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

$$q = UA(t_b - t_i) \tag{32}$$

where

q = heat transfer rate, Btu/h
U = coefficient of overall heat transfer between adjacent and conditioned space, Btu/h·ft^2·°F
A = area of separating section concerned, ft^2
t_b = average air temperature in adjacent space, °F
t_i = air temperature in conditioned space, °F

U-values can be obtained from Chapter 27. Temperature t_b may differ greatly from t_i. The temperature in a kitchen or boiler room, for example, may be as much as 15 to 50°F 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 the difference between the outdoor air and conditioned space design dry-bulb temperatures minus 5°F. In some cases, air temperature in the adjacent space corresponds to the outdoor air temperature or higher.

[redacted]

Floors

For floors directly in contact with the ground or over an underground basement that is neither ventilated nor conditioned, sensible heat transfer may be neglected for cooling load estimates because usually there is a heat loss rather than a gain [redacted]

6.6 CALCULATING COOLING LOAD

The **instantaneous cooling load** is the rate at which heat energy is convected to the zone air at a given point in time. 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 a time dependency to the process that is not easily quantified. Radiation is absorbed by thermal masses in the zone and then later transferred by convection into the space. This process creates a time lag and dampening effect. The convective portion, on the other hand, is assumed to immediately become cooling load in the hour in which that heat gain occurs.

[redacted]

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, lighting, appliances, and equipment, to the radiant exchange. RTS further simplifies the HB 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 in the HB 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 14 contains recommendations for splitting each of the heat gain components into convective and radiant portions.

RTS 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 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:

[redacted]

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.

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

7. HEATING LOAD CALCULATIONS

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, with the following exceptions:

[redacted]

- Temperatures outdoor conditioned spaces are generally lower than maintained space temperatures.
- Credit for solar or internal heat gains is not included
- Thermal storage effect of building structure or content is ignored.

Thermal bridging effects on wall and roof conduction are greater for heating loads than for cooling loads, and greater care must be taken to account for bridging effects on U-factors used in heating load calculations.

[redacted]

	[redacted]
	[redacted]
[redacted]	
[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	
[redacted]	[redacted]
[redacted]	[redacted]
[redacted]	[redacted]

[redacted]

[redacted]

	[redacted]	[redacted]
	[redacted]	[redacted]
[redacted]	[redacted]	
[redacted]	[redacted]	
[redacted]	[redacted]	
[redacted]		[redacted]
	[redacted]	
[redacted]	[redacted]	
[redacted]	[redacted]	

[redacted]

[redacted]

[redacted]

[redacted]

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. Therefore, the near-worst-case load is based on the following:

- Design interior and exterior conditions
- Including infiltration and/or ventilation
- No solar effect (at night or on cloudy winter days)
- Before the periodic presence of people, lights, and appliances has an offsetting effect

Typical commercial and retail spaces have nighttime unoccupied periods at a setback temperature where little to no ventilation is required, building lights and equipment are off, and heat loss is primarily through conduction and infiltration. Before being occupied, buildings are warmed to the occupied temperature (see the following discussion). During occupied time, building lights, equipment, and people cooling loads can offset conduction heat loss, although some perimeter heat may be required, leaving infiltration and ventilation as the primary heating loads. Ventilation heat load may be offset with heat recovery equipment. These loads (conduction loss, warm-up load, and ventilation load) may not be additive when sizing building heating equipment, and it is prudent to analyze each load and their interactions to arrive at final equipment sizing for heating.

7.1 HEAT LOSS CALCULATIONS

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

1. Select outdoor design conditions: temperature, humidity, and wind direction and speed.
2. Select indoor design conditions to be maintained.
3. Estimate temperature in any adjacent unheated spaces.
4. Select transmission coefficients and compute heat losses for walls, floors, ceilings, windows, doors, and foundation elements.
5. Compute heat load through infiltration and any other outdoor air introduced directly to the space.
6. Sum the losses caused by transmission and infiltration.

Outdoor Design Conditions

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

Indoor Design Conditions

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

The indoor design temperature should be selected at the lower end of the acceptable temperature range, so that the heating equipment will not be oversized. Even properly sized equipment operates under partial load, at reduced efficiency, most of the time; therefore, any oversizing aggravates this condition and lowers overall system efficiency. A maximum design dry-bulb temperature of 70°F is recommended for most occupancies. The indoor design value of relative humidity should be compatible with a healthful environment and the thermal and moisture integrity of the building envelope. A minimum relative humidity of 30% is recommended for most situations.

Calculation of Transmission Heat Losses

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

$$q = A \times \mathrm{HF} \tag{34}$$

$$\mathrm{HF} = U\,\Delta t \tag{35}$$

where HF is the heating load factor in Btu/h·ft^2.

Below-Grade Surfaces [redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

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

$$HF = U(t_{in} - t_b) \tag{42}$$

Infiltration

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

Infiltration is treated as a room load and has both sensible and latent components. During winter, this means heat and humidity loss because cold, dry air must be heated to design temperature and moisture must be added to increase the humidity to design condition. Typically, during winter, controlling indoor humidity is not a factor and infiltration is reduced to a simple sensible component. Under cooling conditions, both sensible and latent components are added to the space load to be treated by the air conditioning system.Procedures for estimating the infiltration rate are discussed in Chapter 16. The infiltration rate is reduced to a volumetric flow rate at a known dry bulb/wet bulb condition. Along with indoor air condition, the following equations define the infiltration sensible and latent loads.

[redacted]

7.2 HEATING SAFETY FACTORS AND LOAD ALLOWANCES

Before mechanical cooling became common in the second half of the 1900s, and when energy was less expensive, buildings included much less insulation; large, operable windows; and generally more infiltration-prone assemblies than the energy-efficient and much tighter buildings typical of today. In the past, allowances of 10 to 20% of the net calculated heating load for piping losses to unheated spaces, and 10 to 20% more for a warm-up load, were common practice, along with other occasional safety factors reflecting the experience and/or concern of the individual designer. Today such safety allowances are more conservatively applied with modern construction practices. A combined warm-up/safety allowance of 20 to 25% is common but varies depending on the particular climate, building use, and type of construction. Engineering judgment must be applied for the particular project.

Today's more efficient buildings have smaller overall heating loads at peak design conditions. The spare capacity in absolute terms needed for warm-up may not be any less in the highly efficient building than a more traditional building, assuming building mass is the same in both cases. Simply adding the same percentage factor to a lower peak load value may not result in enough spare capacity for timely warm up of the building. Transient models can be used to calculate both the quasi-steady state peak heating load and the spare capacity needed for building warm-up, but this level of analytical rigor may not be possible for a typical building design process. Experience and judgment must be applied, especially in cases where the base heating system capacity has been reduced through enhanced insulation and infiltration reduction.

7.3 OTHER HEATING CONSIDERATIONS

Calculation of design heating load estimates has essentially become a subset of the more involved and complex estimation of cooling loads for such spaces. Chapter 19 discusses using the heating load estimate to predict or analyze energy consumption over time. Special provisions to deal with particular applications are covered in the 2019 *ASHRAE Handbook—HVAC Applications* and the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*.

8. SYSTEM HEATING AND COOLING LOAD EFFECTS

The heat balance (HB) or radiant time series (RTS) methods are used to determine cooling loads of rooms within a building, but they do not address the plant size necessary to reject the heat. Principal factors to consider in determining the plant size are ventilation, heat transport equipment, and air distribution systems. Some of these factors vary as a function of room load, ambient temperature, and control strategies, so it is often necessary to evaluate the factors and strategies dynamically and simultaneously with the heat loss or gain calculations.

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

8.1 ZONING

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

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

Grouping rooms together to reduce the number of HVAC systems or zones is called **thermal zoning**. Zoning choices can affect the HVAC system peak load as well as system energy performance. [redacted]

8.2 VENTILATION

Consult ASHRAE *Standard* 62.1 and building codes to determine the required quantity of ventilation air for an application, and the various methods of achieving acceptable indoor air quality. The following discussion is confined to the effect of mechanical ventilation on sizing heat removal equipment. Where natural ventilation is used, through operable windows or other means, it is considered as infiltration and is part of the direct-to-room heat gain. Where ventilation air is conditioned and supplied through the mechanical system, its sensible and latent loads are applied directly to heat transport and central equipment, and do not affect room heating and cooling loads. If the mechanical ventilation rate sufficiently exceeds exhaust airflows, air pressure may be positive and infiltration from envelope openings and outdoor wind may not be included in the load calculations. Chapter 16 includes more information on ventilating commercial buildings.

Depending on ventilation requirements and local climate conditions, peak cooling coil loads may occur at peak dehumidification or enthalpy conditions instead of design dry-bulb conditions. Coil loads should be checked against all those peak conditions.

8.3 AIR HEAT TRANSPORT SYSTEMS

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

On/Off Control Systems

On/off control systems, common in residential and light commercial applications, cycle equipment on and off to match room load. They are adaptable to heating or cooling because they can cycle both heating and cooling equipment. In their purest form, their heat transport matches the combined room and ventilation load over a series of cycles.

Variable-Air-Volume Systems

Variable-air-volume (VAV) systems have airflow controls that adjust cooling airflow to match the room cooling load. Damper leakage or minimum airflow settings may cause overcooling, so most VAV systems are used in conjunction with separate heating systems. These may be duct-mounted heating coils, or separate radiant or convective heating systems.

The amount of heat added by the heating systems during cooling becomes part of the room cooling load. Calculations must determine the minimum airflow relative to off-peak cooling loads. The quantity of heat added to the cooling load can be determined for each terminal by Equation (8) using the minimum required supply airflow rate and the difference between supply air temperature and the room indoor heating design temperature.

Constant-Air-Volume Reheat Systems

In constant-air-volume (CAV) reheat systems, all supply air is cooled to remove moisture and then heated to avoid overcooling rooms. *Reheat* refers to the amount of heat added to cooling supply air to raise the supply air temperature to the temperature necessary for picking up the sensible load. The quantity of heat added can be determined by Equation (8).

With a constant-volume reheat system, heat transport system load does not vary with changes in room load, unless the cooling coil discharge temperature is allowed to vary. Where a minimum circulation rate requires a supply air temperature greater than the available design supply air temperature, reheat adds to the cooling load on the heat transport system. This makes the cooling load on the heat transport system larger than the room peak load.

Mixed Air Systems

Mixed air systems change the supply air temperature to match the cooling capacity by mixing airstreams of different temperatures; examples include multizone and dual-duct systems. Systems that cool the entire airstream to remove moisture and to reheat some of the air before mixing with the cooling airstream influence load on the heat transport system in the same way a reheat system does. Other systems separate the air paths so that mixing of hot- and cold-deck airstreams does not occur. For systems that mix hot and cold airstreams, the contribution to the heat transport system load is determined as follows.

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

$$\frac{Q_h}{Q_c} = (T_c - T_r)/(T_r - T_h)$$

2. From Equation (9), the hot-deck contribution to room load during off-peak cooling is

$$q_{rh} = 1.1 Q_h (T_h - T_r)$$

where

Q_h = heating airflow, cfm
Q_c = cooling airflow, cfm
T_c = cooling air temperature, °F
T_h = heating air temperature, °F
T_r = room or return air temperature, °F
q_{rh} = heating airflow contribution to room load at off-peak hours, Btu/h

Heat Gain from Fans

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

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

Duct Surface Heat Transfer

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

The effect of duct heat loss or gain depends on the duct routing, duct insulation, and its surrounding environment. Consider the following conditions:

- For duct run within the area cooled or heated by air in the duct, heat transfer from the space to the duct has no effect on heating or cooling load, but beware of the potential for condensation on cold ducts.

- For duct run through unconditioned spaces or outdoors, heat transfer adds to the cooling or heating load for the air transport system but not for the conditioned space.
- For duct run through conditioned space not served by the duct, heat transfer affects the conditioned space as well as the air transport system serving the duct.
- For an extensive duct system, heat transfer reduces the effective supply air differential temperature, requiring adjustment through air balancing to increase airflow to extremities of the distribution system.

Duct Leakage

Air leakage from supply ducts can considerably affect HVAC system energy use. Leakage reduces cooling and/or dehumidifying capacity for the conditioned space, and must be offset by increased airflow (sometimes reduced supply air temperatures), unless leaked air enters the conditioned space directly. Supply air leakage into a ceiling return plenum or leakage from unconditioned spaces into return ducts also affects return air temperature and/or humidity.

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

Ceiling Return Air Plenum Temperatures

The space above a ceiling, when used as a return air path, is a ceiling return air plenum, or simply a **return plenum**. Unlike a traditional ducted return, the plenum may have multiple heat sources in the air path. [redacted]

As heat from these sources is picked up by the unducted return air, the temperature differential between the ceiling cavity and conditioned space is small [redacted]

[redacted]

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

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

[redacted]

Ceiling Plenums with Ducted Returns

Compared to those in unducted plenum returns, temperatures in ceiling plenums that have well-sealed return or exhaust air ducts float considerably. In cooling mode, heat from lights and other equipment raises the ceiling plenum's temperature considerably. Solar heat gain through a poorly insulated roof can drive the ceiling plenum temperature to extreme levels, so much so that heat gains to uninsulated supply air ducts in the plenum can dramatically decrease available cooling capacity to the rooms below. In cold weather, much heat is lost from warm supply ducts. Thus, insulating supply air ducts and sealing them well to minimize air leaks are highly desirable, if not essential. Appropriately insulating roofs and plenums' exterior walls and minimizing infiltration are also key to lowering total building loads and improving HVAC system performance.

Underfloor Air Distribution Systems

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

• [redacted]

Plenums in Load Calculations

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

8.4 CENTRAL PLANT

Piping

Losses must be considered for piping systems that transport heat. For water or hydronic piping systems, heat is transferred through the piping and insulation (see Chapter 23 for ways to determine this transfer). However, distribution of this transferred heat depends on the fluid in the pipe and the surrounding environment.

Consider a heating hot-water pipe. If the pipe serves a room heater and is routed through the heated space, any heat loss from the pipe adds heat to the room. Heat transfer to the heated space and heat loss from the piping system is null. If the piping is exposed to ambient conditions en route to the heater, the loss must be considered when selecting the heating equipment; if the pipe is routed through a space requiring cooling, heat loss from the piping also becomes a load on the cooling system.

In summary, the designer must evaluate both the magnitude of the pipe heat transfer and the routing of the piping.

Pumps

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

9. EXAMPLE COOLING AND HEATING LOAD CALCULATIONS

To illustrate the cooling and heating load calculation procedures discussed in this chapter, an example problem has been developed. The objectives of this example are to demonstrate (1) the component cooling load calculation procedures for a room using the radiant time series (RTS) method, (2) how orientation of opaque envelope and fenestration affects the magnitude and timing of peak room loads, (3) how a block load accounts for load diversity among rooms, and (4) the component heating load calculations for a room.

Table 26 summarizes RTS cooling load calculation procedures.

9.1 SINGLE-ROOM DETAILED COOLING LOAD EXAMPLE

The objective of this example is to calculate the cooling load for the office shown in Figure 16 for July 3:00 PM local standard time. This corner office is on the second floor of a two-story office building.

Room and Weather Characteristics

Opaque envelope: See Table 27 for surface areas, orientations and construction assembly details for floor, roof, and wall elements of the space.

[redacted]

[redacted]

Fenestration: See Table 27 for surface areas, orientations, window construction, and performance data.

Internal heat gain: See Table 28 for heat gain and schedule data.

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

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

Weather data: This example uses the Example City weather data found in Chapter 14, Table 1: Latitude = 33.64° North, Longitude = 84.43° West, elevation = 1027 ft above sea level. For heating load calculation, the heating design dry-bulb temperature is 21.9°F. For cooling load calculations the 5% monthly design dry-bulb and coincident wet-bulb temperature data from Chapter 14 Table 1 is used. This is statistically equivalent to a 2% annual cooling design condition. See Table 29 for the 24 h temperature profiles calculated per Chapter 14.

Cooling Loads Using RTS Method

Traditionally, simplified cooling load calculation methods such as the radiant time series (RTS) Method have estimated the total cooling load at a particular design condition by independently calculating each component load (wall, windows, occupants, lighting, etc.) and then summing the component loads. Although the actual heat transfer processes for each component do affect each other, this simplification, known as the **principle of superposition**, is appropriate for design load calculations and useful to the designer in understanding the relative contribution of each component to the total cooling load.

On the following pages RTS procedures will be demonstrated for calculating (1) load due to internal heat gain, (2) exterior wall load, (3) load for windows with no shading, (4) load for windows with internal shading, (5) load for windows with internal and external shading, and (6) the total room load. All loads will be calculated for July 3:00 PM local standard time. Equations used in these calculations are summarized in Table 26.

Part 1. Cooling load due to internal heat gain.

Objective: Calculate the cooling load due to overhead lighting heat gain at 3:00 PM local standard time.

Solution: Calculation of the lighting load involves the following steps: (a) calculate the 24 h heat gain profile, (b) split those heat gains into convective and radiant components, (c) determine the radiant portion of the load by applying appropriate RTS factors, and (d) sum the convective and radiant load components to determine the total lighting load.

The heat gain profile is calculated using Equation (1) for each hour of the day. Calculations are shown in columns b through e in Table 30. Each heat gain is designated as q_i. For example, for 3:00 PM (hour 15):

[redacted]

Part 2. Wall cooling load.

Objective: Calculate the cooling load for the spandrel wall section facing 60° west of south for July 3:00 PM local standard time.

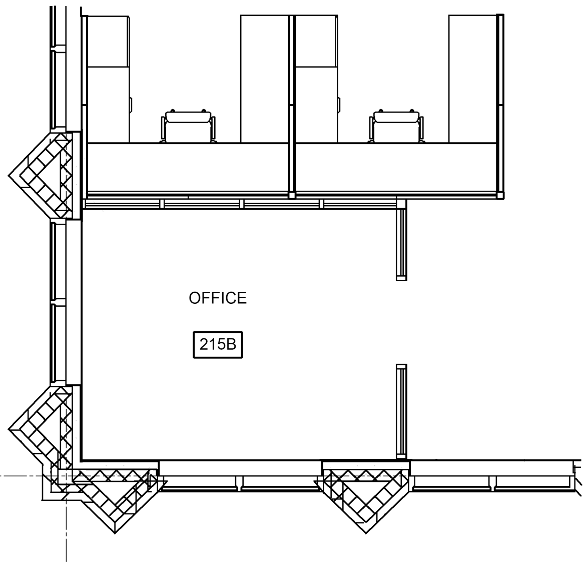


Fig. 16 Single-Room Example Office

Solution: Determining the wall cooling load requires calculation of: [redacted]

[redacted]

[redacted]	[redacted]	[redacted]	[redacted]	[redacted]
[redacted]	[redacted]		[redacted]	[redacted]
[redacted]	[redacted]		[redacted]	[redacted]
[redacted]	[redacted]		[redacted]	[redacted]
[redacted]	[redacted]			[redacted]
[redacted]	[redacted]		[redacted]	[redacted]
[redacted]	[redacted]			[redacted]
[redacted]	[redacted]		[redacted]	[redacted]
[redacted]	[redacted]			[redacted]

[redacted]

[redacted]		[redacted]
[redacted]		[redacted]
[redacted]	[redacted]	[redacted]
[redacted]	[redacted]	[redacted]

[redacted]

[redacted]

[redacted]

This procedure is used to calculate the sol-air temperatures for [redacted]ach hour. Because of the tedious solar angle and intensity calcula[redacted]ons, using a simple computer spreadsheet or other computer software can reduce the effort involved. A spreadsheet was used to calculate a 24 h sol-air temperature profile for the data of this example. See Table 33A for the solar angle and intensity calculations and Table 33B for the sol-air temperatures for this wall surface and orientation.

Next. use the sol-air temperature to calculate the heat input at the exterior surface of the wall using Equation (30). Table 33B shows the results of this calculation in column *f*. The calculation for 3:00 PM is

[redacted]

[redacted]

[redacted]

Part 3. Window cooling load without internal or external shading

Objective: Calculate the cooling load for the 40 ft^2 window facing 60° west of south for July 3:00 PM local standard time, without considering internal or external shading. Internal and external shading will be considered in Parts 4 and 5, respectively.

Solution: Determining the window cooling load requires calculation of: (a) the 24 h window heat gain profile, (b) the convective and radiant portions of the heat gain, (c) the conversion of the radiant heat gain into cooling load using RTS factors, and (d) the total window load as the sum of convective and radiant loads.

[redacted]

[redacted]

Part 4. Window cooling load with internal shading

Objective: Building on the window load calculation in Part 3, calculate the window load considering internal shading due to light-colored mini-blinds. Consider the same 40 ft^2 window facing 60° west of south for July 3:00 PM local standard time.

Solution: Calculation of the window cooling load requires the same four steps used in Part 3, but with different application data. Calculate: (a) the 24 h window heat gain profile, (b) the convective and radiant portions of the heat gain, (c) the conversion of the radiant heat gain into cooling load using RTS factors, and (d) the total window load as the sum of convective and radiant loads.

[redacted]

[redacted]

[redacted]

[redacted]

Part 5. Window cooling load with internal and external shading

Objective: Calculate the cooling load for the window in Part 4 with the addition of a 5 ft overhang shading the window.

Solution: As in Parts 3 and 4, determining the window cooling load requires calculation of (a) the 24 h window heat gain profile, (b) the convective and radiant portions of the heat gain, (c) the conversion of the radiant heat gain into cooling load using RTS factors, and (d) the total window load as the sum of convective and radiant loads.

[redacted]

[redacted]

[redacted]

[redacted]

Part 6. Total room sensible cooling load.

Objective: Calculate the total sensible cooling load for the example office for July 3:00 PM local standard time.

Solution: To calculate the total sensible cooling load for the office, cooling loads for all ten heat gain components must be calculated and then summed.

[redacted]

9.2 EFFECT OF ORIENTATION ON PEAK COOLING LOAD MAGNITUDE AND TIME

A room cooling load is the combination of multiple load components, each driven by separate, independently varying heat gains. The peak room load occurs when the sum of all component loads is largest. This is often at a time when many of the individual component loads are not at their largest value. Among the heat gains driving loads in a room,

- There are multiple different internal sources of heat gain whose intensity varies with time.
- Opaque envelope heat gains are a function of the thermal properties of the assembly, surface area, outdoor and indoor dry-bulb temperature, and the orientation of the surface. Orientation affects when the surface is exposed to solar irradiance of varying intensities.
- Fenestration heat gains are a function of the thermal and optical performance of the fenestration assembly, surface area, outdoor and indoor dry-bulb temperature, and orientation of the surface. Like opaque envelope components, orientation affects when the fenestration is exposed to solar irradiance of different intensities.

Orientation can significantly affect the peak time of individual opaque envelope and fenestration component loads. For example, driven by solar irradiance, an east-facing room will have peak opaque envelope and fenestration loads earlier in the day than a

[redacted]

west-facing room. Orientation can also affect the time of year when peak loads occur. In the northern hemisphere, a south-facing window will experience a peak load during a fall month due to the solar altitude angle being lower so solar irradiance on the window is more intense than during summer months. Because of the effect of orientation, the time of peak load for individual envelope load components and for the room itself can be difficult to predict from intuition or experience for a specific building project. As a result the common practice is to calculate loads for a range of times of day and times of year, and then examine the results to identify the true peak load and time of load for individual rooms. Room load calculations are tedious by hand, but a multiple-hour calculation is feasible using a spreadsheet or software application for load calculation. When software is used, this typically involves calculating room loads for 24 h design cooling days corresponding to the 21st of each month. This section will demonstrate the importance of calculating loads for a wide range of times of day and times of year to determine the true peak load for rooms. If the range of times is too constrained, the engineer risks missing the true peak load and therefore under sizing airflow and thermal cooling capacity for the room.

In section 9.1, a detailed cooling load calculation was presented for the example office for July 3:00 PM. This time was chosen because it is the time of peak outdoor air dry-bulb temperature in design-day weather profiles and historically has been assumed to be the time of peak room load [redacted] Note that the internal load components are the same in July and September. The wall loads are smaller due to the cooler design-day outdoor temperatures in September and in spite of the higher solar irradiance on the walls. The roof load is also lower, due both to the cooler outdoor temperatures and lower intensity of solar irradiance on the horizontal roof surface. But the window loads are much larger due to the more intense solar irradiance, and this overcomes the reduction in wall and roof loads to result in a higher total room load than in July.

[redacted]

[redacted]

• [redacted]

9.3 EFFECT OF COOLING LOAD DIVERSITY ON PEAK BLOCK LOAD

Previous sections of this example focused on calculation of peak cooling loads for individual rooms. Room loads are important for sizing supply airflow rates for individual rooms and cooling capacity when using room by room equipment such as water source heat pumps or fan coil units. However, the calculation of the peak "block load" also has relevance for building projects. The peak block load is the largest simultaneous load among all rooms in a building, or in a portion of a building such as a floor. The peak block load can be used for a preliminary assessment of airflow capacity or thermal capacity for the project. For example, a central VAV AHU only needs sufficient airflow capacity to meet the simultaneous peak of rooms it serves, rather than the sum of individual room peak airflow rates. A central chiller plant only needs enough capacity to meet the simultaneous peak load of AHUs or fan-coil units it serves rather than the sum of individual AHU or fan-coil peak loads.

When determining the block load it is important not to sum the individual peak room loads. Rather, room loads should be calculated for a wide range of times of day and times of year. Then for each hour, sum the room loads to obtain an hourly block load. Review the resulting hourly block loads across all hours calculated to identify the peak value. This section will illustrate the concept of calculating the peak block load.

[redacted] This difference between the peak block load and the "sum of the peaks" load is known as **load diversity**. Although the diversity o[redacted]s relatively small in this simple example, it can be much larger in a full scale building. The ultimate effect of load diversity is that central HVAC equipment serving multiple rooms can be sized for a peak load that is often considerably less than the sum of the individual peaks of those rooms.

[redacted]

[redacted]

[redacted]

9.4 SINGLE-ROOM DETAILED HEATING LOAD EXAMPLE

Although the physics of heat transfer that creates heating loads is identical to that for cooling loads, a number of traditionally used simplifying assumptions facilitate a much simpler procedure for peak heating load calculation. As described in the section 7.1, Heat Loss Calculations, design heating load calculations typically assume a single outdoor temperature with no heat gain from solar or internal sources, under steady-state conditions. Thus, space heating load is determined by computing the instantaneous heat transfer rate through building envelope elements (*UA* Δ*T*) plus heat required because of outdoor air infiltration.

Room heating load.

Objective: Calculate the peak heating load for the example office.

Solution: Calculation of the individual component heating loads and the total room heating load is shown in Table 41.

[redacted]

9.5 CONCLUSION

The example problem illustrates key issues that should be understood and accounted for in calculating peak room cooling and heating loads:

- Room peak cooling and heating loads result from many independently varying sources of heat gain or loss.
- For cooling loads, the sensible heat gain profile must be determined first and then divided into convective and radiant components. The load due to the radiant component of heat gain is calculated considering the dynamic conversion of radiant heat gain to load using RTS factors. Finally, the sensible room load is computed as convective load plus radiant load.
- Peak room cooling loads occur at different times of day and times of year depending on the orientation of exterior walls and fenestration. Calculating loads for a single point in time may miss the true peak load and therefore risks under sizing supply airflow and thermal cooling capacity for the room. Instead, peak room loads should be calculated for a range of times of day and times of year to identify the true peak cooling load.
- The relative importance of each cooling and heating load component varies, depending on the portion of the building being considered. Characteristics of a particular window may have little effect on the entire building load, but could have a significant effect on the supply airflow to the room where the window is located and thus on the comfort of the occupants of that space.
- The peak block load is valuable for preliminary assessments of airflow or thermal capacity for an entire building or a portion of the building such as a floor. To accurately identify the peak block load, room loads must first be computed over a range of times of day and times of year and then summed for each hour to obtain the block load for each hour. The peak block load is identified from the profile of hourly block loads.

10. PREVIOUS COOLING LOAD CALCULATION METHODS

Procedures described in this chapter are the most current and scientifically derived means for estimating cooling load for a defined building space, but methods in earlier editions of the ASHRAE Handbook are valid for many applications. These earlier procedures are simplifications of the heat balance principles, and their use requires experience to deal with atypical 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. Experience of the practitioner can never be ignored.

The primary difference between the HB and RTS methods and the older methods is the newer methods' direct approach, compared to the simplifications necessitated by the limited computer capability available previously.

The **transfer function method (TFM)**, for example, required many calculation steps. It 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 *Handbook of Fundamentals*. Originally intended 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 radiant components had to be averaged over a time representative of the general mass of the building involved. Because perception of thermal storage characteristics of a given building is 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)** attempted to simplify the two-step TFM and TETD/TA methods into a single-step technique that proceeded directly from raw data to cooling load without 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 "cooling load temperature differences" and "cooling load factors" for use in traditional conduction ($q = UA\Delta t$) equations. The results are approximate cooling load values rather than simple heat gain values. The simplifications and assumptions used in the original work to derive those factors limit this method's applicability to those building types and conditions for which the CLTD/CLF factors were derived; the method should not be used beyond the range of applicability.

Although the TFM, TETD/TA, and CLTD/CLF procedures are not republished in this chapter, those methods are not invalidated or discredited. Experienced engineers have successfully used them in millions of buildings around the world. The accuracy of cooling load calculations in practice depends primarily on the availability of accurate information and the design engineer's judgment in the assumptions made in interpreting the available data. Those factors have much greater influence on a project's success than does the choice of a particular cooling load calculation method.

The primary benefit of HB and RTS calculations is their somewhat reduced dependency on purely subjective input (e.g., determining a proper time-averaging period for TETD/TA; ascertaining appropriate safety factors to add to the rounded-off TFM results; determining whether CLTD/CLF factors are applicable to a specific unique application). However, using the most up-to-date techniques in real-world design still requires judgment on the part of the design engineer and care in choosing appropriate assumptions, just as in applying older calculation methods.

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